An Investigation Into Twin Skeg Hullforms In Auxiliary Tanker Design

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INTRODUCTION

Vessel efficiency and fuel costs are becoming a key consideration for navies of the world. The design of modern naval replenishment tankers includes both twin screw and single screw design configurations. Whilst twin screw offers redundancy advantages, many potential customers draw comparison with commercial tankers where the single screw hullform is the most common arrangement. Lower purchase cost is often cited as the reason for favouring a single screw design, based on favourable resistance, ease of build, simplicity of the propulsion architecture and commonality with commercial vessels. However, there are also compelling arguments in favour of a twin skeg, twin propeller design: redundancy, lower thrust per propeller and thus higher efficiencies.

The decision as to which form is best for a particular application therefore requires an appraisal of operating conditions, hydrodynamic performance, powering requirements and machinery arrangement. Some recent studies suggest that benefits in hull efficiency can be realized when adopting twin skeg hullforms in larger ships such as LNG carriers (Reference 1). Vessel size, payload, speed, propulsor and machinery selection and engine-room layout also play into this decision and must be carefully considered to achieve a viable, cost-effective design.

This paper discusses the issues and design aspects of a single and twin skeg hullform, and the relative merits of each. The paper goes on to derive a single and twin screw concept with dimensions typically found for naval auxiliary tankers. Characteristics of each hullform are discussed, and resistance and propulsion estimates, both based on model tests, are prepared for comparison and to inform purchase costs and through-life cost trade-offs.

HULLFORM

Single screw hullforms remain the most common arrangement for commercial, high-block ships usually as a result of lower costs both upfront and through life. In fact single screw remains the arrangement of choice for many, more slender, faster ships, such as container ships. However, there are a growing number of twin screw designs appearing in the press (Reference 2 & 3). In the ongoing pursuit of greater economies of scale the size of new container ships continues to grow and with it the required installed power and propeller loading. The high loading of propellers can lead to reduced propulsive efficiency and an increased risk of cavitation and vibration. Moving to a twin screw design reduces the propeller loading and therefore increases propulsive efficiency. Indeed, Maersk’s Triple-E container ship, the largest container ship in the world at 18000 TEUs, is a twin screw design. Maersk claim that its twin screw propulsion system consumes approximately 4% less energy (due to the combined effect of a lower top speed and greater propulsion efficiency) than the 11000 TEU Emma Maersk’s single screw propulsion system (Reference 4).
Clearly this is somewhat larger than the average naval replenishment tanker and at 23kts is 
farther as well. However, the same argument holds true. As propeller loading increases the 
incentive to choose twin screw over single screw in order to maintain propulsive efficiency is 
compelling. In the case of naval replenishment tankers, it is common for the speed 
requirement to be in excess of that of the equivalent commercial tanker in order for it to keep 
up with the task group. This may be compounded by the need to maintain speed in a seaway 
for Replenishment at Sea (RAS) operations. With the propeller diameter limited by the 
minimum draught, which in turn may be limited by world-wide access requirements, the 
result is that the propeller loading on a naval replenishment tanker is likely to be higher than 
that of the equivalent commercial tanker. Simply implementing a twin, open shaft 
arrangement is likely to incur an efficiency penalty due to the high appendage resistance from 
the struts, bearings and shafts. However, a twin skeg hullform eliminates the need for these 
appendages and can provide favourable hydrodynamic performance particularly for full-
bodied ships (Reference 5).

For the purposes of this study the hullforms for two comparable naval auxiliary ship designs 
were identified for which model test resistance data was available. The first was a single skeg 
design with a displacement of approximately 24500te and the second a twin skeg design of 
approximately 26800te (Figure 1).

Unsurprisingly the principal characteristics of the two designs differed to a degree and 
therefore the single skeg hullform was scaled to match the principal dimensions of the twin 
skeg hullform. The resulting hydrostatics of each hullform are presented in Table 1.

<table>
<thead>
<tr>
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<th>Single Skeg</th>
<th>Twin Skeg</th>
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<tbody>
<tr>
<td>Length Between Perpendiculars (m)</td>
<td>167.4</td>
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<td>Waterline Length (m)</td>
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<td>Vertical Centre of Buoyancy (m, above baseline)</td>
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<tr>
<td>Transverse Metacentre above Keel - KMt (m)</td>
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<td>11.4</td>
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Table 1 – Hydrostatics Comparison

Once scaled by waterline length, beam and draught the characteristics of the two hullforms 
are fairly close. It can be seen that the displaced volume of the twin skeg form is slightly 
larger than the single skeg form. This is due to the inclusion of two, albeit finer, skegs versus 
one and is reflected in the block and prismatic coefficients. The extra displacement is in part
required in the twin skeg form to accommodate the extra weight of the twin screw propulsion system and additional structure associated with integrating two skegs rather than one.

The additional displacement will manifest itself as an increase in resistance which will be compounded by the increased wetted surface area of the twin skeg form. However, the impact can be tempered by taking advantage of the opportunity for a longitudinal centre of buoyancy positioned further aft which would be expected to reduce wave-making resistance. The real benefits of a twin skeg form come in terms of efficiency. Employing two skegs, each more slender than the equivalent centreline skeg of a single screw design, gives more control over the flow into the propellers to the hydrodynamicist and therefore a better wake field for the propellers, less cavitation and lower induced pressure pulses to the hull. Coupled with the more favourable LCB position, efficiency improvements in the region of 2 – 3% (Reference 5) can be expected.

The twin skeg design does have some drawbacks. The skegs are a complex shape and therefore a great deal of care must be taken in defining the shape and orientation at the design stage to realise the efficiency improvements. It is not just the hydrodynamics where the complex shape can cause problems. Structural integration and therefore build can also be complex. Finally, the internal arrangement in terms of providing access to the skegs and accommodating gearboxes can prove challenging. However, all of these challenges can be overcome as has been demonstrated on the UK Tide Class (MARS Tanker) which is now entering the detailed design phase (Reference 6).

RESISTANCE

To demonstrate the difference in performance between the single and twin skeg hullforms, the model test results for the two hullforms presented above were used to create comparable full scale resistance predictions. The differences in original principal dimensions were accounted for by extrapolating the results of the single skeg model tests to a full scale waterline length equal to that of the twin skeg model tests. The resulting differences in draught and beam were corrected using the Mumford indices (Reference 7). The resulting characteristics of the basis hullforms (Cb and Cp) were similar and no correction was made for the comparison.

![Figure 2 - Effective Power](image-url)
As can be seen in Figure 2 the effective power of the single skeg form was found to be marginally lower (3 – 5%) than that of the twin skeg form over the speed range investigated. This would seem to support the higher displacement and wetted surface area of the twin skeg form resulting in higher resistance.

**POWERING & PROPULSION**

Continuing the process of extrapolating the model test results it was also possible to create full scale powering predictions for the two hullforms from self-propulsion tests. However, this approach over-simplifies the problem as the propeller tested on the single skeg model reflected the choice for the original, full scale, single skeg ship and was therefore not necessarily optimal for the new ship principal dimensions. Therefore further investigation was carried out to develop the full scale powering predictions and improved propeller parameters for both hullforms.

**Design Arrangements**

The twin screw design had each shaft driven by a medium-speed main engine through a single reduction gearbox. The single screw design comprised a combining gearbox driven by two medium speed engines. In each design the same diesel generator (DG) set installation provided power to the Ships Electrical Load (SEL).

**Assumptions**

Based on the aft end geometry of the scaled, single skeg hullform, the propeller diameter for the single screw design was selected as 5.8m with the twin screw design being 5.4m diameter. The maximum allowable shaft speed of the twin screw design was 90rpm with a propeller pitch to diameter (PD) ratio of 1.2. The single screw's higher loaded propeller had a maximum shaft speed of 92rpm and a PD ratio of 1.3. The propeller for each design was developed to match the specific loads using a basic design approach. In short, the single screw design was defined to be optimal to allow a valid comparison with the twin screw design. With a larger propeller diameter and higher PD ratio it has the best possible advantage to the twin screw design.

When comparing designs with different propulsion arrangements, it is important that issues such as noise are acceptable. The higher loading on the single propeller could lead to undue noise and a lower Cavitation Inception Speed (CIS). The Blade Area Ratio (BAR) of the single screw design was increased compared to that of the twin to reduce noise and cavitation. However, there is a practical limit to the BAR due to manufacturing and hydrodynamics. Increasing the BAR incurs higher frictional losses on the propeller so normally there is a trade between efficiency and noise.

The relative rotational efficiency was assumed to be the same for both designs. This will not be the case in practice but the impact of the differences, especially at full speed, were believed to be small.

**Results**

Figure 3 presents the efficiencies of the two designs. As noted earlier, the relative rotative efficiency of each design was assumed to be the same. The hull efficiency of the single screw
form is higher than that of the twin screw form (1.2 vs 1.1). In contrast the lower loading of the twin screw design achieves better open-water propeller efficiency (69% versus 55%). When these are combined it can be seen that the Quasi-Propulsive Coefficient (QPC) of the twin screw design is greater than that of the single screw design. Therefore, whilst the twin screw design has a higher effective power than the single screw design, the better propulsive efficiency results in a lower delivered power demand. Figure 4 shows that the delivered power of the twin screw design at 18kts, is approximately 12% lower than that of the single screw design.

![Figure 3 - Efficiencies](image1.png)

![Figure 4 - Delivered Power](image2.png)

The results indicated that below 12 knots the single screw design exhibited marginally better fuel consumption. However, at speeds greater than 12 knots the benefits of the lower loaded propellers and higher efficiencies of the twin screw design were realised. Clearly the design with the best actual annual total fuel consumption will depend on the match of the ship's operating profile to the fuel consumption across the speed range. Put simply, more time
above 12 knots favours the twin screw design whilst more time below favours the single screw design.

![Figure 5 - Operating Profile and Annual Fuel Burn at Each Speed](image1)

An indicative operating profile for a typical naval replenishment tanker was assumed and is presented in Figure 5. The cruise speed was assumed to be between 13 and 15 knots with a maximum speed of 18 knots. The annual time spent at sea was 2,000 hours. The minimum continuous speed was 6 knots as this was assumed to be the minimum speed required for good steerage in confined waters. Combining this with the fuel consumption across the speed range gives the annual fuel consumption for each design also presented in Figure 5. The annual fuel consumption and therefore fuel cost is lower for the twin screw design by 3%.

**Through Life Costs**

An estimate was made of the machinery acquisition costs and the through life costs due to fuel consumption and engine upkeep. The Whole Life Cost (WLC) balance with these cash flows alone is shown in Figure 6.

![Figure 6 - Machinery Whole Life Cost](image2)

Figure 6 is purely indicative, based on BMT estimates, as the acquisition costs and the fuel costs (estimated as £700/tonne) can vary between suppliers and with time, respectively. However, the total cost of both designs, *ceteris paribus*, will have a similar WLC at the end of
the life of the ship. In practice it is likely that the rises in fuel costs will be greater than those in machinery and the twin screw design will have the overall lower whole life cost.

**MACHINERY ARRANGEMENT**

The length of a tanker can be considered in three discrete sections, the area forward of the cargo, the cargo section and the area aft of the cargo section. The minimum length of the ship is then dictated by the combined minimum lengths of each of these self-contained sections. The minimum length of the area forward of the cargo is dictated by the length of the bow thruster space and regulations that specify the location of the collision bulkhead and the presence of a cofferdam between the cargo and bow thruster machinery space. The minimum length of the cargo section is dictated by the quantity of cargo to be carried and the minimum inner hull separation required by MARPOL. It is in the section aft of the cargo that the designer has the opportunity to optimise the arrangement and therefore minimise ship length and to some degree cost.

As a minimum, under SOLAS the main machinery space(s) must be isolated from the cargo section by the cargo pump room and/or a cofferdam. In addition accommodation spaces must be aft of this isolation space. Therefore the length of the aft section of the ship becomes a balance between machinery space length, driven by propulsion arrangement, and accommodation block length, driven by complement, lifeboats and any other capability to be incorporated such as aviation.

In the case of a commercial tanker, it is common for the accommodation block to fit within the length of the main machinery space. Therefore it is the machinery space, driven by the length of the main engine, and separation from the cargo section that drives the aft end length. A single screw hullform allows the machinery to be fitted well aft to keep the machinery space length to a minimum. In the case of a twin screw arrangement, it is likely that the machinery space can actually be shorter as the installed power is now provided by two smaller engines arranged side by side rather than a single larger engine. However, the slenderness of the twin skegs may prevent the gearboxes from being positioned as far aft as in the single screw arrangement. Therefore it is critical that the positioning of the gearboxes is considered during the development of the hullform to avoid unintentionally compromising the aft end arrangement and in turn driving ship length.

In comparison, the aft end length driver for a naval replenishment tanker is far less clear cut. The power generating capacity of a naval replenishment tanker is likely to be greater than that of a commercial tanker. Higher Manning levels, greater complexity and additional capability and systems all combine to increase hotel load. Furthermore, the requirement to conduct Replenishment At Sea (RAS) operations, where cargo and ballast pumps are run whilst underway, places a significant demand on the power generation system. The issue is further compounded by the need for greater availability of power generation which adds a layer of redundancy beyond that found on a commercial tanker. It is therefore likely that a dedicated generator space is required rather than fitting generators on a mezzanine in the main engine room as in the case of a commercial tanker. Now the total length of the machinery space is dictated by the length of the gearbox, main engines and the generators.

However, it is not only the machinery space that is longer. The inclusion of a flight deck will drive the accommodation block forwards and the length of the accommodation block on a naval tanker is also likely to be greater than that of a commercial tanker. This is due to higher Manning levels, the inclusion of a hangar and associated spaces and the need to site lifeboats
aft of the cargo section but clear of the flight deck. In summary, the aft end length driver of a
naval replenishment tanker is therefore a balance between the combined length of the flight
deck and accommodation block and the combined length of two machinery spaces.

The specification of a redundant propulsion system can also have an effect. Lloyd’s
Register’s (LR) Propulsion and Steering Machinery Redundancy (PSMR) notation
(Reference 8) ensures propulsion power and steering is maintained following the loss of a
single item of equipment. Whilst it is not impossible to achieve the PSMR notation with a
single screw design it does present some challenges. A risk-based failure mode assessment
would be required was well as assurances to ensure that adequate redundancy is supplied.
This significantly increases the risk to the design and requires close consultation with the
classification societies. It is clear that the PSMR notation can be achieved much more simply
in a twin screw arrangement. Shaftlines and steering systems are already duplicated so it only
remains to ensure that the supporting services and systems are also independent.

An even greater level of redundancy can be achieved where 50% of the installed propulsion
system is maintained following the loss of a single compartment due to fire or flood such as
LR’s PSMR* notation. The distinction is essentially that the redundant shaftlines must now
be installed in separate compartments. This could be a watertight boundary on the centreline
of the ship separating the main engines and therefore have little effect on overall arrangement.
However, experience shows that meeting military stability standards, which usually feature
transverse damage extents to the ship’s centreline, is extremely challenging. It is more likely
that the main engines must be separated longitudinally. On face value this would add
significantly to the ship’s length. However, it has already been stated that a second
machinery space is required to house the generators. Therefore an extremely flexible and
redundant power generation and propulsion system can be achieved with little additional ship
length by careful arrangement of the main engines and generators across the two main
machinery spaces.

OPERABILITY & REDUNDANCY

The operational considerations which impact on the choice of a single or twin screw solution
for a naval replenishment tanker include:

1. Manoeuvrability in relation to a fast breakaway during Replenishment At Sea (RAS);
2. Directional stability when alongside another vessel;
3. Redundancy should a propulsion equipment fail during RAS;
4. Manoeuvrability in port;
5. General ship availability and redundancy.

Safety During RAS Operations

In general, the replenishment ship is nominated as the “control ship” and maintains a steady
course and speed. The receiving ship is responsible for conducting the approach and
maintaining station on the control ship. This is not always true, as in the case of large aircraft
carriers which will generally be the control ship or consolidation between replenishment ships
when one of the replenishment ships has to conduct the approach (Reference 9).

During the RAS operation, it is important that both ships can maintain a straight course, i.e.
be directional stable. There are strong interactions that occur between the vessels when at
close positions, due to the venturi effect, and this will require a degree of rudder angle to counteract and corrections to maintain distance; this is more fully explained in (Reference 10). Good directional stability is advantageous as the ships behave in a more predictable manner. Both single and twin skeg designs can achieve adequate directional stability and recent model tests for the UK have confirmed that a twin skeg form behaves in a manner consistent with current vessels.

However, the advantage of a twin screw design is its enhanced manoeuvrability when required to rapidly steer away from the other vessel and in its inherent redundancy. Generally both the approach and separation manoeuvres are conducted with small adjustments to direction, but when necessary, a rapid emergency breakaway manoeuvre may be required to increase separation quickly. This is essentially an accelerated separation but clearly the enhanced manoeuvrability provided by a twin screw / twin rudder configuration enables both greater control and the potential for more rapid separation.

Another safety aspect to consider is the failure of the propulsion train during a RAS operation. If one of the vessels loses propulsive power, it will experience a rapid loss of speed and will fall behind the other ship. The issue is the speed with which the ships can disconnect the jackstays before this misalignment becomes a significant safety issue (i.e. lines fouling structure or equipment, damaging winches or simply parting under tension and “snapping” across the ships decks). To enhance safety in these conditions, a redundant propulsion system can be specified such that speed loss can be arrested before the situation becomes dangerous. In the case of LR’s Rules and Regulations for Naval Ships (Reference 8), it is a requirement of the “RAS” notation that the ship also achieves the “PSMR” notation. The Rules further note that “As a guide, ships are to be designed to be capable of maintaining continuous course and speed for at least one minute to facilitate safe emergency breakaway in the event of equipment failure in propulsion and steering systems”.

Some analysis of the effect of losses of propulsion has been conducted for a twin screw ship. In Figure 7, the effect on ship speed of loss of one transmission line (i.e. one screw) is presented. On loss of the shaft, initial activities include communications between ships, the crash stop of pumping activities and the commencement of disconnection. In parallel, the power on the remaining shaft is increased to maximum and rudder angles adjusted to prevent the loss of station keeping. The figure shows that with a twin screw design, minimal speed loss can be achieved. In comparison, even if a single screw design had achieved PSMR notation, the failure of a critical part of the propulsion system not duplicated (e.g. gearbox)
would result in loss of all propulsion power, significant speed loss and a potentially dangerous situation would arise.

**General Manoeuvrability**

In addition to the above, replenishment ships may also have other aspects to be considered, notably manoeuvrability in a threat environment and manoeuvring in port. As a replenishment ship has a need to berth in commercial oil terminals, naval depots and a wide range of commercial ports as part of its wider naval roles, it is likely to be a requirement that the vessel has good berthing manoeuvrability and ideally can berth in many conditions without the need to rely on tug assistance. This will require the fitting of side thrusters, common place on the current generation of replenishment ships and tankers. A single screw vessel is likely to require both bow and stern thrusters whilst a twin screw vessel will only require bow thrusters as the twin screw configuration can generate the required side thrust.

**Overall Availability**

Finally, it is observed that a twin screw vessel has a greater degree of redundancy and therefore offers higher availability (although conduct of RAS on one shaft is highly unlikely, reasonable cruising speeds can be achieved) and lower vulnerability. Both of these factors will be attractive in a replenishment ship design as the impact of equipment failure, whether by accident, unplanned maintenance or action damage must be considered.

**CONCLUSION**

The choice between twin and single screw for a naval auxiliary tanker is a complex one and requires a review of many factors. Much depends on the specific requirements of the ship in question and the anticipated operational profile. Opting for a twin skeg solution is likely to result in a higher initial procurement cost as a result of the greater complexity of the hullform and structure and the extra installed machinery. However, whilst the single skeg design yields comparable or slightly lower resistance, the greater propulsive efficiency of the twin skeg design results in overall lower delivered power and annual fuel cost savings in the region of 3%. Therefore, the additional initial cost will be offset by a lower through life cost resulting in little to choose between the single and twin skeg designs on cost alone. Increasing fuel costs will only add weight to the argument in favour of a twin skeg design. Add to that the greater manoeuvrability, flexibility, availability and vulnerability afforded by a twin screw design and the argument for a twin screw solution becomes more compelling.

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