Reduction of Greenhouse Gas Emissions by Propeller Design for Ship-In-Service Conditions.

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Abstract

A significant current topic in the shipping industry is the reduction of greenhouse gas (GHG) emissions. The International Maritime Organisation (IMO) is introducing mandatory mechanisms intended to ensure energy efficiency standards for ships, effectively requiring ships to become more energy efficient.

A ship's propeller is usually optimised around a steady-state "trial-conditions" design point. When the ship is in service-conditions, the propeller's loading will be different, due to the ship's environment, and the response of the ship to her environment. Thus, in the day-to-day operation of the ship, the propeller will be operating away from its optimal design point.

The aim of this paper is to determine to what extent GHG emissions (in terms of gains in propulsive efficiency) can be reduced by design for more realistic operating conditions. A simulator has been developed which is used to model the effects of the environment on the ship's motion response, and response of the propulsion system. Using this simulator, different operating scenarios may be developed, representing different shipping routes and different ship types. By examination of the subsequent propeller loading, a propeller can be optimised around a more realistic design point.

Conclusions are drawn on energy savings from design for in-service conditions, and the cost effectiveness of retrofitting new propellers, if the ship changes its mission profile.

Keywords: Ship Propeller In-Service Efficiency

1 Introduction.

At the initial stages of propeller design, a basis-propeller is usually chosen from a ‘stock-series’, whose parameters are then adjusted as the analysis progresses.

The initial basis-propeller is generally chosen to be optimised around the steady-state, trial-conditions scenario, id est, the propeller loading is calculated assuming the ship is in calm water.

Compared to the calm water, trial-conditions scenario, a ship in service-conditions will experience additional loading from her environment. The question that this paper addresses is: For what loading condition should the propeller be optimised around?

2 Powering and Performance Methodology.

The ship’s mission profile is kept constant throughout the analysis, only changes in loading from the environment are considered, symbolising different legs of a route. The Kriso Container Ship (KCS) has been chosen for analysis, representing a modern efficient design, whose mission profile is that of a liner service requiring the ship to make port within tight time restrictions.

2.1 Ship Resistance in Service Conditions.

Additional resistance components for a ship in service-conditions include wind and waves, which in turn impart other resistance terms on the vessel such as increase in form drag (due to sailing at some drift angle), and rudder drag (if course correction is required). A seaway by nature is unsteady, resulting in unsteady oscillating ship motions. These ship motions also contribute to the total resistance of a ship in service-conditions. Expected environmental conditions can be obtained from statistical sources like BMT Fluid Mechanics Global Wave Statistics (last accessed June 2012). These conditions can be input into a simulator (Trodden and Woodward, 2012) to obtain the manœuvring motion of a ship in service-conditions.

As a study of preliminary basis-propeller selection, this paper focuses on the steady-state, pure surge components of a ship in a seaway. That is, wake velocity vectors are constant, non-oscillatory in nature, whose direction is in-line with the ship’s centre line.

For the sake of clarity and brevity, this paper will make use of hypothetical sea-margins to represent the resistance of a ship in service. A sea-margin is a factor by which the calm water resistance is multiplied by, to obtain the in-service resistance. The calm water resistance of a ship can be readily estimated from the work carried out by Holtrop and Mennen (1982) and Holtrop (1984). The in-service condition is obtained by multiplying the calm water resistance by the sea-margin. In practice, the actual resistance to motion in pure surge for a ship in-service may be estimated using the simulator, from this the sea-margin can be calculated and used in a similar way to the following analysis.

2.2 Basis-Propeller Selection.

The Wageningen B-Screw series propellers are a general purpose, fixed pitch propeller which is widely used for design and analysis. Extensive studies have been conducted on this propeller series (Oosterveld and Oossanen, 1975), resulting in easy to use, verifiable methods for calculating thrust and torque coefficients. It is the B-Screw series that are used for analysis in this paper.

Figure 1 shows the method used in this investigation to select an optimum basis-propeller suitable for preliminary design purposes.

2.2.1 Cavitation and the Blade Area Ratio.

Unless otherwise stated, throughout this text the Expanded Blade Area Ratio is assumed to be approximately equal to the Developed Blade Area Ratio.

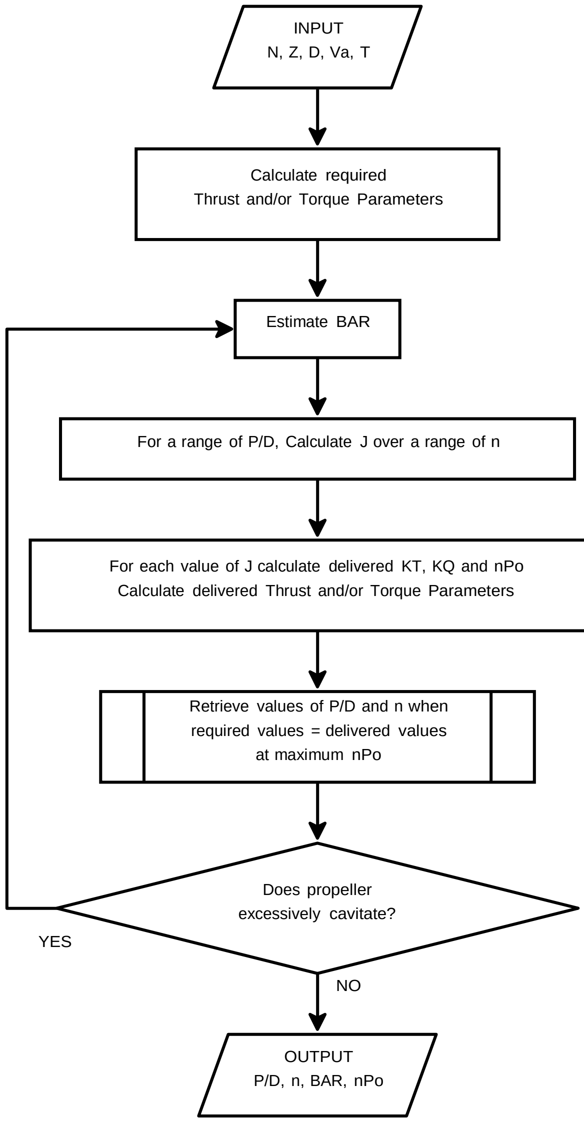

Figure 1: Algorithm for Basis-Propeller Selection.

Cavitation is the term used when the local fluid pressure drops to the vapour pressure (i.e. the pressure at which the fluid vaporises), resulting in "bubbles" within the liquid. Cavitation can occur on propellers when the pressure on the suction side of the blade falls below the vapour pressure.

The effects of cavitation are:

1. Flow along the surface of the propeller blade is disturbed, effectively modifying its profile, resulting in a change in coefficients of lift and drag, thus reduction in thrust and torque and loss of efficiency.

2. Hydrodynamic pressure fluctuations are set up on the hull due to growing and collapsing cavities. This may cause unacceptable levels of inboard noise and vibration.

3. Erosion of the blade will occur if the collapse of a cavitation bubble (due to the bubble moving back into a higher pressure region) occurs on the blade surface.

The onset of thrust breakdown due to cavitation occurs at a later stage than the vibration and erosion stages. Thrust breakdown does not usually occur before approximately 10% back cavitation has been developed (Gawn and Burrill, 1957).

In general, the smaller the Blade Area Ratio (BAR), the higher the efficiency (due to lower frictional resistance). Too small a blade area will however result in excessive cavitation.

Two commonly used methods to obtain the minimum required Blade Area Ratio are Keller's formula, and Burrill's cavitation diagram (Burrill and Emerson, 1962 – 1963). Burrill's method is described in Appendix B.

Keller's formula for the minimum Expanded Blade Area Ratio to avoid excessive cavitation is:

$$\frac{A_E}{A_O} = K + \frac{(1.3 + 0.3Z) \times \text{Thrust}}{D^2 (p_o + \rho g h - p_v)} \tag{1}$$

Where K is a constant of between 0.0 to 0.1 for twin-screw ships, and equals 0.2 for single-screw ships. $p_o + \rho g h$ is the static pressure at the shaft centre line, and p_v is the vapour pressure.

It has been found that the BAR as estimated using Burrill's Cavitation Diagram produces more conservative results than ones obtained using Keller's formula. That is, Burrill's method results in a propeller with a lower open-water efficiency, however, the resulting propeller is shown not to cavitate to the extent where thrust breakdown will occur for a higher range of sea-margins compared to the resulting propeller as obtained using Keller's formula.

Burrill's method is used in this work for estimating the required BAR to avoid excessive cavitation.

In this analysis, any amount of cavitation that results in thrust breakdown during the ship's voyage is considered unacceptable, and the propeller that causes this is rejected.

A propeller selected for maximum efficiency requires a minimum BAR, subject to cavitation avoidance. It can be seen from equations 1 and 2 of Appendix B that any increase of thrust from the design point will result in cavitation of greater than about 5% back cavitation. This may, or may not result in detrimental effects. The final choice depends, to some extent, on the mission profile. Cruise liners for example will want to minimise vibration for the sake of their passengers, on ore carriers, vibration may not matter so much.

If it is decided at the design stage that at no point in the ship's normal operation should the propeller's back cavitation exceed 5%, then there will be only one acceptable propeller selection, and that being the propeller that is selected to be optimised around the maximum significant expected sea-margin.

If it is decided during the design stage that vibration does not matter, then the 'best' propeller would be one whose design point would cause thrust breakdown slightly higher than the maximum significant expected sea-margin.

2.3 Engine-Propeller Matching.

It is important to match the engine power to the power required by the propeller, to avoid over/under loading the main engine, and producing excessive amounts of GHG emissions.

When the ship is in service, the engine is usually designed to operate between 85% and 90% of its Maximum Continuous Rated (MCR) Power. This is done to ensure the engine does not wear out too quickly, and also to provide extra power if it is required. In the following analysis, the engine margin has been selected to operate at 90% MCR in service conditions.

Figure 2 shows the KCS's propeller demand curve superimposed on a Wärtsilä 10 Cylinder RT-flex82C engine layout (Wärtsilä Switzerland Ltd., 2011). The propeller demand curve represents the power required of the propeller to propel the ship at the desired speed. The propeller demand curve may be obtained from the aforementioned simulator, or roughly estimated using the cubic 'Propeller Law' curve. If the resistance of the ship increases then the propeller demand curve will shift to the left and vice versa.

In figure 2 PD represents the propeller's design point about trial-conditions. PD^1 is the propeller's design point at service conditions. SP is the service propulsion point, and MP is the engine's maximum continuous rated power. The grey area in figure 2 shows the engine's overload range.

The light running propeller demand curve represents the power required to achieve the corresponding speed when the ship is in trial-conditions, that is the hull and propeller are smooth. The heavy running propeller demand curve represents the power required after the ship has been in the water for some time, when the hull and propeller are to some extend fouled. The difference between the heavy running and light running curves are usually between 4% and 7% and depends upon the ship's mission profile, dry docking interval and time between engine overhauls. This light running margin is usually selected from experience and in figure 2 the light running margin is 5%.

As can be seen in figure 2 the heavy running propeller whose design point is at P D will not provide enough thrust in service conditions to achieve the ship's service speed, therefore it is prudent to design the propeller to operate a few revolutions faster (the light running margin) than the service speed demands.

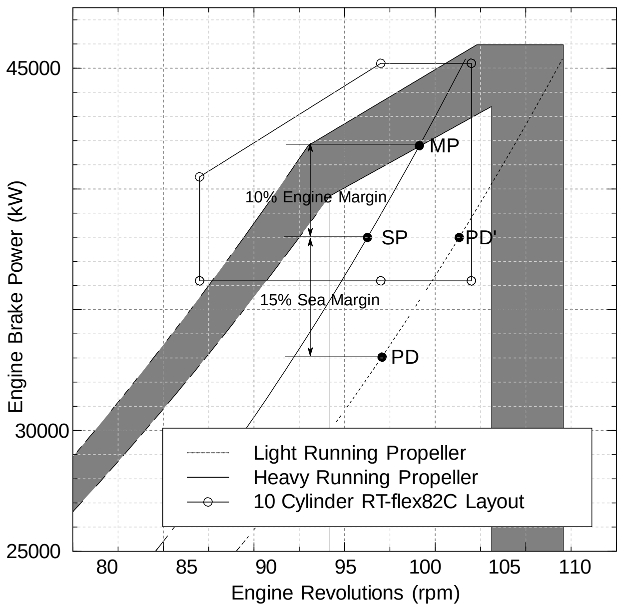

Figure 2: Wärtsilä 10 Cylinder RT-flex82C Main Engine & Propeller Demand Curve for KCS.

3 Propeller Performance in Off-Design Conditions.

Results for propeller performance in off-design conditions can be found in Appendix C. As can be seen from these tables, a propeller selected for lower expected sea-margins exhibits a greater open-water efficiency than one selected for a higher sea-margin, however it will also cavitate more at higher sea-margins.

The difference in open-water propulsive efficiency between the propeller whose design point was trial-conditions, and the propeller whose design point was service-conditions is just 0.006. This suggests that a propeller should be selected which is optimised around the sea-margin for which the ship is expected to sail, not from an efficiency point of view (which is somewhat negligible), but from improved cavitation properties.

4 Conclusions.

It is prudent to select a propeller whose design is optimised around service-conditions, rather than one optimised around trial-conditions. This is due to superior cavitation avoidance properties, rather than efficiency. The difference between the two propeller's efficiency is negligible.

From the viewpoint of retrofitting a new propeller to an existing ship on the basis that a propeller designed around service-conditions is "better" than one whose design point was trial-conditions, then it is probably not cost effective to do so. This holds true if the mission-profile remains constant, id est, her service speed and loaded displacement remain unchanged, and the existing propeller is not unduly cavitating.

5 Further Work.

This paper has focused on the steady-state scenario. Further optimisation may possibly be achieved by designing the propeller not just for the non-uniform wake field produced by the ship's hull (which is usually carried out at a further design stage), but also wake field fluctuations due to the motion of the ship in a seaway. This may be a productive area of future investigation.

6 Acknowledgments.

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A Nomenclature.

η_H Hull efficiency.

η_{Po} Propeller open water efficiency.

η_R Relative rotative efficiency.

η_t Transmission (Shafting) efficiency.

ρ_{sw} Density of salt water. (Kgm^{-3})

σ Cavitation index.

BAR Propeller Blade Area Ratio, usually expanded unless otherwise stated.

D_p Propeller diameter. (m)

J Advance coefficient.

KCS Kriso Container Ship.

K_Q Propeller torque coefficient.

K_T Propeller thrust coefficient.

L_{oa} Length overall. (m)

L_{pp} Length between perpendiculars. (m)

L_{wl} Length along the waterline. (m)

MP MCR propulsion Point.

N Number of propellers.

n Propeller revolutions. (usually rps)

P Pitch of propeller. (m)

PD Propeller Design point at trial conditions.

$PD^{\prime}$ Propeller Design point at service conditions.

P_{mcr} Maximum Continuous Rated Power. (kW)

Q Torque on the Propeller. (Nm)

SM Sea-Margin.

SP Service Propulsion Point.

T Thrust produced by the Propeller. (N)

V_a Speed of advance. (ms^{-1}).

V_s Service speed of ship. (ms^{-1})

Z Number of blades on propeller.

B Burrill's Method for Estimating Blade Area Ratio.

The thrust loading coefficient, τ_c can be calculated from:

$$\tau_c = \frac{T}{0.5\rho A_P V_R^2} \tag{2}$$

The thrust loading coefficient's upper limit for merchant ships (2% to 5% back cavitation), $\tau_{c_{limit}}$ on Burrill's cavitation diagram is given by:

$$\tau_{c_{limit}} = 0.28\,(\sigma - 0.03)^{0.57} \tag{3}$$

Where σ is the local cavitation number:

$$\sigma = \frac{\rho g h + P_{ATM} - P_V}{0.5\rho V_R^2} \tag{4}$$

and V_R is a reference velocity taken at 0.7 of the propeller's radius:

$$V_R = \left(V_a^2 + (0.7\pi n D)^2\right)^{0.5} \tag{5}$$

The Developed Area, A_D may be estimated using Burrill's empirical formula:

$$A_D = \frac{A_P}{(1.067 - 0.229P/D)} \tag{6}$$

From equations 2, 3, 4, 5, and 6 the required Blade Area Ratio can be estimated.

C Propeller Selection & Off-Design Performance Tables.

Propeller selected for KCS optimised around SM = 1.00				
N = 1, Z = 5, D = 7.9m, P = 8.45m, BAR_e= 0.808, η_{Po} = 0.640, n_{opt} = 97.07 rpm				
In-Service Prediction				
Sea-Margin	Delivered Power (kW)	η_{Po}	n (rpm)	Cavitation
1.00	32374.29	0.640	97.05	< upper limit
1.04	33897.60	0.636	98.07	> upper limit, <10% back
1.08	35496.97	0.631	99.12	> upper limit, <10% back
1.12	37123.09	0.627	100.15	> upper limit, <10% back
1.16	38635.47	0.623	101.09	> upper limit, <10% back
1.20	40214.46	0.618	102.05	> upper limit, <10% back

Table 1: Geometry and Performance of Propeller Optimised around SM=1.0

Propeller selected for KCS optimised around SM = 1.08				
N = 1, Z = 5, D = 7.9m, P = 8.59m, BAR_e= 0.871, η_{Po} = 0.628, n_{opt} = 97.98 rpm				
In-Service Prediction				
Sea-Margin	Delivered Power (kW)	η_{Po}	n (rpm)	Cavitation
1.00	32605.07	0.637	96.05	< upper limit
1.04	34082.21	0.633	97.02	< upper limit
1.08	35630.10	0.628	98.01	< upper limit
1.12	37252.75	0.624	99.02	> upper limit, <10% back
1.16	38929.68	0.619	100.03	> upper limit, <10% back
1.20	40461.52	0.615	100.94	> upper limit, <10% back

Table 2: Geometry and Performance of Propeller Optimised around SM=1.08

Propeller selected for KCS optimised around SM = 1.0				
N = 1, Z = 5, D = 7.9m, P = 8.45m, BAR_e= 0.903, η_{Po} = 0.619, n_{opt} = 100.99 rpm				
In-Service Prediction				
Sea-Margin	Delivered Power (kW)	η_{Po}	n (rpm)	Cavitation
1.00	32647.49	0.635	97.22	< upper limit
1.04	34185.83	0.630	98.24	< upper limit
1.08	35500.49	0.626	99.27	< upper limit
1.12	37407.71	0.622	100.28	< upper limit
1.16	38932.43	0.618	101.21	> upper limit, <10% back
1.20	40523.79	0.614	102.16	> upper limit, <10% back

Table 3: Geometry and Performance of Propeller Optimised around SM=1.15