On ship powering predictions
and roughness allowances

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Abstract

After having overcome most of the fundamental deficiencies of Froude's method of ship model and full scale testing by resorting to a rational theory of hull-propeller interaction and quasi-steady trials and monitoring, my present exercise is concerned with the inherent problems of traditional ship powering predictions based on smooth hull results, maybe based on computations or (still?) on results of model tests.

1 Background

1.1 Froude's approach deficient

Although suffering from various fundamental deficiencies Froude's method of ship powering prediction based on physical model tests has been standardised only in the 2010s by ITTC, ISO and IMO.

The fact, that bare hull towing, propeller open water and propulsion tests on model scale produce three incoherent sets of data, has in the 1930s been Fritz Horn's incentive to propose a method to get at least rid of the 'disturbing' concept of rotative efficiency.

But the subsequent tests and results of Horn's method at Berlin and at Wageningen, presented at the 4th ITTC at Berlin 1937, suffered from the lack of adequate conceptual, experimental and computational tools at that time and they were soon after disrupted by the Second World War.

The fact, that Froude's method cannot be applied on full scale, definitely not under service conditions, has since the 1980s been my own incentive to develop a quasi-steady test procedure getting along solely with propulsion tests and coherent conventions replacing bare hull towing and propeller open water tests.

An early proof of the pudding took place during the trials of the German research vessel METEOR in the Greenland Sea in November 1988, the results published and in detail discussed at the '2nd INTERACTION Berlin '91'.

1.2 Traditional trials analysed

As an interlude, since the end of the 1990s I have developed a less ambitious rational method, permitting reliably to evaluate traditional steady trials without any reference to ship theory and prior data, as it must be for trustworthy results.

Applications of the method in two delicate cases have been published in every detail in the first two volumes of my Festschrift, commemorating the quasi-steady full-scale test with the METEOR of 1988 in the Greenland Sea.

1.3 Quasi-steady trials

Further the method plays a dominant role in the subsequently developed extremely efficient quasi-steady method for trustworthy ship powering trials and monitoring. The development, published in the third volume of the Festschrift, has been based on data acquired during a quasi-steady ‘model’ trial performed in 1986, prior to the full scale test with the METEOR.

The essential steps are the identification of the quasi-stationary states ‘passed’ during the quasi-steady trials and their analysis down to the propulsive efficiencies! In case of the ‘model’ trial ten such states have been passed during the test of only two minutes duration!

When comparing my results with those obtained by traditional analysis, I already noticed, that the frictional deduction applied on model scale resulted in unresolved problems, to be discussed in the following exercise.

By the way it is mentioned here, that the evaluation of the quasi-steady ‘model’ trial, has been continued until 2017. The ‘final’ results published on my website demonstrate, that even the thrust can be identified reliably, ‘impossible’ to be measured routinely under service conditions. A full scale application of this extremely efficient procedure is still pending, waiting for somebody interested to acquire a doctor’s hat.

2 Opening operations

2.1 Problem grasped

'Modelling' the difference between model and full scale, smooth hull resistances by an external force and subsequently additively applying power allowances for roughness and others are of course the crudest conventions one can dream up.

According to my memory, whenever the ship sizes increased, these conventions have been found unsatisfactory by my colleagues in charge at VWS, the Berlin Model Basin.

The problem is particularly serious in case of large, slowly 'steaming' vessels, where the values of the wave resistance, at carefully down scaled model speeds, are only very small compared to those of the frictional resistance, which is accounted for only in the crudest way possible.

2.2 Model conceived

Traditionally propulsion is naively treated in terms of forces. The propeller behind the hull is conceived to produce thrust to overcome the prevailing resistance, including the suction caused by itself. One disadvantage of this approach is, that the various forces cannot be measured, routinely not even the thrust of the propeller.

A more adequate approach is to treat the propulsion in terms of powers. In this case the propeller is conceived as a pump feeding energy into its inflow in order to establish the condition of vanishing momentum flow out of the ship system. And the various powers can be identified, even in case of the very inefficient traditional steady trials!

Concerning the approach in terms of powers a ‘remarkable’ statement by Leonid I. Sedow, translated by Georg Weinblum for the benefit of his German colleagues and students, has been quoted in my opus magnum of 2009 on pages 1218 f.

2.3 Goal defined

The goal of the present exercise is to develop the essentials of the latter approach and discuss some of its implications.

2.4 Plan derived

The development is proceeding in the following steps:

1. Abstract model
2. Configuration efficiency
3. Energy wake fraction
4. Jet powers requires
5. Traditional approaches
6. Model test procedure
7. Discussion of results

3 Jet powers required

3.1 Abstract model

In order to provide for an efficient discussion of the problems concerned I am adopting the model of an ideal propeller P behind the hull and its equivalent propeller, conceptually 'far behind', solely in the energy wake.

By definition the ideal equivalent propeller Q of disc area A_Q has the same flow rate Q_J and same pressure rise Δe_J, *i. e.* in terms of pumps the same 'head', as the ideal propeller P behind the hull. For the following exercise it is sufficient to assume uniform wakes.

The advantage of talking in terms of the equivalent propeller is the fact, that the thrust of this smaller propeller equals the total resistance, the thrust deduction fraction ‘vanishing’ with the displacement wake fraction.

The displacement wake fraction and the corresponding thrust deduction fraction at the propeller P behind the hull are energetically neutral! This condition permits to derive a thrust deduction theorem and by global approximation to arrive at a simple thrust deduction convention. The details, documented on my website, are not of interest in the present context; but see a pertinent remark below.

The propulsive efficiency of the equivalent propeller Q, the ratio of the power required and the power supplied at the shaft, is the product

$$\acute{\eta}_{RS} \equiv P_R / P_S = \acute{\eta}_{RQ}\, \acute{\eta}_{QJ}\, \acute{\eta}_{JS} ,$$

of the hull ‘efficiency’

$$\acute{\eta}_{RQ} \equiv P_R / P_Q = 1 / (1 - w_E) ,$$

the ratio of the total resistance power required, traditionally only on model scale to be identified as ‘hull towing power’, and the propulsive power of the equivalent ideal propeller Q operating in the energy wake,

of the jet efficiency

$$\acute{\eta}_{QJ} \equiv P_Q / P_J = 2 / [1 + \mathrm{sqrt}(1 + c_{EQ})] ,$$

the ratio of the propulsive power of the propeller Q and the jet power, being the same for the ideal pumps, alias propellers P and Q,

and of the hydraulic efficiency

$$\acute{\eta}_{JS} \equiv P_J / P_S ,$$

the ratio of the jet power and the power supplied at the shaft.

The same break down applies of course on model and full scale. In order to avoid confusion, additional indices *mod* and *ship*, respectively, will be necessary.

Details of the notation, of the 'Rule driven symbols developed', are to be found in the 'News flash' on my website under 'Very happy end of a very long story'.

In the present context it is noted, that the name ‘hull efficiency’ for the magnitude in question is grossly misleading engineering jargon. The magnitude is not an efficiency proper, but a measure of hull influence, of value usually larger than 1. Accordingly the magnitude should adequately and correctly be called ‘hull influence ratio’, ‘Rumpf-Einflussgrad’ as is ‘standard’ practice in Germany.

Further it is mentioned, that the term ‘magnitude’ for any measurable concept, standardardised as ‘Grösse’ in Germany and standardised as ‘Grandeur’ in France, is ‘slightly less’ misleading than the term ‘quantity’, as many physical magnitudes are not quantities proper in physical sense, but ‘only’ in mathematical sense.

3.2 Configuration ‘efficiency’

In the present context the hydraulic efficiency, the measure for the energetic quality (!) of the propeller design in the given configuration, is not of interest and will for simplicity be assumed to be the same on model and full scale, typically of value 0.8 .

Of interest is the configuration ‘efficiency’

$$\acute{\eta}_{R\,J} \equiv P_R / P_J = \acute{\eta}_{R\,Q}\, \acute{\eta}_{Q\,J} \,,$$

the measure for the energetic quality of the ship design. This fundamental measure, which I introduced in two papers already in 1968 and 1970, is still not being 'accepted' and not used by naval architects.

As in case of the ‘hull efficiency’ the ‘configuration efficiency’ is not efficiency proper, but a measure for the quality of hull-propeller configurations, though in practice (!) with values smaller than 1.

For the ideal configuration of a deeply submerged body of revolution with an ideal propulsor exactly absorbing the whole energy wake the quality ratio becomes

$$\acute{\eta}_{R\,J} = 1 / (1 - w_E / 2) \,.$$

In general this measure explicitly becomes

$$\acute{\eta}_{R\,J} = 2 / [(1- w_E) + \mathrm{sqrt}\{(1- w_E)^2\, (1 + c_{E\,Q})\}]$$

and further

$$\acute{\eta}_{R\,J} = 2 / [(1- w_E) + \mathrm{sqrt}\{(1- w_E)^2 + c_{E\,C})\}] \,,$$

with the load ratio of the equivalent propeller

$$c_{E\,C} = \Delta e_J / q_C \,,$$

normalised by the dynamic head

$$q_C \equiv \rho\, V_{H\,C}^{\;2} / 2$$

referred to the hull speed through the water, the energy wake cancelling out!

For the following discussion it is noted, that the head of the equivalent propeller equals the total resistance of the ship divided by the disc area of the equivalent propeller

$$\Delta e_J = R_{T\,C} / A_Q$$

and the total resistance

$$R_{TC} = R_F + R_A$$

is separated into the frictional hull resistance and the sum (!) of all additional resistance components!

Accordingly the second parameter of the configuration efficiency

$$c_{EC} = c_F + c_A ,$$

is separated into the normalised frictional resistance

$$c_F = R_F / (A_Q\, q_C)$$

and into the normalised physically different additional resistance components

$$c_A = R_A / (A_Q\, q_C) .$$

3.3 Energy wake fraction

The relationship derived highlights the importance of the energy wake, in engineering jargon due to the frictional resistance of the rough (!) hull. The fact, that the energy wake cannot (yet) reliably be identified, as shown even in the final evaluation of my quasi-steady ‘model’ trial of 1986, does of course not change its fundamental theoretical importance.

For the present exercise the energy wake can be crudely estimated based on the (ideal) momentum flow far behind the hull

$$\rho\, A_F\, V_{HC}\, (1 - w_E)\, V_{HC}\, w_E = R_F ,$$

i. e. the normalised momentum balance

$$(1 - w_E)\, w_E = ά\, c_{RF} / 2.$$

with the area ratio

$$ά \equiv A_Q / A_F .$$

It is explicitly noted, that the effect of the frictional hull resistance is ‘similar’ to that of a water turbine extracting energy from the surrounding flow, ‘very’ different (!) from the effect of the propeller similar to that of a water pump ‘feeding’ energy into the surrounding flow. And in this context it is also mentioned, that the resistance values of profiles have been obtained by integration over their energy wakes.

For the numerical exercise, ideal propeller designs digesting exactly the ideal energy wake are assumed. Accordingly the area ratio in question can be derived using the invariance of the flow rate

$$A_Q\, V_{DQ} = A_F\, V_{HC}\, (1 - w_E)$$

with the flow velocity through the disc of the equivalent propeller

$$V_{DQ} = V_{HC}\, (1 - w_E) / ή_{QJ} .$$

Accordingly the area ratio in question simply becomes

$$ά ≡ A_Q / A_F = ή_{QJ} .$$

Thus inversely the efficiency of the equivalent propeller may also be expressed in the format

$$ή_{QJ} = 2\,(1 - w_E)\, w_E / c_{RF}$$

subsequently be eliminated, resulting in the equation

$$2 / [1 + \mathrm{sqrt}(\,(1 + (c_{RF} + c_{RA})\,(1 - w_E)^{-2}] - 2\,(1 - w_E)\, w_E / c_{RF} = 0$$

for the energy wake fraction.

3.4 Jet powers required

In the present exercise only the jet power is of interest, arrived at according to the rule

$$P_J = (R_F + R_A)\, V_{HC} / ή_{RJ} ,$$

i. e. normalised

$$c_{PJ} ≡ P_J / (A_Q\, q_C\, V_{HC}) = (c_{RF} + c_{RA}) / ή_{RJ} .$$

Thus if the jet power is predicted for a given sum of additional resistances it finally depends solely on the frictional resistance.

In the numerical exercise accompanying this note it has been shown, that the function

$$c_{PJ} = f\,(c_{RF} , c_{RA})$$

can be linearly approximated

$$\mathbf{c_{PJ} = a_0 + a_F\, c_{RF} + a_A\, c_{RA}}$$

with very narrow standard deviation.

At this stage it is worth noting, that the normalisation of the magnitudes involved does not need to be based on the disc area of the equivalent propeller Q, but that the known disc area

$$A_P = A_Q / (1 - t)$$

of the propeller P serves the purpose as well.

No reference to the thrust deduction fraction t is necessary! By the way it is mentioned here, that the derivations of the thrust deduction theorem and of the simple thrust deduction convention are to be found in the ‘3rd, virtual INTERACTION 2017’ on my website.

In order to provide an impression of the values a numerical analysis has been carried out for the ranges

$$c_{RF} = 0.15, \ldots\ 0.31$$

and

$$c_{RA} = 0.40, \ldots\ 1.20 ,$$

resulting in the constants

$$a_0 = -0.105, \quad a_F = 0.921, \quad a_A = 1.330$$

and consequently in the difference

$$a_{AF} = a_A - a_F = 0.409\ .$$

In the present context it is only of interest, that the influence of the frictional resistance is considerably smaller than that of the external resistances.

3.5 Traditional approaches

Traditional procedures of ship powering predictions are not directly arriving at the power required for the propulsion of hulls, but via the power required for the propulsion of the smooth hulls (index 0) and 'allowances' for rough hulls (index 1) are additively applied. This approach is not only followed, if predictions are based on model tests, but is even followed in computational predictions.

Accordingly the jet power predicted becomes

$$c_{P\,J.\,trad.\,ship} = c_{P\,J.\,0.\,ship} + a_A\, \Delta c_{R\,F.\,ship}\ ,$$

with the difference in ship hull frictional resistance

$$\Delta c_{R\,F.\,ship} \equiv c_{R\,F.\,1.\,ship} - c_{R\,F.\,0.\,ship}$$

while the rational approach results only in

$$c_{P\,J\cdot\,rat.\,ship} = c_{P\,J.\,0.\,ship} + a_F\, \Delta c_{R\,F.\,ship}\ .$$

Thus the traditional approach over-estimates the jet power required according to the rule

$$\boldsymbol{\Delta c_{P\,J\cdot\,ship} = a_{A\,F}\,(c_{R\,F.\,1.\,ship} - c_{R\,F.\,0.\,ship})}$$

and a corresponding negative (!) roughness 'allowance' is necessary, to 'account' for this systematic error.

3.6 Model test procedure

In the forgoing example it has been assumed, that the configuration efficiencies are obtained computationally for smooth full scale hulls, instead of directly for rough full scale hulls.

If the efficiencies are based on data acquired during model tests, traditionally with external forces simulating 'frictional deduction', the situation is much more involved. In order to avoid any confusion the values on model and full scale have carefully to be distinguished.

In this case the only additional normalised forces to be accounted for on model scale are the normalised frictional deduction and the normalised wave resistance, the latter being the same on model and full scale according to Froude scaling obeyed.

The resulting law for the jet power shows, that the crucial frictional term, dominant for large, slow 'steaming' ships, requires prior knowledge that is not readily available for new types of ships.

With the frictional deductions based on smooth (index 0), hull resistances

$$c_{T\,F.\,mod} = c_{R\,F.\,0.\,mod} - c_{R\,F.\,0.\,ship}\ ,$$

where the second terms are 'initially' unknown, the jet power of the model becomes

$$c_{P\,J.\,0.\,mod} = a_0 + a_F\, c_{R\,F.\,mod} - a_A\, c_{T\,F.\,mod} + a_A\, c_{R\,W}\ ,$$

i. e. explicitly

$$c_{P\,J.\,0.\,mod} = a_0 - a_{A\ F}\, c_{R\,F.\,mod} + a_A\, c_{R\,F.\,ship} + a_A\, c_{R\,W}\ ,$$

while the jet power for the smooth ship hull is

$$c_{P\,J.\,0.\,ship} = a_0 + a_F\, c_{R\,F.\,ship} + a_A\, c_{R\,W}\ .$$

Thus in this case the smooth hull jet power of the ship is under-estimated, according to the rule

$$\boldsymbol{\Delta c_{P\,J.\,0.\,ship} = -\, a_{A\,F}\, c_{T\,F.\,mod}\ .}$$

Together with the former over-estimation

$$\Delta c_{P\,J\cdot\,ship} = a_{A\,F}\, \Delta c_{R\,F.\,ship}$$

this results in the systematic error

$$\Delta c_{P\,J\cdot\,ship} = a_{A\,F}\, (\Delta c_{R\,F.\,ship} - c_{T\,F.\,mod})$$

i. e. explicitly

$$\boldsymbol{\Delta c_{P\,J\cdot\,ship} = a_{A\,F}\, (c_{R\,F.\,1.\,ship} - c_{R\,F.\,0.\,mod})\ .}$$

Thus the rule is as simple as before, but, contrary to the case considered before, the sign of the total systematic error may be positive or negative and thus the roughness allowances negative or positive, respectively.

3.7 Discussion of results

The first result demonstrates, that predicting the required powers for smooth hulls and crudely accounting for hull roughness over-estimates the configuration, thus counter-intuitively requiring 'negative' roughness allowances to correct the systematic errors.

The second result demonstrates, that the traditional model technique, crudely applying the estimated frictional deductions as external forces, under-estimates the ship smooth hull jet powers required. In this case the total allowances to correct the systematic errors may be negative or positive.

The reason for the fundamental systematic differences is the fact, that the frictional resistance cannot 'simply' be treated like an external force.

4 Concluding operations

4.1 Exercise evaluated

The present exercise, concerning the traditional approach to ship powering prediction is based on the rational theory of ship hull-propeller interaction, on the conception of equivalent propellers out-side the displacement wakes. The resulting two simple, practical rules of thumb confirm and explain the repeatedly reported ‘requirement’ of counter-intuitive negative roughness allowances,

4.2 Approach assessed

So far I have applied the rational theory of ship hull-propeller interaction primarily to analyse ship powering trials and monitoring on model and full scale. The present exercise demonstrates that it is not limited to these applications, but of course permits the rational discussion of other fundamental problems of ship powering a well.

Essentially the present exercise is ‘not more’ than a model based dimensional analysis, an ‘inspectional analysis’ as the mathematician Garrett Birkhoff has called it in his famous ‘Study in Logic, Fact, and Similitude’ of 1950. And it is another contribution to my work on the evaluation of propulsors and propulsion lasting now for more than fifty years.

4.3 Decision taken

As this exercise, triggered by a personal discussion with Dr.-Ing. Karsten Hochkirch of DNV GL on October 18, 2017, and indebted to the continuous scrutiny by Dr.-Ing. habil Klaus Wagner of Rostock, is outside the ranges of problems I have primarily been concerned with and thus outside the ranges of data I have at my finger-tips, I am expecting professional response and maybe suggestions for further joint research.

5 References

Selected pertinent references, complete references to be found on my website.

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