SESSION ON POWERING PERFORMANCE

Chairman: Dr. H. Lindgren

Technical Committee Memberships: H. Tanibayashi (Chairman) - W.-C. Lin (Secretary) - G. Collatz - B. Della Loggia - J. Holtrop - D. Bailey - T. Munk - B. Nizery - K. Varsamov.

Discussion of the Report and the Draft Recommendations of the Performance Committee. (Cf Proceedings, Volume 1, p. 271-333.)

I. DISCUSSIONS

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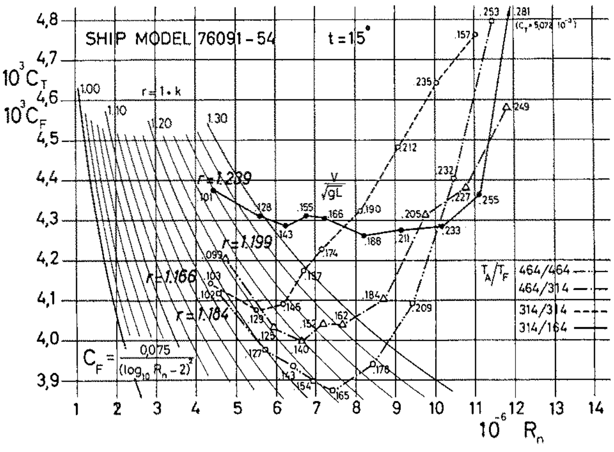
ON THE REPORT OF THE PERFORMANCE COMMITTEE

Remarks Regarding the Form Factor

The form factor method (p. 274) for extrapolation of model test results is a part of the 1978 ITTC Performance Prediction Method, but it is a part causing shipowners and shipyards trouble.

The Propeller Committee writes indeed in its Report (p. 158) that "there exists an international agreement on a common scaling method and this work is straightforward", but in my opinion this is not the case. Different methods are applied by towing tanks for determination of the form factor r = (1 +k) but the methods do not give the same results.

In Fig. 1 an example is given. A model (δ_P = 0.712) has been tested in different conditions and the form factors have been derived by the towing tank. If the applied methods are correct the C_T-curves have to coincide with the corresponding C_F (1+k)-curves at low Froude numbers. For the shipyard it is difficult here to see that this is the



1984 08 SHIP MODEL 76091-54

Figure 1

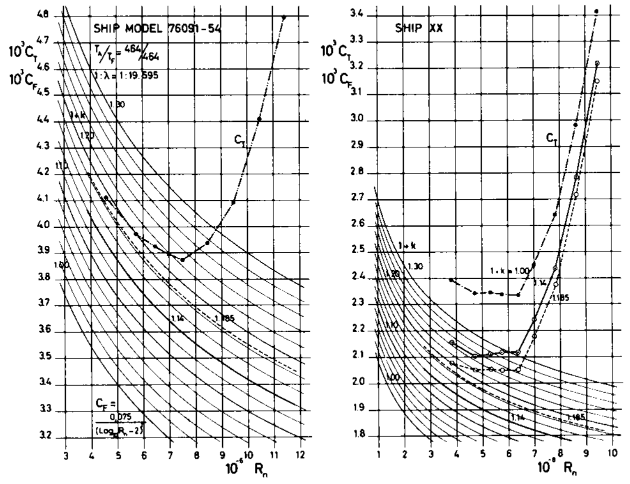


Figure 2

case for all four conditions and that there is a system in the variation of the (1+k)-values and of the C_T-curves. If the (1+k) for the model in full loaded condition (see Fig. 2) has been estimated to 1.14 using one method and to 1.185 using another method the predicted C_T-values of the ship will at a Froude number of 0.21 be about $2.43 \cdot 10^{-3}$ and $2.38 \cdot 10^{-3}$ respectively. The corresponding values of the residuary resistance coefficient C_R are $0.63 \cdot 10^{-3}$ and $0.51 \cdot 10^{-3}$. For another model ($\delta_p = 0.826$), which has been tested at different tanks, (1+k)-values from 1.08 to 1.186 were observed for the model in a ballast condition. Such a difference means at $R_n = 1.1 \cdot 10^7$ a difference of about $0.3 \cdot 10^{-3}$ on C_R when the same C_T has been registrered. This means that the method applied has only a minor influence on the prognosis of the towing tank, but it has an essential influence on the statistics of the shipyards if for instance they want to use the C_R in their future design work.

In my opinion it is to the advantage of the shipyard having the experimental results (not only the predicted values of the ship!) presented in a form analougous with the form used in Fig. 1. Then the shipyard can easily compare experimental results from different towings tanks and in their own way transform the results from model to all ships geometrically similar to the model. Furthermore in doing so the influence of (1 + k) can be minimized. The diagram also gives the shipyard an impression of the experimental uncertainty at model tests.

Perhaps the ITTC ought now to consider the following:

1. to standardize the method of deriving (1+k)
2. to propose a method or methods which could be used by shipyards for an easy and quick estimation of the power for coming ships on basis of results from towing tests earlier carried out at different model basins.

M. CARLIER - Canal de Experiencas Hidrodinamicas, El Pardo, Spain.

ON THE PERFORMANCE COMMITTE REPORT

I have two contributions related to the form factor and its determination:

1.

At El Pardo Basin, we have somtimes found problems in the form factor determination in ships of different types, specially in connection with the presence or absence of rudder in the resistance test. We have examples of single and twin screw ships in which resistance at low speed (and, consequently, form factor) is lower with rudder than without it.

I would appreciate the Committee's opinion about the convenience of using resistance results and form factor derived from resistance tests performed with or without rudder, in connection with self-propulsion test, that of course are proformed with appendages, when applying the ITTC-78 Performance Prediction Method.

2.
The other question is related to Prohaska's method. We have applied it, in its generalized form, with exponents of Froude number different from 4, in the calculation of form factors from more than 400 resistance tests, contained in our computerized databank. These calculations were carried out automatically, by a computers program, selecting the exponent leading to the best least squares fitting of the data points in the Prohaska's graphic. We can present several conclusions of this work:

a. The Form Factors evaluated by this method show a very good correlation with main form parameters: Block coefficient, Length/Beam ratio and Beam/Draught ratio. This, in our opinion, supports the use of this generalized Prohaska's method.

b. In a high percentage of cases, the optimum exponent to linearize the Prohaska's graphic was found to be very high, about 8 and even higher. Nevertheless, the influence of the variation of exponent in the value of the form factor so obtained was not too large.

c. In a large number of cases, in the range of Froude number from 0.18 to 0.20, the values of resistance obtained showed the presence of a too large wave resistance component to be included in the form factor's determination. Therefore, we have adopted the standard criterion of using in our computerized calculations only points in the range of Froude number between 0.12 and 0.18.

The Committee's comments about these points should be appreciated. Thank you.

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ON THE TREATMENT OF HULL SURFACE ROUGHNESS

The author would like to make some comments on the treatment of hull surface roughness effect in the analysis of ship powering, in order to avoid a mis-reading of the Committee Report.

In applying eq. (1.7) in the Report, it should be noted that eq. (1.7) holds only under the condition of constant roughness Reynolds number, that is,

$$\Delta C_F'/C_F^2 = \text{const. for } Uk_s/\nu = \text{const.} \quad (1)$$

Therefore, the roughness problem is essentially a two-parameter problem, at least, i.e., Reynolds number and either the roughness-height Reynolds number Uk_s/ν or the roughness-height ratio k_s/L. Then we have to be very careful about what parameters are constant when we change ship speed or ship length.

The author [1] has recently derived an alternative expression for eq. (1.7),

$$\Delta C_F' = 0.042\ C_F^2\ Uk_s/\nu$$

$$= 1.80\text{x}10^{-5}\ R_n^{-1/4} Uk_s/\nu \qquad (2)$$

which is applicable for the roughness height k_s of 0.1 mm to 0.8 mm, and for Reynolds number of actual ship scale. Eq. (2) also agrees well with the data analysed by *Baba* and *Tokunaga*.

Lastly, if one considers the model-ship correlation of the roughness effect, or if one makes a model experiment with any rough surface, the problem should imply an additional parameter of Froude number, besides the preceding two parameters. For the sake of resolving the complexity in such a case, the author [2] has also proposed an alternative new parameter,

$$h = (k_s/L)\ (UL/\nu)^{2/3}$$

$$= k_s\ F_n^{2/3}\ (g/\nu^2)^{1/3} \qquad (3)$$

which depends only on the roughness height and Froude number, and therefore is constant between the ship and the model with same roughness.

References

[1] HIMENO, Y.: "A Study on Frictional Resistance of Painted Rough Surface", Jour. Kansai SNJ, No. 191 (1983) (in Japanese)

[2] NAKATO, M. and others: "Resistance Increase due to Surface Roughness", 15th Symposium on Ship Hydrodynamics, Hamburg (1984)

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SOME REMARKS ON LERBS EQUIVALENT PROFILE METHOD AND A LIFTING LINE EQUIVALENT PROFILE METHODS AS AN ALTERNATIVE

Introduction

Lerbs, [1] , proposes two methods for the calculation of the equivalent profile characteristics which according to him give equivalent results. The aim of this paper is threefold.

a) to compare the two methods of Lerbs with a modified, nonlinear, version of Lerbs first method, b) to present sensitivity analysis results of Lerbs' methods with respect to the choice of the equivalent radius, c) to propose a lifting line equivalent method as an alternative to Lerbs' methods.

Comparative results of Lerbs' methods

Figures 1 to 4 present the C_L-α and C_D-α curves for two model propellers. Both of them have diameter 0.2 m, four blades, expanded area ratio 0.628 and chord over radius at 0.75R equal to 0.675. The first has a tip skew 60° deg. and nearly constant pitch equal to 0.7 x D. The second has loaded tip, i.e. a strong pitch increase near the tip region with P/D≈0.84. The curves marked with LFM, Fig. 1 to 4, indicate the results of Lerb's first method. The curves marked with LSM indicate the results of Lerbs second method (which corresponds to the block diagram presented by the 17th ITTC Performance Committee § 1.2.2). The curves marked

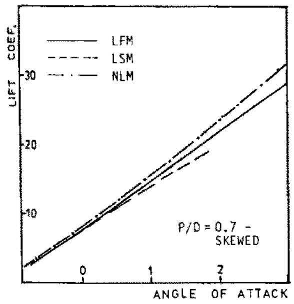
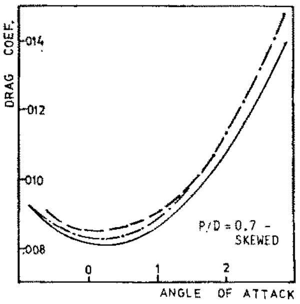


Figure 1



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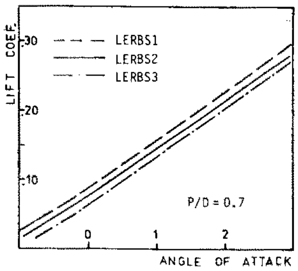

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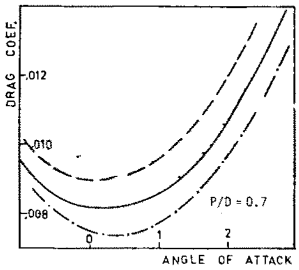

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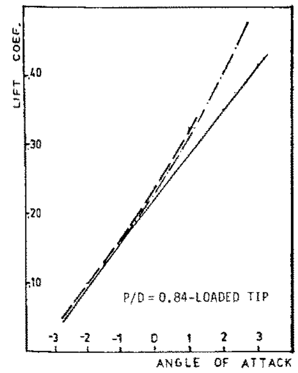

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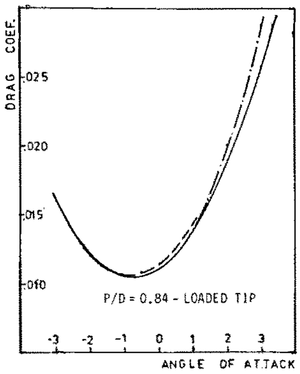

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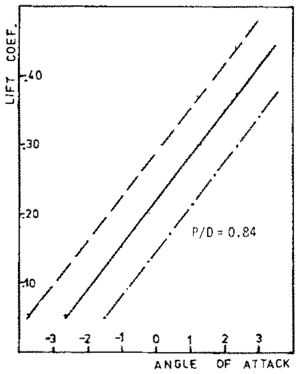

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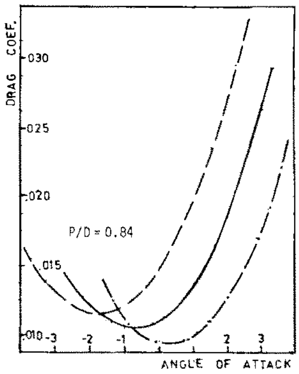


Figure 8

with NLM correspond to a "nonlinear" version of the first method of Lerbs described in [2] and which does not assume that the quantity $(\beta-\beta_i)$ is small.

From Figures 1 to 4 the following conclusions can be drawn:

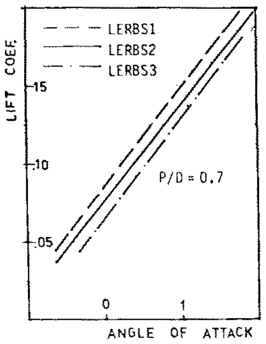
a) There are non-trivial differences among the lift curves obtained by the two Lerbs' methods and the nonlinear version of Lerbs first method which are greater for greater values of the angle of attack. b) The differences in the drag coefficient are generally smaller with the greater differences observed for the skewed propeller.

Sensitivity analysis of Lerbs' method

Figures 5 to 8 present a sensitivity analysis of Lerbs first method for three values of the equivalent radius x_o=0.7, 0.75, 0.8. Similarly Figures 9 to 12 and 13 to 16 present a sensitivity analysis of Lerbs second method and the nonlinear version of Lerbs first method respectively. On Figures 5 to 16 the curves marked with LERBS1 refer to equivalent radius x_o=0.7. Similarly the curves marked with LERBS2 and LERBS3 refer to equivalent radii equal to 0.75 and 0.8 respectively. The other curves contained in Figures 13 to 16 correspond to the proposed lifting line method, briefly explained in the next paragraph.

With regard to these figures the following conclusions can be drawn:

a) For the skewed propeller the C_L-α and C_D-α curves move downwards with



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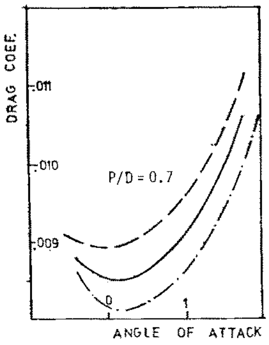

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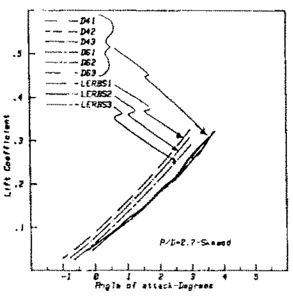
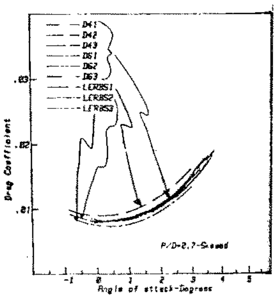


Figure 13



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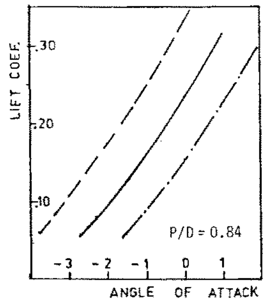

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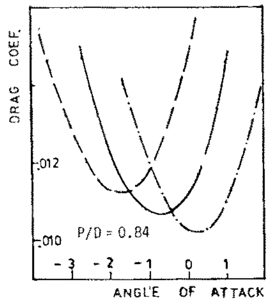

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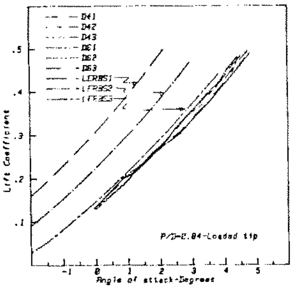

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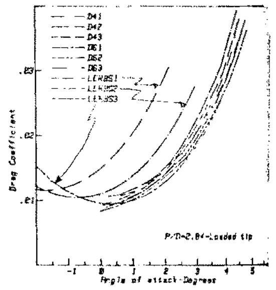

Figure 16

an increase in the equivalent radius. b) For the loaded tip propeller the same trend as in (a) is observed for the lift curve. However, the dependence of the results on the equivalent radius is much greater. c) For the loaded tip propeller the C_D-α curve moves substantially to the right with an increase in the equivalent radius.

The above observations lead to the conclusion that both of Lerbs' methods cannot be considered dependable for use in all cases and for unconventional propellers their results seem to depend greatly on the choice of an "appropriate" equivalent radius, which is different for each application.

Lifting line equivalent profile method

The theoretical shortcomings of the two methods of Lerbs, with have been recognized by the 16th ITTC Performance Committee, are discussed in [2] and [3]. The main assumption in both of Lerbs' methods is that the Goldstein theory can be used to predict the ideal propeller performance, not only at the design point but also at off-design conditions. To avoid such an "unsuitable for a performance problem" assumption, a lifting line equivalent profile method has been developed and described in [3].

Figures 13 and 16 (curves marked with D41, D42, D43, D61, D62, D63) have been taken from [3] and contain some sensitivity analysis results of the lifting line equivalent profile method with regard to changes in the equivalent radius in the range of 0.7R to 0.8R. From these figures we can conclude that

a) the lifting line equivalent profile method is insensitive to changes in the equivalent radius. b) Lerbs methods with an equivalent radius equal to 0.75R give in general C_D-α characteristics very close to the lifting line equivalent profile method. (This result is especially true for constant pitch propellers.)

However, for the loaded tip propeller an equivalent radius of 0.8R is necessary for agreement. c) Lerbs method always overestimates the lift coefficient in comparison to the lifting line equivalent profile method. (This statement is also true for constant pitch propellers.) d) An equivalent radius of 0.8R used with Lerbs' methods seems to give a better agreement of the equivalent profile C_L-α characteristics in comparison to the lifting line equivalent profile method for both the skewed and the loaded tip propellers.

References

[1] LERBS, H. W.: "On the effect of scale and roughness on free running propellers", Journ. ASME, 1951.

[2] POLITIS, G. K.: "On the choice of the equivalent radius in Lerbs' equivalent profile methods", Report No 6/1984, Dept. of NAME, N. T. U. A.

[3] POLITIS, G. K.: "A lifting line equivalent profile method for propeller calculations". Accepted for publication in the Journal of Ship Research, Febr. 1984.

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ON A PROPOSAL FOR A COOPERATIVE PROGRAM FOR THE DEVELOPMENT OF C_L, C_D-α REGRESSION POLYNAMIALS USING THE LIFTING LINE EQUIVALENT PROFILE METHOD

Subject of the proposal

Oossanen in [1] has carried out a Regression analysis for the C_L-α and C_D-α curves obtained by Lerbs' method for the Wageningen B-screw series and used this in his lifting line performance program. In [2] it is shown that Oossanen's method can not work for non-constant pitch and/or highly skewed propellers. On the other hand the lifting line equivalent profile method proposed in [2] gives C_L, C_D-α characteristics which, when used in a lifting line performance analysis, reproduce exactly the experimental results for the full range of J values. The good result obtained by the new theory encourage us to propose to the Performance Committee to adopt and support the following cooperative program among interested I.T.T.C. Member Organizations:

The Program

Each participating organization will provide the Dept. of N.A.M.E. of National Technical University of Atens with the geometrical details and test results of a substantial number of model propellers.

The collected data will be analysed using the lifting line equivalent profile method to obtain C_L-α and C_D-α relations for each case. Then, the so relations so derived will be further analysed using regression analysis techniques to obtain C_L-α, C_D-α expressions appropriate for Lifting Line and/or Lifting Surface (for C_D) performance calculations.

At the end of the Program each participating organization will be provided by both the regression analysis results and the Lifting Line Performance Prediction Program.

Type of the data needed

a) Detailed propeller geometric characteristics in numerical form. That is:

- propeller diameter
- hub diameter
- type of blade sections (thickness and camber forms)
- spanwise variation of pitch, chord and maximum thickness
- coordinates of the generator line
- distance from the generator line of leading and trailing edges

b) Open water propeller characteristics K_T, K_Q-J curves in numerical form. The Reynolds number of each K_T,K_Q-J curve as well as any special information regarding the experimental conditions such as the use of leading edge vortex generators should be given.

References

[1] OOSSANEN, P. van.: "Calculation of performance and cavitation characteristics of propellers including effects of non-uniform flow and viscosity", Ph. D. Thesis, Delft University, 1974

[2] POLITIS, G. K.: "A lifting line equivalent profile method for propeller calculations". Accepted for publication at the Journal of Ship Research, Feb. 1984.

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ON DETERMINATION OF EQUIVALENT CONDITION OF SCREW PROPELLERS OPERATING IN OPEN WATER AND BEHIND CONDITIONS

Introduction

As is known, in their routine practice many experimental tanks use two different methods for determining the open water propeller hydrodynamic characteristics.

- at Reynolds numbers equal to those obtained in behind conditions
- at overcritical (maximum attainable) Rn.

The using of open water propeller hydrodynamic characteristics obtained when applying both methods and subsequently being different for both cases, leads to certain differences in the interaction coefficients (w and r_R) as well as in predicted performance characteristics [4].

The absence of an uniform approach concerning this matter is reflected even in the 1978 ITTC Performance Prediction Method for Single Screw Ships [5], accepted as a standard one, where the requirement for $R_n > 2.10^5$ allows the use of open water screw propeller hydrodynamic characteristics determined by both methods. In this way, a possibility is created for differences in predicted performance characteristics reflecting ultimately on the evaluation of C_p and C_n coefficients.

Brief description of the proposed approach

Subject of the present work is the proposed approach for determination of Rn when carrying out open water tests thus ensuring a conformity of conditions when operating in open water and in behind conditions. The development of a boundary layer on propeller blades depends on the flow parameters in which they operate as well as on the degree of flow turbulence (DFT).

In the present stage, the conditions of screw propeller operation in open water and in behind conditions are characterized solely by the Reynolds number,

$$Rn_{0.75R_{OW,B}} = \frac{C_{0.75R}\sqrt{V_A^2 + (0.75\pi Dn)^2}}{\nu} \qquad (1)$$

Basing on the fact (confirmed by the experimental results given further on) that the DFT in behind conditions of a single screw model (ε_B=17.3%) is considerably higher than the one in open water (ε=2%), a relation between flow velocity $\sqrt{V_A^2 + (0.75\pi Dn)^2}$ and its degree of turbulence is searched, ensuring equal development in both cases of boundary layer on screw propeller blades.

The effective screw propeller Rn is defined in a way similar to the one used in aerodynamics:

$$Rn_{ef} = Rn \cdot K_t \qquad (2)$$

where Rn - screw propeller Rn in open water and behind conditions (defined by eq. (1), and where
K_t - turbulence coefficient

The K_t coefficient is defined as a relation between Rn critical numbers, for a sphere, in a flow with a standard degree of turbulence and those of a flow in which the screw propeller is operating.

Assuming that, as a result of systematic investigations, a standard DFT in open water test (ε_{ST}) is chosen, then for all other experiments of screw propellers in open water

$$Rn_{ef_{OW}} = Rn_{OW} K_t \qquad (3)$$

As a result of sphere's calibration, the relation between the Rn critical number and the DFT can be determined

$$Rn_{cr} = Rn_{cr}(\varepsilon) \qquad (4)$$

At screw propeller operating in behind conditions where the DFT ε_B differs from ε_{ST}, the turbulence coefficient and the effective Rn of screw propeller in behind conditions can be determined as follows:

$$K_t = \frac{Rn_{st_{cr}}(\varepsilon_{ST})}{Rn_{B_{cr}}(\varepsilon_B)} \qquad (5)$$

$$Rn_{ef} = Rn_B \cdot K_t \qquad (6)$$

Taking into consideration the fact that the development of the boundary layer on propeller blade depends on propeller

Reynolds number and on the DFT, the conformity in operation conditions in behind conditions and in open water can be determined from the equality of the effective Reynolds numbers:

$$Rn_{ef_{OW}} = Rn_{ef_{B}} \qquad (7)$$

Substituting (3) and (6) in (7), we can obtain:

$$Rn_{OW} = Rn_{B} \cdot K_{t} \qquad (8)$$

Equation (8) allows the determination of the Rn at which the screw propeller should be tested in open water (at known screw propeller Rn and degree of turbulence in behind conditions), guaranteeing equivalence of operation conditions in both cases.

Preliminary experimental results

The measuring of the flow velocity and DFT is carried out with a wedge-shaped film probe DISA 55R32 and a thermoanemometer unit DISA 55M. The results obtained are recorded on a tape and are statistically processed by a computer.

Experimental Determination of the DFT in Propeller Plane in Behind Conditions.

The measurements are carried out in the propeller plane of a single-screw 29900 TDW tanker model with block coefficient C_B = 0.800 without a rudder. The basic dimensions of the ship & model are presented in Table 1.

The distribution of the DFT is presented in Fig. 1. The mean value is ε_B = 17.8%.

TABLE 1

CHARACTERISTICS	SYMBOL	MODEL	FULL SCALE
Length between perpendiculars	L_{pp}(m)	5.786	162
Length of waterline	L_{WL} (m)	5.954	166.7
Breadth	B (m)	0.929	26.0
Draught at FP	T_F (m)	0.391	10.95
Draught at AP	T_A (m)	0.391	10.95
Volume dispalcement	Δ (m^3)	1.682	36923
Wetted surface	S (m^2)	8.559	6711
Scale	(-)	1 : 28	-
Block coefficient	C_B(-)	0.8001	0.8001
Midship section coefficient	C_M (-)	0.993	0.993
Longitudinal centre of buoyancy rel. to L_{pp}	LCB(%)	+2.4	+2.4
Prismatic coefficient	C_P(-)	0.8057	0.8057
Stern hull form	-	U - V	U - V

The influence of the radial and circumferential velocity components on the thermoanemometer readings is neglected.

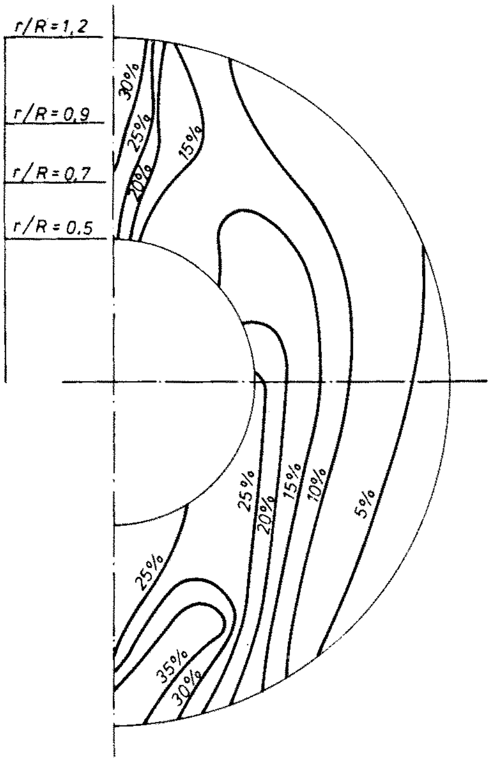

Figure 1

TABLE 2

CHARACTERISTICS	SYMBOL	MODEL
Diameter	D(m)	0.1864
Pitch ratio	$(P/D)_{0.75R}(-)$	0.677
Expanded blade area ratio	A_E/A_0 (-)	0.675
Number of blades	Z	5
Direction of rotation		clockwise

Experimental Determination of the DFT and the Nonuniformity of the Velocity Field in Open Water Tests.

The geometrical characteristics of the ship model propeller are presented in Table 2. The DFT and the nonuniformity of the velocity field are determined simultaneously with the carrying out of the routine propeller open water tests. The DFT is $\varepsilon_{OW} = 2$ %, and the nonuniformity of the velocity field is approximately 4÷5%.

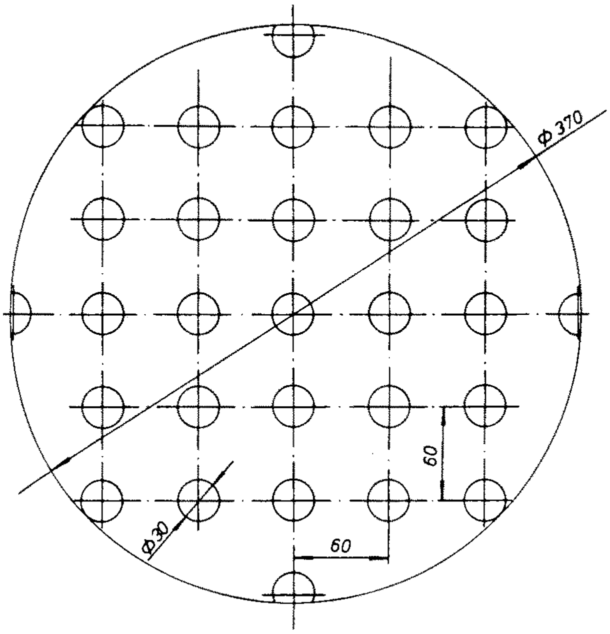

Figure 2

Influence of the DFT on the Propeller Hydrodynamic Characteristics.

A plate with circular holes is chosen to serve as a turbulence generating device. The geometrical characteristics of the plate are presented in Fig. 2. This device provided a mean value of $\varepsilon = 14{,}6$ %.

The results (presented in Fig. 3) from the open water tests, with and without turbulence generating device, show that the propeller open water hydrodynamic characteristics strongly depend on the DFT at Rn equal to the corresponding value in behind conditions. For $Rn > Rn_{cr}$ the propeller open water hydrodynamic characteristics practically do not depend on the DFT.

On the basis of the experimental results the following conclusions can be drawn:

The model propellers of single screw ships operate in behind conditions in flow with high degree of turbulence.

In the open water test condition without application of special turbulence

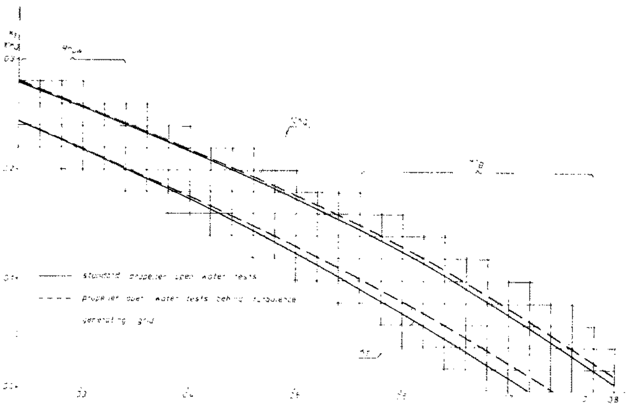

Figure 3

generating devices the screw propellers operate in flow with definitely lower degree of turbulence than when in behind conditions.

The propeller open water hydrodynamic characteristics, obtained at Rn equal to Rn in behind conditions, strongly depend on the DFT.

Conclusion

The preliminary experimental results obtained so far prove the necessity of introduction and practical implementation of effective Rn for determination of equivalent propeller operating conditions in open water and in behind conditions.

After a thorough experimental investigation and verification, the suggested approach can be incorporated as a standard procedure into the "1978 ITTC Performance Prediction Method for Single Screw Ships" [5].

Acknowledgement

The authors wish to express their deep gratitude to Dr. K. Varsamov, from BSHC, for his valuable support in the realization of the present investigation.

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[2] TAMURA, K.: "Speed and Power Prediction Techniques for High Block Ships Applied in Nagasaki Experimental Tank", Mitsubishi Technical Bulletin No 103, 1976.

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K. NAKATAKE and R. YAMAZAKI -
Kyushu University, Department of Naval Architecture, Fukuoka, Japan

ON RUDDER DRAG

It is supposed in the 1978 ITTC Method that the thrust deduction fraction due to the rudder drag is 0.04 as seen in Eq. (1.18) of the Committee Report. We would like to state briefly the results of our researches in the hull-propeller-rudder interactions.

At first, we talk about the propeller-rudder interaction problem [1]. The section shapes of the rudder models used are shown in Fig. 1. They have rectangular planforms and NACA wing sections with three kinds of thickness-chord ratios, i.e. 0.09, 0.15 and 0.25, which are named MR-1, MR-2 and MR-3, respectively. Fig.2

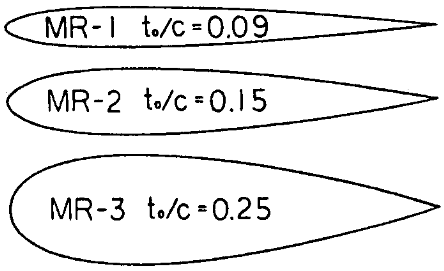

Fig. 1 Shapes of Rudder Sections

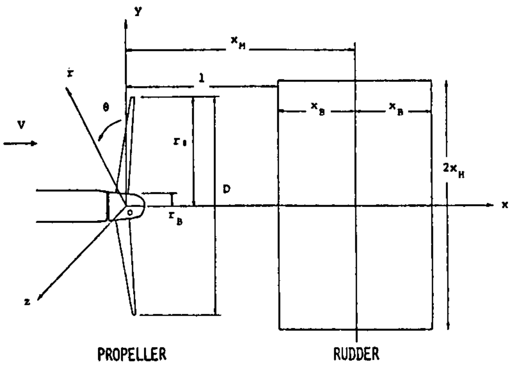

Fig. 2 Arrangement of Propeller and Rudder

shows the arrangement of the propeller and the rudder. Experiments were performed in Ship Research Institute of Japan by late *Mr Mariyama* by changing the rudder and the distance between the propeller and the rudder l. Then he measured the thrust and the torque of the propeller, and the rudder drag which is shown by K_{FT} (=Drag/$\rho n^2 D^4$) in Fig. 3. As the ratio l/D decreases, the thrust and the torque increase because of the rudder wake and at the same time the rudder drag increases. Moreover, as the thickness of the rudder increases, the rudder drag increases. It is very interesting to note that the drag of the thin rudder MR-1 becomes negative, i.e. a thrust. The corresponding calculations are performed by using the infinitely-bladed propeller and the rudder which is represented by the center-plane source distribution and the vortex distribution.

Results are shown by the solid lines in Fig. 3. From good agreements between experimental and calculated results, we confirm that the rudder drag consists of the pressure drag, the frictional (or viscous) drag and the leading edge thrust.

Next we treat the rudder drag in the hull-propeller-rudder interaction problem [2]. Fig. 4 shows the arrangement

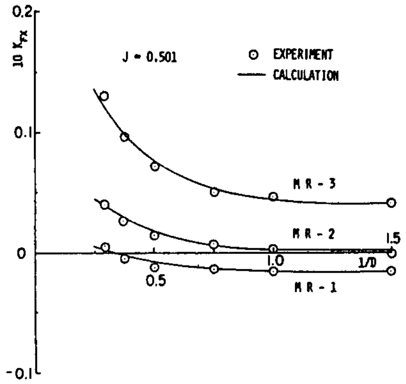

Fig. 3 Changes of Rudder Drags versus l/D

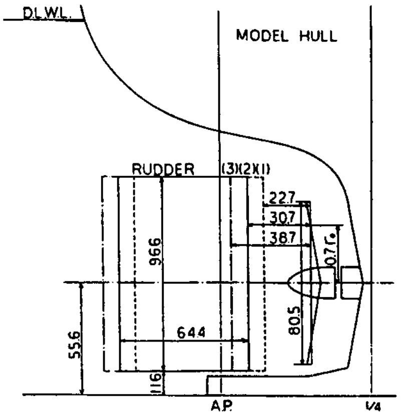

Fig. 4 Arrangement of Hull and Propeller and Rudder

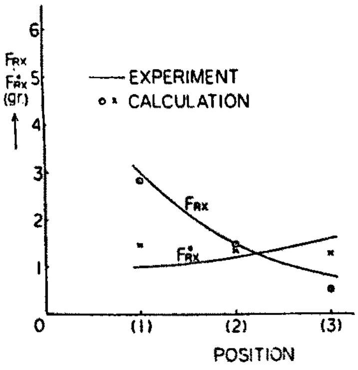

Fig. 5 Rudder Drags in Towed and Self-Propelled Conditions(MR-2)

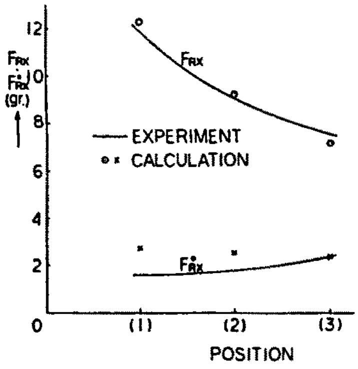

Fig. 6 Rudder Drags in Towed and Self-Propelled Conditions(MR-3)

of the three parts. The hull is an oil tanker model (L_{PP} = 3.00). Experiments were performed in Kyushu University by changing the rudder and the position of the rudder. We measured the rudder drag in the self-propelled conditions. Their results are shown in Figs 5 and 6 by solid lines. F^{O}_{Rx} and F_{Rx} mean respectively the rudder drag in the towed and the self-propelled conditions. We see that the thicker rudder MR-3 experiences much larger drag than MR-2, and the drag changes considerably with the distance between the propeller and the rudder. By using the source-represented hull and the above-mentioned propeller and rudder models, we calculate the propulsive performance of the model ship including the rudder drag. Calculated results of the rudder drag are also shown in Figs. 5 and 6 by circles. These agree well with experimental values. And we know that the rudder works quite differently in the self-propelled and the towed conditions.

At last, as to the thrust deduction fraction due to the rudder drag, we obtained 0.039 for A ship (bulk carrier, to/c=0.190, 1/D=0.246) and 0.048 for B ship (bulk carrier, to/c=0.155, 1/D=0.198). Therefore we may conclude that the 1978 ITTC Method estimates well the rudder effect in the thrust deduction fractions.

References

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[2] NAKATAKE, K., UEDA, K, ODA, K. and YAMAZAKI, R.: "On Hydrodynamical Force Acting on Rudder", Trans. of West Japan S.N.A. No. 63, 1982.

O. BJÖRHEDEN - KaMeWa Marine Laboratory, Kristinehamn, Sweden

ON A SLIGHT IMPROVEMENT IN SHIP MODEL TESTING

As we all know, some of the shortcomings in the ship model testing technique are the so called scale effects due to the large difference in Reynold's number in between the model and the ship. These scale effects, influencing hull resistance and propeller characteristics as well as the hull wake field and the interaction between propeller and hull, must be taken into account by various kinds of corrections which are applied to the model tests results.

A great deal of the problems is related to the viscous boundary layer along the hull which is relatively seen much thicker in the model than in the full scale. Consequently, with the hull model made properly to scale, which has so far been common practice, the total displacement of the hull and the boundary layer is relatively seen much larger in the model than in full scale, in particular towards the stern region. Some improvement in the situation should be possible to achieve by modifying the shape of the hull model, in principle such as to obtain the same outer configuration of the hull plus the boundary layer in model and full scale. Assuming turbulent flow the boundary layer thickness for model and ship can be estimated at various stations along the hull, e.g. by the approximate formula

$$\frac{\delta}{x} = 0.34 \cdot (R_{nx})^{-1/5}$$

where x is the distance from the bow and R_{nx} is the local Reynold's number. With a model speed according to the normal Froude law it appears that the ratio of model and ship relative boundary layer thickness will be proportional to the scale factor to the power of 0.3, i.e.

$$\frac{(\delta/x)m}{(\delta/x)s} = \lambda^{0.3}$$

Knowing this, a reduction of the transverse dimension of the model hull is obtained by applying a "width correction"

$$\Delta w = (\frac{\delta}{x})_s \ (\lambda^{0.3} - 1)$$

perpendicular to the hull surface.

A hull model modified as sketched above would get a shape which is gradually tapered off towards the stern, i.e. its breadth and draft would gradually become smaller in comparison with the nominal hull and finally even its stern contour would fall slightly ahead of the nominal contour. The propeller model and other appendages such as the rudder should of course be kept to scale and at their nominal positions.

With the proposed arrangements there is reason to expect some improvements in the scaling of several resistance and propulsion factors. In the first hand the wave pattern along the hull and the wavemaking resistance (the determination of which is the whole purpose of the towing test) ought to be more correct. Secondly the risk for "erroneous" flow separation taking place in model scale only would be slightly reduced. Moreover, the scale effect on the hull wake would be slightly reduced although most of the error in frictional wake remains. Finally the interaction between the propeller and the hull in the self-propulsion test will be more correct, in particular in cases where the propeller blade tips tend to interfere with the hull boundary layer in the top (12 o'clock) blade position.

M. SCHMIECHEN - Versuchsanstalt für Wasserbau und Schiffbau, Berlin, FRG

ON MODEL TESTING AND ANALYSIS METHODS

The present discusser would like to thank the Performance Committee for a most thorough survey including recent attempts to improve model testing and analysis methods. Concerning the account and evaluation of his own proposal the discusser would like to make only a few comments, an extensive exposition having been presented and discussed at the ONR Symposium on Naval Hydrodynamics in Hamburg last week.

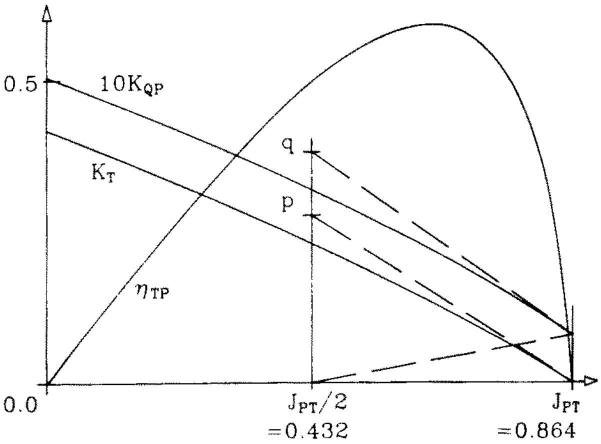
Quite clearly the method proposed is a conventional one, as is the traditional, and the conventions are explicitely stated as a coherent axiomatic system in accordance with modern methodology. The fact that elementary hydrodynamic theory of hull-propeller interaction is a model or interpretation of this set of conventions supports the plausibility of the latter, but does not support the interpretation of the Performance Committee that the conventions are (nothing else but) that elementary theory.

The interpretation of the proposed conventions in terms of measurements turned out to be much more complicated than expected. The first solutions, which have been tried at MARIN and found insufficient, suffered in fact from too many unsatisfactory ad hoc hypotheses. In the meantime this problem has been solved based on the concept of the equivalent state of vanishing thrust.

Without repeating last week's presentation just the resulting procedure for the determination of the propeller advance speed, i.e. the wake, may be shown for the benefit of the those who did not attend the ONR-Symposium. The problem to be solved is to reconstruct the propeller advance speed from thrust and torque measurements at different frequencies of revolution. The solution and its verification using the open-water results of a randomly selected CP-propeller is shown in the Figure enclosed.

Verification of the interpretation
rule using open-water tests
e.g. VWS Propeller No. 1349

$J_{PT}=(2\pi/10)*q/p=0.8635$



The rule shown is so simple that it has been and can be tested rapidly on large numbers of propellers. In any case the value of the velocity determined in the way proposed was very close to the value measured.

In the paper mentioned an extensive error analysis of the procedure and an application to quasisteady propulsion tests have been reported in detail. It is hoped that the results are convincing not only the Performance Committee that the whole approach of determining wake and thrust deduction from propulsion tests alone is now ready for testing.

M. ABE - Akishima Laboratory, Mitsui Engineering & Shipbuilding Co, Tokyo, Japan

ON THE PERFORMANCE PREDICTION METHOD AND UNCONVENTIONAL HULL-PROPULSOR COMBINATION

It is highly appreciated that the Committee has successfully worked for investigating the present technologies and the expected future works.

Among those the most concerned will be the performance prediction method. Many shipowners and shipbuilders were obviously interested in the method and intended to use it for their ships.

In this situation, I would like to ask the Committee to prepare an explanatory document of the 1978 ITTC Performance Prediction Method, so that it will be open for the outside of ITTC and make clear its definition and restriction as well as the tolerance allowed in the method.

In the document the following terms should be noted:

(i) The kind of ship and loaded condition, to which the method is to be applied.

(ii) The correlation allowance including hull roughness and the scale effect of wake should be stated so as not to ignore the discrepancies really found among the individual institutions and shipbuilders.

(iii) It is noticed that stock model propellers are currently used at some institutions. In the case the indication should be noted how to correct to the full scale.

The other interest is that the Committee has discussed the rational method of the performance prediction for unconventional hull-propulsor combinations.

The basic concept should be discussed in the theoretical background or in empirical aspects. However, the Committee has commented that the tank DHP is to be applied to judge the possibility of it at the primary stage. This disregards the hydrodynamic complex present in hull-propulsor system, especially when there is a large change of wake.

The facts obtained at the tank give fundamental informations in resistance and propulsion as well as hull-propeller interaction, and those informations are analytically processed for predicting the full scale performance.

Thus I strongly insist that the Committee should recommend the relevant analysis method for unconventional hull-propulsor combinations without admitting the tank DHP.

M. W. C. OOSTERVELD - Maritime Research Institute Netherlands, Wageningen, The Netherlands

ON TESTS WITH SHIP MODELS HAVING DUCTED PROPELLERS

With the increasing demand for energy saving there is a growing interest in alternative propulsion devices. In this respect the application of ducted propellers, symmetrical and asymmetrical ducts before the propeller, semiducts and tunnels, is in many cases considered. These propulsion devices have in common that on one hand they increase the momentum flow through the propeller disk which leads to a higher propeller efficiency, on the other hand they improve the after body flow which leads to a lower ship resistance.

The Performance Committee proposes that in comparative tests with different "hull propulsor" arrangements the same friction deduction or tow rope force is applied.

The mentioned propulsor arrangements are intended to improve to a certain extent the afterbody flow with the result that the resistance of the hull decreases. This lower resistance determines the form factor and this leads to a different friction deduction.

Another point of concern is the extrapolation of the results of tests with ducted propellers. At the normal self-propulsion point of the ship, the propulsion system is overloaded due to the wake-scale effect and this leads to a too high predicted contribution of the duct. It seems advisable therefore that the prediction of the propulsive performance is based on a test condition in which the thrust coefficient, K_T, is the same for model and full-scale.

We like very much to have the comments of the Committee with respect to our remarks concerning differences in friction deduction in case of improved afterbody flow by the propulsor and the extrapolation of the results of tests with ducted propellers.

FU-SHENG CHEN and QU-TAO QIN -
Shanghai Ship and Shipping Research Institute, Shanghai, China

ON THE WATER DEPTH FOR MEASURED MILE TRIAL OF RIVER VESSELS

Generally the performance in deep water is taken as the main characteristics of a river vessel. But it is very difficult to find the adequate waterway without shallow effect for measured mile trials in the river; therefore, there is a practical question raised, i.e. how to judge if the trial results should be corrected for shallow water effect, and whether it is suitable to apply the ITTC requirement or not.

The shallow/deep water resistance test data of 16 river vessels are analysed. These models are made of wood, about 2.5 m in length and fitted with trip wires of 1 mm diameter at No. 18 and 19 stations. These models were tested in five water depths at the model speed corresponding to $F_h = 0.5-1.1$. The main particulars of 16 models are listed in Table 1.

Table 1

Ship model	L/B	B/T	C_b
Y.P.series no. 7	6.003	5.2	0.5024
" 15	4.761	5.2	0.5604
" 17	6.003	3.6	0.5604
" 18	6.003	4.4	0.5604
" 19	6.003	5.2	0.5604
" 20	6.003	6.0	0.5604
" 23	7.245	5.2	0.5604
" 31	6.003	5.2	0.6184
M.S. 333	5.675	8.66	0.776
334	5.150	7.50	0.611
335	4.7727	7.333	0.650
River tug 1330	4.6296	4.50	0.6173
GRP pass. boat	5.50	6.6667	0.5483
River pass. boat	5.2174	7.6667	0.5323
Li-jiang pass. boat	4.5714	15.556	0.6895
River pass. ship	6.2222	7.5	0.540

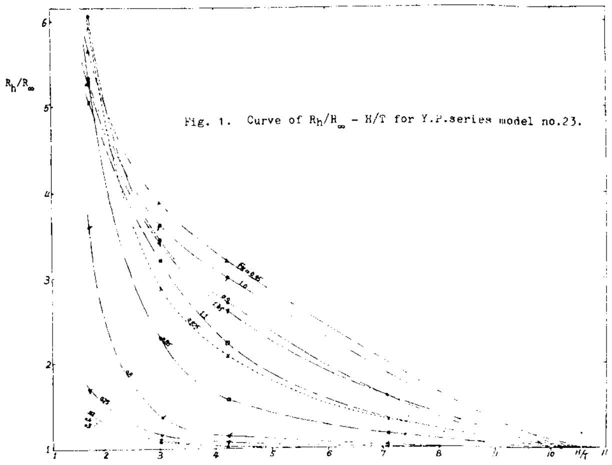


Fig 1 Curve of R_h/R_∞ - H/T for Y.P.series model no.23.

Fig. 1 shows an example of relation between resistance ratio R_h/R_∞ and water depth/draft ratio for one vessel. The curves for vessels of different hull forms show much similar shape but significant difference in R_h/R_∞ values. It shows clearly that the effect of hull form should be taken into account in considering quantitatively the shallow water effect.

Applying the regression techniques for the h/T values at $R_h/R_\infty=1$ obtained from the figures for these models, it gives

$$(h/T)_{min} = 91.0101\ F_L^2 - 0.086878\left(\frac{L}{B}\right)^2 + 0.25793\left(\frac{L}{B}\right)\left(\frac{B}{T}\right) - 0.75276\left(\frac{B}{T}\right)C_b - 89.947\ F_L^3 - 2.7971\left(\frac{B}{T}\right)F_L^2 - 0.20757 \quad (1)$$

There are 189 points to be used in regression and the deviation of $(h/T)_{min}$ is $\sigma=0.4826$ which corresponds to $\sigma/(h/T)_{aver} = 0.0657$.

Table 2

Ship model	Required min. h/T for 16 river vessels by (1)	by ITTC	by SNAME
Y.P.series no. 7	8.1295	10.1465	11.5517
" 15	6.6687	6.8411	10.3925
" 17	5.3708	5.6921	10.3925
" 18	6.4861	6.9489	10.3925
" 19	7.1731	8.2123	10.3925
" 20	7.8603	9.4757	10.3925
" 23	7.4095	9.9114	10.3925
" 31	6.2002	6.8411	9.2333
M.S. 333	6.2664	8.8284	6.08116
334	7.5028	8.2800	9.3811
335	6.3638	8.1240	8.6016
River tug 1330	5.448	6.364	9.2534
GRP pass. boat	8.2672	10.1008	10.6344
River pass. boat	8.9497	11.6904	10.9536
Li-jiang Pass. boat	9.6812	10.5621	7.8086
River pass. boat	9.6649	13.2565	10.7990

Use $C_b = 1.08 - 1.68\ F_L$ to get F_L for each of 16 models, then calculate $(h/T)_{min}$ by equation (1) and compare them with those required by ITTC and SNAME: this gives Table 2.

It can be concluded from the comparison that

(1) ITTC requirement is suitable for river vessels in case of sufficient water depth.

(2) In case of restricted water depth, the suggested regression equation may be used as the limit depth without shallow water effect correction for river vessels.

FU-SHENG CHEN, GUANG-YUAN ZHU and GUO-XIONG ZHANG - Shanghai Ship and Shipping Research Institute, Shanghai, China.

ON SOME CORRELATION RESULTS OF SINGLE SCREW SHIPS

Trial data of 12 actual ships of three different types were collected as shown in Table 1. The wood models, about 4.5-7.0 m long of these ships, were

Table 1 Principal particulars of ships

Ship type		B	F	G
No. of sister ships		6	3	3
L_{bp}	M	147.18	145.0	213.7
L_{wl}	M	157.00	148.0	218.3
B	M	20.4	23.0	32.24
T	M	9.2	9.0	12.5
∇	M^3	19440	21818.	70760.
C_b		0.693	0.727	0.8243
C_p		0.704	0.730	0.8276
Cm		0.985	0.996	0.996
C_w		0.829	0.883	0.903
Lcb	% L_{bp}	-0.34	1.96	2.341
Propeller dia.	M	5.51	5.25	6.9
Pitch ratio		0.926	0.786	0.8486
Model scale		23	33	30

Table 2 No. of trials for load and ballast conditions

	Condition	
Type of ship	Full load	Ballast
B	6	5
F	-	3
G	-	3
Total no. of trials	6	11
Total no. of ships	12	

Table 3 Average value of Correlation Factors for each type of ships

a) Full load condition

Ship type	No.of data points	C_p	C_n	$\sigma n(C_p)$	$\sigma n(C_n)$
B	14	1.093	1.0343	0.0395	0.00675

b) Ballast condition

Ship type	No.of data points	C_p	C_n	$\sigma n(C_p)$	$\sigma n(C_n)$
B	16	1.1330	1.0494	0.0648	0.02468
F	7	0.991	0.9882	0.0332	0.0155
G	15	→0.867	1.001	0.0325	0.0086

tested at the corresponding condition of trials in No. 2 tank of SSSRI. The traditional method and the 1978 ITTC prediction method were used in analyzing for these ships.

The analyzed results for these ships were given in the form of mean values as shown in Table 3. All the figures were carefully checked and the values of C_p for these ships are of the normal order. There may be a question to be raised whether it is reasonable that the longer ship gives a higher ΔC_f according to the 1978 ITTC prediction method. The question may not be so clear to us, since the data of only one ship is available, and it is strongly hoped to get more trial data for ships of 200 m and more in length.

WANG HUAI - Marine Design & Research Institute of China, Shanghai, China

ON THE REPORT OF PERFORMANCE COMMITTEE

The extension of "1978 ITTC performance prediction program" to fine ships and to twin screw ships is reported in p. 295 of Proceedings Vol. 1. MARIC has used 1978 prediction program for ballast conditions of bulk carriers; the results are not so good as in the full load conditions. As is well known, the contract between ship owner and shipyard only gives the trial speed of dry cargo ships in ballast condition, since it is very expensive to conduct a speed trial of dry cargo ship in full load condition. Therefore the accuracy of prediction in ballast condition is very important to ship designers. MARIC asks Performance Committee to pay some attention to this problem.

H. LACKENBY - Formerly BSRA, Wallsend, United Kingdom

ON MEASURED MILE TRIALS

I should like to say a few words about the guide for Measured Mile Trials referred to on p. 291 and particularly the question of approach runs.

It is stated here that these are not specified but the following is suggested.

(a) High Speed Cargo Liners ~25 ship length
(b) Tankers 65.000-100.000 tdw ~40 ship lengths

This seems rather rough and ready and inadequate to me. I immediately ask myself what the background to this is:

(i) What fraction of the terminal speed is achieved - which of course is related to the accuracy of measurement.
(ii) What reduced speed torque characteristics are assumed for the propelling machinery? Is it constant torque as for a Diesel Engine or constant power as for a Turbine? There is an important difference which affects the answer.

(iii) How about ships other than tankers and high speed cargo liners? For example ferries, tugs, warships and so on.

I do not want to give the impression that I am being difficult here because I am happy to say that the work has already been done to cover any type of ship and both types of machinery.

Moreover, the results were generalized and put in such a form that the required approach run could be readily determined for any ship and any required degree of accuracy.

It is all given in a paper to the Institution of Engineers and Shipbuilders in Scotland in 1952 by H. Lackenby.

W. A. CRAGO - British Hovercraft Corp. Ltd., Test Facilities, Isle of Wight, United Kingdom.

ON FORM FACTORS

I would like to refer to the section of the Committee Report dealing with "form factors".

I have always thought that form factor methods become fashionable mainly because of the influence of workers in the field of aerodynamics. However, they do not seem to work too well in tankery, primarily because they can not be determined accurately and uniquely. I have pointed this out at a number of our conferences and *Prof. Harvald* has now provided further evidence to support my contension.

The Committee has given more space to this subject in their Report on this occasion but I submit that this part of the Committee's work lacks both authority and assurance. My view appears to be supported by the list of significant objections to the form factor method given in the Report itself on page 275.

Imagine the reaction of shipowners, operators and builders - our customers - on reading this list of nine objections to the method and then being asked to pay more for their model tests to cover the experiments necessary to derive the form factor!

If the Committee can not offer a simple, practical method of deriving a unique value for the form factor from model experiment I suggest that they recommend tank practioners to abandon the use of form factors until such time they may be better understood.

K. R. SUHRBIER - Vosper Thornycraft (U.K.) Lud., Portsmouth, United Kingdom

ON THE REPORT OF THE PERFORMANCE COMMITTEE

I like to thank the Committee for their interesting report. I appreciate very much the steps taken to extend the 1978 procedure for twin screw ships and in particular also the discussion on appendage resistance.

With regard to the discussion on model/ full scale correlation, I would like to make a few points:

1) As far as resistance experiments are concerned, the water of some test tanks is regularly treated against the influence of biological effects (of long-chain molecules changing with temperature or time of the year) on the model skin friction, in others it is not. This may, or may not, be important for the comparisons made, but they can be significant. This subject has been discussed by the ITTC many years ago and I wonder wether the Performance Committee together with the Resistance Committee (or, maybe, the Resistance Committee only) should not again look at this. I beleive we have no real recommendation on this subject, and the monitoring of this effect - if done at all - may sometimes be left to some subjective judgements.

2) In the case of propulsion tests for ships with rudder (s) in the propeller race, the associated 'appended resistance' tests are either carried out with the rudder (s) fitted - as recommended to the ITTC several years ago - or without. (The latter considers the rudder and propeller arrangement as one propulsion unit.) These difference in techniques used in different facilities can have an effect on the propulsive efficiency components and the predicted power, certainly in the case of twin screw ships with finer forms. May I ask whether this has been considered in the comparisons given and which approach has been used? I also suggest that this problem should be adressed by the next Performance Committee.

As far as the propulsion factors of high-speed craft are concerned, I feel that some (or more) crossreferences to the reports of other Committees might have been appropriate. This subject has in some depth also been dealt with in the sections on High-Speed Propulsion of the Cavitation Committee of the 16th and the 17th ITTC (in particular as far as thrust deduction and compatibility of approaches is concerned in the case of propeller shaft inclination, referred to on page 307).

M. A. ABKOWITZ - MIT, Cambridge, Mass. U.S.A.

ON HULL ROUGHNESS

The Committee reports the difficulty in obtaining an adequate expression for estimating the increase in drag due to roughness. I should like comment on the general form that the expression should take when the physical aspects of the fluid mechanics involved are taken into account.

The increase in drag due to roughness, ΔC_f, results from turbulent eddies which are produced by the fluid flow over the roughness protruding above the smooth surface. However, because of the boundary condition that the velocity component normal to the wall must vanish at the wall, turbulent flow can not exist extremely close to the wall. This roughly explains the existence of a thin laminar sublayer, close to the wall, within the relatively much larger turbulent boundary layer. If the height of the roughness is smaller than the thickness of the laminar sublayer, the flow over the roughness will be laminar - no eddies - and therefore there is no significant increase in drag. However, if the height of the roughness is larger than the sublayer thickness, the flow over the rough geometry will be turbulent with the resulting eddy drag producing a significant roughness drag ΔC_f. Obviously the more of the individual roughness which protrude through the laminar sublayer, the greater the magnitude of ΔC_f. From that rather simple reasoning discussed above, the magnitude of ΔC_f should depend on the height of the roughness as compared to the thickness of the laminar sublayer and the surface density of the roughness (number of roughness elements per unit area). As with all boundary layer phenomena, the height of the sublayer is a function of *Reynold's* number. Figure 1 shows approximately the relationship between the laminar sublayer thickness and the Reynold's number R_n for flow over a flat plate. In the figure, we are only concerned with the two plots δ/x and y_1/δ which are the turbulent boundary layer thickness δ divided by the distance from the leading edge of the plate x and the thickness of the laminar sublayer y_1 divided by the thickness of the turbulent boundary layer. Hence, the thickness of the boundary layer in relation to distance along the plate, y_1/x, is given by the product of the two curves. The product y_1/x is of large magnitude at low Reynold's number, rapidly drops off as Reynold's number increases, and is approximately a constant low value at Reynold's numbers greater than 10^8.

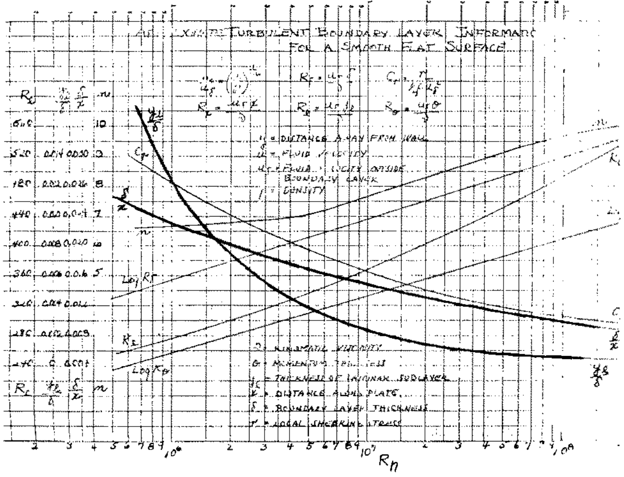


Figure 1

If we move up in Reynold's number by moving aft of the leading edge at constant velocity, we will find a significant portion of the forward part of the plate where the roughness is below the laminar sublayer and therefore contributing nothing to ΔC_f; at some distance down the plate the roughness will protrude through the sublayer (which has become thinner) and contribute to a significant ΔC_f. If the plate is long enough, than the sublayer thickness and the roughnesses in this region do not add significantly to the ΔC_f. Of course, the effect of the roughnesses ahead of the far downstream part of the plate can alter quantitatively the behavior far downstream but not qualitatively. If we fix the length of the plate and increase the Reynold's number by increasing the speed, then at very slow speeds (low Reynold's number) the roughness all along the plate will be below the laminar sublayer and ΔC_f will be zero or small. As we increase speed (and Reynold's number) the roughness elements at the far end of the plate will emerge above the sublayer and cause a significant ΔC_f, and at a higher speed the roughness elements on the entire plate will protrude and the value of ΔC_f will reach a maximum. A further increase in speed (Reynold's number 10^8) will cause the sublayer at the aft end of the plate to thicken beyond the height of the roughness and begin to cause a reduction in ΔC_f.

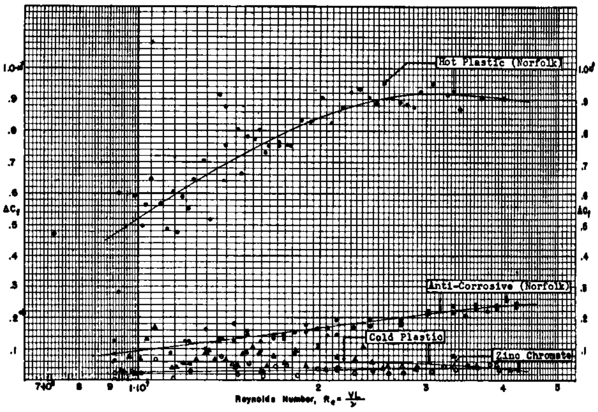


Fig 2 Increase of Resistance Coefficient of Friction Plane with Paint Roughness

Figure 2, taken from a paper by *Todd* as published in the Transactions of SNAME in the early 1950's, entitled "Skin Friction Resistance and the Effect of Surface Roughness", shows the results of a drag test on 20 feet long plank when coated alternately with a primer (zinc chromate) anticorrosive, cold plastic and hot plastic paints. Although similar phenomena exist for the other paints but to a lesser degree, the hot plastic paint curve is selected for analysis. This type of paint has a rather heavy density of roughnesses of the order of 0.01 inches in height. Since the plate has a fixed length of 20 feet the increase in Reynold's number is a result of increased speed. Here we find exactly what was predicted in the previous paragraph. ΔC_f tends to zero at the very low Reynold's numbers, increases steadily with Reynold's number, becomes saturated at a high Reynold's number (max. ΔC_f) and tends to begin to drop off slowly with increasing Reynold's number at high Reynold's number.

The above arguments tell us that ΔC_f should depend on roughness height, roughness density and the Reynold's number based on length (R_x) or

$$\Delta C_f = f\left(\frac{k}{L}, \text{density}, R_x\right)$$

It can not be a simple function of R_x since it must reflect an increasing ΔC_f for an increase in Reynold's number at low R_x, then a saturated or constant value for a significant range of R_x and slightly decreasing values at very large Reynold's numbers.

Experiments involving pipe flow appear to have limited usefulness in the determination of ΔC_f for ship hulls because both the turbulent boundary layer thickness and laminar sublayer thickness are constant along the length of pipe precluding a change in boundary layer thickness as a function of R_x which is necessary to simulate exterior flow along a surface.

M. ABKOWITZ - MIT, Cambridge, Mass., U.S.A.

ON SHIP SPEED TRIALS USING SYSTEM IDENTIFICATION TECHNIQUES

The lack of data on the resistance of a flat surface or of a ship's hull at the very large Reynold's number at which ships operate has seriously limited the ability to properly correct for scale effect in the prediction of ship resistance from the resistance measured on the model. The "k" factor was introduced several years ago in order to avoid a negative ΔC_f which was obtained for several high-block large tanker vessels when the previously used *Froude* method was employed in the process of making the comparison between full scale trial results and predictions from model test results. As stated in the Committee Report, we seemed to have reached a significant level of confusion and uncertainty in the recent and current use of the "k" factor method.

Recent work at MIT has indicated a great potential for a method of "measuring" the resistance of a ship from rather simple full scale trials. The method involved does not have the disadventages of the usual full scale trial for estimating the ship's drag, such as installing a thrustmeter on-board, noisy thrust measurements, uncertainty about the speed and the use of model measured wake and thrust deduction factors (scale effect). The trial maneuvers consist of running the ship up to speed, maintaining the steady speed for a short while on a straight heading, then cutting power allowing the propeller to "windmill" and repeating the same sequence when heading on an opposite course (180° difference). During these runs the forward speed over the ground and the propeller RPM are measured (for data) along with the rudder deflection and heading (to assure a reasonably straight course). The speed and RPM data are analyzed by system identification programs developed at MIT, which were very successfully applied to the data obtained from the Esso Osaka maneuvering trials in "measuring" (identifying) the linear and non-linear coefficients for the ship to be used in proper maneuvering simulation equations of motion.

On the trial runs for measuring resistance there are two coefficients that need to be identified - the total resistance coefficient C_T and the added mass for forward acceleration, $-X_{\dot{u}}$, along with the magnitude and direction of the current. Two conditions are required for the ship(s) selected for these trials, (1) the ability to measure speed relative to water or over the ground with sufficient accurancy and (2) the ability to "windmill" the propeller. The *Doppler* sonar log does have the capacity to measure speed with sufficient accuracy, as demonstrated in the Esso Opaka trials, and a ship equipped with steamturbine engines or a clutch has the capacity to windmill the propeller. Every ship has a rudder indicator, an R.P.M counter, and a gyrocompass.

The potential success of this trial method has been demonstrated through the use of "noisy simulated data" which was processed through the system identification programs in the same manner as was done in the case of determing the potential success of the "Esso Osaka" maneuvering trials before these trials were carried out. Figure 1 shows the results of using "noisy simulated data" (with the noise level of the Doppler sonar log as observed during the Osaka trials) in the identification procedures. The simulated trial run lasts 800 seconds. Initial value estimates (at t=0) were purposely set well away from the actual values of C_R, $m' - X'_{\dot{u}}$ (apparent mass) the magnitude of the current and the direction of the current, which were used to generate the noisy data. One can observe from the figure that all four quantities are identified with good accuracy.

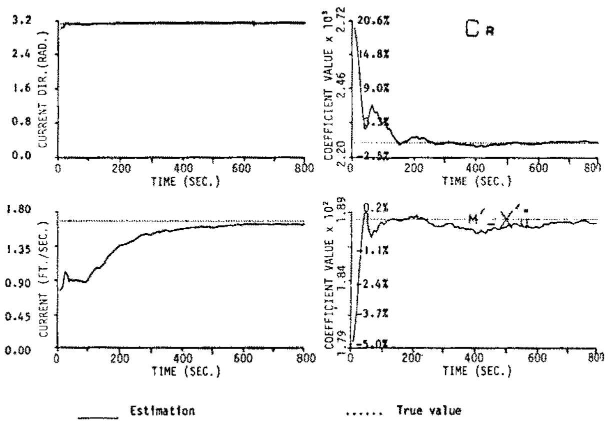


Fig 1 Results of identification. The data files of two maneuvers in opposite directions are parallelly processed to estimate the resistance coefficient C_R together with $m'-X'_{\dot{u}}$. Analysis done on simulated noisy trial data.

The resistance, as identified, includes the parasitic drag of the "windmilling" propeller and a correction needs to be made for this effect. Rough estimates give the parasitic drag of a locked propeller of a tanker in the order of 25% of the ship's drag and the "windmilling" propeller in the order of about 5 %. The parasitic drag of the propeller can be estimated from the propeller characteristic curve obtained from model tests since the J-value is known from the measurements of speed over the ground, current and RPM. Even if the estimate is off by 20% the resistance coefficient is only affected by 1 % (0.20x0.05=0.01).

These simple trials can be carried out during normal ship operations on a calm day during the voyage, thereby minimizing windage corrections and the extraneous effects of waves. Since these tests require no additional instrumentation on board the ship and can be done during a regular voyage they represent essentially no cost to the shipowner. This should reduce the reluctance of the owners to conduct such trials which can significantly contribute in the long run to obtaining better prediction from model test to the benefit of the ship owners and maritime technology. The above is not just conjecture. With little difficulty the owners of a large fleet has agreed to furnish us a ship on which these trials will be carried out during a routine voyage of the ship. There is an additional benefit to the shipowners in the fact that similar trials conducted say 6 months apart can indicate the effect of fouling on the resistance.

S. GORANOV - BSHC, Varna, Bulgaria

ON THE DIRECT CALCULATION OF AVERAGE THRUST AND TORQUE VALUES

My comment is concerning the approaches for direct calculation of time and volumeric average thrust and torque values of the propeller operating in nonuniform wake field behind the ship hull.

Prof. Ikehata suggests such a method capable of ship performance prediction. As a basic reason for the certain disrepancies between calculated and experimental results the author points out the neglection of the effective velocity field, i.e. the influence of the operating propeller on the nominal wake field.

In BSHC an attempt is made for practical application of Prof. Ikehata's method, extended to consider the effective wake radial distribution. The latter is estimated using the method developed by Dyne, modified in BSHC. Thus, on the basis of predicted full scale effective wake radial distribution the propeller time and volumetric average thrust and torque are calculated and ship performance prediction is realized.

An extract of the results (which will be published in more details in the Proceedings of 13th SHMSS of BSHC) is shown on Fig. 1 and Fig. 2. From the table, incorporated in Fig. 1, is evident that the coincidence between w_{TM} and w_{EM}, calculated acc. to the approach developed in BSHC, is rather better than in case of use of Prof. Ikehata's approach (where these figures are, respectively, 0.605 and 0.663).

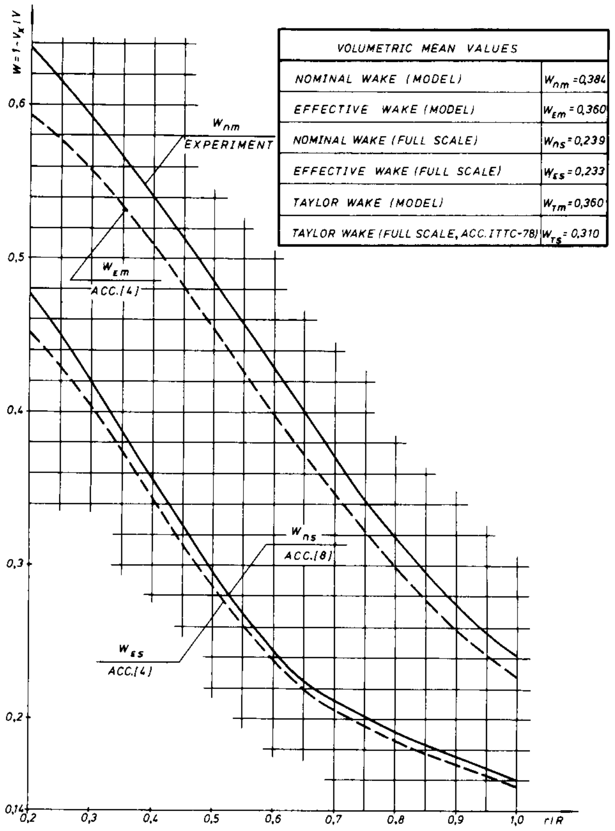


Fig 1 Radial Distribution of Nominal and Effective Wakes

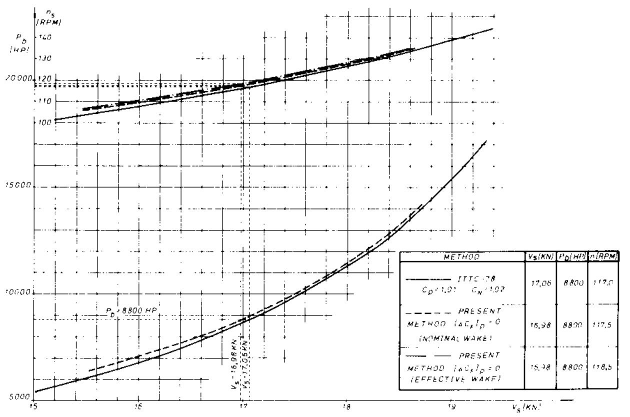


Fig 2 Performance Prediction of Full Scale Ship. $T_F = T_A = 9.01$ m, $P_{b_{nom}} = 8800$ HP, $n_{nom} = 117$ RPM

On the basis of the investigation performed the following conclusions could be drawn:

- The consideration of the effective velocity distribution in Ikehata's approach improves the accurance of calculated model scale propeller hydrodynamic characteristics (T and Q), while the positive effect from such a consideration for the full scale characteristics depends mainly on the adequate scaling of the nominal wake field.

- For the ship treated in Prof. Ikehata's work and in BSHC the methods of *Sasajima*, on the one hand, and of *Tanaka*, on the other, overpredict the nominal wake field scale effect. For this reason the possible routine application of performance prediction methods of Prof. Ikehata's type necessitates adequate numerical prediction on full scale nominal wake field.

And finally I would like deeply to support the Recommendation No 7 of the Performance Committee's Report.

II. REPLY BY THE PERFORMANCE COMMITTEE

The Committee is grateful for the considerable interest shown in its Report, as evidenced by the large number of discussions received, and would like to thank all of the discussers for their contributions

Prof. Harvald has commented on the consistency of subdividing the resistance components by means of the form factor. The examples given clearly show that the determination of the form factor from low-speed resistance measurements by Prohaska's method involves uncertainties. The Performance Committee considers the indicated deviation of the form factor to be an exception. Nevertheless, it illustrates that a different interpretation of the bulb-wave effects and the influence of laminar flow may cause a considerable deviation, even when test results from the same experiment are analysed. In our Report it has been pointed out that attempts have been made to reduce the uncertainty in the determination of the form factor by introducing more complex expressions in deriving the best fit to the model data. It is expected that these methods will eventually reduce the deviations of the form factors. However, the basic problem of the dispersion of (1 + k) remains.

Statistics of (1 + k) and C_R may well serve the purpose of ship designers, but individual experimental C_R values will always be inaccurate due to the uncertainty in (1 + k). A possible solution to the problem raised by Prof. Harvald would be to use (1 + k) values determined from an empirical formula. However, it has been shown in the past that the use of individual form factors from model experiments provides more accurated power predictions.

The Committee supports the idea of a statistical approach, but only after an improved reliability in (1 + k) is made.

The Committee Members have not experienced the problems with form factor determination encountered by *Mr. Carlier* in cases with or without rudders for single-screw ships. For twin-screw ships with centre rudders the removal of the rudder behind the skeg may introduce flow separation and vorticity, and hence a resistance increase. In this case the rudder should be installed on the model. Further investigations are needed to determine whether or not rudder(s) should be fitted to the model for resistance tests as a standard practice.

Regarding Mr. Carlier's analysis of data from more than 400 resistance tests, the Performance Committee would be most interested in learning more about these results. In this respect we refer to Section 2.6 of the Performance Committee Report of the 16th ITTC, in which it was shown how the exponent in the curve-fitting affected the form factor. However, low-speed data should be very carefully considered, particularly in ballast draught conditions and in cases of bulbs fitted at the bow.

We would like to thank *Prof. Himeno* for commenting on Equation (1.7) and offering and alternative formula for ΔC_F. We hope that this alternative forula will be further checked against existing data and that the results can be discussed at the next ITTC.

Prof. Loukakis has asserted that Lerbs' equivalent profile method is sensitive to the reference radius of this profile, especially for unconventional propellers, such as highly skewed or tip loaded. This was not the case when Prof. Lerbs originally proposed his method, because at that time the pitch distribution of the propellers was more or less constant. However, propellers designed today have considerably different pitch distribution, and Prof. Loukakis has proposed a new approach called a "lifting line profile method" for application to such propeller designs. Furthermore, he offers to carry out additional research work in this regard and invites Member Organizations of the ITTC to supply him with propeller data, including respective open water results, to make a regression analysis.

The Committee highly appreciates Prof. Loukakis' proposal and hopes that he will receive cooperation from as many institutions as possible. Then the next Committee - perhaps in cooperation with the Propeller Committee - should review and evaluate the new procedure and assess its adaptability to a performance prediction program.

Messrs. Goranov and *Lazarov* have suggested carrying out propeller open water tests at an "effective Reynolds number" (which is the product of the Reynolds number itself and the turbulence coefficient) in order to avoid scatter in the determination of the model wake and the relative rotative efficiency. This requires measurement of the turbulence level behind the model, but would, as claimed, guarantee an equivalence in propeller open water and behind conditions. In the meantime, a thorough experimental investigation is being carried out at BSHC, Varna, to verify the suggested approach. The Committee is eager to learn about the results and to assess the suitability of proposing such a method as a standard procedure.

The Performance Committee would like to thank *Professors Nakatake* and *Yamazaki* for their interesting results concerning hull-propeller-rudder interaction. It is interesting to learn that their results have confirmed the effect of the rudder as included in Equation (1.18)

Mr. Björheden's proposed new technique is very interesting from the point of view of future research. We are looking forward to experimental evidence together with a detailed discription of the new technique used. However, we feel changes in hull geometry, especially in the afterbody area, may alter the propeller-hull interaction characteristics and lead to real difficulties.

The Committee thanks *Prof. Schmiechen* for giving a further explanation of his concept, in which the problem of ship propulsion is handled in an unconventional way. Particulary appreciated is the clarification of his most recent findings that were presented at the ONR Symposium in Hamburg. The Performance Committee suggests that the first step should be to gain practical experience with this method. Specifically, the reliability of extrapolating the results of overload tests to the zero thrust condition should be investigated carefully. A promising feature of this nontraditional approach is that it can be applied to a full-scale ship as well.

The Committee agrees with *Mr. Abe* that a document explaining the 1978 prediction method would be useful, particularly to ship-owners and builders. In preparing such a document the opportunity to clarify and remove uncertainties should be taken.

With reference to one of the specific points mentioned by Mr. Abe, namely that of correcting a prediction based on the use of model stock propellers, we are of the opinion that the next Committee should investigate this problem. In particular, research should be conducted to determine the extent to which stock propeller geometry may deviate from the final design propeller geometry without introducing significant prediction errors in the model propulsion tests.

With regards to the second half of Mr. Abe's discussion, there seems to be a misunderstanding. The Committee stresses that the comparison between the tank DHP results from different propulsors as advocated is intended to be no more than an <u>initial</u> assessment. Final judgement will be possible only after scale effect factors have been introduced to the final analysis. These initial results should always be included in test reports for reference purposes. This is in line with Dr. Edstrand's statement during the Opening Ceremony, namely that customers expect to get the same answer to the same technical problem from different Institutions. All scale effects factors should be treated separately and in such a manner that readers are able to understand why and how those respective corrections have been made.

We would like to thank *Dr. Oosterveld* for his comments. In a configuration where a device is fitted to reduce hull resistance, it is recommended that test reports present a comparison based on the same friction deduction. In addition, a comparison can be given on the basis of an adjusted form factor including a description of the assumptions involved. Also, experience with correlation of similar arrangements should then be stated. On the second comment regarding the loading of the propulsor in the model test, the Committee agrees that loading is very important, especially for ducted propellers. In principle, the same problem applies to normal open propeller configurations, but in such cases the difference in loading is accounted for in the ITTC-1978 method by supposing that the wake and other propulsion factors are independent of the loading over a small range. In the case of ducted propellers, however, the usual propulsion factors are poorly defined or dependent on the loading. For that reason, the Committee has proposed that the effect of the loading should be measured for at least one speed. By so doing, a prediction can be made based on the condition at which the loading of the model propulsor is equal to that at full scale.

Messrs. Chen and *Qin* have established an empirical expression for the water depth-draught ratio beyond which shallow water effects appear negligible. The Committee values this work and encourages such research by Member Organizations for future evaluation, both for correcting speed trial results and also for predicting resistance increase due to shallow water effects. However, we feel that the suggested empirical boundary curve, indicating vanishing shallow water effects, may require further validation.

The delegates from Shanghai Ship and Shipping Research Institute (*Mr. Chen*) and MARIC, Shanghai (*Mr. Wang*) have both remarked on the inferior results that can arise when predicting performance of ships in ballast draughts. Not only this Committee, but previous ones also, have recognized this problem. There are several reasons that may be put forward. Flow separation at light ballast draughts can sometimes occur, especially in full forms, and profoundly affect the prediction results. It is also found that accurate determination of the form factor becomes difficult when a bulbous bow is fitted to the hull. In cases where trials are conducted at the ballast draught only, as has been mentioned by Mr. Wang, it is clear that difficulties can arise in the correlation. Future Committees should try to extend the applicability of the 1978 method to include light ballast conditions. However, such extension is possible only if additional trial data are available for such ships. The Committee is thus grateful to the Shanghai Ship and Shipping Research Institute for placing on record such data for twelve sister ships. It is encouraging to note that apart from the C_P value in type G, all of the correlation coefficients obtained are close to current experience. The delegates from SSRI also question the value of ΔC_F obtained by the 1978 method for longer ships. As is made clear in our Report, the Committee does have reservations over the determination of ΔC_F for all ships and hopes that the results of recent work will soon lead to a better understanding. However, we would point out that the use of a length criterion alone in assessing ΔC_F is an oversimplification. All aspects of the problem should be considered.

The Committee appreciates the comment by *Dr. Lackenby* with regard to a guide to measured mile trials. Our remark on the length of approach run was intended primarily as guidance to good trial practice but we are grateful to Dr. Lackenby for reminding us of his work.

Mr. Crago suggests abandoning the use of form factors until a universal method of determining them is devised. He refers to our nine objections which may confuse shipbuilders & shipowners. The Committee stresses again that there are definite advantages in using the form factor approach which lead to greater accuracy in the prediction, as has been repeatedly reported by past Performance Committees. Without form factor a three-dimensional extrapolation with regard to the viscous and potential components of resitance might not be possible. Further, the Committee refers to some experiments on full ship models tested in low and high water temperatures, where the deviations experienced were made to collapse satisfactorily only after use of the form factor method (Ref [62]). Therefore the Committee recommends the continued use of the form factor since no better alternative exists today. Nevertheless attempts should be made to develop a more rational and practical solution which hopefully will take account of the effects of the different components of resistance.

Mr. Suhrbier's comments are appreciated. Concerning the biological effects on resistance test result, the Committee thinks that such phenomena could exist and takes the opportunity to call this to the attention of Member Organizations.

Mr. Suhrbier's question on rudders relates to that of Mr Carlier, and, because we feel this is important, a modification to one of the Recommendations to the next Committee has been made to include the need to examine the effect of the presence of rudder or rudders in the resistance tests. With regard to Mr. Suhrbier's reference to high speed craft the Committee is interested in cooperating with the HSMV Committee in examining the problems peculiar to these types of vessel.

Prof. Abkowitz has presented interesting information related to hull roughness and full scale ship resistance. In particular the method of determining ship resistance by system identification techniques needs further investigation, and the Committee looks forward to the validation of results that may be forthcoming.

Dr. Goranov's comments on the direct calculation of time and volume mean thrust and torque of the propeller operating in non-uniform wake field should be appreciated. The Committee agrees with the discusser in that a more rational approach should be made on effective wake, and hopes that this subject should be studied by cooperation of the next Powering Performance and Propulsor Committees.

All these three years we attempted to contact many of the Member Organisations to carry on our work, either by writing correspondence or visiting interview. All the time we were received warmly with the spirit of cooperation although the individual views did not always agree with ours. Indeed the quality of our Report, we believe, has been improved greatly by such informal communication.

We think this spirit of mutual cooperation forms the basis of ITTC activities, and we must stress this now all the more because a strong concern has been expressed on the future of ITTC.

Thank you very much indeed.