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PERSPECTIVES FOR A WIND ASSISTED SHIP PROPULSION

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SUMMARY

For three different wind propulsion technologies, the energy saving potential of sea going cargo vessels are discussed: a kite, a Flettner rotor and a Dynarig-sail. The energy saving potential can be increased significantly if the route can be optimized when using a wind assisted ship propulsion. The increase of travelling time due to a route adoption is within the frame of the commonly accepted uncertainty in supply chains and can be limited or adjusted in the route optimization software as a parameter. The calculated saving potential depends on several parameters: the considered wind propulsion system, the route, the kind of ship (bulker, multipurpose carrier, tanker), as well as the ship speed and the weather. The cost-effectiveness of the installation of a wind propulsion system strongly depends on the fuel price, the ship speed and the international policy concerning the ship emissions.

1. INTRODUCTION

The most important cost factor in ship transportation is the fuel price, 40 % to 60 % , depending on the global fuel price situation (bunkerindex, 2016; Bunker Fuel Prices).There are several approaches to minimize the fuel consumption of ships as slow steaming, operating the engines in the most effective range, ship design, hybrid technologies for ship propulsion. One option for a hybrid system is the Wind Assisted Ship Propulsion WASP using wind for a part of the propulsion systems, which means that the engine power can be reduced using a Wind Propulsion System WPS, when the weather conditions are suitable. This kind of hybrid propulsion is not new, but in the past it came up from another technological direction: In the beginning of the 19th century the steam engine started to replace the sails on seagoing ships. For instance, in 1854 /56 the “Great Eastern” used steam engines to support the sail propulsion. In the beginning of the 20th century the steam engines and the last sails were replaced by fuel driven motors, due to the easy availability of this fuel and the enormous saving potential by ship design (no rags on deck, smaller machineries) and personnel reduction. The perspectives for the use of a WASP is guided by the strong pressure for cost reduction, by the environmental impact from ship emissions (CO_2, SO_2, NO_x, particles ...), and by the long-term view on the availability of oil.

So the use of wind propulsion systems as a WASP comes up under new perspectives (Otto, 1992; Schenzle and Hollenbach, 2010; Schenzle, 2016; Traut et al., 2014). However, the transport capacity, the cargo handling and the amount of personnel on the ship should not be negatively influenced by the use of a WPS.

The energy saving by a WPS depends on the weather conditions along the ship’s route: wind produces propulsion energy by the WPS, but it also produces waves, which consume propulsion energy, when the ship is steaming against wind, and waves. When using WASP it is important to adopt the route for an optimal use of the wind energy. However, this route optimization shall not lead to an essentially longer travelling time, which might decrease the profitability of the cargo transport. So it cannot lead to the ancient ship routes for sailing ships. Based on these conditions appropriate WPS and a suitable route optimization are discussed in this paper. The uncertainty in weather prediction leads to an uncertainty in the route optimization for the saved energy and the estimated time of arrival (ETA) and can be taken into account (Zastrau, 2016).

2. CONCEPT

For a realistic discussion of the future of WASP systems modern WPS, suitable ship types and typical ship routes are selected.

2.1 WIND PROPULSION SYSTEMS (WPS)

The future use of WPS for WASP is under discussion since many years (Otto, 1992; Schenzle, 2010; Dykstra Naval Architects, 2013; Allenström, 2012). Here the following criteria for the use of a WPS in a WASP are considered:

- New interesting features for WPS (with high efficiency, low impact on the ship performance)
- State of the technology (already realized for instance in pilot installations)
- Suitability for installation on existing cargo ship types
- The influence on ship performance (loading, unloading, handling during the steaming period)
- No additional crew necessary

As a result, the following WPS were chosen (Figure 1):

- a kite
- Flettner-rotors
- a Dynarig

Figure 1: Different wind propulsion systems (WPS) used for wind assisted ship propulsion (WASP): a kite (top-left), Flettner rotors (bottom-left), soft furling square sails (Dynarig sails on the Maltese Falcon, bottom-right), solid foldable square sails with a similar function as the Dynarig (Shin Aitoku Maru, top-right).

2.2 SHIPS

The following criteria are considered for suitable ship types:

- cargo ships with a common speed of about 13 – 15 kn. Ships with a higher speed as e.g. 20 kn or more (Container ships, Ferries, Passenger Ships, …) can only use a small part of the wind spectra (wind speeds > 6 Bft) for WASP.
- The ship types should be suitable for the installation of WPS without changing the function of loading and unloading of a cargo or the capacity of the ship.
- The ship types considered should realize an essential part of the international cargo transport.
- The transport time of the ships should have a certain tolerance, which is given for the tramp shipping (not valid for container ships, ferries, ..)

For the simulation the following ships are chosen as typical examples of the three chosen ship types:

- a multi-purpose carrier (MPC): 17 500 DWT, length pp: 133m, width: 22.8 m
- a bulker: 37 600 DWT, length pp: 183 m, width: 28.5 m
- a tanker: 114 000 DWT, length pp: 240 m, width: 44 m

2.3 ROUTES

The efficiency of a WASP system depends on the weather conditions on a route. There are typical differences in the weather situation on different areas of the oceans: the areas near the poles have higher average wind speeds than the areas near the equator (not counting special situations like hurricanes). The following criteria are considered to select appropriate shipping routes:

- The distance (long and short distances)
- The weather conditions (dominant high or low wind speeds)
- Routes used for an essential part of the global good transport

Therefore, the following routes are chosen:

- Germany – North America: Wilhelmshaven – Baltimore, a medium range route in a strong and turbulent wind field area with a dominant wind direction (west-east)
- Germany - South-America: Emden - Nueva Palmira, a long-range route with rather stable weather situations with normally low wind speeds.
- Germany – Norway: Wilhelmshaven – Bergen, a short-range route with turbulent weather conditions in both directions.

3. WIND PROPULSION SYSTEM

3.1 SIMULATION CONCEPT

The propulsion force of the discussed WPS is modelled based on available literature data. The energy saving potential is simulated using models for the ship resistance and the ship propulsion, taking into account the additional resistances due to waves and wind for the actual weather situation (Bentin *et al.*, 2016). The propulsion force FP of a WPS is given by

$FP \sim A * FPN(AWS, AWA, C_l, C_d)$

A characterizes the dimension of a WPS.
C_l, C_d lift and the drag coefficients respectively
AWS apparent wind speed, AWA: apparent wind angle

FPN(AWS, AWA, C_l, C_d) describes the „normalized“ propulsion force of a WPS as function of the direction and the speed of the apparent wind (AWA, AWS) and the WPS characterizing parameters C_l and C_d.

The propulsion force FP of the WPS will provide an effective additional propulsion power $P_e(WPS) = FP*Vs$ at a certain ship speed Vs through water (STW).

The necessary effective propulsion power to overcome the resistance due to the ship speed through calm water (cw) and due to wind and waves is

$$P_e = P_e(cw) + P_e(wind) + P_e(waves)\ .$$

With the WPS and the reduced engine power (for wind assisted hybrid propulsion) $P_e(WASP)$ the effective propulsion has two components

$$P_e = P_e(WASP) + P_e(WPS) \quad or$$
$$P_e(WASP) = P_e - P_e(WPS)$$

The corresponding break power P_B supplied by the engine is

$$P_B(WASP) = \frac{1}{\eta_T} Pe(WASP) = \frac{1}{\eta_T}(P_e(cw) + P_e(wind) + P_e(waves) - P_e(WPS)) ,$$

With η_T = total efficiency between break power delivered by the engine and the effective propulsion power. The break power is related to the fuel consumption m of the motor via the Specific Fuel Oil consumption SFOC:

$$m = SFOC\ P_B .$$

The reduction of consumption due to P_e(WPS) is

$$\Delta m(WPS) = SFOC\ \frac{1}{\eta_T} P_e(WPS)$$

η_T may change due to the reduced load on the propeller by the WPS. However we use a constant η_T in the simulation, which might be realized by a propeller with an adjustable pitch (see E-Ship1). The contribution due the calm water resistance P_e(cw) is taken from the model test report of the specific ship and those due to wind and waves are simulated according to software / formula available in literature and evaluated by actual ship measurements over 2 years (Schlaak, 2016; Bentin *et al.*, 2016).

3.2 KITE

The function and the use of a kite has been described in (Schlaak *et al.*, 2009). The model is based on the energy calculation for wind turbines. The wind power

$$P_{wi} = \frac{1}{2} * \rho * V_{wi}^3 * S_{wi}$$

(V_{wi} = wind speed, S_{wi}=effective wind surface, ρ = air density) is related to a "wind force F_{wi}" by

$$P_{wi} = (\boldsymbol{F_{wi}}) * (\boldsymbol{V_{wi}})$$

(bold letters signify vectors). The wind force F_{wi} causes a force on the kite: FP. The efficiency of the energy transfer from the wind field to the kite is considered by ε

$$FP = \frac{1}{2} * \rho * AWS^2 * \varepsilon * S_{wi} * FPN$$

By comparing with experiments, the model is fitted giving

$$FP = 27N \left(\frac{A}{m^2}\right) \left(\frac{\text{AWS}}{m^s}\right)^p FPN \quad \text{with}$$

$$FPN = \left(cos\frac{\alpha}{2}\right)^2 (cos\ \delta)^2 = \left(cos\frac{180°-\text{AWA}}{2}\right)^2 (cos\ \delta)^2$$

FPN normalized propulsion force as function of the direction of the wind, independent of the size of the kite and the wind speed
AWS apparent wind speed
A dimension (surface) of the kite

α = 180°- AWA

δ = 30° elevation angle of rope from ship to kite

p dependent of the speed of the kite: 1- 2

The kite is sailed on a laying 8, the central point is positioned in the direction of half angle between ship course and AWA.

The used weather data refer to 10 m above sea level. Since the kite is sailed well above the ship (e. g. 100 m) the wind speed V_{Wi} (h) in the height h is calculated by

$$V_{Wi}\ (h) = V_{Wi}\ (10m)\ \frac{ln\frac{h/m}{10^{-2}}}{ln 10^3}$$

The normalized propulsion force FPN characterizes the behavior of the kite as a function of the apparent wind angle AWA (Figure 4). The propulsion power P_e (WPS) is (V_S = ship speed through water STW)

$$P_e(kite) = FP * V_S$$

And the corresponding reduction of the engine power (brake power)

$$\Delta P_B = \frac{1}{\eta_T} P_e(kite)$$

To drive the kite on the 8 an electrical motor is used, consuming a certain small amount of electrical energy P_{EE}, which reduces the fuel saving by the kite $\Delta m(kite)$ with a specific motor efficiency SFC

$$\Delta m(kite) = SFOC\ \frac{1}{\eta_T}\ P_e\ (\text{kite}) - SFC\ P_{EE}$$

but is considered in the simulation by a reduction of the break power saving by the kite

$$\Delta P_B\ (kite) = \frac{1}{\eta_T}\ P_e(\text{kite}) - \frac{1}{\eta_{EE}} P_{EE}\ (\text{kite})$$

η_{EE} characterizes the efficiency of creating electricity by a fuel burning machine on board.

The final break power for the hybrid system is:

$$\begin{aligned} P_B(WASP) = \; & P_B(withou\,WPS) - \Delta P_B\;(kite) \\ & = \frac{1}{\eta_T}\big(\;P_e(cw) + P_e(wind) \\ & + P_e(waves)\big) - \frac{1}{\eta_T} P_e(kite) \\ & + \frac{1}{\eta_{EE}} P_{EE}(\text{kite}\,) \end{aligned}$$

3.3 FLETTNER

The Flettner-rotor creates a force perpendicular to the direction of the apparent wind (Magnus effect based on Bernoulli, see e.g. (Seifert, 2012)). This effect depends on the rotation speed of the rotor and the wind speed. The optimal rotation speed depends on the apparent wind angle. Often the relation between wind speed and the peripheral surface speed is in the range of 3 to 4 (Wagner *et al.*, 1985; Seifert, 2012).

As a ship propulsion, the Flettner rotor was first installed in 1924 on the "Buckau" (2 Flettner rotors) and in 1926 on the "Barbara" (3 rotors). Both ships were sailing with the Flettner rotors for several years. As the fuel driven ships became more economic WPS systems such as the Flettner were not used anymore. The rising fuel price and the impact of the ship emissions on the environment has brought the Flettner rotor back into discussion. In 2008 the E-ship 1 of ENERCON was constructed with 4 Flettner rotors (see Figure 1).

The propulsion force can be described by

$$\begin{aligned} FP \;& = \frac{\rho}{2} A_F\; AWS^2\;(C_l \sin\beta - C_d \cos\beta) \\ & = \frac{\rho}{2} A_F\; AWS^2\; FPN \end{aligned}$$

with

η	density of air
AWS	apparent wind speed
A_F	relevant surface for the rotor = L * D with L = height of rotor, D = diameter of rotor
D_s	diameter of top disc with $D_S \geq 2D$ (to avoid an early break down of the flow structure)
β	angle between ship course and the direction of the apparent wind AWS
C_l, C_d	lift and drag coefficients

For the calculation of the Flettner characteristics $FPN = f(C_l, C_d,$ AWA) the experimentally established dependencies for C_l and C_d are used (Wagner *et al.*, 1985)

$$C_l = C_l\left(\frac{u}{AWS}\right) und\; C_d = C_d\left(\frac{u}{AWS}\right)$$

u peripheral speed of the rotor (m/sec)

For the Flettner rotor the C_l and C_d coefficients depend on the relation between peripheral speed of the rotor surface u and AWS: $x = \frac{u}{AWS}$. For x = 4 there is a maximum for C_l, so that for the model the following conditions are set: $x = \frac{u}{AWS} \leq 4$. There is a limit of the rotor speed $n = n_{max}$ due to technical limitations:

$$u \leq u_{max} = \pi D n_{max}$$

The calculated polar based on these C_l, C_d values is shown in Figure 4.

The propulsion power contribution of the Flettner Rotor is

$$P_e(Fl) = FP(Fl)\; V_S$$

It decreases the propulsion energy required by the machine to keep Vs (brake power P_B) by:

$$\Delta P_B(Fl) = \frac{1}{\eta_T} P_e(Fl)$$

<u>Rotation energy of the rotor</u>

The Flettner rotor needs electrical energy for the rotation of the cylinders. This energy is supplied by electrical motors and is mainly needed to overcome the friction due to the rotation. The friction can be described by (Wagner *C. et al.*, 1985):

$$F_R = c_R * F_N$$

c_R	friction coefficient, dependent on the kind of the bearing
F_N	weight force of the rotor weight

The power to overcome the friction is
$P_R = F_R u = F_R\; 2\pi R n = c_R\; m_R\;\; g\;\pi\; D\; n$

u	peripheral speed of the rotor surface shell
m_R	$= d\pi DH\rho_M$ (mass of the rotor)
d	thickness of the rotor shell
D	diameter of the rotor,
H	height of the rotor
ρ_M	density of the rotor material
n	revolution speed.

The formula is compared with the value given for the E-Ship 1 (ENERCON, 2013) and fitting the C_R value gives

$$P_R = RFL\;\left(\frac{D}{m}\right)^2 \left(\frac{H}{m}\right)\left(\frac{n}{\frac{1}{min}}\right) kW = \; P_{EE}$$

with $RFL = \frac{1}{400} = 2{,}5\; 10^{-3}$

The aerodynamic friction is neglected, estimated to be essentially smaller than the mechanical fraction. The effective reduction of the ship propulsion power of the

Flettner rotor has to be reduced by the rotation energy needed, supplied by an electricity generator as P_{EE}

$$\Delta P_B(Fl) = \frac{1}{\eta_T} P_e(Fl) - \frac{1}{\eta_{EE}} P_{EE}(Fl)$$

η_{EE} for the efficiency of creating electricity by fuel burning on board.

The final break power for the wind assisted hybrid system is:

$$\begin{aligned} P_B(WASP) = \; & P_B(without\ WPS) - \Delta P_B\ (Fl) \\ & = \frac{1}{\eta_T}\big(P_e(cw) + Pe(wind) \\ & + P_e(waves)\big) - \frac{1}{\eta_T} P_e(Fl) \\ & + \frac{1}{\eta_{EE}} P_{EE}\ (\mathrm{Fl}\ ) \end{aligned}$$

3.4 DYNARIG

The Dynarig is a development of the square sail into the direction of the Bermuda sail for a more effective wind use for sailing close to the wind. The sail is constructed as a segment circle attached to the yards and uses the Bernoulli effect for ship headings near to the apparent wind direction (Figure 2). The Dynarig was designed by Prölss and researched by Wagner (Prölss, 1970; Wagner, 1966). In 2006 the Dynarig was realized on the “Maltese Falcon ”, a 88 m yacht (www.decaboyachtpainting.com; Maltese Falcon, 2016) (Figure 1). A transport ship with the Dynarig (”The Ecoliner”) is discussed (Dykstra Naval Architects, 2013).

The apparent wind causes a lift force L and a drag force D at the Dynarig (Figure 2):

$$L = C_l \frac{\rho}{2} A_S\ AWS^2$$
$$D = C_d \frac{\rho}{2} A_S\ AWS^2$$

A_S surface of the sail
C_l, C_d lift and drag coefficients

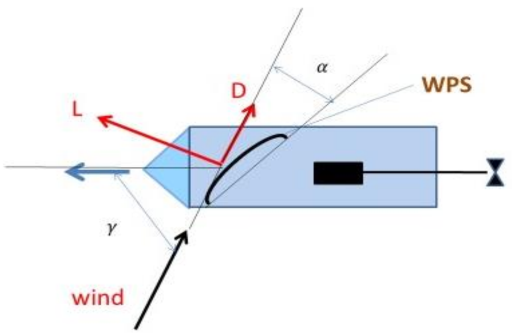


Figure 2: Forces (L, D) and angles at the Dynarig

The propulsion force results in

$$\begin{aligned} FP(DR) &= L\sin\gamma - D\cos\gamma \\ &= \frac{\rho}{2} A_S\ AWS^2\ (C_l\ \sin\gamma - C_d\ \cos\gamma) \\ &= \frac{\rho}{2}\ A_S\ AWS^2\ FPN(DR) \end{aligned}$$

The normalized force FPN is a function of γ, the direction of the apparent wind in relation to the ship course and the position of the sail α in relation of the AWA. C_L and C_d depend on the angle between the direction of the AW to the sail. For the modelling of the function of the sail the C_L and C_d values are taken from Figure 3, based on experimental measurements in the wind tunnel.

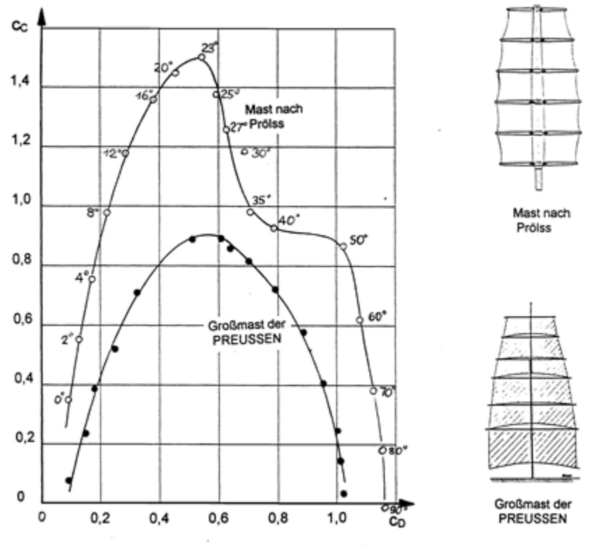


Figure 3: Diagram of C_l (here Cc) and C_d (here C_D)values as function of α , the orientation of the sail to the apparent wind direction AWA form (Wagner, 2000)

$\alpha = 90°$ means pure drag function of the sail. For $\alpha = 23°$ the course close to the wind C_l has a maximum. For the calculation of the polar diagram $FPN = f(\gamma, \alpha)$ (Figure 4) γ is varied and α is choosen for an optimal performance.

The propulsion power of the Dynarig

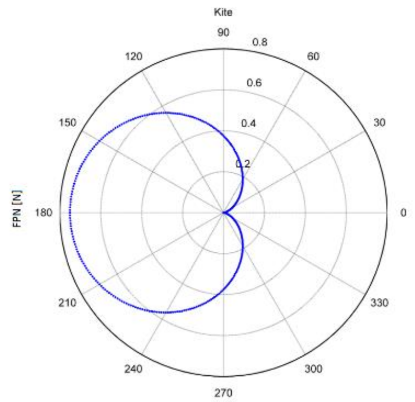
$$P_e(DR) = F(DR)\ V_S$$

reduces the machine propulsion power by

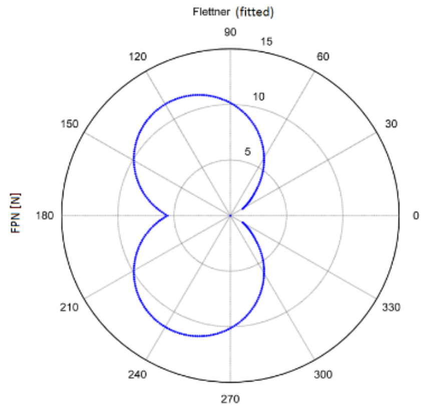
$$\Delta P_B(DR) = \frac{1}{\eta_T} P_e(DR).$$

The final break power for the hybrid system is:

$$\begin{aligned} P_B(WASP) = \; & P_B(without\ WPS) - \Delta P_B\ (DR) \\ & = \frac{1}{\eta_T}\big(\ P_e(cw) + P_e(wind) \\ & + P_e(waves)\big) - \frac{1}{\eta_T} P_e(DR) \end{aligned}$$



(a)



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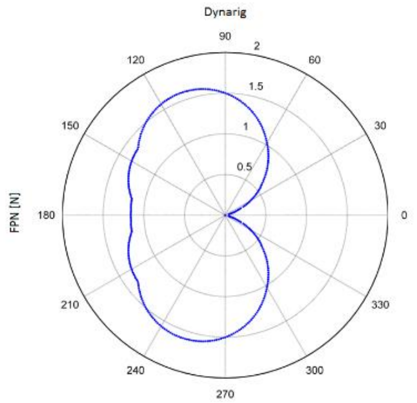

(c)

Figure 4: The normalized propulsion force FPN [N] for the discussed WPS as function of the apparent wind direction AWA. (a) kite: elevation angle δ = 30°; (b) Flettner rotor: values for AWS = 10 m/s, the asymmetry of the polar is due to the C_l and C_d values selected for optimal performance; (c) Dynarig: For stern winds, the positioning of the sail α is not adopted continuously, but only stepwise, a practical approach to avoid resonance effects due to movements of the ship in the sea.

3.5 SPECIFICATIONS OF THE DISCUSSED WPS

While the kite shows high efficiency for stern winds, the Flettner has maximum efficiency for lateral winds, the Dynarig not only shows a high effect for lateral winds (Bermuda sail characteristics) but also a reasonable efficiency for stern winds (Figure 4).

To simulate the energy saving potential of the discussed WPS they have to be dimensioned and positioned on the selected ships. For the dimensions, somewhat realistic values were selected based on the realization of the Flettner rotors on the E-ship 1. The dimensions of the kite and the Dynarig are chosen so that the wind force is about the same for the three systems for lateral winds. Interactions between different units (rotors or masts) on the ship are neglected.

DR: Dynarig
A = 800 m² for each mast
H = 30 m: average height of area exposed to wind
N = 3: number of masts

KI: Kite
A = 800 m²: area of the kite
H = 150 m: mean sailing height of the kite
δ= 30° elevation angle
P_{EE}=2 kW: energy for the kite motor, positioning and moving the kite
η_{EE} = 0.9 efficiency to produce electricity on board

FL: Flettner
H= 25 m height of the rotors
D = 4 m: diameter of the rotors
DS = 8 m: diameter of the end disc on the rotor:
DS = 2* D for the C_d, C_l values used
H= 30 m: mean height of the wind field used:
n_{max}= 200(1/min) maximal rotation speed
η_{EE} = 0.9: efficiency to produce electricity on board
N=4: number of rotors

4. ROUTE OPTIMIZATION

For an optimal use of the WPS on a route the course of the ship can be adjusted to select the route with the best weather conditions for the WPS. An appropriate route optimization tool (including the influence of wind and waves directly on the ship) has been developed (Bentin *et al.*, 2016). Figure 5 shows an example for a shortest route (great circle) and an optimized route. Routes with a travelling time longer than 20% compared to the travelling time on the great circle GC and routes, where the ship encounters bad weather (e.g. waves over 5 m height,) are discarded.

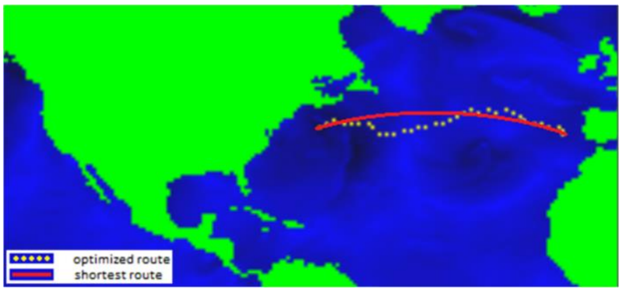

Figure 5: Ship course changed from the great circle GC (red) to an energy optimized route RO (dotted line), when using a WASP (Bentin *et al.*, 2016)

5. RESULTS

The energy saving potential depends on the ship, the ship performance (ship type, ship speed, ship course), the technology and size of the WPS and the weather conditions. To demonstrate the saving potential by the discussed WPS the energy consumption of the 3 discussed ship types and the three discussed WPS have been simulated on the three selected routes (chapter 2). The energy saving is simulated for courses on the great circles (GC) and for optimized routes (RO). The results are presented for a given fixed ship speed of 13 kn (STW). Figure 6 shows the simulation results for a multi-purpose carrier (MPC) on the route from Baltimore (USA) to Wilhelmshaven (Germany). Mean values and the distribution of the simulated values are calculated for one journey per week with historical weather data from the year 2008. As weather data, the historical analysis data of the "Deutscher Wetterdienst" DWD were taken along the route at the specific time, the ship being at that position of the route.

The results for the following scenarios are presented:

(a) Energy saving by RO without WPS ("Sea-margin")

The additional energy needed due to wind and waves (sea margin) is shown in Table 1 for the 3 selected routes. With the route optimization, the sea margin can be reduced significantly on routes with strong winds (North Atlantic).

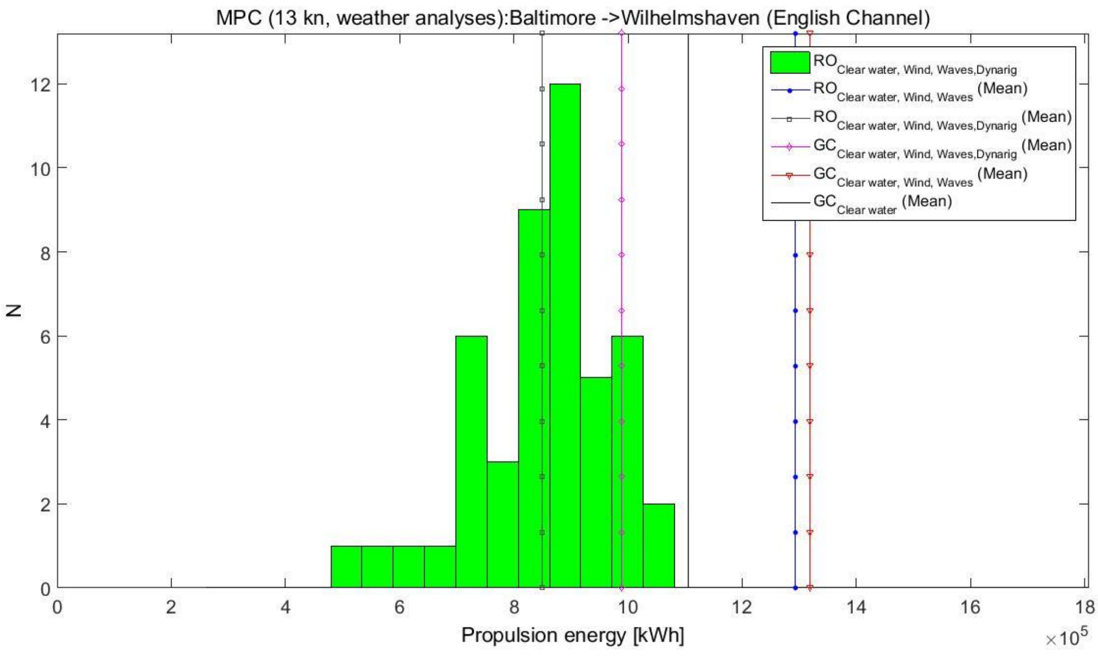

Figure 6: Energy consumption of a MPC sailing with 13 kn from Baltimore to Wilhelmshaven once a week in the year 2008. The perpendicular lines in the histogram represent long-term mean ship propulsion energy values for the respective settings that are shown in the legend. The mean values for the energy consumption are given for clear water resistance only, shown as well are the case with and without Dynarig and with route optimization (RO) and without route optimization (using the great circle route GC), but considering additional resistances by wind and waves. The histogram shows the distribution of energy values for a ship with a Dynarig considering clear water, wind and wave resistances; weather data = analysis data 2008

Table 1: Sea margin (in % of total energy consumption) for the 3 discussed routes, each route with RO and without RO (GC). Multipurpose carrier (MPC), 13 kn, mean values for one year (2008) with one journey / week; GC=great circle; RO = route optimization for minimum energy; > = from - to

Course / energy demand [%] of total energy	Sea margin on GC	Sea margin for RO	Saving by RO
Baltimore>Wilhelmshaven via Engl. Ch. Wilhelmshaven>Baltimore via Engl. Ch.	21 39	17 31	4 8
Nuevo Palmira>Emden Emden>Nuevo Palmira	20 13	18 12	2 1
Bergen>Wilhelmshaven Wilhelmshaven>Bergen	15 16	13 14	2 2

Table 2: Saving potential of the propulsion energy by different WPS in % (Flettner, Dynarig, Kite) on the route BA(Baltimore) –WHV(Wilhelmshaven) via English Channel, Vs=13kn, for a course on the GC and on an energy optimized course (RO)

Course /saving[%]	Flettner Rotor	Dynarig	Kite
BA>WHV; GC	21	25	23
BA>WHV; RO	31	35	29
WHV>BA; GC	14	21	10
WHV>BA; RO	24	33	19

Table 3: Saving potential with the Flettner Rotor on the 3 routes (in % to the sailing considering wind and waves, without WPS) for a MPC, Vs=13kn. compared for sailing on the GC of on the optimized route (RO).

Course / saving[%]	Flettner on GC	Flettner by RO
Baltimore>Wilhelmshaven via Pentland Wilhelmshaven>Baltimore via Pentland	24 15	35 27
Nuevo Palmira>Emden Emden>Nuevo Palmira	7 11	13 15
Bergen>Wilhelmshaven Wilhelmshaven>Bergen	15 16	20 19

(b) Energy saving by WPS on the North Atlantic route

In Table 2 the saving potentials (in % of the energy without WPS) for the 3 discussed WPS are compared on the North Atlantic route (for both directions), all WPS showing lower savings from east to west (upwind). The Dynarig gives the highest saving (up to 35%), especially on the route from east to west.

(c) Energy saving for different routes

In Table 3 the saving potential is shown for the three discussed routes when using a Flettner rotor. The North Atlantic route gives the highest saving of about 27– 35 %, when travelling on the optimized route. On the route to South America the weather is usually rather stable with moderate winds, yielding a saving potential of about 13% - 15%, rather independent from the shipping route direction. On the North Sea route (WHV-Bergen) the saving potential is about 20%, rather independent on the shipping route direction since the route travels perpendicular to the dominant wind direction (from west to east).

(d) Travel time

Figure 7 shows the travel time for the route Wilhelmshaven – Baltimore with and without RO. The increase in travel time due to the RO is in the order of about 5 hours as yearly mean value (maximum is 17 hours). Compared to the total travel time of 294 hours. it is about 2%. This is within the scope of the usual uncertainties of tramp shipping. It could be brought to zero, when adjusting the ship speed for a defined arrival time.

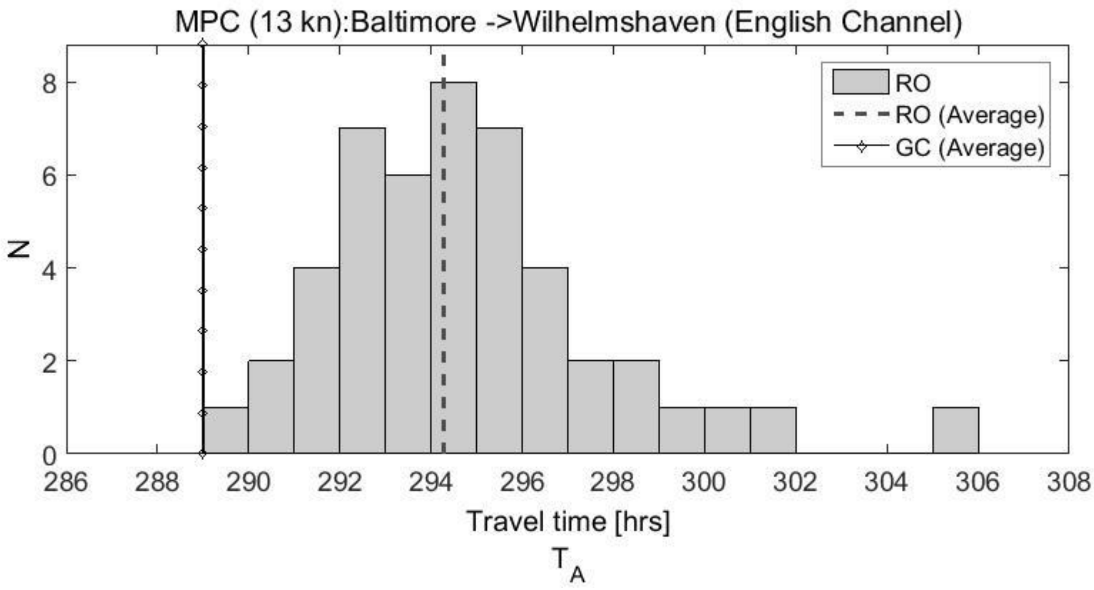

Figure 7: Distribution of the travelling time with the use of a WPS (here a Flettner rotor) with route optimization RO, compared to the travelling time on the great circle GC. Data obtained for sailing once a week from Baltimore to Wilhelmshaven; weather data = analysis data of 2008

6. UNCERTAINTY

The saving potential has been calculated by using the "actual weather data" (analysis data) along the route at the specific time. When planning an optimized route, it will be done on the basis of forecast data. A method to estimate the uncertainty of the energy saving with forecast data is developed as well (Zastrau, 2016).

7. ECONOMY

The realization of a WASP will mainly depend on the economy, but also on the impact of ship emissions on the environment (legal restrictions).

The actual cost-effectiveness mainly depends on

- The fuel price,
- the additional investment for a WPS,
- additional operating costs due to a WPS.
- The ship's speed

For a rough estimation of the investment for a WPS (without developing costs) it can be shown that a WPS on the discussed ships on windy routes such as the North Atlantic might be profitable for a fuel prices of the order of 500 € /to (profitability after 5 years) or 250 €/to (profitability after 10 years) (Schlaak, 2016). In these calculations, no costs due to the environmental impact are considered. Details will be discussed elsewhere.

8. CONCLUSION

The limited availability of oil in the future, connected with expected rises of fuel prices, and the impact of the emissions on the environment by nonrenewable energy, leads to the necessity to look for alternatives to the actual ship propulsion technology. There are several technologies under discussion as the use of LNG, or the use of wind assisted ship propulsion (WASP). Here the saving potential of WPS is calculated by simulating the energy consumption of selected ship types using a kite, or a Flettner rotor or a Dynarig sail as additional propulsion aid. The saving of energy by the different WPS are compared. It can be shown that on windy routes (as on the North Atlantic) a saving potential up to 35 % on optimized routes could be achieved with WPS which do not change the actual function of the discussed ships (speed, performance). With the price of the actual fuel used in seagoing ships a profitability is not yet given. However, considering the needed changes in the future due to the impact on environment and expected fuel prices developments, it can be deduced from this reported research that a development of WASP technologies might be reasonable.

9. ACKNOWLEGDEMENT

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AN ICOR APPROACH TOWARDS SHIP MAINTENANCE SOFTWARE DEVELOPMENT

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SUMMARY

Ship maintenance is one of the key processes to improve system performance and reliability onboard. Maintenance onboard ships is required to be performed in a planned manner because safe and efficient operation of a ship is very much depending on equipment reliability in operational level. A planned maintenance system (PMS), mainly supported with software, is established to monitor the maintenance implementations on board ships. This study aims to assess the effectiveness of onboard maintenance and limited with the procedures and specific softwares employed for the maintenance on board. Specifically, the process analysis approach is developed to execute the existing ship PMS phases based on four dimensions such as input, control, output, resources (ICOR). Besides the critical points to be improved in ship maintenance, this research reveals recommendations to increase the functionality of selected software in terms of workload balance, smart scheduling and safe working environment. To demonstrate the functional improvements, an illustrative case study on an auxiliary diesel generator maintenance routines are conducted. The contributions of the study are expected in both ship operations management and marine software development.

NOMENCLATURE

ICOR	Input, Control, Output, Resources
IMO	International Maritime Organisation
ISM	İnternational Safety Management
OM	Opportunistic Maintenance
PMS	Planned Maintenance System

1. INTRODUCTION

Maintenance is recognized as one of the major jobs facing an organisation to sustain the businesses effectively. It serves as a force multiplier and deemed as the primary contributing job to all processes of an organization to provide services or products at a higher level than its competitors. It also increases the reliability of all the systems used in service providing and in the production processes. Failure in doing maintenance adequately can cause significant costs. These costs are mainly caused by unpredictable failure of the system that is being used because of the missing maintenance and sometimes been unaffordable for a company in due course.

Due to the importance of maintenance, this challenging issue has forced many ambitious industrial organisations to place a high priority on preventive maintenance. In modern industrial organisations, maintenance departments have important roles because of their ability to have an economic balancing effect (Dekker, 1996). Even if there are specifically designed maintenance organizations and specific importances on maintenance processes, in many cases in actual practice, serious accidents and interruptions in many industries are related to poor maintenance, or, more specifically, to poor maintenance planning (Reason and Hobbs, 2003). Maritime industry faces unique challenges in the execution of scheduled and unscheduled maintenance; the mere fact that ships spend significant periods at sea far from the logistically supported areas (Rustenburg et al., 2001). A ship at sea is isolated from onshore repair and maintenance facilities, and, if a failure occurs during the passage, the required replacement parts may not be available on board. The rising cost of ship operation is a problem, since the failure of a vital piece of equipment can be very expensive and may put the safety of the whole ship at risk. Added to this is the cost of out of service (off hire), when the ship is in the downtime period because of the failure. Additionally, it may cause increases of the cargo expenses and may pose danger to the environment as well (Rothblum, 2000).

Maintenance is a crucial factor in a ship‘s performance and, in turn, can heavily affect the shipping company‘s revenue. There should be an optimal maintenance level for all the equipments on ships and therefore a balance between maintenance cost and over-maintenance. Over-maintenance is the term used for the case of excessive use of maintenance activities more than required level. Thus, establishing a good framework with which to measure maintenance performance to have an optimal level and to plan maintenance policy has vital importance for the shipping organization. In fact, approximately 40% of the operation costs in a maritime shipping organisation are attributable to maintenance (Alhouli et al., 2009). To keep the ship operating safely and efficiently with the lowest possible operation cost, ship maintenance is a critical process due to the various factors. Maintenance planning and its costs have great effects not only on technical management but also on commercial management of ships. Factors that affect the PMS for the planned maintenance optimization are vital to ensure that the maritime companies are able to save from the cost of maintenance budget. Indeed, developing a cost effective maintenance framework for ships has a critical importance in shipping organizations. The PMS is basically based on the calendar times of the objects or on actual working hours. The maintenance intervals are determined based on the calculation of the average failure times between the two maintenance periods (Mobley, 1990).

This study aims to develop a framework to aid decision makers in the maritime shipping industry to optimize maintenance planning for their organizations. Accordingly, the main objectives of the paper are as follows;

a) To identify the need and importance of maintenance with a specific focus on the maritime industry; and to extract and assess the factors that affects the decision-making process for ship maintenance planning.
b) To evaluate and compare different maintenance types for indicating the best maintenance planning method for an effective scheduling.
c) To identify basic & advanced tools of process analysis and improvement for using of developing maintenance planning system onboard.
d) To investigate the implementation principles of the ship planning maintenance system onboard and the software supports.

The paper is organized as follows: The initial part underlines the importance of equipment maintenance on board. Then, the existing theoretical developments in maintenance management and ship maintenance applications are discussed in the literature review. After providing brief information about process improvement techniques, ICOR approach is structured through ship maintenance process. After data analysis, findings are recognized and listed to utilize in maintenance planning and software functional development. Additionally, a specific maintenance planning of chosen equipment (i.e. a diesel generator) and its application is taken as an example in the study and evaluations made are compared with the other findings to produce results.

2. LITERATURE REVIEW

Recently, maintenance operations management is a highly cited research theme in the literature. The industrial representatives have closely interested in the studies provide new approaches to the ship maintenance supported with the field studies. Garg and Deshmukh (2006) encouraged the researchers and maintenance professionals to understand the importance of maintenance management.

Regarding the theoretical insights, various number of review studies on the developing concepts are addressed such as e-maintenace (Levrat et al., 2008), machine prognostics (Peng et al., 2010), maintenance optimiztion (Sharma et al., 2011). In detail, the maintenance process seeks for the innovative approaches and models to manage maintenance performance (Simões et al., 2011). In addition, maintenance strategy selection (Bevilacquaa and Bragliab, 2000; Al-Najjara and Alsyoufb, 2003; Wang et al., 2007; Arunraj and Maiti, 2010; Chemweno et al., 2016) is a focused optimization problem in different industries. Ab-Samat and Kamaruddin (2014) introduced the applications of opportunistic maintenance (OM) as new advance maintenance approach and policy. On the other hand, the importance of human factor in maintenance deals with the human error, macro-ergonomics, works planning and human performance (Sheikhalishahi et al., 2016).

In literature, there are limited number of research studies on the ship and offshore maintenance aspects. Deris et al. (1999) developed an algorithm to enable ship availability via maintenance scheduling process. The effects of maintenance to ship reliability was studied by Jiang and Melchers (2005), Lazakis et al. (2010) while Lee et al. (2013) suggested RFID supported cloud technology towards ship maintenance inspection. Goossens and Basten (2015) have cooperated with the owners, operators, shipbuilders and equipment manufacturers from the maritime community to identify the parameter triggers a maintenance action. Lee et al. (2016) proposed a maintenance system for LNG-FPSO via equipment-oriented condition based maintenance methodology. Regarding with the integration into safety management system, Akyuz and Celik (2017) promoted an e-PMS concept to monitor and imrove ship systems' conditions. Besides theoretical insights, the ship operators and shipboard personnel have interested in the utility of the maintenance systems onboard ships that are so critical issue. Especially, software tools to transform ship maintenance are highly important (Ali, 2015). The end user requirements of interfaces influence the effectiveness of maintenance system implementations. On this issue, the ongoing efforts of maritime research institutions, marine technology manufacturers and classification societies have been addressed. Therefore, this study is to be focused on identifying the critical points to be improved on ship planned maintenance system supported with software. The advanced studies (Lazakis et al., 2016; Lazakis and Olcer, 2015; Varelas et al,, 2013) through ship maintenance, operations and inspections onboard are also available in literature.

3. PROCESS IMPROVEMENT

3.1 TECHNIQUES

The definition of a process consists of structured tasks that produce a specific service or product to address a certain goal for a particular actor or set of actors (Glykas, 2013). The development of a system depends on the improvement of the processes in this system. The new and different ideas, views, and methodologies are required for development. Accordingly, the process improvement has the following characteristics: implementing incremental changes, starting with the existing process, carrying out one-time/continuous application, etc. (Damij and Damij, 2014). In process improvement, the practitioners have been using various tools and techniques. Just to name a few; DRIVE (define, review, identify, verify, execute), Process mapping, ICOR (inputs, outputs, controls and resources), Process flowcharting, Force field analysis, Cause & effect diagrams, CEDAC (cause and effect diagram with addition of cards), Brainstorming, Pareto analysis, SPC (statistical process control), Control charts, Check sheets, Bar charts, Scatter diagrams, Matrix analysis, Histograms, etc. (Assi, 2016; Boutros and Cardella, 2016).

In the application of techniques, it is required to break down the core processes into sub-processes, activities and tasks. Once an organization has defined and mapped out the high-level core processes, people need to understand what activities are required within these core processes and how they combine at operational levels (Oakland, 2014). This study takes the advantage of ICOR (inputs, controls, outputs and resources) in process analysis of ship maintenance.

3.2 PROCESS IMPROVEMENT TOOL: ICOR (INPUTS- CONTROLS- OUTPUTS- RESOURCES)

Considering the process improvement tools and techniques, it was deemed that ICOR is a convenient method for analyzing planned maintenance system onboard ship. ICOR (inputs, controls, outputs, and resources) is a process analysis methodology allows processes to be broken down into simple, manageable and more easily understandable units. During the analysis, all the sub-processes that constitute the main process and the components in the pool are examined. All the sub-processes in the pool are distrubuted seperately to the components of the ICOR system, Input, Control, Output, Resources. For instance, the regulations and relevant management systems are considered as Control while the Resources represent competencies in the different levels of organization (Badiru and Osisanya, 2013). By doing so, important shortcomings and the sub-processes needs to be improved to solve the problems could be easily defined for a PMS.

3.3 ICOR APPLICATION TO SHIP MAINTENANCE

The sub-processes, cases and the add-ons affecting the planned maintenance system onboard ships are as follows:

- Equipment conditions
- Running hours
- Spare Parts (Supply Chain)
- Manpowers (Number of Manning)
- Financial Budget
- Marine Suppliers (Vendor Evaluation)
- Knowledge & qualification (Training of Crew Members)
- Audits (internal Audit, External Audit)
- Safety Check List (Deficiencies)
- Performance Checks (Malfunctions)
- Procedure & Regulations (Documentation, Administration Requirement)
- Maintenance Schedule (Smart)
- Operation Planning
- Scope of Maintenance

To be able to analyze the PMS process by the ICOR method, it is necessary to determine in which group the above-mentioned sub-processes and annexes belong to and under which previously determined components they should be grouped. After sub-processes and annexes are grouped, the diagram in Figure 1 is drawn to show the analysis of what the sub-process affects and what other processes are needed to be developed.

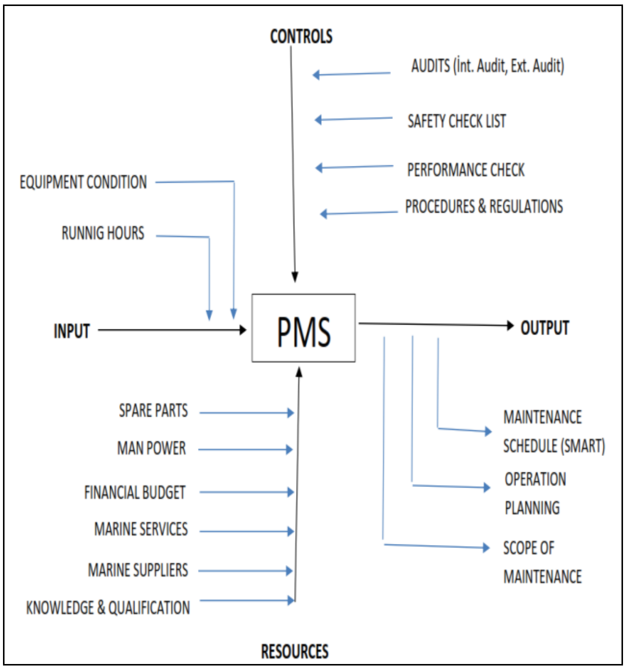

Figure-1: ICOR Analysis on Ship PMS Implementations

In the analysis, excessive and unbalanced workload has been attracted a special importance to identify and to solve the problems. Additionally, improper and untimely maintenance planning and safe maintenance operation problems detected and reported by 3rd party inspections and company / sector experience were noted. For analysis purposes, an ICOR control matrix was developed and shown in the Table 1.

In the first row of the table, the problem to be improved is inserted. The second row contains the corresponding sub-process in the Input group of the ICOR drawing, the third row of the corresponding sub-process in the Control group, the fourth row of the corresponding sub-process in the Output group, and the fifth one sub-process of the Resource group.

Table 1: ICOR Control Matrix

PMS SUBJECT	Workload Balance	Smart Schedule	Safe Maintenance Operation
INPUT	Equipment Condition Running hours of Engine	Equipment Condition Running hours of Engine	Equipment Condition
CONTROL	Safety Checklist	Performance Check	Safety Checklist Procedures & Regulations
OUTPUT	Scope of Maintenance	Operation Planning	Operation Planning
RESOURCE	Manpower	Spare Parts Financial Budget	Manpower Marine Services Knowledge & Qualification
ACTION	Safety checklists must be prepared specifically for ship equipment& crew members. (Critical & stand by equipment) Running hours of the engine and equipment conditions must be considered to determine the scope of maintenances done by the crew members, Sufficient manpower must be provided considering related equipment condition.	Performance checks of equipments must be considered Operation plan should have no conflict with the work schedule. Supply of Financial budget and spare parts for all equipments considering their Running hours has priority in all activities of PMS	Safety checklist properly must be prepared prior to maintenance Safety procedures must be adopted Qualified manpowers must be provided Marine service evaluation must be considered

The categorized processes in the table were analyzed to determine the necessary corrections for the targeted improvement. In order to overcome workload imbalances, it was found that preparation of safety check lists, scope of maintenance and amount of manpower are the three main items needed to be improved. For smart scheduling, possible corrections to be done are centered on the improvement of performance checks of equipments, consistency of operation plan and PMS work schedule and supply of spare parts. The safe maintenance operation deficiencies were analysed and timely preparation of the safety checklists, adaptation of PMS procedures to the ship safety system application, qualified manpower needs and requirements to outer management support subjects were found as sub items to be improved. All those sub items are discussed in the following subsection.

3.4 FINDINGS

The objective of the aplication was to find what should be improved for solving the problems and to restructure an enhanced planning maintenance system onboard ship. By applying the ICOR method to planned maintenance systems onboard, three important shortfalls have been analyzed as explained in the previous section.

3.4 (a) Workload Balance

Workload balance is the first of the above-mentioned shortfalls. Balancing workload onboard is the key component of defining the number of crewmembers onboard the ship. Minimum safe manning certificate is the document of the vessel which is issued by the flag state of the vessel that shows the minimum number and competency of crewmembers for engaging intended voyage in international waters. Most of the times, number of crewmembers indicated on the minimum safe manning certificate is not enough for maintenance and routine operation of the vessel and this situation does not comply with regulating international documents such as Maritime Labor Convention- 2006 of ILO.

Ship's management companies, if it is the case, generally issue common safety checklists and control forms for their ships under management. Unfortunately, those control forms sometimes cause a waste of time for responsible persons. Also this may lead to insufficient or unnecessary controls of equipment & tools. Hence, checks lists must be prepared by considering the specification of the ship equipment and crewmembers to reduce the workload (for critical-stand by equipments).

In order to identify the exact required number of employees onboard, ship's master and operators should consider the ship condition, voyage pattern and number of running equipment by taking into account the allowable maximum number of persons onboard which is indicated in the ship safety equipment certificate.

3.4 (b) Smart Scheduling

Regularly done performance checks or using of condition monitoring systems are the main ways of structuring maintenance programmes of ships. This ensures that, focusing to the maintenance of which equipment needed. Operation planning is another important part of the smart scheduling of the maintenance onboard ship. Most of the times, planned maintenance made without considering vessel's operation may cause that maintenance could not be carried out within planned periods and remain of overdue maintenance. On the other hand, lack of consideration for the shipboard operations and tool availability causes delays and suspensions of shipboard operations. An effective maintenance planning ensures the maintenance of ship's equipment & tools is successfully performed without suspension of the operation of the vessel.

Smart scheduling also requires some kind of supply activity. Before preparing a maintenance plan and starting maintenance of equipment & tools, necessary spare part must be available onboard. Therefore, at the beginning of the fiscal year, ship budget must be prepared considering needs based on the principal maintenance plan.

3.4 (c) Safe Maintenance Operations

Providing safety is not only an aim but also a process of maintenance onboard. Safety measures to be taken have special importance in the maintenance activities on board ship. Carefully prepared procedures for before, during and after maintenance activities and safety control check lists play key roles of the safe maintenance operations.

Some of the maintenance operations onboard are carried out by third parties like as marine technical services. Under these circumstances, quality of marine technical services and approval of specific manufacturer authorizations play a key role. Additionally, Company's marine technical services' evaluation reports provide important feedback for future operations. Their qualification, experience and knowledge levels on safety issues provide important inputs to ensure safe maintenance operation.

3.4 (d) Other Issues

In addition to the findings above, todays contemporary ship management compared to past experiences becomes a more important job in which has considerable amount of detail is managed and completed with many factors. Main goal of any given company to achieve profit with simple methods has become difficult for the shipping industry. Safety, quality, assessment and measurement, follow ups, procurement, giving swift yet true decisions are only some of the critical business requirements. These goals are slowly but steadily becoming impossible to achieve mainly by human operators; thus, data processing automated systems are becoming a necessity. Multiple ship management companies working in collaboration within this application framework to resolve these critical issues to attain a perfect standardization would result in optimum solutions.

3.5 INVESTIGATION OF FINDINGS

After analyzing the problems, the main functions of the system to be developed were found as follows:

- F1: Reducing operating cost and increasing operation time of equipment.
- F2: Reducing defects and downtime,
- F3: Defining root cause of the problem,
- F4: Ensuring safe operations of ship.

In order to develop the planned maintenance functions described above, the process improvement techniques denominated common goals are utilized. When examining both basic and advanced improvement techniques, it was realized that the common goals of many techniques could be as follows:

- M1: Strengthen cross-functional teamwork, problem solving, and collaboration capabilities,
- M2: Develop effective process controls to prevent recurrence of known problems,
- M3: Analyze the process for efficiency, effectivity, and waste,
- M4: Enhance process effectiveness,
- M5: Simplify planning and scheduling,
- M6: Establish standardization.

Under the light of the above consideration, Table 2 has been drawn. Common goals and functions that are problematic in the systems are compared with each other and the evaluations are reflected in Table 2.

Table 2 Functional Improvement Options Matrix

	Improvements					
Functions	*M1*	*M2*	*M3*	*M4*	*M5*	*M6*
F1	X	X	X	X		X
F2	X	X	X	X	X	
F3	X	X		X		
F4	X	X		X	X	X

Providing in Table 2, defined improvements can be applied to variety of functions in the PMS. The implementation of specified improvements could be made on the selected software.

Within the scope of these improvement options, not only the development of the PMS but also the development of a complex ship management system, and the optimization of critical safety problems could be utilized by software development teams and shipping company's technical managers. Increasing the reliability of automation systems with the help of ship management software is an important subject of today for the requirements of contemporary maritime industries are met. The optimization and development of more effective PMS systems which are out of the scope of this study could be addressed in the future.

4. SOFTWARE DEVELOPMENT

4.1 EXISTING PRACTICE

Today, PMS is performed by many of the shipping companies with support of the computer aided systems. As an example, one software application was chosen to produce meaningful results in this study. The ICOR analysis results were used to investigate and to improve the maintenance software functionality, which is used in the ship management company. In practice, all the equipments onboard are included in the planned maintenance system. Maintenance intervals of each equipments are planned and scheduled by PMS with support of special software. As an example, in the study, a specific planning and execution of a diesel generator maintenance was examined to check the findings.

An engineer onboard has to be capable of doing overhaul of a diesel generator during routine maintenance or in case of emergency situation. This competency is compulsory for all level engineers working on ships. Overhauling of generator not only means cleaning and removing carbon from parts and spaces involved in combustion of fuel but also includes checking, fixing and renewing of parts involved in power transmission like connecting rods, con-rod bearings, main bearings etc. According to a research (Wankhede, 2013)**,** 50% of marine engineers who are at the initial stages of their sea careers are unaware of the procedure for overhauling generators. Many of them do not have enough confidence to carry out the overhauling alone or are reluctant to take the responsibility without the assistance of senior personnel. This shows that capable person in doing an effective maintenance onboard has a special importance.

4.2 ILLUSTRATIVE EXAMPLES THROUGH SOFTWARE FUNCTIONS IMPROVEMENTS

Overhaul process of a diesel generator which is examined is as follows:

a) Monthly reports sent by ship to the company as paper form including performance records and working hours are recorded to the PMS.
b) Overhaul is planned by considering performance records and running hours of the engine.
c) Necessary spares are listed and their availability are checked. Requirements are defined and ordered via company procurement and supply branch.
d) Time frame and location of the overhaul is planned.
e) Overhaul process is executed according to its scope.

The fact is that overhauling of ship's generators is not a part of a daily routine task of engineers. It is a long and tiresome procedure which is carried out when the generator reaches the time for scheduled maintenance except of sudden breakdown of it. Overhauling of generators requires hands-on knowledge and experience along with thorough understanding of step-by-step procedures for maintenance of auxiliary engine overhaul which this complexity was solved by help of software. Before and during the overhauling process, a variety of tests are performed on various tools and parts of the generator such as; hydraulic jack test, cylinder head test, bearing cap test, con-rod bolts test, connecting rod bend test, fuel Injector test, starting air valve testing, relief valve test, the current test, alarm and trips test.

Ensuring that safe working practices are followed while carrying out generator overhauling is important for engineers on board ship. As such processes usually require working as a team, miscommunication in the maintenance team can result in unfortunate accidents. It is therefore necessary to follow systematic guidelines and step-by-step procedures for carrying out such complex work in the engine room. The following procedures to be checked to figure out the problems show the complexity of the diesel generators' overhaul. There are many reasons, which will lead to reduced power output and performance of a diesel engine on board ship such as; fuel oil pressure too low, type of fuel used onboard, fuel leakage,fuel temperature, firing pressure difference, blocked filter, wrong valve clearance, damaged exhaust valve, high exhaust back pressure, contaminated passages, insufficient fresh air supply, high suction air temperature to T/C, charge air pressure too low, wrong charge air temperature, charge air cooler contaminated, air cooler S.W temp high, blower turbine or nozzle ring worn/ damage, scavenge air leakage, wrong tappet clearance setting etc.

4.3 REVIEW OF IMPROVEMENTS

After examining the maintenance application of diesel generator onboard, the software application on PMS and the ICOR findings, for a qualified planning and execution of maintenance, the following areas should be considered:

a) Auxiliary diesel engine breakdown may occur anytime each year. Because there is a thin line between the starting of a problem and the problem taking the shape of major issue and it BE was only a ship's engineer who can assess this situation. Regularly done performance checks or using of condition monitoring systems are the main ways of structuring maintenance programmes of ships. This ensures that, focusing to the maintenance of which equipment needed. Thanks to software equipment performance and running hour informations received as simultaneously on basis daily. M2 and M6 improvements from Functional Improvement Options Matrix require special attention.

b) Major overhauling operation should not only be decided by taking into accout the equipment performance and running hours for a ship used to for commercial activities, but also the commercial activities and agreements except sudden failure and unfortunate accident case. Although safety and security of the ship is primary concern, it is vital importance to evaluating the ship commercial activities. The software which had been structured with ICOR philosophy should meet this requirement. This requirement is also dictates the improvement of M3 and M4 of Functional Improvement Options Matrix.

c) Overhaul of an auxiliary diesel engine brings about many problems to be solved. More systematic structure is necessary for accurate and quickly determination in problem solving due to nature of the problem. There may be encountered some problems that require immediate actions during overhaul of diesel generators. Simultaneous follow-up and records of maintenance history provide holistic view and sharp forecast about the root cause of the problem. So, defining the scope of the overhaul, necessary spares for the completion of the operation more certainly were known. In other words, it has helped for mentioned preparation as decision support system. Before preparing maintenances plan and starting maintenance of equipment & tools, necessary spare part must be available onboard. Therefore, at the beginning of the fiscal year, ship budget must be prepared with considering needs based on the principal maintanance plan. By means of integration between ship to shore and software data has contributed improvement of M1 of Functional Improvement Options Matrix.

d) Planning of the overhaul operation is made without interrupting any other ship operation by taking into account the voyage and operation information data which are obtained from the integrated system. This could be achieved only with having the maintenance be finished within planned time frame. Most of the times, planned maintenance made without considering vessel's operation may cause that maintenance could not be carried out within planned periods and remain of overdue maintenance. On the other hand, it could cause suspension of operation of vessel due to the maintenance lacks of equipment & tools when maintenance activities have been started without considering operation of the vessel. An effective maintenance planning ensures the maintenance of ship's equipment & tools is successfully performed without suspension of the operation of the vessel.

e) A successful and safe overhaul operation depends on the certainty of the scope of the overhaul operation and arrangement of proper manpower. Equipment performance and running hour data helps to identify scope of overhaul. Then, this scope defines the necessary average manpower reuirement. Holistic approaches ensured decision support mechanism about to make equipment overhaul operation within the planned time interwal by either the crewmember or outsourced manpower considering crewmember's other duties and their working hour of them. Applicability of the improvements of M5 and M6 were observed here. In order to identify the exact required number of crew to be employed on maintenance activities onboard, ship's master and operators should consider the ship condition, voyage pattern and number of running equipment. Actually, not only the minimum safe manning certificate of the ship but also an effective planned maintenance system defines the number and quality of the ship crew. Shipowners should take into account the requirements of PMS onboard primarily in defining ship crew.

f) Safe maintenance operations are the most important part of the maintenance onboard ship. Carefully prepared procedures for the before, during and after maintenance activities and safety control check lists play key roles of the safe maintenance operations.

5. CONCLUSIONS

This study concentrates on the PMS implementations which influence efficiency, reliability, safety and organizational effectiveness. The planning factors for an effective PMS are consisting of selecting a convenient strategy for maintenance, making the smart maintenance schedule, selecting qualified ship's crew member, selecting convenient shipyard for dry dock of the ship and desinging the ship with consideration of maintenance planning from the construction stage. Indeed, the effective maintenance has great potential to increase the system reliability.

Using of ICOR model in software system as core philosophy ensures solving of each item in detail with holistic approaches which consider interaction with other components. Identified improvements has found applicable from M1 to M6 to reach the targets identified from F1 to F4 shown in the Functional Improvement Options Matrix defined in the previous part of the study.

Practical applications reveal various operational improvements such as workload balance, smart scheduling and safe maintenance operation. Through the integrated system software with using of ICOR method philosophy not only lessons for the development of PMS but also contributions for the improvement of the preventive / predictive maintenance were obtained. Thus, early and sharp forecasts about reason of the breakdown and or malfunctions could become available.

The study also demonstrates the several points to be improved along with the existing fuctions of the utilized softwares, given as follows: i) Reducing operating costs and increasing operation time of equipments, ii) Reducing defects and downtimes, iii) Defining root cause of the problems, iv) Ensuring safe operations of a ship.

ICOR method is not only used for maritime sector but also can be used in many branches of the industries such as logistic, transportation and nuclear energy. Furthermore, it could be used as infrastructure of artificial intelligence or decision support systems, for analysis of enhanced integrated data such as new source, outputs, inputs and controls day by day. Todays industries are getting more complex each day because global economy provides new inputs, outputs and sources to be managed for them. So, new and talented management systems are not only requirement for planning and execution of maintenance for shipping companies but also general requirement for all the industry.

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A METHODOLOGY TO SUPPORT EARLY STAGE OFF-THE-SHELF NAVAL VESSEL ACQUISITIONS

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SUMMARY

This paper describes a research programme to construct a Model-Based Systems Engineering (MBSE) methodology that supports acquiring organisations in the early stages of Off-the-Shelf (OTS) naval vessel acquisitions. A structured approach to design and requirements definition activities has been incorporated into the methodology to provide an easily implemented, reusable approach that supports defensible acquisition of OTS naval vessels through traceability of decisions. The methodology comprises two main parts. Firstly, a design space is developed from the capability needs using Set-Based Design principles, Model-Based Conceptual Design, and Design Patterns. A key idea is to employ Concept and Requirements Exploration to trim the design space to the region of OTS designs most likely to meet the needs. This region can be used to specify Request for Tender (RFT) requirements. Secondly, the methodology supports trades-off between the OTS design options proposed in the RFT responses using a multi-criteria decision making method. The paper includes an example implementation of the methodology for an indicative Offshore Patrol Vessel capability.

NOMENCLATURE

ADO	Australian Defence Organisation
AHP	Analytical Hierarchy Process
C&RE	Concept and Requirements Exploration
CONOPS	Concept of Operations
DBB	Design Building Block
DSE	Design Space Exploration
ESWBS	Expanded Ship Work Breakdown Structure
INCOSE	International Council on Systems Engineering
IPSM	Integrated Platform System Model
KPP	Key Performance Parameter
M&S	Modelling and Simulation
MAUT	Multi-Attribute Utility Theory
MAV	Multi-Attribute Value Analysis
MBCD	Model-Based Conceptual Design
MBSE	Model-Based Systems Engineering
MCDM	Multi-Criteria Decision Making
MDAO	Multidisciplinary Design Analysis and Optimisation
MOP	Measure of Performance
MSC	Medium Security Cutter
NFR	Non-Functional Requirement
OEM	Operational Effectiveness Model
OTS	Off-the-Shelf
OTSO	Off-the-Shelf Option
OWV	Overall Weighted Value
PBSE	Pattern-Based Systems Engineering
RFT	Request for Tender
ROC	Rank Order Centroid
ROM	Rough Order of Magnitude
RSM	Response Surface Method
SBD	Set-Based Design
SE	Systems Engineering
SME	Subject Matter Expert
SysML	Systems Modelling Language
UNTL	Universal Naval Task List
US	United States of America
USCG	United States Coast Guard
VT	Virginia Polytechnic Institute and State University
WSAF	Whole-of-System Analytical Framework

1. INTRODUCTION

Given the increasing complexity and interoperation of military systems, the acquisition of new materiel solutions, such as naval vessels needs to be undertaken in the context of the overall national defence strategic setting (Hodge and Cook, 2014). Furthermore, it is recognised that developing requirements for a defence capability is a design process, the output of which is the definition of the materiel need (Hodge and Cook, 2014; Coffield, 2016; Cook and Unewisse, 2017) along with all the non-materiel aspects of capability. Systems of Systems Engineering approaches are gathering momentum in defence organisations around the world to capture and co-ordinate the wider defence context and are routinely used to define new capability needs. However, it is customary to find that this work needs to be enhanced by a project-specific capability design process performed by the individual capability acquisition project offices.

An important constraint on the capability acquisition process for naval vessels is the adoption of strategies that give preference to Off-the-Shelf (OTS) designs. This has become commonplace in countries with modest Defence budgets like Australia. In fact, the Australian government recently mandated the selection of a 'mature design' for naval vessel acquisitions (Defence, 2017), which has been interpreted to mean OTS solutions. OTS strategies change the nature of defence acquisition projects from the traditional top-down requirements-driven approach to a middle-out approach. This approach is based on defining the functions that are needed (capability goals) and then searching through existing OTS offerings to find the one that best satisfies the needs with the lowest level of customisation.

The OTS acquisition strategy for naval vessels appears to be analogous to the 'repeat', or 'modified-repeat' naval vessel design approach, since they both rely on adapting an existing design to address a naval capability gap. The modified-repeat design approach uses an existing design as the parent hullform, which is modified (to varying degrees) into what is assumed to be a 'mature' design (Keane Jr and Tibbitts, 2013). This is similar to many OTS naval vessel acquisitions, where the OTS design (the parent) is modified (to varying degrees) into what is promoted as a mature design. Both modified-repeat design and OTS acquisition have been perceived as a means of reducing the acquisition cost and schedule risks for naval vessel capability acquisition programs (Saunders, 2013 and Keane Jr and Tibbitts, 2013). An analysis of the cost and schedule benefits associated with the modified-repeat ship design approach showed these perceptions can be realised if the operational requirements for the new design are nearly identical to the existing design (Covich and Hammes, 1983). Furthermore, to maximise the potential of these approaches the existing vessel design will ideally still be in production, since evolving legislative requirements can necessitate significant design changes for older parent vessels (Covich and Hammes, 1983). Hence, to realise the benefits of lower acquisition cost and schedule risks in OTS naval vessel acquisitions, the project will need to identify existing OTS designs, or a region in the OTS design space, with very similar operational and legislative requirements to those for the new vessel and then specify tender requirements accordingly. Unlike the navy undertaking a modified-repeat design approach to address a capability gap, the OTS acquirer will not have knowledge of the parent design's requirements and design data. These aspects, as well as the aforementioned middle-out nature of OTS acquisitions, mean the OTS constraint presents a rather different class of challenge to the acquisition community; one that requires a different class of procurement approach and related methods, processes, and tools.

This paper describes a Model-Based Systems Engineering (MBSE) methodology constructed to support key OTS naval vessel acquisition project activities such as: defining requirements, selecting the preferred technical solution, developing and managing the early stage design information, and maintaining requirement and decision traceability. Figure 1 illustrates the temporal focus of the methodology described in this paper against various system lifecycle models. The emphasis is on the Risk Mitigation and Requirements Setting Phase of the Australian Defence Organisation (ADO) lifecycle and the corresponding early stages in other lifecycles. Andrews (2013) notes "it is often acknowledged that the initial (or concept) design phase is the most critical design phase, because by the end of this phase most of the cost is incorporated in the design…" The methodology seeks to improve the quality of the output of these early design stages using an easily implemented approach to support defensible acquisition of OTS naval vessels. The methodology comprises two main stages. The first stage is a model-based approach to ship Concept and Requirements Exploration (C&RE). This stage focuses on assisting stakeholders to build knowledge about possible OTS solutions to the capability needs. Knowledge is gained by exploring and progressively narrowing an existing OTS design space that is linked through appropriate Measures of Performance (MOPs) and Key Performance Parameters (KPPs) to the capability needs and constraints. This knowledge of the OTS design space supports the elucidation of a set of feasible and traceable request-for-tender (RFT) requirements. The second stage of the methodology is a model-based approach to option evaluation. This stage supports final design activities to refine the existing OTS design as well as the selection of a preferred design from those offered and refined in response to a RFT.

The paper opens with a review of some elements of early stage naval vessel acquisition that are incorporated into the methodology. These elements include Model-Based Conceptual Design (MBCD), Set-Based Design (SBD), Modelling and Simulation (M&S), Design Space Exploration (DSE), Multi-Criteria Decision Making (MCDM) and Pattern-Based Systems Engineering (PBSE). Together these provide defensible support to decision making during the early stages of naval vessel acquisition. After presenting the overall methodology, a brief exemplar implementation is given for a United States Coast Guard (USCG) Concept of Operations (CONOPS) for a Medium Security Cutter (MSC). Following a discussion on the findings from implementing the methodology, the paper concludes with suggestions for further work.

2. ELEMENTS OF EARLY STAGE DESIGN RELEVANT TO NAVAL VESSELS

The latest in a long line of reviews of the Australian Department of Defence, the First Principles Review, highlighted a number of recurring themes from earlier reviews (Peever, 2015: p. 92). Three of these themes provide the impetus for the guiding principles used in the construction of the proposed methodology:

1. Maintaining traceability to the original, strategic intent of the vessel being acquired in order to ensure a defensible outcome.
2. Assisting the stakeholders to make defensible decisions that account for competing goals and objectives.
3. Maximising the capacity to reuse elements – thereby reducing subsequent acquisition efforts to implement the methodology and the resources required to manage these projects.

With these principles in mind, a review of the literature identified six key elements for inclusion in the methodology. These are described in the following sub-sections.

2.1 MODEL-BASED CONCEPTUAL DESIGN

A key recent practice in early stage design is Model-Based Conceptual Design (MBCD). Reichwein et al. (2012: p. 1), state that "Model-based concept(ual) design is often used to allow engineers to describe and evaluate various system aspects". They highlight the wide range of models that can be used during conceptual design, which include (Reichwein et al., 2012: p. 1): mathematical models, geometric models, software models, system models, control system models, multi-body system models, requirement models and function models. An INCOSE MBCD Working Group (WG) was chartered in 2013 and has defined MBCD as "...the application of MBSE to the Exploratory Research and Concept stages of the generic life-cycle defined by INCOSE..." (Robinson, 2013: p. 1). Using MBSE during conceptual design has been found to provide a "clearer understanding of the problem space" (Morris et al., 2016: p. 11). Campbell and Solomon (2011) list some of the benefits of an MBSE approach, particularly in the Defence context as: process independence, reduction in overall effort required, improved accuracy of the output, provision by the tools of a central repository, and traceability throughout the whole project/product lifecycle.

In the Australian defence context, the Whole-of-System Analytical Framework (WSAF) MBCD approach has been applied to the early stages of many complex system acquisition projects (Cook et al., 2015). While naval vessel concept design has been described as having a 'wicked' nature (Andrews, 2013), the OTS constraint serves to effectively bound the initial problem space to one that can be clarified through the use of approaches from other domains. Since MBCD can incorporate MBSE and provides understanding of the problem space, WSAF succeeds in meeting the first and third principles outlined in Section 2. However, WSAF was not intended to support the engineering design and engineering analysis aspects of MBCD and additional elements are needed to cover these activities.

Although no specific mention of MBCD was identified in naval architecture literature, several examples of applying model-based methodologies during naval vessel conceptual design have been found. These methodologies and the features of MBCD they include are summarised in Table 1. While there are some issues associated with implementing MBCD in terms of engagement within organisations (Morris et al., 2016), the structure and traceability provided through MBSE, as well as the ability to reuse models, means MBCD adheres to the three guiding principles for constructing a methodology to support early stage OTS naval vessel acquisitions.

2.2 SET-BASED DESIGN

SBD is an emerging paradigm in naval vessel design (Singer et al., 2009). SBD differs from the traditional point-based iterative approach to design by using sets of values of design parameters, rather than a single value (Hannapel, 2012). Arising from a study of Toyota's automobile design approach in the mid 1990's, the features of the SBD process have been identified as (Parsons, 2003): broad sets are defined for design parameters to allow concurrent design to begin, these sets are kept open much longer than typical to reveal trade-off information, the sets are gradually narrowed until a more global optimum is revealed and refined.

SBD is claimed to offer two main advantages over the point-based approach (Hannapel, 2012). Firstly, the amount of design rework is reduced as SBD uses narrowing sets of design parameters rather than iterations of a single set of design parameters that may change from iteration to iteration. Secondly, design decisions are made when more information is available as the decisions are purposely delayed in the SBD approach.

In other literature covering naval vessel conceptual design, the principle of "*requirements elucidation*", rather than requirements engineering (Andrews, 2011), emerges. In this approach "…the initial design phase is characterised by the need to elucidate what the requirements should be…" (Andrews, 2012: p. 895). Andrews (2012: p. 895) also notes the consistency between the European *requirements elucidation* principle and the US SBD approach with the statement "…this more realistic emphasis in requirements elucidation can then (be) seen to be consistent with the approach of deferred commitment or SBD…"

SBD appears to build upon a proposal to use "concurrent engineering design" for ships from the 1990s. Both concurrent engineering design and SBD share themes regarding the benefit of having more information on which to base design decisions. Mistree *et al* state (Mistree et al., 1990: p. 567): "Conceptually, it is evident from any perspective that as a design process progresses and decisions are made, the freedom to make changes as one proceeds is reduced and knowledge about design increases … at the same time, there is a progression from soft to hard information." Both concurrent engineering and SBD are descriptive, rather than prescriptive models of design, hence their utility for the designer is diminished (Mistree et al., 1990: p. 567). However, they seem well suited to the early stages of OTS naval vessel acquisition as they focus on informing stakeholders on a conceptual design space, rather than providing information on a single point in that space (Morris, 2014). This means SBD adheres to principle two described in Section 2. In OTS acquisitions, there is no need to pursue a point-based approach, since the role of the acquiring organisation is to develop requirements that specify suitable OTS designs from within the OTS design space, as well as to identify any capability risk arising from the OTS constraint, not to produce a specific design.

2.3 MODELLING AND SIMULATION

Modelling and simulation (M&S) has been identified as being valuable for conceptual design for many years. Aughenbaugh and Paredis (2004) term the conceptual design phase of the system development lifecycle, *decomposition* as it aligns with the left hand side of the SE "vee" model (See Forsberg and Mooz, 1991 and Elliott and Deasley, 2007). Aughenbaugh and Paredis, while referring to early stage exploratory design, assert that M&S can help reduce the likelihood that requirements will not be satisfied later in the lifecycle by "supporting exploration of the design during the decomposition process" (Aughenbaugh and Paredis, 2004: p. 2). They also argue that M&S can inform decisions on trimming the design space during conceptual design by helping to "estimate the (system) attributes that would result from a particular decision" (Aughenbaugh and Paredis, 2004: p. 3). This means M&S can be used in OTS acquisitions to build knowledge of the performance characteristics of OTS designs without having specific design details. In turn, this knowledge could support the identification of a region within the design space containing OTS designs with similar operational requirements to those of the acquisition project.

M&S is an element of all of the naval vessel MBCD methodologies in Table 1. However, only two of the MBCD methodologies in Table 1 incorporated integrated MBSE and M&S (WSAF (Morris, 2014) and OTS C&RE (Morris and Thethy, 2015)) to combine MBSE's traceability benefits with the analytical rigour of M&S. It is worth noting these two MBCD methodologies utilised simple M&S models (parametric and surrogate models) to build a Rough Order of Magnitude (ROM) design space. The other naval vessel MBCD methodologies utilised more complex M&S models (Operational Effectiveness Models (OEMs) or the Design Building Block (DBB) model, which provides a vessel representation that can be simulated) and maintained either separate M&S and MBSE models, or no MBSE model. When discussing effective implementation of MBSE, Haveman and Bonnema state: "ideally, all models must be able to interact", whilst also noting: "currently, there are few approaches that effectively integrate high-level models in MBSE" (Haveman and Bonnema, 2013: p. 296). If MBSE and M&S models can be integrated, this will align with principles one and two as MBSE will facilitate traceability to the strategic intent of the capability. In addition, application of M&S during conceptual design can provide evidence to aid defensible decision making.

2.4 DESIGN SPACE EXPLORATION

Kang *et al.* define Design Space Exploration (DSE) as "the activity of discovering and evaluating design alternatives during system development" (Kang et al., 2010: p. 1). Other authors, such as Spero et al. (2014), along with Ross and Hastings (2005) refer to DSE as Tradespace Exploration, with Ross and Hastings defining the tradespace as "the space of possible design options" (Ross and Hastings, 2005: p. 2).

In naval vessel concept design, DSE is synonymous with C&RE, or "requirements elucidation" depending on which side of the Atlantic Ocean the author resides. Brown states: "During C&RE we use a total systems approach, including an efficient search of the design space…" (Brown, 2013: p. 2). Similarly, McDonald *et al.* (McDonald et al., 2012: p. 210) state: "the issue in the initial design of complex ships, such as naval combatants, is that the exploration should be as wide as possible so that all conceivable options are explored and the emergent requirements are "elucidated" from this comprehensive exploration." All the naval vessel MBCD approaches reviewed in Table 1 contained DSE in either a value-driven (C&RE), data-driven (RSM and WSAF), or informal manner, where a range of solution options within the design space were evaluated (SubOA, IPSM and DBB). In the OTS acquisition case, the concept exploration will be constrained to a search of the existing vessel design space.

2.5 MULTI-CRITERIA DECISION MAKING

An evaluation of responses to a request for tender (RFT) to select the most viable design needs to be performed prior to the acquisition stage of an OTS naval vessel acquisition. This evaluation is likely to be a focus of any oversight committee due to the typically large amount of taxpayer money at stake. The evaluation of naval vessel design options is a decision problem where consideration will need to be given to a number of competing objectives (e.g. performance and cost), as well as the views and knowledge of a range of stakeholders (Buede, 2000: p. 360). Multi-Criteria Decision Making (MCDM) is a field of research that has grown since the late 1970s (Mollaghasemi and Pet-Edwards, 1997) to deal with such decision problems. MCDM methods have been developed "to help the decision maker think systematically about complex decision problems and to improve the quality of the resulting decisions" (Mollaghasemi and Pet-Edwards, 1997: p. 3).

MCDM approaches typically fall into two categories: one to address either multiple-objective problems or multiple-attribute problems (Mollaghasemi and Pet-Edwards, 1997: p. 4). Multiple-objective problems are those with a large number of feasible solution alternatives, whereas multiple-attribute problems have relatively fewer solution alternatives (Mollaghasemi and Pet-Edwards, 1997: p. 4). Naval vessel option evaluation during tender evaluation, where the number of alternatives is small and there are a relatively large set of attributes to consider, is an example of a multiple-attribute problem.

Methods of MCDM for multiple-attribute problems include; scoring methods, multi-attribute value analysis (MAV), multi-attribute utility theory (MAUT) and the Analytical Hierarchy Process (AHP) (Mollaghasemi and Pet-Edwards, 1997). The MAV method appears to be the most suitable for naval vessel option evaluation leading up to and during tender evaluation. This is due to there being no need at this stage of the acquisition to incorporate the uncertainty aspects, such as requirements and technology maturity that are included in multi-attribute utility theory. MAV also uses value functions for the evaluation criteria, which are not included in simple scoring methods. These provide a means of representing the relative value of evaluation criteria over a range of values between the minimum acceptable value (threshold) and goal value (objective). Common value function curves for increasing and decreasing value can be found in references such as Buede (2000), which are shown in Figure 2. The need to make pairwise comparisons of attributes in the AHP, make it infeasible for naval vessel evaluation due to the large number of attributes that will be considered. MCDM strongly aligns with guiding principle two for the construction of the methodology to support the early stages of OTS naval vessel acquisitions.

2.6 PATTERN-BASED SYSTEMS ENGINEERING

PBSE has its foundations in the design patterns used by architects and planners in the late 1970s, which were adopted by software engineers in the early 1990s (Pfister et al., 2012: p. 322). Pfister *et al.* describe design patterns as "...a way practitioners can represent invariant knowledge and experience in design" (Pfister et al., 2012: p. 323). Schindel and Peterson (2014) assert that their approach to PBSE, which they call the S*Pattern approach "…includes not only the platform, but all the extended system information (e.g., requirements, risk analysis, design trade-offs & alternatives, decision processes etc.)" (Schindel and Peterson, 2014: p. 5). This means that using PBSE adheres to principle three described in section 2. Architectural design patterns, and design patterns of the associated system information could be reused in subsequent naval vessel acquisitions and reduce the effort required to define the capability.

While none of the naval vessel MBCD methodologies explicitly included PBSE, evidence of patterns in naval vessel design was found. Naval vessel physical architectural patterns included the Expanded Ship Work Breakdown structure (ESWBS) (Cimino and Tellet, 2007). Naval vessel functional architecture patterns were also found, including the work of Andrews (2006), who describes a functional breakdown comprising categories of float, move, fight/operation, and infrastructure. A pattern of naval mission tasks and associated measures of effectiveness is provided in the Universal Naval Task List (UNTL) (CNO, 2007). (However, the utility of the measures provided in the UNTL for naval vessel concept design can be variable as they appear to be more suited to operational testing and evaluation.) Using a design pattern comprising a predetermined list of naval vessel non-functional requirements (NFRs) is suggested by Gabb and Henderson with the statement (Gabb and Henderson, 1995: p. 13):

"All NFRs need to be considered and specified. The use of a comprehensive checklist by Navy would assist in this regard."

It is conceivable that these separate patterns could be amalgamated into a single pattern through the use of an appropriate MBSE metamodel.

3. PROPOSED METHODOLOGY TO SUPPORT EARLY STAGE OTS NAVAL VESSEL ACQUISITIONS

The early stages of naval vessel acquisitions, regardless of whether they are OTS or developmental programs, can be seen as a design activity or process (Finkelstein and Finkelstein, 1983). Finkelstein and Finkelstein (1983: p. 216) state for design in general:

"The design process consists of a sequence of stages starting from the perception of a need and terminating in a final firm description of a particular design configuration. Each stage is itself a design process..."

When considering how to perform early naval vessel design when the OTS constraint has been applied to the solution space, a useful counterpoint is provided by Kroll (2013: p. 180) with:

"Innovative design should be considered a discovery process and not a search over an existing solution space."

Recalling the earlier analogy between the modified-repeat ship design approach and the OTS naval vessel acquisition strategy, it follows from the statement of Kroll above, that early stage design in OTS acquisitions *should* comprise a search of the existing, or parent design space. Using SBD principles, an existing design space that is linked through mission performance measures to the capability needs can be developed. From this, acquisition stakeholders will gain an understanding of the vessel characteristics of OTS, or parent designs that are likely to meet the capability needs of the acquisition project without the need to have detailed parent design data. Exploration of this existing design space allows the acquiring agency to conduct trade-offs and identify the most suitable regions for the capability needs, as well as elucidate a set of RFT requirements and constraints in a 'middle out' SE manner. This is essentially the screening stage of the Kontio et al. (1995) OTSO process shown in Figure 1. Once responses are received for an RFT, the acquiring agency will then need to perform final design activities as well as a design option evaluation to select the preferred tenderer.

Using this reasoning, along with the elements outlined in the previous section that were identified as having alignment with the guiding principles, a methodology to support the early stages of OTS naval vessel acquisitions is proposed in Figure 3 and Figure 4. Figure 3 captures the first part of the methodology for conducting C&RE pre-gate one in the ADO capability lifecycle. Figure 4 captures the design option evaluation stage of the methodology to support tender evaluation between the decision points at gates one and two in the ADO lifecycle.

MBSE underpins the methodology as it facilitates traceability between the military roles of the capability need and early stage acquisition activities. This traceability is shown for the example covered in the next section in Figure 5. The methodology adopts and extends the WSAF MBSE metamodel described in Section 2.1 to include an analysis domain. The inclusion of the analysis domain (shown as a red package in the upper right corner of Figure 5) facilitates the analysis and design activities undertaken when implementing the methodology. It also allows these activities to be managed and design information to be retained within the MBSE model as shown in the model package elements within the analysis domain in Figure 5.

4. TESTING THE METHODOLOGY USING A USCG MEDIUM SECURITY CUTTER (MSC) EXAMPLE

The methodology has been tested by implementing it for an indicative Patrol Vessel capability. The implementation used a descoped CONOPS for a USCG Medium Security Cutter (MSC) (USCG, 2008) found on the internet as its basis. The test implementation was covered in detail in earlier papers by the lead author ((Morris and Thethy, 2015) and (Morris and Cook, 2017), so only key aspects and refinements to the methodology are provided here.

For the test implementation, the hullform was constrained to be of a monohull displacement/semi-displacement type of less than 80 metres in length and the main machinery was assumed to be high-speed marine diesels. The patrol vessel was assumed to be of low-end warfighting capability. While this can be seen to be limiting concept exploration, these constraints are representative of those typically imposed on naval vessel acquisitions in the authors' experience. Such constraints could arise from the need to berth the vessel using existing infrastructure, commonality across fleets and navy doctrine.

4.1 CONCEPT AND REQUIREMENTS EXPLORATION

4.1 (a) Establish Mission Scenarios and KPPs

The first step in the C&RE stage methodology is to define the operational and support mission scenarios. It is vital that these scenarios capture all of the operational needs for the capability to be procured. From the set of mission scenarios, the operational activities and KPPs can be identified using Subject Matter Expert (SME) input, or an appropriate design pattern of naval missions and activities. The KPPs, which are the "minimum number of performance parameters needed to characterise the major drivers of operational performance, supportability and interoperability" (Roedler and Jones, 2005: 11), are also used as the mission performance evaluation criteria during option evaluation.

The MSC CONOPS (USCG, 2008) contained the high-level roles for the vessel and missions it would perform. These missions were entered into the MBSE model and traced to the design pattern of naval operational activities found in the UNTL (CNO, 2007). Within the MBSE model, these operational activities were then decomposed and traced through a ship functional architecture design pattern, to the KPPs. An overview of the MBSE model that shows the mapping from the missions through to the mission performance KPPs for the MSC implementation is given in Figure 5.

4.1 (b) Determine Relationships between KPPs and Design Parameters

Relationships between KPPs and design parameters can be developed using parametric or surrogate modelling. Parametric modelling is a commonly used method in engineering design for making initial estimates of system design parameters such as physical, performance, engineering characteristics, and costs (ISPA, 2008). The estimates are based upon relationships between the design parameters and are typically generated using linear regression or other curve fitting techniques from the historical data of similar systems (Parsons, 2003).

In the case where a sufficient set of historical data is unavailable for developing a parametric model, this can be overcome by running a range of validated simulations of mission performance where the system/sub-system design parameters are systematically varied. Surrogate modelling techniques can then be used to take the results of such a set of simulations across a design space to construct an approximate relationships between design parameters and responses (Mavris and Pinon, 2012). Parametric and surrogate techniques have been utilised previously in a naval vessel concept exploration model by Eames and Drummond (1977), who also discuss constraining concept exploration and using it to identify suitable parent designs (Eames and Drummond, 1977: p. 30):

"The concept exploration model provides a rapid way of exploring all reasonable boundaries of dimensions and hullform … It is comparatively crude, but used with intelligent caution, it can assist the designer to select the most appropriate basis ship…"

Parametric and surrogate models are relatively straightforward to develop compared to high-fidelity physics-based simulations, which makes them suitable for resource constrained acquisition environments. Furthermore, using existing OTS design data as the basis of these models will help ensure the existing design space is feasible, which in turn, should lead to realistic RFT requirements being developed.

For the MSC test implementation, both parametric and surrogate modelling techniques were used to develop relationships between the KPPs identified in the previous step, and ship design parameters. These relationships were provided in Table 4 of Morris and Thethy (2015).

4.1 (c) Develop Simple Numerical Model

In this step, the relationships between KPPs and ship design parameters are built into a numerical model that can be exploited to construct an existing design space for use in subsequent steps. In this test implementation, numerical models were initially built using Excel® (Microsoft, 2010). Subsequently, numerical models of the parametric and surrogate relationships were implemented using *Mathematica*® (Wolfram, 2011), which were wrapped into Phoenix Integration's ModelCenter® (PI, 2014). This approach was taken to enable the analyses to be managed from the MBSE tool, which can also be wrapped into ModelCenter®. As covered in Morris (2014), the addition of an analysis domain into the Whole-of-System Analytical Framework (WSAF) metamodel, facilitates the management and execution of the analysis from within the MBSE tool. The analysis domain containing the executable elements used in the MSC implementation can be seen in the upper right corner of Figure 5.

4.1 (d) Design and Conduct Experiments

This step of the methodology requires consideration of the number of design parameters and how they can be used to develop the ROM design space from the viewpoint of the mission KPPs. When there are more than five variables for an experiment, Schmidt and Launsby (2005) recommend splitting the experiment into two parts: screening and modelling experiments. However, the approach of using parametric and surrogate techniques to build a simple numerical model does not impose a significant computational overhead, which facilitates jumping straight to modelling experiments. There is a need to account for unrealistic combinations of design parameters when conducting the experiment so infeasible regions of the design space are not generated. In the MSC test implementation, unrealistic combinations included:

- High propulsive power with low displacement or length
- Low propulsive power with high displacement or length
- Low displacement with high length/high displacement with low length

A Monte-Carlo design was used for the modelling experiment in the MSC test implementation.

4.1 (e) Build and Explore the ROM Design Space

Using the results from the modelling experiment for the MSC test implementation, the statistical and graphical methods available in the ModelCenter® software, primarily a prediction profiler, were used to build a view of the design space. Hootman (2003) describes a prediction profiler as "not (the) most elegant method of presenting information, but it is one of the most informative ones" (Hootman, 2003: p. 73). A prediction profiler provides a matrix of graphs where the KPPs (responses) are plotted on the vertical axes and the design parameters (inputs) are along the horizontal axes. The slope of the lines in the graphs represents the change in effect the design variable has on the KPP. The prediction profiler developed for the MSC example is presented in Morris and Thethy (2015).

To walk through how the design space can be explored and requirements elucidated for a specific example, the original design space for the endurance time KPP is shown in Figure 6a. Each red point in the design space is a "design" with the combination of ship length and endurance speed (horizontal axes) resulting in an endurance time KPP on the vertical axis. The design space is the result of a 1000 run Monte-Carlo experimental design, with the length ranging from 30-80 meters and the endurance speeds ranging from 8-30 knots. This was the corresponding range of speeds from the existing patrol vessel designs we could find within the Jane's Fighting Ship vessel database (IHS, 2014).

The application of two threshold KPP values for a minimum endurance time and range trims the design space as shown in Figure 6b. In this figure, designs that meet the KPPs are in red whereas those that do not are shown in grey and would not be considered further. From Figure 6b, it can be seen that the smallest length that can meet these threshold values is 45 meters and that there are a larger number of red designs at the higher end of the length scale. This suggests larger vessels are better suited to the capability needs and there is a capability risk associated with the smaller vessels.

On the other hand, if competing KPPs are considered, such as the annual lifecycle cost KPP, for which the constrained design space is shown in Figure 6c, it can be seen that a trade-off between endurance time and the annual lifecycle cost needs to be made. A vessel that can be deployed for longer will need to be larger, which will result in higher sustainment costs. Once all KPPs are considered, stakeholders could use the design space, in combination with their preferences to specify a requirement, such as a minimum length that would help ensure responses to an RFT would be more likely to meet capability needs. Since the KPPs are traceable to the capability needs and the existing design space developed using sound techniques, these requirements will be traceable and defendable. For the MSC example, Concept and Requirements Exploration highlighted that the most suitable designs for the capability needs would be those with a size at or near the upper limit of 80 meters in length.

4.2 OPTION EVALUATION

4.2 (a) Set Evaluation Scope

Once responses to a suitable RFT are received in a naval vessel acquisition (which will occur between gate one and two in the ADO capability lifecycle), an evaluation of the design options provided needs to be performed. When setting the option evaluation scope, Pahl and Beitz note that the evaluation criteria "…must cover the decision relevant requirements and constraints as completely as possible (Pahl and Beitz, 2007: p. 110). The competing objectives of performance, costs, schedule and growth potential will typically be present in naval vessel acquisitions. There may be various strategic factors that have the potential to influence the evaluation as well. The top-level scope of naval vessel option evaluations are likely to include:

- mission performance factors
- economic factors
- schedule and technical risk factors
- non-functional requirements factors
- strategic factors

All these factors were used in the MSC example and their importance weighted in a subsequent step.

4.2 (b) Establish Traceable Evaluation Criteria

For the MSC example, traceable mission performance criteria were established using the KPPs from the first step of the Concept and Requirements Exploration stage of the methodology.

Economic factors capture the cost objectives of the project. The evaluation criteria for economic factors proposed for the USCG MSC evaluation were: acquisition costs and operating costs over the USCG MSC lifecycle. The approach for capturing the traceable technical and schedule risk evaluation criteria can be linked to the risk management activities of the acquisition project.

Evaluation criteria related to non-functional requirements (NFRs) are important for naval vessel acquisitions and as noted by Andrews (2017: p. 72) are "a key hidden decision in the ship's style from the beginning of any ship design study". NFRs have been termed *quality attributes, constraints, goals, or extra functional requirements* (Chung et al., 2000) or "*ilities"* (Mirakhorli and Cleland-Huang, 2013). NFRs relevant to naval vessels could include: reliability, availability, maintainability, logistic supportability, compatibility, interoperability, training, human factors, safety, security and resilience.

In addition, strategic option evaluation factors need to be considered. These can include strategic partnerships and other influencers such as domestic and international politics. Strategic partnerships are likely to wield significant influence on the success (or otherwise) of any major project, however, they are not easy to make traceable! Strategic partnerships can be formed between the acquiring government and other entities including: the designer, the shipbuilder, the in-service support entity, and other navies that operate the same design.

4.2 (c) Determine Evaluation Criteria Value Functions and Weights

The weights and value functions for the evaluation criteria were elicited from Navy and naval architecture SMEs for the MSC test implementation. The threshold and objective values for the evaluation criteria were determined either from the MSC CONOPS, or based on engineering judgement. Weights were derived from the SME rankings of the criteria importance using the Rank Order Centroid (ROC) technique, which has been demonstrated to produce accurate weightings (Buede, 2000: p. 368). SME's selected a value function from the set of eight shown in Figure 2 for each evaluation criteria.

4.2 (d) Estimate Evaluation Criteria Values for Each Design Option

The fourth step in the option evaluation stage of the methodology is to estimate the evaluation criteria value for each option. This can be done using either: designer data from a submitted tender response, M&S, or parametric and surrogate relationships developed for KPPs using curve fitting techniques. For the Medium Security Cutter example, two design options at the upper limit of the size range (which was identified as being the most suitable region of the design space during C&RE), were identified from an internet search and the evaluation criteria values sourced from freely available internet searches. Where values for the design could not be found, they were estimated using engineering judgement or parametric relationships.

4.2 (e) Calculate Overall Value and Compare Options

In evaluating technical products, weighted summation of the evaluation criteria, provided they are reasonably independent, is the usual method of calculating the overall value (Pahl and Beitz, 2007). In the MSC test implementation, the overall weighted value (OWV) for the mission performance factors evaluation criteria (KPPs) was calculated as a weighted summation using a spreadsheet. The spreadsheet was wrapped into a Systems Modelling Language (SysML)-based MBSE model via model integration software. This allowed the evaluation criteria values, ranks and value curve identifiers to be held as value properties in SysML blocks. When executing the evaluation, the value properties were read from the MBSE model and sent to the spreadsheet that calculated the weighted values for each evaluation criteria (column *w.v(KPP)* in Table 2) and the OWV that was subsequently stored back in the model. The mission performance subset of the evaluation is shown in Table 2. From the green highlighted cells in Table 2, it can be seen that design option B had a higher OWV than design option A for the mission performance evaluation criteria shown.

4.2 (f) Estimate Uncertainty and Identify Weak Spots

Since the evaluation criteria values were estimated rather than provided as RFT response data, the level of confidence in the MSC example evaluation is low. However, the values were sufficient for a test implementation of the methodology as it was the methodology, rather than the designs that were under evaluation. To investigate the sensitivity of the OWV to changes in the evaluation criteria rankings, a Design of Experiments study was conducted. The study found the OWV result changed in less than 33% of the experiments due to changes in the criteria rankings.

Weak spots in each design option can be identified by looking for relatively low values of individual evaluation criteria (Pahl and Beitz, 2007). These are particularly important for promising design options that exhibit good overall value. Once identified, these weak spots can be addressed through design changes (Pahl and Beitz, 2007). The yellow highlighted cells in Table 2 indicate the largest differences between the two designs for the mission performance evaluation criteria considered. These highlighted cells indicate there are relative weaknesses of option A for the Endurance Time, Range and Seaboat Average Size KPPs. A weakness of option B relative to design option A is the Crew Accommodation Capacity KPP. If there was scope to change design B to accommodate more crew, this could be a change worth pursuing to increase its overall weighted value for mission performance. It is worth noting that while a design change technically violates the OTS acquisition strategy, changes to OTS designs are commonplace where value or legislative compliance issues need to be considered. It is worth noting any design change will be highly constrained and may impact on other design aspects, the effects of which may not be revealed until the vessel is in service.

5. DISCUSSION

It is worth noting that several of the C&RE methods reviewed for this research included multidisciplinary design analysis and optimisation (MDAO). Due to the OTS constraint, it was assumed there is no need to optimise the design space in order to converge on single point design during the early stages of the lifecycle. Hence, it was not included in the methodology. Furthermore, there is some disagreement on the value of optimisation during the early design stages. Andrews (2006) notes the need to recognise the limitations of optimisation during conceptual design to achieve a creative and divergent approach. Rhodes and Ross also note this challenge with MDAO (and Design Space Exploration), together with a potential conflict between MDAO, Design Space Exploration and system resilience (Rhodes and Ross, 2014: p. 37-38).

The application of Set-Based Design principles in the methodology provides a means of presenting sets of ship design parameters to build an existing design space. This requires less human and computational effort to implement than several of the Concept and Requirements Exploration methodologies referenced in Table 1, which utilise ship architecture models to synthesise multiple single point conceptual designs. The reduction of effort is primarily due to the use of parametric and surrogate models to build the ROM existing design space. Furthermore, there is a large amount of design data available for monohull surface warships and parametric design method has been used in ship concept design since before the use of computers in ship design (Parsons, 2003). Notwithstanding this, there is uncertainty associated with parametric modelling due to inaccuracy in the historical data used in the generation of relationships between design parameters, the correlation between the relationships developed and the historical data points upon which they are based. In the case where curve fitting is used to generate relationships, statistical techniques can be utilised to quantify the level of correlation (Parsons, 2003). Using Set-Based Design principles in the methodology also facilitated the exploration of the design space. During the exploration, trends between the design parameters and KPPs were readily identifiable from the plots. This supports identification of the most suitable combinations of design parameters for the capability needs. The trends also support identification of combinations of design parameters that present capability risk. These aspects suggest SBD is well suited to the conceptual design stage to build knowledge and to inform decisions on combinations of design parameters to take forward into preliminary and detailed design.

5.1 NOVELTY AND CONTRIBUTION

The novelty of the research covered in this paper stems from the incorporation of several different methods into a MBSE based methodology. Through the introduction of the analysis domain into the WSAF metamodel, the research extended the use of MBSE to establish, manage and guide the early stage acquisition, analysis, and tender evaluation activities, whilst maintaining traceability to strategic guidance and requirements. As shown in Figure 5, this extension will allow acquisition project stakeholders to demonstrate the links between capability needs and design activities, thereby building in 'contestability' and SE rigour into the acquisition process. The traceability that has been set up in the methodology also allows for rapid investigation of the impact of requirement changes. Reversing the traceability path allows for an assessment of the impact on requirements of vessel design changes. These contributions should result in better outcomes for naval vessel acquisitions that employ the methodology.

The inclusion of design patterns in the methodology enables reuse of MBSE models and domain knowledge in naval vessel acquisition projects, thereby reducing the level of effort required, provided the original domain knowledge is suitable and accurate. Pre-existing MBSE models could be exploited in subsequent acquisition efforts to rapidly trace through from naval missions to operational activities and their KPPs. Furthermore, reuse of knowledge from previous projects could also inform acquisition stakeholders of previous sources of risks and opportunities during early lifecycle activities (Morris and Cook, 2017). The example implementations performed for this research provides a starting point for building implementation knowledge from the MBSE models that were developed.

6. CONCLUSIONS

This paper covered a body of research undertaken to construct an MBSE methodology to support the early stages of naval vessel acquisitions. These stages of the lifecycle are vital to the success of the project but are difficult and have a history of being poorly performed in Defence acquisitions. The recent proliferation of oversight and contestability functions is evidence of this history and suggests that methods of supporting naval vessel acquisitions are required. Constraining the solution space to OTS naval vessels also presents a challenge to the acquisition community due to the 'middle-out' nature of requirements development. In a similar manner to the modified-repeat design approach, the OTS acquisition strategy is likely to have a higher success rate if the parent OTS vessel is based on a design with similar operational requirements. The methodology proposed in the previous sections seeks to address these challenges by leveraging a range of features from various disciplines. Firstly, a design space linked to the capability needs is developed using set-based design principles, model-based conceptual design, pattern-based systems engineering, and modelling and simulation. Secondly, Concept & Requirements Exploration is used to identify regions within the design space of combinations of design parameters from existing designs that are most likely to have similar performance characteristics to those derived from the OTS acquisition's capability needs. This region can be used to inform the RFT requirements in an OTS naval vessel acquisition in a traceable and defendable manner. Finally, the methodology supports trade-offs and the final design of the OTS design options proposed in RFT responses using a MCDM method.

Testing of the methodology has highlighted that the need to undertake naval vessel design activities, to understand and explore the existing design space, does not diminish when adopting OTS acquisition strategies. These design activities are essential to ensure the requirements released to industry are realistic and that any capability risks associated with the OTS constraint are identified early.

Further work to refine the approach would include fully implementing the methodology for another naval vessel acquisition project in order to gain more stakeholder feedback on its utility and or weaknesses. A final recommendation for further work is to include the development of a 'clean-sheet' concept design option for the capability needs as part of the C&RE process. This could be done using higher fidelity ship architectural or geometry models coupled with M&S tools as in the approaches of Andrews and Pawling (2003) or Dwyer and Morris (2017). Comparing the KPPs and other evaluation factors of the clean-sheet design option to the OTS design options could provide additional information and support to the acquisition stakeholders to determine whether the OTS constraint is likely to be value for money.

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8. COPYRIGHT

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APPENDICES

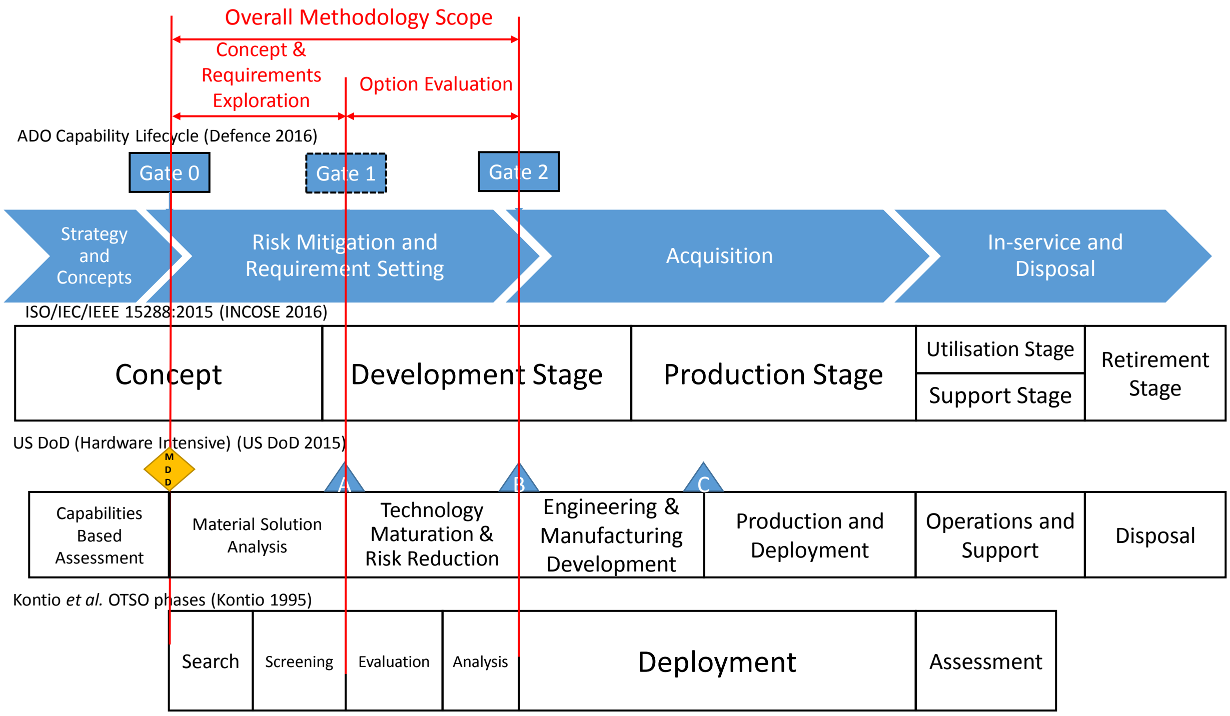

Figure 1: Various system lifecycles and the stages of interest for the research covered in the paper. The methodology constructed as part of the research covers the ADO risk mitigation and requirements setting stage as shown.

Table 1: Summary of naval vessel MBCD methodologies reviewed and the features they include

MBCD Methodology and Key References	MBSE	M&S	DSE	Other	Comments
VT C&RE (Brown and Thomas, 1998), (Kerns et al., 2011a), (Kerns et al., 2011b) and (Brown, 2013)	X	X	X	Also uses Multidisciplinary Design and Analysis.	Uses MBSE to manage ship and mission architecture, Separate ship synthesis, OEMs and MDAO models to analyse effectiveness and optimise. Value model (AHP) used for Overall Measure Of Effectiveness.
OTS C&RE (Morris and Thethy, 2015)	X	X	X	Uses integrated MBSE and M&S.	Uses MBSE for requirements, architecture and parametrics, along with integrated M&S and DSE. OTS Option analysis can be performed during DSE.
RSM Approach (Hootman, 2003) and (Fox, 2011)		X	X		Approaches use separate ship synthesis and OEMs to build concept design space. No explicit link to requirements.
WSAF (Morris, 2014)	X	X	X		MBSE integrated with M&S via parametrics.
SubOA/IPSM (Nordin, 2015)/(Harrison et al., 2012)		X	X		Both approaches use OEMs for submarine option/configuration evaluation during conceptual design. No integration with MBSE models.
DBB (Brown and Thomas, 1998) and (McDonald et al., 2012)		X	X	Uses CAD models	Approach facilitates rapid synthesis of a CAD hullform based on ship functions. Hullform's performance (e.g. seakeeping, resistance and stability) can then be simulated. No integration with MBSE.

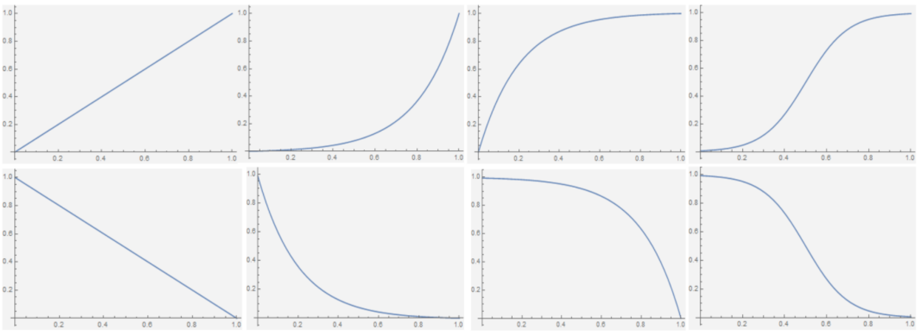

Figure 2: Common value function ($v_i(x_i)$) curves, normalised between threshold and objective values, for increasing utility (top) and decreasing utility (bottom) (Buede, 2000).

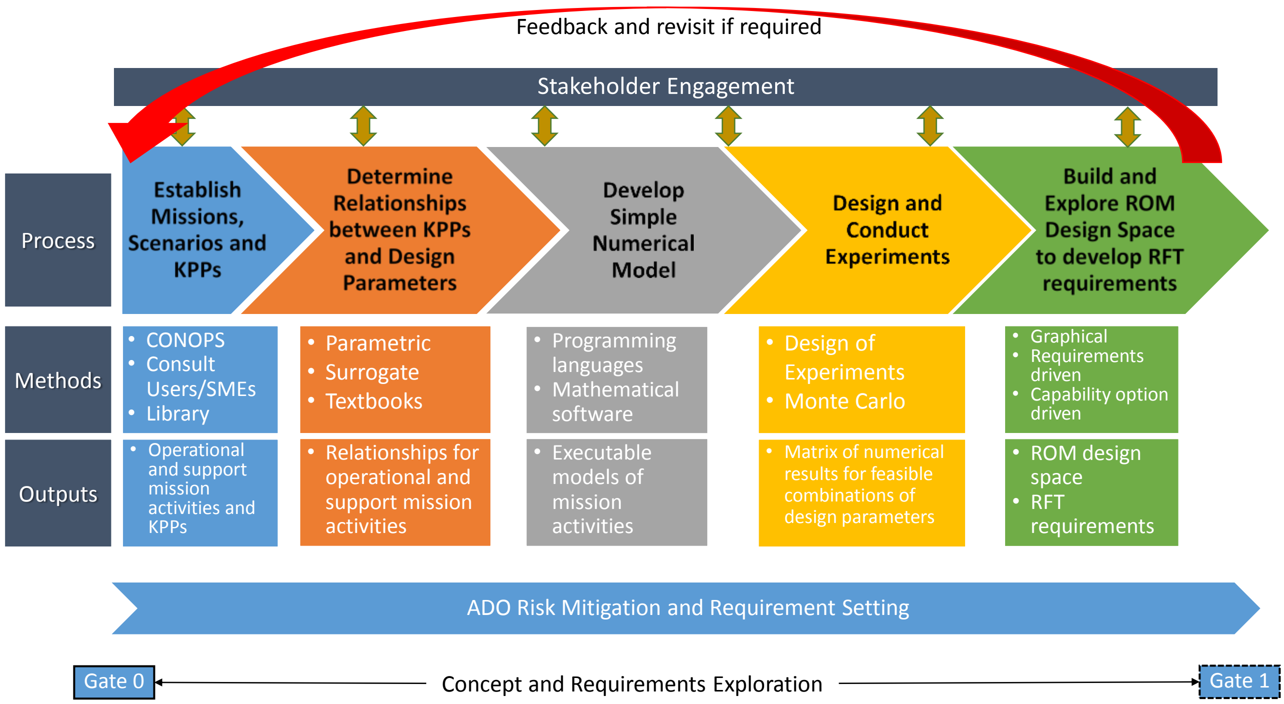

Figure 3: Concept and Requirements Exploration Stage of the methodology supports activities between decision points at Gate 0 and Gate 1 of the ADO capability lifecycle.

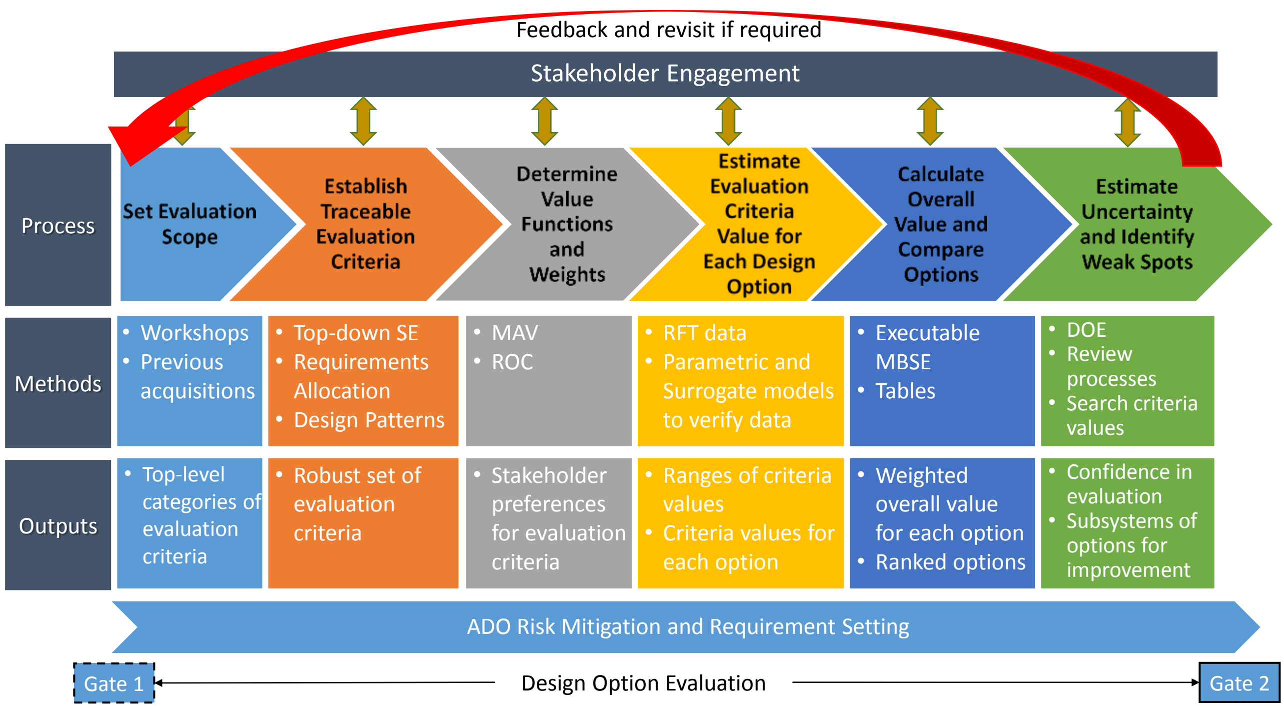

Figure 4: Design Option Evaluation stage of the methodology supports activities between decision points at Gate 1 and Gate 2 in the ADO capability lifecycle.

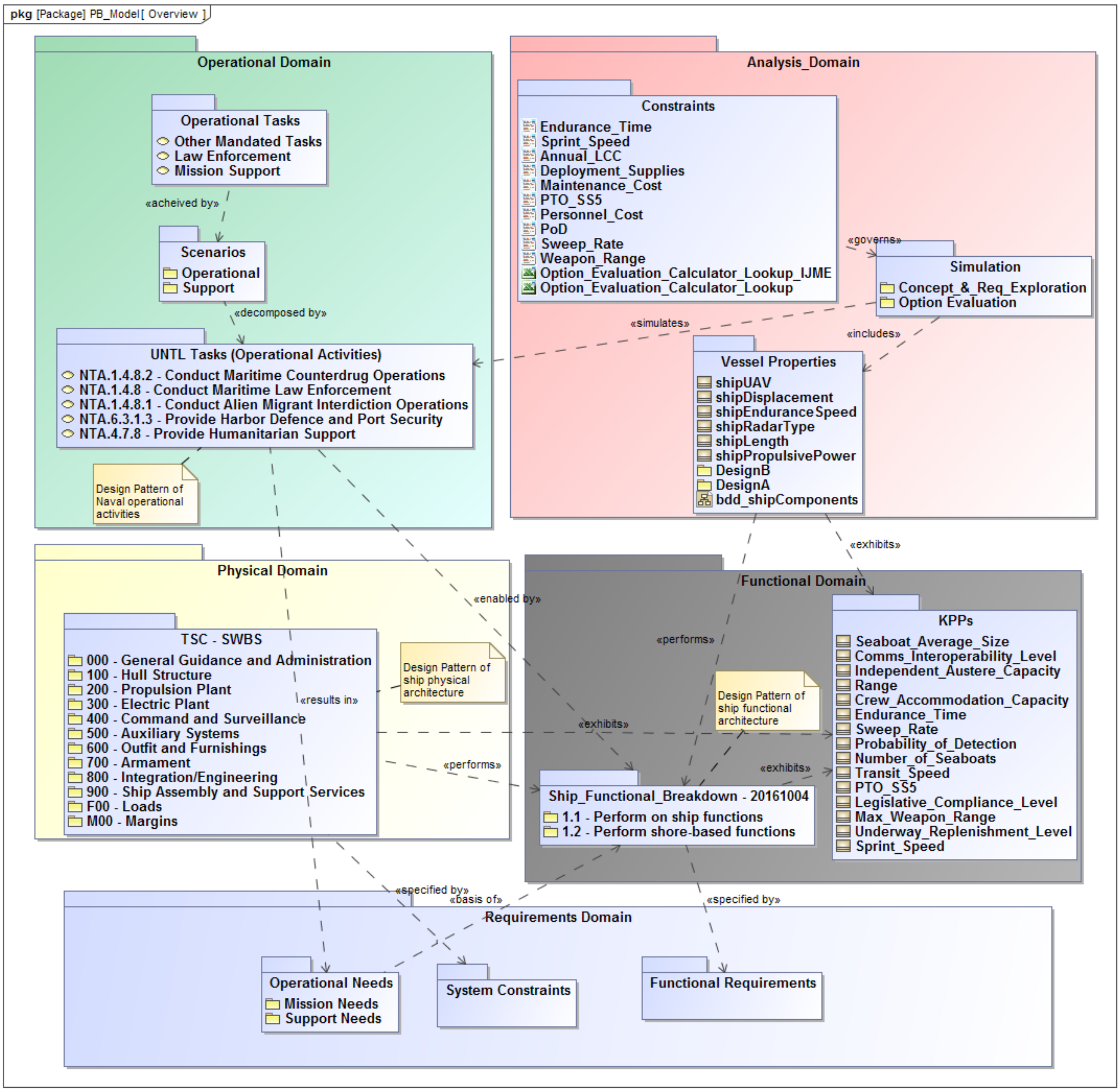

Figure 5: Overview of the MBSE model developed for the USCG MSC implementation. The figure shows the different domains in the extended WSAF metamodel and the relationships between the elements within each of these domains.

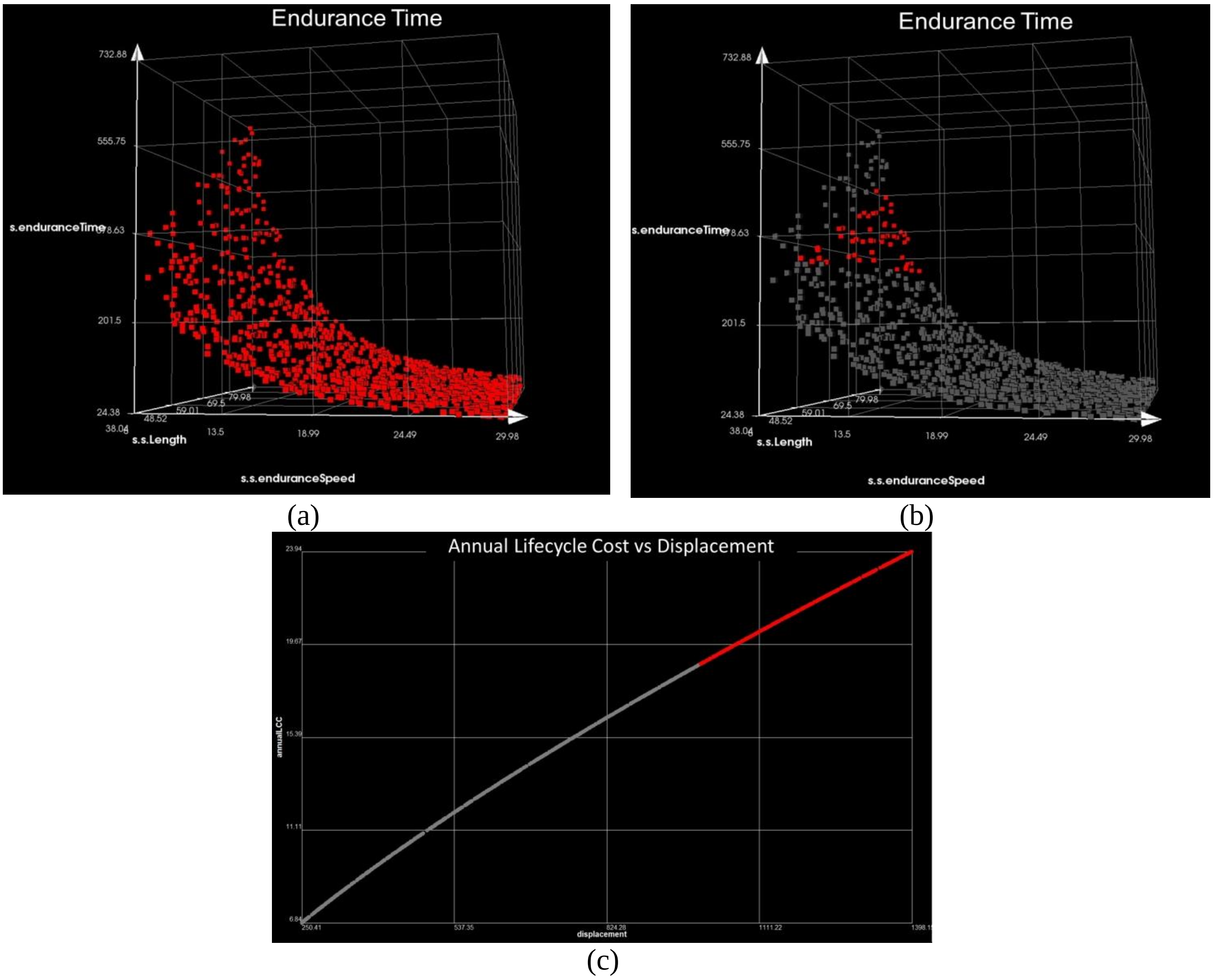

Figure 6: (a) Design space for the Endurance time KPP before constraining the design space to threshold values, and (b) after constraining the design space to threshold values of endurance time and range, and (c) the constrained design space for the competing annual lifecycle cost KPP. The red design points in (b) and (c) represent combinations of transit speed, ship length and displacement design parameters that will achieve the threshold endurance time and range values based on relationships from existing ship design data.

Table 2: Option evaluation table for mission performance criteria. The largest differences between the two designs indicate the weak spots. The weak spots of option A relative to option B are for the Endurance Time, Range and Seaboat Average Size KPPs. The weak spot of option B relative to option A is the Crew Accommodation Capacity. The higher OWV for option B infers it is the most suitable of the two designs for the evaluation criteria considered.

KPP	**Rank**		**KPP**				**Option A**		**Option B**	
Name		**ROC Weight (*w*)**	**Units**	**Threshold**	**Objective**	**Value Curve ***	**KPP**	***w*v (KPP')* ^+**	**KPP**	***w*v (KPP')* ^+**
Seaboat_ Average_ Size	3	0.0929	Metres	5	11	1	6	0.0155	8	0.0464
Comms_Interoperability_ Level	3	0.0929	Ordinal Scale: 1 - Poor 5 - Excellent	2	5	7	4	0.0781	4	0.0781
Independent_ Austere_ Capacity	15	0.0044	Persons	20	50	1	20	0.0000	30	0.0015
Range	3	0.0929	Nautical Miles	7500	10000	5	7500	0.0000	8600	0.0329
Crew_Accommodation_ Capacity	7	0.0375	Persons	30	55	1	54	0.0360	30	0.0000
Endurance_ Time	1	0.1879	Hours	336	672	7	504	0.0939	672	0.1866
Sweep_Rate	7	0.0375	km^2/hr	100	400	7	350	0.0362	350	0.0362
Number_of_ Seaboats	3	0.0929	Number	1	3	7	2	0.0464	2	0.0464
PTO_SS5	1	0.1879	Percent	50	90	7	80	0.1736	80	0.1736
Probability_ of_ Detection	7	0.0375	Probability	0.3	0.75	7	0.7	0.0367	0.7	0.0367
Transit_ Speed	7	0.0375	Knots	8	12	5	12	0.0375	12	0.0375
Legislative_ Compliance_ Level	13	0.0118	Ordinal Scale: 1 - Poor 5 - Excellent	2	5	5	4	0.0114	4	0.0114
Underway_ Replenishment _Level	13	0.0118	Ordinal Scale: 1 - Poor 5 - Excellent	1	5	1	4	0.0088	4	0.0088
Sprint_Speed	7	0.0375	Knots	20	30	5	20	0.0000	22	0.0238
Max_Weapon_ Range	7	0.0375	Metres	6500	15500	5	13800	0.0371	15500	0.0375
							OWV	**0.6112**	**OWV**	**0.7575**

* Value curves 1, 3, 5 and 7 are the increasing utility value curves in the top row of Figure 2. Value curves 2, 4, 6 and 8 are the decreasing utility value curves in the bottom row of Figure 2.
^ *KPP'* is the normalised value of the KPP over the range between its threshold and objective values.
\+ *v(KPP')* is the ordinate of the value function at the normalised KPP abscissa.

URANS PREDICTION OF THE SLAMMING COEFFICIENTS FOR PERFORATED PLATES DURING WATER ENTRY

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SUMMARY

The slamming coefficients for perforated plates of various perforation ratios and layout configurations were predicted using Unsteady Reynolds-Averaged Navier-Stokes (URANS) solver STAR-CCM+. The numerical model was validated by comparing with experimental measurements of slamming coefficient for a circular cylinder. The slamming coefficients and free surface profiles of perforated plates were then predicted at full-scale. It was found the air compressibility plays an important role by studying flat plate water entry phenomena. For perforated plates with small gap length/width ratios, the ability of the trapped air to evacuate through the space between the bottom of the plate and free surface is similar. For perforated plates with different gap number at a fixed perforation ratio, the slamming coefficient is increased with the increase in gap length/width ratio. However, a further increase in length/width ratio may impose a negative impact on the escape of trapped air due to the increase of gap number.

NOMENCLATURE

R	Radius of cylinder (cm)
V	Drop velocity (m/s)
G_B	Mesh Base size (mm)
T	Temperature (℃)
Q	Volume energy sources
$\mathbf{u}$	Velocity field in Cartesian coordinates (m/s)
∇	$(\partial/\partial x, \partial/\partial y, \partial/\partial z)$
p	Pressure (N m^{-2})
$\mathbf{g}$	Gravitational acceleration (m/s^2)
ν	Kinematic viscosity (m^2/s)
ρ	Density (kg m^{-3})
ρ_{air}	Density of air (kg m^{-3})
ρ_{water}	Density of water (kg m^{-3})
μ	Dynamic viscosity (N s m^2)
μ_{air}	Dynamic viscosity of air (N s m^2)
μ_{water}	Dynamic viscosity of water (N s m^2)
e	Specific sensible energy (J/kg℃)
λ	Transport coefficient
α_i	Volume fraction
S_{α_i}	Source of sink of each phase
$D\rho_i / Dt$	Lagrangian derivative of the phase density
U_{SN}	Numerical uncertainty
U_I	Iterative uncertainty
U_G	Grid uncertainty
U_T	Time-step uncertainty
r_k	Refinement ratio
ε_{k32}	differences between coarse-medium solutions
ε_{k21}	differences between medium-fine solutions
S_{k3}	Coarse solution result
S_{k2}	Medium solution result
S_{k1}	Fine solution result
R_k	Convergence ratio
S_U	Maximum oscillation solution result
S_L	Minimum oscillation solution result
$\delta^*_{RE_{k1}}$	Numerical error
δ^*_{k1}	Numerical error if C_K is close to 1
p_k	Order of accuracy
C_K	Correction factor
p_{kest}	Limiting order of accuracy
S_c	Benchmark value
U_{kc} / U_k	Grid, time-step or iterative uncertainty
C_S	Slamming coefficient
F_S	Force acting on the object in vertical direction (N)
A_P	Project area of object normal to the direction of oscillation (m^2)
v_S	Slamming impact velocity (m/s)
DNV	Det Norske Veritas
VOF	Volume of Fluid
URANS	Reynolds-Averaged Navier-Stokes

1. INTRODUCTION

In recent years, oil and gas companies are actively searching for oilfields in deeper waters. In comparison to shallow water marine operations, higher safety standards are essential, which require subsea structures to be lowered across the splash zone and installed safely. This is especially true for the subsea structures with large horizontal surfaces such as baseplates, mud mats and horizontal subsea trees

(Faltinsen, 1990). During the installation process, if the water impact load (i.e. slamming force) occurring within a short duration when a structure penetrates the water surface, is larger than the self-weight of subsea structure in air, a 'weightlessness' phenomenon will occur. This large impact force will result in hoisting line slack, which leads to a snap force thereafter. To avoid this situation and ensure a successful lowering operation, it is critical to estimate the slamming forces during the design process. DNV-RP-H103 (2011) provides a series of benchmark values for the slamming coefficients for simple objects, which includes cylinders and flat plates. However, the applicability of these recommended values is limited since subsea structures have become more complicated in terms of geometrical shapes and sizes (i.e. suction anchor) for deep water zone. In this paper, the object of interest is the perforated plate, which is an important component of typical subsea equipment, for example, a protection shell for the manifolds or a baseplate acting as a loading supporting device.

In the late of 1920's, Von Karman (1929) started studying the slamming problem analytically for a seaplane from the view of classical mechanics. He applied the law of conservation of momentum to determine the maximum pressure. However, many variables were neglected such as gravity, the compressibility of air and water; and especially the local uprise of water. Therefore, to consider the effect of the free surface, Wagner (1932) then introduced potential flow theory to describe this phenomenon. Based on his theory, the body wetted length contacting with the free surface is extended longer due to local uprise of water. Following these two analytical approaches for investigating a free-falling circular cylinder (Faltinsen, 1990), both of them are considered to have their limitations in predicting the slamming coefficients by comparing against the experimental results (Campbell and Weynberg, 1980). In addition, both Von Karman and Wagner's approaches are two-dimensional methods. Therefore, it is difficult to claim which approach will give a better result when considering three-dimensional effects, such as the gravity, side effect and air cushion. However, it should be pointed out that the Wagner's approach provides more details of the flow at the spray zone.

Numerous experiments have been performed in the last 50 years and the most widely studied tests are the free drop test and constant drop test. Verhagen (1967) and Chuang (1966, 1970) studied the air cushion effect, whereby the air trapped beneath the flat plate was observed during water entry while having wedges with a deadrise angle of less than 3º. They also found that the trapped air deformed the water surface. Which significantly reduced the expected maximum impact pressure as the air pocket was forced out from the bottom of the plate. Campbell and Weynberg (1980) conducted a constant drop test using a circular cylinder, and a formula was derived from the experimental data to calculate the slamming coefficients. A more recent experiment conducted by Huera-Huarte *et al*. (2011) showed that asymptotic theory could predict loading on the flat plate well, only when the deadrise angle is larger than 5º. This proves that air compressibility plays an important role especially in areas where the deadrise angle is very small. In addition, they also reported that the slamming coefficient is strongly related to the impact velocity. These experimental results are considered to be sufficiently accurate at the time and are still commonly cited as benchmarks for validating numerical results in the recent numerical research works.

With recent advancement in computing technology, there is an increase in the adoption of numerical methods, especially Computational Fluid Dynamics (CFD), to investigate large and complex structures. Iwanowski *et al.* (1993) investigated the two-dimensional air cushion effect by coupling compressible viscous air and incompressible viscous water. Later, Korobkin (1996) took the effect of water compressibility into consideration when estimating the pressure distribution via acoustic approximation and normal modes, and the result showed the water compressibility has little influence on the slamming force. Fairlie-Clarke and Tveitnes (2008) simulated water entry of wedge-shaped sections with various deadrise angle ranging from 5º to 45º. Later in their experiments (Tveitnes *et al.*, 2008), their experimental results agreed well with the CFD results.

In relation to real engineering application, Næss *et al.* (2014) simulated a suction anchor crossing splash zone, the importance of entrapped air and water were particularly well presented by CFD method while they were hardly observed by the analytical or experimental methods. Swidan *et al.* (2014) simulated a free-falling wave-piercing catamaran model in the calm water to estimate the slamming load, its corresponding motions and flow visualisation. However, there is a lack of published work with regard to perforated plates.

In this study, the water entry process of perforated plates was modelled using the URANS equations and Volume of Fluid (VOF) method. Verification and validation of the numerical setup were achieved by comparing with experiments results (Campbell and Weynberg, 1980). The influences of various perforation ratios and layout configurations on the slamming coefficients were studied. In addition, this paper also examined the importance of air compressibility by modelling flat plate water entry.

2. NUMERICAL METHOD

2.1 BASIC GOVERNING EQUATIONS

The governing equations for the two-phase incompressible flow problem are given by the following equations (Jasak, 1996):

$$\frac{\partial \mathbf{u}}{\partial t} + \nabla(\mathbf{uu}) = -\nabla p + \mathbf{g} + \nabla \cdot (\nu \nabla \mathbf{u}) \qquad (1)$$

here, $\mathbf{u}$ = (u,v,w) is the velocity field in Cartesian coordinates, ∇ is $(\partial / \partial x, \partial / \partial y, \partial / \partial z)$, p represents the pressure, $\mathbf{g}$ is the gravitational acceleration. ν is the kinematic viscosity. As the flow is effectively incompressible, then the continuity equation will be:

$$\nabla \cdot \mathbf{u} = 0 \tag{2}$$

When considering compressible gas, the continuity equations need to be solved with an additional energy equation together to predict the temperature. Therefore, the compressible Navier-Stokes equation and continuity equation will be given as follows:

$$\frac{\partial \rho \mathbf{u}}{\partial t} + \nabla \cdot [\rho \mathbf{u}\mathbf{u}] = \rho \mathbf{g} - \nabla (p + \frac{2}{3}\mu \nabla \cdot \mathbf{u}) + \nabla \cdot [\mu (\nabla \mathbf{u} + (\nabla \mathbf{u})^T)] \tag{3}$$

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0 \tag{4}$$

The energy equation is:

$$\frac{\partial \rho e}{\partial t} + \nabla [\rho e \mathbf{u}] = \rho \mathbf{g}\mathbf{u} - \nabla (p\mathbf{u} + \frac{2}{3}\mu (\nabla \cdot \mathbf{u})\mathbf{u}) + \nabla \cdot [\mu (\nabla \mathbf{u} + (\nabla \mathbf{u})^T)\mathbf{u}] + \nabla \cdot (\lambda \nabla T) + \rho Q \tag{5}$$

where ρ is the density, μ is the dynamic molecular viscosity, e is the specific sensible energy, λ is the transport coefficient, T is the temperature and Q represents the volume energy sources. Also, for the laminar model, the terms which are associated with turbulence can be ignored (Lasrsen, 2013).

2.2 VOLUME OF FLUID (VOF) METHOD

VOF method is one of the well-known mesh-based methods to deal with the free surface. The volume fraction is defined below to describe the spatial distribution of each phase (CD-adapco, 2014):

$$\alpha_i = \frac{V_i}{V} \tag{6}$$

Only 2 phases are studies in this research, therefore the physical properties can be determined by volume fraction by following two equations:

$$\rho = \rho_{air}\alpha_{air} + \rho_{water}(1 - \alpha_{air}) \tag{7}$$

$$\mu = \mu_{air}\alpha_{air} + \mu_{water}(1 - \alpha_{air}) \tag{8}$$

The mass conservation equation describing the transport of volume fraction is:

$$\int_V \left(s_{\alpha_i} - \frac{\alpha_i}{\rho_i}\frac{D\rho_i}{Dt} \right) dV = \frac{\partial}{\partial t}\int_V \alpha_i dV + \int_S \alpha_i (\mathrm{v} - \mathrm{v}_g) da \tag{9}$$

here, s_{α_i} and $D\rho_i / Dt$ are the source of sink of each phase and the Lagrangian derivative of the phase density, respectively.

3. BENCHMARK CASE STUDY

Verification was performed to demonstrate the accuracy of the CFD model and gain confidence in numerical results. A benchmark study was first performed to simulate an experiment involving the water entry of a circular cylinder. The verification process focuses on the mesh independence study and time-step size independence study to estimate the numerical errors.

3.1 WATER ENTRY OF QUASI-CIRCULAR CYLINDER

To validate the proposed method, a circular cylinder with a radius of R=10cm was simulated to come in contact with the free surface vertically at a speed of 5m/s. The results were compared with the experiment conducted by Campbell and Weynberg (1980).

The mesh quality and the computational domain are illustrated in Figure 1, whereby the information on the mesh (i.e. base size, G_B) is shown in Table 1. The refined mesh region around the cylinder with a cell size of $0.125G_B$ is considered to be sufficient in computing the water entry, especially at the region between cylinder bottom and the free surface. An anisotropic mesh was applied in the z direction at the free surface to capture wave deformation. The background domain was built with the consideration of the effect of the reflected wave and boundary conditions.

The cylinder was initially located 1cm above the free surface as shown in Figure 2. The bottom boundary was defined as a velocity inlet, to specify a velocity equal to the constant drop velocity of the case being modelled; and a pressure outlet was defined at the top of the domain. The left boundary was specified as symmetry plane to save computational resources, while the rest walls were defined as slip wall.

3.2 VERIFICATION AND VALIDATION

Verification and validation were conducted by following the generalised Richardson's Extrapolation method described by Wilson *et al.* (2001) and Stern *et al.* (2001). As shown in equations below, the numerical uncertainty U_{SN} is composed of iterative uncertainty U_I, grid uncertainty U_G and time-step uncertainty U_T.

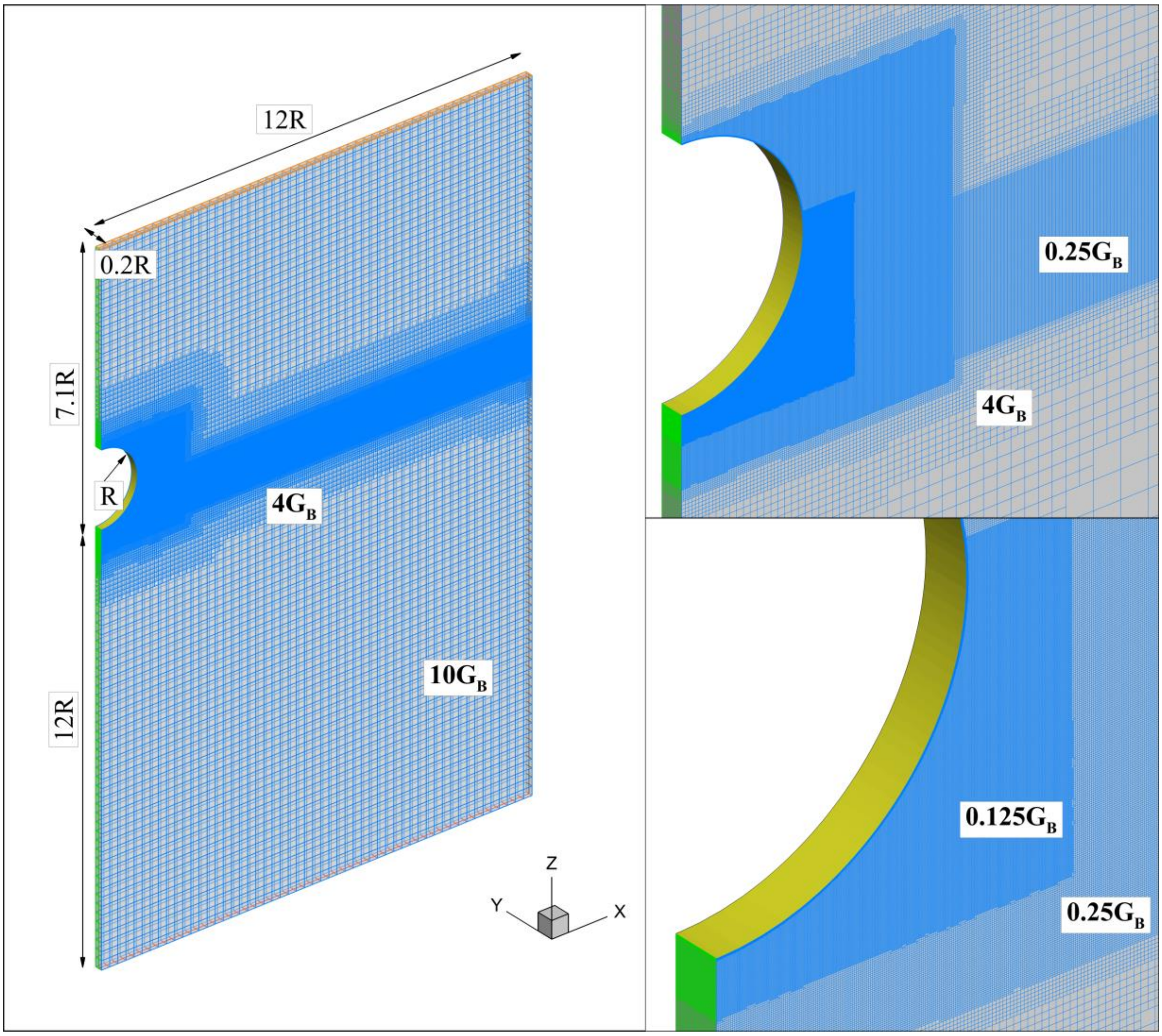


Figure 1: Illustration of computational domain and grid for water entry of cylinder

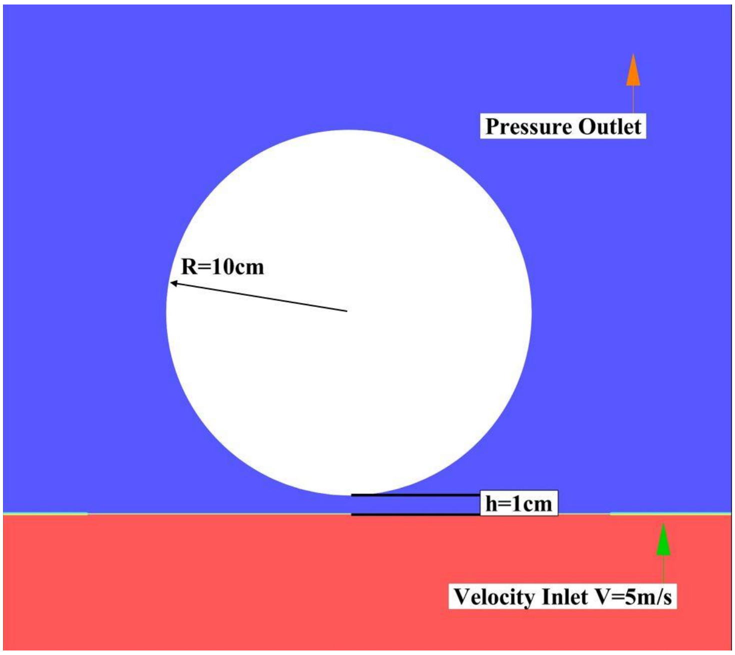


Figure 2: Initial location of cylinder

Table 1: Numerical simulation setup information

Case	**Fine**	**Medium**	**Coarse**
Base size G_B (mm)	2	2.828	4
Inner trimmer size (mm)	0.25	0.3536	0.5
Time step size (s)	6.25E-7	-	-
Grid cell number	19,626,083	7,375,210	2,774,724
	Small	**Medium**	**Large**
Base size G_B (mm)	2.828	-	-
Time step size (s)	1.5625E-07	3.125E-07	6.25E-7

The iterative uncertainty is neglected because all simulations have converged to result in a small magnitude, therefore, only the grid and time step convergence study are performed.

$$U_{SN}^2 = U_I^2 + U_G^2 + U_T^2 \tag{10}$$

With regards to the convergence studies, three different grid and time-step configurations were used with a refinement ratio, $r_k = \sqrt{2}$ and 2, respectively. Detailed configuration is given in Table 1 below. Solution variations should be assessed based on the differences between coarse-medium ($\varepsilon_{k32} = S_{k3} - S_{k2}$) and medium-fine ($\varepsilon_{k21} = S_{k2} - S_{k1}$) solutions as given below:

$$R_k = \varepsilon_{k21} / \varepsilon_{k32} \tag{11}$$

According to the R_k value, three convergence conditions are possible as follows:

i. Monotonic convergence: $0 < R_k < 1$
ii. Oscillation convergence: $R_k < 0$
iii. Divergence: $R_k > 1$

For condition (iii), error and uncertainty cannot be estimated and therefore the simulation result is not valid; for condition (ii), the uncertainty is simply calculated based on the maximum oscillation solution S_U and S_L minimum oscillation solution as follows:

$$U_{ii} = 0.5(S_U - S_L) \tag{12}$$

For condition (i), the generalised Richardson's Extrapolation method can be used to calculate the numerical error and order of accuracy as shown below:

$$\delta^*_{RE_{k1}} = \frac{\varepsilon_{k21}}{r_k^{p_k} - 1} \tag{13}$$

$$p_k = \frac{\ln(\varepsilon_{k32} / \varepsilon_{k21})}{\ln(r_{k21})} \tag{14}$$

A correction factor is introduced to determine the proximity of the solutions to the asymptotic range and defined as follows:

$$C_k = \frac{r_k^{p_k} - 1}{r_k^{p_{kest}} - 1} \tag{15}$$

where p_{kest} is the limiting order of accuracy, based on the assumed theoretical order of accuracy or solutions for simplified geometry and conditions. If C_K is close to 1, it indicates that the solution is close to asymptotic range. Thus, the numerical error δ^*_{k1}, benchmark value S_c and the uncertainty U_{kc} / U_k can be estimated as follows:

$$\delta^*_{k1} = C_k \cdot \delta^*_{RE_{k1}} \tag{16}$$

$$S_c = S - \delta^*_{k1} \tag{17}$$

$$U_{kc} \begin{cases} \left[2.4(1-C_k)^2 + 0.1\right]\left|\delta^*_{RE_{k1}}\right|, |1-C_k| < 0.25 \\ \left[|1-C_k|\right]\left|\delta^*_{RE_{k1}}\right|, |1-C_k| \geq 0.25 \end{cases} \tag{18}$$

If the correction factor is away from unity, only the numerical uncertainty can be estimated:

$$U_k \begin{cases} \left[9.6(1-C_k)^2 + 1.1\right]\left|\delta^*_{RE_{k1}}\right|, |1-C_k| < 0.125 \\ \left[2|1-C_k| + 1\right]\left|\delta^*_{RE_{k1}}\right|, |1-C_k| \geq 0.125 \end{cases} \tag{19}$$

The simulation results for grid and time-step conditions along with their uncertainties are listed in Table 2.

3.3 COMPUTATIONAL RESULTS

The slamming coefficient for the circular cylinder is predicted using Eq (20) and the results are compared with the experimental value as shown in Figure 3 and Figure 4.

$$C_S = \frac{2F_S}{\rho A_P v_S^2} \tag{20}$$

where F_S is the force acting on the object in vertical direction at the initial time, A_P is the project area of object normal to the direction of oscillation and v_S is the slamming impact velocity.

It can be observed from Figure 3 and Figure 4 that a large impulsive force occurs within a very short time (in the order of 200-250μs) when the bottom of the cylinder came in contact with the free surface. After the cylinder penetrated the free surface, this force (directly proportional to the slamming coefficient) reduced rapidly to a small magnitude. The results also show that good spatial and temporal convergences were achieved since the differences in the results were very small. In addition, the computed free surface deformation exhibits a close correlation with that observed in the experiment conducted by Greenhow and Lin (1983) as shown in Figure 5. The uncertainty study and its results are listed in Table 2.

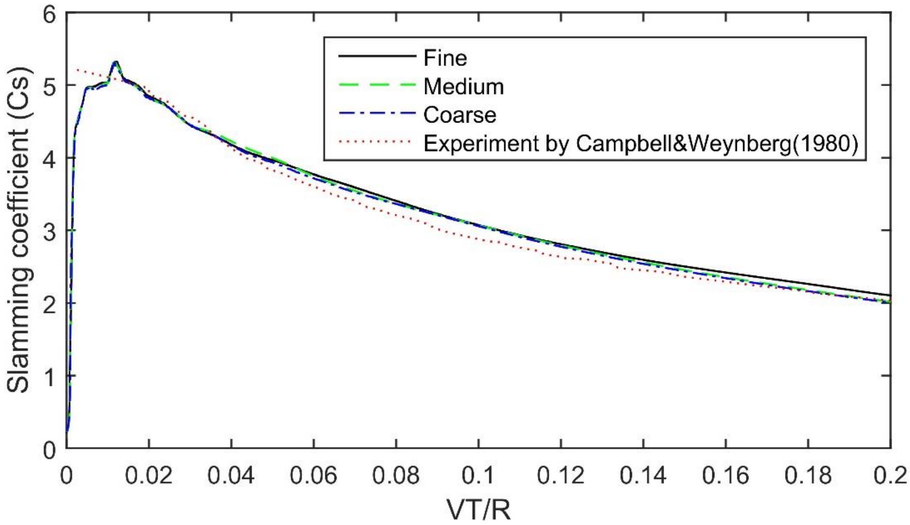


Figure 3: Slamming coefficient for three mesh sizes during initial water entry period

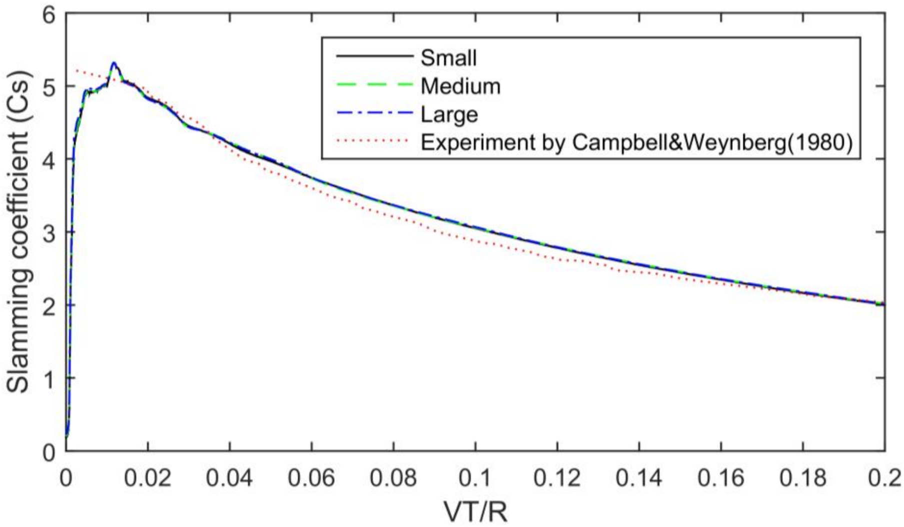


Figure 4: Slamming coefficient for three different time step sizes during initial water entry period

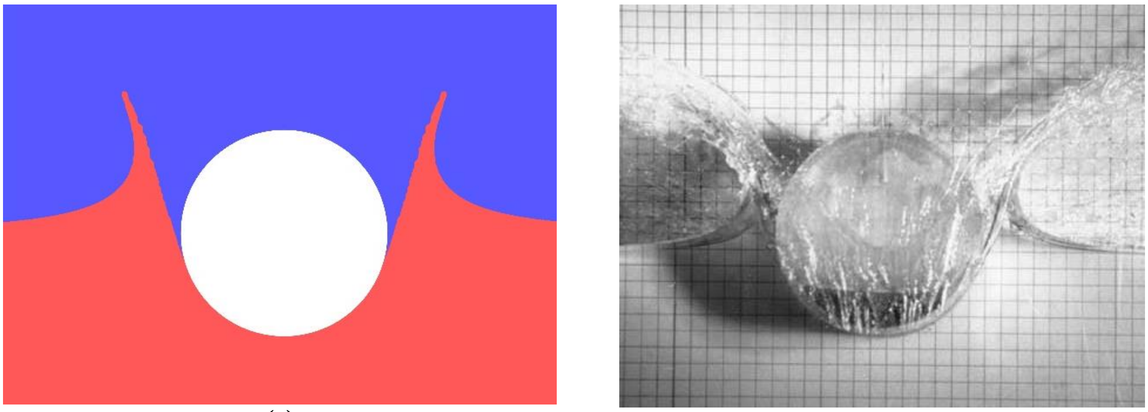


Figure 5: a) Numerical model (b) Experimental results by Greenhow and Lin (1983) at t=0.330s, almost VT/R=1, free drop

Table 2: Grid and time-step uncertainty study for circular cylinder

	r_k	Solution			R_k	U_{GC}/U_T	$\%S_1$
		S_1	S_2	S_3			
Grid	$\sqrt{2}$	5.323	5.320	5.315	0.6	0.01	0.19
Time step	2	5.315	5.320	5.289	-0.6	0.0025	0.05

It can be seen from the results listed in Table 2, that the slamming coefficients of each configuration are very close. Therefore, both the estimated corrected grid uncertainty U_{GC} and time step uncertainty U_T are reasonably small, and their value ($0.19\%S_1$ and $0.05\%S_1$) are considered to be acceptable. The difference between numerical and experiment values (Campbell and Weynberg, 1980) is small (i.e. 3.3%), which indicates that the slamming force will be well predicted if similar grid and time-step sizes were selected. Therefore, the numerical model is proven to have the ability to accurately predict the slamming force for the perforated plate.

4. SYSTEMATIC COMPUTATIONS

4.1 COMPUTATIONAL MATRIX

The geometries of two series of models are shown in Figure 6 and Figure 7 with key parameters listed in Table 3. The first series of models consist of perforated plates with different perforation ratios ranging from 0%, 20%, 30% to 40%. The second series of models are perforated plates with different layouts (i.e. gap numbers) with fixed ratio of 30%. All plates have the same external dimensions of 13m×10m×0.7m. The plate is positioned 0.1m above the free surface and dropped vertically with a constant velocity of 0.5m/s.

Uncertainty analysis was performed for Case 2 with the generalised Richardson's Extrapolation using systematically refined grid-spacing and time-steps, and Table 4 shows the calculated grid and time step uncertainties. Since $0<R_k<1$, the monotonic convergence can be confirmed by both two studies. The grid and time step uncertainties are small, and their values ($6.3\%S_1$ & $11.5\%S_1$) are both considered to reasonably acceptable for three-dimensional simulations. It should be noted that, although further refinement of mesh size and/or time step size will produce a more accurate result and smaller uncertainties. However, the required computational resources will increase significantly. Therefore, from the resource point of view, it is sufficient to accept a reasonable solution by comparing with a quasi-two-dimensional simulation of the circular cylinder studied above with small variations.

4.2 INFLUENCE OF AIR COMPRESSIBILITY

The compressibility of air should be taken into account in terms of the flat plate without any gaps as the trapped air will be forced out from the underneath of the flat plate (Chuang, 1966, Faltinsen, 1990, Huera-Huarte et al., 2011). Therefore, Case 1 was investigated to examine the importance of air compressibility. The velocity contour of air is presented in Figure 8, before the flat plate came into contact with the free surface. It can be seen that the local air velocity has been accelerated to a value larger than 102m/s at the spray root, which translates to a Mach number that is greater than 0.3.

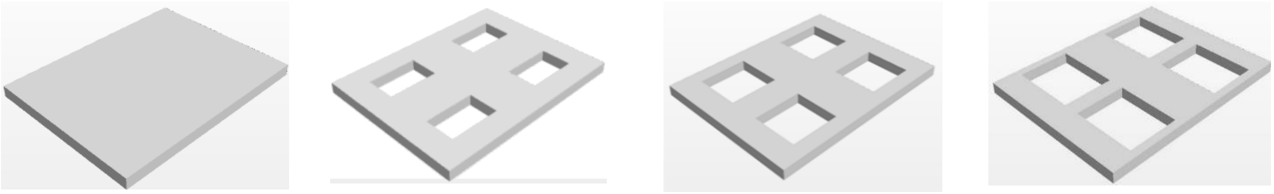


Figure 6: Perforated plates with varied perforation ratio, from left to right: 0%, 20%, 30% and 40%

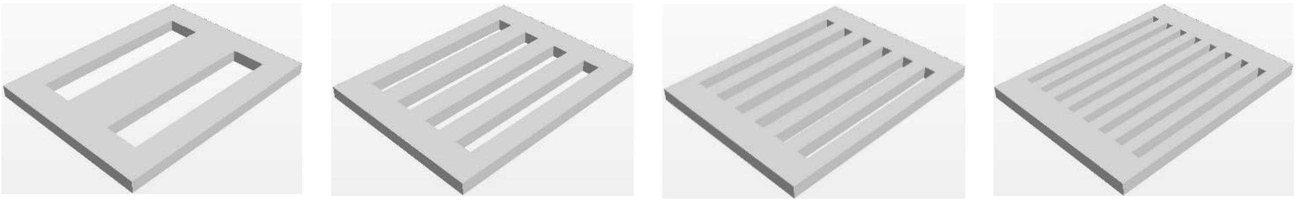


Figure 7: Varied layout configuration with fixed perforation ratio of (30%). From left to right: 2 gaps, 4 gaps, 6 gaps and 8 gaps

Table 3: Simulation cases for layout configuration and perforation ratio

Case no.	**Perforation ratio (%)**	**Gap No.**	**Gap Length/Width Ratio**
1	0	0	-
2	20	4	1.625
3	30	4	1.076
4	40	4	1.230
5	30	2	4.875
6	30	4	9.750
7	30	6	19.500
8	30	8	39.000

Table 4: Uncertainty study for Case 2

	r_k	Solution			R_k	U_{GC}/U_T	$\%S_1$
		Cs_1	Cs_2	Cs_3			
Grid	$\sqrt{2}$	14.03	14.5	19.22	0.1	0.89	6.3
Time step	2	14.03	15.32	17.18	0.69	1.62	11.5

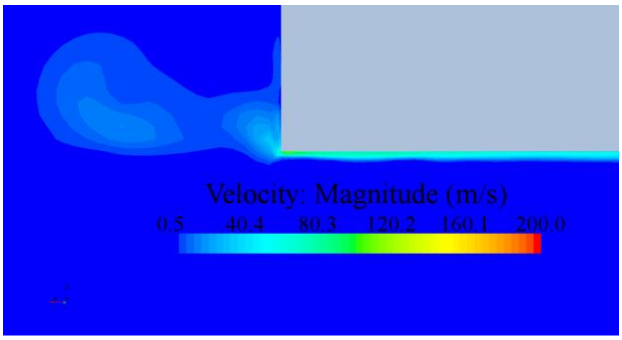


Figure 8: Front view of velocity contour at 0.11s before water entry, grey part represents the corner of the flat plate

Therefore, it is necessary to consider air compressibility when the air cushion is generated before the slamming impact for the flat plate (Anderson Jr, 2010). The slamming coefficients predicted by compressible and incompressible URANS equations are shown in Figure 10, respectively. It was found that the slamming coefficient was significantly reduced and the impact time was extended when the compressibility of air was considered. As shown in Figure 9, the free surface is deformed by the large air velocity and its pressure, some region is suppressed by the air cushion while other region is lifted up and accelerated to impact the bottom of the flat plate. This deformation leads to the occurrence of slamming impact in advance.

With reference to Figure 10 showing the time history of slamming coefficient, the compressible solver produced a negative force, which followed the maximum vertical force after the entire bottom surface of the flat plate had come in contact with the free surface. A similar result was also obtained by Huera-Huarte *et al*. (2011) in his experiments. This can be explained by the pressure contours in Figure 11, where a large low pressure zone occurred underneath the flat plate, thus, leading to a large downward force on the flat plate, when air compressibility was considered, while this phenomenon was absent without the consideration of air incompressibility.

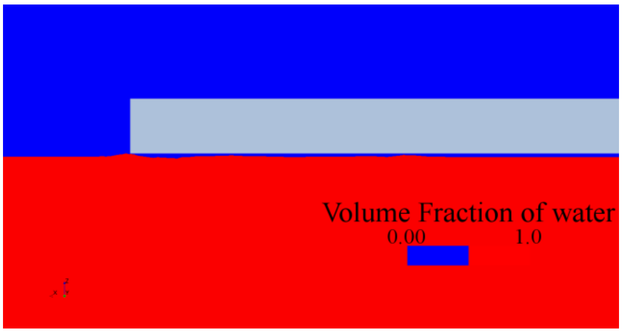


Figure 9: Front view of free surface deformation at 0.12s when flat plate came in contact with free surface

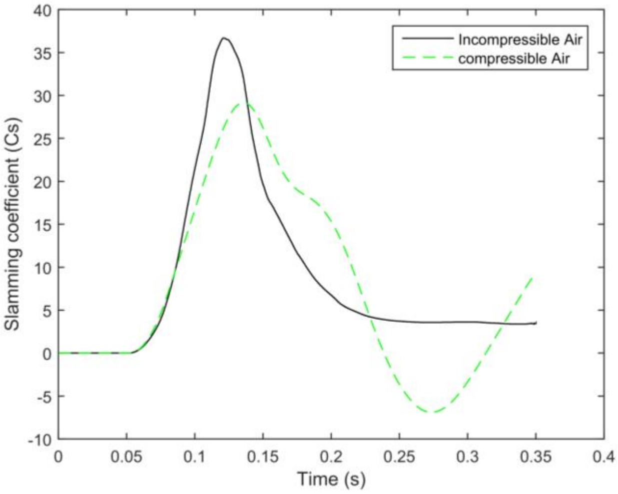
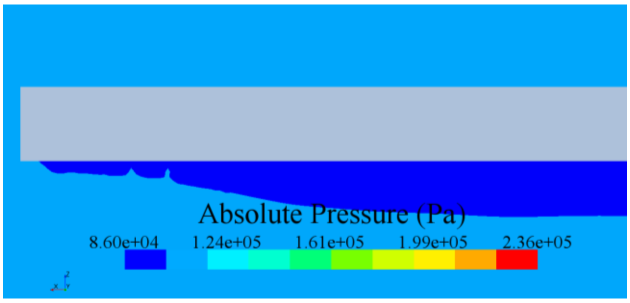
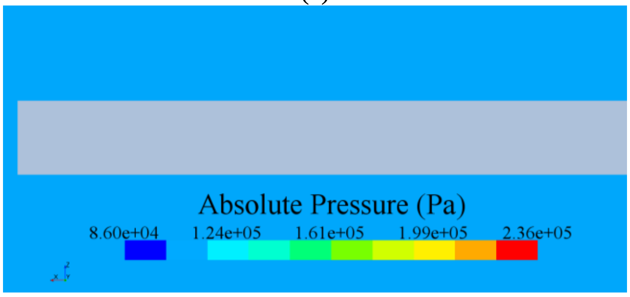


Figure 10: Time history of slamming coefficients of flat plate for compressible and incompressible air



(a)



(b)

Figure 11: Front view of pressure contour showing low pressure region during water entry at 0.26s, from top to bottom: (a) Compressible air, (b) incompressible air

4.3 INFLUENCE OF PERFORATION RATIO

The results of slamming coefficient predicted for different perforation ratios using the studied solver are shown in Figure 12. It can be seen that the slamming coefficient drops significantly from 29.0 at the ratio of 0% to 15.0 at the ratio of 20%, which demonstrates that the perforated structure has a great advantage in reducing the load on the hoisting line during lowering operation. Consequently, the slamming coefficient slightly decreases from 15.0 to 14.4 as shown in Figure 12. The differences among last three ratios (2.6% and 3.3%, respectively) are very small. The free surface deformation and water velocity are presented in Figure 13 and Figure 14, respectively. The plot presented in Figure 13 depicts the free surface deformation when the perforated plate came in contact with the free surface with trapped air. The contact location was found to be at the edge of the plate due to the uprise of water.

The vertical velocity of water underneath most of the plate was significantly suppressed as seen in Figure 14. While the velocity of water at the edge of the plate had a large value, indicating again that the trapped air accelerated the water nearby when it was trying to evacuate from the edge and gaps. However, by comparing the vertical velocity of the free surface, on the bottom surface of the flat plate in Figure 14 (a) with that of perforated plates in Figure 14 (b)-(d), it was found that vertical velocity of the free surface near the four edges of flat plate was accelerated and much higher than the other plates in the same locations. Therefore, the estimated slamming coefficient value for the flat plate is much larger than that for the perforated plates as shown in Figure 12. In addition, the velocity contours of three different perforated plates presented in Figure 14 (b)-(d) are similar which indicates that the ability of the trapped air to escape through the gaps and edges between the bottom of the plate and free surface, is comparable to each other. Therefore, the slamming coefficients of different ratios are very close to each other, if the gap is not slender in shape (length/width ratio smaller than 1.625).

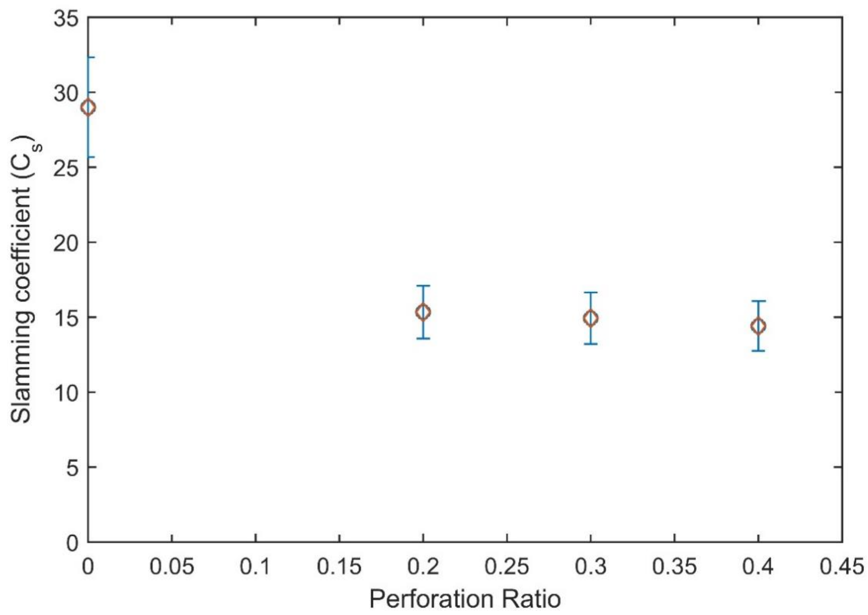


Figure 12: Slamming coefficient of different perforation ratios on fixed layout configuration

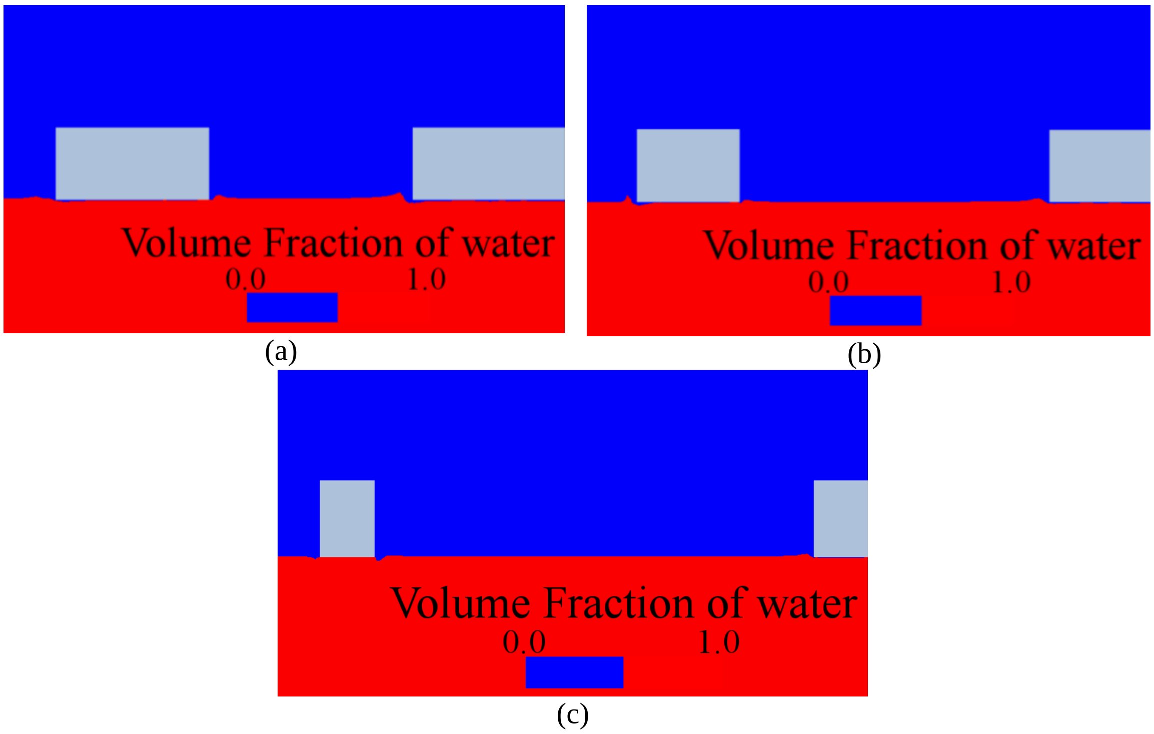


Figure 13: Front view of free surface deformation with respect to different ratios at 0.2s, constant y=3.25m: (a) 20%; (b) 30%; (c) 40%

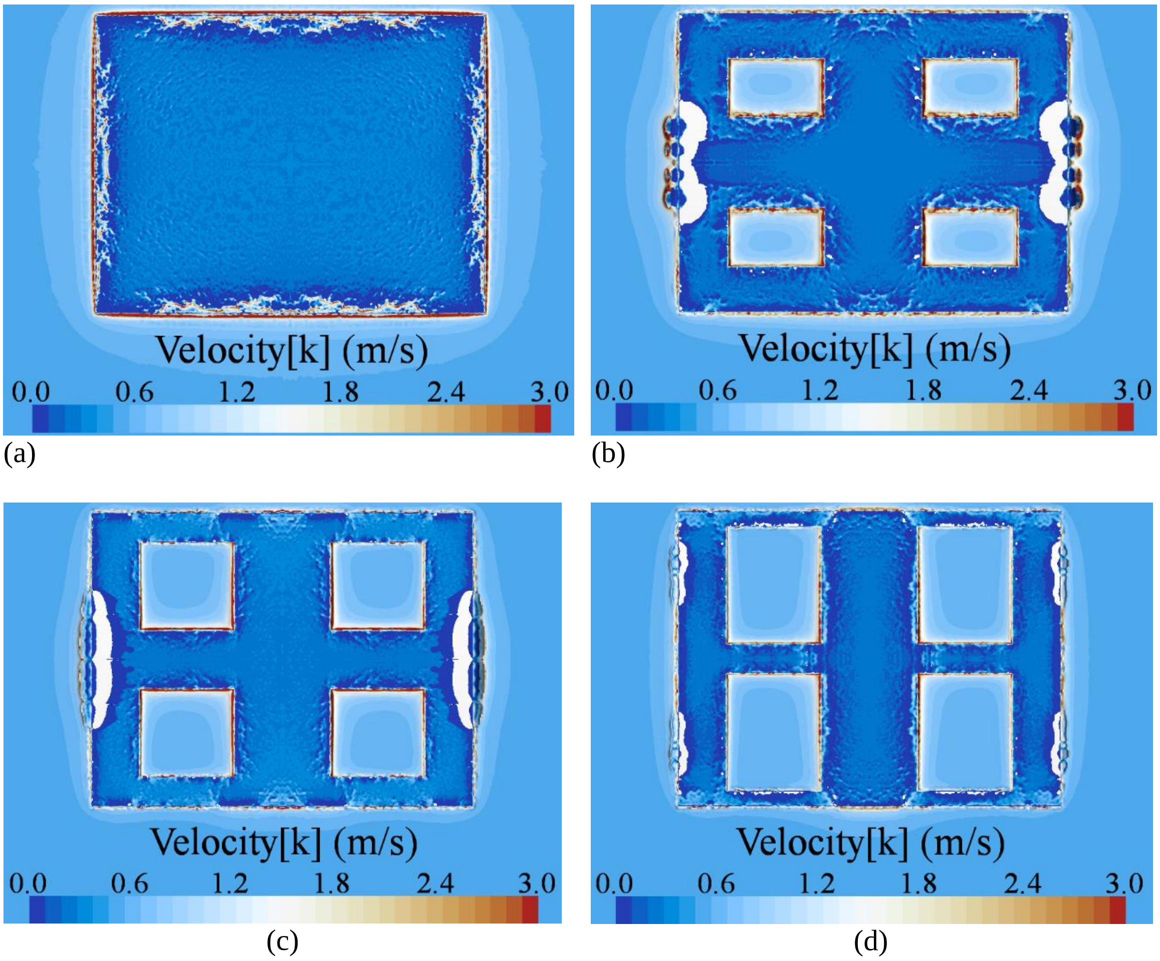


Figure 14: Contour plots of free surface velocity distribution with respect to different ratios at 0.2s: (a) 0%; (b) 20%, (c) 30%; (d) 40%

4.4 INFLUENCE OF GAP LAYOUT CONFIGURATION

The second series of water entry of plates with different layout configuration were simulated as given in Figure 7. As shown in Figure 15, the slamming coefficient increases linearly from 9.7 to 11.3 along with the increase of gap numbers with fixed perforation ratio until 6 gaps layout. Thereafter, the slamming coefficient reduces from 11.3 at 6 gaps to 8.6 at 8 gaps, where the magnitude of reduction rate is 24%. To explain this trend, the corresponded free surface deformations are plotted in Figure 16. The free surface elevations were compared to illustrate the change in slamming coefficient for different gap configurations. It can be seen that there is less interference between plate edge and water in Figure 16 (d), indicating that different layout configurations have a significant impact on the free surface deformation. The free surface contour plots also demonstrate that the air underneath the plate is trying to escape through the gaps and edge as discussed in the previous section. In addition, the reduction in the slamming coefficient for the case of the 8-gaps layout can be explained based on Figure 17. A large low pressure area can be seen under the gaps as compared to the 6-gaps configuration. It is suggested that a large air group isolates the water-plate interaction which leads to a smaller force due to the presence of this low pressure region. Therefore, a conclusion can be made, whereby for a fixed perforated ratio, the slamming coefficient will increase with the increase in length/width ratio until it reaches 19.5. However, a further increase in length/width ratio may impose a negative impact on the air escape due to the increase in gap number. Combining with Figure 18, which shows the vertical velocity of the free surface. The water velocity is clearly suppressed below 0.5m/s for all four layouts, which will lead to less water-plate interaction and a lower pressure. This highlights the importance of the air cushion effect. Additionally, the contact location was seen to shift from the edge of the perforated plate to the middle of the gaps with the increase in gap number.

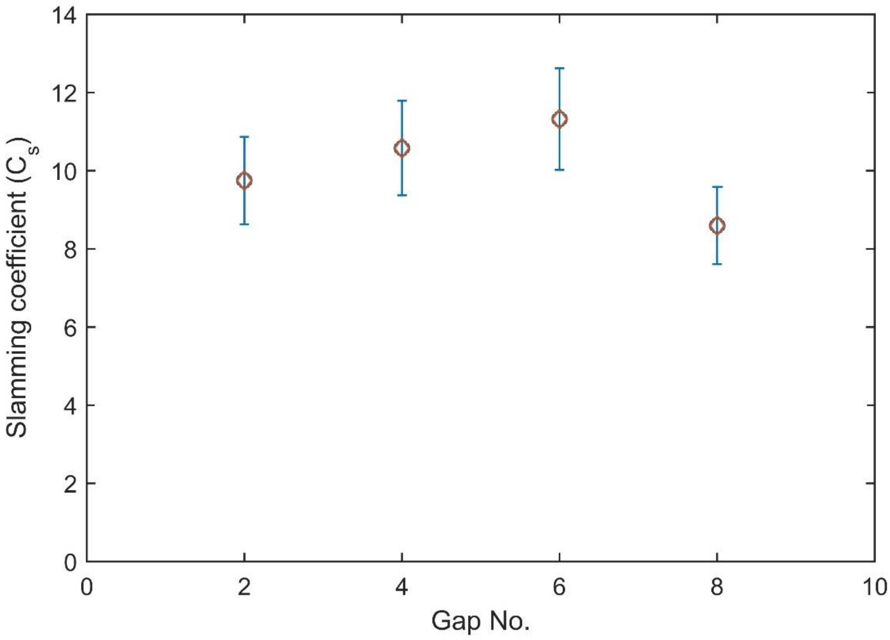


Figure 15: Slamming coefficient of different plate layout configurations on fixed perforation ratio

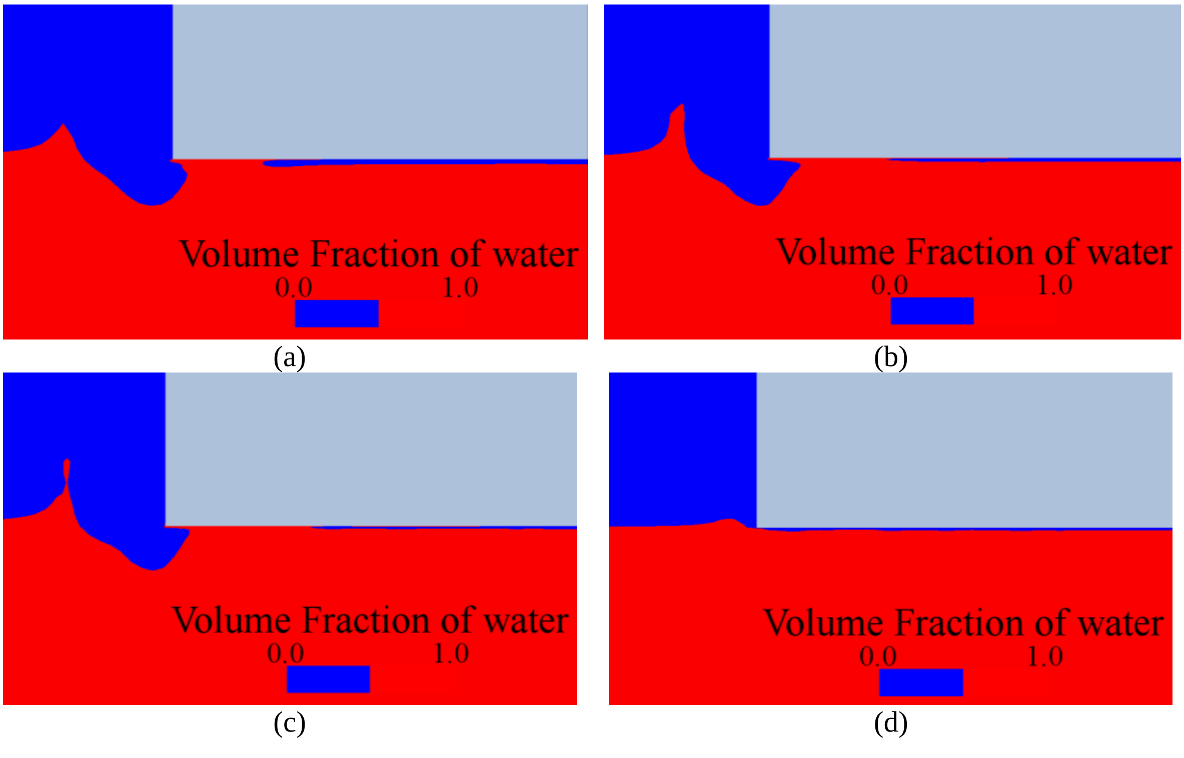


Figure 16: Front view of free surface deformation with respect to different gap configurations at 0.2s: (a) 2 gaps; (b) 4 gaps; (c) 6 gaps; (d) 8 gaps

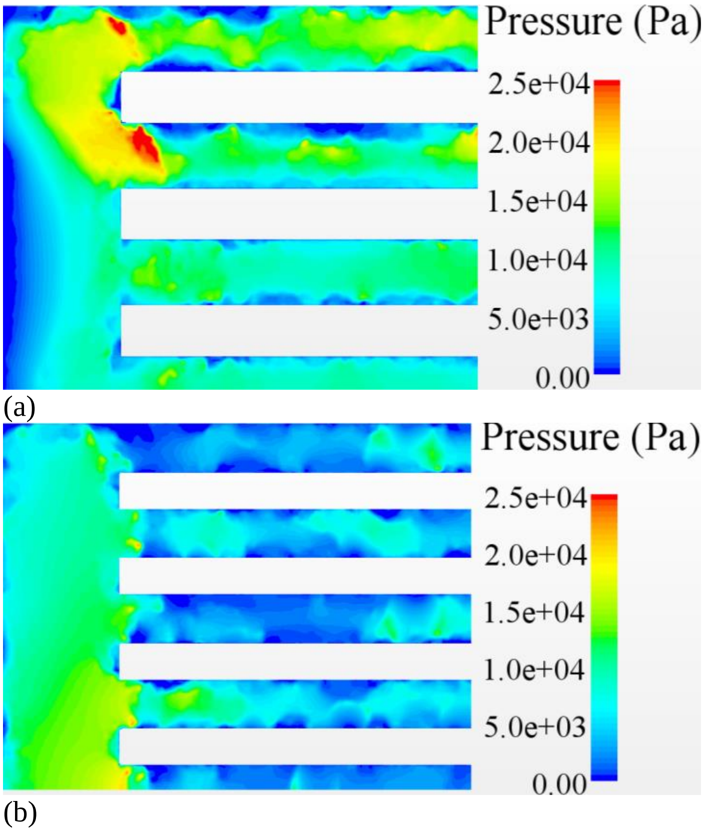


Figure 17: Quarter bottom peak pressure distribution contours with respect to 2 gap configurations: (a) 6 gaps; (b) 8 gaps

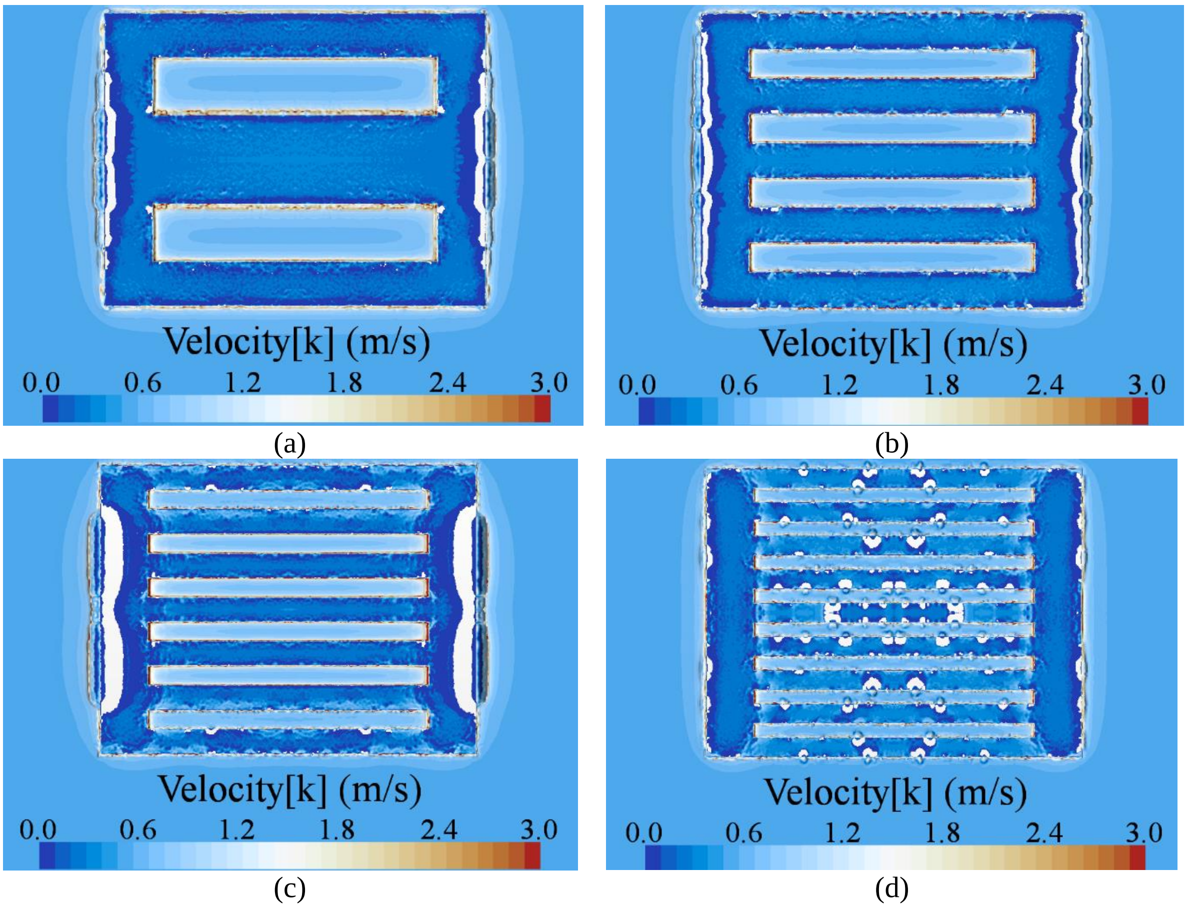


Figure 18: Contour plots of free surface velocity distribution with respect to different gap configurations at 0.2s: (a) 2 gaps; (b) 4 gaps; (c) 6 gaps; (d) 8 gaps

5. CONCLUSIONS

The water entry and slamming coefficients of perforated plates with different ratios and layout configurations were studied and reported in this paper using CFD. The numerical model was verified and validated by simulating a benchmark experiment for water entry of a circular cylinder (Campbell and Weynberg, 1980). From the benchmark study, a large impulsive force occurs within a very short time was found, when the bottom of the cylinder came into contact with the free surface. Besides this, the time history of slamming coefficient agreed well with the experiment value. In addition, the computed free surface deformation exhibited a close correlation with that observed in the experiment, indicating that numerical model can be used for prediction of slamming force.

Upon studying different plates with various perforated ratios and layout configurations, the following conclusions can be drawn:

- The slamming force was significantly reduced and the impact time is extended when the compressibility of air was considered by studying water entry of flat plate.
- For perforated plate with different ratios, if the gap had a small length/width ratio (smaller than 1.625), the ability of the trapped air to evacuate between the bottom of the plate and free surface through gaps was similar.
- For perforated plates with different layout configurations, the slamming coefficient increased with the increase in length/width ratio until it reached 19.5. However, a further increase in length/width ratio may impose a negative impact on the escape of air due to the increase of gap number.
- The series of simulations showed that the URANS equations and VOF method had the ability to model the water entry of large and complex subsea structures.

6. ACKNOWLEDGEMENTS

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THE INFLUENCE OF CENTRE BOW LENGTH ON SLAMMING LOADS AND MOTIONS OF LARGE WAVE-PIERCING CATAMARANS

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M R Davis, University of Tasmania, Australia

SUMMARY

The effect of various centre bow lengths on the motions and wave-induced slamming loads on wave-piercing catamarans is investigated. A 2.5 m hydroelastic segmented model was tested with three different centre bow lengths and towed in regular waves in a towing tank. Measurements were made of the model motions, slam loads and vertical bending moments in the model demi-hulls. The model experiments were carried out for a test condition equivalent to a wave height of 2.68 m and a speed of 20 knots at full scale. Bow accelerations and vertical bending moments due to slamming showed significant changes with the change in centre bow, the longest centre bow having the highest wave-induced loads and accelerations. The increased volume of displaced water which is constrained beneath the bow archways is identified as the reason for this increase in the slamming load. In contrast it was found that the length of centre bow has a relatively small effect on the heave and pitch motions in slamming conditions.

1. INTRODUCTION

1.1 WET DECK DESIGN APPROACHES FOR CATAMARANS

Wave-Piercing Catamarans (WPCs) with centre bows are becoming more widely used for their superior stability, large deck area for loading and unloading cars and trucks, high speeds and ability to avoid deck diving. Figure 1 shows a 112 m INCAT Tasmania wave-piercing catamaran with the centre bow.

Tunnel clearance, centre bow volume, reserve buoyancy and unprotected area of centre bow are key parameters of the WPC hull form. The unprotected area of centre bow, as shown in ……. is defined as the projected area of the centre bow which is visible from the side and is thus not constrained by the demi-hulls during slamming. The water displaced by bow entry is able to exit sideways above the demi-hull bows as the bow enters the water.

Figure 1. A photo of Mols-Linjen 112 m wave piercing catamaran with centre bow built by INCAT Tasmania (www.INCAT.com.au, 2017)

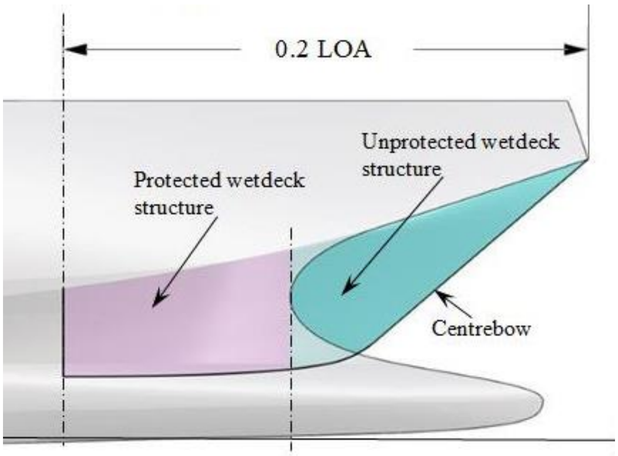

Figure 2. Fore section of an Incat wave-piercing catamaran, LOA = 112m showing the protected and unprotected structures at the bow section with approximately 20% of LOA (Swidan, 2016)

By examining the slamming mechanism from full-scale trials of WPCs (Thomas et al., 2003b, Jacobi et al., 2014), it was proposed that centre bow volume and tunnel clearance have a significant influence on seakeeping and slamming behaviour. These two parameters account for the crucial differences between the two common strategies taken by catamaran designers to reduce or avoid slamming. As depicted in Figure 2 the two approaches to decrease or avoid slamming are either using a large wet deck height or introducing a centre bow with significant reserve buoyancy above the calm water line. Too high a tunnel clearance could result in operational problems when loading and unloading, and too low a tunnel clearance could increase the risk of high slam loads.

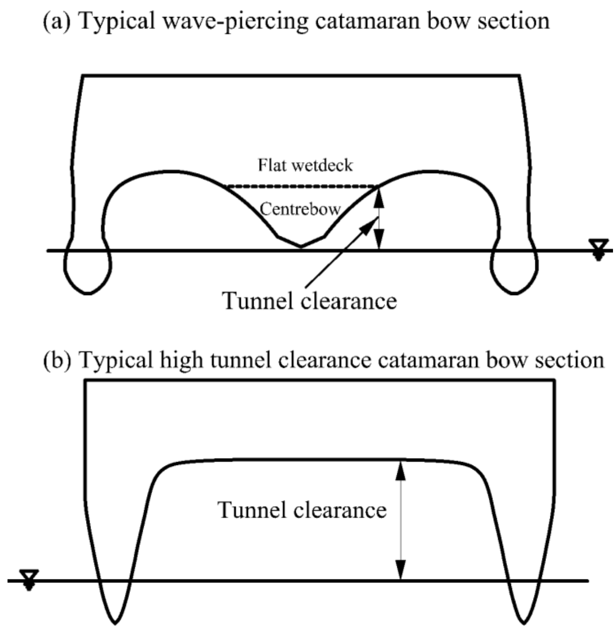

Figure 3. Typical bow profiles of WPCs and conventional catamarans. Narrow water-plane area of demi-hulls, low tunnel clearance and centre bow are the main characteristics of INCAT Tasmania large WPCs

In the case of the WPC configuration adopted by INCAT Tasmania, identification of the optimal configuration of hull form specifically focuses on the centre bow shape and the volume required to improve seakeeping, avoid deck diving and reduce slam loads. By keeping other hull parameters, such as tunnel clearance, constant, one of the major objectives of this work was to investigate the effect of changes in centre bow volume on slamming behaviour including vessel motions and wave-induced loads.

1.2 INVESTIGATION METHODS FOR EFFECT OF HULL FORM ON SLAMMING

Full-scale measurements are an important method for investigating slamming behaviour (Jacobi et al., 2014, Kapsenberg, 2011, Thomas et al., 2003a). However, there are several practical and analytical issues:

- Complicated instrumentation and measurement processes are involved in acquiring data during sea trials, which makes such experiments expensive and time consuming.
- The environmental factors cannot be controlled and so it is difficult to select the required seaway conditions.
- The measurement of the encountered waves is difficult and has significant uncertainties.
- It is difficult to collect pressure data on the hull as the owners/operators are reluctant to drill holes in the vessel hull.
- The key difficulty is to relate the structural response of the vessel to slam loads. A complicated process using a finite element model can be used to obtain a load case by a "reverse engineering" process (Amin et al., 2008).
- It is not possible to investigate the influence of variation of hull form on slam behaviour.

Model drop tests are a useful means for detailed pressure and flow observations on water entry sections. Swidan et al. (2017) conducted a series of three-dimensional controlled speed water impact tests to evaluate the slamming loads of a 3D WPC's bow section using two interchangeable centre bows. However, the three-dimensional effect on slam peak pressures, loads and impact pulse timing has been investigated by Swidan et al. (2016) and Whelan (2004) and Davis and Whelan (2007), showing that simplifying the wetdeck slam phenomenon as a quasi-2D problem can be considered to be an invalid assumption for such complex hull forms. Therefore, application of the results of these tests and relevant two-dimensional theories is not a satisfactory method to investigate the complex vessel sections such as WPC bow sections.

An effective experimental alternative is model testing in a towing tank, where hydroelastic segmented model tests are necessary since the duration of slam loads and the period of whipping are similar and the loading of the ship structure by slamming wave impact is influenced by both the hydrodynamic transient and transient structural response. The results of these experiments can define slamming occurrence criteria and identify motions, slam loads and vertical bending moments (VBM) during slamming and the subsequent whipping vibrations (Lavroff et al., 2013).

Whilst the effect of bow flare shape and general body form for the seakeeping and slamming of monohulls has been investigated e.g. by Kapsenberg and Brizzolara (1999), Hermundstad and Moan (2005) and Bereznitski (2001), there is insufficient data available on the effects of catamaran hull form on seakeeping, slamming and structural loads. Ge et al. investigated wet-deck slamming of a high speed conventional catamaran using a three segment model (Ge et al., 2005). Dessi et al. recently designed and tested a hydroelastic segmented catamaran model measuring motions and slam loads (Dessi et al., 2016). However, the full 3D effect of hull form changes, in particular the effect of variation of bow geometry of WPC has not been investigated. In this paper hydroelastic segmented model experiments are used to investigate the effect of variation of centre bow length and volume.

1.3 FLEXURAL RIGIDITY AND VIBRATORY RESPONSE OF THE MODEL

In view of the 100 m length of the towing tank at the Australian Maritime College, which is 3.5 m wide and 1.4 m deep, an overall model of length of 2.5 m was selected to represent the 2500 tonnes, 112 m INCAT

Tasmania WPC, giving a scale ratio of 1/44.8. Following the approach adopted for an earlier hydroelastic segmented model (HSM01) of the 112 m INCAT vessel (Lavroff et al., 2013, Thomas et al., 2012, Thomas et al., 2010), the new variable geometry Hydroelastic Segmented Model (HSM02, as shown in Figure 4) was designed with each demi-hull split into three segments, the segment cuts being at 33% and 56% of the model length from the transom. The three segments of each demi-hull were connected using short, rectangular section, aluminium elastic links with strain gauges fitted to the upper and lower surfaces of each link (see Figure 5). The two strain gauges on each link were then connected in a single bridge circuit enabling direct measurement of the vertical bending moment in the link.

Figure 4 shows the fully assembled model in the wet dock of the towing tank. As can be seen the mid and aft segment demi-hulls are connected using two carbon-fibre box wet decks and there are two aluminium beams connecting the forward demi-hull segments demi-hulls to the centre bow.

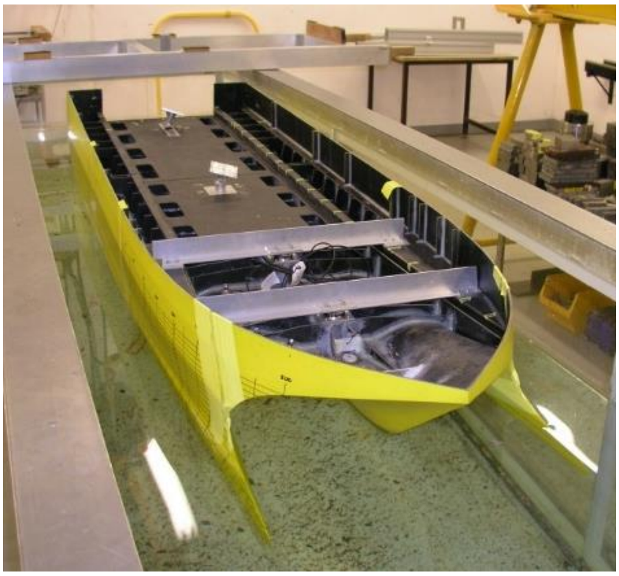

Figure 4: Hydroelastic model HSM02 representing an INCAT 112 m vessel in the wet dock of Australian Maritime College towing tank

The hydroelastic behavior of the model was adjusted to represent the full scale vessel with a first longitudinal mode whipping frequency of approximately 2.2 Hz (Matsubara, 2011). According to model scaling principles the modal frequencies are as given by

$$f_m = f_s \sqrt{\frac{L_s}{L_m}} \tag{1}$$

where f_m and f_s are the model and full-scale frequencies respectively. The desired model wet modal frequency of the model is therefore 14.7 Hz. By assuming rigid segments in the model, the wet vibratory response is determined by the stiffness of the elastic links, the mass distribution of the model, the body form and the surrounding water. Figure 5 shows a drawing of one of the aluminium elastic links used in the model. Both ends of the elastic links were machined to fit inside the rectangular hollow backbone beams mounted within each hull segment with three aluminium bolts (two vertical and one horizontal). Using the method described by (Lavroff et al., 2007) for a three degree of freedom model, the appropriate dimensions of the elastic links (i.e. the dimensions of the square cross section and the length of the link) were obtained to give the required model stiffness.

Since the exact mass distribution of the model was not known before construction and since the segments were not totally rigid, exact prediction of the whipping frequency of the model was difficult. Therefore, impulse experiments were conducted with various elastic link dimensions so as to measure the whipping frequency directly, changing the stiffness by modifying link dimensions. The final frequency was achieved with a stiffer link than originally designed, having a square section of 15 mm and length 13 mm between the two ends. The nominal stiffness of the link based on its cross section and length was calculated to be 2271.6 Nm/rad and the average whipping frequency obtained in water at zero speed was 14.7 Hz which corresponded to 2.2 Hz at full-scale. The average damping ratio for the first few cycles was 0.01, which is similar to the damping ratio observed in full-scale anchor drop tests and in sea trials with severe slamming (Thomas et al., 2010, Thomas et al., 2008).

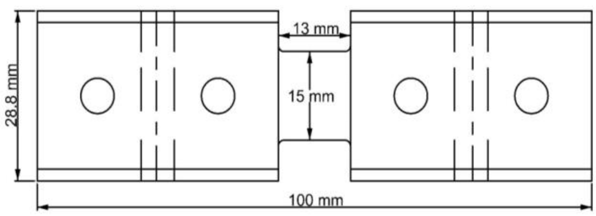

Figure 5: a schematic top view of an elastic link

To ensure rigidity of the connection between the segment links and the demihull segments, aluminium square hollow section backbone beams were build into the demi-hulls and attached to the carbon-fibre hulls at multiple frames. The aluminium square sections for the backbone beams were 32×32×1.6 mm in cross section and had lengths of 225 mm in the forward segment and 260 mm in the aft segment. As can be seen in Figure 6, the backbone beams were supported at least at three points by the transverse frames and bulkheads. Carbon-fibre longitudinal return decks were also assembled on top of the transverse frames to provide increased rigidity to the segments.

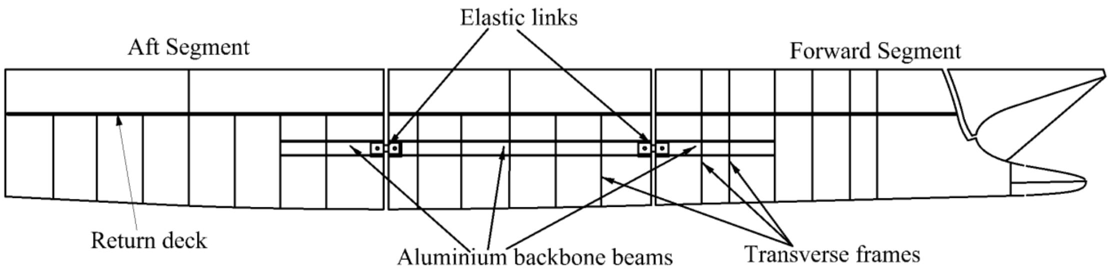

Figure 6: Backbone beams and the transverse frames arrangement in 2.5 m hydroelastic catamaran model HSM02

The model links were calibrated by static loading in both the upright and inverted positions and the calibration factors were averaged for hogging and sagging conditions. Calibration tests were carried out before and after the tank testing programme and only very small changes were observed, these being mainly due to deformation of soft sealing latex covering the small gaps between the hull segments.

2. DESIGN OF THE HYDROELASTIC SEGMENTED MODEL FOR TESTING VARIATIONS OF HULL FORM

The first INCAT Tasmania prototype wave-piercing catamaran was built and tested in 1983. Although following the characteristics of the early design, the geometry and size of the centre bow in INCAT WPCs have varied as these vessels have evolved. By reviewing the different designs adopted, as Figure 7 shows, the centre bow volume has increased in proportion to the vessel size, up to a vessel length of approximately 90 m.

For the recent larger vessels, the centre bow volume has decreased slightly in proportion to the overall hull, becoming approximately constant for vessels larger than 100 m in length. The centre bow is usually truncated at its largest sectional area within the forward third of the vessel overall length. The easiest and most effective way to alter centre bow volume without interfering too much with the bow shape and streamlining is to cut the centre bow volume from its aft end thus changing the centre bow length. The centre bow length is defined as the projected distance between the hull most forward point and the truncation of the centre bow. The Centre bow Length Ratio (CLR) is defined as Equation (2):

$$Centre\ bow\ Length\ Ratio\ (CLR) = \frac{Centre\ bow\ Length}{Demihull\ Length} \tag{2}$$

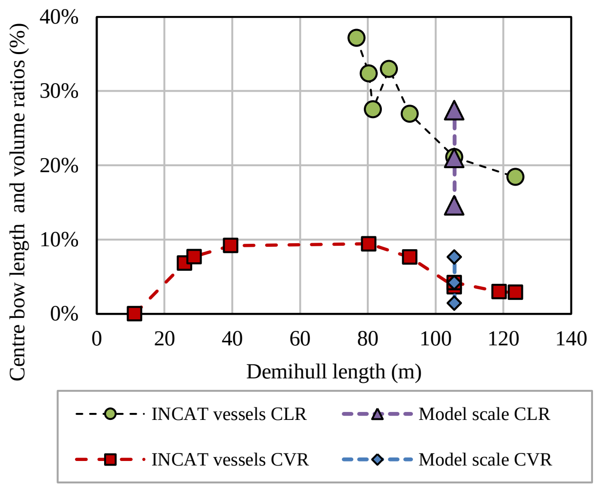

Figure 7: Centre bow length ratio (CLR) and centre bow volume ratio (CVR) of INCAT large wave-piercing catamarans and the ratio variations tested

As seen in Figure 7, the centre bow length ratio (CLR), has decreased from 37% to 18% as the vessel length has increased from 80 m to 130 m. The 2.5 m HSM02 is modelled on the INCAT Tasmania 112 m catamaran that has a 21% CLR and is referred to here as the parent centre bow.

Three different centre bow lengths were considered and the new centre bow lengths were created by removing a 6.72 m (150 mm model-scale) section from the aft end of the parent centre bow model to make a short centre bow and by extending the body lines of the parent centre bow 6.72 m by adding model centre bow segments under the wet deck to make the long centre bow. As seen in Figure 7 the CLR covered by this variation is 6.4% higher and lower than the parent design. Figure 7 illustrates how the added and removed sections create the three centre bow test lengths.

As shown in Figure 8, the centre bow volume is the volume of the centre bow bounded by the keel line of the centre bow and the flat horizontal main wet deck plane extended forward. The centre bow volume varied significantly between the three centre bow test lengths. The key characteristics of these three centre bow lengths are given in Table 1. The model total displacement volume was 0.027 m^3.

Table 1: Three centre bow lengths and volumes selected for experiments with the 2.5m WPC model HSM02

Model Name	Centre bow Length (m)	Centre bow Length Ratio (CLR)	Centre bow Volume (m^3)	Centre bow Volume Ratio (CVR)
Short Centre bow	0.348	14.6%	4.0696×10^{-4}	1.44%
Parent Centre bow	0.498	21%	11.8934×10^{-4}	4.22%
Long Centre bow	0.648	27.4%	21.5579×10^{-4}	7.64%

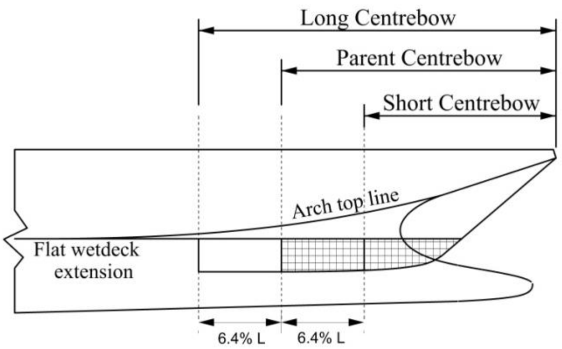

Figure 8: The creation of three centre bow lengths by adding and removing 150 mm pieces from the model parent centre bow truncation. The hatched area shows the parent centre bow volume

The centre bow Volume Ratio (CVR) is defined as,

$$Centre\ bow\ volume\ ratio\ (CVR) = \frac{Centre\ bow\ volume\ below\ the\ wet\ deck\ leve}{Total\ vessel\ displacement} \quad (3)$$

Figure 9 shows the final three centre bow lengths before painting. The extensions to the centre bows were attached to the hull section using three bolts and the gaps filled with plasticine to make the surface smooth and sealed. The weight of the extension pieces was 0.2 kg, but the total centre bow weight and centre of gravity remained constant by using counterweights. For HSM02 the centre bow and arch areas are isolated as a separate segment to capture the slamming loads.

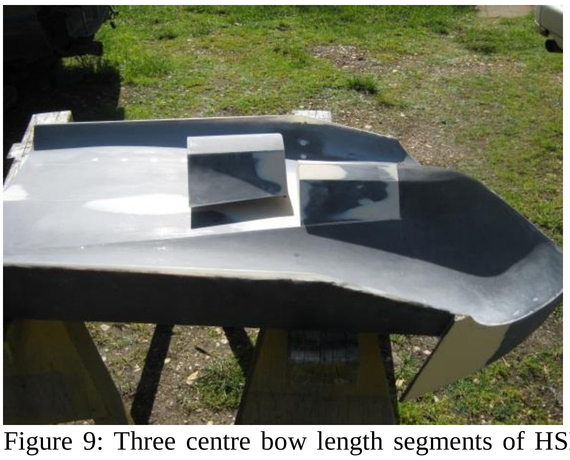

Figure 9: Three centre bow length segments of HSM02 under construction. The short centre bow was achieved by cutting 150 mm from centre bow aft end and a separate segment was built to form the long centre bow.

As seen in Figure 9, the cuts were made in locations in the demi-hulls to include all the centre bow and archways and having minimum water flow disturbances. The centre bow segment extended at its outboard edges to a location where the hull cross section became vertical, the demi-hulls becoming broader below that edge. The centre bow segment was supported by two aluminium transverse beams to the demi-hulls and attached by two 6-degree of freedom load cells to measure the slam forces transmitted by the bow onto the demi-hulls.

3. EXPERIMENTAL PROCEDURE

The experiments were conducted in the Australian Maritime College (AMC) towing tank which incorporates a carriage that runs on rails positioned on the tank walls with a maximum speed of 4.5 m/s. The model was towed using a two post towing system, allowing the model to freely heave, pitch and roll. The average water temperature was 18^{o}C, the average air temperature was 21^{o}C and the density of the water was 998.85 kg/m^3 during the experiments.

The model was ballasted at level trim and the LCG of the model and each segment were measured by balancing the model. The pitch radius of gyration of the model was determined by using the Bifilar technique (Lloyd, 1989). Table 2 shows the model dimensions and parameters. As can be seen, the mass was increased slightly by 0.58 kg compared to the initial design value due to the addition of an extra motion sensor at the LCG.

During the tests, up to 32 channels of signals could be recorded via the data acquisition (DAQ) Card. The recorded channels for which data is presented here were: carriage speed, two LVDT signals on the tow posts to determine heave and pitch, a B&K 4370 piezoelectric accelerometer to measure the centre bow vertical acceleration, one static wave probe and two moving probes one in line with the model LCG and one in line with the model centre bow truncation, four strain gauge channels measuring VBMs at the segment joints and twelve channels recording the signals from the two 6 degree of freedom load cells on which the bow was mounted. The sampling rate was set to 5000 Hz for all the data channels. Figure 10 shows the model located underneath the carriage in AMC towing tank.

Table 2: HSM02 Model Particulars and Equivalent Full Scale Values

Item	Model scale	Full-scale
Scale factor	1/44.8	1
Length overall	2.5 m	112 m
Demihull length	2.35 m	105.6 m
Displacement	27.7 kg	25530 tonnes
LCG	0.941 m from transom	42.15 m from transom
Radius of gyration	0.67 m from LCG	30.16 m from LCG
Forward segment mass, LCG	8.37 kg, 1.804 m	-
Mid segment mass, LCG	7.57 kg, 1.298 m	-
Aft Segment mass, LCG	11.76 kg, 0.381 m	-
Trim	0 degrees	0 degrees
Fundamental structural modal frequency in calm water	14.7 Hz (measured)	2.2 Hz

Figure 10: The AMC towing tank carriage and the HSM02 in the water

The three centre bow configurations were tested at 1.53 m/s and 60 mm wave height, which correspond to 20 knots and 2.688 m of wave height at full-scale. Due to the complexity of the instrumentation and model set up for testing in different conditions, the main aim for the experiments was to obtain high quality results for a limited range of conditions. Regular waves were generated at encounter frequencies from 0.4 Hz to 1.1 Hz, with intervals of 0.5 Hz or 0.25 Hz. Better resolution of model motions and loads was achieved around the frequencies of peak motions by conducting tests at smaller frequency intervals. Selected runs were repeated multiple times to determine repeatability of the obtained results e.g. in peak motions.

4. MOTION RESULTS

Following ITTC guidelines (ITTC, 2014), motion results for seakeeping experiments are presented as non-dimensional response amplitude operators (RAOs), the heave RAO being the ratio of heave to wave height and the pitch RAO being the ratio of pitch value to maximum wave slope. Wave slopes for the longest waves were corrected for shallow water effects based on semi-empirical dispersion formula provided by (Fenton and McKee, 1990, Fenton, 1990). The motion RAOs are shown as functions of the non-dimensional encountered wave angular frequency (ω_e^*),

$$\omega_e^* = 2\pi \times f_e \times \sqrt{L/g} \qquad (4)$$

where L is the vessel overall length, g is the gravitational acceleration and f_e is the encountered wave frequency observed by the moving wave probes. Figure 11 shows the heave RAOs for the three centre bows. As expected, the heave response in high frequency waves is very small and in the long waves (low frequencies) the heave response RAO tends to one. There is a local maximum around $\omega_e^* = 3.7$. Slamming was visually observed in the mid-range of frequencies $(3.7 \leq \omega_e^* \leq 5.4)$ for all the three centre bows. It was found that less than 1% standard error was observed in the non-dimensional heave values for three repeat runs.

The heave results show that for high and low frequencies, the results for the three centre bows are quite similar although in the frequency region of the local peak in RAO $(3.4 \leq \omega_e^* \leq 4)$, the short centre bow has about a 13% higher heave motion than the long centre bow.

Figure 11 shows the pitch RAO with the three centre bows. As can be seen, the pitch RAO also tends to one at low frequency and reduces to zero at high frequency. The maximum pitch RAO occurs between $\omega_e^* = 3$ and 3.4 which is a slightly lower frequency than the frequency of maximum heave RAO. Similarly, less than 1% standard error was observed in the non-dimensional pitch values for three repeat runs.

The pitch motions show only small differences between the three centre bow lengths near to the frequencies of maximum pitch RAO. There is also a slight shift in the frequency of maximum pitch RAO between the three centre bow lengths, with the short centre bow having the smallest frequency for maximum pitch ($\omega_e^* = 3.14$). The small change in the frequency of the maximum magnitude of the RAOs appears to imply that the effect of a change in hydrodynamic stiffness is greater than any change in inertia forces in waves due to added mass effects. That is, the lesser volume in the short centre bow creates less pitch stiffness compared to the other two centre bows. For frequencies between $\omega_e^* = 3.7$ and $\omega_e^* = 5$ where severe wet deck slamming occurred during the runs, the pitch response of the short centre bow is on average 5% less than for the long centre bow. The reason for this will be discussed in more detail in the following sections when the slam loads are considered.

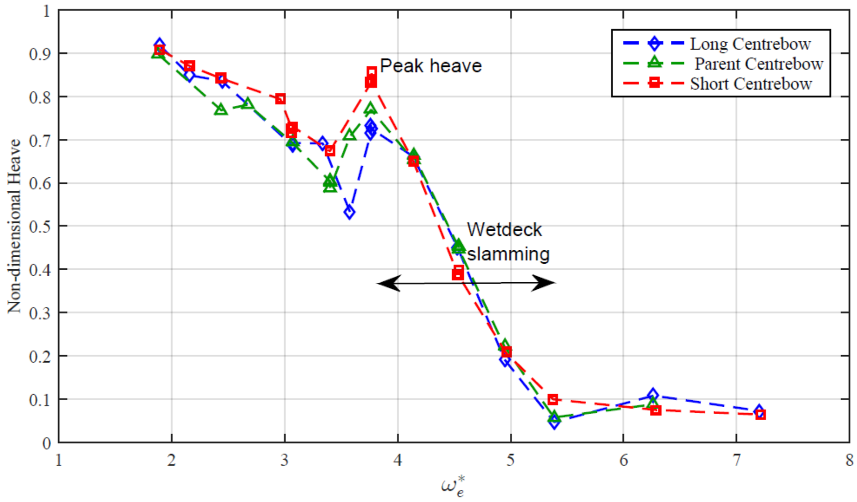

Figure 11: Non-dimensional heave response (RAO) for various centre bow lengths on HSM02 ($H_w = 60$ mm, speed=1.53 m/s)

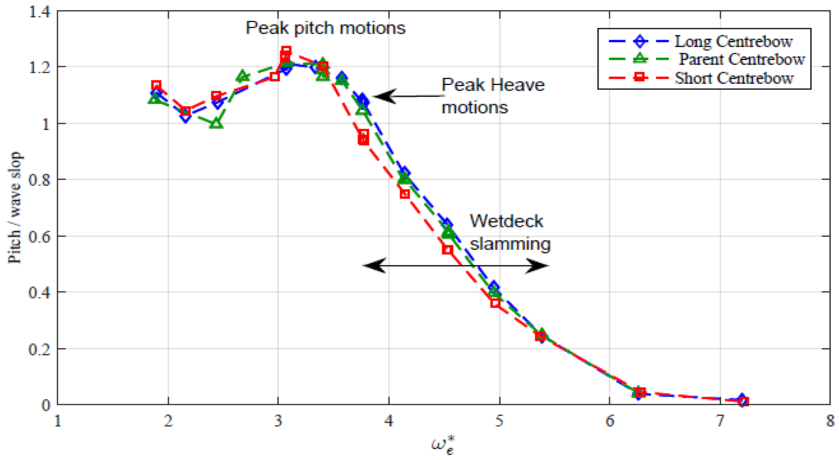

Figure 12: Non-dimensional pitch response (RAO) for the three centre bow lengths in HSM02 ($H_w = 60$ mm, speed=1.53 m/s)

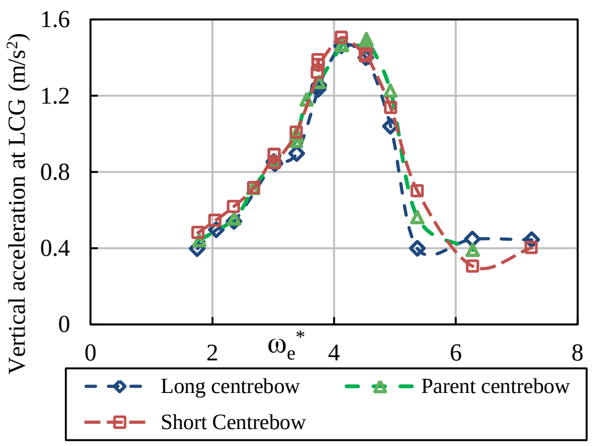

Figure 13: Accelerations measured at the LCG of the model for the three centre bow lengths in HSM02

Figure 13 shows the acceleration results at the LCG calculated from the two LVDT motion signals. The signals were low pass filtered with 4th order Butterworth with 5 Hz cut off frequency to remove the high frequency components of noise and structural vibrations. As seen, the peak LCG accelerations occurred around $\omega_e^* = 4.13$ which is higher than the heave and pitch peak frequency but a little less than peak slamming conditions. There is no significant difference between magnitudes of peak accelerations with the differing bow lengths.

Figure 14 shows the accelerations at a location 37% of LOA forward of LCG, corresponding to the forward end of the bridge at full scale, measured by a B&K accelerometer. Accelerations at this position are significantly greater than at the LCG due to the hull pitching motion. At $\omega_e^* = 4.13$ where peak accelerations are observed, it is evident that the larger centre bow volumes increased the accelerations at this more forward position. The results show that the long bow accelerations are around 7% higher compared to the vessel with the parent bow. Similarly, parent bow accelerations are 9% higher than the short bow. The reason for this difference between the accelerations would be the variation in slam induced load severity in the forward bow areas of the vessel. To investigate the loads induced to the model, vertical bending moments are analysed in the next section.

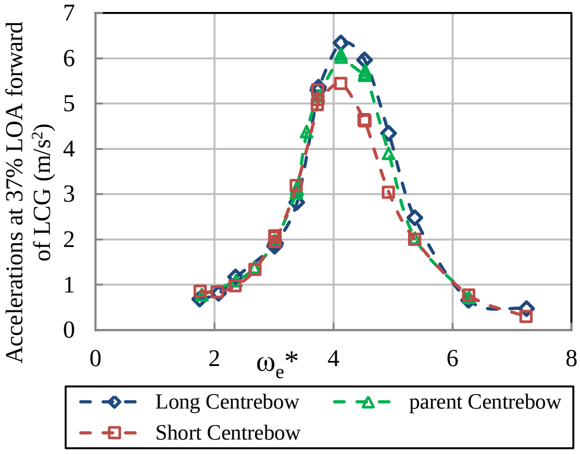

Figure 14: Acceleration measured (and filtered) at 37% LOA forward of LCG for the three centre bow lengths in HSM02 (H_w = 60 mm , speed=1.53 m/s)

5. VERTICAL BENDING MOMENTS (VBMs)

VBMs were derived from the strain gauges on the elastic links at the transverse cuts. The forward cut is at 1409 mm (56% of LOA) from the transom and the aft cut is located 820 mm (33% of LOA) from the transom. In the sagging situation VBM is considered positive and in the hogging VBM is negative and the VBM at each cut is the addition of the VBM from the starboard and port side elastic links.

Figure 15 shows sample time histories of the VBM recorded at the forward cut in (a) run 45 ($\omega_e^* = 4.13$) and (b) run 37 ($\omega_e^* = 6.28$) both with H_w = 60 mm and speed=1.53 m/s. In calm water and at zero speed, VBMs were first removed in each run to eliminate instrumentation bias signals before testing in waves. Figure 15(a) shows the global bending moments due to the encountered waves with only modest levels of whipping or bending vibration being evident. Figure 15(b) clearly shows the slam induced sharp VBM peaks and the consequent strong vibrational loading due to hull whipping. The peaks are visible after a peak slam sagging VBM and are followed by large whipping oscillations around the mean line.

Conducting Fast Fourier Transform analysis on the slam induced VBM signals revealed that the whipping vibratory response of the model to wave slam is at 12.82 Hz, somewhat lower than the impulse response at 14.7 Hz in calm water at zero speed. The whipping effects are significant and they are not fully damped by the time that the next slam is experienced. The average damping ratio of the VBMs for this run is 0.23, which is significantly greater than the value of 0.015 for the model in calm water and at zero speed. Full-scale slam decay coefficients calculated by Thomas (Thomas et al., 2008) on INCAT Hull 050 (96 m length) ranged between 0.05 and 0.4. This means that the model structural damping ratio is in a similar range to that of full-scale vessels. Figure 16 shows a sample strain gauge record from a full-scale keel plate on INCAT Hull 042 (86 m length) after being excited by a slam (Thomas, 2003). The comparison between the model raw data and full-scale data demonstrates the effectiveness of the model in replicating the dynamic behaviour of the full-scale vessel.

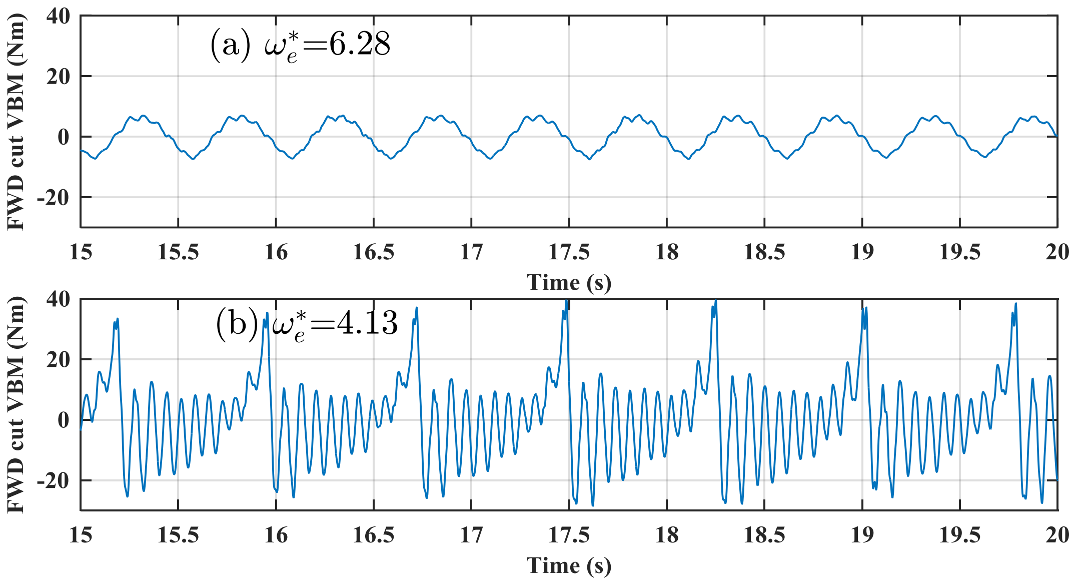

Figure 15: Sample recording VBM data of forward cut (56% LOA) for H_w = 60 mm, speed=1.53 m/s for (a) run 37 with $\omega_e^* = 6.28$ and (b) run 45 with $\omega_e^* = 4.13$

The whipping frequency is not an exact multiple of the slam or wave encounter frequency of course. Hence, each slam-induced VBM peak can vary as they are influenced by the slam load severity and also by the whipping effects from the previous slam. Observing such variations in VBM peak values (as seen in Figure 15(b)) demands presentation of the loads in a probabilistic manner to show the uncertainty involved. In a given environmental condition, both the distribution of load values and the extreme loads can be important.

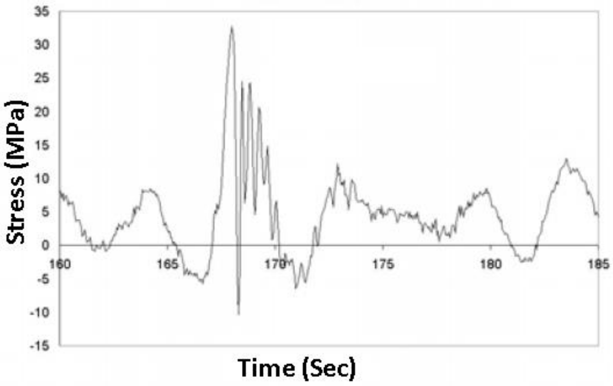

Figure 16: Keel plate stresses frame 24.5 of full-scale measurements on INCAT Hull042 catamaran (86 m length) (Thomas et al., 2008).

Depending on the encounter wave frequency, the number of regular wave load cycles observed in each towing tank run varied between 7 and 25 peaks. In the runs where slamming occurred the number of slams was between 12 and 20. To identify the distribution of peak values as an example, the results of aft cut sagging VBM peak values from four similar runs with $\omega_e^* = 4.53$ (the encounter frequency with the largest slam loads) in the parent centre bow configuration were analysed. Figure 16 shows the histogram of these VBM peak values where from the total of 78 peak values the mean is 38.48 Nm and the standard deviation is 2.79 Nm. It is seen that the samples are somewhat evenly distributed around the mean value and this suggests that a normal distribution can be assumed for the VBM values for a particular run.

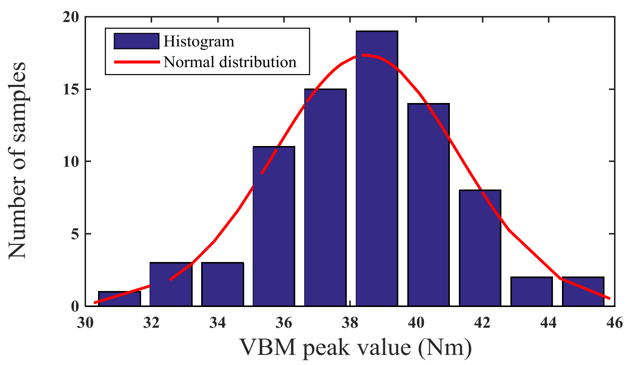

Figure 17: The histogram of slam induced VBM peak values for the forward cut during similar 4 runs (H_w = 60 mm, speed=1.53 m/s, $\omega_e^* = 4.53$). The mean is 38.48 Nm and standard deviation is 2.79.

To verify this assumption, a Kolmogorov–Smirnov test (Justel et al., 1997), a nonparametric test of the equality of continuous, one-dimensional probability) was performed on the data, which showed that with 98.7% confidence a normal distribution can be accepted for this distribution. Figure 18 shows the cumulative distribution function for the normal distribution and the sample data where good concurrence is observed.

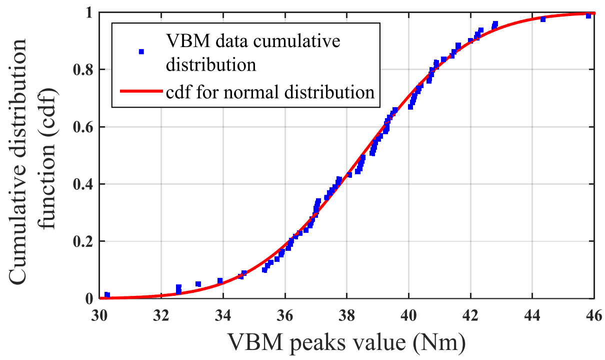

Figure 18: The cumulative distribution of the VBM peak values of parent centre bow during 4 similar runs compared to the normal distribution (H_w =60 mm, speed=1.53 m/s, $\omega_e^* = 4.53$)

The VBM values in hogging or sagging condition were calculated as the mean of the peak values in each encountered wave when the run was in steady wave conditions. The non-dimensional bending moment, VBM* is calculated by Equation (5),

$$VBM^* = \frac{VBM}{\rho g H_w \nabla} \qquad (5)$$

based on the method used by Colwell (Colwell et al., 1995), where ρ is the water density, H_w is the wave height and ∇ is the vessel displaced volume. Figures 19 and 20 show the measured peak bending moment in forward cut (56% of LOA) and aft cut (33% LOA) for the three centre bows with 95% confidence intervals as a function of the non-dimensional encounter frequency. As can be seen for high and low encounter wave frequencies the measured VBM results tend to zero and there is negligible difference between the three centre bows in those regions. In the frequency range of slamming ($3.7 \leq \omega_e^* \leq 5$) the VBM values increase and there is a clear difference between the responses of the three centre bows. Although there is not a sharp peak, the VBM due to slamming is close to a maximum when $4.13 \leq \omega_e^* \leq 4.53$.

The hogging VBMs are smaller than sagging VBMs in the frequency range of slamming. The hogging peaks are of course the consequence of the slam events which induced the preceding sagging peaks when the archways are filled up after a wet deck slam. The hogging moments arise as a combination of two effects: the commencement of whipping vibration and downward loads on the bow as it moves upwards out of the water after a slam event, the latter being associated with added mass of the bow whilst in the water.

Figures 19 and 20 also show that the average VBM peak value for the long centre bow was 52% and 40% higher than the short centre bow at the forward and aft cuts respectively. This difference can be attributed to higher total slam forces imposed on the longer centre bow compared to the shorter centre bow.

Figure 21 show the non-dimensional VBM recorded in severe slamming conditions at the forward and aft cuts as a function of Centre bow Length Ratio (CLR) for different non-dimensional encounter wave frequencies. As can be seen, the VBM peaks in slamming conditions increase monotonically with the centre bow length ratio. This is an important trend, which shows the slam-induced loads on the structure increase linearly with the centre bow length.

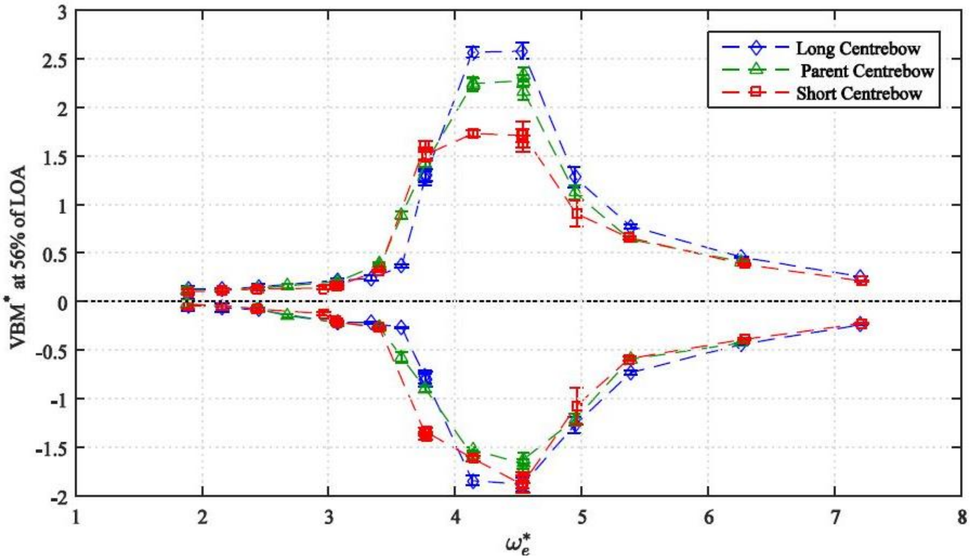

Figure 19**:** The non-dimensional forward cut (56% LOA) peak vertical bending moment ($VBM*$) for the three centre bow lengths ($VBM^* = \frac{VBM}{\rho g H_w \nabla}$)

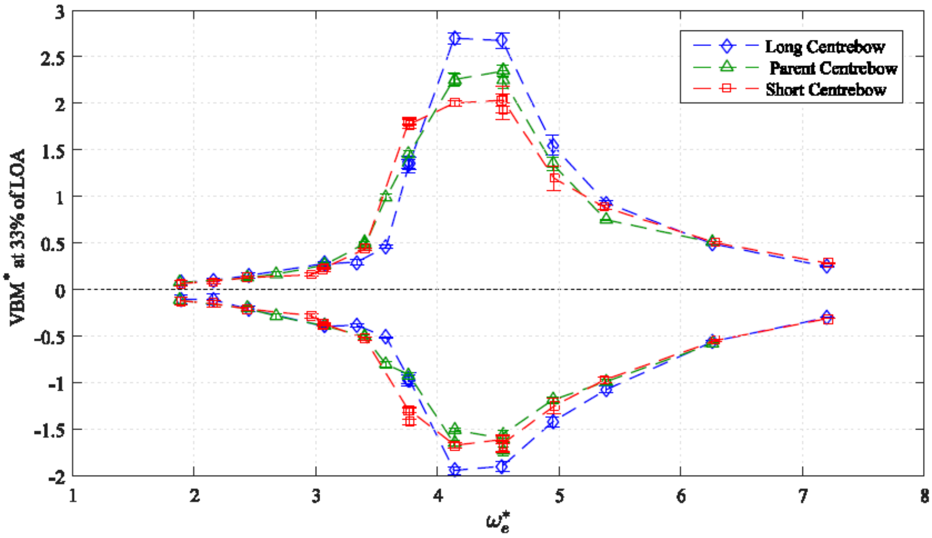

Figure 20**:** The non-dimensional aft cut (33% LOA) vertical bending moment (VBM^*) for the three centre bow lengths ($VBM^* = \frac{VBM}{\rho g H_w \nabla}$)

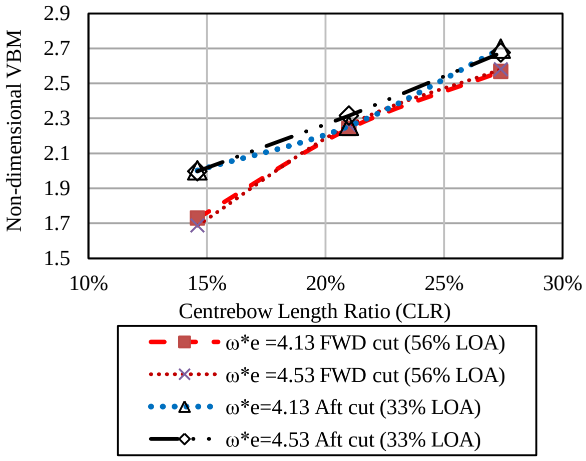

Figure 21: Non-dimensional vertical bending moment for varying centre bow length at two locations along the model

In summary, for the tested condition (equivalent to 20 knots forward speed and 2.68m waves) it has been seen that the heave results showed up to 17% difference between the three bow lengths, whereas the pitch results varied by only about 5%. The bow accelerations were about 25% less for the shorter centre bow than the longer centre bow under slamming conditions. The VBMs were also significantly lower for the short centre bow and increased monotonically with the centre bow length. The main reason for the lower slamming loads with the short centre bow is the lower volume of water being constrained under the wetdeck within the archways as most of the water displaced by the shorter bow can be displaced outwards in a sideways direction over the projecting demi-hull bows in the unconfined areas of the bow. Although this sea condition and speed does show a representative example of slamming for the WPC design, other speeds and wave heights need to be examined to establish a more thorough conclusion.

6. CONCLUSIONS

Measurement of motions and loads for each of the three models of varying centre bow length was undertaken successfully in regular waves of 60 mm height (2.68m full-scale) and at 1.53 m/s speed (20 knots full-scale) at different wave encounter frequencies. The results can be summarised as following:

- The heave motions of the vessel showed a maximum at dimensionless encounter frequency $\omega_e^* = 3.7$. The peak heave of the short centre bow was higher than the parent, and the parent centre bow was higher than the long centre bow. At other frequencies, the difference in the heave RAO between the centre bows was much reduced.
- The maximum peak of the pitch response of the vessel was in the range $3 \le \omega_e^* \le 3.5$. Only a small difference was observed between the peak values of the three model configurations. However, a slight shift of the frequency of maximum pitch for the short centre bow toward lower frequencies was observed. The reason was likely due to the reduction in model hydrostatic stiffness with the short centre bow. The main difference in pitch response was in the frequency range of slamming ($4 \le \omega_e^* \le 5$) in which shorter centre bows had an average 5% less pitch compared to the longer centre bows.
- The peak LCG acceleration differences between the three bows were small at the LCG but at the forward end of the bridge position the accelerations showed about 20% differences between the three bows, being higher for the longer bows.
- The VBM peak values due to slamming increased as the centre bow length increased.

For the conditions tested, the shorter centre bow was shown to be more effective in slamming conditions by reducing both motions and loads. The reason is that with shorter centre bows the displaced water exits from the sides over the forward bows of the demi-hulls and does not become constrained under the archways. The larger slam forces arise for longer centre bows due to this constraint and induce larger upward pitch motions which can then intensify the slam condition further. This however, does not mean that the benefits of centre bow length reduction will increase if the centre bow is removed completely. Designers should note that although shorter centre bows may give less loads and motions in slamming conditions, the clear advantages of having a centre bow to prevent bow diving and the potentially more extreme nature of slams on a flat wet deck should not be overlooked. The main conclusion is that constraining the water between the centre bow and demi-hulls should be minimised as much as possible. This conclusion leads to a recommendation for future work to assess experimentally more diverse hull forms in the bow region, such as moving the jaw (the point where the bow connects to the upper edge of the forward wave piercing demi hull) further aft. However, due to the nonlinearity of the vessel motions with respect to the wave height it is difficult to extrapolate the results observed in the present tests to higher wave heights and for different wet deck clearances. Therefore more extensive testing in large wave height and higher speed conditions with variable wet deck clearances is a priority for future work.

7. ACKNOWLEDGMENTS

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CHOOSING THE STYLE OF A NEW DESIGN - THE KEY SHIP DESIGN DECISION

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SUMMARY

As a former senior designer of naval vessels and, more recently, a leading researcher in ship design, the author has previously presented a description of the ship design process in terms of the important decisions a ship designer makes in concept exploration. Such decision are made consciously or unconsciously in order to produce a new design or, preferably, any design option. It has been contended in many publications that the first real decision that a ship designer makes, in order to proceed, is the selection of the "style" of the design study or of a specific design option. This term was adopted in order to cover, not just a host of design issues and standards implicit in a given study, but also, at this very initial step, the overall characteristics of any particular study. So the term style could be said to be doubly important.

The current paper considers the nature of the early ship design process for complex multi-functional vessels and then retraces the origins of the particular use of the term, where it was seen as the last of the five elements in Brown and Andrews' 1980 encapsulation of the ship design issues that matter to the naval architect, incorporated in the term "S to the 5th". This leads on to consideration of the various aspects of design style, many of which could be considered "transversals" as they apply across the naval architectural sub-disciplines and to the component material sub-systems comprising a ship. One of the distinctive advantages of the architecturally driven ship synthesis or Design Building Block approach is that it can address many of these style issues in the earliest descriptions of an emergent design study. Examples, drawing on a range of built Royal Navy ship designs, are presented to show their top-level style characteristics, followed by a series of ship design research studies illustrating how the impact of specific component style aspects can be investigated in early stage ship design, using the UCL Design Building Block approach. Finally, recent research led investigations into integrating ship style into early stage ship design are summarized to demonstrate why the choice of "style" is seen to be The Key Ship Design Decision.

"In matters of importance, style is everything" Oscar Wilde, (quoted by Rybczynski, 2001)

1. INTRODUCTION – STYLE IN EARLY STAGE SHIP DESIGN

There is an issue with the term "style" when used to describe the design of large scale products, in that style can often be seen as somewhat superficial and highly subjective, hence Oscar Wilde's aphorism, above. It is often taken in engineering or architectural design to be synonymous with appearance and therefore, in most ship designs, seen to be of relatively little importance compared to economic or operational considerations. This is despite the realisation by the US Navy in the Cold War that appearance does matter (Roach and Meier, 1979).

Architects, in straddling the arts and the sciences, historically were seen to adopt particular styles. Thus the American writer on architecture, Witold Rybczynski (2001), says architects are uncomfortable about style, because:

a. their suspicion of it, inherited from reactions to the Modern Movement;
b. adopting a particular style is seen as putting an architect into an " uncreative box";
c. talking of "style" makes architecture (which is a serious business) sound frivolous;
d. seeing style as "subject to the whims and fancies of fashion".

He considers all this is unfounded "as an architecture that recognises style – and fashion – is an architecture for the rest of the world". While this may seem rather rarefied to most design engineers, the issue of subjectivity and a necessary emphasis on human factors, are both key to the nature of style as it is considered it should be addressed in complex ship design. It is in this rather wider sense of style that is used in the current paper.

"Style" was explicitly incorporated as a characteristic of a ship design by Brown and Andrews (1980) and is the fifth "S" of the "S^5" ship design characteristics, seen to be of importance to naval architects in ship design. The other characteristics are those of Speed, Seakeeping, Stability, and Strength (see Figure 1 with a selected example of each S^5 characteristic). Each example ship design in Figure 1 is considered to be revealing of that characteristic. Thus for "Speed" (really Resistance and Propulsion) HMS SPEEDY, a hydrofoil Offshore Patrol Vessel, is an example of an attempt to radically change the approach to offshore protection by recourse to employing a very high speed "interceptor" (Brown & Marshall, 1978). The "Seakeeping" example is of a Leander Class frigate slamming in high sea state, despite the hull form of that class being designed from World War II experience in the North Atlantic. The "Strength" example shows a Weapon Class destroyer being subject to an underwater explosive test (UNDEX), to emphasise that naval ship structural design is about surviving weapon effects not just the wave loading from sea. "Stability" is still a comparative measure with the mid-Victorian disaster that instigated statical stability practice, HMS CAPTAIN, with that too radical design's inadequacies revealed by its GZ Curve (Brown, 1983). Finally the example is that of "Style" and is Baker's St LAURENT design produced when Sir Rowland Baker was seconded to the Royal Canadian Navy. His frigate design is an example of a consciously "stylised" configuration, showing Baker's intention "to put the RCN's first indigenous design on the map" (Brown, 1983), and so it is discussed further in Section 4.

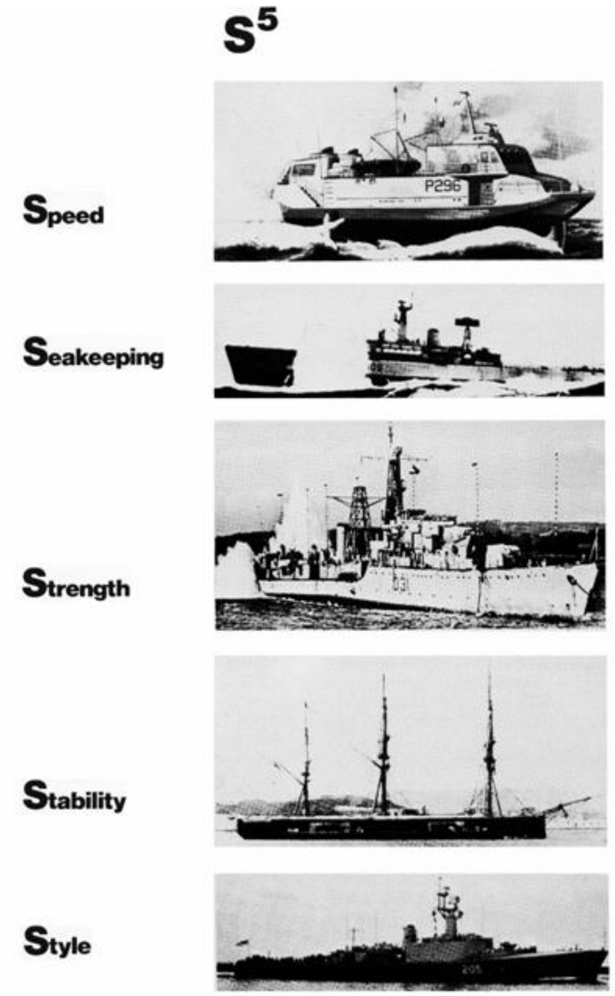


Figure 1: The "S^5" topics that characterise the naval architect's concerns in ship design (Brown & Andrews, 1980)

However adopting the term style to denote a type of information or ship design characteristic, that sets it apart from other characteristics (or even the accepted sub-disciplines) of naval architecture, such as "speed" or structural strength, has some resonance with architectural usage. A slightly different take by Pawling et al (2013) considers the design issues generally grouped under style, to be conceptually different to the other naval architectural disciplines, not because many are unsuited to mathematical analysis (the same was once true of Seakeeping or structural vibration analysis) but because style is a cross-cutting concept. Thus a decision, by the ship designer, on a discrete aspect of style explicitly influences a wide range of solution features. Stylistic information also has the key property of addressing uncertainty, since the term is intended to cover both "hard" knowledge (such as adoption of specific structural standards) and "soft" knowledge (such as guidance on ship internal layout). Such knowledge can then be conceptually connected or grouped. In addition, style choices may also be reflected in any weighting factors chosen, should there be multiple and highly disparate criteria involved in the ship designer selecting a design preference from amongst a wide range of design options.

An example of a transversal style choice would be the level and extent of survivability adopted in a naval ship design. A decision on the level of survivability can influence a wide range of overall and detailed design features, such as the choice of signatures and defensive systems (to prevent a hit), the spacing of bulkheads (to resist weapon effects and any subsequent flooding), the mutual arrangement of compartments (to protect vital spaces and aid in recovering from damage) and structural details (to resist the result of underwater shock on the structural hull girder). This survivability example also illustrates another feature of style in that it is cross cutting across the responsibilities of the engineering disciplines involved in a ship design (such as naval architecture, marine engineering, combatant system engineering). In this regard style choices could be said to be particularly critical in decision-making at the crucial earliest stages of complex ship design. It could be argued, however, that the difference between style and the other components of "S^5" is a matter of degree, given that all the aspects of ship design interact to a greater or lesser degree with each other.

The paper next discusses, at the macro and major levels of design decision making, the transversal nature of style and presents a specific categorisation of many of the style aspects appropriate to the design of complex ships. This requires outlining some of the characteristics in the overall design process for such vessels. In order to emphasise the importance of choosing the overall style of a new ship design option, in the earliest exploratory stage of early stage ship design (ESSD), Section 3 considers some actual ship designs in which a distinct choice appears to have been made with regard to the overall style of the emergent design. These examples are taken from a range of UK naval and auxiliary ship classes, some of which the author had design involvement in (or in the case of the last, prospective, example the author poses some pertinent questions on its viability as a style choice). Section 4 considers the specific cross cutting feature of ship architecture as a means of highlighting the significance of style choice. The subsequent section outlines examples of design studies undertaken by the author's ship design research group at UCL, as each of chosen studies investigated an aspect of style identified earlier in the paper. The paper concludes with an outline of some recent generic research as to how style, as an overarching design consideration, might be better addressed in the ESSD of complex vessels.

2. STYLE ADDRESSING THE TRANSVERSALS AND CATEGORIES OF STYLE

The style to be adopted in a specific design option is seen to be the key design decision for that option and so is the first design decision (beyond deciding that a certain range of solution options is to be investigated). This is indicated in the overall ship design process representation shown in Figure 2 and taken from the author's COMPIT 2013 paper (Andrews, 2013), where each step or decision selection is explained more fully in the appendix to that paper. Thus Selection of the Style of the Emergent Ship Design is the first design choice and can be seen to impact at the macro, major and micro levels. Macro level denotes the overall style of a design or preferably a design option, whether it is, for example, a conventional warship, a more utility or austere design or a radical configuration, such as a trimaran or SWATH. Below the macro level there can be seen to be some major style choices, such as adopting commercial design standards for a utility helicopter carrier (e.g. HMS OCEAN) or low underwater signature for an ASW frigate (e.g. the Type 23). This level can also cover generic style choices, such as being robust or highly adaptable, or having high sustainability or low manning.

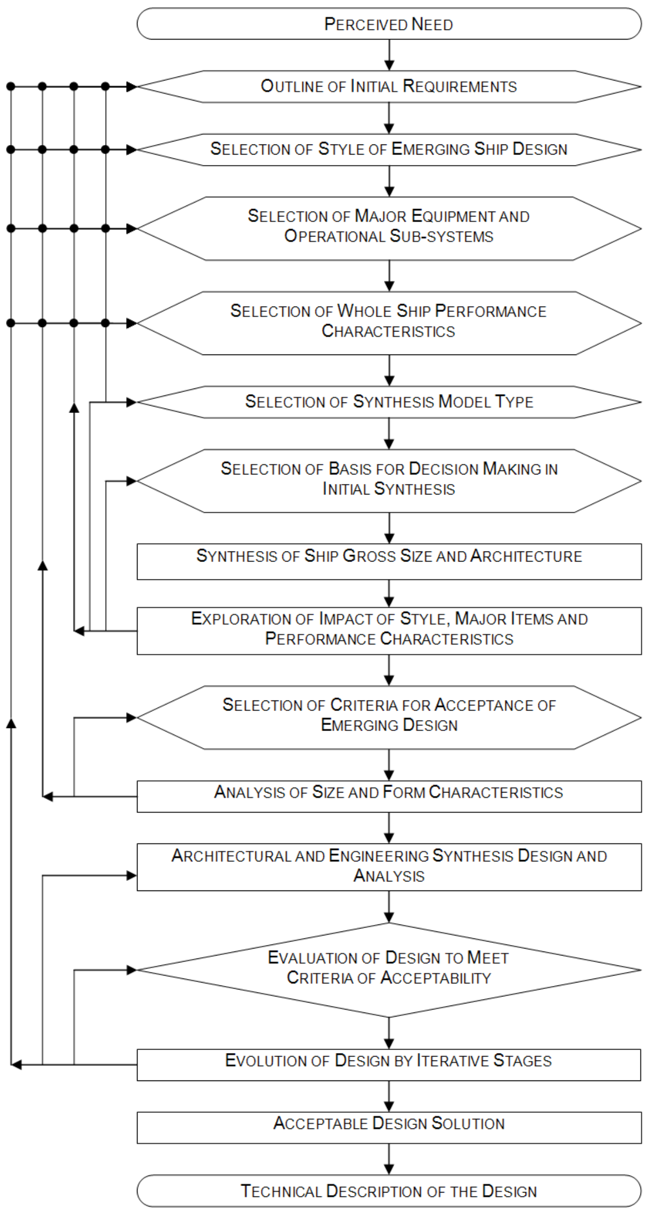


Figure. 2: A representation of the overall ship design process emphasising key decisions with Style as a critical initial choice

While adopting such style issues is inherent in commencing any design study or a specific option in a series of more exploratory studies, it is important that this is done consciously. This is good design practice since each choice has implications for the eventual design outcome and therefore ought to be investigated before that style aspect is incorporated or rejected. Beyond major style choices are a host of minor style decisions often predicated by the first two levels. These in some sense can be seen as reflecting Ferguson's (1992) observation on engineering design practice, that "Design layout and calculations require dozens of small decisions and hundreds of tiny ones". However, lack of coherence regarding style can mean at all three levels these decisions are not always made with consistency.

The term design style was originally proposed to distinguish a host of disparate issues distinct from the classical engineering sciences applied to ship design. Many of those issues could be seen to be on the "softer" end of the scientific spectrum drawing on the arts and humanities (Broadbent, 1988), whereas the first four terms under the "S^5" umbrella are, historically, the principal naval architectural (engineering sciences) sub-disciplines associated with a ship's technical behaviour. Thus Style was devised to summarise those other design concerns, which for the case of the naval ship are listed in Table 1. This very disparate range of issues, have been categorised under some six headings that (ship) designers understand. Thus, for example, concurrent engineering concerns, such as Producibility and Adaptability, are encompassed by the heading Design Issues in Table 1.

Importantly these style issues can make a substantial difference to the final outcome of a design, so their relative impact ought, in the case of a complex ship, to emerge from a proper dialogue between designer and client (or in the naval ship design case, the operational requirements owner). Furthermore, most of these issues have been difficult to take into account early in the design process because, usually, initial design exploration has been undertaken with very simple and, largely, numeric models summarising the likely eventual design definition and giving a (often dubious) feel for the cost to acquire the fabric of the ship (Andrews, 1994). That dialogue can now be informed by also having a graphical representation of the ship's configuration and internal architecture, as is reflected in the process summarised by Figure 2. This process reflects the architecturally (rather than solely numerically) driven synthesis and is propounded in the author's Design Building Block (DBB) approach (Andrews, 2003). At the critical early design stages, such a computer graphics based approach can then enable the ship designer to take account of many of the significant issues, many of which are listed in Table 1. That these are diverse and not readily or consistently quantifiable means that designers need to exercise judgement, which with stakeholder dialogues are best achieved with an architecturally driven synthesis.

Table 1: Listing of style topics relevant to a naval combatant design

Stealth	**Protection**	**Human Factors**	**Sustainability**	**Margins**	**Design Style**
Acoustic signature	Collision	Accommodation	Mission duration	Space	Robustness
Radar cross section	Fire	Access	Watches	Weight	Commercial
Infra-red	Above water weapon effect	Maintenance levels	Stores	Vertical centre of gravity	Modularity
Magnetic	Underwater weapon effect	Operation automation	Maintenance cycles	Power	Operational serviceability
Visual	NBC contamination	Ergonomics	Refit philosophy	Services	Producibility
	Shock		Upkeep by exchange	Design point (growth)	Adaptability
	Corrosion			Board Margin (future upgrades)	
	Damage control				

The categories adopted in Table 1 reveal the heterogeneous nature of the specific individual style issues, for a complex naval vessel. Thus the various items under Stealth can be seen to be the many different signatures, which a ship has and then need to be reduced to avoid detection, while the Protection items are largely aspects worth incorporating in the ship to mitigate the results of weapon effects, should the Stealth (and any "hard kill" self-defence) fail to be totally effective. However, some of the Protection items are required for normal ship practice, such as corrosion control or for non-weapon considerations, such as collision and fire-fighting. The Human Factors aspects are little less coherent (and it might be argued rather more solution oriented than those considered by the urban architectural theorist Broadbent (1988)). Broadbent addresses some 21 "human sciences" that he considers are relevant to human habitation – and hence also likely to be appropriate to HF in ships. HF concerns also relate to the important growth area of automation, which along with micro-ergonomics (e.g. console design) has a strong input to the Protection category, specifically in regard to modern bridge design. Sustainability is a major consideration in naval ship design and could be said to be a major driver, and hence a key hidden decision in the ship's style from the beginning of any ship design study. The list of Margins just makes the point that there are many features and considerations beyond simple margins on the weight/VCG estimates to ensure the ship's stability is adequate beyond the day it is accepted into service. Table 1 also distinguishes those margins required for unplanned (but consistently observed) growth in weight and rise in VCG in-service from Design Margins. The latter are more rightly a Design Issue in Table 1, given they address many measures of uncertainty in design estimates. These margins, across all the weight/space groups, are intended to be absorbed, but not exceeded, as the design and build process progresses to completion.

The last category in Table 1 is clearly the most broad and heterogeneous. Also, generally, such topics have the biggest impact on the final ship design. But this means they need to be recognised as choices and then properly considered with the owner/requirements team from the beginning of studying any design option. Some of these have been the objects of particular investigations by the author's research group at UCL. They are discussed further in Section 5 as examples of the impact on the ship design of considering separately some of these particular issues, where each could be seen as the specific driver of a design from its initiation. It is noticeable that certain of these issues can only be adequately investigated in the Concept Phase if the architectural synthesis assumed in Figure 2 is adopted. The other aspect to most of the Design Issues listed is that they have a qualitative or fuzzy nature. Thus, say, Robustness implies a greater degree of that quality than the "norm" for that type of vessel. This then raises the point that such a "norm" for a given new design option ought itself be defined but is often just accepted (or inferred) as being "current practice" or by the adoption of existing standards. There are also exceptions in the listing of the Design Issues category, like Aesthetics, which for most vessels, other than mega yachts and some cruise ships, is seen to be "a luxury". However, even this can be seen to be a simplification, as in the Cold War there was considerable debate in the US naval ship community as to whether the physical appearance of such a ship was part of its political "armament" (Roach and Meier, 1979).

The nature of the design of complex ships, such as cruise ships and naval combatants, is such that the need to emphasise the importance and difficulty of early representation of style issues is seen to be a further complication in the practice of designing such vessels. This is due to there being, additionally, a wide range in the practice of such design. This arises from the degree of design novelty adopted in a specific design option, as is indicated by Table 2. This shows a set of examples, across the field of ship design, where the sophistication in the design undertaken ranges from a simple modification of an existing ship, through ever more extensive variations in design practice, to designs adopting, firstly, radical configurations and, beyond that, radical technologies.

Table 2: Types of Ship Design in terms of Design Novelty

Type	**Example**
second (stretched) batch	RN Batch 2 Type 22 frigate and Batch 3 Type 42 destroyer
simple type ship	Most commercial vessels and many naval auxiliary vessels
evolutionary design	a family of designs, such as VT corvettes or OCL container ships
simple (numerical) synthesis	UCL student designs
architectural synthesis	UCL (DRC) design studies (see below)
radical configuration	SWATH, Trimaran
radical technology	US Navy Surface Effect Ship of 1970s

Although in first of the last two categories of Table 2, radical configuration with current technology is often explored, such options are still rarely built, due to the risk of unknowns (usually exacerbated by the lack of a real prototype). Furthermore, radical technology solutions are even more rarely pursued. In part this rarity arises because such radical technology solutions require recourse to design and, indeed, manufacturing practice much more akin to that appropriate to the aerospace industry. Thus new major aircraft projects, typically, require massive development costs (including full scale physical prototypes, some tested to destruction) and additionally need tooling and manufacturing facilities to also be specifically designed and then built, before extensive series production of each new aircraft design can commence. This is of course quite unlike most ship design, be it the ubiquitous bulker or the most sophisticated naval vessel. Such distinctions as those of Table 2 for the design of complex ships suggest any discussion of style needs, at least, to recognise the spectrum of design approach resulting from the novelty of the specific design option being pursued. Such choice on design novelty is key to the initial style choice for a given design study or a variant option in a properly conducted concept exploration (Andrews, 2013).

3. EXAMPLES OF STYLE CHOICES IN ACTUAL SHIP DESIGNS

The following are brief summaries of a series of actually built naval ship designs, where the style of an overall configuration has been a distinct choice and also some more specific style choices have been adopted, reflecting some of the more significant issues amongst those listed in Table 1. After the historic examples in the first sub-section, clearly contrasting overall style, several individual programmes are considered, with the first two examples being of commercial style for naval vessels and the next of a short life (margin-less) design intent that was not held to in practice. These can all be seen as Design Issues from Table 1 ‘though with some specific issues from other style categories, also highlighted.

3.1 1ST AND 2ND RATE ROYAL NAVY (R.N.) SHIP DESIGNS

In an early study into the nature of ship cost Brown and Andrews (1980) drew on a series of R.N. ship designs to point out that ship classes, which had been specifically designated “First rate” or “Second rate” designs, invariably showed that the latter were poor value for money (VFM). This applied to the Queen Elisabeth Class and Revenge Class Battleships, where the latter ”cheaper” versions were far less effective and clearly less value for money, in their inability to be upgraded over a thirty years life (Brown, 1999). In WWII the early convoy escorts, the Flower Class corvettes were again poor VFM compared to the later Castles and Lochs. Post War first and second class frigate classes were produced and, while the former (Type 12 Class) led to the very successful Leander Class, the latter (Type 14 Class) were soon disposed of due to their lack of adaptability beyond their design intent as convoy escorts (see Figure 3). All these comparative designs could be seen as excellent examples of the overall style choice of each of these designs, from which all the capabilities followed.

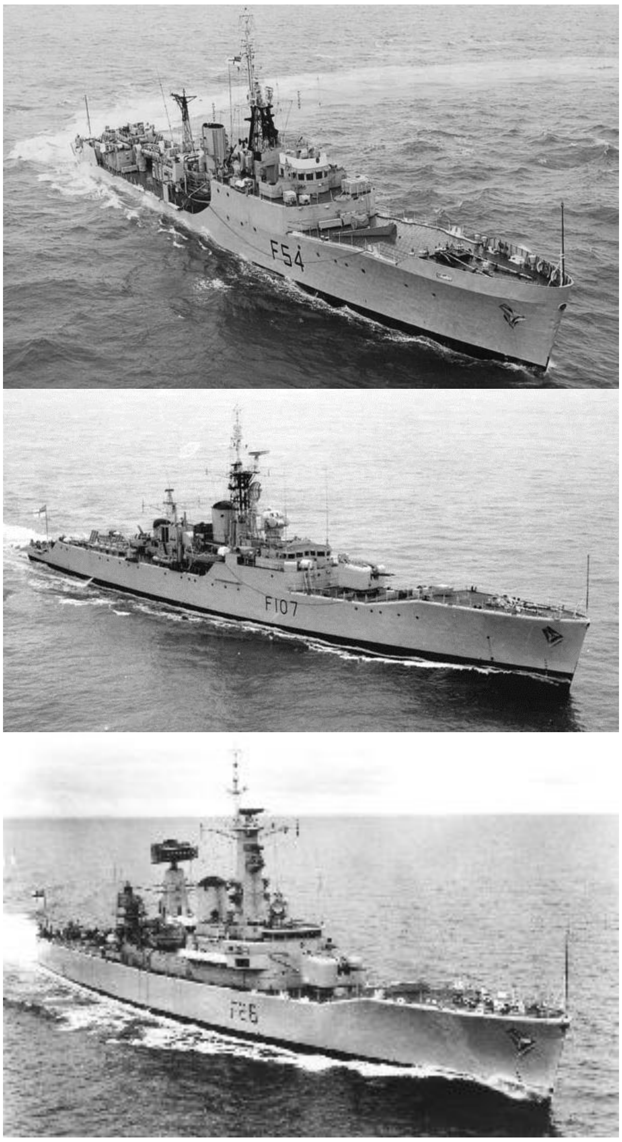


Figure 3: First and Second Rate Post –War Frigates – Type 14 vs. Type 12 and LEANDER Class

3.2 HMS OCEAN

Commercial standards were mandated for this helicopter carrier (Figure 4) without this being assessed through any proper concept and feasibility studies. This arose from the adoption by senior naval personnel, not ship designers, of a false costing based on belief that (as yet delivered) the simpler conversion of a merchant ship to a training role (see Section 3.3) could be readily extended to a major amphibious warfare vessel. The Project Manager (the author in 1986-1990) strongly argued with the naval staff over the survivability consequences of adopting such non-naval standards for an essentially high value unit (given its “cargo” of hundreds of troops plus associated equipment and 12 Commando helicopters).

After an abortive capped purchase price acquisition attempt, the PM managed to get the purchase budget raised but not sufficiently enough to cover the incorporation of quite limited naval standards. The eventual purchase was subsequently criticised by the Ministry of Defence chief marine engineer as this commercial practice substantially increased the engine support requirement for the fleet (due to this one ship's unique engine fit). Needless to say, these support costs were not shown in original cost based decision, given this was obsessed with direct initial procurement cost rather than the "true ownership cost" of the design solution. Interestingly, many have argued that HMS OCEAN has been "good value for money", as it appeared to provide a substantial amphibious lift capability. However that capability has only been exercised in "peacetime roles". The ship has not been used in operations of full naval warfare (rather than usefully in peace keeping), so the jury must be out as to whether this commercially engineered vessel constitutes a viable "cheap" solution, which is capable of discharging a major naval capability "for real". The extent to which many of the detailed "style issues", largely listed in Table 2, were predetermined by the style choice of a "commercial ship" emphasises the importance of the choice of overall design style.

Figure 4: Amphibious Helicopter Carrier HMS OCEAN

3.3 RFA ARGUS

This ship was procured by the major conversion of a Ro-Ro containership to helicopter training ship (Figure 5). Having taken on this project in its acceptance phase the author as Project Manager had subsequently to defend to the parliamentary Defence Committee (the HCDC) the payment of huge cost overruns on a Fixed Price contract (H.M.S.O., 1989). This case proved naval ship acquisition is a lot more than just engineering design and even conversion to a support role, such as a training vessel, can be demanding. The resultant procurement failure was largely due to several unwise acquisition edicts imposed on the project team, early on in the project acquisition and resulted from a style choice, which could only be described as incoherent in the rush to get to sea what looked like a "quick win". An important negative lesson on how crucial the style decision, in this case to modify a built merchant-ship to naval operational capability, can be.

Figure 5: Helicopter Training Ship RFA ARGUS

3.4 TYPE 23 FRIGATE

This frigate class evolved from the 105m "towed array tug" concept, which then grew in steps (112m, 118m) to 123m long general-purpose frigate post-Falklands War (and after the official Concept Phase). It was the first flared R.N. hull form (for radar cross section minimisation reasons – see Figure 6) and pioneered a combined diesel-electric and gas turbine (CODLAG) propulsion fit (for ultra-quiet acoustic signature to operate the towed array). Both these features were incorporated from the Concept Design studies and were the two most fundamental ship design decisions, retained from the concept studies despite the very significant growth in size, post-Concept. The style of the design was politically mandated to be short life and "margin less", when everyone in the concept team "knew", despite the Navy Minister's edict, this would not be held. Many ships in the class will be in R.N. service for at least 28 years, rather than the mandated 18 years ship life, and the through life cost of this shows the impact of an ill thought through and then not sustained key style attribute.

Figure 6: Type 23 Frigate

3.5 TYPE 31e LIGHT FRIGATE

This is a new “exportable” light frigate concept proposed in a recent report to the UK Government by the eminent industrialist and naval architect, Sir John Parker. It is seen as an approach to breaking the ever increasing cost of procuring warships and, through adopting more commercial standards and acquisition approaches for a “Second Rate” naval combatant, enabling the Royal Navy to maintain a numerically sufficient surface combatant force (Parker, 2016). The “style” to achieve this could be seen as a return to the concept of a Second Rate, outlined in Section 3.1. Whether in an era of austerity the concept will be more successful than it has been for previous Second Rates must await its design development and then its introduction into service. Given one measure of success of a naval design could be taken to be the longevity of the design in R.N. service, it will be for historians rather than current ship designers to assess whether it follows the examples of Section 3.1 or not.

4. STYLE WITH AN ARCHITECTURALLY BASED DESIGN SYNTHESIS

Many of the style issues listed in Table 1 should first be exposed in considering the architecture of the ship. This is both in regard to the overall form not just driven by the underwater hydrodynamics and hydrostatics, relevant to the first three S^5 aspects, but also by the overall configuration, which includes the internal layout disposition or architecture. As regards the overall configuration, there are style choices such as whether a multihull is selected or more typically a mono-hull, where there are still high level design choices to be made, such as in the extent and nature of the superstructure configuration (see Piperakis (2013) who considered the survivability of a large hulled/small superstructure frigate option). The manner in which exploration of ship internal configuration and layout can help open up many of the more protracted and less readily analysable aspects of ship design, largely under the style designation, has been taken further by the author. This was firstly proposed in the original exposition of the integration of configuration into early stage ship design ship design (Andrews, 1981) and has been developed right up to recent outlines of this approach, such that it has now been adopted in current text books of naval architecture (Tupper, 2013). This section addresses the approach to ship layout or the architecture of ships, for several distinct ship types or “styles”. It shows the complexity of issues encompassed by style once the architectural component is given its rightful weight in design synthesis and then into the rest of early stage ship design.

4.1 THE EXAMPLE OF FRIGATE ARCHITECTURE

The eminent naval ship designer and historian D K Brown’s paper on “The Architecture of Frigates” (Brown 1987) drew on his experience of preliminary warship design and on research undertaken by Andrews (1984) and various post graduate students at University College London (Hutchinson, 1981, King, 1985). Brown's paper was largely a comprehensive survey of many of the aspects and constraints impinging on frigate layout design through the various phases of design (termed levels by Brown), from initial design concept (Level 1) through to detailed General Arrangement (Level 3). The design constraints were indicated in his figure reproduced as Figure 7 where an outer ring shows “problem areas” directly affecting a frigate's architecture (e.g. access, noise, vibration, hydrodynamics, structural continuity, survivability, stealth, aesthetics and through life issues). These can be seen to be a mixture of Style aspects (Table 1) and the naval architecture sub-disciplines, showing the complexity of any taxonomy for such an interdependency of issues.

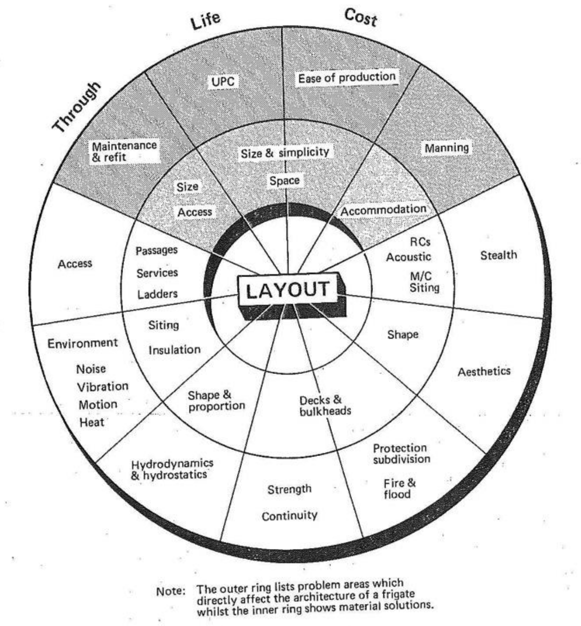


Figure 7: Design Constraints affecting Frigate Layout (Brown 1987)

The inner ring of Figure 7 shows elements of the material solution (e.g. accommodation, decks & bulkheads, shape & proportions, passages, ladders, services & machinery arrangements) that are the components of the ship’s internal architecture. In keeping with concept of ship style, Brown discussed the range of style-related issues relevant to the layout of a given design (i.e. ship role, modular/cellular features, margins, zoning). He emphasised how, for his Level 1 (for a frigate and similar combatant vessels), the key to the internal layout is the design of the upper or weather-deck disposition of weapons, helicopter arrangements, radars, communications, bridge, boats, seamanship features, machinery uptakes and down-takes, and the access over the deck and into the ship and superstructure. Figure 8 shows an updated version of Brown's frigate configuration, identifying many of the generic weapon, sensor and ship issues in arriving at a balanced ship architecture (Andrews 2003).

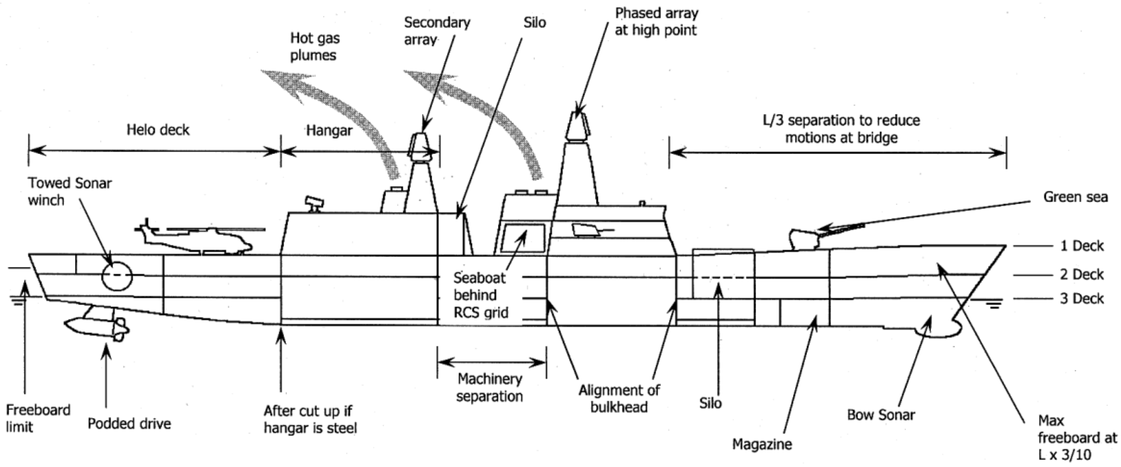


Figure 8: Frigate Layout Considerations (updated from Brown (1987) in Andrews (2003))

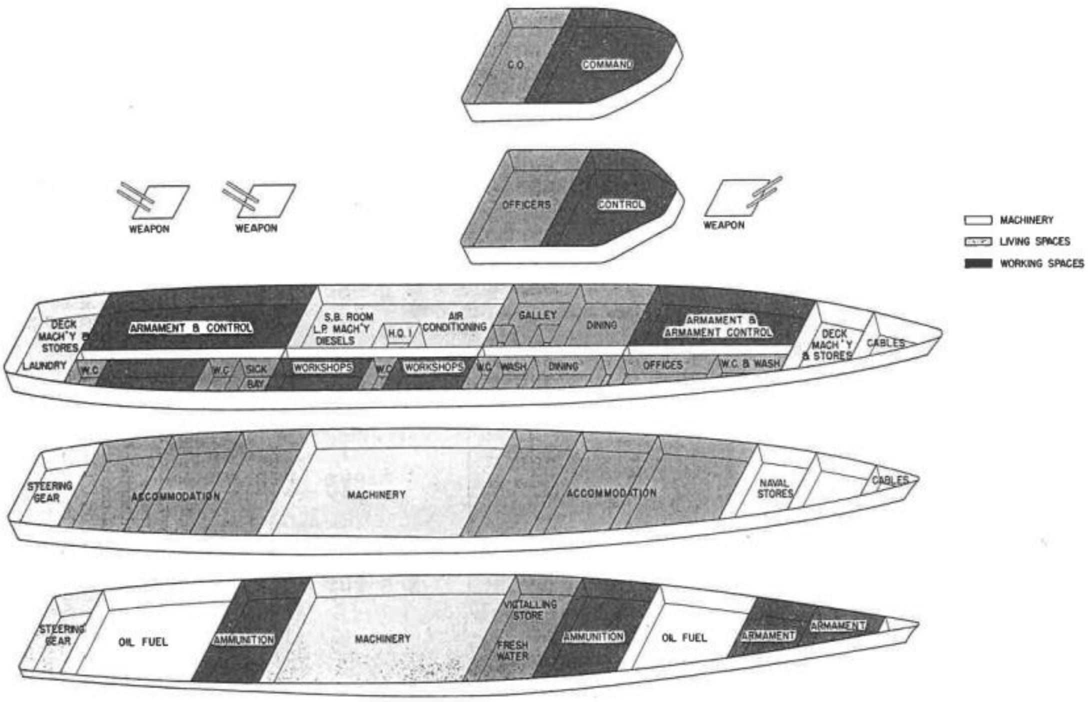


Figure 9: The "Stylised" Layout of Baker's St LAURENT Class Frigates (Baker 1955)

When one considers the internal configuration of a tightly packed multi-role vessel – the General Arrangement – it is often hard to discern a clear logic, due to the usual practice of trying to pack every compartment in the long narrow "box' resulting from high Froude Number driven hull dimensions. However a good actual example of a coherent layout is that in the first indigenously designed class of frigates for the Canadian Navy - the St LAURENT Class, produced in the 1950s and already shown in Figure 1. This was a highly innovative design with a continuous passing deck (unlike previous destroyers and frigates), which also introduced a central cafeteria messing system (Baker, 1955). The design was due the, seconded, Constructor Commodore Baker (later to be Sir Rowland Baker of Dreadnought and Polaris fame (Brown, 1983)). Baker's vision and management of the design commenced with the concept design, which was used to maintain design coherence throughout the design process, despite the high profile nature of the project (Brown 1983). One of the means that Baker used to exercise the control was in the physical architecture of the ship. Thus Baker conceived of a “stylised approach” to the functional arrangement of the ship’s layout, see Figure 9, which was robustly maintained throughout the design process. This was despite numerous attempts by “stakeholders” to impose “improvements” – something all too common in complex (and politically sensitive) major projects. That this design was highly successful was demonstrated by its repeat design (RESTIGOUCHE Class) and by the ability of the design to be modernised in service to successfully accommodate the large SeaKing ASW helicopter on board a relatively small ship.

4.2 CONFIGURATION DRIVEN SHIP DESIGN

Although the author has long postulated that the design of all warships (and most commercial service vessels) should be driven in large measure by their internal (and upper deck) configuration (Andrews 1981, 2003), it will be recognised that the concept design of certain ship types has to be approached by firstly configuring the spaces required to achieve the primary function(s) of that vessel. Thus, the physical description of a passenger, cruise or ferry ship, can only be produced by commencing with the arrangement of the public spaces and cabins (Levander, 2003). Similarly the configuration of certain large naval vessels, such as aircraft carriers and amphibious warfare vessels, are driven by the spaces required to accommodate the primary “cargo”, whether on the hangar and flight decks or the well dock and vehicles decks in those specific cases. A prime example of this aspect was presented in the RINA Transactions paper on the INVINCIBLE Class carriers (Honnor & Andrews, 1982) in a diagram reproduced at Figure 10. This shows schematically the personnel routes, equipment removal routes and stores routes around and directly below the two decks, which dominate any aircraft carrier design, i.e. the flight deck and hangar deck. That paper discussed the need for access from the main through deck, below the hangar, and around the side of the hangar, taking into account the other spatial demands for machinery inlets, outlets and removal routes, as well as features, such as boat arrangements and ship ventilation. That paper also pointed out, however, that some important military features also had to be accommodated in the arrangement but had been deliberately omitted from this figure. These included:

- Magazines and weapon movement routes;
- Other important aircraft support spaces and stores;
- The location of ship and force command, control and communications;
- Damage control features.

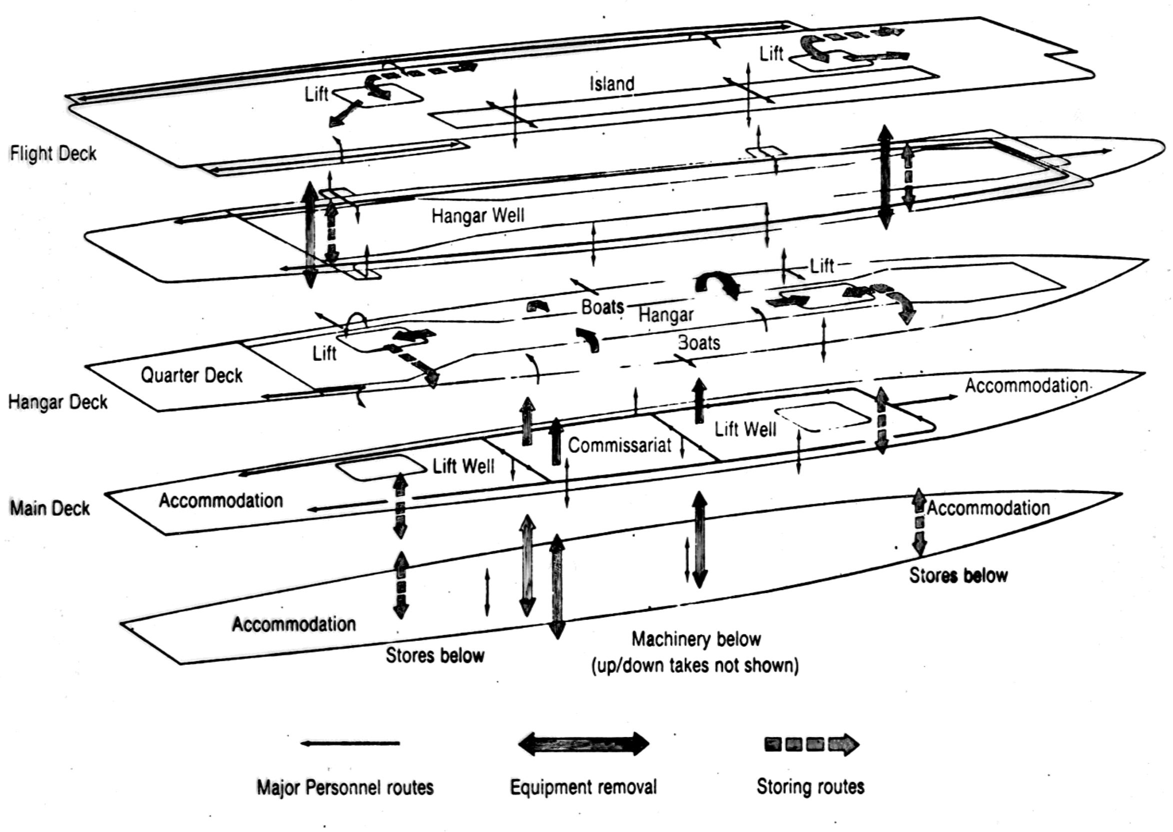


Figure 10: Schematic of INVINCIBLE Class Internal Arrangement (Honnor & Andrews 1982))

Although these features would need to be included in order that the evolution of such a complex ship configuration could be properly appreciated, this example - and the previous frigate case – are considered to demonstrate the author's contention in regard to the centrality of a ship's architecture in the early design process and, in style terms, the essentiality of a three dimensional functional integration, being key to the ship design.

4.3 DESIGN OF UNCONVENTIONAL HULL CONFIGURATIONS

A further type of ship style, which necessitates a significantly distinct ship design process is that identified by the penultimate category in Table 2. While unconventional hull types are relatively infrequently adopted as solutions for ocean going ships, they should nevertheless be included in the options considered in any comprehensive exploratory stages of a new ship design. In particular, in the case of the normally displacement-borne multi-hulled configurations - like the catamaran, SWATH and trimaran - the architectural design is highly significant. When the initial sizing of ocean-going multi-hulled vessels is considered, in determining dimensions and form parameters, it is apparent that their sizing is not circumscribed by the relatively narrow range of hull parameters, that typical apply to mono-hulls essentially driven by the Froude wave making effect. Consequently the designer, of say a SWATH or trimaran, has to size these vessels on the basis that it is the configuration of their major spaces and how they are disposed between the hulls and the broad cross deck structure, which constitutes the main driver for determining the vessel's dimensions and principal form parameters (Andrews, 2004). As can be seen from the items listed in the box in Figure 11, the size and shape of the trimaran combatant shown are driven significantly by the disposition of the major operational and habitable spaces, particularly those in the box structure.

4.4 APPLICATIONS OF THE DBB APPROACH TO OVERALL STYLE INVESTIGATIONS

The established PARAMARINE ship design system (Munoz & Forrest 2002) was made able to accept a new module, known as SURFCON, which implemented the UCL Design Building Block approach in a fully integrated manner. Thus the architectural approach to ship synthesis was fully incorporated within a practical CAD system. The manner in which the SURFCON tool is structured was described following its beta testing by Andrews and Pawling (2003). Two features incorporated in the PARAMARINE version of SURFCON, which were already part of the UCL prototype, were:-

(1) A functional breakdown of the design building blocks adopted for ship description. The categories of the building blocks (i.e. float, move, fight/operation and infrastructure) can be distinguished by their four characteristic colours in the example screen shot of the SURFCON system in Figure 12, plus purple for the main access routes. This breakdown of the Design Building Blocks was introduced to foster the exploration of more innovative configurations as part of Requirements Elucidation (Andrews, 2011), where choice of style is key;

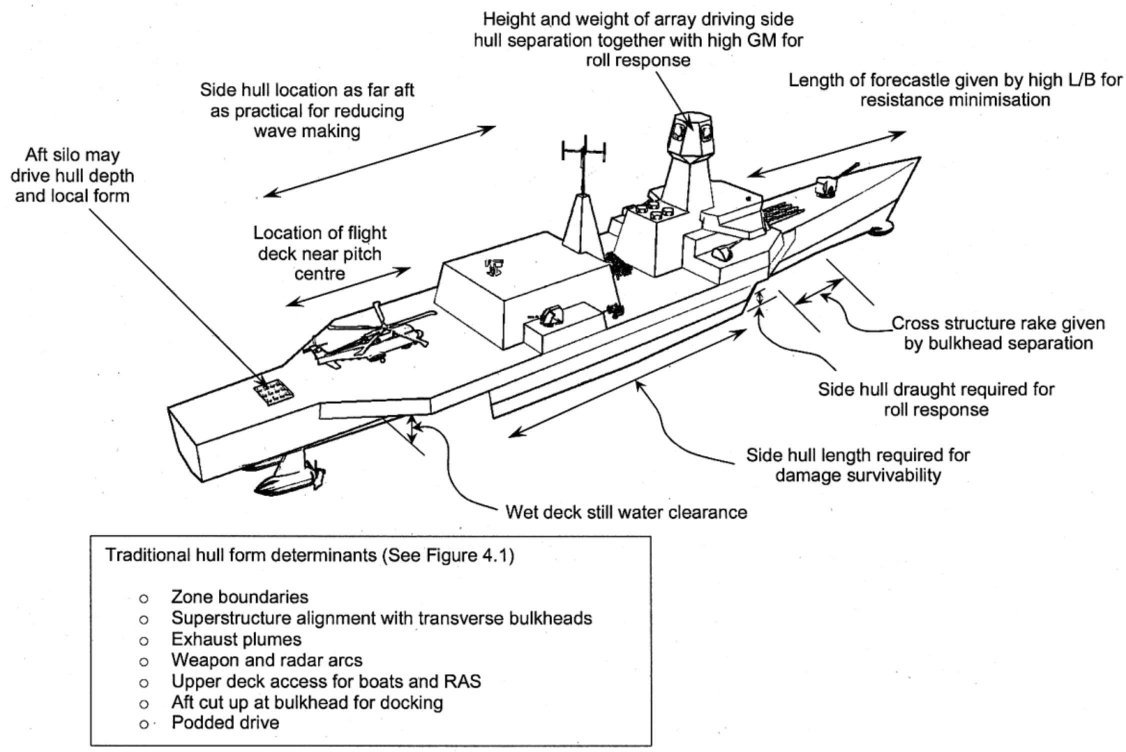


Figure 11: Trimaran Configuration Drivers for a Combatant (Andrews 2003)

(2) Use of the term Master Building Block to indicate how the overall aggregated attributes of the DBBs would be brought together so as to provide the numerical description of the resultant ship design. The advantage of using the design building block capability of SURFCON as an adjunct to the already established ship design suite of PARAMARINE, is that the audited building block attributes assembled within the Master Building Block could be used directly by the Paramarine analytical modules, thereby enabling the necessary naval architectural calculations to be performed to ascertain the balance, or otherwise, of the configuration being derived by the designer.

The DBB approach is intended to foster innovative design solutions and so is neither "hard wired" nor employing predetermined routines to achieve naval architecturally balanced ship solutions. It does need to be used by a capable ship designer, who can then exploit the capabilities of the system to produce coherent and balanced ship design studies. The system, in auditing a new configuration of building blocks, will report to the designer the state of the design. Rather than automatically changing the dimensions and or the hull parameters, which might be the case with a "black box" system, Paramarine-SURFCON will tell the designer where a design study is no longer balanced. Thus the designer can make the appropriate decision on how to proceed with the design at that point in time, drawing on the evidence afforded by the system as to the imperatives for the study at that juncture, including due regard to the chosen style for that design option. In that respect it can be seen to be responsive the subtlety of style based decision-making.

The general procedure adopted in undertaking a new ship design study using the Paramarine/SURFCON system can be summarised as follows:

(1) A very broad intent and tentative outline requirement is identified and a design style proposed;
(2) A series of design building blocks are defined or selected (from a library or newly created), containing geometric and tentative ship size and a set of hull dimensions postulated;
(3) The design building blocks are located as required within a prospective or speculative configurational space and tentative hull form(s) taking into account the preferred or proposed style;
(4) Overall weight and space balance and performance (e.g. stability, powering) of the design are assessed, using the PARAMARINE naval architectural analysis routines and any specific style analyses;
(5) The configuration is then manipulated until the designer is satisfied with both the configuration, reflecting the perceived style issues, and the naval architectural balance;
(6) Decomposition of the design building blocks to ever greater levels of detail is undertaken as required, and ship balance / performance maintained at the appropriate level, often exploring further critical style aspects.

Table 3 shows the manner in which this overall procedure is evolved in stages. From this the GA can be drawn as part of the final design stage for a concept design, as Figure 12 indicates for a (trimaran) naval combatant. (See Andrews and Pawling (2008) for the details behind each step for that design study.)

Table 3: The Stages in DBB approach to Ship Synthesis (with the number of blocks indicated for each step for the design shown in Figure 12)

Design Preparation
Selection of Design Style
Topside and Major Feature Design Phase (18 to 47)
Design Space Creation
Weapons and Sensor Placement
Engine and Machinery Compartment Placement
Aircraft Systems Sizing and Placement
Superstructure Sizing and Placement
Super Building Block Based Design Phase (47 to 110)
Composition of Functional Super Building Blocks
Selection of Design Algorithms
Assessment of Margin Requirements
Placement of Super Building Blocks
Design Balance & Audit
Initial Performance Analysis for Master B.B.
Building Block Based Design Phase (110 to 343)
Decomposition of Super Building Blocks by function
Selection of Design Algorithms
Assessment of Margins and Access Policy
Placement of Building Blocks
Design Balance & Audit
Further Performance Analysis for Master B.B.
General Arrangement Phase
Drawing Preparation

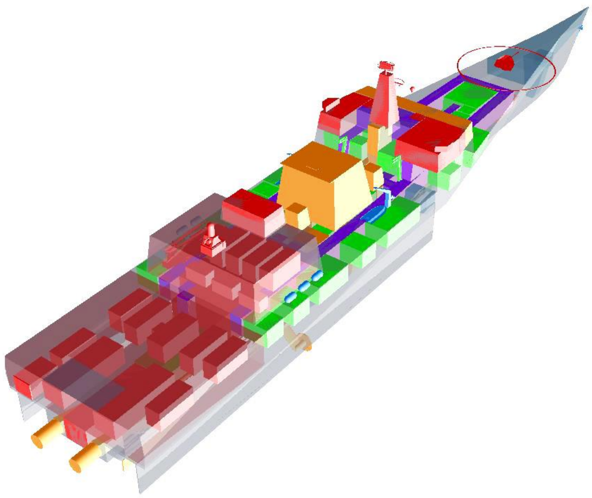


Figure 12: SURFCON Model of UCL DBB Study of the US Navy Littoral Combatant Ship (Andrews & Pawling 2008). (See on-line version for full colour scheme referred to in text at Section 4.4 (1)).

Each design building block, as the fundamental component of the SURFCON approach, can be regarded as an object in the design space and as a "placeholder" or "folder" containing all the information relating to a particular function within the functional hierarchy. The ways in which the design can be manipulated on the screen are described by Andrews and Pawling (2003). Importantly the "block definition" object permits the designer to add whole ship margins and characteristics, such as accommodation demands, once the “block summary” object has summarised all the information in the top level block in the building block hierarchy. (In effect this is the Master Building Block object.) The "design audit" object then allows the design description to be audited for any of the characteristics selected for monitoring, which typically will include style aspects alongside prime naval architectural capabilities. Results can be displayed using the functional group hierarchy; this "design audit" object is assessed for a range of design infringements, by other objects in the design space, and for the balance of the overall ship design from the whole ship characteristics listed in the Master Building Block.

A further advantage of the architecturally driven approach is that it enables the concept designer to look at quite different physical configurations for the same set of broad operational requirements. An excellent example of this was the investigation into the operational concept of small littoral craft being transported oceanographically by a fast mothership (Andrews and Pawling, 2004). Figure 13 shows the SURFCON representations of five configurations (e.g. dock ship, heavy lift ship, crane ship, stern gantry ship and open stern ship) and two variants. This shows well the issue of overall style choice in the five distinct configurational options produced. To the extent that they all aim to reach the same operational performance the style issue becomes one of whole ship configuration exploration.

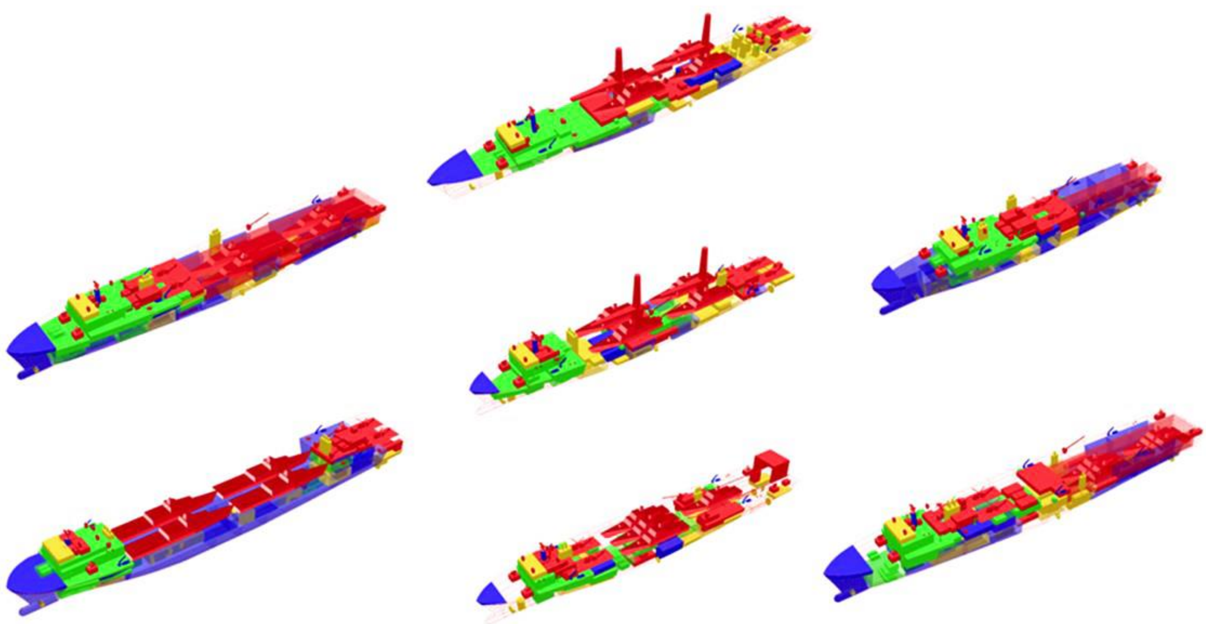


Figure 13: SURFCON Models of UCL DBB Studies for Littoral Mothership Investigation (Andrews & Pawling 2004)

5. EXAMPLES OF UCL CONCEPT STUDIES EXPLORING ASPECTS OF STYLE

This section outlines results of investigations into a selected set of the topics in Table 1, which have been the subject of discrete ship design studies by the author's research team at UCL over the last two decades. The items outlined are not a comprehensive analysis of the Table 1 topics, but are intended to show how some of these various design issues can be seen as key design choices over which the ship designer ought to be more informed. From such research investigations the designer could make the key first decision step in the ship design process, summarised in Figure 2, more overtly and in a better informed manner.

5.1 MARGINS

An exploration into the validity of the Design Spiral, as a representation of the nature of ship design especially in ESSD, considered Growth and Board Margins for a naval combatant study (Andrews et al, 2012). Two distinct design styles were investigated: a conventional frigate style and a large hulled/small superstructure variant. The former showed a linear behaviour in solution size with increasing Board Margin (i.e. the allowance for future capability updates), while the latter showed a discernible step change in size for the same variation in Board Margin (see Figure 14). Thus the choice of overall style was shown to be key to any such investigations. This was only revealed by the use of the DBB approach, as a set of simple numerically ship syntheses, which had been undertaken several decades previously (Andrews, 1984), and also varied Board margin failed to show.

5.2 COMMERCIAL STRUCTURE VERSUS SHOCK ROBUSTNESS

In considering the extent to which commercial standards might be introduced into naval ship design, in the search for reductions in initial (procurement) cost, this UK EPSRC CASE funded project (Bradbeer and Andrews, 2010) first considered the effect of missile attack on a small frigate built to commercial standards. It was found that survivability was affected by the density of outfitting, where the latter could be considered as due to a style decision, akin to building in robustness or adaptability.

A second investigation (Bradbeer and Andrews, 2012) looked at the effect of underwater shock and varied the structural style from normal naval scantlings to commercial practice (see Figure 15). This meant changing from closely spaced "Naval Tee bar" stiffeners, adopted to reduce the structural weight fraction, to typical commercial larger bulb and flat bar stiffeners more widely spaced. The survivability to very high (hull lethality) shock levels due to adopting such differing scantling styles, yet with the same longitudinal bending strength, was found to be considerably less for the heavier but less structurally effective commercial style. This is an example of more detailed analysis than would not normally be undertaken in ESSD, but reveals that a style decision taken early in design can make a major difference in a key ship's capability, which has been seen to be fundamental in a naval combatant.

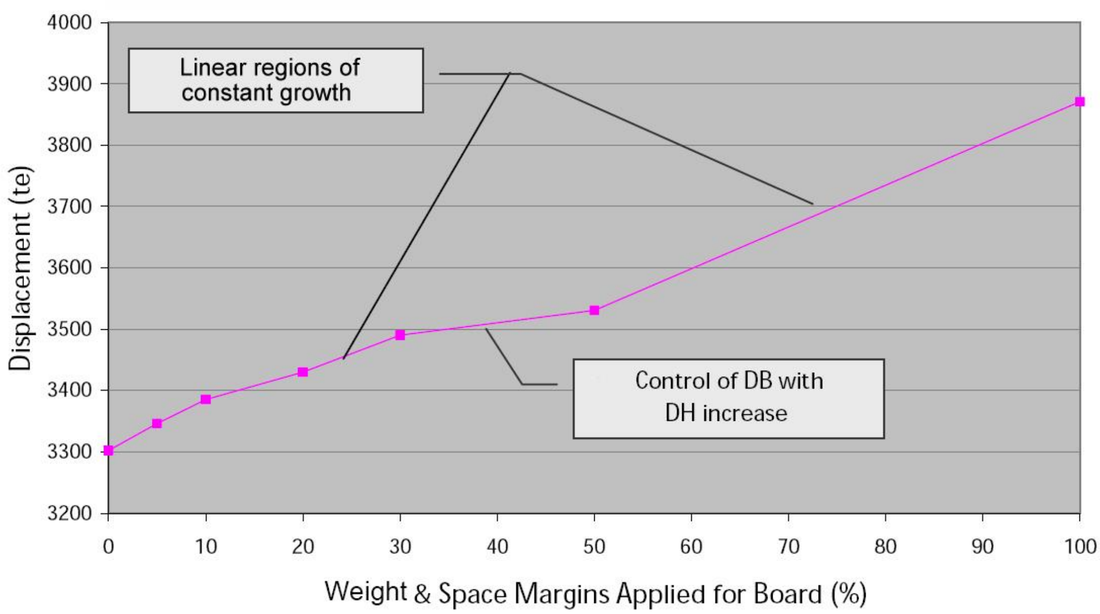
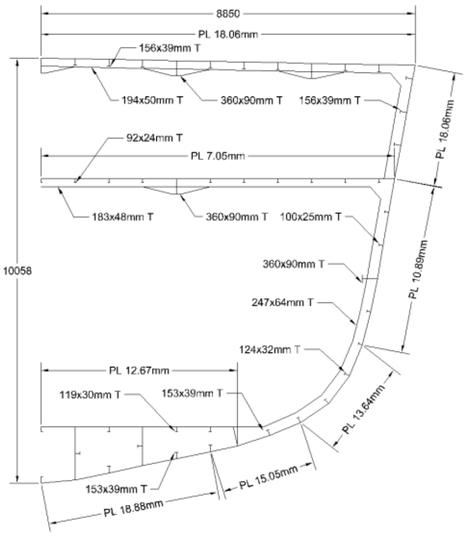
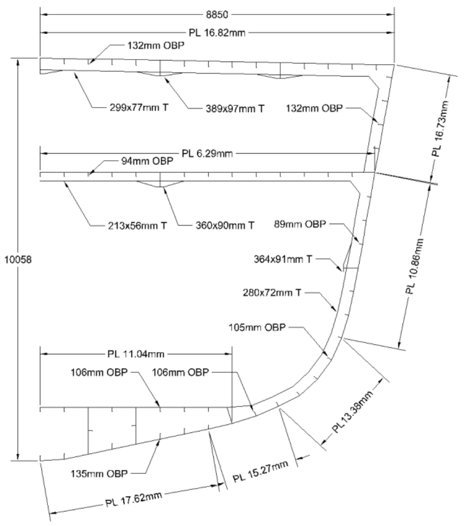


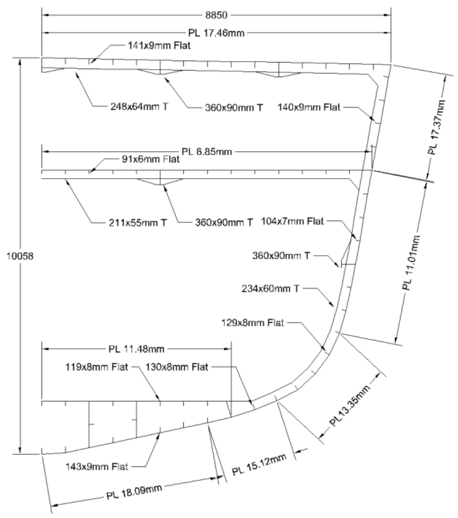
Figure 14: Example of Non-Linearity in Margin growth as Style Choice (Andrews et al, 2012)



a. Naval Tee bars



b. Commercial Offset Bulb Plate sections



c. Commercial Flat bar sections

Figure 15: Comparison of Naval and Commercial Structural Styles (Bradbeer & Andrews, 2012)

5.3 DESIGN FOR PRODUCTION – DESIGN STYLE DOMINATED

In a research project funded by the UK Shipbuilders and Ship-repairers' Association (Andrews et al, 2005) a study was undertaken on both a commercial vessel and a naval ship in concept design to improve the architecture of ships with the objective of reducing the cost of outfitting. This was a novel study in that much of large commercial ship cost is in steelwork, whereas for complex ships, such as the Corvette and the Offshore Support Vessel (OSV) in this study (see Figure 16), much of the cost is in outfitting, and hence amenable to architectural exploration. The styles of both ship types were investigated, with rearranged machinery location and more spacious passageways to fleet in modular cabins, respectively, which could be seen as specific style choices.

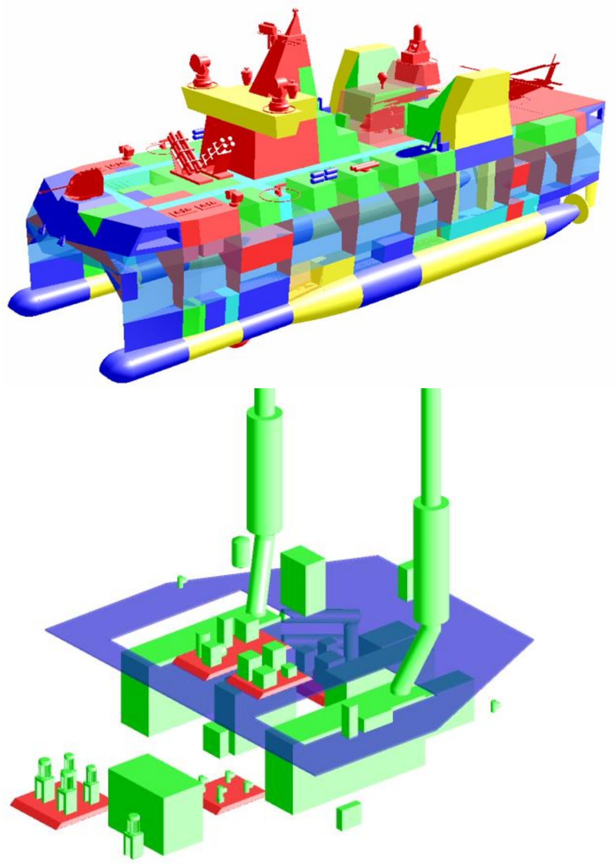


Figure 16: Examples of Style emphasising Producibility in ship configuration (Andrews et al, 2005)

5.4 OFFSHORE PATROL VESSEL (OPV) CONFIGURATION STYLE

This ESSD investigation (Pawling and Andrews, 2010) was undertaken to show that in addition to the conventional OPV, based on the style of small naval combatants, it was worth exploring more radical ship configurations, such as an OSV commercial design, a trimaran OPV and a very wide stern (Ramform like) mono-hull (see Figure 17). These alternative configurations could be seen to be addressing style issues, such as commercial design style and different hull forms, in exploring solutions which might be found to be more appropriate for stowing and deploying sizeable autonomous vehicles, especially in regard to the launch and recovery from the vessel's stern.

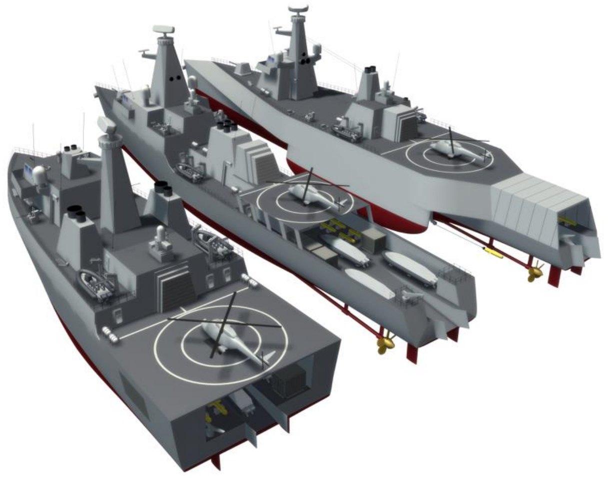


Figure 17: Examples of different Ship Configuration Styles for an OPV Study (Pawling and Andrews, 2010)

5.5 SURVIVABILITY

There are many style choices related to survivability in the design of naval vessels. In fact it could be argued that beyond the carriage of combat related systems, it is a heightened emphasis in the ship design on Vulnerability reduction measures that truly distinguishes the naval ship from its commercial cousins. As part of that style choice, features addressing most of the issues listed in Column 2 of Table 1 need to be selected, or at least allowed for, very early in the concept design of any such options. Thus features such as the water tight sub-division, adoption of a zoning policy or even the configuration of the main passageways, can be seen as significant style choices. Figure 18 shows a comparison of main passageway arrangements explored in a personnel movement exploration for a Type 22 Frigate, which is characterised by having two passing decks in the main hull over the machinery spaces. This investigation compared personnel movement evolutions for the original two central passageways with a proposed double passageway re-design (Andrews et al, 2008).

5.6 MOTHERSHIP CONFIGURATION / OPERATIONAL STYLE

A novel solution to the fast Littoral Combatant concept was seen to be the transporting of several small craft on a large fast vessel. This study (Andrews and Pawling, 2004) with five distinct ship configurations with different launch and recovery methods has already been shown in Section 4.4. The alternative styles were proposed as an exploration of the operational options through alternative ship design configurations or styles.

Figure 18: Examples of Single Central and Double Passageways for Type 22 Frigate (above) and variant (below) (Andrews et al, 2008)

6. RECENT DEVELOPMENTS OF RESEARCH INTO SHIP STYLE

A further feature of style is that, if it can influence multiple areas of design, then it must itself represent the "grouping" of multiple sources of information in some way. Developing an ontology and taxonomy for style is seen to offer potential advantages to the practice of ship concept design, as it could allow for more efficient storage, retrieval and application of potentially disparate pieces of information or decisions (Pawling et al, 2013). It has been proposed that this could be combined with the semi-automatic layout generation methods, such as that developed by the Technical University of Delft (van Oers, 2011), to allow a broader exploration of the impact of stylistic decisions in ESSD than the point based architecturally design approach using DBBs with a CAD tool like Paramarine.

From the some of the above examples of ship architecture, it is considered that for some of these designs there were significant choices made which could be seen to be highly stylistic. Furthermore, such decisions as the number of masts on a frigate, or how the various functions might be disposed around an enclosed hangar on an aircraft carrier (see Figure 10), are highly cross-cutting in impact. This is because such style choices can have both direct and indirect implications on a wide range of overall and detailed design features.

There are seen to be two aspects where a more focused consideration of style might improve ESSD (Pawling et al 2013). Firstly, the development of semi-automatic methods of generating sufficiently detailed designs but based on an initially simplified layout, rather than ab initio and genetic algorithm based such as the TU Delft approach above. The former would allow the designer to better focus on the overall style of the arrangement. Secondly, a style focused taxonomy could be used as a method for describing and storing the data and rules that permit such semi-automatic tools to be used in developing more detailed layouts, such as that in Figures 12 and 16. The designer could then apply a wide range of changes to a design option by selecting different style aspects (e.g. different survivability levels or the extent of through life adaptability – see Table 1). These could then be compared to give insights into major design and cost drivers in ESSD.

6.1 A PROPOSED INTEGRATED APPROACH TO BETTER STYLE DRIVEN DESIGN

An approach, to better consideration of style choices, was made by the UCL design research team, in conjunction with its research partners the University of Michigan and TU Delft (Pawling et al, 2013). Current early stage design techniques for initial general arrangement definition focus predominantly on spatial compartment allocation. When evolving general arrangements, there is a need to be able to introduce detailed style decisions once the selected overall ship style has been made (see Figure 2). Given that style at the level below that overall configurational choice can be defined as the combination of whole ship performance metrics and local system metrics (see Table 1), information can be drawn from different domains, much of which may be ill-defined knowledge (such as many human factors aspects). Style is representative of design intent and the designer's engineering judgment in the early stages of ship design constraint definition, layout generation, and in the evaluation of the evolving layout. With the ability to account for style definition, the designer might then be able to create concept designs that integrate a larger body of design intent, without the need to explicitly describe its characteristics.

In an effort to better incorporate style into the early stages of concept design, an iterative method using three primary levels in the design process has been proposed. Figure 19 shows those levels as the style elucidation and input definition level, design layout generation method level, and post-generation style analysis level, respectively. Multiple components of coupled analysis would allow the cross cutting of knowledge to capture style attributes over multiple domains of the design within each level of the suggested process.

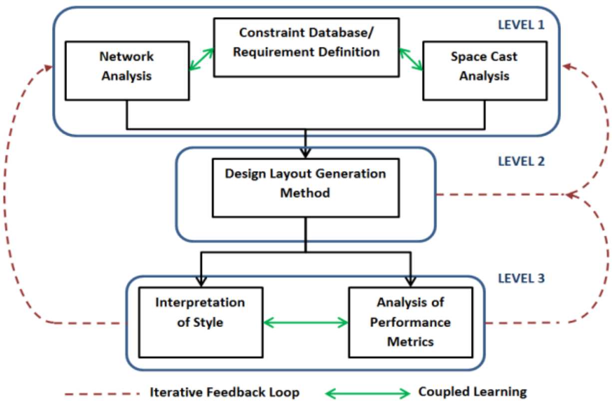


Figure 19: Proposed integrated approach for style definition in early stage design (Pawling et al, 2013)

The first level of Figure 19 highlights the style choices and the production of the inputs for the design generation method. Capturing the style is done through the explicit and implicit definition of the parameters that will drive the analysis of performance metrics, and their stylistic or architectural features. Definition of these constraints and requirements has not been seen to be a trivial process as they evolve throughout the early stages of ship design and then right through to detailed design. One increasingly popular ways to define constraints and requirements is through the use of network analysis. A network, in its broadest sense, is a collection of points joined by lines which can then be represented in a matrix form and therefore able to be analysed. Pawling et al. (2016) provide a summary of networks, including some applications to ship design, and Newman (2010) gives the underlying theory to network analysis. At the first level, the relationships between spatial, geometric, and global location preferences are iteratively updated with each completed loop of the integrated approach. The relationships could be investigated abstractly through a network analysis alongside geometric allocation. Insights gained during this definition level could then be used to guide the elucidation process towards the novel definition of style intent in ESSD. Currently networks are being applied to both vessel configurations and distributed ship service systems (Gillespie, 2012, Collins et al, 2015).

Style at this level could identify any ill-defined knowledge early in ship design and capture metrics as inputs for disparate topics, many of which are hard to quantify. The definition of style could be carried through to Level 2 and Level 3 of Figure 19, coupling ship performance metrics to the generated architectural layout from which appropriate designs can be down selected. With proper definition of inputs and a clearly selected ship design style, the parameters of the constraints and requirements would give the potential to produce designs with higher integrity for the subsequent phases of ship design beyond ESSD.

7. CONCLUSION

The paper has focused on an important part of design decision-making, that of style choice. This has been addressed through discussing actual ship designs in history and specific ship research investigations on discrete ship style related issues, recently undertaken at UCL. A way forward proposed in an earlier joint paper with collaborators has been seen as a means to further emphasise this paper's assertion that choice of overall design style is probably the key design decision. A clearer decision choice on style should be made (hopefully) explicitly at the earliest step in starting any design option to ensure better design exploration and, hence, a better focused downstream process.

8. ACKNOWLEDGEMENT

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MODEL PREDICTIVE CONTROL OF AN AUV USING DE-COUPLED APPROACH

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SUMMARY

This paper considers the decoupled dynamics and control of an Autonomous Underwater Vehicle (AUV). The decoupled model consists of speed, steering and depth subsystems. Generally AUV model is unstable and nonlinear. The central theme of this paper is the development of model predictive control (MPC) for underwater robotic vehicle for ocean survey applications. The proposed MPC for decoupled structure can have simple implementation. Simulation results have been presented which confirm satisfactory performance. Decoupled approach is well suitable for applying control.

NOMENCLATURE

m	Mass of the vehicle (kg)
$X_{\dot{u}}$	Hydrodynamic added mass (kg)
$Y_{\dot{u}}$	Fluid inertia in the lateral y direction due to time rate of change of sway velocity (kg)
I_z	Moment of inertia along z-axis (kgm^2)
$N_{\dot{r}}$	Fluid inertia moment about vertical body axis due to time rate of change of yaw (kgm^2)
υ	Linear motion in y direction (m/sec)
ψ	Yaw angle of spheroidal underwater vehicle (deg)
r	Yaw angular velocity of spheroidal underwater vehicle (deg/sec)
q	Angular velocity about y-axis, rad/sec
θ	Angular position about y-axis, rad
z	Linear position along z-axis, m

1. INTRODUCTION

Underwater vehicles help human to understand ocean in a new ways. Important advances in underwater robots are improved efficiency, low cost and reduced risk in marine operations. Underwater vehicles play an important role in scientific, industry and military operations (Eski & Yildirim, 2014); (Bong, 2015). Underwater Vehicles are categorized into several groups based on their performance characteristics. According to the method of control, underwater vehicles are classified into two categories namely manned and unmanned (Chen-W, Jen-Shiang & Jing-Fa, 2013) (Kim & Yuh, 2011). Unmanned vehicles are further classified into Autonomous Underwater Vehicles (AUV's) and Remotely Underwater Vehicles (ROV's).

Modeling of an AUV is a difficult process. It consists of hydrodynamic, hydrostatic, electrical and mechanical parameters. In addition with these difficulties, there are some uncertainties, parameter variations due to ocean currents, waves and environment (Sorenson, 2005), (Fossen, 1994). So it is very difficult to analyse the overall model of an AUV. AUV is decoupled into three subsystems namely speed, steering and diving control (Antonelli *et al*, 2003) (Herman, 2009). The design of an AUV for the motion control must consider motion stabilization and maneuvering. So controller must be robust to withstand model uncertainties and parameter variations (Filaretoy & Yukhimets, 2012), (Kim & Yuh, 2011). Model Predictive Control (MPC) has been applied for coupled model of an AUV (Budiyono, 2011). The application of MPC on decoupled model of AUV has not been attempted earlier.

Few control techniques have been applied on diving and steering control of an AUV. Sliding mode control has been applied on diving plane of AUV (Huizhen & Fumin, 2012), (Healey & Lienard, 1993). H infinity control was also applied on diving plane and steering plane (Santhakumar & Asokan, 2012). Fuzzy Logic Control and neural network control techniques have been applied recently on decoupled AUV system (Eski & Yildirim, 2014), (Lynch & Ellery, 2014).

This paper has five sections. Section 1 deals with the introduction about decoupled system and modeling. Section 2 discusses about Decoupled model of an AUV. Section 3 is about proposed controller i.e., Model Predictive Control. Section 4 deals about results and Conclusions are highlighted in Section 5.

2. DECOUPLED MODEL OF AN AUV

The AUV model is decoupled into three sub systems: Speed, Steering and Depth, as shown in figure 1. Model Predictive Control has been applied on each subsystem of AUV. Reference (Ref.) inputs and outputs (o/p) of an AUV are speed, steering and depth. Table-I presents the states and inputs of AUV subsystem.

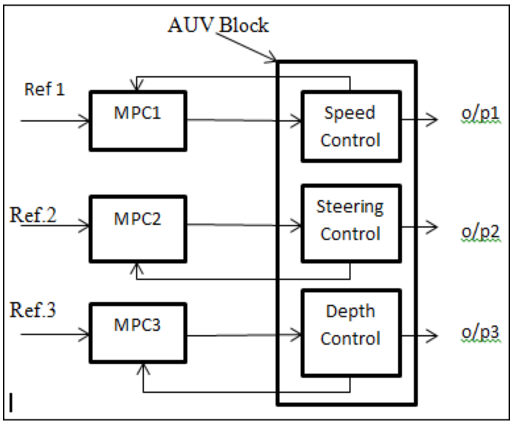

Figure.1 Implementation of MPC on De-coupled system of AUV

Table I AUV subsystem states and inputs

Sub systems	states	Inputs
Speed	$x(t) = u(t)$	$n(t)$
Steering system	$x_1(t)=\upsilon(t)$; $x_2(t)=r(t)$; $x_3(t)=\psi(t)$	$\delta_r(t)$
Diving system	$x_1(t)=w(t)$; $x_2(t)=q(t)$;$x_3(t)=\theta(t)$;$x_4(t)=z(t)$	$\delta_s(t)$

Vehicle model (Eski & Yildirim, 2014) is taken as a case study for the analysis of decoupled control system in this work.

2.1 SPEED CONTROL SYSTEM

This subsystem considers only surge equation of motion of AUV. Assuming that the interactions with sway, heave, roll, pitch, yaw motions are neglected. The AUV has homogeneous mass distribution and xz-plane symmetry so that $I_{xy}=I_{yz}=0$. The surge equation of motion can be written as (Eski & Yildirim, 2014), (Fossen, 1994).

$$(m-X_{\dot{u}})\dot{u}(t) = X_u u(t)-X_{ul}u(t) + (1-tp)\ T$$

or

$$M\dot{u}(t) + R\ u(t) = F \tag{1}$$

where $u(t) = \frac{dy(t)}{dt}$ is speed and $\dot{u}(t)$ is acceleration,

$$F = (1-t_p)\ T$$

The above equation can be written as

$$M\frac{d^2y(t)}{dt^2} + R\frac{dy(t)}{dt} = F \tag{2}$$

Equation (2) can be written in state model as

$$\begin{bmatrix}\dot{x}_1(t)\\ \dot{x}_2(t)\end{bmatrix} = \begin{bmatrix}0 & 1\\ 0 & a_1\end{bmatrix}\begin{bmatrix}x_1(t)\\ x_2(t)\end{bmatrix} + \begin{bmatrix}0\\ b_1\end{bmatrix}F \tag{3}$$

$$y(t)= [0\ \ 1]\begin{bmatrix}x_1(t)\\ x_2(t)\end{bmatrix}$$

Equation (3) is in the form of $\dot{x}$ =Ax+Bu

$$a_1= -\frac{R}{M};\ \ b_1=\frac{1}{M};\ x_1(t) = y(t);\ x_2(t) = \frac{dy(t)}{dt}$$

$$M= (m-X_{\dot{u}}),\quad R= X_{ul}- Xu$$

where 'm' is the vehicle mass. $X_{\dot{u}}$ is the added mass. The coefficient X_u and X_{ul} are the linear and quadratic damping term in surge. t_p is the thrust deduction coefficient and T is the thrust (Bong, 2015), (Bessa, Dutra & Kreuzer, 2010).

2.2 STEERING CONTROL

Considering AUV moves in the horizontal plane, yaw moment on the vehicle is caused due to the change in rudder angle and results in changing the heading direction of the vehicle. The three states related to steering control are sway $\upsilon(t)$, yaw $r(t)$ and yaw angle $\psi(t)$. The control input is the deflection of the rudder angle $\delta_r(t)$. The fluidic forces are linearized and the equations of the steering subsystem are

$$\begin{bmatrix}m-Y_{\dot{\upsilon}} & mx_G-Y_{\dot{r}} & 1\\ mx_G-N_{\dot{\upsilon}} & I_z-N_{\dot{r}} & 0\\ 0 & 0 & 1\end{bmatrix}\begin{bmatrix}\dot{\upsilon}\\ \dot{r}\\ \dot{\psi}\end{bmatrix}-\begin{bmatrix}Y_\upsilon u_0 & Y_r u_0 & 0\\ N_\upsilon u_0 & N_r u_0 & 0\\ 0 & 1 & 0\end{bmatrix}\begin{bmatrix}\upsilon\\ r\\ \psi\end{bmatrix}=\begin{bmatrix}Y_{dr}u_0^2\\ N_{dr}u_0^2\\ 0\end{bmatrix}\delta_r(t) \tag{4}$$

In vehicle dynamics, assuming cross coupling terms in the mass matrix, x_G and y_G are zero. This assumption has been considered based on vehicle symmetry and rudders are very close to equidistance from body centre. So the above vehicle dynamics can be written as

$$\begin{bmatrix}m-Y_{\dot{\upsilon}} & 0 & 0\\ 0 & I_z-N_{\dot{r}} & 0\\ 0 & 0 & 1\end{bmatrix}\begin{bmatrix}\dot{\upsilon}\\ \dot{r}\\ \dot{\psi}\end{bmatrix}-\begin{bmatrix}Y_\upsilon u_0 & Y_r u_0 & 0\\ N_\upsilon u_0 & N_r u_0 & 0\\ 0 & 1 & 0\end{bmatrix}\begin{bmatrix}\upsilon\\ r\\ \psi\end{bmatrix}=\begin{bmatrix}Y_{dr}u_0^2\\ N_{dr}u_0^2\\ 0\end{bmatrix}\delta_r(t) \tag{5}$$

Compactly,

$$M\dot{x}-C_d x=Du \quad \text{or} \quad \dot{x} = (M^{-1}C_d)\ x+(M^{-1}D)u \tag{6}$$

which is in the form of $\dot{x}$ =Ax+Bu, and taking the parameter values from (Eski & Yildirim, 2014) the matrices become

$$A=\begin{bmatrix}-1.0094 & -0.6794 & 0\\ -0.5366 & -0.8274 & 0\\ 0 & 1 & 0\end{bmatrix};$$

$$B=\begin{bmatrix}0.2191\\ -1.1868\\ 0\end{bmatrix};$$

2.3 DEPTH CONTROL

Assuming AUV moves in the longitudinal plane, the pitch and depth can be controlled by changing the stern planes (two horizontal fins) deflection. The depth control will depends on pitch control. When the AUV moves at constant speed, the pitch angle change will results rising or diving of the vehicle and finally changes the depth of the AUV (Watson & Green, 2014).

Four states related to depth control of an AUV are heave velocity w(t), pitch rate q(t), pitch angle $\theta(t)$ and depth z(t).

The equations associated with dive plane are (Eski & Yildirim, 2014)

$$\begin{bmatrix} m - X_{\dot{u}} & -(mx_g + Z_{\dot{q}}) & 0 & 0 \\ -(mx_G + M_{\dot{w}}) & I_{yy} - M_{\dot{q}} & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} \dot{w} \\ \dot{q} \\ \dot{z} \\ \dot{\theta} \end{bmatrix} - \begin{bmatrix} Z_w & mU + Z_q & 0 & 0 \\ M_w & -mx_G U + M_q & 0 & M_\theta \\ 1 & 0 & 0 & -U \\ 0 & 1 & 0 & 0 \end{bmatrix} \begin{bmatrix} w \\ q \\ z \\ \theta \end{bmatrix} = \begin{bmatrix} Z_{\delta_s} \\ M_{\delta_s} \\ 0 \\ 0 \end{bmatrix} \quad (7)$$

State vectors are defined as x= $[w\ q\ z\ \theta]^T$ and the input vector is $u=[\delta_s]^T$. Matrix equation takes the form

$$M\dot{x} - C_d x = Du$$

or

$$\dot{x} = (M^{-1}C_d)x + (M^{-1}D)u \quad (8)$$

which is in the form of $\dot{x}$ =Ax+Bu, considering the numerical values from (Eski & Yildirim, 2014)

$$A = \begin{bmatrix} -2.38 & 1.57 & 0 & 0 \\ 4.23 & -1.19 & 0 & -0.70 \\ 1 & 0 & 0 & -1.54 \\ 0 & 1 & 0 & 0 \end{bmatrix}; \quad B = \begin{bmatrix} -1.37 \\ -3.83 \\ 0 \\ 0 \end{bmatrix}$$

3. MODEL PREDICTIVE CONTROLLER

MPC technique has been applied on decoupled model of AUV for controlling speed, steering and depth assuming ocean currents and other disturbances. An algorithm has been developed for maintaining the vehicle with desired set points. The key elements of MPC are cost function and constraints. The cost function is minimized using an algorithm and applied on controller for desired response (Mayne, Seron & Rakovic, 2006). Constraints can be chosen on inputs, states and outputs. Model is important in MPC.

MPC calculations are developed at each sampling instant, these predictions are used for set point calculations and control calculations. The constraints on input and output variables can be applied on MPC calculations. The model predictive control calculations determine the appropriate sequence of control moves so as to get the optimal results (Budiyono, 2011), (Liuping, 2009).

Considering the AUV model to be controlled as

$$\begin{aligned} \dot{x}_m(t) &= A_m x_m(t) + B_m u_m(t) \\ y_m(t) &= C_m x_m(t) \end{aligned} \quad (9)$$

where $x_m(t)$ is the state vector of dimension n_1. A_m, B_m and C_m have dimension $n_1 \times n_1$, $n_1 \times m$ and $q \times n_1$, respectively.

Now this model is converted to axillary model with axillary variables for MPC formulation

$$\begin{aligned} z(t) &= \dot{x}_m(t) \\ y(t) &= C_m x_m(t) \end{aligned} \quad (10)$$

The augmented state space model using the derivative of control system is

$$\begin{bmatrix} \dot{z}(t) \\ \dot{y}(t) \end{bmatrix} = \begin{bmatrix} A_m & 0_m{}^T \\ C_m & 0_{q\times q} \end{bmatrix} \begin{bmatrix} z(t) \\ y(t) \end{bmatrix} + \begin{bmatrix} B_m \\ 0_{q\times m} \end{bmatrix} \dot{u}(t) \quad (11)$$

$$y(t) = \begin{bmatrix} 0_m & I_{q\times q} \end{bmatrix} \begin{bmatrix} z(t) \\ y(t) \end{bmatrix} \quad (12)$$

which is in the form of

$$\dot{x}(t) = Ax(t) + B\dot{u}(t)$$

$$y(t) = Cx(t) \quad (13)$$

where

$$A = \begin{bmatrix} A_m & 0_m{}^T \\ C_m & 0_{q\times q} \end{bmatrix}(t), \quad B = \begin{bmatrix} B_m \\ 0_{q\times m} \end{bmatrix}(t),$$

$$C = \begin{bmatrix} 0_m & I_{q\times q} \end{bmatrix}$$

The cost function considered for optimization is

$$J = \eta^T\Omega\eta + 2\eta^T\psi x(t_i) + \text{constant} \quad (14)$$

where η is Laugree coefficient vector, ψ, ω are matrices which can be computed from A, B, C and weighing matrices (Zhen & Jing, 2010)

With the help of receding horizon control principle (i.e., the control action uses only the derivative of the future control signal at τ=0,) the derivative of the optimal control for the unconstrained problem with finite horizon prediction is

$$\dot{u}(t) = \begin{bmatrix} L1(0)^T & 0_2 & . & . & . & o_m \\ o_1 & L2(0)^T & . & . & . & o_m \\ . & . & . & . & . & . \\ . & . & . & . & . & . \\ . & . & . & . & . & . \\ o_1 & o_2 & . & . & . & Lm(0)^T \end{bmatrix} \begin{bmatrix} \eta_1 \\ \eta_2 \\ . \\ . \\ . \\ \eta_m \end{bmatrix}$$

where L is the Laguerre function For an arbitrary time t, $\eta = -\Omega^{-1}\psi\ x(t)$, the continuous time derivative of the control is

$$\dot{u}(t) = -\begin{bmatrix} L1(0)^T & 0_2 & . & . & . & o_m \\ o_1 & L2(0)^T & . & . & . & o_m \\ . & . & . & . & . & . \\ . & . & . & . & . & . \\ . & . & . & . & . & . \\ o_1 & o_2 & . & . & . & Lm(0)^T \end{bmatrix} \Omega^{-1}\psi x(t)$$

$$= -K_{mpc}x(t)$$

where the feedback gain matrix is

$$K_{mpc} = \begin{bmatrix} L1(0)^T & 0_2 & . & . & . & o_m \\ o_1 & L2(0)^T & . & . & . & o_m \\ . & . & . & . & . & . \\ . & . & . & . & . & . \\ . & . & . & . & . & . \\ o_1 & o_2 & . & . & . & Lm(0)^T \end{bmatrix} \Omega^{-1}\psi$$

From the above equation, the receding control law is of state feedback control because of the dependence on the current state variable x(t). The augmented state space model (Mayne, Seron & Rakovic, 2006) is

$$\dot{x}(t) = A\ x(t) + B\ \dot{u}(t)$$

From augmented state space model where the input is $\dot{u}(t)$. The closed loop control system is

$$\dot{x}(t) = (A - BK_{mpc})x(t) \qquad (15)$$

The closed loop eigenvalues of the predicted control system can be evaluated from the equation (15).

The control law is calculated for each sub system of AUV from equations (3), (6), (8) and simulated in MATLAB environment. Table II gives the simulation parameters used in Model Predictive Controller Design.

The following steps involved in proposed MPC tuning

1. Model Horizon: The model horizon N should be chosen such that $N\Delta t \geq$ open-loop settling time. Values of N can be chosen between 20 and 70
2. Prediction Horizon: It calculates how far into the future the control objective reaches. Increasing Np results in a more conservative control action but it increases the computational effort. Np=N+Nu. Np is prediction horizon and Nu is the control horizon
3. Control Horizon: It determines the number of control actions calculated into the future. Too large value of NU results in excessive control action. Smaller value of Nu yields a controller relatively insensitive to model errors.
4. In optimization usually weighting factors should be in proper limits. Larger values of weights penalize the magnitude of Δu more and results less vigorous control actions.

Table II Simulation Parameters of MPC used in Decoupled AUV

Simulation Parameters	Value
Desired Depth, m	100
Desired Position, m	1
Desired Speed m/sec	1
Sampling time s	0.2
Time in sec	(0-10) & (0-40)
Control horizon	5
Prediction Horizon	50

4. RESULTS

All the simulations are carried out in MATLAB environment.

a) In Figure 2 set point for depth control is taken as 100 m below the sea level. The vehicle is also following the same set point
b) In Figure 3 and 4 step and sinusoidal inputs are taken as set points. Steering system of Vehicle is tracking in the same set point.
c) In Figure 5 vehicle is following forward surge speed motion. It has good transient and steady state response.

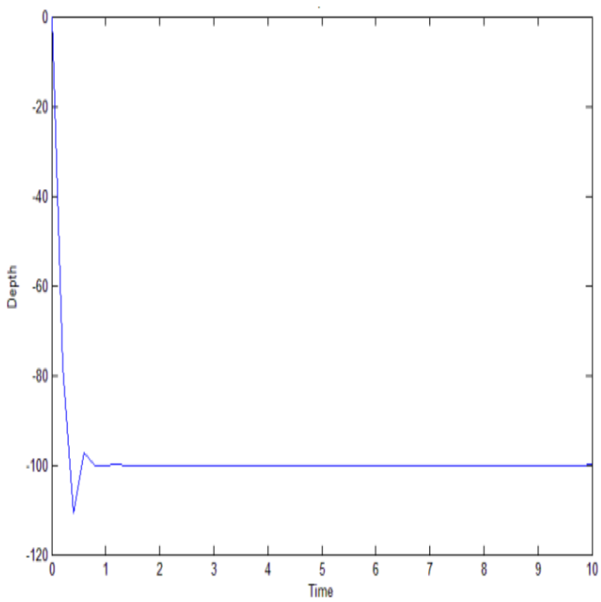

Figure.2 Depth Control of an AUV

Figures 3 and 4 are steering control with respect to step and sinusoidal input.

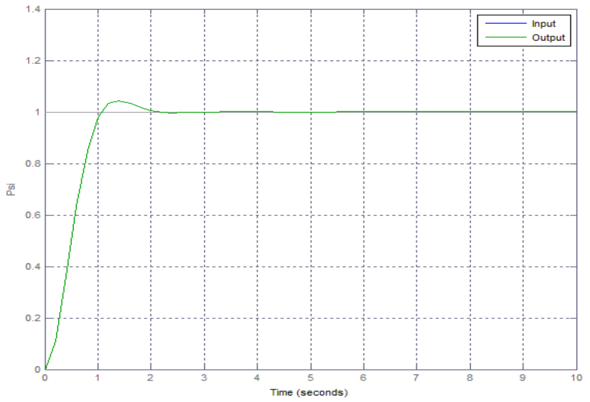

Figure.3 Heading Control of an AUV

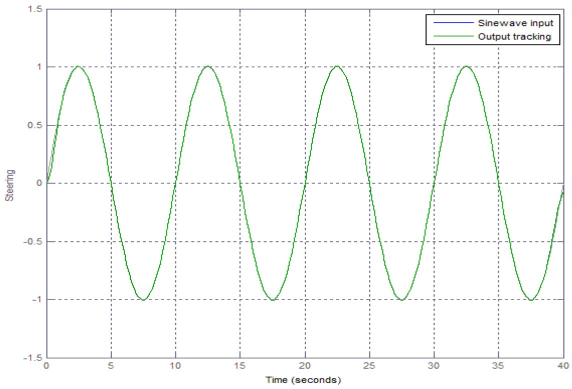

Figure.4 Steering Control of AUV

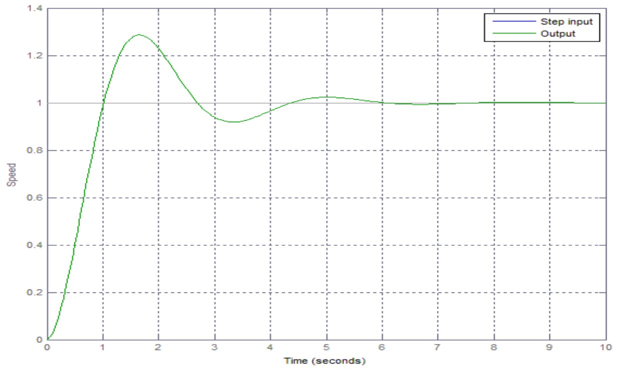

Figure.5 Speed Control of AUV

5. CONCLUSION

Dynamics of Decoupled system of an AUV is considered in this paper. Control can easily applied on decoupled model compared to overall model. The presented MPC algorithm is based on a state space model of an AUV and is therefore flexible to be used for decoupled systems. Concept of moving horizon is used to find the control law. Simulation has been carried out in MATLAB environment. The simulation results demonstrate the effectiveness of the proposed control method. Further the performance can be improved by considering the higher level of MPC.

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