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Exploring the Interfaces among Hydrodynamics, Mechanical Engineering and Controls

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ABSTRACT

Hydrodynamics affects everything the propulsion system designer does. Increasingly shipowners are calling for reduced fuel consumption and emissions together with reduced levels of noise and vibrations in passenger accommodation and crew's living quarters.

Propeller/hull/machinery interactions are important factors in achieving this, and the challenge is greater if the vessel has to operate under widely differing conditions.

Propulsors may be subjected to high transient hydrodynamic loads, which are transmitted to the mechanical elements and control system. Examples include air ingestion into waterjets and similar ventilation effects for various thruster types. This may be influenced by the design of the vessel and the integration of thrusters in the hull, or by the operational modes of the vessel.

A case study deals with propeller design related topics influencing noise and vibration levels for a cruise ferry with very strict comfort requirements. The case considers how different operational modes affect the controllable pitch blade design and cavitation performance.

Another study examines the results based on direct comparison in fully-instrumented sea trials on a vessel, fitted first with azimuth propulsors in pushing propeller mode, thereafter the same trials with azimuth units in pulling mode (tractor propellers).

A further case study shows how full scale monitoring can help to provide insight into the links between hull- and propulsor dynamics in a seaway.

Keywords

Propulsion, vibrations, transmissions, thrusters, course-keeping.

1 INTRODUCTION

The desire to minimise the required engine power for a given speed, the fuel consumption and emissions have impact on both investment and operational costs. These market and business drivers are of increasing importance in the design strategy of a new vessel.

The propulsion concept, the design layout and the power drive train arrangements are important as they influence crucial techno-economic factors such as payload, hull dimensions and machinery investments through inboard space utilisation and the effectiveness of energy transfer from fuel via engines to generation of propulsive and manoeuvring forces.

The associated activities require involvement and co-operation from a variety of disciplines including hull-internal and external design, machinery assessments, propulsor type and size selection as well as the transmission arrangement to provide efficient transfer of propulsive forces under the actual operating condition(s). The propulsor must also be interfaced the hull and the steering device(s) where mutual interactions between hull, propulsor and steering system must be taken into consideration.

Even though the quest for propulsive efficiency in the interest of fuel economy and emissions remains a dominating consideration, reducing noise and vibration is also important for most vessels nowadays. The propulsors are, beside the engines, the most important source of noise and vibration on a vessel. Parts of the propulsion plants, such as the gear transmission and the propeller, can produce annoying noise if their components such as gearwheels and blades are not properly designed and adapted to the operating conditions of the ship and the properties of the prime movers.

Whilst vibration and noise were earlier a comfort issue paid attention to primarily in the cruise-ship industry, these characteristics have now become important for most vessel types. It has been realised that discomfort and rest quality on board influence the work quality of the crew and as such noise/vibration levels are important for safety and health.

In addition, the propeller is an important source of vibratory loads that influence the lifetime of mechanical elements in the drive train. This is particularly important for azimuth propulsor systems where the distance between the two excitation sources (propeller and prime mover), is short compared to conventional propellers with long shaftlines. This leads to different mass and stiffness properties which affect the response in the drive train from the excitation sources.

The optimisation of the propulsion plant and the propulsor itself occurs with different perspectives and at different levels. Compromises are made between conflicting demands such as budgetary issues, the need to improve operational efficiency, redundancy/safety and comfort.

A challenge in the decision making process is to assign a value to the parameters involved in the optimisation and to assess the adequate method or tool to be used to obtain the actual figures. Should empirical/analytical based estimates, new experiments, computations or a combination of these methods be applied, and what is the margin of error for the case being considered?

A focus of the paper is that the propulsor interacts with the hull, machinery and "the environment" in various ways, and each aspect of the interactive systems involved needs proper understanding to develop efficienct and smooth-running propulsion and manoeuvring systems.

2 COMPUTATIONS, MODEL TESTS AND FULL SCALE

In many areas of propulsor design and hull integration optimisation, the question arises about the adequacy of tools and methods available. From model testing, empirical/analytical tools and computational models, what is appropriate for the case in question? The choice of (or combination of) method(s) depend on several factors. The level of detail required, the number of variables involved in the optimisation and its purpose. Different methods and tools may be required, for instance, in the design the propeller itself and those required to provide information about the effect of the propulsor on the design of the afterbody. The choice of method will also often depend on the level of accuracy required and on the time available.

The "novelty" of the project and the specific requirements form part of these considerations. In main propulsion arrangements, propulsors are selected and fine tuned to the unique characteristics of the ship they are to move. Hence, the properties of the hull itself and the associated non-uniformity of the propulsor inflow, the vessel's operation profile as well as the properties and constraints of the prime mover must be evaluated.

In new projects, where these factors are within the reference frame of the designer's experience, well established methods can be applied to develop, for instance, a properly designed CP-blade, provided it is handled by qualified and experienced people who are able to interpret and correct the results.

In the other case (when demanding performance requirements and new features are introduced in the hull or machinery or a combination thereof), new experiments or more advanced computations may have to be applied.

Over the past decade, there has been a shift towards much greater use of computations and physical-based simulations within the field of propulsor design and development as well.

With regard to cavitating propulsor flows, the MPUF code is widely used in Rolls-Royce in the iterative design process when new blade designs for open propellers are being developed. This method solves the unsteady potential flow around a cavitating propeller. It is based on a vortex-lattice method which was introduced for predicting wetted unsteady propeller performance by (Kerwin and Lee 1978) and was further developed by (Breslin et al 1982). The most recent version MPUF-3A, (H.Lei et al 2007), has undergone several improvements and is now a robust solver that predicts the extent and volume dynamics of unsteady sheet cavitation on the suction and pressure side fairly well in most cases.

When more complex cavitation phenomena are expected which may be critical for particular performance criteria and durability, such as erosive cavitation or acoustic emissions generated by certain flow patterns with cavitation, model experiments and full scale are still need to be applied in the learning and development process.

Nevertheless, ongoing studies on computational modelling of tip vortex cavitation and re-entrant cavitating jets in the vicinity of the blade surface appear to be promising. But longer term research will be required before such codes can be included in application engineering.

Viscous flow phenomena are important in many cases. For the past 10-15 years, methods that involve solvers based on RANSE (Reynolds-Averaged- Navier-Stokes-Equations) have become an integral part of propulsor development and research in Rolls-Royce. Compared to potential flow solvers, these methods (hereafter CFD for brevity), are much more computer demanding but offer more detail when necessary to explore flow phenomena that are not well known.

CFD in the Rolls-Royce marine business is applied in many areas in combination with experiments. These range from the detailed level such as studying how flow and performance are influenced by the shape of a CP hub, the effect of varying the lip shape of a waterjet inlet to the investigation of the integral performance of propellers and rudders. Examples are presented by (Seil 2003), (Arén & Lundberg 2008) and (Petterson & Nerland 2007).

CFD is also an indispensable tool for evaluating Reynolds number dependent scale effects on the propeller (Stanier 1998) and (Müller & Maksoud 2005). In addition to improvements in performance at design condition, CFD can also be beneficial in estimating steady and unsteady loads in off-design conditions such as azimuth propulsors in oblique flow or CP propellers at off-design pitch settings and advance ratios.

With regard to exploring the factors that influence on high level system design and optimisation; there are shortcomings in both model experiments and computations. This concerns phenomena that contain coupling between various physical mechanisms in the interface in the dynamics of hull- and propulsors and between the latter and the mechanical response in the drive-train including the prime mover.

One example relates to the coupling between various cavitation phenomena and corresponding acoustic emissions. Tip vortex cavitation, for example, may be critical because it can generate low-frequency broadband hull pressure fluctuations. Although the magnitude of these excitations may be low, they can lead to resonant vibrations which often cause annoyance.

Lack of insight into the underlying physical mechanisms and a corresponding lack of theoretical models have led to the development of semi-empirical methods that estimate inboard noise based on the tip vortex characteristics (e.g. (Ræstad 1996)). This topic is an inherent part of the discussions dealt with in Sec.4.1.

Another example is the relationship between dynamic course stability effects and the integrated propulsion power demand. In experiments, steering/-manoeuvrability and self-propulsion are separately tested to provide results for different purposes. In the latter, steering devices are fixed in their neutral position, but this is never the case in reality. In fact, the degree of course-instability affects the rudder- (or azimuth propulsor) angle dynamics, which again influence on the propulsion power demand. This important interlink is illustrated in case study 4.2.

A third example, is the coupling between transient response to hydrodynamic loads such as those caused by propulsor out of water effects. In model scale, it is held that these transient excitation forces can be adequately extended to full scale provided that the most important dynamic similarity conditions are complied with. But the response in the transmission system and the properties of the electric motor providing the torque will be different at full scale.

Full scale monitoring surveys of real in-service operations have traditionally been costly. But advances in sensor technology and general development in IT equipment, including data storage and transfer systems, have made full scale measurements more cost-effective. In Sec. 4.3 the features of simultaneously measured time histories of hull motions and associated propulsor response to quartering sea wave encounter fluctuations are discussed.

It is important to consider the strengths and the shortcomings of computational models, scale experiments and full scale measurements but at the same time to exploit the synergies between them.

3 MAIN PROPULSION ARRANGEMENTS

The choice of main propulsion arrangement comes early in any ship design process. It involves numerous considerations that affect investment costs, running costs, redundancy and reliability, in addition to the propulsive efficiency and manoeuvrability aspects both in transit and low speed. Some of the high level considerations are :

- Single, twin, triple or quadruple screw configurations including hybrid arrangement, i.e. combination of conventional shaft propellers and azimuth systems.
- Shaftline propellers and rudders or Azimuth propulsors.
- Controllable Pitch Propellers (CPP) or Fixed pitch Propellers (FPP) ,
- Open versus ducted systems.

There are several factors that influence the selected option (a discussion that goes far beyond the scope of this paper). A performance gain, however, is always a prime criterion whether it is regarding low speed manoeuvrability or transit speed performance.

As a manufacturer of all the thrust- and steering devices and transmission systems required to compose the abovementioned propulsion arrangements, some trends are put forward and the basis for selection of arrangement are discussed in the light of our experience and understanding of the mission profile of the vessel.

For all vessel types, the number of propulsors is closely related to the demanded speed and/or the towing pull which again is linked to the machinery selection and power requirements.

The shape of the ship's stern and the lines are to a large extent determined by the propulsor choice. A most important factor in minimising excitations from the blades is to improve the quality of the incoming flow (or wake) as much as possible. This is extremely important in achieving a ship with low vibrations transmitted to the hull, shafts and other mechanical elements since unsteady blade loading and transient cavitation is strongly influenced by the wake field.

Single screw vessels have normally a more uneven inflow to the propeller compared to twin screw conventional shafted systems where brackets and shafts are the main upstream components that lead to a disturbance of the flow. The degree of non-uniformity in the wake field is also important as it may affect design compromises such as the desire to improve propulsive efficiency and at the same time minimise noise and vibration excitations. This is among other things linked to the size selection (i.e. diameter) and the blade design.

Azimuth propulsion systems are becoming more powerful and their potential in the main propulsion role has increased during the last two decades. These propulsors can be divided into two main classes:

- the so-called propulsion pod where the electric motor (enclosed in a pod) drives the propeller through a short shaft.

- The azimuth thrusters where the prime mover is located within the hull and the propeller is driven through shafts including one or two sets of spiral bevel gears (SBG).

Apart from the size and the power transmitted by the two systems, the design challenges are much the same in a hydrodynamic perspective. Both types of azimuth propulsors can either be arranged in pushing mode, with the vertical leg upstream of the propeller, or in pulling mode, with the leg/pod downstream of the propeller (when operating in normal ahead mode). Both systems can also include a ducted propeller, but only in pushing mode due to the excessive steering torque required for ducted azimuth thrusters in pulling mode.

Whilst the first system requires diesel-electric machinery, the second can use both diesel-direct and electric driven systems and they can also be equipped with either CP or FP propellers.

The pod systems have so far mainly been applied to large cruise liners but they have also a potential for other ship types. In (Mewis 2001) the suitability of electric pod propulsors for various ship types is discussed.

The azimuth propulsors that include gears are nowadays used on a variety of vessels. These have a unit power range from typically 0.8 to 8 MW, and the ones in the lower power range are also arranged as retractable or swing-up devices that can be moved into the hull when not being used. These are auxiliary or low speed manoeuvring devices and will not be dealt with here.

Different propulsion systems are favoured for various vessel types, for example:

- **Cruise-/passenger ferries**. This segment covers a wide span in size and speed. During the 1990s, many of the large cruise vessels were equipped with propulsion pods but during recent years, although there has been a general decline in new-buildings, part of this market has returned to the traditional twin screw CP Propeller arrangement with rudders.

- **Yachts**. Twin screw with inclined shafts is a common solution for speeds below 30 knots overlapping with waterjets which are mainly applied for speeds over 25 knots. Some yachts have latterly also been ordered with twin pulling azimuth propulsors driven through gears.

- **Tankers.** Large ships have typically single direct drive low speed diesel engine and FP propeller. Shuttle tankers are usually equipped with twin CP propellers and rudders. Various propulsion arrangements are available where a RP notation (Redundant Propulsion) is stipulated. Smaller product and coastal tankers are also diverse with regard to propulsion arrangements and systems. The latter operate frequently in harbours and restricted areas. Some have therefore moved to twin azimuth thruster main propulsion.

Whilst the ship types mentioned above are diversified with regard to the systems that transfer engine power to propulsive and manoeuvring forces, in the following classes of vessels, one type of propulsor or arrangement dominates. In the first four vessel categories below, azimuth thrusters are dominant:

- **Semi-submersibles rigs and FPS(O)s**.

  Azimuth thrusters with ducted propellers are the major system in this market since their main role is to generate omni-directional thrust at low speed for position keeping purposes. Apart from a few cases with pods, azimuth thrusters dominate this market. Power range is typically 3.5.-5.5 MW per unit with normally 4-8 units per vessel.

- **Double ended Ferries**.

  Almost 100% are equipped with azimuth thrusters. In the most common speed range, 12-15 knots, with one unit at each end with various kind of pushing or hybrid (push/pull) azimuth systems. These include contra-rotating propellers on azimuth thrusters in pushing mode. The latter have high efficiency for the relevant speed range. Among those operating at higher speeds, there are five catamaran hulls equipped with four pulling azimuth thrusters on each ferry (one at each end of the demi-hull). These operate with 21-22 knots service speeds.

- **Platform supply, Well-intervention, Ice-breaking and Subsea construction vessels**.

  Several vessels of this type have recently entered the market and the majority has twin azimuth main propulsor arrangement of various kinds. Power range is typically 2 to 7 MW per unit.

- **Tugs.**

  Twin azimuth thruster systems with FP propellers rotating in nozzles are major solutions. Also many twin screw/rudder systems and several cycloidal propeller tractor tugs.

- **Anchor handling and multirole towing, Seismic survey and Coastguard/ Emergency towing vessels.**

  These vessels are dominated by conventional twin screw shaft systems with CP propellers designed to rotate in a nozzle and often with a flap rudder behind each propeller/nozzle. For the offshore vessels developed for various heavy towing duties such as rig-moves, anchor handling and “trenching”, the bollard pull figure is an important competitive factor.

  Figure 1 present the statistics of measured full scale pull versus power data for several vessel in this category. The statistics include the most powerful offshore towing vessels in operation and show the specific pull in N/W (Newton per Watt) versus the specific disc power (kW per area of a disc of diameter equal to the propeller). This is based on the recorded engine output. The linear regression fitted to the measurement reflects the gradual reduction in the ability to transfer power into thrust as power density increases.

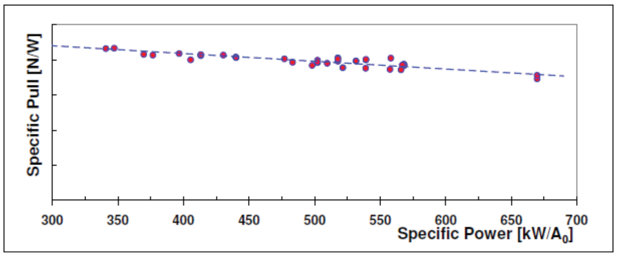

Figure 1. Specific bollard pull versus specific propeller disc power.

Considering the ones with highest energy-flux and thrust-load coefficient (which is infinite by definition in the bollard pull case), there are indeed challenges involved. The shape of the aft body, the individual clearances between nozzle and hull and between the propeller and rudder as well as the detailed design of each component must be carefully evaluated to obtain the specified criteria and at the same time to avoid any detrimental effect from cavitation.

A typical high performance towing vessel (anchor handler) is fitted with a twin screw installation coupled to Twin Input Single Output (TISO) gears and two plus two engines, often in a father and son configuration per gearbox Power at each propeller shaft is typically in the range 7-10 MW.

Transmission systems and interactions

For most ships below 10,000dwt, the shaftlines are now combined with a reduction gearbox and one or more prime movers being medium speed diesel engines or electric motors. These gearboxes are of the parallel shaft type with cylindrical helical gears. One of the positive features with a shaft line is the rather low torsional stiffness of the connecting element between the propeller and the gearbox/prime mover. This is of great benefit to the fatigue loading of the gearwheels and also to the shaft line itself. When the ship is operating in rough weather and waves, high transient loads occur when the propellers are ventilating and re-entering into the sea. These loads are dampened out by the propeller shaft acting like a huge torsion spring.

The basis for azimuth thruster development was the production of high quality case hardened Spiral Bevel Gears (SBG). Today various types, sizes and configurations exist of this type of thruster. The normal type is the pusher type with nozzle which was originally designed for low speed manoeuvring only, but newer versions having more slender bodies and nozzle shapes, are fitted to tugs that occasionally operate at speeds up to 14 knots.

Typical configurations are the Z-type, where there is a top gearbox with horizontal input shaft, or L-type where the input is vertical with an electric motor. When there is a top-gear, the prime mover can be either an electric motor or a diesel engine.

One of the significant advantages of the azimuth thruster is the enhanced manoeuvrability both in manual mode and in DP mode, meaning for example that offshore support vessels can operate with improved manoeuvring accuracy and under rough weather conditions.

Some years ago, the pulling (tractor) azimuth thruster was developed by Roll-Royce partly based on a non-azimuthing geared high speed propulsor with pulling propeller (Halstensen & Leivdal 1990). By its nature the propeller is entering undisturbed water and therefore can be designed for significantly better performance than when working behind a gear housing, especially as speed increases. The result is better performance and reduced vibration and noise compared with a pushing thruster or conventional shafting system and rudders.

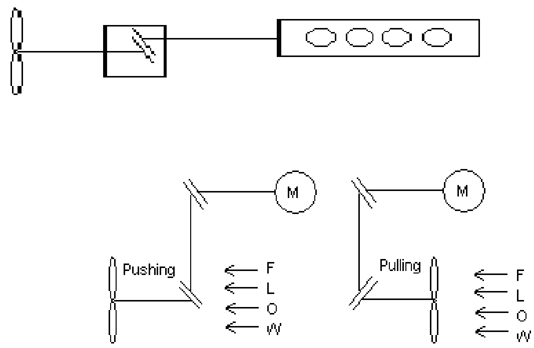

Figure 2.

Sketch illustrating common transmission drive trains. Above. Shaftline system with single-in single out gearbox. Below. Azimuth thrusters with Spiral Bevel Gears (SBG).

When it comes to dynamic response in the transmission line, there are some generic differences between an azimuth thruster and a shaft line with an inboard parallel shaft gearbox. These are small in calm water, but can be significant in a seaway.

The tougher the weather the more the ship is heaving, rolling, yawing and pitching, and the more often the propellers suffer from ventilation and out of water effects.

Measurements in model and full scale have shown that the above mentioned effects can lead to high transient blade loads that are transferred into the shaft. These loads may be the same if the ship has a propulsion system with shaft lines or is fitted with azimuth thrusters. But the responses to these loads are different.

A ship with an installed power of 4000kW and fitted with an 11 m long propeller shaft line system, the torsional stiffness will be about 12 E6 Nm/rad. On an azimuth thruster with the same propeller shaft diameter but length only 1.7m, the stiffness is 190 E6 Nm/rad. The stiffness ratio is some 15 times higher for the propeller shaft of the thruster. The masses of the two systems are also different but the masses operate in favour of the long shaft, damping the transient excitations from the propulsor. The implication of the above is that any load impact coming from the propeller may be more severe in a thruster transmission than the same load magnitude in a shaft line transmission including the gearbox.

There are also dynamic loads coming from the prime mover, but these loads are small, predictable and have traditionally been covered by torque multiplier factors. These factors have been revised as a result of improved understanding of combined propulsor and prime mover excitations/response. Normally there is an elastic coupling between the diesel engine and the input shaft or shaftline reducing the dynamic response to engine excitations, but if impact loads come from the propulsor end of the transmission an elastic coupling is of little help.

Waterjets are also prone to partial or full air ingestion. Prevalence and frequency depend strongly on hull design, wave conditions and heading angle compared to wave direction. The effect can be a rapid change of impeller torque. As air enters the inlet duct, the torque drops quickly down to almost zero, and stays there until the air supply is cut off. It then takes some time for the flow to recover. Air is trapped in the upper part of the inlet duct, and evacuated in portions, causing a fluctuation in input torque. Fig. 3 from (Aartojärvi & Häger 2001) shows a typical example with a sudden load reduction, some time at very low load and thereafter rising torque with a more vibratory pattern. Shaftline components and engine are subjected to rapid changes in torque, and the engine control system must cope with this.

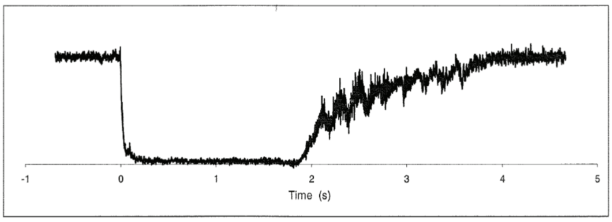

Figure 3. Measured torque on the shaft of an experimental waterjet impeller setup during a ventilation event.

Thruster ventilation has similar effects on other parts of its transmission system. Through Rolls-Royce University Technology Centre research (UTC) in Trondheim, the effect of ventilation on thruster blade and shaft dynamics has been studied. The time history in Figure 3 bears a resemblance to similar ventilation episodes with thrusters as found in (Koushan 2006). Just after ventilation inception a sudden drop is often seen in the thrust and torque signal which is followed with a gradual rise in the load at the same often including some excessive blade frequency dynamics during the torque/thrust recovery phase.

Regarding gear design, all classification society rules to day are based on various ISO standards for gear sizing (or derivatives of ISO standards).

Rolls-Royce uses the BECAL software (Baumann 1998) SBG design and analysis. Allowable Hertzian stress is in the region of 1050-1150 MPa, with the safety factor of 1.2-1.4, giving a safety factor in torque of 1.44–1.96. Allowable bending stress in the root fillet is in the region of 450-550 MPa with safety factor of 1.55-1.8.

There are various types of failure modes on gear-wheels.

Scuffing is a sort of frictional welding between the mating gear flanks that gives severe and irreversible damage to the flank surface. Some years ago scuffing was a gear damage normally caused by severe overload, manufacturing inaccuracies, elastic misalignment, wrong gear oil or combinations thereof.

Some classification societies have requirements when it comes to scuffing. The calculated safety factor towards scuffing is around 1.5 as a minimum (DNV Class Notes 2003). For gearboxes with cylindrical helical gearing the incidence of gear damage is very low.

When it comes to thruster drives and spiral bevel gears, the overall gear damage situation is more complex. Although the majority of azimuth thrusters operate trouble-free on most vessels, there are certain cases where damage may occur. The SBG-gear within the thruster-body is very close to the propeller. Consequently the load regime of the propeller blades is more or less copied into the drive train.

Example of a crown wheel with scuffing marks on six teeth in four distinct sectors is shown in Figure 4.

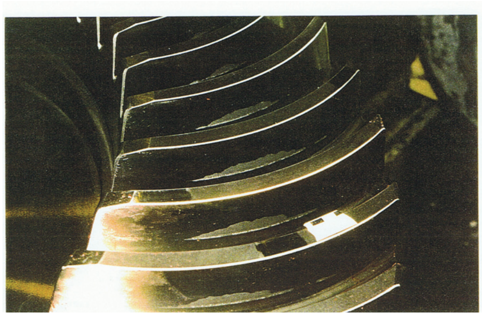

Figure 4. Picture of scuffing damage on six teeth of a crown wheel of diameter 1060 mm.

This was a pushing azimuth thruster with a four bladed CP propeller designed to operate at low speed. The four damaged areas on the gear-wheel corresponded to the angular sectors where the blades sweep through the region of reduced flow downstream of the vertical leg. There is some evidence suggesting this damage occurred during a long transit at high power. As opposed to the low speed manoeuvring case where the inflow is induced by the propeller and nozzle, the effect of the uneven inflow due to the leg will increase with the speed.

Knowing that the scuffing capacity of modern gear oil is very high, the torque necessary to produce scuffing is in the range of 1.5-2.5 times nominal load.

Pitting failure is very rare in SBG drives. Sometimes when there is scuffing damage, pitting is experienced as secondary damage in the scuffed area.

The most frequent failure today however, is TIFF (Tooth Interior Fatigue Fracture). This failure mode was identified in the automotive industry on the gears in a series of trucks (MackAldener 2001). The initial damage starts sub-surface, normally in the transition zone between hardening layer and core material and the result is that the upper part of the tooth is lost. This failure mode is not covered in any standard like DIN, ISO or AGMA, and at present there exists no rating method to estimate the risk of such failure.

An example of TIFF has been seen on broad beamed vessels that have the thrusters fitted at the very aft corners. It is well known that these thrusters suffer from out of water loads. Often only one tooth is affected, indicating that the transient load has been rare, of short duration, and high.

4 CASE STUDIES

4.1 Noise and vibration control - Cruise ferry with several operational modes

The shipping industry is setting ever higher standards for comfort on board vessels. The requirements may being either those set by the classification societies or the special needs of a shipowner. Recent Scandinavian cruise ferries have achieved very high standards under challenging conditions.

A large cruise ship with car decks operates on a 360 nautical mile route. Half of the distance is in shallow water and both port approaches are quite narrow. Entering one harbour stern first is the most challenging manoeuvre of the route. When the vessel was being planned several propulsion systems were considered, but for a variety of reasons the owner settled for a conventional twin screw propulsion system with 4-blade CP propellers. The propulsion machinery has three operating modes: cruising, manoeuvring and port. Design of the propellers was complicated by the fact that the shipyard and owner wanted to reach high Comfort Class level under normal operating conditions both in deep and shallow water and in various engine modes. With four engines this could be two engines on each shaft or one engine per shaft or an asymmetric one plus two. Achieving the guaranteed level of propeller induced hull pressures in one plus one, and asymmetric one plus two modes (total 67% power), was particularly difficult due to the risk of pressure side cavitation on the propellers turning at high speed with reduced pitch. The requirement was DnV Comfort Class COMF-V(1) on 2+2ME and 85% power, implying a noise level below 55dB(A) in public spaces and 44 to 49dB(A) in cabins. In terms of vibration it corresponded to less than 1.5mm/s in the passenger accommodation. The task was made more difficult still because the main restaurant with panorama windows is located right at the stern of the vessel just above the car decks and in the most difficult region to achieve low noise and vibration levels with conventional propulsion systems.

The shipyard had developed a 'wave damping afterbody', using CFD methods to evaluate several hundred candidate shapes, that modified the wave profile along the hull, reducing generated waves and also the power required. From the propeller design standpoint a positive feature was the large propeller to hull clearance available: 40% of propeller diameter. Diameter was limited to 5.2m by the risk of air drawing.

The propellers excite the hull in different ways. The low frequency excitation is felt as vibration and the higher frequencies as noise. The excitation can be divided into two different types:

Fluctuating forces and moments transferred from propellers to the ship

- via the shaft system (1^{st} order of blade harmonics).
- Pressure fluctuations transferred to the hull through the water.

The latter can be divided into:

- Pressures generated without cavitation
- Pressures generated by cavitation on the blades
- Pressures generated by the cavitating tip vortex. For a propeller for this type of ship this is the most important factor in noise generation.

In the 2+2 ME mode the sheet cavitation on the suction side as well as the strength of the tip vortex should be kept to a minimum. In addition to the volume of the blade itself, sheet cavitation is the main source for pressure pulses normally at 1^{st} to 4^{th} blade harmonics. These blade harmonics create vibrations in the hull structure at the corresponding frequencies. The suction side tip vortex creates a broadband noise on board, normally in the frequency range between 4^{th} to 8^{th} order of the blade frequency. For the vessel under consideration this is 35 to 70 Hz. In the 1 ME mode some pressure side cavitation is normally allowed, however it should be kept to a minimum. If this cavitation is excessive, the noise in the same frequency range as for the tip vortex will increase. This cavitation can also be erosive for the blade surface. In shallow water in the 2+2 ME mode, the suction side sheet cavitation will increase. In the 1 ME mode in shallow water any pressure side cavitation will be less compared with operation in deep water. The P_D-n diagram is useful in assessing the limiting conditions predicted for the critical cavitation inceptions with regard to noise and erosion. In addition, the constraints of the engines with regard to load and rpm range can be included in the same diagram. The diagram is therefore also a useful base for the design of the combinator (i.e. the combined pitch-RPM controls).

Figure 5 shows an example of a P_D-n diagram when the ship operates in shallow water. The resistance of the hull is increased in shallow water operation, the ship speed is reduced and a lower propeller pitch is required for the same power compared with deep water conditions. The propeller shaft power demand curves corresponding to the chosen combinators for the 1 ME and 2 ME modes of operation are included in the diagram.

In addition to the selection of the design point, the radial and chord-wise load distribution and the detailed design of the leading edge of the blade profile is optimised to achieve good cavitation performance at the different operating conditions while at the same time efficiency is kept at the high level.

If the load at the blade tip is reduced, the cavitation, and consequently the vibration and noise, in the 2+2 ME mode will be reduced. A reduced load at the blade tip will also reduce the efficiency of the propeller.

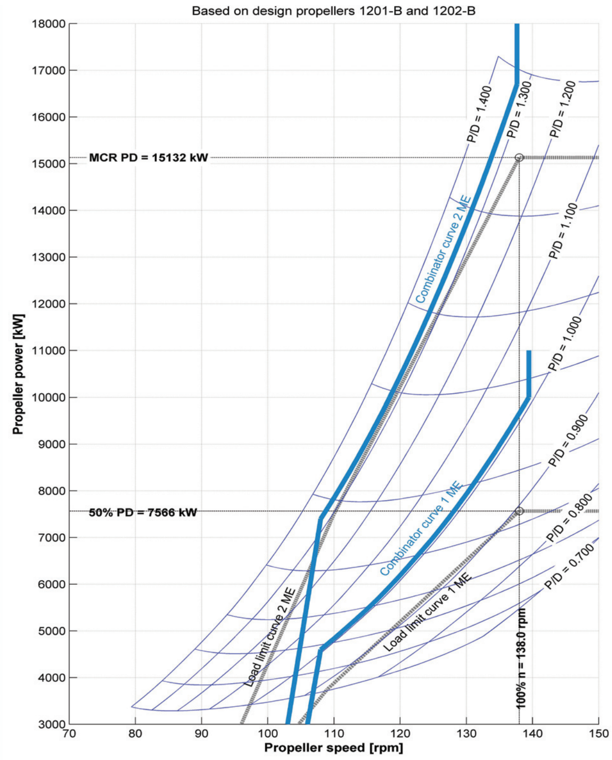

Figure 5. P_D-n diagram for shallow water and trial condition including combinator curves for 2ME operation and 1ME operation.

Extended model tests were carried out to obtain the best possible hull lines and appendages for deep and shallow water. Wake measurements, propulsion and cavitation tests were made. The cavitation pattern on the propeller was video recorded and the propeller-induced hull pressures measured at three different conditions.

During the sea trial, propeller induced hull pressures were recorded by means of transducers installed in the hull above the propeller. Cavitation observations were made. The propeller was observed in stroboscopic light through windows installed in the hull plating above the propellers, with the boroscope technique used for video recording. At 2+2 ME 85% power, the cavitation consisted of only a thin suction side tip vortex as the blade passes through the wake peak. A sketch of the vortex cavitation pattern is shown in Figure 6 and a corresponding picture in Fig. 7.

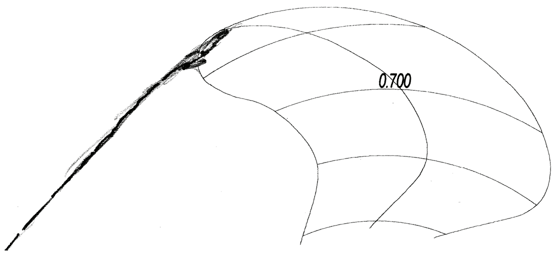

Figure 6. Sketch of full scale cavitation pattern at 2+2 ME 85% MCR.

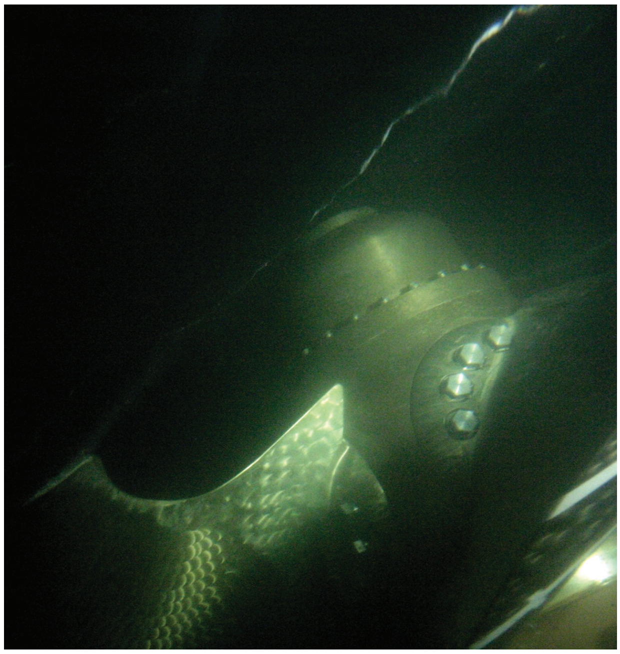

Figure 7. Full scale cavitation observations. Note the thin tip vortex. Condition: 2+2 ME 85% MCR

4.2 Powering performance and course-stability - offshore service vessel in calm water.

Rolls-Royce developed a new azimuth thruster in a pulling propeller configuration based on mechanical elements from its well established thruster range (Vartdal et al 1999). This propulsor can be equipped with either FP or CP propellers on a hub located upstream of a slim underwater body with a profiled skeg below a torpedo shaped part enclosing the lower spiral bevel gear. This shape leads to a high effective aspect ratio which is important to obtain good “rudder efficiency”.

This azimuth propulsor was designed to operate efficiently in speed ranges above those for the conventional pushing azimuth thrusters. The first size was used in a series of three 22 knot catamaran double ended ferries (Vartdal & Bloch 2001).

The offshore industry expressed interest in the concept for supply boats and in 2003, when larger size of the system went into production, the first two units were retrofitted to an offshore supply vessel. This ship was equipped with conventional pushing azimuth units from the beginning and the vessel had been in service since summer 2002.

We had valuable co-operation with the shipowner, who placed the ship at our disposal for two days, one before and one after the conversion. It was possible to run extensive tests over the same measured mile and luckily under essentially the same ideal calm weather conditions as the first sea trial with the pushing azimuth units. The two trials took place 10 days apart in October 2003.

A comprehensive measurement survey was planned with the intention of obtaining a deeper insight into the powering, speed and manoeuvring performance aspects relating to the two different azimuth propulsor systems.

The actual hull had a length over all of almost 90m, beam waterline 18.8m, and for both trials the draft was adjusted to 4.3m at even keel.

Each of the two surveys included the following tests.

- Speed-power runs with simultaneous recorded torque and rpm (by strain gauges, also vibratory torque), azimuth feedback and 6 DOF hull motions.
- Manoeuvring tests including zig-zag, pull out, crash stop and turning circle tests.
- In all these tests, structural vibration accelerations were recorded at different points close to the steering gear and at the lower support bearing of the steering column.

Samples of seawater temperature and salinity were taken at each trial but differences were insignificant. Wind direction and speed were also continuously logged in both occasions, but it was almost completely calm, just some light gusts now and then. Therefore, any differences in the results between the two systems were not likely to be related to any environmental factor.

The diameters of the CP propellers were the same for the pushing and pulling system as also were the shaft revolution speeds (same transmission and gear size).

The blade geometries, however, were different, with the first set of blades adapted to operate in the inflow disturbed by the vertical leg. Therefore, those blades had moderate skew-back with slightly higher area ratio compared to the blades for the pulling azimuth units. The latter were designed to work in a much cleaner inflow and, as opposed to the first blade set, the pulling propeller blades had forward rake to increase the distance from their outer part to the leg. The pulling propeller will operate in a wake having only slight retardation due to the hull boundary layer.

During dry docking when the propulsor retrofit work was carried out, the surface roughness on the blades of the pushing unit was measured on several points in order to be able to give an estimate of its influence on the efficiency. The new blade sets for the pulling propellers had a surface smoothness according to ISO R484 Class I requirement (which also was the delivered manufacturing class and corresponding smoothness for the first set of blades).

The blade surface roughness related to Ra, and the corresponding “Rubert gauge” (Molland 2008), was used along with data from (Phoenix Marine 2003) to provide an indication of the efficiency loss due to surface roughness on the blades of the pushing azimuth thruster.

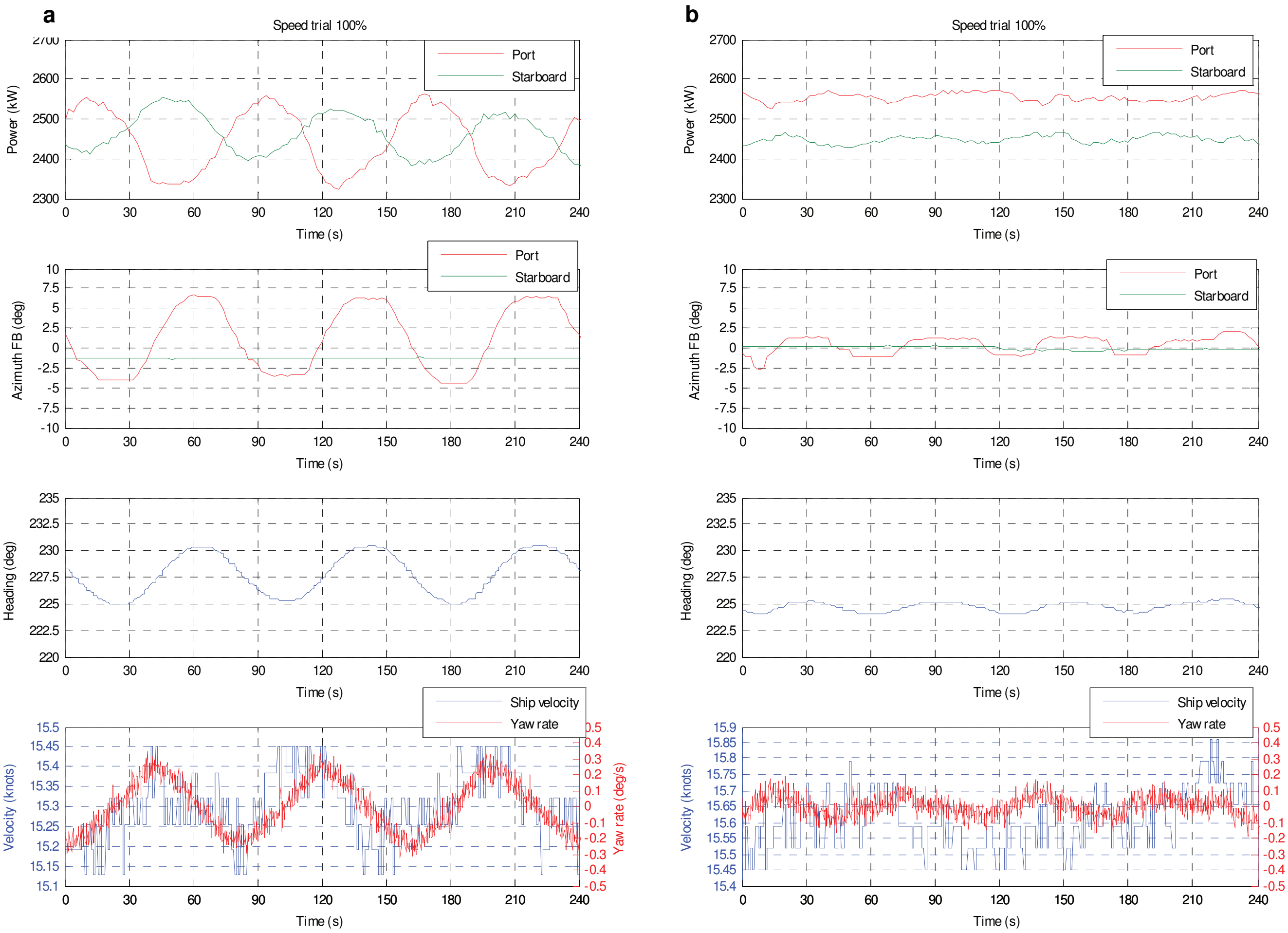

Figure 8 a) Time series from speed trial with pushing thrusters, b) Time series from speed trial with pulling thrusters. From top, the following variables are depicted: Shaft power, azimuth angle, Heading, Yaw rate and ship speed (DGPS).

The hull was not cleaned during the drydocking, apart from a limited surface around the thruster interface area

There were no model scale self-propulsion experiments performed with the hull and actual propeller systems.

An estimate was made of the power-speed prognosis founded on assumptions about interaction coefficients and differences in open water efficiency based on model tests performed with the pulling azimuth unit, and similar stock propeller efficiency data for an equal shaped pushing azimuth propulsor. This exercise was redone recently as more data from open water and propulsion tests with similar shaped afterbodies, propellers and thruster-shapes have become available. The latter predictions led to slightly different results.

But the measured power versus speed data proved to give a reduction in propulsion power demand at same speed amounting to 19-21% ! This conclusion could be drawn on the basis of three double runs on a measured mile with each azimuth propulsor system at approx. 50, 75 and 100% power.

The time history of some central variables are compared in Figure 8 for the two systems based on measurements from the southbound speed runs at 100% power.

In both trials, the starboard unit was fixed in its neutral azimuth position whilst the port side unit was under autopilot control. The autopilot settings remained the same in both trials.

The most noteworthy observation is that the hull performs yawing fluctuations with double amplitudes of 5 degrees with the pushing azimuth units, whilst yaw motions with the pulling thrusters were reduced to below 2 degrees.

The corresponding steering response is also much higher with the pushing units. While the latter perform alternating azimuth rotations between 4 to 7 deg. to keep the course, the pulling azimuth units requires normally less than 2 deg. and the mean course is maintained with much less yawing.

The yaw rate for the hull when steered by the pushing units, oscillate ±0.3 deg. per second whilst it is reduced to ±0.1 deg/s with the pulling azimuth units.

From the same figure it is seen that the speed oscillations tend to coincide with the yawing speed and this is more evident for the pushing than for the pulling thruster units.

It is obvious that these factors contribute to reduced efficiency and a corresponding speed loss.

The tradition in propulsor design is to evaluate the nominal wake measurements in model scale based on a straight course. Likewise, the effective wake and other hull interaction factors applied are also derived from self-propulsion model tests under the same idealised condition for a straight course with fixed steering devices.

The propulsor efficiency considering the effect of oblique inflow, based on available thrust and normal force data (in the coordinate system of the propulsor), can be converted to a force component along the mean heading axis (x) for the vessel. This efficiency may be expressed as:

$$\eta_x(J_a,\alpha) \sim c \cdot \frac{K_X(\alpha,J_a)}{K_Q(\alpha,J_a)}$$

J_a advance ratio.

α angle offset propeller shaft vs. the onset flow direction

$K_x(\alpha,J_a)$ Propulsor unit thrust coefficient .

$K_Q(\alpha,J_a)$ Shaft torque coefficient.

This means that both the thrust and torque is a function of inflow speed and direction. This dependence (on torque) is manifested through the fluctuations as seen in the uppermost time traces of Figure 8 (as pitch and rpm were kept the same through each speed run). However, the "behind efficiency" associated with the oscillating power and azimuth angles in Figure 8 can not be predicted. Of course, the actual inflow velocity and angle of attack seen by the propulsor (α) is not the same as the measured azimuth feedback also because it is affected by the hull. In addition the time-varying effective inflow (and J_a) is generally unknown.

But let us assume that the mean effective inflow angle, α, is in the range 4-6 degrees for the run in Figure 8a, and applying open water data for a similar azimuth thruster as a function of α, it is estimated that the efficiency loss is 3-5% for the predicted mean advance ratio for the pushing azimuth units. Based on similar open water data with the pulling azimuth unit, which is less sensitive to changes in thrust and torque at small inflow offsets around zero, the efficiency loss for the corresponding smaller flow-incidence fluctuations is estimated to maximum 1%.

There is also another hydrodynamic energy loss component which varies between the two speed runs shown in Figure 8. This component is related to the difference in resistance due to the different yawing and sideslip dynamics of the hull. The flow incidence towards the hull itself increases the viscous resistance and also inertial force components arising from time rate-of-change in speed and yaw which comes in addition. These additional resistance components are also believed to be sensitive to hull shape and we have not attempted to quantify them here.

With regard to the aforementioned 19-21% reduced power demand with the pulling azimuth propulsors, there are of course uncertainties such as measurement errors, efficiency loss estimates due to blade surface roughness and assumptions about propulsion factors used and those relating to the open water model data (scaling and application of open water characteristics for geometries with slight deviation from the real full scale version).

Nevertheless, it can be concluded that the difference in the course-keeping dynamics caused by the two azimuth propulsors is significant and this difference has also a strong influence on the integrated propulsion power demand. This of course has a strong impact on emissions. In the particular case, feedback from actual in-service operations confirms that the annual fuel reduction is 16%

corresponding to a reduction of 35 tonnes NOx and 1700 tonnes CO_2 per year.

There are some interesting features which are most evident in the time series of Figure 8a, considering the fact that only the port azimuth thruster was used by the autopilot for steering whereas the starboard thruster was fixed. This means that power oscillations on the starboard unit can not be related to the change of azimuth angle. Then it has to be correlated to the motion of the vessel. We see that the power signals have opposite phase: when the power at the starboard thruster has its maximum, the power on the port unit is at minimum. There seems to be a link to the vessel's yaw rate. The power on the starboard thruster is increasing when the yaw rate is increasing and the power is decreasing when the yaw rate is decreasing. (Positive yaw rate means that the vessel turning with the bow towards starboard).

Knowing that torque/power normally increase as the advance ratio is reduced, an explanation to this observation is as follows: The power on the starboard thruster is increasing when the vessel is decelerating a port turn and starts accelerating to starboard. During the deceleration the inflow seen by the starboard unit has changed from the fairly undisturbed waters outside of the hull towards the centre line.

When turning to starboard, the port unit sees the inflow coming from the centre area where the flow is likely to be slowed down to a greater extent and hence the local advance number reduces.

On the contrary, when the vessel is about to change back from a starboard turn and start accelerate to port, the starboard thruster is experiencing a more undisturbed inflow from outside. Probably this causes the inflow velocity to increase, i.e. the advance number rises and consequently the propeller will absorb less power again.

To comprehend the power oscillations on the port unit, similar explanations apply as for the starboard unit, but the varying azimuth angle lead to additional flow dynamics. It may be seen that the power oscillates with slightly larger amplitudes on the port unit than for the starboard unit, with the power minima being slightly lower. This is likely to be caused by higher dynamics in the onset flow.

It was also observed that the period of the yaw motion in the different speed runs changed in accordance with the characteristics of vortex induced motions. This was consistent in both the three north- and southbound runs. In other words the yawing period was inversely proportional with the ship speed. Again, this was most notable on the pushing azimuth thruster as can be seen on Figure 9 for the three southbound runs. The yawing period for speed runs at 50, 70 and 100% power were respectively 118, 99 and 82 seconds.

A discussion of oscillatory ship motions is found in (Faltinsen 2005). It is mentioned that a characteristic time scale of the transverse ship velocity oscillations may be from half a minute to three minutes which is within the same time scale as observed in Figure 9.

During the trials, the most common standard IMO manoeuvres (International Maritime Organisation) were carried out so that manoeuvring performance of the two azimuth systems could be compared.

The results from a zig-zag 10/10 test and a zig-zag 20/20 test are presented in Figure 10a and b. The results from the zig-zag maneuvers are quite unique since the vessel, its loading conditions and the environmental conditions are the same. The only difference is the propulsors which in this case are also steering devices. As can be seen from Figure 10 there is a great difference in overshoot angles and so is also the time response to azimuth angle changes.

The 1st and 2nd overshoot angles on the zig-zag 10/10 test were respectively 94/64 degrees during the manoeuvre with the pushing thrusters and 14/24 degrees when the pulling thrusters were used. For the zig-zag 20/20 test, the 1st overshoot angle is respectively 54 deg. and 25 deg. with the pushing and pulling units.

The comparison clearly indicates that the side force generated by the pulling azimuth thrusters is substantially larger than that by the pushing thrusters. The difference is even more evident on the zig-zag 10/10 tests than on the zig-zag 20/20 tests.

From the aspect of safe ship operation, this study has demonstrated the impact of main propulsion azimuth thruster type on manoeuvring capabilities.

It should also be mentioned that course-stability is a property that is influenced by the combined features of the hull and the steering devices (and their interactions). A more straight-line stable hull than the one shown in this section is expected to give less difference in power demand, since less yaw-checking would be required to maintain the course.

Furthermore, it should also be mentioned that ducted pushing thrusters have better rudder effect than the similar units with open propellers as featured in this comparative study.

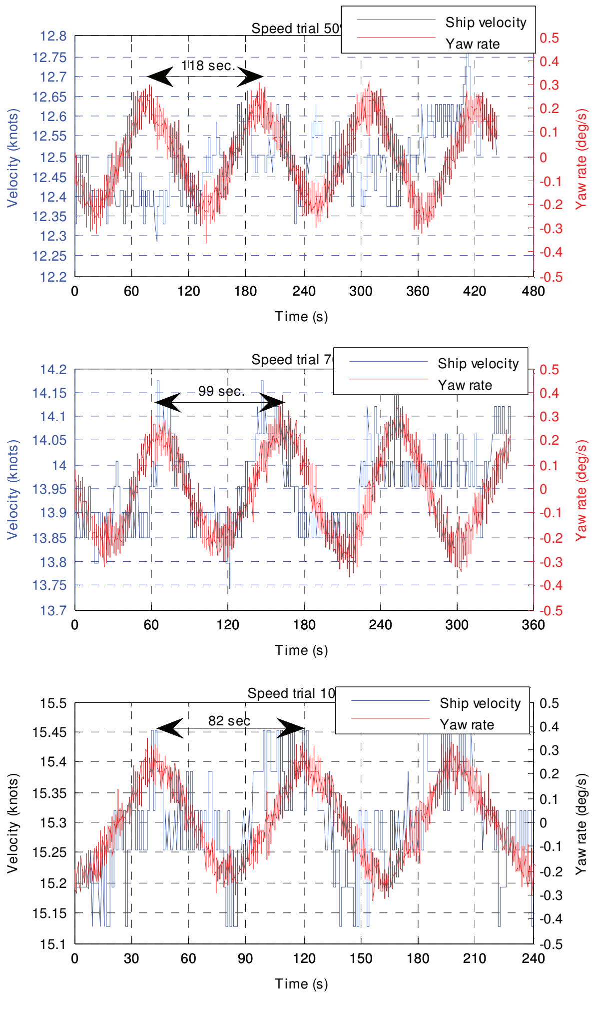

Figure 9. Change in yawing period with ship speed.

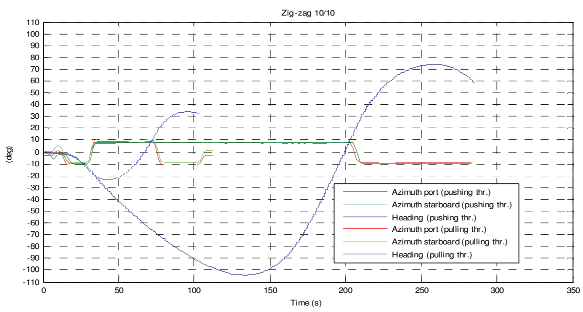

Figure 10 a) Results from zig-zag 10/10 test.

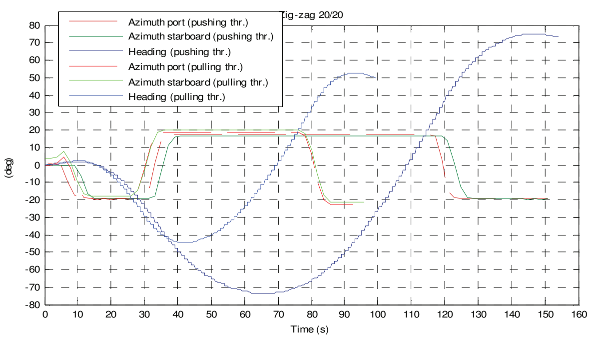

Figure 10 b) Results from a zig-zag 20/20 test.

4.3 Operational research- a propulsion in seaway case

The previous section demonstrate results from calm sea surveys that show the importance of having simultaneous measurements of multiple variables in order to be able to analyse and get insight into the physical effects that influence the relationship between speed and propulsion power demand. But real ship operations seldom takes place in such benign conditions as were experienced in the two trials described in Sec. 4.2.

Rolls-Royce has a system that records and transfers several variables from machinery and propulsors, including the hull motions. The system collects the data on a continuous basis during in-service operations for automatic transfer to shore. These data are very useful for various operational research aspects and they give insight into the coupled hull-propulsor-machinery behaviour in a seaway. Such information can serve as basis to improve system design, potentially to re-define design criteria and to improve control logic or develop more advanced control algorithms.

In the following, time histories will be shown for some key propulsor and hull motion variables during rough North Sea conditions on a vessel that is equipped with a prototype of the abovementioned system.

The case being considered here is stern quartering waves, regarded as one of the most critical conditions with regard to dynamic stability (Kat & Thomas 1998). Considering the size of the ship, the wave conditions were quite severe. Instrumental wave data from a fixed offshore installation about 50 km away indicated significant wave heights between 5 and 6 metres with mean wave period of 8 seconds. The actual measurements are taken on a platform supply vessel (PSV) having the following main particulars:

L_{oa}= 94 m, B=21 m, d_{max} =6.6m

The vessel is propelled and steered by the same type and size of pulling azimuth systems as those retrofitted to the other PSV presented in Sec. 4.2.

The event is shown in Figure 11 and covers a period of about 90 seconds. The variables depicted in the figure are, from the top (sampling frequency [Hz] in parentheses): Propulsion motor torque signal (0.5 Hz), Azimuth feedback angle (0.5 Hz), Heading and wind direction rel. to the ship (1.0 Hz), roll and pitch angle (1.0 Hz), Ship speed log (0.5 Hz) and wind velocity rel. to the ship (1 Hz).

The wind direction is about 260 degrees relative to the vessel, which means that the wind is directed towards the vessel's port side partly from aft (corrected for the vessel speed). No directional wave measurements are available for the location in question. However, since the wind was fairly strong and the combination of wave heights and wave periods indicate wind driven seas, the wave direction was supposedly close to the wind direction.

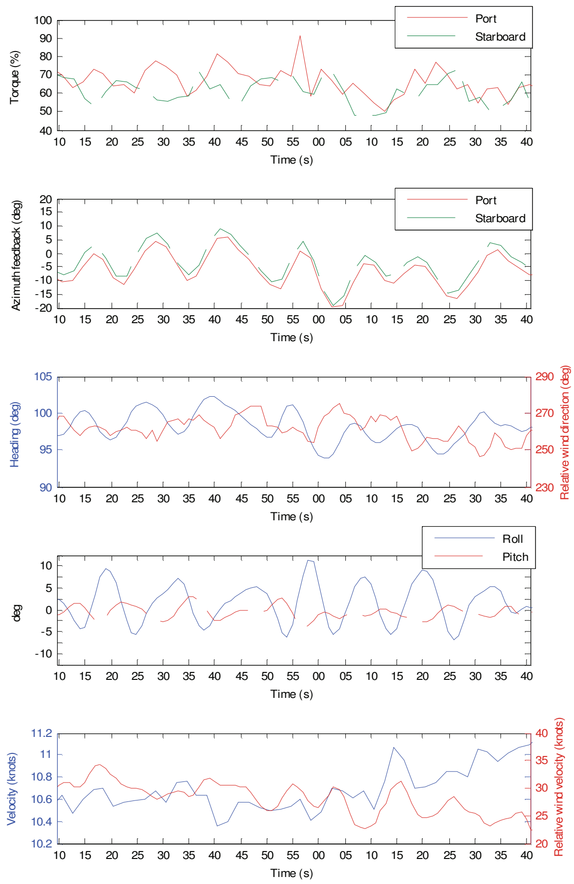

Figure 11. Propulsor and hull motion dynamics in a seaway with stern quartering waves.

The following features are observed in the time histories:

- It is evident that the dominating oscillations of most variables coincide with the wave encounter period.
- The vessel is steered by autopilot, and it is noted that both azimuth units control heading (also at the wave encounter).
- Both helm signals have a bias towards negative azimuth angles which means that the thrusters attempt to steer the bow to starboard.
- The occurrence of notable yaw and roll coupling in wave conditions like this is not uncommon. But it is uncertain to what extent the extensive azimuth angle activity influences on these motions. Within a short time span of about 5 seconds the heading has changed about 7 degrees. The cause-effect relations in these couplings are also unclear.
- The shaft torque oscillations are highest on the port side which corresponds to the windward and “wave-ward” side.
- It is noted that during the calm water speed trials (Sec. 4.2) with the same size pulling thrusters, the one steered by the autopilot was hardly used to correct the course, but in this seaway the azimuth feedback maxima are nearly 20 degrees.
- The calm water yaw motion periods mentioned in the previous case study are probably still there, but masked by the influence of the waves and wind which dominate in the situation dealt with in this case.

CONCLUSIONS

The main objective has been to emphasise the various dynamic interactions that exist between the prime mover, propulsor, steering device and hull. The message is that every aspect of these interactive systems needs proper understanding, not just the sub-systems and their individual performances.

The ship must be considered in its surroundings and the traditional calm water optimisations are useful but not always sufficient.

Good progress has been made in recent years in computational methods, and in obtaining close correlation between calculated and model test results in some areas. But there are shortcomings in both model experiments and computations.

General experience supported by comprehensive full scale measurements as presented in the paper has demonstrated that a propulsion system has inherently several interacting mechanism that depend on the properties of the propulsor, the hull motion dynamics, speed operating profile and the environmental conditions.

Class societies have traditionally provided application factors on torque fluctuations in transmissions which have largely been based on prime mover characteristics. In some cases these may not be adequate when excessive excitations come from the propulsor end.

Advanced condition monitoring systems linked to real-time navigational data is a valuable tool for operational research and to help understand how the hull, machinery and propulsors work as a whole. This, in combination with improved model experiments and computational tool, will potentially lead to better system integration with reduced wear and tear of machinery and transmission system plus reduced fuel consumption and emissions.

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