Effect of Displacement and Configuration on MSI and Global Bending for High Speed Multi-hull Craft

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ABSTRACT: Recent years have seen the emergence of a range of new passenger ship designs of moderate displacement operating at medium and high speeds. This has raised the operating Froude number based on ship length for many displacement hulls into the range between 0.3 and 0.8. Consequent upon this has been the need to develop computational methods which have been validated over a wide range of Froude number. In this paper the various methods which have been developed to compute motions and loads and the shortcomings of these methods will be reviewed. It will be shown that time domain, spatially fixed strip theory gives good predictions over a wide range of speeds for mono-hulls and multi-hulls. The method takes account of the forward speed effect for slender hulls and solutions incorporate the steady wave pattern, operating trim of the design and large wave amplitude effects where sections of the hulls may leave the water. When applied to a family of geometrically similar high speed displacement hulls of varying displacement in the range 500 tonnes to 20,000 tonnes operating at speeds between 5 and 25 m/s the method shows how Motion Sickness Incidence varies with displacement and speed for mono-hulls, catamarans and trimarans. Finally, the variation of vertical bending moment with displacement and speed is also demonstrated.

1 INTRODUCTION

The prediction of ship motions and loads during wave encounter is commonly based on the strip theory of Salvesen, Tuck and Faltinsen (1970). The method considers the hull to be made up of a number of two dimensional strips along the hull length, orthogonal to and fixed in the hull. The hydrodynamic motion is determined as the solution of a two dimensional problem in which each individual ship section is assumed to be in regular periodic motion with radiated waves extending to infinity. The forces on the hull are determined and resolved into a component opposing the acceleration (that is an added mass per unit length of hull) and a component opposing the vertical velocity of the section (an added damping per unit length). The overall ship motion is then computed as he response of the hull and added terms to the excitation due to hydrostatic and Froude-Krylov wave forces integrated over the entire hull. In addition to requiring that the hull be slender to validate the two dimensional assumption the method also neglects the forward motion and the development of the radiated waves from the bow to the stern of a finite length hull since the hydrodynamic solution for each hull section is considered in isolation.

To overcome the limitations of low speed strip theory a variety of alternative techniques have been developed. Broadly these can be divided into several categories. High speed slender strip theory, such as that applied by Hermundstad et al (1999), introduces terms which correct for the forward motion effect whilst retaining the general approach of low speed strip theory based on frequency domain analysis. Three dimensional methods have been based on Rankine source methods, as described by Nakos and Sclavounos (1991) for example, requiring paneling of sources over the water surface and hull. Three dimensional periodic Green functions require paneling of the hull surface (Inglis and Price, 1981). Time domain methods involving so called 2.5 dimension solutions have used sectional strips fixed in the hull or fixed in space and have also been based on both Rankine source (Zhu and Katary, 1998) and two dimensional transient Green functions (Holloway and Davis, 2006). Davis and Holloway (2003) reviewed the published outcomes of all these methods by evaluating the ratio of each individual author’s computed and measured maximum heave and pitch relative to the wave surface motion. These are the response amplitude operator functions which approach unit value at low encounter frequency and zero value at high encounter frequency. Whilst the asymptotic
values at high and low encounter frequency are thus not particularly indicative of the validity of a method, the maximum values of the heave and pitch RAO at the encounter frequency of maximum motion are a good test of a method. Figure 1 therefore shows the error between computed and measured heave and pitch in each author’s published data at the frequency of maximum motion. A full list of cited sources is given in Davis and Holloway (2003). We see that the heave predictions are generally rather high across a range of Froude number and show that predictions of heave often are as much as 70% more than the measured value or as small as 40% below the measured value. For pitch there is a tendency for the discrepancy between prediction and measured data to increase with increasing Froude number. It is thought that this is due to flow effects around and aft of the transom which, if not represented properly in an analysis, have a significant effect on hull loading near the stern and so affect mainly the pitch motion. It can be noted that the two dimensional time domain method based on transient Green function solutions for spatially fixed strips performs well for a range of Froude number.

Measurements of hull bending moment are not common but have been reviewed by Davis, Holloway and Watson (2005) who give a full list of citations. Figure 2 shows the error between computed and measured vertical bending moment at the frequency of maximum bending load, generally near or at the midship location. Once again there appears to be a trend for greater discrepancy at higher Froude number and this also might be attributed to the influence of the separated flow near and aft of the transom not being correctly allowed for in the various analytical methods, in particular three dimensional methods where the entire flow field including that aft of the transom is involved in the solution method. The data shown in figure 2 is based on comparison of the authors’ computed prediction with tank test data in head seas except in the case of the time domain Green function results (solid triangles in figure 2) where the data was obtained by Davis Watson and Holloway (2005) in sea trials of an 86 m INCAT hull in bow quartering seas and was therefore somewhat more approximate.

In general it is again evident that the two dimensional transient Green function time domain method works well over the full range of Froude numbers of relevance to high speed ship motion and load prediction.

2 THE TIME DOMAIN PREDICTION METHOD

The time domain method used in the present work is based on the transient Green function solution for spatially fixed sections of the water perpendicular to the direction of motion (Holloway and Davis, 2006).
The solution for each fixed strip in the water begins as the bow enters the fixed water section and concludes as the stern leaves that section. The Green function is that given by Wehausen and Laitone (1960) as

\[ f(z,t) = \frac{Q(t)}{2\pi} \ln(z - c(t)) - \frac{Q(t)}{2\pi} \ln(z - \bar{c}(t)) \]

\[ -\frac{g}{\pi} \int_0^\infty Q(t') \left[ \frac{1}{\sqrt{gk}} e^{-\sqrt{gk}(t - t')} \sin(\sqrt{gk}(t - t')) \right] dk \, dt' \]

where \( c \) is the source location, \( \bar{c} \) is its complex conjugate, \( Q \) is the source strength and \( k \) is the wavenumber. Application of the hull boundary condition leads to a set of equations for the source strength terms at each time step in terms of the hull motion and the previous time history integrals. The Green function satisfies the linearised free surface boundary condition and therefore it is only necessary to locate sources on the hull surface itself and not over the free surface of the liquid. This simplifies the computation and avoids problems associated with the boundaries of the free surface distant from and at the hull. The effect of finite depth is represented by placing fixed panels to represent the water bottom. The computation of the Green function is relatively time consuming in computational terms.

Figure 3 Variation of computing rate (time steps/second) with CPU speed for a catamaran with 40 sections and 28 panels per section.

Figure 3 shows how the computational speed varies with CPU speed of a Windows system running Lahey Fortran 90 code through a DOS window. The speed shown is for a catamaran design with 40 water strip sections and with a total of 28 panels (20 on each hull and 8 for the water bottom). The computation time varies approximately in proportion to the number of time steps, the square of the number of sections and the square of the total number of panels at each section. For a monohull with 40 sections and 14 panels per section in deep water it is possible to compute a complete RAO involving solutions at 12 wave encounter frequencies using a 2.4 Ghz quad CPU processor system in about 1 hour computing time. At each panel on the hull surface the above Green function solution yields the pressure so that the total force on the hull can be determined by integration over the hull surface at each instant in time. Treating the hull as a rigid body the instantaneous accelerations in heave, roll, pitch, yaw and sway can then be found. Surge has not been included in the solutions here as it involves a range of other issues related to the hull frictional effects. In principle it is then possible to integrate the accelerations in order to determine the motion of the hull through time. However, the hydrodynamic forces on the hull include contributions which in other solution methods would be treated as added mass or damping, terms which are not used in present solution method. The effect of the implied added mass in particular means that there are mass and acceleration terms on both the right hand side (ie due to the hydrodynamic forces acting on the hull) and the left hand side (the hull mass and the acceleration being calculated) of the motion equations. This makes the equations stiff and numerical integration leads to unstable solutions. This problem has been discussed by Holloway and Davis (2006) who introduced a method in which the change of hull position over each time step is taken as a weighted combination of the movement evaluated from the computed acceleration and the movement evaluated from the acceleration computed at the previous time step. It was shown that this yields accurate motion solutions and that weighted fractional contributions from the current acceleration which are about 30% of the total yield stable and accurate results for the motion. The error in the motion solution was shown to be much less than 1%. This was regarded as acceptable in the present context since the effect of finite panel size was similar or larger. With 40 hull sections and 14 panels around each section errors of about 3% arise. As the motion solution proceeds in time the hull is free to sink and trim and also the steady flow disturbance due to the hull is included in the solution.

Figure 4 shows typical transient solutions obtained in calm water. It can be seen that the solution approaches the steady trim and sinkage as computed and that the initial transients pass after about two ship lengths of motion (10.8 seconds) in this case. It should be noted that the during the first ship length of motion the solution is not complete and is still being developed in terms of the convolution...
integrals from bow to stern. If the initial sinkage and trim are selected with care then it is clear that the initial transient can be reduced in magnitude.

Figure 5 shows transient solutions during regular wave encounter. Solutions such as these are used to determine the response amplitude operators by evaluating the magnitude and phase of the computed motion once the solution has stabilised into a regular periodic form after decay of the initial transient. Generally about six cycles were found to be sufficient, although this depends of course on the degree of precision sought and shorter solutions would in many cases be quite adequate. The time step adopted here is such that the hull moves forward by one strip in each time increment, that is a duration of the (ship length)/(ship speed x number of sections). For the examples shown at 48.9 knots with 40 sections and a 136m length the time step is 0.13 seconds. Clearly as the speed is reduced so the time step increases and ultimately it becomes necessary to increase the number of sections at low speed. It is found that 40 sections is sufficient to give stable solutions for length based Froude numbers greater than about 0.35. This increases the computing time required for solutions at low speed. At high Froude number stable solutions can be obtained with fewer sections and so are more rapid. Further, if the initial sinkage and trim are carefully selected or if the encountered waves are relatively large then the solution reaches a regular motion to a greater degree of relative precision more rapidly and so it is possible to compute the regular motion and determine a complete RAO as a function of frequency from a number of separate regular solutions more rapidly.

This computational method has been validated (Holloway and Davis, 2002) with reference to analytic solutions for the wave-maker problem and the transient response of a floating cylinder. Large amplitude motions can be handled by this method since the hull cross sections are panelled to the instantaneous incident wave free surface as the solution proceeds, this requiring a redistribution of panels at each time step around each hull section. Thus the dominant nonlinearity associated with large motions, that is variation of immersed cross sections, is included in the solution. However, the Green function involves the linearised free surface boundary condition and other non-linear effects associated with large motions of the free surface are not accommodated. In general yaw and sway are
quite small and have negligible effect on the heave, pitch and roll motions.

Motions will be influenced by frictional effects, particularly in model scale tests at small Reynolds numbers. It has been found that computed response amplitude operators functions often exceed measured results, particularly in the region of the maximum for heave at high Froude number (Holloway and Davis, 2006). Since frictional effects are not included in any potential solutions, the approach adopted here has been to introduce a sectional friction coefficient $C_s$ for the hull to represent the overall frictional effect due to skin friction, flow separations, flow at the transom, vortex shedding and wave breaking. The transverse (vertical) damping force on each section is $D = \frac{1}{2} \rho U B \nu$, where $D$ is the force per unit length, $B$ is the sectional beam, $U$ the forward speed of the ship and $\nu$ is the vertical velocity of the section relative to the local water surface. Since the main effect is to bring about modest reductions in maximum heave with smaller reductions in pitch, this simple approach is sufficient to match the test data to the RAO maxima. The values required for $C_s$ vary somewhat according to the particular hull under test, but are usually less than 0.15. In particular the value required for this frictional correction is larger for smaller models in general and for the full scale computations to be carried out in the present paper a value of 0.035 has been selected.

The full ship motion prediction method has been validated here by comparison with the tank test data of Wellicome et al (1995). Figure 6 shows a comparison of measured and computed heave response amplitude operator functions whilst figure 7 shows a comparison of the pitch motions. It is clearly evident that the trends with increasing Froude number are well predicted and in particular the maximum in the heave RAO which emerges at high Froude number is accurately represented in the computations. As discussed in section 1, Figures 1 and 2 also show validations of the time domain
method in terms of the maximum heave and pitch (solid triangles) for a range of Froude number when compared to the tank data of Wellicome et al (1995) and for bending moment when compared with sea trials data. It is clear that there is good agreement over a range of Froude numbers of relevance to intermediate speed and fast ships. As mentioned, Davis and Holloway (2003) found that the amplitude nonlinearity effect for the NPL hullform was relatively small and so linearity with wave height has been assumed in the following section.

3 MOTION AND LOADING IN A SEAWAY

The response in a seaway of any vessel depends strongly on the correspondence between the frequency response (or RAO as a function of frequency) and the spectrum of the encountered waves. In the work which follows we will consider the response of families of hulls of varying displacement which are geometrically similar to the NPL hull form used by Wellicome et al (1995) and which were the basis of the validations discussed in section 2. It has been shown by Davis and Holloway (2003) that the nonlinear effect of wave height on the response amplitude operator functions of this hull form was extremely small. Therefore in the work to be presented here it will be assumed that the response amplitude operator functions do not alter with wave height. The parent NPL hull form was used to generate families of monohulls, catamarans and trimarans with the following overall parameters:

Monohull family:
Length/beam =10.0, beam/draught=4.0.

Catamaran family:
Length/beam =15.0, beam/draught=1.5,
offset of hulls from vessel centerline = 14% of overall vessel length.

Trimaran family, main hull:
Length/beam =15.0, beam/draught=2.5,
displacement=90% of vessel.

Trimaran family, outer hulls:
Length/beam =20.5, beam/draught=1.5, offset of hulls from vessel centerline = 14% of overall length.

Figure 8 Motion response amplitude operator functions for a monohull family of varying displacement at 20 m/s in head seas.

Figure 9 Midship vertical bending moment response amplitude operator functions for a monohull family of varying displacement at 20 m/s in head seas. (Ratio of bending moment amplitude to the product of hull weight and wave amplitude)
To illustrate the effect of displacement on the response to encountered head seas, Figure 8 shows the heave and pitch response amplitude operators for the monohull family at a fixed forward speed of 39 knots (20 m/s), and midship bending moment. Figure 9 shows the midship bending moment RAO function: the parameter shown is the ratio of midship moment amplitude in regular waves to the product of vessel weight and wave amplitude. We see that both the motion and bending RAOs, shown here as a function of encounter frequency (Hz), have a progressive reduction in the range of frequency for which there is a significant response as vessel size increases. This is a direct consequence of the increase of vessel length. Also, there a change in the form of the response function which is a consequence of reducing Froude number as the vessel size increases. The combination of these two effects results in a substantial reduction in vessel response with respect to both motions and bending as the vessel size increases for constant speed of operation.

The spectrum of encountered waves for varying wave height is shown as a function of encounter frequency (Hz) for the family of monohulls at 20 m/s and for various wave heights in figure 10. It is assumed here that the spectrum is of the JONSWAP form with average wave periods which increase with increasing wave height as shown in table 1. As would be expected the frequency of maximum wave energy becomes lower as the wave height increases as shown in figure 10.

<table>
<thead>
<tr>
<th>Significant wave height</th>
<th>1.5 m</th>
<th>3.0 m</th>
<th>6.0 m</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average wave period</td>
<td>6.05 s</td>
<td>7.60 s</td>
<td>9.33 s</td>
</tr>
</tbody>
</table>

Table 1: Assumed variation of average wave period with significant wave height

The spectrum of encountered waves for varying wave height is shown as a function of encounter frequency (Hz) for the family of monohulls at 20 m/s and for various wave heights in figure 10. It is assumed here that the spectrum is of the JONSWAP form with average wave periods which increase with increasing wave height as shown in table 1. As would be expected the frequency of maximum wave energy becomes lower as the wave height increases as shown in figure 10.

Figure 11 Vertical acceleration spectra (m²/s³) for a 4270 tonne monohull at 20 m/s at locations along the centerline in a 3m significant wave height head sea.

The response of the vessel will be determined by the relative distributions of the RAO functions and the wave energy spectrum depending upon vessel displacement, speed and wave height. For the conditions shown in figures 8, 9 and 10 the consequent vertical accelerations on the vessel centerline are shown in figure 11 for positions at the LCG, aft at the transom and forward of the LCG by a distance equal to the distance between LCG and transom. The most severe conditions occur at the forward position and least severe at the LCG.

Figure 12 Vertical bending moment spectrum ((Nm)²/Hz) at the midship position for a 4270 tonne monohull at 20 m/s in a 3m significant wave height head sea.

The spectrum of the midship vertical bending moment is shown in figure 12 for the same 4270
tonne monohull in a 3m head sea at 20 m/s. As with the acceleration spectra the location of the spectral peak is controlled by the encountered wave spectrum whilst its magnitude is determined by the RAO function as well as the wave spectrum. The distribution resembles the wave energy spectrum owing to the narrowness of the encountered wave spectrum.

Computations of the rms vertical acceleration have been carried out for the monohull, catamaran and trimaran families for a range of design displacements, forward speeds and wave heights as described above. The formula for Motion Sickness Incidence given by O’Hanlon and Macaulay (1974) has been applied to the spectra of the computed vertical accelerations at the vessel LCG and the MSI thus predicted. Figure 13 shows the results for the three vessel configurations. It is evident in Figure 13 that the MSI becomes significantly large and exceeds 10% for the conditions selected (3m seas, 20 m/s) for monohulls of about 10,000 tonnes or less displacement, for catamarans of about 20,000 tonnes displacement or less and for trimarans of about 5,000 tonnes or less. This outcome is mainly related to the lengths of the three configurations, the trimarans being longest for a given displacement and the catamarans the shortest for a given displacement. Figure 14 shows the effect of wave height on Motion Sickness Incidence for the monohull configuration (which may be compared also with Figure 13 (a)). In a 6m sea even the largest vessel shows just over 10% MSI but only at a very high speed which is outside the domain of existing vessels. In a 1.5 m sea only vessels of under 2000 tonnes give MSI values in excess of 10 %. The results of figures 13 and 14 give a broad overview of conditions where Motion Sickness Incidence becomes a serious issue. Whilst these results have of course been computed for the vessels having the NPL hull form, in general one would expect similar outcomes for other hull forms although the precise detail would differ of course.

Predicted normalised vertical bending moments at the midship position for the monohull, catamaran and trimaran families are shown in figure 15. The normalised bending moment is defined here as the ratio of the root mean square bending moment at the midship position to the product of vessel weight and significant wave height. It should be noted that this normalisation on the basis of significant wave height for a random sea leads to smaller numerical values than are found with the response amplitude operator functions shown in figure 9 where the normalisation is in terms of the wave amplitude of a regular sea.

We see from figure 15 that, as with the MSI results, the length of the vessel clearly has a significant effect. The catamarans have the smallest bending moments for a given displacement and the trimarans the greatest. In a sea of a fixed height the larger vessels have the smaller bending moments relative
to weight and wave height. The effect of wave height on the normalised vertical bending moment is shown in figure 16. These results can be compared also with Figure 15(a) for 3 m seas. In larger seas the normalised bending moment is reduced significantly.

4 CONCLUDING DISCUSSION

A review of published data where measured motions and loads are compared with values predicted by various methods has shown that there are significant discrepancies in the response amplitude operator functions at the frequency of encounter where the maximum response occurs. Prediction of the response near to this frequency is regarded here as the most significant test of the accuracy of a method, particularly with regard to the motion responses where there are well defined unit and zero asymptotic values at low and high frequency of encounter which all methods should readily predict. The source of these discrepancies most likely varies with the method used, conventional frequency domain strip theory omitting forward speed effects and three dimensional theories experiencing effects due to the complicated flow aft of the transom. The time domain, spatially fixed strip theory method based on the transient Green function used here avoids both of these problems and gives good predictions over a wide range of Froude number. The method is based on a slender hull assumption of course and also relies on a linearised free surface condition. However it can accommodate the main source of amplitude nonlinearity due to variation of the immersed hull cross sections during motion and wave encounter. The method has been found to work well when compared with both model tests and full scale trials data.

Figure 14 Variation of Motion Sickness Incidence (%, at LCG) with significant wave height in head seas for monohull vessels

(a) 1.5 m significant wave height sea

(b) 6.0 m significant wave height sea

Figure 15 Normalised vertical bending moment in 3m significant wave height (head seas, midship).
In a random seaway the motions and loads experienced by a vessel depend strongly on the spectral distribution of the encountered waves relative to the frequency distribution of the response amplitude operator functions for vessel motions and loads. In the present paper the responses of families of monohulls, catamarans and trimarans has been presented where the hulls all have a parent NPL round bilge high speed hull form. This particular hull form has been shown previously not to have a strong amplitude nonlinearity in its wave response. The present study has been restricted to head seas and as a consequence the trimaran configuration which has the longest length shows the smallest motion sickness incidence figures. In general vessels of less than 10,000 tonne displacement will show significant MSI levels at speeds of 30 knots or greater, depending on significant wave height. With regard to the midship bending moment the trimaran configuration, having the longest length, shows the greatest bending loads relative to weight and wave height.

5 ACKNOWLEDGEMENT

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6 REFERENCES


