devoted to the measurements on each ship is divided in two parts, the first part concerning mild weather until Beaufort 5, the second part relating to worse weather. The service trials comprise for the cargo ships:

Victory Tervaete	3 voyages measured total 38 days	of which 7 days in bad weather
Lubumbashi	4 voyages measured total 30 days	of which 11 days in bad weather
Jordaens	4 voyages measured total 32 days	of which 13 days in bad weather
Dart Europe	7 voyages measured total 76 days	of which 22 days in bad weather

many croccings Ostend-Dover by cross-channel motor ships Koningin Elisabeth and Reine Astrid

3 voyages around the south corner of South America on ore-carrier Mineral Seraing
8 voyages around the south corner of Africa on tanker Reine Fabiola
3 voyages in Artic waters together on side trawlers Belgian Lady and John
1 voayge to South Africa on stern trawler Narwal.

All the data were published in the RINA transactions, exception being made for the tanker trials published in ATMA and the trawler trials published in the NECIES transactions.

Measured mile trials were run on all ships, moreover in loaded condition on Tervaete, Lubumbashi, Jordaens, and Reine Fabiola. The large number of voyages, some of them separated by drydockings, allowed in many cases the effect of fouling to be established. After 5 years service the Lubumbashi was sandblasted and renewed measured mile trials were run. Hull roughness measurements were made, not only in newly-built condition, but also when at intervals the ship drydocked. These measurements were related to the sea data and the effect of fouling and ageing described in 3 RINA paper. From photographs taken by the Geodesy Laboratory of Delft profiles were derived which were analysed according the BSRA method, giving roughness before and after sandblasting. Having said this, I am enthusiast of the prospect measurements be made in service on many "unmanned" ships - I mean by that the absence of expert technicians - fully instrumented. Measured mile trials possibly should be carried out in loaded condition. Using an admiralty coefficient fuel includes the engine ageing indeed. There are however two objections:

1°) This engine ageing remains unknown;
2°) Measuring fuel consumption is not so easy.

Therefore it is suggested to add a torsionmeter to the instrumentation. This valuable tool, which ought not be calibrated, will allow, in combination with speed and rpm measurement, to investigate the increase of wake with fouling. Unfortunately, the maintenance of a torsionmeter, on bord an unmanned ship, will be difficult. On the other hand a pressure instrument allowing to measure draught on board might be useful. Photographs may be helpful when evaluating hull roughness. But measuring hull roughness by means of a BSRA gauge will add to this evaluation. Fitometer profiles give the increase of frictional resistance as result of fouling. Measuring by eye the wave height is an intricate task, therefore a wave meter undoubtedly will be useful. Ref.268 of Seakeeping Committee mentions for the containership Dart Europe a reasonable agreement between motions and power measured on the model and calculated from the record obtained by the Tucker wave recorder. Bearing in mind that in severe weather conditions the ability of the crew is abated, this shipborne instrument may be recommended. Instalment is expensive, but maintenance has no problems.

If an extensive instrumentation is needed, analysis of the data will be intricate because many parameters ought to be considered. It is so difficult to establish a speed loss that for the criterion suggested to the 12th ITTC (Ref. 69 of Performance Committee) only length of ship was taken as a parameter. It is clear however that in a second stage parameter C_B might be considered when establishing the speed loss, at least for ships beyond 160 m in length. Applying the above mentioned criterion to a ship 218 m in length gives in Beaufort 8 head seas a speed loss 35%. It is suggested however that for the fine ship C_B=0.60 (containership Dart Europe) 30% is a better approximation and for the bluff form C_B=0.80 (ore-carrier Mineral Seraing) 40% suits more this sea state. This however for the light-loaded ship only. For the full-loaded containership 35% loss is all-right. This ship stands better slam than green seas which might wash containers over board.

There is finally the so controversable decision making of ship master. One can argue at length on the thresholds of slamming (in light-loaded condition) and shipping of water (in full-loaded condition) at which ship master decides to reduce speed. No shipowner dares here to instruct his captain. In Ref.220 of Seakeeping Committee fig. 1 shows that in ballast condition on the maiden trip of ore-carrier Mineral Seraing ship master in a storm endured the whole time 5 slams per 100 pitch oscillations. During the second voyage in a similar sea state a relief captain— and just because he was a relief captain and would not do any harm to the ship of his colleague-reduced speed to 2 slams per 100 oscillations. As a result the difference in speed between the captains in a head sea Beaufort 7 amounts to 1 knot. Larger differences are met at higher Beaufort numbers. Therefore it is recommended to mention in the log-book the salient slamming and water shipping data. For instance: acceptable slamming might be noted S, but when it produces conspicuous whipping of the hull girder W. When much spray is seen on forecastle please note s, and when green seas are washing the upperdeck, do not forget to mark G (green seas). Again instruments might do better: a strain gauge on the upperdeck amidships for slamming, another strain gauge for counting the number of green seas (of the latter I have unfortunately no experience).

I wish full success to this endeavour for the profit of the profession.

A.A.ROUSSETSKY - Krylov Shipbuilding Research Institute, Leningrad, USSR

THE EFFECT OF OPERATING CONDITIONS ON THE PROPULSIVE PERFORMANCE OF SHIPS AND THE MODEL - SHIP CORRELATION OF THE PERFORMANCE OF HYDROFOIL VESSELS.

It has justly been stressed in the Performance Committee Report that data obtained from operational tests are very important for the estimation of margins required to attain a given sea speed as well as for choosing screw design conditions.

And the most interesting for a well-grounded determination of these margins are not the test results of individual specially instrumented ships, but the data on the performance of a large number of ships in normal operation. In this case the main source of information are ships log records based on the reading of standard ships devices. Though the volume of information and its accuracy in this case prove to be substantially less for each particular ship, this is largely compensated for by the mass character of information and repetition of similar operating conditions.

To obtain this kind of systematic information, in the USSR log data had been processed for over 250 single-screw ships built from 21 type designs, the analysis made from a number of ships covering four to five years of operation. These ships included tankers, ore carriers, timber carriers, and dry-cargo vessels. Consideration was given to data on the motion of ships in loaded condition as obtained both in calm and rough water characterized by the force and direction of wind recorded in the log.

For the purpose of analysing the results obtained, particularly for reducing test data to specification values of displacement, use was made of the model test data, design data, as well as data on trials of ships. The results were represented as curves of relative thrust increment required for ensuring speed values, attained during the trials. Then curves were plotted against the time of ship operation, some of the curves being plotted for the wind of force 3 and below, and for wind force 4, 5, 6 and 7, respectively. Similar curve plotted for dry-cargo ships of about 10000 tons displacement are given in Fig.1. The lower curve shows the thrust increment required in still water, which is due to fouling and corrosion, while the difference between this curve and the one corresponding to various wind forces determines the additional thrust conditioned by rough water and wind effect. When plotting the curves of additional thrust in waves, use was made of the point obtained for various directions in relation to wind, id est the curves represent the increment values averaged over the relative direction. The operating conditions corresponding to a deliberate reduction of speed by the navigator were ruled out of data processing.

Similar curves were plotted for all the ships surveyed.

In the analysis of the plots it should be borne in mind that the curves corresponding to lower sea states are more reliable, the experimental points being greater in number and having less dispersion due to absence of relative direction effect. At the same time account must be taken of the fact that the increment of the thrust in still water is not only by the growing resistance of ship but also by the drop in screw efficiency due to fouling, which factors are difficult to divide. From this point of view, processing of data on the value of additional thrust retained after drydocking and associated with corrosion of plating is more reliable since the screw characteristics may be expected to practically restore in the course of drydocking. The nature of variation in additional thrust versus period of operation is given in Fig.2, a sharp decrease of thrust as shown by the curves being associated with drydocking. Fig.3 shows the curves of residual thrust increment for the surveyed ships, which is determine by corrosion of plating against ships operating age. Fig. 4 shows additional thrust increment versus operating age curves, between two dry dockings, the increment being associated only with fouling of the hull and ships screw.

The curves for an increment of the requisite thrust due to wind and rough water effect are given in fig.5. These data, as also the data on increment of thrust in calm water are in good agreement with the results obtained by Aertssen and Kent. It appears that the results of this analysis may be used for the assessment of service margin when specifying the delivery trial speed to the designed ships.

The second item I would like to touch upon refers to correlation of full scale and model test data for high-speed vessels.

It has justly been mentioned in the Committee Report that the full scale test data related to high-speed vessels are less reliable. It is felt however, that this remark is true for hydrofoil vessels but to a lesser extent. By way of illustration data are presented on Coefficients C_p and C_N for two types of hydrofoil ships "Taifun" and "Kometa".

Table 1

Calculation of coefficient C_p and C_N for hydrofoil "Taifun"

V_s	n_{nat} rpm	n_{cal} rpm	$C_N = \frac{n_{nat}}{n_{cal}}$	N_{nat} hp	N_{cal} hp	$C_p = \frac{N_{nat}}{N_{cal}}$
34.3	1640	1585	1.034	910	870	1.045
38.5	1785	1760	1.014	1125	1060	1.061
41.25	1920	1895	1.013	1285	1260	1.02
44.45	2060	2065	0.998	1570	1560	1.006
44.85	2060	2092	0.985	1570	1610	0.975
45.3	2095	2115	0.990	1650	1650	1.0

Table 2

Calculation of coefficient C_p and C_N for hydrofoil "Kometa"

V_s	n_{nat} rpm	n_{cal} rpm	$C_N = \frac{n_{nat}}{n_{cal}}$	N_{nat} hp	N_{cal} hp	$C_p = \frac{N_{nat}}{N_{cal}}$
33.6	1625	1680	1.015	872.5	910	0.96
33.2	1615	1550	1.04	830	840	0.99
32.2	1540	1450	1.065	790	760	1.035
30.2	1400	1370	1.02	705	650	1.08

Prediction of performance for the hydrofoil ship "Taifun" equipped with a Z-drive propulsive system was based on the towing test of a model with all its appendages and the cavitation tunnel test of the propeller in the open water and in the presence of the Z-drive with strut and propulsion pod. The latter tests were carried out in order to determine the interaction factors. Towing tests and open water test were also conducted for the hydrofoil "Kometa" which has inclined propeller shafts. The propeller characteristics were corrected for oblique flow. Corrections for interaction effect were not taken into account.

As is seen from the tables, the values of correlation coefficients for the hydrofoil in question are quite near unity.

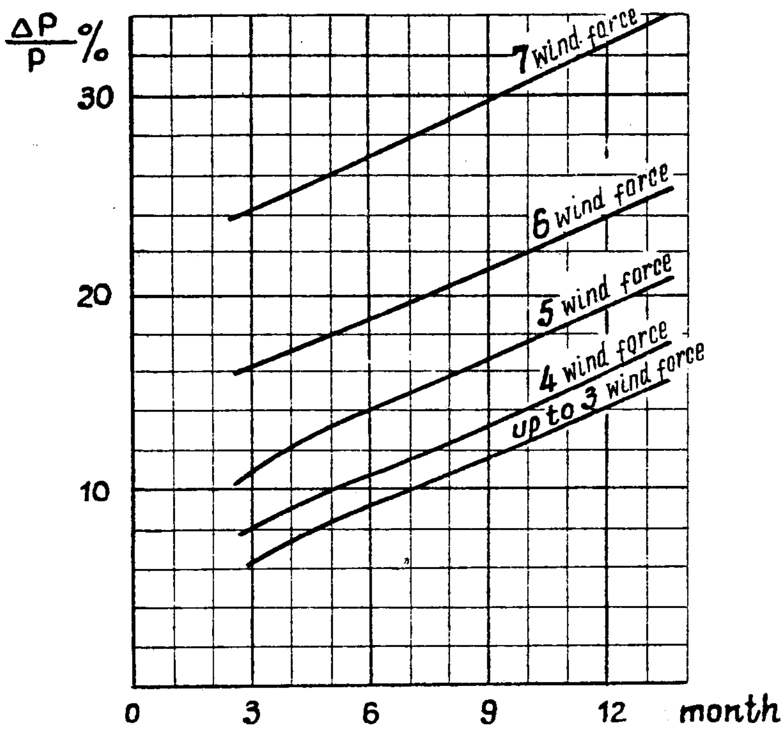
ΔP/P %
30
20
10
0
3
6
9
12
month
7 wind force
6 wind force
5 wind force
4 wind force
up to 3 wind force

Fig. 1

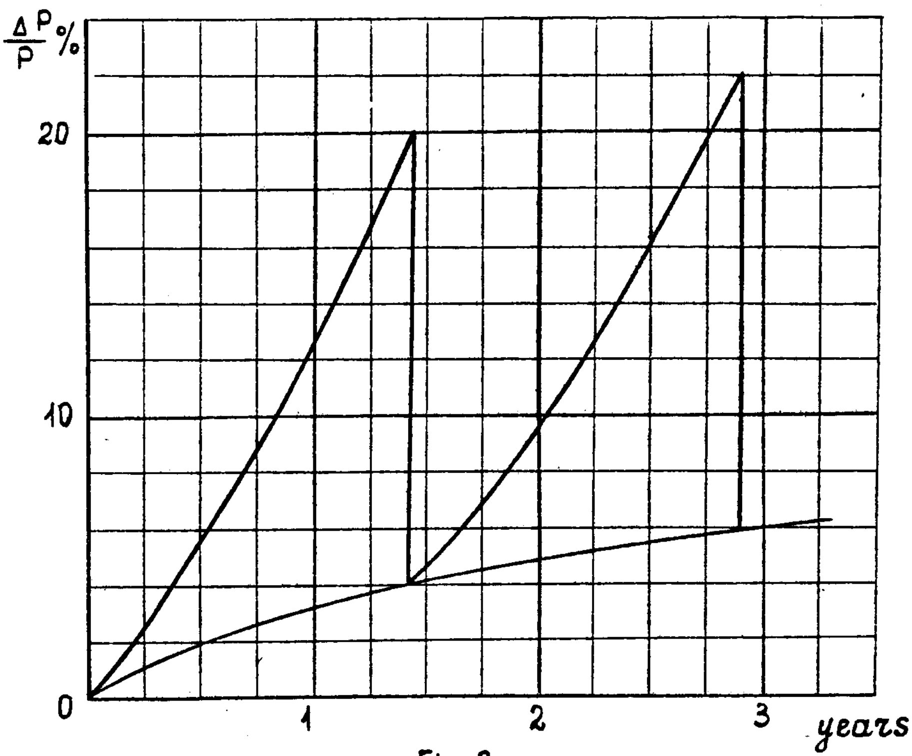
ΔP/P %
20
10
0
1
2
3
years

Fig. 2

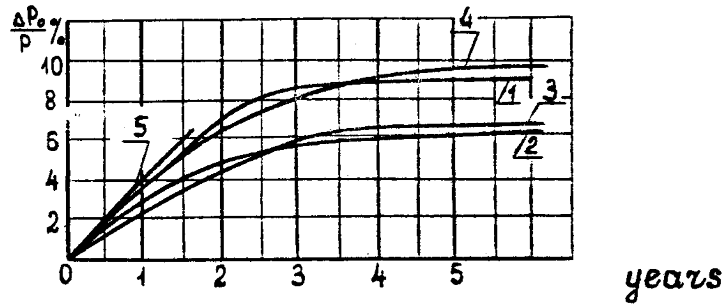

Fig 3.

1 – Ore Carriers
2 – Timber Carriers
3 – Dry cargo ships
4 – Tankers
5 – „Mineral Seraing"

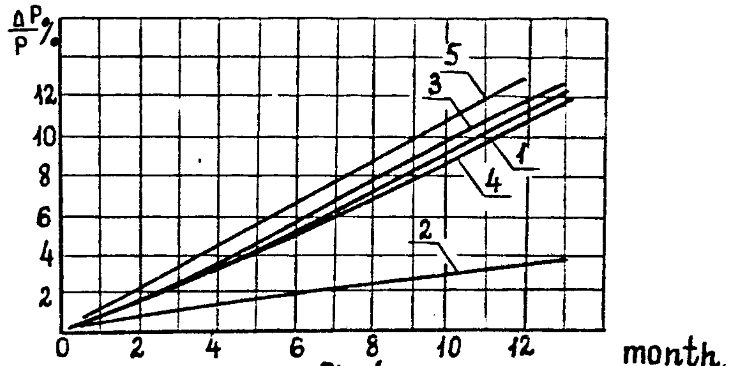

Fig 4

1 – Ore Carriers
2 – Timber Carriers
3 – Dry cargo ships
4 – Tankers
5 – „Mineral Seraing"

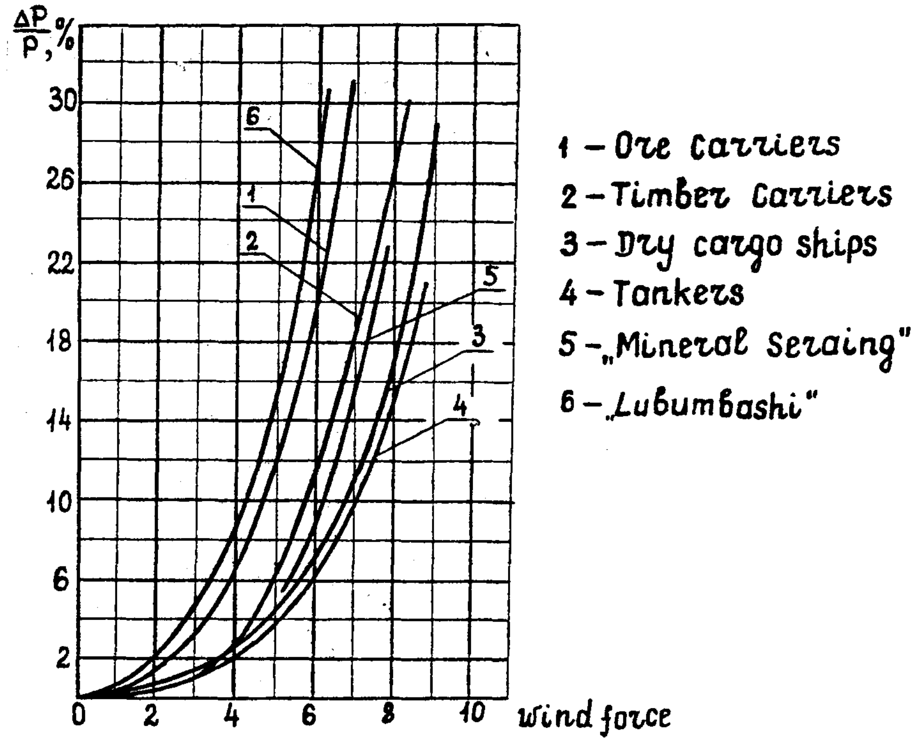

Fig 5

G.DYNE - Swedish Maritime Research Centre SSPA, Göteborg, Sweden

May I first correct a misunderstanding in section 8.2 of the report. When we determine the form factor at SSPA we use not only the self propulsion test results as indicated in the report but a combination of results from resistance and self propulsion tests at low speeds. The thrust deduction is then assumed to be a simple function of the speed.

The curve fitting procedure described in ref. [89] is a result of a cooperation between the Swedish, Norwegian, Danish and Finnish towing tanks. One of the basic characteristics of this procedure, not mentioned in the report, is that it is directly adapted to the 1978 ITTC Method. By means of the open water characteristics the results from the resistance and the self propulsion tests are faired simultaneously in such a way that the thrust deduction, the effective wake and the relative rotative efficiency become simple functions of speed or propeller load. Of course, this method gives somewhat larger standard deviations than if the different measuring quantities were faired separately, but this drawback is in most cases small compared to the advantage of obtaining consistent values of the propulsion coefficients.

The 1978 ITTC Method also forms the basis of another method developed within the Scandinavian cooperative study, viz. correction and analysis of trial test results. In the first part of this method the open water characteristics of the full scale propeller are calculated from test results with corresponding model propeller. The influence of different scales as well as deviations in diameter, pitch and blade area is then considered.

With the aid of these characteristics and with the assumption that the relative rotative efficiency and the thrust deduction are the same as in model scale, the uncorrected ship resistance and the advance speed are determined from shaft power and RPM for the individual runs in the double run. This resistance is then corrected for wind, sea and rudder movements. Assuming the wake fraction to be the same in both directions, this wake and an approximate value of the current are obtained. The mean values of the speed and the corrected resistance are calculated after which additional correction for deviations in draft, temperature and density are made. With the known value of $K_T J^2$ as input the advance ratio J_{TS} and the torque coefficient K_{QS} are read from the propeller characteristic curves. J_{TS} then gives the final rate of revolution and K_{QS} the torque and the shaft power

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I have a comment on page 151 d) of Proceedings vol 1.

The equation $\frac{C_T}{C_F} = (1+K) + Y \frac{F_N^n}{C_F}$ of a twin screw ship with appendages is not a straight line due to separation, so we can not get the form factor K by plotting. Dr. Prohaska gives

$$C_T = (1+K+b)\,C_F + a + Y F_N^n$$

for twin screw ships. We can use double model and test it in wind tunnel or towing tank. By plotting C_T against C_F, we can get the slope (1+K+b) and the constant a. MARIC will try to use this method to predict the performance of twin screw ships.

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

ON THE ESTIMATION OF FORM FACTOR OF FULL FORM SHIPS

1. As recommended by I.T.T.C., the three-dimensional extrapolation should be used for full form ships, the value of form factor K plays an important role in the process. The revised Prohaska's method was recommended as standard by the 15th I.T.T.C., but there is still some uncertainty left for discussion.

In our small basin, 50 M long X 6 M X 2 M water depth, the Froude two-dimensional extrapolation method is adopted for routine works. We are intending to follow the recommendation from 15th I.T.T.C. and attempting to verify whether the revised Prohaska's method is suitable for such a small basin of our's or not. Three geosim models of 50000 tons dwt tanker "Westlake" were tested in our small basin for this purpose.

The principal particulars of the tanker were:

Lwl = 215.0 M	Cb = 0.825	L/B = 6.774
Lbp = 210.0 M	Cm = 0.994	B/T = 2.583
B = 31.0 M	Cpa = 0.7661	Lr/Lbp = 0.40
T = 12.0 M	Lcb = 1.735% LbpF	

The scale ratios of the geosims were 52, 65 and 85, and the lengths of models were 4.0385 M, 3.2308 M and 2.4706 M respectively.
The date and water temperature of these tests are given in table 1.

Table 1

Test No.	Model No.	Date of test	Water temperature
5001	M5085	Dec. 9, 1977	12.0 C
5002	M5085	Aug. 4, 1978	27.6 C
5003	M5065	Aug. 18, 1977	26.0 C
5004	M5052	Aug. 21, 1977	25.6 C
5005	M5052	Aug. 10, 1978	27.8 C
50052	M5052	Feb. 1, 1979	6.6 C

The blockage correction of the resistance test results was applied according to Schuster's formula which was checked by testing 10 models of various fullness in two tank sections in our basin.

The 1957 I.T.T.C. correlation line was used in the analysis.

There was significant inconsistancy between the residual resistance curves by the two-dimensional extrapolating method of the 6 tests of geosims. But it was remarkably improved when three-dimensional method combined with suitably estimated form factor was applied.

The value of K deduced from geosim test results is not a constant throughout $0.12 \leq Fn \leq 0.20$. It decreases when $Fn > 0.15$. Its tendency is similar to that given by Dr. Taniguchi.

Comparison between calculated values of 1+K by Prohaska's and revised Prohaska's methods were made as shown in table 2.

It can be seen that the value of 1+K deduced from the revised Prohaska's method is more satisfied.

The wave-resistance coefficient Cw curve deduced from this geosim test may be expressed as

$$C_W = 77.5687 F_n^{6.96}$$

It gives m=6.96, but the values of m shown in table 2 were 6.70-10.34 and 5.23-12.39 for two different Fn ranges respectively. Since the Fn at designed speed of most full form ships were less than 0.18, it seems that the calculated range of Fn for full forms and hence the value of index m should be subjected to further investigation.

Table 2

Fn	0.12-0.20			0.12-0.18		
Method	Rev. Prohaska's		Prohaska's	Rev. Prohaska's		Prohaska's
Fric.L.	1957 I.T.T.C.			1957 I.T.T.C.		
	Index m	1+K	1+K	Index m	1+K	1+K
T.5001	8.84	1.2482	1.2018	5.2343	1.2347	1.2244
T.5002	9.28	1.2518	1.1976	12.3914	1.2571	1.2283
T.5003	10.34	1.2323	1.1873	6.3316	1.2247	1.2127
T.5004	8.59	1.2315	1.1765	9.9684	1.2351	1.2044
T.5005	6.89	1.2186	1.1856	7.4192	1.2201	1.1981
T.5006	6.70	1.2169	1.1822	6.6942	1.2165	1.1952
Average		1.2332	1.1885		1.2314	1.2105

2. Since the accuracy of control system, especially in the low speed range, is quite different from each other, and the characteristics of ship lines design were also different from one basin to another, the empirical formulae from different sources can't be satisfactorily used without any revision, and it is better to establish such formula on the basis of the testing data from their own model basin. Therefore, by utilizing the data of 93 ship models of full forms tested in our small towing tank in recent years, the analysis of the significance of each parameter , such as L/B, B/T, Cb, Cm, B/Lr, Lcb, etc, and comparison with other existing formulae were made. By utilizing the methods of least square, regression, and logarithmic regression analysis, a new formula was found, i.e.

$$K = 91.20\, Cb^{8.7}\, Cm^{66.72} / (L/B)^{1.99} (B/T)^{0.847} (B/Lr)^{0.964} Lcb^{0.122}$$

The value of 1+K calculated from this formula for the tanker "West Lake" coincides quite well with that obtained from the geosim tests.

C.P.SHENG - Shanghai Chiao-Tung University, China

A NOTE ON THE FORM FACTOR

Recently, five geosim models with lengths 3.197 m, 3.673 m, 4.316 m, 5.232 m and 6.640 m respectively have been tested at the Ship Hydrodynamics Laboratiry of Shanghai Chiao-Tung University. The prototype is a 24,000 DWT tanker with $L_{pp} \times B \times d = 170 \times 25 \times 9.5\,m$, $C_B = 0.778$.
The models are made of wood and carefully finished off. For resistance tests a 1 mm dia, trip wire is fitted at 1/20 L_{pp} abaft the F.P. The ship models are without bilge keels and rudders.

Fig.1 gives the C_T-C_F (1957 ITTC) curves. Fig. 2 shows the relationship between (1+K) and F_n.

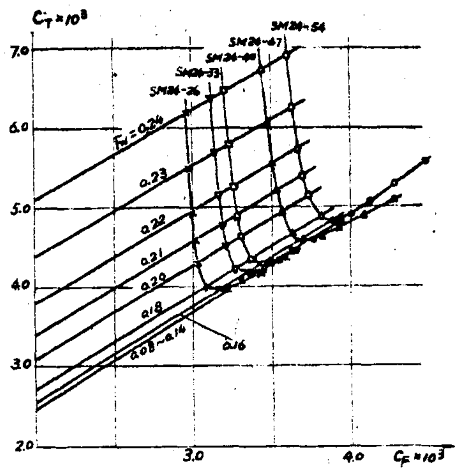

Fig 1

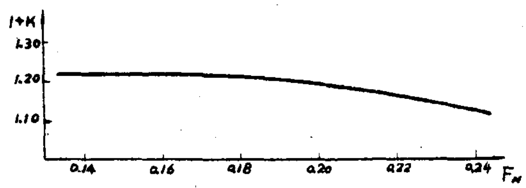

Fig 2

It indicates that the form factor (1+K) is almost constant within the range of F_n = 0.16, while in the higher speed range (F_n > 0.16), the form factor (1+K) decreases as the Fn increases.

The form factors of individual models have been determined both by Prohaska method and 15th ITTC method. The results are as follows:

Model No.	SM24-54	SM24-47	SM24-40	SM24-33	SM24-26
(1+K) Prohaska	1.1657	1.1737	1.1814	1.1555	1.1769
(1+K) ITTC	1.1929	1.1940	1.1948	1.1929	1.1983
Exponent m	6.9	6.5	5.1	7.1	5.6

The results show that the ITTC method is more reasonable than Prohaska's method as the differences of (1+K) values of five models obtained by ITTC method are less than 1%. But the problem is that the exponent m values of five models are quite different and this is hardly to be explained in accordance with the law of similarity of wave resistance.

In determining the form factor by the ITTC proposed formula

$$\frac{C_T}{C_F} = (1+K) + C\left(\frac{F_n^m}{C_F}\right),$$

the minimum standard error does not exist in analyzing some model resistance test results and this would cause the uncertainty of (1+K) value. However, after making a careful examination it is found that no matter whatever the m value is, the largest descrepancy always happens at the same spot. As this spot is considered to be "wild spot" and not taken into account in determining form factor, the minimum standard error and the correct (1+K) value thus can be obtained.

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THE APPLICATION OF 1978 ITTC PREDICTION METHOD

The Ship Hydrodynamics Laboratory of Shanghai Chiao Tung University has reanalyzed the speed trial data of seven single-screw ships and their corresponding model experimental results in accordance with the "1978 ITTC Prediction Method".

The Principal dimensions of seven single-screw ships are listed in table 1.

No.	Item	Ship No.						
		A	B	C	D	E	F	G
1	L_{PP} (m)	147.18	147.0	147.0	148.0	148.0	170.0	172.0
2	L_{WL} (m)	155.87	152.8	152.1	153.195	153.195	172.65	176.19
3	B (m)	20.40	20.4	20.4	21.2	21.2	25.0	23.2
4	T (m)	8.15	8.587	8.025	9.14	9.14	9.5	9.8
5	C_B	0.677	0.676	0.668	0.662	0.662	0.778	0.809
6	C_M	0.984	0.984	0.984	0.974	0.974	0.9924	0.994
7	S (m^2)	4222.0	4198.9	4092.2	4489.76	4489.76	6117.1	6330.0
8	S_{BK} (m^2)	100.32	108.594	96.52	92.394	92.394	192.0	203.912
9	A_T (m^2)	349.18	348.78	348.78	369.54	369.84	392.90	409.10
10	L_{CB} (%L)	0.883	−0.42	−0.425	−0.858	−0.858	1.99	2.91
11	Model Scale λ	50	50	50	40	40	45	50.6
12	D (m)	5.51	5.51	5.51	5.47	5.47	5.58	5.9
13	P/D	0.926	0.926	0.926	0.932	0.932	0.7885	0.75
14	$C_{0.75R}$ (m)	1.49	1.49	1.49	1.471	1.471	1.845	1.87
15	$(t/c)_{0.75R}$	0.0498	0.0498	0.0498	0.052	0.052	0.0406	0.047
16	Boss Dia. Ratio	0.225	0.225	0.225	0.172	0.172	0.18	0.18

$\Delta C_{PC} = -0.1858$, $\Delta W_C = -0.076$,

or $C_P = 0.9725$, $C_N = 1.0234$.

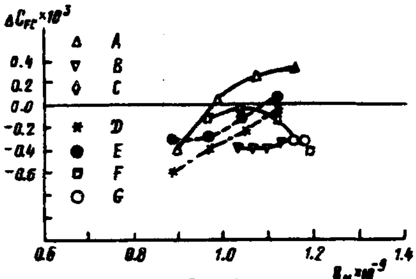

Fig 1

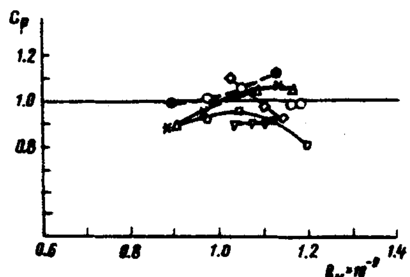

Fig 3

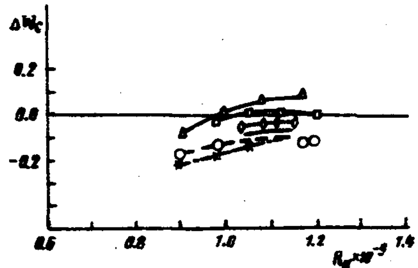

Fig 2

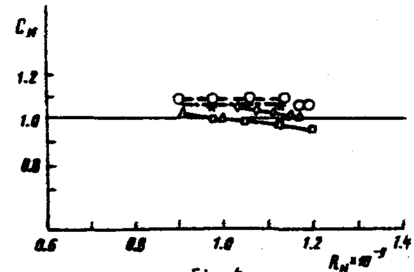

Fig 4

The individual corrections are given in Figs 1-4. It seems that the individual corrections obtained have certain discrepancies and the following data might be taken as the average values of the CTU towing tank.

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UNSTEADY PHENOMENA DURING MODEL PROPULSION TESTS OF MEDIUM SIZE BULK CARRIERS & TANKERS

Unsteady phenomena during model propulsion tests of full ship models have been investigated during the 14th and 15th ITTC. The 15th ITTC Performance Committee states that it is essential for predicting the performance of ships with high block coefficient to identify the occurrence of the phenomena and to provide adequate model-ship correlation corrections to allow for it, even if it does not reflect on a serious error in the prediction of speed or the propeller design due to the fact that design and prediction are usually concerned with full load while the unsteady phenomena occur mainly at ballast drafts.

At the Akishima Laboratory of Mitsui Engineering & Shipbuilding Co., sixty model propulsion tests were conducted in the past two years with full and high block coefficient ships which were built at various ship yards in the world and were in operation already. Among these ships, the unsteady phenomena were observed in seven cases which were all at ballast conditions of medium size bulk carriers or tankers with moderate fullness except one ULCC. As already seen, the unsteady phenomena have scarcely observed on extremely full ships and at lighter drafts which do not exceed fifty percent of full displacement.

The occurrences of the unsteady phenomena on the mentioned seven ships are shown in Fig.1 in the form of effective wake analyzed from model propulsion test results. Non-dimensional characteristics of the model ships are shown in Table 1. In this figure, Model Ship Nos. 1121, 1131, 1154 and 1231 indicate the typical case in which two stable flow fields are often seen during a single run of the propulsion test and the analyzed effective wake is expressed by two independent mean-lines with respect to the Froude Number. On the other hand, in cases of Model Ship Nos. 1097, 1129 and 1134 the plots of effective wake were scattered around a single mean-line. In these cases, the measured thrust, torque and the revolution speed of propeller were continuously being changed, and it was difficult to obtain a consistent value of effective wake. These phenomena were confirmed in several cases by conducting side force measurements and flow observations on the surface of water at stern of a model ship during their propulsion tests.

In order to reply the question that the unsteady phenomena were definitely observed on the seven particular ships among the sixty full and high block coefficient ships, the hull geometries, lines, draft conditions relative to full load, relative propeller diameters to hull geometries, model scales and temperatures at testing etc. were investigated. In consequence, there were little informations obtained concerning the unsteady phenomena. However, in fact the phenomena scarcely occur in ULCC class ships above 200,000 DWT, but are observed but in medium size bulk carriers or tankers, and the hull geometries and lines are not much different each other between ULCCs and medium size ships. On the other hand, the model scale is large enough because a 200 mm diameter propeller is used and the model length is selected corresponding to the scale ratio of propeller at the standard propulsion tests of the Akishima Laboratory. It seems that the occurrence of the unsteady phenomena and the difference in the type of flow field do not directly relate hull geometries, it may be reflected by the history of the flow into the propeller disk plane. In this point of view, detailed analytical approaches should be followed to make clear the phenomena on the particular ships.

As regards the seven ships shown in Fig. 1, propulsion tests with the Mitsui Integrated Duct Propeller (1) were conducted simultaneously. In case when the duct was fitted, the unsteady phenomena were completely removed and a row of steady plots of effective wake was obtained. The mean-line shown in Fig.1 is drawn in the similar tendency with the case when the duct was fitted. Supposing that the test results with the duct were not available, at was difficult to obtain the mean-line shown in Fig.1 An example of effective wake comparing between the hulls with and without the duct is shown in Fig.2 in case of Model Ship No. 1131.

In our experience, the higher mean-line of effective wake shown in Fig.1 will indicate a reliable trend of model effective wake to obtain the model-ship correlation correction. Fig.3 shows $(1-w_{TM})$ at the design speed as well as the band of scatter of $(1 - w_{TM})$, and $\Delta(1 - w)$ which denotes the difference between $(1 - w_{TS})$ obtained at sea trials and $(1 - w_{TM})$. The results of both model propulsion tests and sea trials were carefully analyzed in the standard method closely simi ar with the 15th ITTC Prediction Method. Fig.3 gives a fairly good information about $\Delta(1 - w)$ so that it seems to show a reasonable plot with respect to the fullness coefficient at after part of ship defined by Sasajima (2). When the lower mean-line in Fig.1 is used falsely for $(1 - w_{TM})$, $\Delta(1 - w)$ moves to the point at the head of arrow to be overestimated in our experience with the ships which did not show the unsteady phenomena.

The authors reported their experiences in the unsteady phenomena during model propulsion tests in connection with the model-ship correlation correction. However, furt theoretical and experimental investigation are necessary to establish the performance prediction method when the unsteady phenomena are found in model propulsion tests with full and high block coefficien ships such as medium size bulk carriers a tankers.

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Table 1 Characteristics of Model Ships

MS No.	1097	1121	1129	1131	1134	1154	1231
Kind of ship	TANKER	TANKER	TANKER	BULK	BULK	BULK	BULK
DWT $\times 10^{-3}$	80	150	230	120	160	110	160
L/B	6.8	5.0	5.8	6.0	6.2	6.3	6.3
CB	0.81	0.78	0.83	0.81	0.83	0.83	0.83
Δ/Δ_F (*)	0.59	0.53	0.53	0.43	0.66	0.52	0.49
Model length (m)	6.8	7.8	7.3	7.8	7.7	7.6	7.5

Note: (*) Δ, Δ_F = displacement at ballast & full loads

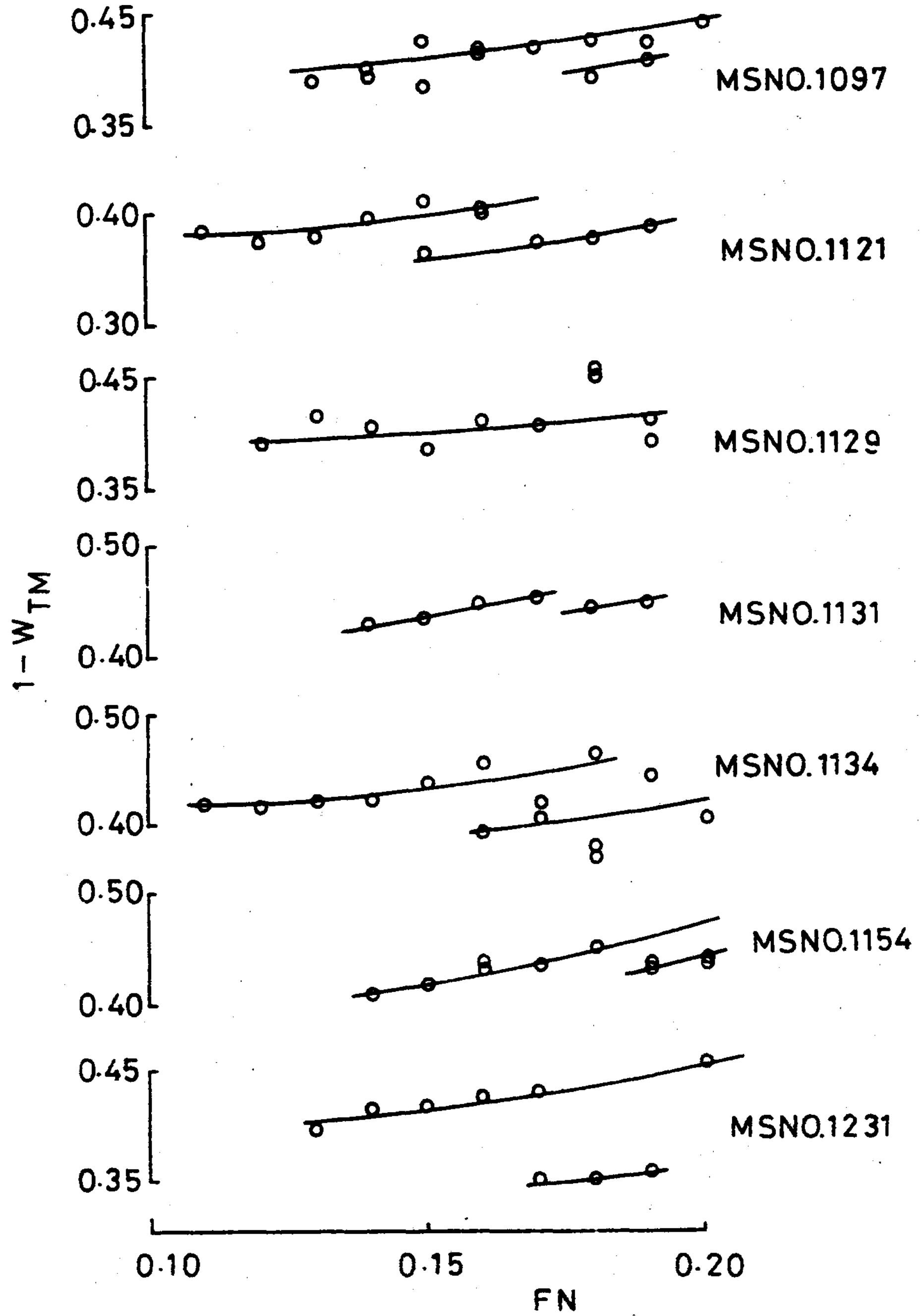

Fig. 1 Unsteady Phenomena on $1 - w_{TM}$

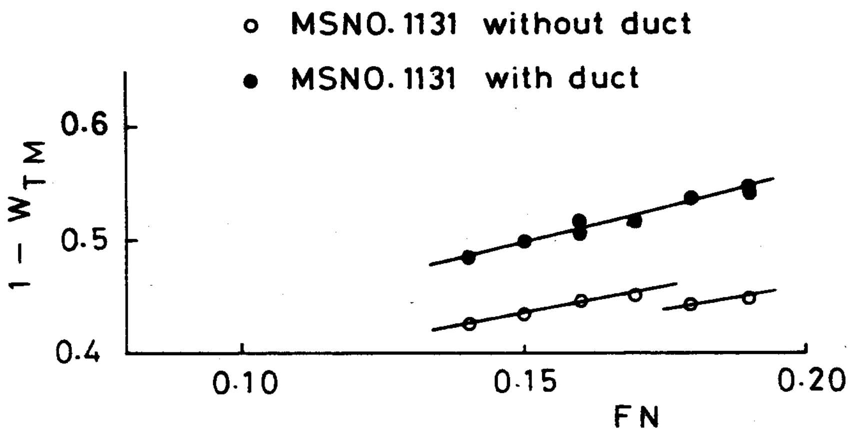

Fig. 2 Effect of the Duct on $1 - w_{TM}$

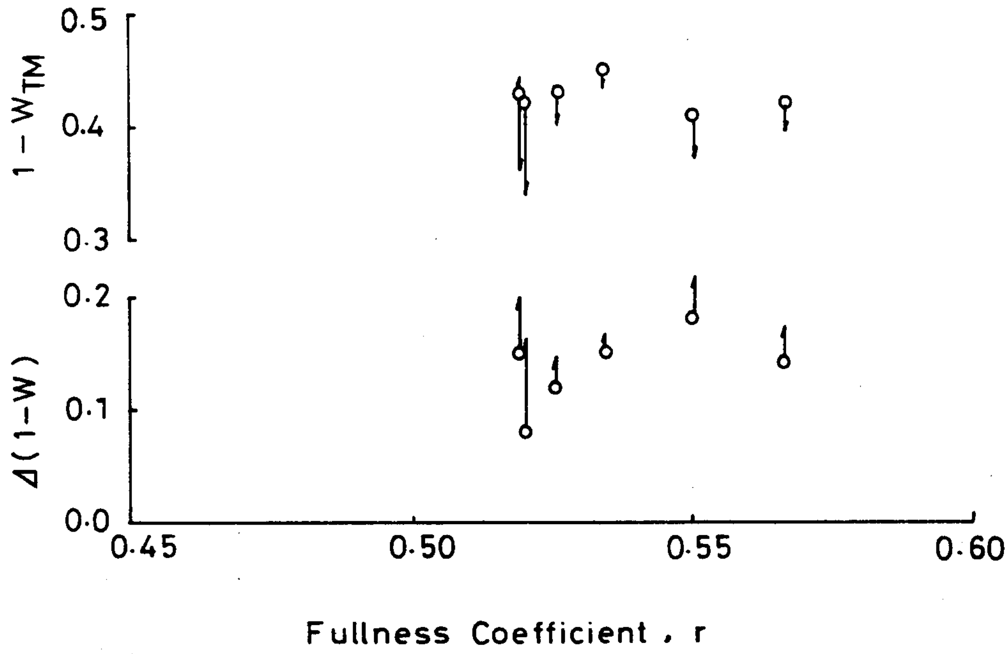

Fig. 3 $1 - w_{TM}$ & $\Delta(1 - w)$ at Design Speed

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This is a discussion about a new prediction method of the propulsive performance for a single screw ship.

Recently several prediction methods for the propulsive performance of ships have been presented based on the theoretical ship-hydrodynamics. [1] In the SHIP RESEARCH INSTITUTE, we have developed a method which can estimate the propulsive performance factors. It fixes the propulsive factors as function of propeller load, that is, it utilizes the propeller loading test concept. However, at present, the method is partly based on the experimental results in towing tank. The propeller loading test technique is a useful tool for understanding the propulsive properties of the propelled bodies from the view point if the ship-hydrodynamics.

One of the aims of our method is to estimate the thrust deduction and the wake fraction. We assumed that only the integrated averaged quantities play important roles in the propulsive performance of the ship. Then the non-uniformity of the inflow at the propeller disk is not considered. At present, we need the experimental results of the resistance test and the wake data at the propeller position in our method.

Consider a ship advancing straight with a constant speed U and with a constant propeller revolution n. The total resistance of the ship is expressed as. [2] [3]

$$R = R_{MO} + F_{RX} + T = R_C + F_{PH} + F_{RH} + F_{PW} + F_{RX}$$

where

R_C : resistance when T=0,

$T = \rho A_p \Gamma_0 (U_{a0} + \Gamma_0/2)$: propeller thrust,

$F_{PH} = \rho B_0 \Gamma_0$: resistance augmentation due to hull-propeller interaction,

$F_{RH} = \rho E_0 \Gamma_0$: resistance augmentation due to hull-rudder interaction,

$F_{PW} = \rho D_0 \Gamma_0$: wave-making resistance augmentation due to propeller loading,

F_{RX} : drag of rudder due to propeller-rudder interaction,

ρ : density of water,

$A_p = \pi D_p^2/4$: propeller disk area,

U_{a0} : averaged mean inflow velocity at propeller,

Γ_0 : strength of propeller bound vortex.

In the experiment for the loading test, $(R_{MO}+F_{Rx})$, T and F_{Rx} are measured by force measuring apparatus. F_{PW} is obtained from the wave analysis. The interaction forces F_{PH} and F_{RH} are calculated numerically by setting a boundary value problem based on the potential theory [2] [3]. F_{Rx} is also calculated theoretically with the viscous correction on the rudder [4].

In our method the thrust deduction and the wake fraction coefficients are given by the following equations.

$$t = G(t)/T \qquad (2)$$

where $G(t) = F_{PH} + F_{RH} + F_{PW}$ And,

$$1 - W = \bar{U}_{a0} + \bar{C}_0 \left(-\bar{U}_{a0} + \sqrt{C_T + \bar{U}_{a0}^2}\right) + \Delta U_A/U \qquad (3)$$

where

$$\bar{U}_{a0} = U_{a0}/U, \quad C_T = T/(1/2)\rho A_p U^2.$$

The second term of the right hand side in (3) denotes the effect of the propeller loading and the third term denotes the effect of the displacement of the rudder on the hull. According to the theory, $\Delta U_A/U$ can be written as, [4]

$$\Delta U_A/U = \varepsilon \bar{U}_{a0} + \bar{F}_0 \left(-\bar{U}_{a0} + \sqrt{C_T + \bar{U}_{a0}^2}\right) \qquad (4)$$

where ε and F_0 are constants determined numerically.

Now that, if we know the constants B_0, $\bar{C}_0$, D_0, E_0, $\bar{F}_0$ and ε , the loading test will be predicted. We need only the experimental results for R_C and the nominal wake fraction which are not easily estimated from the theory even now. Usually R_C is estimated from the resistance R_0 of the ship without propeller. It is known that R_C is larger than R_0 and the difference (R_C-R_0) is assumed to be a function of propeller diameter D_p.

We show the R-T diagram for a container ship with a rudder in Fig.1, together with the drag of the rudder and the wave pattern resistance R_{WP} obtained by the wave analysis. The predicted curve is also shown on the same diagram. It shows fairly good agreement with the experiments. In Fig.2, the thrust deduction, the wake fraction and the trim of the ship are shown as the function of the propeller thrust. So far the change of 1-t and 1-w have been observed, but the change of the trim has not. The change in trim is remarkable.

Estimating theoretically the constants, we can predict the propulsive performance factors such as 1-t and 1-w. The R-T diagram obtained from the propeller loading test is useful to examine the validity of the proposed methods estimating the propulsive performance.

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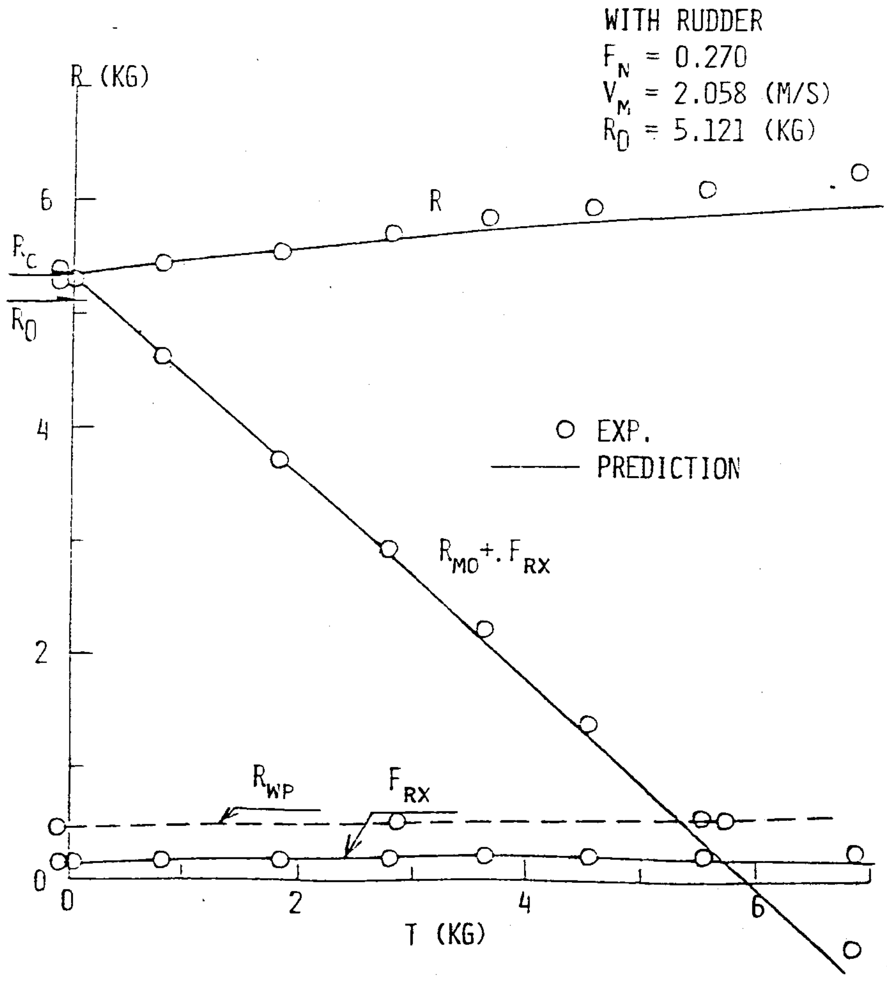

FIG. 1

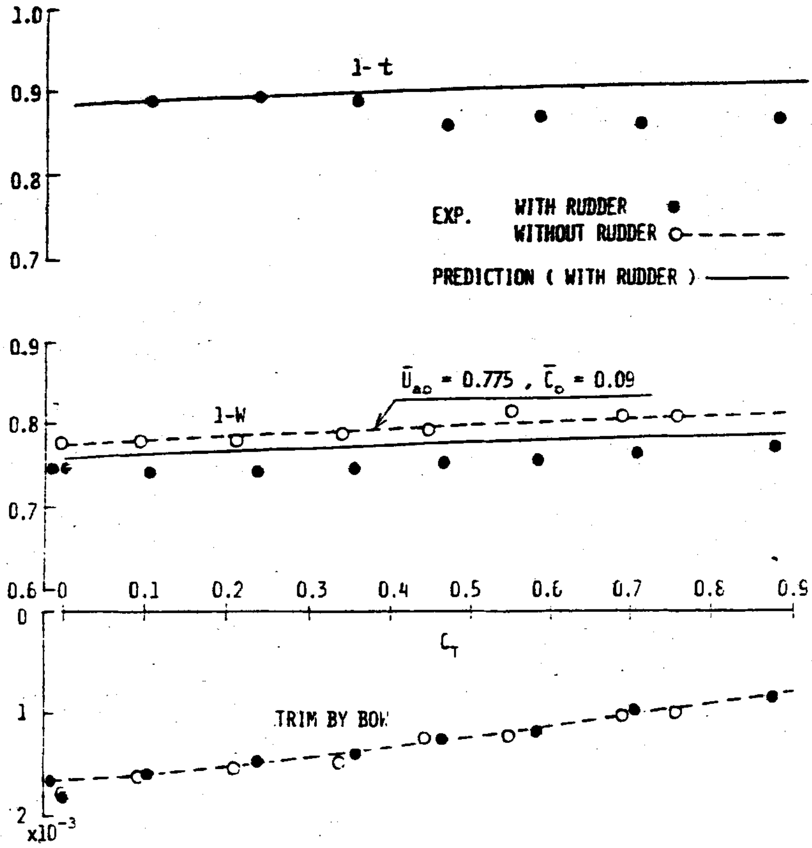

FIG. 2

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Madrid, Spain

I want to present some comments on the Committee Report:

a) With a small basin as we in ASINAVE use, scale effects and separation may be so important that resistance tests become nonsense, especially at low speeds. Therefore we prefer propulsion tests, even if the propellers are very small, and get the form factor, which will be used for extrapolation, at zero thrust.

b) I strongly endorse the 2nd recommendation, regarding propulsion tests to be carried out at more than one loading. I agree on the reasons given in the report. But my point is that, besides, there is some interest on knowing propulsion data with different loads. Either for added resistance - wind and waves - or for "deducted resistance", as motor sailing. On this regard I should like to know whether the Committee has given some thoughts to this subject.

c) Finally there are always some problems related with different Committees, I may mention one of them, which could maybe better treated in the menoeuvrability session but which is close related with propeller, load: the changes rudder forces may experience with it. I give an example from a series of tests we did some years ago.

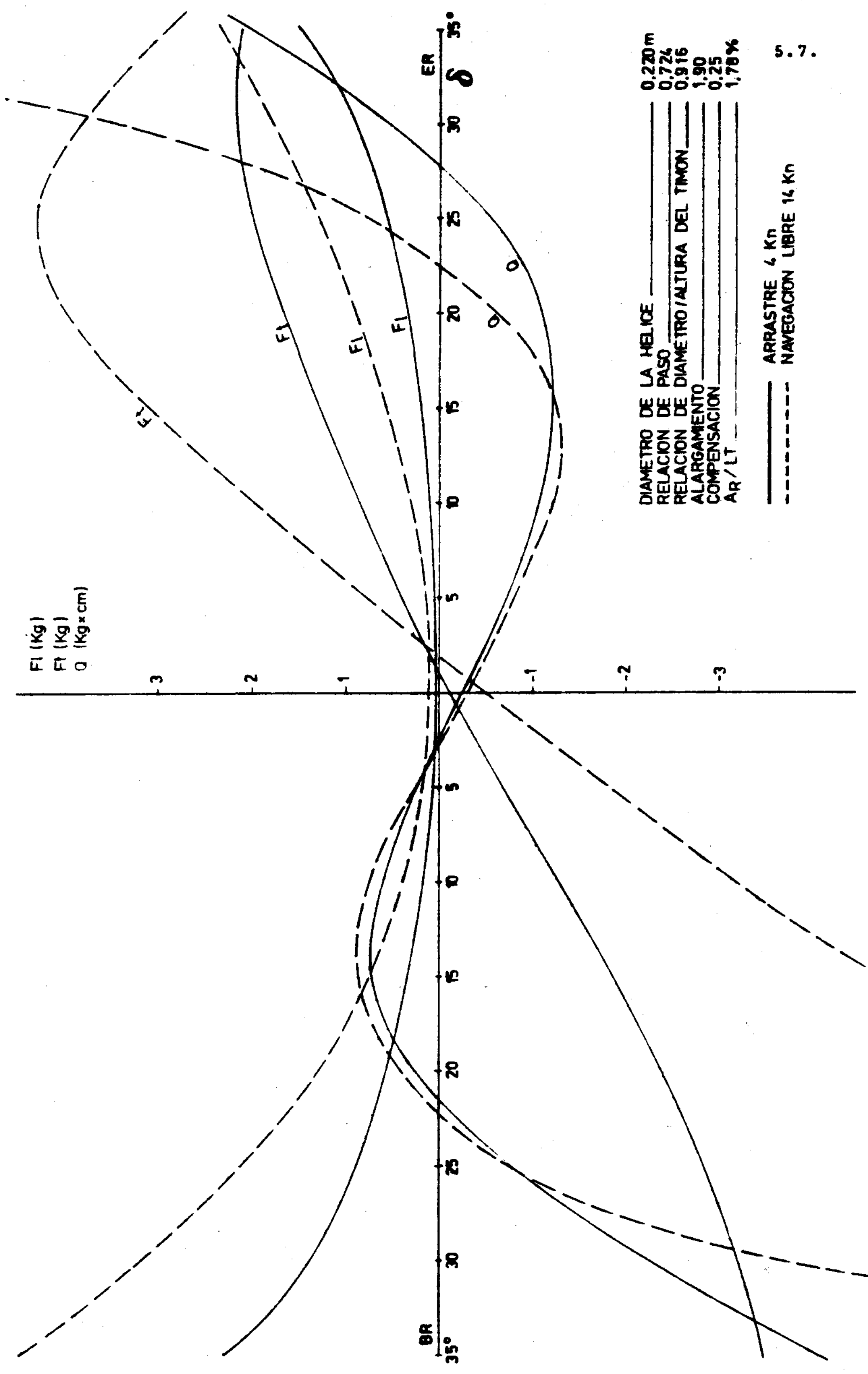
Fl (Kg)
Ft (Kg)
Q (Kg x cm)
ER
BR
35°
δ
Fl
Ft
Q
DIAMETRO DE LA HELICE 0,220 m
RELACION DE PASO 0,724
RELACION DE DIAMETRO/ALTURA DEL TIMON 0,916
ALARGAMIENTO 1,90
COMPENSACION 0,25
A_R/LT 1,78%
ARRASTRE 4 Kn
NAVEGACION LIBRE 14 Kn
5.7.

FIGURA 5.4.-

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A PROCEDURE FOR RESISTANCE, PROPULSION AND MANOEUVRING TESTING OF SCALED MODELS - AN APPLICATION OF FRACTIONAL ANALYSIS-

A model of a phenomenon represents a schematization of the true occurrences. The degree to which this is the case depends on the general design and purpose of the analysis and on the completeness and accuracy required in the solution. In the case of testing of scaled physical models the rigour of the analysis depends on the degree of satisfaction of the laws of similitude. Within the confines of the classical similitude theory the requirements are the fulfilment of geometrical, kinematic and dynamic similarity between the scaled model and the prototype. To achieve this end a dimensional analysis is carried out in terms of important protype and environment parameters by using Buckingham's II Theorem and the resulting non-dimensional parameters for the scaled model and the protype are made identical.

Although the application of classical analysis appears fairly easy, the method of implementation may bring appreciable differences both in the overall testing strategy and in the interpretation of results. Following Schuring [1] the methods available for the derivation of similarity relationships may be considered in three groups, depending on the degree of rigour of the analysis (see Table 1). They are:

(1) Use of postulated relationships.
(2) Use of force ratios.
(3) Use of governing equations and boundary/initial conditions.

In the first case a functional relationship between the required quantity, and the prototype and environment parameters (which are assumed to be important) is postulated as

$$R = f(P_1, P_2, \ldots, P_n)$$

where R is the quantity to be determined and P_1, $P_2 \ldots, P_n$ are parameters which may be primary or secondary quantities, cf. Langhaar [2] . The use of force ratios is the second alternative where the kind of forces relevant to the problem are considered and written in terms of their dimensional equivalents, and ratios of these forces yield the non-dimensional parameters. Ratios of common forces in the field of fluid mechanics are presented in Table 2 as an example of its application (borrowed from Kline [3]).

The third option makes use of the considerable information inherent in the governing equations and the initial and/or boundary conditions in an explicit manner. Here, however, certain modifications of the classical dimensional analysis and similitude theory becomes necessary. However, before studying the importance of this third choice it is worth examining the shortcomings of the first two approaches.

When a relationship is postulated in terms of system parameters the validity of such a relationship depends very much on the foresight of the investigator and consequently some redundant parameters may be included while some important parameters may be left out. As a result non-dimensional parameters derived in this way may not achieve the actual similarity. The use of force ratios reduces the degree of arbitrariness in the previous approach. However, it cannot colve the problem completely because some parameters, such as the ratio of specific heats cannot be expressed as force ratios and some forces, such as forces arising from viscous-inviscid interaction, are not easy to express in equivalent non-dimensional form. An example of a possible failure is the use of the Reynolds number in turbulent flows. It is known that apart from the wall region the role played by the kinematic viscosity of the fluid is negligible. It is therefore meaningful to ask whether the eddy viscosity should be used in the definition of the Reynolds number in a turbulent flow field and it is also worthwhile to remember that, contrary to the kinematic viscosity, eddy viscosity is not a fluid property and it varies within a given turbulent boundary layer depending on location. Furthermore, as shown by Huntley [4] , in deriving dimensional equivalences one length scale may not be sufficient. Huntley points out that the lengths in three coordinate directions have different roles similar to those of the components of a vector and cannot be used to cancel each other in a dimensional analysis.

In addition to the possible shortcomings mentioned above, in a number of physical scaled model tests not all the derived non-dimensional parameters can be made the same for the scaled model and the prototype. Pankhurst [5] states that "when only some of the similarity conditions can be satisfied simultaneously in the scale-model experiment, one must proceed by varying chosen groups of the dimensionless parameters independently of the others, and appeal to other sources for information about the degree to which the effects of the variations can be regarded as independent: available sources may be theoretical analyses, previous experience or - if all else fails - unashamed but frankly acknowledged extrapolation".

The accepted method of testing a scaled ship-model for resistance and performance prediction, as will be discussed later, is based on the use of a postulated relationship and can only achieve partial similarity since both the Froude and the Reynolds number for the model and the ship cannot be made the same in a towing tank. As a result both extrapolation and empirical correction become necessary, cf. Moor [6] . The aim of this note is to exploit the advantages offered by the use of governing equations and boundary/initial conditions in deriving the rules of similarity and to offer an alternative ship model testing procedure for resistance and performance prediction which will, at least, reduce the shortcomings apparent in the traditional testing procedure.

The term "fractional analysis" is used to indicate that in many science and engineering problems one cannot obtain complete solutions and has to be content with a partial or fractional analysis. It is a procedure which employs analytical and physical information, in addition to the usual similitude theory and dimensional analysis, and seeks to obtain information to provide approximate equations from which an approximate analytical or numerical solution can be obtained. In the light of this explanation it can be said that the use of postulated relationships and force ratios are inferior forms of fractional analysis.

However, in order to obtain the maximum advantage of fractional analysis by the use of governing equations and initial (and/or boundary) conditions the classical similitude understanding needs to be generalised. This generalisation is provided by Kline's postulate of similitude [3] which states that:

"If two systems obey the same set of governing equations and conditions and if the values of all parameters in these equations and conditions are made the same, then the two systems must exhibit similar behaviour provided only that a unique solution to the set of equations and conditions exists".

In employing a fractional analysis scheme it is important to appreciate that the more complete and detailed the governing equations and the side conditions the more information can be obtained. These equations and conditions must be sufficient to provide a solution (preferably a unique solution) to the problem. Another important part of the fractional analysis is the need to have a clear understanding of the physical information inherent in governing equations and of the mathematical models of physical events. The importance of this latter issue cannot be over-emphasised as without this understanding the rest of the analysis will be nothing more than a purely mathematical exercise.

Having satisfied the conditions of applicability of Kline's postulate of similitude, the first step in the fractional analysis is transformation of equations and side conditions in such a way that both the dependent and the independent variables occurring in the governing equations and side conditions are made non-dimensional and normalised. An example of the transformation procedure can be illustrated with the linearised free surface condition of the wave resistance theory which is given by

$$\frac{\partial^2 \varphi}{\partial x^2} + \frac{g}{V^2} \frac{\partial \varphi}{\partial z} = 0 \qquad (1)$$

where φ is the velocity potential, x is the longitudinal coordinate, z is the vertical coordinate, V is the forward speed of the ship and g is the gravitational acceleration. Introducing L as the ship length, T as the ship's draught, and V^* and λ as the wave making velocity and length respectively, such that

$$|\varphi|/(V^*, \lambda) \leq 1$$

and defining the new normalised independent and dependent variables as

$$\xi = x/L, \; \zeta = z/T, \; \phi = \varphi/(V^* \lambda)$$

equation (1) can be written as

$$\frac{V^*\lambda}{L^2} \frac{\partial^2 \phi}{\partial \xi^2} + \frac{g}{V^2} \frac{V^*\lambda}{T} \frac{\partial \phi}{\partial \zeta} = 0$$

which after dividing the equation with the coefficient of the first term reduces to

$$\frac{\partial^2 \phi}{\partial \xi^2} + \frac{L/T}{Fn^2} \frac{\partial \phi}{\partial \zeta} = 0 \qquad (2)$$

where $F_n = V/\sqrt{gL}$ is the Froude number.

The advantage of the transformation is self-evident from the form of equation (2) as it now provides a guidance on the order of magnitude of each term in terms of L/T and Fn.

The transformed equations can then be used to reduce the computation if a solution can be obtained, or to determine the form of the approximate solution if a direct solution is not possible. If the governing equations and side conditions are complete and appropriate the parameters found by the transformation process form a necessary and sufficient set for the scaled modelling. So far as this note is concerned the main interest is in the latter use of the fractional analysis and this is the subject of the next section in relation to resistance and propulsion testing.

Testing the scaled ship models for resistance and performance predictions is based on an assumed relationship that the ship resistance is composed of two components: one is determined by the viscosity and the other is determined by the gravity. Hence the resistance can be expressed as [2,7] :

$$R = R_v + R_w = C_v(Rn)\frac{\rho S V^2}{2} + C_w(\psi, Fn)\rho g \nabla \qquad (3)$$

where the subscripts v and w represent , respectively, the viscous and wave making contributions, ρ is the density of water, g is the gravitational acceleration, S is the wetted surface area, ∇ is the displacement volume, V is the velocity of the model, ψ is a form parameter, and Rn and Fn are Reynolds and Froude numbers, respectively.

Initially $C_V(Rn)$ was considered to be the frictional component and the wave component C_W was called the residuary resistance coefficient. Further experience indicated that it would be better to express C_V as

$$C_V = C_f + C_F \qquad (4)$$

where C_f is the skin friction coefficient and C_F is the form drag coefficient.

Since however both C_F and C_W are functions of both Reynolds and Froude numbers, it became necessary to introduce correction factors for full-scale predictions, cf. Gross and Watanabe [8] . Similar difficulties are also encountered in the prediction of propulsion factors from scaled model tests and the ITTC 1978 extrapolation procedure [9] is a typical example of such correction methods.

When viewed in the light if fractional analysis, it can be noticed that the inconveniencies and drawbacks created by the assumption expressed in equation (3) need not be made and a more plausible testing technique can be obtained by the use of the governing equations and the Kline's postulate of similitude.

Let us consider the flow field around a ship's hull (see Fig.1). As a first approximation this field can be considered as an interacting viscous-inviscid flow problem. In the viscous field, the effect of Reynolds number appears in the development of boundary layer through its thickness and in the viscous sub-layer (see Fig.2). It is known from the boundary layer measurements that the pressure variation within the viscous sub-layer is negligibly small. Therefore, if the outer parts of the boundary layer of the scaled model and the ship behaved in the same way, both the form resistance and the wave resistance coefficients would be the same. This can only be achieved if the edge surface conditions of the boundary layer for the model and the ship are the same since this will imply self-similar boundary conditions for the boundary layer development.

It is known from the boundary layer theory that if the body surface is replaced by a pseudo-body surface by adding the boundary layer displacement thickness δ^* the flow outside the boundary layer can be recovered by means of a potential flow calculation. Therefore if one wishes to distort the scale model of the ship to obtain similarity for the boundary conditions at the edge of the boundary layer, the geometric similarity between pseudo-bodies of the ship and the scaled model can provide a first approximation. Figure 3 illustrates the distribution of displacement thickness on the scaled model of a ship. Using this first approximate distribution, similarity in boundary layer edge conditions can be obtained by an iterative scheme and the converged solution yields the deformed hull shape which satisfies the Kline's postulate of similitude.

Achievement of similarity in the governing equations and the boundary conditions in model testing therefore ensures that only the contribution from the skin friction needs to be extrapolated. The advantages gained by this approach are not limited to the resistance prediction. As the proposed procedure will also ensure the boundary layer similarity, apart from a narrow region around the wake centre plane (thickness $\approx Rn^{-\frac{1}{2}} L$), the scaled model and the full scale wakes will be similar and therefore the model propeller will operate in a flow field quite close to the full scale. Hence the need for scaling the propulsion factor will be removed. Additionally, the flow into the rudder and secondary flow effects in manoeuvring will be much closer to their full scale counterparts and consequently predictions obtained by model testing will become more reliable.

Discussion presented in this note indicates that advances in the theory of similitude make it possible to devise model testing procedures which will require less restrictive assumptions than those employed in the present testing methods. The use of the procedure proposed in this note will reduce the amount of extrapolation needed for the full scale predictions to a minimum for resistance, performance and manoeuvring estimates.

Since many organisations now have a three-dimensional boundary layer calculation program, the determination of a displacement thickness distribution for a first approximation to the pseudo-body is readily achieved. The program suite GEMAK [10] developed at BSRA already employs the pseudo-body concept in viscous-inviscid interaction by a blow-out procedure. Inverted use of this methodology will suffice for the determination of the deformed scaled model hull form.

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Table 1

LEVELS OF APPROXIMATE MODELLING

(1) Postulated Relationships

$R = f\,(P_1, P_2, \ldots, P_n)$

Aids: Classical Dimensional Analysis.
Huntley's Extension.
Staicu's General Dimensional Analysis.
Information-Theoretical Methods.

Outcome: Global Approximations.

(2) Governing Laws for Proncipal Contributors (Force Ratios)

Aids: Generalised Dimensional Analysis.
Theory of Similitude.
Lie Algebra.
Information-Theoretical Methods.

Outcome: Procedure for Design and Analysis of Scaled Model Tests.
Extrapolation Procedures for Model Data.
Refined Approximations.

(3) Governing Equations and Boundary and/or Initial Conditions (Fractional Analysis)

Aids: Generalised Dimensional Analysis.
Fractional Analysis.
Method of Trial Solution.

Information-Theoretical Methods.

Outcome: High Level Approximations.
Extrapolation Procedures for Model Data.
Procedure for Design and Analysis of Advanced Scaled Model Test.

Table 2

RATIOS OF COMMON FORCES IN FLUID MECHANICS

$F_V \hat{=} \mu VL$	$F_P \hat{=} \Delta p L^2$	$F_C \hat{=} E_s L^2$	$F_S \hat{=} SL$	$F_G \hat{=} \rho L^3 g$	
$\frac{F_I}{F_V} = \frac{\rho VL}{\mu}$ Reynolds number	$\frac{F_P}{F_I} = \frac{\Delta p}{\frac{1}{2}\rho V^2}$ Pressure coefficient	$\frac{F_I}{F_C} = \frac{V^2}{\frac{E_s}{\rho}}$ Cauchy number	$\frac{F_I}{F_S} = \frac{\rho V^2 L}{S}$ Weber number	$\frac{F_I}{F_G} = \frac{V^2}{gL}$ Froude number	$F_I \hat{=} \rho V^2 L^2$
	$\frac{F_P}{F_V} = \frac{\Delta p L}{\mu V}$ Stokes number	$\frac{F_C}{F_V} = \frac{E_s L}{\mu V}$	$\frac{F_S}{F_V} = \frac{S}{\mu V}$	$\frac{F_G}{F_V} = \frac{\rho L^2 g}{\mu V}$	F_V
		$\frac{F_C}{F_P} = \frac{E_s}{\Delta p}$	$\frac{F_S}{F_P} = \frac{S}{\Delta p L}$	$\frac{F_G}{F_P} = \frac{\rho L g}{\Delta p}$	F_P
			$\frac{F_S}{F_C} = \frac{S}{E_s L}$	$\frac{F_G}{F_C} = \frac{\rho L g}{E_s}$	F_C
				$\frac{F_G}{F_S} = \frac{\rho L^2 g}{S}$	F_S

The six forces most often encountered in fluid flow:

Inertia forces	$\hat{=} F_I$
Viscous forces	$\hat{=} F_V$
Pressure forces	$\hat{=} F_P$
Compressive forces	$\hat{=} F_C$
Surface tension forces	$\hat{=} F_S$
Gravity forces	$\hat{=} F_G$

These forces may be expressed dimensionally:

$F_I = \rho V^2 L^2$

$F_V = \mu V L$

$F_P = \Delta p L^2$

$F_C = E_s L^2$

$F_S = SL$

$F_G = \rho L^3 g$

where

ρ = mass density
L = length
V = velocity
μ = viscosity
E_S = issentropic bulk modulus of compression
S = coefficient of surface tension
g = local acceleration of gravity

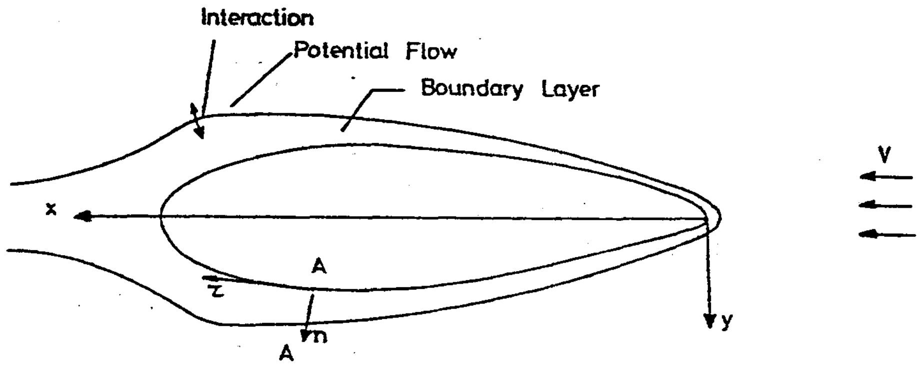

FIGURE 1 SCHEMATICAL ILLUSTRATION OF THE FLOW AROUND A SHIP

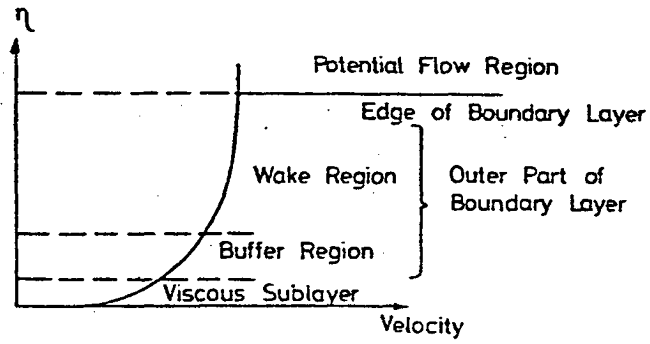

FIGURE 2 FLOW REGIMES IN SECTION A-A

G.COLLATZ - Hamburg Ship Model Basin,
Federal Republic of Germany

My contribution refers to chapter 10 of the report: Separation. We were informed about flow visualisation tests with a tanker model during propulsion tests using a right handed propeller. The flow became asymmetric due to the propeller action and two kinds of asymmetric flow have been observed, one with distinct flow separation on the port side, the other at which separation occurs mostly on starboard side. In the latter case thrust and torque became comparatively smaller.

Here arises the question whether we can take advantage from this phenomenon by forming the stern suitably . In other words, can we improve the vessels performance by designing the aft body asymmetrically and with that predescribe on which side separation shall occur?

Recently a series of model tests was carried out at the Hamburgische Schiffbau Versuchsanstalt with two different container ship models, one having a rather small block coefficient, the other model was of moderate fullness. Both models were first tested with conventional symmetric aft bodies. Then the aft bodies were replaced by asymmetric ones and the models retested. Resistance, propulsion and wake measurement were done at two different drafts.

In all cases the results were very promising. At full load draft a power reduction between 5 and 9% was measured. At ballast condition the results were less good but still better than with the symmetric sterns.

As a result, a German shipyard is now building a container ship (of medium size) with such an asymmetric aft body. Needless to say that I am anxiously looking for the vessels' trial trip. Because still one question is open these good results will be confirmed in full scale.

S.D.SHARMA - Institut für Schiffbau,
Hamburg, Federal Republic of Germany

My comment concerns not the content of the Performance Committee Report but the logical structure of its Recommendations. According to my experience, in the past it has been a source of considerable confusion whether the Recommendations traditionally appended to every ITTC Committee Report were recommendations of the Committee to the Conference or vice-versa. So the present Performance Committee should be commended for taking the lead in trying to sort out this confusion. Apparently, it is the only Technical Committee of the 16th ITTC that makes a logical distinction between recommendations to member organizations and recommendations for the future work of the committee itself. However, I do not quite agree that its Recommendations Nos. 8 and 10 shouls be addressed to the next Committee. These are calls for additional research and should therefore be properly directed to the entire scientific community. The Committees as such can only be expected to compile, collate or review the existing material and not to carry out specific research to fill in the gaps of existing knowledge. To sum up then, all Technical Committees should be encouraged to distinguish clearly between:

(i) Recommendations to the Towing Tanks about the routine application of securely established techniques,

(ii) Recommendations to the future Committees about the compilation and review of scattered and/or controversial information of recent origin, and

(iii) Recommendations to the scientific community at large about the need for specific research to produce urgently required new knowledge of importance to the ITTC.

S.MARSICH - Genoa University, Italy

In applying the modified Prohaska method I found in many cases that the determination of the n exponent presented significant differences, if the absolute or the relative error is minimized.

It is my opinion that it will be correct to minimize the relative error and my thinking is confirmed by the more close results obtained in this case in the successive procedure for the interpolation and the fairing of the experimental data.

Furthermore in this way the field of the useful data can be enlarged to Froude numbers larger than 0.20 including points with accidental errors of more little weight.

But the scope of this intervention is mainly to point out this ambiguity and suggest to the Committee the opportunity to specify what kind of error must be minimized.

B.NIZERY - Institute de Recherches de la Construction Navale, Paris, France

I have only a short comment about the Performance Committee Report.

I am very happy that the Committee has pointed out the importance of sea trials and analysis methods. I would like to emphasize the necessity of having reliable data about sea trials and making a thorough analysis of the measurements in order to eliminate the unrelevant figures and to correct them when it is possible due to the continuous variation of the different parameters.

D.LACKENBY - British Ship Research Association, United Kingdom

Just a brief comment: in spite of the further work done over the past three years, there has not been a very substantial reduction in the average dispersion of the power prediction factors. We are still left with a dispersion of the order of $\pm$ 6 or 7% in C_p which is on the large side from the commercial point of view.

In view of the known importance of hull roughness on resistance, I wondered whether some impression might be made on this by taking detailed account in the trials analysis of the hull roughness for the individual ships instead of assuming average values of 150 microns for the mean apparent amplitude. If there is some justification for using an average value from limited sample of data, it may not apply generally.

On a more general matter: in the light of discussions in previous sessions, this question of the accuracy of the prediction of power from model tests is the very essence of ITTC work and one of the reasons why the organization was set up. It is clearly of considerable importance and it is hoped that this work will receive all the support it can be given.

R.B.COUCH - Department of Naval Architecture, University of Michigan, Ann Arbor, Michigan, USA

I would like to receive the answers from the Committee on the following questions:

1. Has the Committee found any experience

with transient conditions in full scale trials?

2. Obviously Trial and Model results must be used in large numbers to obtain statistical data. If only full scale trial data in perfect weather and ships fitted with torsionmeters and thrustmeters and models retested with exactly the same propeller, hull form, and drafts are used in the analysis, how many are available.

P. O'DOGHERTY - Canal de Experiencias, EL PARDO, Spain

I want first to thank the Performance Committee for the excellent report they have presented and the large number of full scale data that have been analysed in order to investigate the C_P and C_N values, obtained in the 1978 - ITTC procedure.

As Mr. Minsaas has very rightly pointed out, there is a correspondence between the ΔC_F values that are required (or the C_P, C_N values for that matter) and the number of simplifying assumptions that are made.

Referring to form factors, we are conducting at El Pardo a systematic analysis of form factors for all model hulls that are tested, having found the following:

- The n exponent in formula 2.2, for Prohaska revised method, is very variable.
- The 1 + k values for single-screw ships with bulbous bows are generally somehow larger than those for vessels without bulbous bow, although a reduction in total resistance is obtained at the design speed range, when the bulbous bow is properly designed.
- It should be convenient in many cases to establish a Fn dependence of 1+k, specially for finer ships with bulbous bow and at light draughts.
- For fine vessels, the concept of factor has a very doubtful meaning, requiring much research before it becomes applicable with some reliability, for accurate power prediction.

For the extrapolation of propeller open water characteristics, depending on scale effects, it is felt that more efforts should be made in order to obtain reliable open water results at full scale,as the extrapolation to full size may entail great departures that will produce undue increases in the corresponding C_P, C_N values.

I fully agree with Prof. van Manen's remarks, stressing the importance of a proper evaluation of propeller-hull interaction. In this context, propeller interaction is largely influenced by flow conditions astern, so that great care should be exercised in avoiding flow separation phenomena, which will appear when the exit angle of the waterlines is somehow large, giving rise to a pressure reduction and hence boundary layer separation. In this respect, the slope of the buttock lines aft must be considered very carefully.

In the matter of correlation based on either thrust or torque identity, although well aware of the theoretical convenience of thrust identity methods, I feel that very few reliable thrust measurements are made at full scale to support the practicity of using thrust-identity correlation.

More efforts should be made, in order to render a routine practice the measurement of thrust at sea trials. In the meantime, we feel more convenient to base the model-ship correlation on the comparison of torque values that are measured in a reliable way, both at the model and full size ranges.

At El Pardo Tank, the correlation procedure for single-screw ships is based on the

1957 : ITTC frictional line, assuming, as the 1978 - ITTC correlation method, no scale effects on t or ℓ , having obtained from analyzing more than 100 full size sea trials results, a correlation allowance Ca that is defined as
C_A=f(LBP, B/Tm, C_b.B/LBP, Δ , Beaufort number)

This method provides a good agreement with full size sea trials results, with small average C_P, C_N values, using only normal assumptions as to the wake scale effect.

On the matter of ships service performance, we do not support the idea of obtaining service power from the fuel consumption, as it is well known that in the case of propeller overloading - what is usual after some years of service, or in bad weather or when the hull is heavily fouled - the specific fuel consumption per HP/Hr increases clearly, sometimes as much as 15%.

B.MÜLLER-GRAF - Versuchsanstalt für Wasserbau and Schiffbau.

1. In chapter 6 of the report concerning the performance Prediction for high speed craft, the Performance Committee considers High Speed Craft to be those which operate at speeds exceeding the "hump speed".

Why does the Committee not consider the semi-displacement or the semi-planing hull as a high speed craft?

This type of vessel runs above the hump speed and cannot be treated like a slow single screw displacement ship.

2. Relating to the resistance of a hydrofoil craft, the Performance Committee considers on Page 153 the foils to be parts of the appendages. Therefore the statement is made that the appendage drag is the main part and in the foilborn condition the total part of the resistance of the craft. This is not true.

As shown in Fig. 17 of the report of the High Speed Marine Vehicle Panel the main resistance component in the foilborn condition is the drag due to lift, the induced drag which amounts to nearly 60 percent of the total drag. The appendage drag, caused by propeller shafts, pods and shaft barrels reaches a value of nearly 8 percent and the drag of the non lift generating struts D_{struts} amounts to about 15 percent. The wave drag of the struts D_w or R_w is very small and disappears at $F_{nc} > 1$. Therefore an appendage drag with a value of 65 percent of the total resistance as given in Table No.2 of the report does not occur at hydrofoil craft especially with semi-submerged foils. I suppose to change the value of R_{lift} with the value of R_{AP} and R_V. R_V should be correctly replaced by D_p the profile or section drag according to the proposals given by the High Speed Marine Vehicle Panel.

3. Finally I am missing a comment of the Performance Committee on the procedures which are taking into account the effects of oblique propeller inflow conditions on the propulsive coefficient as described in the Report of the Cavitation Committee. These effects are also occurring at speeds near F_n = 0.5 due to the increased running trim.

I hope, this procedure will be recommended in near future as a standard procedure in calculating the wake fraction, the thrust deduction fraction and the propeller efficiency of high speed semi-displacement round bilge hulls and planing craft.

K.R.SUHRBIER - Vosper Thornycroft (UK), Ltd., Portsmouth, United Kingdom

I would like to congratulate the Committee on their work and excellent report. I was quite interested on their comments on the resistance increase due to hull roughness.

Although the ITTC 1978 expression, Formula 3.2 in the Report, for the allowance contains, as stated, other than roughness effects, it appears nevertheless as a formula for the change of skin friction due to roughness. This may be misleading, and it seems to me that this formulation ought to be looked at again. It can - if used for assessing power increase due to (paint) roughness - in my opinion overestimate the effect of roughness quite considerably. Other approaches seemed to give better results when applied to trials analysis.

I feel also that it would have been better if the designation C_A, adopted previously, had been kept for Formula 3.2.

LAZAROV - Bulgarian Ship Hydrodynamics Centre, Varna, Bulgaria

I would like to congratulate the Committee for the impressive and valuable work done by it. I quite agree with the conclusions and recommendations given, particularly the ones concerned with twin screw ships. In the aspect of recommendation 2 on p. 183, although it is time consuming, very often in the shallow water towing tank at BSHC, we are forced to do tests with different loadings with an aim of getting more reliable test data scaling them to full scale ship. I would like to suggest to the Committee, that a recommendation should be included, that more investigations and information are needed for ship performance in shallow and confined waters. This concerns not only ship resistance and propulsion, but the squat, as well, which in many cases can be dangerous for depth/draft ratio $H/T < 1.5$.

The other goal of such investigations is also to provide data for designers of inland displacement vessels, particularly tugs, pushboats, semi-tunnel limited draft ships, etc.

I realize that the problem is rather complex, but nevertheless, I consider that it is important and should be given attention by the Conference.

II. REPLY OF THE PERFORMANCE COMMITTEE

The Committee thank *Prof. Aertssen* very much for his valuable contribution on the service margin of ships. We fully agree with his statement that the measurement of service performance on instrumented ships is preferable. However, as pointed out in the Committee Report, the amount of data obtained from such instrumented ships is still limited (although he kindly refers to some data already published) and the Committee think it is also important to deal with analysis methods based on log books.

The use of fuel consumption, instead of engine power, is suggested in the Report only as an alternative when measurements or estimates of engine power are not avalaible. Fuel consumption is determined every day on board and is easily available from log books.

We also agree with Prof. Aertssen that performance in mild weather (up to Beaufort 5) is most important from the point of view of service margin. Further study of the seakeeping qualities of ships in such mild weather is required. As far as voluntary speed loss in severe head seas in concerned, we strongly support the efforts of the Seakeeping Committee to obtain limiting criteria.

Prof. Roussetsky reports the very comprehensive data on service performance derived from more than 250 single-screw ships. As stated in the Committee Report, the Committee consider it important that analysis and prediction methods for service performance should be based on a rational extension of the 1978 ITTC Method. The Committee suggest it would be of great value if Prof. Roussetsky could utilize his large data set for further analysis along these lines.

Prof. Roussetsky also gives information on the ship - model correlation of a hydrofoil ship. The model test procedure for such ships is not yet standardized, however, and many alternatives are possible even for resistance tests. For this reason, the Committee would appreciate it very much if Prof. Roussetsky could explain the model test method in more detail and give the results of the measurements.

We wish to thank *Dr. Dyne* for his comments on Chapter 9 of the Committee Report, and for the valuable addition to our summary of the preliminary version of report ref./89/ on the cooperative study on the treatment of experiment data carried out by the Danish, Swedish, Norwegiar and Finnish towing tanks. We consider the work reported in ref. /89/ of much importance to the ITTC and therefore suggest (recommendation 9) that the future work of the Committee should include evaluations of the final results of this cooperative study and of the work currently being carried out by Holtrop at the NSMB, results of which have been reported in ref./87/.

With regard to Dr. Dyne's remark on the determination of the form factor, we wish to confirm that we have misinterpreted his Trondheim paper ref./25/. We were not aware of the fact that at SSPA the results of both propulsion and resistance tests at low speeds were used to obtain the form factor from equation (8.3) of our report. Notwithstanding this misinterpretation, Holtrop's efforts to solve equation (8.3) from propulsion test results alone through regression techniques, and his conclusion that the numerical stability of the regression equation with three coefficients is sufficient for routine applications, remains of interest as are his further comments referred to in Section 8.2 of the Committee Report.

Dr. Wang's proposals for dealing with the form factors of twins-crew ships are interesting. We are pleased to learn that he intends to carry out double-model tests in a wind tunnel to evaluate his procedure and we look forward to seeing the results. However, should separate scaling of appendage resistance for a twin-screw ship become necessary in order to attain the desired extrapolation accuracy, then separate form factors for different appendage components would be more desirable.

We are grateful to *Dr. Chen* for providing details of the geosim tests carried out at the Shanghai Research Institute. The results are especially interesting in that the model tests were carried out for water temperatures covering a wide range (the temperatures varied from 6.6 to 27.8 degrees Celsius). It is encouraging to the Performance Committee that Dr. Chen has stated that the agreement between the six sets of geosim results was improved by using the form factor method.

Dr. Chen's results show that form factor varies with Froude number and decreases for values greater than 0.15. The Committee asks Dr. Chen if he would provide more details of the variation so that an assessment can be made of its significance when making predictions of ship speed and power.

The results given in the contribution show that the index of the wave resistance term, in equation 2.2 on page 139 of the Committee report, varies considerably and this is in agreement with the findings of the Committee. An index for one of the sets of results was as high as 12.39, but it should be noted that this was for the smallest model and it is possible this was influenced by some inaccuracy in the resistance measurements. Although the index showed large variations it was found that the changes in the (1+k) values were relatively small and this provides further confirmation of Hughes' statement that the value of the index has only a small effect on the magnitude of the form factor.

Dr. Chen gives an empirical equation which may be used to estimate form factor. In the equation, k is expressed as a function of geometric parameters defining the hull forms. The Committee considers that this type of equation is not very satisfactory for estimating form factors and they agree with Dr. Chen's observation that such equations are not suitable for general use.

The Committee thank *Prof. Sheng* for his interesting report on the geosim tests made at Shanghai Chiao Tung University. Prof. Sheng's tests were for geosim models of a tanker, as were those of Dr. Chen, and the Committee asks both the contributors if they would state whether the models were fitted with ram bows. Obviously this information is important when interpreting the results of the tests.

Prof. Sheng has found that (I + k) is constant up to Froude numbers of about 0.16 but it then decreases at higher Froude numbers. The variation of form factor with F_n certainly requires further examination and this should be one of the tasks of the next Performance Committee. One of the conclusions stated in the report of the present Committee is that the data used to derive the form factors should usually be limited to the F_n range 0.12 to 0.20. It will also be noted that the change in (I + k) for the results of Chiao Tung University is from about 1.22 at F_n of 0.16 to 1.12 at F_n of 0.24, a change of about ten per cent. The investigations of the Committee have indicated that a variation of this magnitude usually brings about changes in C_P of less than 2 per cent and changes in C_N of less than 0.5 per cent. This is discussed in pages 140 - 141 of the Committee report.

Prof. Sheng also notes that the index of the wave resistance term varies for the five geosim models and states that this is hardly in accordance with the laws of similarity for wave resistance. However, it will be seen from equation 2.2 on page 139 of the Committee report that the wave resistance coefficient is a function of the slope "c" as well as the index "n". It can be shown that if the index becomes larger, the F_n term in the equation will be smaller, since F_n is less than unity, and the slope term "c" will become larger. The Froude number and slope terms counterbalance each other and the wave resistance coefficient remains practically constant such that the similarity law is still valid. Furthermore, it should also be remembered that resistance measurements for form factor analyses are carried out at relatively low Froude numbers with the express intention of keeping the wave resistance small and, therefore, it may not be very meaningful to place too much emphasis on the precise magnitude of the wave resistance coefficients obtained from an analysis of this kind.

In his final comment, Prof. Sheng referred to the Committee's proposal that the index should be selected such that the standard deviation is a minimum when equation 2.2 on page 139 is fitted to the experiment data. He states that a minimum standard deviation does not always exist and the Committee notes in its report that it has also encountered this problem. The Committee agrees with Prof. Sheng's conclusions that this is probably due to inaccuracies in the measured experiment data.

The Ship Hydrodynamics Laboratory of Shanghai Chiao Tung University have actively supported the work of the Performance Committee over the last 3 years, and a summary of results they obtained using the 1978 Method is given in the Report. It is interesting that the re-analysis of the trials data presented by *Prof. Sheng* has led to no changes in the mean C_P or C_N values from those previously submitted to the Committee. Prof. Sheng has provided full details of the ships and diagrams of the results and they are a valuable addition to the Committee Report.

Mr. Abe and *Mr. Kishimoto* present much valuable information on the occurrence of unsteady flow phenomena in model tests. The Performance Committee is very grateful for this information and would like to ask other member organizations of ITTC to offer similar material to the Committee whenever available.

Coming to the contents of the contribution, we would like to comment as follows:

In our opinion more study is required to justify the conclusion that the phenomena do not seem to be related directly to hull geometry. The question arises as to what causes the change in the history of flow into the plane of the propeller disc. If this is not related to the hull geometry, but to such factors as test condition (draft and trim), propeller loading and model size, there is a possibility that almost all models of this type may show the same phenomena at a patricular combination of the above factors. The reason why the phenomena do not occur in ULCC models should be investigated. It seems more likely that there is a considerable difference between the hull geometry near the stern of a ULCC and that of a medium size tanker or bulk carrier due to the difference of propeller diame draft ratios. It is noteworthy that the phenomena completely disappeared when the duct was applied to a model. However, it is not clear why it is acceptable to draw a mean line through the wake data for the model without duct on the basis of the trend derived for the same model with duct.

As far as correlation of wake fraction is concerned, the Committee cannot answer the question as to why the flow state with separation in the starboard side is correlated with that without separation. Certainly there is much still to be learned of these phenomena.

In reply to *Dr. Adachi* and *Mr. Moriyoma's* interesting contribution, we agree that one way to improve the techniques for making speed predictions may be to apply appropriate mathematical models for propeller-hull interactions in combination with model testing. We have already referred to such attempts in our report and hope that the work of Dr. Adachi and Mr. Moriyoma is another step in the right direction. It is difficult to judge their method from their contribution to this Conference, as the theory is only presented in a general way without giving many details, but we look forward to seeing more results obtained by the method.

In answer to *Prof. Mazarredo's* comment concerning the use of small models for propulsion tests, the Committee agree that the action of the propellers may modify the flow in such a way as to reduce some scale effects. However, such benefits will be offset by large scale effects on the propeller itself and the difficulty in measuring propeller shaft thrust and torque with sufficient accuracy.

The Committee agree that there are many reasons for carrying out propulsion tests at more than one loading. In addition to those mentioned in the Report and raised by Prof. Mazarredo, tests in ice-free water with the propeller overloaded are being used for comparison with model propulsion tests in ice, the difference between the two sets of results giving information on the effect of ice on propulsion coefficients. The effects of propeller loading on manoeuvring qualities are recognised and this is an area of direct concern to the Manoeuvring Committee.

Dr. Odabasi proposes a radical alternative to the procedures presently used for predicting ship performance from model experiments. The contribution is detailed and it is not possible to explore properly the possibilities of Dr. Odabasi's proposal in a brief reply.

It appears that to achieve success, the approach requires an accurate computation of the boundary layer, and the Committee question whether the calculation methods for three dimensional boundary layers are yet adequate for this task. Indeed, although Dr. Odabasi approach may also be applicable to propulsion and manoeuvring, the first step must be the prediction of ship resistance and the Committee feel that, in the first instance consideration of the proposals would be better directed to the Resistance Committee.

Dr. Collatz presents valuable additional information on the occurrence of asymmetric flow on a tanker model.It is also very interesting to learn of the tests on an asymmetric stern for a container ship in Hamburg. However, in the case of the container ship, we are not able to judge whether it makes use of the occurrence of asymmetric flow separation at the stern, or if it utilizes the pre-swirl effect of the asymmetric stern of the propeller. The Committee would welcome further information on the results for both the model and the full scale ship.

Prof. Sharma comments on the presentation of the recommendations. The Committee gave considerable though to the best way in which its recommendations could be put before the Conference. However, whether a breakdown of recommendations such as suggested by Dr.Sharma should be generally adopted is a subject which would be better discussed elsewhere during the Conference.

It is not clear from *Prof. Marsich's* discussion what is meant by absolute or relative error in form factor computations. The Committee has pointed out on p.139 of the Report that the exponent n is selected to obtain the best linearisation of the resistance data. In other words, lines are fitted to the resistance results, when expressed in the form of equation 2.2 of the Report, for various values of n and the standard deviations are calculated for each line. The (1+k) and n values are taken for the line which gives the minimum value of standard deviation.

The Committee thank *Dr. Nisery* for endorsement of their views on the collection of data from model tests and full-scale trials as expressed in recommendations 3 and 5 of the Report.

The Committee agree with *Dr. Lackenby* that there is still scope for a reduction in the dispersion of the C_p and C_N values, and its efforts have been directed to this objective.

The specific question of the improvement which might be gained by using actual hull roughness instead of the average value of 150 microns was tested on a sample of data by the 14th ITTC Performance Committee. As described in Appendix III, paragraph 3b, of their report, no improvement was obtained by this procedure. It should be remembered that hull roughness is presently defined by a single-parameter, usually mean apparent amplitude, and it is unrealistic to expect to obtain a very reliable measure of the effects on ship resistance. It is important to obtain better definitions of hull and propeller roughness and the hydrodynamic influences associated with them.

In answer to *Prof. Couch's* first question the fullscale evidence of unstable flow available to the Committee comes from both manoeuvring and speed trials. This is summarized on pages 171 and 172 of the Report.

(Dr. Collatz, in a verbal contribution, mentioned results from the PIONEER Class of Blöhm and Voss which give additional evidence).

In answer to Prof. Couch's question on the quantity of good correlation data available, there is a considerable quantity of data for which the model is an exact representation of the ship in so far as draft, propeller and so on are concerned. For example, calm-water tests corresponding to the trial conditions are often carried out at the NMI and St. Albans tanks in the U.K. Unfortunately, the effects of weather during trials are not so readily represented or assessed. Nevertheless, the Committee consider that the limited amount of data available for trials under good weather conditions is very important and recommends that such trials should be analysed in detail to improve the prediction methods.

Adm. O'Dogherty, like *Prof.Sheng* and *Dr. Chen* earlier, notes the variability in the values obtained in form factor calculations and the Committee refer him to their earlier comments on the aspect. The Committee consider that form factors for ships with nonconventional bows should be derived with caution, since the resistance curves in these cases may be distorted at low speeds. Nevertheless the determination of form factor in not as critical as might be supposed, since as shown in Figure 2 of the Committee's Report, fairly large changes in (1+k) have only a small effect on the power and revolutions predicted for the ship when using the 1978 ITTC method.

As far as the question of predictions based on torque rather than thrust identity are concerned, thrust identity is generally used and is adopted in the 1978 Method because of the difficulty in measuring thrust on the full size ship. It is one of the assumptions of the 1978 Method that there is no scale effect on relative rotative efficiency and thrust can be derived indirectly from the measured shaft torque.

The Committee is interested in the method of trial prediction in use at El Pardo, and would like to see a comparison between such predictions and those made using 1978 ITTC Method.

The use of fuel consumption in analyses of ship service performance has been dealt with earlier in our reply to Prof. Aertssen.

Mr. Müller-Graf comments on the section of the Report concerning high speed craft. This section on the report was written in close collaboration with the Panel on High Speed Vessels.

It is quite correct that high speed craft cannot be treated in the same way as other displacement ships, and reasons for this are discussed in detail in Section 6 of the report.

In the case of planing craft with a chine bilged hull, the wetted surface and trim angle change as the forward speed changes. The same will also be true in the semi-planing condition but to a lesser extent.

The question of the appendage resistance of hydrofoil boats is a matter of definition. The Committee agree that it may be more logical in the foil-borne condition, in which the hull itself is out of the water, to include in the appendage drag only the resistance due to shafts, ducts, pods and so on. The proportion of resistance (viscous and induced) due to such appendages in roughly as great in the case of the hydrofoil craft as for other high speed vessels.

The importance of oblique inflow conditions on propulsive coefficients is recognised by the Committee and has been briefly noted on page 154 of the Report.

In reply to *Mr. Suhrbier*, /Equation 3.2/, was derived using data for ship trials which were carried out in good weather conditions. However, it is probable, as noted in the Committee Report, that the equation may reflect the effects of phenomena other than hull roughness. It is also pointed out in the Report that the trends exhibited by the ITTC equation are different from the roughness effects determined from the analyses carried out by Karlsson and Grigson of results obtained in the laboratory. These comparisons confirm Mr. Suhrbier's statement that the ITTC equation probably overestimates the influence of hull roughness. The Committee would like to emphasize that the equation should only by used in the context of performance prediction methods.

The Committee agree with *Dr. Lazarov* that prediction of ship performance in shallow or restricted waters is of importance, and that attention should be given to the specialized vessels which often operate in such conditions. In response to Dr. Lazarov's request, consideration will be given by the Committee to the inclusion of an additional recommendation to cover this subject area.