

The Powering Performance Committee

Committee Chair: Prof. K. Nakatake

Session Chair: Mr. D. C. Murdey

I. DISCUSSIONS

M. Abe
Akishima Laboratories
(Mitsui Zosen), Inc., Japan

APPENDAGE DRAG EXTRAPOLATION METHODS CONCLUDED IN PPC REPORT

I appreciate that the Committee has continuously studied appendage drag extrapolation methods. I have a somewhat different view on this extrapolation method from some of the Committee's conclusions.

At the last ITTC, I pointed out that the form factor concept is not necessarily promising when appendages are shaped as stream lined, namely faired to hull. The reason is that the increase of ship hull resistance due to appendage drag depends on the wetted surface area by appendage fitting (ΔS_{APP}), and the local Reynolds number corresponding to ΔS_{APP} . With respect to this point, in equation (1) in the Committee Report, p.260, the additional drag for appendage seems to be contained in a sense of apparent increase of ship hull form factor in the expression of K_{APS} defined by total ship frictional resistance coefficient (C_{FS}). It may be mentioned from this view that equation (1) does not show an exact extrapolation method of appendage drag physically.

The point is how to decide on a realistic appendage drag extrapolation method. I agree with the Committee's opinion that it is difficult to run the geosim test for every appended ship, and even if it is possible, form factor scale

effect will remain unknown in predicting the appendage drag by equation (1).

From this point of view, I would like to make the same proposal offered at the 19th ITTC, that both bare hull and appended model tests should be conducted and every appendage extrapolation line should be scientifically decided. Then, ITTC could accumulate such data to utilize not only for statistical process, but also to assort in the type, shape and extent of appendages so as to establish the standard appendage drag extrapolation guide.

R. Rocchi

A FORM FACTOR SEMI-EMPIRICAL PREDICTION METHODOLOGY BASED ON HULL BODY-LINES: ITS APPLICATION TO MEDIUM AND HIGH "CB"'S SHIP HULLS AND TO STREAMLINED BODIES OF REVOLUTION

The ITTC '78 extrapolation methodology requires the determination of the $(1+k)$ values of the new ship hull-form, and tankery people know that until now it has been a quite complicated job.

The proposed method allows the evaluation of the $(1+k)$ value prior to its experimental determination. The errors of estimates resulted less than 2.5% (Rocchi, 1992a).

If the model has no separation, no laminar flow and no wave breaking, it is possible to predict its value very accurately by performing regression analysis of the archived $(1+k)$

experimental data selected "around" the model hull form by clustering of the derived sample according to some special form criteria (Rocchi, 1992b). To verify the reliability of the approach see Rocchi, 1992a and 1993. In the Figure data taken from Rocchi, 1992a and 1993 are reported. Regression equations derived in the case of absence of free surface (Rocchi, 1993) permit errors a half of those made in the case of modern tanker ships hull form of medium and high CB.

The validation of the proposed methodology allows its proposal as a useful instrument of anomalous flow conditions' detection during the experimental determination of the $(1+k)$: if the new hull form is prone to flow anomalies the predicted value will show for CT model values higher than those resulting during the low speed towing test.

The author submits his proposal to the consideration of this Committee for evaluation and criticisms.

Table 1. Percentage of Errors of $(1+k)$ Estimates

Series 58			
	1st Approx	2nd Approx	3rd Approx
max	4.8	2.2	1.2
min	- 4.9	- 2.6	- 0.9

with % $\epsilon > 2.5$: 5 cases, > 1.5 : 4, > 1.0 : 1
with % $\epsilon < 2.5$: 19 cases, < 1.5 : 20, < 0.5 : 17

values taken from Table 4.1.1 (Rocchi, 1992b)

Tankers			
	N=24 Sample A	N=30 Sample B	N=54 Sample A+B
max	2.4	2.4	5.1
min	- 1.8	- 1.8	- 4.8

Bulbous Bow CB < 0.80 w/ % ϵ 1.5: 19 cases
Cylindrical Bow CB > 0.80 w/ < 1.5: 26
Both Types CB < 0.88 CB < 0.88, w/ < 2.5: 46

values taken from (Rocchi 1992a).

References

- Rocchi, R., 1992a, "Full Ships' Form Factor: A suggested Method for its Determination from Experimental Collected Data to be Included in the ITTC '78 Extrapolation Process", NAV'92 Proceedings, Genova, Italy.
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A.F. Molland
University of Southampton
United Kingdom

I should like to comment on two areas within the Powering Performance Committee Report under the headings of rudder drag and catamaran powering:

Rudder Drag

My first point concerns rudder drag when operating downstream of a propeller which is, for example, mentioned in the report on the work of the JSRA.

We are involved in an extensive investigation of rudder-propeller interaction at the University of Southampton and, like others such as Nakatake and Kracht, have observed significant changes in rudder drag due to changes in propeller thrust loading, different rudder types and relative positions of rudder and propeller. These changes in rudder drag can be monitored, but changes in the propeller operating environment, and subsequently propeller performance also occur, which have implications for the overall powering estimate.

Efforts to understand scale effect on appendages seem to center on bossings and shaft brackets, with less attention being paid to scale effect on rudder drag. Manœuvring requirements and possible legislation have lead to an

increased interest in manoeuvring devices/different rudder types. If "correct" conclusions are to be drawn as to their effect on overall powering then the question of method of analysis and extrapolation to full scale needs to be addressed. Methods of analyzing self-propulsion tests, such as treating the propeller plus rudder as the propulsion unit and basing the analysis on the propeller plus rudder open water results, and the application of likely scale effects on rudder drag will require renewed consideration. Hopefully, also, commonality of approach will be adopted by the various test tanks. It would seem prudent for the Powering Performance Committee to include these as areas which should be monitored and/or reopened for more detailed consideration.

Catamaran Powering

My second point relates to powering estimates for catamarans and the possible use of form factors for such vessels. Some model-ship correlation work carried out at Southampton (unpublished) indicates that scaling methods using $k=0$ can lead to significant overestimates of full scale power. We understand this phenomenon has been found elsewhere.

The catamaran types to which we are referring invariably have transom sterns which are running wet with a confused aft end flow at lower speed and which will be running dry at higher speeds. The use of a form factor defined in the usual way is perhaps questionable. However, at Southampton we are investigating the components of catamaran resistance and our findings indicate that viscous form effect is present. This is due in part to viscous interaction between the hulls as well as to the form effect of the individual hulls. This has been determined and broadly validated by measurements of wave pattern and total viscous resistance. (Our work was mentioned at the 19th ITTC and is referenced in the current PP and HSMV Reports under Insel and Molland 1991). We are currently varying demi-hull breadth/draught ratio in a systematic way for our catamaran model series which we hope will shed some further light on viscous interaction.

If it is assumed that some kind of allowance for form effect is to be used for catamarans then the problem arises to how it may be determined in a routine and practical manner. The PPC questionnaire to member organizations indicates that some members use a form factor k , but not

how it is determined. Confused flow aft of the transom and transom drag effectively eliminates the use of low speed tests. We have experimented with low-speed bow-down tests with transom clear, but consider that too great a change in effective hull shape has taken place.

We would advocate the use of wave pattern analysis (which can now be carried out by most tanks in a routine manner) and the derivation of C_v from $C_T - C_{wp}$. This may be based on an analysis in the higher speed range (Fn 0.7 - 1.1) which is generally of interest, and clear of any transom effects. Our use of this technique has however lead to relatively high viscous form effects or form factors (even with the monohull tests), although in the case of the catamaran this would be part justified as viscous interaction. These form factors were derived in speed areas where the transom was running dry and where insignificant wave breaking and spray were present, conditions not unusual for relatively fine round bilge hulls. A broad validation was also carried out by means of the total viscous wake measurements mentioned earlier. Some decrease in derived form factor does occur if the analysis is based on running wetted area (rather than static), but the changes are not large and the final extrapolated full scale resistance is of course the same. We do therefore still remain a little uncertain as to the full physical justification of such results, and continue to pursue a full solution.

The commercial application of higher speed semi-displacement catamarans continues to grow in popularity. If the test tank community is to maintain its credibility then some commonality of approach to the extrapolation of catamaran resistance should be adopted which, by deduction, requires a better understanding of their resistance components. It naturally follows that I would agree with the draft recommendations to Conference (Section 2.3.1) that the Powering Performance Committee should monitor experience with powering predictions for catamaran forms.

J. Holtrop
MARIN, Wageningen
The Netherlands

First of all, let me convey my congratulations to the Committee for preparing a fine report. As comments, I first wish to make a brief remark on the appendage drag problem, which

goes on to occupy various committees. The conclusion that the form factor could be a good way of treating the scale effect on the appendage drag is received as a welcome statement and it confirms findings and cumulative experience at MARIN. There is, however, one problem which should be addressed and that is how form factors of appended models can be determined with confidence from either the resistance or load variation tests without turning to low speed testing where the Reynolds' numbers of the appendages become extremely low. Therefore, I venture to suggest investigating if more stable, consistent form factors of appended models can be derived from the higher speed points by subtracting the wave resistance of the bare hull form, a condition where the form factor can be found with a reasonable accuracy.

My principal comments refer, however, to Chapter III of the Committee's Report and aim to enhance the quality of the model propulsion experiments. Where the Committee concludes that the load-variation test in many cases, (1.2.1), it is suggested to maintain the open water test, not as an indispensable means to establish various factors needed for the extrapolation of the results, but rather as an element through which the accuracy of the predictions can be improved. On the next three items the propeller open water test is thought to contribute to the accuracy and reliability:

1. Propeller scale effects (1.5 and 2.5).
2. Analysis of loading dependent flow promoters as the wake fraction, (2.2 and 3.4).
3. As an independent check of the equipment as regards the level and gradients of the thrust-torque relationship of a propulsor.

In the following paragraph only the aspect of the propeller scale effect is discussed, since the last two items speak for themselves. In earlier committees it was argued at which Reynolds' number the propeller open water test was to be carried out. The answer given was that the same extent of laminar flow should be achieved as in the self propulsion experiment because only then could reliable wake fractions be expected. No definite solution could be offered at which Reynolds' number this condition would appear, though it was recommended to adhere to a certain minimum ($2 \cdot 10^5$) to avoid the most sensitive Reynolds' number range, a

restriction which can have adverse effects in acquiring reliable wake fractions. Through the years it became apparent, however, that laminar flow occurs almost same in the propulsion experiment as in the open water tests at comparable rotation rates. On the other hand, a variation of the Reynolds' number in the open water tests over a much wider range than achievable in the propulsion test revealed that unexpected large variations in K_t and K_q can occur in some types of propellers (high-skew propellers and sections which have wide cavitation buckets). As a standard, MARIN examines the sensitivity as to Reynolds' number effects on model scale to be warned if predicted full scale rotation rates should be suspected. The observation that the found tendencies do not often follow a systematic pattern, indicates that simple rules for calculating the scale effects, both lift and drag oriented, fail to take into account the special flow behavior and the specific profile characteristics. Hence, these rules will be unsuccessful to provide a general solution to the problem despite the general bearing of conclusion 1.5 of the Report.

It is recommended to carry out the propeller open water test at that rotation rate which gives indeed the same amount of laminar flow as in the corresponding propulsion test and to extend this test by some points at the highest attainable Reynolds' number to determine the differences in K_t and K_q over the largest range possible at model scale. The propulsion test results could then easily be upgraded to the level of the highest Reynolds' number attainable in the open water test by adding the measured ΔK_t and ΔK_q to the K_t and K_q in behind condition. After having done this, the propulsion test is thought to form a sounder basis for applying propeller scale effect corrections. By this procedure the Reynolds' number range of the greatest variability is covered by the open-water experiment and subsequent scale effect corrections will be smaller and more certain.

Finally, some remarks are made on the draft recommendations: Is recommendation 2.3.2 on water jets the same as 3.6? Recommendation 2.5, to which the above comments refer, could imply the evaluation of such methods as a task for a future PPC. Is recommendation 2.6.2 the same as 3.1? Recommendation 2.7.2 suggests that the weak point of CFD is rather the availability of experiment results, than the CFD itself. As this topic concerns the scaling problems where viscous flow is concerned, I had

expected the opposite. Anyhow, can the ITTC offer the service of identifying "public" experimental data available for correlation with CFD? A call to release available experiment results by member organizations could be made. The wording of the recommendation 3.3 suggests that this is a standard to which some activities should obey. An outline of these activities as a task for a next Committee should be given as clarification. As regards recommendation 3.5, is there an overlap with 2.1.1, or is 3.5 thought to be directed towards the hull form resistance of viscous origin? There will also be the Froude number influences. How does this match to corresponding activities within the Resistance and Flow Committee?

C.W.B. Grigson
Marine Consulting, Norway

AN IMPROVED METHOD OF PREDICTING THE PERFORMANCE OF FULL SHIPS

Experimental

The towing test provides (i) k , (ii) $C_R = C_{TM} - (1+k) C_{FM}$.

The propulsion test is carried out by making a set of runs each at a different n , propeller speed, for each u , hull speed. It measures (iii) $F(n)$, (iv) $T_m(F)$, (v) $Q_m(F)$. These are the propulsion functions, F is the towing force, T_m , Q_m thrust and torque; subscripts m or lower-case symbols mean model scale. From the propulsion functions behind-propeller characteristics are derived, $k_{tb} = \phi_t(j_b, F_n)$ and $k_{qb} = \phi_q(j_b, F_n)$ in which j_b is based on hull speed and F_n is Froude's number.

Need for Behind Characteristics

From ϕ measured, construct a second kind of behind characteristic $\psi(j, F_n)$ which can be compared with the open-water propeller characteristics $f(j_0)$. Corresponding points on ϕ and ψ have the same ordinates, but the argument of ψ is $j = (1-w)j_b$ (Grigson, 1990). At model scale w relates the actual axial velocity \bar{v}_m at the screw to the hull speed. Thus

$$(1-\omega)u = \bar{v}_m \quad (\text{model}) \quad (1)$$

$$(1-\Omega)U = \bar{v} \quad (\text{ship}).$$

This is at full power, so that \bar{v}_m and \bar{v} include axial velocity. $j = \bar{v}_m / nd$ and $j = \bar{v} / ND$ are advance numbers in the actual non-uniform flow field at the sterns and are not the same as open-water advance numbers.

Usually ω is unknown. But from the measured curves of ϕ , construct families of curves of ψ with ω as parameter. If $\psi(j)$ should be identical with $f(j_0)$ a value of ω will be found which brings the two curves into coincidence.

Unfortunately, if $C_B > 0.7$ - most merchant ships - ψ_t and ψ_q are not the same as f_t and f_q . This is particularly true of the behind torque characteristic.

Proper Model Test Conditions

The model thrust coefficient ought to be the same as that of the ship:

$$C_{th} = (8/\pi) k_{tb} / j^2 = (8/\pi) K_{TB} / J^2 = C_{TH} \quad (2)$$

where j and J apply in the actual velocity fields of the sterns. Let ratio

$$(1-\Omega)/(1-\omega) = G, \quad (3)$$

then C_{th} will equal C_{TH} provided the towing force is given the value

$$F_* = \frac{1}{2} \rho_m u^2 (C_{TM} - C_{TS} / G^2). \quad (4)$$

Corresponding Full Scale Quantities

When the towing force takes its proper value, the measured model propeller speed and thrust are

$$n_* = n(F_*)$$

$$T_{m*} = T_m(F_*).$$

Provided $F = F_*$, then at ship scale

$$N_* = n_* G_* / \sqrt{\lambda} \quad (5)$$

$$T_* = T_m(F_*) \cdot \rho / \rho_m \cdot G_*^2 \lambda^3 \quad (6)$$

l is the geometrical scale factor.

The thrust deduction fraction of the ship is

$$t_* = 1 - \frac{1}{2} \rho S U^2 C_{TS} / T(F_*) \quad (7)$$

Further, and always provided $F = F_*$,

$$J_{B*} = j_{b*} / G_* \quad (8)$$

From (1) and (2) the proper test condition leads to

$$K_{TB*} / J_{B*}^2 = G_*^2 (k_{tb*} / j_{b*}^2) \quad (9)$$

The equation of steady motion of the ship is

$$K_{TB} / J_{B*}^2 = C_{TS} S / (2D^2(1-t)) \quad (10)$$

All the quantities on the right are determined. From (9), express the left side in terms of model quantities, known from measurement.

$$k_{tb*} / j_{b*}^2 = C_{TS} S / (2G_*^2 D^2(1-t_*)) \quad (11)$$

On solving (11), j_{b*} is found. This will lead to the full scale behind torque coefficient for the measured function ϕ_q of the model at the advance no. j_{b*} :

$$K_{QB*} = k_{qb}(j_{b*}) - \Delta k_q \quad (12)$$

$$P_D = 2\pi N_* Q_*$$

Quantities denoted by asterisks are valid only in the proper test conditions.

Thrust in Behind

Thrust in behind operation is

$$T = \rho N^2 D^4 \psi(J, F_n, \sigma, R_n) \quad (13)$$

Here the dominant group is $\bar{\psi} / ND$. Of the others, Froude number is made the same on the model as on the ship. The effect of s should be small, or tests can be made in a model basin at scaled pressure so that s is the same at both scales. The R_n group represents the scale effect of blade friction on thrust, also small (Δk_f is about 1% of the rating k_{tb}). consequently F_n , s , and R_n may be dropped in (13) in the special case of model/ship comparison in the propulsion test. Thus in this special case

$$T / \rho N^2 D^4 = \psi(J) \quad \text{ship} \quad (14)$$

$$T_m / \rho_m n^2 d^4 = \psi(j) \quad \text{model} \quad (15)$$

Set these two equations equal by requiring that $j = J$.

Then the thrust coefficient at model scale must equal that at full scale. Moreover equations (4) - (6) and (8) all follow; see Grigson, 1990.

The Ratio G

Provided $F = F_*$, a good approximation to G is

$$G_* \approx (1-W)/(1-w) \quad (16)$$

where W and w are effective, traditional wake fractions found in the usual way from the open-water characteristics. In the present method the values of W and w are calculated at each scale from Holtrop's regression formula (Holtrop, 1984).

Other Scale Effects

The scale effect on behind torque coefficient is important, and the formula of the ITTC 1978 method is used to calculate it. The drag coefficient of the model blades must be based on a smooth turbulent friction equation; that of the blades at full scale is dealt with as in Grigson (1990).

Scale effects on form factor, on t and on the lift coefficient of propeller blades, are all assumed to be negligible.

Test Cases

31 propulsion tests were available for eight different designs, of which some particulars are given in Table 1; the after body lines of these vessels are shown in Grigson (1990). They have a wide range of shapes as implied by ratios of B/T from 2.6 up to 5.7.

The model tests were not at all designed to suit the method but for quite other purposes. However each test composed at least four sets of measurements of n , F , T_m and Q_m spread over a sufficient range of n to lead to quite good estimates of the propulsion functions. Good trials data were available for all the designs except M (which suffered from weather); and

could be put in the form of power laws, $P_D = aU^b$ and $P_D = cN^3$. From which observations corresponding with the speeds of the model tests were calculated.

In determining $F(n)$, $T_m(F)$, $Q_m(F)$, Prohaska's line, the behind characteristics ϕq and ϕt , and the power laws, analytical fits to the data were made by least-square error methods.

A correct prediction of N_* requires accurate estimation of C_{TS} . To this end, a friction law claiming improved accuracy was employed (Grigson, 1992). Effects of roughness were calculated as in (Grigson, 1990), with hull parameters h and m , blade parameters h_p and ℓ_p , where ℓ_p was assumed twice h_p . Values of the parameters and of the corrections for blade roughness, $\Delta k_q/K_{QB}$, $\Delta k_t/k_{tb}$ at service speeds

are given in Table 2. K_{QB} and k_{tb} are the operating values of the coefficients.

Table 1. Some particulars of the designs tested.

Hull/Screw	C_B	model ℓ_{wl} , m.	B/T	λ
A/A	.883	12.3	2.59	27.4
C/C	.810	6.61	2.63	47.5
D/D	.831	12.3	2.69	29.2
D/E	as D/D			
G/A	.805	12.0	3.99	27.4
I/C	.772	6.48	5.48	47.5
J/E	.781	11.7	5.71	29.2
M/M	.677	6.25	3.80	19.6

Table 2. Roughness parameters and corrections for scale for blade friction.

Hull/Screw	h , mm	m	h_p , μ	$\Delta k_q/K_{QB}$, %	$\Delta k_t/k_{tb}$, %	Speed, kn.
A/A	0.1	40	5	8.15	0.74	16
C/C	0.15	12	5	11.3	1.02	16
D/D	0.12	8	5	11.2	0.97	16
D/E	0.12	8	20	11.3	0.95	16
G/A	0.1	40	5	8.50	0.75	17
I/C	0.15	12	5	11.7	1.06	17
J/E	0.12	8	20	9.30	0.79	18
M/M	0.15	14	45	4.32	0.45	17

Table 3 summarizes the statistics of the prediction method. C_N , C_p and C_Q are respectively the ratios of the observed to the calculated values of N , P_D and K_{QB} .

Table 3. Statistics of C_N , C_p and C_Q

	mean	standard deviation
C_N	1.003	0.0098
C_p	0.988	0.0338
C_Q	0.980	0.0302

The sample size in this work of course is rather small. It ought certainly to be enlarged. The author, in complete commercial confidence, would gladly analyze data for any institution interested. This data could today be considerably improved compared with what

was available earlier, because the tests would be designed to suit the method. However from the point of view of statistical behavior, a sample size of 31 is too large for the results to be due to chance. The mean values here are close to unity; the standard deviations are half those of previous methods.

Acknowledgments

The model data were measured by the Danish Tank, NSMB, Ede, the Ship Division of NPL and the Swedish Tank.

References

- Grigson, J., Ship Res., 34, pp 262-82, 1990.
- Holtrop, Int. Shipbuilding Prog. 31, 1984.
- Grigson, RINA Paper W5, 1992.

Table 4 gives each result. R_m is the drag of the model hull in the configuration of the propulsion test.

Table 4

Case	Hull/ Screw	Scaled pressure?	Froude number	G_*	F_*/R_m	C_N	C_P	C_Q
1	A/A	Yes	.152	1.104	.529	.9948	.9768	.9923
2		No	.152	1.104	.537	.9836	.9353	.9828
3		Yes	.143	1.106	.535	.9934	.9810	1.001
4		No	.143	1.106	.537	1.003	1.002	.9922
5		Yes	.134	1.109	.543	1.016	1.069	1.018
6		No	.134	1.106	.539	1.009	1.038	1.009
7	C/C	No	.149	1.110	.530	1.002	.9578	.9523
8		No	.139	1.113	.539	1.004	.9624	.9513
9		No	.128	1.116	.548	1.010	.9901	.9599
10		No	.120	1.118	.554	1.012	.9781	.9425
11		No	.111	1.121	.561	1.016	.9694	.9250
12	D/D	Yes	.139	1.149	.531	1.011	.9591	.9284
13		No	.139	1.149	.531	1.005	.9396	.9271
14		Yes	.131	1.153	.539	1.022	1.015	.9514
15		No	.131	1.153	.539	.9931	.9125	.9317
16		Yes	.122	1.157	.546	1.001	.9333	.9308
17	D/E	Yes	.139	1.147	.527	.9943	.9895	1.007
18	G/A	Yes	.154	1.187	.567	1.013	1.034	.9939
19		No	.154	1.187	.578	.9913	.9740	1.000
20	I/C	No	.176	1.211	.572	1.010	1.001	.9693
21		No	.169	1.214	.581	1.024	1.051	.9805
22		No	.161	1.218	.589	1.009	1.007	.9801
23		No	.156	1.222	.596	1.001	.9888	1.003
24		No	.146	1.226	.603	.9971	.9850	.9935
25		No	.131	1.235	.618	.9900	.9723	1.002
26	J/E	Yes	.160	1.236	.567	.9939	.9995	1.018
27	M/M	No	.252	1.068	.363	.9963	.9936	1.005
28		No	.229	1.071	.376	.9950	.9988	1.104
29		No	.208	1.073	.385	.9951	.9918	1.007
30		No	.186	1.075	.402	.9961	.9919	1.004
31		No	.165	1.078	.410	1.006	1.024	1.006

S. Cordiez and F.X. Dumez

RESISTANCE COMPONENTS OF DISPLACEMENT AND SEMI- DISPLACEMENT HULL FORMS

Introduction

The ITTC cooperative resistance tests performed by several Japanese towing tanks [Tanaka et al. 1990], and full scale powering estimates [19th ITTC PPC report] on a semi-displacement, Vee-shaped hull showed that important problems associated with the power prediction of this type of hull form remain. In particular, there is a large uncertainty in the value of the form factor for this type of hull form for which standard experimental methods are not applicable. In an attempt to answer some of these questions, the Bassin d'Essais des Carènes began a research program on the

determination of the resistance components. The two hull forms chosen were the Series 60 ($C_B=0.6$) and the semi-displacement passenger ferry *OLIVE* used by Tanaka et al. 1990.

Experimental Program

Following the classical approach [Baba, 1969], a momentum balance was conducted on a control volume far from the model to identify the wave resistance and the viscous resistance. To this end, detailed information was obtained on the velocities, the static pressures, and the wave heights (longitudinal cuts) in the wake of the models towed in our No. 3 tank (13m x 5.5m x 260m). This contribution presents the results on the determination of the form factors and resistance components of these two hull forms. The models used and the test performed are described in table 1 below.

Table 1: Characteristics of the models tested and of the test program

Model	Scale	L _{pp} (m)	H _T /L _{pp}	Resistance, Wave pattern	Wake Surveys
Series 60	1/25	4.877	1.13	0 < Fn < 0.35	LDV Fn=0.25 Pitot Fn=0.25, 0.35
Semi- displacement*	1/7.16	3.24	1.54	0.5 < Fn < 1.0	LDV Fn=0.8 Pitot Fn=0.6, 0.8, 1.0

*Full load, without appendages, towed parallel to baseline at rest
Turbulence stimulation on both models using roughness strip

Viscous Resistance

The viscous resistance of the two models was measured by a 3D LDV system and by a rake of 13 pitot tubes. Using the static pressure data, the measured velocities (by LDV or pitot tubes) were corrected for the potential flow component present in the wake due to the wave field to yield the viscous velocity component. The product of measured and viscous velocity components integrated over the wake area gives an estimate of viscous resistance. Comparison of the viscous resistance and the friction resistance based on the ITTC friction line and the measured wetted surface yields a form factor which can be calculated at each Fn. For the Series 60 model a value of form factor was determined based on low speed tests using Prohaska's method.

The results shown in Figures 1 and 2 for the form factors of each model show a slight dependence on Froude number for the Series 60 model as reported by previous authors. This dependence is significantly increased for the semi-displacement hull form (Fig. 3) where the form factor is close to 1.0 below Fn=0.9 and smaller above this value. The data obtained is significantly lower than the values presented in Tanaka et al. 1990 using Hughes' method which is based on the analysis of numerous tests of geosim models. However, the results obtained are in agreement with the value which yields the best model to full scale correlation in the 19th ITTC PPC report. These values are also close to 1.0 which is recommended by the ITTC for this type of hull form.

Wave Resistance and Resistance Components

The wave pattern resistance was calculated for each model over a large range of F_n based on 4 longitudinal cuts [Eggers et al. 1967]. The results were added to the viscous resistance found by viscous wake analysis and are presented versus Froude number in Figures 3 and 4. In the case of the Series 60 model for which there is no wave breaking (Fig. 3) there is an excellent correlation between the sum of these components and the load cell measurements which demonstrates the validity of the procedure used. In the case of the semi-displacement hull form (Fig. 4), a large gap exists between the sum of viscous and wave pattern measurements and the total resistance. This component is presumably attributable to spray and wave breaking resistance which will be estimated in the near future based on hull pressure and spray wake velocity measurements.

As direct measurement methods gain in accuracy and speed, they will become a useful tool in the evaluation of new types of hull forms for which little data is available.

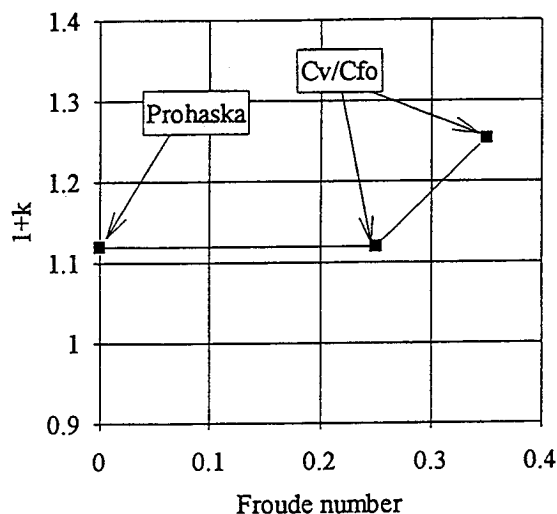


Figure 1. Form factor of Series 60 model

Acknowledgment

We wish to thank Dr. Tanaka of SRC for providing us with the lines and experimental data on the *OLIVE* semi-displacement hull-form. This work was funded by the DRET (Direction des Recherches, Etudes et Techniques).

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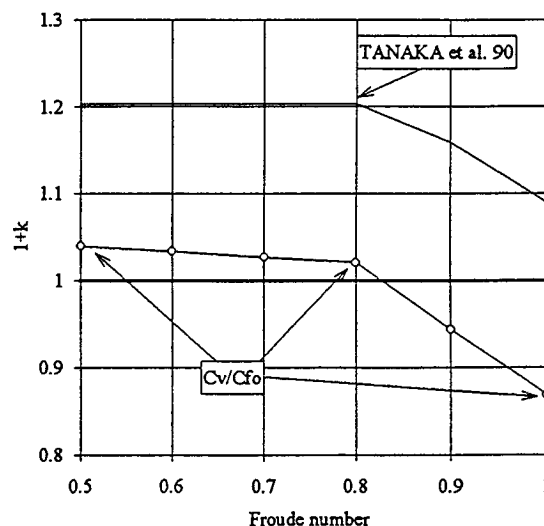


Figure 2. Form factor for the semi-displacement model

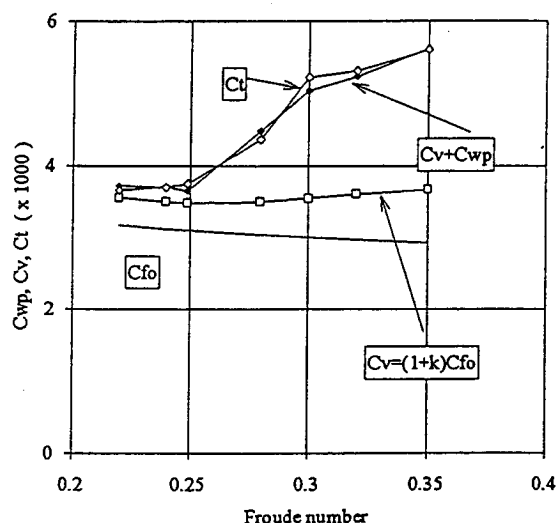


Figure 3. Resistance components of Series 60 model

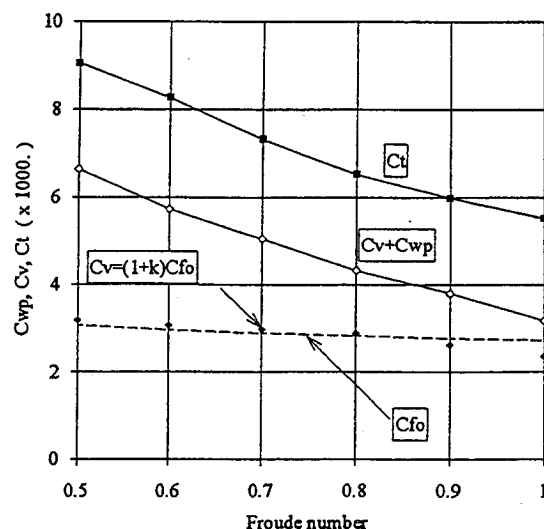


Figure 4. Resistance components of semi-displacement model

M.B. Wilson
David Taylor Model Basin
Bethesda, Maryland, U.S.A.

There is an interesting interaction resistance problem that arises with two-strut per side configurations of SWATH ships. When the struts are arranged in tandem, the aft strut operates directly behind and therefore in the wake of the forward strut. The question I have in mind is how can the viscous part of the drag on the aft strut be properly estimated. At least three aspects of interaction important to resistance are present:

- The aft strut could be thought to be running in a slower onset flow velocity than free stream because of the velocity deficit behind the forward strut. From this point of view, the Reynolds number would be effectively smaller, and the estimated friction coefficient higher. What would be the net change in viscous drag compared with the isolated strut case?
- The surface wave system from the forward strut provides elevation changes from the calm water line on the aft strut and thus alters the wetted surface on the aft strut compared with the isolated strut case.
- Turbulence in the wake of the forward strut could cause the boundary layer growth on the aft strut to reach a more mature turbulent

state earlier along the body length, and thus could alter the net viscous drag on that strut. Could anyone on the Powering Performance Committee comment on their idea of how to estimate the net effect of all these influences on the friction drag of the tandem strut arrangement?

S. Miranda And J. Szantyr
Università degli Studi di Napoli
Federico II, Napoli, Italia
Institute of Fluid Flow Machinery
Polish Academy of Sciences
Gdansk, Poland

LIFT SCALE EFFECT CORRECTIONS FOR PROPELLER OPEN WATER CHARACTERISTICS

Nomenclature

c	-	propeller blade section chord length
$c_{.75}$	-	propeller blade section chord length at radius 0.75
C_L	-	lift coefficient
C_D	-	drag coefficient
k_p	-	blade surface roughness
K_T	-	thrust coefficient
K_Q	-	torque coefficient
ΔK_T	-	drag based correction for thrust coefficient
ΔK_Q	-	drag based correction for torque coefficient
M	-	subscript for model conditions
$P_{.75}$	-	blade pitch at radius 0.75
R_N	-	blade section Reynolds number
S	-	subscript for ship conditions
t	-	maximum thickness of the blade section
Z	-	number of blades
x	-	radius fraction
α	-	blade section angle of attack
α_0	-	blade section angle of zero lift
α_{x_0}	-	attack angle of blade section at radius x_0
β	-	advance angle
β_i	-	hydrodynamic pitch angle
ϕ_{x_0}	-	geometric pitch angle at radius x_0
η_0	-	screw efficiency
λ	-	advance ratio

Introduction

The purpose of this contribution is to demonstrate the effects of application of the lift-based propeller scale effect corrections to the propeller open water characteristics. Two different approaches are described: one employing a lifting line procedure for calculation of propeller open water characteristics and another making use of the equivalent propeller blade section characteristics.

Theoretical Considerations

The standard ITTC '78 procedure employs the following drag-based propeller scale effect corrections, which are applied to the propeller open water characteristics measured in the model scale:

$$\Delta K_T = -0.3 \Delta C_D \frac{P_{.75} c_{.75} Z}{D} \quad (1)$$

$$\Delta K_Q = 0.25 \Delta C_D \frac{c_{.75} Z}{D} \quad (2)$$

$$\Delta C_D = C_{DM} - C_{DS} \quad (3)$$

$$C_{DM} = 2 \left[1 + 2 \left(\frac{t}{c} \right)_{.75} \right] \left[\frac{.044}{R_N^{1/6}} - \frac{5}{R_N^{2/3}} \right] \quad (4)$$

$$C_{DS} = 2 \left[1 + 2 \left(\frac{t}{c} \right)_{.75} \right] \left[1.89 + 1.62 \frac{c_{.75}}{k_p} \right]^{-2.5} \quad (5)$$

A thorough analysis of the experimental characteristics of aerofoils, for example those published in Abbott & Doenhoff (1959), shows that the scale effect on lift has a much more limited character than the scale effect on drag. In contrast to the drag coefficient, there is no continuous change of the lift coefficient with the Reynolds number. Practically, the scale effect on lift demonstrates itself only on two occasions, namely:

- in the value of the maximum lift corresponding to the beginning of stall
- in the value of the lift coefficient corresponding to the mixed laminar/turbulent flow at relatively low Reynolds numbers

Both these conditions may be present in the model tests, but their unsteady and unpredictable character makes introduction of any effective correction very difficult.

A team of Russian authors has developed formulae based on the aerofoil data from Abbott & Doenhoff (1959). These formulae, published in Bavin et al. (1983), are employed in the present analysis. According to this approach the lift coefficient for every propeller blade section is determined in the following way:

$$C_L = 2\pi\mu(\alpha + \chi\alpha_0) \quad (6)$$

$$\mu = \left(1 + 0.87 \frac{t}{c}\right) \left[1 - \exp\left(-0.0691 + 12.46 \frac{t}{c} - 1.8551 \ln R_N\right)\right] \quad (7)$$

$$x = 1.015 \left[1 + \frac{t}{c} \left(\frac{t}{c} - 0.05\right)\right] / (0.04664 \ln R_N - 0.4378)^2 \quad (8)$$

These formulae are strictly applicable for $C_L \leq 0.3$, $t/c \leq 0.1$, $R_N \geq 10^5$.

Equivalent Profile Approach

The formulae (6) - (8) have been used for calculation of α_{x_0} by solving the equation:

$$\varphi_{x_0} - \alpha_{x_0} + f(\alpha_{x_0}, R_{NX_0}) = \arctan \frac{\lambda}{\eta_0 x_0} \quad (9)$$

obtained according to the hypothesis that the local efficiency of the section at x_0 equals the total efficiency of the propeller (Lerbs 1951). The function $f(\alpha, R_N)$ is given by:

$$f(\alpha, R_N) = \frac{C_D}{C_L} \quad (10)$$

where the drag coefficient of the section has been determined by the formula (Bavin et al 1983):

$$C_D = 0.05808 \left(1 + 2.3 \frac{t}{c}\right) / R_N^{1.458} \quad (12)$$

Assuming that α_{x_0} has the same value both for model and full scale equivalent sections, it is possible to calculate by (6) C_{LM} and C_{LS} values. These values have been inserted into the following formulae for calculation of the propeller open water characteristics both for model and full scale propellers:

$$K_T = m(x_0) \left(\frac{dK_T}{dx}\right)_{x_0} =$$

$$m(x_0) \frac{\pi^2}{4} \left(\frac{c}{D}\right)_{x_0} Z \lambda^2 C_{LX_0} \left[A^2 \frac{\cos(\beta_i + \gamma)}{\sin^2 \beta_i \cos \gamma} \right]_{x_0} \quad (14)$$

$$K_Q = m(x_0) \left[\frac{1}{2} x_0 \left(\frac{dK_T}{dx}\right)_{x_0} \tan(\beta_i + \gamma)_{x_0} \right] \quad (15)$$

where

$$A = 1 + \frac{\cos \beta_i \sin(\beta_i - \beta)}{\sin \beta}$$

and according to Lerbs (1951) $m_{x_0} = 0.5277$ at radius 0.75.

The mean values of the resulting differences in the values of thrust and torque coefficients are regarded as lift-based scale effect corrections to model open water characteristics.

Lifting Line Approach

In this method the formulae (6) - (8) have been inserted into a lifting line/lifting surface computer routine for calculation of the propeller open water characteristics. This routine forms a part of the computer system for design and analysis of propellers, early version of which is described in Szantyr (1987). The calculations of open water characteristics are performed both for model and full scale propellers. The resulting differences in the values of thrust and torque are regarded as lift-based scale effect corrections, which may be applied to the experimental model scale open water characteristics of propellers.

Results of Calculations

The analysis using the above described methods has been performed for six screw propeller, main particulars of which may be found in Table 1. An example of results of propeller P1 is presented in Table 2. This table contains the propeller open water characteristics in four variants:

1. as measured in towing tank
2. corrected using drag-based scale effect corrections (ITTC '78)
3. corrected using drag and lift based corrections (lifting line)
4. corrected using drag and lift based corrections (equivalent profile)

Conclusions

Introduction of the lift based scale effect corrections results in increase of both full scale propeller thrust and torque for region of high and medium propeller loading, while in some cases both these parameters may be reduced for low propeller loading. The magnitude of corrections is a function of propeller loading and of the differences between model and full scale propeller Reynolds numbers.

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Table 1. Main Parameters of Propellers

Propeller	P1	P2	P3	P4	P5	P6
Z	4	5	4	4	4	5
D (m)	5.60	7.30	3.605	4.90	3.80	9.70
A_E / A_0	0.640	0.611	0.486	0.574	0.525	0.484
$P_{0.75} / D$	0.897	0.796	0.884	0.786	0.679	0.744
R_{NM}	$0.60 \cdot 10^6$	$0.53 \cdot 10^6$	$0.41 \cdot 10^6$	$0.48 \cdot 10^6$	$0.54 \cdot 10^6$	$0.46 \cdot 10^6$
R_{NS}	$6.39 \cdot 10^7$	$3.61 \cdot 10^7$	$2.30 \cdot 10^7$	$4.38 \cdot 10^7$	$2.71 \cdot 10^7$	$3.40 \cdot 10^7$

Table 2. Open Water Characteristics of the Propeller P1

J	VARIANT 1		VARIANT 2		VARIANT 3		VARIANT 4	
	K_T	$10K_Q$	K_T	$10K_Q$	K_T	$10K_Q$	K_T	$10K_Q$
0.10	0.4188	0.5984	0.4198	0.5894	0.4326	0.5923	0.4324	0.5982
0.15	0.3960	0.5730	0.3970	0.5640	0.4093	0.5666	0.4096	0.5728
0.20	0.3723	0.5465	0.3733	0.5375	0.3873	0.5405	0.3858	0.5463
0.25	0.3479	0.5193	0.3489	0.5103	0.3624	0.5131	0.3615	0.5191
0.30	0.3234	0.4917	0.3244	0.4827	0.3369	0.4852	0.3370	0.4915
0.40	0.2749	0.4362	0.2759	0.4272	0.2875	0.4294	0.2885	0.4360
0.50	0.2279	0.3808	0.2289	0.3718	0.2399	0.3740	0.2415	0.3806
0.60	0.1823	0.3250	0.1833	0.3160	0.1938	0.3180	0.1959	0.3248
0.70	0.1370	0.2670	0.1380	0.2580	0.1473	0.2598	0.1506	0.2669
0.80	0.0892	0.2038	0.0902	0.1948	0.0974	0.1961	0.1028	0.2036

T. Loukakis
National Technical University
of Athens, Greece

The Committee's recommendations to the Conference include "CFD calculations for a standard model to investigate scaling" and "investigations of the Reynolds number dependency of form factor using experimental data." In view of the above I would like to bring to the attention of the Committee a recent journal article (1993) by Prof. Tzabiras of NTUA which addresses both problems. Two tanker models (HSVA & SSPA) have been tested numerically at low (5×10^6) and high (2×10^9) Reynolds numbers, with and without a propeller (actuator disk). The calculations ignored the free surface. Resistance (frictional and pressure) and nominal wake were computed. When the results were compared to the traditional scaling laws, it was found that Froude's method overpredicted, whereas the form factor method underpredicted. Resistance and effective wake were also computed with the propulsor in operation, allowing the examination of the scaling effects on the wake fraction and the thrust deduction. Among the conclusions of the paper are:

- The traditional extrapolation methods are questionable.
- The thrust deduction factor remains remarkably constant.
- Stern section of the U or V type, experiences opposite effects with respect to hull resistance when the hull is towed or self-propelled.

With the above in mind, I would like to suggest:

- a) that CFD is nearing the point where it could be incorporated in tankery work,
- b) that the problem to solve is the real one of ship propulsion at full scale and not the traditional one of model resistance and therefore, the Resistance and Powering Committees should work closely together or, may be merged into one committee.

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A. J. Musker
Defense Research Agency
Gosport, United Kingdom

I feel I must make a response to the remarks made by Prof. Loukakis in connection with the work described in the Committee Report. I agree that the method of extrapolating results from model to full-scale is full of questionable empiricism and that the long term objective of CFD must be to calculate at full-scale directly. In this way, no scaling laws at all will be required. At the same time, we must of course remember that our problem (or at least one of them) is to predict full-scale effective wake rather than model-scale nominal wake. I believe that, at the moment, faith in full-scale calculations is misplaced. How can they be believed when:

- a) there is a lack of full-scale validation data of known accuracy?
- b) we know that the methods fail for the simpler case of model-scale?

The question that needs to be addressed is whether the errors associated with extrapolating model-scale results (experimental or computational) to full-scale are smaller or greater than the errors associated with performing a CFD calculation directly at full-scale. Whilst the latter process is the one we should be striving to follow, I believe it is premature to claim we can do it.

S.H. Van, M.C. Kim & J.T. Lee
Korea Research Institute of Ships and
Ocean Engineering, Daejeon, Korea

SOME REMARKS ON THE POWERING PERFORMANCE PREDICTION METHOD FOR A SHIP EQUIPPED WITH A PRESWIRL STATOR-PROPELLER SYSTEM

Introduction

A preswirl stator is located in front of a propeller for the purpose of recovering the rotational kinetic energy in the propeller slipstream by generating a counter-swirl component to the propeller inflow velocity. Since the preswirl stator-propeller system is based on the clear hydrodynamic principle and has a very simple mechanism (compared to a CRP), it

might be considered as one of the most efficient and reliable energy saving propulsion systems.

On the way to developing an optimum combination of stator-propeller system for a VLCC, five stators were designed using lifting surface method and tested at the towing tank and at the cavitation tunnel of KRISO.

The first four stators have radially constant pitch angles, although each blade has a different pitch angle setting which is selected as adaptive to the ship's wake. The final stator (S5) has a radially varying pitch angle distribution for each stator blade, which is designed to be adaptive to the estimated ship's effective wake. The perspective view of the final stator-propeller system is shown in Fig. 1.

Improvement of the propulsive efficiency is clearly shown from the model tests results, especially for the final stator-propeller system. There remains some discussion though, on the powering performance prediction method for the full scale ship.

Powering Performance Prediction of a Ship with a Preswirl Stator-Propeller System

A series of model tests, consisting of resistance tests for the hull with and without stator, open-water tests for the stator-propeller system and for the propeller alone, self-propulsion tests with the measurement of the stator drag, were performed in the towing tank. Only the results with the stator 4 and 5 will be given here, because the better powering performances are obtained with them. Geometric characteristics of the stator 4 are similar to those of the stator 5, except its radial pitch angle distribution of each blade is constant. More details about the design concepts and the test results can be found in (Lee et al. 1992, Kim et al. 1992, Lee et al. 1993, Van et al. 1993).

The tank power (P_{DS}), that is directly scaled up from the measured torque and rpm in self-propulsion test using the following equation, is calculated and compared for a ship with and without stator.

$$P_{DS} = P_{dm} \lambda^{3.5} \frac{\rho M}{\rho S}$$

As shown in Table 1, the tank power with

the stator-propeller system for a 300,000 DWT VLCC is decreased remarkably compared with that of the single propeller.

Table 1 Comparison of tank powers for different combinations of propulsion system at the design speed ($V_S = 15.5$ knots)

stator	tank power (PS)	%
without	23,149	100.0
S4	22,256	96.2
S5	21,684	93.7

However, the tank power is not considered as the actual power that is needed for the full scale ship. A kind of powering performance prediction method should be applied. As shown in Tables 4 and 5, if the 1978 ITTC powering performance prediction method (Report of the Performance Committee, 15th ITTC 1978) is used without any modification to extrapolate the present model test results, the predicted values for the self propulsion factors are somewhat unreasonable, and also the power gains are too small compared to those of the tank power. It is believed that the interaction between the stator and propeller is not properly considered in the 1978 ITTC method.

It is evident that an alternative prediction method should be provided for the extrapolation of the model test results with the preswirl stator. The only references to the powering performance prediction method for full-scale ships, found by the discussors, are the papers by Takekuma (1980, 1981) whose method is based on the JTTC method.

Two alternative methods basically following the 1978 ITTC model, are considered here. In method A, a propeller together with the stator is considered as a propulsion system, so the propeller open-water test results with the stator are used as the open-water characteristics of the propulsor. In the other method (method B), all the procedures are the same as the 1978 ITTC method, except that the wake fraction is scaled up using the following formula.

$$\omega_s = (t_{MO} + 0.04)$$

$$+ (\omega_{MO} - t_{MO} - 0.04) \frac{C_{FS} + C_A}{C_{FM}} + (\omega_{MS} - \omega_{MO})$$

where

w_s : ship wake
 w_{MO} : model wake without stator
 w_{MS} : model wake with stator
 t_{MO} : thrust deduction factor without stator

When the stator is located upstream of the propeller, the angle of attack for the propeller blade section is increased because the tangential velocity component is generated by a stator and mainly considered as the potential part, so the quantity $(\omega_{MS} - \omega_{MO})$ is assumed to be the same for model and ship (Van et al. 1993). The self-propulsion factors and the power and rpm for a full-scale ship without stator predicted using the 1978 ITTC method are shown in Tables 2 and 3, and those for a full scale ship by using the above methods are shown in Tables 4 and 5, respectively. Compared to results of the tank power, almost the same amount (in the sense of the percentage to power without stator) of decrease in DHP is predicted by the proposed methods.

Table 2 Self-propulsion Factors for the VLCC without Stator Predicted Using the ITTC Method

Method	t	ω_M	ω_s	η_R	η_o	η_D
ITTC	.217	.434	.320	.983	.622	.704

Table 3 DHP and RPM for the VLCC without Stator predicted using the ITTC Method

Method	EHP (PS)	RPM	DHP (PS)	%
ITTC	18,058	75.92	25,635	100

Table 4 Comparison of Self-Propulsion Factors for the VLCC Predicted Using the Different Prediction Methods

($V_S = 15.5$ knots, with the stator 4)

Method	t	ω_M	ω_s	η_R	η_o	η_D
ITTC	.285	.581	.416	1.003	.566	.696
A	.247	.457	.348	1.042	.612	.736
B	.275	.582	.468	1.007	.540	.741

($V_S = 15.5$ knots, with the stator 5)

Method	t	ω_M	ω_s	η_R	η_o	η_D
ITTC	.271	.594	.412	1.018	.572	.722
A	.236	.460	.342	1.050	.616	.750
B	.272	.595	.481	1.018	.534	.762

Table 5 Comparison of DHP and RPM for the VLCC Predicted Using the Different Prediction Methods

($V_S = 15.5$ knots, with the stator 4)

Method	EHP (PS)	RPM	DHP (PS)	%
ITTC	18,222	74.35	26,189	100.2
A	18,058	71.55	24,546	95.8
B	18,222	71.97	24,579	95.9

($V_S = 15.5$ knots, with the stator 5)

Method	EHP (PS)	RPM	DHP (PS)	%
ITTC	18,222	74.02	25,250	98.5
A	18,058	71.22	24,082	93.9
B	18,222	71.34	23,908	93.3

Model Tests

For the application of method A, open-water test for stator-propeller systems are necessary. The dynamometer and accompanying shaft system, designed for a CRP (Contra-Rotating Propeller), is used to fit the stator upstream of a propeller. During the open-water test and the self-propulsion test, the thrusts of propeller and stator are measured simultaneously and their sum is used as a total thrust of this propulsion system. In the case of applying method B, an open-water test for the propeller alone is necessary, but the resistance and self-propulsion tests without stator should be performed in addition to those with a stator. The measurement of stator thrust is not necessary in this case and no special dynamometer and shaft arrangement is needed.

Discussion

Although the prediction methods proposed here are not verified and completed yet, those methods enable more reasonable prediction than the 1978 ITTC method for a ship equipped with a stator-propeller system. Based on the results obtained here, more than 6.0% saving in delivered power is expected if the properly designed preswirl stator-propeller system is installed. Especially, as far as the model scale values are concerned, the optimistic effect of stator is confirmed clearly.

The stator can be thought of as a promising energy saving device because the shaft system is relatively simple compared to that of CRP. KRISO would continue more extensive studies to understand the flow field around a stator-propeller system and to set up a more reliable prediction method.

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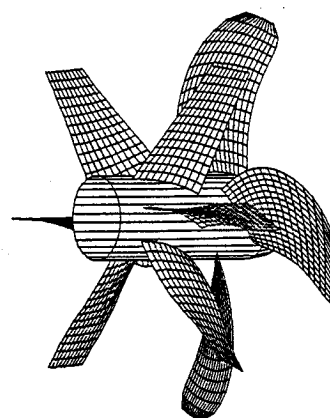


Figure 1. Perspective view of a preswirl stator-propeller system (S5 +P1).

II. WRITTEN DISCUSSION

J.W. English

A NOTE ON RUDDER FORCE A ZERO HELM

1. Notation

A logical abbreviated notation for the presence or non presence of a rudder in various model states is introduced to economize on words, i.e.,

rudder _O	-	rudder in open water (no propeller)
rudder _H	-	rudder behind model hull (no propeller)
rudder _{OP}	-	rudder behind propeller in open water
rudder _{HP}	-	rudder behind propeller and hull

A similar notation applies in describing the presence or non presence of the rudder on the performance of the operating propeller.

propeller _O	-	propeller in open water (no rudder)
propeller _B	-	propeller behind hull (no rudder)

noting that,

$$\begin{aligned} \text{rudder}_{OP} &= \text{propeller}_{OR} \\ \text{rudder}_{HP} &= \text{propeller}_B \end{aligned}$$

The corresponding rudder force F , propeller thrust T and propeller torque Q attract the same suffixes to describe their magnitudes, for example,

T_{OR} = propeller thrust in open water with rudder present, similarly for T_O , T_B , T_{BR} and the propeller torques, Q_O , Q_B , Q_{OR} and Q_{BR} .

For the rudder force F we have,

F_O	-	rudder force in open water (no propeller)
F_B	-	rudder force behind hull (no propeller)
F_{OP}	-	rudder force in open water (with propeller)
F_{BP}	-	rudder force behind hull (with propeller)

The propeller thrust and torque increments due to the presence of a rudder are given by,

$$\Delta T_{OR} = T_{OR} - T_O$$

$$\Delta T_{BR} = T_{BR} - T_B$$

and similarly for ΔQ_{OR} and ΔQ_{BR}

This note is intended to refer mainly to large single screw displacement ships where the rudder force is barely distinguishable in the total resistance.

2. Introduction

In this note the commonly used term 'rudder drag' and what is meant by it is discussed. It is recognized, however, that the nett fore and aft force on a rudder at zero helm in the slipstream of an operating propeller can be a thrust, albeit usually small and only at low $J = V/nD$. Because of this the term rudder drag is more appropriately called rudder force, using the sign to discriminate a thrust (+ve) from a drag (-ve), although in free running ship service conditions the force is invariably a drag.

3. Fundamentals of the Problem

Consider first a rudder *not* in a propeller slip stream and at zero incidence, or rudder_H in abbreviated notation. In this case drag comes from the tangential friction force, the form or pressure drag due to the normal pressures acting on the rudder surface and the induced drag (also a pressure force), which should be zero for a symmetrical rudder and flow).

For the same rudder in a propeller slipstream (rudder_{OP} and rudder_{HP}) the flow over it is quite different. Firstly, this flow, which is dominated by the operating propeller in most cases, has a cyclical component at blade passage frequency $n.z$, where n is the shaft rotational speed (rev/s) and z is the number of propeller blades. This effect diminishes with increasing z and it is much lower for ducted propellers as opposed to conventional unducted ones. The cyclical component also depends on the pitch-diameter ratio of the propeller and the loading. In this state the friction and pressure forces on the rudder are different to those without the propeller. They are obviously cyclical and because the flow is inclined to the symmetrical rudder, as a result of the slipstream swirl,

the pressure force on the rudder can be thrust instead of a drag. Additionally, there could be a small induced drag on the rudder in the rudder_{HP} case because of the asymmetrical nature of the flow into the screw behind a hull and in the propeller slipstream.

The rudder_{HP} force (drag or thrust) can be measured directly but it would be extremely difficult with routine measurements to separate this force into its tangential frictional and normal pressure components. On the assumption that this force can be separated into these components, they could in principle be found by integrating a distribution of local measurements, but this would not be attempted in such complicated and unsteady conditions. Furthermore, even if the axial force due to normal pressures were known it would not be possible to split this into drag and thrust components. As a consequence of this only comparisons of the nett force on the rudder_{HP} can be made between alternative rudders under a stipulated set of externally imposed conditions that make such a comparison fair. It is concluded, therefore, that it is not practicable to attempt to evaluate the components of rudder_{HP} force by experimental methods. As a corollary, scale corrections to the rudder_{HP} force cannot be made, because the unknown force components scale according to different laws.

The assumption sometimes made, that the rudder_H force, when a drag, can be taken as a basic drag which can be deducted from the rudder_{PH} force to get the force on the rudder due to the propulsor seems crude, not tenable in a rigorous sense and of doubtful validity as an engineering approximation.

IT SEEMS FAIR TO CONCLUDE, THEREFORE, THAT RELIABLE PREDICTIONS OF FULL SCALE CONVENTIONAL RUDDER FORCE CANNOT BE MADE AS A RESULT OF TRADITIONAL SHIP MODEL RESISTANCE AND PROPULSION EXPERIMENTS.

This is not such a radical conclusion as it may first appear when it is remembered that bilge keels, for instance, are not normally included in a propulsion model fit-up simply because the resistance they create on the model is excessive.

All the above remarks refer to what may be described as conventional rudders of the spade

or Mariner type, but when specialized rudders are installed for superior manoeuvrability the scaling problem can be compounded. This is likely to be the case with the proprietary 'Schilling Rudder' for example, where top and bottom endplates are used to enhance rudder manoeuvring performance. It is reasonable to assume in this case that a payment must be made for carrying these plates and a question is, how can this be quantified? It appears that the technique of conventional resistance and propulsion experiments leads to penalties which are claimed to be excessive and the adequacy of this procedure for evaluating powering performance is questioned.

As a possible alternative approach one could contemplate theoretically calculating the rudder_{PH} force and then scale correct the component parts. Attempts at making such calculations have been made in the past for VTOL aircraft with propellers on wings and Isay (1965), and probably others, have tried to theoretically calculate the interaction between marine propellers and rudders, however the simplifying assumptions that are made to obtain numerical results detract from this approach.

Whilst this note highlights the problem of scaling rudder force it does not produce a solution to it. However, by highlighting it and hopefully engaging the interest of the International Towing Tank community a step forward in improving the means of dealing with it may be forthcoming.

4. The Rudder In A Ship Model Propulsion Test

4.1 Background. Despite the existence of the ITTC and its past attempts to standardize ship model procedures so that customers of different Tanks can cross reference results more readily, the earlier situation has hardly changed. Exceptions to this are, the introduction of the ITTC 1957 model-ship correlation line and its acceptance by many but not all Tanks, a so-called 'scientific' propulsion analysis method, the 1978 ITTC Performance Prediction Method which incorporates a form factor approach, scaling of the wake fraction and the model propeller efficiency. However, apart from one European tank and some recent tanks, without a lengthy history, this is still an alternative to most in-house propulsion test analysis methods.

As regards experiments in the tank and the role the rudder takes in these, there are alternative viewpoints. At one extreme the rudder is an appendage, lumped in with the others, whilst at the other extreme it is regarded as an integral part of the propulsion device. As a consequence it is treated differently and the scaling difficulty is exaggerated compared with that of the ship as a whole.

As examples of what is meant consider the even more difficult cases of a duct in a ducted propeller (kort nozzle) or a water jet intake. Are these items accountable to the resistance or propulsive side of performance, or both? The duct can be built into the hull in which case, like the water jet intake, it is not possible to test it in open water and the manner of its treatment is open to discussion.

The duct of a ducted propeller that is **not** built into the hull but is supported by struts or the like can be tested in open water however and can, and should, properly be regarded as an integral part of the propulsor. The flow around the duct is dominated by the action of the propeller, added to which the duct can contribute substantial thrust in propulsion despite this compelling argument for the duct configuration, others have classified the duct as an appendage even though the flow around it in the absence of the propeller is completely different and bears no relationship to that in the propelling condition.

4.2 Rudder Classification - Propulsor Attribute or Appendage? The rudder in a single screw ship is in a somewhat similar category to the duct or water-jet in that it reacts closely with the propeller upstream. Although the rudder_{HP} force is invariably a drag at normal service speeds this force is the sum of the drag and propulsive components which cannot be separated. In addition to this the propulsor thrust and torque are also changed by the presence of the downstream rudder.

The propulsive effect of a rudder was described by Suhrbier (1974), and the rudder was considered as part of the propulsor by English (1975) where it was shown how the hull-factors changed under this assumption. This approach is still regarded by the writer as the most appropriate because the nature of the close interactions in the rudder_{HP} condition which also occur in the rudder_{OP} condition and, in the limit, when the aft end behind flow approached

uniformity rudder_{OP} the case is recovered. This is not the case when the rudder and propeller are separated. It is not difficult, however, to understand why regarding the rudder as part of the propulsor did not find favor and acceptance generally; this is simply because more work is involved in conducting the tests in the rudder_{OP} and rudder_{HP} conditions, as a sensitive transducer has to be used to measure the rudder force in both cases. Furthermore it would have produced a step jump in results which is always unpopular when compiling a subsequent data base.

the alternative view has prevailed amongst the Tanks in general and the rudder is, and is likely to continue to be, regarded as a drag generating appendage. Stierman (1989a, 1989b) discusses and makes proposals on this approach and similarly Kracht (1992) also follows this approach.

4.3 Rudder as Drag Generator. The ramifications of considering the rudder as a drag generator in the analysis of resistance and propulsion experiments are quite deep when considering the diversity of propulsion analysis methods in use and how appendages are treated within them. For example, some Tanks install the rudder in the rudder_{HP} resistance experiment whilst others do not. The form factor k is used by some tanks and not others and this can change with different rudders in some cases. There could be other differences in local testing methods and there are certainly differences in the model to ship extrapolation process not to mention the ship model correlation factor that is judgementally applied in predicting the ship power.

The scale effect of ship appendages in general is one of the least satisfactory aspects of ship model testing and this is particularly true for large, heavy displacement merchant ships when scaled models are used. As already mentioned bilge keels are not normally fitted in resistance and propulsion experiments because their drag is excessive. Most other appendages have a sizable empirical scale correction factor applied to the drag increment when it is scaled to the ship. for example, from a model test the non dimensional drag of upstream propeller shaft bossings and A brackets could be factored by 1/2 in extrapolating to the ship. On the other hand, some Tanks estimate the appendage drag independently of the model test using simple empirical formulae for this.

With most appendages the model drag can be found fairly precisely by conduction 'with' and 'without' appendage tests, however, this cannot be done with rudders in slipstreams because in addition to the model rudder force there is a noticeable increase in the propeller thrust and torque coefficients.

In the light of the conventional and somewhat imprecise treatment afforded to appendages in general, it seems incongruous that the rudder drag penalty measured on a ship model should be applied to the ship without correction. Furthermore if it is applied with correction what should the correction be?

As it is generally agreed that the interaction between the propeller and the rudder is much greater than that between the **propeller + rudder** and **hull** the former interaction deserves emphasis. (The fact that the thrust deduction t can change with a change of rudder is not because of a change in t in the conventional sense, i.e. a pressure reduction on the stern, but is because of the change in thrust due to the conditions downstream of the propeller). Therefore, perhaps the most productive way of investigating the rudder force problem is by using large scale model tests in water tunnels and tanks in the rudder_{OP} mode, as reported in Suhrbier (1974), English (1975), Stierman (1989a, 1989b), and latterly in a wind tunnel in Molland and Turnock (1992).

In passing it may be mentioned that numerous so-called efficiency enhancers are fitted on ships today even though an enhancement has not been demonstrated at the model scale and sometimes the reverse has been found. This displays faith or trust by the owner in the advocacy of the body making the recommendation in the first place. This is even more remarkable when it is realized that in some cases it appears difficult to justify the hydrodynamic claims made for installing the 'enhancer' in the first place.

5. Stern Arrangement

From the Towing Tank viewpoint the scope for optimizing the stern arrangement of high powered single screw ships is fairly limited, as frequently the critical dimensions such as position of the rudder stock (Aft Perpendicular), aft face of the stern tube and height above shaft to hull are frozen at an early stage in the design.

This takes account of structural considerations, propeller diameter and clearances following from the main engine specification. There may be some latitude to change the line to improve the flow to the propeller and help with minimizing propeller vibration excitation and sometimes the position of the screw in the clearwater stern can be varied by a small amount to try and minimize power.

Although the impact of the rudder on manoeuvring may not have extended beyond fixing its area and dimensions as a fraction of the underwater profile area, there is usually little scope for anything other than minor adjustments to the rudder and aperture at the model test stage.

If it is decided to test an alternative rudder type at a later stage in the model test programme the constraints already present usually have to be observed, in which case the alternative rudder design may not show to its best advantage from the drag viewpoint.

6. Treatment Of Appendage Drag

The treatment of appendage drag varies considerably depending on the Tank conducting the work. It seems without exception that bilge keels are not fitted on ship models and as regards the other appendages some Tanks fit them and make empirical scale corrections, whereas, at the other extreme model experiments are not used at all and are replaced with empirical formulae.

An up-to-date review of the treatment of Appendage Resistance by Tanks would be welcomed by their customer, including the method of scaling from model to ship where appropriate. This is particularly important in the case of rudders in general, where scale corrections are often not applied, and in the case of some high performance rudders where an even greater uncertainty exists in predicting powering performance on the ship from model tests. This subject assumes more importance because of the current attempts to formalize manoeuvring standards for ships.

7. References

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III. REPLIES

Reply to M. Abe

The Committee is grateful to Dr. Abe for his contribution. As stated in the report, if the influence of the appendages on the wavemaking resistance is considered negligible at low and moderate Froude numbers, the variation of the viscous resistance due to the appendages is obtained from variation of the form factor with respect to that calculated for the bare hull. But as was indicated in the report of the Powering Performance Committee of the last ITTC (page 250 of Vol. I), both form factors, those of the appended and the bare hull models, must be calculated using the bare hull wetted surface in both cases. Really, as Dr. Abe says, the use of Eq. (1) of page 260 is only an approach to the extrapolation of this formula to the "Lucy Ashton" data, and it shows that to take into account scale effect on the form factor is a promising way to be investigated. The difficulty is how to know the form factor scale effect. A better knowledge of this problem should be very interesting, as stated in the recommendations of the Committee and, in this

way the proposal of Dr. Abe about the interest of a well documented data base is very valuable.

Reply to R. Rocchi

The Committee very much appreciates Mr. Rocchi's contribution, and agrees with him, that a well documented data-base is very useful to obtain regression equations which enable the determination of some unknown parameters (in this case the form factor) with a very narrow gap of error, if the sample is well selected. Nevertheless, the use of statistics has the risk to choose a lot of variables as a consequence of the use of powerful software or to choose some variables which correlate very well but with a doubtful meaning in representing the problem. In Mr. Rocchi's contribution and referring to his papers, the accuracy obtained in the prediction of 1+k values, indicates that the proposed equations depend on well selected parameters. One objection is that some of the independent variables used in the equations which represent the hull geometry are not explained and that is a restriction to the generalization of his procedure.

Reply to A.F. Molland

Thank you very much for your comments. We also received Dr. English's contribution on the rudder force. The title of the paper is "A Note on Rudder Force at Zero Helm". He says about the complex feature of rudder force, "It seems fair to conclude that reliable predictions of full-scale conventional rudder force cannot be made as a result of traditional ship model and resistance experiments." As to rudder drag, we want to say that rudder action is very complicated. That is, in the axial flow, only drag acts on the rudder, but in the slipstream, which consists of axial and rotating flows, both drag and lift act on it. Also, the rudder affects the flow field at the propeller plane due to the rudder thickness, and then the thrust and torque of the propeller increase. These are caused by interference effects. In this respect, the theoretical work of Kyushu Univ. group may be useful to understand the mechanism. Also, they theoretically treated the propeller-rudder combination as one propulsor. This is surely an interesting idea, but it is difficult to perform the open-water test of such combination. We expect that Southampton's investigations will

clarify the entire mechanism of propeller-rudder combination.

As to catamaran powering, we thank you for your report on experience of viscous form effect of catamaran with transom stern. The flow field around such transom stern changes drastically at high speed. Then we agree that form factor can not be determined from resistance tests at low speeds. We think some members applied form factor to catamaran operating at relatively low speeds. We expect member organizations will report to the PPC their experiences with powering of catamarans.

Reply to J. Holtrop

We would like to thank Dr. Kuiper for presenting the contribution of Mr. Holtrop. Secondly, we would like to thank Mr. Holtrop, who is the immediate past Chairman of the PPC, for preparing his discussion, and for his continuing interest in our Committee. Mr. Holtrop proposes an interesting concept to increase our confidence in determining form factor for appended models. He proposes to use the higher speed points of the resistance or load varying test of the appended model and subtract the wave making resistance of the unappended model assuming it is similar to that of the appended model. The Committee agrees that this approach is worthy of consideration by the member organizations.

The Committee also agrees with Mr. Holtrop's principal comments regarding retaining the open water propeller test as an element through which the accuracy of prediction can be improved. Our conclusion specifically stated, "for routine testing and power prediction, the load variation test can replace additional and inconsistent hull resistance and open water tests in many cases." We agree that the open water test contributes to the accuracy and reliability of powering prediction for each of the reasons cited by Mr. Holtrop, and particularly in the area of propeller scaling. The Committee did not intend to suggest that the weak point of CFD is the availability of experimental results as interpreted by Mr. Holtrop. What we were recommending was that more precise model and full scale test data are needed in order to verify CFD results. It is interesting to note that the Resistance and Flow Committee also recommends more precise benchmark experiments to validate CFD codes. As regards Mr.

Holtrop's question on identifying public experiments for CFD correlation, we refer him to the Resistance and Flow Committee report, page 54, which identifies a set of experimental data available upon request.

Mr. Holtrop questioned three of our recommendations to the Conference being similar to, and overlapping recommendations for future work of the PPC. The Committee hopes that these recommendations to the Conference will stimulate work by member organizations which can then be provided to the PPC for consideration in its future work. Finally regarding Mr. Holtrop's comment on ISO-9000, as we heard earlier from the Quality Control Group, ISO-9000 is upon us. We share Mr. Holtrop's concerns regarding implementation. Hopefully the Quality Control Group will be providing all of the committees the guidance needed in order to apply ISO-9000 to our work. Again, the Committee would like to thank Dr. Kuiper, and especially Mr. Holtrop for his continued interest in our work.

Reply to C.W.B. Grigson

The Committee very much appreciates Dr. Grigson's efforts towards improved performance prediction for full ships. They are in many items very much in line with most of the other new proposals as reviewed by the Committee such as emphasis on:

- Full hull form applications
- Load variation tests
- Propeller behind conditions
- Scale effects of
 - form factor,
 - wake fraction,
 - propeller characteristics and so on.

Our special interest however is directed to three novel items in Grigson's approach,

- A new proposal for the model self-propulsion test point to fulfill equality with full scale of:
 - thrust loading based on propeller advance velocity, rather than
 - thrust coefficient based on ship speed, as conventionally practiced.

- A new proposal for an improved friction line basically reducing the slope of ITTC-57 line, especially in the model range of Reynolds' numbers, quite similar to Schoenherr's line but even stronger.
- And, a new proposal for the hull roughness effect according to his consistent efforts in this field.

These new proposals deserve serious attention and analysis since evidence from the presented 31 cases of model-ship trial comparisons opens promising prospects for developing improved propulsion prediction methods, also for the next Committee.

Reply to S. Cordier and F.X. Dumez

We thank you for your report on form factor and resistance components of Series-60 and semi-displacement hulls. The last PPC considered the powering of semi-displacement craft. If spray drag is evaluated by elaborate wave and wake analyses, the scale effect of spray drag may be clarified.

Reply to M. B. Wilson

The Committee thanks Dr. Wilson for his comments on the interaction of tandem struts of a SWATH vessel and for his explanation of the physical phenomena associated with such a configuration. It is the opinion of the Committee that both struts should be considered as a system integral with the submerged hull and hence powering predictions should be made as in the case of powering predictions for multiple appendages located one behind the others.

Reply to S. Miranda and J. Szantyr

We thank you for your report on the lift-based scale effect on propeller characteristics. This is read as an appendix to section II.5.

Reply to T. Loukakis

The Committee thanks Prof. Loukakis for his contribution. He mentioned that Prof. Tzabiras had published his paper recently which involved CFD in scaling problems and gave some interesting results. The scaling

problem is one of the most important items and CFD is one of the ways for solving this problem. We think that the ITTC will pay more attention to this field. We agree with Prof. Loukakis's suggestion that the Resistance and Flow Committee and the Powering Performance Committee should work closely together in the future.

Reply to A.J. Musker

The Committee is very pleased to receive Dr. Musker's remarks on the present status of CFD regarding full-scale powering prediction. We agree with his comment that for the time being CFD calculations should be used as a supplementary tool.

Reply to S.H. Van, M.C. Kim and J.T. Lee

Thank you very much for your contribution. You are addressing exactly what we were looking for in the Committee with respect to unconventional devices, namely scaling methods to predict the powering performance with unconventional devices. We noted that there is fair agreement in the efficiency gains concluded from your methods A and B and direct scaled up tank power, whereas the ITTC-78 extrapolation leads to different results. Looking to the level of improvement we can see the figures to be in line with other data on stator-propeller systems mentioned in our review. The arguments for the alternative prediction are not so well founded. However, the statement that the difference in wake between conditions with and without stator is of potential origin, and thus the same for model and ship, seems a reasonable assumption. We agree with you that the method is to be evaluated further, and this fits exactly in the recommendation for the next Committee to concentrate especially on methods for CRP and propellers with stators.