

Ship Hydrodynamics Knowhow derived from Computational Tools

--- Some Examples

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ABSTRACT

This paper discusses how the use of today's computational tools can lead to a quick advance of the field of ship hydrodynamics, by answering long existing questions, indicating simple models and demonstrating design trends. Some examples are given of subjects for which analysis of computational results, from RANS and free-surface potential flow codes, has led to improved understanding of the flow. The first example describes how better understanding of ship wave making and its dependence on the hull form has been obtained from analysis of potential-flow calculations. The resulting insight is still used in the context of CFD-based hull form optimisation. The second example describes how questions regarding the model-to-ship extrapolation of experimental results could be solved using RANS computations. The last example shows how computational study of shallow-water effects has led to a method to correct for tank width effects in model measurements; to an improved model-to-ship extrapolation procedure for shallow water tests; and to a simple trial correction method for moderate shallow-water effects. The examples are meant to illustrate and promote this sort of research, and subjects are mentioned for which similar progress can probably be made using CFD methods.

Key words: CFD; ship wave making; shallow water; model-to-ship extrapolation

1. INTRODUCTION

Ship hydrodynamics is a field of science with a respectable history. The oldest towing tanks were built in the 18th century, and the Froude hypothesis underlying all model testing work dates from 1868. Important empirical knowledge has been built up over time. Predictions of ship full-scale performance were, and usually are, typically based on model tests, using model-to-ship extrapolation procedures based on empirical knowledge and a large database of full-scale correlation data. Ship hull form design was classically done by experienced designers, using their knowledge and empirical insights in the flow physics determining resistance and propulsion.

Today, CFD predictions play an increasing role in the design process. More and more complete and accurate computations are made. But prediction alone does not lead to a better ship. Insight in the flow physics remains essential if we are to design more efficient ships. Furthermore, basing the final full-scale performance prediction just on CFD computations for the self-propelled ship is not considered sufficiently reliable yet; therefore, model testing still is a standard component for larger ships.

A smooth transition from the empirical to a computational process, and continuous improvement of design quality, can best be guaranteed by a good integration of the two approaches. On the one hand this means that CFD predictions cannot simply replace those based on model tests, unless with much attention for modelling and numerical accuracy, full-scale validation and determination of correlation data; and that CFD-based design and optimisation should exploit existing knowledge of the flow physics and design trends. On the other hand, today's computational methods offer unprecedented possibilities for extending our insight in the flow around the hull, scale effects and design trends. Quick progress beyond the classical knowledge can be made; e.g. by analysing computed flow fields and pressure distributions, by performing scale effect studies and integrating CFD use in the power prediction based on model tests, or by CFD-based optimisation leading possibly to new design solutions.

The present paper will discuss some examples of how computational methods have contributed to ship hydrodynamics knowhow. All are taken from the author's research at MARIN over the years; not to present latest results but to illustrate and propagate this use of computational tools and, hopefully, provide inspiration to others. The CFD work is entirely unspectacular, but in the examples mentioned, the computational results still have enabled to identify most practical insights and simplified models that help understanding mechanisms and trends.

2. UNDERSTANDING AND REDUCING SHIP WAVE MAKING

The first example we are considering is, how computations have contributed to the insight in the mechanism of ship wave making and to the effectivity of wave resistance minimisation.

Early theoretical developments on ship wave making were based on analytical methods. Until around 1970, this mostly concerned methods in which not only the free-surface boundary conditions were linearised, but also the hull boundary condition. Thin-ship, flat-ship and slender-ship approximations have been derived and studied. Neither of these was applied widely and successfully in ship hull form design, because of the large deviations of the calculated resistance for real ships. While the insights derived on the properties of ship wave patterns are still essential knowledge today, the approximations made precluded learning much about the relation with the hull form. For design, rather the guidelines from systematic series of model tests were used, along with simple theories regarding interference, as proposed by Wigley.

With the introduction of bulbous bows, attempts have been made to understand their action from the available theories, but with little success. In practice, designing and optimising a bulbous bow was a matter of 'trial and error', through many model tests with modifications of the model shape. Semi-theoretical approaches have been tried to find the best bulb size and longitudinal position based on just few experiments (Sharma and Naegle, 1970), but no indication was thus found of the desired bulbous bow shape.

Around 1970-1975, 'slow-ship linearised' free-surface methods were being proposed. There has been some debate on their formulation (Newman, 1976), and evaluating these theories numerically was a challenge at that time. But Dawson's method (1975) did provide a practical formulation permitting efficient computation, with promising results for wave resistance and wave pattern. At MARIN we started using this method in practice from 1987 (Raven, 1988). Like several others we found that quite puzzling results could be obtained for wave resistance. The subsequent development of nonlinear free-surface panel methods (Jensen, 1988; Raven, 1992,1996; Janson, 1997) led to a much better generality and accuracy, removing the main shortcomings of the wave resistance prediction.

Even so, at least with panel densities computationally feasible at that time, the accuracy of that resistance prediction was still a problem. But the predicted flow field was in good agreement with reality. For us, a breakthrough came when we found a way to base ourselves on the computed flow field and wave pattern rather than just trying to reduce wave resistance by trial and error. A combined visualisation of wave pattern, hull form, pressure distribution and streamline direction was not usual in those days, but became available and helped a lot in analysing the computed fields. The hull pressure distribution in particular, which is hardly feasible to measure in a model test, proved to be an essential result.

Through analyses of many calculated results and tentative hull form adjustments, insight in the mechanism was obtained. This was later more formally described as a procedure (Raven, 2010; Larsson and Raven, 2010) that conceptually splits the relation of a hull form and the wave pattern it generates, into two steps:

1. the relation of a hull form and the pressure distribution on the hull and still-water surface it generates;
2. the wave pattern generated by that pressure distribution.

While the steps are not disjunct and the second step does affect the first, for not too high Froude numbers this is fairly limited. The advantage of the separation into two steps is that each of them follows some rather straightforward rules, very briefly summarised as follows:

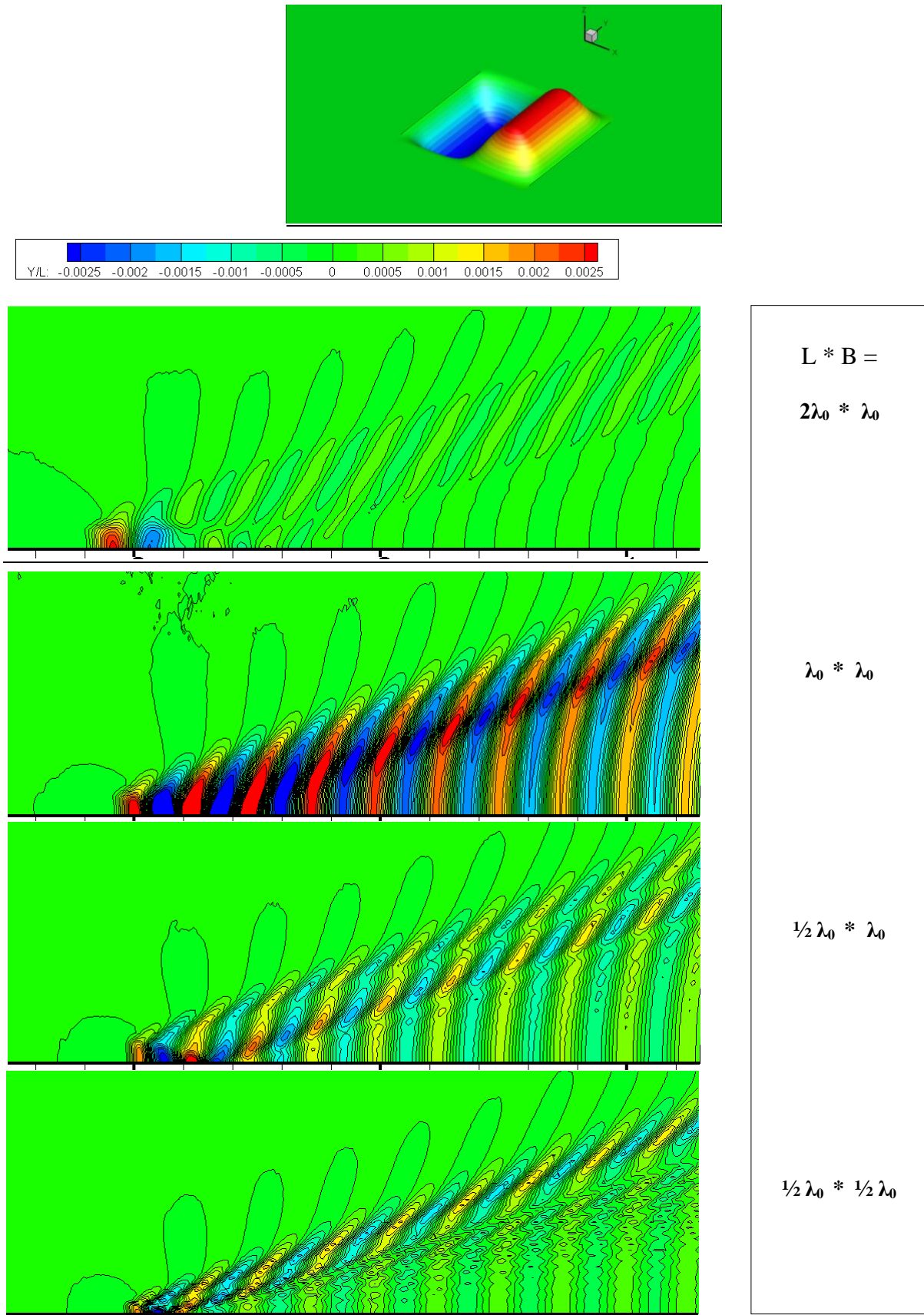


Figure 1 Waves generated by a pressure patch travelling over the free surface.
Top to bottom: shape of pressure distribution; Wave patterns for patch width λ_0 and patch length $2\lambda_0$, λ_0 , and $0.5\lambda_0$, respectively; and patch width $0.5\lambda_0$, patch length $0.5\lambda_0$.

1. Hull form to hull pressure distribution: This is dominated by streamline curvatures; so, by the hull surface curvature in the streamline direction, and also by the slope of the hull surface (as that affects streamline curvature at a distance).

Hull pressure distribution to pressure distribution on still water surface: This is determined by the ratio of the distance to the still-water surface, to the length scale of the hull pressure distribution. If that ratio is large, the hull pressure feature is hardly felt at the free surface, and its length scale has increased; if the ratio is small, pressure disturbances are felt at the surface without much change.

2. Surface pressure distribution to wave pattern: the wave components preferentially generated by a surface pressure distribution are those that have comparable length scale and shape; longitudinally and transversely.

This second step is of course nothing revolutionary. It is well approximated by the analytical expression from linear theory, known since a century, but its use in the present context was not common. The meaning of the second step can easily be illustrated by the wave patterns generated by a free-surface pressure distribution. In Fig.1, from (Raven, 2010), the pressure imposed on the surface has a sinusoidal shape lengthwise, constant crosswise, as shown; and has equal amplitude in all figures. In the first three figures its width is $\lambda_0 = 2\pi V^2/g$ and its length is $2\lambda_0$, λ_0 and $\lambda_0/2$, respectively (so the Froude number based on the pressure patch length is 0.28, 0.40, 0.56). The first does not fit any wave well, and there is little wave making. The second generates a dominant transverse wave and causes much wave resistance. The third has a length that fits a diverging wave, but its width causes two sets of such waves, and some transverse waves. The last is short and narrow and causes a pattern dominated by diverging waves. The figures show how the wave pattern is entirely determined by the length scales of the pressure distribution. As linear relations dominate the wave pattern, these considerations apply as well to components of the pressure distribution induced by the ship, thereby giving useful information on the cause of wave components and measures to reduce them.

We also note that the idea of this two-step approximation of ship wave making is already described in Lighthill (1980), without any reference or validation, and is used to discuss the action of a bulbous bow. Therefore, the components were there but the application in design appeared essentially unknown.

To assess the usefulness of this two-step consideration for practical cases, in (Raven, 2010) a strict computational equivalent was set up: first, the still-water pressure distribution generated by the hull in double-body flow was computed; next, the wave pattern generated by minus that pressure distribution imposed on the free surface, in absence of the hull. This is somewhat simpler than Dawson's method, as it omits the effect of the curved and variable double-body velocity field on the wave propagation, but it corresponds essentially with an older slow-ship linearised theory by Baba and Takekuma (1975). Fig.2 shows that the wave pattern for the Series 60 $C_b=0.60$ hull as found from that simple method corresponds fairly well with that of a full nonlinear potential flow calculation, thus supporting the validity of this two-step analysis approach --- not for prediction, but for understanding. The same was found for various other cases.

As shown in (Larsson and Raven, 2010), the qualitative guidelines for both steps explain much of what is known and observed on ship wave patterns. Both steps are simple enough to be easily estimated and understood. The disregard of near-field effects and nonlinearities hardly matters for their practicality as long as we remain aware of those. The simple analysis procedure, which at MARIN we use since around 1990, still plays an important role in our ship design work. We use it, informally, to understand a computed wave pattern; and to decide what hull form modifications to make to eliminate the dominant wave components.

A new dimension of this possibility to investigate and understand the dependence of the flow and wave pattern on hull form features came with the development of a visualiser for computations for parametric hull form variations. Fig.3 is a screenshot of the user interface, in which the hull form can be varied by moving the sliders top right, and the resulting wave pattern, flow direction and pressure distribution for continuous variation of the hull form parameters are instantly shown. In this case, it interpolates the flow data in an underlying set of $5 * 5 * 5$ computed flow solutions for combinations of the 3 parameters --- just as if the CFD result is computed real-time. This tool gives an unprecedented insight in the detailed design trends and the mechanisms determining the flow and wave making. Contrary to a usual optimisation, we do not just obtain the trend of the objective function, e.g. resistance, but also how this is related to changes of the flow

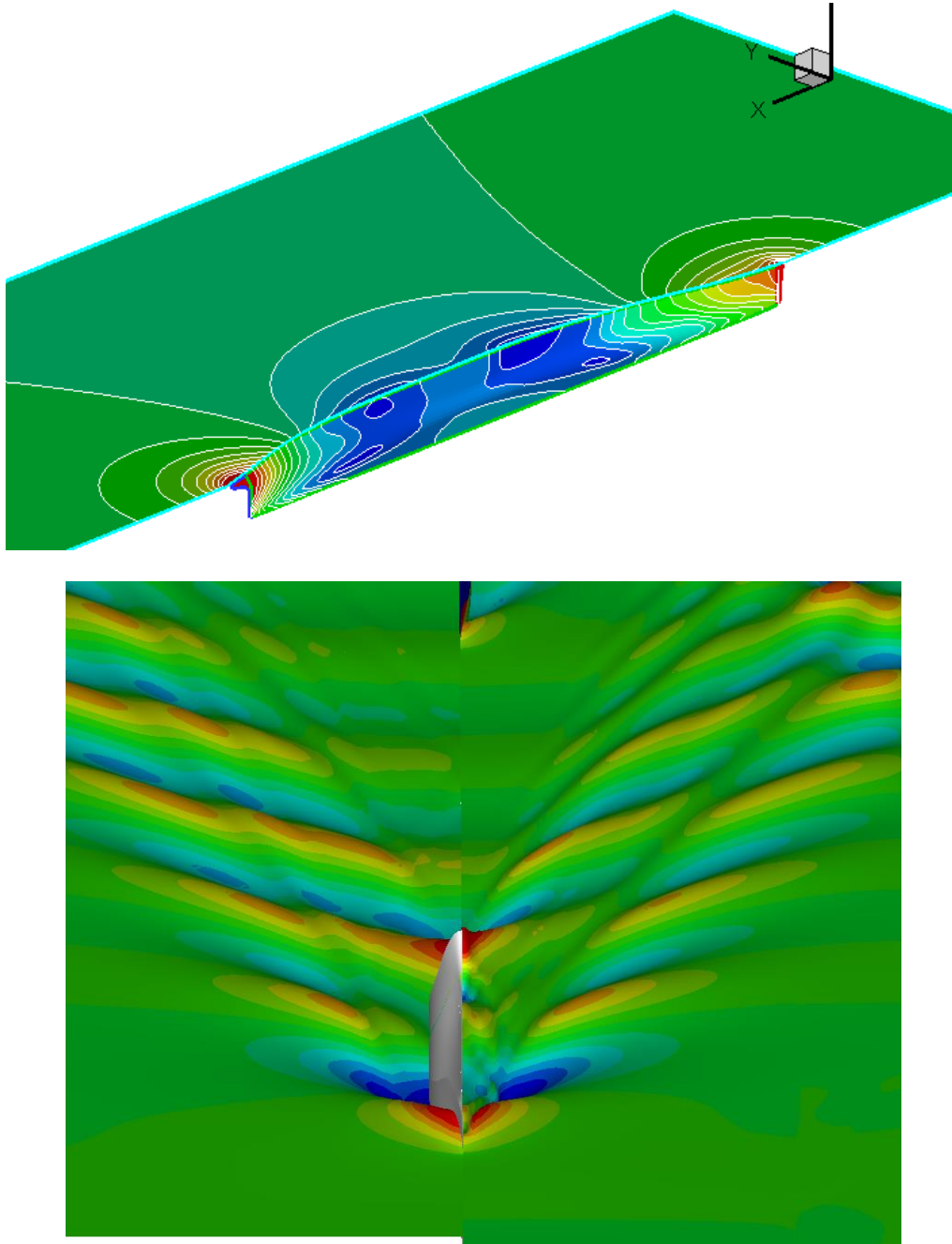


Figure 2 A check of the two-step procedure, for Series 60 $C_b=0.60$ ship at $Fr = 0.316$. Top: double-body pressure distribution on hull and still-water surface, Bottom: right side: wave pattern generated by minus this double-body pressure distribution in absence of the hull. Left side: Wave pattern of the same hull, computed by RAPID

field. This visualiser was initially developed for potential-flow results, and later extended for viscous-flow computations using the PARNASSOS code. Fig.4 illustrates its use for inspection of limiting streamlines on the stern and nominal wake field dependent on hull form parameters (Raven, 2014).

In a computational optimisation of ship hull forms, when dealing with wave resistance, we select parametric deformations of an initial hull form which are based on the same hydrodynamic considerations: we aim at using a set of hull form parameters that directly addresses the dominant wave components of the initial hull form. In this way, with a limited set of parameters significant improvements can often be achieved (Raven, 2017). While today many publications appear in which hull forms are optimised for minimal wave resistance

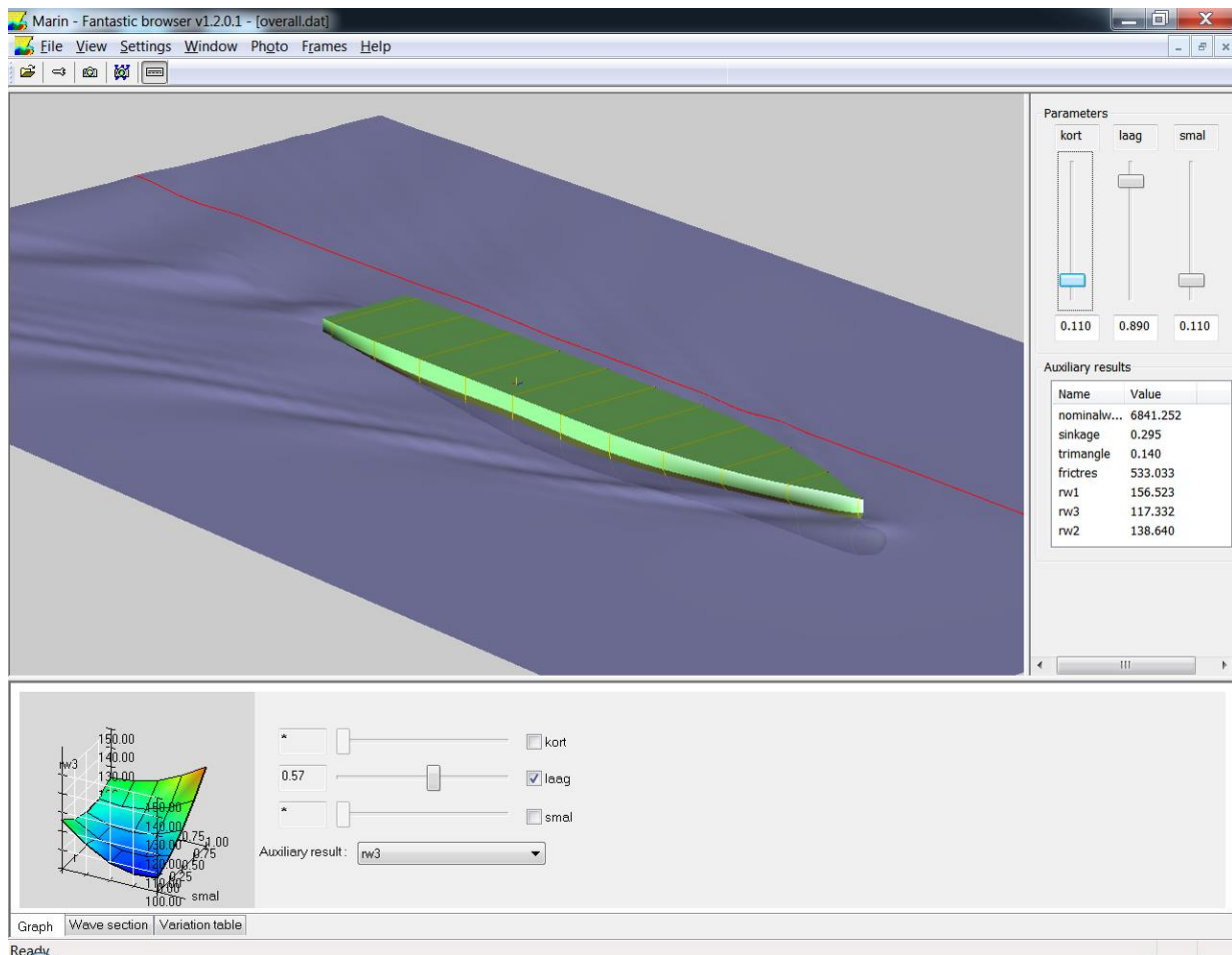


Figure 3. Visualiser for flow fields of systematically varied hull forms. Wave pattern, from free-surface potential flow code, for 3-parameter bulbous bow variation.

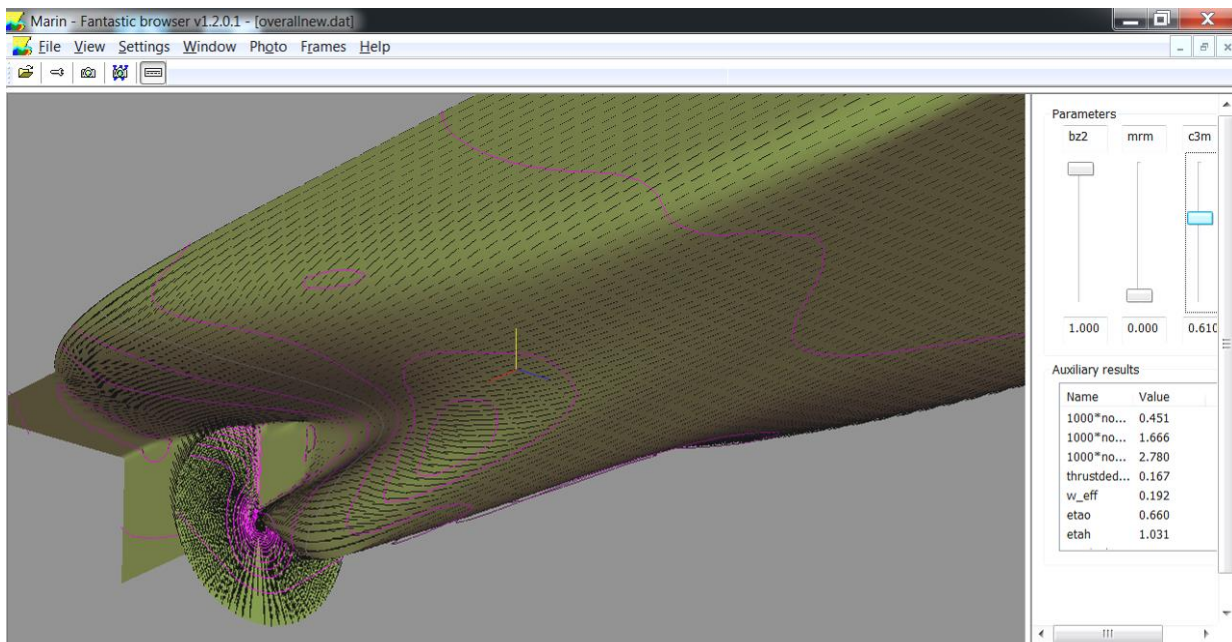


Figure 4. Visualiser for flow fields of systematically varied hull forms. Limiting streamline directions and wake field, from RANS code PARNASSOS, for 3-parameter stern variations.

using a set of parameters just selected from geometric considerations, it is a pity that in that way no use is made of the existing knowledge on ship wave making, and the theories and insight developed in the past.

Thus, this example shows how close inspection of systematic, or also unsystematic, computational results can provide information and insights unreachable otherwise. Existing linear theories provided the proper background to develop the simplified model and insight in design trends. The model and its proper use only became apparent from computational results, in this case from a free-surface potential flow code. The availability of calculated hull pressure distributions along with the wave patterns has been instrumental in understanding and, in a way, demystifying, ship wave making.

3. SCALE EFFECTS ON SHIP RESISTANCE

As a next example of a field where CFD computations have provided essential information, we consider resistance scale effects. To predict the resistance of a ship based on model tests, the Froude hypothesis is used, which, in modern terms, approximates the resistance coefficient as the sum of a viscous resistance just dependent on the Reynolds number, and a wave resistance just dependent on the Froude number:

$$C_t(Fr, Re) = C_v(Re) + C_w(Fr).$$

From a model tests performed at equal Froude number, the ship resistance coefficient is then calculated by correcting for the difference in viscous resistance coefficient resulting from the Re-difference:

$$C_{ts} = C_{tm} - C_{vm} + C_{vs} = C_{tm} - \Delta C_v$$

In the usual model-to-ship extrapolation, the viscous-resistance coefficients are supposed to be proportional to the frictional resistance coefficients of a flat plate at equal Re:

$$C_v(Re) = (1+k) C_{f0}(Re)$$

So the form factor $1+k$ is supposed to be equal for model and ship. This supposition was sometimes doubted; at least for fuller hull forms. But until the availability of accurate CFD, there was no way to really check it. Geosim testing (testing models at several scales) just covers a most limited part of the required range of Reynolds numbers. Full-scale trials only provide measured propulsion power and RPM for a given speed, and any link to a viscous-resistance scale effect is most indirect.

Instead, computing form factors using CFD is in principle straightforward. By making a double-body flow calculation, the resistance found is purely viscous, properly representing the limit for $Fr \rightarrow 0$. Dividing it by the frictional resistance coefficient of a flat plate, taken from a ‘plate friction line’, directly provides the form factor, and, if done for model and full scale, its scale effect.

However, we note here that for quite some time, very few full-scale computations using RANS codes were published. For most CFD codes, a stable and converged computation for full-scale Re with $y^+ < 1$ was unachievable for a long time, and still seems not feasible for some. Avoidance of this problem by using wall functions has the significant disadvantage of possibly distorting the scale effects found.

Probably the first paper on full-scale ship viscous-flow computations without wall functions was that by Eça and Hoekstra (1996). The flow around tanker sterns at model and full scale were shown, including grid refinement studies indicating a good grid-independence. The code used, PARNASSOS, easily permitted such computations owing to its particular formulation, in which the continuity and momentum equations are solved in fully coupled form. Extreme cell aspect ratios are no problem for this code, and the grid for full scale can be obtained from the model-scale grid simply by a stronger contraction towards the hull.

In a study on form factor scale effects in 2008, double-body flow computations have been made with this code for some standard test cases at model and full scale (Raven *et al*, 2008). For a first example, the

Hamburg Test Case, we obtain the following form factors relative to the friction values from the ITTC'57 line: $1+k_{model} = 1.14$, $1+k_{ship} = 1.22$, therefore, not equal at all. Similar form factor increases had already been obtained for several other ships. Therefore, is the form factor method invalid?

However, the scale effect on the form factor depends on the slope of the plate friction line that is used. Table 1 shows that the marked increase of $1+k$ from model to ship occurs for the two most popular lines, ITTC'57 and Schoenherr, but much less so for the more modern lines, from Grigson and Katsui. So, how can we decide on the scale effect of $1+k$?

Table 1 Calculated form factors for 'Hamburg Test Case' with different plate friction lines.

Plate friction line	1+k model	1+k ship	Difference
ITTC'57	1.138	1.222	7.4%
Schoenherr	1.161	1.221	5.2%
Grigson	1.154	1.157	0.3%
Katsui	1.179	1.202	2.0%
Numerical friction line	1.213	1.227	1.2%

A plausible answer was obtained as follows. We want to know whether the viscous resistance of a ship is proportional to that of a flat plate over the Re range. Therefore, we compare the computed value for the ship with the computed value for a flat plate, using the same code and the same turbulence model. Different turbulence models do give somewhat different results, but that influence should largely be eliminated in this way. To this end, we use the numerical friction lines (Eça and Hoekstra, 2008) that had been derived by very careful RANS computations for flat plates, using the same code and several turbulence models.

Table 1 shows that relative to the plate friction line for the same turbulence model (Menter's $k - \sqrt{k} L$ model in this case), the form factor is almost equal for model and full scale; a conclusion that has been confirmed for many other ships. At the same time, this result, and the comparison with other plate friction lines, suggests that the ITTC'57 line, when plotted against $\log(Re)$, has a too large slope. The lines from Grigson and Katsui are a lot closer to the numerical plate friction lines and lead to a more constant form factor.

For the KVLCC2 tanker, having some flow separation at model scale, a slight decrease of the form factor from model to ship was found even when using the numerical friction line.

Therefore, from this and other cases we conclude:

- That the form factor concept itself is found largely valid: CFD computations indicate a good proportionality of the viscous resistance of a ship hull form, with the frictional resistance of a flat plate --- both being computed by the same code and the same turbulence model; unless flow separation occurs;
- that the same is true if one of the modern friction lines is used, but not the ITTC 57 or Schoenherr line;
- that using a fixed form factor together with the ITTC 57 line (i.e. the method formerly recommended) leads to a prediction of full-scale viscous resistance that is significantly too low. This systematic underestimation was implicitly compensated by a part of the correlation allowance, but in view of the variables used in the expression for that allowance, that compensation could only be approximate.

For MARIN's model-to-ship extrapolation procedure this study and later checks have led to the replacement of the ITTC '57 line by the Grigson line (along with a complete revision of the correlation coefficients). Besides, form factors are now always computed from double-body viscous flow computations using PARNASSOS; and as a check, this is always done both for model and for full scale, such that also for cases with deviating scale effects, the right ship viscous resistance is estimated.

More recently, extensive studies have been done by Korkmaz (2020), which confirmed that a computational determination of the form factor is as accurate and often more practical than the experimental determination. Also, numerical friction lines have been derived for various turbulence models, and it was demonstrated for an example that using those lines removes the clear underestimation of the full-scale viscous resistance found with the ITTC'57 line. Moreover, the combination of computed form factors with numerical friction lines led to a reduced scatter of correlation coefficients for a large trial database. In a joint study of many institutes

worldwide (Korkmaz *et al*, 2021), similar conclusions were reached. Thereby, this careful and comprehensive research has given support to a proposed revision of the recommended model-to-ship extrapolation procedure by the ITTC, using this ‘Combined CFD/EFD method’ as an option.

Therefore, in this example, the use of CFD has answered some of the oldest questions in ship hydrodynamics; confirming the validity of the form factor concept, indicating the replacement of the plate friction line; and improving the accuracy of predictions for cases with flow separation or other special scale effects.

4. SHALLOW-WATER EFFECTS ON SHIP RESISTANCE

4.1 Introduction

The third example concerns the effect of shallow water on the flow and resistance; both in model tests and at full scale. Around 2006, it was reported to us that during trials in shallow water, ships were running significantly faster than had been predicted based on shallow-water model tests. Several studies have then been done to clarify and remove the causes of this. Up to that moment, shallow-water tests were essentially analysed like deep-water tests; and predictions made either by a Froude extrapolation, or by a form factor extrapolation. Some empirical estimation methods for shallow-water effects on resistance or power had been published; such as the methods by Schlichting (1934) and Lackenby (1963). Besides, some theoretical results had been published, which only addressed the effect on the wave resistance (Kirsch, 1966). As we know now, some of these papers have led to misunderstanding.

4.2 Tank Width Effects

Our first suspicion was that the limited width of the shallow-water model basin (although, 16 m at MARIN) could cause the predictions (for shallow water of unlimited width) to be pessimistic. Tentatively, the measured resistance curve was shifted to a higher speed based on the overspeed (return flow) found from Kreitner’s method (Kreitner, 1934). This on the one hand confirmed that channel effects could be significant in these tests; on the other hand, the correction appeared too large, incidentally resulting in a lower resistance curve in shallow water than in deep water.

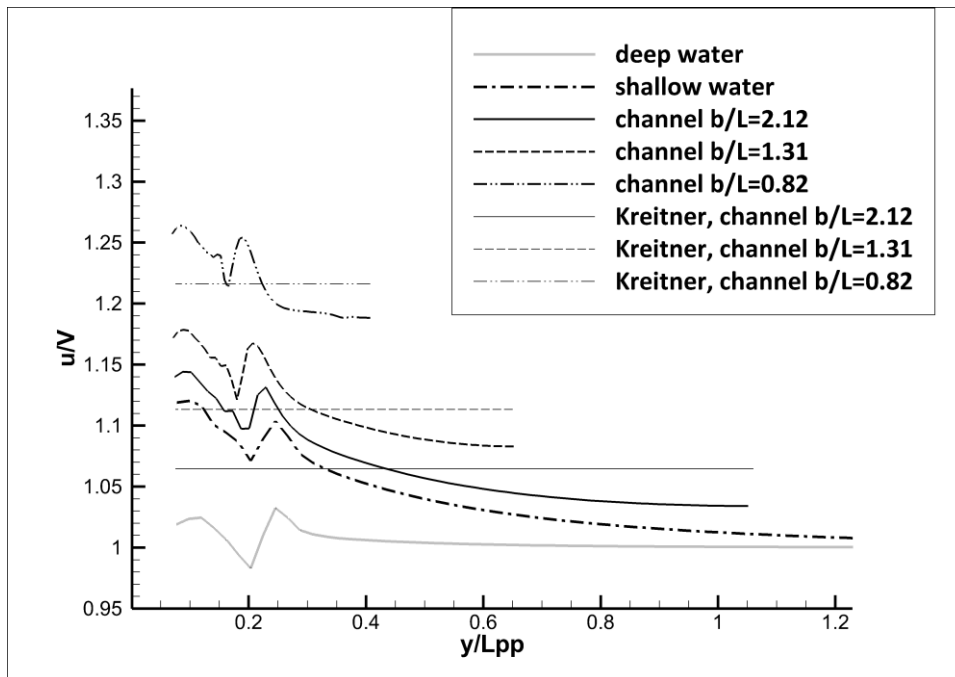


Figure 5. Distribution of overspeed along the ship, at the midship section, against distance from centreplane; in deep water, shallow water, and shallow channels of same depth.

In Kreitner's method it is assumed that the overspeed along the model is uniformly distributed over the channel cross section. As a first step, the actual distribution has been calculated for various channel dimensions and ship speeds, using the potential-flow solver RAPID. Fig.5 (Raven, 2019a) shows the crosswise distribution of the overspeed next to the ship's midship section, at the free surface, for shallow water of infinite width, and for some channels of equal depth but variable width. The narrower the channel, the higher the overspeed. Clearly the overspeed is nonuniform, highest close to the ship and decreasing with distance. But remarkably, the overspeed *increase* due to the channel walls, compared to that in shallow water of infinite width, is fairly uniform. This fact was just observed owing to some simple computational work. This then led to a new formulation of the channel effect. The ship's midship section A_m is an obstruction to the flow; the overspeed next to it must therefore carry an excess volume flux $A_m \cdot V$. In a channel, all this excess flux must be transported through the limited channel cross section; but in shallow water of infinite width, a part of it, Q_{out} , is transported by the overspeed outside the channel wall position. The overspeed increase $\Delta\gamma = \Delta u/V$, supposed uniform over the channel cross section, must be such that it compensates this part of the excess flux. But this overspeed increase also leads to a lowering of the water level and resulting increase of the sinkage of the ship, thereby a further reduction of the available channel cross section. Thus, we get a 3rd degree algebraic equation for $\Delta\gamma$, similar to but different from Kreitner's equation:

$$\Delta\gamma \left(1 - \beta + \frac{1}{2} Fr_h^2 (1 - 3\bar{\gamma}^2 - 3\bar{\gamma}\Delta\gamma - \Delta\gamma^2) \right) = \frac{Q_{out}}{VA_c},$$

in which β is the blockage, and $\bar{\gamma}$ is the overspeed ratio in shallow water of infinite width, averaged over the width of the channel. We need Q_{out} and $\bar{\gamma}$, which are derived from a single shallow-water calculation using RAPID for one relevant speed. Then we solve for $\Delta\gamma$, and apply a speed shift derived from this to the measured resistance curve. Thereby, the resistance measured in the model basin for a speed V_{tank} is supposed to be valid for a slightly higher speed $V_{shallow}$ in shallow water of equal depth but unlimited width. Besides, a correction is applied to the measured sinkage of the model.

Applying this correction for the limited tank width has been found to be absolutely essential to determine the true water depth effects (Raven, 2019b). The tank width effect causes a resistance increase that grows quickly for decreasing water depth. Therefore, without such a correction the apparent water-depth effect can easily be doubled. It is very likely that older empirical estimates for water-depth effect on resistance or sinkage have strongly been affected by the limited tank width used.

4.3 Model-to-Ship Extrapolation

Still, using these corrections of the model-test data did not entirely remove the pessimistic full-scale predictions. Therefore, in a next study the scale effects in the viscous resistance for ships in shallow water have been computed. Double-body flow RANS computations were done using the PARNASSOS code (Hoekstra, 1999; Van der Ploeg *et al*, 2000), for several ships, each in a range of water depths and for model and full scale. Thus, a clear picture was obtained of the shallow-water effect on viscous flow and resistance, and on its scale effect (Raven, 2012).

It was observed that in shallow water, as long as $T/h < 0.67$ or so, the change of the streamline pattern over the hull is rather limited; but due to the lower pressure along the midbody, pressure gradients at fore and aftbody increase. For $T/h > 0.67$, more drastic flow changes occur, with the flow over the hull following a more horizontal path, resulting in a larger apparent hull fullness, and still larger pressure gradients possibly leading to flow separation.

Regarding the viscous resistance, we best plot the results as $C_v/C_{v_{deep}}$ against T/h , Fig.6. This figure includes not only computations done for this study, but also a collection of results from computations in practical projects, covering a variety of ships; plus a set of experimental data from Millward (1989). We observe that for $T/h < 0.5-0.6$, the relative increase of the viscous resistance (or form factor) for all cases available is in a rather narrow band. In this regime, the precise hull form does not matter much for the increase, and also the model and full-scale increases are nearly equal. The mean line plotted is the relation

$$C_v/C_{v_{deep}} = 1 + 0.57 (T/h)^{1.79}$$

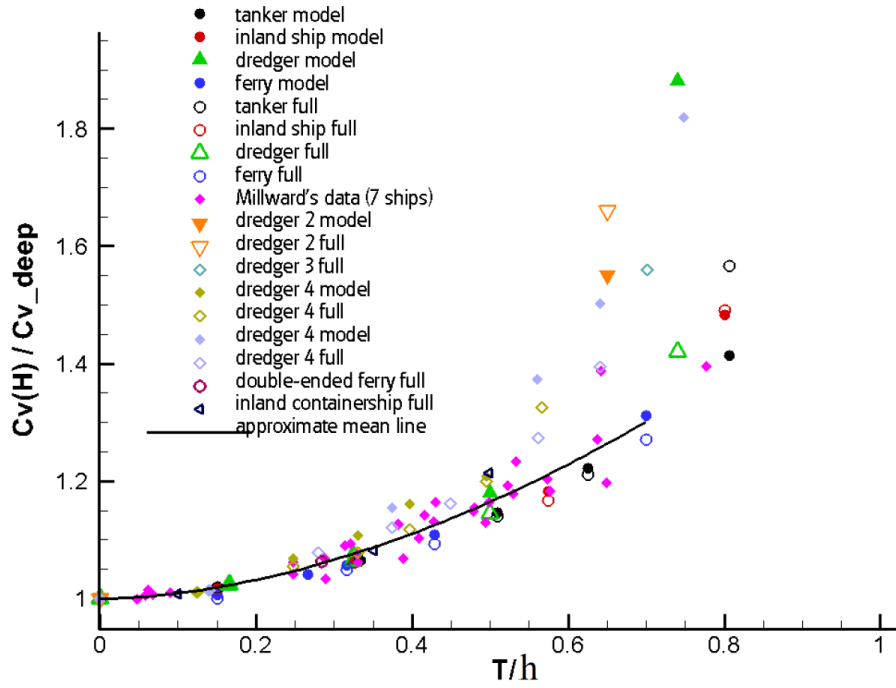


Figure 6. Computed relative increase of viscous resistance coefficient in shallow water, for various ships and models (Raven, 2016)

However, in the shallower regime, where the flow direction over the hull is more strongly changed, there is a larger dependence on the hull form, and also a larger difference between model and full scale.

This also tells directly how shallow-water model tests need to be analysed to get a full-scale prediction:

- first, the measured resistance curve needs to be corrected for tank width effects;
- next, the model resistance can be extrapolated to full scale using a water-depth dependent form factor; or a deep-water form factor multiplied by the mean line value shown in Fig.6;
- if, however, $T/h > 0.5$ and/or the ship is not slender, larger form factor scale effects can occur and it is safer to compute the viscous resistance coefficient in shallow water both for the model and for the ship, by simple double-body RANS calculations.

(Raven, 2019b) gives an example of how large the differences can be due to these steps. The procedure described is being used at MARIN since 2012.

With some additional work (Raven, 2016), the same research led to what is called the ‘Raven shallow-water correction’, a simple correction method for incipient shallow-water effects in full-scale speed trials (ITTC, 2017a, 2017b). If the contractual speed for a new ship is specified for deep water, but the trials have been performed in a slightly limited water depth, a correction may be applied. This correction uses the mean line shown in Fig.6 for the increase of viscous resistance, along with considerations on the change of the sinkage, wave resistance and propulsive efficiency. The set of simple empirical relations based on the computational data permits to estimate limited effects of shallow water on the resistance, for model and ship alike.

Fig. 7 (Raven, 2019b) illustrates what we learned in the process. It shows how, from a deep-water resistance curve for a ferry model, we estimate the shallow-water resistance curve (the blue dot-dash line) for $T/h = 0.46$, by adding the two components of the shallow-water correction: the change of viscous resistance, and the additional change of resistance due to the increased sinkage in shallow water. However, to estimate what would be measured in the model basin, we apply the speed shift for the tank width effect, from the method of Section 4.2, and get the red line. The black markers represent the actually measured points, which are in very good agreement with the estimates. The (expected) slight deviation at higher speeds can be attributed to the shallow-water effect on wave resistance that sets in around $Fr_h = 0.70$ (15.4 kn in this case).

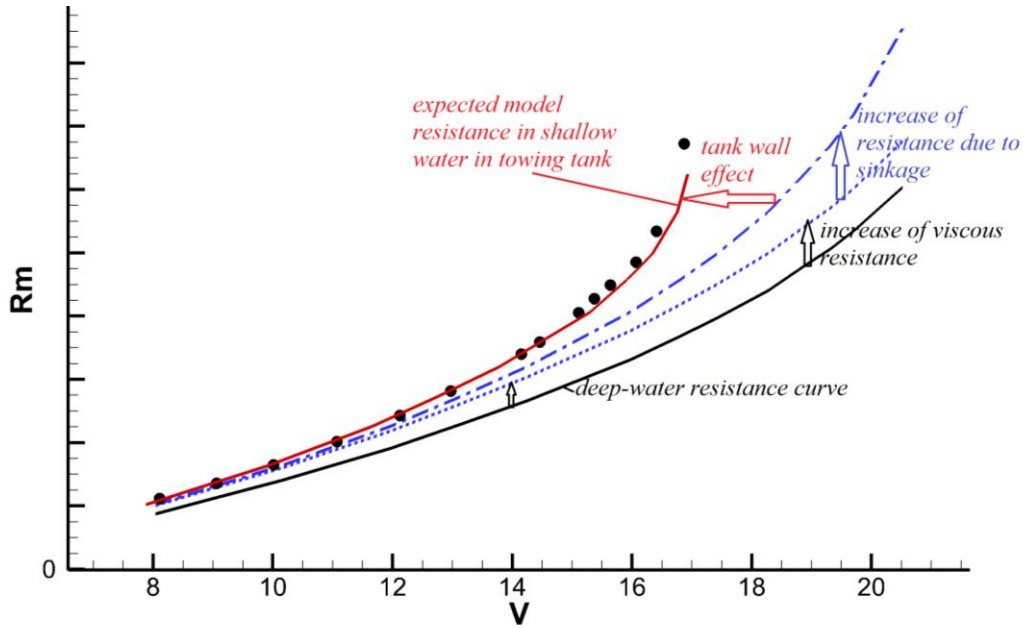


Figure 7. Decomposition of difference between shallow-water and deep-water model resistance curve. The components are derived from the empirical shallow-water correction, and the tank width correction. Black markers are actual measurements in shallow-water model basin. Model of ferry with $T=6$ m, in water depth 13.1 m.

Therefore, not only we have derived some simple models based on the computed data; but also we gained understanding: the difference in measured model resistance in shallow and deep water, which formerly just had to be taken for granted, can now be decomposed into its separate contributions, providing a clear view of what physically is taking place. Here again, some of the classical problems in steady ship hydrodynamics could be well solved using today's computational methods.

5. OTHER CHALLENGES AND UNSOLVED PROBLEMS

In ship hydrodynamics, even for the familiar resistance and propulsion field, there are still several unsolved problems, and subjects for which more information or validation is desired and could well be obtained from computational results. We mention a few.

The scale effect on the effective wake fraction (Taylor wake) plays an important role in the full-scale power and propeller RPM prediction based on model tests. Currently used approximations are very simple and possibly not always adequate. Computational studies are desired. This requires a RANS solver for the hull flow, coupled with a propeller model, either by RANS or by a boundary element model for the propeller. In this way, a validation of the simple relations now used could be carried out, and perhaps a better model could be derived.

Thrust deduction is a fairly easily predictable quantity that affects the required power of a ship. Perhaps, more attention for its dependence on the hull form could help reducing that power. It can easily be computed using a RANS solver coupled to a propeller model or body-force disk; and experimentation with hull forms should provide helpful design trends. As thrust deduction and effective wake tend to show similar variations but with opposite effect on required power, both should be computed and the effect on overall propulsive efficiency estimated.

Scale effects on propeller characteristics play a role in the full-scale predictions, and a way to correct for them is included in standard methods. But usually these corrections are based on just the drag scale effect due to friction on the propeller blades, and disregard the scale effect on the lift. Computational studies can clarify this, and work is being done on this subject.

With today's free-surface RANS codes, a study of wave/viscous interaction effects is accessible. Viscous and scale effects on stern wave making exist, and their importance depends strongly on the case. While we have done some computational studies on this, these did not yet lead to a full picture of the physics or a clear view of the optimal design to aim for. Possibly, there would be further room for improvement of hull form design if we would have a better understanding of the interaction effects and their meaning for ship resistance.

The effect of wave breaking on the trailing wave system and flow, and thereby on ship resistance and radiated wave height, is a quite unknown aspect. While small amounts of breaking seem to mean little for ship resistance, for a class of vessels with strongly breaking waves it is of importance. So far, trustworthy computational predictions with RANS codes seem absent for breaking ship waves. In this case, computational work might not provide a solution in a short term.

Therefore, a further advance of the knowledge on steady ship hydrodynamics can well be obtained from computational work, for several subjects but not for all.

12 CONCLUSIONS

The three examples discussed were meant to illustrate how a quick advance of the science of ship hydrodynamics, even for its most classical field (resistance and power), can be achieved by dedicated studies using CFD or other computational tools. The focus needed is on understanding the physics and using that to understand design trends, or to set up simple models. This is a different focus than to aim at the most complete or advanced prediction tool, although of course a large numerical accuracy is essential. The present paper is meant to advocate such work, as an important step towards further improvement of ship and propeller designs.

The first example given showed how, based on inspection and analysis of results of free-surface potential-flow computations, and being aware of simplified ship wave making theories, a conceptual model could be deduced that helps understanding the mechanism of ship wave making. The same understanding is still used in the context of CFD-based hull form optimisation.

The second example addressed the scaling of viscous resistance. Double-body RANS computations and numerical plate friction lines quickly resolved some questions that before could hardly be disentangled; confirming the validity of the form factor scaling method, but just when based on a proper plate friction line.

The third example, on shallow water effects, showed how with easy computational work, both potential flow and RANS, towing tank width effects can be eliminated, the model-to-ship extrapolation was revised, a shallow-water trial correction was derived and insight was collected on the contributions to the shallow-water effect on resistance.

Of course, many similar developments have been done by others, and much progress has been made. All three examples were largely taken from our own research done at MARIN. On a personal note, I want to mention that the present symposium almost coincides with my official retirement, after over 40 years involvement in the development and application of computational methods for ship resistance and flow. The subject of this invited session, "Use of computational methods to advance steady ship hydrodynamics", has been a main theme during my work. I hope this session and the present paper may serve to inspire others to continue contributing to the field of ship hydrodynamics and hydrodynamic ship design.

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