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Prceedings

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KEYNOTE PAPER 1

New Challenges in the Design of Offshore Support Vessels (OSVs)

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ABSTRACT

This paper describes systematically the salient features of new challenges in design of offshore support vessels (OSVs). The revolutionary introduction of ULSTEIN X-BOW® has been a remarkable design feature since inception of offshore support vessel in Gulf of Mexico more than a half century ago. Advantages of X-BOW have been provided compared to conventional raked bow with or without a bulb. Various merits of X-BOW in regard to hull design, resistance & speed, etc. have been put forward.

Other challenges like NOx & SOx emission control through fuel efficiency have been described in relation to propulsion power and machinery configuration. Various aspects of diesel mechanical (conventional), diesel electrical and diesel mechanical & electrical (HYBRID) have been mentioned.

Various ideas like wet exhaust system, all around (360 degrees) views, etc. have also been mentioned. The continuous demand in more deck area & cargo carrying capacity for platform supply vessels (PSVs) and deep water operation resulting in demand of larger anchor handling/tug supply (AHTS) vessels are additional design challenges. ULSTEIN has initiated to come out with solutions like Multi-Application Cargo Solutions (MACS), and in collaboration with strategic partners such as EVOMEC (Norway), ULSTEIN has contributed to the development of novel concepts for moveable deck extensions (MODEX™), safe hose handling systems etc., which are important milestones in innovative design solutions for the OSV industry.

An OSV being neither a cargo vessel nor a passenger vessel is often subject to SPS (Special Purpose Ship) Code of IMO. But recent changes in MSC 235(82) have brought new changes in damage stability criteria with further stringent criteria in MSC 266(84). Further, DNV and NMD are talking about introduction of anchor handling notations and legislations.

With the opening of new frontiers in arctic areas, OSV design cycles now meet again new challenges of low temperature, reduced light, dealing with sea ice (ice class and ice breaking capability), crew comforts, etc. as added features in design of OSVs.

Efficient and Reliable Propulsion Systems for Offshore Support Vessels

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ABSTRACT

Offshore Support Vessels have increasingly developed into larger and more powerful ships responding to new operational requirements and the need for more reliable and multipurpose designs. Efficient utilization of the machinery installation for different operational modes is also becoming more important with today's increasing fuel prices and focus on reduction of exhaust gas emissions.

Wartsila has for many years been strongly involved in developing a wide range of propulsion concepts for Offshore Support Vessels meeting different vessel requirements. The systems are ranging from basic diesel mechanic installations to advanced diesel-electric systems. Selection of the most suitable propulsion concept, either if it is mechanic-, electric- or a combined solution (hybrid), should always be based on the specific operational requirements for each individual vessel designs. This paper will address the propulsion system selection and optimizing process, and discuss which elements and operational parameters that need to be included in order to assure the most efficient and reliable propulsion installation.

The Wartsila Low Loss Concept (LLC) will be presented specifically. The LLC is an advanced diesel electric propulsion concept with a number of advantages compared with conventional systems. The main characteristics of the LLC are reduced electrical losses, increased redundancy and higher reliability. Better fuel efficiency is achieved, and the architecture of the electric power distribution gives less single failure consequences, meaning higher safety and reduced operational risk. Finally, the system gives a substantial weight and space saving for the installation.

INTRODUCTION

Diesel electric (DE) propulsion has over the year's gained more market share onboard offshore support vessels. Both the technology development in diesel electric systems and the increasing demand for safety and operational flexibility have supported this development. Traditionally, diesel mechanic (DM) propulsion systems have been considered as more fuel efficient, as the electric losses in DE systems may be substantial. Consequently an additional fuel penalty of about 10% has been claimed for the DE systems. However, this fuel penalty is not taking into account the complete picture for a DP classed vessel with different operational modes. Here propeller efficiencies and main engine performance needs to be included for the different running conditions to make a complete fuel efficiency comparison between DM and DE systems. From such investigations a DE system may in some cases be a better alternative also with regard to total fuel economy.

GENERAL DESCRIPTION OF DIESEL ELECTRIC SYSTEMS

Typical DE systems for OSV vessels will consist of 4-6 diesel-generators producing all required power for propulsion and other consumers onboard. Total installed power onboard will of course depend on the vessel size and type of operation, but normally it will be between 6 and 10 MW.

Figure 1 shows a typical 8 MW DE system for an OSV. The electrical power produced by the diesel-generators will be distributed throughout the total installation to the different consumers via transformers, switchboards and frequency converters.

The total power system must be monitored and controlled with one or several control systems depending on the level of system integration. Extensive use of power electronics and a modern Power Management Systems (PMS) is essential to control the dynamic electrical network in a DE system. The PMS can either be a stand alone system or it can be integrated in a total Vessel Automation system.

Typically for these vessels the propulsion power requirement in economic transit speed is in the range 3000 – 4000 kW. During DP operation the power requirement will be very weather dependant, in the order 600 – 1500 kW in calm weather increasing to 3000 – 5000 kW in rough weather

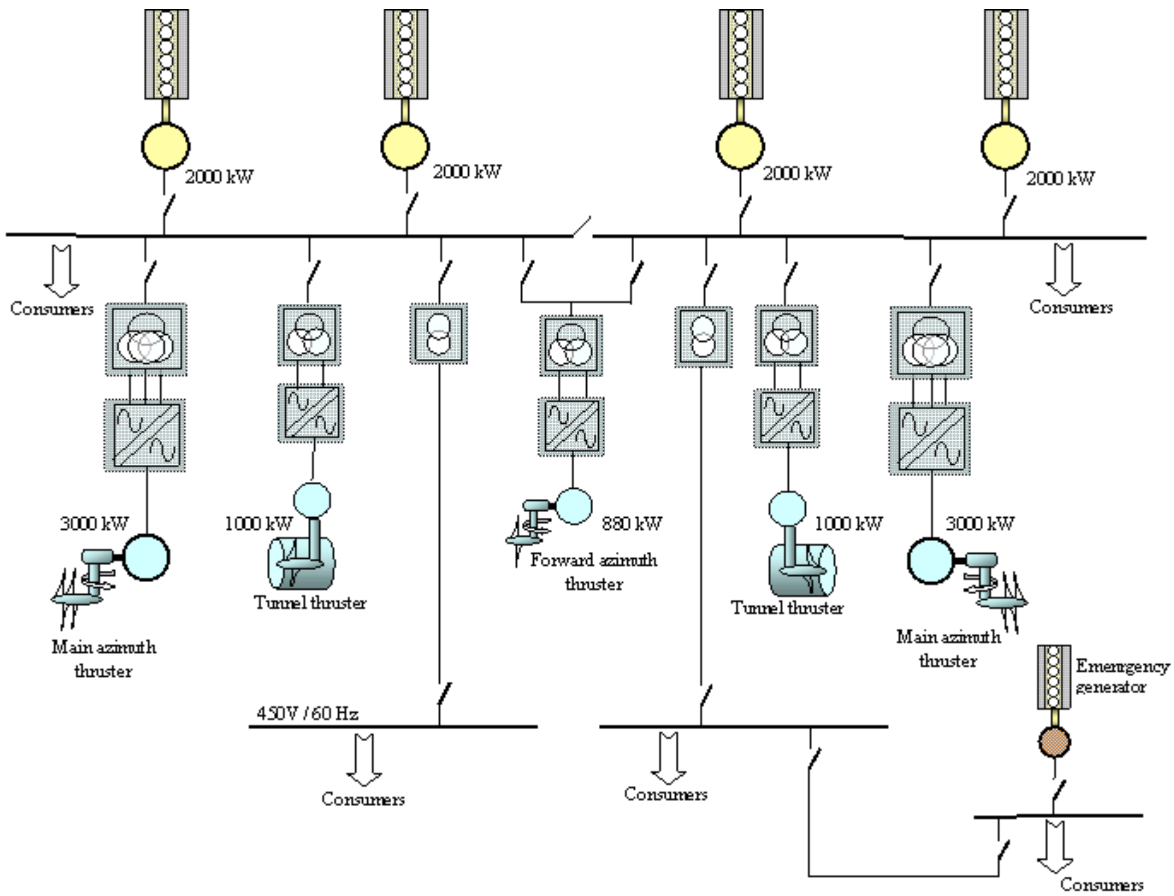

Fig 1. Typical single line diagram for OSV

Dynamic Positioning System (DP) requirements

Requirements for systems onboard DP classed vessels are regulated by the IMO Guidelines for Vessels with Dynamic Positioning Systems. Here the DP systems are grouped in 3 different equipment classes according to the level of redundancy. For equipment class 2 & 3 the positioning keeping system must be redundant, meaning that a single failure in components or systems must not lead to a loss of vessel position.
For class 2 and 3, IMO also requires an online consequence analysis system during DP operation. This function must continually perform an analysis of the vessel's ability to maintain its position and heading after a predefined single worst case failure during operation. Possible consequences are based on the actual weather condition, enabled thrusters and power plant status. Typically worst case failures are:

- Failure in the most critical thruster
- Failure in one thruster group
- Failure in one switchboard section

The consequence analysis will warn the operator if the weather and systems conditions are such that the single worst case failure will cause position drift-off. Pending on the criticality of the actual operation, if such a warning occurs, the DP operation may need to be aborted.

From the above it is obvious that one measure to increase a vessel's DP capability is to seek for arrangements in the system design that will reduce the worst single failure consequence.

Power Management System (PMS) requirements

The Power Management System (PMS) is based on intelligent control principled in monitoring and control of electric power production and consumption. The system controls and monitors the engine driven generators, switchboards and consumers. In the case of an electrical system fault the power management system shall restore power in a minimum of time.

- PMS is the sum of human experience and an efficient automated control and monitoring system
- PMS must secure a safe, reliable and efficient monitoring and control of the electrical power supply to important vessel functions in all operational conditions.

PMS Functionality

Due to the DP class 2and 3 redundancy requirements the PMS is divided in two independent parts, PMS A and PMS B. This is shown schematically in Fig. 2. Feedback from the bus tie breakers will determine the switchboard mode (Split or Connected mode). With open bus tie, each half of the switchboard has a PMS that operates independently.

Both PMS A and PMS B (two separate controllers) are getting active power (kW) signal from all generators and calculates available power for the plant. Available power signal is given from each PMS to each frequency converters. If PMS A load control fails the PMS B load control will take over.

When the bus bars are connected they will operate as one system regarding PMS functionality.

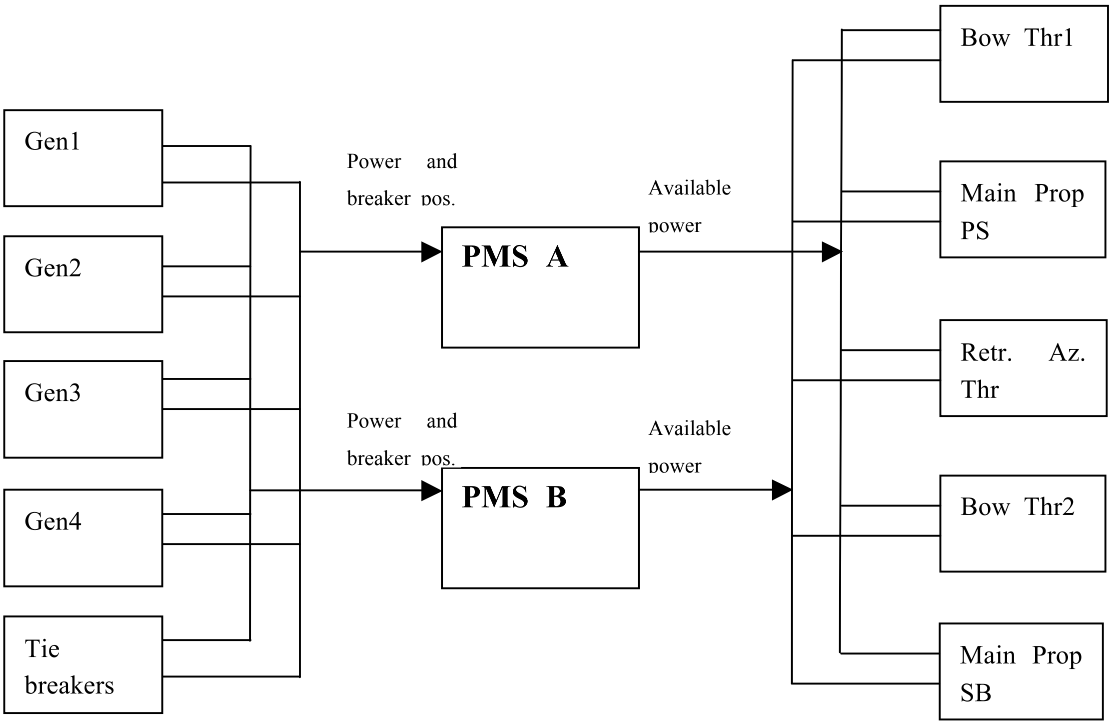

Fig 2. PMS Redundancy Architecture

WARTSILA LOW LOSS CONCEPT (LLC)

The Wärtsilä LLC is a specially designed and patented Diesel Electric system primarily developed for Offshore Support Vessels. Main features of the concept are reduced electric losses, increased reliability and less space requirement.

The basic principle of the LLC is shown in Fig. 3. In traditional systems the frequency converters are protected from the harmonic distortion in the net by having transformers at the power intake for each individual converter. In a LLC system these transformers are removed and a LLC transformer is installed and connecting the switchboards. Power for each frequency converter is duplicated, and supplied from individual switchboard sections. The LLC transformer performs phase vector difference enabling 12-pulse supply to the frequency converters. This special arrangement will reduce the total harmonic voltage distortion (THD) in the system, i.e. propulsion transformers can be omitted. Also, in normal operation almost no current is passing through the LLC transformer, meaning that transformer losses present in conventional DE systems are to a large extent eliminated.

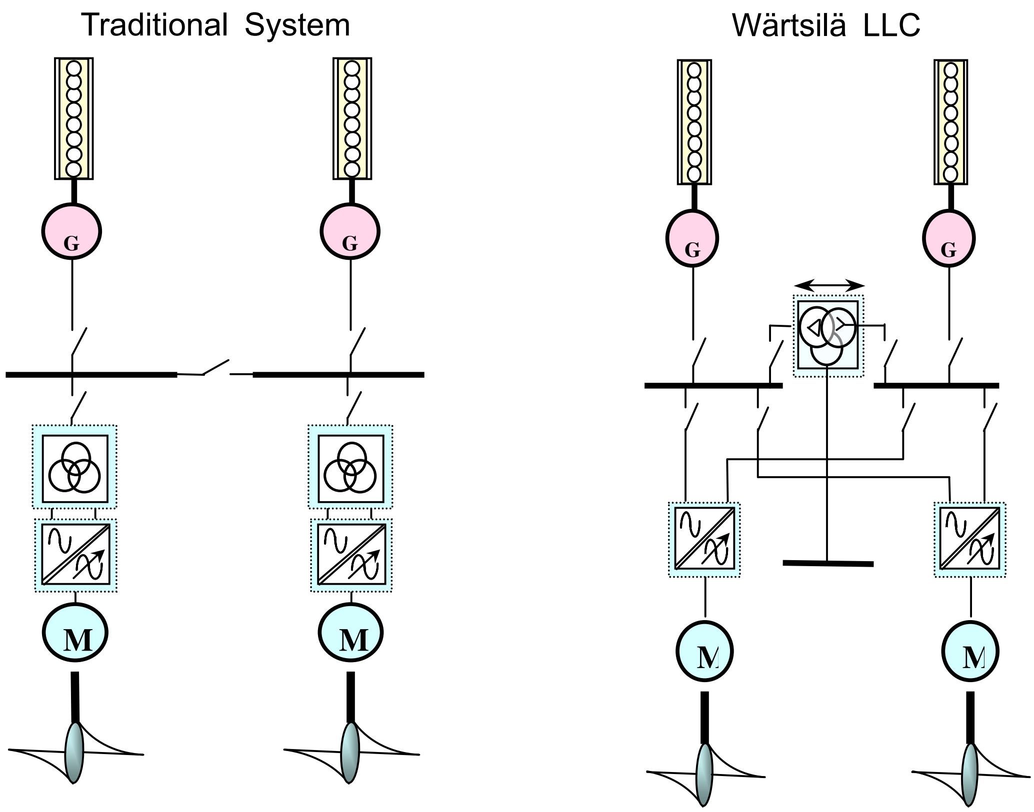

Fig 3. LLC basic principle

A typical LLC Low Voltage system for and OSV is shown in Fig 4. It is based upon a symmetrical design for the port and starboard side of the power generation, power distribution and propulsion supply systems.

The power generation system consists of four (4) diesel engines as prime movers of four (4) low voltage generators. These generators are supplying power symmetrically to a switchboard system of four (4) main 690V switchboards. These main switchboards are connected via two (2) LLC transformers.

On the port side generator G1 is connected to switchboard A1. Generator G2 is connected to switchboard A2. On the starboard side generator G3 is connected to switchboard B1. Generator G4 is connected to switchboard B2. Bus-tie breaker connects switchboards A2 and B1. Bus-link connection connects switchboards A1 and B2.

Frequency converters and the LLC transformers can be situated in the switchboard rooms.
This will simplify installation and commissioning, and make operation and maintenance easier.

No transformers or converters need to be located in the thruster rooms, i.e. more space will be available for other purposes.

The advantages of the LLC can be summarized as follows:

- The LLC gives increased robustness in DP mode by providing more available propulsion and thruster power at the occurrence of a single failure
- The main switchboard is segregated into four sections with dual bus connection through bus-link and bus-tie breakers. This increases operational flexibility and availability
- The LLC gives lower fuel consumption and reduction of environmental pollution by reduction of the electric losses in the system
- Personnel safety is significantly increased due to reduced short circuit levels on the 690V switchboard
- Weight and space requirements for the electric system components are significantly reduced.

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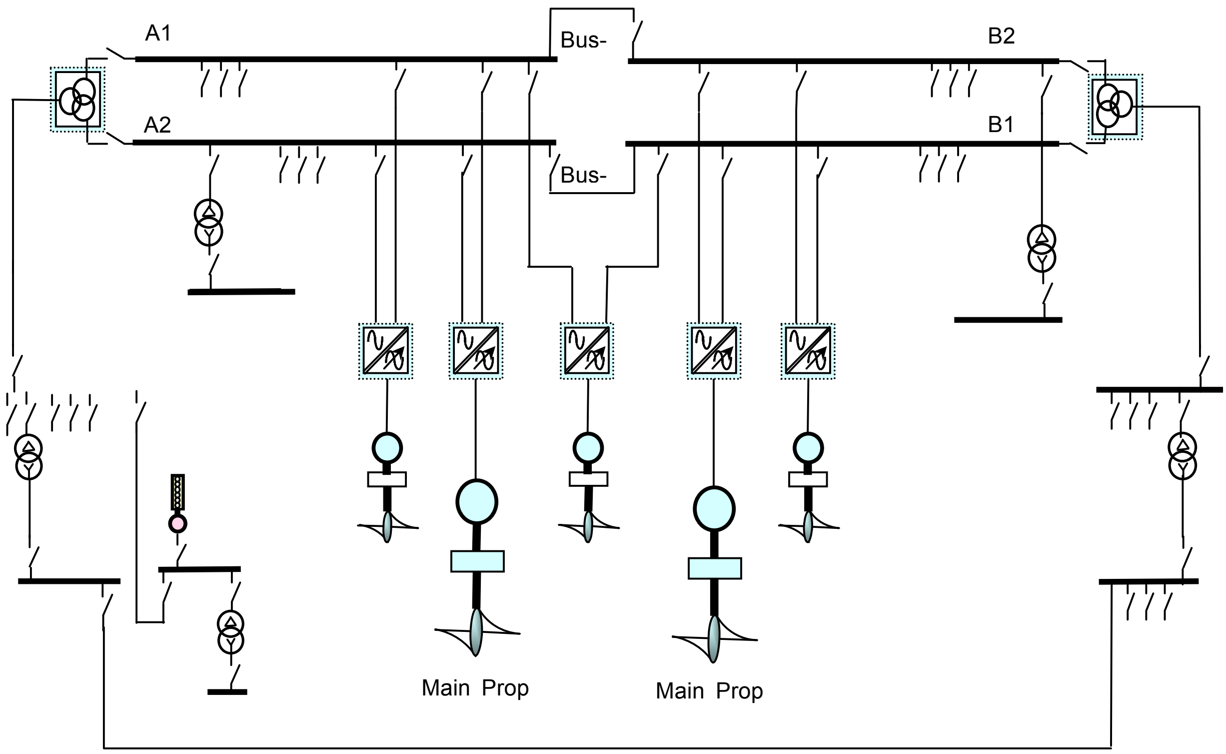
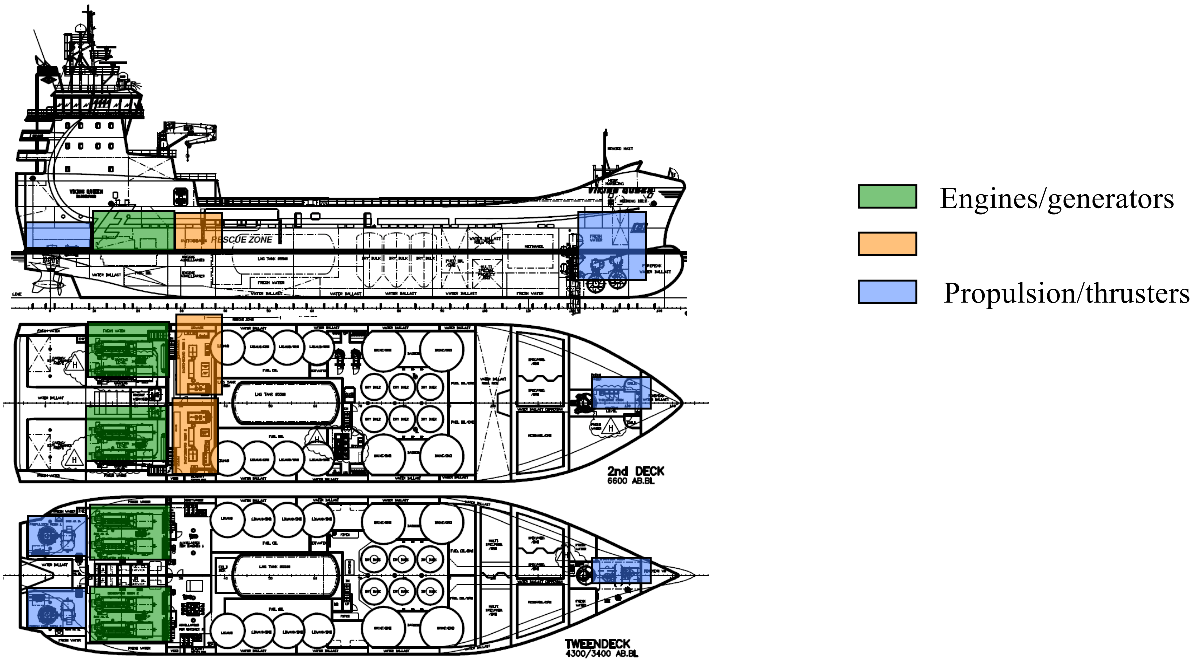


Fig 4. Typical DE system for OSV based on Low Loss Concept

EXAMPLE CASE - LLC ON A PSV VESSEL

The first vessel with Wärtsilä LLC entered service in 2005. Since then about 10 vessels have been delivered, and about 50 vessels are under construction with the LLC.

As an example of a recent Wartsila LLC delivery, the propulsion system onboard *M/S Viking Queen* will be described more in detail. *Viking Queen* is a Multipurpose Platform Support Vessel of Vik Sandvik design serving oil rigs, oil platforms and other offshore installations. The vessel has LNG gas/diesel (dual fuel) electric propulsion, is 92,2 meter long and has accommodation for 25 persons.
The vessel is built by WestCon shipyard in Norway for Eidesvik Shipping AS. She was delivered in January 2008 and is operating in the North Sea on a long term time charter for StatoilHydro. Main machinery and propulsion particulars are given in figure 5.



Vessel Design..……………………: VS 493 Avant LNG
Vessel Dimension (LxBxD)……….: 92.2 x 21.00 x 9.60 m
Main diesel-generators…………….: 4 x Wartsila 6L32DF (4 x 1950 kW)
Main propulsion…………………...: Pulling type thrusters – 2 x 2300 kW
Tunnel thrusters forw……………...: 2 x 1200 kW
Azimuth thruster forw……………..: 880 kW
Integrated Automation and PMS…..: Wartsila
DP System…………………………: Kongsberg DPC-2
DNV DP Class…………………….: AUTR (IMO DP Class 2)

Fig 5. Viking Queen – Machinery and Propulsion arrangement

Arrangement

One of the characteristics with the LLC is less space requirement for the equipment. No transformers or frequency converters need to be installed in the propulsion rooms, and this feature has been fully utilized in the vessel design for Viking Queen. The LLC transformers and propulsion converters are placed in the two switchboard rooms (ref. Fig. 5), making it possible to minimize the size of the propulsion rooms, and thus increasing the available under-deck volume for cargo. This increased volume is naturally of high value for a PSV.

The main 690V switchboard (SWB) is arranged in four physically separated sections with A1/A2 in port SWB room and B1/B2 in starboard SWB room, ref. single line diagram in Fig. 6. The switchboard sections are connected via bus tie / bus link breakers. The SWB rooms also contains the two 1600 kVA LLC transformer units, each of them including a 640 kVA/450 V distribution transformer for other non-propulsion consumers onboard. All thrusters are of fixed pitch design driven by frequency controlled electric motors. The frequency converters are all made by Wartsila. These are 12 pulse type based on IGBT transistor technology with PWM and advanced vector control.All switchboards onboard are made by Wartsila. They are designed to withstand the high short circuit current that may occur when all main generators sets, LLC transformers and 230V transformers are running in parallel.

The electric system is designed in such a way that it will give a selective disconnection of any electrical fault on the main bus bars and main feeders, with protection of the individual electric components. For instance a short-circuit on bus bar A1 will open bus bar breaker towards bus bar A2 as well as bus link towards bus bar B2, before trip of generator breaker.

The 690V main switchboard is designed and constructed for operation with both open bus tie / bus link and with closed bus tie / bus link. Bus bar A1/A2 and B1/B2 will in normal operation be connected via the LLC transformers. For DP2 class operation, the safest set-up will be with bus tie breaker in open position. However, closed operation is also allowed, giving a more efficient utilisation of the diesel generators, with better engine running condition and lower fuel consumption. The switchboards can be controlled and monitored locally or automatically from the IAS/PMS. The PMS is arranged for automatic operation for both open and closed bus tie.

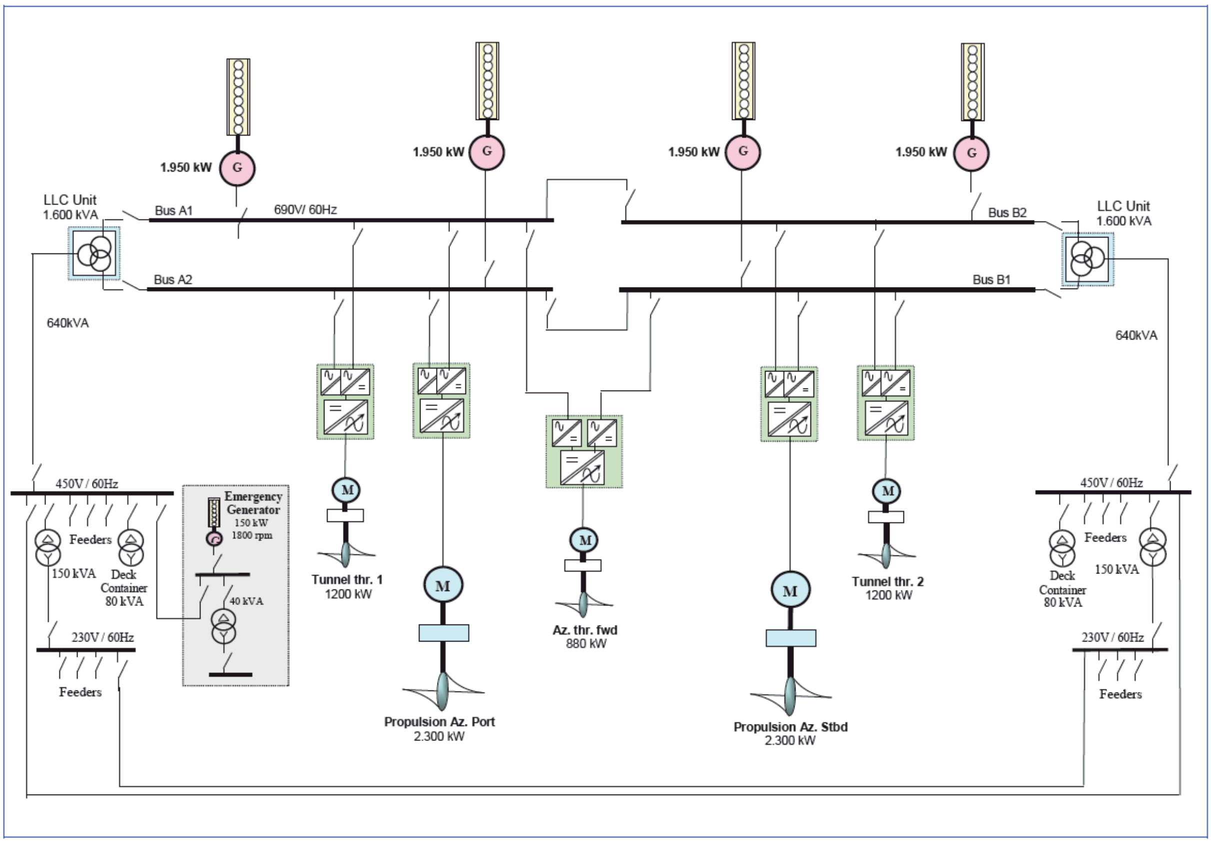


Fig 6 . Viking Queen – Machinery and Propulsion arrangement

Worst Case Single failure Consequences – FMEA test

The FMEA analysis and tests carried out onboard Viking Queen has proven the special features of the LLC. One of the conclusions from the FMEA report is as follows:

“From the design documentation, we can not find any single fault that will stop or disconnect more than 1-one generator set simultaneously. If short circuit in 1-one of the 4-four 690V busbars, or in the LLC transformer occurs, then, the bus-tie breaker will open and prevent generators on other parts of the main switchboard from tripping”. The above conclusion was tested and proven during the FMEA tests. Table 1 summarizes the remaining thruster capacity during loss of individual busbar sections. For all of these busbar failures, the remaining generator capacity will be 5850 kW (3 x 1950 kW) which is 75% of the full installed power.

The total remaining propulsion capacity during loss of busbar A1 or B1 will be 5690 kW, which corresponds to 73% of total installed power. Loss of busbar A2 or B2 will have even less consequences, as 79% (6130 kW) of the full power still will be available for propulsion.

	Remaining thruster capacity (%)					Remaining
Loss of Busbar	**Main PS**	**Main SB**	**TT 1**	**TT 2**	**Fwd Azim**	**prop. power (kW / %)**
A1	50	100	50	100	50	**5690 / 73**
A2	50	100	50	100	100	**6130 / 79**
B1	100	50	100	50	50	**5690 / 73**
B2	100	50	100	50	100	**6130 / 79**

Table 1. Remaining thruster capacity during loss of one busbar

Full scale measurements of electric Losses

Electric losses in the propulsion system for Viking Queen have been measured in full scale. The principal setup of the measurement is shown in Fig 7. The losses have been measured between the output from the diesel engines and the output from the electric propulsion motors by use of strain gauges. The measurements were carried out on the port distribution side, and in order to eliminate the influence from the utility power (power for hotel and other auxiliary consumers) all these consumers were supplied and isolated to the starboard side by running with open switchboard connections.

Thus during the tests the port diesel-generators G1 and G2 were only supplying power to the port main propulsion frequency converter and a minor amount of power (3-10 kW) to the port thruster utility system (lubrication and steering). This thruster utility power has been excluded from the efficiency calculation in order to get the most correct results for the pure electric losses.

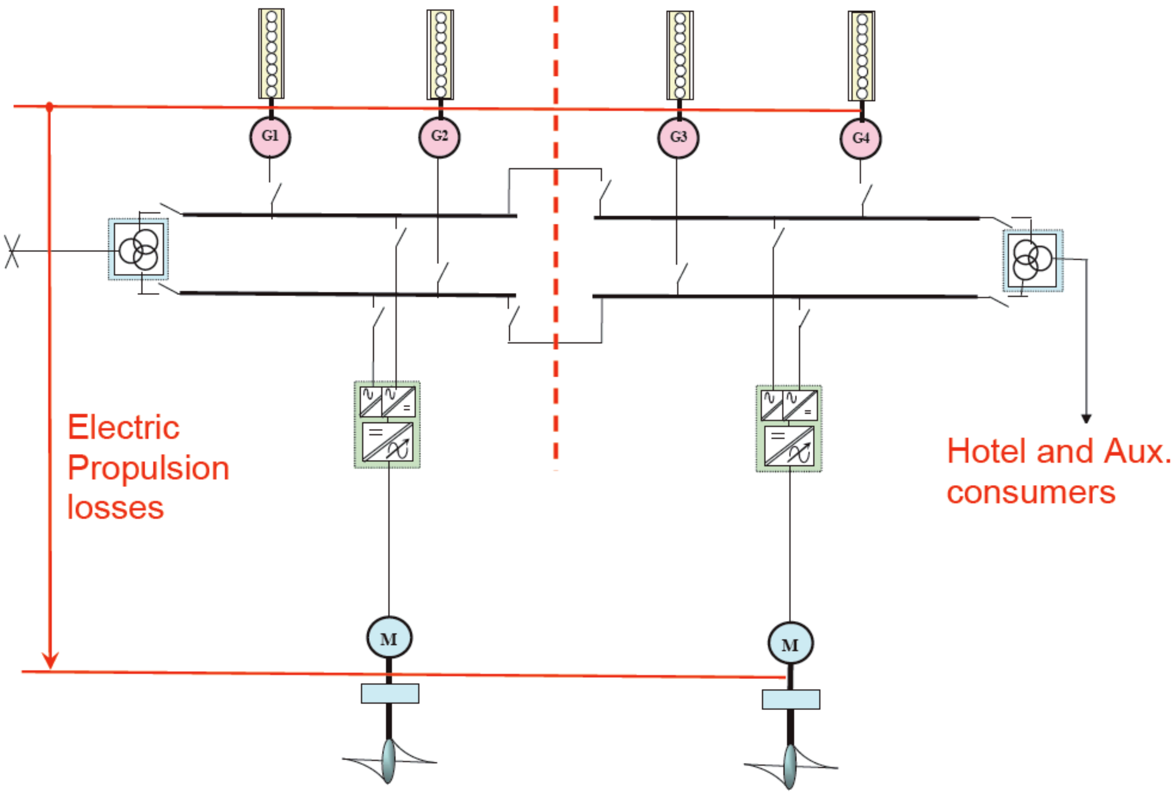


Fig 7. – LLC - Full Scale measurements of electric propulsion losses

A summary of the results from the full scale measurements are shown in the table 2 below. As can be seen the measured electric losses are in the range of 4 – 7 % which is remarkably low. Also the losses are higher in the tests where only one generator (G1) takes the whole load on port switchboard. This is as expected as in this case some of the power to the propulsion converter needs to pass through the LLC transformer giving some additional losses. When the electric load is equally distributed between G1 and G2, practically no power is passing via the LLC transformer, which explains the lower losses measured during these tests.

	Test 1		Test 2		Test 3		Test 4		Test 5	
	Port	Stb.	Port	Stb.	Port	Stb.	Port	Stb.	Port	Stb.
Generators running	**G1**	**G3**	**G1 G2**	**G3**	**G1**	**G3 G4**	**G1 G2**	**G3 G4**	**G1 G2**	**G3 G4**
Diesel Power (kW)	1261	1526	1253	1531	1692	2021	1676	2078	2253	2588
Propulsion Power (kW)	1176	1139	1198	1155	1578	1555	1603	1592	2129	2121
Electric losses	**6.74%**		**4.39%**		**6.74%**		**4.36%**		**5.50%**	

Table 2. LLC - Results from Full scale measurement of electric propulsion losses

SELECTION OF THE OPTIMUM PROPULSION CONCEPT

Selection of the most suitable propulsion concept should be based on the specific operational requirements for each individual vessel designs. This chapter will address the propulsion system selection and optimizing process, and give recommendation to which elements and operational parameters that need to be included in order to assure the most efficient and reliable propulsion installation. The major differences between a diesel-mechanic- (DM) and a diesel electric (DE) system will be highlighted

Figure 8 shows schematically the different power losses in a propulsion train. The overall fuel efficiency can be defined as the ratio between the effective thrust power produced by the propeller(s), and the power (energy per time unit) in the fuel supply to the diesel engines.

The diesel engine losses are typically around 50%, and will depend somewhat on the running condition (load and rpm) of the engines. Hence in an optimized system the engines should run in conditions giving the lowest possible overall specific fuel consumption. An optimum engine running requirement is favoring a DE system as the individual engine loading for a given load condition may be optimized by the selected number of engines running.

On the other hand transmission losses are normally higher for a DE system compared with a DM system. Losses in generators, switchboards, transformers, converters and electric motors add typically up to more than 10%. But as we have seen with the Wartsila LLC, losses below 5% have been achieved in a DE system. For a DM system with direct connection to the propellers via reduction gear and shaft line, the losses are in the range of 3%.

The propulsion losses are influenced by several factors. Selection of size and type of propulsors, together with hull interaction may have a significant effect. Also, the propellers should be operated in the most optimum way. For CP propellers, operation at high rpm and low pitch should be avoided as it gives high propeller losses. This situation may be a challenge during operating of DM installation with shaft generators as the “zero-pitch” losses for a CP propeller at full rpm, may be as high as 15-20% of the full propeller power.

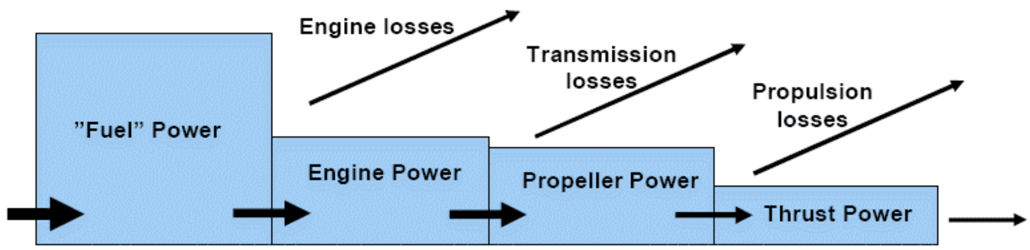


Fig 8. Power losses in a propulsion system

For vessels with a large variety of operation modes a combination of Diesel Electric- and Diesel Mechanic propulsion may be a favorable solution (Hybrid Propulsion System)

This system combines both DM end DE propulsion, thereby facilitating the following operation modes:

- Main propellers driven by diesel engines only (DM)
- Main propellers driven by both diesel engines and electric motors (Booster mode)
- Main propellers driven by electric motors only (DE)

In a hybrid system, the best characteristics for DM and DE systems are combined. At higher propeller loads DM propulsion is used taking advantage of the lower transmission losses, and the operation is outside the "zero-pitch" losses area. At lower propeller loads, the main propellers are run diesel electric with full frequency/rpm control. Thus the "zero-pitch" losses in DM systems are eliminated.

Figure 9 shows an example of a LLC Hybrid Propulsion System for a 300 T Anchor Handling Tug Supply Vessel (AHTS).

The main propellers can be operated diesel-electric by the two 2400 kW electric motors connected to the reduction gear boxes. For the full 300 ton bollard pull condition, the 2 x 8000 kW main diesel engines and electric motors are running in parallel (booster mode) giving a total propulsion power close to 21000 kW

The reason for selecting a Hybrid Propulsion System for this AHTS vessel is the wide operation area with a big span in required propulsion loadings (ref. the vessels mission profile in figure 10). 60% of the total operation time is with low propeller power. The differences in power consumption for the main propellers in "DP light" mode have been calculated, and the difference between DM and DE mode is shown in figure 11. As can be seen from de figure the "zero-pitch" losses are substantial for DM propulsion. Converted to fuel consumption, the saving by using DE propulsion in "DP light" mode is about 50% or 5 ton /24 hours operation.

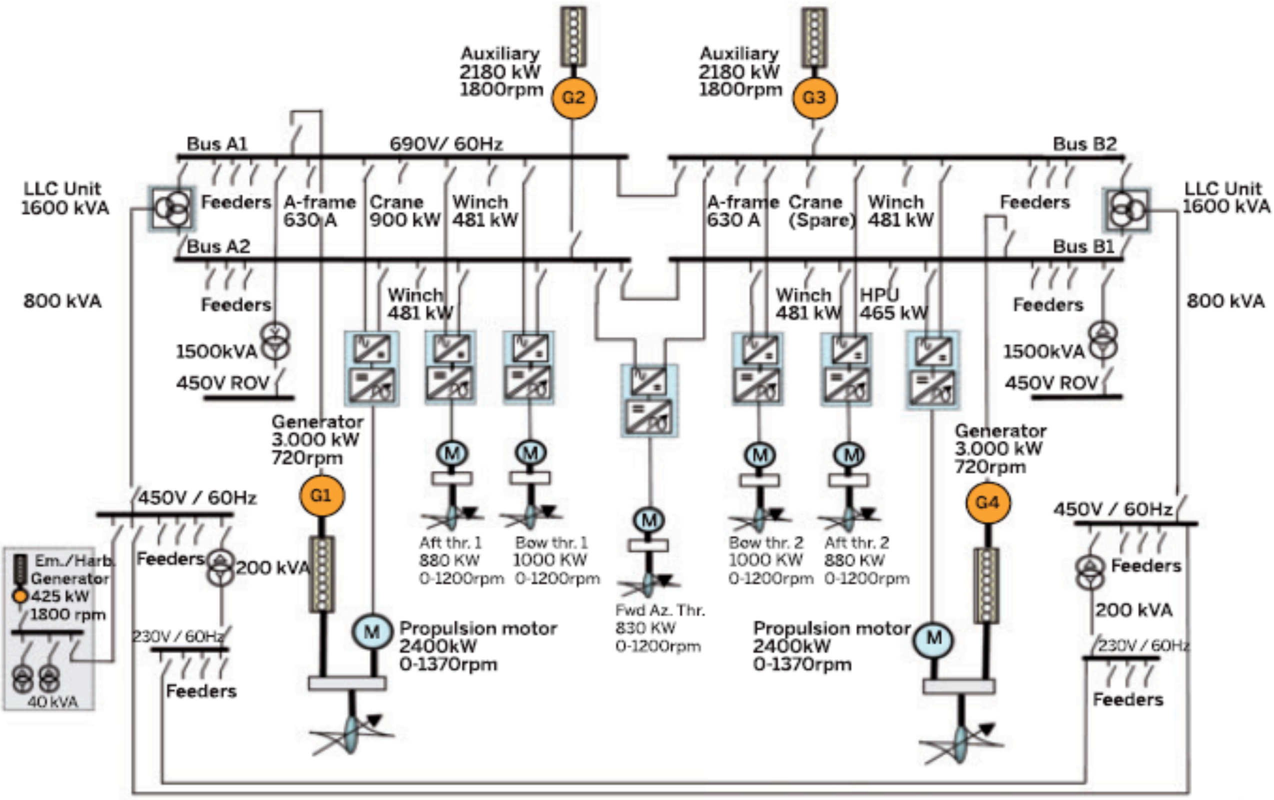


Fig 9. Hybrid Propulsion System for a 300 T AHTS

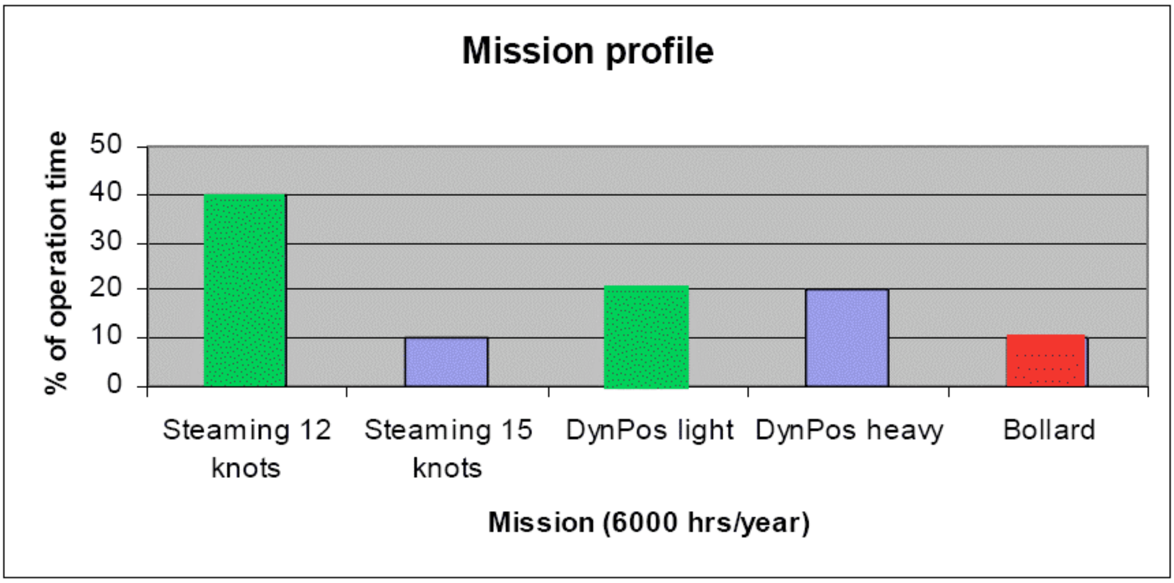


Fig 10. Mission Profile for a 300 T AHTS

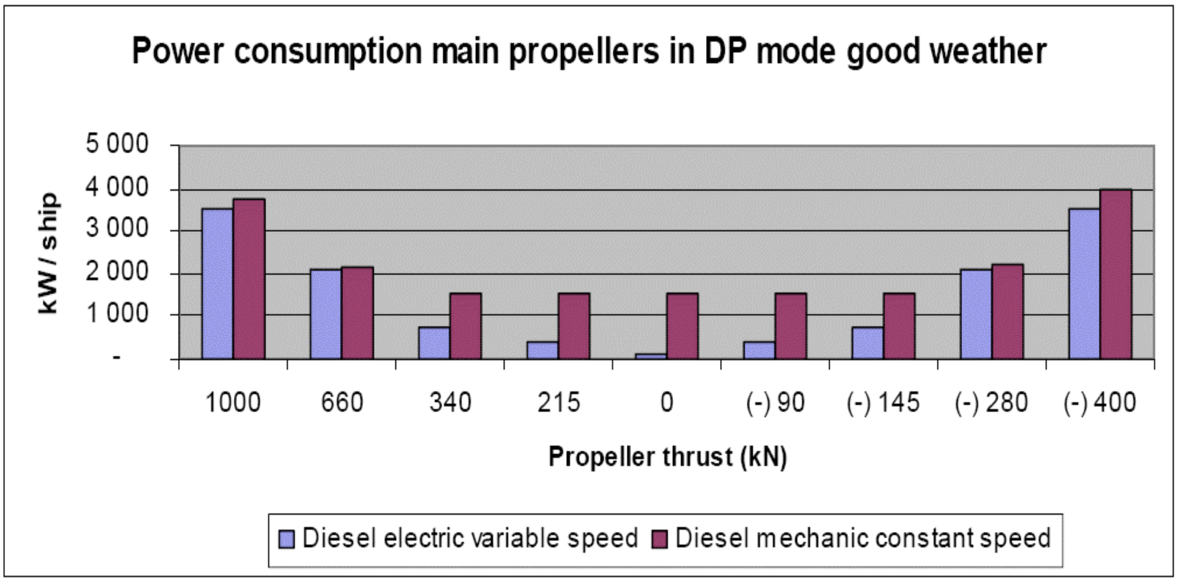


Fig 11. Power consumption for main propellers in DP mode – DM versus DE propulsion

CONCLUSION

Diesel electric propulsion systems are widely used in Offshore Support Vessels. This is primarily caused by the need for high flexibility and reliability during operation. As a further development in this field, the Wärtsilä Low Loss Concept (LLC) for diesel electric propulsion has been introduced. This system gives even higher overall reliability in the electric power distribution, with less single failure consequences compared with conventional systems. Thus DP vessels fitted with the LLC will be able to increase the DP capability. The LLC is also reducing the electric losses in diesel electric propulsion systems, a feature that has been confirmed by full scale measurements. Total electric losses of about 5% have been demonstrated, which is significantly lower than common figures for conventional diesel electric systems.

For vessels with extremely different operational conditions like AHTS vessels, the advantages of the low transmission losses with diesel mechanic propulsions can be maintained by a hybrid DE/DM solution. For other types of vessels a pure mechanical propulsion system may still be the preferred solution. Therefore, selection of the most suitable propulsion system for a given vessel should always be done by taking the operational requirements into account.

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Fuel Saving and Reduction of Environmental Emissions in OSV and AHTS by use of Electric and Hybrid Propulsion

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ABSTRACT

Electric Propulsion is used in a wide range of vessel types and applications. In Offshore Support Vessel (OSV) segment, a large portion of new buildings are equipped with electric power plant with variable speed electric motors to control the main propulsion and thrusters.

These days we also see a tendency that advanced Anchor Handling Tug Supply (AHTS) beeing more and more equipped with a combination of Diesel Mechanical and Diesel Electric propulsion (HYBRID) because economical analysis shows that potential operational benefits over the vessels life time are even higher than for Platform Supply Vessels (PSV). During the last years, the focus and restriction on environmental emissions has been, and is further expected to be strengthened. The use of electric propulsion will also contribute to reduction of green house gases due to the lower fuel consumption. New techniques for control and power conversion are made to further improve the environmental footprint of the vessels, such as the use of active rectifiers in low voltage installations

Electric propulsion can prove substantial savings in fuel costs and this will hopefully stimulate the oil industry to select the most optimum and innovative solution, and thereby also reduce the environmental impact for offshore operations.

INTRODUCTION

Since mid 1990's, OSVs have been equipped with electric propulsion /1/, Fig. 1, where the main propulsors and station keeping thrusters have been driven by variable speed electric motor drives, being supplied from the common ship electric power plant with constant frequency and voltage. Thrusters and propulsors are normally of fixed pitch propeller design (FPP) that reduces the mechanical complexity of the units, and the electric power is normally supplied from fixed speed combustion engines; diesel, gas, or dual fuel.

The electric propulsion has shown to give a significant benefit of reduced fuel consumption and environmental emissions from the fleet of PSV's and also in other ship types in the fleet of offshore support vessels with typically 15-25% savings depending on the operation profile, and 40-50% in DP operations. Electric propulsion has become the primary choice of vessel designs in many of the offshore oil and gas fields, including the North Sea and Brazil, and increasingly being specified by oil companies in new areas in order to reduce the operational costs and emissions.

With the proven fuel saving in the PSV application, it is yet some kind of paradox that the large fleet of anchor handling vessels, mainly are designed with direct mechanic propulsion system; even though the same effects that contributes to the fuel savings in the PSV vessels, are also existing and even to a larger extent in anchor handling, tug, and support vessels (AHTS). The reason for this is mainly due to the higher investment costs, which must be paid back through the day rates of the charters. If the charterer is not willing to differentiate the pricing between vessels with high and low fuel consumption, and just pick up the bill for fuel and environmental emissions, ship owners will not have many other options than to go for the cheapest solution in order to be competitive; even though knowing that this is not the optimal solution with regards to life cycle costs.

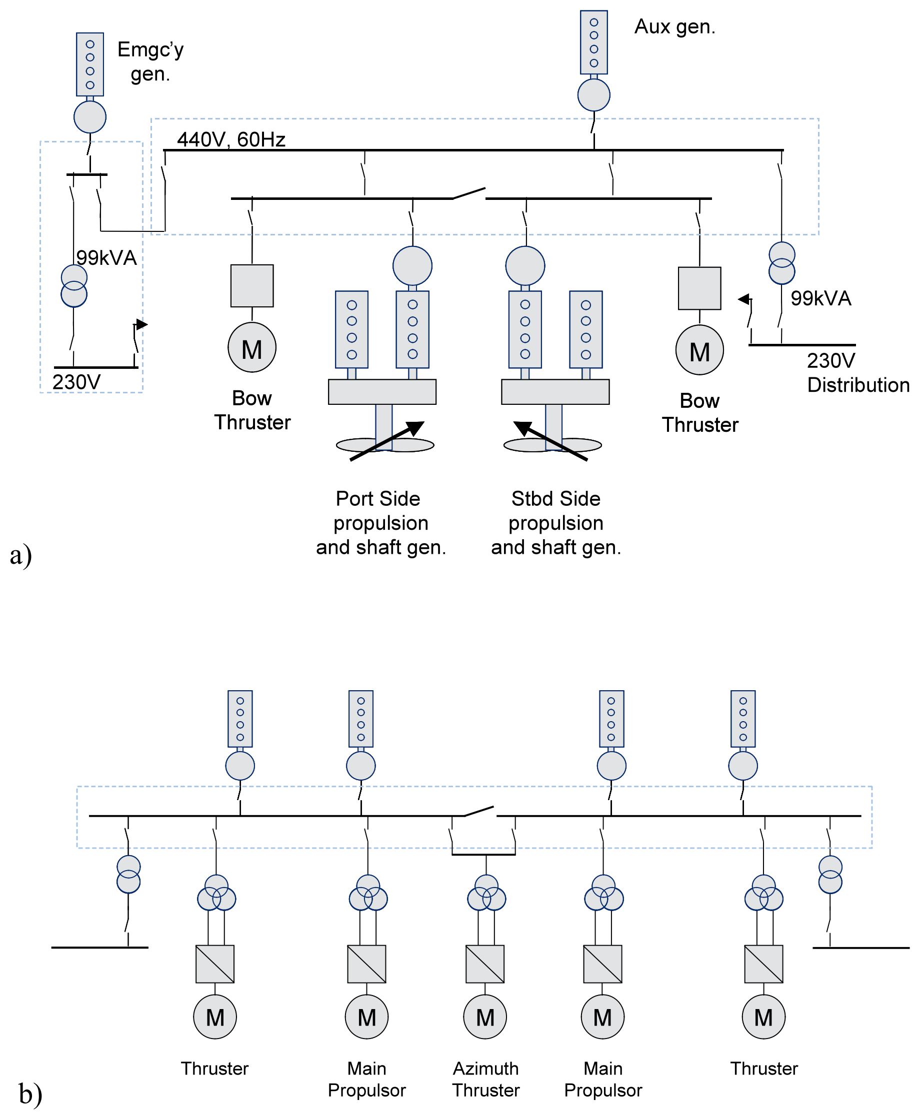


Fig. 1: a) Conventional direct mechanical propulsion, and b) electric propulsion concept for OSV.

THE ENVIRONMENTAL CHALLENGE

Not many years ago, the environmental aspect was rather absent in the marine industry. In the global agreements on green house gases, the marine industry has "been lucky" not been involved in the balance sheet and limitations of emissions. But this is changing.

Not only is the industry themselves now a driver for fuel reduction, as it has a clear economical benefit too, but also the society will not allow the marine industry to be unaffected of the common environmental challenges.

So far, IMO and legislative restrictions have mainly been made on NO_x, SO_x, and particle emissions, which are local or regional concerns, but severe enough for the affected areas. It is though clear signs, that also CO_2 emissions from the marine industry will be included in the coming global greenhouse gas reduction agreements.

While waiting for the regulations to come, it is worth to notice, that fuel saving and emission reduction is cost saving already today. Studies, as /2/, shows that there are a ample amounts of measures that can efficiently reduce the fuel consumption and emissions with profitable payback with today's fuel prices. It is therefore not only an issue of technological development, but also a need to further increase the awareness in the industry, and ensure that the one who pays for the initial investment, also directly gets the benefit of cost saving.

VARIABLE SPEED DRIVES FOR ELECTRIC PROPULSION

The variable speed drive (VSD) for propulsors and thrusters is one of the most essential components in a power plant for electric propulsion.

The VSD consists of:

- Electric motor, normally asynchronous (induction) motors, but also synchronous motors for the high power range. Other types of motors used in special applications; such as permanent magnet motors and DC motors.
- Frequency converter, converting the fixed voltage and frequency of the network to a variable voltage and frequency needed to adjust the speed of the electric motor.
- Optionally line filters or transformers, depending on configuration for reducing the harmonic distortion of currents flowing into the network and voltage adjustment where applicable.
- A control system, consisting typically of a motor controller and an application controller for the propulsion / thruster control, taking care of the control functions as well as monitoring and protection of the VSD.

For the power level needed for OSV propulsion, the Voltage Source Inverter (VSI), Fig. 2, is the dominating topology of frequency converters and used by most suppliers to this market. DC drives, Current Source Inverters (CSI) and Cycloconverters are rarely used and being phased out from new buildings of OSVs. Therefore, this paper only considers the VSI in various configurations.

The voltage source inverter consists of a rectifier, a DC link with voltage smoothing capacitors, and an inverter unit as the main components. The DC link may where required be equipped with a breaking chopper to dissipate wind-milling power from the propeller in rapid speed variations or in crash stop conditions of the vessel.

As the propulsors and thrusters are driven with electric power, they are essentially decoupled from the power source, which can in principle be anything that produces electric power. This opens for the use of new power sources, but also for flexibility in adapting to new sources that should be available in the future, throughout the life time of the vessel.

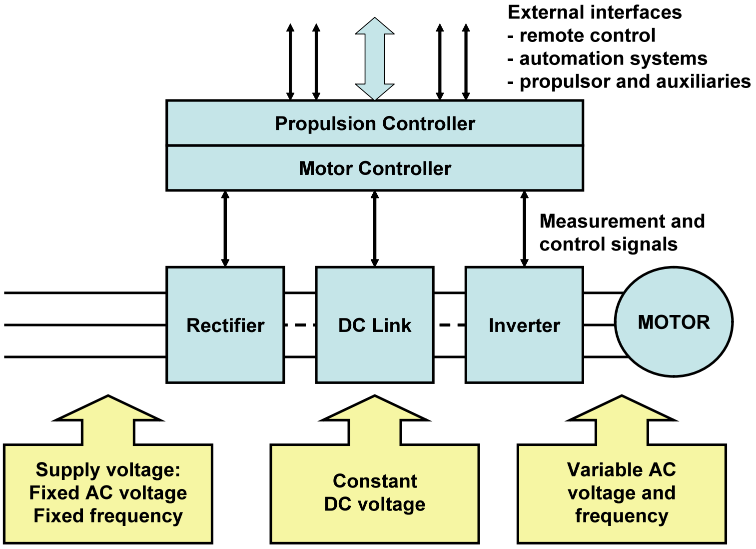


Fig. 2: The basic modules for a Voltage Source Inverter (VSI).

SYSTEM CONFIGURATIONS FOR ELECTRIC PROPULSION

As previously shown, the basic topologies for the VSD are relatively similar among the various suppliers to the application of electrical propulsion.

From a ship application perspective, one of the main technical differences are related to how these products are put together in a system configuration for electric power generation, distribution, and propulsion / station keeping. Several system configurations are applied, of which the most common ones are shown in Fig. 4. For each main configuration, there may be several variants for optimization to the actual requirements for each vessel.

The main challenge in system design is to meet the class requirements and ship specific requirements at a minimum total cost including equipment and installation costs, and with a best possible life cycle economy. Each vessel may have its own specific requirements, e.g. whether space is a scarce resource or not in the design. Propulsion transformers are large and heavy equipment, and the 6-pulse, active rectifier, and the Q12-pulse with phase shifted main voltages are examples on “transformer-less” solutions, but not without other penalties.

The 6-pulse and Q12-pulse solutions normally will require some kind of harmonic filter installations, unless significant restrictions and constraints of operations shall be applied, which may cause deterioration of the fuel economy of the prime movers and limitations of operational windows for the vessel. The Q12-pulse solution in particular depends on a complex main switchboard with two feeders to each frequency converter for balanced loads that is a necessity for maximum performance. Each feeder will carry a 6-pulse current that to some extent will enter the respective generators on the switchboard, or flow through the two primary sides of the distribution transformers, and give additional losses that to some extent will counteract the benefits of avoiding the losses in the drives transformer.

The active rectifiers increase the number of active components in the installation, and the complexity of the installation as each of the rectifiers requires a HF harmonic filter that introduces resonance modes of the installations that should not be excited by the switching frequency of the rectifier. Also, the size and costs of the frequency converter itself will increase, as well as the power losses in the rectifier, counteracting at least partly the benefits of the transformer-less design.

Hence, there exists no one “ideal” design for all vessels. The different solutions have different characteristics, and only when considering the requirements and limitations for a vessel design, the best solution can be applied.

HYBRID CONFIGURATIONS

An alternative to the full electric solution is the combination of mechanical and electric propulsion systems, the so-called hybrid propulsion, Fig. 5. Here, the vessel can be operated in either;

- Full electric propulsion, for low speed maneuvering, transit, and DP
- Full mechanic propulsion, for tugging and high speed transit
- Hybrid (combined) electric and mechanic propulsion, where electrical equipment can be used as booster for the mechanical propulsion system, used to obtain maximum bollard pull.

In terms of installation costs, such hybrid solutions will be cheaper than pure electric solutions, and will in fuel cost calculations be quite similar in fuel consumption to the electric solution. Therefore, several of the new AHTS designs are based on such hybrid solutions, especially for AHTS vessels with high bollard pull.

However, one should not disregard the increased mechanical complexity of such hybrid systems, where it is required that the crew more actively and manually selects operational modes optimal for the conditions. In pure electric propulsion systems, it is much easier to automatically optimize the configuration of the power and propulsion plant, ensuring that the system always will operate closest possible to optimal conditions without or with reduced manual interactions.

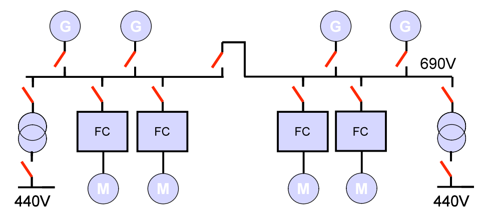
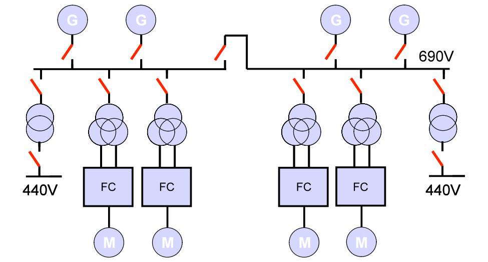
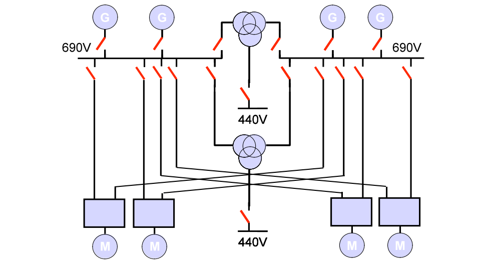
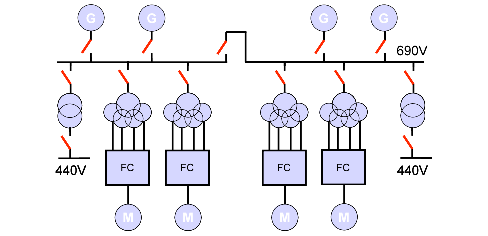
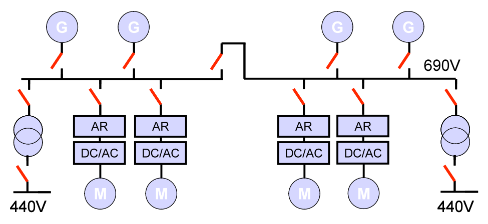
	6-pulse: • No drive transformers • Harmonic filters needed to get THD<5% • Weight: Low • Footprint: Low • Operational constraints: Medium • Total efficiency: Approx: 90-91%
	12- and quasi 24-pulse • 3-winding transformers, phase shift for Q24 • Harmonic filters for 12-pulse, not Q24 • Weight: High • Footprint: High • Operational constraints: Low/medium • Total efficiency: Approx: 90%
	Quasi 12-pulse with phase shifted mains voltages /5/: • No drive transformers, oversized distribution transformers for power transfer • Weight: Medium • Footprint: Medium • Operational constraints: High • Total efficiency: Approx: 90% included harmonic losses in generators and distribution transformer
	24-pulse: • 5-winding transformers (or 2 x 3-winding) • No harmonic filters • Weight: High • Footprint: High • Operational constraints: Low • Total efficiency: Approx: 90%
	Active rectifiers: • No drive transformers • High frequency input filters for harmonics • Weight: Low • Footprint: Medium • Operational constraints: Low / Medium • Total efficiency: Approx: 90-91%
Glossary:	690V: Main switchboard voltage 440V: Main distribution voltage G: Generator M: Motor (Propulsors and thrusters) FC: Frequency Converter AR: Active Rectifier DC/AC: Inverter

Fig. 4: Alternative system configurations with main characteristics. 690V Main SWBD voltage is shown, high voltage, e.g. 6.6kV is used when generator capacity typically exceeds about 10MW.

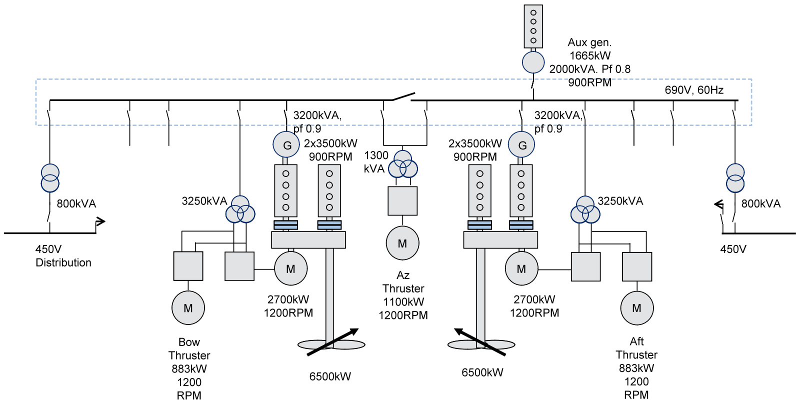


Fig. 5: Hybrid electric and mechanical propulsion for 200+ metric ton AHTS.

A NEW ELECTRIC POWER SOURCE INTERFACE

The electric power sources, i.e. the generators, produces 60Hz (or 50) at a standard voltage of typically 450 or 480V, 690V, and in larger installations, 6.6kV or 11kV. This is based on traditional design, where standard industrial components have been applied.

But with the use of electric propulsion system and frequency converters, that nevertheless transfer the majority of the electric power, it is not necessarily the optimal interface for the electric power source. This could equally well be a variable frequency, or voltage, or fixed DC voltages. We can call it the "EPSI"; the electric power source interface.

In ship applications, typically 80% or more of the generated power is rectified already in order to control the speed of the propulsion motors and thrusters. Why not then redefine the EPSI to a new standard that better suits the frequency converters, such as DC; e.g. 1000VDC or 4500VDC.

An example of a configuration where a DC voltage is used as the EPSI; and with the potential to connect multiple power sources into the grid with very different characteristics and control dynamics is shown in Fig 6.. This is yet not a solution available today, as there are certain issues regarding selectivity and fault tolerance in the DC distribution that needs to be clarified with class societies. The configuration, though, has a lot of advantages, such as higher efficiency, flexibility for future power sources, and fewer single points of failures (except the DC distribution).

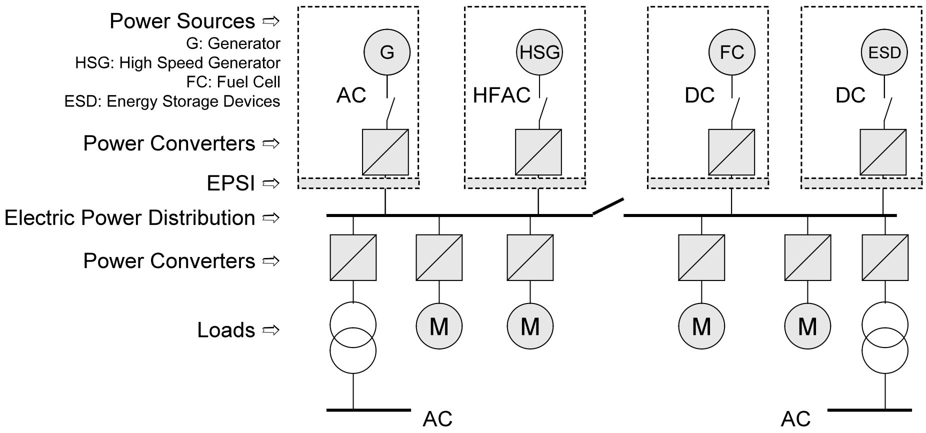


Fig. 6: Future: Defining a new standard for electric power source interface (EPSI); e.g. 1000V or 4500V DC; simplifies the use of multiple power sources and decouple the control and basic electric characteristics of them.

HARMONIC MITIGATION

Frequency converters are inherently non-linear components due to the switching characteristics of the rectifier components, meaning that they do not draw sinusoidal currents from the network, even though being fed by sinusoidal voltages.

The non-sinusoidal currents into the rectifier consist of a fundamental voltage, and a series of harmonic components with a wide content of frequencies which depends on the rectifier type and system configuration. For the type of converters that have the highest level of distortion in the currents, typically those with 6-pulse, 12-pulse, and Q12-pulse rectifiers, the level of harmonic distortion in the currents may lead to voltage distortion above the class limits. Most class societies now have adapted the IEC 60092-101 requirement,.

When the limit of the applicable regulation will be exceeded with the decided frequency converter and system configuration also after optimizing the design of the generators and transformers in the plant, there are still several ways to manage the harmonic distortion level.

A harmonic filter can be applied. There are two main different types for harmonic filters; passive LC filters (alternatively damped LCR) and active filters /1/. For ship applications, passive filters are more commonly applied, due to their lower costs; especially since they can be used at lower voltage levels in the distribution system to filter the voltage distortion not necessarily in the complete installation, but for the sensitive equipment only.

ACTIVE RECTIFIERS

Active rectifiers are primarily used for three purposes;

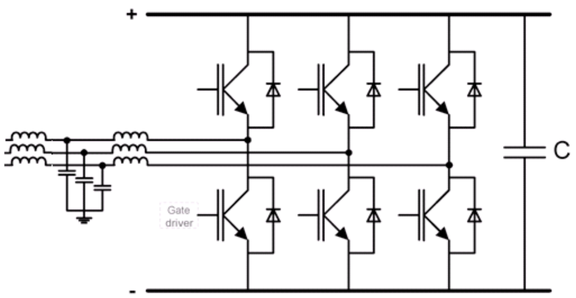
- Feeding power back to the network
- Reducing network voltage distortion
- Reducing weight and footprint compared to standard and multi-pulse drive systems; this has been the main driver for installation of active rectifiers in ship applications.

Active rectifiers have been widely used in industrial applications for several years, but not as much in ship systems as there have been some concerns on using rectifiers that can feed power and noise back to a relatively weak network on board the vessel. However, the latest years, active rectifiers have been more used also in ship applications, both for thrusters and propulsors, as well as various auxiliary systems.

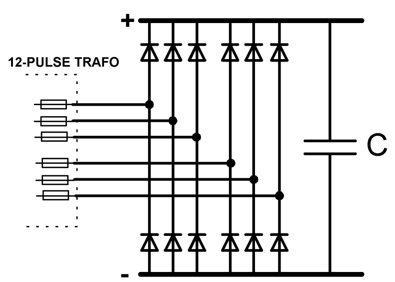
It should be noted and carefully considered, though, that the use of active rectifiers in weak networks, e.g. ship applications, requires special attention to system engineering.

In the active rectifier, Fig 7 a), the actively controlled IGBTs are shown, in contrast to the passive rectifiers with uncontrolled diodes in Fig 7 b). The IGBTs are used to control the network current to nearly sinusoidal; not that the LCL filter is required to filter out the switching harmonics in the input current, and that therefore the active rectifiers are not without the use of harmonic filters, however, the filters are normally integrated in the drive line-up and hence not so visible for yard and owner.

For the system integrator, this is essential to consider, as these filters may have resonance points in the high frequency areas, but also in the lower frequency spectrum that can be cumbersome to handle should there be passive rectifiers in the same network, which normally is the case.



a)



b)

Fig. 7: a) Active (IGBT) rectifier. b) Passive (diode) rectifier.

CONCLUDING REMARKS

In design of PSV, electric propulsion has become the new standard.

Although the same characteristics, with even higher potentials for saving, are present in AHTS vessels, still the majority of AHTS are made with conventional propulsion. This is mainly due to the lack of awareness of that the one party paying for installation, is not liable for fuel costs, and vice versa. This leads to non-optimal solutions, with higher fuel consumption and more environmental emissions than necessary.

New configurations are available; hybrid designs are used for reducing the difference in investment costs and make more flexibility in operations, although also more complicated for operation; and active rectifiers are used to reduce the footprint of the electric equipment.

This paper has presented the most commonly applied solutions in electric propulsion with the objective to give the decision makers background information to better understand the concepts and to make the most beneficial selection for the specific vessel's requirements; and hopefully contributed to a higher awareness of the environmental impact of the use of vessels. Further, that there not necessarily is a contradiction between economy and environmental awareness – fuel saving is both cost reduction and environmental beneficial.

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A Review of DP Station Keeping Incidents and Systematic Safety Management

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Abstract

Recent station keeping incidents are presented and their causes discussed, with a view to identifying limitations in DP station keeping ability. This is put into perspective for the users of DP vessels. Areas of improvement are suggested, including FMEA style, format and integration with a systematic ship safety management system in a wider ranging risk identification process.

INTRODUCTION

Ship/s management systems have evolved throughout recent years. Initially, the ISM Code was introduced as a way to address the increasing incidence of human related failures. More recently, Risk Assessment has been prescribed for analysis of high speed passenger craft (HSC) and Failure Mode and Effect Analysis (FMEA) has been prescribed for dynamic positioning (DP) systems.

Technology has rapidly changed and systems are more complex.

Technology has outpaced human experience. It is no longer sufficient or possible to rely on compliance with Classification Society Rules to ensure adequate redundancy and reliability. Reliability is now more critical in DP systems, particularly in diving operations.

As DP station keeping incidents still occur, it is now necessary to relook at the “usability” of the FMEA and its integration into the ship’s systematised safety management system to make the job of the crew easier and more effective.

HOW COMPLEX SYSTEMS FAIL

According to Richard I Cook (Bibliography Ref. 7), systems are inherently and unavoidably hazardous by their own nature. The frequency of hazard exposure can sometimes be changed but the processes involved in the system are themselves intrinsically and irreducibly hazardous. It is the presence of these hazards that drives the creation of defences against hazards that characterise these systems.

The high consequences of failure lead over time to the construction of multiple layers of defence against failure. These defences include technical components (e.g. redundancy “safety” features built in) and human components (e.g. training and knowledge) but also a variety of organisational, institutional and regulatory defences (e.g. policies, procedures, certification, work rules and drills). The effect of these is to provide a series of shields that normally divert operations away from accidents.

FAILURE MODE AND EFFECT ANALYSIS

A sudden DP vessel position loss is potentially very dangerous. Therefore active redundancy must be built into each part of the total DP system, and the vessel should not be operated in conditions where a single failure of any component would lead to a loss of position.

Redundancy in power supplies is commonly provided by having several prime movers, and splitting the generators and electrical supplies to thrusters and references into various switchboard sections, with suitable electrical fault protection systems.

Power supplies to DP computers and the low voltage side of position references are normally provided by one or more uninterruptible power supply (UPS) systems, which have battery back ups

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for main supply loss. Where it is required for DP Class 2 Notation, a vessel will have an FMEA Manual.

A FMEA is carried out to ensure that the total system operates as designed, such that no single fault will have a knock-on effect which will take out all generators, thrusters or position references.

In marine and offshore systems, the use of FMEA is particularly useful as a way to identify risk exposure and to see the interrelationship between factors that combine to allow a failure to occur. Whilst a qualitative approach is satisfactory, the benefit of FMEA can be particularly enhanced if probabilities are input into the model.

The FMEA example (see Fig 1 below) is extremely useful to identify all of the events leading to a failure, and to identify the criticality, or whether a single failure in a system would cause a catastrophe.

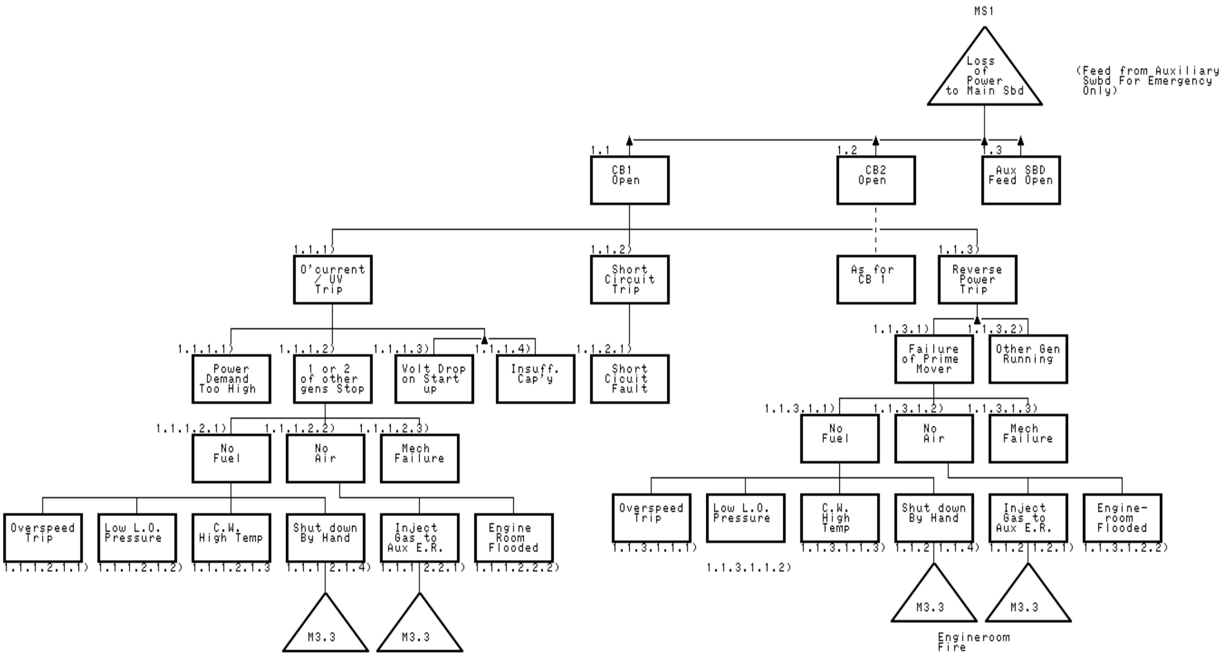


Fig 1 Failure mode and effect analysis example

HAZOP STUDIES

A simple and effective tool for the identification of hazards and the analysis of risk is through the listing of hazards and the study of cause and effect. Preventive or corrective measure can then be identified.

Fig. 2 Basic Hazard Analysis

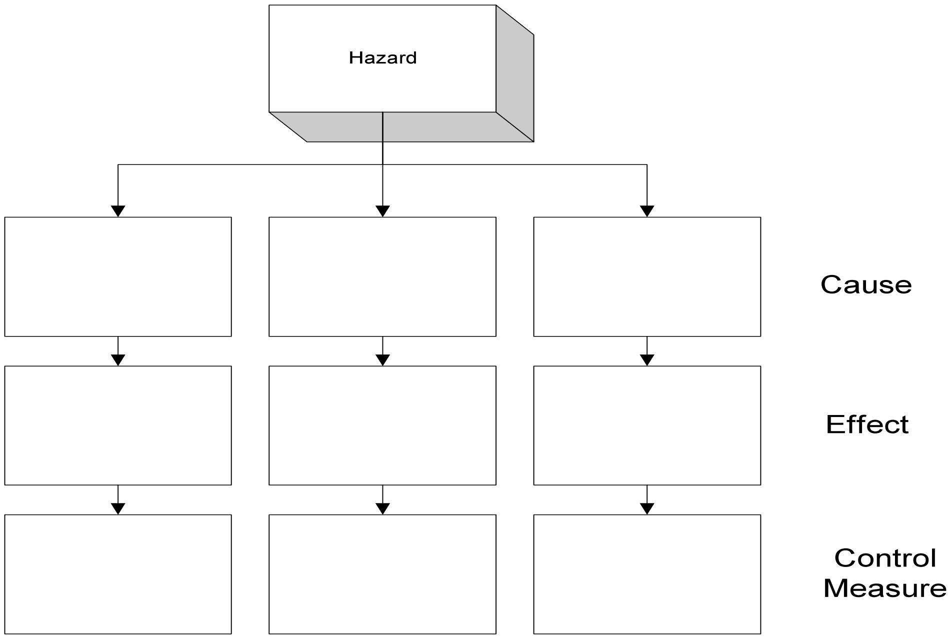


Fig 3. Typical HAZOP Table Example

Hazard	Cause	Effect	Corrective or Preventive Measure
Loss of sea water cooling when only one sea strainer is open and one is choked	1. Choked sea strainer, and 2. One of two sea chests closed	Failure of all engines, tripping of all generators, loss of electrical power to thrusters and loss of station keeping, extreme danger to property and life	1. Provide procedure for sea water cooling 2. Two sea chests must supply to the sea main during DP operations 3. Procedure for blowing the sea chest 4. Instructions to operator on emergency action to be taken

RELEVANT WEAKNESSES IN SYSTEMATIC MANAGEMENT SYSTEMS

According to the HSE "*Review of Methods for Demonstrating Redundancy in Dynamic Positioning Systems for the Offshore Industry*" (Bibliography Ref. 3), several weaknesses have been identified. Indicated below are those that are relevant to documented systematic management systems:-

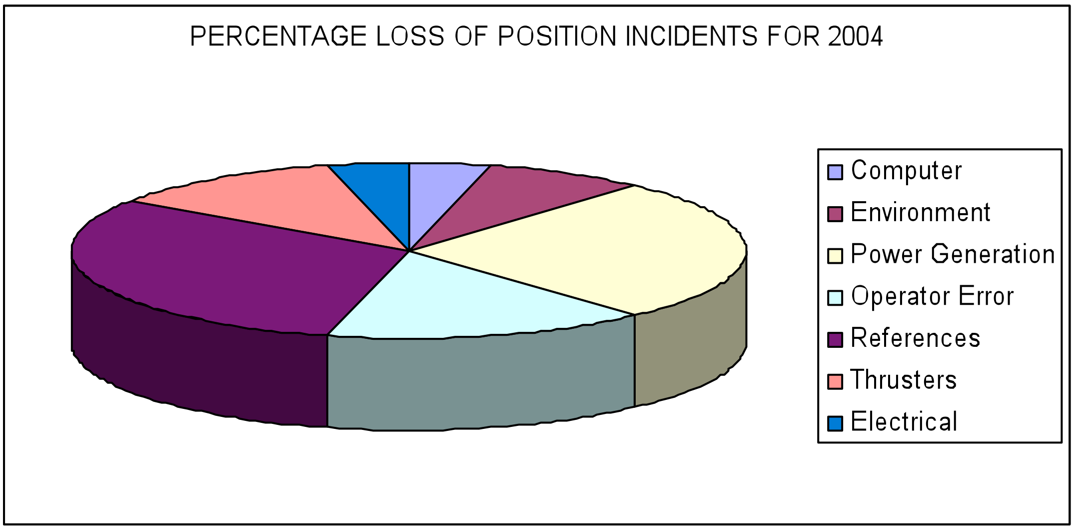
1. When FMEA is used to demonstrate that no critical single point failures can occur, there is a danger that failures may be overlooked.
2. Many FMEAs do not follow a systematic procedure for considering all relevant failure modes.
3. FMEAs mainly address technical failures. The human operator and the shore management are excluded from the definition of the DP system.
4. There is sometimes a lack of information about the failure modes of bought-in systems such as DP control systems and power management systems.
5. It is well known that some vessels are not operated in the way that is assumed in their FMEA.

STATION KEEPING INCIDENTS REPORTED FOR 2004

According to the International Marine Contractors Association (IMCA) (Ref 1 of Bibliography) the largest percentage of cases of lost position (see chart below) are caused by problems with references, such as wind and position. This is based on the understanding that accurate position depends on reference inputs. Most of the reference problems are related to the design or software deficiencies of the DP system.

Power generation is the next largest, and this is understandable, considering the complexity of the various pieces of equipment that the power depends on. Such failures may be related to operator error (combined with lack of procedures), or design and software errors.

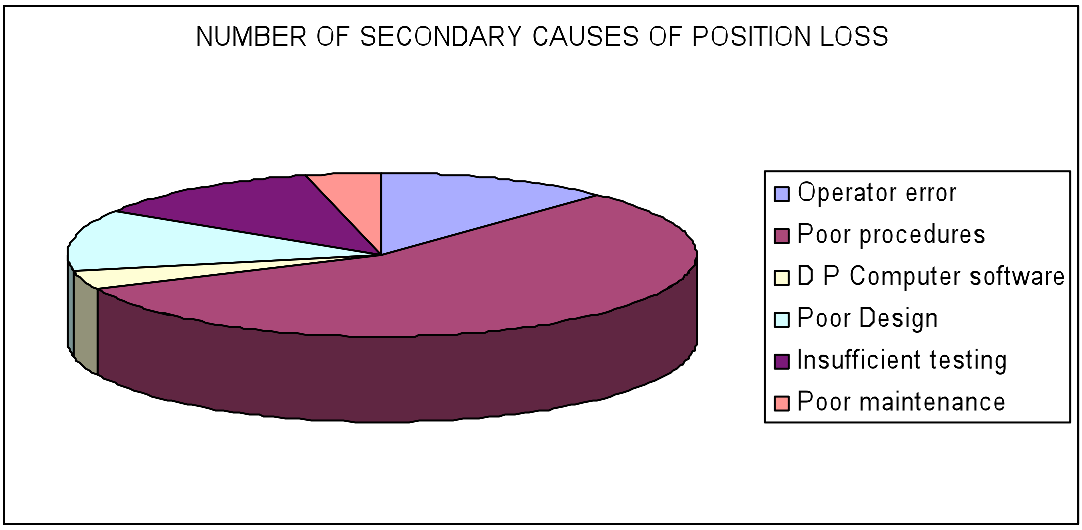
Those errors that are directly attributable to operator error may be related to a lack of procedures or human error.



If the secondary (or prime) cause of the incident is analysed (see chart below) then the largest culprit (other than operator error) is poor procedures or lack of procedures. This is a clear result of lack of anticipation of possible problems and the vessel operating procedures (whether engineroom, electrical or deck department) being non existent or insufficient.

This illustrates the weaknesses in systematic risk management mentioned above as follows:-

1. Individual failures or failure modes have not been anticipated in the FMEA and precautions were not put in the form of operating procedures, for avoiding the occurrence of such failures or failure modes.
2. The human operator and shore management are excluded from the definition of the DP system in the FMEA.
3. There is sometimes a lack of information about the failure modes of bought-in systems such as DP control systems and power management systems.
4. It is well known that some vessels are not operated in the way that is assumed in their FMEA.



SELECTION OF INCIDENTS

Seven incidents (Ref 1 of Bibliography) have been selected and shown in the appendix below.

These are chosen to provide an example of as many types of incident as possible.

PREVENTIVE ACTIONS

The purpose of FMEA analysis is to make the incidence of failure as low as possible. However there will always be failures that are not anticipated. The Space Shuttle failures are an example where millions of Dollars are spent on FMEA analysis and tragic failures still occur, despite all of the efforts being made to prevent such scenarios.

Maritime risk management has much scope for improvement and the following can be considered to improve the situation:-

1. Integration of the vessel's deck, electrical and bridge operating procedures with the FMEA, with cross referencing between elements, so that a preventive precaution can be traced to a failure mode or effect.
2. Introduce procedures to ensure integration of the FMEA and operating procedures into the ship's and Company's safety management system to ensure that the system is constantly being reviewed and upgraded.
3. As there is sometimes a lack of information about the failure modes of bought-in systems such as DP control systems and power management systems, purchasers and operators need to be mindful of the need to request these, should the various suppliers fail to provide.
4. Systematic identification of hazards. Perform "brain storming" among a group of experts to maximise a list of possible failures as possible. Identification of possible hazards and also HAZOP (Hazard and operability) studies may be useful here.

5. The FMEA is required to be user friendly and ship specific by the IMCA Guidelines, and there are guidelines for the creation of FMEAs, however there is no requirement for the manual to be more instructive and to facilitate the Master and Chief Engineer to use the FMEA or the operations procedures in a “back to back” transparent manner.

DP OPERATIONS MANUAL

General

Operations (or operating) manuals have three main purposes.

Firstly a technical manual (a requirement of some Classification Societies) is intended to provide guidance for the DP operator about the specific dynamic positioning installations and arrangements of the vessel.

Secondly, a procedures manual (recommended by the International Marine Contractors Association (IMCA)) is required, which describes actions to be taken by personnel.

Thirdly, reference documents are necessary to provide historical and model data to compare present performance with past performance and also a source of fault tracing data.

Technical Operations Manual

This DP operations manual is intended to provide guidance for the DP operator about the specific dynamic positioning installations and arrangements of the specific vessel. The DP operations manual is to include but not be limited to the following information.

1. A description of all the systems associated with the dynamic positioning of the vessel, including backup systems and communication systems
2. Block diagrams showing how the components are functional related
3. A description of the different operational modes and transition between modes.
4. Definitions of the terms, symbols and abbreviations
5. A functional description of each system, including backup systems and communication systems
6. Operating instructions for the normal operational mode (and the operational modes after a failure) of the DP electrical or computer control systems, manual position control system, manual thruster control system, DP equipment (thrusters, electric motors, electric drives or converters, electric generators, etc.)
7. Operating instructions for the systems and equipment, indicated in the above paragraph, during failure conditions
8. References to where more specific information can be found onboard the vessel, such as the detailed specific operation instructions provided by the manufacturer of the DP electrical or computer control systems, manufacturer’s troubleshooting procedures for vendor-supplied equipment, etc.

Operational Procedures Manual

The manual will be ship specific and consist:-

1. Organisation and Responsibility
2. DP Philosophy
3. Standing Orders regarding DP Operation
4. DP Guidelines
5. DP Checklists
6. DP Trials Procedures
7. Change Control System
8. Control Station Security

Reference Documents to be Carried Onboard

1. FMEA Manual
2. Customer Acceptance Tests
3. DP Trial Results
4. Capability Plots
5. Vendors' Maintenance And Operating Manuals And Drawings
6. Ship and DP Interface Drawings
7. Ships drawings
8. IMCA Publish data on failure modes and station keeping incidents.

CONCLUSIONS

A systematised and integrated management system will ensure that risks, hazards and non-conformities are accurately identified and reported to preventing initiation or further repetition of a failure.

In Fig 1, loss of propulsion can be the effect of various events. As an example, main propulsion could fail because of such occurrences as choked sea strainers or a failure of a standby pump to start-up. Suitable operational procedures or checklists can significantly reduce the possibility of these events occurring.

Little has been mentioned about the role of operability and reliability as factors in the cause of losses. As systems are more integrated between FMEA and operating procedures, operability and reliability functions will be identified as having an important contribution to the state of affairs where an accident could occur.

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3. John Spouge, DNV Consulting, "*Review of Methods for demonstrating redundancy in Dynamic Positioning Systems for the Offshore Industry*" — HSE Books, UK HMSO 2004
4. B Jenner, '*Risk Control Through Quality Management (Future Development And Tools)*', IMAS 92: Quality of Shipping in the year 2000, I.Mar.E Conference, Volume 104, 3, paper 16, pp 125— 132, Marine Management (Holdings) Ltd (1992).

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6. ABS Rules, American Bureau of Shipping
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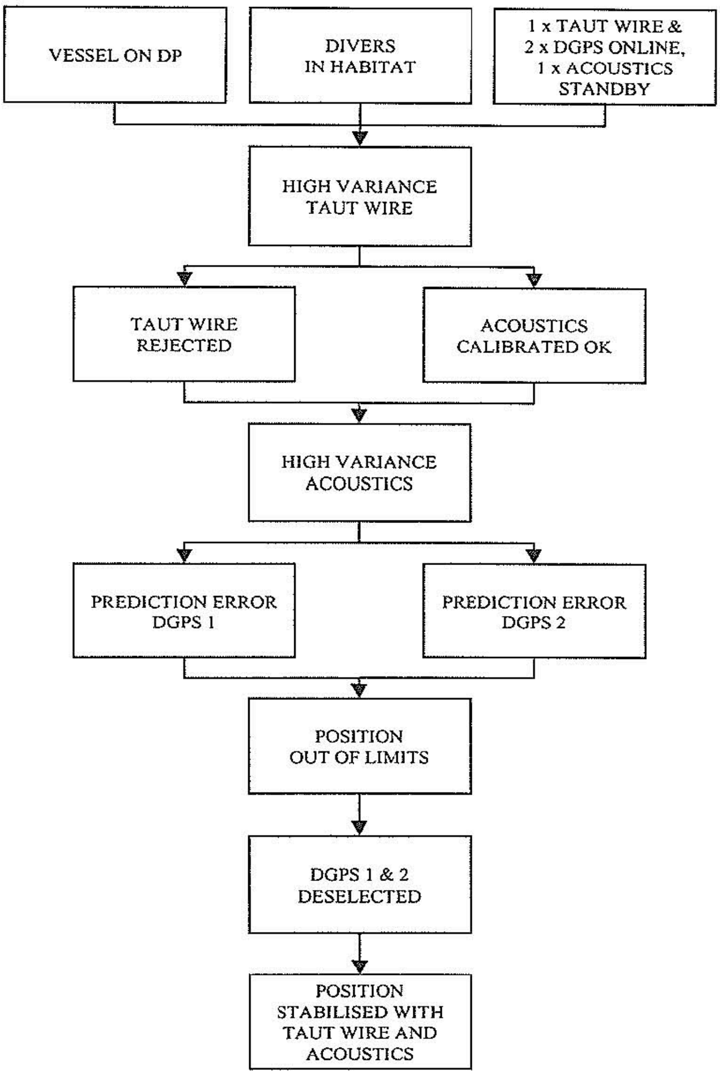
APPENDICE - Selected Loss of Position Incidents

Incident # 0403



Comments

At the time of the high variance on the LTW, the vessel was only operating on 2 x DGPS and the LTW. The acoustics were selected but almost immediately showed high variance because the vessel was moving, although the DGPS informed the model it was not. An investigation by the DGPS supplier showed that at the time there was an uplink failure to the satellite network. The second failure was that the data string for both DGPS inputs had been changed but not the software so the loss of differential was seen as steady fix, hence the drifting astern and 'perfect' reference symptom. The vessel will no longer use 2 x DGPS position references for dive support work but three independent position references.

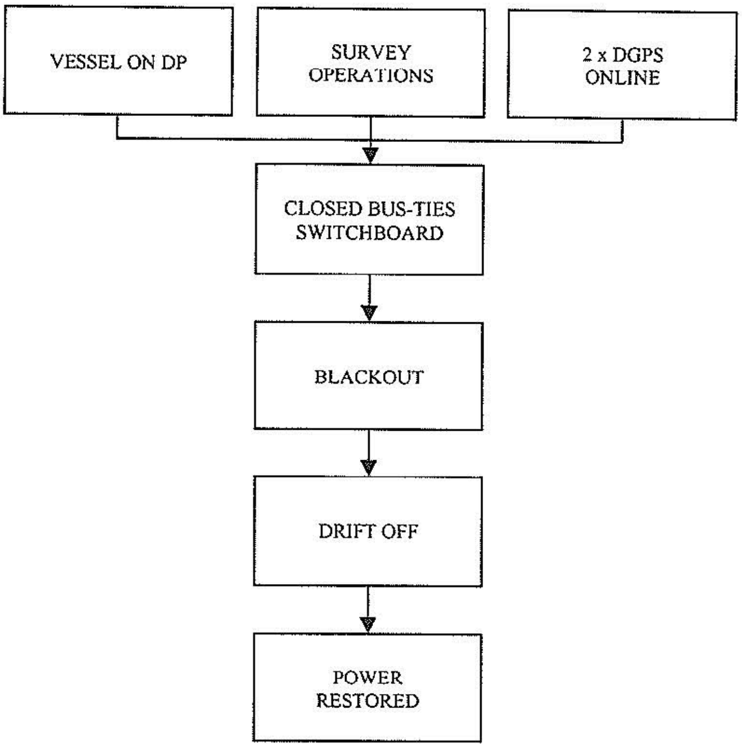
Trigger

References (DGPS)

Secondary Cause

DP Software

Incident # 0405



Comments

The same power arrangement was run for 24hrs without finding any problems. The two diesel generators that tripped were mega-tested and found to be OK. The PMS software was changed when the vessel was next in port.

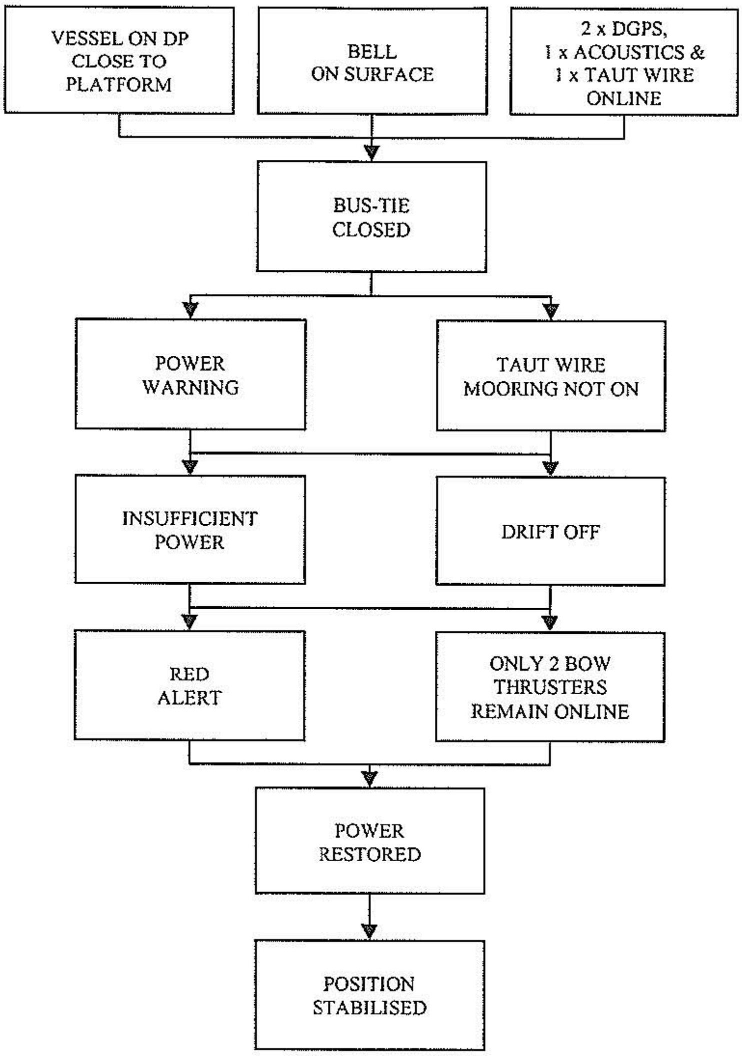
Trigger

Power Generation
(AVR)

Secondary Cause

Poor Design

Incident # 0406



Comments

The quick close valves for two of the three engines were inadvertently closed. These two engines slowed down leaving the power to be provided by the third healthy machine only. The two DGs then tripped along with one bow thruster and one aft azimuth thruster due to the tripped 6kV/440V transformer. However, the other aft azimuth thruster also tripped probably from the large voltage and frequency fluctuation and thus the vessel was left with just two bow thrusters.

The vessel resumed work with all four diesel generators on line and 6kV bus tie open.

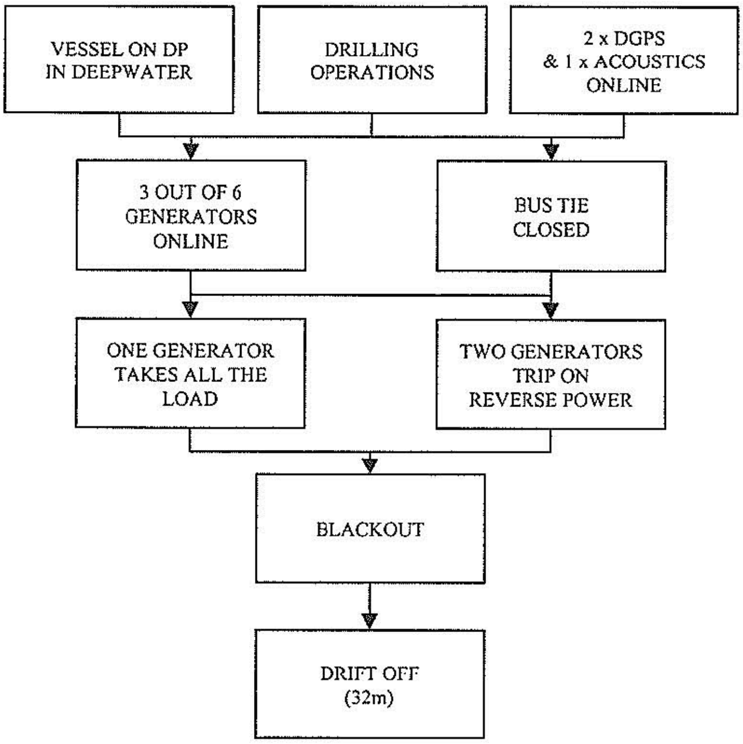
Trigger

Operator Error
(ERO)

Secondary Cause

Poor Procedures

Incident # 0410



Comments

One actuator on one diesel generator caused this blackout even though other factors contributed. The weather was calm and the drillers were pulling out of the hole at the time. Changes were made to standing instructions and procedures for governor actuator maintenance.

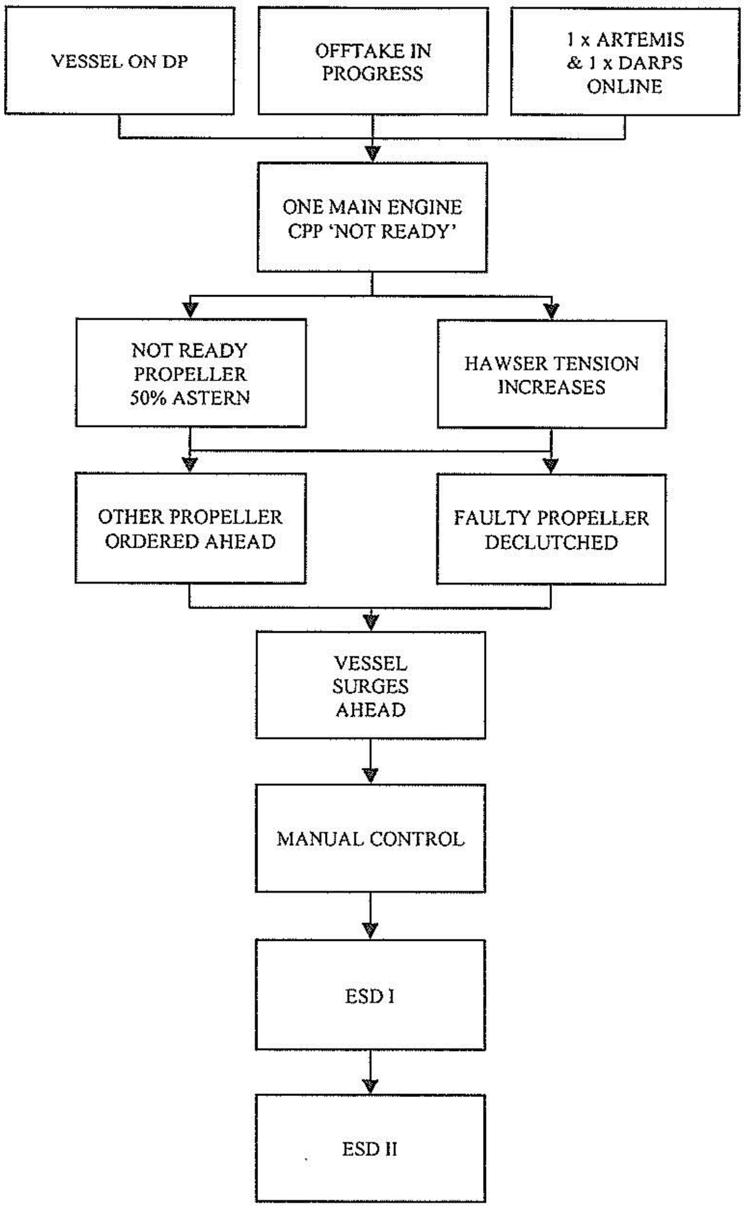
Trigger

Generator
(Actuator)

Secondary Cause

Poor Procedures

Incident # 0412



Comments

The DP control system started to compensate for the astern movement. When the astern thrust of 50% was stopped this caused a surge ahead which brought the tanker within 4m of the loading point. The CPP hydraulic oil was found to be contaminated (bacteria) and the CPP system worn such that the solenoid valve malfunction was thought to be the cause. The problem reoccurred 7 days later.

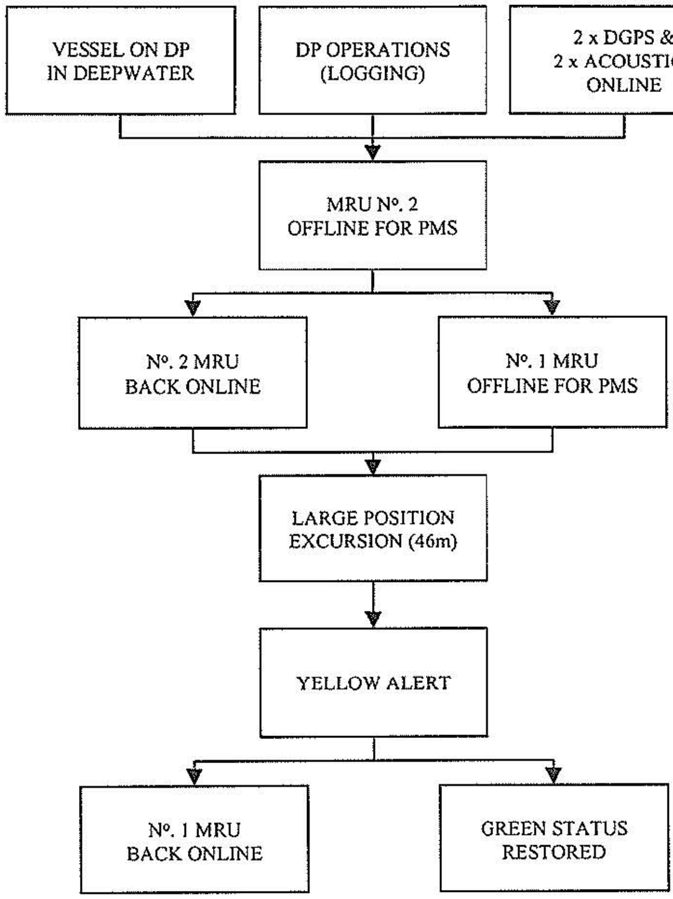
Trigger

Thruster
(CPP Control)

Secondary Cause

Poor Maintenance

Incident # 0414



Comments

No. 1 MRU was being used by the Short Base Line Acoustics and hence the Acoustic fixes were lost. The jump in position was probably significant if the Acoustics were the reference origin.

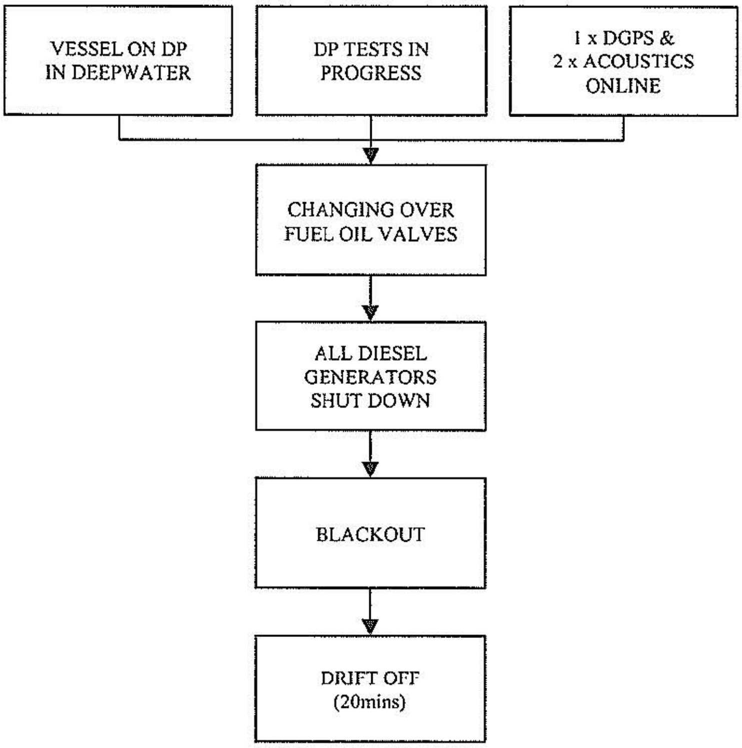
Trigger

References
(MRU)

Secondary Cause

Poor Procedures

Incident # 0415



Comments

This is included as an incident as it was not part of the test but part of a fuel oil transfer operation where a new crew member operated valves in the wrong sequence. The valves have been fitted with locking devices and the operation is now included in the permit to work system.

Trigger

Operator Error
(ERO)

Secondary Cause

Poor Procedures

Ten Major Design Evolutions of the Humble OSV

Mr. B.H.Wong F.I.Mar E
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Abstract
This short paper tries to highlight the design changes on the simple OSV brought about by onerous demands in the fast evolving Oil and Gas drilling and production industry.
The presentation will discuss the ten most important factors influencing the design of a multifunctional OSV.

1. Deep water anchor handling and support services,
2. The problems with the disposal of toxic backloads, and drill cuttings from platforms,
3. Other environmental pollution issues and their effects on the OSV design,
4. Safety issues and the high number of Special Purpose personnel,
5. Noise and vibration control for crew and personnel comfort,
6. Will Diesel-Electric take over the traditional direct engine shaft propulsion?
7. Sub-sea services and the effect on the OSV design,
8. Is it wise to incorporate the well intervention function on the OSV?
9. Is Dynamic Positioning (AAA) mandatory for some functions?
10. The 'Bourbon Dolphin' disaster and the ensuing design changes.

Background
The OSV is an Anchor Handling and offshore supply vessel specially designed to provide anchor handling services and to tow semi submersibles and barges loaded with offshore platforms and production modules. The vessels are also used as standby rescue vessels for oilfields with production platforms and are often equipped for fire fighting, rescue operations and oil recovery. The AHTS is also used in general supply service for all kinds of platforms, transporting both wet drilling fluids, acids and chemicals and dry bulk in addition to deck cargo. The vessels' Hull, winch and propulsion engine capacities kept increasing gradually as drilling and production activity moved into deeper waters. In order to offer a safe and efficient anchor handling operation in deep water conditions in excess of 2000M, various design changes were necessary. As subsea technology improved and became more economical, the OSV was considered as a platform for subsea installation, construction, repair and maintenance, including well intervention work.

Taking into account the large OSV's additional equipment to meet the increased functions we see the vessel's cost approaching that of small jack-up or a VLCC.

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Deep Water Anchor Handling

Anchor handling tugs rely on the main propulsion horsepower to tow drilling rigs and perform anchor handling operations. The increase in deep water exploration has led to even higher Engine horsepower to handle the heavier Rig anchoring gear. The high-horsepower anchor handling/tug/supply vessels were needed to move large new sophisticated drilling rigs, handle their anchors, chain and mooring lines, and meet all kinds of service demands of the new generation deepwater rigs and production platforms.

Now the total main Engine BHP has exceeded 25,000BHP with the Bollard Pull exceeding 300T with Stern Rollers in excess of 4M diameter. The anchors that are handled by these vessels require winches exceeding 600T brake capacity and Rig Chain Locker capacities approach $1000M^3$ on vessel hull beams exceeding 25M. Dynamic positioning has become the norm and most platform operators would insist on DP(AA) or DP(2) as a minimum.

Some large anchors need an A-Frame on the aft deck in order to reduce the Winch tension. Robotic Cranes have taken over the handling of the outsize gear to provide a safer working environment. Normally two cranes one on each crash rail are installed and their booms and robotic arms are adjustable to work any part of the main deck. The Anchors and cables being handled have become so large and heavy surpassing the level for safe human handling.

The Supply function and Back Loading Problems

Due to more sophisticated drilling fluid technology through the use of more chemicals, acids and oils mixing to enhance the properties of the mud, oil majors now need to consider National environmental awareness. Waste and certain drill cuttings cannot be dumped overboard and have to be shipped ashore for treatment and disposal.

The OSV Code (IMO Res.A.863(20))"Code of Safe Practice for the Carriage of Cargoes and Persons by Offshore Supply Vessels" does not directly address the hazards and pollution of certain chemical cargoes.

Generally, most Offshore Platform Supply Vessels including the multifunctional AHTS are used for the transportation of stores, materials, equipment or personnel to and from and between offshore installations. In its outward bound journey, the vessel carries fresh drilling mud and brines. These are normally not hazardous nor are they pollutive except for certain brines containing zinc salt which have to be certified under Category B of Annex II MARPOL 73/78 from 01/01/2007.

The additional need to transport more hazardous and pollutive chemicals (listed in the Annex II) to the platform therefore creates a need to comply with these new regulations in the outward bound supply voyage to the platform.

Guidelines For The Transport And Handling Of Limited Amounts Of Hazardous And Noxious Liquid Substances In Bulk On Offshore Support Vessels according to IMO Res. A673(16)

This resolution applies in full to OSV's keel-laid on or after 19/04/90. For OSV's keel-laid before this date, certain exemptions to the vessel's design may be granted by the Flag Administration.

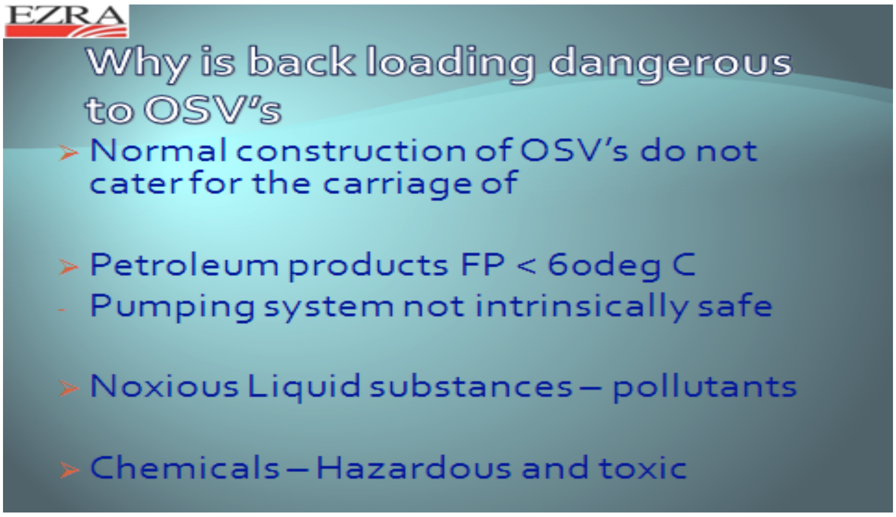
IMO had to prepare separate guidelines for OSV's due to the increased demand in the offshore industry for servicing and resupplying mobile offshore drilling units and offshore platforms.

Furthermore the industry regulators realized the dangers of the continuing and increasing need for offshore support vessels to carry limited quantities of noxious and hazardous liquids in bulk in the normal course of their operations.

Backloading Hazards

Due to environmental pressures and national legislation (eg. OSPAR convention 1992 to protect the marine environment of the North East Atlantic) most countries prohibit indiscreet discharges to sea or shore. This means that the back loads of contaminated mud/brines and drill cuttings have to be loaded on supply vessels for transport to shore facilities to be treated before disposal. However unless specially designed, the vessel is normally not equipped to carry back loads from the platforms because they may contain low flash point oil or flammable toxic gas. The tanks on OSVs and the pumping system generally only cater for oil flash point above 60^{o}C. Major design changes have to be made to cater for low flashpoint oil and IMO Res. A.673(16) discusses this.

Another more noxious gas that may be present is hydrogen sulphide. Any vessel with such hazardous cargoes will prompt the Port Authority to refuse the OSV entry into port.



Applicable Regulations and Guidelines for Supply and Backloading

Latest Legislation on Marpol Annex II applicable from 1st Jan 2007 issued by IMO requires OSVs carrying fresh brine with zinc salts and intending to carry backloads have to be certified under IMO Res. 673(16) and issued with a Certificate of Fitness and carry a Shipboard Marine Pollution Emergency Plan.

All backloads now need to go to reception facilities, therefore countries need to provide reciprocal arrangements for full compliance with IMO. Backloads can also be "washed" clean or treated at the platform and certified safe.

The British Maritime and Coastguard Agency, Environmental Quality Branch has also issued the Marine Guidance Notes No.283 on the back loading of contaminated bulk liquids from offshore installations. These are very clear instructions to the platform operator and the Master of the OSV on the safe carriage of any backloads from offshore installations to shore facilities.

The Clean Notation

Environmental concerns are addressed by Classification Society Rules (eg. By DNV "Clean Notation") and the IMO MARPOL 73/78 Convention Annexes.

These are in response to the increasing pressure on ships to minimize the impact of their operations on the Environment.

On the Environment aspect the "Clean" Design notation provides for limited emissions and discharges to air and sea respectively specifically limited are antifoulings containing TBT, the sulphur content of fuel oils, on-board refrigerants containing global warming promoting CFCs such as Freon, engine NOx emissions, plus discharges of grey/black water, and contaminated ballast.

The Class environmental standards are in excess of the current statutory requirements in anticipation of stiffer future legislations by certain countries. Eg. Class Limit engine Nox emissions to less than 80% of the Marpol standards.

Great care has been taken to ensure both the engineering and environmental integrity in the on-board systems used for carrying and transferring methanol, brine, mud, base oil, ballast and drill water. Mud water washings from hot water heater cleaning of mud and brine tanks in the cement room are pumped to slop tanks where the mud is settled, leaving the water to be re-circulated for further washing. After final use, washing water is pumped ashore for disposal. Bilge drainage from the mud and brine rooms are to be separated from the ship's system. The mud and brine pumps are not to be located in the Engine room. The provision of double hull coupled with the location of fuel oil and other cargo tanks inboard confers considerable damage resilience and hence contributes to environmental protection.

The IMO Marpol convention Annex 1 Reg 12A requires a Double Skin Construction for all Oil Fuel tanks if the total oil fuel capacity on board exceeds $600m^3$. As another safety measure, water ballast tanks surrounding the ship's two large coated steel special product tanks are filled with sea water when methanol is carried.

Solid waste is separated according to type, compacted and then incinerated or kept until a port is reached. Food waste is ground to small particles which, mixed with water to concentrations of less that 100ppm, can be released into the sea.

Code Of Safety For Special Purpose Ships

The additional functions performed by the OSV eg: subsea construction and maintenance, diving support, well intervention, well stimulation etc, necessitated the housing of large number of personnel required for such 24/7 operations on board the vessel. These personnel were neither crew nor passengers, hence such ships were neither cargo ships nor passenger ships and the requirements for either of them could not cater to these ships. Cargo Ship requirements were insufficient in Life saving & Fire whilst Passenger ship requirements would make the project costs forbidding. Flag states questioned the Safety standard of such a ship with supernumeries greater than 12 and this resulted in the formulations of IMO A.534, Code of Safety for Special purpose ships. This code laid down specific requirements for deterministic damage stability by which the vessel had to survive one compartment damage. The assumed extent of transverse damage was B/5 (considerably higher than the 760mm for supply vessels) and the engine room had to be considered as damaged when the number of special personnel exceeded 50. The vessel had to comply with SOLAS Passenger vessel requirements when the number of special personnel exceeded 200. Such special purpose ships had wing tanks of breadth equal to B/5 compared to the 760mm wide wing tanks for supply vessels. They also had shorter engine rooms compared to supply vessels. Designers even resorted to using diesel electric propulsion to achieve the shorter engine room lengths necessitated by damage stability. The code also set down requirements for bilging, navigation, fire and safety and lifesaving by referring to relevant sections of SOLAS.

This code has recently been revised by the Maritime Safety Committee to bring it up to date with the amendments to SOLAS. This revised code is designated as the 'Code of Safety for Special Purpose Ships (2008)'. Under this new code, damage stability had to comply with probabilistic damage as required for SOLAS Passenger vessels with required index equal to R when the vessel carries not less than 240 persons. The R value assigned is 0.8R when the vessel carries not more than 60 persons. Again, the requirements for bilging, navigations, fire and safety and life saving were stipulated by referring to the relevant sections of SOLAS. The vessels would now need to be designed in line with a totally different damage stability philosophy and this would mean some modifications in watertight compartmentation. This code like its precursor is not mandatory and is left to the discretion of the flag authority. However, it may soon become the industry standard for oil majors and port states.

Crew Comfort

Offshore work is physically and mentally demanding due to shipboard and environmental hazards. Therefore the STCW regulations stipulates minimum rest hours. Special attention is paid in the design and construction to limit the vibration and noise levels within the ship to those generally accepted and which will not result in discomfort or annoyance to the crew.

The Comfort Class notation given by certain classification societies takes crew comfort to a higher level. According to this notation, which is optional, the noise and vibration levels on the ship as well as the indoor climate on board are rated on a scale of 1 to 3 with 1 being the highest level.

These rules are based on international standards, including the ISO Standard 6954-1984 version "Mechanical vibration and shock – Guidelines for the overall evaluation of vibration in merchant ships", as well the IMO Noise Code, Resolution A.468(XII),"Code on noise levels on-board ships"

The noise and vibration criteria as specified in the Comfort Class are to be met for all power settings of the main propulsion machinery up to 85% MCR during a normal transit condition. The specified criteria should also be met during maneuvering/dynamic-positioning (DP) with at least 40% load on the thrusters.

The vibration level should not cause damage to the main propulsion system or lead to malfunction of other shipboard machinery and equipment.

Special attention shall also be taken to avoid machinery and equipment having local vibrations causing risks for malfunction or damages when the ship is in service. The vibration levels of machinery, equipment, radars, structure etc. shall be in accordance with the recommended limits given by classification.

All vibration and noise restriction levels are addressed at the design stage. Structural separations, insulation, recreation rooms, cabins location, machinery mountings are all included, and all Flag State crew accommodation regulations are taken into account.

The Diesel Electrical Propulsion evolution was achieved through new electric motor technology making them more compact to fit the tight propulsion space. No high voltage system is required and the 690V motor with the two into one configuration using smaller converters and transformers is cost effective and suitable for straight shaft or Azimuth propeller drives.

The Diesel Electric drive eliminates the long propeller intermediate shaft. The main engines can be replaced by the Alternator Engines which can be housed at a higher deck out of the Engine room thereby increasing lower deck dry bulk capacities.

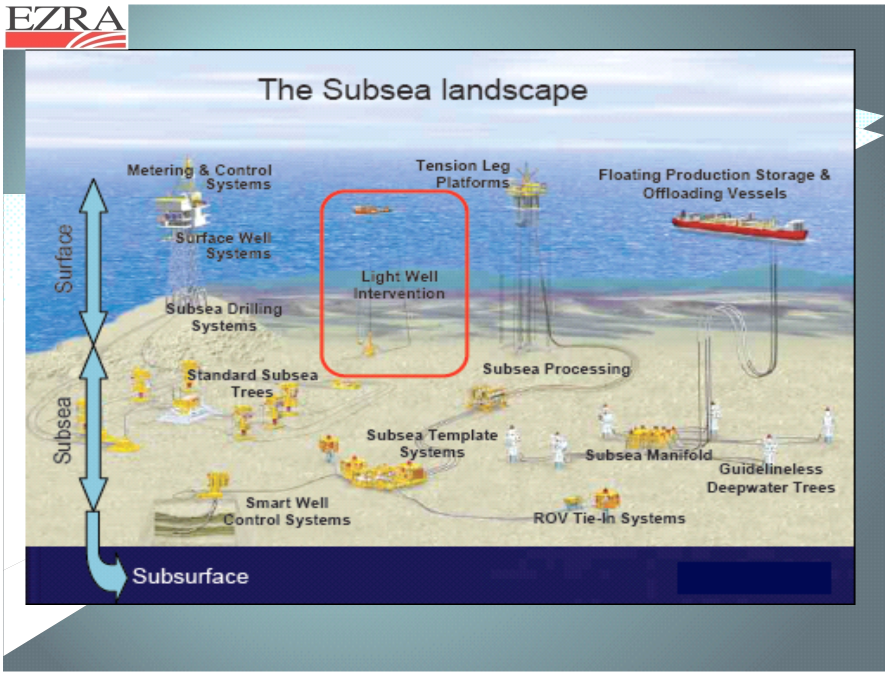
Large main engines driving PTO (power take-off) shaft generators at low speeds produces low fuel efficiency, high emissions and more engine maintenance. The Diesel electric engine eliminates all these and will have an improved life cycle and maintenance costs not to mention reduced fuel consumption.

Another important factor is the reduction of the size of the main engine room to comply with the SPS code previously mentioned.

If the use of a higher voltage system (eg6.6Kv) is selected including larger switchboard and generators and the use of transformers, specially designed air conditioned compartments will have to be provided.

These compartments are also protected against fire and flooding.

High voltage systems in todays technology can be considered very safe and will provide for less bulky equipment and cabling and greater efficiency.



Added value in the provision of sub sea services
From the Owners perspective the design of a support vessel that can complement subsea and well intervention work makes it more deployable. Subsea activity in Exploration and Production has increased due to high cost of platforms in deeper waters vis a vis improved costings for Subsea hardware and installation. Large OSVs can be fitted with Subsea Heave compensated offshore cranes and 300 Ton 'A' frame and moonpools, ROVs etc.

In the Subsea design individual wells drilled are connected by a control system of valves and pipelines on the sea floor. The drilling platform is then moved to another location. Oil and gas will be produced via the valves on the sea floor, transported in pipelines along the sea floor to a central gathering location and then transported to shore.

The added advantage is that there is no permanent surface platform. Subsea systems have become an essential part of offshore E & P due to their improved technology and lower costs.

Some cost comparisons: (Relative only and figures are not updated)
Cost for Subsea manifold with Xmas trees for four Subsea wells is estimated USD 15m and can be installed in less than a year. Twenty years ago the cost would be USD 500m requiring two years installation time.

Therefore Lower Capex requirements can be seen in the following figures
Four wells Subsea installation USD 15m – 50m
Compare Topside production platform USD 100m plus

Why Oil and Gas operators are opting for the subsea installation.

- Subsea gas and water injection improve oil recovery.
- Ability to tie back wells to platforms over vast distances.
- Shorter construction cycle
- Need fewer offshore platforms in deeper waters.

To follow this trend for subsea installations, we can expect more larger OSV's to be equipped to meet this demand.

Multifunctional including Light Well intervention
A well intervention vessel does not need to be directly connected to a live well. Riserless means that we do not have rigid risers, thus the control of the well is by use of umbilical hoses and the blow-out scenario with massive return of hydrocarbons to the vessel is minimized but some regulations still require a blast wall to protect the accommodation area. A riserless well intervention vessel will in worst-case only meet a blow-out hazard through the umbilical, which can be disconnected or cut-off. Riserless well intervention is also commonly named light well intervention services provided by a well Intervention Unit I (by Norwegian Rules)

The more severe operation is Well Intervention with risers (or heavy intervention or Intervention unit II) that have direct connection to live wells. Unit II intervention vessels could therefore easily work as a drilling vessel as well. No definite Rules and Regulations spell out that Unit II vessels need DP3 or the MODU Code. Should the accommodation block be separated by a "Blast Wall" and located 30M away from the Moonpool?

Is it practical to jumboise an OSV with a Moonpool situated at the aft end? Obviously the Anchor Handling function will have to be fore gone. The safest option will be to have a dedicated well intervention vessel but this is not the most economical or cost-effective due to initial cost and difficulty in getting long term charters for this vessel type.

DP (AAA) Yes or No?

The next major design evolution was to include the multifunctional AHTS with Dynamic Position 3 or DP3 as commonly known. This basically calls for a duplication of the Dynamic Positioning controls, Power Supply, bilge drainage, vessel stability, fire safety which is tantamount to being able to operate the vessel fully in the event of a fire, flood or explosion. This decision has to be taken at the design stage because to retrofit a DP3 system would not be cost effective unlike upgrading from DP1 to DP2 after the vessel is built.

There is still a lot of uncertainty as to the Rules and Regulations and a few questions are being asked to determine whether DP3 is necessary or not to meet current regulations or future legislation.

For Diving Support vessels what are the recommendations?

For ROV Subsea construction, maintenance and inspection work, do we require DP3?

For Well intervention work, any future legislation?

A common misunderstanding is that DP3 gives additional redundancy compared to DP2 which is not the case. The redundancy level is similar for DP2 vessels and DP3 vessels. The only differences are that DP3 vessels require physical separation between compartments in case of fire or flooding. DP3 gives an increased safety level compared to DP2 due to fire explosion and flooding hazards.

With DSV's equipped for Saturation Diving, the author fully supports the recommendations for a DP3 vessel because in an accident on board, the vessels needs to be operable for at least long enough to get the divers up to safety.

DP3 is not an absolute requirement for subsea construction and installation work today. According to industry information, there will be no such requirement or need for DP3 for these services in the future. Most ROV sub-sea construction and installation vessels are only DP2 equipped. Even if DP3 is insisted upon the conversion of an existing vessel will be too extensive to be practical.

Then came the "BOURBON DOLPHIN".

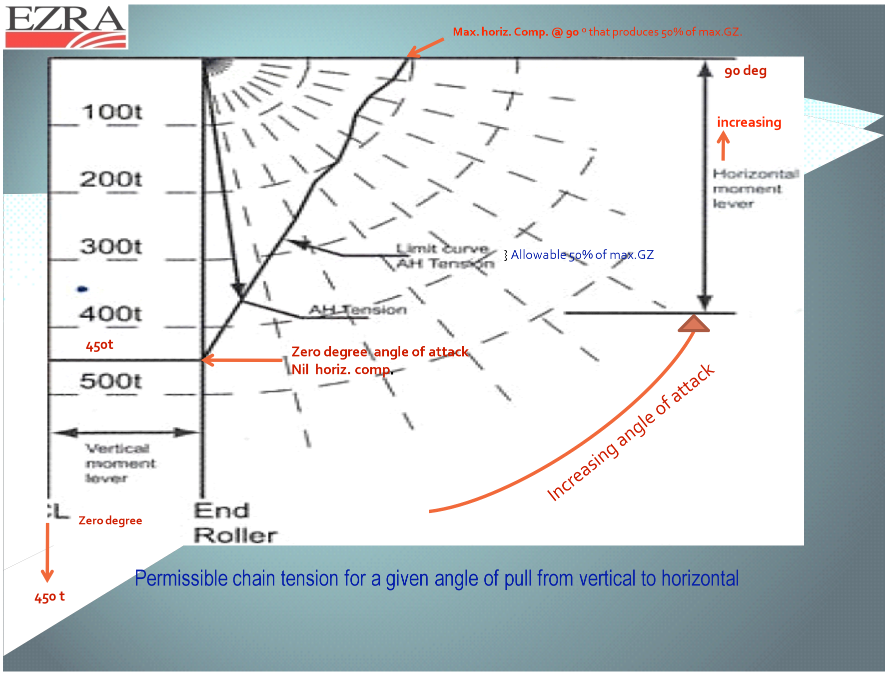
The "Bourbon Dolphin" was an anchor handling tug supply vessel which capsized on 12th April 2007, while performing anchor handling for a drilling rig in the North Sea off the coast of Shetland. Eight lives were lost in the accident including that of the master and his 14 year old son. This accident set off alarm bells in the offshore supply vessel sector and has forced a reevaluation of the design and operations of large anchor handling tug supply vessels. The Norwegian minister of Justice and Police named a special Inquiry commission to investigate the causes of the accident and to recommend countermeasures to prevent such occurrences in the future. The main findings of the commission are summarized below

DIRECT CAUSES:

a) External forces from the weather & current condition
b) Unfavorable heading of the ship in relation to external forces
c) Machinery black out and the consequent reduction of maneuverability
d) Depressed towing pin which changed the angle of attack
e) Current loading condition and vessel's stability characteristics.

INDRECT CAUSES:

a) Weakness in vessel design
b) System failures on the part of many players



RECOMMENDATIONS

Stability Book

Specific anchor handling conditions shall be prepared with 100% and 10% bunkers and shall be shown to comply with an additional stability criterion. As per this criterion, the first intercept of the heeling arm curve with the GZ curve shall not exceed 50% of the maximum GZ.

Specific KG-limit curves shall be prepared for anchor handling to take into account the above criteria.

Instructions to the Master shall be vessel specific (and not generic as of now) and shall incorporate concrete operational restrictions, capacities for given operations and other operational factors significant to the vessel's stability. Examples are the use of roll reduction tanks, ballast tanks, maximum manageable force from the mooring line, maximum capacity of deck cargo etc.

Operations

Ship's crew to be trained in towing and anchor handling operations preferably by a vessel specific simulator.

The bollard pull certificate shall indicate two values of bollard pull. The maximum continuous bollard pull by the vessel's main engines and a reduced bollard pull taking into account the operation of the shaft generator. This will indicate the actual capability of the vessel and prevent the vessel from undertaking jobs which may be beyond it.
The functionality of the winch under all operation conditions shall be tested prior to installation on the basis of a type approval. A quick release function in a casualty situation is to be considered.

Winch operators shall be adequately trained preferably in collaboration with the winch manufacturer.

Equipment

The placement and installation of rescue floats to be improved to ensure their floatability in various casualty situations.

Survival suits to be improved so that they can be donned easily even in a listed condition.

Stability Book

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Operations

It is to be noted that the above recommendations remain guidelines unless they are ratified by either NMD (Norwegian Maritime Directorate, which was the flag authority for Bourbon Dolphin) or IMO. However it is in the interest of AHSV owners and charterers to pay due attention to these recommendations.

Concluding Remarks :

Today's multifunctional OSV incorporating all the above additional operating functions, Safety and Crew comfort features will cost up to USD 150M to build.

When the oil price picks up and serious deep water E and P resumes, these multi functional OSV will be in demand in South America, the North Sea and Australia to name a few places, and their day rates will exceed USD150,000.

The simple OSV will find itself participating at a different level in the E and P activities in deep waters.

END

REFERENCES and ACKNOWLEGEMENTS:

IMO Resolutions A673(16)

MCA GN 283 Marine Coastguard Agency U.K.

Norwegian Government Commission of Enquiry on the 'Bourbon Dolphin'

New IMO requirements for coating of ballast water tanks - challenges and solutions

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ABSTRACT

The International Maritime Organisation (IMO) Performance Standard for Protective Coatings (PSPC) applies to all ships with more than 500 GRT where the building contract was placed on or after 1 July 2008. This means that the IMO PSPC may be relevant for many Offshore Supply Vessels (OSV). This paper outlines the motivation and background for the IMO PSPC and describes its main elements. Furthermore, it discusses the main implications for shipyards and ship owners, in particular some of the implementation challenges in relation to OSVs.

INTRODUCTION

IMO has approved the Performance Standard for Protective Coatings (PSPC) of dedicated seawater ballast tanks in all new ships and of double-side skin spaces of large bulk carriers (2006). The IMO PSPC applies to all ships with more than 500 GRT where the building contract was placed on or after 1 July 2008. This means that the IMO PSPC may be relevant for many OSVs. The target useful coating life of the new requirements is that the coating system remains in GOOD condition for 15 years. The IMO PSPC specifies how coating systems are to be approved, how surfaces are prepared prior to coating and how the coating process is to be carried out and monitored. Furthermore, there is a requirement to document materials and process in what is called a coating technical file (CTF). To comply with the new requirements, shipyards are upgrading their production facilities and work processes.

The aim of this paper is to outline the motivation and background for the IMO PSPC and describe its main elements. Next, it will discuss the main implications for shipyards and ship owners. In particular it will discuss some of the implementation challenges in relation to OSVs. Finally examples of possible solutions for design and manufacture will be discussed.

MOTIVATION AND BACKGROUND FOR THE IMO PSPC

Some of the recent accidents with tanker such as Erika and Prestige triggered the development of new regulation to make these types of vessel safer. Structural design was improved by developing common structural rules for bulk carriers and crude oil tankers. At the same time new requirements for the corrosion protection of seawater ballast tanks were developed. In 1998, first regulation was put in place for coating of water ballast tanks: SOLAS Ch. II-1/Reg. 3-2 – Coating of ballast tanks. However, this was not followed up in the intended way and a new resolution was agreed upon in 2006 to impose stricter requirements on the coating activities in water ballast tanks.

- RESOLUTION MSC.215(82), adopted on 8 December 2006: Performance Standard for Protective Coatings for Dedicated Seawater Ballast Tanks in all types of ships

and double-side skin spaces of bulk carriers (PSPC)
- RESOLUTION MSC.216(82), adopted on 8 December 2006: implementation of MSC.215(82) in SOLAS Reg.II-1/3-2

As of 1st July 2008 the IMO PSPC applies to the protective coatings in dedicated seawater ballast tanks of all types of ships of not less than 500 GRT and double-side skin spaces of bulk carriers ≥ 150 m in length. Fishing vessels and naval craft are exempted from IMO PSPC.

Coating is now considered a safety issue. The main aim of IMO PSPC is to achieve a target useful life of 15 years. This is the time from initial application of the coating over which the coating system is intended to remain in "GOOD" condition. IMO PSPC defines "GOOD" condition as a surface having only minor spot rusting as defined in resolution A.744(18). IACS made this definition more specific in its procedural requirement PR 34. There it is stated that "Good" is defined as: Condition with spot rusting on less than 3% of the area under consideration without visible failure of the coating. Rusting at edges or welds, must be on less than 20% of edges or welds in the area under consideration. One example is shown in Figure 1.

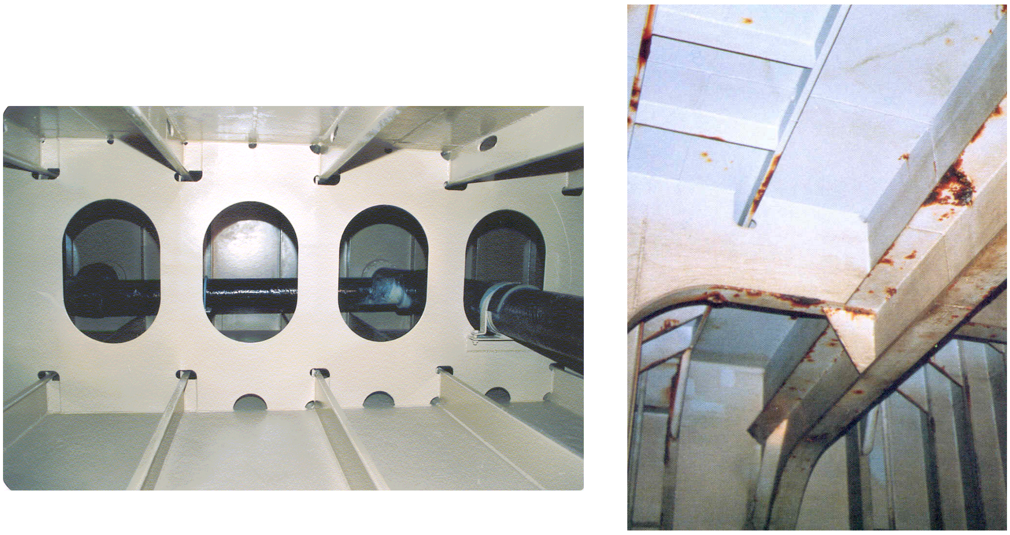

Figure 1 New ballast water tank & in "GOOD" condition after 15 years

MAIN ELEMENTS OF THE IMO PSPC

The IMO PSPC (2006) specifies in Section 4.4 the basic coating requirements *for protective coating systems to be applied at ship construction for seawater ballast water tanks and double-skin spaces for bulk carriers of 150 m in length and upwards*.

<u>Primary surface preparation</u>

Steel plates are to be blast cleaned to Sa 2½ (ISO 8501-1) and primed with a shop primer. The shop primer shall be of an inhibitor free zinc silicate type and shall be compatible and pre-qualified with the main coating system.

<u>Secondary surface preparation</u>

One of the main requirements is that sharp edges are removed from all free edges and rounded to a radius of 2 mm. Alternatively one can use three pass grinding. Intact shop primer may be retained if pre-qualified to be compatible with the coating system. Primer that is not prequalified has to be removed (at least 70%) by blast cleaning to Sa 2. Steel imperfections are to be treated with manual grinding to grade P2 according to ISO 8501-3. Damaged shop primers and along welds the surface is blast cleaned to Sa 2½. The surface cleanliness is assessed visually according to ISO 8501-1.

Surface preparation after erection

Erection weld lines and damages to the coating after erection may be repaired manually for small damages up to 2% of the area under consideration. The required surface cleanliness is St3. For contiguous damages over 25 m^2 or more than over 2% of the area under consideration, blast cleaning to Sa 2½ is required.

Miscellaneous requirements

In addition to the process specific requirements there are also general requirements on the environmental conditions. Blast cleaning and painting shall be carried out at relative humidity of ≤ 85% and at surface temperatures 3 C^o above the dew point. The dew point is the temperature at which air is saturated with moisture. The conductivity of soluble salts on the surface is measured in accordance with ISO 8502-6 and ISO 8502-9, and compared with the conductivity of 50 mg/m2 NaCl. If the measured conductivity is less then or equal to the conductivity of 50 mg/m2 NaCl, then it is acceptable. All soluble salts have a detrimental effect on coatings performance. ISO 8502-9:1998 does not provide the actual concentration of NaCl. The % NaCl in the total soluble salts will vary from site to site. Minimum readings to be taken are one reading per block/section/unit prior to applying.

Main coating system

The coating system used is usually epoxy based with light colour. Epoxy based systems are used exclusively today, even though there are possibilities to qualify alternative systems. The prequalification of the system is documented by a Type Approval Certificate (TAC).

There shall be a minimum of two stripe coats and two spray coats, except that the second stripe coat, by way of welded seams only, may be omitted if it is proven that the NDFT can be met by the coats applied. Any reduction in scope of the second stripe coat shall be fully detailed in the CTF. Two stripe coats are applied prior to coating of the water ballast tanks. Stripe coating is painting of edges, welds, hard to reach areas, etc., to ensure good paint adhesion and proper paint thickness in critical areas. Stripe coats should be applied as a coherent film showing good film formation and no visible defects. The application method employed should insure that all areas that require stripe coating are properly coated by brush or roller. A roller may be used for scallops, ratholes etc., but not for edges and welds.

Two coats are applied with a nominal dry film thickness (NDFT) of ≥ 320 μm according to the 90/10 rule. 90/10 rule means that 90% of all thickness measurements shall be greater than or equal to NDFT and none of the remaining 10% measurements shall be below 0.9 x NDFT.

Items of importance in the IMO PSPC

- Coating system approval (sect.5*)
- An Inspection Agreement to be established (sect.3.2*, also required earlier)
- A Coating Technical File (CTF) shall be prepared (sect.3*)
- Coating inspection during coating preparation and application (sect.6*)
- Verification (sect. 7*)

** Refers to the relevant section in the PSPC*

Maintenance

IMO PSPC requires that all repair of the coating of the water ballast tanks is recorded in the CTF. IMO is currently preparing a guideline on how to carry out maintenance. It is based on IACS Recommendation No. 87.

IMPLICATIONS FOR OSVs

The IMO PSPC has originally been conceived for large oil tankers. Hence, it comes as no surprise that it is not always straight forward to apply to Offshore Supply Vessels. One item that is receiving particular attention is the definition of the seawater ballast tanks. IMO PSPC applies to "dedicated" seawater ballast tanks. In many cases OSV's have combined tanks that can also carry e.g. drilling fluids. To clarify this matter, IACS has submitted a unified interpretation to the IMO's "Sub-Committee on Ship Design and Equipment" at its 52nd session (DE52) in March 2009 (2009a). Here IACS proposes that:

The following tanks are not considered to be dedicated seawater ballast tanks and are therefore exempted from the application and requirements of the IMO PSPC:

1. *ballast tank identified as "Spaces included in Net Tonnage" in the 1969 ITC Certificate;*
2. *seawater ballast tanks in passenger vessels also designated for the carriage of grey water.*

The proposal was considered at the DE 52. However, it is not quite clear what was actually agreed upon at the meeting (as of May 2009). A report by DE52 to the maritime safety committee (2009b) states that the Sub-Committee considered document DE 52/17/6 (2009a) and, having supported the interpretation in principle, agreed to take no further action on the matter. This does not seem to give an accurate picture of what was discussed. The report by the IACS representative present at the DE52 meeting (2009c) states that the plenary discussion was not very accurately documented in this statement. It notes that a number of delegates opposed the 'derogations' (as they were seen) especially for grey water tanks in passenger ships.

The IMO PSPC defines a minimum quality standard and it is quite possible to exceed these requirements if desired. The coatings used for combined ballast tank are known to be of higher quality than conventional corrosion prevention coatings for seawater ballast tanks. However, these coatings are not usually type approved according IMO PSPC.

In the absence of an agreed interpretation, most OSVs are handled on a case by case basis where the status of the ballast tanks is agreed upon with the Flagstate. This is today's praxis in Norway and DNV.

MAIN IMPLICATIONS FOR DESIGNERS, SHIPYARDS AND SHIP OWNERS

Consequences for ship designer

The application of IMO PSPC is usually considered a production issue to be taken care of by the shipyard. However, as mentioned in Section 3 General Principles, subsection .3.2 of the IMO PSPC (2006), there are also many opportunities already in the design phase of a vessel to make coating friendly design that are easier to produce and maintain. The main focus should be towards reducing the length of free edges in ballast water tanks, accessibility of the tanks and the avoidance of complex joints within the ballast water tanks. Some of the ideas suggested here require optimisation of ship structures as changing frame spacing will lead to different scantlings such as plate thickness.

Free edges need to be rounded to a 2mm radius which can involve considerable manual work. By using fewer stiffener and using profiles that already have the correct radius a considerable amount of time may be saved.

Accessibility is often a problem in many BWT. Hence ensuring easy access not just for the painter but also their equipment will increase the quality and efficiency of the surface preparation, coating application and required quality control inspections.

By reducing the number of complex joints, the need for NDT inspection and documentation will be reduced considerably. Further reduction can be achieved by using fewer stiffeners as discussed above. NDT coating thickness inspection is manual task and therefore designers can reduce the time spent on NDT inspection by modifying their ship designs.

Consequences for shipyard

There are a number of logistical and administrative tasks for shipyards. While there are already shipyards, in particular working for the offshore industry, that meet the technical requirements, there are few that already have suitable systems and procedures in place to meet the PSPC requirements for documentation.

Shipyards are required to prepare the inspection agreement, and the CTF. A first draft of the CTF and the inspection agreement is required for the plan approval.

There is usually a need to upgrade the shipyards production system. Approved coating systems need to be specified, including compatible and approved shop primers. More work needs to be done on surface preparation with clear targets on cleanliness and surface roughness. Furthermore, 2 stripe coats need to be applied. In addition new coating halls may need to be built and additional qualified staff for coating is required, including certified coating inspectors.

Furthermore there are challenges regarding production planning, workflow and material selection: How can blocks be dimensioned to minimise congestion in the paint shop? Furthermore, one should re-assess the criteria for selecting coating systems to achieve the fastest production throughput.

Consequences for ship owners

Ship owners will get an active role in maintaining the CTF while the ship is sailing. There is a requirement to maintain the CTF which has to be kept on board the vessel (see sections 3.4.3 to 3.4.5 in (2006)). IMO is finalising a guideline on how this could be done. The CTF shall be inspected by the Administration.

Standard paint specifications of ship owners will have to be adopted to make sure the coating system is type approved. Another requirement is to provide Permanent Means of Access (PMA) to facilitate inspection and maintenance of the water ballast tanks (2008). The PMA have to follow IMO PSPC for parts that are integral to the ship structure.

These measures may lead to an increases of the initial price of the vessel but is expected to result in reduced maintenance costs and possibly enhanced resale value. Furthermore, ship owners are better prepared for evaluation by Vetting and Rating agencies.

CONCLUSIONS

IMO PSPC will affect the way OSV’s are going to the built. While there are still some uncertainties as to which ballast tanks are to be include under the IMO PSPC it is clear that both shipyards and owners will be affected by the new requirements. While there are new requirements on workmanship it seems that the requirements to document the coating process may turn out to be the most demanding challenge. It was also pointed out that there are opportunities to optimise ship designs to make them more coating friendly and thus cheaper to produce and operate.

REFERENCES

(2006) Resolution MSC.215(82) - Performance Standard for Protective Coatings for dedicated Seawater Ballast Tanks in all types of ships and double-side skin spaces of bulk carriers. IN COMMITTEE, M. S. (Ed. London, IMO.

(2008) MSC.1/Circ.1279 - Guidelines for corrosion protection of permanent means of access arrangements. London, IMO.

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Voith Schneider Propeller (VSP) - Investigations of the cavitation behaviour

Dr. Dirk Jürgens (Voith), H.-J. Heinke (Potsdam Model Basin)

1 Introduction

Voith Schneider Propellers (VSP) are used primarily for ships that have to satisfy particularly demanding safety and manoeuvrability requirements. Unique to the Voith Schneider Propeller is its vertical axis of rotation. The thrust is generated by separately oscillating, balanced propeller blades. Due to its physical operating principle and its design, thrust adjustments can be done very quickly. The VSP, a controllable-pitch propeller, permits continuously variable thrust adjustments through 360°; combining steering and propulsion. Rapid step-less thrust variation according to X/Y coordinates improves ship handling. Voith Schneider Propellers operate at a comparably low revolutional speed. Currently, VSPs are used primarily on Voith Water tractors (VWT), Offshore Support Vessels (OSV), Double-Ended Ferries (DEF), Mine Countermeasure Vessels (MCMV) and Buoy Layers.

A question which has been posed several times: What is the cavitation behaviour of the VSP and are there any differences when compared to the cavitation effects of screw propellers?

Cavitation tests have been carried out by Voith Turbo in its own tank from 1933-1976 [1], which had a quite small measuring section (0.75 x 0.25 m²). Further tests have also been carried ou at MARIN [2] and at the HSVA [3]. Investigation into the influence of the cavitation on VSP blades with regard to the hydro-acoustic have also been done by SSPA [4] and the KRYLOV Institute [5].

In the last years, the demand on the input power of the VSP relative to the propeller sizes has increased. There is only limited knowledge available about the cavitation effects for the current propeller loads and future increases of the input power. Cavitation can, for high input power not be avoided, but what effects does the cavitation have?

Voith has gained over the years the experience that cavitation does not created damage on the VSP blades. There is no detailed insight available why the cavitation does not create erosion on the VSP blades. Nowadays a deep understanding of cavitation phenomena can be achieved by using high speed camera technology. There are no reliable measurements available of the present VSP loads as functions of the cavitation number. As yet, new developed blade profiles have not been investigated. by extensively experiments.

VOITH and Potsdam Model Basin (SVA) have carried out together investigations about the cavitation behaviour of Voith Schneider Propellers.

Two main questions have been in the focus of the research activities:

1. Why does the cavitation of the VSP blades not create erosion?
2. At which cavitation number does a thrust deduction occur?

Different profiles of VSP blades had been investigated in tests with single blades and in propulsion and cavitation tests with a VWT model. Selected results of these tests had been presented at the Hydrodynamic Symposium – Voith Schneider Propulsion in March 2006 [2].

The cavitation tests have been carried out in the large circulating and cavitation tunnel UT2 at the Technical University of Berlin and in the cavitation tunnel K15A of the SVA Potsdam. The aim of

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the bollard pull measurements at cavitation identity in the large circulating and cavitation tunnel UT2 of the TU Berlin was to study the influence of cavitation on the forces and moments of the Voith Schneider Propellers.

Investigation into cavitation behaviour of single VSP blade and of the Voith Schneider Propeller at high thrust loading coefficients have been carried out in the cavitation tunnel of the SVA Potsdam. The optical access in the SVA's medium sized cavitation tunnel (test section 850 mm x 850 mm) is better than in the test section of the UT2 (test section 5000 mm x 3000 mm). In addition high-speed video technique could be used for the cavitation observation.

2 The technical and hydrodynamical principal of the VSP

For a better understanding of the later presented hydrodynamical investigations, a short explanation of the technical principle of the VSP is required. A detailed description can be found in [1], [6], [7] and [8].
The thrust of a VSP can be applied very fast applying X/Y-logic. Figure 1 shows the sectional drawing of a VSP and figure 2 the installation of two VSPs in a Offshore Support Vessel.

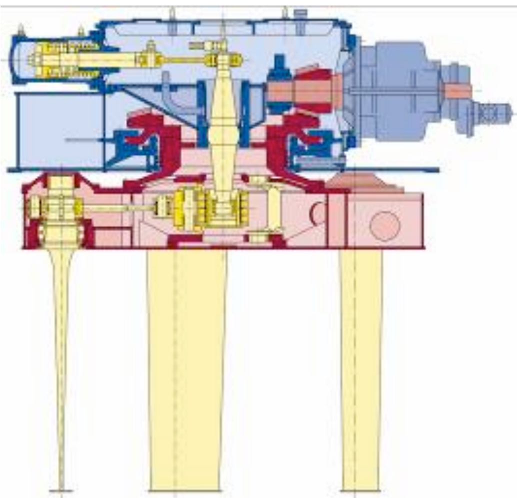


Fig.1: Sectional drawing of a VSP

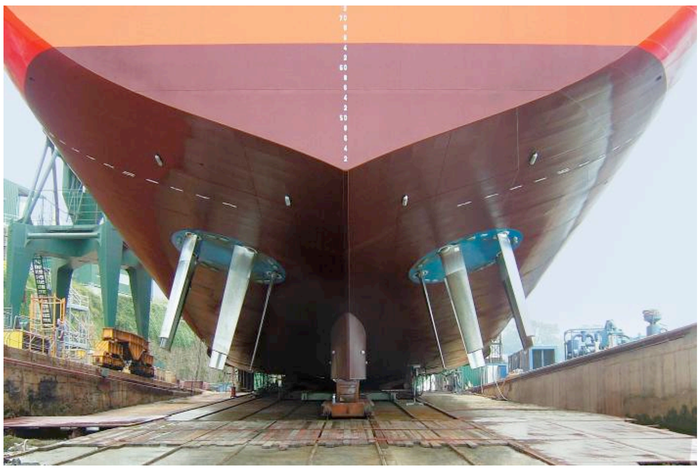
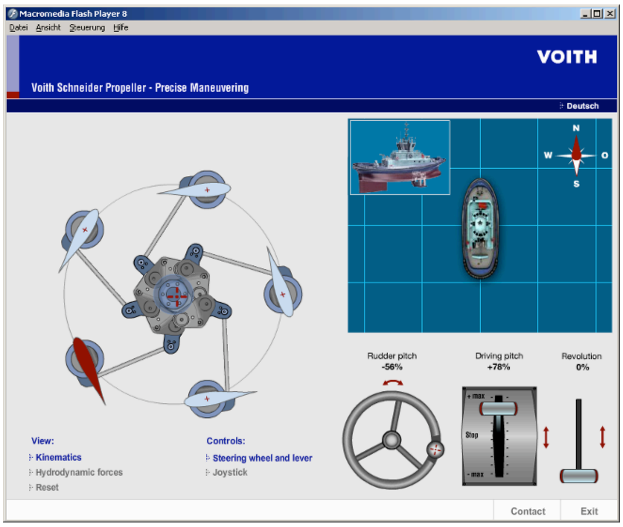


Fig. 2: Two VSP installed in a Offshore Support Vessel

The thrust is created by vertically mounted blades in a rotor casing. While the rotor casing is rotating, the blades are oscillating. The blades oscillation is steered by the law of intersecting normals. Figure 3 shows the mechanical principle of the VSP and figure 4 the corresponding hydrodynamic thrust creation. A descriptive simulation program showing the technology of the VSP and the hydrodynamics can be downloaded at www.voithturbo.com/marine.



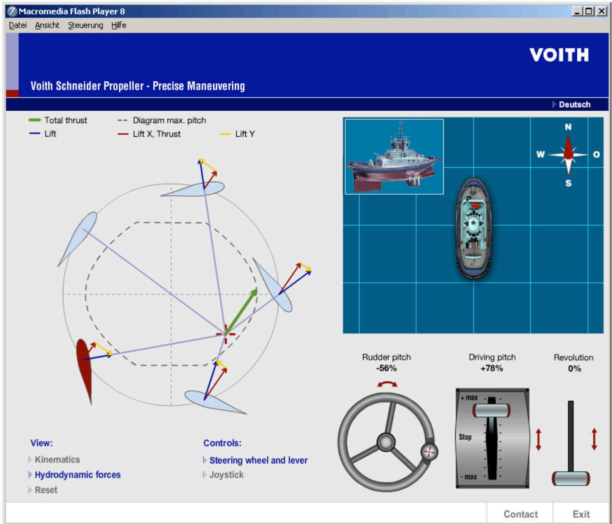
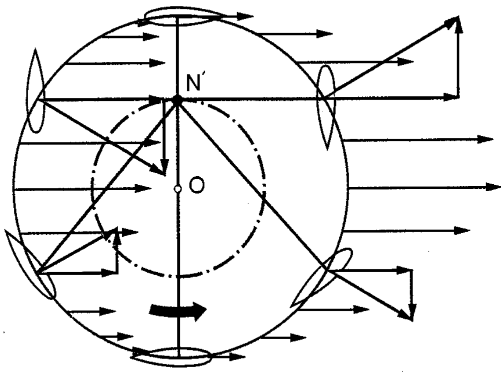


Fig. 3: Mechanical principle of the VSP

Fig. 4: Hydrodynamic principle of the VSP

Figure 5 shows the forces acting at the propeller for selected blade positions. Lift changes during revolution due to the nonstationary flow at the propeller blades. The forces acting across the desired direction of thrust cancel each other, whereas the forces acting in thrust direction are added over the propeller circumference. Figure 6 shows the lift as a function of the cycloid path for a stationary observer.



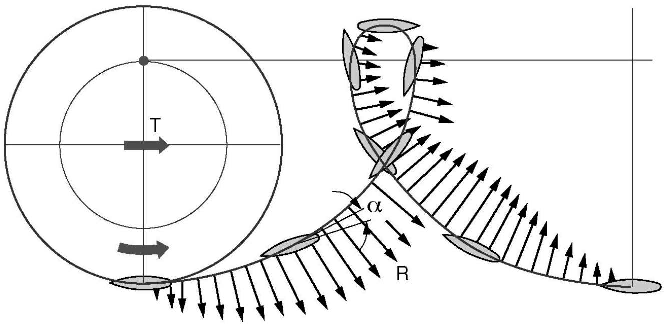


Fig. 5: Blades forces for different blade position

Fig. 6: Lift generation on a blade as a function of the cycloidal blade path

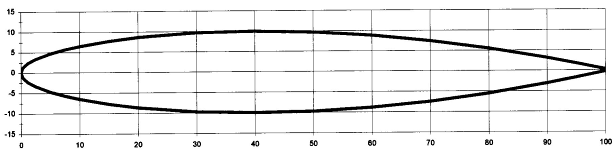
3 Cavitation tests with single VSP blades

The inflow of the VSP blades is complex. The inflow speed as a function of the chord length consist in a vectorial addition of the advanced speed, the rotational speed and the inflow velocity due to the oscillation of the blades around the shaft axis. Firstly the blades had been tested in a parallel flow and stationary angles of attack to get a better understanding of the complex situation
The profiles of VSP for VWT are high lift profiles. The profiles are based on the DVL profile Series and HSVA profiles, but they have a modified leading and trailing edge. Voith Turbo Marine has varied the profile shape systematically. Five profiles (Figure 7) have been investigated in model tests for different angles of attack and cavitation numbers [9]. The main parameters of the profiles are:

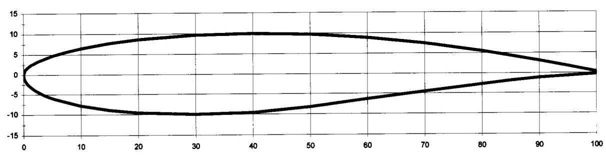
Profile length	L	[mm] :	390.00
Chord length	c	[mm] :	150.00
Relative thickness	t/c	[-] :	0.16

basic profiles

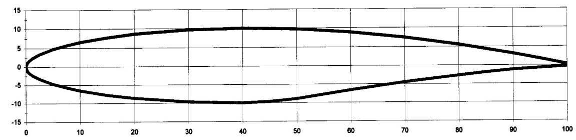
modified profiles



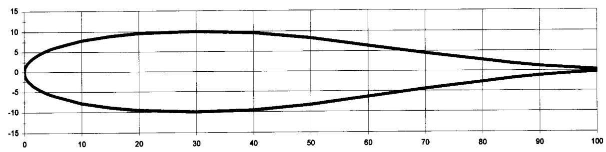
Profile 2 (DVL)



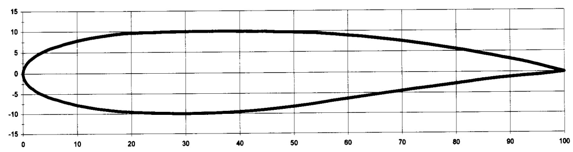
Profile 6



Profile 9



Profile 4 (MP 73)



Profile 14

Fig. 7: Profiles for single VSP blade tests

The cavitation behaviour of the profiles was analysed with high-speed videos. A comparison of the cavitation inception at similar lift coefficients shows a better cavitation behaviour of the MP 73 profile at higher lift coefficients compared to the DVL Profile.

The figures 8 and 9 present video prints of the cavitating MP 73 profile at the angles of attack 12 and 19 degrees. Sheet cavitation appears at the leading edge on the suction side of the profile during the test with an angle of attack of 12 degrees. The cavitation separates from the profile surface after a chord length of about 25%. Two cavitating vortices appear at the profile sole.

At the angle of attack of 19 degrees the flow separates at the suction side. Cavitating vortices can be observed in the separated flow. A cavitating vortex appears on the profile sole.

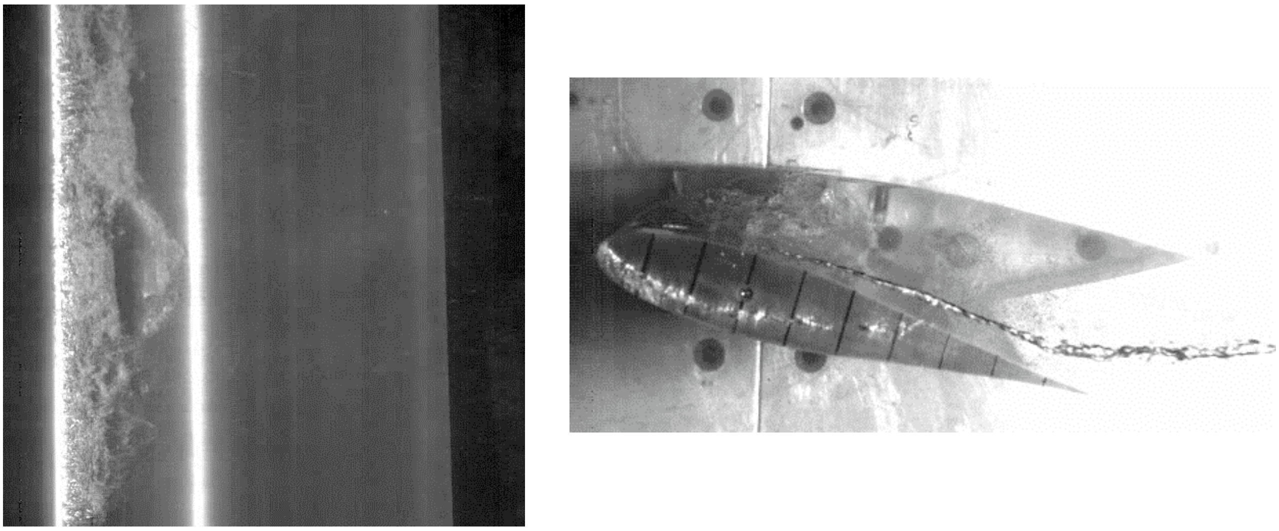


Figure 8: Cavitation observation, angle of attack = 12°, V_M = 4.00 m/s, σ_V = 2.40

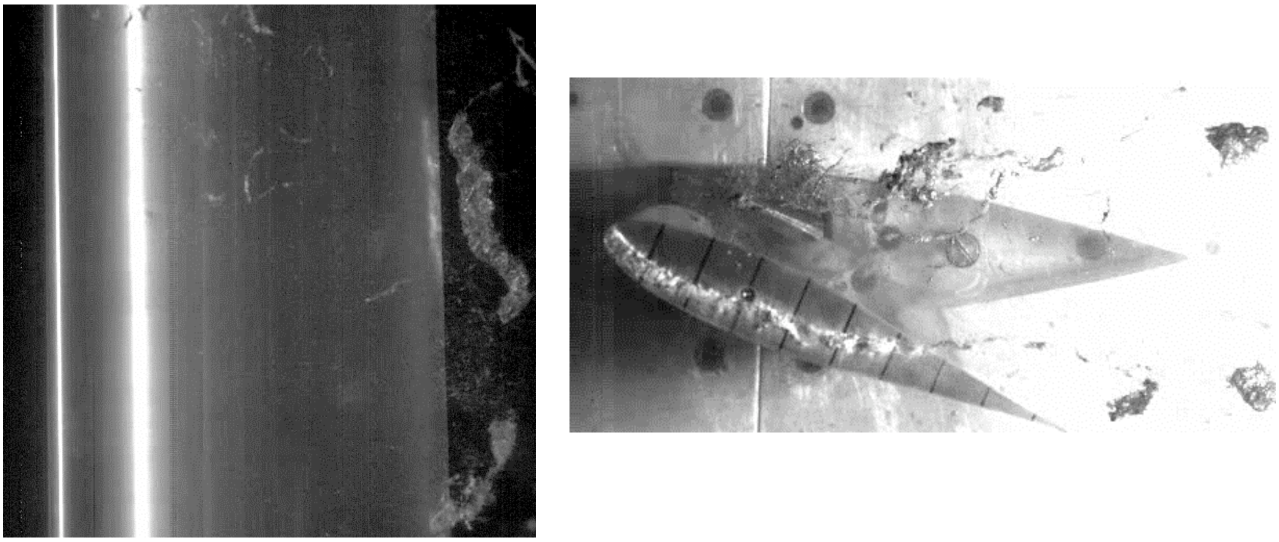


Figure 9: Cavitation observation, angle of attack = 19°, V_M = 4.00 m/s, σ_V = 2.40

4 Cavitation tests with a VSP model

4.1 Measuring device, test arrangement

SVA Potsdam has developed a special thrust measuring system (3-component balance) for VSP model drives (Figure 10 & 11) [8], [10]. The VSP-balance allows the measurement of the longitudinal and transverse forces of the VSP during open water and propulsion tests.

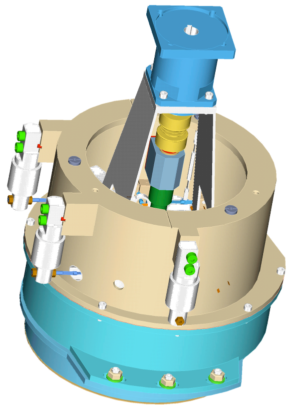


Fig. 10: VSP-balance of the SVA Potsdam for open water and propulsion tests in the towing tank

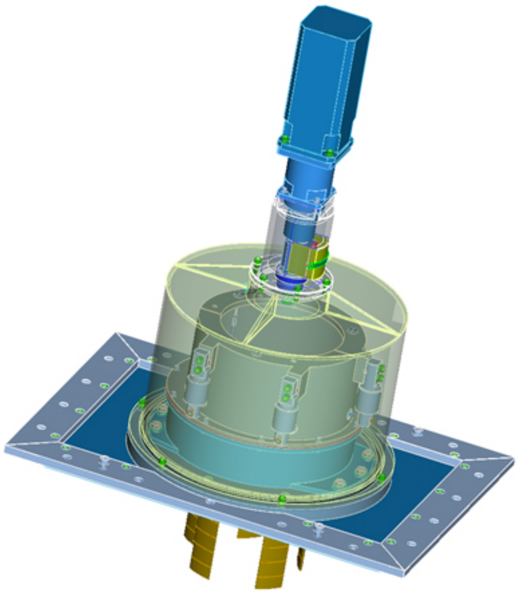
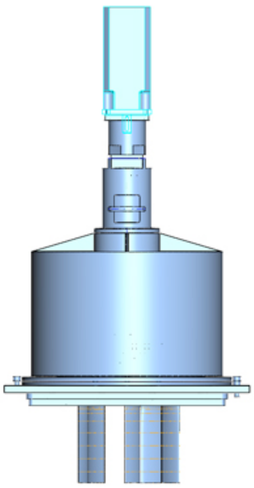


Fig. 11: VSP-balance of the SVA Potsdam for cavitation tests

The VSP-balance was also used for the test set-up in the cavitation tunnel type K15A from Kempf & Remmers. The VSP-balance is arranged on a plate by a test section window. A pod surrounds the balance. The driving motor and a torque measurement device are arranged on the top of the pod (Figures 11 and 12).



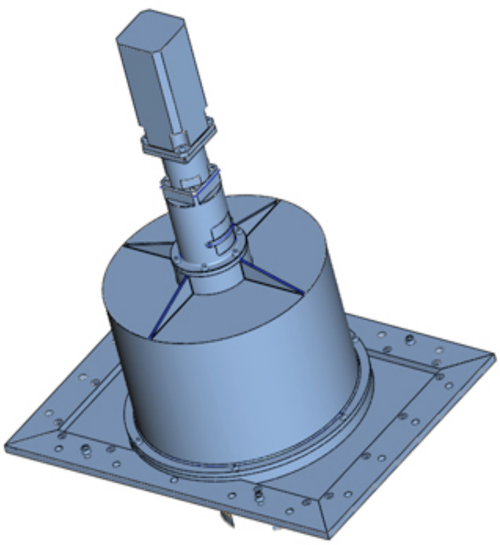


Fig. 12: Views of the VSP-balance for cavitation tests

The driving shaft was designed in a way that no forces will be transmitted on the measuring system. The pod with the balance is filled with water during the measurements. This is necessary to minimise the take of air in the gap between rotor casing and the plate.

The test set-up for the cavitation tests has been tested successfully. The measurement of the forces was possible at overpressure and low pressure. The measurement of the torque was more complicated due to the influence of the water in the rotor casing on the idle torque. Idle torque measurements in the water with the VSP without blades is necessary.

The cavitation tests have been carried out with the VSP model P9659 from Voith Turbo Marine. The main data of the model VSP P9659 are:

Diameter	D	[mm] :	200.00
Blade length	L	[mm] :	166.50
Chord length	c	[mm] :	40.00
Number of blades	Z	[-] :	5

The blades are characterised by an asymmetrical profile, as shown in figure 13. This profile was developed by Voith Turbo Marine using a numerical optimisation strategy [11], [12] based on CFD calculation.

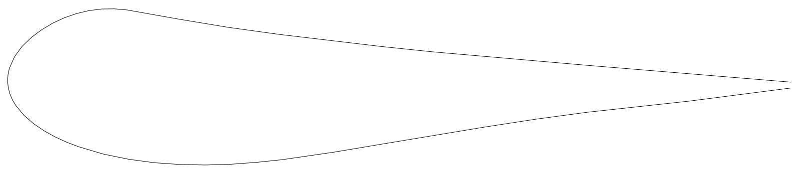


Fig. 13: Profile of the VSP blades, model P9659

4.2 Tests in the cavitation tunnel

The tests have been carried out in the SVA’s cavitation tunnel in the test section No. 2 with 850 mm x 850 mm. The arrangement of the VSP model P9659 in the cavitation tunnel is shown in figure 14. Rotation is clockwise.

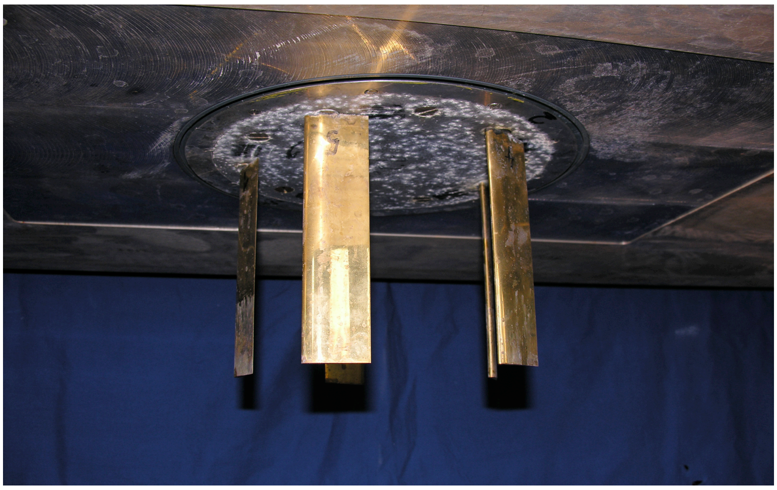


Fig. 14: Rotor casing with VSP blades P9659_R in the cavitation tunnel

The rotor speed of the VSP models is limited to $n_{max} \approx 6$ rps. The cavitation tests have been carried out at rotor speeds in the range between n = 5 to 5.5 rps. Due to the work of the VSP model a minimum water speed of $V_{min} = 0.473$ m/s appears in the test section.

The following coefficients have been calculated on the base of measured values in the cavitation tests:

Circumferential velocity $U = \pi * n * D$

Advance coefficient $\lambda = V_A/U$

Longitudinal force coefficient $K_{SX} = T_X/(\rho/2 * D * L * U^2)$

Transverse force coefficient $K_{SY} = T_Y/(\rho/2 * D * L * U^2)$

Torque coefficient $K_D = 2Q/(\rho/2 * D^2 * L * U^2)$

Cavitation number $\sigma_n = (p - p_V + \rho * g * h)/(\rho/2 * \pi^2 * n^2 * D^2)$

Reynolds number $Re = c/\nu * U$

4.2.1 Force measurements

The cavitation tests have been done at the advance coefficient $\lambda = 0.15$ at pressures between atmospheric pressure and the pressure corresponding to the full size cavitation numbers. The full-scale cavitation numbers of the VSP for a VWT are in the range between $\sigma_n = 2.2$ (water depth rotor casing) and $\sigma_n = 2.6$ (water depth blade end).

The Figure 15 shows the measured longitudinal, transverse and torque coefficients during the cavitation number variation in the cavitation tunnel. The change of the longitudinal force and torque coefficients of the VSP model P9659_R due to the influence of cavitation is small.

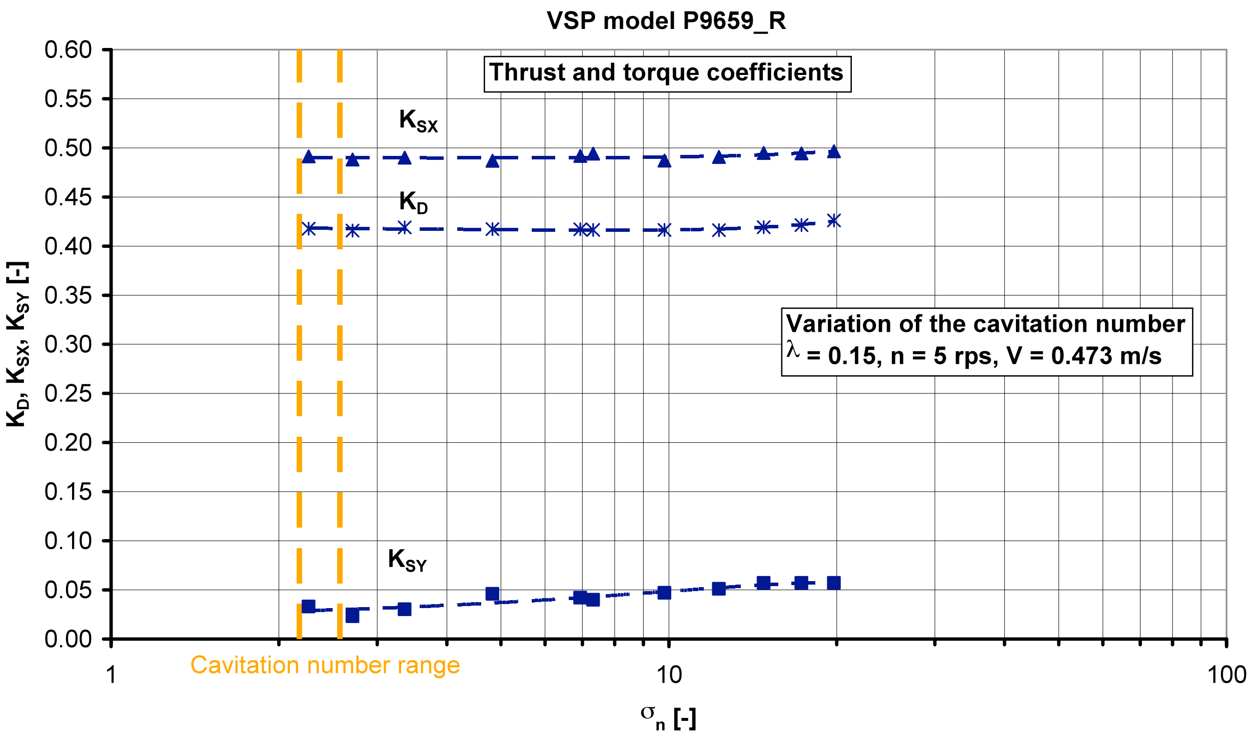


Fig. 15: Influence of the cavitation on the VSP coefficients, model P9659_R, at mode Reynoldsnumber

4.2.2 Cavitation observation

The Figure 16 shows a calculated pressure distribution of a VSP with symmetrical blade profiles acting at a VWT with guard plate [12]. It can be seen, that a low pressure and consequently a large cavitation danger will occur in the angle range 300 to 360 degrees. The CFD calculation with a cavitation model shows that cavitation will appear at the outside and inside of the blades (Figure 17).

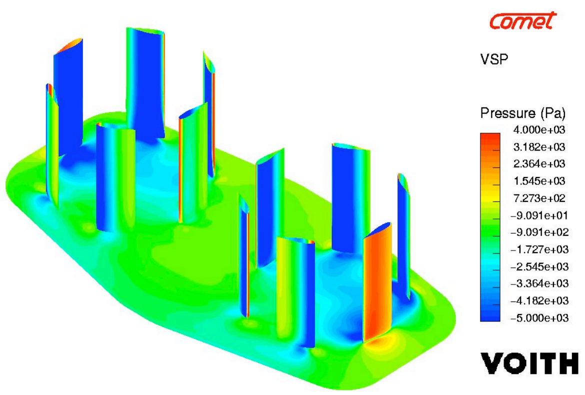


Fig. 16: Calculated pressure distribution at VSP blades

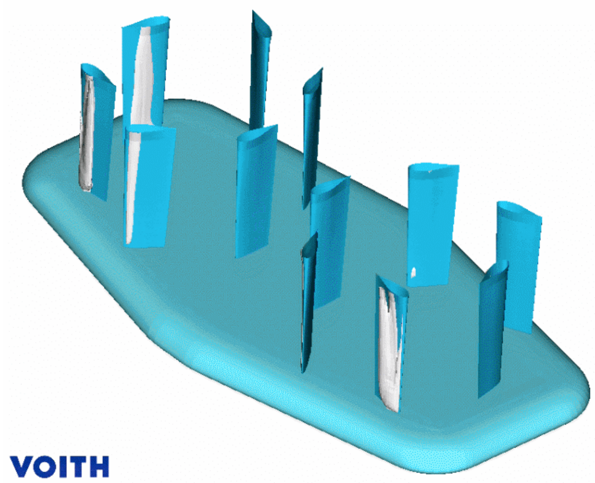


Fig. 17: Calculated cavitation behaviour at high thrust loading

In general the calculations from Voith Turbo Marine show, that the pressure distribution and cavitation danger at the blades can be calculated actually with CFD-methods. A problem is the calculation of the cavitation dynamic and connected separated cavitation. The calculated cavitation allows today no differentiation between bubble, sheet and cloud cavitation. That's why the investigation of the cavitation behaviour of Voith Schneider Propellers is still a task for model tests.

The following video prints from the high-speed camera videos show examples of cavitation patterns on the VSP model P9659 with the asymmetrical profile. The realised advance coefficient of $\lambda = 0.15$ represents the maximum possible load in the cavitation tunnel K15A with the test section 850 x 850 mm.

Due to the asymmetrical profile and the blade angle curve no cavitation at the outside of the blades was observed.

The cavitation inception for the VSP model P9659 has been observed at the cavitation number $\sigma_n = 6.94$. Blade sheet cavitation first appears at the leading edge near the rotor casing (Figure 18).

The cavitation observation at the cavitation number $\sigma_n = 3.36$ shows sheet and vortex cavitation in the angle range 270° to 360°. The cavitation separates from the blade (Figure 19).

At the design cavitation number $\sigma_n = 2.71$ for bollard pull condition sheet cavitation appears at the leading edge of the whole blade length (Figure 20). At the blade end intermittent tip vortex cavitation could be observed. The cavitation separates from the blade and hits the following blade.

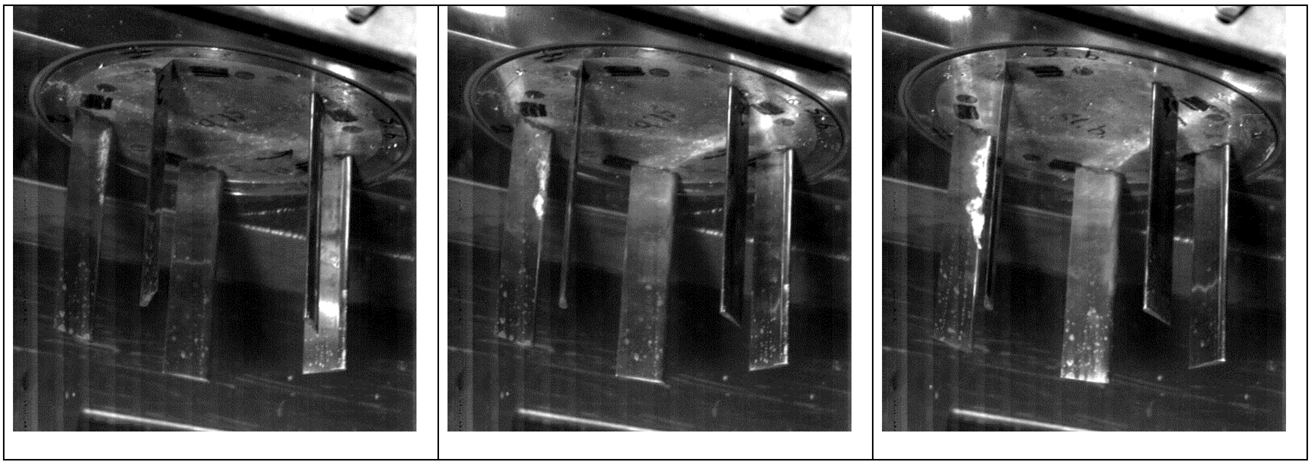


Fig. 18: Cavitation inception, $n_M = 5\ s^{-1}$, $V_M = 0.473$ m/s, $\lambda = 0.15$, $K_{SX} = 0.492$, $K_D = 0.423$, $\sigma_n = 6.94$

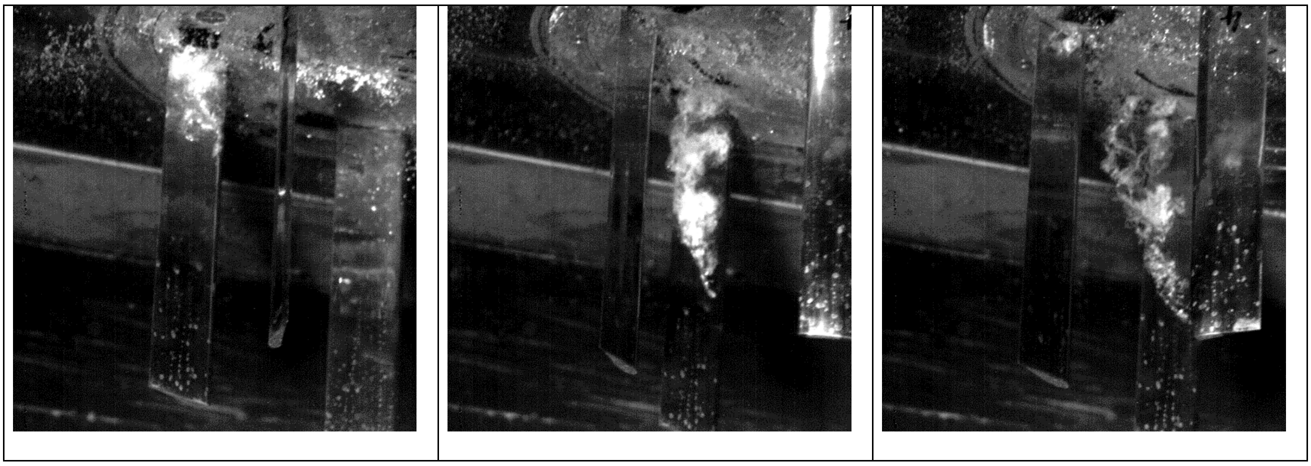


Fig. 19: Cavitation observation, $n_M = 5\ s^{-1}$, $V_M = 0.473$ m/s, $\lambda = 0.15$, $K_{SX} = 0.490$, $K_D = 0.426$, $\sigma_n = 3.36$

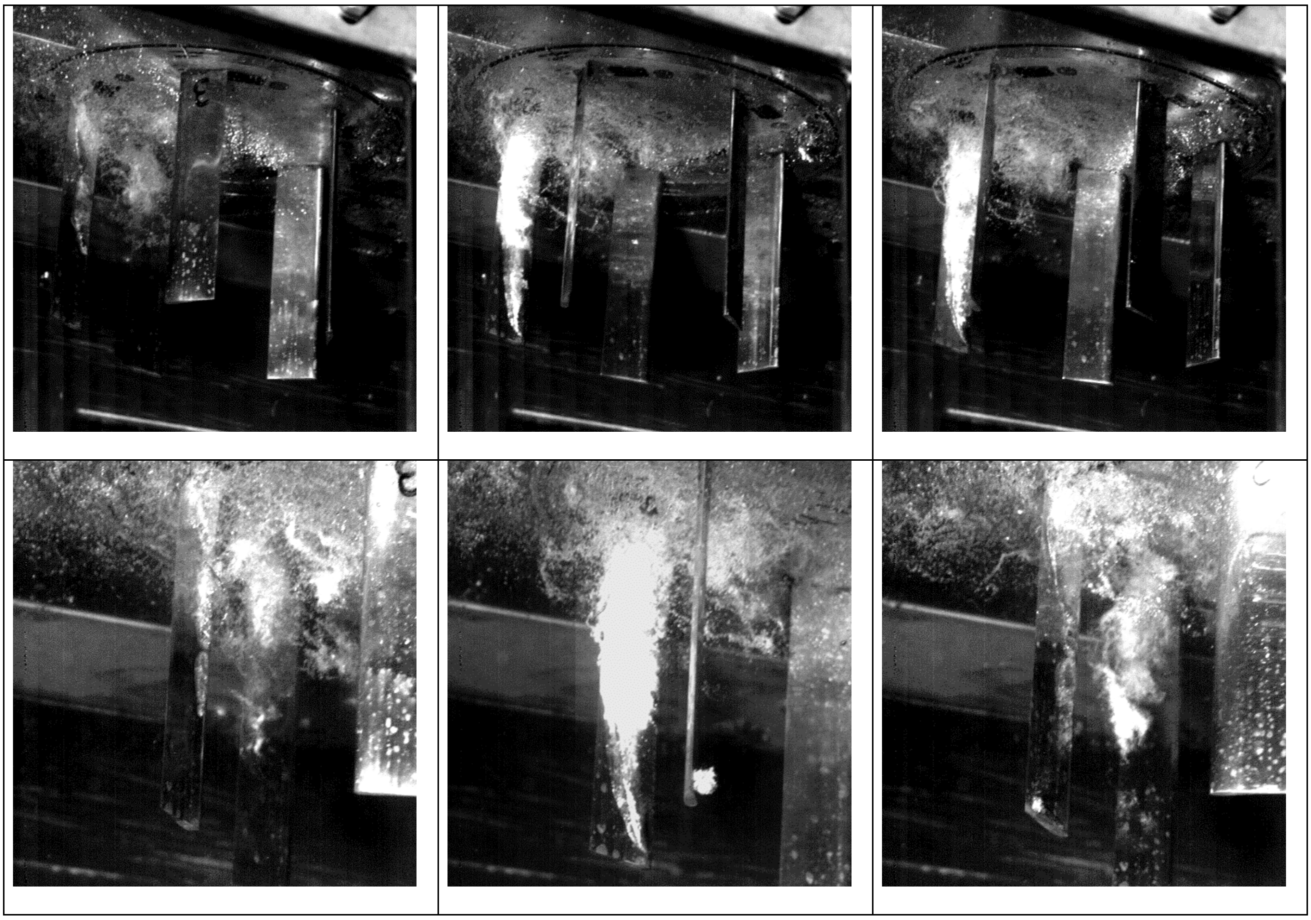


Fig. 20: Cavitation observation, $n_M = 5\ s^{-1}$, $V_M = 0.473$ m/s, $\lambda = 0.15$, $K_{SX} = 0.488$, $K_D = 0.416$, $\sigma_n = 2.71$

4.3 Bollard pull measurements at cavitation similarity in the large circulating and cavitation tunnel UT2

Bollard pull measurements have been carried out in the large circulating and cavitation tunnel UT2 of the TU Berlin, to study the influence of cavitation on the forces and moments [9]. The test section of this tunnel has a length of 11.0 m, a width of 5.0 m and a depth of 3.0 m which allows tests at bollard pull conditions. The photos in Figure 21 shows a VWT model during the installation in the UT2. Measurements in the UT2 have been done at a water velocity $V_S = 0$ and different rotor speeds in the range from $n_M = 1$ to 5.5 rps at atmospheric pressure and at pressures corresponding to the model and full size cavitation numbers.

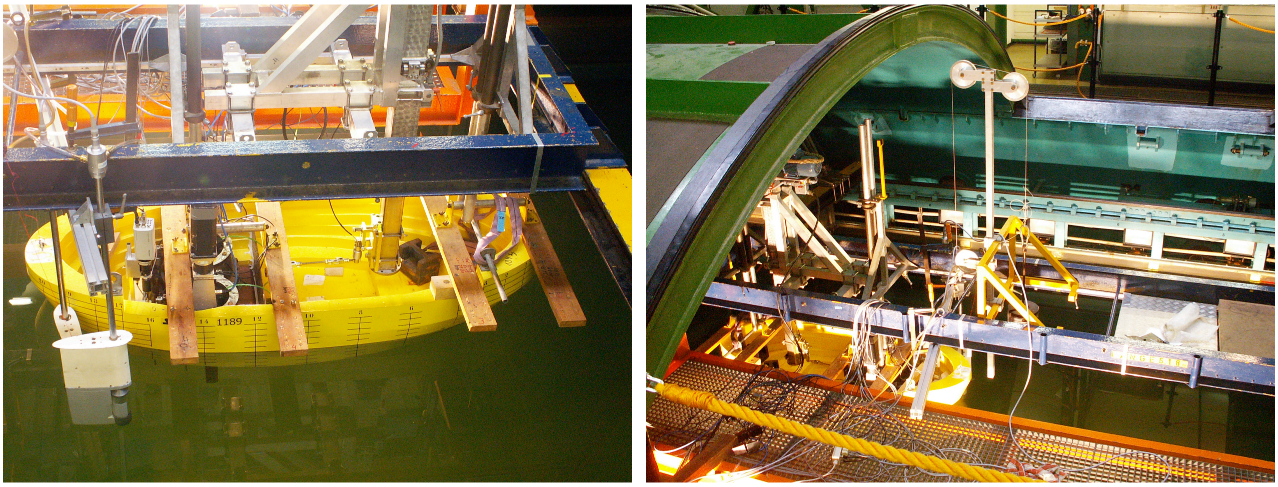


Fig. 21: VWT model in the test section of the UT2

The cavitation numbers of the VSP for the VWT are in the range between $\sigma_n = 2.176$ (water depth rotor casing) and $\sigma_n = 2.573$ (water depth blade end).

The bollard pull investigations with the VWT model have been carried out with different profiles, presented in Figure 7, on the VSP models. The measurements show, that in general the changes in the forces of the VSP due to the influence of cavitation are small. But it could be seen in the bollard pull measurements at cavitation similarity, that the blade profiles influence the cavitation behaviour and also the tendencies in changing of the VSP characteristic.

The Figures 22 and 23 show results of bollard pull measurements for the VSP with profile 4 and with profile 9. All data are at model Reynolds number. There is a very sting effect of the very low Reynolds number.

The cavitation at the VSP with the profile 4 leads to a slight increasing of torque coefficients (Figure 22). The thrust of the VSP with cavitation is a little bit smaller than without cavitation. The result is also a slight decreasing of the bollard pull at the design cavitation number.

The Figures 23 shows results of bollard pull measurements for the VSP with profile 9. The cavitation at the VSP blades with the profile 9 hasn't an influence at the thrust and torque coefficients as well as at the bollard pull.

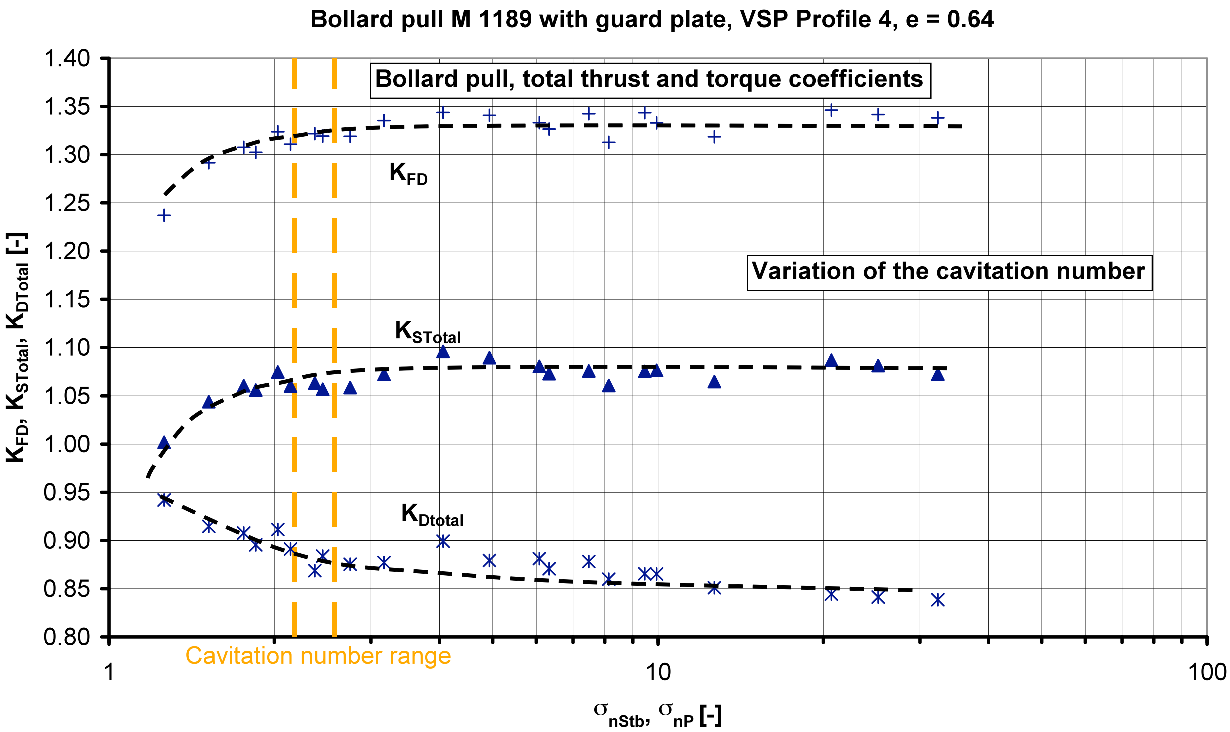


Fig. 22: Bollard pull, total thrust and torque coefficients at different cavitation numbers, VSP with profile 4, data at model Reynolds number

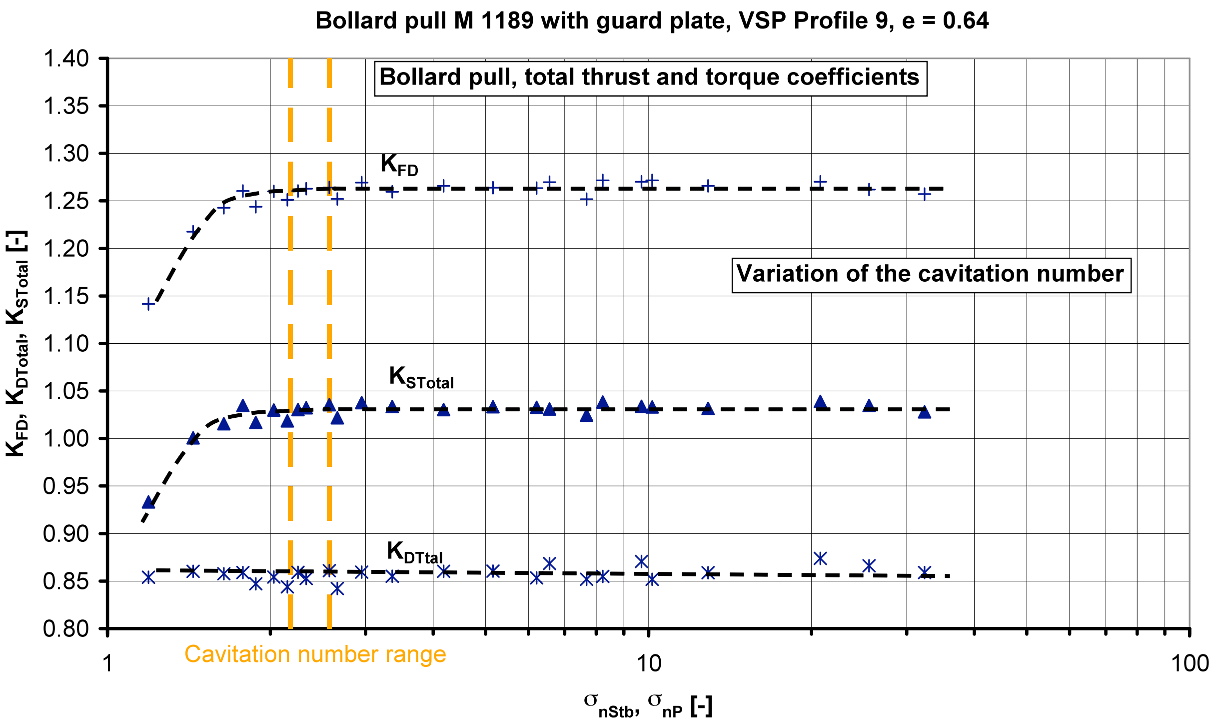


Fig. 23: Bollard pull, total thrust and torque coefficients at different cavitation numbers, VSP with profile 9, data at model Reynolds number

5 Summary

Cavitation tests with single VSP blades and with a VSP model have been carried out in the cavitation tunnel K15A. In addition bollard pull measurements at cavitation similarity have been carried out with a VWT model in the large circulating and cavitation tunnel UT2 of the TU Berlin.

A test arrangement for cavitation tests with a VSP model was developed and tested at the SVA Potsdam. A 3-component VSP-balance was used for the measurement of the longitudinal and transverse forces.

Different VSP blade profiles have been investigated in the cavitation tests. The blades are partly characterised by asymmetrical profiles.

The force measurements showed that the danger of a thrust break down due to cavitation is small. Cavitation at the blades appears only in a limited angle range. The cavitation separates rapidly from the blade and floats with the stream in the VSP area. The cavitating flow hits the following blade and leads to an increasing of the cavitation thickness at this blade.

Sheet cavitation appears mainly at the blades. Vortex cavitation could be observed at the blade end. This kind of cavitation at the VSP is not erosive. Erosive cavitation, like bubble or cloud cavitation, has not been observed on the VSP blades during the different cavitation tests.

The forces measurements and especially the high speed videos give a deep insight into the cavitation phenomena of Voith Schneider Propellers. The nonstationary working mode is the main reason for unique effects. The cavitation separates from the blades and therefore creates neither erosion on the blade surface or thrust reduction.

The cavitation at the blades has to take in account for choose of the profiles. A variation of the high lift profiles of the VSPs can guarantee a high thrust at very low cavitation numbers.

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NUMERICAL PREDICTION OF CAVITATION FLOW ON A MARINE PROPELLER USING A CFD CODE

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ABSTRACT

In this paper, non cavitating and cavitating flow around a B series marine propeller is studied using a RANS CFD code. Incompressible RANS equations are solved with SST k-ω model for the turbulence modelling. The predicted non-cavitating open water performance of the marine propeller agrees well with the analytical-experimental code Propol.

The mixture multiphase model is used in the current work for the numerical simulation of cavitating flows based on the full cavitations model which accounts for all first order effects i.e., phase change, bubble dynamics, turbulent pressure fluctuations, and non-condensable gases developed by Singhal et al. [1]. Before any attempt of computing cavitating propeller flows, we have validated against a benchmark problem for cavitating flows on Clark-Y hydrofoil. The leading edge and cloud cavitation on the hydrofoil is reproduced well and shows good comparison with the well-known experimental data. Finally, the cavitating propeller performance as well as tip and sheet cavitation is presented.

NOMENCLATURE

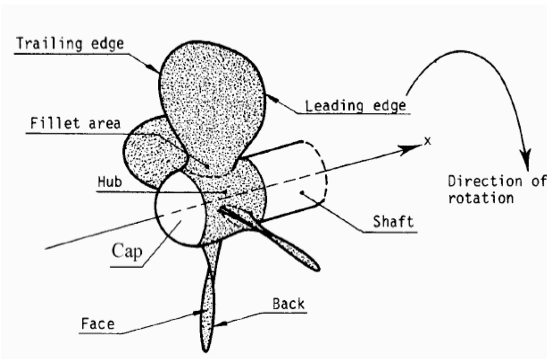
$$\text{Propeller advance coefficient } J = \frac{V}{nD}$$

$$\text{Propeller thrust coefficient } K_T = \frac{\text{Thrust}}{\rho n^2 D^4}$$

$$\text{Propeller torque coefficient } K_Q = \frac{\text{Torque}}{\rho n^2 D^5}$$

$$\text{Propeller pressure coefficient } C_P = \frac{P - P_\infty}{\frac{1}{2}\rho (nD)^2}$$

$$\text{Cavitation number } \sigma = \frac{P_\infty - P_v}{\frac{1}{2}\rho (nD)^2}$$



S Pitch [m]

D Propeller diameter in [m]

n Propeller revolution in [rev/s]

P Static pressure [Pa]

P_∞ Static pressure at far field [Pa]

P_v Vapour pressure [Pa]

ρ Density of water [kg/m^3]

V Mean inflow velocity in [m/s]

K Turbulent kinetic energy [m^2/s^2]

ω Specific turbulence dissipation rate[1/s]

1. INTRODUCTION

Cavitation occurs when pressure surrounding the propeller dips below the water's 'vapor pressure'. This causes the water to produce bubbles or cavities of water vapor - typically at the face, tip, or back of the propeller. Face cavitation usually occurs only on propellers with uncharacteristically low S/D ratios at high vessel speeds. A major threat to propeller corrosion, excessive face cavitation is caused by a negative blade angle of attack. This is generally indicated when the S/D ratio is less than, or close to, the advance coefficient, J. Tip cavitation is typically indicated by excessive tip speeds. Tip cavitation generally does not affect thrust, but can produce noise and contribute to blade corrosion. Finally, back cavitation appears in heavily loaded propellers, and is the principal cause of blade corrosion and thrust loss. Back cavitation is indicated by excessive blade pressure (too much lift) or cavitation percentage, as well as a blade area ratio less than the recommended.

It occurs in nearly all hydraulic machinery. Performance of different applications such as sub-marine propulsion and hydropower turbines are affected by cavitation. Cavitation from submarine propellers generates unwanted noise, while hydro turbine cavitation causes enormous maintenance costs for the hydropower industry.

The numerical modeling of such a cavitation has received a great deal of attention, it is still very difficult and challenging task to predict such complex unsteady and two-phase flows with an acceptable accuracy. Early studies in cavitation modeling were based on the potential flow theory and are still used in various engineering applications.

Tulin [2, 3] first introduced a linear theory to analyze the cavitating flow around a two-dimensional hydrofoil. This theory somewhat resembles the thin wing theory. It assumes that the cavity and the foil thicknesses are thin, compared to the foil chord length. Based on this assumption, the dynamic boundary condition on the cavity surface can be significantly simplified by specifying a constant horizontal perturbation velocity there. Geurst [4] employed the conformal transformation technique to derive theoretical expressions of the lift, drag, and moment coefficients, each of which contained coefficients in integral form. He also studied the special case of a flat-plate hydrofoil and obtained simple analytical expressions of these coefficients in terms of the angle of attack of inflow and the transformation parameters. Nevertheless, all these results apply only to the flow at a small angle of attack, as the assumptions of the linear theory imply.

With the modern evolution of computational methods, several nonlinear numerical procedures of boundary-element type have been successfully developed for the solution of sheet cavitation, based on the early theoretical achievements. In these approaches, cavity surface conditions are usually satisfied on the exact cavity surface that is part of the solution and determined iteratively by proper computational algorithms. Uhlman [5, 6] employed a velocity-based nonlinear boundary element method to obtained solutions for partially-cavitating and supercavitating hydrofoil flows. Kinnas and Fine [7] developed a potential-based nonlinear boundary element method for the non-linear analysis of inviscid cavitating flow around hydrofoils or propeller blades. Wang, et.al [8] experimentally studied stationary and non-stationary characteristics of attached, turbulent cavitating flows around solid objects. Different cavitation regimes, including incipient cavitation with traveling bubbles, sheet cavitation, cloud cavitation, and supercavitation, are addressed along with both visualization and quantitative information. Phenomena such as large-scale vortex structure and rear re-entrant jet associated with cloud cavitation, and subsequent development in supercavitation are described.

CFD methods compliment the experimental tests in design. In conjunction with traditional towing tests and cavitation tests and with analytical methods based on circulation theory and standard series, CFD codes represent a new capability to greatly improve the propeller design and analysis process. RANS methods have been successfully applied not only to viscous flow around ship hulls but also to marine propellers. Watanabe et.al [9] has applied unstructured grid technique to the flow around the Seiun-maru highly-skewed propeller. The agreement with experiment was good both for steady and unsteady conditions. Numerical modelling of cavitation has, until recently, only been possible using cutting edge in-house CFD codes. Special challenges occur since cavitating flows are highly dynamic in nature. Also, such flows are characterized by large gradients in the density field, which is known to cause numerical instabilities.

The work of non cavitating flows [10] is extended here using Fluent 6 [11] to determine the cavitating characteristics of a B series propeller, see [12]; the geometry of this propeller is obtained by Propcad software as given in this website. The profile generation methods are discussed in [13]. Here the output geometry provided in the website [12] is directly taken to build the CAD model in UG. In order to validate the results obtained in Fluent, the software Propol [15] is used to determine the propeller characteristics, see [14]. Propol code uses experimentally obtained open-water characteristics expressed as polynomials, and B-series propellers are tested in [15]. All test data was corrected for Reynolds effects by means of an equivalent profile method. The propeller and its characteristics given in [9] are used in Propol and the results obtained are in good agreement.

The second objective of this study is to investigate the cavitating unsteady flow around a Clark-Y hydrofoil for any attempt of computing cavitating propeller flows. The leading edge and cloud cavitation on the hydrofoil is reproduced well and shows good comparison with the experimental data available in Wang, et.al [8]. Finally, the cavitating propeller performance as well as tip and sheet cavitation is presented.

2. CFD MODEL

The propeller geometry is given in Table 1.

Table 1: Principal Characteristics of B-Series propeller

Propeller type	Fixed pitch
Rotation	Right
Number of blades	4
Diameter [m]	0.508
Pitch [m]	0.508
Pitch ratio at 0.7R	1
Expanded area ratio	0.650
Skew angle [Deg]	9.2
Rake [deg.]	10.00

Fig.1 shows the propeller geometry and the airfoil cross-sectional details at various heights are given in Fig. 2. In all there are 12 cross sections, bottom 8 of them from centerline are placed 1" apart. Top 4 sections are placed at half of the distance of previous sections, i.e., ½, ¼ … There are 28 points per section in building the airfoil. The hub diameter varies from LE 3.7" to TE 3.2". The propeller speed is taken to be 300 rpm in the CFD calculations. The solid model of the propeller is given in Fig. 3.

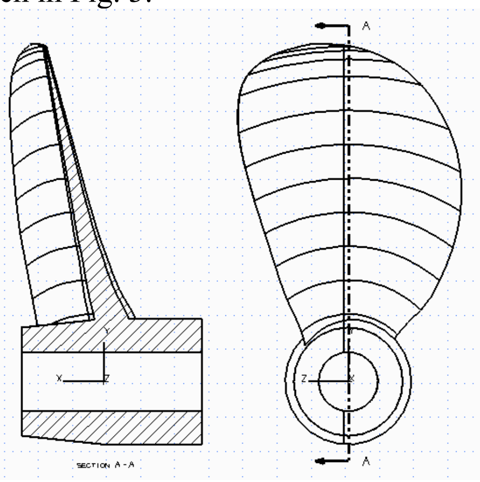


Figure 1: *Series B* Propeller Geometry

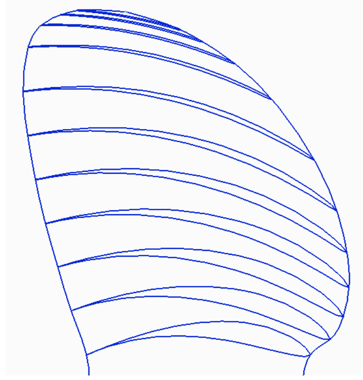


Figure 2: Propeller airfoil cross-sectional details

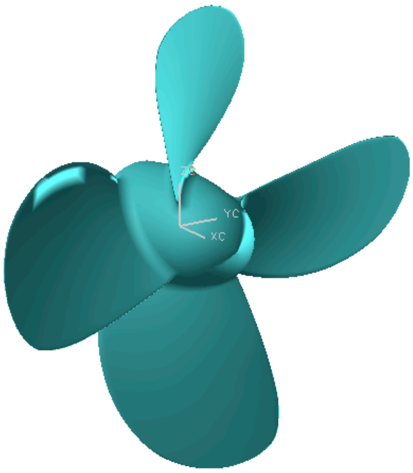


Figure 3: Solid Model of the Propeller

Main problem in computing propeller flows by solving Navier-Stokes equations is the complexity involved in generation of suitable grids. Compared to other lifting bodies, like wings on airplanes, there are additional difficulties associated with strong twisting of the blade central plane and complex shape of modern propellers.

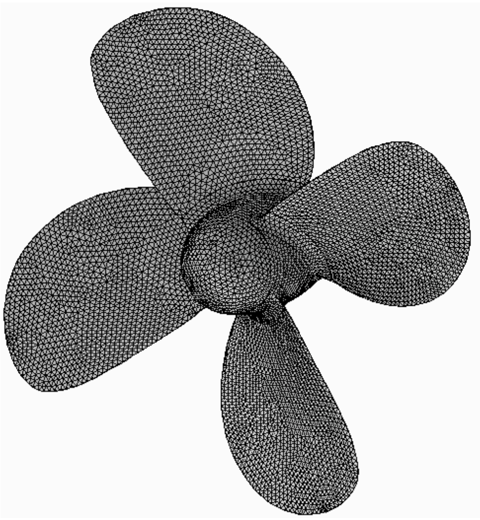


Figure 4: Propeller Surface Mesh

Gambit is used to create the flow domain and the grid. The propeller surface mesh is shown in Fig. 4. Tetrahedral elements are used in the immediate vicinity of the propeller to match the complicated geometry of the blades, while elements are fine near the surfaces of blades in order to improve the resolution of boundary layers on blades. Fig. 5 shows a schematic of the cylindrical computational domain, whose diameter is 2m and length is 5m. Sizing functions were used to control the growth rate of the grid size to obtain a final mesh with size of approximately 1.3 million control volumes, Y+ values are maintained in the range 5-50.A constant free-stream velocity boundary condition was specified at the inlet boundaries. On the exit boundary, the static pressure was set to a constant value zero. On the outer boundary, the no slip boundary condition was imposed. The boundary condition on the propeller, hub and the conical tip were specified as rotating wall, while the shaft was stationary.

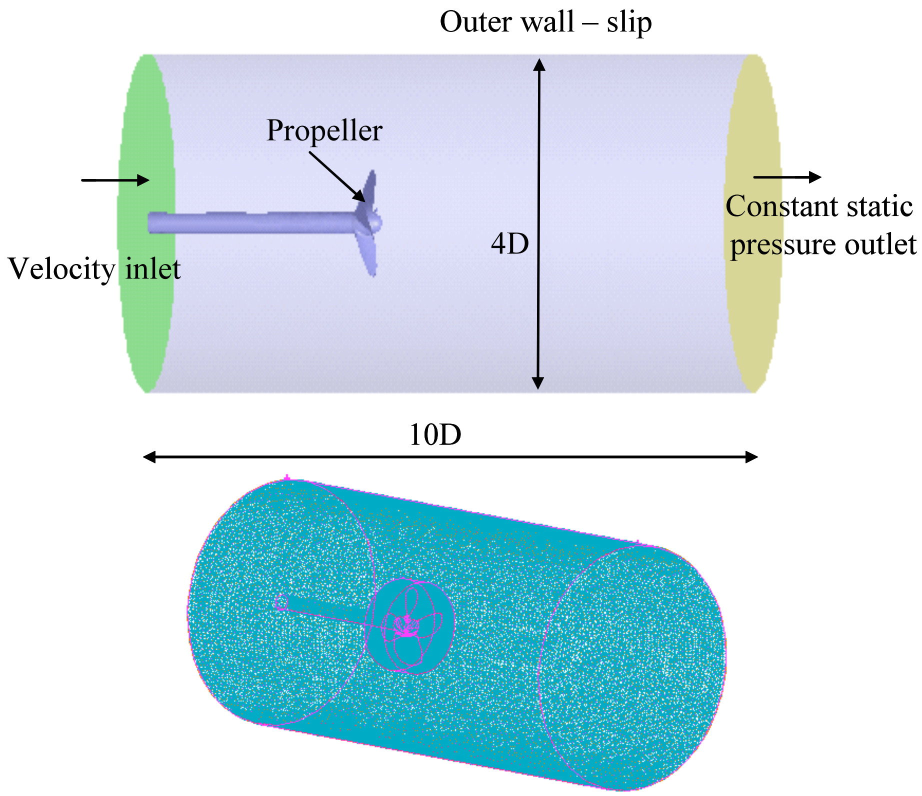


Figure 5: Computational Domain and Mesh

Incompressible RANS equations were solved in Fluent version 6 with SST K-ω model for the turbulence modelling. The K-ω model uses two equations to represent the turbulence, but instead of ε, it calculates a specific turbulence dissipation rate, which can be considered the ratio of ε to K. The SST (Shear-Stress Transport) K-ω model takes into account the transport of turbulent shear stress, and makes a gradual change of solution variables from the standard K-ω model in the inner region of the boundary layer to a high Reynold's number version of the K-ε model in the outer part of the boundary layer. The SST K-ω model is more reliable for flows which have adverse pressure gradients, and most widely adopted in turbomachinery applications. For non cavitating flow, single phase flow is used. Density of water is taken as 1000 kg/m^3 and viscosity is taken equal to 0.001 kgm/s. A segregated solver with SIMPLE as the velocity-pressure coupling was selected, and QUICK scheme was used for the discretization of the momentum equation.

3. SINGLE PROPELLER BLADE RESULTS

First, open water conditions were simulated for the isolated (free) propeller. To impart the airfoil action of the blade, a moving reference frame around the blade as shown is given an angular velocity 300 rpm applied to the direction of rotation of the propeller.

The static pressure distribution on the pressure (face) and suction (back) surfaces of the blade is given in Figs. 6a and b. The difference in the static pressure distribution gives the thrust on the blade. The thrust coefficient obtained for four blades is shown in Fig. 7. The result obtained is more than the experimental result predicted from Propol. Here, periodic boundary condition is not used and therefore the result predicted is upper bound by about 20%.

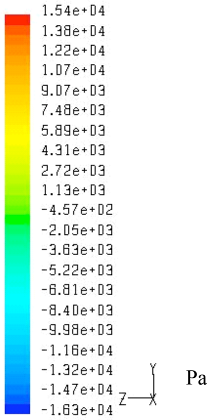


Figure 6a: Static Pressure contours on face surface at J = 0.4

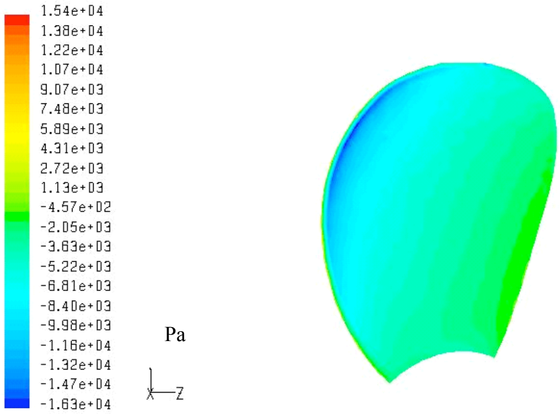


Figure6b: Static Pressure contours on back surface at J = 0.4

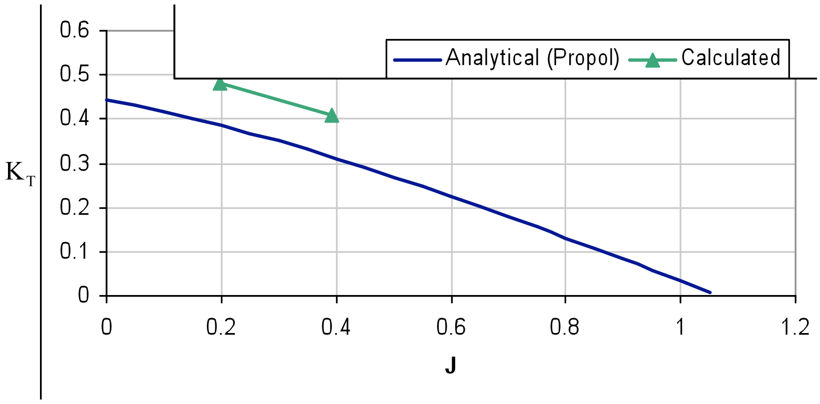


Figure 7: Thrust Coefficient for full Propeller Blade based on Single Blade Analysis

4. FULL PROPELLER RESULTS

The full propeller is analyzed in a similar way as the single propeller blade to determine the propeller open-water characteristics. The static pressure distribution on the pressure (face) and suction (back) surfaces of propeller blades is given in Figs. 8a and b for J = 0.2 (inlet velocity 0.5 m/s). The pressure coefficient on the back surface is given in Fig.9. Fig.10 shows velocity magnitude for the advance ratio J = 0.2. The flow is accelerated as it passes through the propeller. The acceleration of fluid is related to the pressure gradient, which in turn determines the thrust and torque on the propeller. Part of the acceleration occurs upstream of the propeller as the pressure on upstream (suction) side of the blade is lower than the ambient pressure and part of the acceleration occurs downstream as the pressure on downstream (Pressure) side of the blade is higher than the ambient pressure.

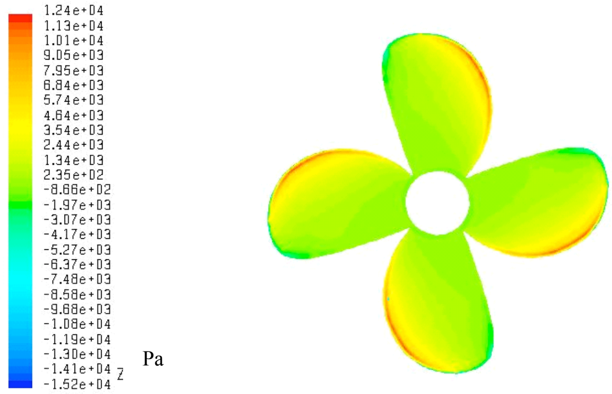


Figure 8a: Static Pressure contours on face surface at J = 0.2

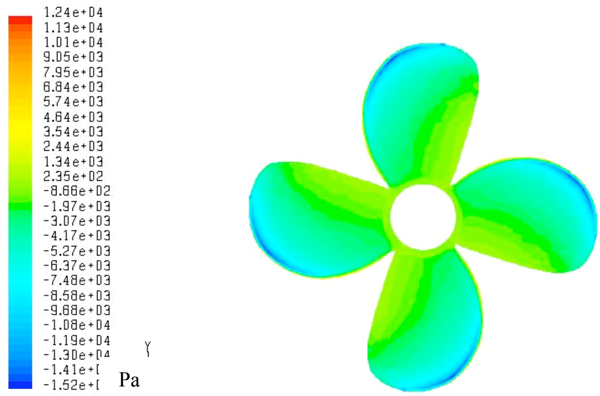


Figure 8b: Static Pressure contours on back surface at J = 0.2

The torque coefficient predicted by the CFD calculations, however, is more than that predicted by *Propol* analytical values. Both the coefficients decreased with J and follow the same trend as generally observed. The inlet velocity is varied from J = 0.2 to 0.8 and the propeller performance map obtained for thrust and torque coefficients is shown in Fig. 11. The analytical results obtained by *Propol* are also shown in the same figure

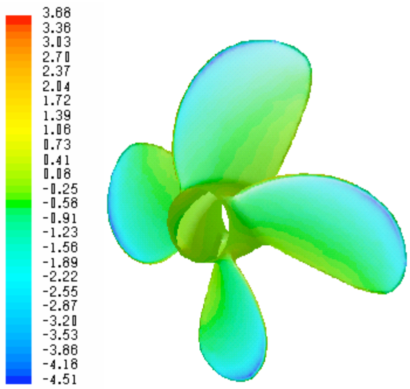


Figure 9: Pressure coefficient contours on back surface at J = 0.2

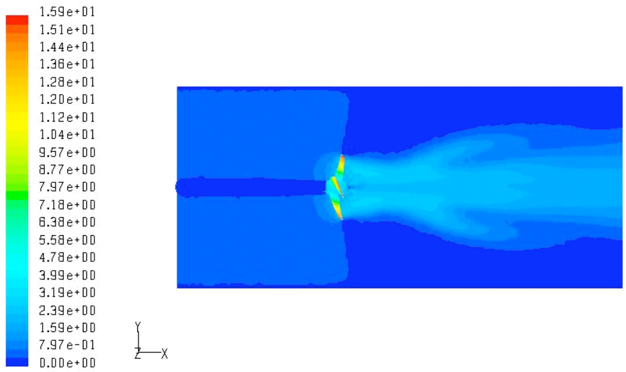


Figure 10: Mid-Plane Velocity Magnitude m/s

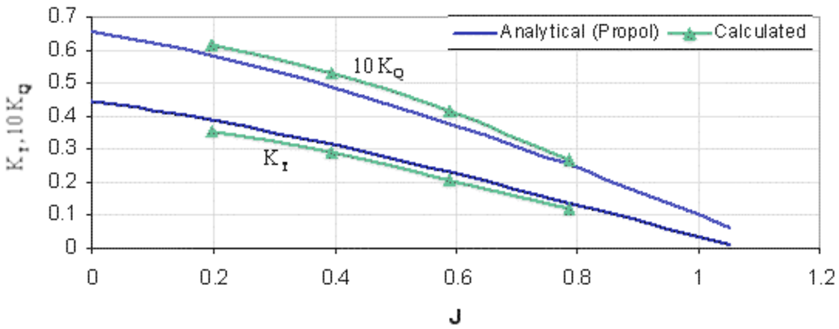


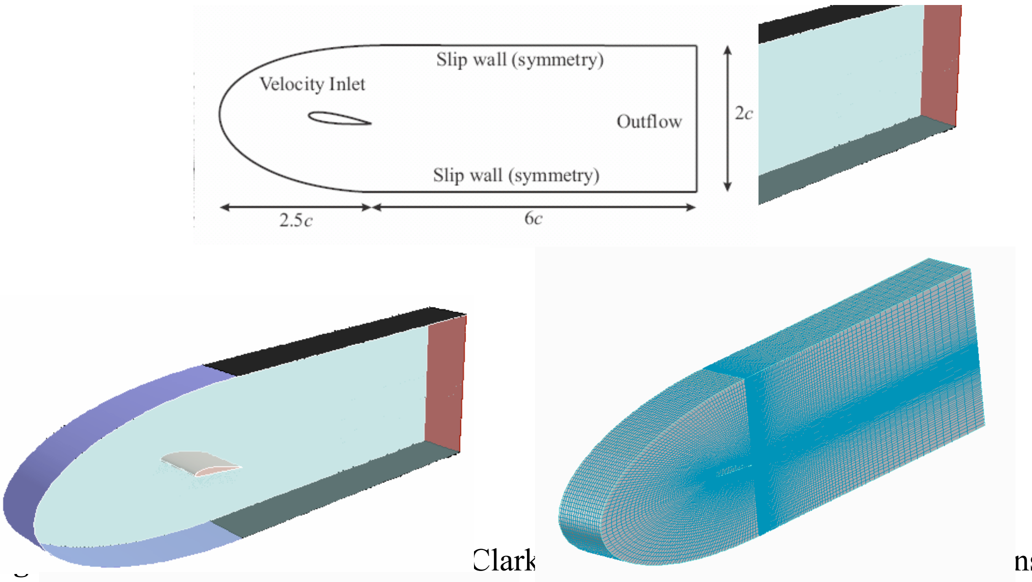
Figure 11: Analytical and Calculated Performance of *B-Series* Propeller Model in Open Water

The results for thrust have now considerably improved over the single propeller blade analysis; in fact, the analytical results predicted from experimental values using *Propol* are slightly more than CFD results which are conservative in nature.

5. CLARK-Y HYDROFOIL RESULTS

Before any attempt of computing cavitating propeller flows, we have validated against a benchmark problem for cavitating flows on Clark-Y hydrofoil and compared with experimental data of Wang, et.al [8]. Fig.12 shows the total 2D computational domain showing lengths and boundary conditions. The domain extends approximately one chord length in both directions from the foil. Upstream, the grid stretches 1.5 chords from the leading edge and 6 chords downstream measured from the trailing edge. The total grid size was approximately 0.6 million grid cells and the first cell is located at y+ <30 along the foil. The chord length of the foil is 70 mm. The inlet boundary condition is specified velocity, using a constant velocity profile. Upper and lower boundaries are slip walls, i.e. a symmetry condition. The outlet used is a constant pressure boundary condition. In the analysis, 8 degree angle of attack is considered. The inlet velocity is set to 10 m/s. This is in the same range as the experimental data.

The mixture model is used in the current work for the numerical simulation of cavitating flows. In this model, the flow is assumed to be in thermal and dynamic equilibrium at the interface where the flow velocity is assumed to be continuous. The cavitation model implemented here is based on the so-called “full cavitation model", developed by Singhal et al. [1]. It accounts for all first-order effects (i.e., phase change, bubble dynamics, turbulent pressure fluctuations, and non-condensable gases).



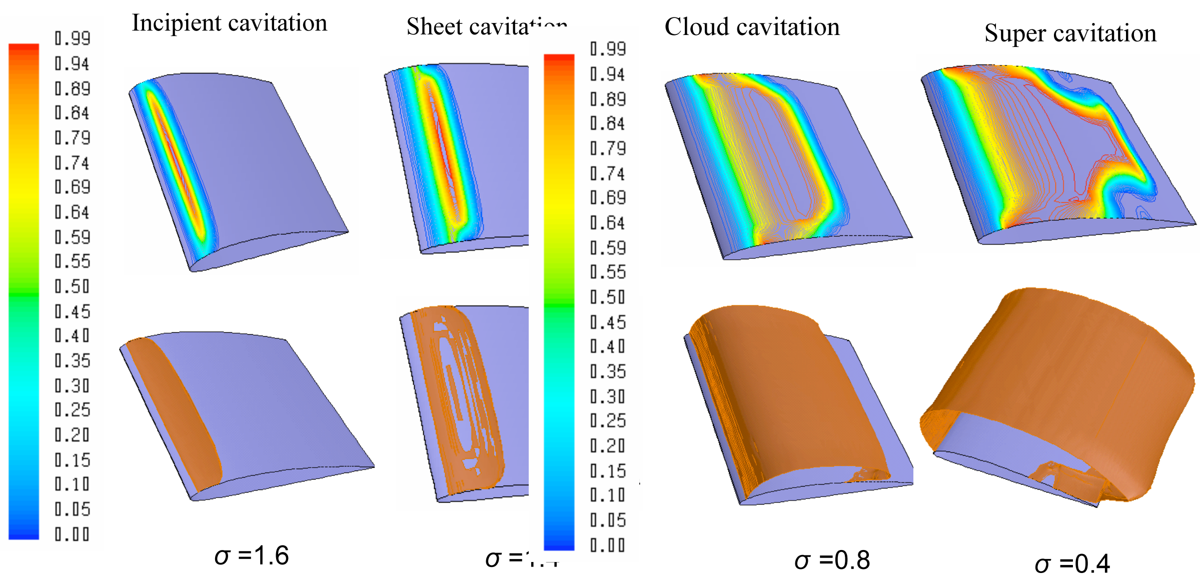


Figure 13: Different cavitations types in a Clark-Y hydrofoil

Fig. 13 shows different cavitation types of cavitating flow in Clark-Y hydrofoil. In the incipient cavitation stage, with non separated flows, the case shown is for cavitation number σ = 1.60. Further lowering the cavitation number, the cavitations changes from the incipient cavitation to a sheet like cavity. In cloud cavitation, the development of the cavity is qualitatively similar to that of sheet cavitations. Supercavitation is the final state of cavitation, which is caused by further decreasing the cavitation number from the cloud cavitation regime. Due to the very low pressure, the cavitating area covers the entire hydrofoil, extending to the downstream region of the solid object. The leading edge and cloud cavitation on the hydrofoil is reproduced well and shows good comparison with the experimental data available in Wang, et.al [8].

6. CAVITATING PROPELLER RESULTS

Finally, the cavitating propeller performance is studied. Boundary conditions for the cavitating cases are set in the same way as for the non-cavitating cases. The only difference is on the exit boundary, where the constant exit pressure is set to match the given cavitation number σ.

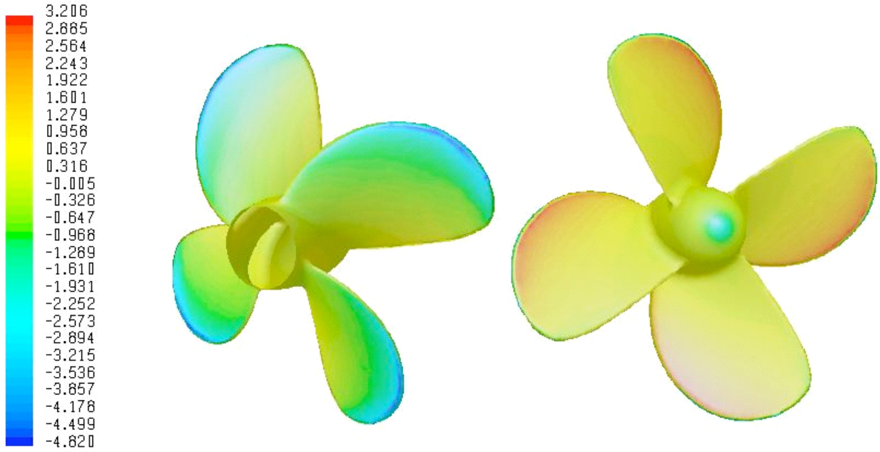


Figure 14: Pressure coefficient contours at J = 0.2 and σ = 1.5

Fig.14 shows the blade back and face pressure coefficient contours at J = 0.2 and σ = 1.5. It is clearly seen that cavitation is to occur in the tip area. This prediction of cavitation inception can be confirmed by the contours of vapor volume fraction on the backside of the blade, Fig.15, in which the high vapor volume fraction area closely matches the low-pressure area in Fig.14. The hub cavitation is also observed in the propeller. The computed, iso-surface of vapor volume fraction of 0.1 is shown in Fig 16. Further lowering the cavitation number, the cavitation changes from tip cavitation to a sheet like cavity. Due to the very low pressure, the cavitating area covers the entire propeller face. Figs.16 and 17 show the blade back vapor volume fraction and iso surface contours at J = 0.2 and σ = 0.5.

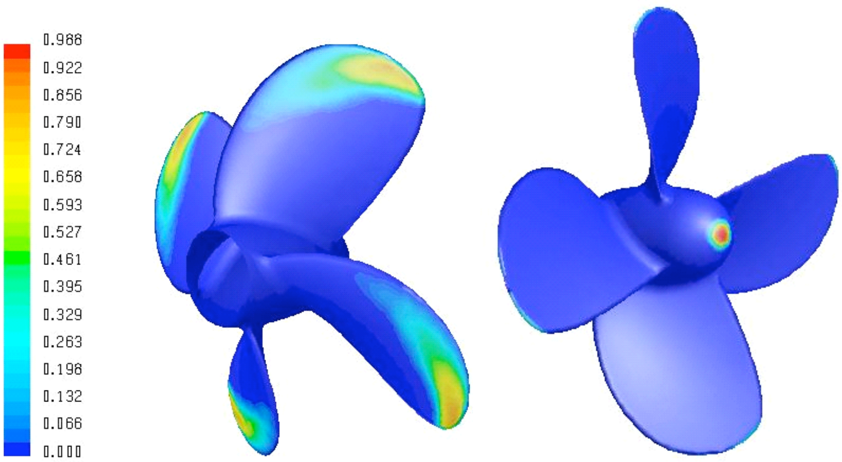


Figure 15: Vapor Volume fraction contours on back and front surface at J = 0.2 and σ = 1.5

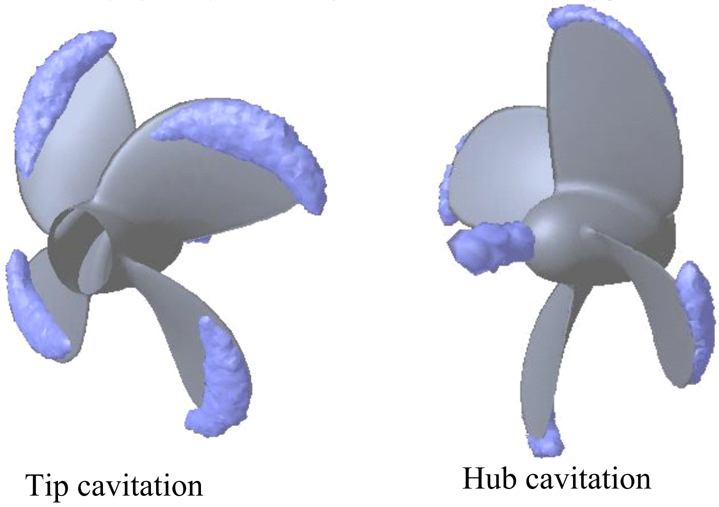


Figure 16: Iso-surface of vapor volume fraction of 0.1 on back and front surface at J = 0.2 and σ = 1.5

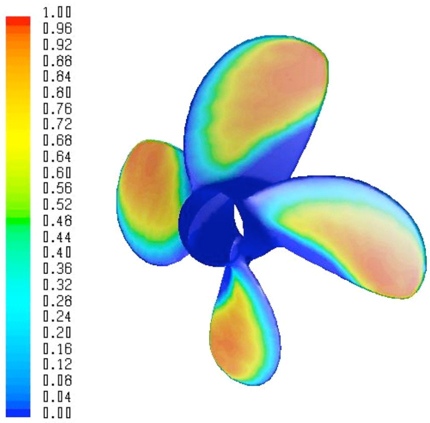


Figure 17: Vapor Volume fraction contours on back surface at J = 0.2 and σ = 0.5

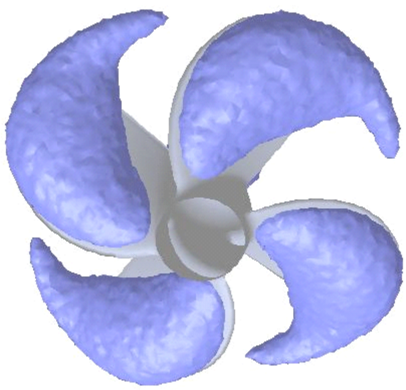


Figure 18: Iso-surface of vapor volume fraction of 0.1 on back surface at J = 0.2 and σ = 0.5

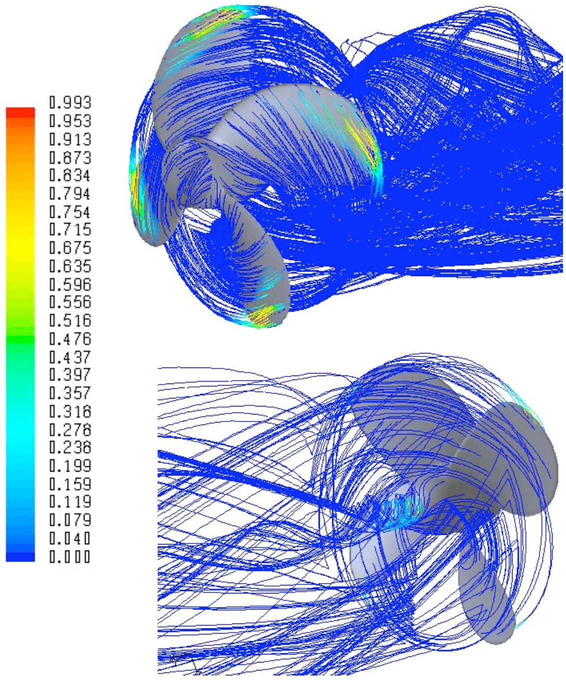


Figure 19: Path lines colored by volume fraction of vapor on front and back surface at J = 0.2 and σ = 1.5

Fig.19 shows Path lines colored by volume fraction of vapor on front and back surface at J = 0.2 and σ = 1.5. Due to the complexity of propeller geometry, it is not easy to generate sufficiently fine grid to resolve the tip vortex. For future extension fine grid refinement is required to capture the complete tip vortex.

7. CONCLUSIONS

In this paper, we have demonstrated the procedure to determine the characteristics of a marine propeller without the hull using Fluent CFD code.

A *B-Series* Propeller Model in Open-Water, whose experimental data is available, is chosen for the CFD calculation. First a single blade is analyzed and the results obtained are in fairly good agreement; CFD results are nearly 20% more than the experimentally predicted value. When the full model is considered, the CFD results are in close agreement with the experimentally predicted values. The thrust coefficient predicted is slightly lower and the torque coefficient slightly higher over the experimentally predicted analysis.

Then the numerical investigation of cavitating flows in a marine propeller using the mixture multiphase model. The cavitation model is validated for the flow around a Clark-Y hydrofoil. The numerical results agree very well with the experiments results from literature. As a result we include that the present cavitational model is able to capture the major dynamics of attached cavitating flows.

As the authors proceed with this research, we are focusing on several areas including: 1) improved physical models for turbulence, 2) extension to coupled simulation with a ship hull and a propeller.

ACKNOWLEDGMENTS

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Class Rules and Complementary CFD Simulations for OSVs

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ABSTRACT

Today's offshore service vessels (OSVs) are larger, more specialized, and more technically sophisticated to meet demands of current deepwater field developments. Analysts forecast a steady growth of the global OSV fleet through 2020. The growing market initiated specific rule developments for such ships that resulted in a new set of class rules presented here. There is increasing recognition that the design and normal operation of such ships differ significantly from those of general cargo ships. As a consequence, comprehensive international regulations are needed that take specific account of the practical constraints of these ships. Some of the difficulties faced by designers of OSVs are caused by the operational requirements in offshore environments. For example, intact and damage stability requirements need to account for the unobstructed stern area needed for cargo handling as well as for duties such as anchor handling and towing. However, prescriptive class rules cannot cover all safety related design issues. Complementary computational fluid dynamics (CFD) simulations are suitable measures to assess safety aspects beyond the scope of class rules, such as operations in severe seas that can adversely affect the ship's controllability, causing loss of stability that ultimately compromises the survivability of the ship. In this paper, we demonstrated the ability of a Reynolds-averaged Navier-Stokes (RANS) solver to simulate the motion behavior of an OSV operating in stern quartering seas as these motions may affect the ship's dynamic stability. Systematic investigations based on a validated CFD technique can be useful not only to develop, but also to continuously update OSV rules. The specification of design loads for OSVs under extreme conditions, for example, is a critical issue. To demonstrate this, we also performed simulations of the OSV in severe head seas to obtain wave-induced slamming pressures in the bow area.

INTRODUCTION

Offshore service vessels have become progressively more technically sophisticated in response to demands of complex deepwater field developments. The global OSV fleet, now comprising some 2500 ships, is forecasted to see a steady growth through 2020. Part of this growth can be attributed to the expanded definition of an offshore service vessel (OSV). This term includes not only traditional supply boats, but also anchor handling tug/supply ships, well stimulation ships, and standby ships, among others. In addition, these ships may be built to carry hazardous and noxious substances, to fight fires, or to occasionally recover oil.

It is increasingly recognized that the design and normal operation of such ships differ significantly from those of general cargo ships. Therefore, comprehensive international regulations are needed that take specific account of the practical constraints of these ships. Originally adopted in 1981, the International Maritime Organization (IMO) Resolution A.469 (XII) was the first document to specifically address OSVs. Although it was revised by MSC.235 (82) in 2006, its scope is limited, primarily addressing stability issues. The IMO adopted other statutory developments to address the OSV sector. For example, IMO Resolution A.673 (16), adopted in 1989, provides guidelines for the transport and handling of limited amounts of hazardous and noxious liquid substances in bulk on OSVs, and IMO Resolution A.863 (20), adopted in 1997, provides a code of safe practice for the carriage of cargoes and persons by OSVs.

As these ships are increasingly involved in specialized work in deep waters at greater distances from shore, they are required to be stationed precisely in the target area. At the same time, they must be able to effectively move away temporarily when the situation arises and quickly return to complete the work. For this kind of operation, the ship would be fitted with a dynamic positioning system. The IMO Resolution MSC.235 (82), superseding IMO Resolution A.469 (XII), documents important issues on positioning systems for OSVs.

However, these resolutions are mostly voluntary and cover selective operational, design, and construction aspects. In view of this, it can be argued that the IMO resolutions are either confusing in the manner in which they must be applied to OSVs or that they do not go far enough. Consequently, designers of these ships have had to make many compromises over the years, requiring builders and owners to apply for regulatory exemptions, on a case-by-case basis, from flag administrations.

Tang et al. (2007) raised the following issues that illustrate the difficulties faced by designers of OSVs: location and arrangements of navigational lights, navigation bridge visibility, guard rails, double bottom, stern tubes, and intact and damage stability. In the second part of their paper, Tang et al. briefly highlighted the various regulatory documents that affect OSVs in their operation, design, and construction aspects. Given that these documents contain requirements that are mostly voluntary, at least one has been made mandatory by the International Convention for the Prevention of Pollution from Ships (MARPOL). The main classification societies have drawn up specific rules for OSVs and, generally, these rules are continuously upgraded to incorporate these regulatory changes.

Operations in severe seas may adversely affect the ship's controllability, causing loss of stability that ultimately compromises the survivability of the ship. Complementary to the standard rule-based safety assessment, such safety aspects may be assessed by, for example, numerical simulations based on computational fluid dynamics (CFD) techniques that directly solve the Reynolds-averaged Navier-Stokes (RANS) equations to predict the behavior of an OSV operating in severe seas. The usefulness of this technique for wave-induced ship-related problems has been amply demonstrated in recent years, i.e., el Moctar et al. (2006) and Schellin and el Moctar (2007). Here we analyzed the behavior of a typical modern OSV in waves representing an anchor handling operation. Carried out on a systematic basis and with results validated against measurements, such simulations can turn out to be a useful, if not an essential, tool for future rule developments. Based on user-coded RANS routines to obtain green water pressure distributions, Kahl et al. (2008), for instance, established new design rules for breakwaters on containerships. With this aspect in mind, we performed additional simulations to obtain wave-induced slamming pressures in the critical bow area of the ship under head sea conditions for comparison with rule values.

OSV CLASS RULES

Class Notation

A classification society generally assigns a class notation, notably Offshore Service Vessel, to sea-going ships specially designed for support service to offshore installations and built to applicable requirements of the society's construction rules. At the request of the owner, ships equally intended for additional services and having the associated functional equipment may be assigned an additional notation as specified, for example, in Table 1. Such services include the capability of carrying specialized stores and cargoes to mobile offshore units and other offshore installations, performing offshore anchor handling and towing duties, stimulating wells, fighting fires, performing standby and rescue operations, and storing and transporting recovered oil.

Table 1 Additional class notations for OSVs

HNLS	Carrying hazardous and noxious liquid substances
AH	Anchor handling
WS	Well stimulation
Fire Fighter	Fire fighting
Standby	Standby and rescue
Oil Recovery	Oil recovery and transportation

Material, design loads, and longitudinal strength of OSVs shall comply with relevant specifications for main class. Ships built in compliance with class rules for OSVs are subject to additional requirements concerning items such as hull arrangement and scantlings, cargo handling arrangement, intact and damage stability, superstructures and deckhouses, and special equipment.

Hull Arrangement and Scantlings

Regarding hull arrangement and scantlings, special requirements need to be fulfilled for fenders, frames, shell plating, and side longitudinals. Where cargo is carried on deck, knot free wooden sheathings should cover the deck to protect the steel plating from mechanical damage, and effective means such as stow racks, steel cradles, or steel or wooden dunnage are to be provided to uniformly distribute the cargo weight in the deck structures. The cargo deck plating itself is to have a minimum thickness of 8 mm, and in deck areas of heavy cargo units, such as drilling rig anchors, the deck structure shall be adequately strengthened. Due regard is to be given to the arrangement of freeing ports to ensure the most effective drainage of water trapped in pipe deck cargoes and in recesses at the aft end of the forecastle.

Cargo Handling Arrangement

Ships that occasionally handle, store, and transport recovered oil from a spill and ships intended for transportation of liquids with a flash point below 60°C shall comply with special requirements. Cargo pumps shall be provided with remote shut down devices and, where cross contamination causes safety hazards or marine pollution, segregation between cargo piping systems shall be by means of spectacle flanges, spool pieces, or equivalent. Where cargo tanks for dry cement or mud are fitted, these cargo tanks are to be separated from the engine room and accommodation spaces by steel bulkheads and decks. Where tanks for hazardous and noxious liquid cargo are fitted, the quantities of cargo are limited, and segregation and construction of tanks for hazardous and noxious liquid cargo are to comply with special requirements.

Stability

One key objective of Resolution IMO A.469 (XII) was to adress stability issues of OSVs. Thus, stability issues play a dominant role in class rules for these ships. Recognized standards of the ship’s intact and damage stability, including subdivision, are to be complied with. However, where compliance with these criteria is impracticable due to the ship’s characteristics, IMO Resolution A.469 (XII) provides alternative criteria that are acceptable for class. For intact stability calculations, specified loading conditions shall be presented, comprising the ship in fully loaded departure condition, in fully loaded arrival condition, in ballast departure condition without cargo and with full stores and fuel, in ballast arrival condition without cargo and with 10 percent stores and fuel, and in worst anticipated operating condition. If the ship is equipped with towing gear, a typical condition ready for towing shall be considered as well.

In addition to meeting intact stability requirements, class rules require that damage stability and subdivision of OSVs comply with recognized standards. As recommended by IMO Resolution A.469 (XII), damage shall be assumed anywhere along the ship’s length between transverse watertight bulkheads. Vertical extent of damage shall be assumed to extend from the underside of the cargo deck over the full depth of the ship. Transverse extent of damage shall be assumed to be 760 mm, measured inboard from the side of the ship perpendicular to the centerline at the level of the summer load waterline. Tunnels, ducts, or pipes that may cause progressive flooding when damaged shall be avoided in the damage penetration zone. If this is impossible, arrangements shall be made to prevent progressive flooding intact spaces. Alternatively, these spaces shall be assumed flooded in the damage stability calculations. Damage stability criteria shall take into account sinkage, trim, and heel, whereby the final waterline shall be below the lower edge of any opening through which progressive flooding may occur. In the final stages of flooding, the heel angle from asymmetric flooding shall not exceed 15 deg. If the deck does not immerge, this angle may be increased to 17 deg. Stability in the final stage of flooding may be regarded as sufficient if the range of the righting lever curve is at least 20 deg beyond the equilibrium position associated with a maximum residual righting lever of at least 100 mm. Limiting values of the vertical center of gravity and the metacentric height calculated on the basis of the ship's characteristics related to damage stability criteria shall be documented as a diagram.

Deckhouses

Due to their location at the forward end of the ship, deckhouses are to be reduced to essentials, and special care is to be taken to ensure that their scantlings and connections are sufficient to withstand wave loads. The fitting of windows and side scuttles shall comply with special requirements. For instance, windows may not be fitted at all locations, only on the second tier and higher above the freeboard deck (a) in the aft end bulkhead of deckhouse and superstructure and (b) in the sides of deckhouse and superstructure that are not part of the shell plating and at the third tier and higher above the freeboard deck in the forward facing bulkheads of deckhouse and superstructure. At all other locations, only side scuttles are acceptable.

Special Equipment

Special equipment installed on board shall satisfy certain class requirements. Thus, if the ship is designed for towing operation, the arrangement shall satisfy the requirements for anchor handling tug/supply ships. The steering gear shall be capable of changing the rudder angle from 35 deg on one side to 30 deg on the other side in 20 s with the ship underway at maximum service speed. Exhaust outlets are to be located as high as is practicable above the deck and are to be fitted with spark arrestors. For ships without means for dynamic positioning, but intended for anchoring close to offshore installations, it should be considered to increase the diameter and length of chain cables above minimum class requirements. Chain lockers are to be arranged as gas-safe spaces, and hull penetrations for chain cables and mooring lines are to be arranged outside gas-dangerous spaces.

STATUTORY RULES

Hazardous Materials

Class rules drawn up for OSVs that carry hazardous materials to and from offshore installations follow the IMO adopted Resolution A.673 (16) and its ammendments Resolution MEPC.158 (55) and IMO Resolution MSC.263 (82). These resolutions provide safety and pollution requirements for OSVs when carrying limited quantities of hazardous materials. Specifically, this means that quantities of such substances carried by OSVs are limited to 800 m^3 or a volume in cubic meters equal to 40 percent of the ship's deadweight, based on a cargo density of unity. Additives not considered hazardous may be carried in limited amounts not exceeding 10 percent of the ship's amximum authorized quantity of products.

Well stimulation vessels that are permitted to carry more than the maximum amounts specified above should be designed to meet the requirements for subdivision and intact and damage stability contained in IMO Resolution MSC.236 (82), which resolution documents guidelines for the design and contruction of OSVs. However, it shall be assumed that damage occurs anywhere along the ship's length at any transverse watertight bulkhead.

Cargo Tanks and Piping

Cargo tanks should be located at least 760 mm measured inboard from the side of the ship perpendicular to the centerline at the level of the summer load waterline. By means of a cofferdam, void space, cargo pump room, empty tank, oil fuel tank, or other similar space, tanks containing cargo or residues of cargo should be segregated from machinery spaces, propeller shaft tunnels (if fitted), dry cargo spaces, accommodation and service areas, and drinking water and stores for human consumption. On deck stowage of independent tanks or installing independent tanks should be considered to satisfy this requirement. Cargoes that react in a hazardous manner with other cargoes or oil fuels should be segregated from such other cargoes or oil fuels by means of a cofferdam, void space, cargo pump room, empty tank, or tank containing a mutually compatible cargo. Furthermore, such cargoes should have separate pumping and piping systems that do not pass through other cargo tanks containing such cargoes unless encased in a tunnel, and they should have separate tank venting systems.

Cargo piping should not pass through any accommodation, service, or machinery space other than cargo pump rooms or other pump rooms. Pumps, ballast lines, vent lines, and other similar equipment serving permanent ballast tanks should be independent of similar equipment serving cargo tanks. Bilge pumping arrangements for cargo pump rooms or for hold spaces where independent cargo tanks are installed should be situated entirely within the cargo area.

Where not bounded by bottom shell plating, fuel oil tanks, a cargo pump room, or a pump room, the cargo tanks should be surrounded by cofferdams. Tanks for other purposes (except tanks for fresh water and lubricating oils) may be accepted as cofferdams. For access to all spaces, minimum spacing between cargo tank boundaries and adjacent ship structures should be 600 mm. Cargo tanks may extend to the deck plating, provided dry cargo is not handled in that area. Where dry cargo is handled on the deck area above a cargo tank, the cargo tank may not extend to the deck plating unless a continuous, permanent deck sheathing of wood or other suitable material of appropriate thickness and construction is fitted.

Further requirements refer to the location of other spaces relative to the cargo space. Additional requirements are to be noted for fire fighting capabilities, cargo tank construction, cargo tank vent sytems, cargo transfer, ventilation of cargo spaces, and emergency shutdown.

A ship certified to carry noxious liquid substances should be provided with a Cargo Record Book, a Procedure and Arrangements Manual, and a Shipboard Marine Emergency Plan, developed and approved for the ship in accordance with Annex II of MARPOL 73/78.

Dynamic Positioning

Following the IMO Resolution MSC.235 (82), which documents important issues on positioning systems for OSVs, specific class rules were drawn up also for dynamic positioning (DP) systems. A DP system comprises components and systems acting together to achieve sufficiently reliable position keeping capability. The consequence of a loss of position keeping capability determines the system’s necessary reliability. To achieve this philosophy, the requirements are grouped into three equipment classes, and for each equipment class the associated worst case failure should be defined. For simple material and supply goods handling offshore and for towing functions, an OSV may have a lower equipment class notation. For other specialized activities, such as diving support or fire fighting duties, the offshore industry increasingly demands owners to provide more reliable and robust positioning systems of a higher equipment class notation.

PRACTICAL SHIP CONSTRAINTS

Guard Rails

The International Convention on Load Lines, adopted in 1966, requires guard rails or bulwarks to be fitted on exposed decks for crew protection. However, anchor handling and towing operations require the stern area of OSVs to be open, making it impracticle to have permanent fixtures such as guard rails or bulwarks. Acceptable practice has been to provide portable rails at the stern area for crew protection.

Double Bottom and Stern Tubes

According to SOLAS Regulation II-1/12-1, double bottoms are to be fitted as far as practicable. However, OSV designs tend to have streamlined hull forms and a large skeg to improve maneuverability and towing function. Acceptable practice has been to require survivability in the event of assumed flooding of the engine room. A one-compartment damage stability analysis regarding the considered space is normally requested when encountering these configurations.

According to SOLAS Regulation II-1/11.9, stern tubes are to be enclosed in a watertight space of moderate volume. Similar space constraints make application of this regulation difficult on OSVs. As an alternative, the acceptable practice has been to ensure the ship survives in the event of assumed damage to the engine room compartment.

Bridge Visibility

According to SOLAS Regulation V/22, ships are to comply with specific requirements with regard to their navigation bridge deck layout and arrangement. In particular, the ship's sides are to be visible from the bridge wings. Most OSVs do not have bridge wings, but have a set-in wheelhouse configuration. Following the intent of the regulation, the as-designed bridge configuration is acceptable insofar as the field of vision and the arc of visibility is satisfactorily addressed.

Navigational Lights

For ships greater than 50 m in length, collision regulations (COLREGs 1972) require the provision of both a forward and an aft masthead light, with a horizontal distance between them of at least half the ship's length. Currently, most OSVs are in the 60 to 70 m range. For these ships, and those greater in length, this requirement would place the aft mast somewhere on the cargo deck.

Furthermore, the regulations require that stern lights, aft anchor lights, and towing lights be placed as close as practicable to the stern. However, this is impractical because of the unique operational requirement for an unobstructed stern area in offshore environments, as well as for duties such as anchor handling and towing. The usual practice in such cases is for the owner to approach the flag administration for waivers, which are normally granted with certain conditions.

COMPLEMENTARY NUMERICAL SIMULATIONS

The RANSE solver COMET (Star-CD, 2002) simulated ship motions in waves and the corresponding pressure distribution acting on the ship by solving the nonlinear rigid body six degrees of freedom motion equations in the time domain. This solver, by implementing the interface capturing techniques of the volume-of-fluid (VOF) type, is suitable for handling nonlinear hydrodynamic phenomena, such as breaking waves, spray, and air trapping. A two-phase formulation of the governing equations models the two-fluid system (Muzaferija and Peric, 1998). No explicit free surface is defined, and overturning (breaking) waves as well as buoyancy effects of trapped air are accounted for. Solving an additional transport equation for the volume fraction yields the spatial distribution of each of the two fluids (Ferziger and Peric, 1996). The flow around the ship is computed, taking into account viscosity, flow turbulence, and deformation of the free surface. Hydrodynamic and aerodynamic forces and moments acting on the ship are calculated by integrating the pressure and friction stresses over the ship's surface. Nonlinear rigid body motion equations are solved, and subsequent time integration yields accelerations, velocities and displacements (Brunswig and el Moctar, 2004). By updating the position of the ship and again computing the fluid flow for the new position and integrating this procedure over time, the trajectory of the ship is obtained.

We simulated motions of a typical OSV in waves as well as the corresponding wave-induced pressure distribution on the hull. Table 2 lists principal particulars of this ship. A numerical grid comprising about 1.7 million hexahedral control volumes surrounded the ship. Outer grid boundaries were located two ship lengths ahead, five ship lengths aft, two ship lengths to port and starboard of the ship's sides, and one and a half ship lengths beneath and above the free surface. Near the ship's hull and at the inlet boundary of the waves, grid density was high to ensure sufficient resolution of the waves, whereas towards the outlet boundary the grid was stretched to dampen the waves, resulting in a relatively course grid density. Figure 1 shows the numerical grid on the ship's surface, and Fig. 2 presents the numerical grid domains surrounding the ship as seen on a vertical centerline plane of the ship at the beginning of the simulations. A zero-gradient (hydrostatic) pressure boundary defined the wake flow, and a no-slip boundary was specified on the ship's surface. The time step size for all computations was 0.02 s.

Table 2 Principal particulars of subject OSV

Length overall	105.0 m
Length bet. perpendiculars	100.0 m
Draft	6.5 m
Molded breadth	22.0 m
Center of gravity above keel	9.9 m
Metacentric height	0.5 m
Displacement	11 0000 t
Gyradius abt. long. axis	5.0×10^8 kg m²
Gyradius abt. tranv. axis	7.6×10^9 kg m²
Gyradius abt. vert. axis	7.6×10^9 kg m²

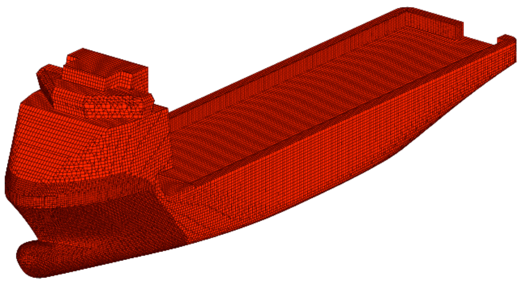

Fig. 1 Numerical grid on surface of the OSV

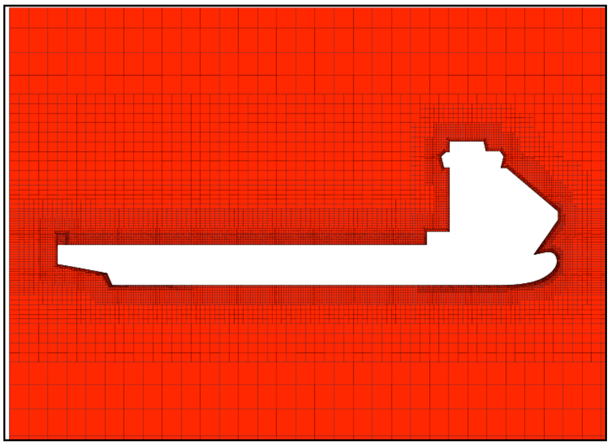

Fig. 2 Numerical grid domains surrounding the OSV

The OSV in Stern Quartering Seas

We investigated the OSV in a natural (irregular) long-crested seaway having a significant wave height of 4.0 m and a mean period of 7.5 s. The quartering seaway approached the ship at a 30 deg angle off the port stern. No current and no wind were assumed acting. The ship was investigated under zero speed conditions to simulate the running of an anchor chain. However, the pull of the anchor chain itself was not accounted for. Although the ship was provided with portable rails at the stern area for crew protection, we left the port side of the aft guard rail open to more realistically simulate the anchor handling operation.

To enable mesh adoption to large amplitude rotational ship motions, we selected the grid morphing technique, treating the ship as a rigid body. Figure 3 shows the undeformed numerical grid domains surrounding the ship as seen on a transverse vertical sectional plane located 90 m from the ship's aft perpendicular. Figure 3 also depicts the corresponding deformed grid after a simulated time of 82 s. Figure 4 shows three selected screen shots of the OSV after simulation times of 35, 55, and 82 s. At 82 s, the ship attained its maximum roll angle of 8 deg. Figure 5 shows the corresponding time histories of computed heave, roll, and pitch motions.

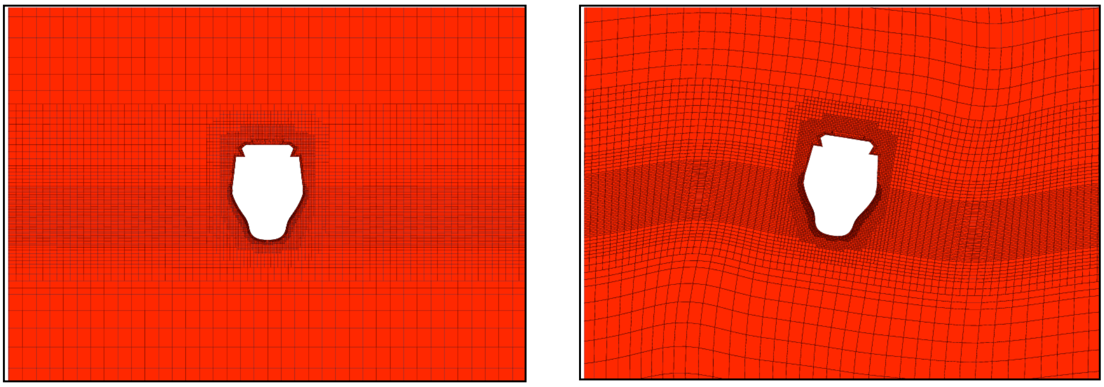

Fig. 3 Undeformed (left) and deformed (right) numerical grid domains

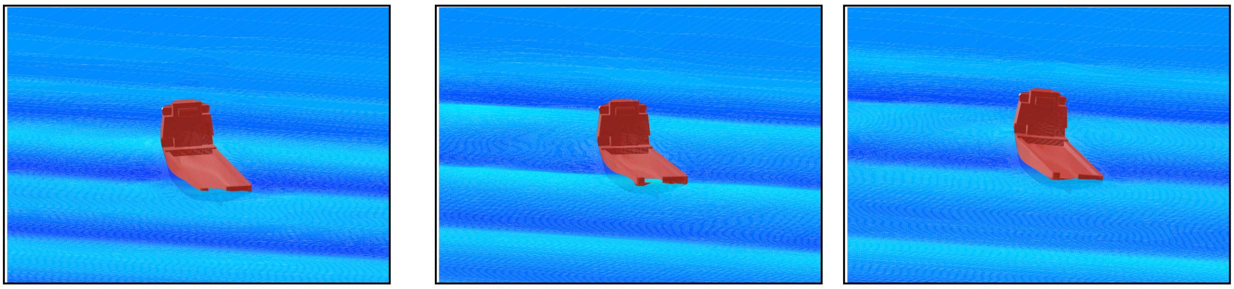

Fig. 4 The OSV in stern quartering seas at times 35 s (left), 55 s (center), and 82 s (right)

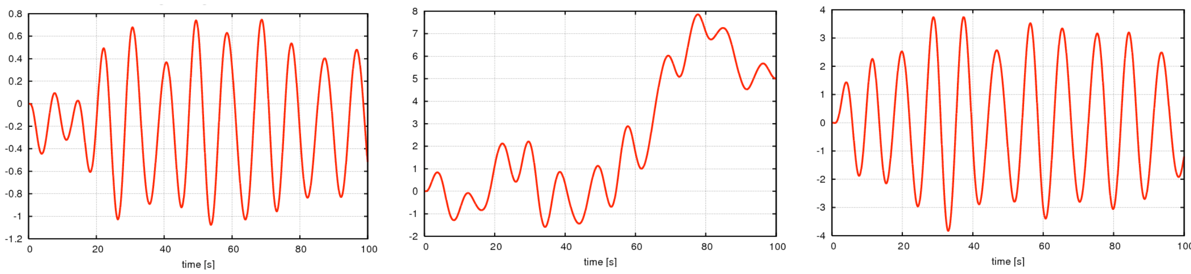

Fig. 5 Time histories of heave [m] (left), roll [deg] (center), and pitch [deg] (right)

The OSV in Head Seas

We investigated the OSV in regular head waves having a height of 6.5 m and a period of 7.5 s. Two constant propulsive forces of 500 kN, acting horizontally at points located 3.0 m ahead of the ship's aft perpendicular, 3.25 m above the ship's baseline, and 5.5 m to port and starboard of the ship's centerline, represented the thrust of the two propellers. Computations were performed under a constant inlet velocity of 2.0 m/s to account for the ship's forward speed. Figure 6 shows a screen shot of the OSV in head seas and Fig. 7 the associated pressure distribution at wave impact, occurring after an elapsed simulation time of 29.5 s.

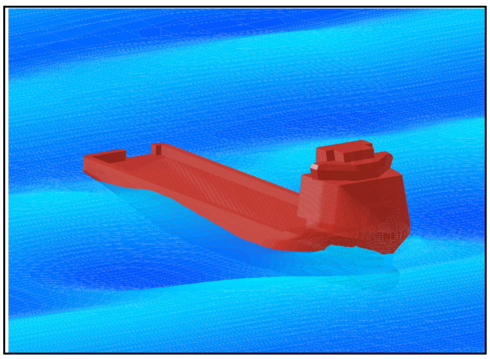

Fig. 6 The OSV in head seas

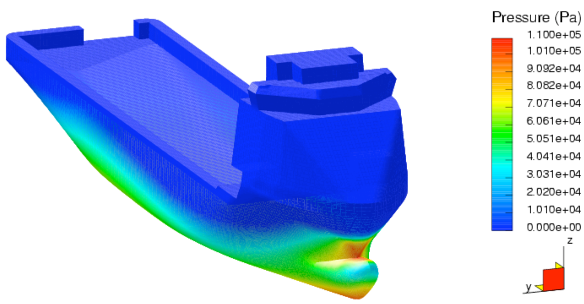

Fig. 7 Pressure distribution at wave impact

Figure 8 shows computed time histories of heave (amidships) and pitch motions and the corresponding pressure acting on the bow at a point located 100.06 m from the ship's aft perpendicular, 0.014 m to port of the ship's centerline, and 6.8 m above the ship's baseline. The functional relationship of the resulting impact-related wave-induced pressure peak of 118 kPa corresponded to the classical slamming pressure. The pressure increased suddenly and then decreased afterwards to about one half of its peak value, finally decreasing further until reaching atmospheric pressure. The comparable rule-based pressure for this ship was 105 kPa (Germanischer Lloyd, 2005).

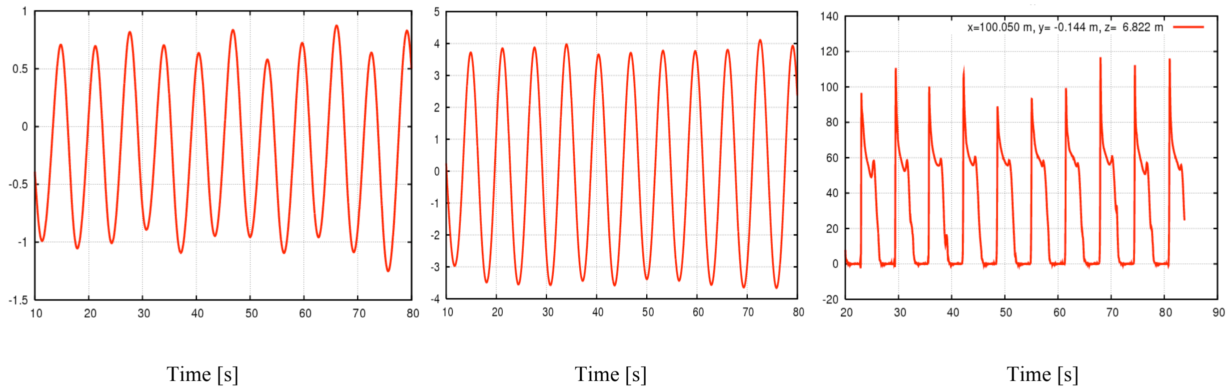

Fig. 8 Time histories of heave [m] (left), pitch [deg] (center), and pressure [kPa] at bow

DISCUSSION AND CONCLUDING REMARKS

In this paper we presented new rules for OSVs. The rules primarily comprise strength related dimensioning requirements and stability requirements. Ships equally intended for additional services and having the associated functional equipment may be assigned an additional notation dedicated to the individual services. Such services include the capability of carrying specialized stores and cargoes to mobile offshore units and other offshore installations, performing offshore anchor handling and towing duties, stimulating wells, fighting fires, performing standby and rescue operations, and storing and transporting recovered oil. However, prescriptive class rules cannot cover all safety related design issues. Complementary computational fluid dynamics (CFD) simulations are suitable measures to assess safety aspects beyond the scope of class rules.

We performed complementary motion simulations of the OSV in stern quartering seas to demonstrate the ability of investigating operations that can adversely affect the ship's controllability and ultimately compromise the survivability of the ship. For the particular condition we analyzed, the ship was not endangered. However, it is conceivable that other wave conditions, not necessarily more severe but propagating from another direction, may lead to water trapped on deck, especially since stern area must be left open for handling an anchor. Such a situation may lead to loss of stability, causing the ship to capsize. Therefore, it is essential that freeing ports be arranged to ensure the most effective drainage of water trapped in recesses at the aft end of the forecastle.

The RANSE solver we used was extensively validated by comparing computed results against experimental measurements obtained from systematic model tests performed at the Hamburg Ship Model Basin (el Moctar et al., 2006). These tests were conducted for a 100 m motor yacht of 3600 t displacement under conditions where the ship experienced bow flare slamming. Figure 9 shows sample time histories of measured pitch motion and vertical acceleration at the forward perpendicular (FP) together with comparable results obtained from RANSE computations. Motions as well as vertical accelerations were accurately predicted.

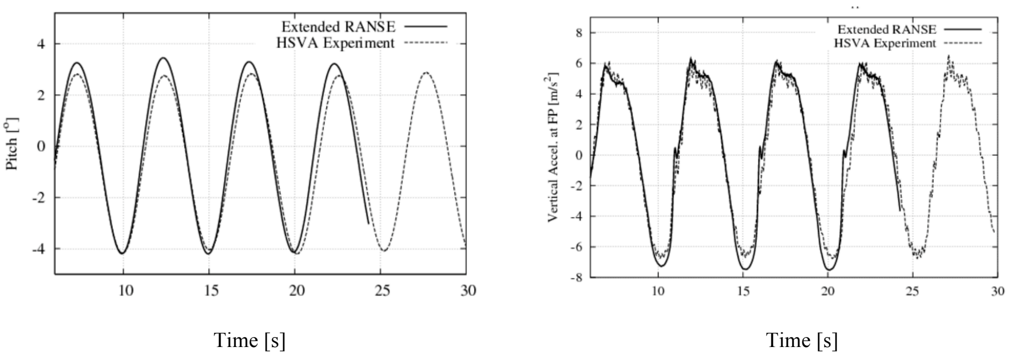

Fig. 9 Comparative time histories of pitch motion (left) and vertical acceleration at FP (right) for a motor yacht in head waves (el Moctar et al., 2006)

Accurate prediction of wave-induced loads continues to be difficult, mainly because of the many parameters involved. Therefore, it was essential to rely on a validated procedure to compute these pressures, especially with the goal in mind of obtaining reliable predictions needed for systematic rule development. The technique we used was extensively validated against seakeeping model test measurements of hydrodynamic loads acting on a forebody hull segment of the bow region of two OSV hulls (Schellin and el Moctar, 2007). This hull segment, separated from the wooden model, was located above the stillwater line and connected to the ship model by a special force balance that enabled measuring six-degree-of-freedom forces and moments acting on this segment as well as on smaller plate fields of this segment. Measured global loads as well as local pressures compared favorably against computed values, as seen by representative sample results shown in Fig. 10. Results in this figure are presented as nondimensional values. Vertical force (F_V) was normalized by a maximum value of F_0 = 2350 kN; pressure (P_{sl}), by a maximum value pf P_0 = 265 kPa; and time (Time), by the wave encounter period of T_0 = 6.0 s.

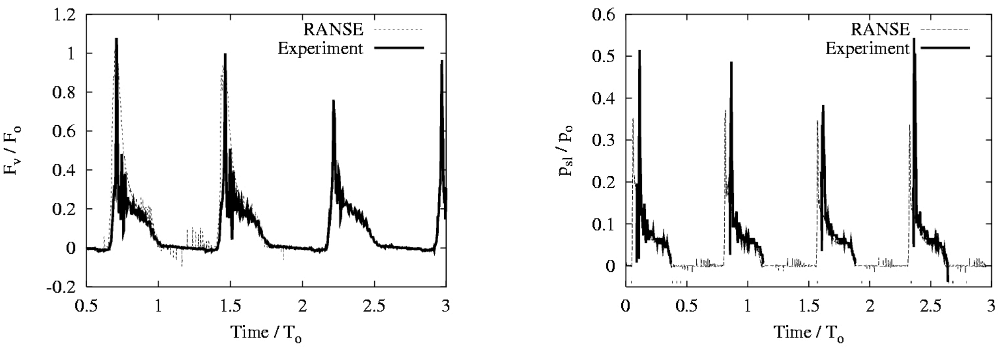

Fig. 10 Time histories of measured and computed vertical forces on bow section (left) and pressures on a small plate field of the bow section of Hull 1 (Schellin and el Moctar, 2007)

Model test measurements as well as computations did not account for the influence of structural deformation although this deformation would have affected the wave-induced slamming loads. Including this effect most likely would have led to somewhat smaller loads. The computed impact pressure peak of 118 kPa, acting on the bow of the subject OSV, should be treated as an equivalent static design pressure. This pressure peak only slightly exceeded the comparable rule-based design pressure of 105 kPa.

ACKNOWLEDGEMENTS

The authors thank A. Köhlmoos for generating the numerical RANS grids and J. Oberhagemann for his assistance in performing the RANS computations.

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KEYNOTE PAPER 2

Developments in Ship Propulsion – Efficiency, Fuel Consumption and Environment

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ABSTRACT

The environmental factor, or the "green factor" has during the very recent years become astronger driver for technology development and design of ships and ships' equipment.
What we have seen so far, is expected to be only the beginning of what will be one of the most critical factors in the years to come.

NOx and SOx regulations already have come into force and will for sure be supplemented by stricter local and international regulations, and IMO rules for ballast water treatment will be ratified within a short time.
Further, global or regional regulations on green house gas (GHG) emissions from ships are claimed to be included in the agreements to be made in the Copenhagen meeting at the end of this year; after being quietly exempted from the target CO2 reductions in the Kyoto agreement.
Being a supplier of electric propulsion system, our solutions are directly influencing the environental footprint of the vessel, in particular to greenhouse gas emissions, as well as NOx and SOx. For a range of vessels, the electric propulsion itself is contributing to significant reduction in the fuel consumption, and hence also the emissions. For OSV vessels, up to 40-50% reduction is achievable in for example DP operations. This also reduces the fuel costs accordingly, and electric propulsion has found its use in not only OSVs, but also in a range of ships where the savings in fuel costs justify the initial investment.

However, there are always room for improvements, and with higher costs of fuel or introduction of taxes on emission, even small efficiency improvements may give large benefits for ship owners and charterers.

This paper elaborates on the issues of environmental footprint of OSVs, and recent developments that will lead to both reduction in emissions and lower operational costs.

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Optimising Propulsion Systems for AHTS Vessels

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ABSTRACT

The paper presents the design approach currently used by MAN Diesel for optimising AHTS performance. It is based on the extensive experience gathered over more than 30 years of supplying propulsion systems for the offshore industry.

The optimisation of the design at various stages of a project is outlined together with the applied tools like CFD and FEM as well as the inclusion of model and full scale test experience.

The newly developed AHT (Alpha High Thrust) propeller nozzle that was presented at the OSV conference in 2005 has since grown into a series of nozzles that can be used for different applications. The requirements and experience using this new nozzle type will be explained, focusing on the hydrodynamics, cavitation and structural aspects.

A series of 120 ton bollard pull AHTS have since been commissioned and will be used as an example to illustrate how the bollard pull can be maximised by following a holistic approach in the design of hull, propeller, and a high efficiency AHT nozzle.

The latest results from an extensive cavitation test series of the AHT nozzle family will be discussed with special emphasis on the influence of cavitation on the performance of nozzles in general and the AHT nozzle in particular.

To ensure that the performance of the final manufactured nozzles are as predicted by calculations a set of quality standards have been introduced. The maximum allowable manufacturing tolerances have been set using CFD calculations of geometries diverting from the theoretical one.

INTRODUCTION

The design of a propulsion system for an AHTS is a challenging task involving not only the physically products like engine, gearbox, propellers and control system but also the interfaces between these components as well as their influence on the vessel's performance. One significant example in this respect is the interaction of the propeller and nozzle with the hull.

Most AHTS's are highly powered and designed as twin screw vessels with ducted CP propellers in order to achieve the required BP and a high manoeuvrability. The other operating conditions seldom play a role in specifying the main engine power. However, the BP is not solely determined by the installed power but also by an optimised propulsion system and hull lines. An optimum solution is characterised by a design where all three items have been addressed.

At MAN Diesel the design of the propulsion system is generally carried out in two phases and supported with different optimization tools and computer codes.

PRE-ORDER STAGE

Hydrodynamic aspects

One of the first questions raised, when starting the design of an AHTS, is how much power is needed to reach a certain specified bollard pull. For years it has been a common practise to use simple rules that would link the bollard pull to the installed power. One rule simply states that each HP will yield 13.6 kg [1].

MAN Diesel developed a more refined method where the bollard pull is determined from the power density i.e. based on both power and propeller diameter [2].

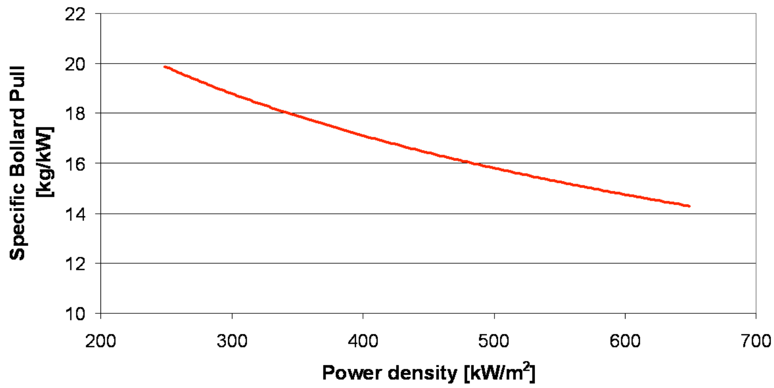

Fig 1: Specific bollard pull versus power density

That power cannot be used as a sole parameter to determine the achievable bollard pull can be demonstrated by comparing three different MAN Diesel propulsion configurations which will all lead to a 90 ton bollard pull.

Engine		Propeller		Power	Specific
Type	Power	Speed	Diameter	Density	Bollard Pull
[-]	[-]	[rpm]	[mm]	[kW/m²]	kg/HP
7L27/38	2380	150	3300	278	13.9
8L27/38	2720	206	2750	458	12.2
9L27/38	3060	276	2400	676	10.8

Table 1: Different propulsion configuration giving 90 ton bollard pull for a twin screw AHTS

Had the simplified ruled (13.6 kg/HP) been applied an underestimation of 10% and 21% would have occurred in the case of the 8 and 9L27/38 propulsion systems.

A further refinement has since been added to account for the nozzle type, Length/Diameter ratio, support type and the influence of cavitation on performance. A more precise determination of the bollard pull is thus possible in the project stage.

An accurate determination of the bollard pull is important as a possible bollard pull guarantee will have to be based on the available figures at this stage.

Structural aspects

The optimum design of the propeller/nozzle arrangements is primarily determined by the requirement of having an optimum hydrodynamic efficient solution and sound structural construction. The latter requirement secures that harmful vibrations and possible structural failures are eliminated.

It is MAN Diesel intention to be a part of the very early design stage where all the important decisions related to the nozzle design are being made [2]. This will make it easier to reach the optimum solution for the propeller and nozzle arrangement in the post-order phase. A basis for a sound design is that lines plan and hull structure drawings are forwarded for evaluation.

In order to reach an optimum solution MAN Diesel has introduced a set of guide rules (Data Request for Nozzle Design) that can assist the hull designer in the structural design of the aft ship.

To optimize the flow to and around the propeller the guide lines specify design parameters which make the nozzle design more efficient and less costly.

The following design parameters should be observed at this stage of the project

Vessel type and operation mode: The vessel type and how the vessel is intended to be operated is essential for the propeller blade and the nozzle design including the interaction in-between the two.

Nozzle type and support: The profile type and the connection to the hull are decided from the operating profile of the vessel, bollard pull requirement, structural possibilities inside the hull and hydrodynamic aspects.

To avoid vibration problems MAN Diesel recommends that the natural frequency of the nozzles should be minimum 20% above or below the 1st order natural frequency of the propeller blades. The stiffness of the nozzle profile itself, the connection type to the hull and the aft ship stiffness forms the basis for this evaluation. A sound design is characterised by having a well distribution of forces and by avoiding stress raisers. The design of the top strut and headbox is a special challenge in this respect. However, the structural aspects must always be balanced by the hydrodynamic requirements.

Figure 2: Strut and headbox support

POST-ORDER STAGE

The detailed design usually takes place after signing the contract when more information is available on the hull lines, engine, gearbox and shaft arrangement.

The items that are usually addressed are:

Aft ship hull form design. The achievable bollard pull depends on the aft ship lines and the propeller and shaft arrangement. In general the water flow around the hull will follow the buttock lines. This means the slope of the buttock lines is of great importance as it will influence the thrust deduction factor.

$$t = 1 - \frac{T_{BP}}{T_{P,B} + T_{N,B}}$$

From the formula it can be seen that the propeller and nozzle thrust in behind condition $T_{P,B}$ and $T_{N,B}$ is reduced by the thrust deduction factor t – leading to a corresponding reduction in the bollard pull. This reduction is mainly caused by the suction of the propeller and nozzle on the adjacent hull surfaces. For that reason the distance from where the shaft protrudes from the hull to the centre of the propeller should be as long as possible.

It is MAN Diesel's recommendation to design slowly raising buttock hull lines of approximately 17-19 degrees. The overall aim is to keep the thrust deduction factor to a minimum.

Furthermore, it must be secured that sufficient water will be present above the propeller/nozzle in order to prevent air suction.

Propeller blade design. The detailed design of the propeller blades will be based on the different operating conditions and the results from the model tests (resistance, self propulsion with stock propeller, wake measurements). The blades will be optimised for the bollard pull condition and checked for different other operating modes (free sailing, towing etc) to ensure that an overall optimum design has been reached.

The final design will be based on a balance between the two major design objectives – efficiency and cavitation/vibration. The detailed design of the propeller and nozzle is made in close cooperation between the hydrodynamic and structural engineer.

For AHTS the shape of the blades will exhibit wide chords at the tip (Kaplan shape) to maximise the bollard pull.

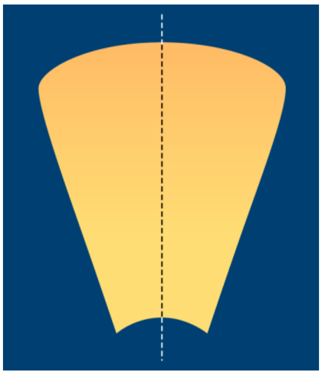

Fig 3: Kaplan blade shape

Nozzle design. The type of nozzle has already been selected in the pre-order phase and the detailed design of the nozzle will focus on the support and hull attachments to minimise the thrust deduction caused by the interaction effects with the hull. Compared to the conventional nozzle types the AHT nozzle will deliver more thrust thus making the design details of the support more important in order to minimise the thrust deduction factor.

Consequently, only a plant specific designed propeller and nozzle including well faired and structurally sound supports will result in an optimum solution. This means that the propeller and nozzle supplier needs to be a part of the very early design stage as already underlined in reference [2].

To verify the potential of the different design alternatives MAN Diesel recommends to make model test of the final designed propeller and nozzle, including test of tilt and azimuth angles of nozzle as well as propeller direction of rotation.

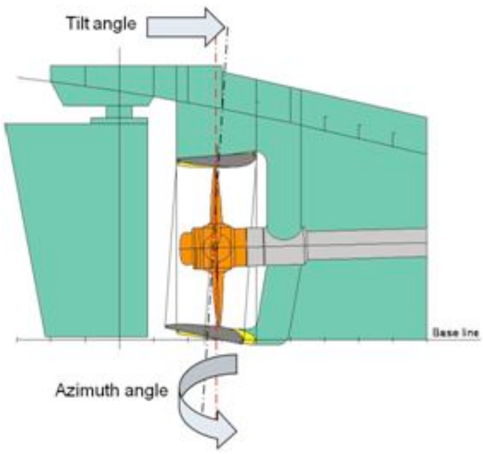

Fig 4: Definition of tilt and azimuth angles.

The possible improvement that can be achieved by following this systematic approach will be exemplified by the following case study.

However, it is important to note that the more aligning requirements that are proposed for the nozzle, the more cumbersome the installation will be. In each case, the gain obtained in bollard pull by introducing an additional nozzle alignment requirement should be carefully judged against the risk of possible misalignment during installation.

In any case MAN Diesel recommends choosing the same supplier for the propeller and the nozzle to optimise the overall performance.

LATEST NOZZLE DEVELOPMENT

Since the introduction of the AHT nozzle in 2005 [2] its range has been extended to include

- Longer and shorter nozzles than the original L/D=0.5 making it possible to select the most optimum size depending on cavitation number and propeller load.
- A simplified and more production version with a strait inner area at the propeller zone.

The nozzle family was developed using CFD calculations on a large number of systematically varied nozzle shapes and with the bollard pull conditions as the prime optimisation objective.

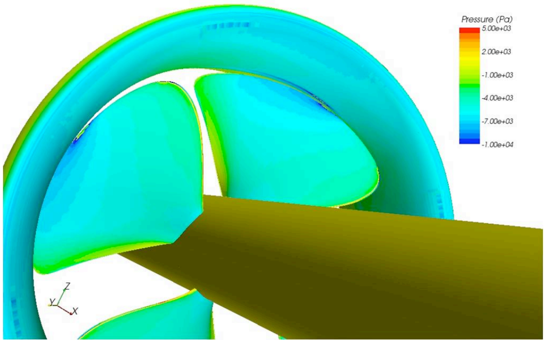

Fig 5: CFD pressure calculation of nozzle and propeller

A major research program was recently undertaken by MAN Diesel to investigate the performance of ducted propellers including the influence of cavitation. Different types of AHT nozzles and the well known 19A nozzles were tested at SVA Potsdam as well as in the Free Surface cavitation tunnel at the University of Berlin.

Most propellers – being open or ducted – are designed with a certain amount of cavitation and if kept within limits the cavitation will only affect the performance marginally. However, this is not true for highly loaded ducted propellers where the present of cavitation reduces especially the nozzle thrust.

One aspect that became clear was the importance of minimising the tip clearance because the tip vortex would disturb the flow at the exit of the nozzle. However, for practical reason a certain clearance is necessary to facilitate the dismantling of the blades inside the nozzle.

An extensive test series was carried out in both non- and cavitating conditions for the AHT series of nozzle as well as the 19A version. The results can be summarised as:

- The AHT nozzles showed superior performance compared to the 19A
- The shorter nozzles are more affected by cavitation than the longer versions
- Air suction from the water surface into the propeller/nozzle reduces the bollard pull significantly. The risk increases with diminishing water height above the propeller and increasing L/D ratios

The backing performance of the different nozzles also formed a part of the investigation and clearly showed the superiority of the new AHT nozzle family. A 20-25% improvement of the astern thrust was measured compared to the 19A type.

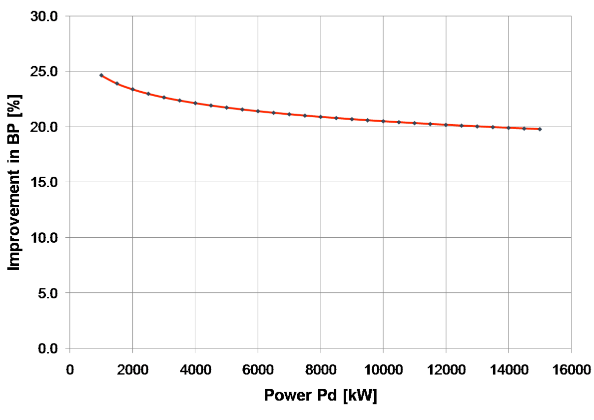

Fig 6: Comparison of astern bollard pull, AHT versus 19A both with L/D=0.5

All the results and findings have since been included in the design procedures at MAN Diesel.

MANUFACTURING STANDARDS

Since MAN Diesel developed its new AHT nozzle series in 2005 it became apparent that the existing manufacturing procedures and tolerances were inadequate to secure the necessary quality and maintain the predicted hydrodynamic performance. One reason is the more complex manufacturing of the double curved surfaces of the AHT nozzles.

As a consequence MAN Diesel decided to develop its own manufacturing standards with respect to tolerances, welding procedures and quality checks.

One of the reasons for the good performance of the AHT nozzle profile is the low and uniform pressure distribution at the inlet of the nozzle and the geometry must therefore be carefully controlled in this area.

As a support to find the permissible deviations from the theoretical nozzle profile, CFD calculations were carried out on different geometries.

Fig 7: An AHT Ø4030 nozzle ready for dispatch. Leading edge of the nozzle is on the floor.

To ensure the overall geometry will be within the specified tolerances, the manufacturing precision of internal stiffeners forming the nozzle profile plays an important role.

Fig 8: An AHT nozzle (ø4030) during assembly.

MAN Diesel developed a standard for the joint connections to ensure that the basis for controlling the outer geometry of the nozzle can be met.

To avoid mechanical failures after the installation and during operation, welding procedures for the main structure of the nozzle were introduced. Based on the experience from Finite Element calculations, welding procedures for the nozzle supports have been worked out including Non Destructive Testing requirements.

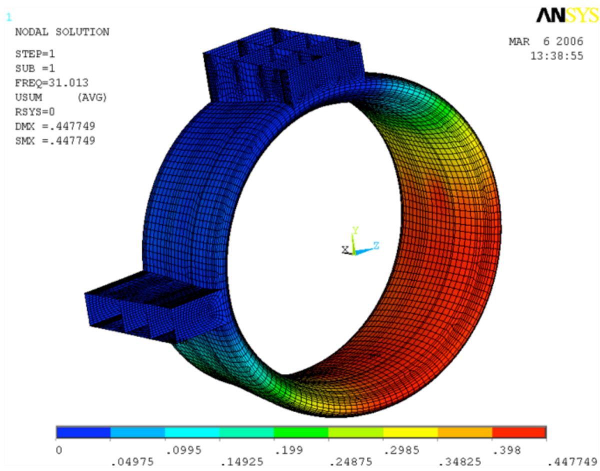

Fig 9: Finite Element vibration analysis of nozzle including supports

These requirements must be documented by the nozzle supplier in addition to the rules of the classification society. The requirements are supported by a set of guide lines that the supplier must adhere to.

One essential dimension is the inner diameter of the nozzle which must be controlled to keep the predicted performance.

The final tip clearance after the installation has been completed depends on the tolerances of the outer diameter of the propeller, the inner diameter of the nozzle and the misalignment of the nozzle itself. Furthermore, the outer diameter of a CP propeller increases as the pitch is changed from the design pitch setting towards zero pitch.

In order to avoid any unpleasant surprises, when installing the propeller into the nozzle, MAN Diesel has put a stricter requirement on the outer diameter of the propeller blades than specified in the ISO 484 manufacturing standard. A strict requirement is also put on the inner diameter of the nozzle by allowing only a plus tolerance. The objective of controlling the tolerances is not only to secure the predicted performance like bollard pull but also to facilitate the installation of the nozzle at the shipyard.

CASE STUDY

The case study concerns a series of AHTS vessels designed to deliver a bollard pull of 120 tons with a MAN Diesel propulsion system.

Main engine:	9L27/38, 3285 kW at 800 rpm
Reduction gear:	AMG55EV, with integrated servo oil system
CP propeller:	VBS980, dia. Ø 3800 mm with AHT nozzle
Controls:	Alphatronic 2000 PCS

Fig 10: Propulsion plant configuration of a 120 ton bollard pull AHTS

The design of the propulsion system followed the procedure as outlined in the previous sections of this paper.

The initial hull lines developed by the naval architect displayed steep buttock lines of approx. 25° exceeding the recommended 17-19 °. The buttock lines were later reduced to 23° by lowering the gearbox followed by a redesign of the aft ship. In addition the distance between the propeller and where the shaft protrudes from the hull is short. Because of these unfavourable conditions the thrust deduction factor ended up being 9.6%.

Fig 11: Buttock lines and propeller location.

A comprehensive model testing program was set up to investigate the possible improvements from not only using the new AHT nozzle type but also including other relevant installation aspects.

Apart from the normal testing with stock propeller the following were added

- Nozzle supports comprising both a headbox and a strut solution
- Propeller direction of rotation
- Nozzle types – AHT and 19A
- Tilting and azimuthing of nozzles

The model testing program was planned in the sequence as described above and lead to an increasing improvement of the bollard pull as the testing proceeded.

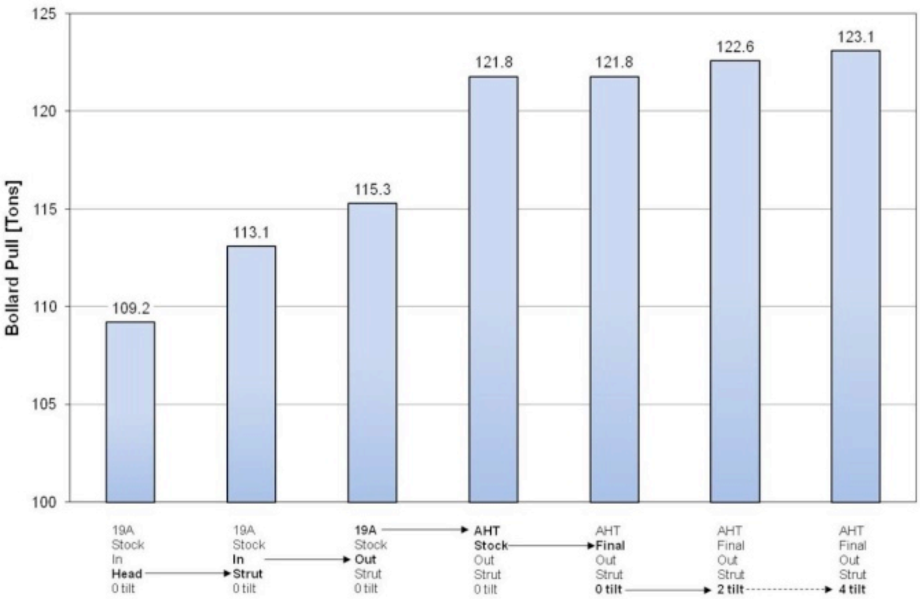

Fig 12: Stepwise improvement in bollard pull for a 120 ton AHTS

Especially the testing with the AHTS nozzle showed a pronounced improvement in bollard pull.

Varying the azimuth angle of the nozzle only resulted in a marginally improvement and was for this reason not applied.

Compared to a standard solution a 13% improvement in bollard was achieved by following this systematic approach!

The full scale testing was conducted as the vessels were commissioned and at the time being 5 vessels had their bollard pull measured.

Vessel No	Measured BP [tons]
1	124.0
2	121.7
3	122.2
4	121.2
5	122.5

Table 2: Full scale measurements of a 120 ton BP AHTS

The full scale figures are as measured and not corrected for the unfavourable conditions at the test site (limited water depth and current across tow line) as required in [5].

This type of vessel falls into the standard 120 ton category of AHTS's which up to now have been characterised by having two 8 cylinder 32 cm bore main engines with a rated power of 4000 kW. Compared to this industry standard the MAN Diesel optimised propulsion solution can suffice with only 2x3285 kW to reach the required bollard pull.

CONCLUSION

The bollard pull of an AHTS depends not only on the power transmitted to the propellers but also on the propeller diameter, nozzle design and their interaction with each other and the hull.

By following a systematic approach and pay attention to details when designing not only the propulsion system but also the hull, it is possible to improve the bollard pull significantly compared to a standard solution. The case study shows a 13% increase in bollard by following this concept which furthermore has been verified by the full scale results.

However, a prerequisite for reaching an optimum solution is a close and open minded cooperation between owner, shipyard, consultant and the supplier because most design proposals are not limited to the propulsion system but affects the overall design of the vessel.

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Offshore Support Vessel Sector Adds New Category - Wind Installation Vessels

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Michael A. Sano – ABS Senior Engineer, Corporate Energy Project Development – Houston, Texas

Abstract

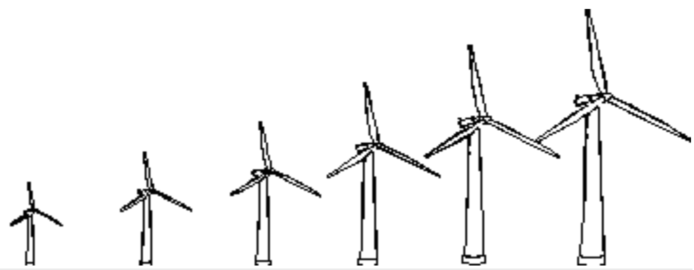
Windpower – Overview of Wind Industry

The global demand for clean, renewable electric power is driving the relatively young wind industry to expand capacity. It is universally understood that energy needed to satisfy current and future world demand must come from many different sources, in addition to the traditional energy resources of oil, gas, coal and nuclear. Wind is one of these optional sources. A 19 May 2009 article in the Wall Street Journal titled ‘Environmental Capital’ stated that, in spite of the current economic climate and world wide recession, the wind industry will continue to grow at more than 20% annually for the next five years. The same article reported that today worldwide wind energy production is 120 gigawatts but is expected to reach 332 gigwatts by 2013. If achieved, this will represent a 276% increase in world wide wind generating capacity over the next five years.

One example of large gains is in China, which has doubled the installed national wind power in each of the previous four years. In addition, Chinese officials with the National Energy Administration have made it known that the new target for China’s wind energy is 100 gigawatts by 2020 (Source: China State Press). The U.S. has lagged behind in fully developing wind energy but is expected to also double total installation to reach approximately 55 gigawatts capacity in one to two years.

Wind farms generate electric power normally in the hundreds of volts range depending on the arrangement of the wind farm, number and size of units installed, etc. power is stepped up to higher voltages (normally in the thousands of volts range) using transformers for more efficient delivery over transmission lines. The power is subsequently stepped down again, to lower voltages for use by consumers.

Wind turbines vary in size. The chart below depicts a variety of historical turbine sizes and the amount of electricity they are each capable of generating (the turbine's capacity, or power rating).



	1981	1985	1990	1996	1999	2000
Rotor (meters)	10	17	27	40	50	71
Rating (KW)	25	100	225	550	750	1,650
Annual MWh	45	220	550	1,480	2,200	5,600

Shore-based

The following are some rough guidelines on the physical dimensions of wind turbine components:

Towers - Tower sections for the common 80 meter (250-foot) wind turbine tower in the United States can weigh more than 150,000 lbs (70 tons), be 36 meters (120 feet) long and have a diameter of 4.5 meters (15 feet). The next generation of 105 meter (330-foot) towers will be 5.4 meters (18 feet) in diameter at the base.

***Nacelles* –** Nacelles, which house the gear box and generator, commonly weigh 50-70 tons and can weigh up to 90 tons or more.

***Blades* -** On commercial scale projects, blades run from approximately 33 meters (110 feet) to 44 meters (145 feet). Blade lengths will continue to grow in the future, particularly for offshore wind projects. The largest blades are just over 60 meters-plus (200 feet) long for a 5-MW turbine.

Currently, eighty countries around the globe have installed shore-based wind farms of varying capacity. The top four countries currently ranked as having the highest installed capacity by year end 2008 per World Wind Energy Report 2008 are the United States, Germany, Spain and China, respectively.

Offshore Wind

The industry is moving from shore-based to offshore installations to take advantage of stronger and more predictable winds. Because of the robust conditions for harnessing wind energy offshore, larger turbines can be installed for the purpose of capturing more available wind energy. For ABS, working with offshore wind installations is a natural outgrowth of its marine and offshore classification business. The installation of wind farms offshore, initially in shallow waters and progressively further offshore into deeper waters, has created a new category of offshore support vessels (OSVs) called Wind Installation Vessels. These vessels will be involved in the installation, maintenance and repair of wind turbine units.

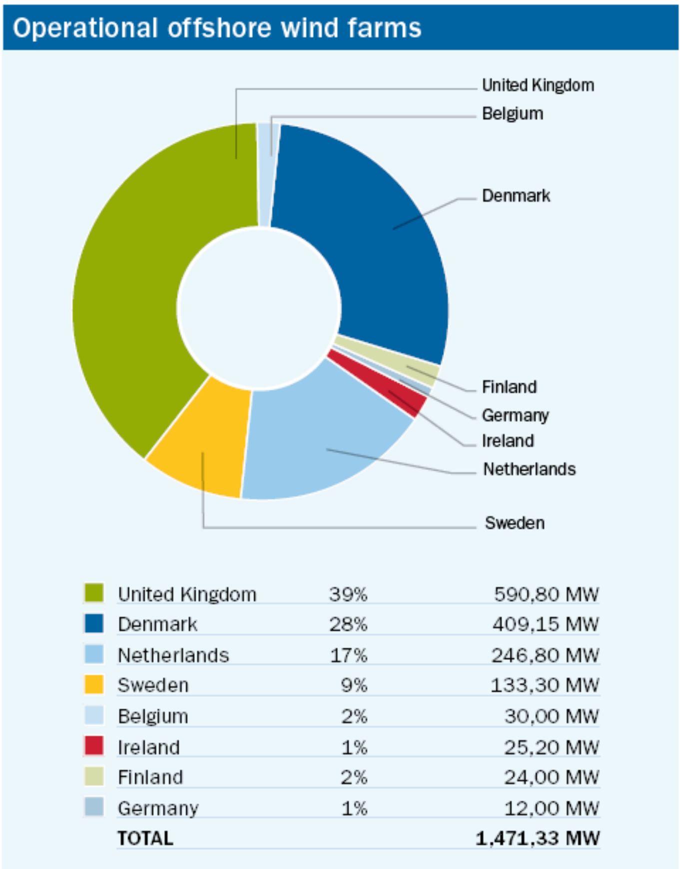
Eight countries have wind turbines installed offshore providing electricity. They are: Denmark, Belgium, Sweden, Finland, Germany, United Kingdom, Netherlands and Ireland. Additional countries with planned offshore wind energy projects by 2015 include France, Italy, Norway, Poland and Spain. The U.S. currently has no existing offshore wind energy projects built but there are a number of prospective projects moving through the approval and development process. In May 2008, the U.S. Department of Energy's report on a 20% Wind Energy Scenario found offshore wind capacity could achieve a capacity of 54 gigawatts.

Offshore energy has several advantages over shore-based wind farms. Offshore wind power generating units can be larger sea transport make the carriage of larger units and components more feasible than road or rail transport of land-based units. Offshore wind turbines can generate more power than shore-based wind units due to the fact that offshore wind speeds are generally higher and the velocity is steadier. For example, European offshore wind farms have been found to generate electricity between 70 and 90% of the time they are operating. Wind tends to be less turbulent offshore, extending turbine life. The ability to locate offshore wind farms closer to demand centers makes for more efficient transmission of power due to the shorter distances required. And offshore wind farms can generate power during periods of high demand taking advantage of the 'sea breeze effect.' [Source – American Wind Energy Association Wind Fact Sheet. **www.awea.org**]

:

The below chart depicts the distribution of offshore wind farms by country. The capacity is shown next to each country listed. The top three offshore wind energy producers are the United Kingdom, followed by Denmark and the Netherlands. While the United States has several projects that are in the planning stages, none to date have come to fruition.

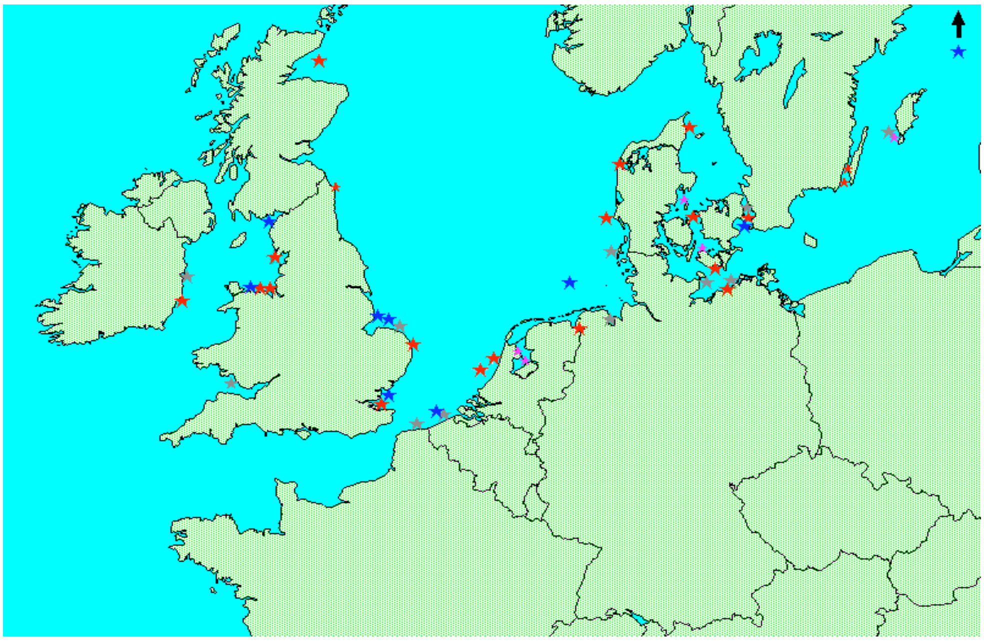
Operational Offshore Wind Farms - distribution by Country



Source – The European Wind Energy Association – Offshore Wind Energy Fact Sheet **www.ewea.org**

Existing Wind Projects

Europe has by far the highest concentration of wind farms as shown by the map below. As other locations and nations around the world construct additional sites, additional vessels will be required to install, maintain and repair the offshore wind turbines.



Wind Farms North-West Europe

Red	Constructed Large Windturbines	Purple	Constructed small Windturbines
Blue	Under construction	Grey	Planned

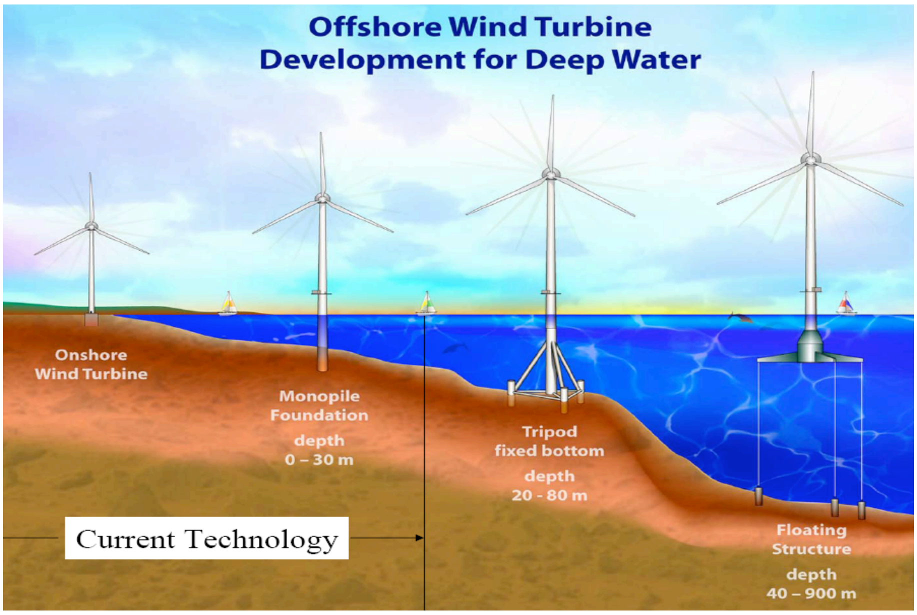
Source: Offshore Wind Energy Europe [www.offshorewindenergy.org]

Types of Offshore Wind Units

For shallow water, up to 30 meters water depth, various installation options exist. A common method is to construct a large round cement foundation into which the base of the mast is inserted once in position. The wind units are then built up in sections. Power connections are made via installed sub-sea wires.

Deep water units, expected to be located in 40 to 300 meter water depth, would be anchored or tethered to the ocean floor.

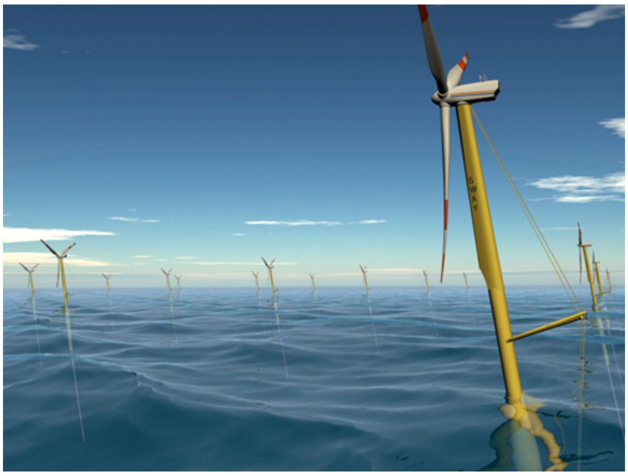
The below illustration depicts current and predicted future trends in offshore wind unit foundations increased water depths.



Deepwater Installations

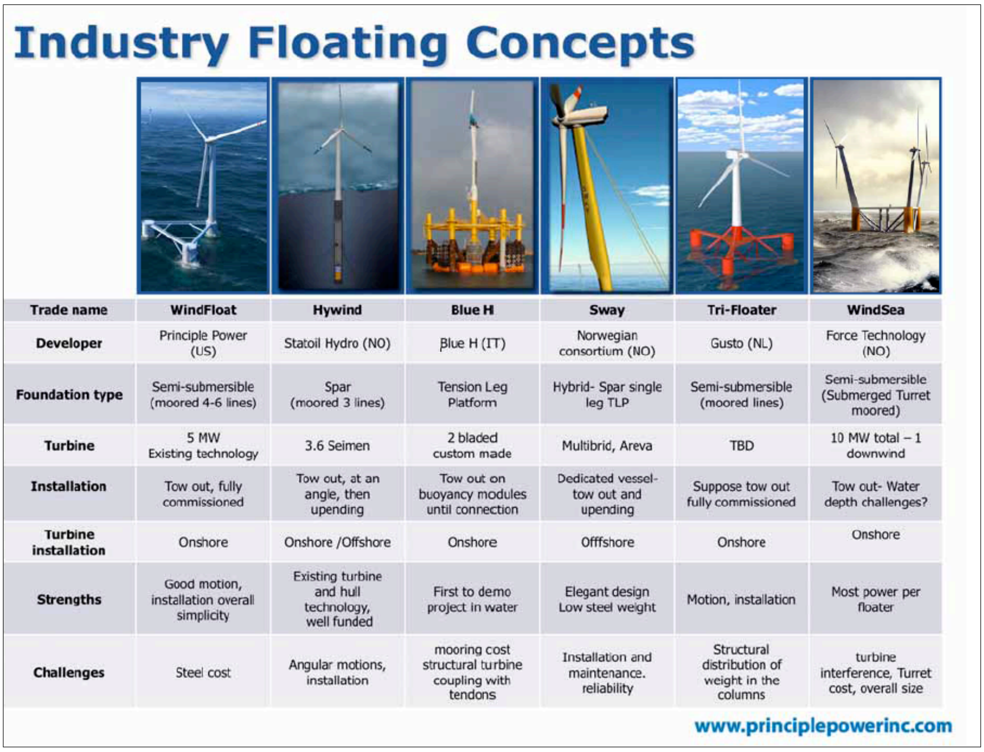
Several designers are now offering deepwater wind unit concepts including WindFloat, Hywind and WindSea among others. An example of how these designers approach the challenge is provided by SWAY A/S of Norway which has developed the SWAY concept for floating wind units. These units differ from traditional land-based or shallow water units in that the turbine is driven by wind from the opposite end of the casing or downwind rotor side. The design calls for them to be constructed as a floating cylinder which is then towed out to the construction site. Upon arrival, the unit is partially ballasted, forcing it to take a vertical orientation. Struts and stays are installed to stiffen the mast. After further ballasting, the power head and blading are installed in one step. The concept calls for both solid material and water ballast to be utilized in this process. The ballast method is also used to lower the suction pile into the seabed. When complete, the unit takes on an equilibrium tilt angle of 5 to 10 degrees from vertical due to the pressure of wind thrust on the rotor. It should also be noted that the entire unit, including the mast, turns from the base on a joint bearing keeping the blading always oriented in the downwind position. These deepwater units are planned to be of 5 MW capacity each.

Envisioned deepwater field of installed SWAY type wind power units.



Source: SWAY A/S Norway

Other examples below of various wind industry floating concepts:



Industry Floating Concepts

Trade name	WindFloat	Hywind	Blue H	Sway	Tri-Floater	WindSea
Developer	Principle Power (US)	Statoil Hydro (NO)	Blue H (IT)	Norwegian consortium (NO)	Gusto (NL)	Force Technology (NO)
Foundation type	Semi-submersible (moored 4-6 lines)	Spar (moored 3 lines)	Tension Leg Platform	Hybrid- Spar single leg TLP	Semi-submersible (moored lines)	Semi-submersible (Submerged Turret moored)
Turbine	5 MW Existing technology	3.6 Seimen	2 bladed custom made	Multibrid, Areva	TBD	10 MW total – 1 downwind
Installation	Tow out, fully commissioned	Tow out, at an angle, then upending	Tow out on buoyancy modules until connection	Dedicated vessel- tow out and upending	Suppose tow out fully commissioned	Tow out- Water depth challenges?
Turbine installation	Onshore	Onshore /Offshore	Onshore	Offfshore	Onshore	Onshore
Strengths	Good motion, installation overall simplicity	Existing turbine and hull technology, well funded	First to demo project in water	Elegant design Low steel weight	Motion, installation	Most power per floater
Challenges	Steel cost	Angular motions, installation	mooring cost structural turbine coupling with tendons	Installation and maintenance. reliability	Structural distribution of weight in the columns	turbine interference, Turret cost, overall size

www.principlepowerinc.com

Source: Principle Power Inc. [www.principlepowerinc.com]

Wind Installation Vessels

In 2003, the first purpose built offshore wind installation vessel was put into service. Since then, the vessel has been purchased by Vroon in the Netherlands which has announced the construction of two more wind turbine installation vessels. Like the growing sophistication we see in the traditional OSV market, wind installation vessels are also quickly gaining in sophistication. Most notably, the cranes are becoming larger, capable of lifting up to 1,000 tons or more, to accommodate increased wind turbine mast and component size. Also they feature larger accommodation facilities, heli-decks and enhanced jacking and station keeping capabilities (DP2 being the industry minimum standard).

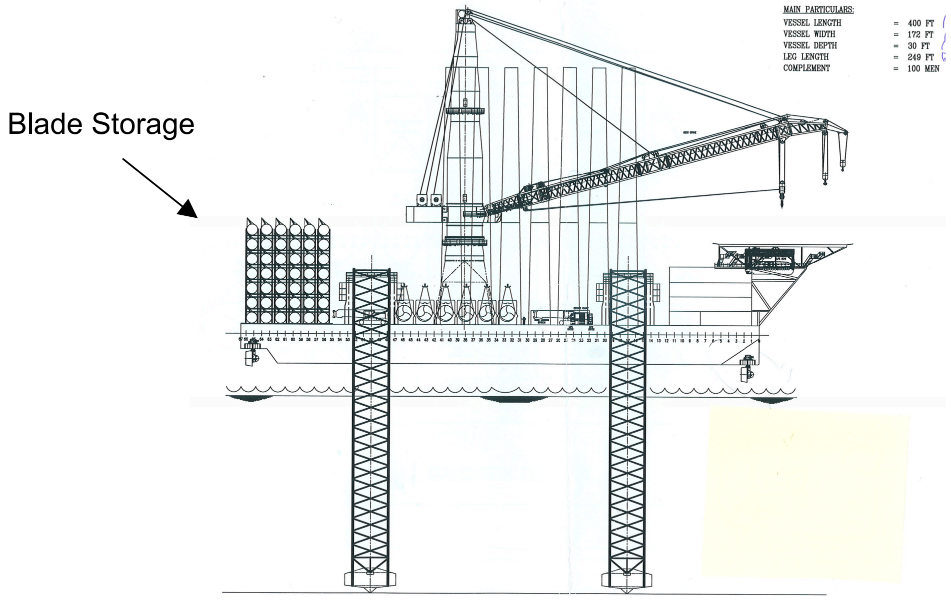
An example of the growing sophistication of these vessels came with the recent delivery by Drydocks World of the OSA Goliath, one of the world's largest multi-purpose offshore construction vessels, which can also support offshore wind installations. This ABS classed vessel is 180 meters LOA. with 32 meters beam and features DP3 along with a 2,000 ton crane (see full details under Sample Wind Vessel Projects below).

The design parameters of wind installation vessels will vary depending upon whether they are intended to service shallow (up to 30 meters) or deepwater (greater than 30 meters) installations, with the key consideration being water depth. As noted earlier, offshore wind units are expected to be of the largest sizes available since they will not be subject to space or transportation restrictions that may be applicable to shore-based installations. Offshore turbine designs now under development will have much larger rotors with one design incorporating a 110-meter rotor diameter. It is the size of the wind units and of the key components including the mast sections and blade lengths that will drive the requirements for the size and loading capacities of the vessel work and storage decks, crane capacity and reach required to load, transport then finally, carry out the offshore installation.

Types of Installation Vessels

Shallow Water

For fixed type wind unit installations, the preferred support vessel design appears to be an OSV configured vessel which is self-propelled and self-elevating. The design should include a large working and high load capacity deck for component storage. Relatively large accommodations should be provided, as even comparatively small wind installation vessels may carry in excess of 100 personnel. A forward mounted heli-deck should be fitted. Lifting capacity is provided by two large capacity, same-side mounted pedestal cranes which may be rated between 400 – 1000 tons. Purpose built units will likely have the highest ratings and reach. The next generation wind towers are expected to be at least 120 m above the water, possibly higher. The vessel will be self-propelled with thrusters for precise positioning. DP2, considered as a minimum, with trends indicating a growing preference for DP3. Jacking units (four or six legs) will be robust in capacity and speed. One planned unit has a listed speed rating of over 30 meters/hour (100 ft per hour) or over 0.5 meters/min (1.67 ft/min) and jacking capacity load rating of 2,850Te/leg.



<u>Deep Water</u>

Existing floating wind installation vessels have the basic OSV configuration. Future vessels, which will be purpose built, are likely to be more efficient and feature specific key design features which are incorporated from lessons learned from actual installation experience and feedback from vessel crews.

For floating wind units, the wind installation vessels will be operating further offshore where ship shape hulls are better suited. Vessels will require; towing capability; large area work and storage decks suitable for large wind unit ; large, high capacity pedestal mounted cranes with active heave compensation; a minimum DP2, possibly DP3 capability; anti-roll tanks or roll reduction capability, mounted heli-deck and a large accommodation space for construction crews, wind technicians and vessel staff.

Class Notations

Class Societies recognize that these early wind installation vessels represent the beginning of a new type of OSV. They will only become more specialized in the segment as time goes on and as purpose built vessels are constructed they will include design features unique to the requirements of the offshore wind industry.

Currently, ABS has the following existing notations which cover the present vessels. New Rules and notations are being considered for development as the requirements and design considerations evolve for these vessels. It is expected that the new Rules will address the unique aspects of wind installation vessels, such as; active / passive heave compensated lifting appliances; new specialized deck machinery which support installation operations; and dedicated structures that support wind turbine installation.

Rules & Notations

Current existing Class rules that apply are as follows:

Fixed Shallow Water Wind Installations

Applicable

✠ A1 MOU, ✠ DP2, CRC

Lift Vessel – Mobile Offshore Unit

- MOU - Mobile Offshore Units July 2008
- Crane Certification – Lifting Appliances July 2007

Floating Deep Water Wind Installations

Applicable

✠ A1, Offshore Support Vessel, ✠ DP2, CRC

- Steel Vessel Rules 2009
- Crane Certification – Lifting Appliances July 2007
    - Active & Passive Heave Compensation (to be developed)

For U.S. Flag / USCG Compliant Vessels – Ref: 46 CFR 173 Subpart B (Lifting)

46 C.F.R. PART 173—SPECIAL RULES PERTAINING TO VESSEL USE

Floating Deep Water Wind Installations

Applicable

✠ A1 MOU, ✠ DP2, CRC

Column Stabilized Unit

- MOU - Mobile Offshore Units July 2008
- Crane Certification – Lifting Appliances July 2007
  - Active & Passive Heave Compensation (to be developed)

Wind Turbine Units Fixed and Floating

For the wind units themselves, the possibility exists that they will also be Classed for which ABS has the below criteria. For U.S. waters, ABS can act as the CVA (Certified Verification Agent) to MMS (U.S. Minerals Management Service) which will be required for each of the installations.

Offshore Installations

For Offshore Installations that are generally intended to remain at a particular site, these offshore facilities are known by Class as Floating Offshore Installations or FOI's. They may be buoyant or non-buoyant structures supported or attached to the sea bed. These installations consist of one or more of the following:

- Platform (offshore wind farms incorporate a platform for power collection point and to locate step-up transformers for transmission)
- Offshore Facilities

The following types of platforms are included:
- Pile Support Platform
- Gravity Structure
- Compliant Tower
- Articulated Buoyant Tower
- Tension Leg Platform

Currently, it may be possible to apply the below existing Class notation to the wind unit itself. This may require updating to address new developments in wind unit technology.

✠A1 Offshore Installation - Electric Generating Plant (electric generating plant – export load)

References:

1/1.3.1 of the Rules for Building and Classing Offshore Installations

1-3/5.3 of the Guide for Building and Classing Facilities on Offshore Installations

<u>Sample Wind Installation Vessel Projects</u>

A2SEA – Sea Worker
ABS ✠A1, Self Elevating Unit
Restricted Service
Heli-Deck
LOA 55.5m
Breath - 32.2m
Draft – 5m
Crane 300 Ton
Leg length 78.85m

Seajacks International Ltd
M/V Seajacks Kraken
ABS ✠A1, Self-elevating Unit
✠AMS, ✠ACCU, ✠ DPS-2
Self-Propelled
Heli-deck
LOA: 76m
Breath 36m
Draft: 3.65m
Four Legs
Accommodation 90 persons
Elevating speed: 0.8 m/min

Seajacks International Ltd
M/V Seajacks Leviathan
ABS ✠A1, Self-elevating Unit
✠AMS, ✠ACCU, ✠ DPS-2;
Self-Propelled
Heli-deck
LOA 76m
Breath: 36m
Draft: 3.65m
Four Legs
Accommodation 90 persons
Elevating speed: 0.8 m/min

Coastline Maritime PTE., LTD
OSA Goliath
Offshore Construction Vessel
ABS ✠A1, Fire Fighting Vessel
Class 2, Offshore Support
Vessel,
✠AMS, ✠ACCU, ✠DPS-3;
Self-Propelled, Unrestricted
Service
Heli-deck
LOA 180m
Breath 32m
Hull depth: 12m
Dwt. 22,000 tons
Crane 1,600 tons capacity
Accommodation 250 persons

Self Elevating Platforms N.V.
JB-114 & JB-115 (sister units)
ABS ✠A1, Self-elevating Unit,
Restricted Service
Heli-deck
LOA 55m
Breath 32.2m
Depth 5.0m
Four Legs
Air Gap 13.5m
Jackup Load 3,800 tons
Max Water Depth 40m
Crane 300 tons
Leg Length 73.1m

Conclusion/Summary

The offshore wind industry is only at the beginning of the development phase of its full potential as an additional source of energy for nations around the globe. The size and complexity of units is certain to increase over time as this technology is fully developed. This in turn will drive the need to transport, install, maintain, repair/replace and, at the potential end of field or wind unit life, remove and/or dispose of units. These offshore wind functions will drive the requirements for wind installation / offshore support vessels in the future.

Active Roll Compensation System for Helidecks

Uwe Heim[1*], Tom Christian Dahl [2*]

*TTS Offshore Handling Equipment AS, Ålesund, Norway

ABSTRACT

Helicopter access in due time for crew change is crucial for many Offshore Support Vessels, and at times, delays represent a significant cost driver. Increasing the weather window available for safe landing and take-off, will have a direct impact on operational efficiency of many types of OSV.

TTS Offshore Handling Equipment AS, part of the TTS Marine Group of Norway, have developed an unique motion compensation system for Helidecks, solving some of the critical safety issues of landing a helicopter on moving Helidecks offshore. The TTS Active Roll Compensation system is the world's first motion compensated helicopter deck application. By combining well-proven technologies, with a taste for «less is more» solutions, the system has been developed by TTS-OHE's team in close interaction with experienced offshore helicopter pilots, vessel crew and regulatory bodies.

The patented ARC- Helideck system is installed on two of the world's most advanced seismic vessels; PGS's Ramform Sovereign, and Ramform Sterling.

This paper will look into the ARC systems main benefits, function and development philosophy.

Helicopter Landing Offshore

Helicopter landing is one of the most safety critical operations offshore. Helicopter access to offshore installations is restricted, and marine weather conditions are a significant cost driver in the offshore business as it often causes an unacceptable operational environment and thereby costly downtime.

Motion compensation technology has for many years been a valuable contributor in downtime reduction and operation efficiency of daily tasks in a marine and unstable environment with adverse weather and excessive motions.

Now finally, this proven technology has been adapted for the purpose of increasing the weather window for safe helicopter landing on offshore vessels. This paper will introduce you to this invention, its benefits and development philosophy, but first, some background;

AHC Work Deck

In 2004 a small company in Ålesund, Norway – ICD Projects AS - solved a great engineering challenge. In less than three months, they conceptualized, engineered and constructed the world's first Active Heave Compensated work deck, The system compensates 3 Degrees Of Freedom - roll, pitch and heave movement and was developed for the Norwegian vessel owner Fredrik Odfjell's vessel FOB Junior – a catamaran especially designed for use during maintenance of offshore windmills.

The platform, still serving its purpose well, is in regular use on offshore windmill fields both off Denmark and the UK. Maintenance costs have been reduced with up to 75% for certain operations after introduction of the AHC Work Deck.

3 DOF AHC Work Deck

The 8 x6 meter work deck is mounted on the 15m long x 10m wide catamaran FOB JR. Two cranes, each with a lifting capacity of 7 tones, are mounted on top of the platform. This system prevents shock tension to the windmills winch during loading/unloading with a compensation efficiency of up to 98%. This enables the maintenance crew to safely transfer heavy loads up to 15 tons in sea states up to 2.5 m significant wave height. This application can be modified and scaled up to meet operational requirements.

TTS Marine/ TTS Offshore Handling Equipment

In 2007 ICD Projects AS was acquired by the TTS-Marine Group, and TTS Offshore Handling Equipment AS was established in Ålesund, Norway, to provide the offshore industry with effective and smart handling solutions. TTS Marine ASA has aprox 1500 employees and a turnover in 2008 of NOK 3.5 Billions.

TTS OHE is member of the Norwegian Centre of Expertise- Marine, and situated in one of the strongest Marine/Offshore clusters in the world - the area of Møre in Norway. Originating from a competitive environment driven by boldness and innovation, a great part of the 180 companies related to this industry harvest international acclaim for their efforts.

TTS Offshore Handling Equipment AS

The completion of many offshore operations can benefit from more precise handling, and TTS OHE always aims to facilitate effective operation under the harsh offshore conditions, both on the surface and in the deep sea environment.

In addition to offering “run of the mill” handling systems like Active Heave Compensated Winch systems and Anchor Handling Winches, we apply our core competence of motion compensation technology, cybernetics, and concept development to develop operation specific solutions. These applications or handling modes are especially beneficial for high end offshore support and construction vessels with high day rates.

Active Roll Compensated Helideck

The basic concept of the AHC work deck has often sparked an immediate thought in the minds of experienced offshore personnel. “This must be a perfect solution to ease helicopter landing on offshore vessels”.

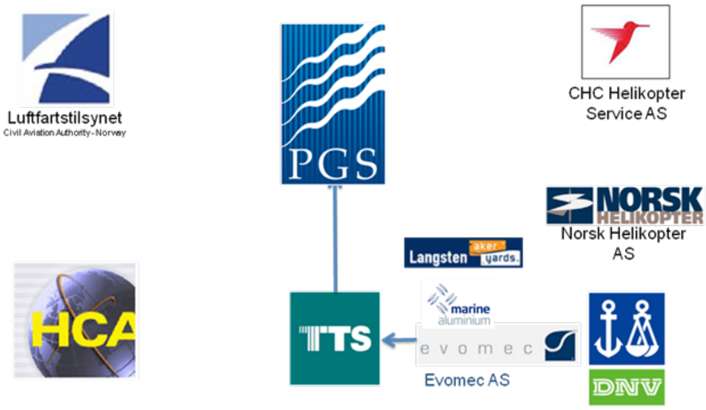
In the case of Petroleum Geo Services, the spark ignited. Heavy costs are involved when a seismic streamer is forced stop production, collect all outboard equipment, and revert to shore for an overdue crew change. Fuelled by firsthand experience; PGS launched a development project together with TTS-OHE AS. The aim; Adapting the AHC platform technology to create the world’s first motion compensated Helideck for PGS’ new vessel Ramform Sovereign.

Purpose and Benefits

The time- the weather window - available for safe helicopter landing on offshore vessels is limited. If the window is missed it can have significant implications in terms of cost and productivity, as vessel crew remain on standby and have to work extra hours in order to complete the task. Cost implications resulting from one single miss easily counts several million US$. Any application that contributes to widening the available weather window will ultimately increase the vessel’s production time and hence the overall profitability of the operation.

<u>Development Partners</u>

As experience show, the technologists initial thought of having a solution ready for the challenge is often wrong. To get a complete understanding of the factual conditions that would influence the choices we needed to make to meet the challenge at hand, a pre-study was initiated. We spoke to the pilots, looked at landing procedures and situations, and analyzed the operational environment and vessel motion patterns such as vessel angles and angular velocities movements.



Members of the Working Group

Throughout the project we had continuous and priceless participation of a working group of Offshore Helicopter professionals and regulative bodies. With the valuable input from these experienced professionals, the pre-study suggested and evaluated several alternative designs. In the end only one stood out as a safe, effective, low maintenance and financially viable concept.

We will present the technical concept shortly, but let us elaborate on the basics first.

<u>North Sea Vessel Classification System</u>

Landing on a marine helideck in the North Sea is only allowed under certain conditions. Weather, wind, wave and vessel movements must all and at the same time be within acceptable limits. This need to be thoroughly documented. Regardless of Class, Helideck take- off from base is triggered only by receipt of a 20 minutes data log being transmitted from the vessel to Helibase. The log must reflect what is been classified as safe landing conditions, The helicopter is then allowed to take off under the presumption, supported by weather forecasts, that the measured weather and wave conditions are relatively stable.

In the North Sea, vessels have been split in four Categories – based on size and what measuring and monitoring equipment they have;

Category A: Large ships (including productions ships) and semi-submersible rigs with measuring- and monitoring equipment deviating from standards described in the Helideck manual.

Category A+: As Category A, but with measuring- and monitoring equipment installed and functional, in accordance with standards described in the Helideck manual.

Category B: Small ships (diving vessels and similar) with measuring- and monitoring equipment deviating from standards described in the Helideck manual.

Category B+: As Category B, but with measuring- and monitoring equipment installed and functional, in accordance with standards described in the Helideck manual.

The limits for these classes are primarily set by vessel inclination angle and the vessel's heave rate.

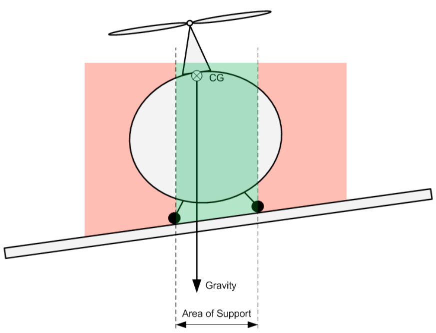
Category	Landing and planning			
	Pitch, Roll / Helideck Inclination		Max Average Heave Rate	
	Day	Night	Day	Night
A	±3°	±2°	1,0 m/s	0,5 m/s
A+	±3° / 3,5°	±2° / 2,5° (*)	1,3 m/s	1,0 m/s
B	±2°	Not approved	0,5 m/s	Not approved
B+	±2° / 2,5°	±1,5° / 2°	1,0 m/s	0,5 m/s

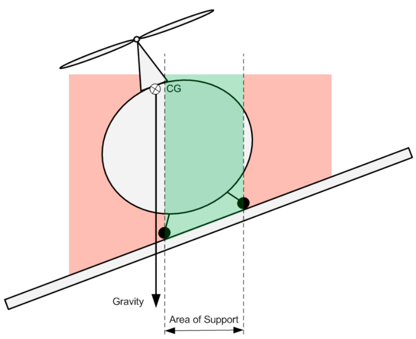
North Sea Vessel Classification System (*) For Semi Submersibles Category A+ the night limit is ±3° / 3.5°

The prime aim for this project was to achieve a reclassification of the Vessel Ramform Sovereign from B+ to A+, this will increase uptime of the Helideck significantly thus allowing landing under conditions that would not have been acceptable if the vessel was equipped with a conventional Helideck solution.

<u>Development Philosophy</u>

As we have seen, the conditions for an acceptable landing are a.o determined by the helideck's tilt angle. Let us investigate this further;





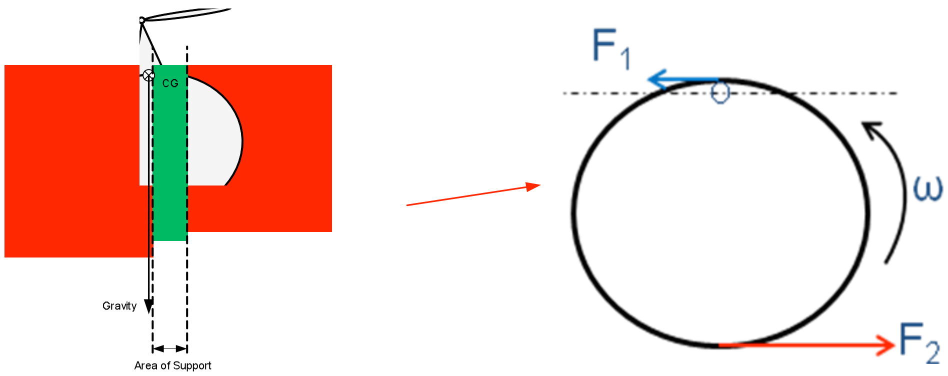
When a helicopter lands on a horizontal Helideck, the undercarriage gives an area of support to the helicopter. If the Helideck is slanted, the area of support is decreased. This area is important in relation with the helicopters Centre of Gravity, which is normally placed high. In an extreme example; if the

Helideck angle is too great, and the helicopter Centre of Gravity is outside the area of support, the helicopter will tip over. This problem is emphasized if the Helideck is accelerating sideways, which it does on a vessel at sea.

On a human male the centre of gravity is normally placed in the vicinity of the diaphragm, and the area of support is on a straight line between the legs. If the person stands on a rug and that rug is pull from underneath his feet, he will fall. As a counteraction to avoid falling, the person will throw his upper body in the same direction as the rug or move one foot to widen the area of support.

Understanding the ARC principle

This analogy is equivalent to the helicopter and the Helideck scenario. When a conventional helideck is in motion, the sidewise acceleration induced by the vessel movements cause the resulting Area of Support to decrease, thereby significantly increasing the tip-over danger.



To reduce the tip-over danger and maintain the helicopters area of support, the acceleration induced by the vessel must be removed.

The conclusion that accelerated sideways forces are very critical in any marine helideck landing situation, was confirmed through feedback from the group of experienced offshore pilots in this project. They showed us that in the landing phase, the most critical issue was not the heave or pitch, but the roll induced sway- sideways displacement.

This is also supported by the regulative bodies, specifically through the introduction of an MSI – Motion Severity Index as an addition to other critical landing criteria as wind and heave roll and pitch measurements

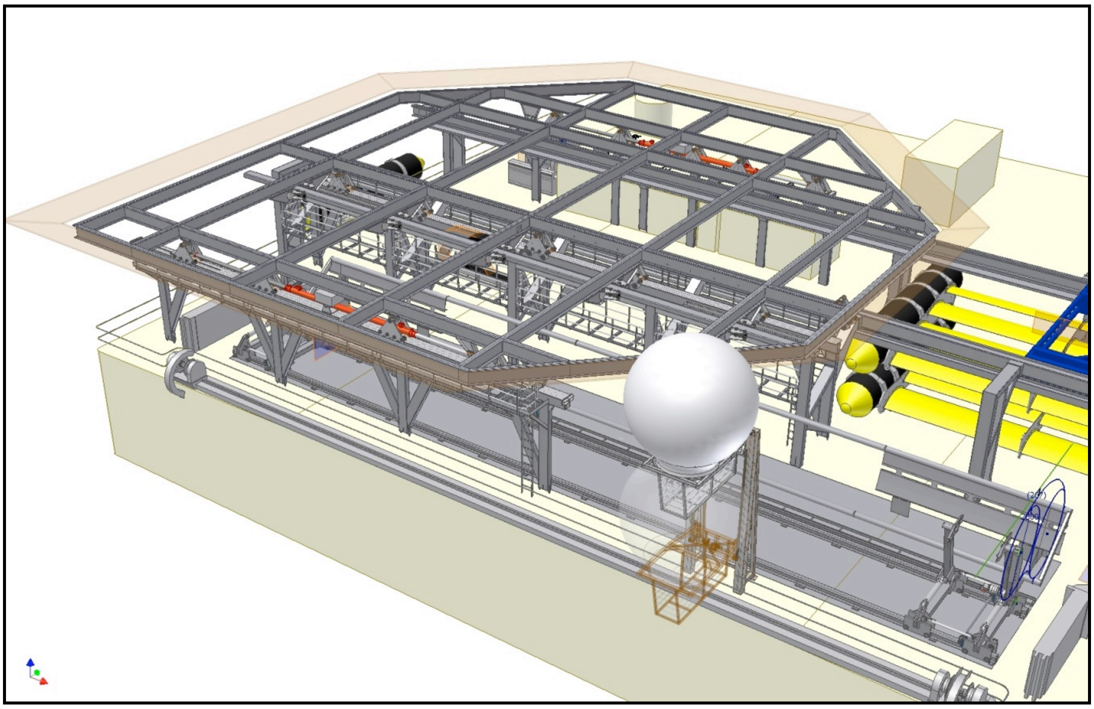
These confirmed design criteria were the basics for the final concept.

The ARC solution on Ramform Sovereign

The ARC system is actually the Helideck support structure. A conventional helideck of 28x28 meter is mounted on a sturdy frame with a horizontal rail system. This frame is connected with the main support structure. It runs on boogies and based on input from the Motion Reference Unit and controlled by the control system, it is moved back and forth with two linear compensators with a stroke length of 4.4 meters.

Thereby we remove the y-motion in the xy-plane projection of the vessels 3Degrees of Freedom roll movement. Various sensors give input on the actual position of the system at any given time.

As vessels are different some initial considerations have to be done to modify and adapt the system/support frame to the specific vessel type. The Helideck has weight of approx. 80 tons, has a maximum velocity of 1.6 m and will compensate for vessel movement up till max 5 degrees.



ARC system

Operation

The helideck has four operational modes. ARC mode, Lock mode, Service mode and emergency mode

ARC (Active) Mode: The system compensates automatically for the helideck's sway motion induced by the vessel's roll movement. It is not possible to run the Service Mode when ARC Mode is active. A 20 minute long log from the helideck monitoring system must be sent to the helicopter land base before the helideck can departure. This log will include the helideck's motion in ARC Mode. The helideck must run until the helicopter departures from the vessel

Lock Mode: The helideck is automatically parked in locked position. When the helideck is locked it will act as a conventional helideck. When leaving ARC Mode the system will enter the Lock Mode automatically

Service Mode: The helideck is driven manually with a joystick from a control panel. This Mode is used during service and maintenance of the helideck. It is not possible to start ARC Mode or Lock Mode when Service Mode is activated.

Emergency Mode: In this Mode the helideck is driven manually by hydraulic handles. This Mode is used when the control system is disabled or shut down. There will be a procedure plate mounted for instructions

System Safety

Marine conditions are harsh, and the ARC Helideck needed a design that was robust and low maintenance. Based on this, the ARC Helideck system design had to be founded on a less is more philosophy. The technological solutions are all well proven – linear compensation has been used for a great number of years in the offshore industry. The Control System Platform CDP is the software also chosen by Rolls Royce Marine for all their control systems. Today, hundreds of safety critical systems are developed on the CDP platform, and runs on everything from Cruisers in the Caribbean to Norwegian Coast Guard vessels. All control system components are standard commercial off the shelf – and easy to Source anywhere in the world. In addition a thorough HAZOP/FMECA was done by DNV with participation of all the collaborating partners including the certifying bodies and The Norwegian Aviation authority. This process ensured that we would prepare the system in such a way that all safety precautions were taken.

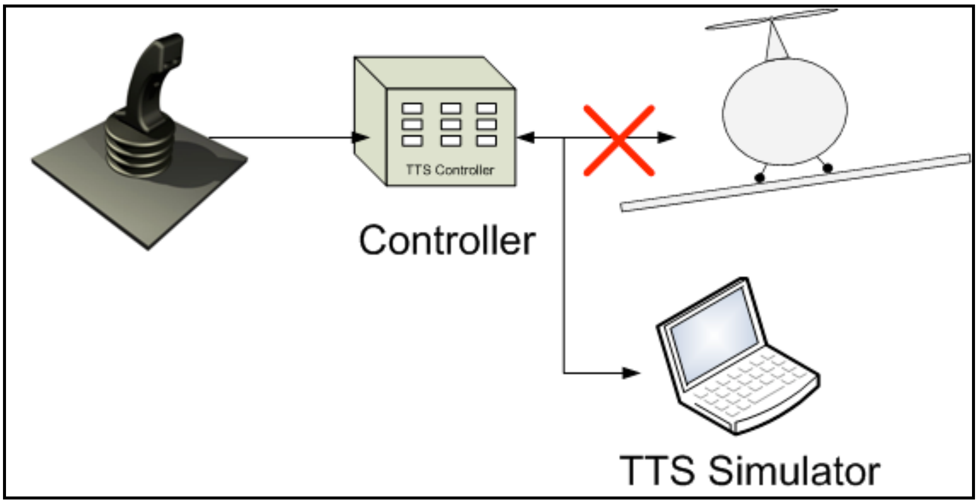
For fast problem solving, the system has remote access features. This means that skilled professionals can access, diagnose and guide any necessary repairs of the system from an onshore location.

Computer Simulation and Hardware-In-the-Loop Testing

Advanced Real Time Simulation tools were used through the whole project, assuring the best possible outcome. First, the control system application was completely tested in the office before any mechanics were involved; all mechanical parts were simulated using CDP software simulation tools. After verification of the control system software, the physical parts of the system were gradually integrated for Hardware-In-The-Loop simulation. Simulated vessel movement was eventually used to test active heave compensation on-shore. The HIL simulation reduces risk and cost, ensuring the functionality of the complete system before going offshore. During the sea trials only the last of fine tuning were necessary.

The definition of a simulator is;"A device that enables the operator to reproduce or represent under test conditions phenomena likely to occur in actual performance", Merriam-Webster Dictionary 2009. Computer simulation is a useful part in mathematical modeling of natural systems such as physics, chemistry, economics, environment, and many more.

In development of systems like the ARC Helideck, computer simulation is used to represent the laws of physics. The model is used to verify the behavior of the system and would typically contain hundreds of time dependent differential equations. The equations are solved using numerical methods on a computer. Computer simulation is an important tool in system development and prototyping. It can, dependent of the focus, be used in strength calculations, fatigue calculations, control system verifications and more.



At TTS OHE computer simulations is used in control system verification and Hardware In the Loop testing. When developing a simulator for Control System verification there are two approaches; one where the Control System and the simulator is built into the same application and one where they are distributed into two applications. If the first approach is used it is not always easy to divide the control system from the simulator model. It can be tedious and prone to errors. With the latter solution, the Control System can easily be separated from the simulator model and it is not necessary to test it again after it has been implemented. This is particularly useful in HIL testing.

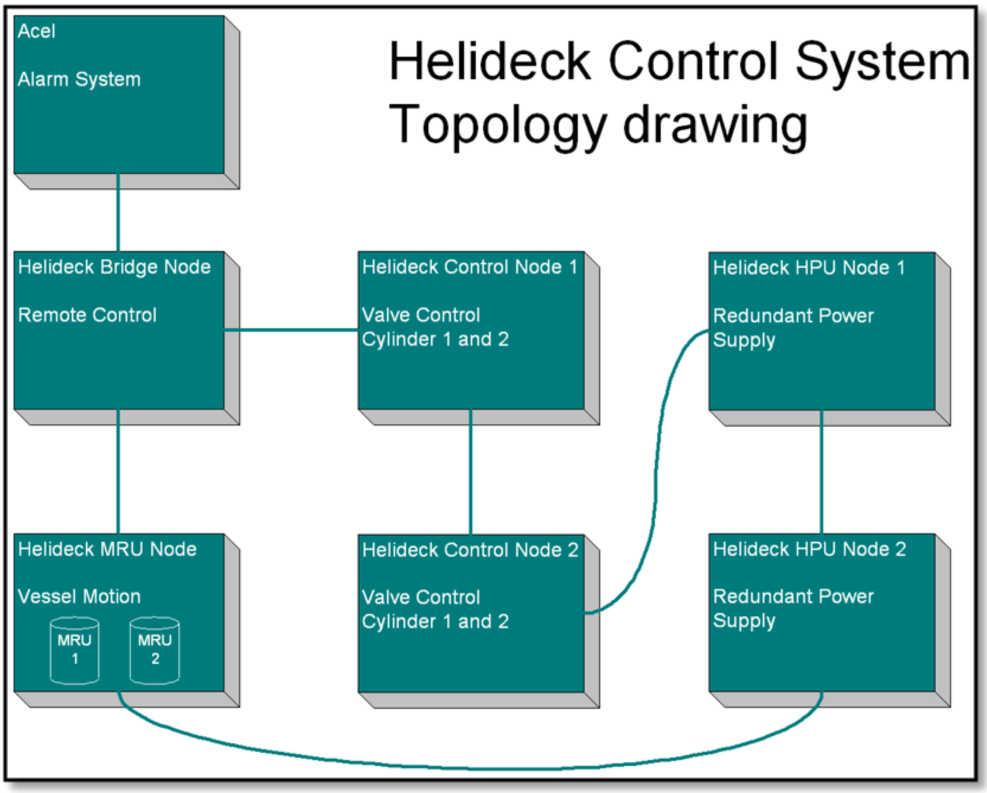
Hardware-In-the-Loop testing is one of the techniques that TTS Offshore Handling Equipment uses as a tool in system development and verification. HIL testing is performed with hardware, such as Controller Units, from the actual system connected to a simulator. With the simulator in place instead of, e.g. the Helideck, it is possible to test the control system for errors and verify behavior.

The advantages of HIL testing are manifold, besides revealing errors there is one in particular that is very useful; the ability to test closed loop control algorithms. Active Roll Compensation contains such an algorithm. This means that the ARC is dependent on feedback from sensors to know in which direction and how far it should compensate. Conventional tests are unable to verify such loops because of the inability to produce applicable encoder signals. The simulator generates such signals making it possible to verify closed loop algorithms at the office and even make a coarse tuning of the system.

Redundancy

All critical parts of the system are fully redundant; the hydraulic system, the power supply and the control system, including sensors.

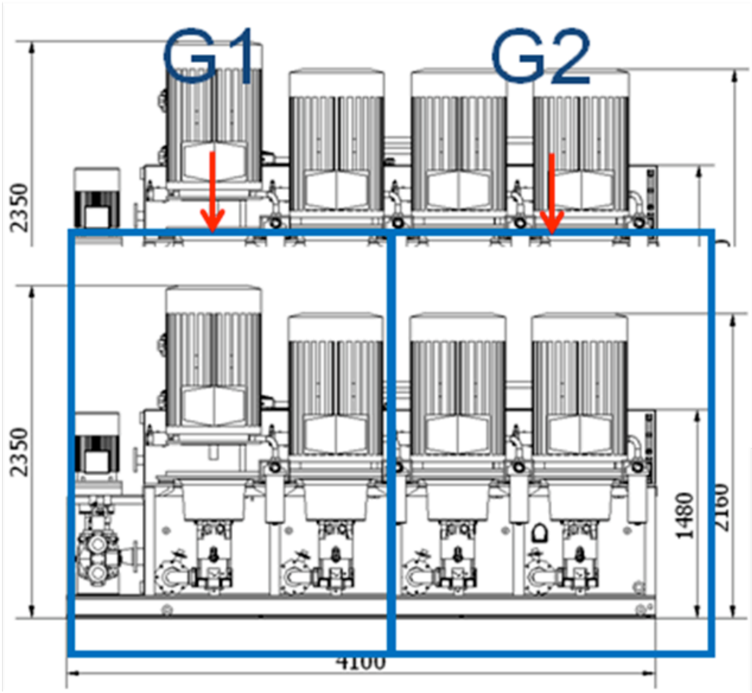
Control System: The ARC equipment has dual MRU's CPU's, sensors and I/O modules. The control system gathers information from different sensors. It analyzes the information and makes decisions in the various distributed nodes in the system. Information is sent and processed with a speed of 100 hertz, some processes even up to up to 2 kHz, -giving real time response in the mechanical/hydraulic system.



The control system is based on the Hot Standby principle with one master and one slave, placed at different locations and connected through a control system loop. A failure of any one of the components will not result in an immediate stop, nor will a broken loop. The system will run as normal. When the active controller for some reason should fail, the standby controller will take over as the new active controller within a given timeframe, providing a seamless switchover.

However and based on the Fail-Safe principle; Should any failure in the system as such result in an alarm,

Energy Supply: The energy supply to the ARC system is fully redundant and set up with a hydraulic system with Dual set of hydraulic pumps. Each set consists of two pumps with separate electrical supply. Two accumulators are in place for storage of oil pressurelanding procedures will be stopped immediately and until full redundancy is reestablished.



Redundant Hydraulic system

The accumulator oil pressure is released in case of a system pressure drop. In addition the system has a Dual set of hydraulic valves. One set is standby. There are two set of hydraulic valves controlling the linear cylinder compensators moving the helideck. Each set is independently controlled and is supplied with electricity from two separate generators. The system has four hydraulic pumps running at all times during normal operation

The system also has redundant electrical supply with two electrical generators and UPS (battery supply for electronic equipment). If one electrical generator is shut down, two pumps will still be running. If both generators are shut down the hydraulic accumulator will provide enough pressure to obtain a controlled helideck shutdown.

Certification Process

Ramform Sovereign was baptized in Ålesund – Norway 12 of March -08 and a landing test program was scheduled to start shortly thereafter. Final certification was expected to be ready by mid 2008, and Ramform Sovereign would be the first vessel in the world that can benefit from the advantages of having a motion compensated helideck**.** Regretfully, but nothing less than what was to be expected in a prototype project as this, our ambitious and maybe optimistic time schedule was delayed. We needed to replace one of the main cylinders – this postponing the whole certification schedule significantly.

Ramform Sovereign had to leave Norwegian waters for a 2 years assignment off the coast of Brazil. This made it difficult to perform sea trails and commence the certification process, especially as the initial approval of the Helideck shall be for the North Sea.

However, there is already a second ARC Helideck on PGS's new build Ramform Sterling Sovereign's sister ship of Ramform Sovereign. Ramform Sterling is ready for delivery from STX Europe AS (former Aker Yards) Langsten in July 2009.

Sea trials of the ARC Helideck will commence shortly after delivery. Trails with the helideck in ARC mode will be initiated with a period of test runs to log Helideck and vessel movements. The data from this logging period will be analyzed and reviewed before any actual landing on the vessels helideck in active mode is allowed.

The next step is a series of helicopter landings on the active deck. These trials will give the pilots view of the landing situation, and reveal any need for amendments or additions to the concept or related systems as markings, lights etc. Also crucial is the landing procedure as such and the interaction the pilots have with the vessel crew. During the trials the involved parties will get a good understanding of which additions needs to be made to adapt standard landing procedures to landing on a helideck with the TTS ARC system.

All parties, including regulatory bodies, trust that the ARC system from TTS will contribute significantly to reducing the risk moment during helicopter landing on movable marine installations, and are strong believers in this project and are confident in its approval.

Final certification is now expected to be ready Q3 2009.

The workgroup involved in the development of the ARC Helideck consisted of DNV, Civil Aviation Authority- Norway, Helideck Certification Agency, CHC Helikopter Service AS, Norsk Helikopter AS, PGS, Evomec AS, and TTS Offshore Handling Equipment AS.

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USBL Systems
Pushing the Performance Boundaries

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1 Abstract

This paper is focused on the usage of Ultra Short Baseline Systems (USBL) on Offshore Supply Vessels (OSV) and how a good system can add significant value to your vessel. It first introduces the USBL system and the tasks it can perform. Focus is then drawn to the sensors, interfaces and technology available for integration in the system. Having identified what a USBL system is and the technology and sensors required, the paper looks into how the system can be installed or improved to meet criteria for USBL tasks. These tasks are then summarised and scenarios are identified whereby the correct USBL system can add value and a good representation of your vessel which will result in prosperous charters above and beyond typical OSV tasks.

2 What is a USBL System?

A USBL system is a subsea range and bearing system. Using acoustic signals in the water column, it can measure to and from a transceiver that is installed on the vessel, and a transducer that is integrated into a transponder on the seabed or subsea vehicle such as an ROV. This can provide a position of the transponder in relation to your vessel in a similar manner that your vessel is positioned in relation to GPS satellites.
It can be used for DP where the vessel is positioned relative to a seabed transponder. The position is taken from the USBL system and fed into the DP desk so that if there are GPS problems, the DP will continue to hold station using the transponders position.

It can also be used to position multiple subsea targets such as ROVs, divers & structures. This second scenario is often overlooked during vessel build and equipment specification and yet it is in this capability that actual monetary value can be added to your vessel.

Modern USBL systems such as Sonardynes Fusion Wideband system also allow simultaneous operations (SIMOPS) to be conducted both with system installed on the vessel allowing it to combine DP and Survey positioning at the same time, whilst also being able to operate within interface range of other vessels. This saves time and money for the field operator and by meeting this performance benefit, your vessel can gain further value.

However, in order to attain this performance and reputation, careful consideration into the system, installation and operation is required.

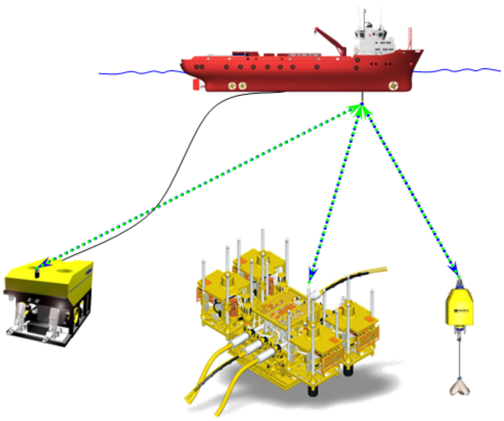

Figure 1 A USBL System

2.1 Simultaneous Operations (SIMOPS)

2.1.1 Vessel SIMOPS

Current USBL systems are now versatile enough to conduct both DP and Survey scenarios simultaneously. They can provide relative range & bearing to a transponder(s) on the seabed for optimal

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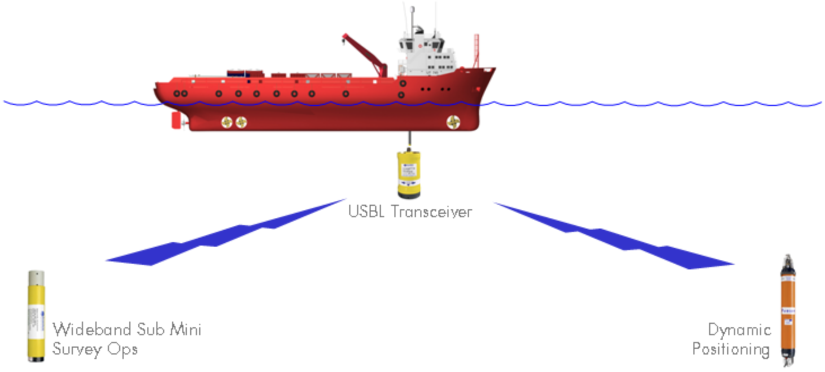
DP classification as well as relative or absolute positioning of seabed & mobile targets for survey operations.

Both parties can use the same hardware & software to meet their individual requirements. However, DP & Survey should maintain independent software systems but share the USBL system as pictured below.

Figure 2 Shared use of the same hardware



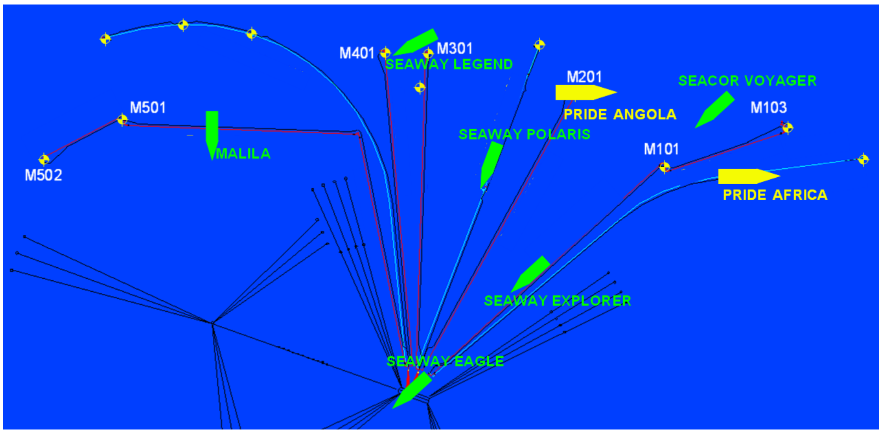
2.1.2 Field SIMOPS

There is also a demand for Simultaneous Operation (SIMOPS) to be conducted in field. SIMOPS offer a big cost saver to the field operator by allowing multiple vessels to work at the same time to get fields online. This can be lucrative for the vessel providing the vessel is capable of SIMOPS without interfering with other vessels.

By installing equipment from a single manufacturer standardises the acoustic signal type making frequency planning easier & minimising conflicts. Equipment from a single manufacturer also means there is only one point of contact for technical support & training and of course sales

As vessel charter rates increase along with the offshore boom the demand for multi-tasking vessels will increase too, making SIMOPS ever more valuable,

Figure 3 SIMOPS - A Real World Scenario



In the above real world example, OSVs worked harmoniously alongside drill ships & barges in field through good project planning. This including an AHV currently operating as a Survey Support Vessel that was being positioned on DP. Simultaneously it was also positioning one ROV for well operations. Another OSV being utilised for survey support was also positioning on DP whilst simultaneously positioning two ROVs and conducting LBL operations with no conflicts.

Figure 4 SIMOPS

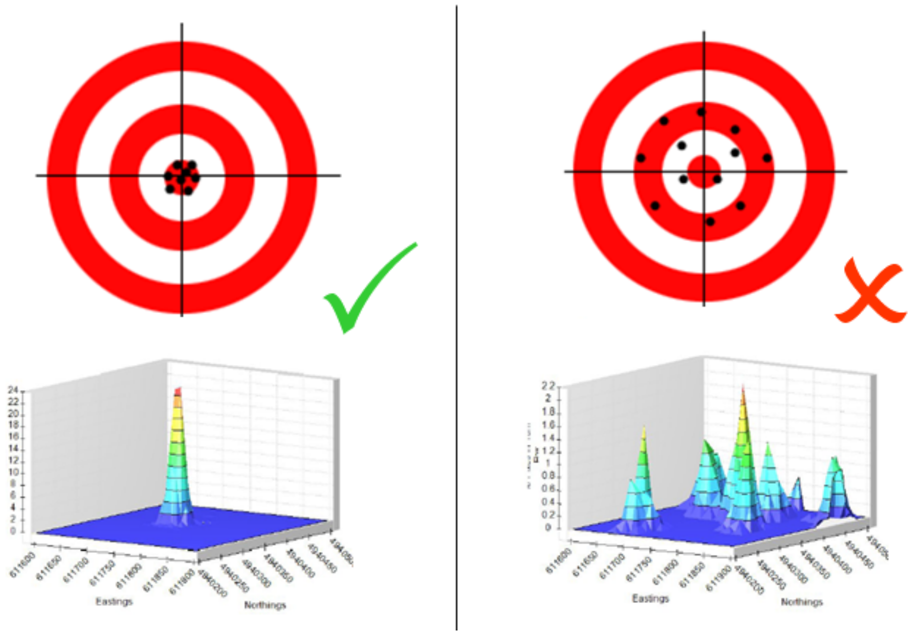
3 Importance of a good system installation

The accuracy of a USBL system is dictated by a combination of system installation factors and real-time factors. By optimising the system installation before operations makes things easier as well as saving time and money.

Figure 5 USBL Performance Chart

Both the method by which the system is installed, and the integrated sensors for its use, have an impact on the reliability and quality of the complete systems. Poorly installed and configured systems produce highly spread data which can be unacceptable for DP, survey or both.

Figure 6 System Precision



3.1 Deployment Poles

A deployment pole offers a rigid interface on which to mount the USBL transceiver on the vessel. USBL performance improves with a secure pole installation and a rigid pole improves the Signal to Noise Ratio (SNR) by both reducing cavitation and securely lowering the transceiver well below the vessel hull. A good SNR value means that the system will perform at its optimum.

A rigid pole also improves the accuracy in which it can be calibrated, the system repeatability when the pole is recovered and re-deployed and also protects the transceiver from the elements which reduces maintenance.

A through hull deployment machine is the recommended system for mounting the USBL transceiver. It offers easy deployment and recovery by the simple press of a button and its high rigidity offers the best possible platform from which to take USBL measurements. A gate valve should be installed during initial vessel build with attention paid to its physical location to avoid thruster noise and cavitation.

Figure 7 A through hull deployment machine

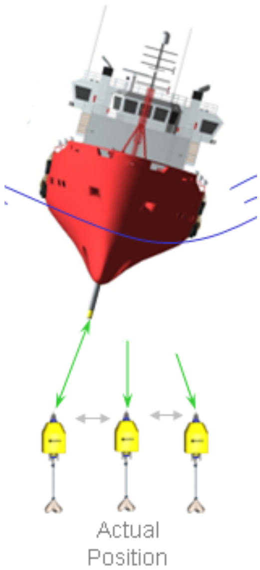
A good over-the-side system can also be sufficient. They offer the benefits of not requiring a permanent system installation and obviously do not require an expensive gate valve. They can easily be mounted on a vessel of opportunity and can be fabricated quickly if required. However, bespoke poles can vary in quality and degrade the USBL performance. Sonardyne offers through hull deployment machines as well as over the side poles which are designed with drag and vortex reduction in mind.

Figure 8 A Sonardyne over the side pole

3.2 Sensors

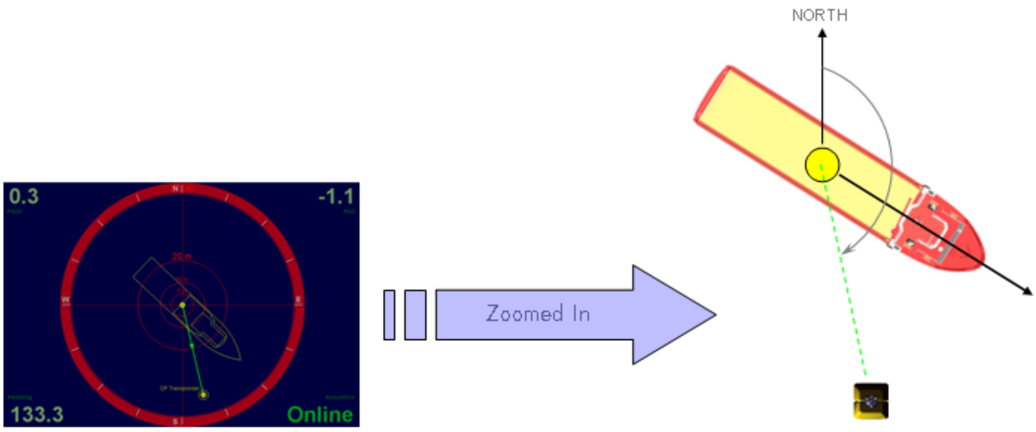
Good USBL performance also requires a pitch and roll sensor input. Imagine a DP transponder static on the seabed. As the vessel rolls the USBL range and bearing measurement from the transceiver to transponder changes. The software then plots the transponder as moving as it will have no knowledge that it is the vessel that is moving and not the transponder. This is incorrect and will cause DP alarms. It could even manifest as a DP run-off. This is solved by interfacing a high grade external pitch and roll sensor.

Figure 9 The effect of vessel motion without an attitude sensor



A good USBL performance also requires a heading sensor input. If a transponder is being positioned without knowing the real heading, it can only be positioned relative to the transceiver's forward mark. This is solved by interfacing a high grade heading sensor so that the actual bearing to the transponder can be calculated.

Figure 10 Application of heading sensor information



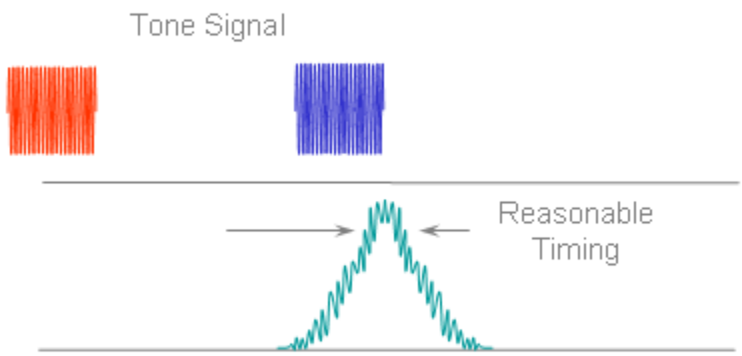
3.3 Tone and Wideband signals

The acoustic technology used in the system also effects the system quality. Traditional acoustic technology used analogue signals to compute the bearing and distance between the USBL transceiver and the transponder. These signals are called tone and consist of a short transmission of constant carrier frequency. One transmission per carrier frequency means that the number of channels is limited.

3.3.1 Tone Signals

When a transponder signal is detected by the USBL transceiver, the receiver sees a rise and fall in signal. The USBL transceiver detects the signal above the background noise which means that the signal processing can be affected by high noise. The timing precision of a tone signal is in the order of 20cm.

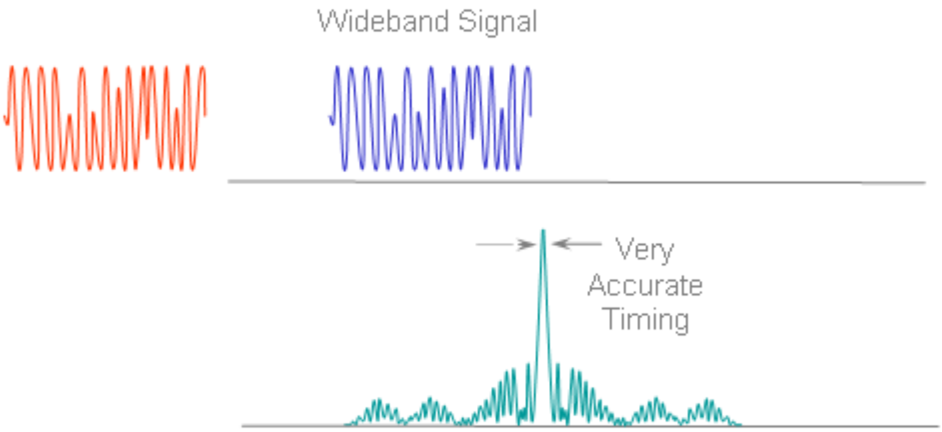
Figure 11 An analogue tone signal



3.3.2 Digital Wideband Signals

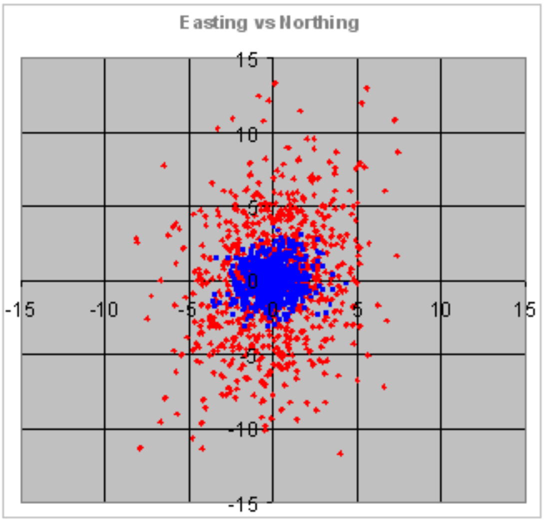
Wideband is Sonardyne's proprietary digital acoustic signal technology. It consists of a short transmission of a modulated carrier frequency. This offers unique codes rather like serial numbers, meaning hundreds of channels are available. When a Wideband signal is detected by the USBL transceiver, it is like fitting a key into a lock. The Wideband signal is more robust than analogue in high noise environments making is extremely useful in the offshore industry.
Wideband timing is also very precise offering range measurements better than 1cm.

Figure 12 A digital wideband signal



Wideband precision and its improved signal to noise ratio offers a noticeable improvement over its analogue counterpart

Figure 13 Tone verses wideband

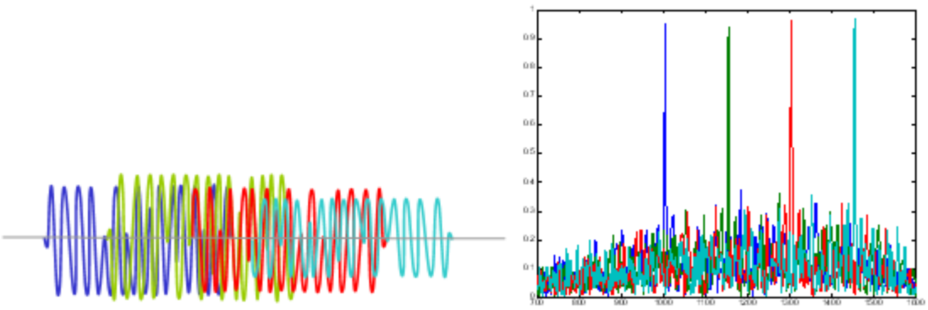


Tone Statistics		
Stdev X	Stdev Y	Stdev Z
2.60	4.22	1.01

Wideband Statistics		
Stdev X	Stdev Y	Stdev Z
1.16	1.18	0.17

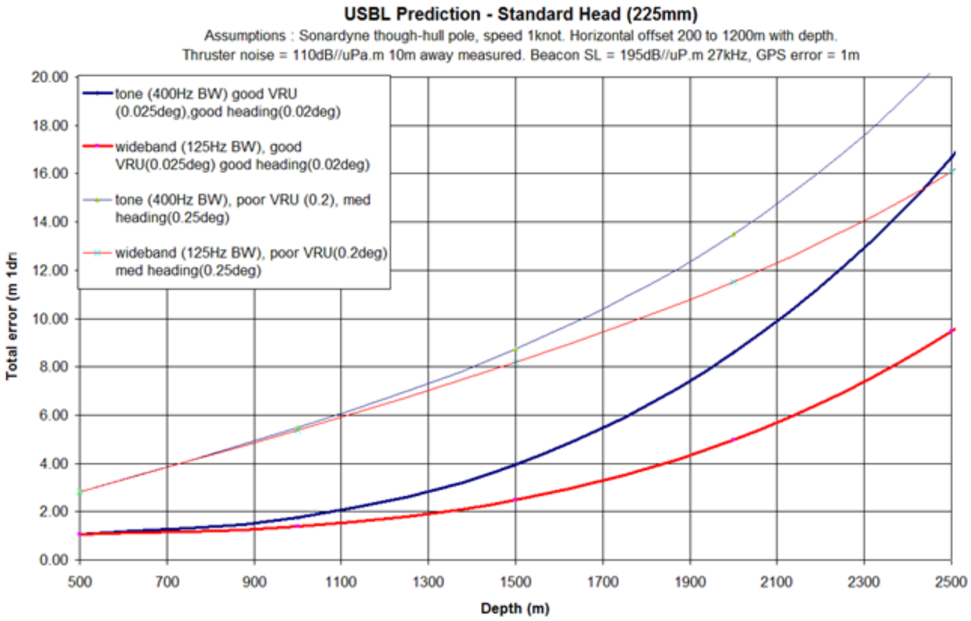
Wideband correlation technique can also detect overlapping codes, making Wideband more resilient to overlapping transmissions. This means that wideband signals are also perfectly suited for simultaneous operations.

Figure 14 Overlapping Wideband signals



So to summarise, the USBL quality will be influenced by the physically installation, signal type and reference sensor quality.

Figure 15 System performance



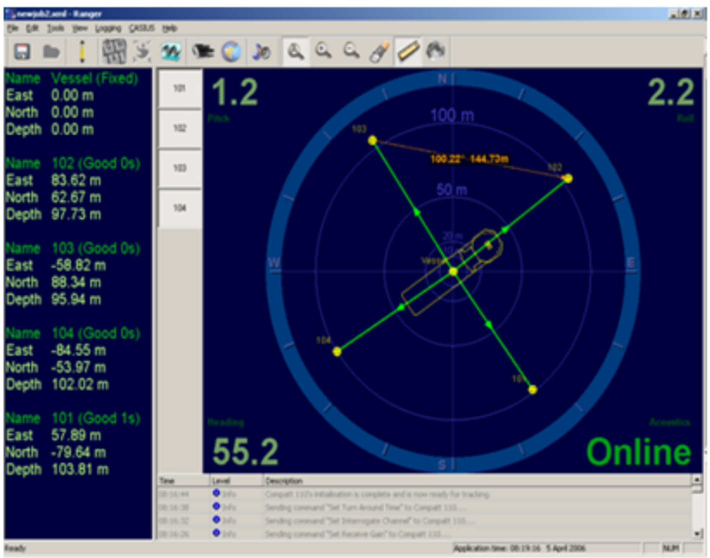
3.4 Controlling Software

Sonardyne offers two windows based topside software products to control the USBL transceiver. Due to their different complexities and features, they are suited for different operators.

Ranger USBL is often used for DP. It is also used for survey operations when the survey contractors already own a powerful navigation suite of software. Ranger USBL offers:

Simple to use
Wideband & Tone Capable
Industry leading Acoustic Performance
1 Hz update rate no matter water depth using “Ping Stacking”
Simultaneous Tracking up to 10 Beacons
Basic observation space Kalman Filter (range gate)

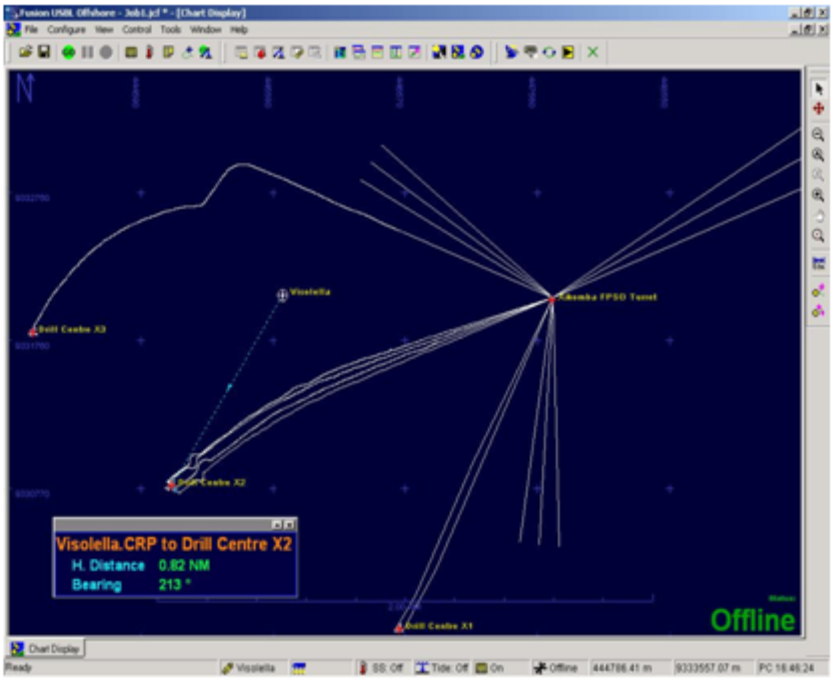
Figure 16 Sonardyne Ranger USBL Screenshot



Fusion USBL is often used for complex survey operations. It offers:

Complete Survey USBL System
Wideband & Tone Capable
Full Geodesy Package
Simultaneous Tracking up to 10 Beacons
Extended Kalman Filter
Guidance AutoCad Backdrops
Reconfigurable to LBL

Figure 17 Sonardyne Fusion USBL Screenshot

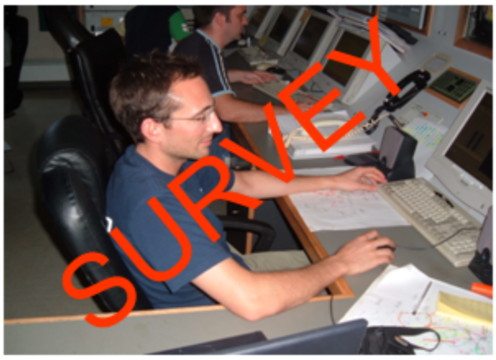


4 Improving your system

A fully optimised Sonardyne Fusion or Ranger USBL closely coupled with Sonardyne AHRS (Attitude Heading and Reference System) and a good GPS reference can achieve.

Acoustic update rate of 1 second regardless of the water depth
Accuracy of 1m in depths to 500m
Accuracy of 5m in depths of 1500m

Figure 18 DP and Survey

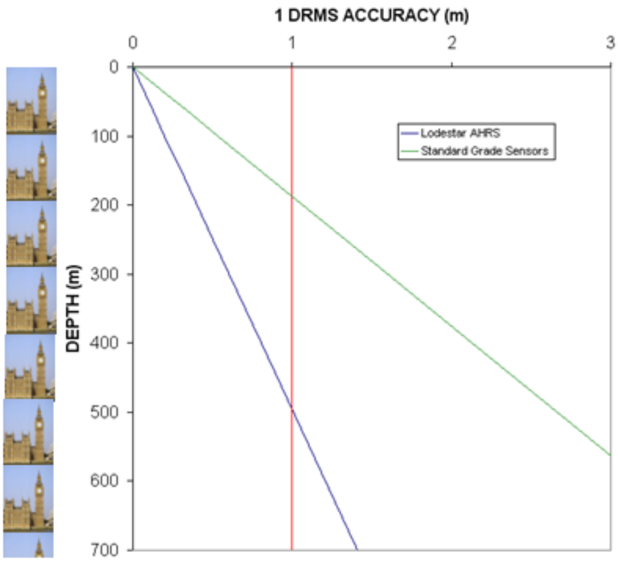


DP systems often look to achieve 1m accuracy in the USBL performance to provide stable DP updates. The effect of using good and poor attitude sensors can dramatically effect the ranges achievable when looking for 1 metre accuracy.

For example, looking at the following sensors:

Lodestar AHRS = Pitch and Roll 0.01°, Heading 0.1°
Standard Grade AHRS = Pitch and Roll 0.20°, Heading 0.6°
Noise at transceiver = 110dB
Transponder Source Level = 199dB

Figure 19 System Performance for DP

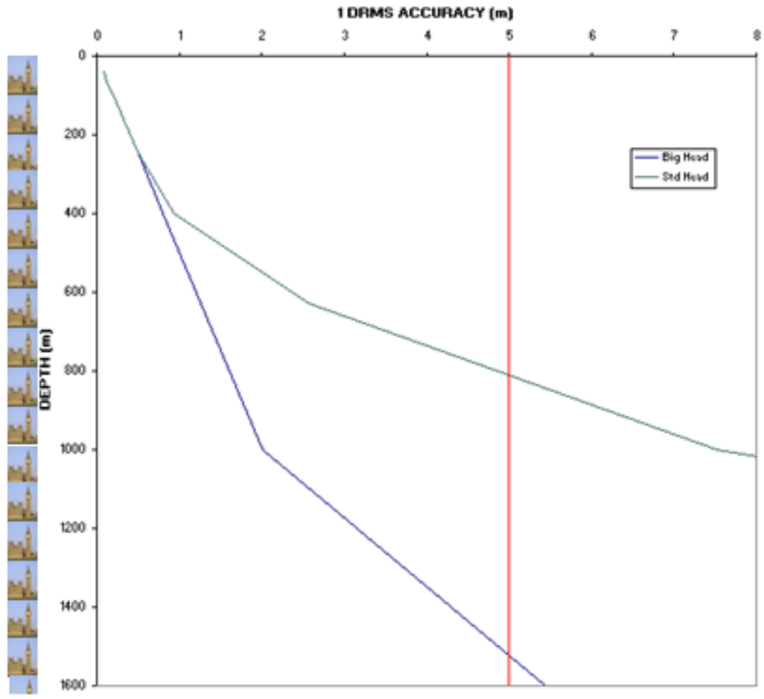


Using standard grade sensors, 1m accuracy is exceeded in 200m water depth. Switching to high grade sensors such as Sonardyne's Lodestar AHRS, the 1m accuracy is not exceeded until 500m which is equivalent to over 5 times the height of the Big Ben Clock Tower. In total, a 2.5 times performance increase.

Survey systems often look at achieving 5m accuracy in deep water. Having established that a good AHRS such as Sonardyne Lodestar provides the best sensor performance, it is then possible to look at the type of USBL transceiver to optimise the system even further.

Lodestar AHRS = Pitch and Roll 0.01°, Heading 0.1°
Noise measured at Big Head Tcvr = 110dB
Noise measrued at Standard Head Tcvr = 123dB
Transponder Source Level = 199dB

Figure 20 System Performance for Survey



Using a Lodestar AHRS, 5m accuracy can be achieved in 1500m water depth using a Big Head USBL Transceiver. That depth is equivalent to the height of 16 Big Ben clock towers.

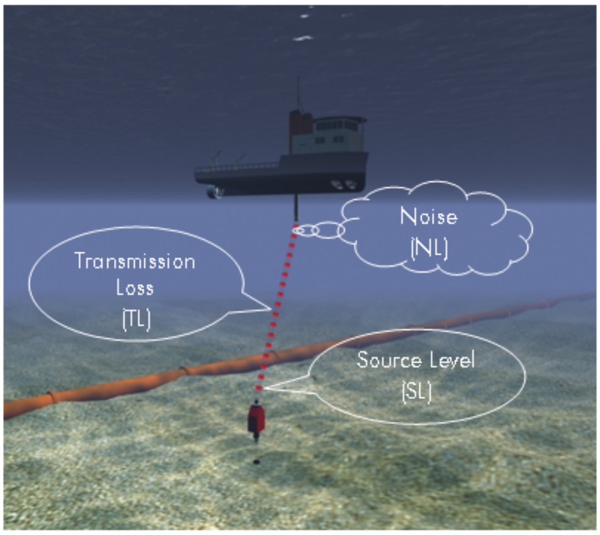
4.1 The importance of the Signal to Noise Ratio (SNR)

Effective operation requires that the strength of a transmitted signal is sufficient to be detected above the noise level at the receiver. This resultant value of detected signal is referred to as the Signal To Noise

Ratio (SNR). Poor SNR, leads to poor reliability, repeatability and accuracy as well as the loss of time, money and reputation. As already mentioned, a good SNR value allows the system to operate at its optimum performance.

The SNR consists of three basic components:
Source Level
Transmission Loss
Noise

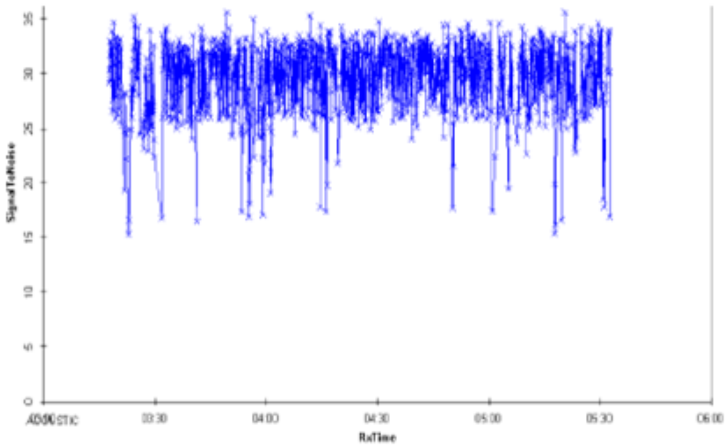
Figure 22 The Signal to Noise Ratio (SNR)



Signal to Noise Ratio (SNR) = SL – (TL + NL)

Locating a USBL pole as far away from the thrusters as possible helps to improve the SNR value. USBL poles should also be located as far away from moonpool as possible. A moonpool causes cavitation / aeration as a result of the vessel motion. This causes interference to the signal path and creates a poor USBL system.
The chart below shows the effect on SNR of a received source level dropping by 30dB because of the signal passing through a bubble cloud and suffering interference. The downward spikes correlate with position jumps

Figure 23 The Affect of cavitation



SNR also improves with Wideband signal processing. The greater the signal to noise ratio, the better the USBL angular estimation is compared to analogue systems. This is due to Sonardyne digital (Wideband) signals having:

- Longer pulse lengths providing more signal strength
- Better timing accuracy
- Improved noise rejection
- Multipath tolerance

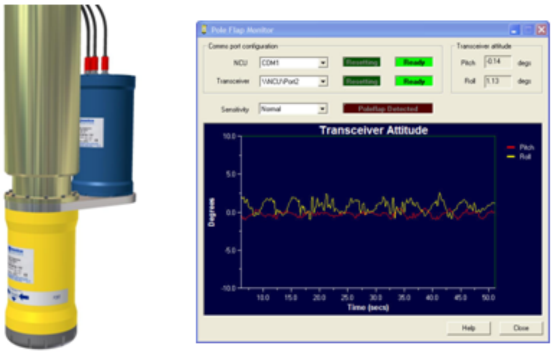
- Improved reliability/robustness

These improvements result in a higher SNR & therefore better positioning in traditionally low SNR conditions. Wideband can make a non-operational system operational when noise issues are a factor.

4.2 Improving an over the side pole

Revisiting pole deployment, when a pole moves (fore/aft/sideways) whilst the vessel is moving, the Pole wobble (Flap) will cause an error in the USBL accuracy. This error increases the more the pole moves and the further the vessel is from the transponder. It can be overcome by co-location of the AHRS attitude sensors to either the top of the pole or next to the transceiver.

Figure 24 Co-locating a AHRS



4.3 Better sensors

The quality, mounting & offset control of sensors also improves system performance. The following sensors are available for installation in the USBL system:

GPS – Required for calibration and for absolute positioning
ATTITUDE (VRU) – CRITICAL IN DEEP WATER
Heading (Gyro) – critical for high elevation
AHRS (Attitude & heading) – Modern inertial sensors offering attitude and heading

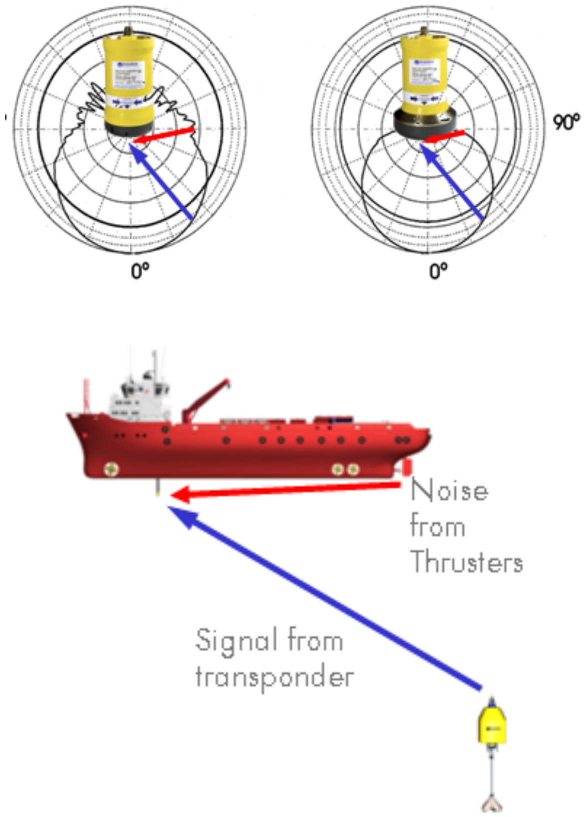
Poor quality sensors limit USBL performance. Sonardyne Lodestar is the highest quality,
low latency sensor optimised for ultimate USBL performance.

Figure 25 Sonardyne Lodestar AHRS

4.4 Selecting the best transceiver

You should also choose an appropriate USBL transceiver type. Sonardyne currently offers two types of digital USBL transceivers namely the standard and big head transceivers. Standard head transceivers offer signal detection to the sides whereas a big head focuses it's signal detection straight down. Standard head are therefore good for tracking at all angles but at the same time, allow a lot of thruster noise into the equation and thus often produce lower SNR values. Big head transceivers are best for DP and ROV work especially in deep water.

Figure 26 Sonardyne Standard and Big Head Transceivers



Signal to Noise	Observed Noise	Accuracy @ 1000m	Theoretical Max Range
Standard Head	123 dB	8m	1600m
Big Head	110 dB	2m	3600m

In the example above, tracking a DP transponder directly below the vessel, a big head transceiver can offer an improvement of 13dB in noise. When used in a predication program, an extra 13dB of signal can theoretically extend the tracking capability from 1600m to 3600m. A 2.25 times increase.

5 Concluding Remarks

SIMOPS offer the field operator potential massive cost savings by operating multi vessels and systems at the same time enabling the time till field completion to be reduced. This is especially due to the increasing offshore boom. Multi systems can be operated simultaneous from the same vessel.

In order for an OSV to operate in a SIMOPS environment it should be capable of sharing its USBL resources. It should also offer a fully optimised USBL system hardware using an optimal pole installation to offer both DP and survey grade quality results.

Warning…
Experience shows that compromising USBL system integrity at any stage results in poor performance and therefore poor vessel reputation. This can detrimentally affect successful bidding of a vessel for a project.

A New Weighted Support Vector Regression and its Application in Ship's Principal Particulars Mathematical Modeling

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ABSTRACT

Mathematical modeling on ship's principal particulars is a very important and useful task. The Support Vector Machines (SVM), a new general machine learning method based on the frame of statistical learning theory, is an effective method of processing the non-liner classification and regression. Because of its solid theoretical background and excellent generalization performance, it has become the hotspot of machine learning. This method can solve those practical problems such as limited samples, high dimension, non-linear problem and local minimum. Recently, Support Vector Regression (SVR) has been introduced to solve regression and prediction problems and widely used in many fields. With the analysis of both advantages and disadvantages of current support vector regression based on Gaussian kernel function, we propose a new weighted support vector regression algorithm in this article, thus the rigorous constraint is overcome that maintains "corresponding parameters of kernel function support vectors should be equal". In this proposed algorithm, a new kernel function is brought forward with weight factors: $K(x_i, x_j) = \exp\left(-\frac{\|x_i - x_j\|^2}{\sigma_i^2 + \sigma_j^2}\right)$, and the weight vector $W = (w_1, w_2, \cdots, w_n)^T$ is decided by the input vector $X = (x_1, x_2, \cdots, x_n)^T$. And based on this new proposed SVR method, we apply it in the offshore support vessel's principal particulars mathematical modeling in scientific research project, and compare the result with ordinary regression method and Neural Network method. The results of experiment show the practicability and effectiveness of this algorithm in the field of ship's principal particulars mathematical modeling.

INTRODUCTION

In the Design and Construction of Naval Architecture, the ship's designing system includes the determination of the main principal particulars, mold line design, arrangement and structural design and so on. It is a complicated organic synthesis with mutual influence. And the determination of ship's principal particulars is the prerequisite without doubt. It is very important in the quoted designing, the preliminary designing and even the following detailed designing. Whether the ship's principal particulars are designed reasonable or not directly influences the ship economy and the technical performance, thus affects the ship's transport business economic efficiency.

One effective way is using the empirical formula to estimate the principal dimensions. By gathering the relevant statistical data and setting some ship parameters as variables, we establish an appropriate mathematical regression model to decide principal dimensions including the ship's length, breadth, depth, design draught, lightweight and so on. Nowadays, regression methods based on the theory of mathematical statistics is widely used, such as multiple linear regression method (Jin Pingzhong,2004) and Neural Network method (SANG Song,2000) and so on.However, in practical design, using the empirical formula sometimes doesn't bring on a very good prediction results and it requires more information about the designing ship.

Therefore, how to use a higher precision and more effective method to predict under less data circumstance has become a very meaningful research. Thus deciding on how to choose an effective prediction method will take on good, practical significance. Recently, Support Vector Machines (SVM) has become an effective approach in solving this problem.

However, in the feature space of existing SVM, the parameters of kernel function corresponding to the inner product between all support vectors and samples are equal. Thus, the existing SVM can only describe the distribution features of data set by the position and weights of the support vectors. Obviously, it cannot effectively describe the distributing features. This problem has been the subject of much attention recently.

With the analysis of both advantages and disadvantages of current Support Vector Regression based on Gaussian kernel function and the essence of describing distribution features in SVM, breaking through the traditional limitation "the parameters of kernel function corresponding to the inner product between all support vectors and samples must be equal",we propose a new weighted support vector regression algorithm in this article and apply it to model the ship's principal particulars.

PROPERTIES OF SVM

The Support Vector Machines (SVM) based on the frame of statistical learning theory(V. Vapnik, 1995) is originally brought forward to solve the small sample problem of pattern recognition, and then extended to the application of regression estimation (V. Vapnik,1997; H. Drucker,1997; Alex. J. Smola,2004). This method is divided into two parts, SVC (Support Vector Machines for Classification) and SVR (Support Vector Regression). Indeed, it exhibits many useful properties (K.R. Muller,2001; B. Schölkopf, 2000; B. Schölkopf,2002). Among these properties are:

1) SVM specially aims at the situation of limited samples, with the goal of obtaining the optimal solution from the existing samples not just the optimal solution from the infinite samples.

2) The training process is made by solving a quadratic optimization problem. Theoretically speaking the global optimum solution can be obtained. Hence, local solution can be avoided, but it is unavoidable in the neural network.

3) The algorithm converts the actual problem into high-dimensional feature space by use of the nonlinear transformation. The linear discriminating function is constructed in the high-dimensional feature space to realize the non-linear discriminating function in the original space. This special property can ensure the method will have better generalization performance, while at the same time being able to skilfully solve the dimension problem. Also, the complexity of algorithm has no connection with the sample dimension.

4) The architecture of machine is automatically set by the solution of the optimization problem, which makes the training quite easy, allowing for a very low risk of poor generalization.

Therefore, these properties guarantee in general a learning procedure which lends itself to be tter generalization and more reliability than traditional prediction methods. The SVR algorithm can ensure the high precision, easy modelling and excellent limited-sample performance.

LEARNING PRINCIPLE

In this section we will introduce the learning principle of Support Vector Regression based on Gaussian kernel function. In the discussion below, we support the training data set : $\{x_i, y_i\}$,

$i = 1,2,\cdots,l$, there $x_i \in R^n$ are input variable, $y_i \in R$ are output variable, l is the number of training data. The basic idea of Support Vector Regression is to find a non-linear mapping function from the input space to the output space $\phi(x)$: $R^n \to H$, and map the input data x into the high-dimension feature space H , then adopt the proper kernel function $K(x_i, x)$ instead of inner product of high-dimension space $\phi(x_i)\cdot\phi(x)$,and use the formula below in the feature space to solve the optimal regression function:

$$f(x) = (w\cdot\phi(x)) + b \quad , \quad w \in H \tag{1}$$

whereby $(\cdot)$ means inner product, b is the coefficient.

In this way, linear regression in the high-dimension feature space is corresponding to the non-linear regression in the low-dimension input space. The support vector regression solves an optimization problem:

$$\text{Min} \quad \psi(w) = \tfrac{1}{2}\|w\|^2 + C\sum_{i=1}^{l}(\xi_i + \xi_i^*)$$

$$\text{s.t.} \quad y_i - (w\cdot\phi(x_i)) - b \le \varepsilon + \xi_i ,$$

$$(w\cdot\phi(x_i)) + b - y_i \le \varepsilon + \xi_i^* ,$$

$$\xi_i , \ \xi_i^* \ge 0 , \quad i = 1,2,\cdots,l$$

$$C > 0 \tag{2}$$

Hence majorized function $\psi(w)$ is a typical quadratic programming problem and the constraint conditions are linear. We construct a Lagrange function as follows and introduce the Lagrange multiplier α_i , α_i^* , η_i , η_i^* ,

$$\begin{aligned} L(w,b,\xi,\xi^*) = \frac{1}{2}\|w\|^2 + C\sum_{i=1}^{l}(\xi_i + \xi_i^*) \ &- \sum_{i=1}^{l}\alpha_i(\varepsilon + \xi_i - y_i + (w\cdot\phi(x_i)) + b) \\ &- \sum_{i=1}^{l}\alpha_i^*(\varepsilon + \xi_i^* + y_i - (w\cdot\phi(x_i)) - b) \\ &- \sum_{i=1}^{l}(\eta_i\xi_i + \eta_i^*\xi_i^*) \end{aligned} \tag{3}$$

Here we introduce the kernel function $K(x_i, x_i) = (\phi(x_i)\cdot\phi(x_j))$ to solve the Lagrange function above:

$$\text{Max} \quad W(\alpha) = \sum_{i=1}^{l} y_i(\alpha_i - \alpha_i^*) - \varepsilon\sum_{i=1}^{l}(\alpha_i + \alpha_i^*) - \frac{1}{2}(\alpha_i - \alpha_i^*)(\alpha_j - \alpha_j^*)K(x_i \cdot x_j)$$

$$\text{s.t.} \quad \sum_{i=1}^{l}(\alpha_i - \alpha_i^*) = 0$$

$$0 \le \alpha_i, \alpha_i^* \le C , i = 1,2,\cdots,l \tag{4}$$

The above dual problem can be translated into a normal quadratic programming problem, with regression function calculated as follows:

$$f(x) = \sum_{i=1}^{l}(\alpha_i - \alpha_i^*)\cdot K(x_i \cdot x) + b \tag{5}$$

whereby ε and C are coefficients to control the VC dimension of regression function, which are chosen by user.

Support Vector Regression based on Gaussian kernel function is the Support Vector Regression in which inner product of samples in the feature space is calculated by Gaussian kernel function $K(x_i, x_j)$. Gaussian kernel function (J.A.K. Suykens, 2001) is stated as follows:

$$K(x_i, x_j) = \exp\left\{ -\frac{\|x_i - x_j\|^2}{2\sigma^2} \right\} \tag{6}$$

In the above part, it is the common discussion about the Support Vector Regression based on Gaussian kernel function. This algorithm is quite simple and can achieve the minimum risk, so it has been widely studied and used. But the parameters of Gaussian kernel function of all the samples in this algorithm are all the same. However in some practical problems, the more important the sample is, the more useful it will be during the training process, thereby making the result more reasonable. Therefore, considering the different contribution to the prediction function, the samples with different importance should be given the different weight factors. In order to solve this problem, the author proposed a new algorithm named "Gaussian kernel parameter weighted Support Vector Regression" (GKPW-SVR).

PROPOSED ALGORITHM (GKPW-SVR)

In the GKPW-SVR, we employ this kernel function:

$$K(x_i, x_j) = \exp\left\{-\frac{\|x_i - x_j\|^2}{\sigma_i^2 + \sigma_j^2}\right\} \tag{7}$$

whereby σ_i^2 means the corresponding part of kernel function parameter decided by sample x_i when it calculates the kernel function with other samples. Also, where σ_j^2 means the corresponding part of kernel function parameter decided by sample x_j when it calculates the kernel function with other samples.

In this article, each kernel function parameter corresponding to each sample is calculated by $\sigma_i^2 = w_i \times \sigma^2$. And σ^2 is the parameter of Gaussian kernel function in formula (6), w_i is the weight factor associated with each learning samples.

Otherwise, in the GKPW-SVR, the regression function is described below:

$$f(x) = \sum_{i=1}^{l} (\alpha_i - \alpha_i^*) \cdot \exp\left\{-\frac{\|x_i - x\|^2}{2\sigma_i^2}\right\} + b \tag{8}$$

Decision of Weight Factor

The decision of weight factor w_i is very important, as it concerns the performance of the Support Vector Regression. Many scholars have done much research about the decision of weight factor. These weight factors can be divided into two parts: one part is about time series model, in which the weight factor is decided according to the samples' collection time. The other part is about space model, in which the weight factor is decided by the samples' space distribution characteristics.

In this article we adopt the method below to decide the weight factor. The importance of the training data $x_i = (x_i^1, x_i^2 \cdots, x_i^m)$ is related to the testing data $x_{test} = (x_{test}^1, x_{test}^2 \cdots, x_{test}^m)$ .The more similar, the more important, and the weight factor w_i is more bigger. And here the similarity of x_i and x_{test} is measured by the Euclidean distance described as follows:

$$\|x_i - x_{test}\| = \sqrt{\sum_{l=1}^{m} (x_i^l - x_{test}^l)^2}$$

The weight factor w_i is looked as the function of input variable x_i, and it is calculated as follows:

$$w_i = \tau + \frac{1}{1 + \|x_i - x_{test}\|}$$

where $\|\cdot\|$ denotes the Euclidean distance (Huang, H.P.,2002), τ is a positive real number, $0 < \tau < 1$.

Steps

The steps in the GKPW-SVR are as follows:

Step1 Obtain the learning data set $\{x_i, y_i\}$, $i = 1,2,\cdots,l$;

Step2 Choose the precision parameter ε and penalty factor C for the error;

Step3 Choose the corresponding Gaussian kernel parameter σ_i^2 according to the samples' importance;

Step4 Solve this quadratic optimization problem, obtain the values of α_i, α_i^* and b , and gain the regression function;

Step5 Input the data needed to be predicted in the regression function (8), and we can achieve the prediction result.

APPLICATION IN MODELING ON SHIP'S PRINCIPAL PARTICULARS

In this article, we gathered more than 450 supply vessels. The main principal particulars of those ships are distributed as Fig. 1.

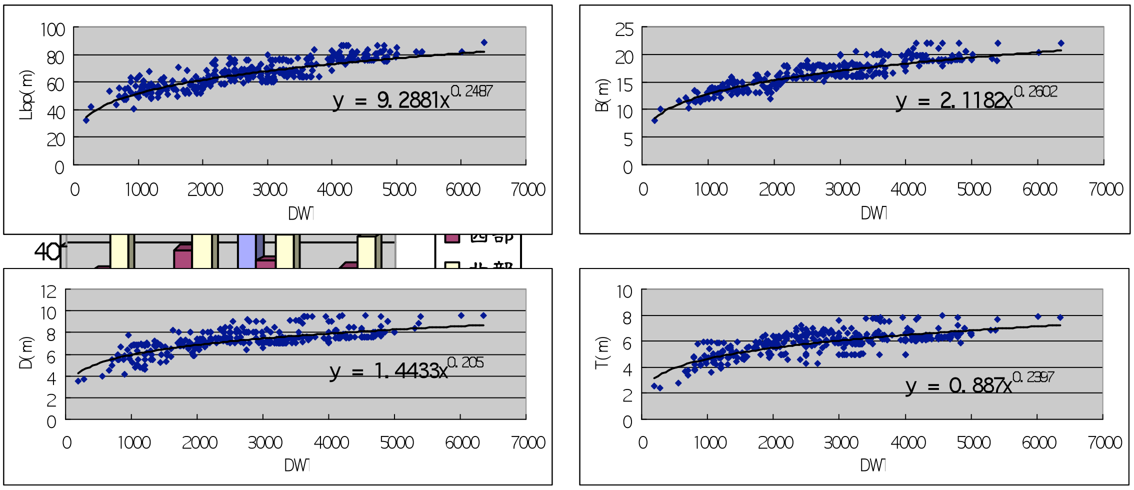

Fig. 1 Distributing curves of 450 vessels' principal particulars

Comparing with the traditional regression methods, we use the GKPW-SVR proposed in this article to establish the regression model, the gathered 450 supply vessels as training data, and eight supply vessels in real application as testing data and shown as Table 1:

Table 1 Testing data of eight vessels

Vessel \ Item	DWT (t)	Lbp (m)	B (m)	D (m)	T (m)
NICE TULIP	196	32.01	7.93	3.51	2.591
AMARCO TIGER	812	47.6	11.5	5.5	3.821
PACIFIC PATRIOT	1200	48.8	13.5	6	4.2
PACIFIC RAPIER	1520	57.25	15	6.7	5.1
FAR SKY	1900	63.6	16.4	8	5.611
LADY ASTRID	2350	65.6	17.2	8.3	6.3
NORMAND MASTER	3627	72.67	20	9.5	7.5
SUBSEA VIKING	6350	88.8	22	9.6	7.85

1) Setting up the model of ship's length

How to choose the length is usually based on the arrangement and the performance of the ship. It should meet the requirements and the arrangements of the ship. What's more, we should try to choose the ship type which is in good condition of corresponding resistance.

Maneuverability performance is closely related to the ship's length. If the ship is too long, the gyre capability is poor, and the control capability in port and sea-route need to be considered. If the total resistance is not very different at the minimum length, the shortest length of ship should be selected when the resistance is not very high in order to reduce the cost of the hull.

Here by using the GKPW-SVR proposed in this article, we choose deadweights of 450 different ve-ssels as input variable and the corresponding length between perpendiculars as output variable to establish the regression model. In GKPW-SVR we proposed, we set C=18.2, ε = 0.001, σ^2 = 21.3. Then we can use this regression model to test the eight ships in Table 1, and compare the prediction result with the traditional regression methods, the result is shown as Table 2 and Fig. 2.

Table 2 Predictive result of length between perpendiculars(Lbp)

Item / Vessel	Lbp (m)	Regression method		BP Network		Regression method		BP Network	
		Predictive Value	Relative error	Predictive Value	Relative error	Predictive Value	Relative error	Predictive Value	Relative error
NICE TULIP	32.01	34.515	0.078	33.504	0.047	32.840	0.026	32.329	0.010
AMARCO TIGER	47.6	49.151	0.033	48.370	0.016	48.137	0.011	48.014	0.009
PACIFIC PATRIOT	48.8	54.165	0.110	51.192	0.049	51.085	0.047	49.692	0.018
PACIFIC RAPIER	57.25	57.445	0.003	57.549	0.005	56.871	-0.007	57.059	-0.003
FAR SKY	63.6	60.723	-0.045	61.415	-0.034	64.728	0.018	63.958	0.006
LADY ASTRID	65.6	64.019	-0.024	64.377	-0.019	66.474	0.013	66.211	0.009
NORMAND MASTER	72.67	71.316	-0.019	71.327	-0.018	71.756	-0.013	73.151	0.007
SUBSEA VIKING	88.8	81.974	-0.077	86.742	-0.023	90.474	0.019	89.719	0.010

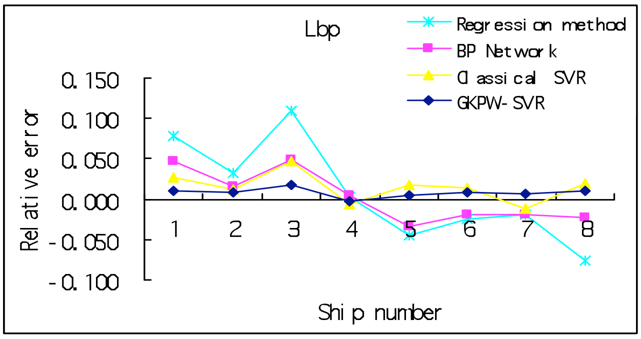

Figure 2 Comparisons between relative errors(Lbp)

2) Setting up the model of breadth

There are lots of factors which have great impact on determining the breadth of ship. And the different factors are in contradiction, such as stability and rolling. It requires wider breadth so that the initial height of metacenter and the restoring moment will be increased when considering the initial stability. But if we want to relax the rolling, the breadth should be reduced appropriately so that the initial height of metacenter will be cut down to increase the rolling period. This shows that the two requirements of the breadth are self-contradictory. For the ship in design, it is necessary to find primary contradiction to determine the breadth.

Similarly, we choose deadweights of 450 different vessels as input variable and the corresponding breadth as output variable to establish the regression model. In GKPW-SVR we proposed, we set C=51.0, ε = 0.001, σ^2 = 12.8.Specific results of models by various methods are shown in Table 3 and Fig. 3.

Table.3 Predictive result of breadth(B)

Vessel \ Item	B (m)	Regression method		BP Network		Regression method		BP Network	
		Predictive Value	Relative error	Predictive Value	Relative error	Predictive Value	Relative error	Predictive Value	Relative error
NICE TULIP	7.93	8.364	0.055	8.128	0.025	8.074	0.018	8.016	0.011
AMARCO TIGER	11.5	12.107	0.053	11.914	0.036	11.652	0.013	11.604	0.009
PACIFIC PATRIOT	13.5	13.402	-0.007	13.632	0.010	13.389	-0.008	13.415	-0.006
PACIFIC RAPIER	15	14.252	-0.050	15.421	0.028	15.201	0.013	15.148	0.010
FAR SKY	16.4	15.104	-0.079	15.897	-0.031	16.624	0.014	16.537	0.008
LADY ASTRID	17.2	15.963	-0.072	16.553	-0.038	16.742	-0.027	16.923	-0.016
NORMAND MASTER	20	17.872	-0.106	20.646	0.032	19.643	-0.018	19.787	-0.011
SUBSEA VIKING	22	20.675	-0.060	22.429	0.019	21.685	-0.014	21.801	-0.009

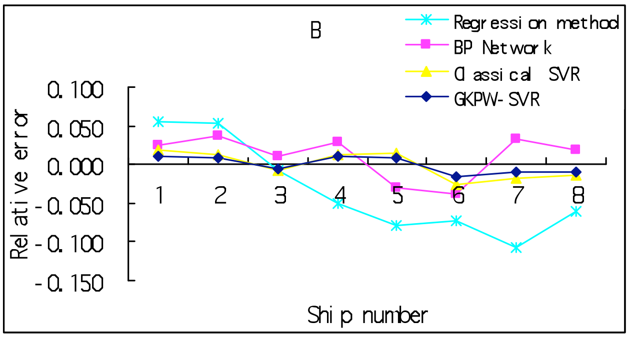

Figure 3 Comparisons between relative errors(B)

3) Setting up the model of molded depth

The size of molded depth plays an important part in the safety and performance of ship. At a certain draft, molded depth will be increased when freeboard is increased. When we choose the molded depth, the capacity, arrangement and performance should be considered. What's more, safety and the overall longitudinal strength and the impact of the cost should be also considered.

Similarly, we choose deadweights of 450 different vessels as input variable and the corresponding molded depth as output variable to establish the regression model. In GKPW-SVR we proposed, we set $C = 37.4$, $\varepsilon = 0.001$, $\sigma^2 = 55.2$. Specific results of models by various methods are shown in Table 4 and Fig. 4.

Table.4 Predictive result of molded depth (D)

Vessel \ Item	D (m)	Regression method		BP Network		Regression method		BP Network	
		Predictive Value	Relative error	Predictive Value	Relative error	Predictive Value	Relative error	Predictive Value	Relative error
NICE TULIP	3.51	4.259	0.213	3.604	0.027	3.573	0.018	3.547	0.011
AMARCO TIGER	5.5	5.699	0.036	5.328	-0.031	5.605	0.019	5.552	0.009
PACIFIC PATRIOT	6	6.174	0.029	6.106	0.018	6.094	0.016	6.069	0.012
PACIFIC RAPIER	6.7	6.481	-0.033	6.859	0.024	6.792	0.014	6.625	-0.011
FAR SKY	8	6.784	-0.152	8.315	0.039	7.854	-0.018	7.876	-0.016
LADY ASTRID	8.3	7.086	-0.146	7.872	-0.052	7.976	-0.039	8.141	-0.019
NORMAND MASTER	9.5	7.746	-0.185	9.891	0.041	9.185	-0.033	9.698	0.021
SUBSEA VIKING	9.6	8.688	-0.095	9.183	-0.043	9.467	-0.014	9.514	-0.009

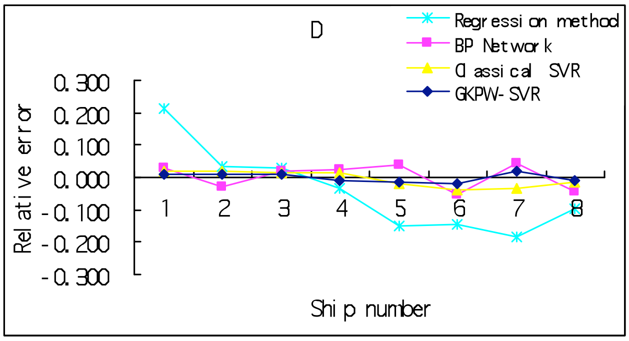

Figure 4 Comparisons between relative errors(D)

4) Setting up the model of design draught

The change of design draught will affect the value B/d. The residuary resistance increases when B/d increases under the same displacement.The frictional resistance takes a large part in the total resistance in the low-speed. In the ship design of large marine vessels, the draught is always limited by the depth of sea-route and port. Therefore, when determining the draft; we should pay attention to the environmental factors, especially the changes in water depth of sea-route and port. At the same time, the draft's determination should consider the buoyancy and seakeeping capacity.

Similarly, we choose deadweights of 450 different vessels as input variable and the corresponding draught as output variable to establish the regression model. In GKPW-SVR we proposed, we set $C = 58.0$, $\varepsilon = 0.001$, $\sigma^2 = 8.6$. Specific results of models by various methods are shown in Table 5 and Fig. 5.

Table.5 Predictive result of design draught (T)

Item / Vessel	T (m)	Regression method		BP Network		Regression method		BP Network	
		Predictive Value	Relative error	Predictive Value	Relative error	Predictive Value	Relative error	Predictive Value	Relative error
NICE TULIP	2.591	3.143	0.213	2.709	0.046	2.641	0.019	2.612	0.008
AMARCO TIGER	3.821	4.419	0.157	4.005	0.048	3.697	-0.032	3.858	0.010
PACIFIC PATRIOT	4.2	4.853	0.155	4.504	0.072	4.321	0.029	4.247	0.011
PACIFIC RAPIER	5.1	5.136	0.007	5.052	-0.009	5.143	0.008	5.063	-0.007
FAR SKY	5.611	5.418	-0.034	5.473	-0.025	5.519	-0.016	5.662	0.009
LADY ASTRID	6.3	5.701	-0.095	6.105	-0.031	6.204	-0.015	6.218	-0.013
NORMAND MASTER	7.5	6.326	-0.156	7.867	0.049	7.712	0.028	7.421	-0.011
SUBSEA VIKING	7.85	7.235	-0.078	7.611	-0.030	7.685	-0.021	7.769	-0.010

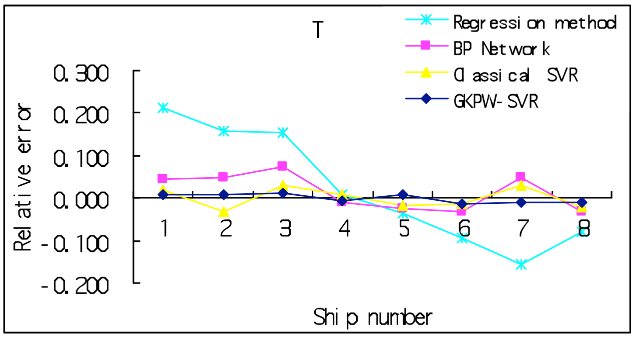

Figure 5 Comparisons between relative errors(T)

In short, the results from above shows that: It can bring on smaller fitting error and higher forecast accuracy and better generalization performance than the traditional regression analysis and neural network when using support vector machine regression algorithm and the GKPW-SVR proposed in this article.

CONCLUDING REMARKS

In recent years, with the development of the statistic theories and methods, various engineeri ng fields use SVM for dates forecasting and emulation and have made some achievements in scientific research. In this paper, we propose a new weighted support vector regression algorith m in this article named "Gaussian kernel parameter weighted Support Vector Regression". This proposal algorithm is a great breakthrough in the development of Support Vector Machines. T he example proves that this new proposed algorithm can make the best of sample information and importance of each sample, and has better generalization performance and higher precision. With the development of theories and methods, it will certainly become an effective tool in t he field of ship's principal particulars modelling.

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