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Paper No 1

Barry Coupland

BSL Barrigan Group, New Zealand

"Continuing Developments with the Mestral Class of High Speed Ferries"

PAPER ONE

CONTINUING DEVELOPMENTS WITH THE "MESTRAL" CLASS OF HIGH SPEED FERRIES

"Monohull Excellent Seakeeping TRAnsport and Leisure"

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J B Coupland	:	BSL Barrian Group, Wellington	:	NEW ZEALAND

Summary

Empressa Nacional BAZAN was commissioned at the end of 1991 by Compania Trasmediterranea to develop a new fast ferry design for the transport of passengers and vehicles between Barcelona and the Baleares Islands in the Mediterranean Sea.

Historically the builder of naval ships for the Spanish Navy, Bazan's San Fernando Shipyard was established in 1711 and the request from Compania Trasmediterranea ideally suited Bazan's planned diversification into fast ferry building.

This paper briefly describes the evolution and continuing developments of the "state of the art" Mestral Class of High Speed Ferries and includes details of design and philosophical criteria, together with operational experience with the design prototype, MV ALBAYZIN, in New Zealand on Cook Strait 1994-95.

The paper has been compiled by J B Coupland, utilising operational experience as Chief Engineer of ALBAYZIN, together with the assistance of J C Sanchez, Commercial Manager, Bazan, Madrid.

Concept

Compania Trasmediterranea proposed the following parameters to Bazan for service on the Barcelona to Palma de Mallorca route:

–	Passengers:	450
–	Cars:	76
–	Speed:	35 knots
–	Distance between ports:	130 nautical miles
–	Daily single trips:	3 in summer time
		2 in the rest of the year
–	Passenger comfort according to ISO 2631/3 for a SS4	
–	Propulsion:	Conventional diesel engines

Environmental statistics for the Barcelona-Palma de Mallorca route showed that the significant wave height is less than 3.0 m for 90% of the year and less than 2.0 m for 70% of the year. Mean wave period is about 6 seconds with a low scatter, and the prevailing wind direction is North - South, very close to the route direction.

Trasmediterranea had anticipated that multihull vessels may fulfil its need, but Bazan soon established that such vessels experienced very high vertical accelerations in the prevailing conditions. Bazan had already studied monohulls in deep-V forms for fast patrol boats and this experience led it to test derivations of the latter.

Deep-V monohulls proved to be the ideal solution for the route being studied. As well as very good performance in waves, the hullform also demonstrated excellent sustained speed characteristics.

A monohull deep-V form vessel was preferred to a multihull for several reasons: in the specified sea conditions, vertical acceleration values in the passenger area (in the monohull solution) were about 50% of those found in an equivalent multi-hull this meant that passenger comfort in the monohull was improved four fold over that expected for a multihull in similar conditions. Hull construction of the monohull is also much simpler and allows a superior engine room layout for maintenance and operation to be achieved.

The preference for the monohull was not infinite, as it could require around 25% more propulsion power than the equivalent multihull. However the seakeeping studies showed that a monohull solution could maintain a specified passenger comfort standard for a higher percentage of time over the year, than that expected for the multihull solution.

Comfort criteria which could be maintained, more than 90% of the time over the year in the operation area, subsequently confirmed by model tests, were: vertical accelerations as stated in ISO-2631/3; transverse accelerations (RMS) of 0.4 m/s^2 or less, roll angle (RMS) of 2.5 degrees or less, and 20 or fewer slams per hour. This exceptional performance is a very important facet for any ship profitability analysis, which is sometimes forgotten when comparisons between different ships are carried out.

A choice between a steel and aluminium hull had to be made as both materials were feasible for a monohull of the size in question. Service speed was a critical factor as the required 35 knots could be achieved with four diesel engines in an aluminium hull but the steel solution's extra 220/250 tonnes of structural weight dictated a gas turbine propulsion solution, not acceptable to Trasmediterranea, and with an accompanying extra initial cost of at least US$4 million, and increased maintenance and fuel costs.

The result of Bazan's evaluations of all parameters was the "MESTRAL" consisting of the following specifications:

Maximum length	95.20 m
Maximum breadth	14.60 m
Depth	8.90 m
Full load displacement	946 t
Full load draught	2.06 m

Description

MESTRAL is a deep - V monohull fast ferry completely built in aluminium and fulfilling of the Trasmediterranea requirements.

MESTRAL is able to transport 450 passengers and 76 cars or a combination of cars and caravans, with 11 caravans as maximum. The total deadweight of the vessel is 172 tonnes and the pay-load (passenger plus vehicles) 127 tonnes.

The trials speed was 35.5 knots at the continuous service rating. An annual average speed of 34 knots in the operational area has been estimated using 90% of the continuous service rating. The vessel has a standard range of 300 nautical miles at 34 knots with 37 tonnes of fuel, allowing a storage of a further 13 tonnes for an extended range of 400 nautical miles.

The vessel including hull, machinery and equipment, has been designed and classified according to Det Norske Veritas (DNVC) "Rules for Classification of High Speed Craft" (January 1991), as amended in January 1993, thus receiving Class notation "+1A1 HSLC R1 Car Ferry A EO".

Furthermore, the design complies with the Code "High Speed Craft" (HSC) of IMO revised on May 1994 for a category "A" ship but, because of the engine room and propulsion system arrangement, can be certified as a category "B" passenger vessel, with only minor modifications to the drencher system.

MESTRAL has an all-welded aluminium alloy structure, using AlMg alloys for the laminated products and specifying Silicon alloys for the extrusions. Bottom, sides and decks are longitudinally stiffened. The bridge deck is rigidly connected to the hull girder to maintain stiffness and stresses within the permissible fatigue values, without compromising the ship weight.

Passenger spaces are separated from the engine rooms by the garage deck, thereby contributing to low noise levels. Passengers are carried on two decks, 68 first-class passengers on the bridge deck and 382 tourist-class passengers on the deck below.

Each passenger is allocated a pullman-style seat, and a split-level arrangement featuring a raised block of seats along the centre of the tourist deck ensures that all passengers have a good view out of the large side windows. A bar and lounge are situated forward of the seating area in each class, and a shop and information desk are located on the tourist deck.

Vehicles access the single drive-through garage deck through the forward door located to port, or the stern door. A centre lane with a height of 3.8 m can accommodate 11 caravans, and the total capacity for cars alone is 76.

Beneath the garage deck the vessel is divided into eight spaces, with seven transverse bulkheads. Two of the spaces form the engine rooms, the aftermost compartment houses the KaMeWa Jet Units, and the forward five spaces are void (except No.1 aft of the forepeak, where the air conditioning machinery is located), providing the required reserve of buoyancy and damage stability. Two adjacent compartments can be flooded without the ship being lost in accordance with SOLAS 74, including the amendments of chapter 11-1, 1988.

Four 16-cylinder Caterpillar 3616 TA medium-speed diesel engines each drive a waterjet via a reduction gearbox. Each engine has a maximum rating of 5420 BkW at 1,000 RPM, and a continuous service rating (CSR) of 5,000 BkW. Only the outer pair of waterjets incorporate steering and reversing mechanisms, which are controlled from the Bridge by way of a "joystick" type of tiller. The inner pair of waterjets are fixed boosters.

Hydraulic power for the outboard steering mechanism is taken from the corresponding gearbox via a PTO. MESTRAL also has a 180 kW bow thruster to assist the waterjets when docking.

Propulsion and electrical plants are divided between two separate spaces for safety and reliability reasons. Each engine room contains two main engines arranged so that each pair drives a port and a starboard waterjet, one of which is steerable and the other fixed. Therefore, if one engine room were to be put out of action by fire or flooding the vessel could still reach port without assistance. Failure of one engine or in one propulsion line still allows service to be maintained.

MESTRAL's stabiliser system comprises two pairs of standard hydraulic active-fin roll stabilisers, sufficient for its operations, but a more complicated ride control system can be installed for other demands.

Two of the three 337 KVA generator sets are located in the forward engine room, and the other one is in the aft engine room. Two generators can meet the complete electrical demand. An emergency genset and switchboard are located on the bridge deck.

MESTRAL is operated from two adjacent consoles in the centre of the bridge, the engineer's to port and the navigator's to starboard.

Bridge equipment is integrated and was specially developed for high-speed craft. navigational equipment includes two interswitchable X-band and S-band ARPA radar sets, GPS receiver, echo sounder, autopilot and gyro compass, and an electronic chart system.

A complete Sailor GMDSS station on the starboard side of the bridge includes VHF/MF/HF transceivers, Navtex, 2,182 kHz watch receiver, and radiotelex equipment. A satcom-C terminal, two 9 GHz radar transponders and three portable VHF sets are also carried.

A colour monitor displays a variety of information from around the vessel as part of a Seamanager system. This can present data such as the status of each engine, relative wind, GPS position, thruster direction, gyro heading, fire detector status, and watertight door position. Measurements of vertical accelerations can be called up on a "comfort meter" display and compared with the ISO 2631/3 standard for levels liable to produce a 10 per cent seasickness incidence.

The engineer's console incorporates two PC displays to monitor propulsion engine parameters, electric plant and auxiliary systems. Both are integrated into the programmable automation system, which fulfils requirements for DNV's EO unattended machinery space notation. Control is effected via a "trackball" type of "mouse" fitted to both PC keyboards.

This distributed alarm/monitor/control system was developed by Bazan and is capable of self-diagnosis to locate faults. It uses fibre-optic cable to transmit all data through a local area network, backed up with redundancy via a wire cable. Each engine room is also monitored by a closed-circuit TV camera.

To minimise time taken for an emergency evacuation, four RFD ramp slides are located on the upper passenger deck, two on each side. Each slide has three inflatable liferafts capable of accommodating 50 people.

Three levels of fire protection are provided: fire detection, passive protection in the form of insulation, and active protection. A fire detection panel is installed on the bridge, connected to sensors in the engine rooms and garage deck. Active fire-fighting measures comprise: a sea-water system, a double-charge CO2 flooding system in the engine room, a fixed and manually operated drencher system in the garage, a foam system with portable applicators on the garage deck, and portable CO2 and dry powder extinguishers distributed throughout the ship.

Seakeeping

One of the most important characteristic of a fast ferry, intended to operate all around the year, is its seakeeping performance. Downtime must be minimised and appropriate passenger comfort must be ensured. A comprehensive investigation was carried out to achieve the best balance between calm water and seakeeping performances.

The test programme was carried out in two different stages. Firstly, three different hull configurations were tested at several speeds in a long towing tank (model scale 1:14), in calm water and in head irregular waves, simulating two different sea states (SS4 & SS5) in two different environmental conditions, i.e. Western Mediterranean sea and open Atlantic ocean.

The following parameters were measured:

Resistance

Heave and pitch motions

Vertical accelerations in five sections

Slamming occurrence and corresponding pressure.

Furthermore, roll decay tests in smooth water and rolling tests in regular waves of different heights were carried out to gather information related to the ship roll stabilisation.

Once the hull was optimised, from a good seakeeping/smooth water point of view, a smaller model (scale 1:24) was built for testing. The model was tested in irregular waves, simulating two different sea states (SS4 & SS5) in different relative headings (0° through 180°) and several speeds.

Besides the foregoing parameters, the following were also measured:

Lateral acceleration in two sections.

Deck wetness occurrence.

Relative motion at the position of waterjet intakes.

In order to have an in depth knowledge of the ship's performance based on the aforementioned seakeeping tests, simulations with the SEAMAN computer program were performed investigating short crested sea effects, stabilising fin effects and manoeuvrability.

The following significant single amplitude values (twice RMS single amplitude values) were obtained for head seas at 35 knots, in waves of Hs = 1.90 m. and modal period of 6.5 s:

Heave	0.22 m
Pitch angle	1.1°
Vertical acceleration (*)	0.95 m/s^2
Added resistance	7%

(*) In the middle of passenger sitting area.

These values correspond to the bare hull, without any type of roll or pitch stabilisation system.

The roll angle obtained with a standard active fin system on was 1.5° (significant single amplitude value).

It is worthwhile mentioning that neither slamming nor deck wetness were observed during the model test at any ship heading or tested sea state.

Based on tank test results and on environmental conditions in some specified traffic areas taken from sea statistics over many years, or from published Wind and Wave data, several downtime analyses were carried out. For example, the analysis performed for the Western Mediterranean sea (between Barcelona/Palma de Mallorca) shows that the Mestral provided with a standard roll stabilising system, only exceeds the seakeeping criteria related to passenger comfort for 8% of the year. The analysis performed with North-Eastern Mediterranean sea data shows that the ship only exceeds that criteria for 8.7% (all year in the Piraeus - Iraklion route) and 4.4% (from May to September in the Piraeus - Khios route).

Manoeuvrability

An extensive and detailed combined model test and computer simulation manoeuvrability study was carried out in order to define the underwater ship appendage configuration and the ship steering system to achieve a good compromise between the different manoeuvring criteria of the fast ferry MESTRAL. The study comprised captive model tests as well as a time domain computer simulation.

The scale 1/24 model was used for the captive model tests and the SEAMAN program for the computer simulation. The coefficients of the non-linear system of equations, which governs the movements of the ship, were calculated based on the results of the model tests. The forces due to surge, sway, roll and yaw motions were measured during the model tests, while the model was free to heave and pitch. The tests were conducted using the rotating-arm techniques for two model speeds and four drift angles.

Two different aft skeg sizes were considered as well as three ship steering system configurations:

a) Two waterjets provided with steering bucket.

b) Four waterjets provided with steering bucket.

c) Two waterjets provided with steering bucket and two rudders located one in each skeg.

The following manoeuvrability characteristics were considered in the study:

- Inherent dynamic stability.
- Yaw checking ability.
- Initial turning/course changing ability.
- Turning ability.

The IMO interim standards for conventional ships were used as a reference, since no manoeuvrability criteria for high speed vessels have been internationally accepted.

Configuration "b" provides slightly better manoeuvrability characteristics than configuration "c". Both configurations largely fulfil the above mentioned standards, configuration "b" being technically simpler. The simpler configuration "a" supplies also sufficient manoeuvrability characteristics to the vessel.

With configuration "b" the following characteristics were achieved:

- Slightly positive course stability index.
- First overshoot (10°/10°): 3.5° at low speed and 7.9° at service speed. (IMO standards ≥ 10°).
- Initial Turning Ability: 1.4 ship lengths at low speed and 1.9 at service speed (IMO standard 2.5).
- Stopping ability: 0.5 ship lengths at low speed and 2.6 at service speed (IMO standard 15).

The turning circle diameter is not so relevant a parameter for this type of ship, however this characteristic was measured obtaining 3L at low speed and 3.5 L at service speed (IMO standard 5L).

Structure

Ship construction normally calls for qualitative precautions when using aluminium as structural material, due to the lower fatigue resistance that this material exhibits with respect to steel. Low level of operative stresses, special care on the structural details to avoid stress concentrations and an exhaustive control of the weld quality are some of the precautions most commonly required by the Classification Societies, based on their past experience and on in-service reports of existing ships.

During the MESTRAL design phase, the structural behaviour was monitored with appropriate simulation models and finite element analysis. Predicted motions of the vessel, in order to obtain the load transfer to the structure, were estimated with programs specifically developed for high Froude number values (DNVC "FASTSEA" program). Slamming pressures, vertical accelerations, bending moments and shear forces, as calculated by FASTSEA, were used with more stringent than Rule values resulting.

In order to obtain a comprehensive knowledge of the stress flow pattern, several finite element models were developed:

- A global 3D model for the overall sea loads. This model was also used to assess the dynamic behaviour of the ship and to provide information on forces or displacements used in more detailed models.
- A detailed 3D model of the machinery room bottom structure.
- A detailed 3D model of the deckhouse.
- A detailed 3D model of the waterjet compartment.
- Smaller models for the window rows, structure surrounding the lateral bow door, bottom girders, skeg, etc.

Besides these studies, a complementary and more rigorous approach from the fatigue point of view has been adopted to ensure the structural reliability of the ship. A specific programme has been implemented to quantify the fatigue phenomena. A damage - tolerance approach is being observed to ensure the level of accuracy with respect to the fatigue life of the structure.

The main objective of this programme is to monitor and control the fatigue behaviour of the most critical elements of the ship structure to ensure structural integrity during its life.

Four different phases have been identified in the programme:

1. Fatigue testing. All fatigue-related parameters are measured for different thicknesses, welding processes and welding zones. Influence of postwelding treatments has also been considered. More than 70 compact type fracture specimens have been tested.

2. <u>Residual stresses measurements</u>. Ultrasonic techniques have been used to determine the level of residual stresses that will normally occur during the ship construction. These stresses are of great importance for the correct consideration of the fatigue stress ratio.

3. <u>Stress monitoring</u>. Fatigue-critical components of the structure have been identified. A complete set of sensors (strain gauges, accelerometers, pressure transducers,...) were mounted during the ship construction, in order to monitor and record information on a portable data logger during service operation of the vessel.

4. <u>Evaluation of the fatigue life</u>. With the data collected in the previous phases, the fatigue life of the different components are able to be predicted. A set of recommendations have eventuated, comprising an inspection programme, corrective actions (where needed) and lessons learned.

   The same programme was envisaged for the first vessel of the series (ALBAYZIN) but since its delivery was moved two months forward, it was impossible to mount all the sensors, and only a small amount of them were installed.

   During the voyage of the ship from Cadiz to Wellington, information was registered on a data logger which post evaluation remains available to rectify any structural problems which may arise during the life of the vessel.

   As a result of all these parameters - simulation of loads, design with mathematical models and a fatigue life assessment programme - confidence in the structural design of the MESTRAL type vessel can be assured.

ALBAYZIN: THE FIRST MESTRAL

The first MESTRAL named "Albayzin" was delivered in October 1994, two months ahead of schedule to fulfil changed circumstances. "Albayzin" was the first of two vessels originally ordered by Trasmediterranea to operate between Barcelona and Palma de Mallorca, reducing the crossing time from 9.5 hours taken by conventional ferries, to 3.5 hours.

In the middle of October 1994, it was confirmed that "Albayzin" had been sold to Argentinean operator Buquebus, and chartered to Sea Shuttles in New Zealand to operate across the Cook Strait between Wellington and Picton.

Before its departure for New Zealand, sea trials were carried out in an extensive and detailed programme where it was verified that the ship's performance was in excess of contractural and anticipated requirements.

The lightship weight according to the inclining experiment results exactly matched the values initially estimated. The vertical centre of gravity margin was not exceeded during the construction, therefore the ship is able to fulfil the IMO wind stability criteria for navigation without restriction.

The maximum mean continuous speed at full load reached during the sea trials was 37 knots which results in more than one knot over the contractual speed. Minor modification in the appendage configuration (aft skegs, sea chest intakes, etc.) have produced in the second and third Mestrals, an increase of more than 1 knot over the original speed.

During the sea trials the ship motions at sea were found to be extremely smooth due to the excellent damping properties of the hull lines. The ship's ability to sustain service speed in waves 2.0/2.5 metres was amply demonstrated.

Noise level in bridge and passenger areas was very low and the recorded vibration level was well below the standard recommended values. All these factors, together with the vessel's superior accommodation and facilities guarantee a high level of passenger comfort on board. The spacious accommodation allows maximum passengers in complete comfort.

ALBAYZIN entered service on Cook Strait in December 1994 and proved to be a near ideal configuration for operation in often difficult conditions. The superior seakeeping, comfort and stability of her monohull design demonstrated that this concept is preferable to any multi-hull operation in the same area.

As with all prototype and initial concepts this is not to say that the design cannot be improved with further development and operational experience.

At time of compilation of this paper it is on record that three failures of equipment systems have occurred. These are:

a)	Reduction Gearboxes	(4)
b)	Stabiliser Fin	(1)
c)	Bow Thruster	(1)

It is apparent that these occurrences do not prejudice the basic excellence of the concept of the Mestral design.

There are several versions of the Mestral available depending upon requirements and further, an enlarged concept referred to as the ALHAMBRA design of some 124.70 m LOA and 40 knot service speed.

Delegates are informed that a range of product brochures are available. Those requiring some or all of this information, should ensure that their business cards are handed to Barry Coupland, Australasian Sales Representative, BAZAN Fast Ferries.

Juan C Sanchez
J Barry Coupland

07 November 1996

PAPER TWO

CONTINUING DEVELOPMENTS WITH THE MESTRAL CLASS OF HIGH SPEED FERRIES

"Monohull Excellent Seakeeping TRAnsport and Leisure"

Presentation Paper

J C Sanchez : BAZAN Head Office, Madrid : SPAIN
J B Coupland : BSL Barrian Group, Wellington : NEW ZEALAND

We are privileged to be able to present and commit to you our papers one and two on continuing developments with the Mestral Class of High Speed Ferries, and wish to compliment and congratulate the organisers and sponsors of Ausmarine '96 on their particularly auspicious theme and venue.

It is hoped that the context of our necessarily brief papers before you has given some insight into the "state of the art" Mestral Class of High Speed Ferries.

We will now develop further on aspects of, and progress with, the equipment failures as referred on the final page of Paper One.

Reduction Gearboxes:

In all four cases, failure occurred in way of the welded connection between the single helical input pinion and shaft, and its hollow section thrust shaft assembly.

Evaluations by the manufacturers and the classification society acknowledged unsuitable welding techniques and evidence of generally unsatisfactory manufacturing procedure and quality control.

In the redesigned assembly the input pinion shaft and thrust shaft is manufactured from a solid forging thus eliminating the need for any welded connections. The initial reasoning behind the original design was one of weight, however the redesigned solution still results in a gearbox of acceptable weight. Four new input pinion shaft assemblies were fitted to the first Mestral ALBAYZIN whilst in Wellington, New Zealand, following her withdrawal from service. Subsequently the vessel navigated the 12,000 nautical miles back to Spain and re-entered service at the beginning of July 1995 in the Straits of Gibraltar.

To date the vessel has completed not less than 6,500 operating hours of commercial service with her re-designed gearboxes without further problems. Therefore there are sound reasons for believing that the Mestral gearbox problems are now solely a matter for development history.

Stabiliser Fin

The first Mestral stabiliser system as briefly described on page four of paper one endured problems in two areas.

Firstly, two occurrences of the cast "Nylon" fins fracturing were experienced with the port forward assembly. Analysis by the manufacturers showed that the fracture was of a repetitive fatigue type, but not a problem of the material specification itself. The fin profiles were enlarged with increased scantlings of the high stress areas and new fins were manufactured and fitted.

Secondly, the stress loadings in way of the individual stabiliser unit mountings and their associated securing bolts were found to exceed the values used by the stabiliser manufacturers.

These areas were submitted to re-design and the resulting increase in scantling and specification has proved satisfactory, with no further problems being experienced to date.

Bow Thruster

On two subsequent occasions, fracture of the thruster propeller and support leg occurred.

The nature of the fractures under analysis showed that the cause was due to impact loadings and not by fatigue. The general tendency for this to occur is as a result of the thruster tunnel openings being devoid of any protective screen or grill with the consequent tendency for fouling of the propeller with flotsam. The propeller blade and boss assembly material specification has been altered from that of cast aluminium to that of bronze.

To date no further failures have occurred and in addition, the provision of a suitable tunnel grill or screen is under study, with the first requirement being that of continued thruster efficiency.

ALBAYZIN: The First Mestral

Following her return to Spain from New Zealand, the vessel was operated in the Straits of Gibraltar over the European summer of 1995 without any further downtime or mechanical problems. Subsequently, the vessel relocated to South America where she performed successfully for owners "Buquebus" on the "River Plate" between Argentina and Uruguay over the period November 1995 to February 1996. From April 1996 until the end of September 1996 she has been operated by "Larvik Line" between Denmark and Norway. Bad weather, combined with a requirement the Danish Authorities to carry out operating tests of the evacuation system in wave heights of two metres, has resulted in some delays to her schedule.

ALCANTARA: The Second Mestral

ALCANTARA is a sister ship of ALBAYZIN and predictably has commenced her service life with all of the developmental experience with ALBAYZIN factored into her build.

She enjoys all of the gearbox, stabiliser and thruster developments, and in addition is fitted with a higher specification stabilising system as briefly mentioned on page four of the paper.

Basically the forward pair of fins have been replaced by hydro-foil type "flap" assemblies.

The provision of these has significantly improved her ride and handling characteristics, and has resulted in a trial speed of 38.5 knots being achieved - up from 37 knots as in the case of ALBAYZIN.

Alcantara entered service on 4 June 1995 between the Balearic Islands and the mainland of Spain. She has had one stoppage due to minor mechanical problems which resulted in a delay of one day. The only other disruptions to her schedule have been due to bad weather during the winder. To date she has completed not less than 4000 operating hours of commercial service and has proven to be a very successful and pleasing vessel for her owners, operators and crew.

ALMUDAINA: The Third Mestral

Almudaina entered service during the European summer in June 1996 and has been operating on the same route as Alcantara. To date she has completed not less than 1400 operating hours of commercial service and has had only one trip lost due to minor mechanical problems.

All three Mestrals have operated with sound commercial success over two Northern summer seasons with the average passenger and freight loadings varying between 70-75% or better. The owners and operators of these vessels (Trasmediterranea) are currently negotiating with BAZAN for further derivatives of the Mestral design.

ALHAMBRA: The Giant Mestral

The first Alhambra named "Silvia Ana" by her South American owners "Buquebus" was launched in July 1996 and handed over at the end of October 1996. Already Buquebus have contracted with BAZAN for the supply of a second Alhambra which should take about 18 months to reach fruition.

ALHAMBRA is a similar but larger concept to ALBAYZIN, in that the main characteristics of deep-V monohull, entire aluminium construction and conventional diesel propulsion are maintained.

Similar IMO and Classification Society requirements are complied with and in addition compliance for B category vessels is achieved.

The following main particulars show the comparison between the ALHAMBRA and Mestral vessels and the formers "economies of scale" are well evidenced.

	Alhambra	Mestral	Coefficient of Increase
Length Overall (m)	125.0	95.0	1.32
Maximum Breadth (m)	18.7	14.6	1.28
Depth (m)	11.2	8.9	1.30
Deadweight (t)	475.0	175.0	2.71
Full load draught (m)	2.45	2.05	1.20
Propulsion power (kw)	33,900	20,000	1.70
Trial speed (kn)	40.0	36.0	1.11
Endurance (nm)	400	300	1.33
Passengers	1,250	450	2.78
Cars	244	76	3.21

Commercial Areas

ALHAMBRA is divided into four decks, the upper two for passenger accommodation and the lower two for vehicles.

Nine transverse bulkheads subdivide the vessel longitudinally and the Bridge has been elevated above the passengers upper level to provide for a 360° vantage.

The main level of the continuous garage area has a capacity for 136 cars and the upper level for 108 cars.

The aft area of the garage is available with double tweendeck height in order to stow up to four trucks/buses of twelve metre length, with a maximum axle load of 10 tonne.

In order to minimise the turn-round time between payloads, external accesses to the garage as well as the intercommunication ramps between levels provide independent and simultaneous operation of both garage levels.

Shopping and food/beverage outlets are conveniently located within the accommodation areas and passenger access from the garage to accommodation area is conveniently and efficiently provided by forward, midships, and aft located stairwells.

Passenger comfort and amenities of a very high standard are provided which include first class and economy areas which are well served with abundant toilet facilities.

Navigation

The vessel may be operated entirely from the bridge and hence is fitted with an integrated monitoring control and metering system which in itself is largely a development of the Mestral's ALBAYZIN and ALCANTARA.

Propulsion

The machinery plant is located throughout the aft most three watertight compartments, the forward two containing the engine rooms and the aft the KaMewa Waterjet Units.

The propulsion machinery consists of five propulsion units, four on the sides and one in the centre. Each lateral propulsion unit consists of one Caterpillar 3616 diesel engine which drives a steerable and reversible KaMeWa 112S waterjet intercoupled by a direct drive reduction gearbox. The central propulsion unit which serves as a booster, available in the ahead mode only, consists of two Caterpillar 3616 engines, which drive a KaMeWa 140B waterjet intercoupled by a direct drive double input and single output reduction gearbox.

Two sets of three diesel generators of 315kw output each, provide the required auxiliary power. In addition an emergency genset which is able to supply all essential equipment is located on the main deck.

Hull Form

Based on the Mestral, the hull form has been developed with the assistance of model tests and in conjunction with Bazan's "MOLAS" computer program.

Head Seas: Regular Waves

Data is collated at sections 5 and 10 (from a total of 20 over the hull length comprising of 10 compartments) so as to give a result representative of the passenger region.

The calculated seakeeping characteristics simulated for the full load condition at 35 knots, and in head seas of height 2.0-3.25m are:

Table 1

Preliminary ALHAMBRA's seakeeping characteristics head sea, full load, 35 knots (significant values)

Sea State	4	5
Wave Height (m)	2.0	3.25
Pitch Angle (degrees)	0.4	1.0
Vertical Acceleration Section 5 (m/s^2)	0.45	1.15
Vertical Acceleration (m/s^2) Section 10 (m/s^2)	0.93	1.95

Head Seas: Irregular Waves

Model tests in head seas of irregular waves have also been carried out utilising the following parameters:

Table 2

Parameters

Sea state	Hs(m)	Tm(s)	Tz(s)
4	2.00	6.5	4.6
5	3.25	7.6	5.4
6	4.50	9.5	6.8

Legend
Hs:- Significant Wave Height
Tm:- Modal Period
Tz:- Mean Zero-Crossing Period
RMS:- Root Mean Square

The following vertical accelerations results, similarly recorded at sections 5 and 10, have been obtained:

Table 3

Results of Model Tests in Irregular Waves

Sea State	**4**	**5**	**6**
RMS acc. at section 5 (m/s^2)	0.193	0.585	1.290
RMS acc. at section 10 (m/s^2)	0.294	0.996	2.006

As is apparent, results at lesser sea states (SS4) compare with results deduced from tests of regular waves, however, at increased sea states (SS5 and SS6) evident non linear effects produce differences between the results of the tests of regular and irregular waves.

With the Ride Control Stabilisation System (RCS) in operation and applying the Alcantara correlation factors existing between the model test and results of her sea trials, the following sea trials results can be anticipated for the *Alhambra* project.

Table 4

Anticipated Seakeeping on Sea-Trials with RCS in operation

Sea State	**4**	**5**	**6**
RMS acc. at section 5 (m/s^2)	0.208	0.631	1.393
RMS acc. at section 10 (m/s^2)	0.209	0.706	1.422

Note that the utilisation of the *Ride Control System* achieves the very desirable effect of regularising the vertical accelerations along the length of the vessel.

Using the preceding data as a conservative basis, it can reasonably be anticipated that the passenger comfort characteristics are 4 times better than the already very satisfactory Mestral.

NB 1:
In summary the vertical accelerations established for *Alhambra* are half of those for *Albayzin* hence it is deduced that passenger comfort will be enhanced "four-fold".

NB 2:
Note that the degree of passenger **comfort** is proportional to the square of the vertical acceleration, therefore if the acceleration is doubled then the degree of passenger **discomfort** is quadrupled.

General Comment

BAZAN has entered into a licence agreement with Advanced Multihull Designs Pty Ltd (AMD) of Sydney whereby the designs of AMD are utilised for BAZAN to develop the project and build the vessel.

Currently BAZAN's catalogue includes two such AMD designs, the B40 and B60 catamarans. Buquebus of Argentina have two B60 vessels on order, the first contract commenced on 1 July 1996 for delivery at the end of 1997 and the second to follow subsequently.

Buquebus also have ordered the second of the Alhambra designs from BAZAN and at time of compilation of this paper the build schedule and delivery date has yet to be finalised.

The time of presentation of this paper almost exactly coincides with the delivery of the first Alhambra to Buquebus and it is regretted that full details of her trials performance are not quite available. However the vessel has achieved or exceeded all of her contractural design and performance criteria.

Subsequent to the completion of the first six months of commercial service Alhambra will be the subject of a further paper dealing selectively with that design which we hope to present next year 1997.

For the interest of delegates BAZAN's current fast ferry product catalogue is tabled below.

Of the five basic designs tabulated, the SERVAL version has been evolved as an intermediate vessel between the Mestral and Alhambra. In addition derivatives of the individual designs are also available to suit owner's requirements.

	MESTRAL	SERVAL	ALHAMBRA	B-60	B-40
Passengers	450	700	1,250	450	350
Cars	80	170	250	50	46
Buses	-	6	4	-	-
Propulsion plant	4 Cat. 3616	4 RK270	6 Cat 3616	2 ABB GT35	2 Cat 3616
Power (Kw)	22.600	27.000	33.900	32.000	10.900
Speed (Knots)	38	37	38	57	38.5
Delivery time	12 months	18 months	18 months	12 months	12 months

Further information is available as regards the commercial connotations of the range, should any delegates so wish to enquire.

Written questions on the subject matter are welcomed and will be dealt with in writing as soon as possible, and if necessary by facsimile.

We recommend to you the Mestral family of High Speed Ferries as **the** solution to their sector of the demanding requirements of commercial shipping for the future.

Thank you for your attention and the opportunity to present our papers One and Two.

Juan C Sanchez
J Barry Coupland

07 November 1996

Paper No 2

Paul McGrath

Chief Executive, Australian Maritime Safety Authority

"AMSA's Role in Maritime Safety"

AMSA'S ROLE IN MARITIME SAFETY

Address to the Technical Forum at Ausmarine '96

Fremantle, 7 November 1996

Paul McGrath
Chief Executive
Australian Maritime Safety Authority

The maritime industry is truly international. The historic right of shipping to ply for trade and to be granted both right of free passage and assistance in times of need are concepts that have influenced the nature of shipping over the centuries. Add to that the dispersed and relatively independent nature of shipbuilding, the enormous range of vessels produced, the variety of commercial and operational interfaces and the vital impact of shipping on national economies, and the result is a very complex international industry.

In some ways the shipping industry can be regarded as a major component of international infrastructure. As with national infrastructure, it also has the major problems associated with the magnitude of capital costs, the need to ensure good economic returns, the requirement to maintain assets in good order and condition, and eventually to provide the ability to replace them.

Then there is the question of accountability. In the case of national infrastructure, if the challenges mentioned above are not met, the responsible government is brought to account by the people. But in regard to shipping, which government is accountable, and who represents the ordinary citizen? In fact, lack of true accountability is the source of many of the problems that the industry is experiencing nowadays.

For many centuries the control of shipping was left in the hands of individual governments or, more likely, individual shipowners. Such an approach did not work, and today the control of shipping is recognised as being an international responsibility.

But is the current system working? If it is, why are so many governments and so many concerned organisations and individuals demanding better performance? Human nature being what it is, there will always be people and organisations prepared to avoid responsibility for reasons of personal or corporate gain. This also applies to sovereign governments which often turn a blind eye to the problems for narrow national advantages.

We now accept international regulation through the International Maritime Organization (IMO). However, we all understand and tacitly accept the "lowest common denominator" nature of IMO outcomes, that 'lip service' is paid by some to IMO regulations and that blatant self interest still drives certain elements of the maritime industry. The real problem is that the international regime has virtually no capacity for sanctions to be applied to countries that flaunt the rules.

In discussing the Australian perspective, I clearly recognise that Australia is not a major shipowning nation; that it is, at best, a medium power with enormous dependence on shipping for trade. It is therefore essential we understand the real issues and produce outcomes that are philosophically sound, but at the same time politically acceptable and industrially pragmatic.

The Priorities of Government

For any regulator, the policies and concerns of its owning government should be clearly enunciated and implemented. I am confident that the current status of the Australian Government's approach can be summed up relatively simply. This listing is the AMSA interpretation.

Australia is a developed country in which human life has highest value. This is applied in a non-discriminatory way, regardless of colour, creed or nationality.

Following large losses of bulk shipping some five to six years ago, Australia reacted to the loss of lives of seafarers. To Australia, the nationality or personal characteristics of the sailor is not important; the worth of the sailor's life and wellbeing is of paramount importance. Hence:

Priority 1 is -To protect the life and wellbeing of all involved in the industry.

Australia is an island nation. We have some 36,000 km of coastline to look after. Some of this coastline is without doubt among the most spectacular and environmentally sensitive on earth. Australians place highest possible values on the sea and the coastal environment. But not only is the coastline recognised from an environmental viewpoint: it is also important to maritime trade, resource exploitation, tourism, recreation and related perspectives.

The marine environment must be afforded the highest protection. Hence:

Priority 2 is - To protect and enhance the quality of the marine environment.

Recognising the truly international nature of shipping, and accepting the fact that this country is unlikely ever to be able to exercise sufficient economic power to directly control international standards, we must look to effective use of the mechanisms available and how best our interests can be protected. Australia cannot succeed in isolation from the world maritime community, and hence:

Priority 3 is - To work within and influence the International Maritime Organization system to the greatest extent possible.

Australia is blessed with abundant resources. To a great extent, we owe our 'first world' standards of living to the value of our commodities and other exports. We are particularly aware that foreign flagged vessels carry more than 95% of Australian cargoes. The overall aim must be to achieve improvement in shipping in a manner which does not adversely impact on our trading position and in a way that provides confidence to the interests with quality vessels that they will be treated fairly in Australia. Hence:

Priority 4 is - To seek to improve maritime industry performance and at the same time to facilitate Australian trade.

Finally, in addressing priorities, the cost of Government regulation and services cannot be overlooked especially as the cost of shipping is reflected directly in the prices of both exported and imported goods. Hence for the maritime regulator:

Priority 5 is - To minimise the cost of regulation and safety services and to recognise the inter-relationship between the maritime industry and the wider economy.

I could clearly develop further priorities, but the five identified should provide an adequate basis for an understanding of the Australian Maritime Safety Authority's objectives.

The Role of the Australian Maritime Safety Authority

The Australian Maritime Safety Authority (AMSA) was created in 1991. The broad scope of the functions of AMSA as enshrined in the enabling legislation was to:

- combat pollution in the marine environment
- provide a search and rescue service
- provide, on request, services to the maritime industry on a commercial basis.

There was no direct mention of the safety role for which AMSA is accountable. However, in May 1995, the Government announced a number of changes to AMSA's corporate structure and method of operating. The AMSA Act has been changed, to reflect the primacy of AMSA's safety objective. While AMSA was originally classed as a Government Business Enterprise, with strong emphasis on commercial objectives, the Government changed AMSA into a self-funded Government safety agency, with its primary objectives relating to safety and marine environment protection, rather than commercial performance. I believe this to be a very pragmatic and appropriate political decision, which means

AMSA's safety and marine environment protection obligations are enshrined as Government policy and explicitly spelt out. Although we had a change of Government in March 1996, the new Government has confirmed its acceptance of the operating environment of AMSA.

Let me further address the authority for AMSA's regulatory function. At the time of creation of the Authority, the Government also changed the *Navigation Act 1912*. This change transferred to AMSA the legislative responsibility for administration of the Navigation Act. We do not work under Ministerial delegation; AMSA, from the Board down, is legislatively accountable for the maritime safety function. AMSA is the Government's regulator - this must be recognised as the ultimate responsibility, and it should always be recognised by those dealing with AMSA: that we see this as our ultimate responsibility to our owning Government and to the Australian people.

Given that AMSA is the maritime regulator, the next question is: "How does the Authority perform this role?"

One way is to be remote from the industry, to concentrate on bureaucratic methods, to administer according to the written rules and to focus on the processes that are often based on history rather than contemporary industry needs.

The regulatory role does not have to be carried out in this way. Indeed, in my opinion, such an approach is not only unlikely to be acceptable from an industry viewpoint, but is also unlikely to achieve effective regulation. The effective modern regulator must have a professional approach to the role, an understanding of the rationale behind the rules, a responsiveness and sensitivity to the industry. It must also accept that it must be responsive to both Government and the industry. However, the ultimate responsibility of the regulator is to Government and the people it represents - a point that industry should always understand. It is particularly important that the regulator avoids, and is seen to avoid, any suggestion of "industry capture".

AMSA has concentrated on becoming an integral element of the maritime industry. We have tried hard to understand all relevant viewpoints, to improve the consultation and communications arrangements, to assist in resolving major issues and, in general terms, to work with industry in a way which does not compromise our regulatory accountability. We have tried particularly to be effective in both our domestic and international roles.

We believe we have been reasonably successful in this approach. but there is still room for improvement. The Government administrative arrangements for AMSA differ significantly from those existing in most other maritime administrations. We believe they can improve substantially the way in which the industry is regulated.

The Relevance Of The Maritime Industry To Australia

Australia has some interesting characteristics which differentiate it from other nations and which influence maritime regulation.

We are an island nation, so international road and rail competition does not exist. Air transportation is only viable for some cargoes. Hence, more than most other nations, we have a very significant dependence on maritime trade.

Australia is remote from most of its export markets - our trade is relatively long haul. Many of our competitors are much better placed geographically and the transport costs of their exports tend to be less significant.

Australia's export products are often low value, high bulk and high density commodities subject to intense competition internationally, such as iron ore and coal. Transport cost is a significant pricing component and transport can be a major factor in obtaining and retaining export markets. Regulatory action taken in Australia must be cogniscent of the potential for there to be significant financial or economic penalty associated with it.

More than 95% of all maritime cargoes are carried in foreign flagged vessels, thus denying Australia direct control over standards. Hence our actions must be directed towards ensuring adequate international standards.

Ideally, the best situation for Australia would involve international enforcement of standards, reasonable economic returns for shipowners, acceptance by flag states of their responsibilities and equitable treatment of shipping in all countries. These are high ideals which are unlikely to be attainable in the near future, if ever.

The Australian Approach

Efficient and effective maritime trade is an important requirement for best economic performance. However on the international scene, Australia is not a major player. This is in spite of Australia ranking about fifth internationally in terms of a km-tonne measure.

The Australian fleet is less than 1% of the international fleet. Australian flag vessels are of high quality, modern, well managed and with competent crews. The cost of operations is generally high, even though Australian crew numbers are less than the OECD average. There would appear to be little prospect of substantial expansion of the Australian flag in the near future.

Given that more than 95% of our cargoes are carried in foreign-flagged vessels, there is a need to ensure adequate quality of these ships.

The most pragmatic way for Australia to achieve its objectives is to work within and influence the international regulatory regime which is based on the International Maritime Organization (IMO).

Australia is proactive within the IMO, and seeks to identify problems, propose solutions and gain cooperation from countries and administrations with similarly high standards. In the international arena, Australia and its shipping industry are generally regarded as being competent and prepared to work to satisfy legitimate needs. We are free of the vested interests of some other countries with a high profile in IMO. Consequently we are often given the chance to act as 'honest brokers'. We intend to do everything possible to protect and enhance this reputation, as it is likely to be in the best interests not only of Australia but of the more conscientious players in the international shipping industry.

"Ships of Shame"

The Australian approach and attitude has been influenced by the "Ships of Shame" report, released by the House of Representatives Committee on Transport, Communications and Infrastructure in December 1992. The Committee learned, among a host of indictments, of the operation of unseaworthy ships, the use of poorly trained crews and false qualification papers; of crews being unable to communicate with each other or with Australian pilots; of flag states failing to carry out their responsibilities under international maritime conventions, and of inadequate, deficient and poorly maintained safety and rescue equipment. The report was accepted by the Government and received general support in the Parliament. It has been shown beyond any doubt the Federal Government, regardless of political persuasion, would support the broad conclusions of "Ships of Shame" and their influence on the current directions being taken by AMSA.

The report of the Committee, which was chaired by Peter Morris, has been accepted as a realistic, non-political assessment of the major problems facing the shipping industry today.

It is also interesting to note the parallels between "Ships of Shame" and the European Commission document "A Common Policy for Safe Seas". In broad terms, the policies of the European Commission are consistent with those of Australia and there is wide accord in other countries that take pride in the quality of their shipping operations.

Put simply, "Ships of Shame" is the backdrop to AMSA's approach. Unfortunately, even an optimist would have to predict that any change for the better will be slow and will not be uniform. While our primary objective is to protect Australia's interests, the problems are international. They must be resolved through concerted international action.

Working with Industry and Unions

AMSA sees a direct benefit in working closely with industry while retaining our statutory independence. This is essential if we are to understand the real issues, identify the most appropriate outcomes and obtain the support of our government and of other governments aspiring to highest standards.

The consultation process with industry is very important. Meaningful consultation can often be very difficult, particularly for a public sector body, but it is not impossible. We must be prepared to be interested in problems, to listen carefully, to work to achieve solutions and, where it is beneficial, to involve industry .

It is relevant to note that the influence of industry in regulatory matters has become greater recently. The extent of direct industry participation in international forums, particularly IMO, proves this.

While there is a need to establish the Australian position before IMO and other international meetings, and to require industry acceptance and compliance if they wish to participate, it is often not difficult to do so. Industry members on our delegations increase our ability to achieve good outcomes. Interaction with AMSA provides industry with a real insight into the way international regulation works, and the problems we face internationally. It builds mutual respect that is essential if Australia's position is to be optimised.

In talking about industry, we also should think about the workers in it and their representative unions. We have been able to work successfully with the unions. Many beneficial changes in the industry over the last decade have been due to the forward thinking and pragmatic attitudes of the unions and their understanding of the major issues that face them and the nation.

In certain of our functions, AMSA has a mediating role between industry and unions. This role requires a careful balance and an honesty to all parties, a toughness to stand up for the correct outcomes and an ability to bring together parties who otherwise might adopt confrontation as a normal approach. We value this role, and will work to ensure that our power to influence in the national interest remains a primary objective.

The Coalition Government has announced a number of major initiatives to improve maritime industry performance. While many of the details are yet to be worked out, it is obvious that some will impact on the operations of AMSA. The AMSA position will be very simple - we exist to serve the Government of the day and will accept any related challenge in an objective and professional manner. At the same time, however, we will strive to maintain positive relationships with all sections of the industry and to use our influence to achieve outcomes that represent best achievable in the circumstances.

Specific Action

AMSA tries to ensure the widest influence in its operation. In so doing there are a number of activities that are specifically directed at improving the performance of an industry that over the last few years could be described as being in crisis. I only intend in this paper to address a few.

Port State Control

Much has been said and done recently in regard to action to protect countries and seafarers from the consequences of unscrupulous shipowners. One major weapon has been port State control, to enforce minimum acceptable standards.

Australia has a reputation for efficiency in port State control. Our inspection rates are among the highest in the world and we target those flags and vessels where problems are prevalent. We pay particular attention to safety equipment, structural aspects and seafarers' conditions, with the aim of protecting life and preventing pollution.

We are sometimes portrayed as too conscientious and too ready to detain a foreign vessel. I can assure everyone that we are more conscientious in relation to our own flag. Fortunately the Australian industry has a good reputation. The Australian Shipowners' Association and AMSA have signed a "Code of Conduct" which confirms the joint approach to appropriate standards being adopted in Australia.

The ships we detain receive significant media attention. However, the media focus on the negative aspects of the industry and often do not recognise the performance of the reputable players - the great majority of shipowners.

The number of ships detained since the vesting of AMSA are: 1991 - 27; 1992 - 61; 1993 - 72; 1994 -153, 1995 - 244, 1996 to 12/9/96 - 184.

While these numbers are significant, they constitute less than 2% of visiting ships. And of this 2%, a relatively high proportion are able to effect repairs before the scheduled departure time, and hence are not delayed.

The percentage of vessels with faults is high. But, in most cases the faults are not serious and can be fixed relatively quickly while the vessel loads.

There is one element of this that continues to surprise me. We are, and will continue to be, efficient in port State control. It is a primary tool to enforce standards. Substandard vessels have a very good chance of being detected and detained. Common sense would suggest to me that the owners of substandard ships would avoid Australia. Why then do we have so many ships detained?

The chances of substandard ships being allowed to trade with Australia without detection are already poor, and we intend to reduce these chances even further. We make no apologies for this. We will continue to enhance port State control as one of the most effective means of making our position obvious to the international maritime community.

The Human Element

Port State control, by its nature, tends to concentrate on physical defects in ships. However, at least 80% of accidents arise from failings of the humans involved. If we extend human error to lack of maintenance, this figure will approach 100%. Two specific international initiatives recently came to fruition and are designed to focus on the human element. These are particularly important to AMSA.

The first is the International Safety Management Code. This underscores the fundamental principle that safety involves all with an interest in the ship, not just the master and crew. The objective is to ensure that each company owning or operating a ship has in place a safety management system which identifies a person in the shore organisation responsible for monitoring the safety and pollution prevention aspects of each ship. He or she also has the task of ensuring that manpower resources are appropriate in terms of numbers, qualifications, experience and training, and that other resources enabling all operations to be carried out safely are adequate.

The safety system addresses compliance with statutory requirements and implements the broader safety and environment protection policies of the company. Guidelines have been developed by industry to complement the Code and to identify areas of normal and safety-critical operations that the safety system needs to address.

Representatives of the Australian maritime industry played a significant role in developing the Code at IMO. In May 1994, the Code was made mandatory for all ships coming within the ambit of the Safety of Life at Sea Convention (SOLAS). It departs from normal maritime practice in requiring that the company, as well as the ships it operates, be issued with appropriate certification.

There will be a phased entry into force of the Safety Management Code. Passenger ships, tankers and bulk carriers will be required to comply by 1998. The relatively long lead time was agreed to within IMO in recognition of the massive amount of work necessary to achieve compliance. The major part of the Australian industry was charged with implementing the Code on a voluntary basis from the middle of 1995.

Another facet of the IMO human element initiatives relates back to port State control. It is "control of operational requirements related to the safety of ships and pollution prevention"; in short, crew competence. Anyone with an interest in ship safety recognises the importance of well trained, competent and contented crews.

Guidelines have been agreed on to assist port States in testing the competence of crews of foreign ships in carrying out their duties in accordance with good and effective operational standards and on-board procedures. The tests cover a range of matters such as a familiarity of crew members with their duties, effective conduct of fire and emergency drills, familiarity of navigation officers with bridge procedures and equipment, and the ability of engineer officers to operate machinery and emergency equipment.

Countries such a Australia have been performing such assessments as part of the control function for many years, especially on passenger ships and tankers. The new guidelines now formalise and standardise these practices, and will be helpful in introducing uniformity so that they can be applied consistently across the world.

It is anticipated that the new focus on the human element of safety and pollution prevention will make a significant contribution to the reduction of accidents and, more importantly, to a reduction in loss of life and injuries to seafarers.

On the subject of pollution, let me now refer briefly to AMSA's Marine Environment Protection Services.

Through this Business Unit, AMSA administers the National Plan to Combat Pollution of the Sea by Oil and has the responsibility, based on international conventions, for minimising the threat of ship-sourced pollution. The National Plan is a combined Commonwealth/State/Oil Industry cooperative arrangement, with funding provided through AMSA's Protection of the Sea levy on all vessels carrying more than 10 tonnes of oil.

AMSA's environmental management responsibilities both at sea and ashore may therefore be considered in three stages - prevention, preparedness and response.

The National Plan was originally developed and implemented so as to provide a viable response to the threat of ship sourced marine oil pollution. This followed the OCEANIC GRANDEUR incident in Torres Strait in 1970, in which 1400 tonnes of oil was lost. The Plan has been in operation since 1973. It involves Commonwealth, State and Territory governments and industry organisations, and provides an effective response to oil pollution incidents in the marine environment. It also manages associated funding, equipment and training programs to support National Plan activities.

In summary, the National Plan involves competent national and local authorities in maintaining and implementing:

- a national contingency plan for preparedness and response which incorporates the operational relationship of the various organisations involved, whether public or private;
- an adequate level of pre-positioned oil spill response equipment, commensurate with the risk involved, and programs for its use;
- a comprehensive national training program designed to familiarise personnel at all levels with the strategies involved in planning for and responding to the needs arising from an oil spill. This program includes the conducting of frequent exercises;
- detailed national, state, local and industry contingency plans and communications arrangements for mobilising resources and responding to an oil pollution incident;
- an awareness by governments, media and the community generally of the limitations inherent in a response to a major spill, with particular emphasis on the need to accept that, other than in exceptional circumstances, there is not currently available the technology to prevent weather driven oil coming ashore on a coastline - and neither is such a solution likely in the near future.

The most significant factor in oil spill response is having the right equipment available. Assessment of equipment requirements for the National Plan is based on a tiered response, as follows:

- tier 1, oil spill of less than 10 tonnes
- tier 2, oil spill of 10 to 1,000 tonnes
- tier 3, oil spill greater than 1,000 tonnes.

Under present National plan arrangements, AMSA provides oil spill response equipment to States and the Northern Territory in the Tier One and Tier Two categories. The oil industry provides equipment of Tier 1 level in oil terminal ports, which may be accessed under National Plan and Marine Oil Spills Action Plan arrangements. In areas where it is justified, AMSA supplies equipment required to mount an initial response capability for a Tier Three spill. This latter requirement provides an initial response until equipment can be brought in from other National Plan or industry sources. These include the Australian Marine Oil Spill Centre at Geelong, recognised as the principal resource centre for a Tier Three spill anywhere in Australia.

National Plan Tier 1, 2 and 3 equipment is strategically located at 37 ports around the Australian coastline. Oil spill risk is taken into account in the location and composition of the National Plan resources centres. A major review, concluded in 1993, assessed these five areas as those of highest oil spill risk. This assessment was based on three factors: the environmental sensitivity of the area, traffic density and hazards to navigation.

Historically, major spills occurring in the coastal zone have resulted world-wide in oil impacting the foreshore. There is no current oil spill response technology in existence that prevents oil spilt in these circumstances from reaching the coastline. However, strategically placed equipment can minimise the effects of a spill by protecting sensitive resources such as mangrove stands, salt marshes, or bird or other animal breeding and nesting sites.

Additionally, AMSA in cooperation with the oil industry is finalising an aerial dispersant spraying contract which will increase response capability and improve response times in certain instances.

We have also developed Memoranda of Understanding with neighbouring countries to ensure enhanced capacity to respond regionally.

Conclusion

In this paper I have set out the factors influencing the Australian approach to maritime regulation. The action we are taking is not directed towards AMSA's self interest, but demonstrates the genuine commitment Australia has to making the maritime industry safer and the seas cleaner.

The clear and consistent message is that reputable shipowners have nothing to fear from the Australian regulator. For the vast majority of shipowners, all we need is continuation of their commitment to quality operations. For the substandard or non-committed owners I would suggest that they either improve their standards before coming here or, alternatively, stay away.

For the pariahs of the industry, Australia will work here and in the international arena with reputable nations to rid the oceans of the rust buckets and/or incompetently crewed "Ships of Shame". With the extension of regional port State control cooperation, we hope to make even the vast oceans too small for poor ships to operate in.

Finally, for the Australian industry, we will seek to maintain standards and to assist wherever reasonable. We see the future of the Australian Maritime Safety Authority as being integral to a quality international industry, with our own fleet at the forefront.

AMSA wants to be accepted and recognised as a competent regulator, with the best interests of the industry and of Australia at heart. We will not, however, equate acceptance with anything less than appropriate quality standards.

AMSA may be somewhat idealistic, but in my mind the fundamental prerequisite is the vision of a quality industry. We ask all of like vision to work together with us.

Paper No 3

Professor Mike Davis

Propulsion Program Leader, AMECRC

"The Fluid Dynamics of Water Jet Propulsion Units"

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THE FLUID DYNAMICS OF WATER JET PROPULSION UNITS

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1. INTRODUCTION

In order to circumvent the difficulties associated with operation of conventional propellers in high speed flows water jets have now been widely adopted as the normal propulsion system for high speed vessels. Whilst there have been suggestions made that the use of propellers might be reconsidered as a possible option in some cases, depending upon speed and power of course, it seems that the water jet will continue to be selected for the large high speed ferries now entering service in increasing numbers. Vessel designs exhibits a surprisingly wide range of configurations, but in general the overall power requirement would be expected to follow the payload and speed of the vessel. Figure 1 summarises this relation and includes a number of vessel designs including monohulls and catamarans with varying hull forms. This extends from conventional high speed designs for Froude numbers between 0.5 and 1.0 approximately to semi-SWATH configurations designed to reduce wave response. Speed and configuration contribute to the variability of outcome, as also does the efficiency of the structural design, and it should be emphasised that criteria such as speed and wave response may feature substantially in the optimal design for a particular requirement. Thus a higher power/payload ratio may often be quite acceptable in exchange for other benefits. The range of power/speed outcomes illustrated in Figure 1 shows clearly that other performance issues may often take precedence to some extent.

A marine propulsion water jet is essentially a mobile pump and is illustrated schematically in Figure 2. It is the mobile nature of the installation that has presented the greatest departure from the fixed pump conditions and which has thus been the source of most practical difficulties given that the design of fixed pumps is a highly developed art. The influence of pump mobility centres primarily on the fluid dynamics of the intake but also extends to the inlet flow interaction with the rotor in a number of critical areas. Downstream of the rotor, the flow in the stator/flow straightener and nozzle sections of the unit tend to be much less influenced by intake or vessel motion effects and are thus approached in terms of well established blade and nozzle design without undue difficulty. Nevertheless, there are some issues that attach to the aft end design of water jet installations which do need special consideration, such as the elevation of the nozzle centreline. Finally, it must be realised that the characteristics of the water jet under vessel operating conditions ranging from dockside manoeuvring to full speed cruising are all of importance to the operator. This further complicates the balancing of priorities in the design process.

Whilst the responsibility for the water jet selection and detail design clearly rests with the water jet supplier, the requirements of a water jet are much more complex than is the case for a fixed pump installation. The unit must perform over a full range of vessel operations including speed, water depth, load, trim, wind and sea conditions and so it is much more difficult for the performance specification to be comprehensive. Complexities in design prediction arise not only from the nature of the interaction of the jet with the inlet flow but also from the considerable complexity of the total vessel fluid dynamics, involving such matters as hull boundary layer development, vessel wave making and dynamic motion. In this situation it is clearly important that all partners in a total project have a good understanding of the technical aspects of the performance of water jet units as installed and operated. If this understanding does not exist then the potential for difficulties to emerge is appreciable and it is clear that all participants need to contribute to the ongoing research and development task.

The aim of this paper is to give a broad overview of the range of issues relating to the operation of water jets, and to focus on particular aspects which have presented real difficulties. The industry has been following a rapid growth curve in terms of performance that has by no means levelled off as Figure 1 illustrates. The potential for the re-emergence of problems that have been resolved at smaller unit size is appreciable as many of these problems have been essentially size related. The search for generality may not bear great fruit under such circumstances involving a variety of issues, ranging from structural vibration to turbulent fluid motion, but the lessons of the past can at least provide broad guidance for the future and point to areas where research and development may be beneficial.

2. OVERALL UNIT SELECTION

2.1 Overall performance considerations

As with any pump the general configuration of the unit will depend upon the power, flow rate through the pump and shaft rotational speed. These overall parameters are combined to form the specific speed (N_s) given in dimensionless form by

$$N_s = NQ^{0.5} / (gH)^{0.75} \qquad (1)$$

where N is the shaft rotational speed (rad/sec), Q the volumetric flow rate (m^3/sec), g is gravitational acceleration (m/sec^2) and H is the total head rise across the pump (m). Treating the specific speed on this basis as a dimensionless quantity in terms of the angular velocity of the shaft it is generally found that the optimal design of a pump changes from a mixed radial/axial flow design to an axial design at a specific speed of approximately 3.0, higher specific speeds requiring an axial design for optimal performance. It should be noted that a considerable variety of other dimensional definitions of specific speeds are often used in the pump industry, but these often tend to confuse the overall situation. For a typical large ferry application the values of the parameters involved might be:

Q = 15 m^3/sec, N= 75 rad/sec (ie. about 700 rpm),
Shaft power = 6 MW and Pump head rise H= 35m .

This leads to a specific speed of N_s = 3.64. This outcome shows that the design will lie close to the conventional pump parameters for which the optimal choice will only just fall in the range of the axial flow pump. We can thus not expect a clear cut optimal design problem and that the actual outcome may depend upon a variety of other considerations. It is therefore not surprising that there is appreciable variation in designs employed in practice, ranging from almost direct axial flow to units which have appreciable radial flow with corresponding centrifugal components in their pressure rise. Each design, whilst giving generally good fluid dynamic performance, will present its own particular benefits. On the whole the peak efficiency is not highly sensitive to specific speed and thus the penalty for varying the flow configuration within reasonable limits is not too severe.

Of course the question of what defines optimal performance may vary, and if unit size or weight is a factor then the optimal decision may as a consequence be shifted. In terms of these aspects this is usually more towards the selection of a machine general design that in fluid dynamic terms is better suited to a lower specific speed than that resulting from the power, mass flow and shaft speed (rpm) of a particular application. That is, towards a more mixed flow design. The power, mass flow and shaft speed can of course translate to head (H), volumetric flow rate (Q) and angular speed (N) in equation (1), the shaft power being related to head and flow rate (Shaft power = QH/efficiency). The efficiency of a particular machine is then determined by its general design, its operating speed ratio (Q/ND^3, where D is the overall pump rotor diameter usually selected as the characteristic dimension of the unit) and its operating Reynolds Number ($\rho ND^2 / \mu$, where ρ is the flow density and μ its viscosity). The general form of the efficiency envelope for pumps is illustrated if Figure 3.

In the case of the water jet we might expect the substantial magnitude of the velocity entering the pump unit to influence the optimal configuration selected. As will be seen in subsequent sections the design of the intake to diffuse the flow velocity to a lower value at entry to the rotor presents appreciable difficulty and it might thus appear that it is better in an overall sense to accept a higher rotor inlet velocity. Such considerations might bias the selected design towards the more axial flow configuration. However against this must be set the consideration of minimum local blade pressures and the tendency towards blade cavitation, which generally tend to be more pronounced in an axial flow rotor which relies strongly on the performance of rotor blade sections as lifting foils in what is usually termed a reaction blade design. Axial flow designs need to be correctly sized to avoid cavitation damage.

Another perspective on selection of pump design is that mixed flow rotors tend to be less influenced by intake flow conditions with the consequence that as vessel speed reduces, due to headwind for example, the pump exhibits a smaller rise of power if shaft speed is maintained than would be the case for an axial flow pump or, more extremely, an open propeller of course. The consequence is that vessels propelled by axial flow jets tend to lose less speed due to headwind and exhibit smaller speed shortfall due to unfavourable errors in the original performance calculations than is the case with a mixed flow jet unit. Such a conclusion presumes that the engine will take up the increased power and hold the shaft rotational speed at a fixed value. If shaft rotational speed cannot be increased to offset such a vessel speed reduction then there is generally a greater exposure to potential for speed loss with a mixed flow jet unit. This needs to be considered carefully at the design stage in terms of the conservatism of the design process. It also needs to be borne in mind that with some hull/pump combinations the most critical condition presented is the capability to traverse the hump speed in the resistance curve and in this regard some pump selections may need to treat this condition as a critical selection case, which may be as much as 15kt below cruise speed.

2.2 Cavitation damage

A dominant consideration in selection of the optimal water jet pump design is the question of avoidance of damaging cavitation. This is in fact a critical design constraint in many cases and will limit the maximum blade loading and unit power which can be achieved. This leads to the general goal of minimising jet size in order to reduce weight as far as avoidance of cavitation will allow, taking account also of the need to avoid excessive rotor tip speeds. In conventional pump design this is usually approached in terms of the net positive suction head (NPSH), this being the amount by which absolute total pressure head in the inlet to the pump exceeds the vapour pressure of the flow at the operating temperature. A clear margin in NPSH is needed to allow for pressure reductions at the blade suction surfaces and this must be sufficient to avoid damaging cavitation on the blades. Some codes (eg DIN ISO 5199, 1987) suggest that a set margin (ie. in meters) should apply depending upon the design of a particular type of pump to avoid cavitation damage. In this context it is important to distinguish between damaging cavitation and cavitation sheets which form near the blades and which do not lead to damage. Water jet blades may often operate satisfactorily in the presence of such general cavitation sheets on the back of the blades and towards the tip over the first half or more of the blade chord. Damage tends to be associated with very localised vortex related cavities attached to the blade surface, sometimes being formed towards the blade root and producing quite strong and localised cavitation damage.

The necessary margin by which the inlet total pressure must exceed the vapour pressure is generally expressed in terms of an Euler or cavitation number,

$$\text{Cavitation number} = \Delta p / \rho g Z \qquad (2)$$

where the pressure difference Δp is taken between the inlet total pressure and the vapour pressure of the liquid. Z is the overall head rise across the pump, although in some definitions this is substituted by the inlet dynamic head ((inlet flow speed)^2/2g).

Alternatively the tendency to cavitation can be related to the suction specific speed (S), given by

$$S = NQ^{0.5}/(gH_v)^{0.75} \qquad (3)$$

where H_v = (Inlet total pressure-vapour pressure)/ρg. Other parameters indicative of the tendency to cavitation can also be used, such as the Thoma parameter ($H_v/H = (N_s/S)^{4/3}$), but in all cases the question remains the same: that is, by how much should the inlet total pressure exceed the vapour pressure so that local pressure reductions on the blade surfaces do not induce damaging cavitation? This depends of course on the general nature of the blade design, axial flow rotors generally operating with a larger blade pressure reduction than mixed flow or centrifugal rotors. Thus larger values of the design cavitation number or Thoma parameter would be expected to be required for an axial flow machine. Some authors suggest that S should be smaller than a value between 3 and 15 for centrifugal pumps, and others that the critical cavitation number which must be exceeded might be in the range from 0.05 to unity. These are wide general propositions and reflect the importance of individual unit detail design as regards cavitation proneness. Alternatively, a somewhat more ad hoc approach shows a diagram as illustrated in figure 4 where the safe operating inlet head varies with pump overall total head gain (H) and specific speed (N_S) of operation. However, it is of course only appropriate to apply such requirements to the particular design of machine which has been tested.

In practice no general values can be given for non damaging values of the various cavitation parameters and it is necessary to evaluate each machine design by testing. This leads to another approach (ISO 5198, 1987) and that is that the requirement for so called acceptable cavitation corresponded to the condition of low inlet pressure for which the machine efficiency reduced below the non-cavitating value by 3%. This approach allows a machine to be tested for efficiency only and avoids the difficult task of actually observing cavitation on a test rotor and seeking to determine what is regarded as an acceptable operating condition. The subjective nature of visualisation inevitably makes it a difficult basis for decision making. The other problem involved in actually testing to establish safe cavitation conditions is that damage my take considerable time to develop. Thus it is difficult and time consuming to carry out a range of tests at different conditions in order to find the relation between inlet pressure and damage for a given design. Such testing must also contend with the great sensitivity of cavitation onset to local surface geometry and thus to relatively minor variations between individual rotors of the same design. Against this background there is obvious advantage in seeking to establish the cavitation damage criterion on a prescribed shortfall in efficiency below the non-cavitating efficiency.

The influence of vessel speed on the tendency for cavitation within the intake duct is illustrated by figure 5 which shows how the intake cavitation number varies with vessel speed and shaft speed of a typical propulsion system. A typical operating envelope is shown and it can be seen that increases in vessel speed contribute appreciably to the alleviation of the tendency to cavitation, whilst the cavitation parameter is lowered substantially if the intake has a reduced area throat at its inlet end. Whilst there is a considerable exposure to cavitation at low vessel speed with a reduced inlet size, this is alleviated by increasing the intake to have constant area.

2.3 Entrained air

It is fairly common practice in the operation of fixed pump and turbine installations to bleed air into the inlet flow to alleviate the effects of cavitation. This approach relies on the fact that very rapid and damaging cavity collapse is related to the high modulus of air-free water. Water containing even small amounts of undissolved air has a much lower modulus (and a much lower propagation velocity of compression waves, which can be as low as 20m/sec when the air content is 50% by volume compared with around 5000m/sec in pure water) and thus the consequences of cavity collapse are greatly moderated. The effects of air content can easily be observed during operation of large water jets under heavy sea conditions when intermittent air entrainment often occurs. During periods of air entrainment observed through viewing ports in the intake duct it is quite apparent that the relatively sharp noises associated with cavitation are substantially reduced. This suggests that the deliberate introduction of entrained air into the intake flow even under calm water conditions might contribute usefully to cavitation alleviation as it does in the operation of some large water turbines. It might also offer a means of alleviating cavitation during dockside manoeuvring thereby increasing achievable shaft speed and maximum manoeuvring thrust. To the knowledge of this author such concepts have not been extensively investigated for water jets, although they have been used on some high speed propellers, and it is suggested that the effect of entrained air on water jet operation merits further research as benefits may accrue from the deliberate introduction of entrained air in limited quantities to the intake flow.

3. INTAKE FLOW DISTRIBUTION AT CRUISE AND MANOEUVRING

3.1 Physical processes which influence the intake flow

It is well known that here have been substantial difficulties with the operation of large water jets in the 5MW power range and that these difficulties have been associated with the non-uniform nature of the flow in the intake. This non-uniformity has its origins in three or four quite distinct features of the intake flow:

- The hull boundary layer which has developed over the full length of the vessel ahead of the intake.
- The geometry of the intake duct, in particular the commonly adopted rectangular initial section with considerable potential for vortex formation along the convex corners where the intake joins the underside of the hull surface and the transition from rectangular to circular cross section.
- The diffusion of the inlet flow from vessel speed to rotor inlet speed in a frame of reference fixed in the vessel.
- The presence of the rotor shaft which must traverse the upper half of the flow as the rising section of the intake duct turns to the horizontal direction.

3.2 The hull boundary layer

The hull boundary layer forms as a consequence of the flow over the upstream hull from the intake and is therefore significantly influenced by the three dimensional form of the hull as the cross section develops from the bow. In some vessels a bulbous bow or semi-SWATH configuration may significantly influence the boundary layer formation. However to gain an approximate estimation of the thickness of the hull boundary layer at entrance to the intake the situation is often represented as if three dimensional effects are not present and the boundary layer is essentially two dimensional in nature. In many cases this is probably a fair working assumption provided that the hull has a substantial length of nearly constant cross section and that the boundary layer thickness remains smaller than the individual hull beam or draught. If this is the case then a variety of representations of the boundary layer in terms of its length based Reynolds Number may be used. Given the necessity to ascribe some effective length to the hull ahead of the intake, which would be somewhat less than the length to the physical bow position since the hull cross section is small near the bow, it is probably sensible not to use an excessively complex approach to the representation of the boundary layer development. However, it is useful to define what is the effective length of the boundary layer against the background that the momentum deficit of the boundary layer is the most fundamental parameter of importance as it influences the intake wake fraction and jet thrust directly.

The hull Reynolds number is around 10^9 for a typical high speed catamaran of about 1000 tons displacement and so the hull boundary layer will be turbulent from very close to the bow. The velocity profile at such a Reynolds number can represented in a momentum integral solution by $u / U = (y / \Delta)^{(1/10)}$ and the resulting solution for the momentum thickness at a distance x from the leading edge of the surface then follows as

$$\theta = 0.015x / (Re_x)^{1/7} \qquad (4)$$

We can now average the calculated boundary layer momentum thickness around the wetted hull perimeter ahead of the intake so that the average momentum thickness $\overline{\theta}$ at that section is

$$\overline{\theta} = (1/S)\int \theta(s)ds \qquad (5)$$

where θ(s) is the thickness at any position around the perimeter (s) and depends upon the distance to the effective origin for that sector of the boundary layer. S is the hull wetted perimeter at the intake entry section. Substituting from (4) into (5) in the integral and for $\overline{\theta}$ in terms of an effective length of hull L_e ahead of the intake gives

$$Le / L = \left((1/S)\int x(s)^{6/7} ds\right)^{7/6} \qquad (6)$$

where L is the total hull length from the extreme wetted bow to the intake. For a known hull form the integral can be evaluated numerically, and for a typical high speed catamaran hull form with a V section near the bow varying to a rectangular section at the intake we find that the effective hull length is 82% of the overall length. For an overall length ahead of the intake of 60m, an effective boundary layer average growth length of 0.82 x 60 = 49.2m and an external flow speed of 20m/s, equation (4) thus predicts a boundary layer thickness of Δ = 0.64m and a momentum thickness of 0.048m.

The extent to which the boundary layer is ingested into the intake depends upon the geometry of the intake itself. Intakes which are relatively wide will tend to ingest a larger component of the hull boundary layer. In most cases the intake is formed initially by a rectangular rising entrance of similar dimension to the diameter of the round internal duct ahead of the jet unit rotor. If we assume that the ingested boundary layer section is thus of a width equal to the duct diameter (D) then for a 1.1m diameter duct we find that the momentum deficit due to the boundary layer is a fraction $(4/\pi).(\overline{\theta}/D)$ of the total intake flow momentum, that is about 6% for a momentum thickness of 0.048m. In fact it appears that ingestion from a larger section of the boundary layer and hull surface roughness increase the momentum deficit to the range from 8% to 15% depending upon conditions. It is important in making measurements of the hull boundary layer to ensure that the velocity traverse is taken out to the local external potential flow value as the surface wavemaking may cause local departures from nominal ship speed in this regard. If the boundary layer traverse is truncated prematurely on account of a rather high local external velocity, as would occur when the intake is located under a trough in surface wave pattern, then it is possible that an unduly thin estimate of boundary layer thickness is made leading to a low estimate of momentum deficit, although this would at least be a conservative outcome in the overall design process.

The momentum deficit at the intake has a significant effect on the overall unit thrust achieved as will be discussed and there is some debate as to the precise magnitude of this quantity. It appears that the rather larger values around 10% or more are often used in thrust design calculations due to the influence of hull detail (such as chines) on the flow and also the geometric intake detail effects on boundary layer ingestion. The definition of the commencement of the intake raises other issues, and if the hull technically included part of the intake such as the rising ramp section then it might be necessary to include the surface pressure distribution in the momentum balance calculation. However if the intake if defined as commencing at the very start of the rising ramp then such effects are ascribed to the intake and jet unit as a whole and need not affect the calculation of the momentum deficit of the intake flow. This latter approach is generally adopted for reasons of simplicity and clarity of definition.

3.3 Diffusion in the intake

It is usually the case that the design axial velocity under cruising conditions at entrance to the rotor is substantially less than the vessel speed and therefore it is necessary that flow diffusion takes place in the intake duct. An ideal design for cruise condition would achieve this by sizing the inlet to the intake duct so that the external velocity prevailed at that section. The flow could then be diffused within the intake duct to best effect by minimising the duct diffusion angle so that a maximum pressure recovery is achieved. Some designs in small high speed vessels adopt this arrangement, in which case the question of flow diffusion is as for any other internal duct a question of ensuring that the duct does not expand too rapidly. However a design constraint which applies to large vessels and ferries is that they must have the capability of delivering adequate thrust whilst manoeuvring alongside a dock. Under such manoeuvring conditions a reduced area throat at the inlet to the intake duct gives rise to more severe pressure reduction in the intake and thus cavitation occurs at a rather lower mass flow, thereby limiting manoeuvring thrust which can be achieved. For such vessels this may be unacceptable, and in many cases the intake duct is designed without any internal expansion of area to provide cruising flow diffusion inside the duct. Flow diffusion for such constant area intakes thus occurs at cruise in the form of a separation of the hull boundary layer on the upper side of the intake duct and subsequent remixing of this separated flow zone within the intake duct itself. In general the single most important parameter which influences the local intake flow patterns is the velocity ratio, that is the ratio of average inlet speed to the rotor to the ship speed, a ratio which has a value of about 0.6 in many cruise cases and which will increase as the ship speed reduces. It has been suggested that variable geometry intakes in which the intake mouth is closed down as ship speed rises might be one way of addressing the range of requirements from manoeuvring to cruise, but the practical complexity of such an arrangement would probably preclude its use except in relatively small vessels.

Typical velocity profiles measured in an aerodynamic model intake are shown in figure 6. The separation near the inlet from the lower surface (curve 1 in figure 6, Y=0) with zero ship speed is quite clear in figure 6(a), whilst at a condition representative of cruising conditions (figure 6(b)) we see that there is a strong region of separated flow on the upper surface of the intake (Y=100 mm in this model test of a rectangular intake duct). As the flow moves downstream into the intake (curve 2 in figure 6) so turbulent diffusion remixes the highly separated flow, but the separated flow effect is more persistent in the cruising case. These results indicate that considerable flow non-uniformity would exist at the rotor inlet under cruise conditions.

Under cruising conditions the criterion for avoidance of separation of the hull boundary layer is based on the reduction of surface shear stress to zero as the boundary layer momentum becomes insufficient to overcome the adverse (rising) pressure gradient due to the diffusion of flow outside the boundary layer. This criterion can be summarised from the momentum integral boundary layer solution in the form that for no separation we require

$$(dU / dx) < U(d\theta / dx) / (2\theta + \delta^*) \qquad (7)$$

where δ^* is the boundary layer displacement thickness and θ is the momentum thickness. For a typical high speed ferry intake the flow might diffuse with a reduction of 10m/s over a distance of 5m, that is $(dU/dx) = 2\ s^{-1}$. For a high Reynolds number boundary layer with n=10 we have $\delta^* = 1.2\ \theta$ and from the momentum integral equation for the layer $(d\theta/dx) \sim 0.007(\rho U\theta/\mu)^{-1/6}$. Thus the right hand side of equation (7) typically takes on a value of $0.1\ s^{-1}$, which is very much less than the left hand side, with U=20m/s and θ=0.048m. It is clear that separation of the hull boundary layer and the formation of a very nonuniform velocity profile within the intake at cruise conditions is almost unavoidable, there invariably being a region of separation and low velocity on the upper side of the intake duct.

When the vessel has zero forward speed the general criterion for satisfactory performance of an intake without significant separation at the lip and without significant pressure loss as a result is that the radius of the lip should be about 0.3 of the intake diameter. For most water jet inlets this condition is nowhere near met as far as the aft lip is concerned, the lip radius often being as small as 2 to 4% of the internal duct diameter. Thus substantial separation of flow will occur inside the intake as the flow passes around the aft lip. This causes significant pressure loss in the intake as the main inflow passes around the large separated flow region on the lower side of the intake and subsequently refills the intake under turbulent mixing conditions. The pressure loss gives rise to cavitation at a lower mass flow than would otherwise occur and thus limits achievable thrust. Typically rotor rotational speed may only reach 65% of the design cruising value before serious cavitation occurs. Also cavitation may occur on the lip itself due to high velocity at the lip if the lip radius is small.

4. OVERALL POWER, THRUST AND EFFICIENCY

An elementary flow analysis based on continuity, momentum and energy considerations leads directly to expressions for the thrust (T), thrust power (P_T), propulsive efficiency (η_p), jet efficiency (η_j) and overall efficiency (η_o),

$$T = \rho\ A\ V_J\ (V_J - V_I) \qquad (8)$$

$$P_T = T\ V_C \qquad (9)$$

$$\eta_p = 2 V_C / (V_J + V_I) \quad (10)$$

$$\eta_j = 2 V_C(V_J - V_C)/(V_J^2 - (1-\zeta)V_C^2) \quad (11)$$

$$\eta_o = 2 V_C / ((V_J + V_I) + 2g\Delta H/(V_J - V_I)) \quad (12)$$

where V_J= jet velocity relative to nozzle of area A and V_I= average inlet flow velocity relative to jet which may be rather less than the vessel cruising speed (V_C) due to the presence of the hull boundary layer. The sum of internal head losses in the jet unit plus the vertical height of the nozzle above the waterline is ΔH. The intake loss coefficient is ζ. Internal pump unit losses might typically lie in the range from 0.04 to 0.1 of the nozzle dynamic head. The overall efficiency (equation 12) represents the ratio of thrust power to shaft power and includes the internal losses in the pump and intake and the effect of height of the nozzle above the waterline. The propulsive efficiency (equation 10) is the ratio of thrust power to shaft power less the internal pump losses and nozzle height effects. The propulsive efficiency provides a clear reminder that operating with a value of the jet velocity much larger than the vessel speed increases energy wastage in the jet and reduces the thrust power for a given unit power. A larger, slower jet is inherently more efficient as the jet velocity reduces towards the vessel speed, at which limit the propulsive efficiency becomes 100%. Such considerations lead to the general outcome that the overall efficiency of the propulsion system tends to rise from around 0.5 at 20kts to around 0.75 at 40 kts. The jet efficiency (equation 11) combines the representation of propulsive and intake loss effects. The flow rate through the pump in relation to the shaft speed determines the optimal operating efficiency of the pump itself, whilst the ship speed may affect the losses in the intake as the intake separated flow pattern varies. All these factors combine to give the best overall operating efficiency point.

The inclusion of the nozzle height above the water line indicates another compromise that often has to be struck, namely avoidance of flow resistance of nozzle components such as reverser or steering gear that may arise with a low mounted nozzle whilst avoiding an excessively large vertical displacement of the nozzle exit flow, the energy of which is not translated into propulsive effect.

Ultimately it is the thrust delivered which is critical to overall performance and speed achieved and it is sometimes easier to consider vessel performance in terms of thrust by measuring the nozzle stagnation pressure directly to determine the jet velocity (V_J). The only uncertain aspect which then enters the identification of causes for departure from design speed is the extent to which the average inlet velocity relative to the jet unit (V_I) is less than the vessel speed (V_C) due to ingestion of the hull boundary layer. Measurements of shaft power, although important in assessing the performance of the pump itself, inevitably introduce other uncertainties into the performance review process. It is suggested here that it is clearest if the two issues of achieved vessel speed and pump efficiency are approached separately by this approach rather than seeking to attribute achieved vessel speed to the shaft power of the jet drive, an association of parameters which introduces both the pump efficiency and the hull boundary layer effect into the calculations simultaneously and will result in uncertainty as to which aspect has caused the achieved vessel speed to depart from predictions. The efficiency of the pump itself will be affected by the inlet flow non uniformity and other intake effects, whilst the overall thrust achieved will only be subject to the question of the momentum deficit in the ingested hull boundary layer.

Representation of the inlet velocity can be made in terms of the wake field of the hull as is done with propellers. Several alternative representations are commonly used, the first being to specify the velocity ratio of inlet velocity relative to the vessel (with average V_I)to the vessel speed (V_C). This velocity ratio (V_I/V_C) is of course less than unity for the average inlet velocity. Taylor's method uses the so called Taylor volumetric wake fraction, which is $w_T = (1- V_I/V_C)$ and thus represents the ratio of velocity difference between intake average speed and vessel speed to vessel speed. Finally, the Froude volumetric wake fraction representation forms the ratio of velocity difference to the relative inlet speed, $w_F = (V_C/V_I - 1)$. Whichever representation is used, equation (8) shows that for a given mass flow reduction of the inlet velocity relative to the vessel (V_I) improves the thrust achieved and that ingestion of the hull boundary layer is beneficial to thrust. This effect can be quite significant in the actual outcome of vessel speed achieved and it is clear that designers need to pay great attention to ingestion of the hull boundary layer and its effect on thrust. Values of the Froude wake fraction as high as 0.15 have been suggested in some cases, much larger than the figure of around 0.06 that is implied by the discussion of section 3.2 above. Use of a lower value for the wake fraction is, of course, the more conservative approach to adopt.

The question of provision of a margin between estimated hull resistance (R) and the estimated jet thrust (T) can also be considered in terms of the thrust deduction factor, $t = (T - R)/T$, which can be seen as a means of expressing the fact that the interaction of the jet with the hull as installed is not accounted for by the hull resistance or jet thrust as determined separately. This leads to another concept, that of the hull efficiency (η_h),

$$\eta_h = (1 - t)/ (1 - w_T). \qquad (13)$$

The hull efficiency can be seen as representing the trade off between the loss of performance due to installed interactions between jet and hull and the advantages gained by ingestion of the hull wake to the jet, the overall ship efficiency then becoming the product of pump, hull and propulsive efficiencies.

5. INFLUENCE OF INLET FLOW ON ROTOR ASSEMBLY

The severe inlet flow non-uniformity has a number of effects on the rotor including its overall efficiency of operation, but the most important operational consequences are to produce substantial asymmetric loads on the rotor and to give rise to vibration and to non-uniform cavitation. Cavitation gives a fairly clear indication of the flow non-uniformity as illustrated in figure 7 for a typical rotor. The cavitation sheet on the back of the rotor blades becomes larger in size on the upper section of the rotor disc plane and as the rotor moves downwards after encountering the region of low axial velocity at the top of the duct where the rotor inlet flow velocity remains less than the average velocity in the intake duct. Owing to the peripheral extent of each blade the cavitation tends to be strongest somewhat after the leading edge has passed through the region of lowest axial velocity. These observations are relatively easy to make if the rotor is illuminated obliquely by a stroboscopic lamp via a suitable viewing port. It is worth noting that the use of a borescope for viewing and recording appears to be an approach with appreciable potential for improving such observations as borescopes have a good depth of field and field of view.

The vibration effects of inlet flow non-uniformity are much harder to quantify and some manufacturers have installed instrumented blades to directly measure fluctuating blade stresses. However it appears to be the case that more axial flow rotor designs with smaller blade numbers present a more severe general vibration condition, especially if the blades of the rotor have near straight radial leading edges. The consequences of vibration are likely to be fatigue of weld joints in the general region of the jet unit and it is therefore important to ensure that the joint and structure designs which are used in this part of the vessel do not concentrate loads at the welds themselves. In some cases fatigue cracks have formed near the roots of the rotor blades themselves.

The asymmetric loads produced on the rotor arise due to the transverse momentum fluxes associated with the passage of the non-uniform inlet flow through the rotor. Significant rotor shaft transverse and bending loads can result. We can estimate these asymmetric loads by considering the momentum flux balance across the rotor when it experiences a non uniform inlet flow axial velocity $v(r,\phi)$, a function of radial position (r) and angular position (ϕ), and when the exit flow from the rotor has a tangential velocity component $v_t(r,\phi)$. The resulting expressions for the horizontal (F_h) and vertical (F_v) transverse forces on the rotor are then written as

$$F_h = \iint_{exit} \rho v(r,\phi) v_t(r,\phi) \sin(\phi) r dr d\phi \qquad (14)$$

$$F_v = \iint_{exit} \rho v(r.\phi) v_t(r,\phi) \cos(\phi) r dr d\phi \qquad (15)$$

whilst the whilst the turning moments on the rotor about horizontal (M_h) and vertical axes (M_v) in the plane of the rotor are

$$M_h = \iint f(r,\phi)\ r \sin(\phi)\ dr\ d\phi \qquad (16)$$

$$M_h = \iint f(r,\phi)\ r \sin(\phi)\ dr\ d\phi \qquad (17)$$

where $f(r,\phi)$ is the axial force per unit area on the rotor disc due to the momentum and pressure fields applied to it. Whilst a precise evaluation of theses forces and moments is of course very complicated it is possible to make estimates of their magnitude and nature. For a typical 5 MWatt near axial flow water jet with a diameter somewhat over 1m the flow non-uniformity can be approximated by a linear variation of axial velocity entering the rotor, with a higher velocity on the lower side. The tangential velocity at exit can be estimated from the rotor geometry and thickness from inlet to outlet. Such calculations lead to the conclusion that the rotor will experience a transverse force of approximately 2.5 tonnes downwards and at about 40 degrees from the vertical in the direction of rotation. The rotor turning moment under typical conditions is estimated to be approximately 5 tonne m in a bow down sense about an axis inclined at about 45 degrees to the horizontal axis through the rotor plane, that is nearly perpendicular to the transverse force vector. These transverse loads can give rise to significant deflection, especially where cutless rather than roller bearings are fitted, and also to significant shaft bending. It has been found that broadly consistent shaft deflections are observed with these predictions and also that the fitting of a diffusing intake to reduce the rotor inlet flow non-uniformity can reduce the transverse loads and consequent shaft deflections substantially. However such intakes significantly restrict manoeuvring thrust due to cavitation.

The non-uniform inlet velocity and load distribution also forms a basis for estimating the unsteady blade loads which can give rise to formation of fatigue cracks in rotor blades. Typical inlet flow non-uniformities thus lead to estimated blade load fluctuations of around three tons per blade on a three bladed 5MWatt rotor at the rotational frequency of around 12Hz.

6. CONCLUDING COMMENTS

Water jets operate essentially as mobile pumps and many of the design approaches applied in fixed pump installations are relevant to the water jet. However, there are particular features associated with the inlet flow that need particular consideration. Most important amongst these are the effect of the hull boundary layer and intake separation on the thrust delivered and the need for designs to accommodate the asymmetric rotor loads which arise from non-uniform inlet flow. Intake designs should seek to accommodate operation at cruise and whilst manoeuvring. This involves a balanced consideration of cruise inlet flow non-uniformity which must be set against the need to maximise manoeuvring thrust to an acceptable level, the resulting compromise establishing the extent to which the intake can be designed as a diffuser at cruise. There is potential for continued improvement of intake design, in particular in respect of the aft intake lip where increases of radius may offer the potential to improve manoeuvring performance without impairment of cruise performance. Also, there is potential to maximise the hull efficiency by appropriate intake design.

ACKNOWLEDGEMENT

The author wishes to acknowledge a long period of cooperative work with INCATS Tasmania Pty Ltd which has provided the general background to the preparation of this paper.

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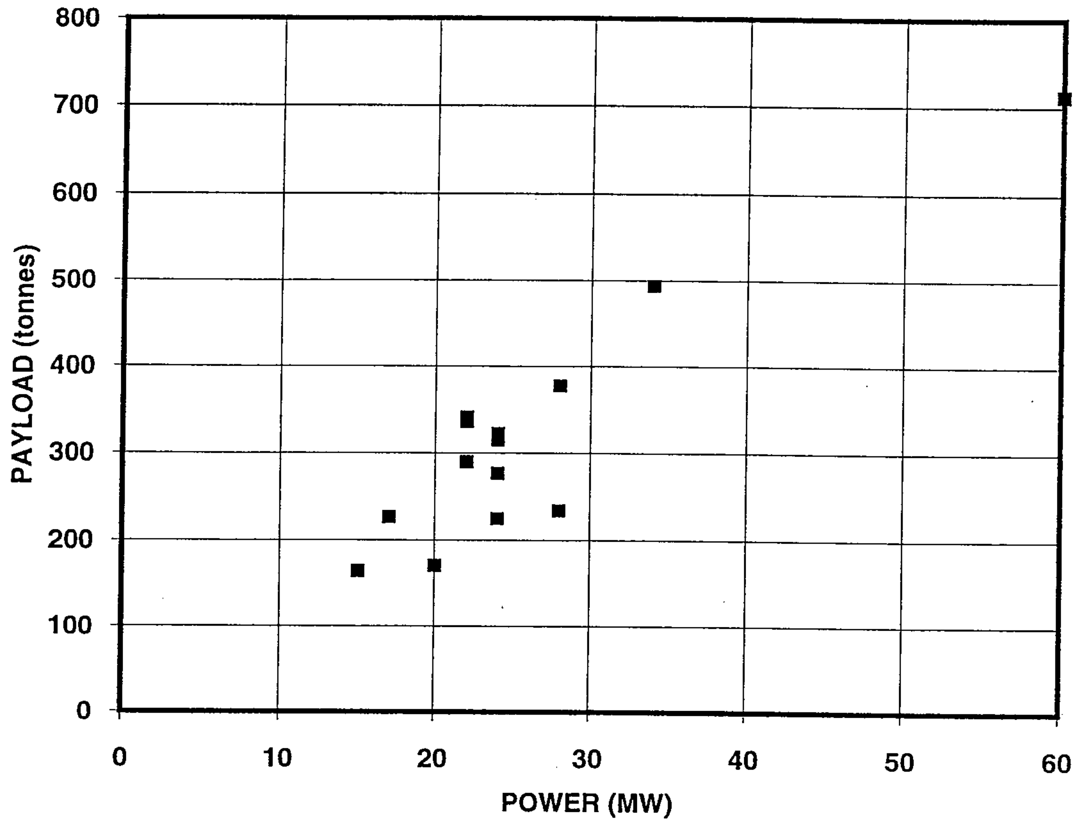


FIGURE 1 Power and payload of high speed ferries in the 35-42 kt speed range

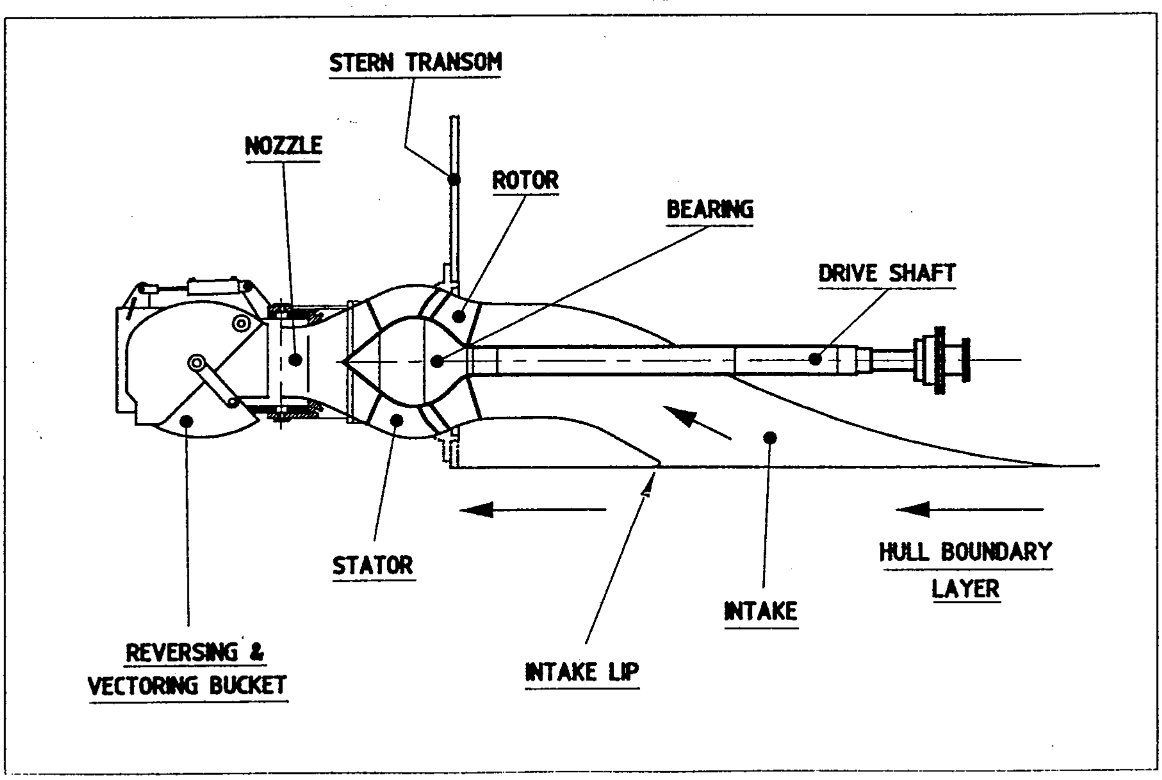


FIGURE 2 General layout of a propulsion waterjet

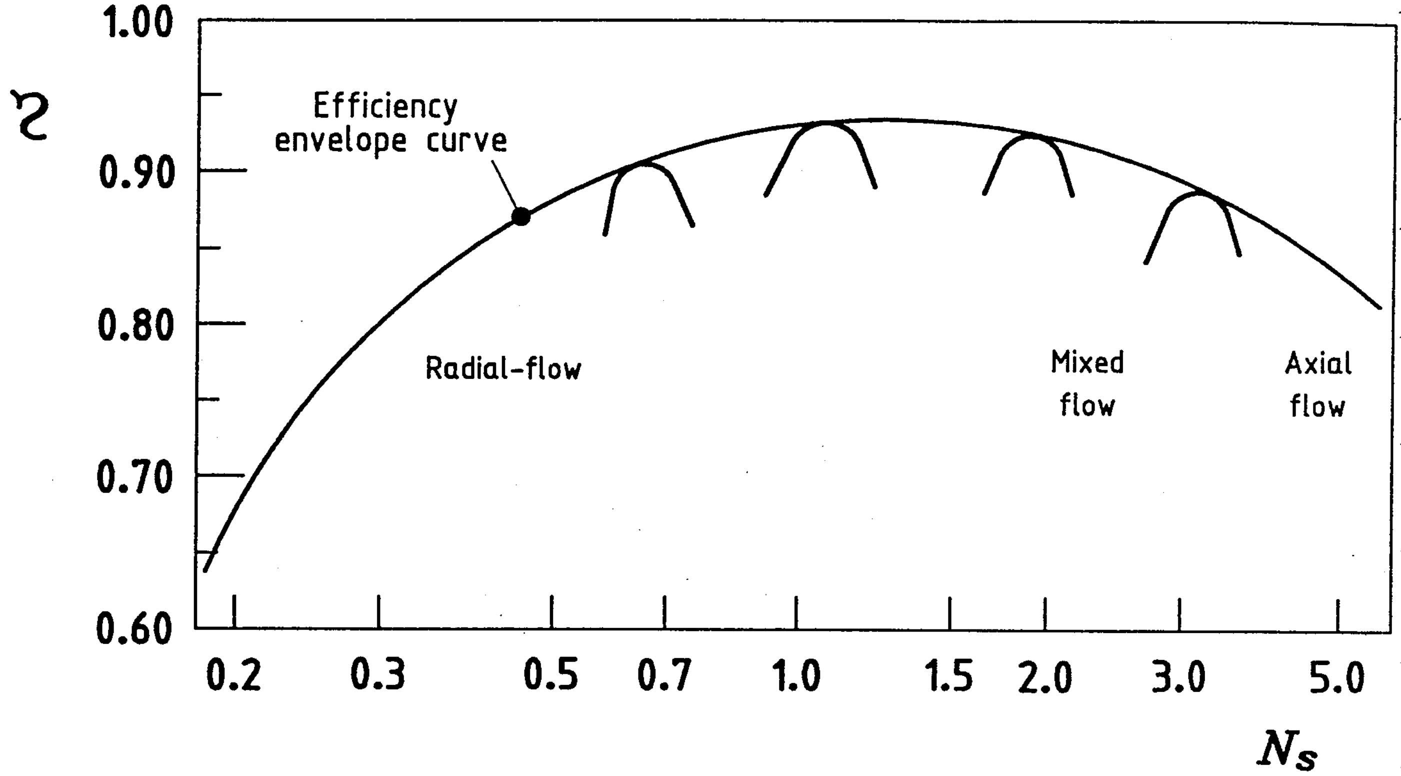


FIGURE 3 Efficiency of pumps and efficiency envelope curve

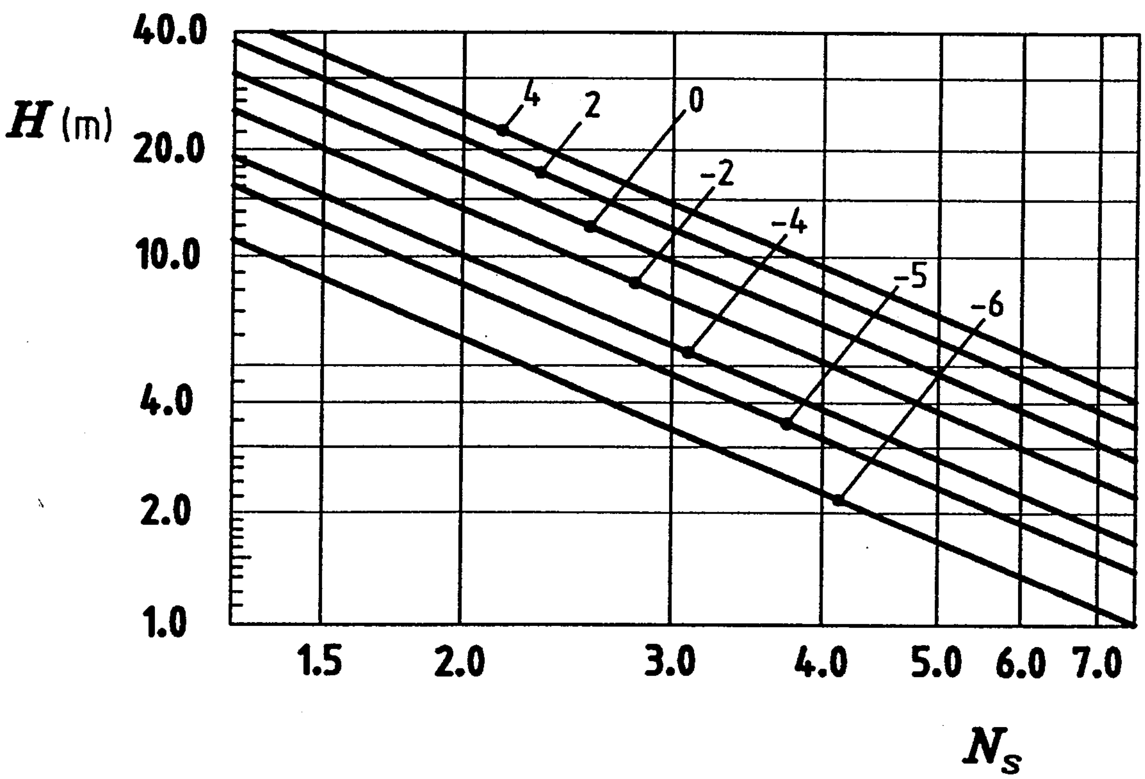


FIGURE 4 Variation of head limitation with specific speed and suction side head for mixed and axial flow pumps (Hydraulic Institute Standards,1969: figures against curves denote suction side head for fresh water at 30^{o}c)

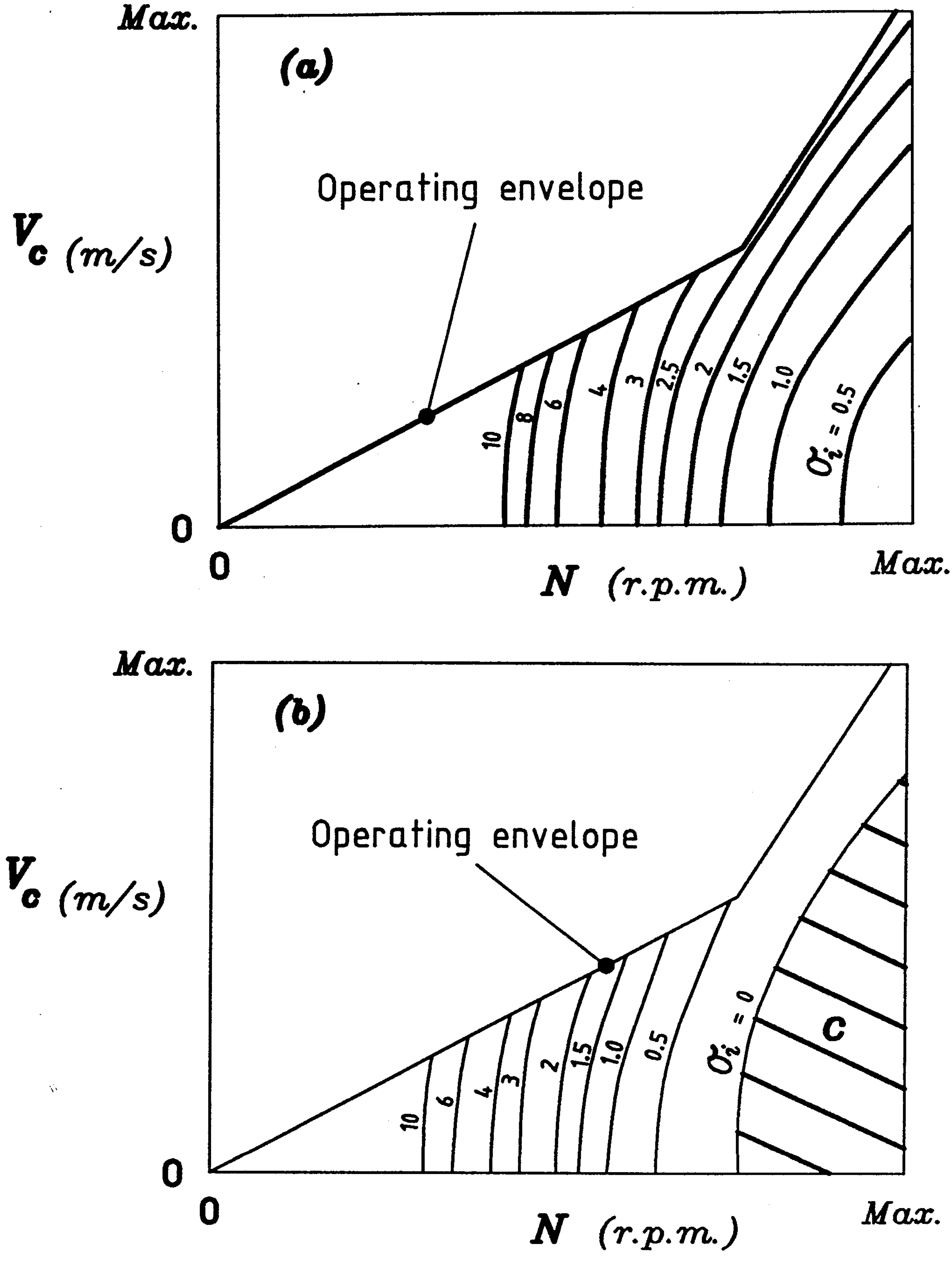


FIGURE 5 Variation of intake cavitation parameter with shaft and ship speeds (intake loss factor = 0.6, ship speed =1.6 x rotor inlet speed, cavitation parameter normalised on intake dynamic head, C denotes zone of bulk intake cavitation). (a) Intake of constant area. (b) Intake with inlet throat sized to give internal flow diffusion.

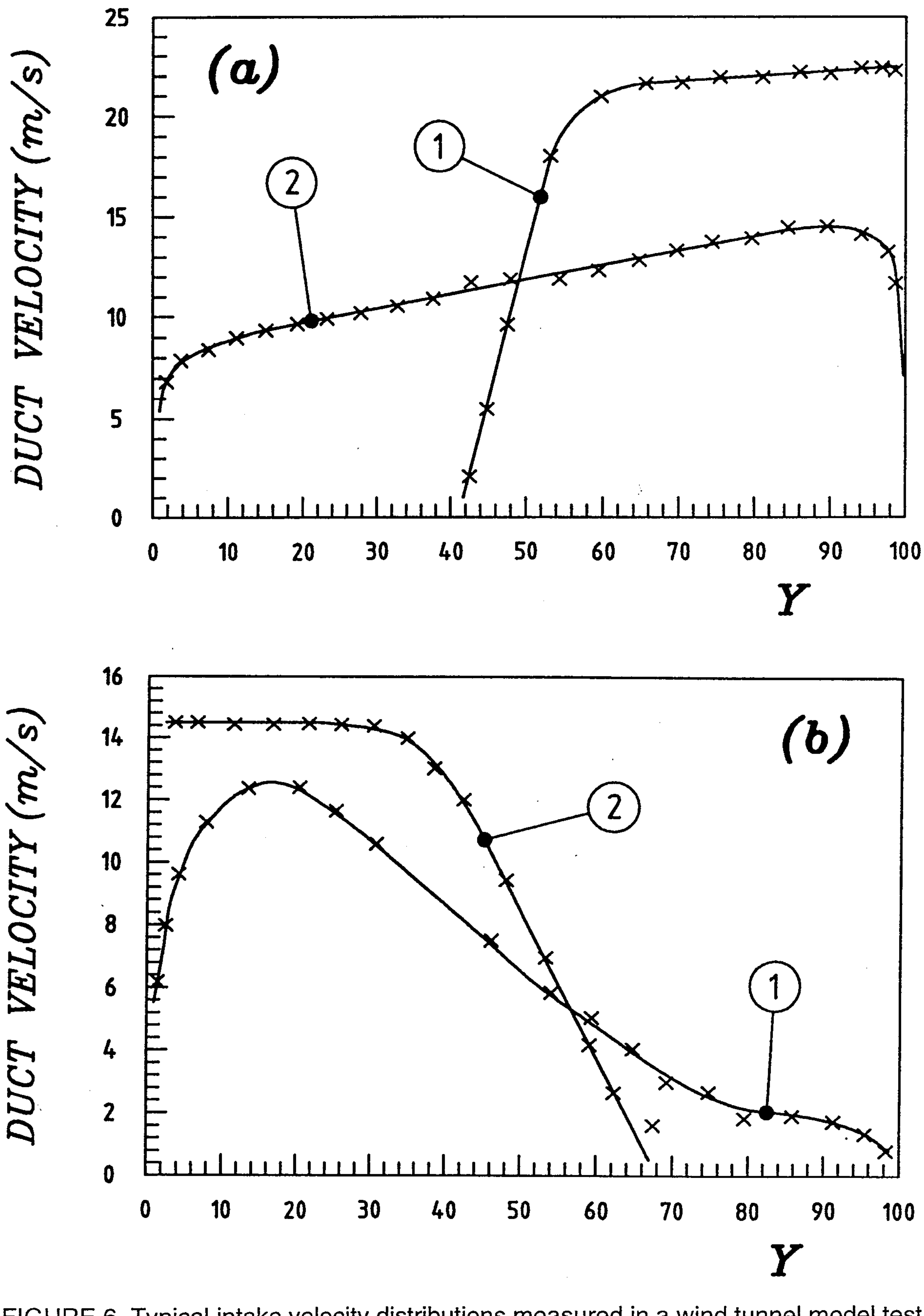


FIGURE 6 Typical intake velocity distributions measured in a wind tunnel model test
(a)Zero ship speed (b) Ship speed = 0.6 x intake mean speed
(Constant area intake, lip radius 2% of duct height : curve 1 - profile at 1.23 times duct width from intercept of intake lower surface line with hull underside; curve 2 profile at 11.23 times duct width from intercept)

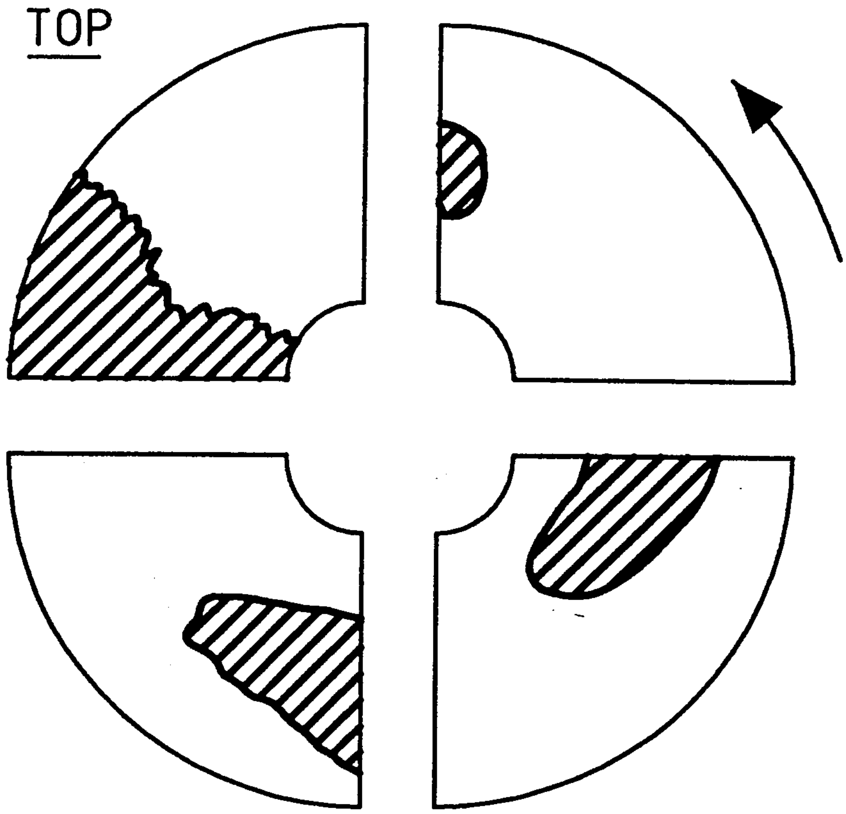


FIGURE 7 Typical distribution of cavitation sheet locations on a water jet rotor around the periphery

Paper No 4

Shane Smiltnieks

Materials Welding and Joining CRC

"Adhesive Bonding in Marine Applications"

Adhesive Bonding in Marine Applications

S.J. Smiltnieks, G.M. Spinks and M.J. Turner

ABSTRACT

The use of adhesive bonding in marine applications is being investigated. Even for minor attachments, the influence of the marine environment and fatigue stress loads cannot be ignored. This project has involved the design and construction of a unique testing facility that simultaneously subjects adhesive joints to fatigue loads and a salt spray. The performance of the testing machine has been evaluated using adhesively bonded aluminium joints prepared with various pretreatments and an epoxy adhesive. Results confirm that the pretreatment is critical to fatigue life and that some relatively simple pretreatments appear to give excellent performance. Preliminary characterisation work has been completed to elucidate the effects of pretreatment on the metal surface using atomic force microscopy.

Introduction

High speed ships and offshore structures, as well as foreshore constructions, contain substantial amounts of minor components that are currently welded in place, often to the detriment of the quality of adjacent paint finishes, nearby combustible materials and so forth. The use of adhesives as opposed to welding for such items as small seats, cable trays, pipe hangers and other minor structure offers potential competitive advantage in terms of: less distortion, less secondary damage due to heat, improved fatigue and corrosion resistance, dissimilar metal connections and greater scheduling freedom all leading to lower cost. Currently, there are some impediments to the introduction of adhesives for marine craft and other offshore structures stemming from the lack of performance data for adhesive joints in the marine environment. The Co-operative Research Centre for Materials Welding and Joining is part of an international collaborative research effort aimed at generating such performance data and so encourage the greater use of adhesives in marine applications.

A comprehensive review of the technical literature has identified that there have been very few reports on the combined effects of an aggressive environment and fatigue loads on the performance of adhesive joints. In contrast, there have been a great many reports on the effects of these conditions on the adhesive joint lifetime and strength as separate studies. It is obvious from these reports that both cyclic stresses and aggressive environments (particularly moist environments) can significantly weaken a bonded metallic sample. The combined effects are not yet clear, however.

The aim of our work is to develop a testing procedure for evaluating adhesives and metal pretreatments for fatigue life in a marine environment and to identify systems that give satisfactory fatigue life.

Experimental

Design of Testing Facility

To adequately study the combined effects of an aggressive environment (e.g. salt spray) and cyclic loads, a purpose built fatigue testing machine needed to be designed and constructed. The following design criteria were decided:

1) Operating Conditions

Samples must be subjected to a salt spray environment similar to that described in Standard ASTMG85-85

Provision for variable temperature experiments should be available

2) Loading Conditions

Low frequency fatigue is to be considered with cycle frequencies of approximately 1 Hz. The loading waveform should be approximately sinusoidal. These conditions approximate the loading induced by wave action on marine craft/offshore structures. It was noted that although high cycle fatigue can produce results more quickly, those results may not be representative of lower frequency loading, due to the viscoelastic nature of the polymeric adhesives.

Load range was to be such that a suitable S-N (stress - number of cycles to failure) curve could be generated.

3) Number of Samples

Simultaneous testing of multiple samples should be available to reduce the time needed to generate the S-N curve

A photograph of the completed testing facility is given in Figure 1.

Materials

Aluminium (5005 Grade) was used as the substrate materials for the fatigue tests. Samples were cut to coupons of 15mm wide and 120mm long. These were bonded with Araldite K106, a 2 part epoxy adhesive from Ciba Geigy, in a lap shear arrangement with an overlap length of 10mm.

Various pretreatments were applied to the samples prior to bonding. The simplest pretreatment was a solvent wipe with a clean cloth dampened with acetone. The Forest Products Laboratory (FPL) etch pretreatment was conducted as specified in ASTMD2651-90. This essentially involves immersing the aluminium in a dichromate acid solution at 70°C for 15 min. The third pretreatment involved immersing the aluminium in an aqueous alkaline solution at 75°C for 45 min. This was termed an alkaline precleaning treatment. Two other lesser reported pretreatments have also been investigated, the first is very similar to FPL and requires the same processing parameters and has been designated acid. The second is a chromate free solution which only requires heating to 45 °C and 15mins immersion.

Results

Loading Conditions

The testing machine was calibrated using a load cell placed in series with each of the loading points. The loading rate was calculated to be 0.3Hz. The load range varied from zero to the maximum load, and the maximum load that could be applied was 750N which translates to 5MPa for a single overlap joint of overlap area 150mm^2.

Fatigue tests were conducted using maximum loads of between 550N and 220N.

Static Joint Strengths

Figure 2 is a bar chart showing the lap shear strengths of aluminium joints bonded with Araldite K106 adhesive that were prepared with various pretreatment techniques: solvent wipe, FPL etch, alkaline preclean, acid and chromate free. The results show that initially the

FPL and alkaline preclean samples show similar strengths which are slightly greater than that of the solvent wiped samples, the two other pretreatments show strength significantly greater than the other treatments. Once exposed to a hostile environment which in this case was immersion in a salt water bath at 50 degrees Celcius it can be observed that there is a very significant decrease in strength of both the solvent wiped sample and the FPL sample. The alkaline preclean and the acid samples retained a large percentage of their strength and the chromate free samples actually increased in strength.

Fatigue Performance in Air

Solvent wiped samples were tested using the fatigue machine at ambient ("air") conditions of approximately 15°C to 30°C and 50% to 80% RH (Figure 3). The S-N relationship follows the log linear behavior typically observed in fatigue tests. No evidence of an endurance limit was observed in the tests conducted, although this may become apparent at lower loads / longer test times.

Fatigue Performance in Salt Spray

Figure 4 compares the performance of solvent wiped samples tested in air and in a salt spray environment. The results show that at high loads the salt spray shortens the fatigue lifetime of the joints. At lower loads there is no significant difference in performance in the two environments for these samples.

Figure 5 compares the fatigue performance in salt spray of all the pretreatments. Only slight improvement was observed with the FPL process compared with solvent wipe pretreatment. Importantly, however, the alkaline precleaned sample showed significantly improved fatigue resistance.

Figure 6 compares the performance of FPL treated samples which were fatigued in both air and salt spray. This test differed slightly from those above in that it was run to in excess of 1.6 million cycles. The results show once again the degrading effect on joint strength of the marine environment. It should be noted though that in the test that was conducted in air one of the samples actually failed through the aluminium rather than at the bond. This occurred at approximately 1.6 million cycles and indicates that the indurance limit of the aluminium is somewhere in this vicinity. Figure 7 shows two different samples, one which failed through the substrate (aluminium) and one which failed through the bond.

Surface Characterisation Using AFM

The surface pretreatment applied to the aluminium appears to significantly affect the joint strength and fatigue performance, as has been widely reported by many other workers. There are many theories that attempt to explain the effect of pretreatment on adhesive joint strength. In this study, Atomic Force Microscopy (AFM) has been used to quantitatively study the effect of pretreatments on the surface roughness of the samples. Figures 8-10 show AFM images of samples that have been solvent wiped, FPL etched and alkaline precleaned. The latter two treatments significantly increase the roughness of the samples surface through an etching processes. The joint strength should, therefore, be increased through an increased interfacial contact area and mechanical interlocking effects.

Discussion and Conclusions

The project has successfully developed a testing facility that will allow the simulation of marine conditions for the testing of adhesive joints. Preliminary results show that the fatigue performance displays the typical log-linear relationship between applied load and cycles to failure. The early results also show that the metal pretreatment significantly affects the fatigue performance.

The results obtained to date have some value in the practical application of adhesives in marine environments, but much more work is required to achieve the project objectives. Fatigue testing to date has been conducted up to approximately 1.6 million cycles which is about one and a half months of continuous testing. This length of test is the minimum required to obtain sound data on the long term performance of the adhesive bonds. To date this length of test has only been performed on one type of pretreatment and hence there are quite a number which still require evaluation although it should be noted that even from the short term tests (100,000 cycles) it can be observed that even after a few days the strength can be reduced by up to 5 times. It is for this reason that we cannot rely only on lap shear samples (static tests) to provide data on the fatigue strength of adhesive bonds particularly when exposed to a hostile environment

Future work will concentrate on longer term testing of joints with a wider variety of adhesives and substrate materials. In addition, work will continue on the evaluation of metal pretreatment methods for increasing fatigue resistance and the characterisation of the adhesive - substrate interfaces.

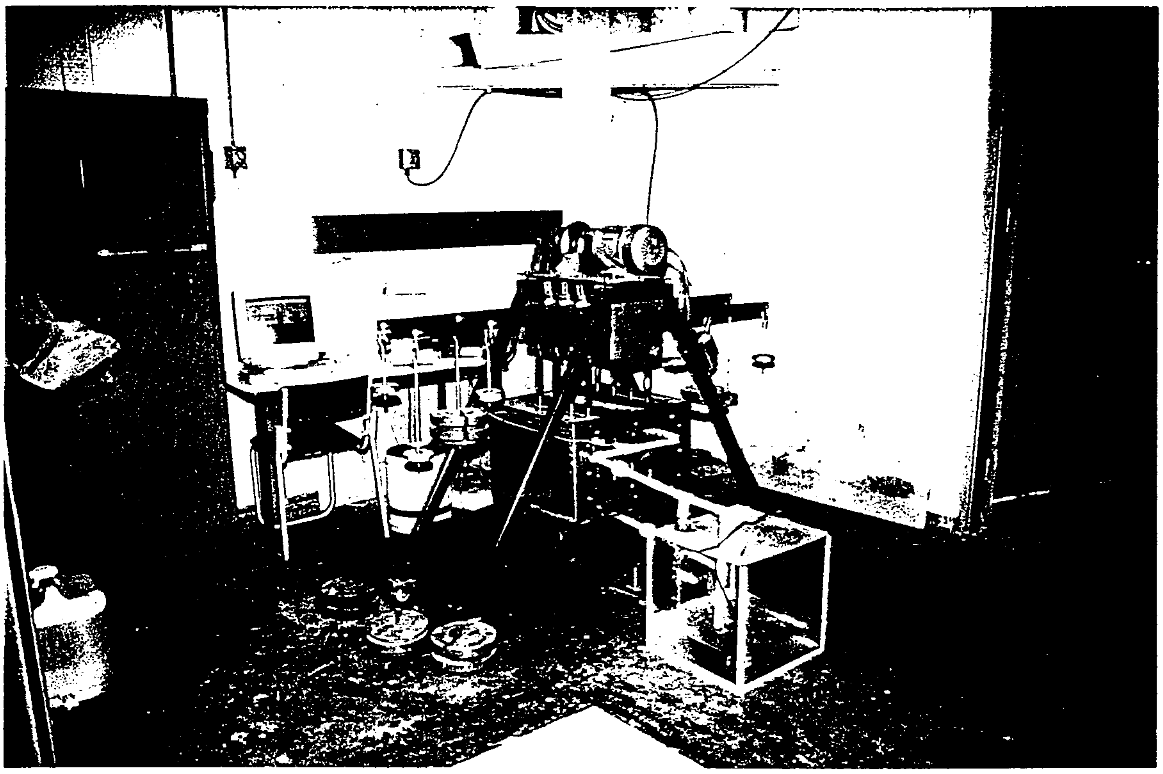


Figure 1

Photograph of fatigue testing facility at the University of Wollongong for simultaneously fatigue testing eight samples in a salt spray chamber.

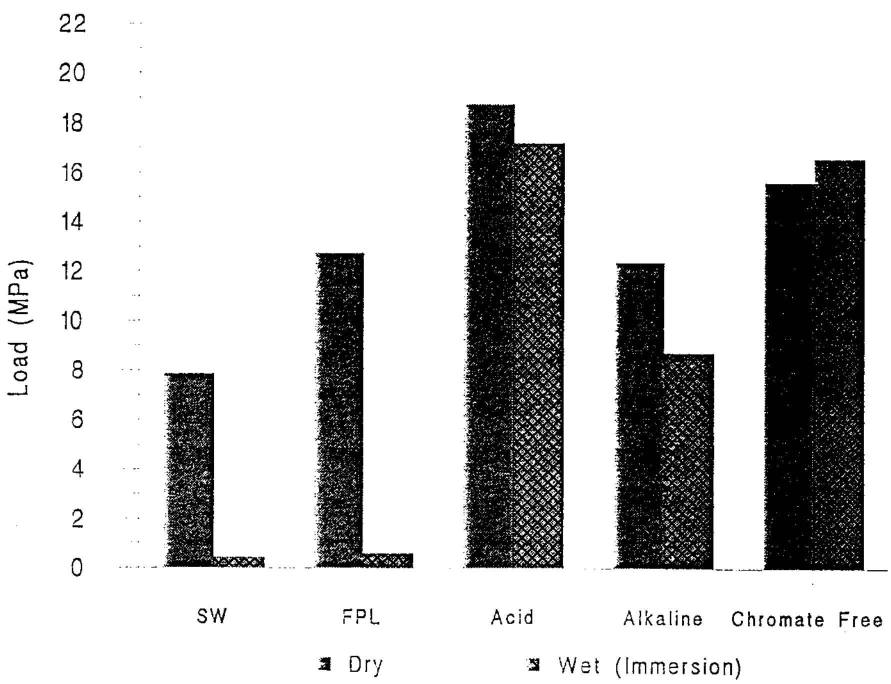


Figure 2

Lap Shear results for various pretreatments both initially and after exposure to a salt water bath at 50 degrees Celius for 10 days.

ACKNOWLEDGEMENT

The reported work forms part of an on-going programme of work at the University of Wollongong which is supported financially by the Cooperative Research Centre for Materials Welding and Joining. Dr Spinks and Mr Smiltnieks are at the University.

The programme interfaces with a European programme within the EUREKA initiative. This programme is managed by TWI (The Welding Institute) UK. Mr Turner is seconded from TWI to the CRC-MWJ.

The authors are grateful for the support of all the bodies which are subscribing to these programmes.

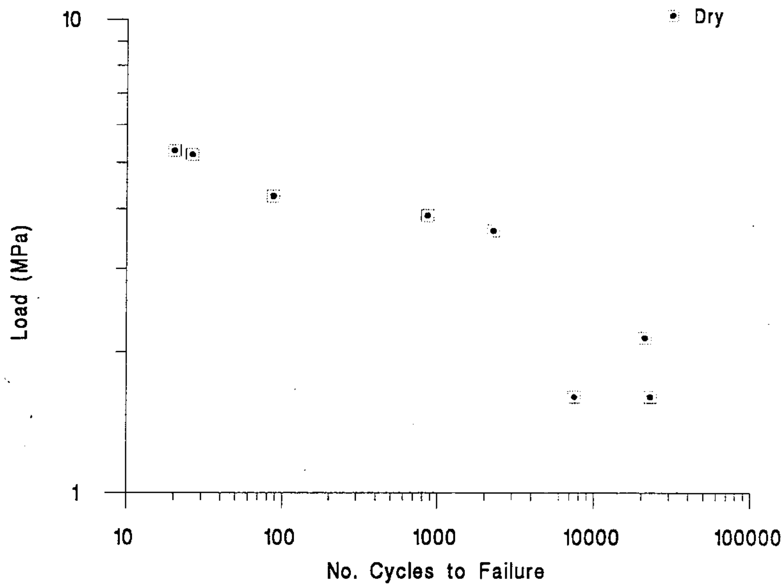


Figure 3

S-N curve of bonded aluminium joints prepared by solvent wiping the substrate with acetone prior to bonding, tested in ambient ("dry") condition.

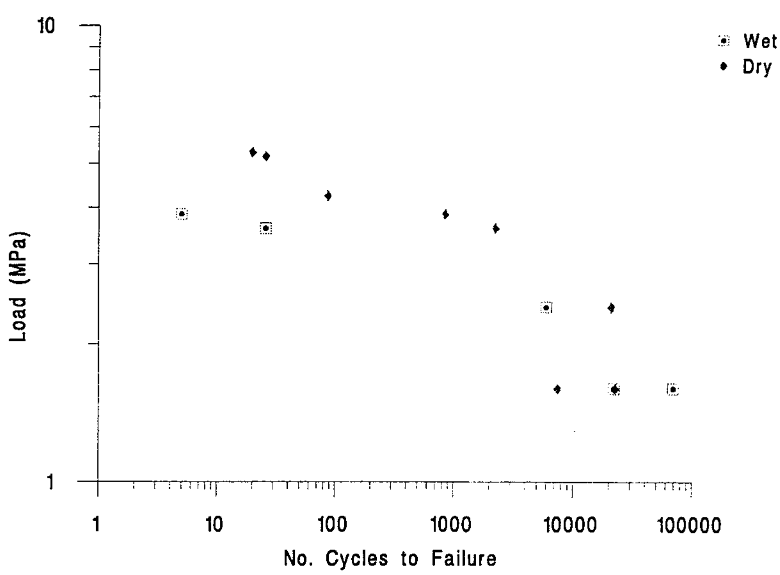


Figure 4

Comparison of S-N curves recorded in ambient ("dry") and salt spray ("wet") environments for solvent wiped samples.

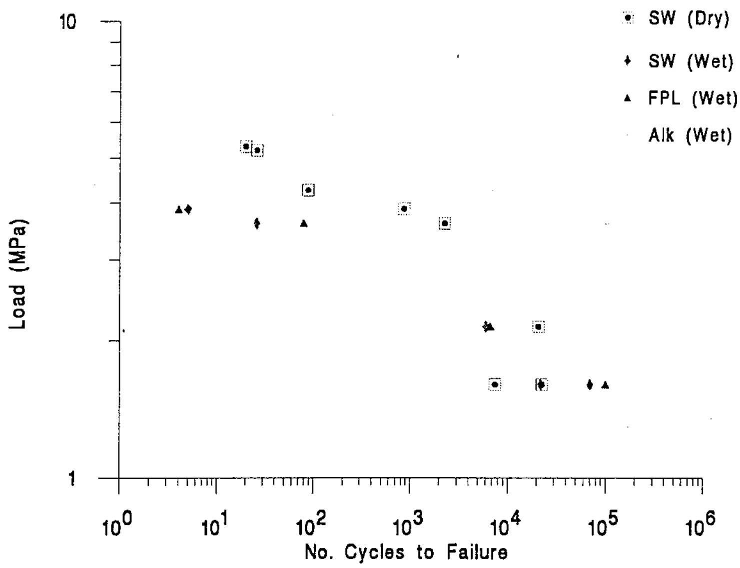


Figure 5

S-N curves for samples tested in salt spray ("wet") environment. Samples having different pretreatments are shown.

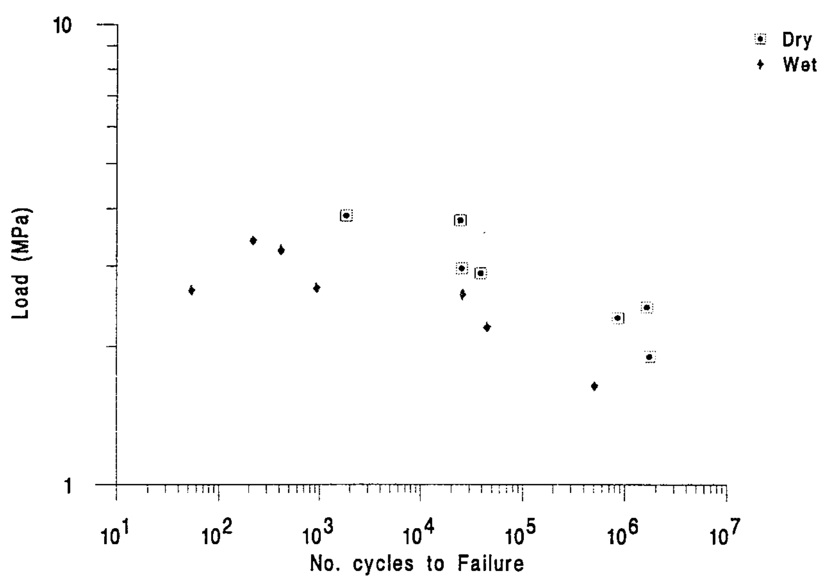
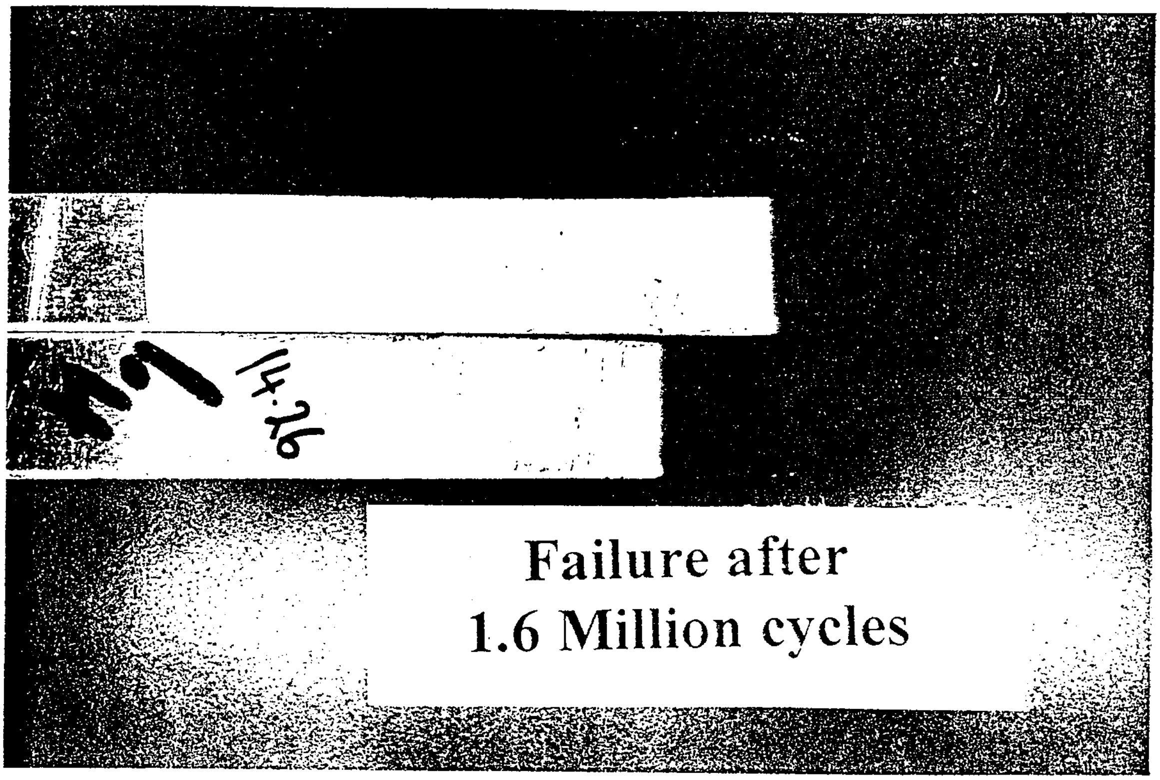
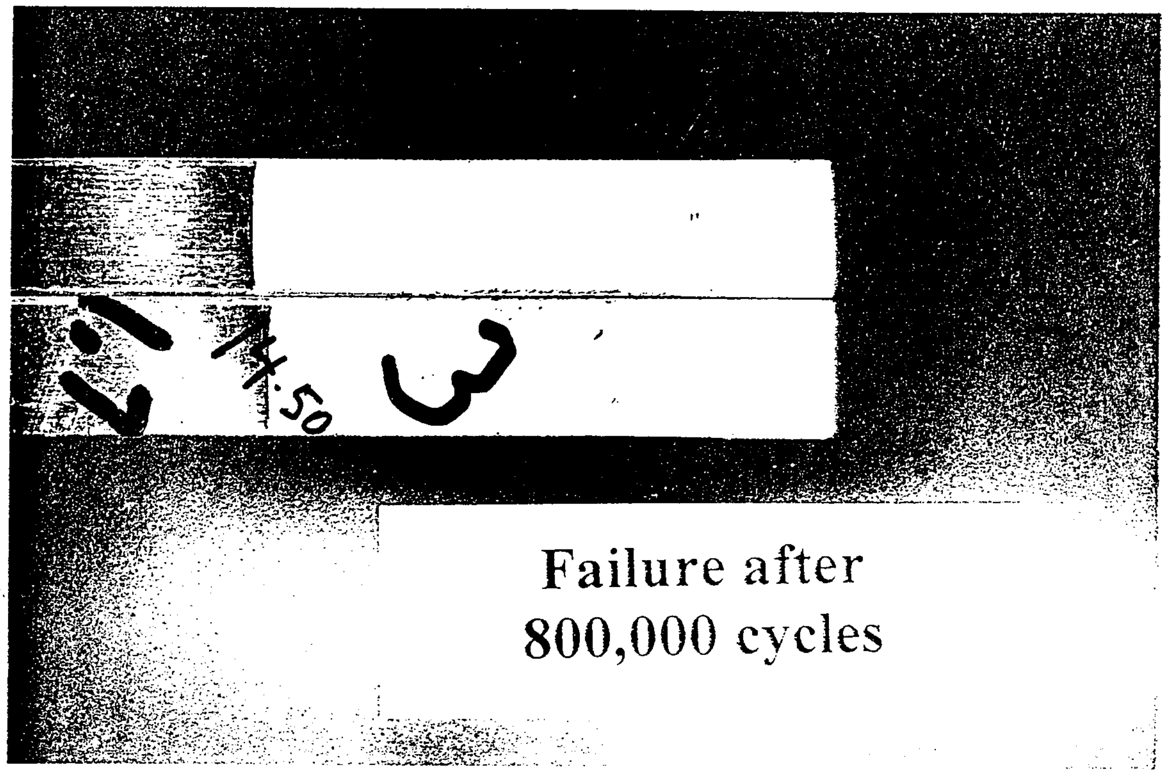


Figure 6

S-N curve for FPL pretreated samples recorded in both ambient ("dry") and salt spray ("wet") environments.





<u>Figure 7</u>

Photos show the difference between failure through the bond (800,000 cycles) and failure through the substrate (1.6million cycles).

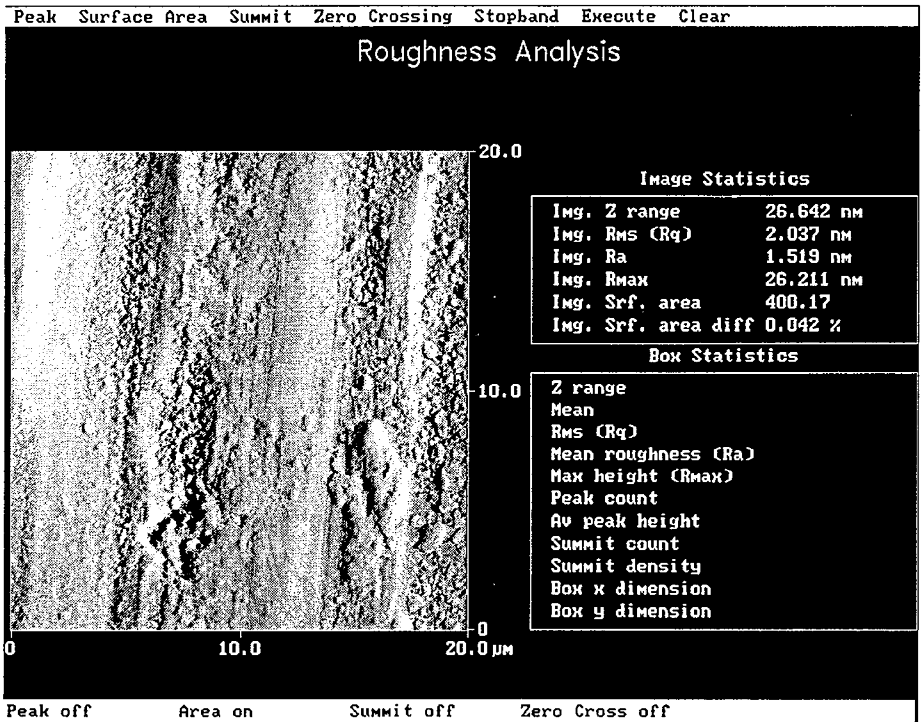


Figure 8

AFM image of an as recieved aluminium sample prior to pretreatment. The average surface roughness (Ra) is 1.5nm.

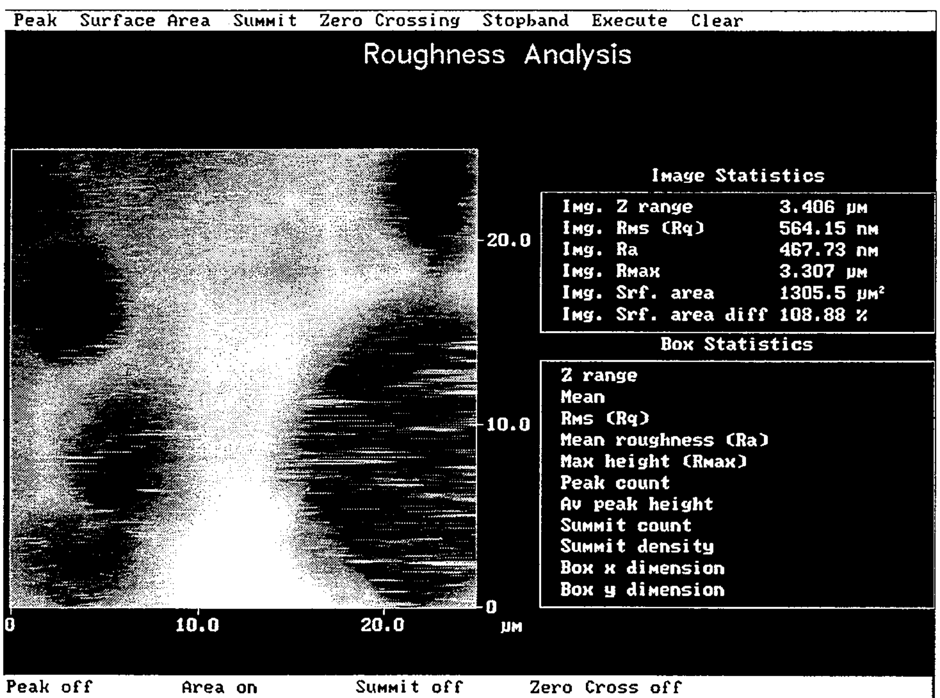


Figure 9

AFM image of an alkaline precleaned sample. The average surface roughness (Ra) is 468 nm.

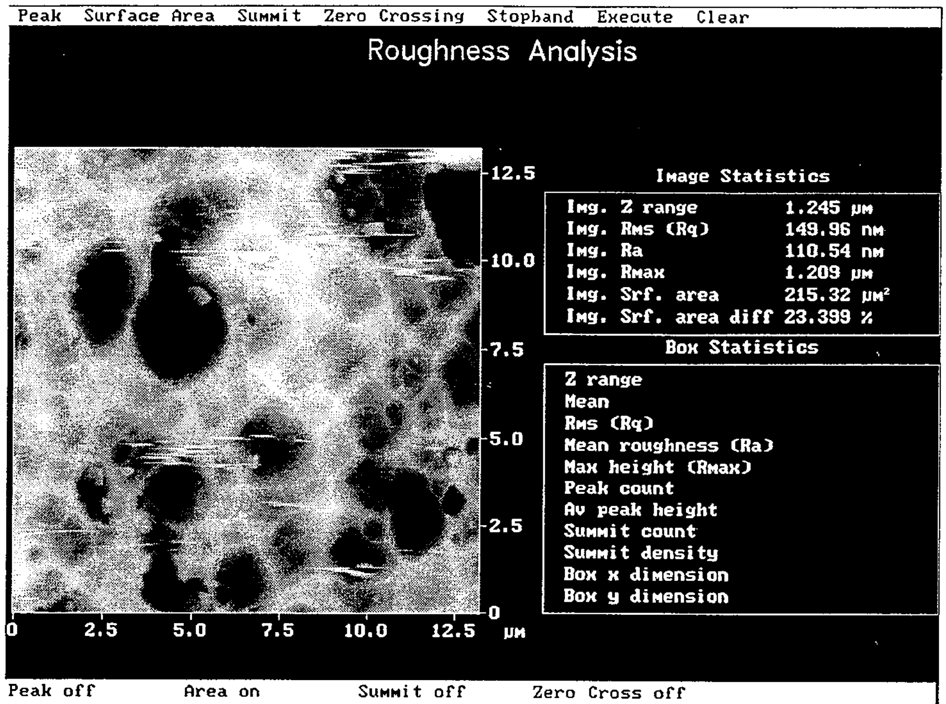


Figure 10

AFM image of an FPL etched sample. The average surface roughness (Ra) is 110 nm.

Paper No 5

Peter Blockley

Managing Director, Maunsell & Partners Pty Ltd

"Offshore Concrete Structures"

This Paper not available at time of printing.

Paper No 6

Dr Patrick Couser

Research Engineer, AMECRC

"Resistance Characteristics of High-Speed Catamaran Forms"

Resistance characteristics of high-speed catamaran forms

by Patrick Couser

Abstract

This paper gives an overview of the work of Molland et al. (1996) which was presented in Southampton at a meeting of the Royal Institution of Naval Architects on 14th November 1995. The work described in this and the aforementioned paper was performed by the author whilst working at the University of Southampton as a Research Assistant in the Department of Ship Science.

An experimental investigation into the calm water resistance components of high speed displacement catamarans with symmetric demihulls has been carried out. The experimental programme was a development and extension of earlier work, at the University of Southampton, in which a small series of three catamaran models were tested.

Total resistance, running trim, sinkage measurements and wave pattern analysis based on a multiple longitudinal cut technique were carried out for ten round bilge hulls derived from the NPL series. The tests were conducted over a Froude number range of 0.2 to 1.0 and separation to length ratios of 0.2, 0.3, 0.4, 0.5 and infinity. Interference effects for the resistance components have been investigated.

The results of the investigation provide a better understanding of the components of catamaran resistance including the influence of demihull separation, length:displacement ratio and length:beam ratio over a wide range of Froude numbers.

1. Introduction

This paper gives an overview of the work of Molland et al. (1996) which was presented in Southampton at a meeting of the Royal Institution of Naval Architects on 14th November 1995. The work described in this and the aforementioned paper was performed by the author whilst working at the University of Southampton as a Research Assistant in the Department of Ship Science.

There is little doubt that the catamaran concept is here to stay. The requirement for faster vessels in a variety of applications is steadily growing; this is especially true for the passenger ferry market. Despite, or perhaps because of the rapid growth of the high speed catamaran market there is little information publicly available for carrying out powering estimates for these vessels, particularly in the high speed range. The exact nature of the calm water resistance breakdown into Froude number and Reynolds number dependent quantities and the exact nature of the interference effects between the catamaran demihulls is not sufficiently well understood. The aim of this work is to clarify some of these points by means of model resistance tests on a systematic series of catamaran forms.

Work on the resistance of high speed displacement catamarans has been ongoing over a number of years at the University of Southampton in an effort to improve the understanding of their resistance components and to provide design data: Insel (1990); Insel and Molland

(1992) and Couser (1996). Other published experimental work on such vessels is limited, but includes that reported in Doctors et al. (1991), Incecik et al. (1991), Matsui et al. (1993) and Müller-Graf (1993).

This paper describes an extensive series of model tests on catamarans in calm water. The experimental programme is a development of the earlier work in which a small series of three catamaran models were tested (Insel 1990, Insel and Molland 1992). The current work has extended the parametric investigation to cover changes in breadth:draught ratio (B/T) and a wider range of length:displacement ratios ($L/\nabla^{1/3}$). As in the earlier work, an approach comprising total resistance measurements together with wave pattern analysis was utilised. A wide range of hull separations was tested and, overall, the experiments covered over 40 model configurations, each over a speed range up to a Froude number of unity. This paper will present some example data and discuss the general findings, for more detailed results readers are referred to Molland et al. (1996) and Molland et al. (1994a) where comprehensive tabulations and plottings of all the test data are given. For a more global overview of catamaran performance, including performance in waves, readers are directed to Couser (1996).

2. Description of models

Details of the models used in the investigation are given in Table 1. (Note that all the values refer to a single demihull.) For all the models the length was 1.6m, the longitudinal centre of buoyancy was 6.4% of the length aft of amidships and $C_B = 0.397$; $C_P = 0.693$; $C_M = 0.565$. Models 3b, 4b and 5b had already been tested some three years earlier and their results published in Insel and Molland (1992) where they are designated C3, C4 and C5 respectively. Some results for these models are included in the present paper for comparison and discussion since they form the basis from which the current, wider series of models was developed.

The models were of round bilge form with transom sterns, Figure 1, and were derived from the NPL round bilge series, Bailey (1976). It is recognised that this hull form is perhaps not in keeping with recent design trends and that better catamaran hull forms exist. However, this hull form is a useful stating point since it extends the existing NPL round bilge series and is broadly representative the underwater form of a number of existing catamarans. In this work we are concerned with the effects of large scale parameters and it may be argued that these effects are largely independent of the detailed hull form design. The models were firstly tested as monohulls and, in the catamaran configurations, with separation:length ratios (S/L) of 0.2, 0.3, 0.4 and 0.5.

The model towing force was in the horizontal direction. The towing point in all cases was situated at the longitudinal centre of gravity and at an effective height one third of the draught above the keel datum. The models were fitted with turbulence stimulation comprising trip studs of 3.2mm diameter and 2.5mm height at a spacing of 25mm. The studs were situated 37.5mm aft of the stem. No underwater appendages or spray rails were attached to the models. For some of the lighter displacement models it was necessary to apply a counter balance. Care was taken with its application whereby the effect on accuracy was negligible.

Table 1: Model details

Model	$L/\nabla^{1/3}$	B/T	L/B	A[m^2]
3b (C3)*	6.27	2.0	7.0	0.434
4a	7.40	1.5	10.4	0.348
4b (C4)*	7.41	2.0	9.0	0.338
4c	7.39	2.5	8.0	0.340
5a	8.51	1.5	12.8	0.282
5b (C5)*	8.50	2.0	11.0	0.276
5c	8.49	2.5	9.9	0.277
6a	9.50	1.5	15.1	0.240
6b	9.50	2.0	13.1	0.233
6c	9.50	2.5	11.7	0.234

* Tested earlier and reported in Insel and Molland (1992).

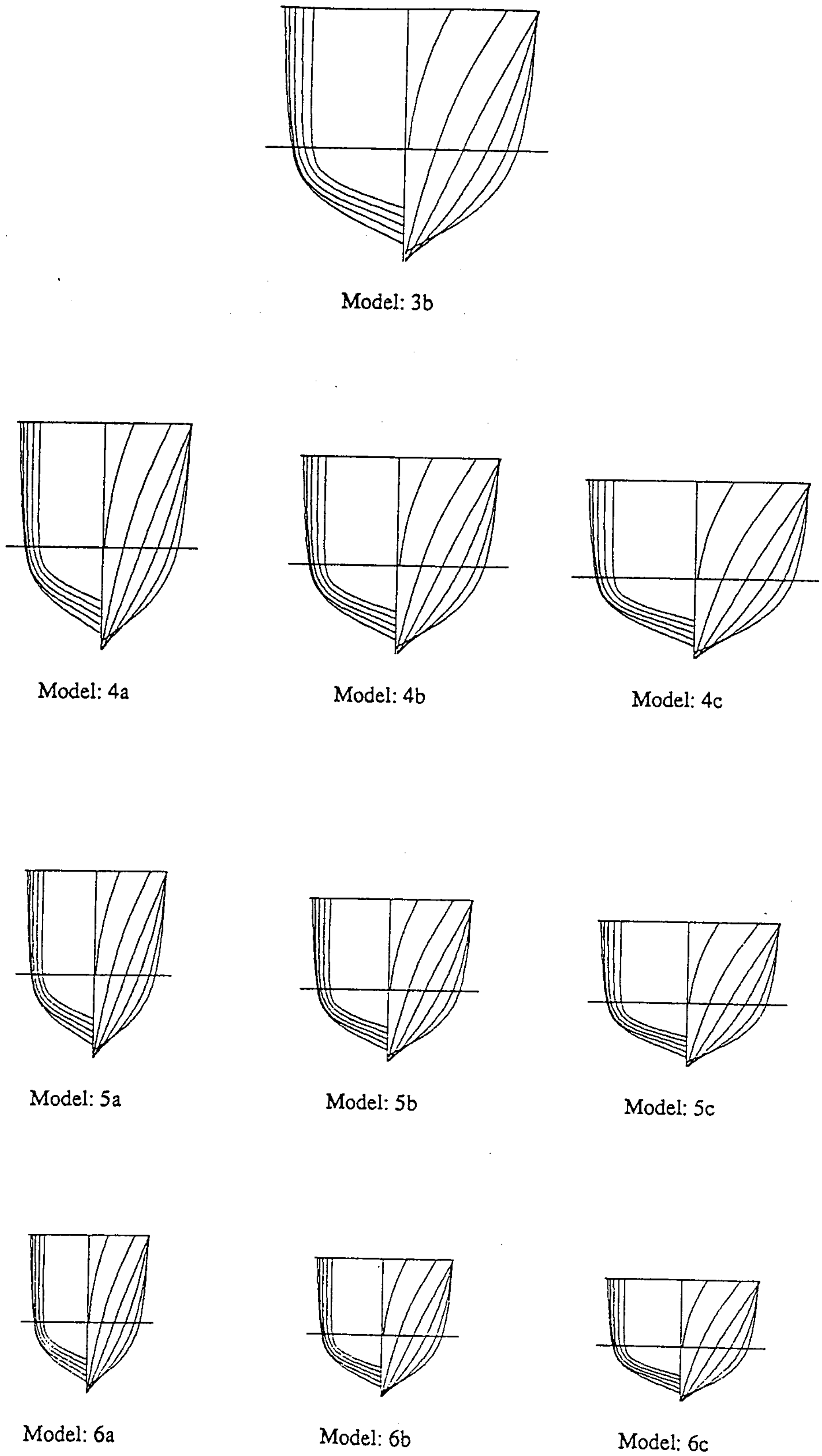


Figure 1: Model body plans and notation

3. Facilities and tests

3.1 General

All the model experiments were carried out in the Southampton Institute test tank. The principal particulars of which are given in Table 2. This tank has very similar dimensions to those of the AMECRC tank in Launceston, Tasmania.

Table 2: Principal tank dimensions

Length:	60.0m
Breadth:	3.7m
Water Depth:	1.85m
Maximum Carriage Speed:	$4.6ms^{-1}$

The tank has a manned carriage which is equipped with a dynamometer for measuring model total resistance together with various computer and instrumentation facilities for automated data acquisition.

Calm water total resistance, running trim, sinkage and wave pattern measurements were carried out for all the models. All tests were carried out, where possible, over a speed range up to a little over $F_n = 1.0$; in some cases, for the catamarans with closely spaced, wide demihull forms, excessive spray precluded high speed runs. Over the Froude number range $0.1 < F_n < 1.0$ the corresponding Reynolds number range for the models was $0.5x10^6 < R_e < 5.5x10^6$.

Total resistance was measured using the carriage dynamometer, the displacement of the flexures was measured using a LDVT. Sideforce was also monitored to ensure that the model was correctly aligned in the tank.

Trim (positive bow up) was measured by means of a potentiometer mounted on the tow fitting; accuracy of this measurement was within ±0.05°. Sinkage (positive downwards) was measured by means of a linear displacement potentiometer with a measurement accuracy within ±0.1mm.

3.2 Wave pattern resistance

A wave pattern analysis based on multiple longitudinal cuts was applied to all the models. The analysis system was fully automated and consisted of four resistance wave probes, a microcomputer based data acquisition system and data analysis software which enabled wave pattern analysis and resistance determination to be performed during standard resistance tests.

The wave pattern analysis used data from four longitudinal wave cuts to determine the Eggers (1955) coefficients of the far-field wave pattern. From these coefficients it was then possible to calculate the energy in the far-field wave pattern and hence the wave pattern resistance. The analysis method took account of the reflected wave system from the tank walls which enabled wave traces of up to 10m to be used in the analysis. The longitudinal positions of the wave probes was carefully chosen to obtain a suitable cosine term in the wave series for every harmonic. A full description of the apparatus and analysis method is given in Insel (1990).

3.3 Low speed tests

Although form factors were principally derived from the total and wave pattern resistance measurements, it was thought worthwhile to attempt to derive from factors from low speed tests using Prohaska's method. Slow speed tests were run with the model in two conditions: first, with the model trimmed bow down with transom emerged and second, with normal trim and the transom immersed. This technique was, for example, mentioned in the discussion to Molland and Insel (1992). It has a number of limitations, but a short investigation into its potential uses was considered worthwhile.

4. Data reduction and corrections

All resistance data were reduced to coefficient form according to standard ITTC practice, see Equation (1), using fresh water density (ρ = 1000.0 kgm^{-3}), model speed (U) and total static wetted surface area (A_{tot} = A, 2A) noting that, for the case of the catamarans, the sum of the static wetted areas of both demihulls was used.

Static wetted surface area as opposed to running wetted surface area was used to non-dimensionalise the resistance data since this was more readily and accurately measured. The routine measurement of running wetted surface area for catamarans with closely spaced demihulls was particularly difficult due to the problems associated with measuring the wave profile along the inside of the tunnel between the two demihulls. Overall it was felt that the measurements of running wetted surface area were not of sufficient accuracy to warrant their use in calculating the non-dimensional resistance coefficients.

$$\text{Resistance Coefficient} = \text{Resistance} / (0.5\ \rho\ A_{tot}\ U^2) \quad (1)$$

The total resistance measurements were corrected to a standard temperature of 15°C. Corrections due to the drag and influence of the turbulence studs were applied. Tank blockage and shallow water effects were estimated using slender body theory. These were found to be negligible below $F_n = 0.60$ and of the order of 4% of total resistance at $F_n = 0.95$, rising rapidly as the shallow water critical speed was approached. However, there is no evidence of the predicted resistance increase due to shallow water in the experimental data and corrections were not applied. More detailed accounts of these corrections, and the justification for using static wetted area are given in Molland et al. (1994a).

5. Presentation of data

The same basic breakdown and presentation of the experimental data has been adopted from the earlier work of Insel and Molland (1992). This is summarised in Equation (2) below:

$$C_{T\,cat} = (1 + \beta k)\ C_F + \tau\ C_W \quad (2)$$

where:

- C_F is obtained from the ITTC-1957 correlation line.
- C_W is the wave resistance coefficient for the demihull in isolation.
- (1+k) is the form factor for the demihull in isolation.
- β is a viscous interference factor.
- τ is the wave resistance interference factor.

It is noted that for the demihull in isolation, β = 1.0 and τ = 1.0.

This resistance breakdown provides a method of determining the resistance of an arbitrary catamaran configuration provided that suitable values for the wave and viscous interference factors and the monhull C_W are known. The interference factors have been calculated and are presented in Molland et al. (1994a) for all the catamaran configurations tested. This provides a relatively simple approach for a first approximation to the resistance of a proposed catamaran design since there is a large database of monohull tank data available. Alternatively C_W or τ C_W may be estimated by numerical methods, e.g: Molland et al. (1994b), Couser (1996). In practice it will often be the case that the final catamaran designs will be tank tested in the appropriate catamaran configuration.

Examples of the measured experimental data are presented in Figure 2. In this figure the wave pattern resistance C_{WP} is plotted downward from the total resistance C_T, in the form (C_T - C_{WP}). The estimates of the form factor, (1 + k) or (1 + βk) for the catamaran, are also shown in the diagrams, these lines being set to the lower envelope of the (C_T - C_{WP}) curves when they settle at an approximately constant level above the ITTC friction line at higher Froude numbers. The values of (1 + k) for the monohulls and (1 + βk) for the catamarans are, for practical design purposes, assumed to remain constant over the speed range.

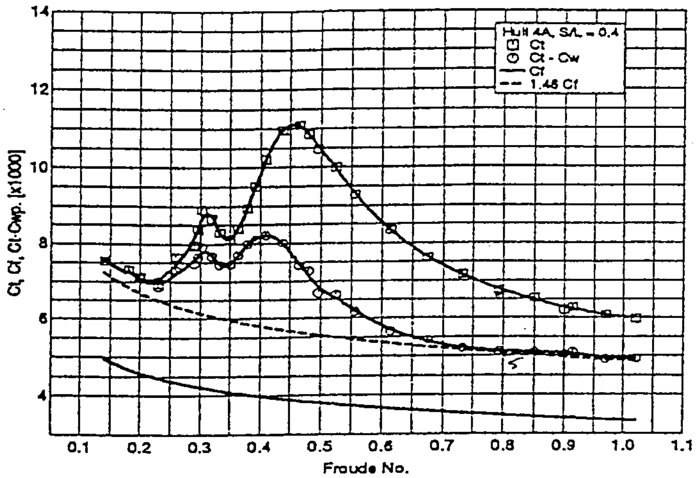


Figure 2: Resistance components, Model 4a, S/L = 0.4

From a practical viewpoint it is not necessary to confine the user to the particular values of (1 + k) or (1 + βk) derived in this work. Following the earlier work some concern was expressed over their magnitudes and application, for example see discussion to Insel and Molland (1992). For these reasons, residuary resistance coefficients C_R (derived from $C_R = C_T - C_{F\ ITTC}$) have been calculated from the experimental data. This presentation provides the data in a form suitable for practical powering applications and an overall comparison of the residuary components for the various hull configurations. The user is able to choose a suitable (1 + k) or (1 + βk) from this work or other sources. Examples of these data are presented in this paper, the complete data may be found in Molland et al. (1996).

6. Discussion of results

6.1 Total resistance and wave pattern resistance

An example of the results of the wave pattern measurements are included in Figure 2, here they are plotted downwards from the total resistance values. The results display a hump (or decrease in measured wave pattern resistance) at a Froude number of about 0.4 before settling down at an approximately constant level above the ITTC correlation line at higher Froude numbers. Observations during the tests indicate that the large hump is due primarily to transom stern and wave breaking effects in this speed range when the transom is just about to run clear. At higher speeds very little, if any, wave breaking was visible except for the lowest length:displacement ratio demihulls at the closet separation:length ratio, suggesting that all the energy associated with the wave resistance component was being transferred to the far-field wave pattern.

The resistance hump and the decrease in measured wave pattern resistance occurs at approximately $F_n = 0.40$ to 0.45. This corresponds to the Froude number at which the transom started to run clear. This also corresponds with the rapid increase in bow up trim and the point at which the maximum sinkage of the centre of gravity occurs; see Section 6.4. The Froude number at which the transom flow started to release was found to be similar irrespective of demihull form or demihull separation. The transom flow regime was found to change very suddenly, and this could be observed during the acceleration phase of each tank run. When the flow had released from the transom a large rooster tail formed behind the transom. The size of the rooster tail diminished and its distance behind the transom was found to increase with increasing forward speed.

6.2 Residuary resistance

Examples of the experimental results are presented in term of residuary resistance coefficient C_R in Figure 3 to Figure 6, where the residuary coefficient has been derived from $C_T - C_{F\,ITTC}$, as discussed in Section 5. This presentation is used in order to provide a readily available tool for powering purposes and a means of comparing the relative merits of changes in the hull form parameters.

The results in Figure 3, for fixed B/T = 0.2 and S/L = 0.3, clearly show the influence of length:displacement ratio as it is increased from Model 3b to 6b. With increase in length:displacement ratio the main resistance hump becomes less pronounced and the Froude number at which it occurs decreases slightly. As might be expected length:displacement ratio was found to be the primary hull parameter affecting residuary resistance. The same general trends were observed for the other hulls tested irrespective of breadth:draught ratio and separation:length ratio in the case of the catamarans.

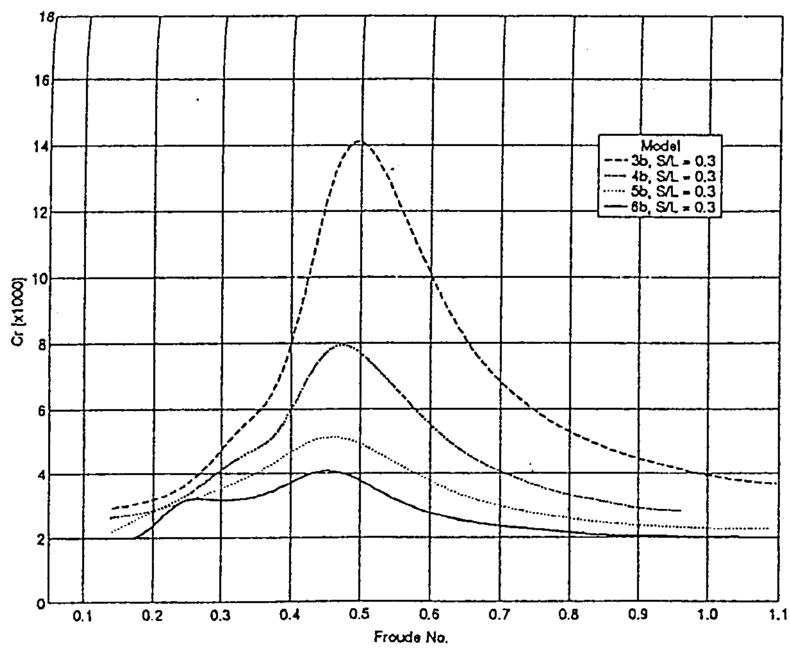


Figure 3: Effect of $L/\nabla^{1/3}$ on C_R, Models 3b, 4b, 5b, 6b (S/L = 0.3)

Figure 4 and Figure 5 show the influence of breadth:draught ratio at two length:displacement ratios. The influence of breadth:draught ratio is seen mainly in the lower Froude number range up to about 0.6, and differences in C_R of up to 10% due to changes in breadth:draught can occur in this region. However trends are not systematic and the lines cross and re-cross. In the higher Froude number range, at speeds often representing service speeds for this type of hull form, the following trends may be observed: For the highest length:displacement ratio (Models 6a to 6c), Model 6a with the smallest breadth:draught ratio tends to have the largest residuary resistance coefficient. For the low length:displacement ratio (Models 4a to 4c) the trend has been reversed and Model 4a (with the smallest breadth:draught ratio) tends to have the lower residuary resistance coefficient. The differences in residuary resistance coefficient are typically of the order of 3% to 4%. Similar trends are found in the Series 64 data, Yeh (1965).

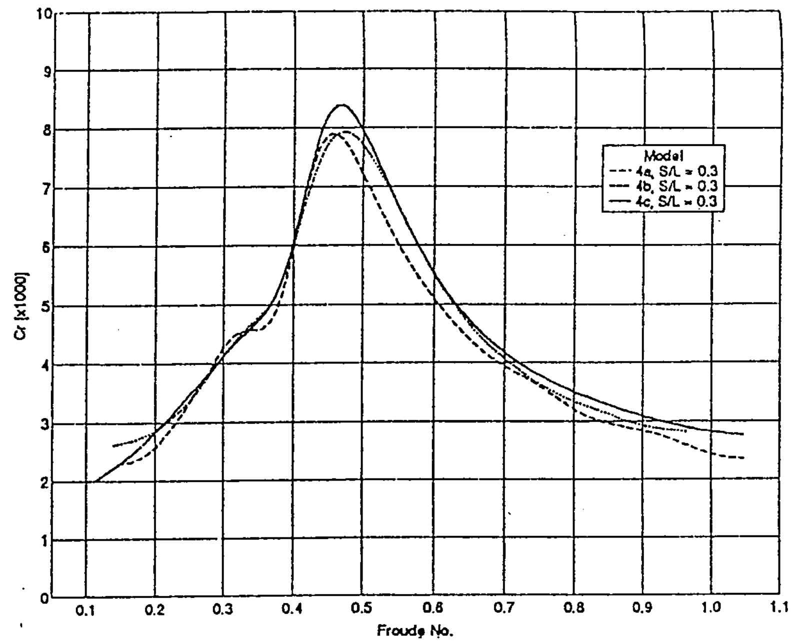


Figure 4: Effect of B/T on C_R, Models 4a, 4b, 4c (S/L = 0.3)

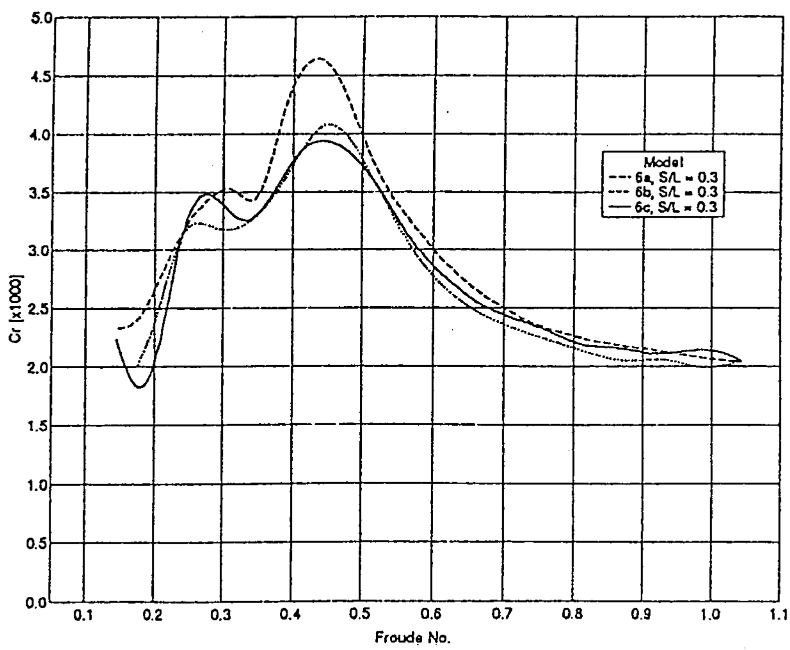


Figure 5: Effect of B/T on C_R, Models 6a, 6b, 6c (S/L = 0.3)

Figure 6 and Figure 7 give the results for each of the catamaran spacings tested for constant length:displacement and breadth:draught ratios. The monohull results are also shown in the figure. The general trend, apparent in all cases, is that as the demihull separation is increased the resistance decreases and the main resistance hump occurs at decreasing Froude number. It is noted that, in the higher speed range, changes in hull separation tend to have a relatively small effect. There is however an increase in residuary resistance for the catamaran compared with the monohull, and this increase becomes a larger proportion of the monohull residuary resistance as length:displacement ratio increases from Models 3 to Models 6. It should be noted, however, that, for the curves of wave resistance coefficient, the results for all the catamarans and the monohull converge to the same value at high Froude number. This is because the residuary resistance coefficient contains some viscous component and it appears that significant viscous interference exists even at the widest separation:length ratio tested. These phenomena are discussed in greater depth in the following section.

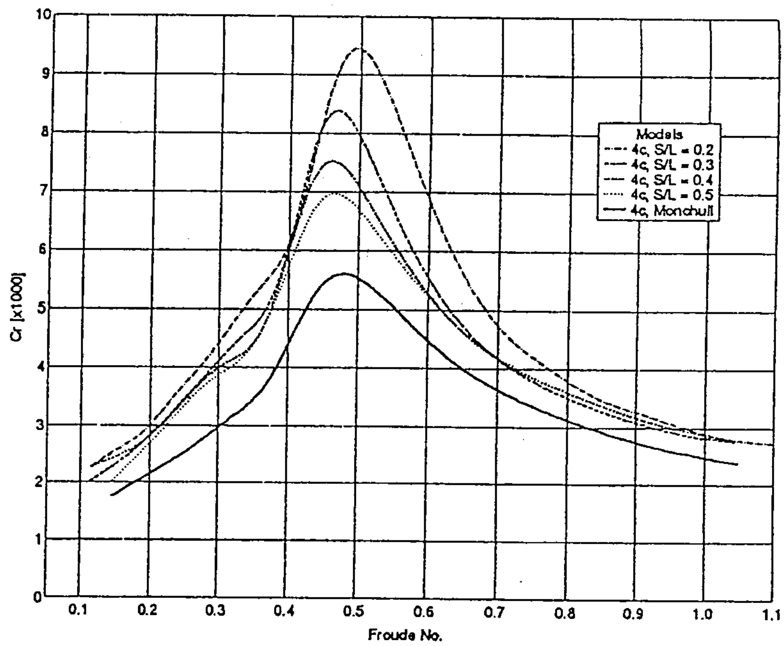


Figure 6: Effect of S/L on C_R, Models 4c

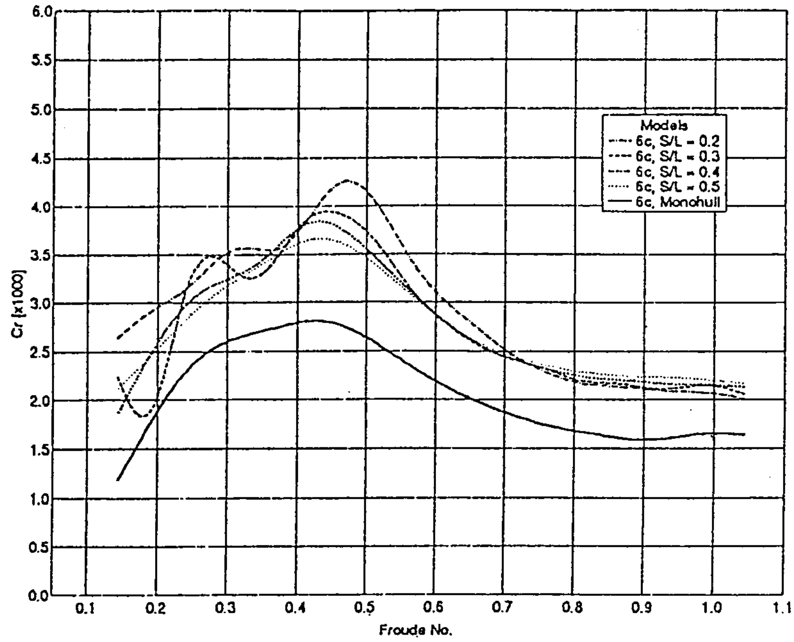


Figure 7: Effect of S/L on C_R, Models 6c

6.3 Viscous resistance and form factors

6.3.1 General

Form factors (1 + k) for the monohulls and form factors for the catamarans including viscous interference (1 + βk) were obtained by deducting the wave pattern resistance from the total resistance as described in Section 5.

The resulting values of (1 + k) and (1+ βk) for the various configurations are summarised in Table 3 and the viscous interference factors (β) for the catamarans are given in Table 4. As discussed in Section 5, these factors need not necessarily be used directly for design or resistance scaling purposes, but they do provide a broad indication of changes in viscous resistance and viscous interference due to changes in length:displacement, breadth:draught and separation:length ratios. It should also be noted that since static wetted surface areas have been used to obtain the non-dimensional resistance coefficients, the form factors are greater than they would be had running wetted surface area been used. The running wetted surface area for these types of vessel may be of the order of 14% to 17% greater than the static wetted surface area at the higher Froude numbers; use of the running wetted surface area instead of the static would lead to corresponding reductions in resistance coefficients and hence form factors.

6.3.2 Monohulls

For the monohulls, reference to Table 3 indicates a decrease in (1 + k) with increasing length:displacement ratio (Models 3 to 6). This was determined by Insel (1990) and Insel and Molland (1992), and would be expected physically. For each length:displacement ratio no statistically significant change in (1 + k) with change in breadth:draught ratio was determined. The monohull form factors were somewhat higher than might be expected for such vessels. However, they were of the same order as those found for a similar vessel tested by Tanaka et al. (1990/91) although much higher than those found by Cordier and Dumez (1993).

6.3.3 Catamarans

For the catamarans, Table 3 indicates $(1 + \beta k)$ values to be higher than the corresponding monohull $(1 + k)$ values, indicating $\beta \neq 1.0$ and hence some viscous interference between the demihulls as well as the form effect of the demihulls. Part of this increase would be negated due to additional wave breaking between the demihulls which would not be present for the monohulls. If this additional wave breaking were the present, less wave resistance energy would propagate to the far-field, leading to decreased values of C_{WP} and subsequent overestimates of $(1 + \beta k)$. Observations at the time of the tests suggest that, in most cases, this effect should not be significant.

Table 3: Monohull and catamaran form factors $(1 + k)$, $(1 + \beta k)$

Model	$L/\nabla^{1/3}$	B/T	Monohull	S/L=0.2	S/L=0.3	S/L=0.4	S/L=0.5
3b	6.27	2.0	1.45	1.60	1.65	1.55	1.60
4a	7.40	1.5	1.30	1.43	1.43	1.46	1.44
4b	7.41	2.0	1.30	1.47	1.43	1.45	1.45
4c	7.39	2.5	1.30	1.41	1.39	1.48	1.44
5a	8.51	1.5	1.28	1.44	1.43	1.44	1.47
5b	8.50	2.0	1.26	1.41	1.45	1.40	1.38
5c	8.49	2.5	1.26	1.41	1.43	1.42	1.44
6a	9.50	1.5	1.22	1.48	1.44	1.46	1.48
6b	9.50	2.0	1.22	1.42	1.40	1.47	1.44
6c	9.50	2.5	1.23	1.40	1.40	1.45	1.44

Table 4: Catamaran viscous interference factors (β)

Model	$L/\nabla^{1/3}$	B/T	S/L=0.2	S/L=0.3	S/L=0.4	S/L=0.5
3b	6.27	2.0	1.33	1.44	1.22	1.33
4a	7.40	1.5	1.43	1.43	1.53	1.47
4b	7.41	2.0	1.57	1.43	1.50	1.50
4c	7.39	2.5	1.37	1.30	1.60	1.47
5a	8.51	1.5	1.57	1.54	1.57	1.68
5b	8.50	2.0	1.58	1.73	1.54	1.46
5c	8.49	2.5	1.58	1.65	1.62	1.69
6a	9.50	1.5	2.18	2.00	2.09	2.18
6b	9.50	2.0	1.91	1.82	2.14	2.00
6c	9.50	2.5	1.74	1.74	1.96	1.91

As with the monhull tests, the form factor $(1 + \beta k)$ was found to decrease with increasing length:displacement ratio. However, there were no apparent, statistically significant trends relating $(1 + \beta k)$ to either breadth:draught ratio or separation:length ratio. The fact that there is no apparent reduction in viscous interference with increasing separation:length ratio is perhaps surprising; thus it must be surmised that there is still significant viscous interaction between the hulls at $S/L = 0.5$, this is after all much less than infinity — the value of separation:length ratio for the demihull in isolation.

It may be noted, from Table 4, that the viscous interference (β) is more important for the most slender demihull forms (Models 6). This is because for these demihull forms the viscous resistance is by far the greater proportion of the total resistance, this is highlighted in Figure 8.

This is particularly true at model scale where the relatively low Reynolds number (compared with full scale) results in larger C_F and hence C_V.

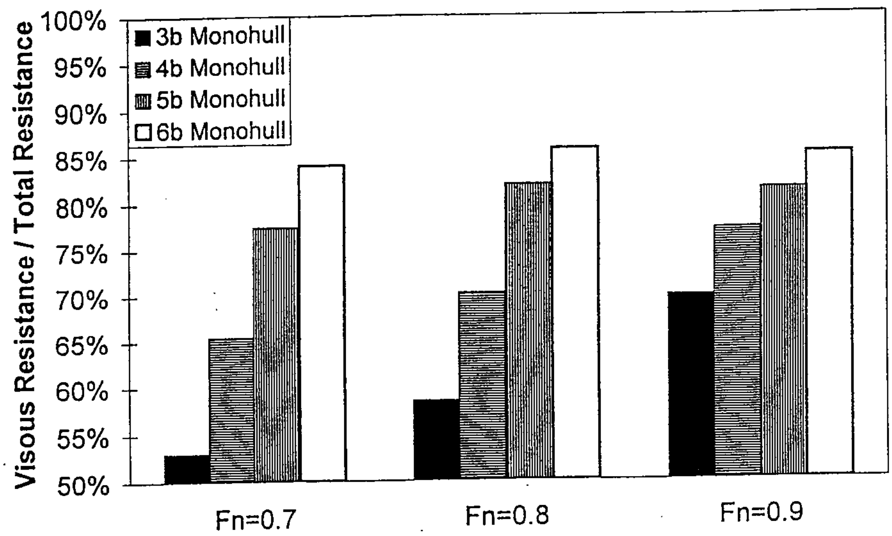


Figure 8: Variation of proportion of viscous resistance with $L/\nabla^{1/3}$, Models 3b, 4b, 5b, 6b Monohull

6.3.4 Low speed tests

The results of the low speed for catamaran Model 4a at S/L = 0.5, with the transom immersed and emerged, are shown in Figure 9. The results with the transom immersed (normal trim condition) are much more erratic than with the transom emerged. This is likely to be due to the highly turbulent, chaotic wake and vortex / eddy shedding in the region of separated flow behind the deeply immersed transom.

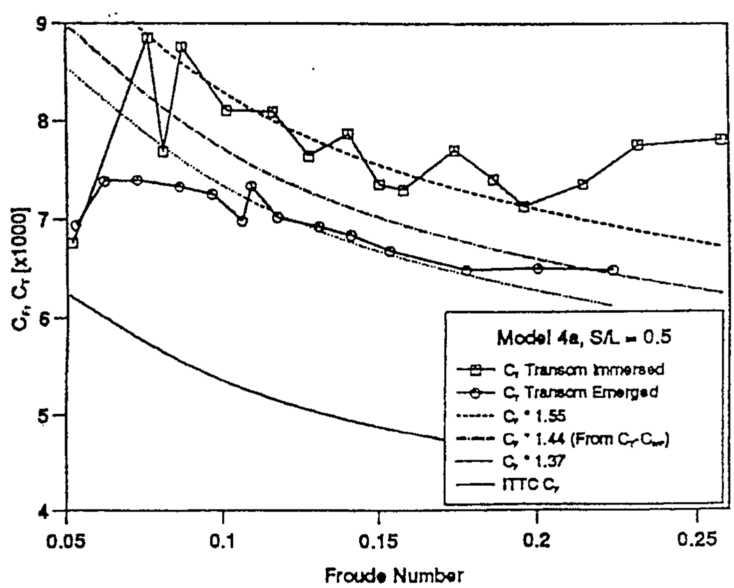


Figure 9: Form factor from slow speed tests, Model 4a, S/L = 0.5

The slow speed tests indicate a $(1 + \beta k) = 1.55$ for the normal trimmed condition and 1.37 for the transom emerged case. Prohaska's method was also applied to these particular data and similar values for $(1 + \beta k)$ were found. Table 3 (where all the $(1 + \beta k)$ values have been obtained by the $(C_T - C_{WP})$ method) indicates a value of 1.44 for this model.

These results tend to confirm earlier deductions that viscous form and interaction effects are present, although they may be smaller than the values suggested by the (C_T - C_{WP}) method; noting that the (1 + βk) value obtained by the (C_T - C_{WP}) method could be somewhat reduced if the resistance coefficients were non-dimensionalised using running rather than static wetted surface area.

Taken overall, and compared with the normal trim condition, the (1 + βk) value derived from the bow down / transom emerged tests was in broad agreement with the value obtained from the wave pattern analysis. In both cases the transom was running clear, indicating that when the transom is immersed and not releasing it has a substantial effect on the flow resulting in an increase in resistance.

It is finally noted that the slow speed bow down / transom emerged tests should be treated with caution due partly to the low resistance forces measured at low speed and the fact that the forward trimmed hull form will be different (although not necessarily significantly) from the actual normal trimmed condition.

6.4 Running trim and sinkage

Typical of the results of the trim and sinkage measurements are presented in Figure 10 and Figure 11 respectively. Trim is given in degrees, with a bow up trim being positive. Sinkage is the vertical downward displacement of the centre of gravity, expressed as a percentage of static draught; positive sinkage implies an increase in draught at the centre of gravity. The overall results and trends are in broad agreement with published monohull data such as Lahtiharju (1991) and Tanaka et al. (1990/91).

In all cases, trim angle interference is important between F_n = 0.3 and F_n = 0.7 where the catamaran displays significantly higher trim angles than the monohull, but generally approaches the monohull trim angle as separation:length ratio is increased. It was found that as length:displacement ratio is increased (when going from Models 3 to 6) there is a decrease in running trim. As breadth:draught ratio is increased for a given length:displacement ratio (when going from Models 'a' to 'c') the changes in running trim are relatively small.

In general, as length:displacement ratio is increased (when going from Models 3 to 6) there is a decrease in running sinkage. It was found that as breadth:draught ratio is increased for a given length:displacement ratio (when going from Models 'a' to 'c') there tends to be an increase in sinkage or lift effects for the fuller models, particularly at higher speeds.

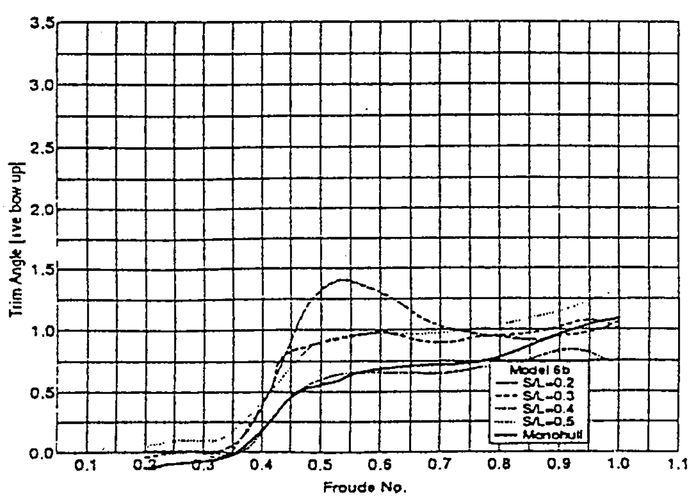


Figure 10: Running trim, Models 6b

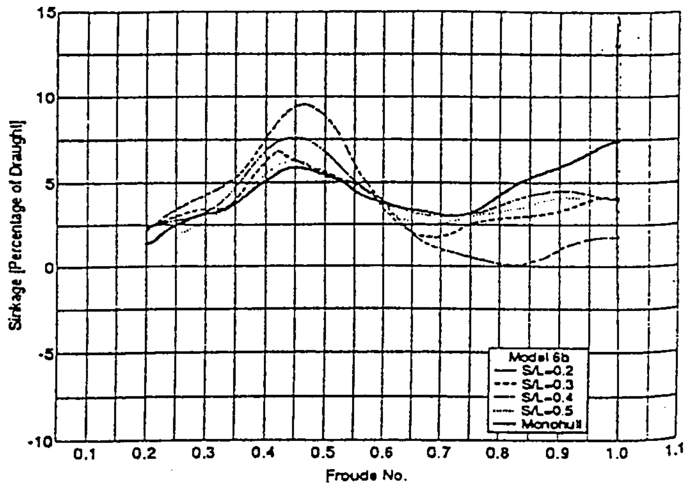


Figure 11: Running sinkage, Models 6b

7. Conclusions

7.1. The results of the investigation provide further insight into the influence of hull parameters on the resistance components of high speed displacement catamarans, and offer a very useful extension to the available resistance data for this vessel type.

7.2. length:displacement ratio was found to be the predominant hull parameter, resistance decreasing with increasing length:displacement ratio as might be expected for higher speed displacement vessels.

7.3. The effect of breadth:draught ratio on resistance was not large. Changes in resistance due to changes in breadth:draught ratio were however identified in particular ranges of speed and length:displacement ratio which could warrant attention at the hull design stage. In the main, increase in breadth:draught ratio led to an increase in resistance in the lower length:displacement ratio range and a decrease in resistance at the highest length:displacement ratio.

7.4. The catamaran displays significantly higher running trim angles than the monohull, but generally approaches the monohull angle as separation:length ratio is increased. Changes in running trim due to changes in breadth:draught ratio are relatively small. As breadth:draught ratio is increased there is an increase in running sinkage / lift effects for the fuller models, particularly at higher speeds.

7.5. Form factors for the catamarans were consistently higher than the corresponding monohulls, suggesting some viscous interference between the hulls as well as the form effect of the demihulls. The form factors were found to be effectively independent of speed and to be dependent primarily on length:displacement ratio and to a much lesser extent on hull separation and breadth:draught ratio.

7.6. Bow down / transom emerged tests indicated that the viscous form and interference factors may be lower than those derived directly from the total minus wave pattern resistance results. Whilst the total minus wave pattern resistance method provides very useful information on the general changes in wave pattern and viscous resistance, further work is required to justify and confirm the magnitude of the total viscous term. It should be noted that the form factors derived by the (C_T - C_{WP}) method would be reduced by the order of 14% to 17% were running wetted surface area rather than static wetted surface area used to non-dimensionalise the resistance. This would bring these form factors into closer agreement with those found by Prohaska's method.

7.7. Based on observations during the tests a significant presence of spray and wave breaking was not apparent. Any presence of either or both of these components would however lead to a reduction in the derived viscous form factors.

Acknowledgements

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Nomenclature

Demihull	One of the hulls which make up the catamaran (in the current investigation all demihulls are symmetrical)
L, L_{BP}	Length on still waterline [m]
A	Static wetted surface area (demihull) [m^2]
B	Demihull maximum beam [m]
T	Demihull draught [m]
S	Separation between catamaran demihull centrelines [m]
∇	Volume of displacement (demihull) [m^3]
U	Velocity [ms^{-1}]
F_n	Froude Number, [$U/\sqrt{(gL)}$]
R_e	Reynolds Number, [UL/υ]
C_B	Block coefficient
C_M	Midship section area coefficient
C_P	Prismatic coefficient
C_R	Coefficient of residuary resistance [$R_R/(0.5\rho AU^2)$]
C_T	Coefficient of total resistance [$R_T/(0.5\rho AU^2)$]
C_W	Coefficient of wave resistance [$R_W/(0.5\rho AU^2)$]
C_{WP}	Coefficient of wave pattern resistance [$R_{WP}/(0.5\rho AU^2)$]
C_F	Coefficient of frictional resistance [ITTC-1957 Correlation line]
$1+k$	Form factor
β	Viscous resistance interference factor
τ	Wave resistance interference factor
g	Acceleration due to gravity [ms^{-2}]
ρ	Density of water [kgm^{-3}]
υ	Kinematic viscosity of water [m^2s^{-1}]

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