AN INVESTIGATION INTO THE HYDRODYNAMIC CHARACTERISTICS OF A HIGH-SPEED PARTIAL AIR CUSHION SUPPORTED CATAMARAN (PACSCAT)

Molland, A.F.,
School of Engineering Sciences,
University of Southampton,
Southampton, SO17 1BJ, UK

Wilson, P.A.,
School of Engineering Sciences,
University of Southampton,
Southampton, SO17 1BJ, UK

Lewthwaite, J.C
Independent Maritime Assessment Associates
Ltd., Fareham, PO14 3HX, UK.

Taunton, D.J.
BMT SeaTech Ltd.,
Southampton, SO14 3TJ, UK.

ABSTRACT

The partial air cushion supported catamaran concept (PACSCAT) can offer an efficient solution when a relatively high payload is required to be carried at relatively high speeds with a restricted draught. Such circumstances reflect potential modes of shallow water operation in inland waterways such as the Rhine and Danube.

In support of the development of the concept, numerical models of the resistance and wave wash characteristics have been developed and extensive tank tests in shallow water carried out. A theoretical model of the cushion has been combined with thin ship theory representing the side hulls. The developed model offers the facility to make performance estimates of the powering and wave wash for various operational scenarios.

The paper also describes the choice of propulsor for the vessel, from the options of open propellers, ducted propellers or waterjets. It is found that the best propulsive efficiency can be achieved with the use of ducted propellers.

NOMENCLATURE

A	Wetted surface area [m^2]
a	Half cushion length [m]
b	Half cushion width [m]
Fn	Froude Number [$V/(gL)^{1/2}$]
Fn_H	Depth Froude Number [$V/(gH)^{1/2}$]
H	Water depth [m]
k	Wave number [m^{-1}]
L	Length on waterline [m]
P_C	Cushion pressure [Pa]
Q_F	Lift fan flow rate [m^3/s]
S	Separation between catamaran demihull centrelines [m]
V	Ship speed [m/s]
W	Channel breadth [m]
R_T	Total resistance [N]
C_T	Coefficient of total resistance [$R_T/½\rho AV^2$]
R_{WP}	Wave pattern resistance [N]
C_{WP}	Coefficient of wave pattern resistance [$R_{WP}/½\rho AV^2$]
g	Acceleration due to gravity [$9.81 m/s^2$]
ρ	Density of water [kg/m^3]
ρ_A	Density of air [kg/m^3]
η_D	Propeller quasi propulsive coefficient
η_T	Transmission efficiency [gearbox and shafting]
η_F	Efficiency of lift fan(s)
σ	Source strength [m^2/s]
θ	Wave angle [deg.]
ζ	Wave elevation [m]
∇	Displacement volume [m^3]

1. INTRODUCTION

1.1 Background

The partial air cushion supported catamaran concept (PACSCAT) can offer an efficient solution when a relatively high payload is required to be carried at relatively high speeds with a restricted draught. In this concept, which was originally developed by IMAA Ltd., the cushion is rectangular in plan form and is contained by the side hulls, a bow seal and a stern flap. Vertical support is provided by approximately 50% buoyant support from the side hulls and 50% from the cushion. This paper describes the hydrodynamic investigation which formed part of a much wider study into the concept including practical, technical, economic and environmental aspects of operation on inland waterways such as the Rhine and Danube. The overall investigation is described in Ref.1.

1.2 Hydrodynamics

As part of the overall investigation into the proof of concept, there was a need to be able to estimate the hydrodynamic performance characteristics of the craft, particularly concerning operation in shallow water. Resistance and wash characteristics are available for high speed semi-displacement catamarans in shallow water as is information on the resistance of fully cushion supported hovercraft, such as those reported in Refs. 2 and 3. Little detailed work has been carried out on partially supported cushion craft. The current hydrodynamic investigation has therefore included an extensive set of physical tank tests, together with the development of a numerical model of the concept. The combined contributions of the experimental results and numerical methods allow performance estimates to be made of the powering and wave wash for various operational scenarios.

The overall powering characteristics of the PACSCAT vessel, at a particular speed, are shown in Fig.1. As cushion pressure is increased from zero, the buoyant (hull) force decreases as more of the ship mass is supported by the cushion. As cushion pressure and lift fan power is further increased, the hull resistance and propulsive power initially decreases rapidly but then more slowly as the wave resistance of the cushion increases. This decrease in resistance and power holds until there is significant air leakage under the seals at which point cushion pressure does not increase any more with fan speed. Any further increase in fan speed and power does not produce a decrease in resistance. It is therefore seen that, at a particular ship speed, there is an optimum cushion pressure. In broad terms, if ship speed is increased, the optimum buoyant support decreases as the fully cushion supported, or hovercraft, concept is approached. If ship speed is decreased, the optimum buoyant support increases, moving towards a displacement craft. For the operating size and speed of the PACSCAT concept, the optimum buoyant support is typically between 40% and 60%. A surface effect ship (SES) will typically have a buoyant support of about 15%, i.e. a cushion support of about 85%.

1.3 Design proposals

Design proposals for operation on the Danube and Rhine rivers were developed using practical constraints such as canal widths, water depths and air draughts, together with operational simulations to indicate overall size and speed. The resulting design proposal has an overall length of 135m [Lwl = 127.5m] and breadth of 22.8m (the largest dimensions permissible on the waterways), a full draught of 2.6m and a draught on cushion of 1.6m, a deadweight of about 2000 tonnes and an operating speed of 18.4 knots (34km/hr). A significant feature of the concept is that the small draught on cushion allows operation in areas which are well below the draught limitations of conventional vessels. An outline profile and plan of the ship is shown in Fig. 2

2. EXPERIMENTS

Extensive tank tests in calm shallow water were carried out in the GKN-Westland tank in the UK and at the VBD tank at Duisburg in Germany. A 2.3m variable geometry generic model of the PACSCAT concept was tested in both tanks. This model was designed in such a way that the hulls, displacement and hull separation could be changed. Thin plastic side plates could also be applied instead of the hulls in order to simulate the moving pressure pad. This model was tested in shallow water (50mm and 100mm) for resistance and wash only.

A 6.8m model, closely representing the final vessel design, was tested in the VBD tank. This model was tested in shallow water (140mm and 200mm) for resistance, wash and self-propulsion experiments. A range of speeds, displacement, trim and cushion pressure were tested and the results used in subsequent power prediction and validation exercises. A number of detailed experiments were also carried out into different bow seal configurations and stern flap angles. Wave cuts are shown in Fig.3 for both the 2.3m and 6.8m models at the same Froude number in the VBD tank. The agreement between the two models, particularly for the leading part of the wave system, is good.

The 6.8m model was fitted with four stock propellers in tunnels. Propulsion tests were carried out over a range of conditions at the design speed. The relatively high ship speed, coupled with a shallow draught and limited propeller diameter, leads to high propeller thrust loadings. It was noted early in the propulsion tests that air was being drawn under the outer side of the outboard tunnels, leading to ventilation and loss of thrust on the outboard propellers, Fig.4. The outboard side of the propeller tunnels were therefore modified and extended downwards closer to the baseline and the fore end carefully faired. Further experiments demonstrated that ventilation was now prevented. Overall, the experiments had identified and solved potential problems with heavily loaded propulsors in this form of layout.

3. NUMERICAL MODEL

3.1 Outline of the model

The hulls are modelled using thin ship theory, a version of which has been developed over a number of years at the University of Southampton Refs.4 and

5. The cushion is modelled using a moving pressure segment theory which was originally developed to model hovercraft, Ref.6. The cushion is rectangular in plan form and the bow seal and stern flap are not modelled.

The modelling process is outlined in Fig. 5. This process uses the WUMTIA lines fairing package ShipShape, Ref.7, to generate the hull input file for a given sinkage and trim. The WUMTIA hydrostatics package is used to determine the wetted surface area and draught for a given lift force (cushion pressure × cushion area). The outputs from the numerical model are wave patterns for the hulls and the cushion consisting of wave heights at given x and y coordinates. The wave pattern due to the combined hulls and pressure pad is the linear sum of the ζ terms from the thin ship theory and the moving pressure pad.

Analysis of the wave pattern allows wave cuts to be interpolated at intermediate lateral positions and the wave pattern resistance to be determined. Since the wave resistance is a function of the wave height squared, the wave resistance needs to be found for the combined hull and cushion wave system rather than for each component and summing.

The output from the numerical model is the wave pattern and wave resistance for the specified run condition. From this wave pattern, statistical values of maximum wave height, wave period, wave energy and divergent wave angle can be determined. The wave pattern resistance can be combined with the viscous and other components of resistance to estimate the total resistance of the PACSCAT vessel.

3.2 Thin Ship Theory

The background and development of the theory is described in Refs.4 and 5. In the theory, it is assumed that the ship hull(s) will be slender, the fluid is inviscid, incompressible and homogeneous, the fluid motion is steady and irrotational, surface tension may be neglected and that the wave height at the free surface is small compared with the wave length. For the theory in its basic form, ship shape bodies are represented by planar arrays of Kelvin sources on the local hull centrelines, together with the assumption of linearised free surface conditions. The theory includes the effects of a channel of finite breadth and the effects of shallow water.

The strength σ of the source on each panel may be calculated from the local slope of the local waterline, Equation (1)

where $\frac{dy}{dx}$ is the slope of the waterline. The hull waterline offsets in the current procedures can be

$$\sigma = \frac{-V}{2\pi}\frac{dy}{dx}dA \qquad (1)$$

obtained directly and rapidly as output from a commercial lines faring package, such as ShipShape, Ref.7.

The wave system is described as a series using the Eggers coefficients, Equation (2). The wave coefficients ξ_m and η_m can be derived theoretically using Equations (3), noting that they can also be derived experimentally from physical measurements of ζ in Equation (2).

The wave pattern resistance may be calculated from Equation (4) which describes the resistance in terms of the Eggers coefficients.

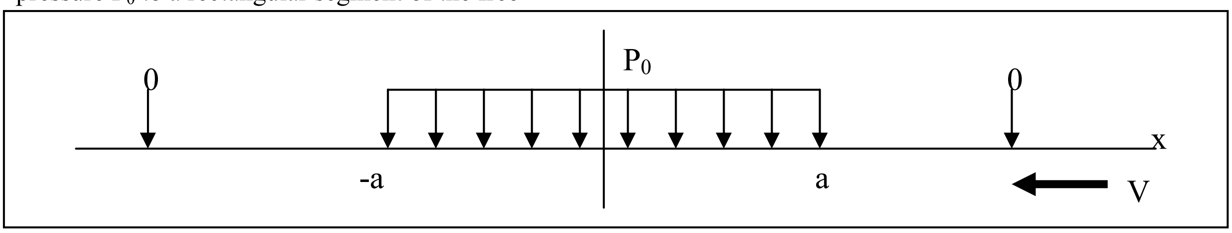
$$\zeta(x,y) = \sum_{m=0}^{m}\left[\xi_m \cos\left(xk_m\cos\theta_m\right) + \eta_m \sin\left(xk_m\cos\theta_m\right)\right]\cos\frac{m\pi y}{W} \qquad (2)$$

$$\begin{vmatrix}\xi_m\\ \eta_m\end{vmatrix} = \frac{16\pi U}{Wg}\frac{k_0 + k_m\cos^2\theta_m}{1+\sin^2\theta_m - k_0 H\,\mathrm{sech}^2\left(k_m H\right)}\sum_{\sigma}\left[\sigma_\sigma e^{-k_m H}\cosh\left[k_m\left(H+z_\sigma\right)\right]\begin{vmatrix}\cos\left(k_m x_\sigma\cos\theta_m\right)\\ \sin\left(k_m x_\sigma\cos\theta_m\right)\end{vmatrix}\left\{\begin{matrix}\cos\frac{m\pi y_\sigma}{W}\\ \sin\frac{m\pi y_\sigma}{W}\end{matrix}\right.\right] \qquad (3)$$

$$R_{WP} = \frac{\rho g W}{4}\left\{(\xi_0^2 + \eta_0^2)(1 - \frac{2k_0 H}{\sinh(2k_0 H)}) + \sum_{m=1}^{M}(\xi_m^2 + \eta_m^2)\left[1 - \frac{\cos^2\theta_m}{2}\left(1 + \frac{2k_m H}{\sinh\left(2k_m H\right)}\right)\right]\right\} \qquad (4)$$

3.3 Modelling the moving pressure pad

The moving pressure pad is modelled by applying a pressure P_0 to a rectangular segment of the free surface defined by $-a \le x \ge a$, and $-b \le y \ge b$, Ref.6. Outside of the pressure segment the pressure is atmospheric.



The pressure P, acting over the rectangular planform a.b is:

$$P(K_m,\theta_m) = \frac{W}{2\pi^3}\frac{Sin(aK_m Cos\theta_m)}{K_m Cos\theta_m}\frac{Sin(\frac{2m\pi b}{W})}{m} \qquad (5)$$

The wave elevation ζ is given by Equation (6):

$$\zeta(x,y) = \frac{-8\pi^2 P_0}{\rho g W}\sum_{m=-\infty}^{\infty}\frac{K_m Cos\theta_m P(K_m,\theta_m) Sin K_m \omega_m}{2 - Cos^2\theta_m(1 + 2K_m H / Sinh 2K_m H)} \qquad (6)$$

The wave resistance of the moving pressure segment is derived by analysis of longitudinal wave cuts.

It is noted that the combined theories (thin ship and pressure segment) provide an estimate of the proportions of transverse and diverging content in the wave system and that the theoretical predictions of the wave pattern and wave resistance can be compared directly with values derived from physical measurements of the wave elevation.

3.4 Comparison with experiments

Figs.6a and 6b show experimental results for tests with the 2.3m model fitted with plate side hulls, compared with the pressure segment theory. At speeds approaching critical there is a large leading bow wave which the numerical model under predicts. At higher speeds, the numerical model produces acceptable predictions. It should be noted that the influence of the bow seal and stern flap were not modelled numerically or allowed for. It is expected that this influence will be larger at lower speeds when the water tends to build up in front of the seal and flap.

The influence of hull spacing on wash is shown in Figs.7a-c. Hull spacings of 300mm, 400mm and 500mm were tested and as the spacing/area increased the cushion pressure was reduced to maintain the same lift force and buoyant fraction. It is noted that the influence of hull spacing on wave height is small. The numerical predictions include the hulls and the pressure pad. Generally, there is reasonable agreement between the numerical and experimental wave heights, although the numerical results are displaced downwards.

The prediction of wave resistance by experimental and numerical methods is shown in Fig.8. There is good agreement between the numerical and experimental results at higher speeds, although differences develop as the critical speed is approached, where the numerical method is less reliable.

A number of experimental and numerical investigations were also carried out into the influences of the length and position of the cushion relative to the hulls, and the potential for favourable wave resistance cancellation. These investigations showed limited success for practical layouts.

4. POWERING

4.1 Total service power

The total required service power is made up of the power required for propulsion and that required for the lift fan:

Total Service Power, P_S:
Ps = Hull Propulsive Power + Lift Fan Power

$$= P_E/\eta_D\eta_T + P_C Q_F/\eta_F$$
$$= R_T.V/\eta_D\eta_T + P_C Q_F/\eta_F \qquad (7)$$

The total resistance coefficient may be expressed as:

$$C_T = (1 + \beta k)C_F + C_W + \Delta C_F + C_S + C_A + C_{APP} + C_M \qquad (8)$$

where $(1+\beta k)C_F$ is the total viscous resistance, C_W is the wave resistance of hull and cushion (derived experimentally or by numerical analysis), ΔC_F is an allowance for hull roughness, C_S is an allowance for skirt and stern flap drag, C_A is air (or profile) drag and C_{APP} an allowance for appendage drag.
C_M is the momentum drag and arises because the horizontal velocity of the air in the atmosphere that is drawn vertically down into the lift fan(s) is increased from zero to ship velocity. That mass flow of air experiences an increase in momentum and the reaction on the ship to the resulting force gives rise to the momentum drag:

$$D_M = \rho_A Q_F V$$

$$\text{and } C_M = D_M / \tfrac{1}{2}\rho A V^2 \qquad (9)$$

Skin friction coefficient C_F may be derived from the ITTC Correlation line:

$$C_F = 0.075 / [\log Rn - 2]^2$$

and the viscous form factor for catamaran hulls, Ref.8, is assumed as:

$$(1 + \beta k) = 3.03\ (L/\nabla^{1/3})^{-0.40} \qquad (10)$$

$$\Delta C_F = \left[105 \left(\frac{K_S}{L} \right)^{1/3} - 0.64 \right] \times 10^{-3}$$

where a mean value of $K_S = 100 \times 10^{-6}$.

Using the above procedure the total resistance is derived as $R_T = C_T \times \frac{1}{2}\rho AV^2$ and the effective power P_E (to tow the vessel) as:

$$P_E = R_T \times V$$

4.2 Choice of propulsor

The available propulsor options amounted to waterjets, conventional open propellers and ducted propellers.
Due to the shallow draught, waterjets would have been an attractive proposition. However, based on the service speed of 18.4 knots, waterjets would be expected to have relatively low efficiency (of the order $\eta_D = 0.52$) and their use was not investigated further. Open and ducted propellers operating in tunnels were considered, and the propulsion experiments were carried out using four open propellers (from stock). The propellers were enclosed in tunnels with the tunnels raised in way of the propellers to allow a maximum propeller diameter to be incorporated, Fig.9. The layout is relatively common practice in inland waterway vessels. The propulsion experiments indicated that the propeller open water efficiency would be very low (of the order $\eta_O = 0.42$), due mainly to the restricted diameter and high thrust loadings. Using ducted propellers indicated that the open water efficiencies η_O could be raised to the order of 0.52 inboard and 0.59 outboard which, when combined with the hull efficiency, led to quasi propulsive coefficients η_D of the order 0.60 inboard and 0.55 outboard. Four ducted propellers in tunnels were therefore adopted for the final design proposal.

5. WASH

Wave heights, as a series of wave cuts, were measured during all the tests in both tanks. Examples of such wave cuts, and numerical predictions, are shown in Figs.6 and 7. These provide near field wave patterns, typically between 50% - 90% of ship length off the ship centreline and the facility to deduce wash wave height. Typical relative values of wash height for the PACSCAT proposed design with change in ship speed are shown in Fig.10. The wave wash height decays as the waves propagate to the shore and the actual heights of the waves approaching the shore will be lower than those shown in Figs.6, 7 and 10.

6. PERFORMANCE PREDICTIONS

Using the extensive experimental data base and numerical model, a number of performance predictions were carried out. These entailed the derivation of resistance, power and wash height over a range of displacements, speeds and water depths suitable for use in operational simulations and investigations into environmental impact. Typical wash and powering predictions for the design displacement are shown in Figs.10 and 11. It is clear that significant changes in wash and power occur near the critical condition. It can be deduced from these results that ship speed will have to be varied for different water depths in order to avoid extreme conditions. In particular, near the critical condition, the speed may have to be increased by a suitable amount or decreased below critical.

As well as the resistance and wash experiments, coursekeeping and manoeuvring experiments were carried out on the self-propelled PACSCAT model in the VBD tank. These were used in support of numerical simulations which verified that the proposed design has excellent coursekeeping and manoeuvring characteristics.

7. CONCLUSIONS

The results of the experimental and numerical investigations into the resistance, wash and manoeuvring characteristics indicate that the PACSCAT design proposal is technically feasible and viable.

The experimental data base and developed numerical tools enable performance predictions to be made for a wide range of variants and applications of the PACSCAT concept.

ACKNOWLEDGEMENTS

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REFERENCES

1. Clements, R., Lewthwaite, J.C., Ivanov, P., Mollamd, A.F. and Wilson, P.A. The Potential for the use of a novel craft, PACSCAT (Partial Air Cushion Supported CATamaran) in inland European waterways. *The 8th Intl. Conf. on Fast Sea Transportation FAST'2005*, St. Petersburg, 2005.
2. Molland, A.F., Wilson, P.A., Taunton, D.J., Chandraprabha, S. and Ghani, P.A. Resistance and wash Measurements on a Series of High Speed Displacement Monohull and Catamaran Forms in Shallow Water. International Journal of Maritime Engineering, *Trans. of the Royal Institution of Naval Architects,* Vol.146, Part A2, 2004.

3. Everest, J.T. and Hogben, N. A Theoretical and Experimental Study of the Wavemaking of Hovercraft of Arbitrary Planform and Angle of Yaw. *Trans. of the Royal Institution of Naval Architects*, Vol.111, 1969.
4. Insel, M., An Investigation into the Resistance Components of High Speed Displacement Catamarans. *Ph.D. Thesis*, Department of Ship Science, University of Southampton, 1990.
5. Couser, P.R., Wellicome, J.F., and Molland, A.F., An Improved Method for the Theoretical Prediction of the Wave Resistance of Transom Stern Hulls Using a Slender Body Approach. *International Shipuilding Progress*, Vol. 45, 1998, 331-349.
6. Wellicome, J.F., A Theory for a Moving Pressure Segment. Personal Communication to A.F. Molland, 2000.
7. ShipShape Lines Fairing Package. User Manual. Wolfson Unit for Marine Technology and Industrial Aerodynamics, WUMTIA, University of Southampton.
8 . Molland, A.F., Karayannis, T., Taunton, D.J. and Sarac-Williams, Y. Preliminary Estimates of the Dimensions, Powering and Seakeeping Characteristics of Fast Ferries. *Eighth International Marine Design Conference, IMDC 2003,* Athens, Greece, May, 2003.

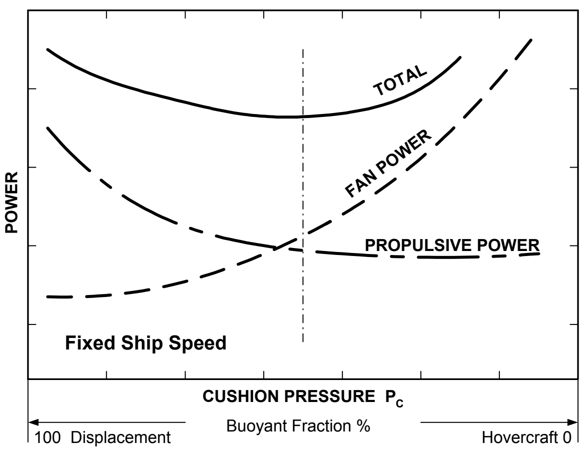

Fig.1 Powering characteristics for partial cushion supported catamaran

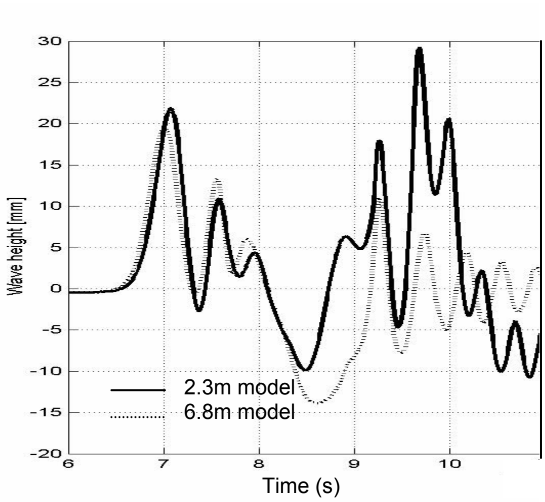

Fig.3 Comparison of wave cuts for 2.3m and 6.8m models

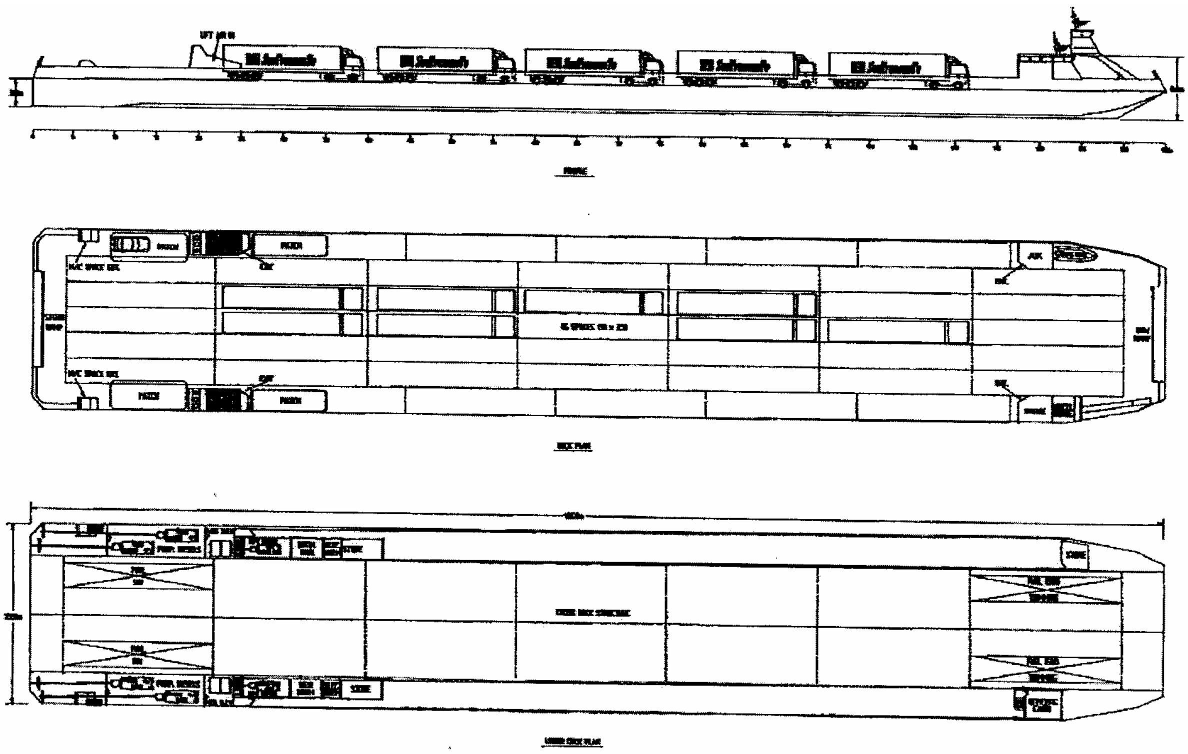

Fig.2 PACSCAT Outline profile and plan

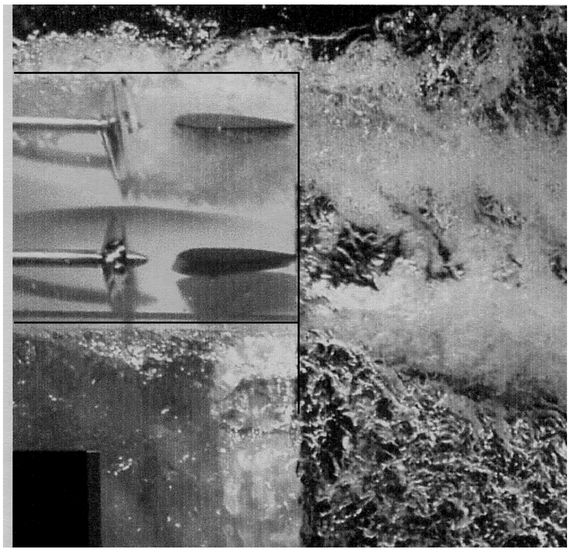

Fig.4 Ventilation on outboard propeller (viewed from below)

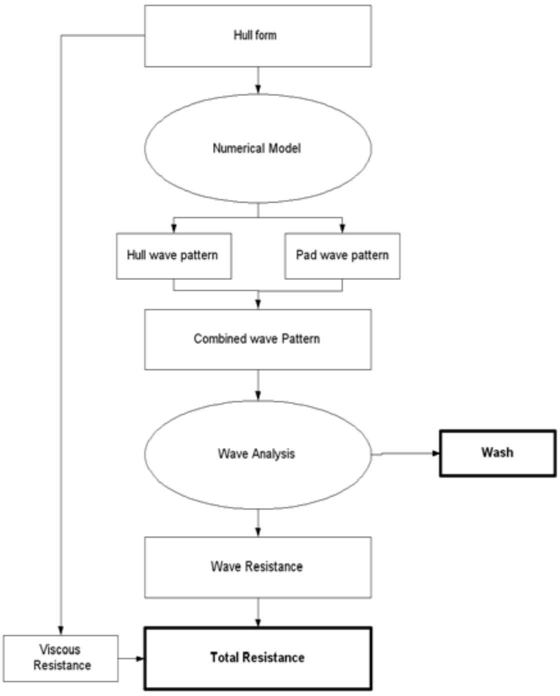

Fig.5 Outline of numerical modelling process

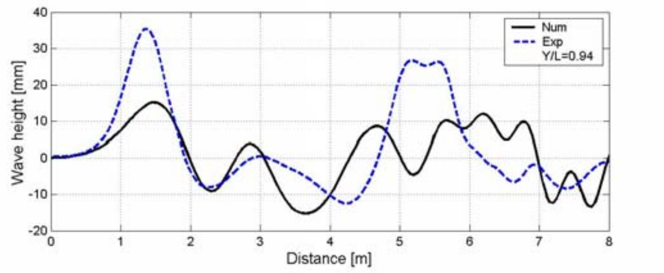

Fig.6a Plate side hulls V=1.4m/s S=500mm, H=100mm

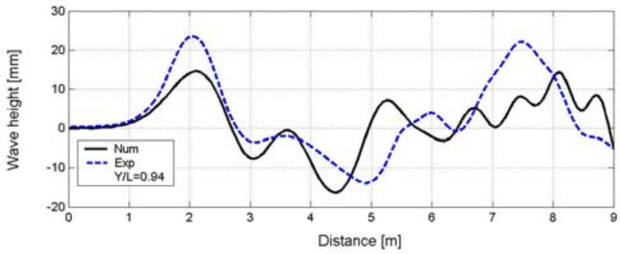

Fig.6b Plate side hulls V=1.6m/s S=500mm, H=100mm

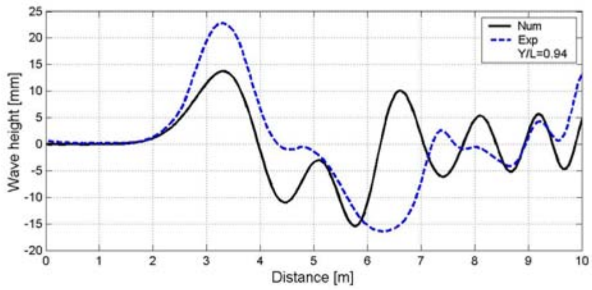

Fig.7a Influence of hull spacing S = 300mm

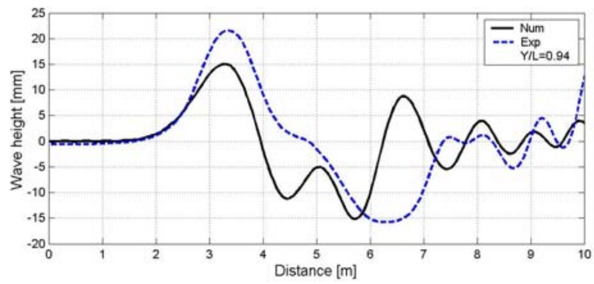

Fig.7b Influence of hull spacing S = 400mm

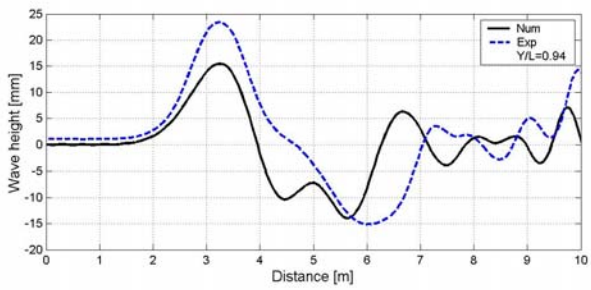

Fig.7c Influence of hull spacing S= 500mm

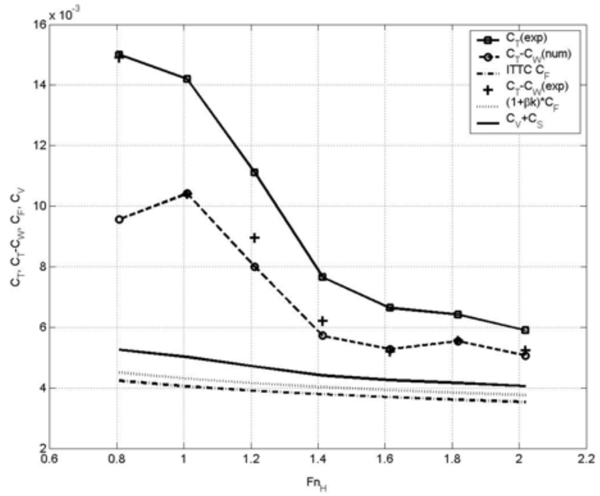

Fig.8 Predicted wave resistance

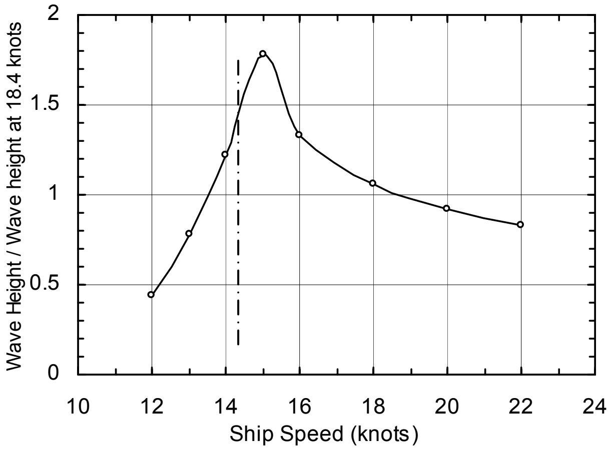

Fig.10 Wash height
(water depth = 5.5m, y/L = 0.5)

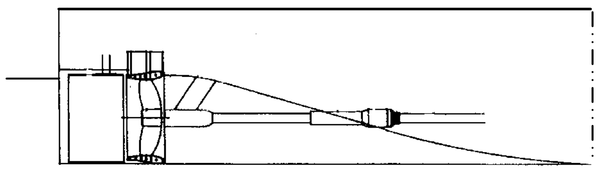

Fig.9 Propeller tunnel arrangement

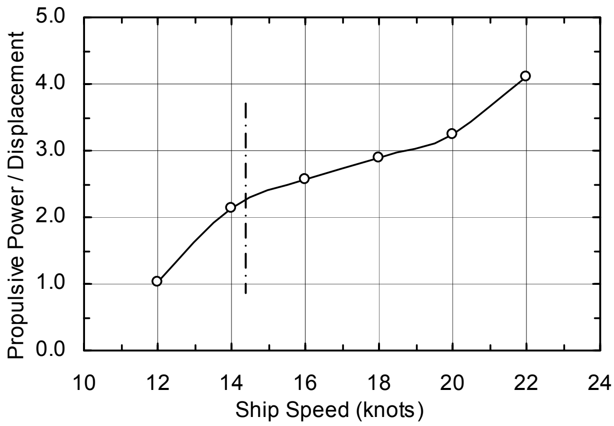

Fig.11 Power characteristics