ON THE EFFECT OF HULL SHAPE ON THE PERFORMANCE OF SOME EXISTING HIGH-SPEED FERRIES

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ABSTRACT

The hydrodynamic characteristics of twelve hull forms of existing high-speed vehicular and passenger-only catamaran ferries with a length greater than 40 metres are presented and discussed, together with 2 other model-tested hull forms, covering both resistance and ship motions. Comment is made on the actual performance on trials when compared against the tank test results, together with the results from various numerical prediction methods. The hull forms include round-bilge, soft-chine and hard-chine shapes, representing semi-displacement hulls and Semi-SWATH hulls, and includes the effects of motion control devices. The effect of bulbous bows and transom sterns is discussed. The hull forms cover a range of speeds up to a Froude number of 1.2. Although not a systematic series, the trends resulting from variations in hull shape and characteristics are evident.

1. INTRODUCTION

There are several papers published on systematic series of hull forms which provide an insight into various performance characteristics, such as resistance, for catamaran arrangements. Most of these use a hull shape that was originally intended for monohull applications, such as the NPL hull used by Molland et al (1995), and others have developed a hull specifically for catamaran usage, such as Müller-Graf (1993). There are also several papers which examine similar issues using hull shapes which are mathematically defined, such as the Wigley hull. As far as is known, none of these hull shapes have been used to build a commercial or military catamaran.

High-speed ferries are typically catamarans, or other arrangements of long and thin hulls, such as the trimaran, because of the advantage that this provides for low resistance and good seakeeping, whilst maintaining a very high level of stability and safety.

Austal Ships is a major manufacturer of high-speed ferries, and has based its approach to the market on the concept of unique solutions to specific Client’s needs. Each ship is therefore generally a unique design, tailor-made for the clients stated speed, cargo capacity, ease of loading and unloading, sea conditions, depth of water, classification society choice, flag survey and engine manufacturer preferences. Sometimes a Client may order multiple craft, in which case these are all the same hull, but each individual Client has generally enjoyed the benefits of a novel hull shape. This could only be done with an experienced design team with the necessary software to enable it to design in a very short time frame.

2. BACKGROUND

Twelve existing commercial hull forms were selected for a comparative analysis of resistance and ship motion. It is important to recognise that some hulls appear to be better than others in resistance and motion because these are not the only two criteria by which a ship should be judged, and there are many other characteristics that lead to the selection of a hull; for example the vessel may be required to operate in rough water, or may have been designed for minimum motions whilst having a limited length or draft for docking purposes.

Ten of the twelve hull forms have been built, and therefore trials results as well as model tests were available. The two hulls that have not been built were extensively model tested. Part-body plans are illustrated in Fig.1.

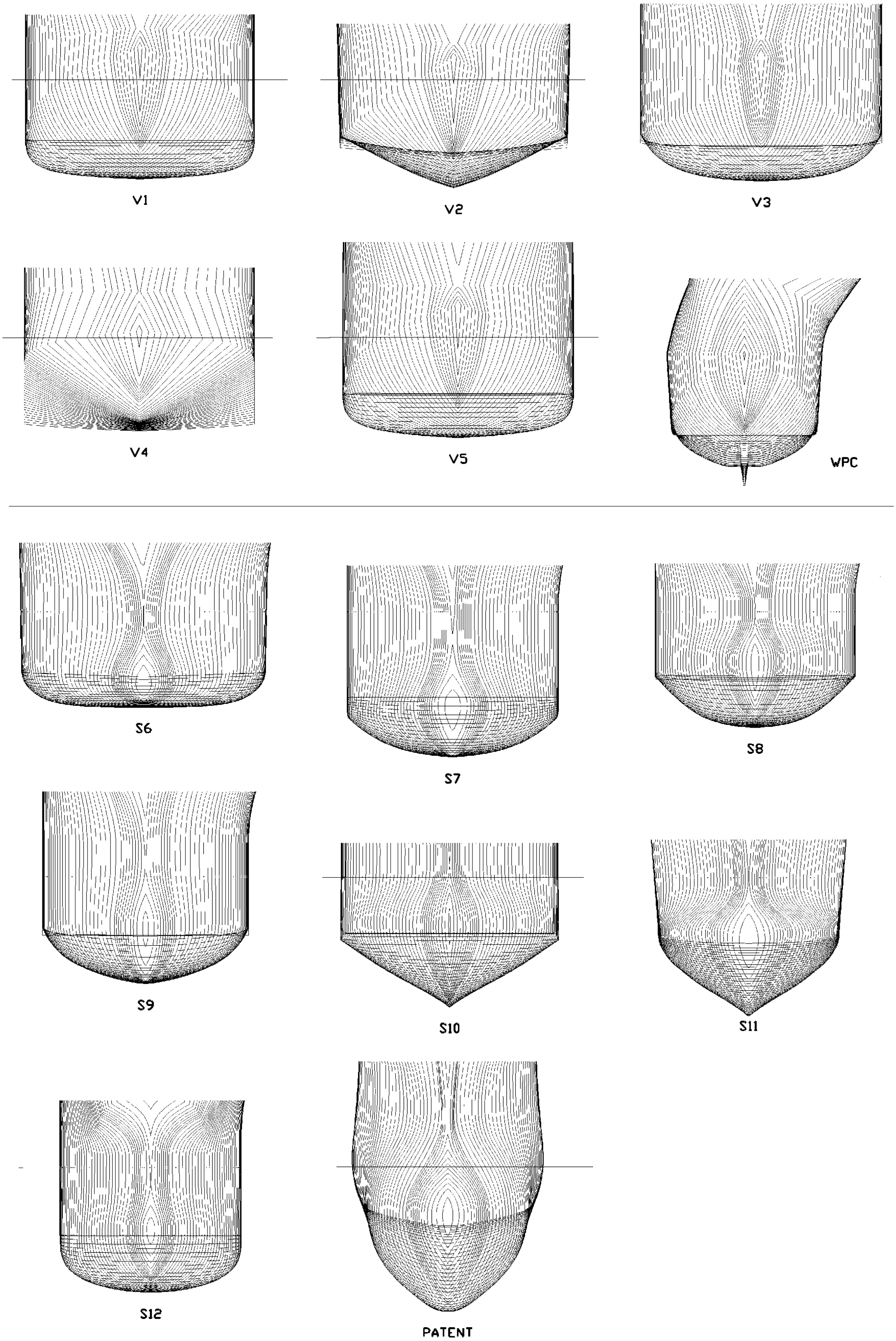

Fig.1: Body plans of the fourteen hulls in this analysis.
The top six illustrate the V-series and the lower eight illustrate the S-series.
Waterlines represent a displacement of 260 tonne.

In addition to these twelve commercial hull forms, two other shapes which have been built by others have also been analysed. These are a catamaran hull shape defined by a patent, Knudsen *et al* (1996), and a wavepiercing catamaran *WPC* similar to several of those in service. The wavepiercing design was an estimated hull shape based on publically-available details including the displacement, draught and demihull length and beam.

Because the fourteen hulls represent various sizes of ferries, from the small passenger-only ferry up to the largest vehicular ferry, there is a spread of displacements and associated capabilities, which makes it difficult to make a fair comparison. It is relevant to discuss these differences and describe the design process undertaken by a commercial shipbuilder, with the following distinct steps:

1. The Client's requirements and market analysis will be used to estimate the length of vessel.

2. An arrangement drawing will be prepared to confirm the vessel length and overall beam. Typically this will involve generating a car deck layout which will accommodate all of the specified vehicles in accessible locations, together with all of the necessary ramps and access ways. If the vehicle space only has access over the stern, then the deck arrangement has to be wide enough to accommodate a turn-around area for the vehicles, or alternatively the stern ramp will need to be wide enough to permit trucks and coaches to back off safely and quickly. The car deck layout must permit a rapid turn-around time. Sometimes a bow door may be fitted to reduce the turn-around time. Equipment such as stern and bow ramps need to be long enough to reach the wharf, and this may impact on the relative lengths of the hulls and the main deck.

3. The overall beam of a vehicle ferry is usually some multiple of 2.4 metres, the average car width.

4. On passenger ferries the passenger cabin length and beam is dictated by the relevant seat pitch and width, together with the necessary access, although the width is flexible because toilets, food preparation and other facilities may be grouped along the centreline. Care needs to be taken that the passenger cabin is not extended so far forward that the ship motion causes excessive seasickness within the cabin.

5. There will always be interference effects between the two hulls of a catamaran, at any spacing. An acceptable level of interference can be inferred from published data on the various craft. A good rule of thumb is to ensure that the aspect ratio of length to clear width of the tunnel space between the hulls is never more than 7.

6. It is generally desirable to have a narrow demihull, and the beam of the demihulls is directly linked to the width required for the waterjet nozzles. Similarly it is generally desirable for resistance to have a minimum transom immersion, and this will correspond with the vertical location of the waterjets in a light ship condition such that the immersion allows them to self-prime.

7. Sometimes the hull beam is the minimum permitted by the width of the main engines and transmission arrangements.

Two of the defining characteristics of a high-speed ferry hull form are the slenderness ratio $L/\nabla^{1/3}$ where ∇ is the demihull volume of displacement, and the thinness ratio *L*/*b* where *b* is the demihull beam on the waterline. This investigation does not include the slenderness ratio, as the analysis of the performance of the hull shapes was based on the same length and displacement. The slenderness ratio will generally follow the trend illustrated in Fig.2. Thinness ratio and beam-to-draught ratio of the analysed hulls are illustrated in Fig.3.

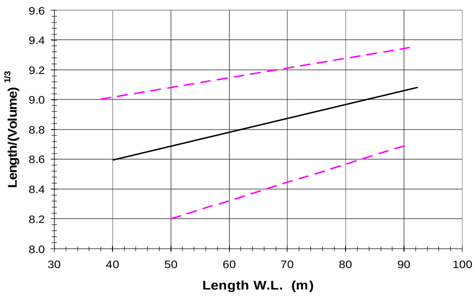

Fig.2: Range of demihull Slenderness Ratio of typical Fast Ferries

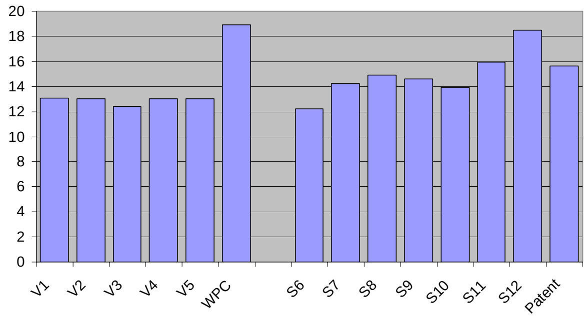

Fig.3(a): Length-to-demihull beam *L/b* of the hulls

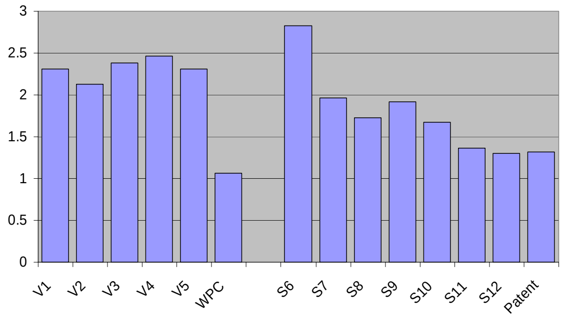

Fig.3(b): Demihull beam-to-draft *b/T* ratio
Displacement = 260t

The analysed hull forms are generally representative of modern-day shapes, characterised into two groups:

- *passenger ferry hull shapes* which are generally designed for high Froude numbers *Fn*, with V-shaped forward sections to generate lift and hence reduce resistance; and
- semi-displacement semi-SWATH shapes typically used for *vehicular ferries*, meaning that they are SWATH-like in the forebody, but more conventional in the afterbody. The afterbody is more full because of the need to fit propulsion machinery and waterjets. The forebody is characterised in that there is a bulbous swelling shape below the waterline, which is intended to dampen ship motions whilst permitting a very narrow angle-of-entrance for minimum wavemaking resistance.

Hulls *V1* to *V5* represent the high *Fn* designs intended to generate dynamic lift. *V1* is very similar to *V5*, although intended to operate at a different Length/ Displacement ratio, and consequently has a slightly smaller beam. Hulls *V3* through *V5* are related in that they were alternative designs for the same project, with *V4* being an entirely hard-chine hull version of *V3*, which was hard chine at the bow but quickly washed out into a round bilge at midships and aft. Hull *V4* also had a considerably deeper transom in order to investigate an anticipated reduction in dynamic trim and hence reduce resistance. Hulls *V2* and *V4* are derived from similar origins as *WPC*. Hull *V5* was a variation on *V3* with the chines being lifted at the bow, minimum practical wetted surface area, and minimum transom area consistent with waterjet priming at lightship.

All of the high-*Fn* hull shapes (*V1~V5*) have elongated hulls, commonly called a *Z-bow*, with the intention of lengthening the waterline without increasing the overall length, and minimising the angle-of-entrance.

Hulls *S7* to *S12*, and the *Patent* hull form, represent semi-SWATH semi-displacement hulls designed for *Fn* values below 0.8. The exception is *S6*, which although intended to have a high *Fn* was deliberately designed as a semi-SWATH in order to better suit the sea conditions for the area for which it was designed. On the other hand, the *WPC* which although intended for “lower” *Fn* has V-shaped sections. It is thought likely that this was done for ease of construction. Hull shapes *V2* and the *WPC* are related, although they have some major differences in distribution of volume because they are designed for very different Length-Displacement ratios, the *WPC* being originally designed at about twice the length of *V2*.

There are a variety of bow bulb shapes. The *Patent* hull is very SWATH-like forward, with a large bulb located quite high. *S11* has a large bulb which is flattened on top in order to investigate potential increased damping whilst pitching. *S6* has a very low bulb.

Most of the bulbous shapes are designed to assist with locating the longitudinal centre of buoyancy *LCB* close to the centre of gravity associated with the ship layout, rather than to obtain the minimum resistance. It is also necessary to avoid encroaching on the patent claims of Bystedt (1991) which describes a semi-SWATH hull form with a bulb, and this generally results in rather narrow bulbs being used on all the S-series shapes summarised in Table 1.

Table 1: Description of the bow bulb shapes

Hull name	Bulb length	Bulb height	Bulb size
S6	medium	v. low	fat & small
S7	short	low	average
S8	short	mid	average
S9	short	mid	thin
S10	medium	mid	thin
S11	medium	mid	large, flat-top
S12	short	mid	thin
Patent	long	high	fat & large

For a meaningful comparison of hull shapes it is necessary that the hulls all have the same length and displacement; therefore all of the hull shapes have been scaled geometrically such that they are geosym models of the full-sized craft but with the same waterline length of 44 meters. The displacement was chosen as 260 tonnes, a mean value such that the waterline for all models was reasonably close to the design location. This scaling does have the unfortunate effect that the waterline will not be exactly at the designed waterline, but the waterline does remain within a reasonable range for each craft, with the exception of model *WPC*, where the waterline is too deep relative to its intended design waterline. The waterline for the *Patent* hull is low and that for *S7* is high compared to their design waterlines.

The hull separation was kept constant at 8.43 meters between demihull centrelines in order to keep constant the hull interference effects in the resistance analysis, and to assist with the analysis of the effect of hull shape on rolling motions with a constant hull beam.

3. RESULTS AND DISCUSSION

3.1 Resistance

Results of calm water resistance from model testing were available for all of the hull forms except for three of the S-series. The numerical potential flow code *Shipflow* was used on all models and a correlation factor was found for residuary resistance coefficient C_R against the model test results. The correlation was based on Froude number Fn and $L/\nabla^{1/3}$, and varied between 1.2 at 20 knots and 1.6 at 40 knots. Different correlation lines were obtained for the V-series hulls and the S-series hulls. The S-series correlation was then applied to the models that had not been tank-tested to obtain a realistic estimate of the resistance of the hulls.

Speed-dependent form factors, based on the form parameters of $L/\nabla^{1/3}$, b/T and C_B in accordance with Armstrong (2000), were applied to all of the model test results to obtain the final values of C_R. Air resistance was not included. Fig.4 shows the C_R values for the various hulls in both V and S-series.

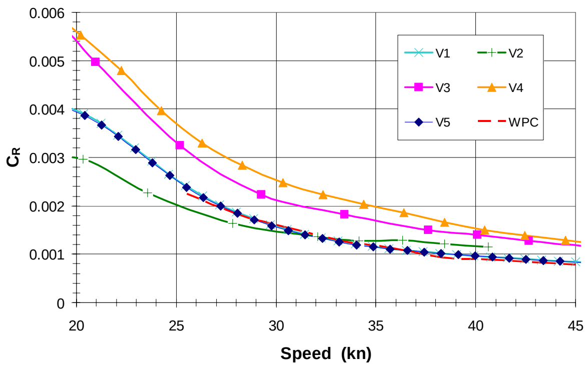

Fig.4(a): C_R values, V-series

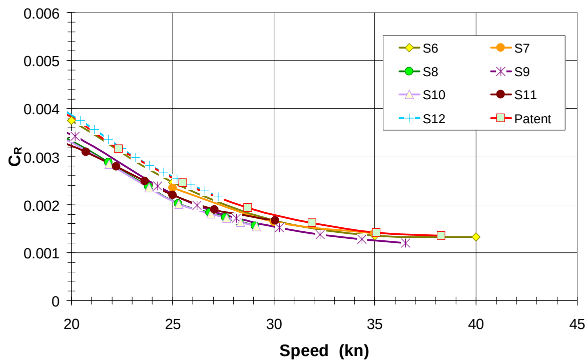

Fig.4(b): C_R values, S-series

The residuary resistance coefficients illustrated above demonstrate a wide variation for the V-series, but close agreement for the S-series. Whilst all models have the same length and displacement, they do not have the same wetted surface area, and neither do they have the same transom area. At the high speeds under consideration, the flow in the region of the transom will separate, and the transom will be at atmospheric pressure, which represents a loss of hydrostatic pressure recovery and therefore can be treated as a resistance component equivalent to $\rho gh.A_{TR}$ where h is the average depth of the transom below the waterline and A_{TR} is the area of the transom, as discussed by Doctors & Day (1997). The resistance owing to flow separation at the transom for each of the hull shapes is illustrated in Figures 6 and 7. An indication of the importance of transom resistance is indicated in Fig.6, where for shape *V4* it has the same magnitude as frictional and wavemaking resistance. The residuary resistance C_R is considered to be made up of wavemaking and transom drag:

$$C_R = C_W + C_{TR}$$

The frictional resistance is a major component of the total drag for high-speed ships, and the successful design of a hull shape with low resistance is obviously a question of balancing the need for a low wavemaking resistance with a low wetted surface area, together with a low transom immersion.

Fig.5 illustrates the difference in the wetted surface area for all models. Generally the V-series have a lesser surface area than the S-series. *WPC, S10, S11, S12* & *Patent* all have relatively larger wetted areas which has a consequential effect on the total resistance.

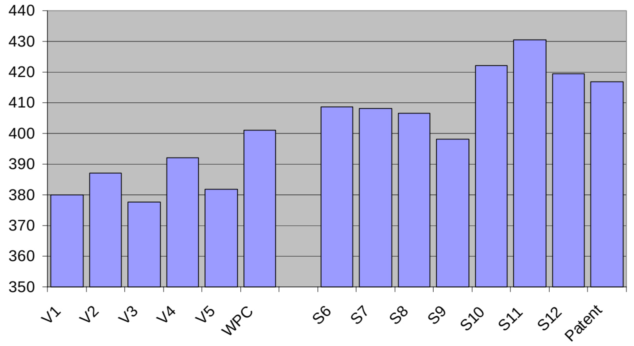

Fig.5: Wetted surface Area (m^2) of all hulls

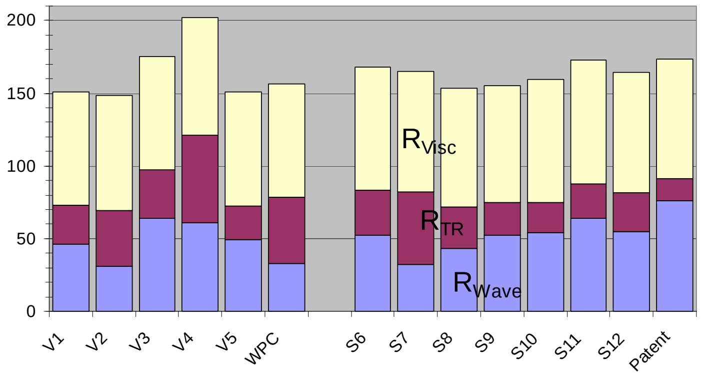

Fig.6: Resistance in kiloNewtons for all 44m hulls, at 30 knots, showing Viscous, Transom and Wavemaking resistance components

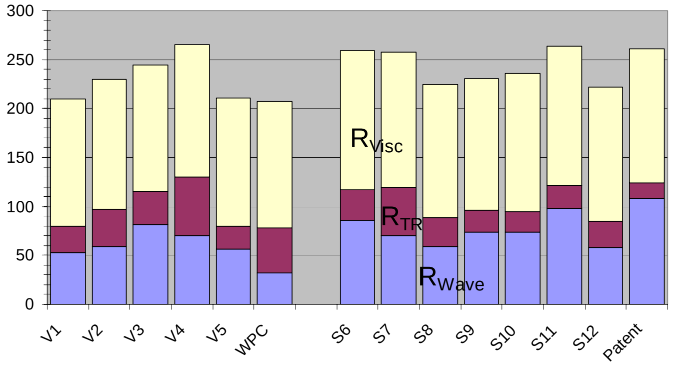

Fig.7: Resistance in kiloNewtons for all 44m hulls, at 40 knots, showing Viscous, Transom and Wavemaking resistance components

The residuary resistance is also dependent on the dynamic trim and sinkage of the hull, as this not only affects the wavemaking resistance but also the apparent transom immersion. It is usually always advantageous to include some device to reduce dynamic trim, such as a wedge or transom flap or interceptor, which can generally achieve level dynamic trim on a fast ferry at *Fn* of about 0.8. The dynamic trim of some of the tank tested models is illustrated in Fig.8. It should be noted that the dynamic trim of the full scale vessel will be different to that of the model tests because of the action of the waterjet, as reported by Armstrong (2000).

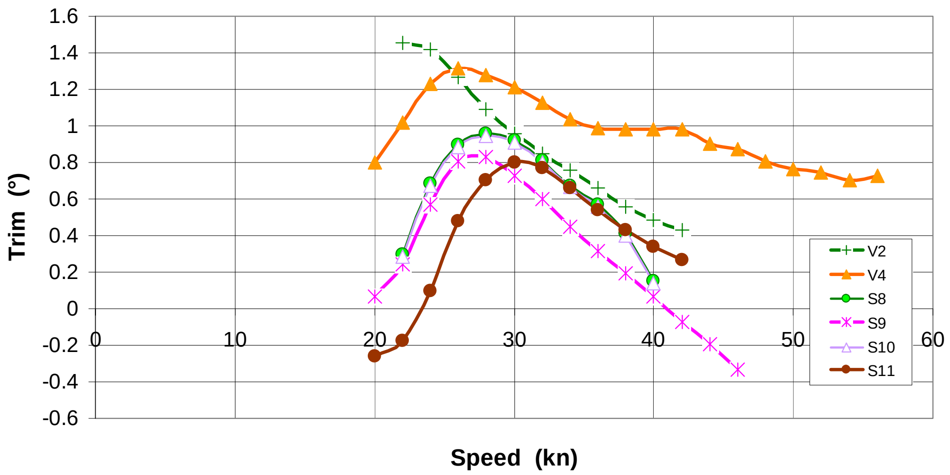

Fig.8: Dynamic trim from model tests. Hull *V4* was not fitted with a wedge. *V2* & *S11* had a small wedge.

Speed trials were carried out in calm water on all of the twelve hull shapes that were built as fast ferries. Equating the predicted thrust from the waterjets with the nominated level of engine power against the resistance curve from the model test results gives the predicted full-scale speed, and this was achieved in all instances. However, as with all fast ferries it is generally necessary to correct the results for shallow water effects, because fast ferries are trialled in calm water in order to minimise speed loss in waves, dictating trials in inshore waters where there is usually a depth effect. Therefore most fast ferries will record a higher speed on trials than when operated in deep water. Corrections can be obtained from trials on vessels in both shallow and deep water, which has been found to fit the theory reasonably well. Once the speed is corrected (reduced) for shallow water effects, it is generally necessary to add a correlation factor as a separate resistance coefficient to allow for marine growth on the hull and waterjet components, and variations in the thrust owing to the main propulsion thrust also being used to steer the vessel, together with several other reasons relating to the practical tolerances of the waterjet. These correlation factors are quite small, but owing to the high-speed of fast ferries the effect can be appreciable. Correlation factors have been found for the twelve hulls analysed here of between C_A = 0.0002 and 0.

3.2 Ship Motion

There are a variety of ways to demonstrate ship motions, but for fast ferries it is passenger comfort that is the most important outcome of ship motions. Therefore four characteristics were chosen by which to assess the influence of hull shape on the overall ship performance.

These were:
Vertical acceleration at the centre of gravity - *VFE*
Lateral acceleration at the centre of gravity - *LFE*
Motion Sickness Index for 2 hr exposure - *MSI*
Roll angle

Vertical acceleration is generally accepted as being a measure of passenger discomfort. Lateral acceleration relate to an ability to stand up or walk without holding on to avoid tripping or sliding or otherwise loosing balance. Roll angle is included because at excessive angles passengers can become concerned about the vessel's ability to remain upright, leading to anxiety and associated discomfort.

The *MSI* levels were determined according to the method of O'Hanlon & McCauley (1976) and McCauley (1977), assuming a two-hour exposure period.

The ship motions were calculated using *VERES*, a strip theory code, which was validated against measurements taken during trials on a number of multihull craft, including one that is included in this study, and from tank testing of a military multihull craft. Similar validations have been reported by O'Dea (2005). In the following studies, the wave environment is assumed to be a 1 metre significant wave height with a Pierson-Moscovitz spectrum in long-crested seas, at a wave period and at the (worst) heading that gave the highest result.

Several of the S-series hulls were specifically designed to have motion control systems comprising of active fins mounted at the bow and active interceptors at the transom stern. A study of passenger comfort is relatively meaningless for a bare hull, as commercial ferries would normally include a motion control system, and therefore additional investigations were conducted into the effectiveness of fitting a 1.66m^2 T-foil at the bow of each hull for several of the models, together with stern interceptors. Table 2 shows the reduction in accelerations, *MSI* values and Roll angles that were achieved with this active T-foil system on all hulls, and which can be applied directly to the results given in Fig.9 to Fig.12.

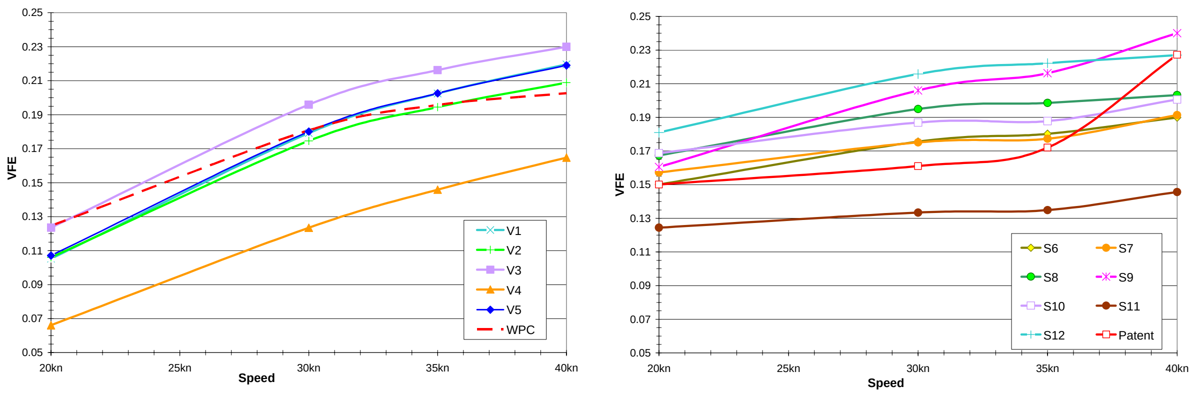

Fig.9: VFE values (rms vertical acceleration m/s^2) for V-series and S-series, no motion control system

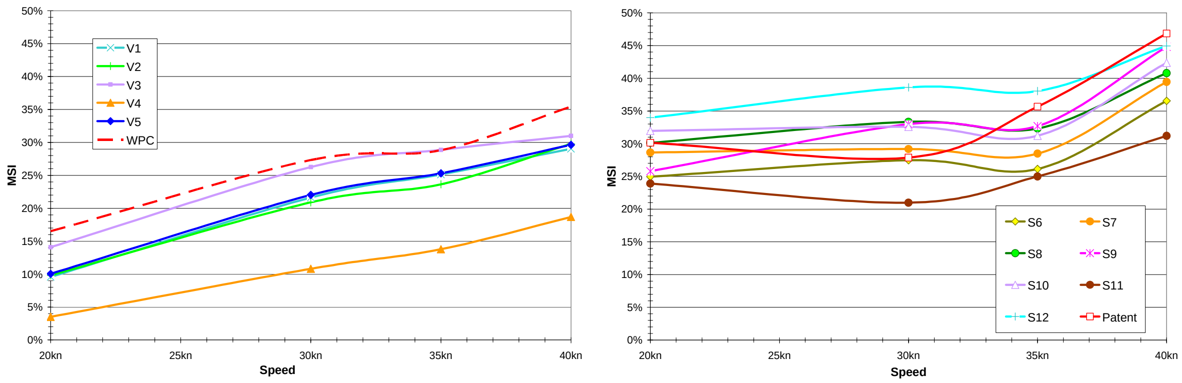

Fig.10: *MSI* values (% passengers sick in 2-hr period) for V-series and S-series, no motion control system

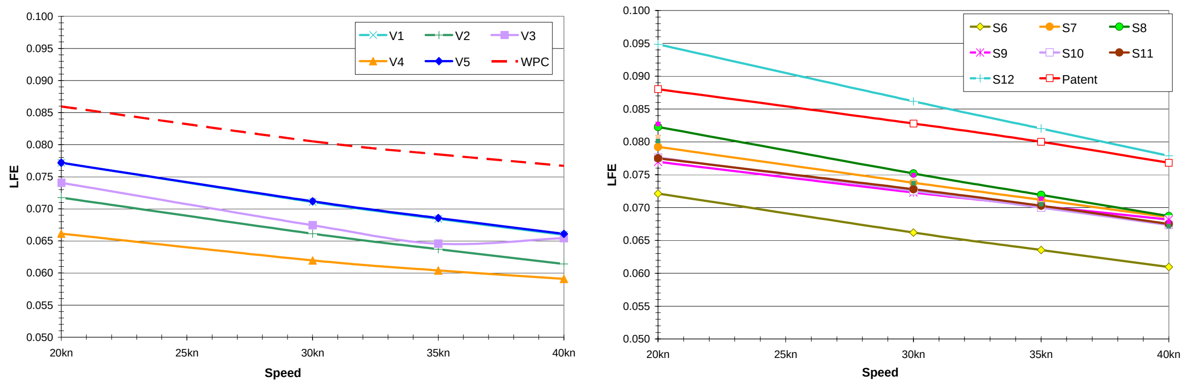

Fig.11: LFE values (rms lateral acceleration m/s^2) for V-series and S-series, no ride control system

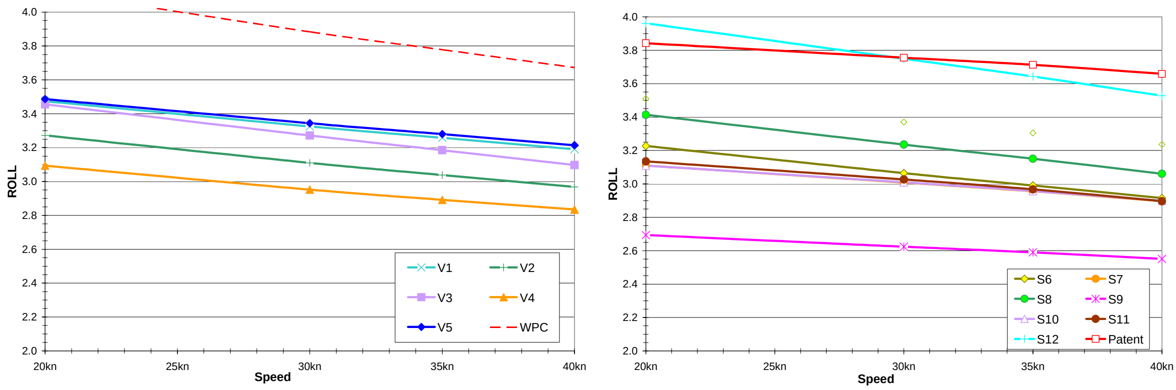

Fig.12: ROLL values (rms roll angle °) for V-series and S-series, no ride control system

Table 2: Reduction Factors to apply to ship motion results in Fig.9 ~ Fig.12 when fitted with a Motion Control System, according to correlated strip theory code *VERES*

VFE				
	20kn	**30kn**	**35kn**	**40kn**
S6	0.78	0.71	0.68	0.61
S8	0.76	0.67	0.64	0.60
S9	0.76	0.67	0.64	0.57
S10	0.76	0.68	0.66	0.59
S11	0.85	0.80	0.77	0.69
S12	0.71	0.63	0.60	0.57

LFE				
	20kn	**30kn**	**35kn**	**40kn**
S6	0.87	0.82	0.80	0.81
S8	0.83	0.72	0.79	0.90
S9	0.86	0.78	0.76	0.83
S10	0.83	0.73	0.79	0.90
S11	0.76	0.72	0.72	0.80
S12	0.81	0.64	0.66	0.75

MSI				
	20kn	**30kn**	**35kn**	**40kn**
S6	0.66	0.54	0.50	0.45
S8	0.64	0.53	0.49	0.35
S9	0.64	0.54	0.49	0.46
S10	0.65	0.53	0.50	0.35
S11	0.72	0.66	0.41	0.36
S12	0.61	0.52	0.49	0.37

ROLL ANGLE				
	20kn	**30kn**	**35kn**	**40kn**
S6	0.84	0.66	0.55	0.42
S8	0.82	0.63	0.51	0.39
S9	0.84	0.67	0.56	0.44
S10	0.82	0.62	0.49	0.36
S11	0.60	0.45	0.36	0.28
S12	0.80	0.60	0.47	0.37

4. CONCLUSIONS

The analysis was conducted at only one Length/ Displacement ratio $L/\nabla^{1/3}$, consistent with a 44-metre length craft. It is known that the results will differ for other ratios, such as will occur with longer vessels like vehicular ferries.

4.1 Conclusion on effect of hull shape on resistance

In this study, at 40 knots, the lowest total resistance was achieved by three of the V-series, *V1, V5 & WPC*. Hull shape *V4* had the highest resistance as a result of the deep transom immersion, the relatively high wetted surface area and possibly the lack of an adequate wedge, closely followed by *S11, Patent, S6 & S7*.

At 30 knots, the best of the V-series and the S-series hull shapes had similar resistance. The best hulls were *V1, V2, V5, S8 & S9*, and the worst hull was *V4*, followed by *V3, Patent, S11, & S6*.

Hull shape *WPC* had the lowest wavemaking resistance coefficient, but was penalised by a high transom immersion and a high wetted surface area. This hull has a high *L/b* ratio and a low *b/T* ratio.

Hull *V3* had the least wetted surface area, but was not a good performer owing to high C_W. This shape has a low *L/b* ratio and high *b/d*. No general conclusion could be reached with regard to the influence of these ratios on resistance; it was noted that the hull shapes with the lowest resistance, *V1 & V5*, had low *L/b* and high *b/T*, but then so did the worst performer *V4*.

S11 had the highest wetted surface area, and had high resistance, especially at higher speeds.

The influence of the bulb was difficult to assess because of the limited data. Generally it was felt that a fat and large bulb such as those on *Patent*, *S11 & S6*, was not effective at any speed. A thin bulb appeared to give good results at 30 knots, but appeared to make no difference at 40 knots.

4.2 Conclusion on effect of hull shape on bare hull motions

Although hull shape *V4* was a poor performer in terms of resistance, it was by far the best performer of the V-series in terms of ship motions, giving the lowest accelerations, sea-sickness and roll angle. Hull shape *WPC* generally gave the highest motions, although *V3* had the highest values of vertical acceleration and *MSI* at high speeds.

Generally the V-series hulls gave lower *MSI* and lateral accelerations than the S-series, but the vertical accelerations and roll angles were similar, except at speeds below 30 knots where the V-series had very low vertical acceleration.

The motions of *S11* were outstanding, and gave the lowest vertical acceleration of all hull shapes at speeds above 30 knots, probably a function of the additional pitch damping created by flat-topped bulb, and the *MSI* for this craft was the lowest of all the S-series. However, hull shape *S11* may come close to conflicting with the patent claims of Bystedt (1993).

S9 demonstrated high vertical accelerations and *MSI*, but the lowest roll angle and very low lateral acceleration.

At 40 knots, the worst performers for accelerations, *MSI* and Roll angle were *S12* and *Patent*.

4.3 Conclusion on effect of bulb shape on bare hull motions

In general terms, thin bulbs appear to have little effect on vertical acceleration and *MSI*, but they appear to be beneficial for lateral acceleration and roll angle.

A large flat-topped bulb gave superior performance to *S11*.

The high, fat and large bulb of *Patent* only gave reduced vertical acceleration at speeds below 36 knots, otherwise it was detrimental to all other ship motions. The low, fat and small bulb of *S6* gave better than average all-round results.

4.4 Conclusion on effect of a Motion Control System on ship motions

The effect of fitting motion control systems on motion sickness *MSI* was to reduce it by more than 50% at speeds above 35 knots. Roll angles were also reduced by more than 50%. Vertical accelerations were reduced by approximately 35%. It can be anticipated that a larger T-foil would give larger reductions in motion.

Although improving performance considerably, the motion control system had a little less effect on the vertical acceleration of *S11* at high speeds and on *MSI* at 30 knots, compared with the other hull shapes. It was also less effective on *MSI* values for *S6* and *S9* than other shapes, although it still achieved a 55% reduction.

The motion control system was effective in reducing lateral accelerations on all hulls, but extremely effective on *V12*.

It was also very effective in reducing the roll angle of *S11*, and overall the effect of the motion control system was to give *S11* the most outstanding performance of all hull shapes.

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