INTERNATIONAL CONFERENCE OF TANK SUPERINTENDENTS THE HAGUE, JULY 13TH AND 14TH, 1933

NEDERLANDSCH SCHEEPSBOUWKUNDIG PROEFSTATION
WAGENINGEN (HOLLAND)

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COLLECTED PAPERS AND NOTES

EDITED BY L. TROOST

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SECRETARY OF THE CONFERENCE

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General approval has been expressed only upon the statements made in the „Decisions'' at the end of the Volume.

PREFACE

During the International Hydro-mechanical Congress, organised by Dr E. FOERSTER and Dr G. KEMPF in Hamburg, May 1932, Mr JOHN DE MEO in the course of an after dinner speech pleaded strongly for international technical co-operation in the field of ship propulsion.

Several leaders of experimental tanks who were present felt the desirability of continuing on the broad lines laid down in the Hamburg Congress.

After conferring with those colleagues who were present, I invited, in answer to Mr. DE MEO's speech, the superintendents of the various experimental tanks to come to Holland in 1933, in order to discuss the form which co-operation between the tanks should take. All those present accepted the invitation.

The Council of the Dutch experimental tank in Wageningen, in agreement with the above, issued an invitation to all experimental tank superintendents and some other specialists for a conference to be held in The Hague, on the 13th and 14th July 1933.

This meeting took place and is to be regarded as a first step towards reaching a standard system of publication of tank results.

With Prof. E. VOSSNACK in the Chair, the discussions, which were of an informal and confidential character, lead to the appointment of a Committee of Four (BAKER, BARRILLON, KEMPF, TROOST) for working out in a more definite way the general conclusions given on page 138.

The intention of the Conference to give the tank officials an opportunity of conferring in an open and confidential manner on their own methods and also on the manner of publication of tank results, has been fully realised.

These discussions were resumed during the Summer Meetings of the Institution of Naval Architects in Londen, 1934, which devoted the principal part of its programme to subjects dealing with tank research work. In 1935, a meeting of tank officials will be held in Paris.

Thus international co-operation in the field of tank research work for ship propulsion is in progress and the pioneer work of

Mr DE MEO, who has been advocating international technical co-ordination on general lines already for years, seems not to have been done in vain as far as tank work is concerned.

The following papers and notes were presented to the Conference by various members and I have collected them in this volume as a work of reference for those who are interested in the future co-operation of experimental tanks. The scope of these papers was indeed much too extensive to be fully treated in the 1933 Conference.

Thanks are due to the Society of Supporters of the Wageningen Tank, whose financial aid enabled the Board of this Tank to organise the 1933 Conference and the printing of this volume, and to the ,,Koninklijk Instituut van Ingenieurs", who kindly placed the Hall of the Institution's Building at the disposal of the Conference.

Wageningen, September 1934
L. TROOST.

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PROGRAMME
OF THE CONGRESS OF TANKSUPERINTENDENTS
THE HAGUE 1933

a. Discussion of the methods of investigating selfpropelled models, i.e. with and without friction correction according to Froude. It is known that different continental tanks make these experiments with a tension correction fitted to the model itself, which is previously calculated from the Froude friction correction formula, while the tanks in the English speaking countries base their S. H. P. on the self-propulsion point of the model.

b. The use of standard means for prevention of laminary flow on the model.

c. The determination of a general formula for the correction of skin friction according to Froude's method with a standard temperature.

d. The use of a standard formula, for instance $\frac{R}{\frac{\varrho}{2}V^2A}$ for the publication of the curves of resistance (R = total resistance).

e. A discussion for the determination of standard symbols and constants.

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16. H. Munday
17. A. van Driel
18. H. M. Weitbrecht
19. M. Legendre
20. H. S. Howard
21. W. P. v. Lammeren
22. W. v. Beelen

FIRST INTERNATIONAL CONGRESS OF TANK EXPERIMENTERS.

The recognition and the soundness of aims towards the stimulation of co-ordinated technical progress for the benefit of the marine community, have received unequivocal support from progressive technicians associated with shipping concerns. It has made acceptable the plea to establish methodical development on research-problems of ship propulsion.

These aims have been widely disseminated through the technical press, while zeal and honesty of opinion necessary to interpret the different thoughts and aspirations, has made myself the exponent of a propaganda campaign inspired by impersonal motives. This, therefore, allows me to-day, at the invitation of the Dutch Committee, the privilege of addressing distinguished technicians from many Countries.

Most of you, Gentlemen, have approved the fundamental items of the preliminary programme, which was drawn up in order to stimulate reaction from foremost colleagues towards the promotion of these pioneer meetings.

The work to be initiated now requires a great deal of care because it affects basic items of fundamental importance in the development of future research. It would avoid well known existing differences in results and the difficulty of favourable reaction for marine application.

Perhaps the lack of procedure in handling professional work, has done much to discourage progressive technicians who have been anxious to explore the possibilities of reaching improved technique. The outgrowing overlap of experimental work and the unbearable accumulation of data on the same aspect of research, made the present system of disseminating information through the channel of occasional papers insufficient to ensure rapid assimilation of progressive efficiency. A remedy could be found through the integrating efforts of indispensable solidarity among the world Tanks to lay down concentrated understandable experimental results.

The organized sectionalization of Tank work with uniformity of method, could avoid the generally lamented overlapping and the difference in data. Obviously this would realize economy and saving of time, ensuring to the Tanks more disponibility for work of peculiar interest to the Countries concerned.

The general opinion towards the possibility of reaching such technical achievement, destined fundamentally to satisfy all aspirations from research-workers and professional aims, foreshadow the usefulness of a "forum of knowledge" where dictates from science and research duly co-ordinated through the homogeneity of method, established upon a common nomenclature and technical language, would make understandable in future sifted contributions clearly integrated, and fill the obviously distressing gaps in matters of ship propulsion.

Undoubtedly, such a forum of knowledge through its court of highly qualified experienced technicians, would ensure technical judgments combining scientific wisdom with complement of practical decisions. It would improve the general level of marine technique thereby preserving a proper basis of individual competence and maintaining the high status of technical qualifications of those professionally occupied in the marine field.

The accomplishment of organized unity of knowledge in dictates of ship propulsion would ensure that research-work no longer suffered from underestimation, while the world technical community and associated concerns of the marine industry would enthusiastically welcome the possibility by which experience and common sense, associated with duly sifted experimental results, would produce the most logical amalgamation to be desired.

The remarkable researches pioneered by FROUDE and established through his dictates, by which Great Britain has benefited past and present generations are everywhere gratefully acknowledged. It demands such tradition legitimately continue, for the acquisition of future complemental dictates and experimental research data harmonized through technical co-ordination. Thus to follow the fundamental fact of modern existence and the rapid evolution of mechanical possibilities of the times emphasizing the need of collective efforts and inherent energies among eminent men of science and research-workers from every organization, collaborating with engineers in practical problems of naval architecture. These efforts are evidently destined to minimize effects from barriers in theoretical and practical fields, actually preventing the full fruition of isolated outstanding technical capacity.

There has never been any question of lack of vision, or possible apathy, among the professional concerns towards justifiable technical reform.

The activities are well known of individual Institutions in every Country, for possible co-operation with other scientific and educational bodies, with views to advance in science of shipbuilding and marine engineering. It is simply a matter of psychological phenomena of professional misconception and individual inertia which have unfortunately clogged, up to the present, the accomplishment of progressive factors of practical mportance equally to all technicians and shipping concerns.

The preceding remarks, suggesting the foundation of one incorporated body of knowledge to federate kindred Institutions all over the world, appear fundamentally justified. It should canvass and correlate scientific material with experimental results and trials, and do these things on a common international ground of high professional status qualification and scientific training, such as would satisfy all professional, scientific and technical aspirations among the marine professions.

The whole technical community has notoriously in mind to stimulate offers of help from the various Governments, by asking the disposal of craft belonging to the auxiliary service or reserve, for the purpose of making the needed investigations and filling the unbridged gaps arising from the impossibility to reproduce normal conditions in laboratory results. It is desirable that such aspiration should be helped forward by the interested Governments here represented, and such action would surely eliminate a present deficiency, almost creating international divergence of opinion in matters of ship propulsion.

The indispensable assemblage of uniform knowledge being of interest to all maritime Nations, the propitiatory character of these meetings is the best pledge that could be offered to encourage from interested Governments and Tank establishments, a farsighted consideration and a willingness to complement such important work with collective financial support. The desired forthcoming of such moral and material help would represent a substantial sympathy, which would be no more than a due recognition of the value of benefits arising from the technical progress thereby assured, and of equal interest to the various Countries facing problems to achieve sound economies.

Detailed representations of the need that assistance forthcoming from Governments and Tank concerns, commonly interested through this great work of technical co-ordination, are here esteemed superfluous, being well known to the Delegates, here assembled, while their peculiar knowledge in this research-branch is the best advocate justifying their active cooperation. Delegates are also well aware of the importance attached to the need of craft being placed at their disposal to meet large-scale experiments, and of the little financial requirements involved in carrying out collectively the peculiar divisional research-work amongst the world Tanks.

This is possible through the help and assistance of the Tank personnel to consolidate results through the common technical forum of knowledge, for the final delivery of conclusions regarding factors of practical economy greatly affecting the Nations concerned.

Such essential consolidation, moreover, is of great importance because these united efforts intentionally integrate the spirit and impulse of fellowship, and combine the legitimate desire from world technicians for the development of knowledge communicated with an absence of personal motive and of National pride.

Doubts and misapprehensions among technical men would be dissipated through such consolidation of collective aims, and the evident goodwill and sincerity in this great work of cooperation with mutual understanding, endeavouring to reach finality of better interpretation of Tank work on common basis, would leave untouched the individuality of the research-workers, while they would remain masters of their own thoughts and developments.

Alea jacta est! — Your work, Gentlemen, begins now by establishing the fundamental classification and nomenclature, to assist the better knowledge from existing and future researches. Your cumulative experience will harmonize future proceedings of this kind, offering complements of knowledge with absence of smoke-screens for the benefit of the whole technicians. There will thus be formed the granitic foundation of the pioneer edifice destined to contribute largely to the welfare of the marine profession, holding indelibly your names in the history records of the marine progress.

The furtherance of this work of clarification for better comprehension of principles and methods would represent, consequently, the product of a real community of interests, by which the harmonious foundation of the edifice will mark a new era of evolution of natural progress, and stamp with universal nomenclature the scientific material, to be delivered to your successors for the developments through the marine activities.

Concluding, Gentlemen, I feel privileged to have addressed you wishing definite success for conclusive fundamentals of technical co-ordination, with persisting reliance in your present and future work of sincere cooperation and understanding.

Starting to-day with this historical initiation, all will perceive that it is surely destined to accelerate the rhythm of genuine progress, bringing finally the universal desire of the technical world to willingly co-operate in marine enterprises.

JOHN DE MEO.

Den Haag–July, 1933.

The William Froude Laboratory
Teddington

FROUDE S.F.C. METHOD

(see Trans I.N.A. 1888; page 310)

(see also „William Froude Laboratory" formulæ previously given)

Let skin friction term in resistance = f
corresponding term in (C) value = (F)

The process of applying skin friction correction consists in deducting, from the total (C) value for model, the (F) value for model, (= $(F)_m$ say) and subsituting that for ship. (= $(F)_s$ say), or deducting the net value $(F)_m - (F)_s$

Assume, for given length between perpendiculars, that skin friction varies as area of wetted skin, and as power 1.825 of speed.

Let L = length (ft.) between perpendiculars.
S = area (sq. ft.) of wetted skin.

From formulæ:

$$(K)^2 = (L)^2 . (M) = (L)^2 . \frac{L}{\triangle^{1/3}}$$

$$(F) = \frac{1000\ f}{\triangle . (K)^2} = \frac{1000\ f}{\triangle^{2/3} . L . (L)^2}$$

For some given length and speed such that (L) = 1.0 let 1000 f = O.S.L. ("O" = "ordinate").

Then for same length and any value of (L),

$$1000\ f = O.S.L.\ (L)^{1.825}$$

$$(F) = \frac{O.S.L.\ (L)^{1.825}}{\triangle^{2/3} . L . (L)^2} = O . (S) . (L)^{-.175}$$

values of "O" for various lengths are given on next page, as also values of $(L)^{-.175}$, from which to construct curves of O_m and O_s, ordinates for model and ship.

$$\text{Then } (F)_m - (F)_s = (O_m - O_s) . (S) . (L)^{-.175}$$

VALUES OF O_m AND O_s, ALSO $(L)^{-.175}$, AS USED AT W.F.L.

model l (ft)	O_m	Ship L (ft)	O_s	(L)	$(L)^{-.175}$	(L)	$(L)^{-.175}$
5	.15485	40	.1004	.1	1.4962	3.1	.8204
6	.1495	60	.0938	.2	1.3253	3.2	.8160
7	.1449	80	.08987	.3	1.2345	3.3	.8114
8	.1409	100	.0871	.4	1.1739	3.4	.8076
9	.1373	150	.0828	.5	1.1296	3.5	.8031
10	.1341	200	.08009	.6	1.0935	3.6	.7996
11	.1312	250	.07811	.7	1.0640	3.7	.7953
12	.1286	300	.07651	.8	1.0398	3.8	.7920
13	.1262	350	.07520	.9	1.0186	3.9	.7881
14	.12405	400	.07404	1.0	1.0	4.0	.7847
15	.1221	450	.07303	1.1	.9835	4.1	.7816
16	.1203	500	.07215	1.2	.9686	4.2	.7784
17	.11875	550	.07135	1.3	.9551	4.3	.7752
18	.1173	600	.07061	1.4	.9430	4.4	.7720
19	.1160	650	.06994	1.5	.9318	4.5	.7686
20	.1147	700	.06931	1.6	.9211	4.6	.7660
21	.1136	750	.06872	1.7	.9116	4.7	.7627
22	.11255	800	.06819	1.8	.9023	4.8	.7604
23	.11155	850	.06769	1.9	.8940	4.9	.7572
24	.1106	900	.06722	2.0	.8857	5.0	.7545
25	.10975	950	.06678	2.1	.8782		
26	.1089	1000	.06637	2.2	.8708		
27	.1081	1050	.06597	2.3	.8642		
28	.1073	1100	.06560	2.4	.8579		
29	.1066	1150	.06526	2.5	.8518		
30	.1059	1200	.06493	2.6	.8460		
				2.7	.8404		
				2.8	.8351		
				2.9	.8302		
				3.0	.8251		

EXAMPLE OF FROUDE S.F.C. CALCULATION

given L = ship length = 436′

l = model length = 17′44

$(S) = 6.223 \left(= \dfrac{S}{(35\triangle)^{2/3}};\ S = \text{wetted skin} \right)$

$p = \begin{cases} .750 \\ \text{(prismatic coefficient)} \end{cases}$

$l_m = 17.44 \quad O_m = .1181$

$L_s = 436 \quad O_s = \underline{.0733} \quad (S)$

$O_m - O_s = .0448 \times 6.223 = .279$ S.F.C. @ $(L) = 1.0$

L	.3	.4	.5	.6	.7	.8	
S.F.C.	.344	.327	.315	.305	.297	.290	$= .279 \times \text{L}^{-.175}$
P	.245	.326	.408	.490	.572	.653	$= \dfrac{\text{L}}{\sqrt{2p}}$

Knots scale: — $\text{P} = \dfrac{.746\ \text{V}}{\sqrt{p\text{L}}} = .04125\ \text{V}.$

It is usual to take the O_m and O_s values from the length between perpendiculars for practically all vessels with raised sterns. For cruiser stern vessels we use length on water line in most cases, but we exercise a little judgment in such extreme cases as when the contour of the cruiser stern only just touches the water for a considerable length or when a large portion of the screw aperture is out of water. In many cases it is immaterial which is used, as the $(\text{O}_\text{m}-\text{O}_\text{s})$ values actually obtained are only slightly different whichever lengths are taken. In every case, however, we take the *total* immersed skin area with the ship at rest, in calculating the S value.

Correction for Temperature.

All C curves are brought to a standard temperature of 55° Fahrenheit. The correction is plotted in conjunction with the "Iris" correction as an addition or deduction as the case may be and is the same for all L values.

It is assumed that for 10° Fahrenheit difference in temperature there is 3 per cent difference in resistance. The increase or reduction is allowed on surface friction which for the standard model ("Iris") at L = 1.0 is .878 C; so that if the temperature is increased 1° Fahrenheit .878 C becomes .87536 C i.e. is reduced 0.3 per cent. This difference divided by S which for the "Iris" is 7.08 gives .000372. Therefore for a difference of N° Fahrenheit from the standard temperature of 55° Fahrenheit, the temperature correction will be .000372 × S × N°.

HAMBURGISCHE SCHIFFBAU-VERSUCHSANSTALT G.m.b.H.

a. Die Hamburgische Schiffbau-Versuchsanstalt G.m.b.H., Hamburg 33, führt ihre Versuche bei den Modellen mit Selbstantrieb unter Benutzung eines Reibungsabzuges nach FROUDE durch und stützt sich dabei auf folgende Überlegung:

Setzt man $S = W = W_f + W_r$.

so wird $s \cdot \alpha^3 = (w - w_r)\, \alpha^3 \gamma_1 \cdot + W_r$.

infolgedessen ist

$$s \cdot \alpha^3 = \alpha^3 \gamma_1 \left\{ w - \left(w_r - \frac{W_r}{\alpha^3 \gamma_1} \right) \right\}$$

$$s = \gamma_1 \left\{ w - \left(w_r - \frac{W_r}{\alpha^3 \gamma_1} \right) \right\}$$

$$s = \gamma_1 \, (W - Ra)$$

Dieser rechnungsmäßig festliegende Betrag des Reibungsabzuges $Ra = w_r - \frac{W_r}{\gamma_1 \alpha^3}$ muß als Zuggewicht bei der Durchführung des Versuches angebracht werden, damit der Rest des Gesamtwiderstandes mit α^3 multipliziert den Gesamtschub des Schiffes entspricht und die dabei ermittelten WPS und Drehzahlen sich den tatsächlichen Verhältnissen für die betreffende Geschwindigkeit anpassen. Der Nennbetrag des Reibungsabzuges ist so groß, daß sich unrichtige Ergebnisse für Schub, Wellenpferde und Drehzahlen ergeben würden, wenn keine Berücksichtigung des Reibungsabzuges stattfände.

Die Berechnung des Reibungsabzuges erfolgt nach der vereinfachten Formel $Ra_2 = f \cdot v_m{}^{1,825} \left(\lambda_m - \frac{\lambda s}{\alpha\ 0,0875} \right)$ worin f die benetzte Oberfläche des Modells in m^2, v_m die Modellgeschwindigkeit in m/sec, α der Modellmaßstab und λ_m und λ_s die Reibungskoeffizienten für Modell und Schiff sind, abhängig von deren Länge. Als Länge wird dabei die größte Länge innerhalb des Tiefganges gewählt einschließlich Ruder.

Die Reibungskoeffizienten sind im Hilfsbuch für den Schiffbau V. Auflage Band 1, Seite 170/171 veröffentlicht.

b. Die Hamburgische Schiffbau-Versuchsanstalt verwendet als Mittel zur Verhinderung des Auftretens von laminarer Strömung am Modell eine mit der Ziehklinge hergestellte Aufrauhung von 2–3 cm Breite, die etwa auf Konstruktionsspant $9\frac{1}{2}$ angebracht ist.

c. Die Hamburgische Schiffbau-Versuchsanstalt wendet im Umrechnungsverfahren für die Reibungskorrektur nach FROUDE bezogen auf die Schlepptemperatur im Vergleich zur Normaltemperatur von 15° die Abänderung des Reibungskoeffizienten für das Modell an. In dem Ausdruck für $Ra = f \cdot v_m{}^{1,825} \left(\lambda_m - \frac{\lambda s}{\alpha\ 0,0875}\right)$ wird das λ_m, der Reibungskoeffizient für das Modell in $\lambda_{m'}$ verwandelt nach der Formel

$$\lambda_{m'} = \lambda_m \left[1 \pm 0,0043\ (15° - t°)\right]$$

worin das Vorzeichen + bei Temperaturen t unter 15°, das Vorzeichen — bei Temperaturen über 15° einzusetzen ist.

Die Temperatur 15° ist die mittlere Normaltemperatur des Atlantischen Ozeans und der Koeffizient 0,0043 der Formel ist das Ergebnis langjähriger Vergleichsversuche der ehemaligen Marineversuchsanstalt Lichtenrade mit einem Cement-Modell bei den verschiedensten Wassertemperaturen.

d und *e.* Die Wiedergabe von Widerstandskurven von Schiffsmodellen und die Besprechung einheitlicher Symbole und Konstanten kann unter Zugrundelegung der von Herrn Weitbrecht eingereichten Liste erfolgen, obwohl bei uns zum Teil andere Bezeichnungen im Gebrauch sind.

Dr. Ing. GÜNTHER KEMPF

Hamburg, den 11. Mai, 1933.

Preussische Versuchsanstalt für Wasserbau und Schiffbau, Berlin

VORLÄUFIGE LEITSÄTZE

BETREFFEND

VERWENDUNG EINES STOLPERDRAHTES ZUR VERMEIDUNG LAMINARER STRÖMUNG AN SCHIFFSMODELLEN

(zu Punkt b der Tagesordnung)

1. Turbulente Strömung ist am Modell vorhanden, wenn bei Auftragung des Gesamtwiderstandsbeiwertes über der Reynoldszahl, also $\zeta_g = f(R)$, bei kleinen Reynoldszahlen die Kurve der Widerstandsbeiwerte ungefähr äquidistant mit den turbulenten Plattenreibungsbeiwerten verläuft.

2. Für Modelle von mehr als 4 m Länge muß die Dicke des Stolperdrahtes mindestens 1,5 mm betragen.

Der Draht ist auf 1/20 $\overline{\mathrm{L}}$ vom Vorsteven zu befestigen.

Zur Berechnung des Modellreibungswiderstandes sind die turbulenten Plattenreibungsbeiwerte zu verwenden.

3. Der Einfluß des Stolperdrahtes zeigt sich nur bei Reynoldszahlen unter $4 . 10^6$.

4. Die Unstetigkeit in den Mitstromwerten bei Reynoldszahlen unter $8 . 10^6$ bleibt auch bei Verwendung eines Stolperdrahtes bestehen.

5. Der Stolperdraht ist der Aufrauhung vorzuziehen, da man jederzeit auf den ursprünglichen Modellzustand zurückgehen kann.

Bem.: Die Leitsätze beruhen auf dem Ergebnis von Versuchen in der Preußischen Versuchsanstalt für Wasserbau und Schiffbau, Berlin. Die Versuche sind jedoch noch nicht abgeschlossen.

Preussische Versuchsanstalt für Wasserbau und Schiffbau, Berlin

VORLÄUFIGE LEITSÄTZE

BETREFFEND

UMRECHENVERFAHREN FÜR DIE REIBUNGSKORREKTUR

(zu Punkt c der Tagesordnung)

1. Der Abfall der Verbindungslinien von Punkten gleicher Froudezahl auf den Kurven der Widerstandsbeiwerte einer Modellfamilie aufgetragen über der Reynoldszahl erfolgt von verhältnismäszig niedrigen Reynoldszahlen ab etwa parallel dem Abfall der Reibungsbeiwerte glatter Platten bis zum naturgroszen, aber völlig glatten Schiff, soweit es sich nicht um scharfe, schmale Schiffskörper handelt.

2. Schematische Anwendung der Umrechnung nach Froude vom Modell auf Schiff kann besonders bei kleinen Modellen und niedrigen Froudezahlen zu groszen Fehlern führen.

3. Die Arbeit von Schlichting zeigt einen Weg, um die Mängel des Verfahrens von Froude zu beseitigen. Jedoch sind die Einzelwerte zahlenmäszig noch nicht genug gesichert. Es ist anzustreben, endgültige Zahlenwerte möglichst bald zu gewinnen.

4. Der Praxis ist mit dem Schema eines neuen Umrechnungsverfahrens ohne gesicherte Zahlen nicht gedient. Es genügt für sie zunächst vollkommen, wenn bei der Anwendung der Umrechnung nach Froude für erkannte Fehler zahlenmäszige Richtigstellungen vorgenommen werden können.

5. Bei der Anwendung der Umrechnung nach Froude sichert:

a. für alle Fahrzeuge mit Ausnahme der unter 5*b* aufgeführten die Einhaltung einer Reynoldszahl von mindestens $4 . 10^6$ beim Versuch vor gröszeren Fehlern.

b. Für schlanke, schmale Fahrzeuge ist die Reynoldszahl mindestens $7{,}5 . 10^6$ zu wählen.

6. Bei Anwendung eines Stolperdrahtes oder einer Aufrauhung kann man für die unter *5a* genannten Fahrzeuge auf Reynoldszahlen von 2.10^6 heruntergehen.

Inwieweit der Widerstand der unter 5 b genannten Fahrzeuge durch Stolperdraht oder Aufrauhung beeinfluszbar ist, musz erst noch durch Versuche geklärt werden.

Siehe Bericht über den Sprechabend der S.T.G. vom 22-2-'33 in der Zeitschrift „Schiffbau" 1933, Heft 7/8.

KURZE STELLUNGNAHME

VON HERRN PROFESSOR DR. HORN

ZU DEN PUNKTEN DES TAGUNGSPROGRAMMS DER KONFERENZ IN DEN HAAG ÜBER SCHIFFSANTRIEB.

a. *Versuchsmethoden bei Modellen mit Selbstantrieb*, insbesondere ob mit oder ohne Reibungsabzug nach FROUDE.

Da bei der Ermittlung des Schiffswiderstandes auf Grund der reinen Schiffsmodellversuche durchweg mit einem Reibungsabzug gerechnet wird — ob nach FROUDE oder sonst wie, bleibt unter c zu erörtern —, so ist es meiner Ansicht nach aus Gründen der Konsequenz wünschenswert, dass die von den Versuchsanstalten auf Grund des Versuchs mit Schraube ermittelte Antriebsleistung der gleichen Basis entspricht. D.h. es muss das Modell bei Selbstantrieb um den dem Reibungsabzug entsprechenden Betrag entlastet werden. Andernfalls entspricht die Belastung der Modellschraube nicht der der naturgrossen Schraube, sondern ist grösser, der Wirkungsgrad der Modellschraube — sowohl der reine Propellerwirkungsgrad wie der gesamte Vortriebsgütegrad — also geringer als der der naturgrossen Schraube.

Wenn demgegenüber angeführt wird, dass der ohne Berücksichtigung eines Reibungsabzugs erhaltene Wirkungsgrad der Propulsion praktisch den tatsächlichen Verhältnissen näherkommt, so bin ich der Meinung, dass das Modellversuchsverfahren zunächst in sich möglichst einheitlich und geschlossen durchgeführt werden sollte. Auch bin ich der Meinung, dass, wenn der *richtige* Reibungsabzug eingeführt wird (der nach FROUDE ist meiner Ansicht nach zu gross, vergl. Punkt c), man auch praktisch den tatsächlichen Verhältnissen völlig ausreichend nahekommen wird. Falls aber, um dies letztere Ziel zu erreichen, doch noch gewisse Zuschläge sich als notwendig bzw. empfehlenswert herausstellen sollten, so sollten solche auf das Endergebnis gemacht werden, weil man nur auf diese Weise einen klaren Überblick darüber behalten kann, was das reine, d.h. einheitlich durchgeführte Modellversuchsverfahren leistet bzw. ob und wie weit seine Ergebnisse noch von der Wirklichkeit abweichen.

b. *Anwendung von einheitlichen Mitteln zur Verhinderung des Auftretens von laminarer Strömung am Modell.*

Die Anwendung solcher Mittel halte ich zur Beseitigung oder Verminderung der laminaren wie auch der Ablösungsströmungen grundsätzlich für wünschenswert. Im übrigen möchte ich mich aber vorläufig noch nicht näher hierüber äussern, da alles davon abhängt, welche positiven Ergebnisse die verschiedenen Mittel zur künstlichen Turbulenzerzeugung bisher gezeitigt haben und da über die positiven Ergebnisse mir bisher zu wenig bekannt geworden ist.

c. *Feststellung eines allgemein gültigen Umrechnungsverfahrens für die Reibungskorrektur* nach FROUDE bei einer einheitlichen Normaltemperatur.

Hinsichtlich dieser Frage verweise ich im wesentlichen auf meine Ausführungen auf dem Sprechabend der STG am 22.2.33, die in Zeitschrift „Schiffbau" 1933, S. 164/65 und ausserdem in dem 2. Forschungsheft der STG S. 16/17 abgedruckt sind. Ich darf kurz die meiner Ansicht nach hieraus zu ziehenden Konsequenzen zusammenfassen:

1. Den Grundgedanken von FROUDE, nämlich einerseits Errechnung des Reibungswiderstandes von Modell und Schiff unter Zurückgreifen auf die Widerstände ebener Platten, auf das Schiff nach dem FROUDE'schen Ähnlichkeitsgesetz, wird man meiner Ansicht nach trotz der bekannten und berechtigten grundsätzlichen Bedenken als notwendiges Kompromiss, zum mindesten für die routinemässige Handhabung seitens der Schiffbauversuchsanstalten, beibehalten müssen.

2. Die speziellen FROUDE'schen Reibungswerte sind jedoch sowohl vom wissenschaftlichen wie vom praktischen Standpunkte aus zweifellos nicht mehr haltbar. Sie sollten nur noch solange — und für diese Zwischenzeit am besten unverändert, um unnötige Verwirrung zu vermeiden — beibehalten werden, bis man sich auf die in Zukunft zu verwendenden Werte geeinigt hat. Hierfür mache ich folgende Vorschläge.

3. Ausgangskurve für den Reibungsabfall wird die neue Göttinger Kurve $\zeta_r = \frac{0{,}455}{(\log R)^{2{,}58}}$. Durch neue Versuche mit Modellfamilien sehr grosser Modelle, deren kleinstes schon oberhalb der laminaren bezw. Ablösungs-Störungszone liegt, ist für einige typische Schiffsformen und Völligkeiten festzustellen, ob und in

welchem Masse der Formeffekt eine Änderung der Abfallkurve mit sich bringt. Im übrigen dürfte zunächst ein prozentualer Zuschlag auf ζ_r für den Einfluss des Formeffekts naheliegen, dessen Höhe gegebenenfalls routinemässig aus der gemessenen Absenkung des Modells abgeleitet werden könnte (vergl. meinen Diskussionsbeitrag zum Thema Reibungswiderstand auf der Antriebskonferenz Hamburg).

4. Der sehr bedeutsame Einfluss der Rauhigkeit sollte durch einen besonderen Zuschlag auf die ζ_r-Kurve berücksichtigt werden, dessen Höhe auf Grund noch vorzunehmender weiterer systematischer Rauhigkeitsversuche nach Art der neueren Göttinger sowie der Kempfschen Pontonversuche zu bestimmen wäre.

Ein den Vorschlägen 3 und 4 entsprechendes Verfahren ist in beiliegendem Diagramm am Beispiel eines schnellen Frachtschiffs erläutert und zwar sowohl ohne als mit Reibungsformzuschlag (siehe 3), hier auf Grund der Absenkungsmessung mit 8% eingeführt). Die dabei gewählte Göttinger Korngrösse der Rauhigkeit von 0,3 mm [1]), würde nach den Kempfschen Messplattenversuchen auf der „Hamburg" annähernd der praktischen Rauhigkeit einer normalen Schiffsoberfläche entsprechen. Der hiernach sich ergebende Gesamtwiderstand liegt rund 15 bzw. 11% höher als er sich nach den FROUDE'schen Reibungswerten ergeben würde. Sollte sich bei Durchführung dieses Verfahrens an einer genügenden Anzahl von Beispielen herausstellen, dass Modellbeiwert und Schiffsbeiwert mit praktisch hinreichender Genauigkeit stets auf ein und derselben Abfallkurve liegen, so könnte man auch in Erwägung ziehen, für praktische Zwecke eine solche Einheits-Abfallkurve für eine normale gute Beschaffenheit der Schiffsoberfläche zu Grunde zu legen.

d. *Wiedergabe von Widerstandskurven von Schiffsmodellen* in Veröffentlichungen in einer einheitlichen Form.

Es dürfte sich meiner Ansicht nach empfehlen, in den von den Versuchsanstalten gelieferten Widerstandsdiagrammen, die ja meist unmittelbar in dieser Form auch veröffentlicht werden, als Grundkurve die Kurve der gemessenen Modellwiderstände (kg. bzw. Pfund) und zwar die unmittelbaren Messwerte durch Signa-

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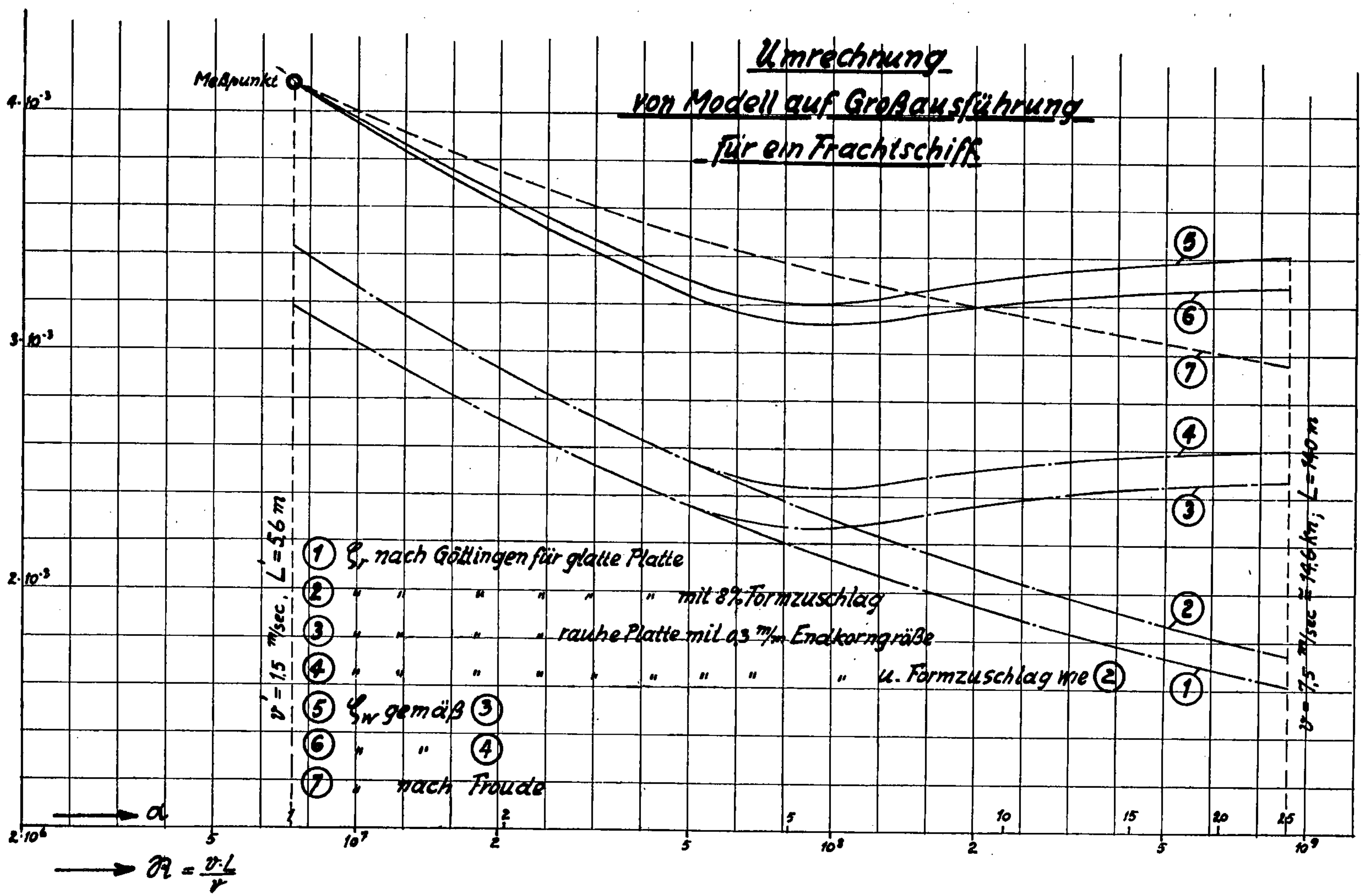
[1]) In meinem Diskussionsbeitrag auf dem erwähnten Sprechabend ist die Korngrösse irrtümlicherweise mit 0.4 mm angegeben; dieser entsprach eine Widerstandserhöhung gegenüber Froude von rund 15%.



Umrechnung
von Modell auf Großausführung
für ein Frachtschiff
Meßpunkt
4·10⁻³
3·10⁻³
2·10⁻³
2·10⁶
5
10⁷
2
5
10⁸
2
5
10⁹
1
2
5
10
15
20
25
α
R = v·L/ν
v = 1,5 m/sec, L = 5,6 m
v = 7,5 m/sec = 14,6 kn, L = 140 m
1 ζr nach Göttingen für glatte Platte
2 " " " " " " mit 8% Formzuschlag
3 " " " " rauhe Platte mit 0,3 m/m Endkorngröße
4 " " " " " " " " " " u. Formzuschlag wie 2
5 ζw gemäß 3
6 " " 4
7 " nach Froude
1
2
3
4
5
6
7

tur gekennzeichnet, über der Basis der Modellgeschwindigkeiten (m/sec bzw. Fuss/sec) beizubehalten, unter Angabe der Temperatur des Tankwassers. Ausserdem müsste das Diagramm die Kurve der daraus, etwa bei 15° C., abgeleiteten Schiffswiderstände W und Schleppleistungen (in PS) über der korrespondierenden Schiffsgeschwindigkeit (in Knoten) enthalten; schliesslich die Kurve des Gesamtwiderstandsbeiwerts $\frac{W}{\frac{\varrho v^2}{2} \cdot O}$ über der FROUDE'schen Zahl $\frac{v}{\sqrt{gL}}$. Das gäbe also im Ganzen 3 verschiedene Abszissenteilungen: Modellgeschwindigkeit, Schiffsgeschwindigkeit, FROUDE'sche Zahl. Eine weitere Teilung nach der REYNOLDS' schen Zahl ist im allgemeinen wohl überflüssig, kann aber im Einzelfalle leicht zugefügt werden.

e. *Einheitliche Symbole und Konstanten.*

Siehe die von Herrn Oberbaurat WEITBRECHT bereits eingesandte vorläufige Liste, die im Einvernehmen mit mir auf Grund eines schon von früher stammenden Vorschlags einer Kommission, bei welcher auch Herr Dr. KEMPF massgebend mitgewirkt hat, entstanden ist.

gez. HORN.

13-6-1933.

Schiffbautechnische Gesellschaft

Berlin, den 14. Juni 1933.

STELLUNGNAHME

DES FACHAUSSCHUSSES FÜR WIDERSTAND UND VORTRIEB DER SCHIFFBAUTECHNISCHEN GESELLSCHAFT

zu den Punkten a *bis* e *der Tagesordnung für die Konferenz von Wageningen (12–15 Juli 1933)*

a. Besprechung der Versuchsmethoden bei Modellen mit Selbstantrieb, bezw. mit und ohne Reibungsabzug nach FROUDE.

Die Versuche sind mit Reibungsabzug zu fahren, da dessen Anwendung entsprechend dem Froudeschen Verfahren folgerichtig und zweckmässig erscheint.

Über die Frage, welcher Reibungsabzug anzuwenden ist, siehe Punkt *c*.

b. Anwendung von einheitlichen Mitteln zur Verhinderung des Auftretens von laminarer Strömung am Modell.

Die Verhinderung laminarer Strömung durch künstliche Mittel am Modell ist grundsätzlich erstrebenswert. Mit welchen Mitteln und unter welchen Bedingungen diese Aufgabe zweckmässig gelöst wird, erscheint noch nicht genügend geklärt.

c. Feststellung eines allgemein gültigen Umrechnungsverfahrens für die Reibungskorrektur nach FROUDE bei einer einheitlichen Normaltemperatur.

Die Froudeschen Reibungswerte bedürfen der Umstellung auf Reynoldsziffern und der Berichtigung. Sie liegen im besonderen für schnelle und lange Schiffe zu niedrig. Zu ihrer Berichtigung erscheint es notwendig, folgende Fragen im Rahmen eines gemeinsamen aufzustellenden Planes zu klären:

1. den Formeneinfluss auf die Reibung,
2. den Einfluss, den die Rauhigkeit der normalen guten Schiffsoberfläche auf die Reibung ausübt.

Von dem Ergebnis wird es abhängen, ob der Reibungswiderstand für die Schiffsoberfläche durch gesonderte Zuschläge für die Einflüsse zu 1 und 2 auf die Werte glatter Platten ermittelt, oder ob und wie weit für den Gesamtwiderstandsabfall (ζ-Werte) zwischen Modell und Schiff eine Einheitskurve mit praktisch ausreichender Genauigkeit zugrundegelegt werden kann.

d. Der Wiedergabe von Widerstandskurven von Schiffsmodellen in Veröffentlichungen in einer einheitlichen Form, z.B. $\frac{W}{\frac{\varrho v^2}{2} \cdot O}$ auf Basis R, (W = Gesamtwiderstand) wird zugestimmt. Es wird jedoch vorgeschlagen, hierzu auch die Froudeziffern anzugeben und ausserdem die Originalmodellwerte über der Modellgeschwindigkeit mit Angabe der Temperatur zu liefern.

e. Zu der Besprechung über einheitliche Symbole und Konstanten wird auf die Stellungnahme von Herrn Prof. Dr. Horn und von Herrn Oberbaurat Dr. Weitbrecht bereits eingesandte Liste verwiesen.

BEZEICHNUNGEN

FÜR SCHIFFSWIDERSTAND UND ANTRIEB

(Auf Grund von Besprechungen vom Jahre 1927 zwischen den Berliner und Hamburger Versuchsanstalten und der Technischen Hochschule Berlin)

Die Maßeinheiten sind die des technischen Maßsystems m, kg, s

I. *Bezeichnungen und Rechnungsgrößen.*

1. allgemein.
   - α = Maßstab = geometrisches Ähnlichkeitsverhältnis.
   - γ = spezifisches Gewicht.
   - g = Beschleunigung der Schwerkraft.
   - ϱ = $\gamma : g$ = Dichte.
   - ν = kinematische Zähigkeit.
   - k = tatsächliche Korngröße der Rauhigkeit.
2. Schiff (Modellwerte gestrichen).
   - L = Länge.
   - B = Breite.
   - T = Tiefgang.
   - $\otimes$ = Hauptspantfläche.
   - O = benetzte Oberfläche.
   - V = Raumverdrängung.
   - D = γ V = Gewichtsverdrangung = statischer Auftrieb.
   - G = Schiffsgewicht.
3. Schraube.
   - D = Außendurchmesser.
   - H = Steigung.
   - F = $\frac{D^2\pi}{4}$ = Schraubenkreisfläche.
   - F_a = abgewickelte Flügelfläche.
   - F_p = projizierte Flügelfläche.
   - n = Drehzahl in der Sekunde.
   - z = Flügelzahl.
   - l = Sehnenlänge des Flügelschnittes (Flügeltiefe).
   - s = größte Dicke eines Flügelschnittes.
   - δ = s/l = verhältnismäßige Dicke eines Flügelschnittes.

II. *Kräfte und Momente* (Modellwerte gestrichen).

A = dynamischer Auftrieb.

D = Gewichtsverdrängung = statischer Auftrieb.

G = $D + A$ = Schiffsgewicht.

P = Kraft (allgemein).

Z = Trossenzug.

W = Widerstand (allgemein).

W_{oo} = Widerstand des nackten Schiffes ohne Anhängsel und ohne Schrauben.

W_o = Widerstand des Schiffes im jeweiligen Schleppzustand ohne Schrauben.

W_a = tatsächlicher Widerstand des Schiffes.
= W_o + Zuschlag für im Modellversuch nicht berücksichtigte Einflüsse (Außenhaut + Anhängsel + Fahrtwind + ...).

S = zu W_o gehöriger Schraubenschub = Gesamtwiderstand des Schiffes im Schleppzustand bei laufender Schraube.

$W_o = W_r + W_f = W_r + W_{we} + W_{wi}$.

W_r = Reibungswiderstand.

W_f = Formwiderstand.

W_{we} = Wellenwiderstand.

W_{wi} = Wirbel- und Ablösungswiderstand.

M = Drehmoment.

III. *Geschwindigkeiten* (Modellwerte gestrichen).

c = Wellen- Fortpflanzungsgeschwindigkeit (gegebenenfalls c_0).

v = Schiffsgeschwindigkeit (gegebenenfalls v_0).

v_p = Fortschrittsgeschwindigkeit der freifahrenden Schraube.
= mittlere relative Zuflußgeschwindigkeit zur Schraube am Schiff also mit Berücksichtigung des Vorstroms aber ohne die von der Schraube selbst erzeugten Zusatzgeschwindigkeiten.

w = Relativgeschwindigkeit (mit verschiedenen Indices).

u = $r\,\omega$ = Umfangsgeschwindigkeit.

ω = Winkelgeschwindigkeit.

c = absolute Geschwindigkeit (mit verschiedenen Indices).

c_a = Axialkomponente der Geschwindigkeit im Schraubenstrahl.

c_u = Umfangskomponente der Geschwindigkeit im Schraubenstrahl.

IV. *Leistungen.*

N_i = indicierte Maschinenleistung.
N_e = effektive Maschinenleistung.
N_w = Wellenleistung = Propellerdrehleistung.
N_o = $W_o . v$ = Schleppleistung ohne Schraube.
N_s = $S . v_p$ = Schubleistung der Schraube.

V. *Kennziffern.*

$F = \frac{v}{g^{1/2}\, l^{1/2}}$ Froudeziffer l = Bezugslänge.

$R = \frac{v\, l}{\nu}$ Reynoldsziffer v = Bezugsgeschwindigkeit.

$\vartheta = \frac{S - W_o}{S}$ Sogziffer.

$\psi = \frac{v - v_p}{v}$ Mitstromziffer.

$\Lambda = \frac{v_p}{nD}$ Fortschrittsziffer.

$\sigma = \frac{nH - v_p}{nH}$ nomineller Slip.

$\sigma_s = \frac{nH - v}{nH}$ scheinbarer Slip.

VI. *Beiwerte.*

1. allgemeine Kraftbeiwerte.

$\zeta = \frac{P}{\varrho/2\, v^2 F}$ = Kraftbeiwert.

$\zeta_a = \frac{A}{\varrho/2\, v^2 F}$ = Auftriebsbeiwert.

$\zeta_w = \frac{W}{\varrho/2\, v^2 F}$ = Widerstandsbeiwert.

$\zeta_r = \frac{W_r}{\varrho/2\, v^2 F}$ = Beiwert des Reibungswiderstandes.

$\zeta_f = \frac{W_f}{\varrho/2\, v^2 F}$ = Beiwert des Formwiderstandes.

$\varepsilon = \frac{\zeta_w}{\zeta_a}$ = Gleitzahl.

(worin v Bezugsgeschwindigkeit, F Bezugsfläche bedeutet).

2. Beiwerte der Schraubenberechnung.

a. Belastungsziffern (aus Freifahrtdiagramm).

$$K_s = \frac{S}{\varrho n^2 D^4} \text{ Schubziffer.}$$

$$K_m = \frac{M}{\varrho n^2 D^5} \text{ Momentenziffer.}$$

$$K_L = \frac{N_w}{\varrho n^3 D^5} \text{ Leistungsziffer.}$$

b. Belastungsgrade.

$$C_d = \frac{S}{\varrho v_p^2 D^2} = \frac{K_s}{\Lambda^2} \text{ Durchmesser-Belastungsgrad.}$$

$$C_n = \frac{Sn^2}{\varrho v_p^4} = \frac{K_s}{\Lambda^4} \text{ Touren-Belastungsgrad.}$$

$$C_{dm} = \frac{M}{\varrho v_p^2 D^3} = \frac{K_m}{\Lambda^2} \text{ Durchmesser-Momentengrad.}$$

$$C_{nm} = \frac{Mn^3}{\varrho v_p^5} = \frac{K_m}{\Lambda^5} \text{ Touren-Momentengrad.}$$

$$C_{dL} = \frac{N_w}{\varrho v_p^3 D^2} = \frac{K_L}{\Lambda^3} \text{ Durchmesser-Leistungsgrad.}$$

$$C_{nL} = \frac{N_w n^2}{\varrho v_p^5} = \frac{K_L}{\Lambda^5} \text{ Touren-Leistungsgrad.}$$

VII. *Wirkungsgrade.*

η = echter Wirkungsgrad (allgemein).

ξ = scheinbarer Wirkungsgrad (allgemein) (Einflußgrad).

$\eta_p = \frac{N_s}{N_w} = \frac{S v_p}{M\omega} = \frac{K_s}{K_m} \cdot \frac{\Lambda}{2\pi} = \frac{K_s}{K_L} \Lambda$ = Propellerwirkungsgrad (freifahrend).

$\eta_{ps} = \eta_p \cdot \xi_a$ = Propellerwirkungsgrad hinter Schiff.

$\eta_z = \frac{Z.v}{N_w}$ = Trossenwirkungsgrad.

$\eta_l = \frac{N_w}{N_e}$ = Wellenleitungswirkungsgrad.

$\eta_m = \frac{N_e}{N_i}$ = mechanischer Wirkungsgrad der Maschine.

$\xi_o = \frac{N_o}{N_w} = \frac{W_o v}{M\omega} = \eta_{ps}\, \xi_s = \eta_p . \xi_a . \xi_s$ = Vortriebsgütegrad.

ξ_a = Einflußgrad der Anordnung.

$\xi_s = \frac{1-\vartheta}{1-\psi}$ = Einflußgrad der Schiffsform.

Bassin de la Marine
Prof. E. G. BARRILLON

PARIS, le 27 Avril 1933.

NOTE

Pour Ir. L. Troost,
Directeur Nederlandsch Scheepsbouwkundig Proefstation,
Haagsteeg 2, Wageningen (Holland)

Monsieur le Directeur,

A. Le Comité organisateur du Congrès de La Haye a demandé aux divers directeurs de Bassins, un commentaire sommaire sur les divers titres du programme. Je réponds ci-dessus en ce qui concerne le Bassin de Paris.

a. Dans l'essai d'autopropulsion, j'estime que les résultats doivent être présentés en donnant les chiffres obtenus sur le modèle lui-même et non pas des chiffres calculés pour le navire. Un compte rendu d'Essais doit donc donner: la longueur du modèle et la nature de sa matière, la vitesse du modèle, le nombre de tours des hélices, la poussée et le couple sur les hélices, l'effort sur le dynamomètre de remorquage. Ce sont là les seuls chiffres permettant une comparaison entre les résultats obtenus dans divers bassins.

Rien n'empêche de compléter par les résultats prévus pour le navire, mais ces résultats faisant intervenir des corrections qui ne sont pas faites par des méthodes uniformes, ne se prêtent pas à des comparaisons entre les différents bassins.

b. L'étude de la turbulence artificielle ne semble pas actuellement assez avancée pour justifier une Standardisation de ce procédé. Il serait très désirable qu'une comparaison d'essais en autopropulsion avec et sans turbulence artificielle fût faite sur des modèles de formes assez différentes, de dimensions variables et surtout avec diverses distances entre les hélices et la carène.

c. Il serait nécessaire de définir un navire idéal qui aurait les mêmes dimensions que le navire réel et qui serait supposé être construit en paraffine. Ce navire idéal est le seul pour lequel la méthode des modèles donne une prévision. Pour le navire idéal, une formule générale de correction de frottement, a un sens bien défini. Les augmentations dues à l'irrégularité de la formation des tôles, aux cans de tôles, aux têtes de rivets, à la rugosité de la peinture, ne peuvent être standardisées.

d. Le type de formule $\frac{R}{\varrho \frac{v^2}{2} A}$ n'est pas employé couramment par les ingénieurs. Il est intéressant au point de vue scientifique, mais pour les ingénieurs il a l'inconvénient de faire intervenir A qui n'est pas déterminé suivant des règles uniformes et est moins bien défini que le volume de carène.

e. Toutes les constantes sont bonnes à condition d'être de dimensions nulles et de n'utiliser que les mêmes unités pour les diverses quantités y figurant. Il est par suite recommandé de ne pas employer simultanément une longueur en pieds et une vitesse en noeuds.

Pour les hélices, il serait intéressant de discuter l'emploi d'un diagramme dans lequel les abscisses sont les valeurs de $\frac{P}{\varrho n^2 D^4}$ et les ordonnées les valeurs de $\frac{C}{\varrho n^2 D^5}$, diagramme sur lequel une seule courbe graduée en valeurs de $\frac{n D}{V}$ représente complètement le fonctionnement d'une hélice, ce qui facilite les rapprochements entre modèle et réel, ou les comparaisons de modèles à diverses échelles.

B. La formule employée au Bassin de Paris pour le frottement est la même pour le modèle et pour le navire idéal. C'est une formule empirique établie à la suite des travaux de FROUDE:

$$R_f = \left(0.1392 + \frac{0.258}{2.68 + \Lambda} \right) A.V^{1.825}$$

Nous avons conservé cette formule qui ne correspond pas à l'état actuel de nos connaissances sur le frottement, parce que nous avons jugé que nos archives seraient plus difficiles à utiliser, si, à chaque époque, nous changions la formule de frottement.

La longueur du modèle est définie comme longueur de la flottaison au déplacement et à l'assiette du plan.

NOTES ON THE METHODS USED IN THE ,,VASCA NAZIONALE", ROME, AND REMARKS REGARDING THE DESIRABILITY OF STANDARDISATION

The following is a short description of the methods employed during experiments at the National Tank in Rome.

a. SELF PROPELLING

The dynamometers were designed and made by Dr Gebers in 1928. They are placed in the model, and register thrust, torque and number of revolutions. Power is supplied by an electro motor placed sometimes in the model and in other cases, for reasons of weight or stability, carried on the towing carriage. In the latter case, the motor and dynamometer are connected by a sliding coupling, which allows the model, free in the direction of the run. The propeller is run up to the requisite number of revolutions to give the model the same speed as the carriage. The model is weighted to take up the additional resistance, which has been calculated previously, as usual in Continental tank practice. The frictional resistance of the model and ship are calculated respectively by expressions:

$$r_a = \lambda_m . \Theta . s_b . v^{1,825}$$
$$R_a = 1.026 . \lambda_n \, S_b . (0,5144 . V)^{1,825}$$

where: λ_m = friction co-efficient of the model varying with the maximum length of the wetted surface, and which has the value as found in Table 1.

Θ = Temperature correction co-efficient calculated from the formula $\Theta = \left(\frac{\nu_{t^\circ}}{\nu_{15^\circ}}\right)^{0,175}$

where ν_{t° and ν_{15° are respectively viscosity co-efficient at t°, the experiment temperature, and at temperature of 15 degrees Centigrade. The values of Θ are given in Table 3.

s_b = Total wetted surface in m^2 for normal forms by motionless ship, and for seaplanes running at speed.

v = Speed of model in metres per second.

r_a = Frictional resistance of the model at experiment temperature. 1,026 = specific gravity of sea water.
λ_n = Friction co-efficient for ship, varying with the length as given in Table II.
S_b = Wetted surface of the ship in m^2.
V = Speed of ship in knots.
R_a = Frictional resistance of ship at a temperature of 15^0 centrigrade in kilograms.

The torque, and consequently the shaft horse power, are calculated at the propeller. The thrust is taken up at the thrust block of the dynamometer. By the calculation of the thrust therefore, neither the weight and the inclination of the shafting, nor the rake of the shafting due to the movement of the model, are taken into consideration. The towing resistance of the propeller boss and the torque absorbed are also neglected, as opposed to the practice in Washington. (See Comm. Harold E. Saunders, Society of Naval Architects and Marine Engineers, Nov. 1932).

The models are of paraffin wax and between 5 and 7 metres long. The propellers are made of a tin alloy, with diameters ranging from 13 to 20 centimetres.

No system of ,,Hanging up'' models on account of the excessive weight has yet been found necessary.

The values of the revolutions and shaft horse powers are taken from the experimental data without any addition for service conditions.

b. LAMINAR FLOW

No attempt is made to correct for this, except keeping the model as large as reasonable. As stated above, the models are between 5 and 7 metres long, and it is seldom that models shorter than this are used.

c. FRICTION CORRECTION FOR TEMPERATURE

Temperature correction is made as set out under *a*)

d. PUBLICATION OF RESULTS

The results of the experiments in the Rome Tank are set out for the customers in the form of diagrams. In addition to this, a book is printed annually containing the result of the year's work as far as the interests of the clients permit, the first number of which appeared in 1931, and the second of which is at the moment being printed. These books (Annali della Vasca Nazionale per le Esperienze di Architetture Navale in Roma) can be obtained upon application.

The dimensions of the models are published, the original results of the experiments i.e., total resistance, thrust, torque, number of revolutions as also the values of frictional resistance and frictional correction, effective horse power, and all the calculated values for the ship.

In the second volume now in preparation, for the convenience of the student, the following particulars are also being supplied in numerical form: total resistance of the model and the power in E.H.P. for the ship, revolutions of propeller p. sec. and torque of the model, and revolutions p. min. and power in S.H.P. for the ship. No use is made of the formula $\frac{R}{\varrho A.V.^2}$ in the resistance calculation, as, so as previously stated, the values of the total resistance of the model are known. For purposes of comparison only, are given:

1°. the main dimensions of the model reduced to one cubic metre displacement.

2°. The relative wave making power also reduced to one cubic metre of displacement:

$\frac{PCE_0}{1.026\, \Delta_1^{7/6}}$ (PCE_0 = wave power for the vessel in H.P.; Δ_1 = volume of vessel in m^3).

3°. The corresponding relative speed in Knots: $\frac{V}{\Delta_1^{1/6}}$

4°. The relative friction power $\frac{PCE_a}{1{,}026\, \Delta_1^{53/48}}$ in functions of relative speed, according to Telfer's formula (E.V. Telfer „On the Presentation of Ship Model Experiment Data". Transactions of the North East Coast Institution of Engineers and Shipbuilders, Newcastle-on-Tyne, 1923).

e. SYMBOLS AND CONSTANTS

The Rome Tank avoids the use of symbols and constants as much as possible, but when unavoidable the following are used:

	Model		Ship	
	Symbol	Unit	Symbol	Unit
Scale	a			
Length over all	l_c	m	L_c	m
Length B.P.	l	m	L	m
Length on W.L.	l_g	m	L_g	m
Breadth extreme	n	m	N	m
Breadth on W.L.	n_g	m	N_g	m
Draft forward	i_{AV}	m	I_{AV}	m
Draft aft	i_{AD}	m	I_{AD}	m
Draft midships	i	m	I	m
Volume	δ	m^3	Δ_1	m^3
Displacement	δ	T	Δ	T
Wetted surface	s_b	m^2	S_b	m^2
Area of immersed midship section	b	m^2	B	m^2
Area of water line	g	m^2	G	m^2
W. L. co-efficient	φ_g	$=\dfrac{g}{l_g.n_g}$	$=\dfrac{G}{L_g.N_g}$	
Midship section co-efficient	φ_b	$=\dfrac{b}{n.i}$	$=\dfrac{B}{N.I}$	
Prismatic co-efficient	φ_l	$=\dfrac{\delta}{l_c.b}$	$=\dfrac{\Delta_1}{L_c.B}$	
Block co-efficient	φ_t	$=\dfrac{\delta}{l_c.n.i}$	$=\dfrac{\Delta_1}{L_c.N.I}$	
Max. diam. propellers	d	m	D	m
Diam. of propeller boss	d_m	m	D_m	m
Face pitch	p_g	m	P_g	m
Number of blades	Z			
Expanded area	s_p	m^2	S_p	m^2
Developed area	s_v	m^2	S_v	m^2
Pitch ratio	p/d		P/D	
Blade area ratio	$\dfrac{s_p}{\pi\dfrac{d^2-d_m^2}{4}}$		$\dfrac{S_p}{\pi\dfrac{D^2-D_m^2}{4}}$	
Speed of advance	v	m/sec	V	knots
Total resistance	r	kg	R	T
Frictional resistance	r_a	kg	R_a	T
Wave making resistance	r_o	kg	R_o	T
Frictional co-efficient	λ_m		λ_n	
Water temperature	t^0	cent.	15^0	

	Model		Ship	
	Symbol	Unit	Symbol	Unit
Correction co-efficient for temperature	Θ			
Effective H.P.				
frictional			PCE_a	HP
wave making			PCE_0	HP
total			PCE	HP
Speed relative of screw ...	v_1	m/sec	V_1	knots
Revolutions..............	n	per sec.	N	per min.
Thrust	s	kg	S	T
Torque	m	kg.m	M	T.m
Thrust coefficient.........	S_c	$= \frac{s}{v_1^2 . d^2}$	$= \frac{S}{V_1^2 . D^2}$ 3683	
Torque coefficient	M_c	$= \frac{m}{v_1^2 . d^3}$	$= \frac{M}{V_1^2 . D^3}$ 3683	
Revolution coefficient	N_c	$= \frac{n . d . 60}{v_1}$	$= \frac{N . D .}{V_1}$ 1,945	
Propeller efficiency open ..	η	$= \frac{60 . S_c}{2 \pi N_c . M_c}$		
Tension correction in self-propulsion experiments	δ_r	kg		
Shaft horse power			PCA	HP
Wake co-efficient	η_s	$= \frac{v}{v_1}$	$= \frac{V}{V_1}$	
Thrust deduction co-efficient	η_r	$= \frac{r - \delta_r}{s}$	$= \frac{R}{S}$	
Apparent propulsive efficiency................	η_a	$= 0{,}159 \frac{s\ v}{n . m}$	$= 6{,}8586 \frac{SV}{PCA}$	
Total propulsive efficiency.	η_t	$= 0{,}159 \frac{(r - \delta_r)}{n . m} v$	$= \frac{PCE}{PCA}$	

In the formulas the specific gravity of the tank water is taken at 1000 kg/m³ and salt water at 1026 kg/m³.

In general, the „Vasca Nazionale di Roma" is in favour of the idea of uniformity in experimental systems and their publication, where this will not lead to confusion in the tanks and amongst their clients. With this latter end in view, the following points are brought forward:

a. SELF PROPELLING

In some tanks, a correction is made to the apparatus as an allowance for the frictional difference between model and ship, while in others this is ignored. Practical comparison between tank results and actual performance at sea can only decide which of these systems is to be preferred, but, as such comparisons are not available, or at any rate, in sufficient numbers, we give preference to the first, because:

1. It has a theoretical basis, which the second has not. If Froude's method which takes into consideration the different law of comparison regarding friction and wave making resistance, is used for calculating the towage resistance for ship, why should this not be done with self propelled tests? In this case it would be more logical to compare the towage of both ship and model without frictional correction.

2. If a series of models of the same vessel to different scales were tested, a varying set of results would be obtained as regards total resistance, which as has often been demonstrated, do not vary as the cube of the scale. For example; the resistance of the model of a vessel of 35 — 40.000 tons at 25 knots, 8 metres long, would be 7,15 times, instead of 8 times the resistance of the 4 metres model.

3. The friction correction is a percentage of the total resistance that varies considerably with the same model at different speeds (from 0,375 with a speed of 15 knots, to 0.194 with a speed of 42 knots, results obtained with the model of a light cruiser in the Tank at Rome).

The ratio between the S.H.P. calculated by the tanks adopting the aforesaid variable correction and the S.H.P. actually measured at sea is appearing to be fairly constant, which is a prove of the rightness of the same method.

4. In any case, a practical co-efficient must be obtained for tank and sea performance.

Again preference is given to the practice followed by the majority of tanks.

b. LAMINAR FLOW

The only way to reduce this is the using of models as large as possible at the highest speeds.

c. FRICTIONAL CORRECTION FOR TEMPERATURES

The ,,Vasca Nazionale di Roma'' regards this point as uncertain. There are great differences between summer and winter temperatures. The tank has at times tested the same models at different temperatures, and obtained inexplicable results. In some cases with large variations in temperature, no variation in resistance was found. It is therefore thought desirable to obtain more information on this point before discussing standardisation.

d-e. PUBLICATION OF RESULTS. SYMBOLS AND CONSTANTS

The Rome tank avoids the use of symbols wherever possible, and favours the writing of the terms out in full, but considers that standardisation can be useful. The standardisation of constants will be of more use, and better still, if the discussion is restricted to the ground covered by the circular, the use of a formula and of a common denominator for those values, mentioned in the foregoing pages as wake co-efficient, thrust deduction co-efficient, propeller efficiency, quasi-propulsion co-efficient, propulsion co-efficient, thrust, torque and revolutions, and the constant for the last three with free running propeller. It frequently occurs that the same denominator is used for different values, but the unit of measure causes the most trouble, i.e. transforming English figures into the metrical system and vice versa.

For this reason, important experimental results often do not receive the attention they deserve.

The Rome Tank does not underestimate the difficulty of changing a system, as each tank has it routine-system of presentation, clients, etc., who would not welcome innovations. Almost every tank publishes a yearly report of its work together with results, and it is suggested that in addition to the publication of the report in the usual form, a second be added using the standardised forms of symbols, constants etc. for the convenience of international comparison. This does not present any grave difficulties, nor great alteration, and the original system of presentation is adhered to for the convenience of clients and students who are accustomed to them.

Table 1. VALUES OF λ_m

„Frictional co-efficient at temperature of 15° C. for models".
Formula: $r_a = \lambda_m s_b . v^{1,825}$.
r_a = frictional resistance of model in kg.
s_b = wetted surface of model in m^2
v = speed of model in metres per second.

Maximum length of bottom	λ_m	Maximum length of bottom	λ_m	Maximum length of bottom	λ_m
		4,20	0,17699	6,20	0,16803
		4,40	0,17580	6,40	0,16740
2,60	0,18937	4,60	0,17467	6,60	0,16684
2,80	0,18750	4,80	0,17374	6,80	0,16630
3,00	0,18574	5,00	0,17270	7,00	0,16579
3,20	0,18401	5,20	0,17180	7,20	0,16521
3,40	0,18239	5,40	0,17101	7,40	0,16470
3,60	0,18085	5,60	0,17019	7,60	0,16426
3,80	0,17950	5,80	0,16940	7,80	0,16380
4,00	0,17822	6,00	0,16870	8,00	0,16330

Table 2. VALUES OF λ_n

„Frictional co-efficient at 15° C. temperature for ships."
Formula: $R_a = \lambda_n S_b . 1,026. (0,5144 . V)^{1,825}$.
R_a = frictional resistance of ship in kg.
1.026 = S.G. of salt water.
S_b = wetted surface of ship in m^2.
V = speed of ship in knots.

Maximum length of bottom	λ_n	Maximum length of bottom	λ_n	Maximum length of bottom	λ_n
10	0,15935	110	0,14182	210	0,13884
20	0,15080	120	0,14147	220	0,13858
30	0,14740	130	0,14114	230	0,13832
40	0,14565	140	0,14082	240	0,13806
50	0,14460	150	0,14050	250	0,13780
60	0,14390	160	0,14020	260	0,13756
70	0,14340	170	0,13992	270	0,13733
80	0,14297	180	0,13964	280	0,13711
90	0,14256	190	0,13936	290	0,13690
100	0,14218	200	0,13910	300	0,13670

Table 3. VALUES OF Θ

„Correction co-efficient for the frictional resistance for the temperatures."

Formula: $r_t = r_{15} \cdot \left(\frac{\nu_t}{\nu_{15}}\right)^{0,175}$

r_{15} = frictional resistance at 15° C.
r_t = frictional resistance at experiment temperature t°.
ν_{15} = viscosity co-efficient at 15° C.
ν_t = viscosity co-efficient at temperature t°.

Water-temp.	Θ	Water-temp.	Θ
1	1,0742	21	0,9743
3	1,0620	23	0,9661
5	1,0500	25	0,9580
7	1,0388	27	0,9503
9	1,0284	29	0,9426
11	1,0186	31	0,9349
13	1,0095	33	0,9273
15	1,0000	35	0,9196
17	0,9915		
19	0,9828		

Imp. Japanese Navy

SUBJECT OF DISCUSSION:

A. ABOUT THE METHOD OF „TENSION CORRECTION" IN CASE OF „SELF-PROPULSION EXPERIMENT"

Answer: As it is reasonable to perform „tension correction" in case of self propulsion experiment, we, of the experimental tank station of the Japanese Naval Research-Institute, generally apply a correction such as FROUDE's, (like several other research-institutes on the European continent). But we have had some uncertainty respecting the friction correction by FROUDE's formula. So we carried out an exhaustive study about frictional resistance by the way of towing several rectangular plates of length: shortest 2′–0″ to longest 75′–0″, several ship's models of length 5′–0″ to 26′–0″, as well as a third class destroyer of length 235′–0″. After a period of five years study recently, we have come to a certain conclusion. The result shall be published on a suitable occasion.

B. ABOUT THE STANDARD METHOD TO AVOID „LAMINAR FLOW"

Answer: The problem could be easily solved in comparing the dimension of ship's models and towing speed. The experimental tank of the Japanese Navy has the following dimensions: Breadth 12,5 m, Depth 6,5 m, Sectional area about 80 m^2. The length of ordinary models is 6,5 m to 8 m. Therefore if the towing speed is not smaller than 1 m/sec, generally we can avoid the laminar flow. Also if we consider the dimensions of models and the experimental tank, we may assume the side and bottom effects to be inappreciable.

The artificial methods to avoid laminar flows, such as the application of wires or nets, are not used by us, because it is difficult, to have similar conditions in actual ships.

C. ABOUT THE GENERAL FORMULA FOR THE TEMPERATURE-CORRECTION OF THE FRICTIONAL RESISTANCE

Answer: In the experimental tank of the Japanese Navy, we have applied hitherto the following formula for the temperature-correction of frictional resistance:

$$R_f = \gamma f \left\{ 1 + \underbrace{0.0043\,(15^\circ - \theta)}_{\substack{\text{temperature correct-}\\ \text{ing factor taking}\\ 15^0\text{ C as standard}\\ \text{temperature.}}} \right\} S.V^{1.825}$$

But as it was necessary to confirm the reliability of this formula, we have performed a test as a part of the researches for frictional resistance as mentioned under subject A, from 1926 to 1927, in a small tank (length 100′–0″, breadth 6′–0″, depth 5′–0″) on brass-plates and wooden models. The water-temperature of the tank was varied from 4° C. to 60° by admixing ice or steam.

From this towing experiment we found that the frictional resistance varies proportionally to $\nu^{0.165}$, where ν is the kinematic viscosity. The viscosity and density are measured in each occasion. From this result, taking 15° C. as standard temperature, the correcting percentage of frictional resistance for each temperature is shown in Fig. 1.

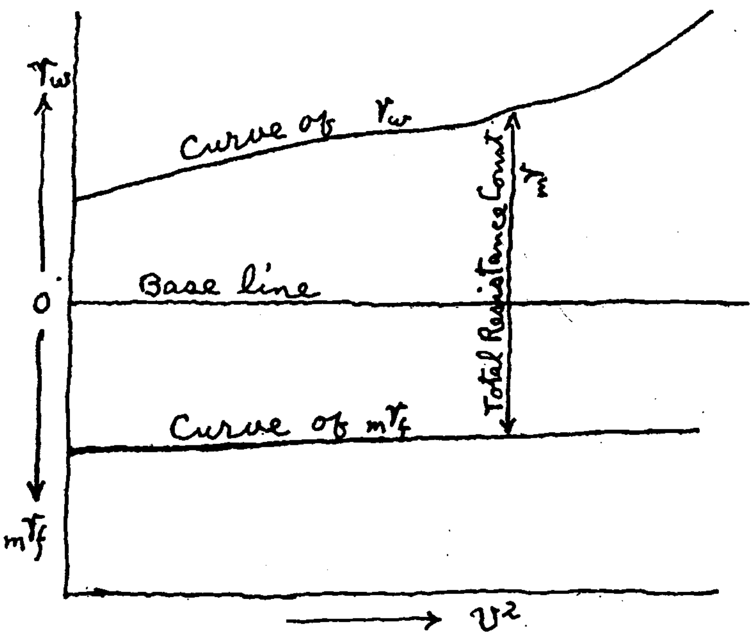
D. ABOUT THE METHOD TO EXPRESS THE RESISTANCE-CURVE BY A FORMULA

We have no objection against the principle of adoption of „dimensionless representation'', though there would be the problem which number REYNOLD's or FROUDE's we should take as a base. We are of the opinion that the most rigorous dimensionless representation for model experiment is to show the value of $R_w/\varrho\triangle^{2/3}V^2$ on the base of $V^2/\triangle^{1/3}g$, because originally the method of FROUDE's model experiment has the purpose to get the value of f in the formula

$$\frac{R_w}{\varrho\triangle^{2/3}V^2} = f\left(\frac{V^2}{\triangle^{1/3}g}\right)$$

where Rw = residual resistance
$\triangle$ = volume displacement
g = acceleration due to gravity
ϱ = density of water
V = velocity of model

In our experimental tank we adopt the following practical method:



(prefix m or s show model or ship, respectively)

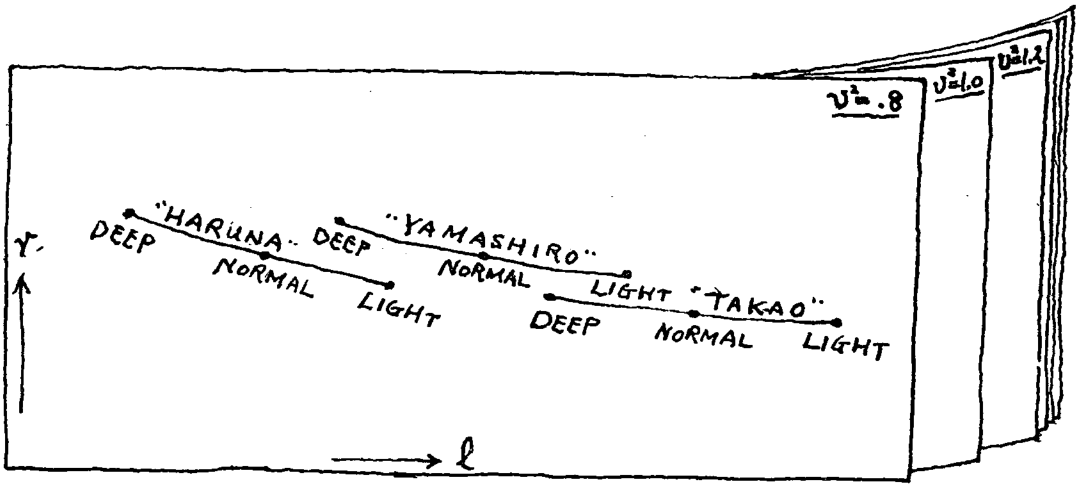
$$v^2 = \frac{_m V^2}{_m \triangle^{1/3} g} = \frac{_s V^2}{_s \triangle^{1/3} g}$$

$$r_w = \frac{_m R_w}{_m \varrho \, _m \triangle^{2/3} \, _m V^2} = \frac{_s R_w}{_s \varrho \, _s \triangle^{2/3} \, _s V^2}$$

$$_m r_f = \frac{_m R_f}{_m \varrho \, _m \triangle^{2/3} \, _m V^2}$$

To compare the results of the resistance experiment of several ships and for the purpose of reference for future designs of ship-types, in our experimental tank, the results of experiments are all converted into *a standard displacement* and represented dimensionless as described before, while in this case the resistance constant is shown only as total resistance constant r, but not divided into r_f and r_w. And the r for the same v^2 value for variable ships are shown on the base of length constant of respective ship $l = \left(\frac{L}{\triangle^{1/3}}\right)$

The following sketch is an example of the diagram:



As shown in the above sketch, the r value of every ship for the same v^2 value is collected in a sheet with the name of ships. Further, in this case we choose 8200 tons as a standard displacement. The reason of the figure 8200 tons is as follows:

$20^3 \times$ Specific gravity of sea water $1.025 = 8.200$

and by this way the length of respective ships converted into standard displacement, is easily got as 20 l because

$$l = \frac{L}{\triangle^{1/3}} = \frac{L}{\left(\frac{\text{Weight dispt.}}{\text{Specific gravity of sea water}}\right)^{1/3}}$$

$$= \frac{L}{20 \times \frac{(\text{Specific gravity of sea water})^{1/3}}{(\;,,\;\;,,\;\;,,\;\;,,\;\;,,\;)^{1/3}}} = \frac{L}{20}\text{ ; or } L = 20\,l.$$

And similarly

$$B = 20b;\; D = 20d;\; S = 20^2\, s;\; V = \sqrt{20g}\,.\,v;$$

where b = breadth constant
d = draught constant
s = wetted surface area constant
v = velocity constant

E. ABOUT THE STANDARD SYMBOLS AND CONSTANTS

About the symbols and constants to represent the result of tank experiments, we adopt the same symbols and constants as determined by „The Experimental Tank Committee In Japan" which are shown in the appendix. The selection of the signs is generally based on the following principle:

1. The symbols for the subjects which have dimensions are represented by capital letters while the subjects without dimension are represented by small letters, but

2. The symbols which have been used hitherto (mainly in English speaking countries) are left as they are, in case they give no great inconvenience.

3. To represent the correspondence between actual ships and models, the symbols for actual ships and models are shown with the prefix s and m, respectively.

The constants which represent the result of resistance experiment (already described in D) are shown together as follows:

velocity constant .... $v = \dfrac{V}{\sqrt{\triangle^{1/3} g}}$

resistance constant .... $r = \dfrac{R}{\varrho \triangle^{2/3} V^2}$

frictional resistance constant ...... $r_f = \dfrac{R_f}{\varrho \triangle^{2/3} V^2}$

residual resistance constant .... $r_r = \dfrac{R_r}{\varrho \triangle^{2/3} V^2}$

length constant .... $l = \dfrac{L}{\triangle^{1/3}}$

wetted surface area constant ..... $s = \dfrac{S}{\triangle^{2/3}}$

sectional area constant .... $a = \dfrac{A}{\triangle^{2/3}}$

We presume the suitable forms for the constants to represent the result of propeller experiments, are as follows:

speed of advance constant.... $v_1 = \dfrac{V_1}{N\ D}$

thrust constant .... $t = \dfrac{T}{\varrho N^2 D^4}$

torque constant .... $q = \dfrac{Q}{\varrho N^2 D^5}$

screw efficiency .... $\eta_p = \dfrac{t\ v_1}{2 \pi q}$

The individual result of propeller experiment is represented by taking v_1, on abscissa and t, q, η_p on the ordinate.

Also the suitable forms to express the result of *series-experiments* of propellers are (by the two methods) shown as follows:

1. *Thrust system diagram* (see Fig. 2).

On a logarithmic section paper, taking v_1, on abscissa and t on ordinate, the results of series experiments are represented in collected form. Further we plot the equiefficiency curve or graph with dimensionless scale for

$$\frac{P_t}{\varrho N^3 D^5} = t\, v_1 \;;\quad \frac{P_t}{\varrho D^2 V_1^2} = \frac{t}{v_1^2} \;;\quad \frac{P_t N^2}{\varrho V_1^5} = \frac{t}{v_1^4}$$

as well as the dimension scale for

$$V_1 \text{ (knots)} = 12.96 v_1$$

$$T \text{ (for sea water in kg)} = 74412\ t$$

$$\text{T.H.P. (for sea water)} = 6616 \frac{P_t}{\varrho N^3 D^5}$$

N (per min) and D (in m)

these are in the standard conditions of

$$N = 100 \text{ per minute};\ D = 4 \text{ metres}$$

$$\varrho = 104.6 \frac{\text{Kg sec}^2}{\text{m}^4} \text{ (for sea water)}$$

2. *Torque system diagram* (see Fig. 3).

On a logarithmic section paper, taking v_1 on abscissa, q on ordinate, the results of series experiments are expressed in collected form. Further we plot the equiefficiency curve or graph with the dimensionless scale for

$$\frac{P_p}{\varrho N^3 D^5} = 2\pi q \;;\quad \frac{P_p}{\varrho D^2 V_1^3} = 2\pi \frac{q}{v_1^3}$$

$$\frac{P_p N^2}{\varrho V_1^5} = 2\pi \frac{q}{v_1^5} \;;\quad \frac{Q}{\varrho D^2 V_1^2} = \frac{q}{v_1^2}$$

as well as the dimension scale for

$$V_1 \text{ (knots)} = 12.96\ v_1$$

$$Q \text{ (for sea water in kg.m)} = 297.692\ q$$

$$\text{P.H.P. (for sea water)} = 6616 \frac{P_p}{\varrho N^3 D^5}$$

N (per min.) and D (in m)

while these 5 terms are of standard condition same as shown in the thrust diagram.

Fig. 4 and Fig. 5 show the example of the two methods above mentioned.

OTHER SUBJECT OF DISCUSSION

1. THE VALUE OF A CONSTANT USED IN THE CASE OF FROUDE'S FRICTION CORRECTION

In our experimental tank the variation of the coefficient of friction by length is taken from the table in the next page. While the formula for frictional resistance is:
$R_f = f.\ S.\ V^{1.92}$ for the plates shorter than 2.44^m (= 8 feet)
$R_f = f.\ S.\ V^{1.825}$ for the plates longer than 2.44^m (= 8 feet)
where, R_f in kg; S in m^2; V in knots.

The variation of coefficient of friction by temperature is shown already in C, so it is omitted here.

2. MEASURING THE LENGTH OF THE SHIP IN TANK EXPERIMENT

As far as referring to resistance problem, we presume Lpp has no meaning, therefore we apply L.W.L.

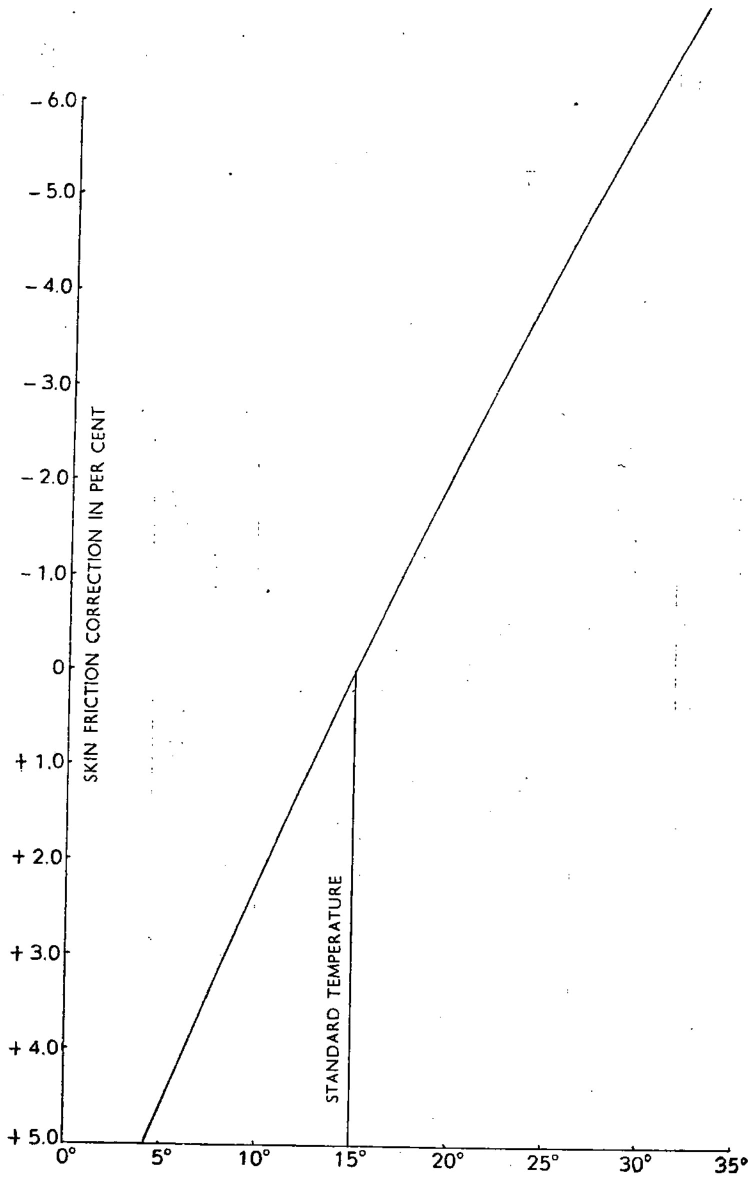
COEFFICIENT OF FRICTION

(Standard temperature = 15^0 C. Length (L) in metre)

L	0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9
2						.055988	.055702	.055419	.055134	.054856
3	.054600	.054349	.054108	.053872	.053646	3426	3213	3007	2805	2613
4	2422	2235	2052	1874	1705	1541	1383	1230	1078	0933
5	0796	0661	0535	0408	0290	0172	0058	.049946	.049837	.049731
6	.049627	.049526	.049429	.049330	.049235	.049141	.049047	8958	8868	8780
7	8693	8612	8532	8453	8378	8303	8230	8159	8090	8025
8	7960	7898	7838	7777	7720	7662	7607	7552	7498	7446
9	7393	7342	7294	7243	7192	7146	7098	7052	7006	6961
10	6915	6871	6828	6785	6742	6702	6660	6619	6579	6541

L	0	1	2	3	4	5	6	7	8	9
10	.046915	.046503	.046141	.045822	.045540	.045284	.045057	.044852	.044666	.044497
20	4341	4201	4072	3955	3847	3749	3657	3574	3494	3424
30	3357	3290	3230	3175	3120	3070	3022	2977	2934	2894
40	2854	2817	2782	2747	2718	2686	2657	2630	2603	2576
50	2552	2528	2504	2482	2460	2440	2419	2399	2379	2361
60	2342	2325	2307	2289	2272	2256	2239	2224	2208	2192
70	2177	2163	2147	2133	2120	2106	2093	2080	2066	2053
80	2040	2027	2015	2003	1990	1978	1966	1955	1943	1931
90	1920	1909	1898	1887	1876	1866	1855	1845	1834	1825
100	1815	1804	1795	1785	1775	1766	1756	1746	1736	1727
110	1717	1707	1699	1689	1679	1670	1662	1652	1642	1632
120	1623	1615	1605	1596	1586	1577	1567	1559	1549	1539
130	1530	1522	1513	1504	1493	1484	1476	1467	1457	1447
140	1438	1430	1421	1412	1402	1393	1385	1375	1367	1357
150	1349	1340	1331	1321	1313	1305	1295	1286	1277	1269
160	1261	1252	1244	1235	1227	1218	1210	1202	1193	1184
170	1176	1167	1158	1151	1143	1134	1125	1118	1110	1101
180	1094	1086	1078	1071	1062	1054	1047	1039	1032	1025
190	1017	1009	1002	.040994	.040986	.040979	.040972	.040965	.040957	.040950
200	.040942	.040934	.040927	921	913	907	899	892	885	878
210	872	864	857	850	844	838	831	823	817	810
220	804	797	791	784	777	771	764	758	751	745
230	738	731	726	719	713	707	701	694	689	682
240	677	670	664	658	652	646	641	635	629	623
250	617	611	605	599	594	589	583	577	572	566
260	561	555	550	544	539	534	528	524	518	513
270	508	503	497	493	487	482	478	472	467	462
280	458	454	449	444	438	434	430	425	420	415
290	410	406	401	397	392	388	383	379	375	370
300	366	362	357	353	348	344	340	335	331	328
310	323	319	315	311	307	303	298	295	290	287
320	283	278	275	272	267	263	260	257	252	249
330	246	241	238	234	230	227	224	220	216	213
340	209	206	202	199	195	192	189	184	181	178
350	174	171	168	165	161	158	155	151	148	145
360	142	138	135	133	130	126	123	120	117	114

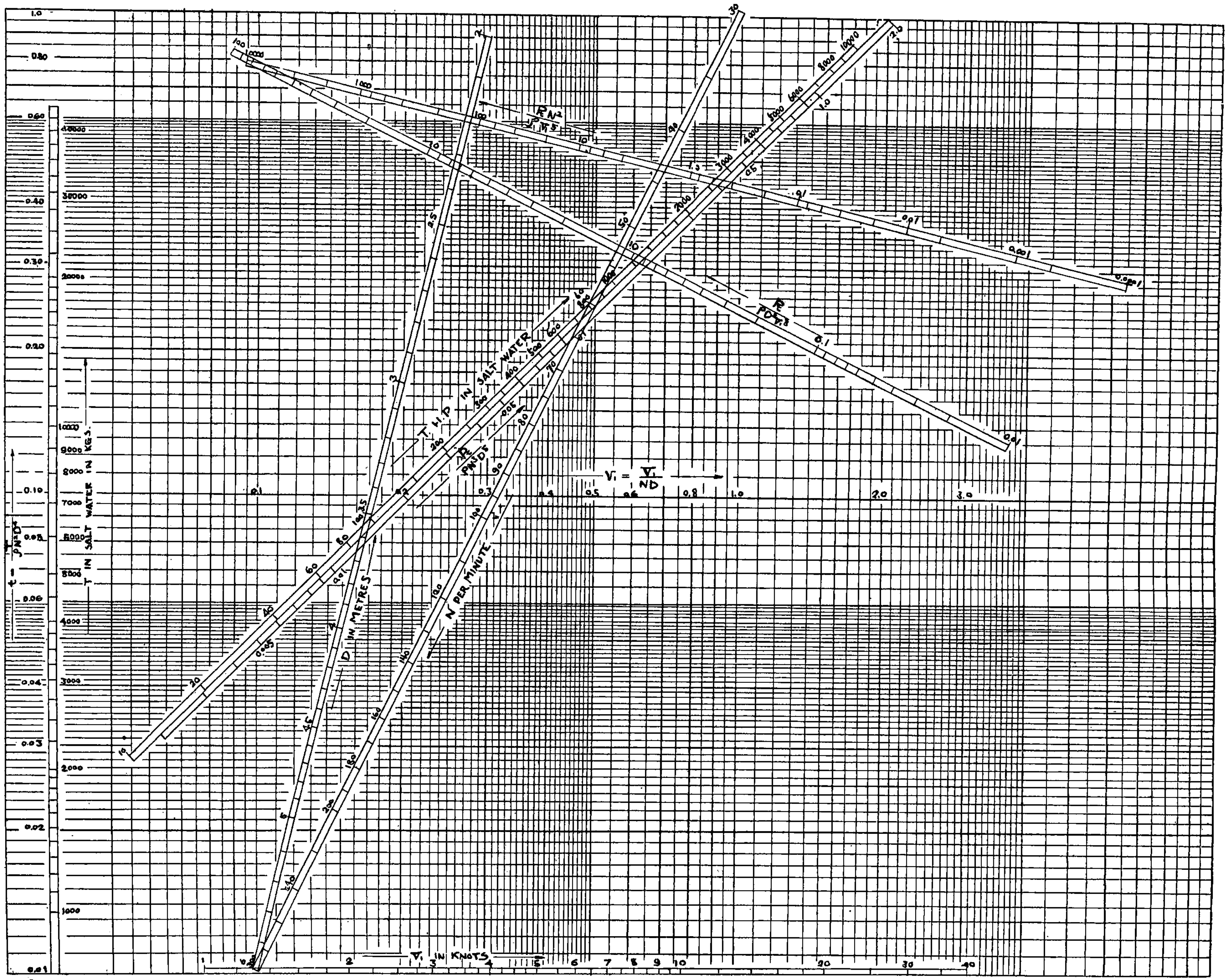
FIG. 1. SKIN FRICTION CORRECTION DUE TO TEMPERATURE DIFFERENCE



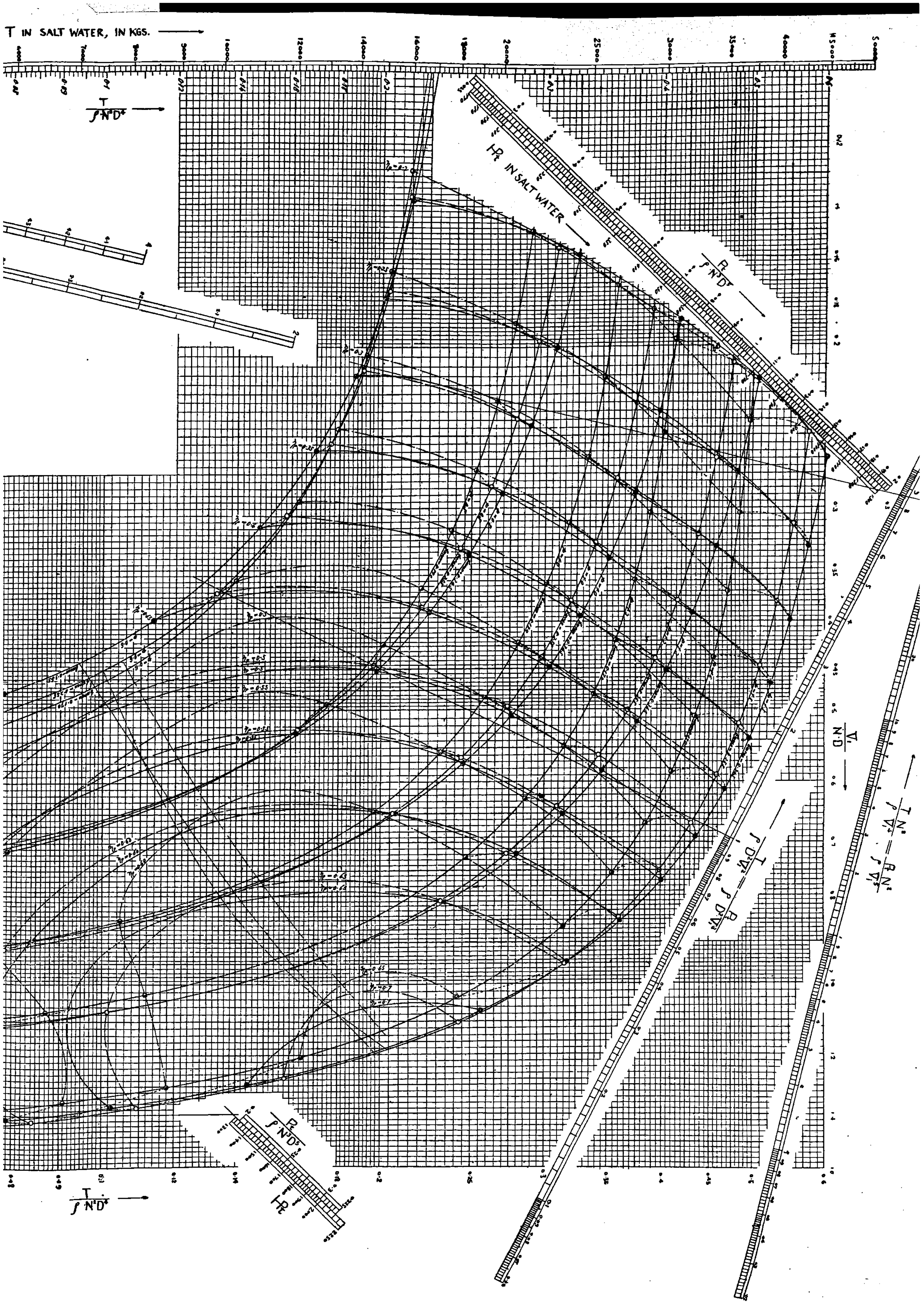
Temperature of water in centigrade

THRUST SYSTEM DIAGRAM

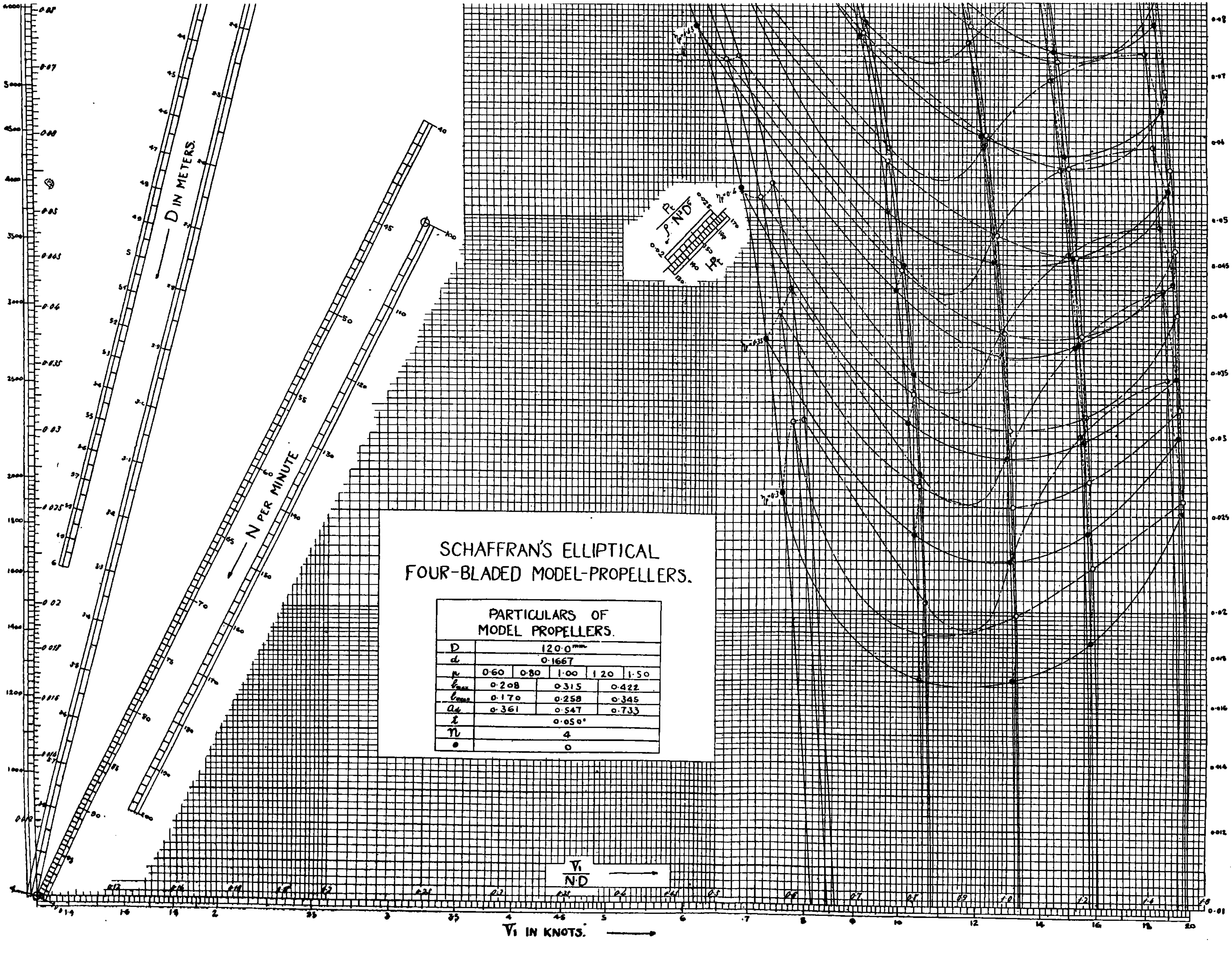
FIG. 2



(試驗水槽成績表現法調査委員會　第二回報告　附圖)



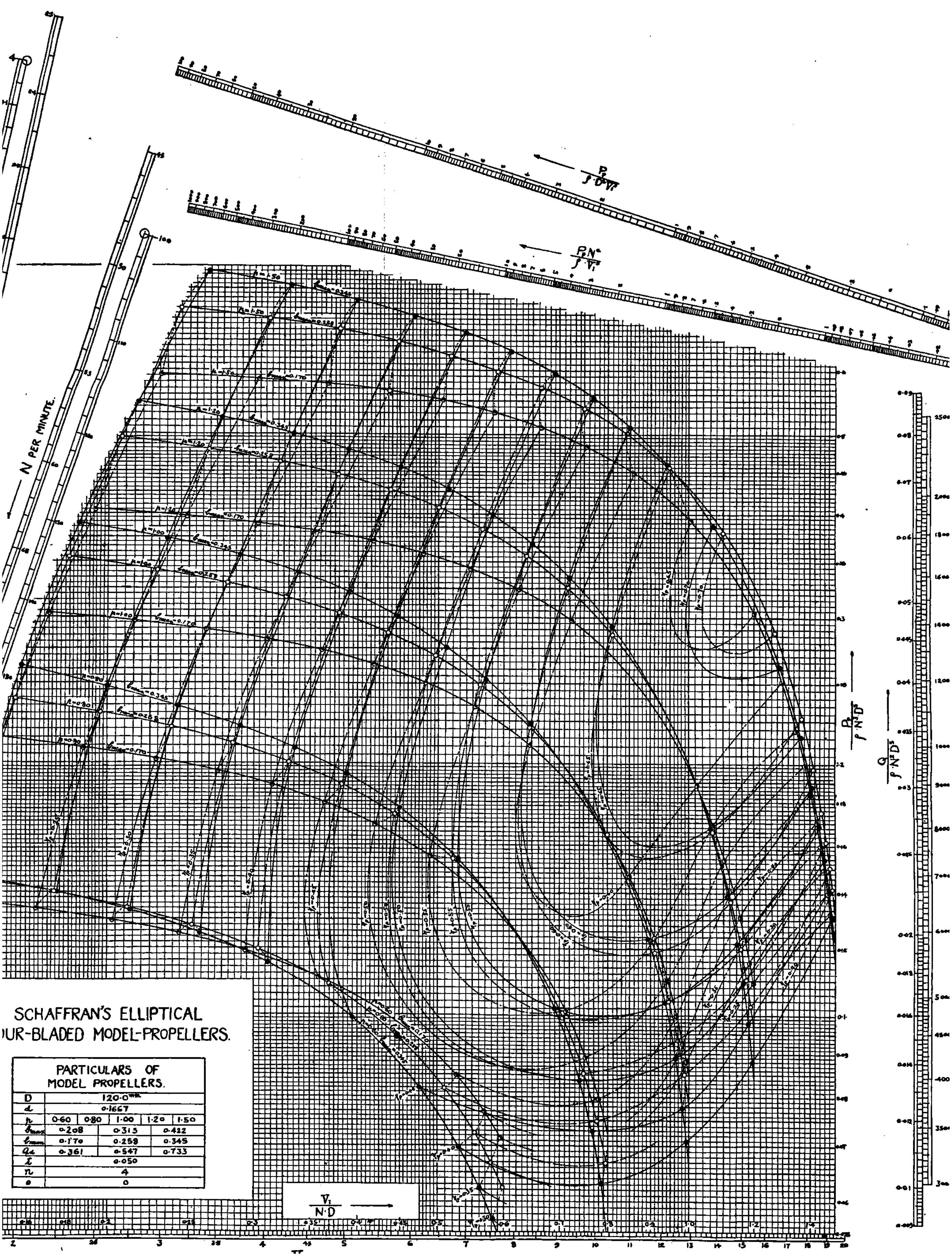
T IN SALT WATER, IN KGS.
HP IN SALT WATER
$\frac{T}{\rho N^2 D^4}$
$\frac{P_t}{\rho N^3 D^5}$
$\frac{V_1}{N \cdot D}$
$\frac{T}{\rho D^2 V_1^2} = \frac{P_t}{\rho D^2 V_1^3}$
$\frac{T N^2}{\rho V_1^4} = \frac{P_t N^2}{\rho V_1^5}$



PARTICULARS OF MODEL PROPELLERS.

D	120.0mm				
d	0.1667				
p	0.60	0.80	1.00	1.20	1.50
b_{max}	0.208		0.315		0.422
b_{mean}	0.170		0.258		0.345
a_d	0.361		0.547		0.733
i	0.050°				
n	4				
e	0				

（試驗水槽成績表現法調査委員會　第二回報告　附圖）



SCHAFFRAN'S ELLIPTICAL
[F]OUR-BLADED MODEL-PROPELLERS.

PARTICULARS OF MODEL PROPELLERS.					
D	120·0 mm.				
d	0·1667				
p	0·60	0·80	1·00	1·20	1·50
b_{max}	0·208		0·315		0·422
b_{min}	0·170		0·258		0·345
q_a	0·361		0·547		0·733
t	0·050				
n	4				
e	0				

APPENDIX

STANDARD SYMBOLS USED IN THE EXPERIMENTAL TANKS IN JAPAN

Technical Term	Symbol
Acceleration	α
Angle of heel	θ
Area,	
wetted skin —	S
Disc —	A_o
Developed —	A_d
Breadth on W.L.	B_{wL}
— extreme	B_{ex}
— moulded	B_m
Coefficient,	
Block —	C_b
— of friction	f
— of viscosity	μ
— of kinematic viscosity	ν
Midship —	$C\otimes$
Prismatic —	C_p
Water plane area —	C_w
Vertical prismatic —	C_v
Thrust deduction —	t
Density	ϱ
Depth	D
Diameter of propeller	D
Dimension ratio	λ
Displacement,	
volume —	$\triangle$
weight —	W
Draught,	
— moulded	d_m
— extreme	d_{ex}
Efficiency,	
Engine —	η_e
Hull —	η_h
Screw —	η_p
Propulsive —	η
Relative rotative —	η_r
Effective pitch ratio	p_e
Gravity,	
Acceleration due to —	g
Horse power,	
Brake —	B.H.P.
Effective —	E.H.P.
Indicated —	I.H.P.
Shaft —	S.H.P.
Thrust —	T.H.P.
Length,	
— between perpendiculars	L_{pp}
— over all	L_{oa}
— on waterline	L_{WL}
— on loadwaterline.	L_{LWl}
— on trial waterline	L_{TWL}
Mass	M
Moment of inertia	I
Number of blades	n
— — revolutions	N
Pitch,	
Effective —	P_e
Face —	P_f
Virtual or effective —	P_e
— ratio	p_f
Power,	
Useful or thrust —	P_u
Absorbed —	P_a

Technical Term	Symbol
Pressure	P
Resistance,	
Air —	R_a
Eddymaking —	R_e
Form —	R_F
Frictional —	R_f
Idle —	R_i
Residuary —	R_r
Total —	R
Wave making —	R_w
Slip,	
Apparent —	s_a
Effective or virtual —	s_e
Nominal or real —	s
Specific gravity	γ
Speed,	
— of ship	V
— of advance of screw	V_1
Thrust	T
Open —	T_o
Time	T
Torque	Q
Velocity	V
Angular —	ω
Virtual pitch ratio	p_e
Wake fraction	x
— percentage	w

Teishin-sho Experiment Tank *May* 1933.
Tokyo, Japan

To Dutch Experiment Tank.

a. At the self-propulsion tests the tension correction is fitted to a ship model, as in Continental tanks.

b. To prevent the laminar flow around a ship model, no special means are adopted, except the daily use of six meter models.

c. The tension correction is calculated by R. E. FROUDE's friction-formula, taking 15° C. as a standard temperature, but for small ships special considerations are paid.

d. The test results are published as follows:

I. Resistance tests. When the effects of weight and viscosity of water are taken into consideration, the following relation for the resistance of a moving body is obtained by the principle of dynamical similarity (the significance of the symbols used is detailed in *e*),

$$\frac{R}{\varrho L^2 V^2} = f\left(\frac{V^2}{Lg}, \frac{VL}{\nu}\right),$$

where f represents some function and L a certain linear dimension of a body under consideration. Since, however, it is practically impossible to make dynamically similar model experiments in accordance with such general condition, we use W. FROUDE's idea of separating the total resistance into the wave making and frictional resistances, transferring only the first one from model to actual ship, but calculating the other for both. In other words, what can be investigated by model experiments is the effect of weight, but not that of viscosity. Under these simplified considerations, the above relation can be transformed to

$$\frac{R}{\varrho L^2 V^2} = f\left(\frac{V^2}{Lg}\right),$$

which is the fundamental relation for the dimensionless presentation of test results.

The linear dimension L may be conveniently replaced by $\nabla^{1/3}$

i.e. the length of the side of the cube having the contents equal to the immersed volume of the ship or its model.

Hence, we have

$$\frac{R}{\varrho \nabla^{2/3} V^2} = f\left(\frac{V^2}{\nabla^{1/3} g}\right),$$

so that the most natural and reasonable dimensionless presentation of the test results is to represent the variation of $\frac{R}{\varrho \nabla^{2/3} V^2}$ on the base of $\frac{V^2}{\nabla^{1/3} g}$, because the object of model experiment is the determination of the function f.

$\sqrt{\frac{V^2}{\nabla^{1/3} g}}$ which corresponds to R. E. FROUDE's speed constant

$$(K) = \sqrt{\frac{V^2}{\nabla^{1/3} g} \cdot 4\pi}$$

is called the ,,relative speed'' and denoted by v, so that

$$v^2 = \frac{V^2}{\nabla^{1/3} g}$$

The ,,relative resistance'' $\frac{R}{\varrho \nabla^{2/3} V^2}$ corresponds to R. E. FROUDE's resistance constant

$$(C) = \frac{R}{4\pi \varrho \nabla^{2/3} V^2},$$

and is denoted by r, so that

$$r = \frac{R}{\varrho \nabla^{2/3} V^2} = \frac{R}{\triangle} \cdot \frac{1}{v^2}$$

In plotting the test results, we use the following procedures:

1. The relative frictional resistances of model r_{mf} are calculated and plotted downwards on the base of the square of the relative speed v. It must be remembered that the position of the r_{mf}–v^2 curve will be altered if either the length of model or the temperature of tank water is changed.

2. The total relative resistances of model r_m are calculated from test results and plotted against v^2 upwards from the r_{mf}–v^2 curve. We have, then, the curve representing the relation of relative wave making resistance r_w and the square of the relative speed v, for it is evident that

$$r_m = r_w + r_{mf}$$

The r_w–v^2 curve thus obtained is equally applicable to the model and actual ship and it represents the variation of residuary resistance following FROUDE's law of comparison as the speed changes.

The comparison of the frictional resistances of different forms may be made by means of the relative skin area

$$f = \frac{F}{\nabla^{2/3}}$$

without difficulty.

II. Propeller tests in open water. When the effects of absolute dimensions, advance speed, number of revolutions, immersion of a propeller and viscosity of water are taken into consideration, the following relation for the thrust of a working propeller is obtained by the principle of dynamical similarity.

$$\frac{T}{\varrho N^2 D^4} = f\left(\frac{V_1}{ND}, \frac{ND^2}{\nu}, \frac{H}{\varrho V_1^2}\right).$$

But it is practically impossible to make dynamically similar model experiments in accordance with such general condition.

Since experiments show that highly polished and deeply submerged propellers do not change their performance with the variation of REYNOLDS' number $\frac{ND^2}{\nu}$ larger than a certain limit, the terms $\frac{ND^2}{\nu}$ and $\frac{H}{\varrho V_1^2}$ in the above relation may be neglected. Then we have the relation

$$\frac{T}{\varrho N^2 D^4} = f_1\left(\frac{V_1}{ND}\right).$$

Similarly, the relation for the torque of a working propeller is

$$\frac{Q}{\varrho N^2 D^5} = f_2\left(\frac{V_1}{ND}\right)$$

These two relations are the fundamental expressions for the dimensionless presentation of open water tests, so the thrust constant $t = T/\varrho N^2 D^4$ and torque constant $q = Q/\varrho N^2 D^5$, calculated from test results, are plotted on the base of the advance constant $v_1 = V_1/ND$.

Moreover, the propeller efficiency

$$\eta_p = \frac{t\, v_1}{2\pi q}$$

and nominal slip ratio

$$s = 1 - \frac{V_1}{NP}$$

are represented in the same diagram against v_1.

Since some change may be expected in the propeller performance represented in the above diagram with the variation of ND^2/ν and $H/\varrho V_1^2$, it is necessary to write in this diagram REYNOLDS' number ND^2/ν and the propeller immersion at which tests were made.

III. Self-propulsion tests. For the sake of simplicity, the method of the publication of test results will be shown only for a single or twin screw ship.

From the results of self-propulsion tests,

$$\text{relative total thrust } t' = \frac{T'_m}{\varrho_m \nabla_m{}^{2/3} V_m{}^2} = \frac{T'_m}{\triangle_m v^2},$$

$$\text{relative total power } p' = \frac{2\pi N_m Q'_m}{\varrho_m \nabla_m{}^{2/3} V_m{}^3} \text{ and}$$

$$\text{relative revolutions } n = \nabla_m{}^{1/3} \frac{N_m}{V_m}$$

are calculated, and plotted on the base of v^2, from which three fair curves are drawn.

Propulsive efficiency, which is the ratio of E.H.P. to B.H.P., may be split up into the various factors given in the following expression:

$$\eta = \frac{\text{E.H.P.}}{\text{B.H.P.}} = \eta_t \frac{\text{E.H.P.}}{\text{S.H.P.}} = \eta_t \frac{RV}{2\pi NQ'}$$

$$= \eta_t \frac{R}{T'} \frac{V}{V_1} \frac{T'V_1}{2\pi NQ'} = \eta_t \frac{1-t}{1-w} \eta'_p = \eta_t \eta_h \eta_r \eta_p.$$

From the curves of t', p' and n, obtained above, it is easy to calculate these factors, together with apparent slip ratio, R.P.M. of actual propellers and admiralty constant. At first the relative total resistance of ship $r_s = \frac{R_s}{\varrho_s \nabla_s{}^{2/3} V_s{}^2} = \frac{R_s}{\triangle_s v^2}$, speed of ship in knots $V'_s = 1.944\, V_s = 1.944\, \nabla_s{}^{1/6} g^{1/2} v = 6.0857 \nabla_s{}^{1/6}\, v$ and speed of model in meters per second $V_m = (.10204/\nabla_m{}^{1/3})^{-1/2}\, v$ are represented in the above diagram. Then,

$$\text{thrust deduction coefficient } t = \frac{T'_s - R_s}{T'_s} = 1 - \frac{r_s}{t'},$$

R.P.M. of actual propeller $N'_s = \frac{60}{\alpha^{1/2}} N_m = \frac{60}{\alpha^{1/2} \nabla_m^{1/3}} n V_m$,

apparent slip ratio $s_a = 1 - \frac{V_m}{N_m P_m} = 1 - \frac{\nabla_m^{1/3}/P_m}{n}$,

S.H.P. $= \frac{1.025\,\alpha^{3.5} \times 2\pi N_m Q'_m}{75} = \frac{1.025\,\varrho_m\,\alpha^{3.5} \nabla_m^{2/3}}{75} p' V_m^3$,

B.H.P. $= \frac{\text{S.H.P.}}{\eta_t}$ (transmission efficiency η_t assumed).

admiralty constant $C = \frac{\eta_t \Delta_s^{2/3} V_s^{3}}{\text{S.H.P.}} = \frac{5.362}{p'} \eta_t$ and

propulsive efficiency $\eta = \frac{\text{E.H.P.}}{\text{B.H.P.}}$

are calculated.

Next, comparing the measured thrusts at self-propulsion tests with those at open water tests, the advance constant v_1, nominal slip ratio s and propeller efficiency (open) η_p are obtained, and from these

wake fraction $\quad w = 1 - \frac{V_1}{V_m} = 1 - \frac{D_m}{\nabla_m^{1/3}} n v_1$,

hull efficiency $\quad \eta_h = \frac{1-t}{1-w}$,

propeller efficiency (behind) $\eta'_p = \frac{T'_m V_1}{2\pi N_m Q'_m} = \frac{t}{p'}(1-w)$.

and relative rotative efficiency $\quad \eta_r = \frac{\eta'_p}{\eta_p}$

are calculated.

Among these calculated values, the thrust deduction coefficient t, apparent slip ratio s_a, propulsive efficiency η, nominal slip ratio s, propeller efficiency (open) η_p, wake fraction w, hull efficiency η_h, propeller efficiency (behind) η'_p and relative rotative efficiency η_r are represented in the above diagram, so that thirteen dimensionless and two dimensional curves are shown in this diagram.

For practical use E.H.P., B.H.P., revolutions per minute, propulsive efficiency and admiralty constant are represented on the base of speed of ship in knots.

e. Symbols used are as follows:

symbols	dimensions kg	m	sec	remarks
ϱ	1	−4	2	density i.e. mass of unit volume of water.
γ	1	−3	0	weight of unit volume of water.
σ	0	0	0	specific gravity of water (for sea water 1.025).
θ				temperature of water in °C.
L	0	1	0	length of ship.
F	0	2	0	area of wetted surface.
V	0	1	−1	speed of ship.
V′				speed of ship in knots.
R	1	0	0	resistance of ship.
R_f	1	0	0	frictional resistance of ship.
R_w	1	0	0	wave making resistance of ship.
$\bigtriangledown$	0	3	0	immersed volume.
$\bigtriangleup$	1	0	0	displacement.
ν	0	2	−1	kinematic coefficient of viscosity of water.
g	0	1	−2	gravitational acceleration.
λ				R. E. FROUDE's coefficient of friction.
v	0	0	0	relative speed.
r	0	0	0	relative resistance.
r_f	0	0	0	relative frictional resistance.
r_w	0	0	0	relative wave making resistance.
f	0	0	0	relative wetted surface.
D	0	1	0	diameter of propeller.
P	0	1	0	pitch of propeller.
N	0	0	−1	revolutions of propeller.
N′				R.P.M. of propeller.
V_1	0	1	−1	advance speed of propeller.
Vw	0	1	−1	speed of wake.
H	1	−2	0	statical pressure on unit area.
T	1	0	0	thrust of propeller.
Q	1	1	0	torque of propeller.
v_1	0	0	0	advance constant.
t	0	0	0	thrust constant.
q	0	0	0	torque constant.
η_p	0	0	0	propeller efficiency (open).
s	0	0	0	nominal slip ratio.

symbols	dimensions kg	m	sec	remarks
α	0	0	0	linear ratio.
T'	1	0	0	total thrust of propeller.
Q'	1	1	0	total torque of propeller.
n	0	0	0	relative revolutions.
t'	0	0	0	relative total thrust.
p'	0	0	0	relative total power.
t	0	0	0	thrust deduction coefficient.
w	0	0	0	wake fraction.
s_a	0	0	0	apparent slip ratio.
E.H.P.				effective horse power.
$E.H.P._f$				frictional effective horse power.
$E.H.P._w$				wave making effective horse power.
S.H.P.				shaft horse power.
B.H.P.				brake horse power.
z				number of blades of propellers.
C				admiralty constant referred to S.H.P.
η	0	0	0	propulsive efficiency.
η_t	0	0	0	transmission efficiency.
η_h	0	0	0	hull efficiency.
η'_p	0	0	0	propeller efficiency (behind).
η_r	0	0	0	relative rotative efficiency.

Note. The suffixes m and s are added to those for model and ship, respectively, if necessary.

NOTES FOR CONFERENCE

METHODS AND CONSTANTS USED AT DUMBARTON

The waterline length is used as standard.

The surface friction for model is based on W. Froude's results for paraffin applied to the calculated wetted surface of the model. These results are assumed to be for 51° F. and the temperature correction, as a result of recent experiments has been fixed at 2,7% inversely per 10° F. departure from standard temperature, for the range of model speed and length in use at Dumbarton.

The ship surface friction is based on W. Froude's results for varnish extrapolated to the length required.

No correction is made on surface friction for curvature, butts, landings or rivet heads etc.

In our opinion the ship E.H.P. as calculated above should not be corrected for sea temperature. The practice or opinion of other superintendents in this connection would be of interest.

No special means are in use at Dumbarton to induce turbulent flow but the model is always chosen of such a length in conjunction with the running speed to make the Reynolds' Number of the model considerably in excess of 3×10^6. Experiments with trip wires and roughening devices at the bow, in our opinion, introduce secondary errors of more importance than the laminar flow error.

Special care is taken to neutralise currents in the tank so as to ensure that speed is as indicated.

The standard method in use at Dumbarton for recording results of model towing experiments is rate at 51° F. for a 100′ ship, i.e. $\frac{200 \times (\text{E.H.P. for } 100' \text{ ship})}{\text{Surface in ft.}^2}$. In conjunction with a group of S.F. Correction curves an estimate is readily prepared for any length of ship.

Self-propulsion experiments are not normally made at Dumbarton.

When screw experiments are made behind a model the Froude

S.F.C. is deducted from the Augmented resistance to obtain the point corresponding to ship self-propulsion.

In deducing the Shaft Horse Power for ship from this point, proper additions are made for bilge keels or other excrescences not fitted to model, super-structure air resistance and tail shaft losses.

Model propeller dimensions are chosen so that the value of Reynolds' Number (developed blade width $\times$ speed of section through water $\div \nu$) at .25 radius is $.6 \times 10^5$ or more, but a propeller of less than $4\frac{1}{2}''$ in diameter is avoided. In our opinion, propeller „scale effect" extends above this point but the foregoing is considered a reasonable working limit for reliable results.

If the propeller model satisfies the above criterion it is generally found that the corresponding ship model is well out of the laminar region.

The question of probable error in the various measurements involved in tank work is one which might well receive attention as it must always be closely associated with results if a true perspective is to be preserved. Each tank could submit a statement to the conference, giving the values of probable error in various routine measurements, particularly resistance, thrust and torque, but also wake, thrust deduction etc. and the comparison of these statements would enable a value to be found for the average percentage error to be associated with any measurement.

Imp. Japanese Navy

PROPOSED SUBJECT:

Among the technical terms used in the analysis of the results of tank experiment, some are given different definitions by different authors, therefore, we propose to standardize the same.

Note: For these ambiguous technical terms ,,The Experimental Tank Committee In Japan" has given the explanation shown as follows:

Symbols follow then as mentioned under Subject e and
E.H.P. = effective horse power for naked hull
E.H.P.a= ,, ,, ,, ,, ship with appendages and so on.

Propulsive efficiency $\eta_1 = \frac{\text{E.H.P.}}{\text{I.H.P.}}$ or $\frac{\text{E.H.P.a}}{\text{I.H.P.}}$

,, ,, $\eta = \frac{\text{E.H.P.}}{\text{S.H.P.}}$ or $\frac{\text{E.H.P.a}}{\text{S.H.P.}}$

Hull efficiency $\eta_h = \frac{\text{E.H.P.}}{\text{T.H.P.}}$ or $\frac{\text{E.H.P.a}}{\text{T.H.P.a}}$

Wake fraction $w = \frac{V-V_1}{V}$

Thrust deduction coefficient $t = \frac{T-R}{T}$

Engine efficiency $\eta_e = \frac{\text{S.H.P.}}{\text{I.H.P.}}$

Transmission efficiency $\eta_t = \frac{\text{D.H.P.}}{\text{S.H.P.}}$

Relative rotative efficiency $\eta_r = \frac{\text{P.H.P.}}{\text{D.H.P.}}$

Propeller efficiency open $\eta_p = \frac{\text{T.H.P.}}{\text{P.H.P.}}$

Propeller efficiency behind $\eta'_p = \dfrac{\text{T.H.P.}}{\text{D.H.P.}}$

Propulsive efficiency $\eta = \dfrac{RV}{2\pi N Q_s} = \left(\dfrac{R}{T}\right)\left(\dfrac{V}{V_1}\right)\left(\dfrac{Q}{Q_s}\right)\left(\dfrac{TV_1}{2\pi N Q}\right)$

$$= \frac{1-t}{1-w} \cdot \eta_r \cdot \eta_p = \eta_h \cdot \eta_r \cdot \eta_p$$

Propulsive efficiency $\eta_1 = \dfrac{\text{E.H.P.}}{\text{I.H.P.}}$

$$= \left(\frac{\text{S.H.P.}}{\text{I.H.P.}}\right)\left(\frac{\text{D.H.P.}}{\text{S.H.P.}}\right)\left(\frac{\text{P.H.P.}}{\text{D.H.P.}}\right)\left(\frac{\text{T.H.P.}}{\text{P.H.P.}}\right)\left(\frac{\text{E.H.P.}}{\text{T.H.P.}}\right)$$

$$= \underbrace{\eta_e \cdot \eta_t \cdot \eta_r}_{\substack{\text{Friction remainder}\\ \text{or mechanical}\\ \text{efficiency.}}} \cdot \eta_p \cdot \eta_h$$

INTERNATIONAL CO-ORDINATION REGARDING TANKWORK

BY PROF. H. R. MØRCH
(TRONDHJEM)

Practical shipdesigners who during many years have made use of the conclusions which systematic tankwork etc. steadily renders, will easily agree on the large and valuable work which has been produced since the days of WILLIAM FROUDE. Designers will however necessarily have felt the hamper, involved by the use of the various methods for putting down the experimental results, the different bases for execution of the experiments and methods for analysis of same. It is therefore very satisfying to learn that attempts now are being made to bring about an international interwork in this respect. Mr. JOHN DE MEO has already forwarded to interested parties a memorandum regarding possible points for discussion on the first international meeting to be held at Wageningen this summer.

Of the experiments which are carried out in the ship experimental tanks none are of greater value to the ship designers as a whole than the systematic series, by reason of their great educational value to the students of Naval Architecture at the universities and practical value to the designing offices. These systematic series embody the hull, propeller and their interwork as other problems regarding ship designing. The detached experiments executed for shipyards, shipowners etc. are of much less importance in this connection. It is the law of resistance against propulsion of hulls, performance of propellers etc. connected with certain definite variations in form, size etc. which are of lasting value to ship designers. In order to deduce such laws, the hulls as well as the propellers have to be treated separately, finally the reactions between hulls and propellers inclusive wake questions. As the number within each group is large, both subjected to developments in form and size with time, the experimental work is overwhelming and can only be solved in its broad features, if co-operation between all good powers could be arranged.

According to invitation by Mr. TROOST, director of the new dutch experimental tank, I hereby put forward some questions regarding the resistance of ships hulls, which according to my opinion should be discussed at the first meeting at Wageningen. These are:

1. Systematic experiments ré towing resistance of ships hulls.
2. The various characteristics of the resistance results.

Ré 1. Systematic work of this kind must be based on a practical and definite definition of the size and form of the submerged and emerged portions of the ships hulls, including the important variations in form etc.

The german Dr. WEINBLUM has shown in his papers, that a mathematical treatment of the hull is possible by the use of rational functions. The arithmetical work will no doubt be reduced when the work is systematiced. Variations in form etc. will by the use of such a system obtain mathematical precision.

A nearly mathematical treatment of this sort has been used at the Washington basin since its start according to D. W. TAYLOR in the Transactions of the International Engineering Congress, San Fransisco, 1915. The sectional area curve as well as the C.W.L. curve in entrance and run are treated mathematically by the use of parabolas of 5th degree. Parameters are the area coefficient, inclination of the curves at stem and stern and the radius of curvature of the curves at the ⊗ section or parallel body. For the transverse sections parabolas of the 4th degree are used where the area coefficients are less than ca. 0,72, ordinary hyperbolas elsewhere. For the transverse sections of parabola form the area coefficient, tumble-home at C.W.L. and dead-rise at keel are the parameters, for the hyperbola sections the area coefficient and the tumble-home at C.W.L. Excepting the choice of the inclination of the transverse sections at keel and at the C.W.L., which values are not faired by the use of mathematical functions, the method of TAYLOR is strictly mathematical.

Excepting the above mentioned 2 mathematical methods of WEINBLUM and TAYLOR the „common trial and error method" is, as far as I know, generally used both in tank- and design work.

If an international agreement is possible regarding systematic towing work of ships hulls, this should also embody a mathematical method of defining the shipform. If so we should in time be able to obtain resistance results corresponding to specified mathematical variations in the form of hulls and thereby increasing our knowledge regarding a possible basis for a broader and more exact arithmetical calculation of the resistance of ships.

The tables 1 and 2 appended contain the most important systematic series regarding the towing of ships hulls, experimented with in Great Britain, U.S.A., Germany and Italy. L is the length of ship, S the area of ⊗ section, V the total volume of displacement, % the ratio between the parallel length of the sectional area curve and the length of ship, F the form of sectional area curves at entrance and run in a 10th divisional diagram, B maximum breadth of ship, d the mean draught and F in table 2 the form of ⊗ section in a 10th divisional diagram.

An international agreement concerning systematic work of this nature will no doubt be difficult to obtain. Nevertheless it ought to be tried and should according to my opinion only consist of a general plan where the main features only are outlined. It must be left to each country or experimental tank to choose those systematic series, types and sizes of ships etc., which are found to be most appropriate to them. Communications ought to be given the public in each case, when such systematic experiments are started and in time the results hereof.

Ré 2. The results of towing hulls for shipyards or shipowners are generally given in the form of a curve, representing the towing horse power of the ship, calculated from the performance of the model. If the ship model has been tested in connection with modelpropellers, the curve of shaft horsepowers is given also.

The publication of towing results ré systematic shipmodel series is based on:

a. William Froude's „method of division" by the so-called „residuary resistance pr. ton displacement" as a function of Froude's speed number $F \backsim \frac{V}{\sqrt{L}}$ or

b. R. E. Froude's „method of correction" of 1888 as „total resistance pr. ton displacement" as a function of a speed number, as inst. $(K) \backsim \frac{V}{\Delta^{1/6}}$. The published values of the total resistance per ton displacement are generally referred to a ship of a certain length, as inst. 400-feet, after certain corrections are done in $(C)_{TM} \backsim \frac{ehp_T}{\delta^{2/3} v^3}$ for model.

Of these two methods *b* has the advantage over *a* in using the real towing force as measured and not a calculated residuary resistance per ton displacement, but only under that special condition, that the published results refer to the model itself.

When measuring the towing force of the model we must be sure of obtaining turbulent flow and that the transverse dimensions of the model tank are such relatively to those of the model, that we get resistance results not influenced by the bottom and walls. According to my opinion these 2 points are of primary importance and should in the near future be further investigated.

My suggestion is therefore that the total resistance per ton displacement of the model should be the basis for the published results of the systematic series in the form of C values for model $\backsim \frac{ehp_{TM}}{\delta^{2/3} v^3}$ as a function of $F \backsim \frac{v}{\sqrt{gl}}$. In connection with each model are published the main dimensions of same, its formcharacteristics, displacement, wetted surface etc. As commendatory should be published in curves- or table form a system according to which the published C_{TM} values may be corrected to any size according to R. E. FROUDE's equation $C_{TS} = C_{TM} - (C_{FM} - C_{FS})$

Index TS equal total resistance of ship.
Index TM equal total resistance of model.
Index FM equal frictional resistance of model.
Index FS equal frictional resistance of ship.

The expressions for speed ought to be given in a form, independent of the chosen system of measurement as $F = \frac{V}{\sqrt{gL}}$ or $K = \frac{V}{\Delta^{1/6}} \frac{\gamma^{1/6}}{g^{1/2}}$. In the same way the C values must be given in the form $\frac{g}{\gamma^{1/3}} \frac{EHP}{\Delta^{2/3} V^3}$. For R. E. FROUDE's surface coefficient S the expression must likewise be $\frac{\gamma^{2/3} F}{\Delta^{2/3}}$.

For the metric system we thus have $F = 0{,}3193 \frac{V}{\sqrt{L}}$ where V in m/sek, L in m.

$K = 1.0096 \frac{V}{\Delta^{1/6}}$ ...., $\gamma = 1{,}00$ and $K = 1.0138 \frac{V}{\Delta^{1/6}}$ ...., $\gamma = 1{,}025$, Δ in kg.

$$\textcircled{C} = 10{,}000 \frac{g}{\gamma^{1/3}} \frac{EHP}{\Delta^{2/3}V^3} = 9810 \cdot \frac{EHP}{\Delta^{2/3}V^3}, \; \ldots . \; \gamma = 1{,}00$$

$$= 9730 \; . \;\; ,, \;\; , \ldots . \; \gamma = 1{,}025$$

$$\textcircled{S} = 100 \frac{F}{\Delta^{2/3}} \; \ldots ., \; \gamma = 1{,}00$$

$$= 101{,}66 \frac{F}{\Delta^{2/3}} \; \ldots ., \; \gamma = 1.025$$

If we choose Schlichting's expression of 1931 $C_F = 0{,}455 \, (\log R)^{-2{,}58}$ to give the change in frictional resistance with size and speed, and using $(\alpha + \beta)$ as expressions for the form- and roughness influence we get the following expressions corresponding to R. E. FROUDE:

$$(\textcircled{C}_{FM} - \textcircled{C}_{FS}) = 30{,}35 \, \textcircled{S} . \left[\frac{(\alpha+\beta)_m}{(\log R_m)^{2.58}} - \frac{(\alpha+\beta)_s}{(\log R_s)^{2.58}} \right] \text{ in}$$

metric measure.

Example. Small combined passenger and cargo boat of main dimensions 44 m $\times$ 7,925 $\times$ 3,20 m, Δ = 600-metric tons, F = = 432,71 m^2. Speed 11 knots. Seams and butts overlapped. $C_{\otimes} = 0{,}86$ and $C_p = 0{,}612$. Model of parafin, scale 1/10 tried in tank. Temperature of tankwater 9,5° C., that of sea water 15° C.

Towing force of model at a speed corresponding to 11 knots = = 3,79 kg. Corresponding $\textcircled{C}_{TM} = 2{,}2127$.

$(\alpha + \beta)_m = \alpha_m$ as β taken = 0; $\alpha_m = 1{,}0948$ according to a chosen formula for form effect — $\alpha = 0{,}85 + 0{,}4 \, C_p$. For ship $(\alpha + \beta)_s$ taken = 1,30 in all.

$R_m = 6{,}7796$ and $R_s = 8.30963$.

$(\textcircled{C}_{FM} - \textcircled{C}_{FS}) = 0{,}4445$.

$\textcircled{C}_{FS} = 2{,}2127 - 0{,}4445 = \mathit{1{,}7682}$. This works out to EHP_T for ship = *234,42*.

According to Hamburgische Schiffbau-Versuchsanstalt's known formula with frictional coefficient for ship of 0,14504 and for model corrected for temperature of tankwater to 15° C: 0,17913, the EHP_{TS} is calculated = 234,5.

Our knowledge regarding frictional resistance of ship forms will no doubt increase steadily with time; the tables or curves of recommendation will therefore have to be corrected from time to time according to knowledge gained.

Regarding the question of propulsion of ships I find the solution of wake velocities of primary importance for design work. Dr. KEMPF's modelwheels by which he finds the mean speed of the water in the various circles about the centre of the shaft, are of special interest and should be used in connection with all systematic models tried. Would not the use of large wheels of similar construction be usefull on some large ships, towed without propellers on a mile, in order to bring about some comparisons between wakespeeds at large ships and their models?

BRIEF NOTES ON THE PRESENTATION OF TANK DATA

By Professor T. B. Abell
University of Liverpool

GENERAL CONSIDERATIONS

In approaching the problem of deciding upon the form in which results of tank data should be presented in order that the best use shall be made of them there seem to me to be three main considerations to be kept in view. It would probably be accepted that the first claim is that of the advancement of the science itself, and that to realise to the full this claim the original experiment measurements should be furnished to individual investigators who are increasing in number as the application of the science increases. The investigator has made important contributions to the science and I think all are likely to benefit by the general publication of results. Specially do I believe that the experiment establishments themselves will profit by such publication.

I would place second the claim of the shipbuilder and shipowner who desires experimental investigation to solve his own particular and specific problem, and third that of the general designer who desires information which will enable him to put his design into a form which may or may not require actual experimental investigation before it can be definitely settled, depending upon the difficulties the actual vessel has to overcome.

Publication of actual measurements of speed, resistance, thrust, thrust torque and revolutions etc.: would I think meet all these claims in a very satisfactory manner (though of course curves or particulars of horsepower units would also be necessary to satisfy a shipbuilder's enquiry). Publication in a uniform manner is of greater importance than uniformity of symbols or of units. For these can be readily expressed in terms of each other and are then available for all time. We cannot hope to obtain a universal language and indeed it might not be beneficial if we could, but much will be achieved if we can obtain a uniform method of presentation.

The subject naturally divides itself into two distinct elements, 1. the towed model and 2. the self-propelled model.

THE TOWED MODEL. CONDITIONS OF EXPERIMENT

If the data published from the different experiment stations is to be immediately useful it is important that the conditions of experiment should be as far as possible uniform. The principal factors governing uniformity of condition are size of model, condition of surface, state of the fluid motion.

Size of model cannot be uniform in all stations or tanks, it thus becomes desirable to supplement the publication of resistance experiments by the addition of the calculated skin friction resistance for the particular model, the calculated skin friction being obtained from data fixed for general international use. This would require an additional line to the table given on Plate III of the Wageningen proposals.

Condition of the surface can in the majority of stations in temperate countries be uniformly paraffin. Provided reasonable care is exercised in preparation of the final surface and also provided that the surface is cleaned after immersion in the water and immediately before test, uniform conditions should be secured.

The state of the fluid motion, if requiring regulation, can so it seems to me, be more satisfactorily regulated by a standard wire of cylindrical or triangular cross section than by introducing an artificial condition to the surface of the model, since uniformity can only with great difficulty be obtained.

PARTICULARS OF MODELS

Whilst the particulars proposed to define the model are ample, I would like to see the water plane at half draught added to the load water plane and curve of sectional areas.

As an alternative to a complete body plan I think sections 1, 2, 5, 10, 15, 18, 19 with tables of percentage ordinates augmented to include water lines 7 and 8 and a table of ordinates of water lines at the ends of the model bow and stern to supplement the contours of bow and stern are sufficient to define the form. A curve of areas of water planes would be useful.

CALCULATION OF WETTED SURFACE

As the complete calculation of wetted surface does not require very much additional work beyond that required to obtain the mean girth it would seem worth while to secure the advantage of determining the actual surface. Where this has not been done the value of S used in the calculation might be stated and equal to the length multiplied by the mean girth.

PLOTTING OF RESISTANCE TESTS

The resistances and speeds are probably measured to the second place of decimals and to the fourth significant figure. It is suggested therefore that no coefficient need be used in which five figures appear. In the calculation of displacement accuracy of 1 in 1000 is rarely obtained though in weighing of course this error is large. Unless then g varies at the different stations to this extent it is justifiable to omit g from the speed length and speed volume ratios. I would prefer plottings in the general form of $\frac{R\ (\text{total})}{\varrho\, S\, V^2}$ to a base of $V/\sqrt{L}$ using model particulars for R. S. and V.

PROPELLERS. OPEN WATER EXPERIMENTS

I do not know whether it is the modern practice to obtain curves of thrust for a complete range of revolutions including that corresponding to $T = o$. In many of R. E. FROUDE's experiments observations were made at revolutions for which T was negative so that n for $T = o$ could be determined. This is an important characteristic of any propeller and if practicable I think an attempt should be made to obtain it for all propellers with which tank experiments are made. There is no fundamental reason why the change of thrust in this region should be irregular and the value of n for $T = o$ should be capable of accurate determination.

The thrust and torque coefficients $\frac{T}{n^2D^2P^2}$ and $\frac{Q}{n^2D^2P^3}$ would then have a more precise significance. Failing this there is probably a greater justification for using $\frac{T}{D^2V^2}$ and $\frac{Q}{D^2V^3}$ with $\frac{V}{nD}$ rather than $\frac{T}{n^2D^2P^2}$ and $\frac{Q}{n^2D^2P^3}$

I would like to see values of $\frac{T}{P\,D^2V^2}$ and $\frac{Q}{P^2D^2V^3}$ and $\frac{V}{nD}$ tabulated and would be prepared to dispense with the curves shown on Plate V.

SKIN FRICTION CALCULATION

Whilst my experience is almost entirely confined to the use of FROUDE's data and that of using GEBERS' data is very limited I can only express an opinion upon it in a very guarded manner. But there is one feature of GEBERS' data which I suggest is an important one. It does not seem as a general rule, so far as my observation goes, to accord in general character with the total resistance curves of models at the low speeds when practically all the resistance is that due to skin friction and eddy resistance due to form, as well as FROUDE's data do.

THE SELF PROPELLED MODEL

I have no comments to make upon this mode of experiment as I have had no actual experience of using it. I should however like to see measurements of the average wake velocity through the propeller disc made for alle models by means of a light propeller of known pitch for the sake of the information so obtained and for comparison with the wake values obtained by a driven propeller yielding a thrust sufficient to drive the model.

REPORT

ON TESTS OF INTERNALLY PROPELLED MODEL 1119 WITH SCREWS B1 AND B8, MADE IN THE THREE TANKS

ALFRED YARROW	WAGENINGEN	HAMBURG
ENGLAND	HOLLAND	GERMANY

§ 1. The purpose of the tests was to compare the results obtained in the three tanks with the same screw propelling the same hull, the model hull and screws used being the actual same in all three tanks. The tests were made possible by the kindness of Messrs. N.V. Stoomvaart Maatschappij „Nederland" who were good enough to transport the large wooden model hull from England to Amsterdam and Hamburg and back again.

§ 2. The model hull (Tank No. 1119) used represented a fairly good form of cargo boat, intended to run at about 12 knots on 400 ft. length, when loaded. It was propelled by a single screw, and had no appendages. The ship dimensions and particulars for a 400 ft ship are given in Table 1. The actual model was 24 feet in length, made in wood, with an enamel surface.

§ 3. Two screw propellers were tried, both of the same diameter, one foot for model, corresponding to ship dimensions given in Table 1. The screws were in type metal and had a reasonably polished surface. Screw B8 had blade sections with circular backs, screw B1 had blade sections with the same face as B8, but $A^3/_8$ aerofoil backs.

§ 4. The model hull was fitted with a stern tube carrying the propeller shaft, and this fitting remained in the hull and was used in all three tanks. The tests were made with the model at one displacement as given in Table 1. Each tank followed its normal routine in carrying out the tests.

§ 5. In the Teddington tank the model was towed by itself at a series of speeds from which the effective horse powers were obtained. The screw experiments were made at speeds corresponding to 12 and 14 knots. The revolutions being varied over a fair range at each of these fixed speeds. In the Teddington tank the propulsive coefficient and Delivered Horse Power have been obtained at the revolutions which would give a power equal to the Effective Horse Power + 10 per cent, but results corresponding to naked Effective Horse Power are given in Table 2 for comparison with other tanks.

The results are presented in two forms, first in detail and then the final figures for ship. Details obtained are given on Sheets 1 and 2 in the form of constants. In Sheet 1 are given all the results with the same screw. In Sheet 2 the comparison of results with the two screws in the different tanks. Owing to the different procedure in the tanks, the comparison has to be made under two distinct conditions:

1. when the screw is just propelling the model;
2. when the screw thrust corresponds to the model resistance with a deduction representing the skin friction correction passing from model to ship.

The constants used are defined at the end of the report. The general comparison of any particular constant — thrust, torque or revolutions — is fairly good between the three tanks, the maximum difference being 5 per cent, but these differences are in opposite directions on thrust and torque, and the propulsive coefficients in the different tanks differ by nearly 10 per cent maximum (Dutch and English). In this particular case the difference is mainly due to torque variation.

§ 6. The comparison of the two screws in the different tanks, is very good as regards individual constants but the Dutch tank shows a 3 per cent improvement of efficiency of B8 over B1 against only 1 per cent in the other two tanks.

The largest discrepancy in this detail comparison obviously lies in the torque measurement, and further tests in this direction seem called for.

§ 7. *Final figures for ship.*

Sheet 3 gives individual estimates of Shaft Horse Power and revolutions for 400 ft. ship at 12 and 14 knots, as obtained at the three tanks, using appropriate methods.

Table 2 gives the comparison, in ideal weather, of Shaft Horse Power based on model results, used in the same manner.

G. S. BAKER,
Superintendent.

The William Froude Laboratory.
26th June, 1933.

DATA

Dimension	Ship (Salt water)	Model (Fresh water)
Speed	V (Knots)	v_s (feet per second)
Displacement	$\triangle$ (Tons)	δ (Lbs)
Resistance	R (Tons)	r (Lbs)
Screw Diameter	D (Feet)	D_m (Feet)
Revolutions	N (Per minute)	n (Per second)
Thrust	T (Tons)	f (Lbs)
Torque	Q (Lbs Ft)	q (Lbs. Ft.)

CONSTANTS

Quantity	Sym-bol.	Ship (Salt water)	Model (Fresh water)
Speed-diameter constant	k_1	$=\frac{V}{DN}\times 101.3$	$=\frac{v_s}{D_m.n}$
Thrust-constant	y	$=\frac{T}{D^2V^2}\times 765.7$	$=\frac{f}{D_m{}^2.v_s{}^2}$
Torque-constant	z	$=\frac{Q}{D^3V^2}\times .3418$	$=\frac{q}{D_m{}^3.v_s{}^2}$
Resistance-constant	e	—	$=\frac{r}{D_m{}^2.v_s{}^2}$
Quasi-propulsive coefficient	ψ	—	$=\frac{ek_1}{2\pi z}$
Thrust deduction	t	—	$=1-\frac{e}{y}$

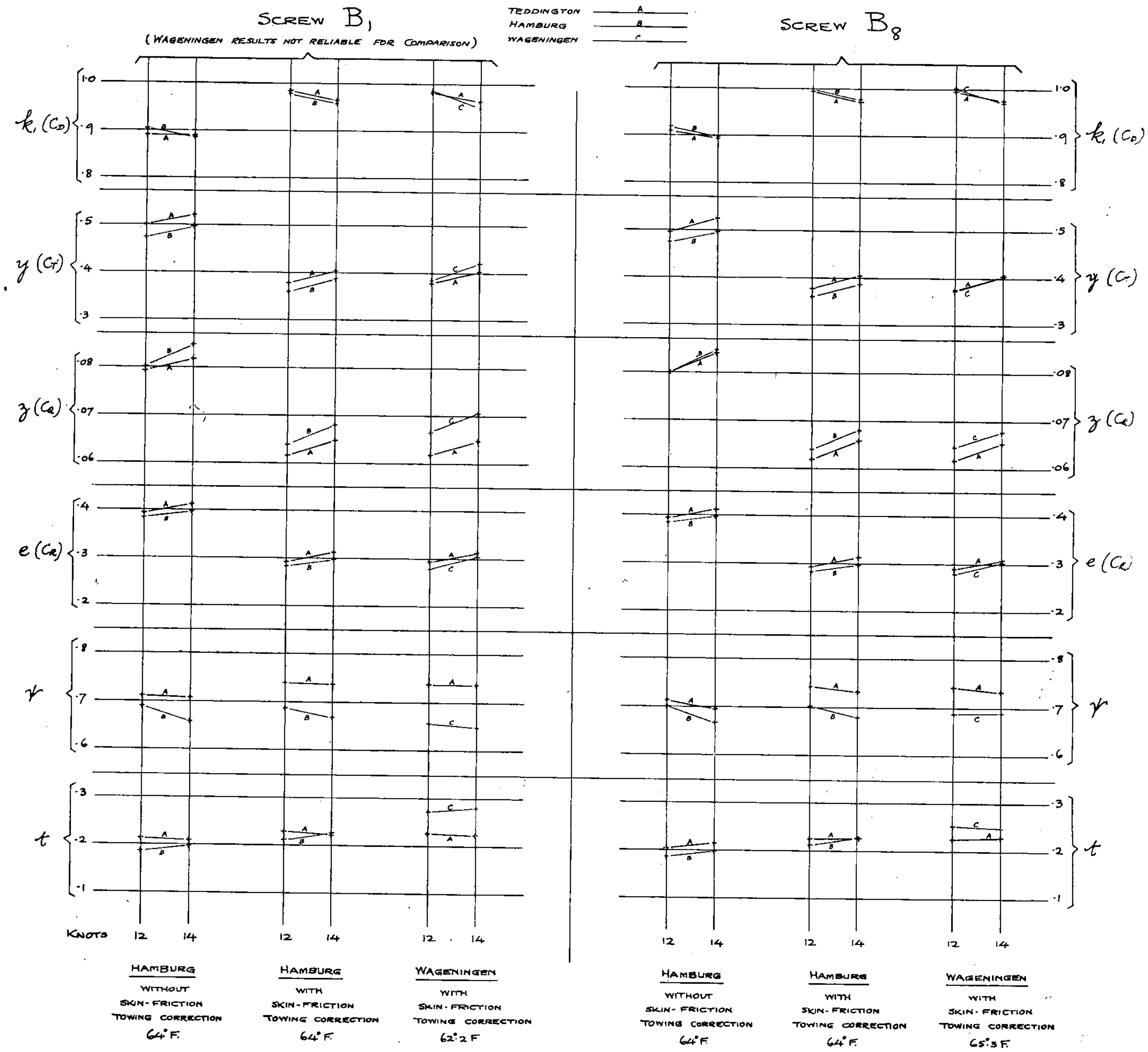
$$\text{Effective Horse Power (E.H.P.)} = \frac{C\,V^3\,\triangle^{2/3}}{427.1}$$

$$\text{Delivered Horse Power (D.H.P.)} = \frac{\text{E.H.P.}}{\psi}$$

The William Froude Laboratory, Teddington, England.

MODEL 1119 AT 4600 LBS: DISPLACEMENT. LEVEL TRIM. PROPELLED BY SCREWS B_1 AND B_8
COMPARISON OF RESULTS WITH HAMBURG AND WAGENINGEN

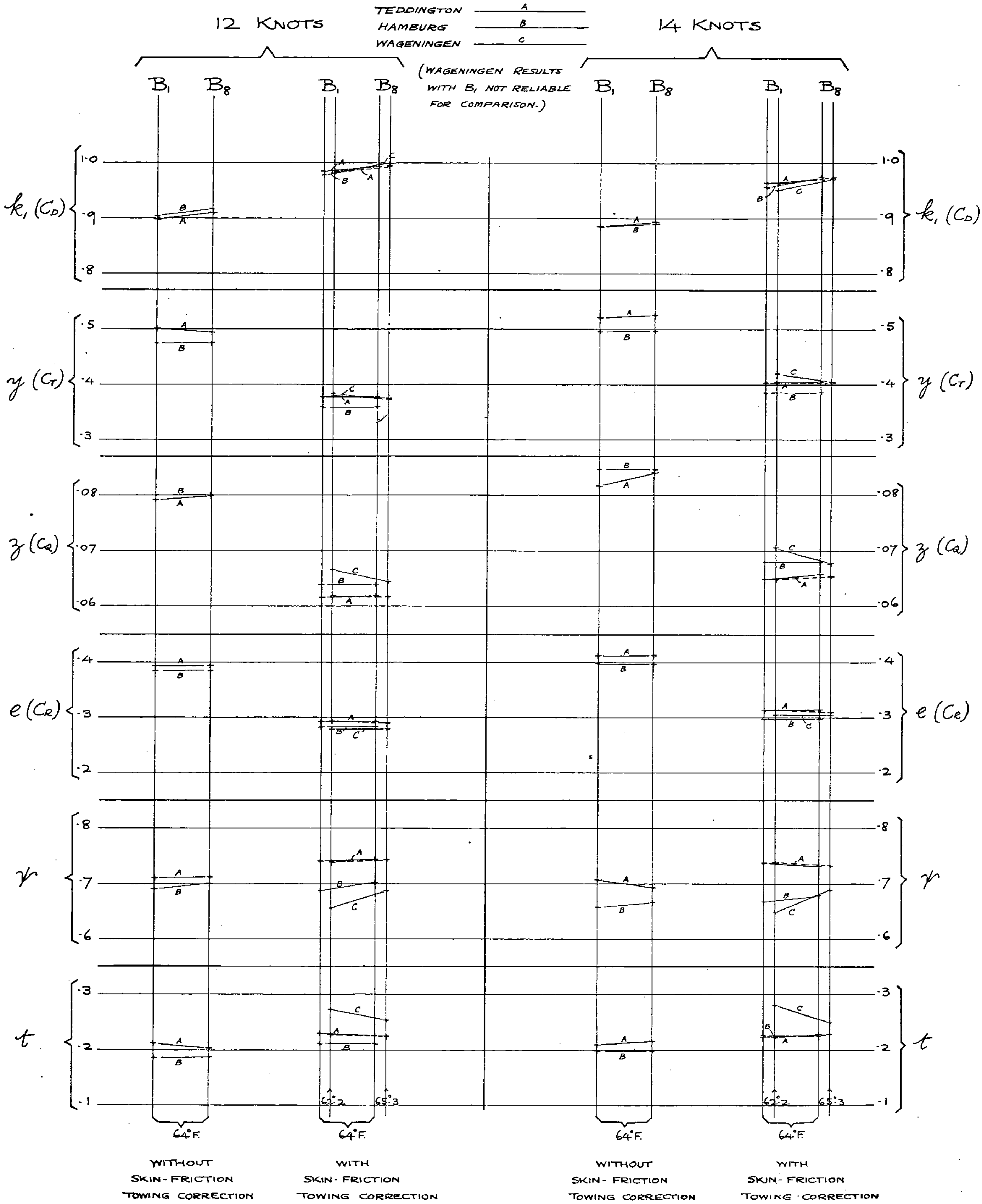
Sheet 1.– Comparison of Various Tank results for same screws



The William Froude Laboratory, Teddington, England.

MODEL 1119 AT 4600 LBS: DISPLACEMENT. LEVEL TRIM. PROPELLED BY SCREWS B_1 AND B_8. COMPARISON OF RESULTS WITH HAMBURG AND WAGENINGEN

Sheet 2.– Comparison of Constants for Model Screws at the same Tank



Screw B1 · Screw B8

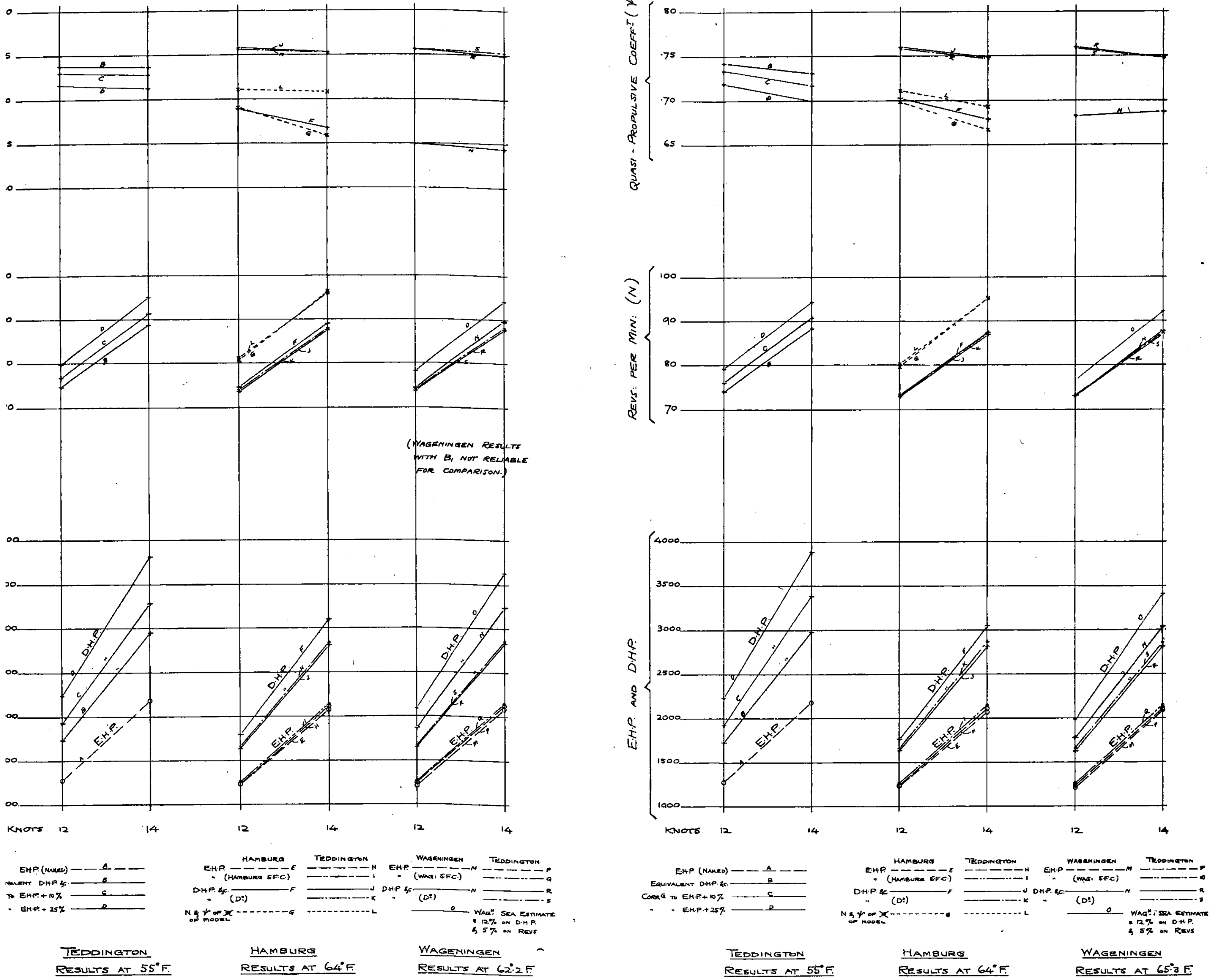


TABLE 1.

DIMENSIONS OF MODEL AND SCREWS

Model 1119: Screws B1 and B8

		Ship	Model
Length B.P.		400.0 feet	24.0 feet
Beam		53 ft. 4 ins.	38.4 inches
Draught		22.0 feet	15.84 inches
Displacement		9750 Tons	4600 Lbs
Trim		Level	Level
Model scale		–	$\frac{1}{16.67}$
Screws B1 and B8	Diameter	16.67 feet	1.0 foot
	Pitch	16.67 feet	1.0 foot
	Developed area (outside boss)	88.6 sq. ft.	45.9 sq. ins.
	Number of blades	4	
	Rotation	Right hand	
	Rake	15°	
Shape of sections	B1	$A^3/_8$ aerofoil	
	B8	Circular back	

TABLE 2.

SHAFT HORSE POWERS FOR 400 FT. SHIP BASED ON MODEL-RESULTS

(With correction for skin-friction)

Basis	Speed in knots	Screw B1			Screw B8		
		Teddington	Hamburg	Wageningen	Teddington	Hamburg	Wageningen
Actual Tank Results:	12	1734	1796	1864	1725	1761	1775
	14	2954	3100	3244	2982	3048	3041
With 10% standard-allowance for wind and appendages:	12	1907	1976	2050	1898	1937	1953
	14	3249	3410	3569	3280	3353	3345
Temperatures of Results:		55° F.	64° F.	62.2° F.	55° F.	64° F.	65.3° F.
TEDDINGTON RESULTS AT OTHER TEMPERATURES							
Actual Tank Results:	12		1641	1655		1637	1630
	14		2809	2839		2823	2817
With 10% standard-allowance:	12		1805	1820		1801	1793
	14		3090	3123		3105	3099

COMPARISON OF RESULTS OF PROPULSION EXPERIMENTS.
SCREWS B1 AND B8 WITH MODEL 1119.
AT HAMBURG, WAGENINGEN AND TEDDINGTON

1. *Hamburg Results* (64° *F.*) („*without friction deduction*")

Screw	B1		B8	
Speed Kts	12	14	12	14
v_s (ft/sec)	4.965	5.793	4.965	5.793
v_s^2	24.65	33.56	24.65	33.56
revs/sec	5.5	6.557	5.42	6.475
C_D ("k_1")	**.9028**	**.8835**	**.9161**	**.8948**
resistance (g)	4300	6040	4300	6040
,, lbs (r)	9.48	13.317	9.48	13.317
$C = r \times \left(\frac{1.528}{v_s}\right)^2$	.898	.927	.898	.927
$C_R = \frac{C}{2.336}$ ("e")	**.3844**	**.3968**	**.3844**	**.3968**
thrust (g)	5280	7540	5280	7540
,, lbs	11.65	16.62	11.65	16.62
C_T ("y")	**.4726**	**.4953**	**.4726**	**.4953**
torque c.m.g.	27270	39260	27270	39260
,, lbs ft	1.973	2.840	1.973	2.840
C_Q ("z")	**.0800**	**.08462**	**.0800**	**.08462**
$\psi = \frac{ek_1}{2\pi z}$	**.690**	**.659**	**.700**	**.667**
$t = 1 - \frac{e}{y}$	**.187**	**.199**	**.187**	**.199**

2. *Hamburg results* (64° *F.*) („*with friction deduction*")

Screw	B1		B8	
Speed Kts	12	14	12	14
v_s (ft/sec)	4.965	5.793	4.965	5.793
v_s^2	24.65	33.56	24.65	33.56
revs/sec	5.08	6.055	4.98	5.95
C_D ("k_1")	**.9775**	**.9567**	**.997**	**.9738**

resistance (g)	3160	4535	3160	4535
,, lbs (r)	6.967	9.999	6.967	9.999
$C = r \times \left(\frac{1.528}{v_s}\right)^2$ ..	.660	.696	.660	.696
$C_R = \frac{C}{2.336}$ ("e") ...	**.2825**	**.298**	**.2825**	**.298**
thrust (g)	4010	5870	4010	5870
,, lbs	8.84	12.94	8.84	12.94
C_T ("y")	**.3586**	**.3856**	**.3586**	**.3856**
torque (c.m.g.)	21770	31530	21770	31530
,, lbs ft	1.575	2.281	1.575	2.281
C_Q ("z")	**.0639**	**.06797**	**.0639**	**.06797**
$\psi = \frac{ek_1}{2\pi z}$	**.688**	**.668**	**.702**	**.679**
$t = 1 - \frac{e}{y}$	**.212**	**.227**	**.212**	**.227**
tow-rope force (g.)	1140	1505		
,, ,, (lbs)	2.513	3.318		
,, C	.238	.231	= S.F.C. as used	
	.247	.240	= S.F.C. from FROUDE	

3. *Wageningen results with screw B1 at 62.2° F.*

Speed Knots	12	14
v_s (ft/sec)	4.965	5.793
v_s^2	24.65	33.56
revs/sec	5.06	6.10
C_D ("k_1")	**.9813**	**.9497**
tow-rope force (lbs)	2.513	3.329
,, C (S.F.C.)	.238	.232
Total C	.889	.939
Propulsion C	.651	.707
$C_R = \frac{C}{2.336}$ ("e")	**.2787**	**.3026**
thrust (lbs)	9.436	14.108
C_T "y")	**.3827**	**.4204**

torque (lbs. ft)	1.638	2.3685
C_Q ("z")	**.06645**	**.0706**
$\psi = \dfrac{ek_1}{2\pi z}$	**.655**	**.648**
$t = 1 - \dfrac{e}{y}$	**.272**	**.280**

4. *Wageningen results with screw B8 at 65.3° F.*

Speed Knots	12	14
v_s (ft/sec)	4.965	5.793
v_s^2	24.65	33.56
revs/sec	4.975	5.975
C_D ("k_1")	**.998**	**.9696**
tow-rope force (lbs)	2.441	3.241
,, ©	.231	.226
total ©	.882	.932
propulsion ©	.651	.706
$C_R = \dfrac{©}{2.336}$ ("e")	**.2787**	**.3022**
thrust (lbs)	9.194	13.512
C_T ("y")	**.373**	**.4027**
torque (lbs ft)	1.589	2.2688
C_Q ("z")	**.06447**	**.06763**
$\psi = \dfrac{ek_1}{2\pi z}$	**.687**	**.6894**
$t = 1 - \dfrac{e}{y}$	**.253**	**.249**

5. *Teddington results without friction deduction of 24.11.32. (Temp. 60° F.)*

Screw	B1		B8	
Speed Kts	12	14	12	14
C_D ("k_1")	**.8962**	**.8834**	**.9064**	**.8894**
C_R ("e")	**.397**	**.415**	**.397**	**.415**
C_T ("y")	**.5045**	**.524**	**.499**	**.528**
C_Q ("z")	**.0797**	**.0825**	**.0806**	**.0848**
$\psi = \dfrac{ek_1}{2\pi z}$	**.711**	**.707**	**.711**	**.693**
$t = 1 - \dfrac{e}{y}$	**.213**	**.208**	**.204**	**.214**

6. *Teddington results of 24.11.32; to compare with Hamburg (at 64° F.); „without friction deduction". Assuming corrn. for temp. diffce. = –.009 C = –.004* C_R.

Screw	B1		B8	
Speed Kts	12	14	12	14
C_D ("k_1")	**.8988**	**.886**	**.9092**	**.892**
C_T ("y")	**.500**	**.520**	**.4945**	**.524**
C_Q ("z")	**.0791**	**.0817**	**.0799**	**.084**
C 55°	.940	.981	.940	.981
C 64° (corrn. –.021)	.919	.960	.919	.960
$C_R = \frac{C}{2.336}$ ("e")	**.3934**	**.411**	**.3934**	**.411**
$\psi = \frac{ek_1}{2\pi z}$	**.711**	**.709**	**.712**	**.694**
$t = 1 - \frac{e}{y}$	**.213**	**.210**	**.204**	**.216**

7. *Teddington results of 24.11.32; to compare with Hamburg (at 64°); „with friction deduction".*

Speed Kts	12	14
Hamburg S.F.C. as used (from 2)	.238	.231
corrg. C_R	.1019	.0989
+ temp. diffce. C_R (.004)	**.1059**	**.1029**
total C at 64° (from 6)	.919	.960
propulsion C	.681	.729
corrg. C_R ("e")	**.2915**	**.312**

Screw	B1		B8	
Speed Kts	12	14	12	14
C_D ("k_1")	**.982**	**.9631**	**.9915**	**.9704**
C_T ("y")	**.3785**	**.402**	**.3765**	**.404**
C_Q ("z")	**.0616**	**.06485**	**.0619**	**.0658**
$\psi = \frac{ek_1}{2\pi z}$	**.740**	**.738**	**.743**	**.732**
$t = 1 - \frac{e}{y}$	**.230**	**.224**	**.226**	**.228**

8. *Teddington results of 24.11.32 to compare with Wageningen for Screw B1. (Temperature 62.24° F.)*

Speed Knots	12	14
temperature correction (C)	–.005	–.005
S.F.C. as used (from (3))	–.238	–.232
total correction (C)	–.243	–.237
total correction $C_R = \frac{C}{2.336}$	**–.1040**	**–.1014**
total (C) at 55°	.940	.981
total (C) at 62.2° (corrn –.017)	.923	.964
S.F.C. as used	.238	.232
propulsion (C)	.685	.732
propulsion $C_R = \frac{C}{2.336}$ ("e")	**.2932**	**.3133**
C_D ("k_1")	**.9803**	**.962**
C_T ("y")	**.3805**	**.4035**
C_Q ("z")	**.0619**	**.0651**
$\psi = \frac{ek_1}{2\pi z}$	**.739**	**.737**
$t = 1 - \frac{e}{y}$	**.229**	**.223**

9. *Teddington results of 24.11.32 to compare with Wageningen for Screw B8. (Temperature 65.3° F.)*

Speed Knots	12	14
temperature correction (C)	–.012	–.012
S.F.C. as used (from (4))	–.231	–.226
total correction (C)	–.243	–.238
total correction $C_R = \frac{C}{2.336}$	**–.1040**	**–.1019**
total (C) at 55°	.940	.981
total (C) at 65,3° (corrn –.024)	.916	.957

S.F.C. as used	.231	.226
propulsion (C)	.685	.731
propulsion $C_R = \dfrac{\text{(C)}}{2.336}$ ("e")	**.2932**	**.3129**
C_D ("k_1")	**.9898**	**.9697**
C_T ("y")	**.378**	**.4055**
C_Q ("z")	**.0623**	**.0661**
$\psi = \dfrac{ek_1}{2\pi z}$	**.741**	**.731**
$t = 1 - \dfrac{e}{y}$	.224	.228

TEDDINGTON RESULTS (SHIP FIGURES)

1. *Usual Method: Using ψ of Model ⊗.*

a. Good Trial Conditions. (E.H.P. + 10%).

Screw	Knots	E.H.P. + 10%	ψ	D.H.P.	N
B1	12	1411	.711	1985	77.3
B1	14	2394	.707	3386	92.1
B8	12	1411	.711	1985	76.6
B8	14	2394	.693	3454	91.6

b. Average Sea Conditions; (E.H.P. + 25%).

Screw	Knots	E.H.P. + 25%	ψ	D.H.P.	N
B1	12	1604	.711	2256	80.0
B1	14	2720	.707	3848	95.3
B8	12	1604	.711	2256	79.2
B8	14	2720	.693	3925	94.6

(Calcns E.H.P. 8/8. & Plottings T_2,/6_A)

TEDDINGTON RESULTS (SHIP FIGURES)

2. *Elaboration of Foregoing Method.* Taking ψ as variable with slip, and obtaining D.H.P. and N from intersection of D.H.P. curves as obtained from C_Q and E.H.P. and ψ.

a. Good Trial Conditions (E.H.P. + 10%).

Screw	Knots	E.H.P. + 10%	ψ	D.H.P.	N
B1	12	1411	.730	1934	76.8
B1	14	2394	.728	3288	91.3
B8	12	1411	.734	1922	76.0
B8	14	2394	.717	3338	90.8

b. Average Sea (E.H.P. + 25%).

Screw	Knots	E.H.P. + 25%.	ψ	D.H.P.	N
B1	12	1604	.716	2239	79.9
B1	14	2720	.713	3815	95.1
B8	12	1604	.719	2230	79.0
B8	14	2720	.700	3887	94.4

(Calcns. A.W.R. 7/37)

TEDDINGTON RESULTS (SHIP FIGURES)

3. *Model Underpropelled by Amount* ≡ S.F.C. (Continental Method) (Figures for 55° F.) No Allowance for Appendages or Sea, etc.)

Screw	Knots	E.H.P.	ψ	D.H.P.	N
B1	12	1283	.740	1734	74.7
B1	14	2176	.737	2954	88.8
B8	12	1283	.744	1725	74.0
B8	14	2176	.730	2982	88.2

(Calcns 7/41)

4. *Results as Above but for 64°* F. (To compare with Hamburg.)

Screw	Knots	E.H.P.	ψ	D.H.P.	N
B1	12	1244	.758	1641	73.7
B1	14	2115	.753	2809	87.7
B8	12	1244	.760	1637	73.0
B8	14	2115	.749	2823	87.0

(Calcns 7/41)

5. *Results at 64° F. but using Hamburg figures for S.F.C.*

Screw	Knots	E.H.P.	ψ	D.H.P.	N
B1	12	1261	.756	1667	74.0
B1	14	2142	.753	2845	88.0
B8	12	1261	.759	1662	73.3
B8	14	2142	.747	2869	87.3

(Calcns 7/44)

6. *Results at 62.2° F. to compare with Wageningen;* (Screw B1 only.) („FROUDE" S.F.C.)

Screw	Knots	E.H.P.	ψ	D.H.P.	N
B1	12	1252	.7565	1655	73.8
B1	14	2126	.749	2839	87.8

7. *Results at 62.2° F. using Wageningen S.F.C.*

Screw	Knots	E.H.P.	ψ	D.H.P.	N
B1	12	1266	.756	1674	74.0
B1	14	2145	.752	2852	88.0

(Calcns 7/45)

TEDDINGTON RESULTS (SHIP FIGURES)

8. *Results at 65.3° F. to compare with Wageningen;* Screw B8 only. („FROUDE" S.F.C.)

Screw	Knots	E.H.P.	ψ	D.H.P.	N
B8	12	1239	.760	1630	72.9
B8	14	2106	.748	2817	87.0

9. *Results at 65.3° F. using Wageningen S.F.C.*

Screw	Knots	E.H.P.	ψ	D.H.P.	N
B8	12	1261	.759	1662	73.3
B8	14	2142	.748	2864	87.3

(Calcns. 7/46)

THE FORM CHARACTERISTIC OF SHIPS HULLS

BY PROF. H. R. MØRCH

(TRONDHJEM)

The realisation of the great value of tank work seems to broaden amongst as well professional men as others. Practical ship designers make more and more use of the suggestions which tank work continuously produces. No ship of any great value is built at present, without being carefully designed and tested in an experimental tank. The rule of the thumb has practically given way in ship design work to methods which satisfy science. It is no longer left to the experimental tanks and the navies alone to plan and carry out systematic experiments with regard to hulls, propellers etc.; also the larger shipowning companies and shipyards do so in their own interest ré specialised types.

Of the experiments which are carried out in the ship experimental tanks, none are of greater value to ship designers as a whole than the systematic series, by reason of their great educational value to the students at the universities and practical value to the designing offices. These systematic series embody the hull, propeller and their interwork as other problems regarding ship designing. The detached experiments, executed for shipyards, shipowners etc. concerning special ships, are of much less importance in this connection. It is the law of resistance against propulsion of hulls, performance of propellers etc., connected to certain definite variations in form, size etc., which is of lasting value to shipdesigners. In order to deduce such laws, the hulls as well as the propellers have to be treated separately, finally the reactions between the hull and propeller, inclusive wake questions. As the number within each group is large, both subjected to developments in size and form with time, it follows that the experimental work is overwhelming and can only be solved in its broad features, if co-operation between all good powers could be arranged. A general plan for such systematic experimental work should therefore be put down and if possible sanctioned by an international conference, consisting besides other interested such as ship designers etc. of experts from our shipmodeltanks and other hydro-and aerodynamical experimental stations.

The conference regarding the hydrodynamical problems under ship propulsion, recently held at Hamburg, has shown possibilities for successful international co-operation in this respect. Before such a co-operation however may be started and before shipdesigners as a whole will be able to reap the full benefit hereof, a common plan ought also to include an internationally adopted method for putting down the experimental results. It is hoped that a general method with factors independent of the british or metric measures soon will find general agreement.

An international agreement concerning co-operation in the systematical experimental work however should be harder to obtain, remembering that each country has its own specialities regarding types of ships etc. The co-operation should therefore follow a general plan where the main features only are outlined. It must be left to each country or experimental tank to choose those systematic series, types and sizes of ships etc. which are found to be most appropriate to them. In my own country Norway — for instance, systematic experimenting might benefitly include all types and sizes of fishing boats, combined passenger and cargo boats for norwegian coastal service and cargoboats up to 6000 ton D.W. Communications ought to be given the public in each case, when such systematic experiments are started and in time the results hereof. In order to illustrate how such a plan might be developed, I will in the following only take up a discussion of the systematic variations which might be done concerning ships hulls.

We distinguish between the emerged and submerged body of the hull. With regard to the resistance to propulsion the emerged body is only of interest in connection with wind resistance and the propulsion amongst waves. The submerged body comprises the main body and the supplements, such as keels, rudders, shaftbrackets, bossings etc. We will in the following only consider the main body. The sheer draught gives exactly the form of the hull. This drawing together with one of the main dimensions-length L, breadth B or draft d, fixes the *size* of the ship, for propulsive purposes the displacement $\triangle$. A suitable factor for the mean value of the displacement pr. unit length of the ship is $\left(\frac{\Delta/\gamma}{L^3}\right)$ = ,,a'', dimensionless in character. It is not possible to-day to express the form of a ship by simple mathematical expressions. It is however possible to do so with regard to the lines of intersection between the hull and certain planes, such as waterlines, transverse sections, buttocks, diagonals etc. and also the sectional area curves for ,,entrance'' and ,,run''. History tells us many attempts in this direction. In the middle of the 18th century the Swedish naval architect Fredric Henrich af CHAPMAN analysed the form of the sectional area curves for the best sailing ships at that time and compared these with known geometrical curves (parabolas). From such analysis he was able to design very good sailing ships. And in the middle of the 19th century the well known british naval architect JOHN SCOTT RUSSEL put forward his ,,wave-principle'' thus obtaining waterlines in the entrance of sinoide shape, in the run of trochoide form. The form of the transverse sections were thus also dependent upon the size and form of the $\otimes$ section, the length of the parallel body and its position fore and aft in the ship. Finally I wish to emphazise D. W. TAYLOR's method, which apparently during many years has been used at the Washington tank and published by TAYLOR at the international conference at San Fransisco in 1915. According to this method the sectional area curve as well as the design waterline (CWL) in their respective entrance and run are given by parabolas of the 5th degree $y = t.x + a.x^2 + b.x^3 + c.x^4 + d.x^5$. By inserting the eligible quantities prismatic coefficient C_p, the inclination of the curve t at both ends of the ship, also the curvature of the curves at $\otimes$ or parallel body $k = \frac{d^2y}{dx^2}$, the equation is reduced to the form $y = A + B.Cp + C.t + k.D$, where the quantities A, B, C and D only contain x and constants. By the use of special tables the values of y (equal sectional area or CWL breadths) are rapidly deduced after values of C_p or C_{WL}, t and k have been chosen. The determination of the transverse sections follow now by employing known geometrical curves, f.inst. hyperbolas for sections with an area coefficient above 0,72 and parabolas for sections of less area coefficient. This work is also performed in tables after the values of inclination of curve at keel, and CWL, also area coefficient, are determined upon and faired up for the whole length of the ship. This method of TAYLOR seems to suit systematic work very well, as it embodies much of mathematical precision.

In order to deduce the systematic variations in the form of hulls which are practically possible, the following remarks may be of some interest. From the results of published systematic experimental work in connection with the tow-rope resistance of ships we may gather that provided the displacement V and length L of a ship are given, the residuary resistance pr. ton displacement will, in order of importance, mainly depend upon:

1. The form of the sectional area curve.
2. The ratio of beam to draft.
3. The form of CWL line.

When this is so, we obtain the great advantage of discussing the 3-dimensional form of the ship by use of 2-dimensional curves, viz. the sectional area — and CWL curves.

The area of the sectional area curve represents the displacement and the form of the curve the distribution of the displacement along the length of the ship. In the same way we have for the CWL curve the area of the waterline and its distribution along the ship's length. The size and form of these curves are determined by the length of the ship L, the max. ordinate $S_\otimes$ or B/2, the position of the max. ordinate fore and aft and the form of the ends — „entrance'' of length L_1 in front of $\otimes$ or „parallel body'' of length L_2 and „run'' of length L_3, aft of $\otimes$ section or the parallel body. The position of $\otimes$ section or parallel body fore and aft is best given by the ratio L_1/L_3. The area under the sectional area curve for the entrance is V_1, that of the parallel body V_2 and that of the run V_3. These areas in relation to the circumscribed rectangles are the prismatic coefficients Cp_1, $Cp_2 = 1{,}0$ and Cp_3. In the same manner we have for the CWL curve A_1, A_2 and A_3, also C_{WL1}, $C_{WL2} = 1{,}0$ and C_{WL3}. As we know, these coefficients give only very imperfectly the distribution of the area along the length of these curves. This distribution may however be perfectly noted by the use of TAYLOR's mathematical method or by a so-called 10-divisional diagram, where the base corresponds to the lengths of the curves, the height to $S_\otimes$ or B/2.

A sectional area curve shows:

$L = L_1 + L_2 + L_3$, viz. $1 = \% + (L_1/L + L_3/L)$ where $\% = L_2/L$

$V = V_1 + V_2 + V_3$ and $Cp = \frac{V}{LS_\otimes} = \% + \left\{Cp_1\left(\frac{L_1}{L}\right) + Cp_3\left(\frac{L_3}{L}\right)\right\}$.

For ships without parallel body % vanishes. Similar relations exist also for the CWL curve.

For a transverse section of the ship we have generally $C = \frac{S}{Bd}$ where S is the value of a corresponding ordinate of the sectional area curve. The form of a transverse section is best given by TAYLOR's mathematical method or by the ratio B/d and a 10-divisional diagram, where the base and height correspond to the half breadth and draft respectively.

Provided the ⊗ section or parallel body remains in the same fore and aft position (L_1/L_3 constant), table I gives the possible combinations which can be made between the 5 determining quantities L, $S_{\otimes}$ (for brevity we write in the table only S), V, % and F. F denotes the form of the sectional area curve, respectively the CWL curve at the entrance or run, given by TAYLOR's method or by a 10 divisional diagram. If F is constant, so are Cp_1 and Cp_3, respectively C_{WL1} and C_{WL3}. The following quantities are given in table I.

$$a = \frac{V}{L^3},\ Cp = \frac{V}{LS},\ m_3 = \frac{L_3}{S^{1/2}} \text{ etc.}$$

This table also includes published experimental work of systematic nature with regard to the residuary resistance of ship together with the form characteristic of the models used. It should be noted that in some of these experiments the ratio L_1/L_3 has been altered. This may influence the general conclusions given to the left in the table I, which provided in general for a constant L_1/L_3 ratio.

Similarly table 2 gives the possible combinations which can be made with regard to the ⊗ section, namely the quantities $S_{\otimes}$ (for brevity we write in the table only S), B, d and F. F denotes the form of the ⊗ section given either by TAYLOR's method or by a 10-divisional diagram. If F is constant, so is $C_{\otimes}$ the area coefficient. $S_{\otimes} = C_{\otimes}$ (B.d). The size and form of a ⊗ section is given when we know F from a 10-divisional diagram and the values of B and d. Table 2 also includes published systematical experimental work regarding residuary resistance of ships.

Provided the 5 quantities in table I L, S, V, % and F are kept constant, a variation in the ratio L_1/L_3, viz shifting the ⊗ section or parallel body fore and aft in the ship, we have: New total displacement V^1 = the former displacement $V + u\, S_{\otimes}$ $(Cp_1 - Cp_3)$, where u is the movement fore and aft of the ⊗

a or M	C_p	C_{p_1}	C_{p_3}	C_{WL}	C_{WL_1}
Var. 168–434	Var. 0,689–0,827	Var. 0,71–0,755	Var. 0,631–0,703		
Var. 221–230	Var. 0,80–0,83	Var. 0,70–0,72	Var. 0,68–0,70		
Var. 200–350	Var. 0,839–0,87	Var. 0,594–0,647	Var 0,56–0,67		
Var. 5,213–5,852	Var. 0,554–0,617	Var. 0,539–0,598	Var. 0,570–0,635		
Var. 49–304 ,, 90–225	Var. 0,55–0,86 ,, 0,66–0,83	Const. 0,526 ,, 0,672	Const. 0,57 ,, 0,638		
Var. 30–180 Variable	Const. 0,56 Const. 0,544, 0,6 & & 0,67	$C_{p_1} = C_{p_3} = C_p$			
& Var. 61–155	Var. 0,53–0,674	$C_{p_1} = C_{p_3} = C_p$			
Var. 20–160 ,, 4,5–1,0 ,, 104–307 ,, 59,5–4,50 ,, 61–155 ,, 59–135	Const. 0,48–0,80 ,, 0,554–0,617 ,, 0,655–0,827 ,, 0,622–0,865 ,, 0,554–0,665 ,, 0,594–0,665	$C_{p_1} = C_{p_3} = C_p$ Const. 0,539–0,598 Const. ,, $C_{p_1} = C_{p_3} = C_p$ $C_{p_1} = C_{p_3} = C_p$	Const. 0,57–0,635 Const. ,,		
Const. 123 ,, 129 ,, 145 ,, 86	Var. 0,57–0,61 ,, 0,60–0,64 ,, 0,51–0,74 ,, 0,59–0,70	Var. 0,552–0,58 $C_{p_1} = C_{p_3} = C_p$	Var. 0,598–0,639		$C_{WL_3}–C_{WL_1} = 0,05$
Var. 144–179 ,, 156–178	Var. 0,664–0,843 ,, 0,72– 0,82	Const. 0,625–0,72 ,, 0,67	Const. 0,58–0,70 ,, 0,66		
,, 175–200	,, 0,74 –0,84				
Var. 92,5–154 ,, 168–222 ,, 119–127 ,, 141–179 ,, 138–146 Const. 167 & 272	Var. 0,48 –0,80 ,, 0,632–0,835 ,, 0,556–0,594 ,, 0,664–0,843 ,, 0,625–0,686 Const. 0,768 & 0,826	Var. 0,48 –0,80 ,, Var. 0,526–0,571 ,, 0,625–0,72 ,, 0,625–0,72 ,,	Var. 0,48 –0,80 ,, Var. 0,586–0,616 ,, 0,578–0,70 ,, 0,578–0,70 ,,		
Const. 57–172 ,, 136–178	Const. 0,68–0,80 ,, 0,68–0,87	Var. 0,46–0,80 ,, 0,59–0,76	Var. 0,47–0,80 ,, 0,53–0,77		
Const. 129 ,, 146 & 175 ,, 140 ,, 216 ,, 60 & 150 ,, 60 & 100	Const. 0,60 & 0,64 ,, 0,691–0,828 ,, 0,655 ,, 0,803 ,, 0,60 & 0,65 ,, 0,55	Const. $C_{p_1} = C_{p_3} = C_p$ Const. 0,672 ,, 0,672 ,, 0,705 $C_{p_1} = C_{p_3} = C_p$ $C_{p_1} = C_{p_3} = C_p$	Const. 0,638 ,, 0,638 ,, 0,713	Const. 0,685 & 0,72 ,, 0,61	$C_{WL_3}–C_{WL_1} = 0,05$ Const. 0,628 & 0,67 ,, 0,57

TABLE 1.

C_{WL_3}	M_3	L_1/L_3	SPEED	REMARKS
	Var. 3,37–4,15	Var. 0,79–0,90	C = 0,4–0,84	Also Tried in Ballasted Condition, Serie P
	Var. 3,05–3,85		C = 0,3–0,80	
	Var. 1,88–2,18		C = 0,3–0,85	
	Var. 4,67–5,26	Var. 1,0–1,13	(K)= 2 –4,8	Table 2 Nr 7
	Const. 3,58	Const. 1,0	(P)= 0,5 –1,15	Entrance & Run Kept Const.
	„ 4,05	„ 1,0	(P)= 0,52–0,9	Entrance & Run Kept Const.
			Vs = 5,20 m/sek	L Varied 93–169 m.
	Variable	Const. 1,0	C = 0,5–1,10	
	Var. 5,5 –8,0	Const. 1,0	C = 0,4–1,1	
	Var. 4,63–16,9	Const. 1,0	C = 0,6–2,0	Table 2 Nr 7
	„ 4 –12	„ 1,0–1,13	(K)= 2 –4,8	Table 2 Nr 7
	„ 2,31–6,70	„ 0,92–1,0	(P)= 0,3–0,85	Draft Const. Beam Varying. Table 2 Nr 10
	„ 1,96–8,48	„ 0,92–1,0	(P)= 0,3–1,0	$B/S^{1/2}$ Const. = 1,694, Table 2 Nr 4
	„ 5,5–8,0	„ 1,0	C = 0,5–1,1	Draft Const. Beam Varying. Table 2 Nr 10
	„ 5,93–8,12	„ 1,0	C = 0,5–1,1	Beam Const. Draft Varying. Table 2 Nr 9
	Const. 5,66	Var. 1,04–1,1	(P)= 0,75–1,05	
& 0,10	Var. 5,73–5,91	Const. 1,0	C = 0,45–1,15	
	„ 5,04–6,3		C = 0,6 –1,0	
			(P)= 0,6 –1,05	
	Var. 2,9 –5,19	Const. 1,0	(P)= 0,3–0,975	
	„ 3,52–4,66	Var. 0,46–1,10	C = 0,3–0,80	
	„ 2,65–5,28	„ 0,56–1,20	C = 0,3–0,80	Tried at 4 Drafts
	Const. 6,08	Const. 1,0	C = 0,6 –2,0	Table 2 Nr 7
	„ 2,66–5,19	„ 0,9 & 1,0	(P)= 0,3 –0,85	Varying B & d, S Const. Table 2 Nr 7
	„ 5,59	„ 1,07	(P)= 0,75–1,05	
	„ 2,9, 4,06 & 5,19	„ 1,0	(P)= 0,3 –0,975	
	„	„ 1,0	(P)= 0,35–1,05	
	„ 3,12 & 4,15	„ 0,81 & 0,905	(P)= 0,30–0,70	Serie Q
	Var. 2 –10	Const. 1,0	C = 0,5 –1,8	
	„ 1,77–5,95	„ 1,0	C = 0,3 –1,0	
& 0,10	Const. 5,73 & 5,91	Const. 1,0	C = 0,45–1,15	
	„ 2,9, 4,06 & 5,19	„ 1,0	(P)= 0,3 –0,80	
	„ 5,80	„ 1,0	(P)= 0,35–1,05	
	„ 3,73	„ 0,857	C = 0,4 –0,625	
Const. 0,75 &0,775	„ 5,55 & 8,36	„ 1,0	C = 0,4 –1,50	$S_{FP}/S\otimes$ = 0–0,20, t = 0–4,0
„ 0,65	„ 8,10	„ 1,0	C = 0,6 –1,30	„ = 0–0,08, t = 0–2,0

NR	CONST.	VARIAB.	CONSEQUENTLY CONST.	CONSEQUENTLY VARIAB.	EXPERIMENTS BY	MAIN OBJECT
1	L	$S\nabla\%F$		$Cp\ a\ M_3$	Todd I.N.A. 1931	Influence of Size &Form
2	S	$L\nabla\%F$				
3	∇	$LS\%F$		a		
4	%	$LS\nabla F$				
5	F	$LS\nabla\%$	$Cp_1\ Cp_3$			
6	LS	$\nabla\%F$		$Cp\ a\ M_3$	Sadler S.N.A. & M.E. 1909	Influence of Size & Form
7	$L\nabla$	$S\%F$	a	Cp		
8	L%	$S\nabla F$		a	Sadler S.N.A. & M.E. 1909	Influence of Size & Form
9	LF	$S\nabla\%$	$Cp_1\ Cp_3$	a		
10	$S\nabla$	$L\%F$		a		
11	S%	$L\nabla F$			R. E. Froude I.N.A. 1904	Influence of Snobbing
12	SF	$L\nabla\%$	$Cp_1\ Cp_3\ M_3$	Cp a	W. Froude I.N.A. 1877 Kent ,, 1915	Influence of Length
13	$\nabla\%$	LSF		a		
14	∇F	LS%	$Cp_1\ Cp_3$	a		
15	%F	$LS\nabla$	$Cp_1\ Cp_3$			
16	$\nabla\%F$	LS	$Cp\ Cp_1\ Cp_3$	$a\ M_3$	Taylor-Hisbook	Influence of Length
17	S%F	$L\nabla$	$Cp\ Cp_1\ Cp_3$	$a\ M_3$	Sadler S.N.A. & M.E. 1914	Influence of Size & Length
18	$S\nabla F$	L%	$Cp_1\ Cp_3$	Cp a		
19	$S\nabla\%$	LF		$Cp\ Cp_1\ Cp_3\ a\ M_3$	Sadler S.N.A. & M.E. 1914	Influence of Length & Form
20	L%F	$S\nabla$	$Cp\ Cp_1\ Cp_3$	$a\ M_3$	Taylor Stand. Serie R. E. Froude I.N.A. 1904 Kent ,, 1919 Kent ,, 1919 Sadler S.N.A. & M.E. 1914 Sadler S.N.A. & M.E. 1914 [1]	Influence of Size ,, ,, ,, ,, ,, ,, ,, ,, ,, ,, ,, ,, ,, ,, ,,
21	$L\nabla F$	S%	$a\ Cp_1\ Cp_3$	$Cp\ M_3$		
22	$L\nabla\%$	SF	a	$Cp\ Cp_1\ Cp_3\ M_3$	R. E. Froude I.N.A. 1905 Taylor S.N.A. & M.E. 1911 Taylor S.N.A. & M.E. 1913 Schaffran Schiffb. 1914	Influence of Form ,, ,, ,, ,, ,, ,, ,, ,, ,,
23	LSF	$\nabla\%$	$Cp_1\ Cp_3$	$Cp\ a\ M_3$	Baker I.N.A. 1914 Robertson S.N.A. & M.E. 1920 Sadler & Bragg S.N.A. & M.E. 1921	Influence of Size & Par. Body ,, ,, ,, ,, ,, ,, ,, ,, ,, ,, ,, ,,
24	LS%	∇F	M_3	$Cp\ Cp_1\ Cp_2$	Taylor Standard Serie Kent I.N.A. 1919 R. E. Froude ,, 1905 Baker ,, 1914 Kent ,, 1915 Todd ,, 1931	Influence of Size & Form
25	$LS\nabla$	%F	Cp a	$Cp_1\ Cp_3\ M_3$	Taylor S.N.A. & M.E. 1909 Sadler S.N.A. & M.E. 1907, 08 & 09	Influence of Par. Body ,, ,, ,, ,,
26	$S\nabla\%F$	L				
27	$L\nabla\%F$	S			Combinations not Possible	
28	$LS\%F$	∇				
29	$LS\nabla F$	%				
30	$LS\nabla\%$	F	$Cp\ Cp_1\ Cp_3\ aM_3$		Taylor S.N.A. & M.E. 1911 Baker I.N.A. 1914 Kent ,, 1915 Schaffran Schiffb. 1919 Taylor Mar. Engg. 1923 Bragg S.N.A. & M.E. 1930	Influence of Form Bow & Run ,, ,, ,, ,, ,, ,, ,, ,, ,, ,, ,, ,, Influence of Form Run Influence of Bulbous Bow ,, ,, ,, ,,

Note S. used instead of $S_{\otimes}$.

[1]) See Also Ackerson S.N.A. & M.E 1930 — Const $Cp = 0.671$, $C_{\otimes} = 0.98$, Cp_1 & Cp_3, C_{WL} etc

OBJECT	%	$B/_d$	$C_\otimes$	a or (M
Size &Form	Var. 5–46	Var. 2,15–2,25	Const. 0,98	Var. 168–43
Size & Form	Var. 30–45	Const. 3–0	Const. 0,98	Var. 221–23
Size & Form	Const. 60	Var. 3,7–6,17	Var. 0,88–0,986	Var. 200–35
Snobbing	Const. 0	Const. 2,59 & 3,47	Const. 0,8775	Var. 5,213–
Length	Var. 0–68	Const. 2,67	Const. 0,90	Var. 49–304
	,, 0–50	,, 2,25	,, 0,98	,, 90–225
Length	Const. 0	Const. 2,925	Const. 0,915	Var. 30–180
Size & Length	Const. 0	Const. 2,50	Const. 0,92	Variable
Length & Form	Const. 0	Const. 2,0, 2,5 & 3,0	Const. 0,914, 0,92 & & 0,936	Var. 61–155
Size	Const. 0	Const. 2,25 & 3,75	Const. 0,926	Var. 20–160
,,	,, 0	,, 2,59 & 3,47	,, 0,8775	,, 4,5–1,0
,,	,, 0–50	Var. 1,69–3,93	,, 0,980	,, 104–30
,,	,, 0–50	Nearly Const.	,, 0,98	,, 59,5–4,
,,	,, 0	Var. 2,0–3,0	,, 0,92	,, 61–155
,,	,, 0	,, 2,5–3,125	,, 0,92	,, 59–135
Form	Const. 0	Var. 2,86–30,2	Var. 0,90–0,926	Const. 123
,,	,, 0	Const. 2,50	Const. 0,96	,, 129
,,	,, 0	,, 2,40	Var. 0,86 –1,10	,, 145
,,		,, 2,84	,, 0,813–0,93	,, 86
Size & Par. Body	Var. 10½–50	Const. 2,25	Const. 0,98	Var. 144–17
,, ,, ,, ,,	,, 15–46	,, 2,26	,, 0,985	,, 156–17
,, ,, ,, ,,	,, 10–50	,, 2,02	,, 0,981	,, 175–20
Size & Form	Const. 0	Const. 2,25 & 3,75	Const. 0,926	Var. 92,5–15
	,, 0,50	Var. 1,81–5,47	,, 0,98	,, 168–22
	,, 0	Const. 2,8	,, 0,888	,, 119–12
	,, 10½–50	,, 2,25	,, 0,98	,, 141–17
	,, 0	,, 2,25	,, 0,98	,, 138–14
	,, 16¼–50	,, 2,15 & 2,21	,, 0,98	Const. 167 & 2
Par. Body	Var. 0–60	,, 2,50	,, 0,96	Const. 57–17
,, ,,	,, 0–70	,, 2,142	,, 0,964 &0,984	,, 136–17
Form Bow & Run	Const. 0	Const. 2,50	Const. 0,96	Const. 129
,, ,, ,, ,,	,, 10½, 30 & 50	,, 2,25	,, 0,98	,, 146 & 1
,, ,, ,, ,,	,, 0	,, 2,25	,, 0,98	,, 140
Form Run	,, 35	,, 2,235	,, 0,981	,, 216
Bulbous Bow	,, 0	,, 3,20 & 3,35	,, 0,92 & 0,99	,, 60 & 15
,, ,,	,, 0	,, 2,94	,, 0,98	,, 60 & 10

p_1 & Cp_3, C_{WL} etc & B/d = 2.25,2.50 & 2.75, *Var.* a = 50—7.50. Table 2 Nr 7.

SYSTEMATIC RESIS

NR	CONST.	VARIAB.	CONSEQUENTLY		EXPERIMENTS BY	MAIN OBJECT	
			CONST.	VARIABLE			
1	S	BdF	B/d C, [Bd]	C, [Bd] C, B/d, [Bd] B/d	Taylor S.N.A. & ME. 1908	Influence of ⊗ Form	B/
2	B	SdF	C	B/d, [Bd] C, B/d, [Bd]			
3	d	SBF	C	B/d, [Bd] C, B/d ,[Bd]			
4	F	SBd	C C, B/d	B/d, [Bd] [Bd]	Kent I.N.A. 1919	Influence of Beam & Draft	C
5	SB	dF		C, B/d, [Bd]			
6	Sd	BF		C, B/d, [Bd]			
7	SF	Bd	C, [Bd]	B/d	Taylor Standard Serie R. E. Froude I.N.A. 1904 Rota „ 1905 Kent „ 1919 Ackerson S.N.A. & M.E. 1930	Influence of B/d „ „ „ „ „ „ „ „ „ „ „ „	C „ „ „ „
8	Bd	SF	B/d, [Bd]	C			
9	BF	Sd	C	B/d, [Bd]	Sadler S.N.A. & M.E. 1914	Influence of Draft	C
10	dF	SB	C	B/d, [Bd]	Sadler S.N.A. & M.E. 1914 Kent I.N.A. 1919	Influence of Beam „ „ „	C „
11	BdF	S	C, B/d, [Bd]		Combinations not Possible		
12	SdF	B	C	B/d, [Bd]			
13	SBF	d	C	B/d, [Bd]			
14	SBd	F	C, B/d, [Bd]				

ESISTANCE EXPERIMENTS TABLE 2.

CONSTANTS	VARIABLES	REFERENCE
$B/d = 2{,}923$, $a = 27–160$, $Cp = Cp_1 = Cp_3 = 0{,}56$ & $0{,}68$, $L_1/L_3 = 1{,}0$	$C\otimes = 0{,}70–1{,}10$, $C = 0{,}6–1{,}80$	
$C\otimes = 0{,}98$ $B/S^{1/2} = 1{,}694$, $\% = 0–50$, $L_1/L_3 = 0{,}9$ & $1{,}0$	$a = 60–450$, $Cp = 0{,}62–0{,}865$, $(P) = 0{,}3–1{,}0$	Table 1, Nr 20
$C\otimes = 0{,}926$, $\% = 0$	B/d 2,25–3,75, $C = 0{,}6–2{,}0$	Table 1, Nr 20 & 24
,, $= 0{,}8775$, $\% = 0$	,, $= 2{,}59–3{,}47$, $(K) = 2\ –4{,}8$	Table 1, Nr 20 & 24
,, $= 0{,}872$, $a = 82$, $Cp = Cp_1 = Cp_3 = 0{,}561$	,, $= 1{,}68–6{,}70$, $(K) = 1{,}5–3{,}8$	
,, $= 0{,}98$, $\% = 0–50$	,, $= 1{,}81–5{,}47$, $(P) = 0{,}3–0{,}85$	Table 1, Nr 24
,, $= 0{,}98$, $\% = 0$, $a = 50–250$, $Cp = 0{,}671$	,, $= 2{,}25–2{,}75$, $C = 0{,}4–1{,}10$	Table 1, Nr 20
$C\otimes = 0{,}92$, $\% = 0$, $Cp = 0{,}544–0{,}665$	$B/d = 2{,}5–3{,}125$, $C = 0{,}5–1{,}10$	Table 1, Nr 20
$C\otimes = 0{,}92$, $\% = 0$, $Cp = 0{,}554–0{,}665$	$B/d = 2–3$, $C = 0{,}5–1{,}10$	Table 1, Nr 20
,, $= 0{,}98$, $\% = 0{,}50$, $Cp = 0{,}665–0{,}827$	,, $= 1{,}68–3{,}93$, $(P) = 0{,}3–0{,}85$	Table 1, Nr 20

section or parallel body. Such a movement will not thus alter the total displacement of the ship, provided the area coefficient (prismatic coefficient) of the entrance and run are the same. If this is not the case, a change in total displacement will follow.

We know that the position of the parallel body or ⊗ section may influence the tow-rope resistance largely. For ships of low speed length ratio the length of the run has an important influence in minimizing the eddy resistance. According to experiments of BAKER, MC ENTEE and SEMPLE $m_3 = L_3/S^{1/3}$ ought to be equal to or greater than 4,10 and the angles of waterlines, diagonals etc. must be kept low. For such slow ships value of L_1/L_3 lies usually below 1,0. As the value of the speed length ratio increases, the importance of the length of entrance for wave making resistance increases also and so the ratio L_1/L_3. As far as I know the following experimenters have studied this point viz. SADLER S.N.A. & M.E. 1909, BAKER I.N.A. 1913, MC ENTEE S.N.A. & M.E. 1918 and SEMPLE I.N.A. 1919. See table 3.

TABLE 3.

	SADLER	BAKER	MC. ENTEE	SEMPLE
	const. 0	const. 10–50	const. 33	const. 30 & 45
l	const. 2,143	const. 2,25	const. 2,25	const. 2,22
)	const. 0,935 & 0,936	const. 0,980	const. 0,98	const. 0,984
r M	const. 107 & 120	const. 116–175	const. 206	const. 164,5 & 176
		nearly const. 0,597–0,828	const. 0,788	const 0,732 & 0,782,
1	const. 0,538 & 0,606	const. 0,52–0,672	VAR 0,68 –0,695	const. 0,635 & 0,640
3		const. 0,584–0,638	VAR 0,673–0,640	const. 0,600 & 0,570
	VAR 4,84–7,25	const. 2,19 –4,06	VAR 2,50 –5,43	VAR 2,25–4,39
L_3	VAR 0,80–1,0	VAR 0,554–1,681	VAR 0,283–1,914	VAR 0,833–1,75
ed	C = 0,4–1,25	K = 0,9–3,0	C = 0,3–0,65	P = 0,38–0,56

The two parent models, used by SADLER, had entrance and run of the same prismatic coefficients, respective 0,538 and 0,606 for the two series. By changing the position of the ⊗ section fore and aft, the total displacement therefore remained constant. Such was not the case with the models of BAKER; hence the displacement varied somewhat and the mean value is given for each serie. In order to keep the displacement constant, Mc ENTEE changed the form of the sectional area curve, at both entrance and run.

There is also published some systematic experimental work with shipmodels, where the form of the transverse sections are varied, keeping the same form of sectional area curve and also in some cases the same CWL curve. See TAYLOR in S.N.A. & M.E. 1914, also in the same transactions MCENTEE 1917, SADLER 1918 and SADLER and BRAGG 1921, also D'EYNCOURT I.N.A. 1919, TODD I.N.A. 1931 en Dr. ING. ZEYSS in Werft — Reederei und Hafen october 1922. The conclusions which may be gathered from these experiments are that the form of the transverse sections has a minor influence upon the tow-rope resistance of ships, provided the sectional area curve and the CWL curve are kept constant, or minor changes take place in the CWL curve.

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CHAPTER I

REMARKS AND PROPOSALS ON THE COORDINATION IN TANKWORK

The more science tries to clarify the numerous problems relating to ship resistance and propulsion, the more it is generally felt, that collaboration among the various experimental tanks is essential to the advancement of the art and the improvement of working methods.

Since WILLIAM FROUDE carried out his pioneer work, the various tanks have worked almost independantly in developing their mode of experiment and methods of deducing ship resistance from that of models.

Up-to-date, each tank has collected a large amount of test data, which have been filed in its private records for general information of the tank authorities.

For comparative purposes, it is necessary to bring the model resistance into some standard form, or to subdivide it into skin frictional resistance and residuary resistance by the application of a temperature correction and a skin friction formula.

The publications of the various tanks demonstrate, that both the temperature correction and the skin friction formula vary from tank to tank.

It is evident, that the results depend on the calculation made; thus direct comparison is only possible if exactly the same methods of calculation and presentation are adapted.

In any case the original measurements, which have a lasting value, are lost in presenting the results in a corrected form.

It is obvious from the above, that valuable information can be satisfactorily given only by publishing the original data of the tests, and it is a matter of primary importance in order to coordinate the results, that they should be presented in the same form.

In order, that the data of different tanks may be made readily comparable it is essential that tests be made under similar conditions.

With regard to shipmodels this signifies that a model with smooth surface of given dimensions and form, with certain appendages, made and tested at the same temperature of water in different tanks, shall always give the same results within the usual limits of error.

Assuming that the resistance is accurately measured, differences may occur in the execution of the model.

The form and appendages can be made from the same drawing to a high degree of accuracy.

However the nature of the surface is liable to variation.

As it is known that frictional resistance is powerfully affected by roughness of surface, all models should be made as smooth as possible and any pores discovered, should be filled up.

As a general rule the dimensions of the model should be sufficiently large to ensure that the turbulent boundary layer is formed over the major portion of the length of hull, so that the fitting of a trip wire or the use of a local roughening of certain portions of the entrance can be dispensed with.

If the use of the latter arrangements cannot be avoided, their dimensions and position should be published with the results, as they cause an increase of the frictional resistance.

Possibly other reasons exist, such as the composition of tankwater, which cause differences in the results obtained and which could be brought to light by more extensive comparative tests in different experimental tanks.

Nowadays, many investigators are studying the problem of ship resistance, with the object of devising an improved method of deducing ship resistance from model experiments.

They have brought to light discrepancies in the method initiated by WILLIAM FROUDE. But at the same time they have shown that the problem is very complicated and that it will be difficult to obtain a universal method in accordance with the modern points of view.

The method of FROUDE and more especially his formula for frictional resistance is used with minor variations by nearly all experimental tanks, and still forms a reasonable basis for calculation and comparison. Thus it is of great importance that the small differences in the various methods of calculating should be eliminated,

Meantime, one of the principal problems facing an active international cooperation is to arrive at an improved and generally acceptable method of calculation.

In view of the foregoing remarks, the following proposals are submitted for consideration, viz:

a. a proposal for the publication of model and propeller tests;

b. a comparison of the different methods of calculation by FROUDE's formula for frictional resistance.

A. THE PUBLICATION OF MODEL AND PROPELLER TESTS

For exhaustive information on the tests made, it is necessary to give not only the results obtained but at the same time all particulars concerning the form of the model or propeller.

Let us consider in rotation:

1. ship model test; and
2. screw propeller tests.

1. *Ship model tests*

The major factors affecting resistance for a given displacement are as follows:

1. the length;
2. the midship sectional area, or by given length the prismatic coefficient;
3. the ratio's $^{L}/_{B}$ and $^{B}/_{D}$;
4. the distribution of the displacement over the length of the ship (position of the centre of buoyancy);
5. the shape of midship section and the sections at the bow and stern.
6. the profile of the ship (bow and stern);
7. the wetted surface with regard to frictional resistance.

Thus the form may be defined by:

1. the non-dimensional characteristics: Length-beam ratio, $^{L}/_{B}$; Beam-draft ratio, $^{B}/_{D}$; Length-volume ratio, $^{L}/_{Vo^{1/3}}$, Wetted surface-volume ratio, $^{S}/_{Vo^{2/3}}$; Prismatic coëfficient (Vo = volume of displacement).

2. The sectional area curve with a division of 20 ordinates

over the length of the ship and each section area given in percentages of the midship section area.

At the same time the length of parallel middle body and the position of the centre of buoyancy should be stated. Similarly the load waterline may be given in percentages of $^{B}/_{2}$ (See plate I).

3. The body plan, which should be accompanied by a table giving some sections at the bow and stern and the midship section, defined by 6 waterline ordinates.

The ordinates may be expressed in percentages of $^{B}/_{2}$.

Thus enabling easy comparison with other sectionform.

4. The contours of bow and stern with waterlines.

The drawing indicates how L is to be measured.

The centre line of shaft is of interest having regard to the point of attachment of the towing dynamometer. (See plate II.)

If the model is fitted with appendages, these should be shown on the body and profile plans.

The form given must be that of the model and not of the corresponding ship.

Thus differences in the workmanship of the model by different tanks, as for instance the finish of stem and stern and, possibly, an addition to the displacement for the shellplating, are discounted.

In this way the form is perfectly defined and for determination of the absolute size of the model only one parameter as e.g. length or volume of displacement need be fixed.

If for some reason, the curve of sectional area's and the body plan cannot be published, the shape of the model can be defined more roughly by giving:

the above non-dimensional characteristics, the length of entrance and run, the prismatic coefficient of entrance and run and tabulating: the load waterline, waterline at half draft, bilge diagonal and quarter beam diagonal.

THE PRESENTATION OF RESULTS

In testing the model at a certain temperature of water, resistance, change of trim and waveprofile can be measured at various speeds.

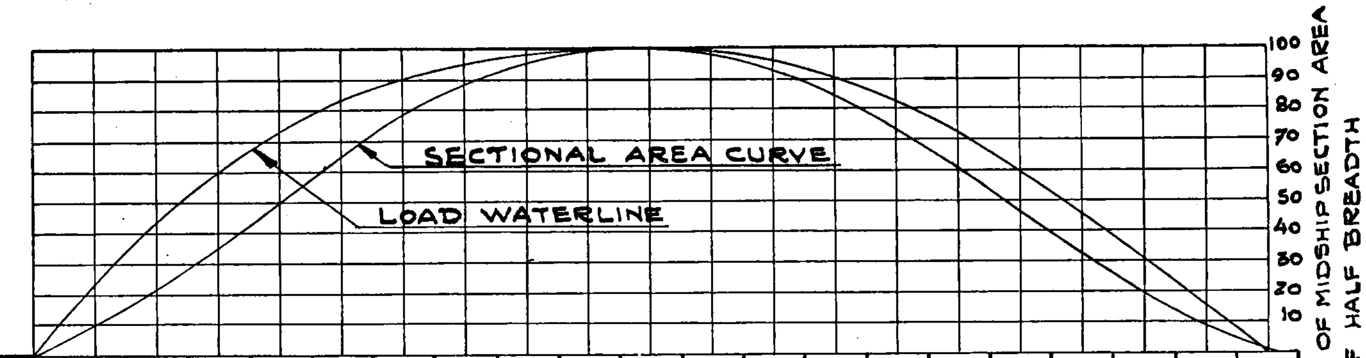
The wave profile may be photographed or measured on the model.

NON-DIMENSIONAL CHARACTERISTICS

$L/B =$ — $B/D =$ — $L/\nabla^{1/3} =$ —

$S/\nabla^{2/3} =$ — $p =$ —

SECTIONAL AREA CURVE AND LOAD WATERLINE



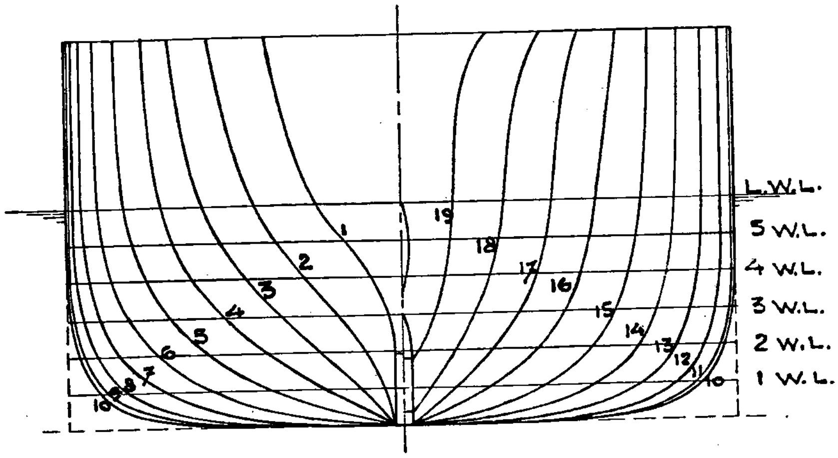
SECTION	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
SECTIONAL AREA	0	9,8	21,4	35	50	65	78,3	88,8	95,4	99,3	100	98,1	92,6	85	73,6	60,6	46,8	32,9	19,4	8,4	0,9
LOAD W.L.	0	24	43,6	59,3	73	83	90,6	95,5	98,3	99,6	100	99	96,2	90,8	82,8	72	59,3	46	31,2	15,7	0

PARALLEL MIDDLEBODY FROM ___ % TILL ___ % OF L FROM ORDINATE 0

CENTRE OF BUOYANCY ___ % OF L FROM ORDINATE 0

Plate I

BODY-PLAN



WATERLINE	1	2	3	4	5	6
SECTION 2	5.9	11.5	19	28.7	37	43.8
SECTION 4	22.7	40	52.6	62.2	68	71.6
SECTION 6	56.8	73	81.5	86.5	89	92
SECTION 10	90	96.2	99	100	100	100
SECTION 14	57.2	70.6	76	79.4	81.4	82.3
SECTION 16	25.8	40.6	48.7	53.7	56.8	58.5
SECTION 18	8	15.9	21.6	26.3	29.3	31

CONTOURS OF BOW AND STERN WITH WATERLINES

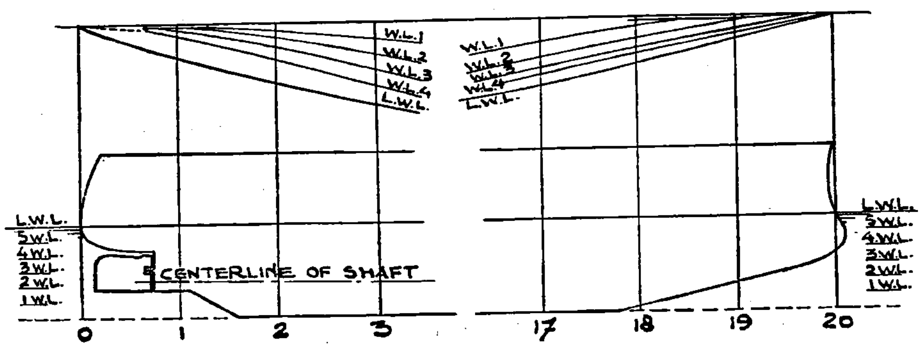


Plate I

In order to render it independent of the various systems of measurement in force in different countries, the test data should be presented in non-dimensional form.

Thus change of trim is divided by the length L.

It is well known that by dimensional reasoning the resistance of similar ships with smooth surface can be expressed by

$$\frac{R}{\varrho\, AV^2} = f\left(\frac{V}{\sqrt{g.L}}, \frac{V.L}{\nu}\right)$$

Thus we may select the following non-dimensional factors to represent the resistance:

the specific resistance $= \dfrac{R}{\varrho/_2\, AV^2}$

the „speed ratio" $= \dfrac{V}{\sqrt{g.L}}$

the „speed product" or REYNOLDS' number $= \dfrac{V.L}{\nu}$

The denominator of the specific resistance is $\varrho/_2\, AV^2$, because $\varrho/_2\, V^2$ equals the dynamical pressure of the liquid.

L = a parameter with dimensions L^1.

A = a parameter with dimensions L^2.

The parameters L and A must be clearly defined by the ship or model.

In comparing similar ships it is not important which dimensions are taken.

If, however, different forms are compared, the coefficients should be composed of the factors that have the greatest influence on resistance.

Thus for the first parameter the length of the ship is taken, as length affects both form resistance and frictional resistance.

The frictional resistance is considered to be directly proportional to the wetted surface. Having regard thereto the second parameter should be S.

As the residuary resistance is considered to be directly proportional to the volume of displacement, which is in most cases the principal dimension, A may be expressed by $V_0^{2/3}$.

In presenting the total resistance it is proposed to give a specific resistance based on both S and $V_0^{2/3}$.

It is essential that for similar ships L is determined in the same way.

If the length of the wetted surface equals that of the form as e.g. for vessels with cruiser sterns, the length on waterline should be used.

For vessels with raised stern the length is to be taken to the afterside of the rudderpost.

It is difficult to give general rules for extreme cases as when the contour of cruiser stern just touches the water for a considerable length or when a large portion of the screw aperture is out of the water.

The length L serves to determine the „speed-length ratio" and the REYNOLDS' number and at the same time is the base for determining the shape of the model.

For S is to be taken the total immersed skin area with the ship at rest.

It is probably impracticable to calculate the wetted surface in motion as at different speeds the wave profile, and hence the wetted surface varies.

As the exact calculation of the wetted surface is a laborious process, most tanks employ aproximative methods with inevitable variations in the results. In „Schiffbau No. 8, 1933, page 163", F. GUTSCHE contends, that for the determination of frictional resistance it would be more correct to introduce in the formula for frictional resistance the length multiplied by the mean girth rather than the exact wetted surface.

As the former is only slightly smaller than the latter and more capable of exact determination it is proposed to adopt this method.

In every case the wetted surface must be known for every method for deducing ship resistance from model resistance.

For the usual case, in which a model is tested at constant temperature of water, the resistance can be plotted in the form of specific resistance to a base of speed-length ratio.

With this curve the temperature of water and the length or volume of displacement of the model must be given.
Instead of speed-length ratio, the speed-volume ratio $V/\sqrt{gVo^{1/3}}$ may be taken for comparison of ships with different volume-length ratio $L/Vo^{1/3}$.

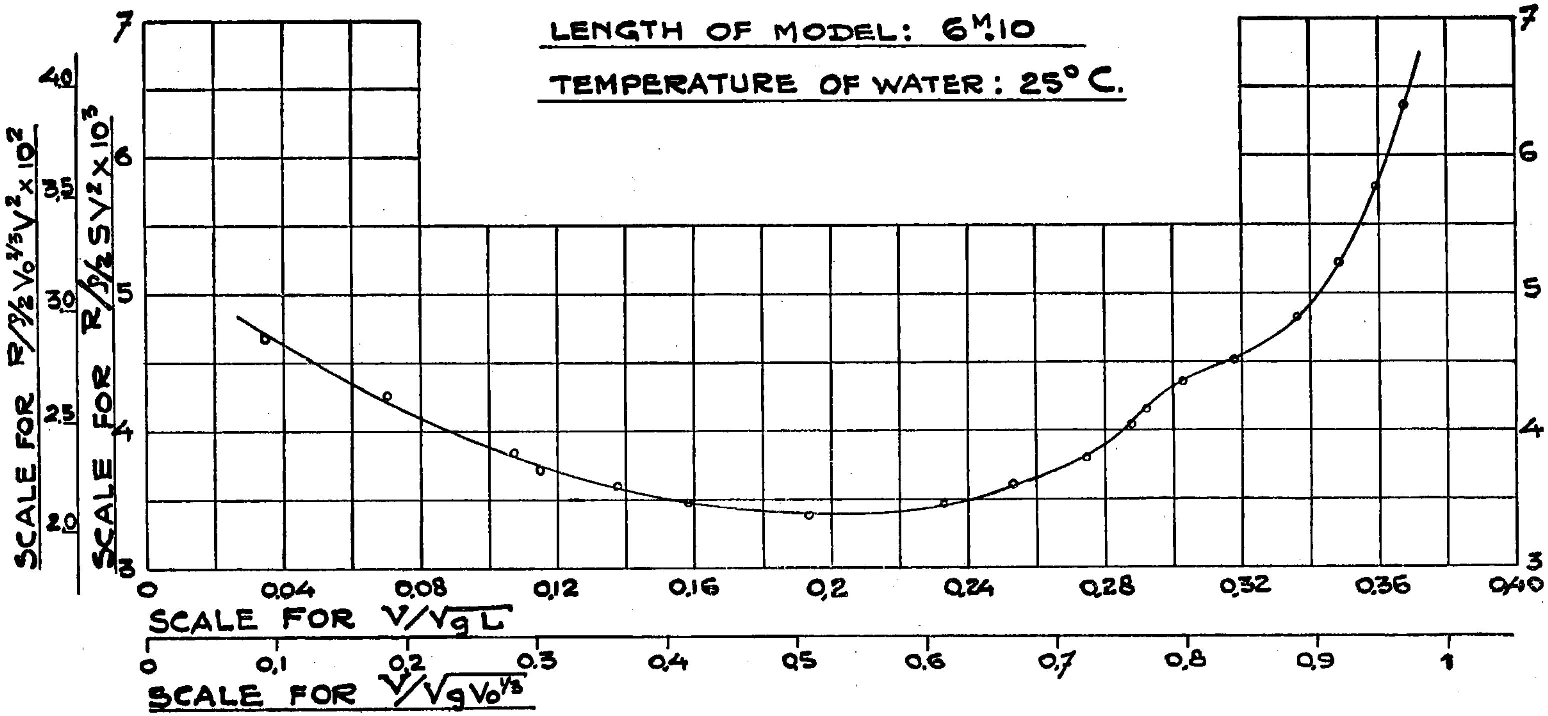
At each speed the REYNOLDS' number can be calculated and eventually used for skin friction correction. (See plate III.)

For investigation on the influence of REYNOLDS' number and

RESULT OF RESISTANCE TEST

LENGTH OF MODEL: $6^{M}.10$

TEMPERATURE OF WATER: 25° C.



$\frac{V}{\sqrt{g.L}}$	0,04	0,06	0,08	0,10	0,12	0,14	0,16	0,18	0,20	0,22	0,24	0,26	0,28	0,30	0,32	0,34	0,36	0,38
$\frac{R}{\rho/2\, S V^2} 10^3$	4,63	4,4	4,09	3,88	3,70	3,56	3,48	3,40	3,38	3,40	3,49	3,66	3,90	4.32	4.55	4.92	5,82	
$\frac{V \times L}{\nu} 10^{-6}$	2,18	3,27	4,36	5,45	6,54	7,63	8,72	9,81	10,90	11,99	13,08	14,17	15,26	16,35	17,44	18,53	19,62	20,71

Plate III

speed-length ratio on the resistance by testing geometrically-similar models at different temperature, the diagram of Telfer may be used, giving the specific resistance on a base of (REYNOLDS number)$^{-1/3}$ with curves of constant speed-length ratio.

In connection with the above it may be mentioned that R. E. FROUDE's Constant Formulae (T.I.N.A. 1888), which are used by the WILLIAM FROUDE Laboratory, Teddington, are also non-dimensional.

They are of the following nature:

$$\text{(M)} = \frac{L}{U} \qquad U = (Vo)^{1/3}$$

$$\text{(K)} = \frac{V}{U^{1/2}}\sqrt{\frac{4\pi}{g}}$$

$$\text{(C)} = \frac{R}{\frac{\text{weight per unit}}{g}\, U^2 \times V^2} \times \frac{1000}{4\pi}$$

$$\text{(L)} = \frac{\text{(K)}}{\sqrt{\text{(M)}}} = \frac{V}{\sqrt{L}}\sqrt{\frac{4\pi}{g}}$$

Thus (M) is the same as the above mentioned length-volume ratio.

(K) $= \sqrt{4\pi}$ or 3,545 times the above mentioned speed-volume ratio.

(C) $= \frac{1000}{8\pi}$ or 39,8 times the specific resistance.

(L) $= \sqrt{4\pi}$ or 3,545 times the above mentioned speed-length ratio.

PHYSICAL CONSTANTS

The above-mentioned physical constants are:

g = acceleration of gravity.

ϱ = density.

ν = coefficient of kinematic viscosity.

The acceleration of gravity is known exactly for every place on earth.

The density of fresh water varies with temperature only, these variations are exactly known.

It may be mentioned that these variations should be considered at higher temperature e.g. at 25° C. the specific gravity equals 0,99708 or is 0,3 % less than at 4° C.

The density of water at 4° C. equals 1,94015 $^{\text{lbs sec}^2}/_{\text{ft}^4}$ = = 101,97 kg $^{\text{sec}^2}/_{\text{M}^4}$.

G. S. BAKER gives 1,938 $^{\text{lbs sec}^2}/_{\text{ft}^4}$ but perhaps for a higher temperature.

As the density of salt water varies in different seas, it may be sufficient for practical purposes to make the assumption that the weight of 35 cub. feet equals 2240 lbs at 10° C.

BAKER gives the density of salt water (35 parts per 1,000) = = 1,988 $^{\text{lbs sec}^2}/_{\text{ft}^4}$.

Thus the specific gravity of salt water $= \frac{1,988}{1,94015} = 1,025.$

The coefficient of kinematic viscosity of fresh water varies with temperature only.

In view of the desirability of obtaining uniformity of practice, the following table of values employed by various investigators has been prepared in order to show the differences at present ruling:

VALUES OF COEFFICIENT OF KINEMATIC VISCOSITY OF FRESH WATER IN $^{\text{M}^2}/_{\text{Sec} \times 10^6}$

Degrees Celsius	Lyle and Hosking	Smithsonian Tables	Johow	Hütte	Baker	Physikalisch-Chemische Tabellen
0°	1.794		1,78	1,775		1,794
5°		1,5188			1,5226	1,522
10°	1,309	1,308	1,30	1,31	1,3108	1,3109
15°	1,144	1,1388	1,134		1,1427	1,143
20°	1,011	1,0068	1,00	1,01	1,006	1,0078
25°		0,8963				0,8952
30°	0,806	0,8042				0,8035

The first column gives the values as quoted by LEES (T.I.N.A. 1916). The second column: SCHOENHERR (T.S.N.A. & M.E. 1932, page 285). The third column: JOHOW. The fourth column: HUTTE. The fifth column: G. S. BAKER, Ship design, resistance and screw propulsion, Vol. I (1933). Table III, page 223. The last column contains the values calculated from Physikalisch-Chemische Tabellen (LANDOLT-BORNSTEIN) as investigated by HOSKING and LYLE (1909).

For salt water the coefficient of kinematic viscosity in $M^2/_{sec} \times 10^6$ we get the following table.

Temperature C.	0°	5°	10°	15°	20°	30°
Baker.................		1,579	1,373	1,200	1,060	
Lees (Hosking and Lyle)	1,78		1,318	1,158	1,025	0,825

The values given by Baker correspond with the investigation of O. Krummel and E. Ruppin (Physikalisch-Chemische Tabellen) with water containing 3,5% salt.

Though the differences are small it would be of great practical value to obtain uniformity in these values.

CHAPTER II

THE PUBLICATION OF SCREW PROPELLER TESTS

The principal formfactors affecting the operation of a propeller of certain diameter are:

1. number of blades;
2. the blade outline;
3. the type of sections;
4. the face pitch at a number of sections;
5. the shape of hub;
6. the rake.

Thus the propeller may be defined by:

1. the non-dimensional characteristics:
   Number of blades;
   Maximum width ratio;
   Blade thickness ratio;
   Face pitch ratio;
   Hub diameter ratio;
   Rake angle.
2. adrawing (see plate IV) showing:
   expanded outline and a number of blade sections;
   pitch diagram if pitch is not constant;
   the shape of hub with strickling line indicating the rake angle, the maximum thickness of blade sections and the tip thickness.

GENERAL REMARKS

A propeller is defined for construction by the form of hub, the position of the strickling line (rake), the shape of sections, the position of sections to the strickling line, the face pitch of these sections and the number of blades.

The developed area is used only for comparative purposes and has no fundamental value.

Instead of the developed area the expanded area may serve equally well. It differs little from the former and is better defined as several methods exist for developing blade area. The expanded contour is directly formed by the sections and no special development is required.

As indicated by TELFER in his paper ,,The presentation of propeller experiment data" (Trans. N.E.C.I.E.S., May 1924) for comparison of similar blade outlines it is more practical to give the maximum blade width ratio $^{B}/_{D}$.

This parameter affords a simple method of designing the propeller if the widths at other radii are known.

It is therefore proposed to adopt this parameter to show the proportions of the blade and to give the ratio's concerning the area of blade (M.W.R. and B.A.R.) for comparison with former test results.

SECTIONS

The sections can be taken at 20, 40, 60, 80 and 90% of D. The widths of sections are given in percentages of the maximum width. The position of sections to the strickling line may be stated as a percentage of the width of each blade as well as the radial thickness which is to be measured on a co-axial cylinder.

The maximum thickness extrapolated to the shaft is to be measured on the axis and not normal to the face.

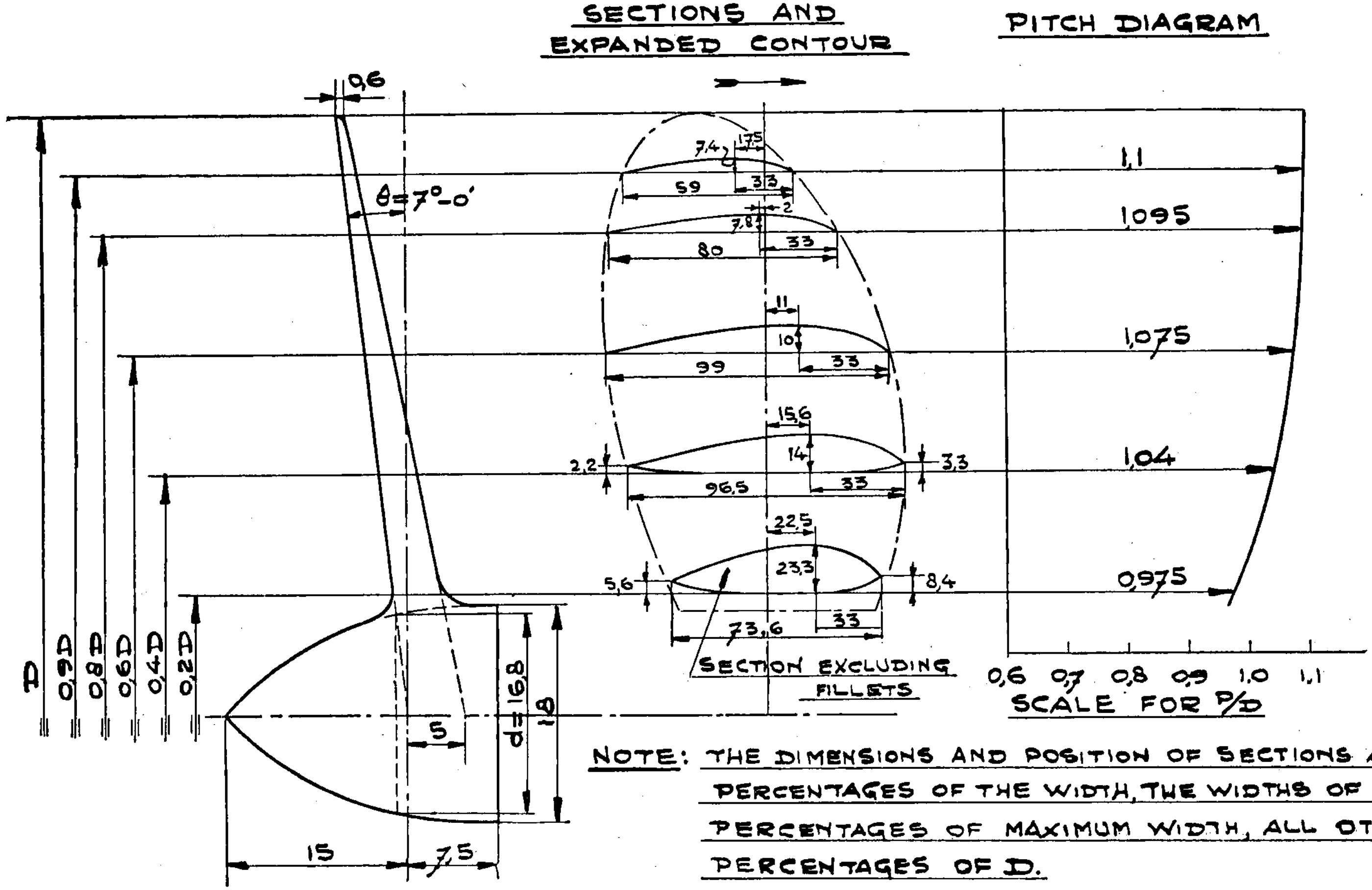
The position of maximum thickness may be given in percentages of the width, as well as other particulars on the sections (see plate IV).

It is proposed to determine the diameter of hub as indicated in plate IV.

GENERAL PLAN OF SCREW PROPELLER

CHARACTERISTICS: NUMBER OF BLADES = 4 – B.W.R = 0,24

B.T.R = 0,05 – P/D = VARIABLE – d/D = 0,168



NOTE: THE DIMENSIONS AND POSITION OF SECTIONS ARE EXPRESSED IN PERCENTAGES OF THE WIDTH, THE WIDTHS OF THE SECTIONS IN PERCENTAGES OF MAXIMUM WIDTH, ALL OTHER DIMENSIONS IN PERCENTAGES OF D.

Plate IV

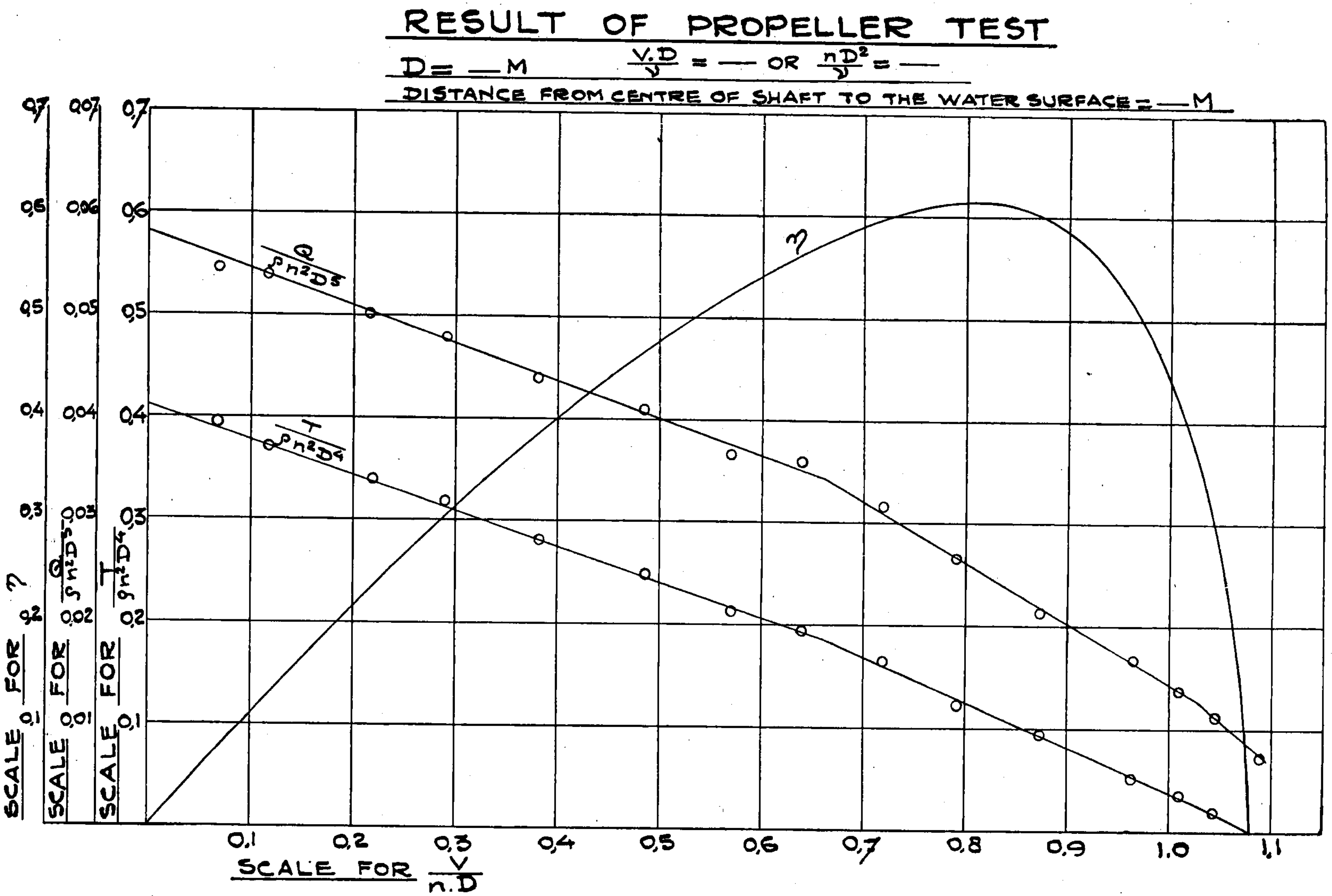

RESULT OF PROPELLER TEST
D= — M
$\frac{V.D}{\nu}$ = — OR $\frac{nD^2}{\nu}$ = —
DISTANCE FROM CENTRE OF SHAFT TO THE WATER SURFACE = — M
$\frac{Q}{\rho n^2 D^5}$
$\frac{T}{\rho n^2 D^4}$
η
SCALE FOR η
SCALE FOR $\frac{Q}{\rho n^2 D^5}$
SCALE FOR $\frac{T}{\rho n^2 D^4}$
SCALE FOR $\frac{V}{n.D}$
0,1 0,2 0,3 0,4 0,5 0,6 0,7 0,8 0,9 1.0 1,1
0,01 0,02 0,03 0,04 0,05 0,06 0,07

Plate V

By dimensional analysis TELFER comes to the following formula expressing thrust obtained with similar highly polished propellers:

$$\frac{T}{\varrho\, n^2 D^4} = f\left(\frac{V}{n.D}, \frac{VD}{\nu}, \frac{h}{\varrho\, V^2}\right), \text{ h = hydrostatic pressure.}$$

He concludes: ,,that the FROUDE propeller law which states that when $\frac{V}{n.D}$ is constant the thrust of similar propellers varies as the square of the revolutions and the fourth power of their diameter, is the best empirical relation which we have as yet as a basis of model experiment work.''

The influence of viscosity expressed by $\frac{V.D}{\nu}$ may be studied in testing geometrically similar propellers at different temperatures of water and at different speeds for the same value of $\frac{V}{n.D}$.

To be free from this influence Dr KEMPF has published the following criterion:

$$\frac{Fa/F}{z}\,\frac{n.D^2}{\nu} > 60.000, \text{ F = disc. area.}$$

Fa = developed area, z = number of blades.

The influence of the static pressure expressed by $\frac{h}{\varrho V^2}$ may be studied in the cavitation tank.

For deeply immersed slow turning model and ship propellers the influence of this term is slight as in both cases the absolute pressure on the back or front of the blade is safely above that of the water vapour.

As the influence of static pressure and viscosity will be always present it will be of interest to state at every experiment the position of the propeller in relation to the water surface and the values $\frac{V.D}{\nu}$ or $\frac{n.D^2}{\nu}$, if the experiment is made at constant speed or at constant revolutions.

Thus, as already adopted by several tanks, the experiment data may be plotted as $\frac{T}{\varrho\, n^2 D^4} \cdot \frac{Q}{\varrho\, n^2 D^5}$ and propeller efficiency $\eta = \left(\frac{T}{\varrho\, n^2 D^4} : \frac{Q}{\varrho\, n^2 D^5}\right) \frac{V}{2\pi\, n\, D}$ on a base of $\frac{V}{n.D}$ (See plate V).

As with ship model test results, a table giving the readings of the original plotting may be added.

Other bases used are:

1. slip ratio s;
2. $\frac{n.D}{V}$.

The slip ratio is not preferable as it is dependent on pitch. Apart from different interpretation of pitch viz. effective and face pitch, the pitch may be variable from section to section.

$\frac{n.D}{V}$ has the disadvantage of becoming infinite if V equals zero; this occuring in dead-pull condition.

For presentation of thrust and torque the following values are used:

1. $\frac{T}{n^2D^2P^2}$ and $\frac{Q}{n^2D^2P^3}$ by Schaffran and Washington tank.
2. $\frac{T}{D^2V^2}$ and $\frac{Q}{D^3V^2}$ by the W.F.N.T.

These coefficients are dependent on the systems of measurement as ϱ is omitted in the denominator.

Furthermore, the former have the same disadvantage as the slip ratio, while the latter become infinite at dead-pull condition.

Plotting $\frac{Q}{\varrho\, n^2D^5}$ on a base of $\frac{T}{\varrho\, n^2D^4}$, as advocated by Mr Barrillon, gives good result for comparative purposes, but is not practical in making the results suitable for general information.

SUMMARY

Having due regard to the systems at present in use an attempt has been made to define a uniform method of publication of ship model and screw propeller experiment data.

As the primary purpose is to produce a basis for co-operation in tankwork, the replotting of research data for the use of practitioners in design, which is so largely a matter of taste, has been left out of discussion.

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E. V. Telfer. The presentation of propeller experiment data. Trans N.E.C.I.E.S., May 1924.

CHAPTER III

FROUDE'S METHOD OF CALCULATION

This method is based on the separation of frictional resistance and residuary resistance.

The latter must be multiplied by $\frac{\triangle}{\delta}$ to get the ship resistance from the model resistance. The frictional resistance of the model and the ship are calculated separately by some formula.

Thus the accuracy of the results obtained depends on the accuracy of the formula employed.

Nearly all experimental tanks estimate the frictional resistance with the aid of R. E. FROUDE's formula: $R = f. S. V^{1,825}$, except the Washington tank, that formerly used TIDEMAN's formula and nowadays GEBERS' formula.

As in the latter the exponents of V are different from that of R. E. FROUDE's formula the results obtained will be different.

For comparison of the methods used by the experimental tanks other than Washington, only the frictional coefficients need be compared.

These coefficients are deduced from the Om and Os values given by R. E. FROUDE (T.I.N.A. 1888). This probably accounts for the occurrence of small differences in these coefficients.

The Om and Os values are used by the WILLIAM FROUDE Laboratory, Teddington.

Comparison of the frictional coefficients from BAKER's book „Ship form, resistance and screw propulsion'' and those from TAYLOR's book „Speed and power of ships'' gives the following result:

Length in feet	50	75	100	200
BAKER	0,0096	0,00935	0,0092	0,00898
TAYLOR	0,00963	0,00936	0,00923	0,00902

Length in feet	300	400	500	700
BAKER	0,0089	0,00883	0,00877	0,00868
TAYLOR	0,00892	0,00886	0,00880	

In checking the calculation the same values, as given by BAKER, were found.

Comparison of BAKER's values with those of continental tanks and the values obtained by the formula of LE BESNERAIS $\lambda = 0{,}1392 + \frac{0{,}258}{2{,}68+L}$ (L is length in M.) gives the following result:

L in feet .... L in M.	10 3,048	15 4,572	20 6,096	25 7,620	50 15,24	100 30,48
BAKER.........	0,18506	0,17460	0,16816	0,16413	0,15341	0,14702
Continental	0,18525	0,17484	0,16840	0,16426	0,15356	0,14731
LE BESNERAIS ..	0,18424	0,17478	0,16860	0,16425	0,15360	0,14679

L in feet .... L in M.	200 60,96	300 91,44	400 121,92	500 152,40	600 182,88	
BAKER.........	0,14350	0,14222	0,14110	0,14015	0,13935	
Continental	0,14386	0,14255	0,14143	0,14043	0,13956	
LE BESNERAIS ..	0,14325	0,14194	0,14127	0,14086	0,14059	

The values are given in M-KG-sec units.

Instead of f, continental tanks use λ and $f = \gamma \times \lambda$ (γ = specific gravity).

The English values have been converted into M-KG-sec. units by the use of the factor: $\frac{0{,}45359}{0.3048^2 \times 0{,}51476^{1{,}825}} \times \frac{1}{1{,}025} = 15{,}98.$

The continental values are a little greater than BAKER's. Perhaps the origin of this is to be found in the confusion of nautical miles and „Seemeilen".

The formula of LE BESNERAIS is used in Paris tank.

The values obtained with this formula show good agreement also with BAKER's values.

As in general the differences are very small the calculation according to the FROUDE S.F.C. method (See T.I.N.A. 1888, pag. 310) must give pratically the same result as a calculation according to the continental method.

For comparison the E.H.P. calculation is given below for one of the model tests published in H. E. SAUNDERS' paper „Test on three geometrically-similar shipmodels". (Transactions of the Society of N.A. & M.E. 1933, page 75.)

In the discussion on page 144 BAKER gives the FROUDE S.F.C. calculation of the following ship and model:

$\triangle = 14500$	model: 3127 (Washington tank)
$V = 21{,}87$ knots.	$\delta = 2105$ pounds
$S = 40355$ square feet	$v = 4{,}46\ \frac{\text{feet per min.}}{100}$
$L = 502$ feet	$s = 66{,}15$ square feet
Unit $= (35 \times 14500)^{1/3} = 79{,}765$	$l = 20{,}33$ feet
$U^2 = 6362{,}5$	$V_m = 4{,}4$ knots
$\text{(S)} = \frac{40355}{6362{,}5} = 6{,}342$	$\delta^{1/3} = 12{,}816$
	Scale $= \frac{1}{24{,}7}$

Assuming the resistance of the model at 15,7 pounds and neglecting correction for temperature of water.

$$\text{(C)} = 15{,}7 \times \left(\frac{15.248}{4{,}46 \times 12{,}816}\right)^2 = 1{,}118.$$

$l_m = 20{,}33 \qquad O_m = 0{,}1144$

$L_s = 502 \qquad O_s = 0{,}0721$

$$O_m - O_s = 0{,}0423 \times 6{,}342 = 0{,}268 \text{ S.F.C. at } \text{(L)} = 1{,}0$$

$$\text{Actual (L)} = \frac{21{,}87}{(502)^{1/2}} \times 1{,}0552 = 1{,}03.$$

$\text{(L)} = 1{,}0 \qquad 1{,}1$

S.F.C. $= 0{,}268 \qquad 0{,}264 \qquad$ take $0{,}267$

Ship $\text{(C)} = 1{,}118 - 0{,}267 = 0{,}851$

$$\text{E.H.P.} = \frac{0{,}851 \times (21{,}87)^3 \times (14500)^{2/3}}{427{,}1} = \underline{\underline{12400}}$$

CONTINENTAL METHOD

$$A = 0{,}00685926 \times \alpha^3 \times \frac{\gamma_1}{\gamma}$$

$$B = 0{,}00203925 \times \gamma_1 \left(\alpha^{0{,}0875}\, \lambda_m - \lambda_s\right) \Omega$$

$\alpha = 24{,}7 \qquad \alpha^{0{,}0875} = 1{,}324 \qquad \alpha^3 = 15069 \qquad \gamma_1 = 1{,}025$

$\Omega = 3749\ M^2 \qquad \lambda_m = 0{,}16805 \qquad \lambda_s = 0{,}14041$

$A = 0{,}00685926 \times 15069 \times 1{,}025 = 105{,}94$

$B = 0{,}00203925 \times 1{,}025 \times 0{,}08209 \times 3749 = 0{,}6431$

$\text{E.P.S.}_{\text{total}} = A \times W_m \times V_s - B \times V_s^{2{,}825}$

$W_m = 15{,}7 \times 0{,}4536 = 7{,}12$ kg
$V_s = 21{,}87$ knots $= 21{,}88$ Seemeilen $\quad V_s^{2,825} = 6107$
$E.P.S._{total} = 105{,}94 \times 21{,}88 \times 7{,}12 - 0{,}6431 \times 6107 = 12577$
$E.H.P. = \frac{12577}{1{,}0139} = \underline{\underline{12404}}$

Thus the results are practically the same.

The ship resistance may be deduced from the specific resistance by the following equation

$$\frac{R_t}{\varrho'/_2\, SV^2} = \frac{r_t}{\varrho/_2\, sv^2} - \left(\frac{r_f}{\varrho/_2\, sv^2} - \frac{R_f}{\varrho'/_2\, SV^2}\right)$$

ϱ' = density of salt water $\qquad \varrho$ = density of fresh water

$\left(\frac{r_f}{\varrho/_2 SV^2} - \frac{R_f}{\varrho'/_2\, SV^2}\right)$ is found by some frictional resistance formula in combination with REYNOLDS' number or may be calculated from FROUDE's formula: – $Rf = f \times S \times V^{1,825}$ or $\frac{Rf}{\varrho'/_2 SV^2} =$

$= \frac{f}{\varrho'/_2 V^{0,175}}$

The FROUDE S.F.C. method is principally in agreement with that mentioned above.

The corresponding values of $\frac{f}{\varrho/_2\, V^{0,175}}$ are here designated $O_m.\, (L)^{-0,175}$ and $O_s\, (L)^{-0,175}$.

The above-mentioned calculation was made without a correction for temperature.

If the original test data viz. modelresistance = 15,18 lbs at 80° F. is taken as the base of calculation, a correction for temperature is required to eliminate the difference in frictional resistance.

Calculation of E.H.P. according to the „WILLIAM FROUDE National Tank" method gives 12700 (see BAKER's above mentioned discussion) and after the continental method 12390. Thus a difference of 2,5%.

It is obvious that this difference is caused by the correction for temperature.

In the first place the standard temperature, at which the above-mentioned frictional constants are considered to be correct is different for various tanks.

At Teddington it is 55° F., continental tanks 15° C. or 59° F. At Paris no correction is made as the water is kept at a constant temperature of about 13° C.

In the second place the correction itself is different.

Though the temperature of water for the example selected is rather high it shows clearly what differences may occur. Thus several corrections will be considered.

CORRECTION FOR TEMPERATURE

1. The continental method.

The standard temperature is 15° C. It is assumed that the frictional resistance increases or decreases by 0,43% per 1° C. decrease or increase in temperature.

The general formula is $\lambda' = \lambda\ (1 \pm 0{,}0043t)$, thus altering the frictional constant.

2. The W.F.N.T. method.

BAKER has published data on temperature correction for a 16ft model. His work shows a mean change in frictional resistance of 2,92% per 10° F. change from 55° F. standard.

For comparative purposes it is assumed here that for 10° F. difference in temperature from 55° F. standard, there is 3% difference in frictional resistance.

The correction is given as a (C) value and equals 0,00372 × × (S) × N°, for a difference of N° F. from the standard temperature of 55° F.

As 0,00372 is a constant, it is assumed that $\frac{Rf}{SV^2}$ is constant for every model at any speed.

This of course is not true.

It would be more correct if O_m were multiplied by a factor $(1 \pm 0{,}003N)$. This correction equals 0,54% per 1° C.

In his paper „Ship resistance similarity" (T.I.N.A. 1927) TELFER obtains almost the same result viz. 2,93% per 10° F change from 55° F. standard.

3. GEBER's correction.

Standard temperature is 10° C. or 50° F.

The correction for temperature is given in multiplying by the factor:

$$\left(\frac{\text{kinematic viscosity of water at temp. of test}}{\text{kinematic viscosity at 50° F.}}\right)^{1/8}$$

In Schiffbau No. 2, 1933 GEBERS gives the following table calculated from this factor.

Temp. of water in ° C.	0–5	5–10	10–15	15–20	20–25	25–30
Mean difference in frictional resistance in % for each degree C.	0,43	0,39	0,35	0,32	0,29	0,27

It is obvious that these corrections fail to show satisfactory agreement.

Fortunately the latest researches on skin friction of smooth plates permit an opportunity of studying the influence of temperature on frictional resistance.

For this purpose fig. 6 from EISNER's paper: ,,Frictional resistance'' (Hydromechanische Probleme des Schiffsantriebes, page 29) has been copied. (See plate VI.)

In this diagram the specific resistance is definitely given on base of REYNOLDS' number for a certain condition of flow in the boundary layer.

If it is assumed that the boundary layer is turbulent, SCHLICHTING's formula:

specific resistance $= 0{,}455 (\log R)^{-2{,}58}$ may be applied.

For the customary speed-product range of each tank the change in specific resistance has been calculated.

This has been done for $V.L = 12$ M^2/sec. and $V.L = 6$ M^2/sec. with the coefficients of kinematic viscosity at 5°, 10°, 15°, 20° and 25° C.

The results are represented as mean difference in frictional resistance in percentages for 1° C. to a base of 15° C. (See plate VII.)

It appears that the continental correction forms a good mean.

If the boundary layer is not turbulent over the whole length, the specific resistance forms a ,,transition'' curve between the turbulent and laminar region.

It is evident from the diagram that for this region the correction is less and may even change, so that resistance decreases with decrease of temperature.

As indicated in the diagram the experiments of GEBERS lay in the ,,transition'' region. Thus it is quite natural that according to his researches the correction should be less.

The investigations on models by WEITBRECHT (Jahrbuch der Schiffbautechnischen Gesellschaft 1933, page 329) show, that by changing REYNOLDS' number the specific resistance is altered

SPECIFIC RESISTANCE OF SMOOTH PLANES (EISNER)

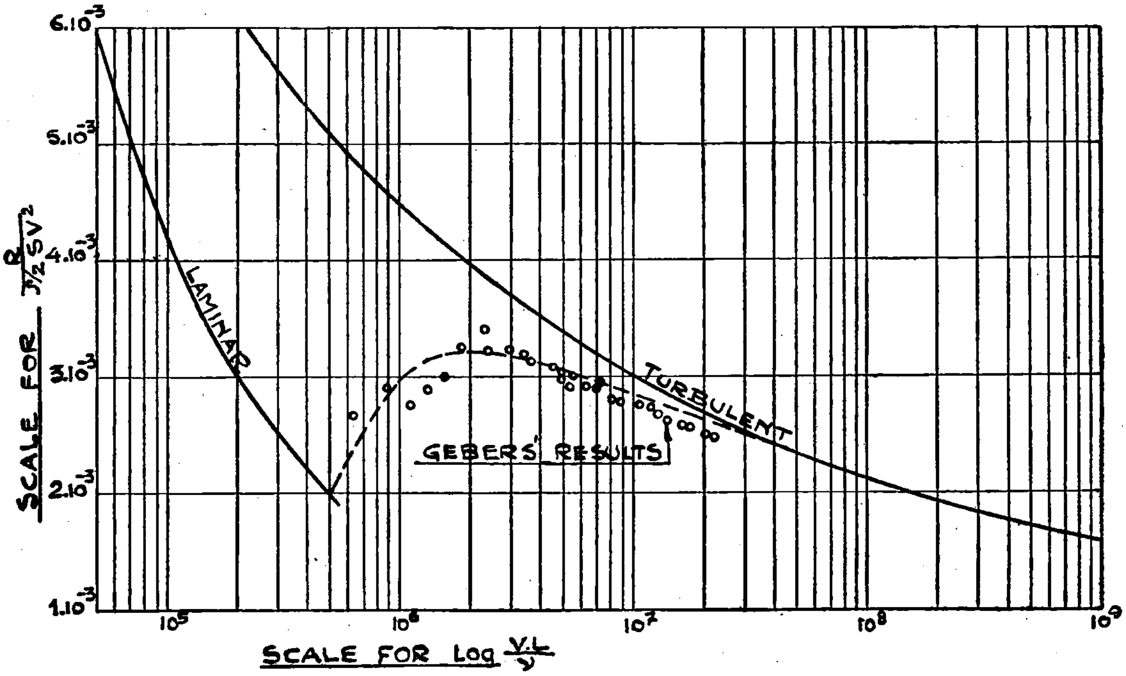
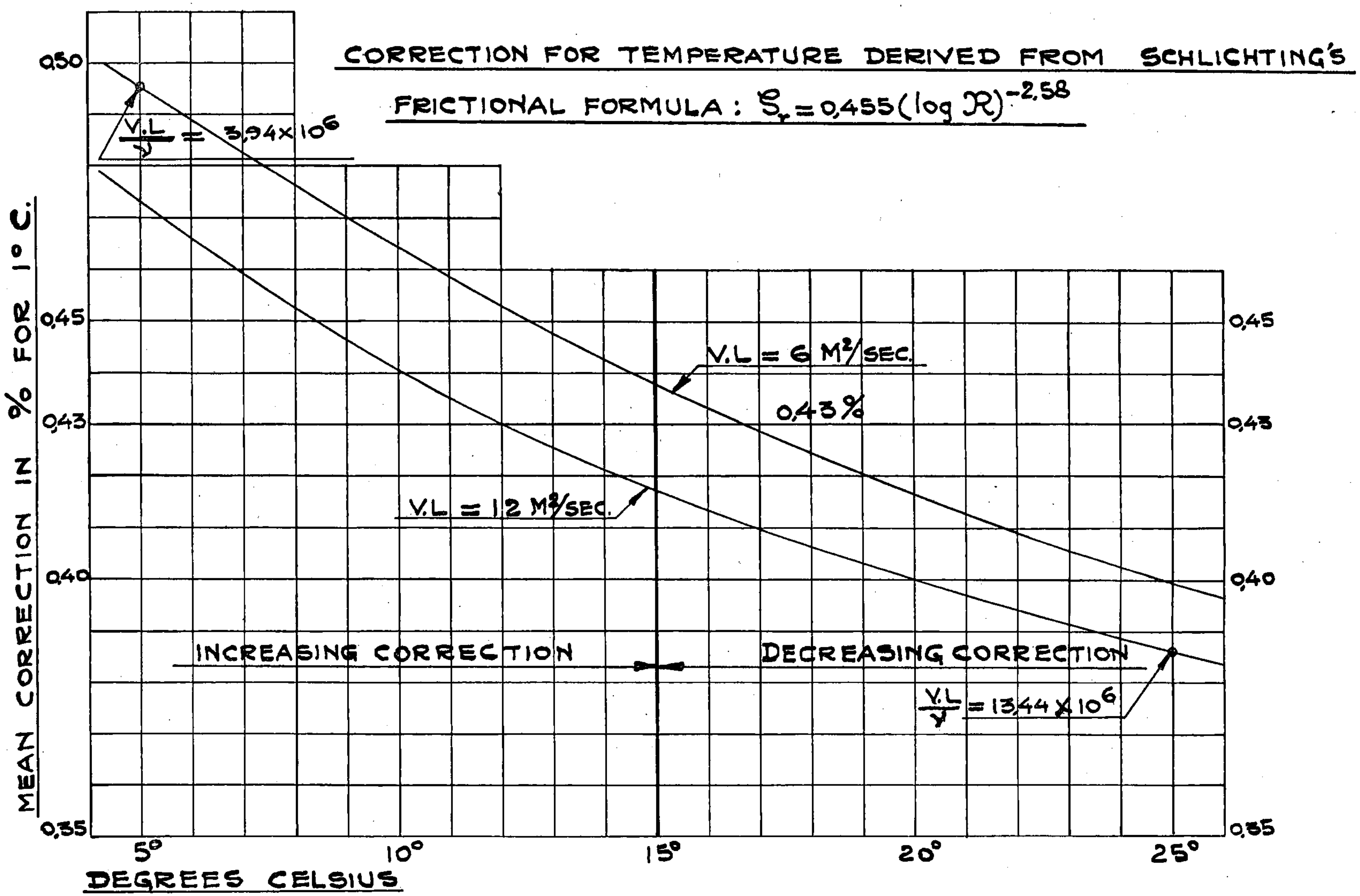


Plate VI



CORRECTION FOR TEMPERATURE DERIVED FROM SCHLICHTING'S
FRICTIONAL FORMULA : $\mathcal{S}_r = 0{,}455(\log \mathcal{R})^{-2{,}58}$
$\frac{V.L}{\nu} = 3{,}94 \times 10^6$
V.L = 6 M²/SEC.
0,43 %
V.L = 12 M²/SEC.
INCREASING CORRECTION
DECREASING CORRECTION
$\frac{V.L}{\nu} = 13{,}44 \times 10^6$
MEAN CORRECTION IN % FOR 1° C.
0,50
0,45
0,43
0,40
0,35
5°
10°
15°
20°
25°
DEGREES CELSIUS

Plate VII

in accordance with resistance of smooth plates with turbulent boundary layer, if REYNOLDS' numbers and speed-length ratio's are high.

At lower REYNOLDS' numbers and lower speed-length ratio's the correction for temperature will be less.

Other differences in the calculation may occur in the determination of S and L to find the frictional constant.

For this it is referred to what has been said above.

At Hamburg almost all models are tested with the rudder and the length for finding the frictional constant is taken as the maximum length under water inclusive of the rudder.

In general it is immaterial which length is used as the effect on the result of the calculation is very small.

CONCLUSION

In connection with the above mentioned it may be stated, that there is good agreement among the different methods of calculating ship resistance according to the formula of R. E. FROUDE.

Rather excessive differences exist in the standardtemperature and the correction for temperature.

Unfortunately the temperature of water is not mentioned in the fundamental paper of R. E. FROUDE (T.I.N.A. 1888).

As uniformity in these respects is desirable it is proposed:

1. to fix the standard temperature at 15° C. or 59° C., as this forms a good mean temperature of basinwater for several experimental tanks;

2. to adopt a temperature correction of 0,43% of the frictional resistance per 1° C. or 0,24% per 1° F.

CHAPTER IV

SYMBOLS USED

No attempt will be made to lay down a uniform system of symbols as these are partly so strongly related to the different languages, partly so familiarised by long use that alteration would be very difficult.

It would already be a considerable advance if a uniform system of symbols were used in several countries with the same language;

so that the symbols in a publication in a particular language were invariable, thus avoiding unnecessary confusion.

For languages other than e.g.: English, German, French and Italian one of these four sets of symbols could be chosen.

Nevertheless there are some up to date coefficients which are susceptible to uniformity as e.g.: specific resistance, speed-length ratio, REYNOLDS' number and the coefficients relating to screw propellers.

Apart from symbols there are also differences regarding terminology as e.g. in the manner of presenting the wake value.

In the following table the symbols given in the foregoing, and which are principally given by the W.F.N.T. system are compared with the German system adopted by the Hamburg and Berlin tank.

	W.F.N.T.		M.K.G.S. Units Berlin and Hamburg	
	Ship	model	Ship	model
General				
specific gravity			γ	
acceleration of gravity	g		g	
density	ϱ		ϱ	
coefficient of kinematic viscosity	ν		ν	
Ship				
Length..................	L (ft)	l (ft)	L	L'
Breadth	B (ft)	b (ins)	B	B'
Draught	D (ft)	d (ins)	T	T'
Midshipsection area	A (sq.ft)	a (sq.ft)	$\otimes$	$\otimes'$
Volume of Displacement ..	Vo*(cu.ft)		V	V'
Displacement	$\triangle$(Tons)	δ (lbs)	D	D'
Prismatic coefficient	p		φ	
Wetted surface...........	S (sq.ft)	s (sq.ft)	O	O'
Total resistance	R (Tons)	r (lbs)	W_o	W_o'
Frictional resistance	R_f*(Tons)	r * (lbs)	W_r	W_r'
Residuary resistance	R_r*(Tons)	r_r* (lbs)	W_f	W_f'
Speed of ship	V (knots)	v $\frac{\text{ft/min}}{100}$	v	v'
Speed-length ratio $\frac{V}{\sqrt{gL}}$...			F	
REYNOLDS' number $\frac{V.L}{\nu}$ ..			R	
Specific resistance $\frac{R}{\varrho/_2\, SV^2}$.			ζ_w	

	W.F.N.T.		M.K.G.S. Units Berlin and Hamburg	
	Ship	model	Ship	model
Propeller				
Diameter	D (ft)	D_M (ft)	D	D'
Face pitch	P (ft)	P_M (ft)	H	H'
Number of blades			z	z
Developed area	B.A.R.*		F_a	F_a'
Disc area			F	F'
Mean width ratio	M.W.R.*			
Maximum blade width ratio	B.W.R.*			
Width of blade section			l	l'
Max. thickness of blade section			s	s'
Blade thickness ratio	B.T.R.*			
Camber ratio			$s/l = \delta$	
Rake	$\theta°$*			
Hub diameter	d*			
Thrust	T (Tons)	f (lbs)	S	S'
Torque (open)	Q (lbs-ft)	g (lbs-ft)	M	M'
Speed of advance	V_1(knots)	v_1 (ft/sec)	v_p	v_p'
Revolutions per min.	N			
Revolutions per second		n	n	n'
Thrust coefficient $\frac{T}{\varrho n^2 D^4}$			K_s	
Torque coefficient $\frac{Q}{\varrho n^2 D^5}$			K_m	
$\frac{V}{n.D}$	C_D		Λ	
slip ratio	s*		σ	
Propeller efficiency (open)	η		η_p	
Hydrostatic pressure	h*			

The symbols marked with * are not given in the W.F.N.T. table.

Instead of B.T.R. BAKER uses B.T.F. (thickness measured square to rake line).

B.A.R. = Blade area ratio $= \frac{\text{total developed area}}{\text{disc area}}$ = D.A.R. = Disc area ratio.

Wageningen, June 15th, 1933

W. VAN BEELEN

L. TROOST

POSITION DE LA DÉLÉGATION FRANÇAISE À LA CONFÉRENCE DE WAGENINGEN

(ASSOCIATION TECHNIQUE MARITIME ET AÉRONAUTIQUE-BASSIN DE PARIS)

Dans les documents préparatoires du congrès se trouvent une notice de MM. TROOST et VAN BEELEN, résumant les réponses faites au questionnaire et un procès verbal du Fachausschuss der Schiffbautechnischen Gesellschaft.

Je m'occuperai principalement des sujets examinés dans ces deux documents et introduirai une seule question nouvelle qui me semble avoir été à tort laissée de côté.

Suivant la notice de MM. TROOST et VAN BEELEN, dans l'ordre des paragraphes, je fais les remarques suivantes:

Page 1 (99). Je souscris entièrement à la proposition de recommander la publication des résultats directs des essais (c'est-à-dire les nombres obtenus sur le modèle) en les présentant sous une forme unique. Ces résultats doivent être exprimés en quantités de dimensions nulles *et en unités cohérentes.* Si l'actuelle réunion arrive à un accord sur ce seul point, un résultat utile pour tous aura été obtenu, et ce ne sera pas en vain que les organisateurs de la réunion auront dépensé leur peine.

Page 2 (100). Le remplissage des pores est-il recommandé à la suite d'expériences ou à la suite de considérations théoriques? Des expériences faites il y a quelques années nous ont montré que les pores de la dimension courante sur les modèles n' avaient pas d'importance. Des expériences analogues ont-elles été faites ailleurs?

A cette question se rattache celle de savoir si les modèles doivent être conservés dans l'eau avant l'essai.

Et puisque nous en sommes à des détails de l'exécution des essais, je poserai encore la question: est-il préférable de peser la modèle pour le mettre dans ses lignes d'eau, ou de le lester d'après des aiguilles indiquant son enfoncement?

A Paris, nous n'attachons pas d'importance à la présence des pores, nous ne conservons pas le modèle dans l'eau avant l'essai, enfin, nous pesons le modèle.

Malgré cette différence de technique, nous avons trouvé dans des comparaisons faites avec les bassins étrangers, que notre façon de procéder conduisait aux mêmes résultats.

Je pose la question suivante: en suivant d'autres techniques, a-t-on constaté expérimentalement que des variations de la technique de l'essai, sur ces trois points, entraînaient des variations dans les résultats et quelle est la grandeur de ces variations.

S'il y a des variations il faudra préciser les conditions dans lesquelles doivent être faits les essais.

Page 3 (101). Une carène doit être définie par ses trois dimensions principales L.l.p et par un plan ou relevé d'ordonnées relatif au modèle.

La proposition de diviser les $^1/_2$ largeurs par $\frac{l}{2}$ est appuyée. Jusqu'ici, nous n'avons utilisé ce procédé que pour des études scientifiques ou des exposés didactiques. Nous estimons utile de l'utiliser pour la pratique.

Les coefficients les plus intéressants sont:

$$\varphi = \frac{W}{B^2L}, \quad \frac{L}{\sqrt[3]{W}}, \quad \frac{\Sigma}{W^{2/3}}$$

on doit donc donner les valeurs de ces coefficients, ou les valeurs de W, B^2, Σ permettant de les retrouver.

Il sérait très utile d'avoir de plus les deux coefficients

$$\sigma = \frac{I}{^1/_{12}Ll^3} \quad \Big\| \quad \eta = \frac{i}{p}$$

donnant, l'un le rapport de l'inertie I de flottaison à celle du rectangle circonscrit, l'autre, le rapport de l'immersion du centre de carène à la profondeur. Ces coefficients permettent de retrouver le métacentre en hauteur pour des carènes similaires. Ils ne sont pas nécessaires pour la pratique des Bassins d'Essais, mais sont indispensables pour l'étude des variantes d'un projet.

Enfin, il est nécessaire de fixer une règle uniforme pour la détermination de L et pour la façon de mesurer Σ. La méthode proposée par GUTSCHE pour Σ est celle que nous employons. La question se pose de savoir si dans l'étude d'une même carène à diverses assiettes et déplacement on doit changer L et Σ comme cela est nécessaire théoriquement ou si pratiquement ces corrections sont sans intérêt.

Pages 4 à 6 (102 à 104). Je préfèrerais la profondeur divisée en 5 parties plutôt qu'en 6.

Page 7 (*105*). Je propose un axe des abscisses gradué en $\frac{V}{\sqrt{gL}}$ et un axe des ordonnées gradué en $\frac{R}{\varrho\, gW}$ avec indication de la courbe de résistance de frottement du modèle. Le graphique donne ainsi à la fois les résistances directes et les résistances totales mesurées sur le modèle, ce qui permet plus tard de faire des corrections si les formules de frottement changent.

La forme $\frac{R}{\varrho/_2\, AV^2}$ présente le même défaut que $\frac{V}{\sqrt{\varphi L}}$: adopter l'une de ces formes est supposer à priori une influence physique de A ou de φ sur le phénomène que l'on étudie.

Page 15 (*113*). Sur cette planche on voit des courbes de

$$\frac{Q}{\varrho\, n^2D^5} \text{ et } \frac{T}{\varrho\, n^2D^4}$$

en function de $\frac{V}{nD}$. Les courbes présentent des cassures systématiques.

Nous n'avons jamais obtenu de cassures de ce genre. Il serait intéressant de savoir si l'auteur donne le sens d'un phénomène réel à ces cassures.

Page 17 (*115*). Contre l'emploi de $\frac{nD}{V}$ comme abscisse, l'auteur indique que les points représentatifs du fonctionnement au point fixe, seraient à l'infini ($V = 0$), mais l'emploi préconisé de $\frac{V}{nD}$ envoie à l'infini le point correspondant à l'hélice remorquée et empêchée de tourner ($n = 0$). Pour éviter l'un et l'autre inconvénient nous employons un axe des abscisses gradué en $\omega = \text{arc tg } \frac{nD}{V}$, ce qui fait que pour ω variant de 0 à 360° on a sur un même graphique de largeur finie, non seulement les points dont il est question ci-dessus mais toutes les circonstances de fonctionnement d'une hélice, en marche avant comme en marche arrière, de l'hélice agissant soit comme propulseur soit comme turbine.

Ce genre de graphique qui est notre type normal a comme le graphique de la planche V l'inconvénient de nécessiter l'emploi de trois courbes séparées, ce qui n'est pas commode lorsqu'on veut comparer des essais faits sur des modèles, à diverses échelles ou lorsque l'on veut calculer des sillages ou des succions, après mesure simultanée de couples et de poussées. Dans ce cas, nous employons le graphique à abscisses $\frac{P}{\varrho\, n^2 D^4}$ et ordonnées $\frac{C}{\varrho\, n^2 D^5}$ Ce graphique a en particulier l'avantage d'indiquer immédiatement si les couples et poussées mesurés sur navire peuvent s'expliquer ou non par l'emploi d'un sillage de translation seulement, il a l'avantage de permettre le rapprochement immédiat des essais effectués à degré REYNOLDS différent.

Page 23 (*121*). La correction de frottement des modèles pour tenir compte de la température ne paraît pas bien déterminée:

Allemagne 0,430 – Baker 0,526 – Gebers 0,335, par degré centigrade.

Sur une planche de 4 m. de long, nous avons trouvé une décroissance de 0,25% par degré de température à la surface et 0,36% par degré de température au fond. Il est peut être prématuré de proposer un accord. La ligne de conduite que nous avons suivie est d'éviter les variations de température.

J'arrive maintenant à la Stellungnahme des Fachausschusses der S. T. G.

Sur le point a: doit-on faire les essais d'autopropulsion avec compensation de frottement? J'indique qu'à Paris nous faisons les essais d'autopropulsion par une méthode qui permet le calcul soit avec, soit sans compensation de frottement, c'est-à-dire que nous pouvons utiliser soit le point d'autopropulsion du modèle, soit le point d'autopropulsion du navire. Nous avons conclu que l'on obtenait d'aussi bons résultats par l'une ou l'autre méthode. Cependant, nous employons normalement la méthode proposée par le Fachausschuss qui nous paraît plus correcte au point de vue théorique. Nous ne pensons pas cependant que cette méthode soit absolument correcte, ni qu'aucune méthode soit absolument correcte par suite de la non similitude cinématique de l'écoulement autour du modèle et autour du navire réel.

Sur le point b: moyens artificiels de provoquer la turbulence. Nous avons étudié, il y a quelques années, l'emploi d'une arête en saillie sur le modèle. Nous avons trouvé que les résultats, aux très petites vitesses, étaient plus réguliers avec l'arête, c'est-á-dire qu'il y avait une moindre dispersion des points expérimentaux entre-eux, mais que, ni le sillage ni la succion n'étaient influencés. Nous n'avons pas employé ce procédé par la suite.

Sur le point c., frottement du réel, nous ne pouvons apporter aucun renseignement et pensons que la question ne pourra faire un pas décisif qu'après un essai de remorquage de navire réel à la mer, comme cela avait été envisagé à l'époque du Skin friction Committee.

Sur d. nous avons déjà indiqué notre avis ci-dessus.

Sur e. nous pensons qu'une discussion orale est seule possible.

J'arrive pour terminer, au point important qui me semble devoir être ajouté à ceux déjà proposés pour discussion.

Indépendamment des questions de notation et de représentation des résultats d'essais sur modèle, je pense qu'il y aurait intérêt à examiner la question de la comparaison des diverses formes de carènes et à mettre de l'uniformité dans les méthodes de discussion à employer chaque fois qu'une nouvelle forme de carène est lancée dans la littérature technique.

Pour juger d'une nouvelle forme de carène, il est nécessaire de tenir compte de deux genres de considérations.

1° Une comparaison entre deux formes n'a de valeur pour un constructeur que si les formes comparées satisfont toutes deux à diverses conditions imposées par la stabilité et par l'emménagement intérieur. Dans un projet de navire, en effet, la forme de la carène n'est pas déterminée uniquement par la condition de moindre résistance à la marche ou de meilleur rendement propulsif.

2° Pour le Congrès actuel, la seule condition de moindre résistance à la marche est le point de vue principal. Il est alors utile d'examiner si un point de repère ferme peut être trouvé en ce qui concerne l'appréciation numérique d'une forme nouvelle proposée. A mon avis, ce point de repère existe dans l'album publié par TAYLOR, et on peut parler sans ambiguité de la carène TAYLOR satisfaisant à la donnée de L., W. et $\frac{1}{p}$ et de plus ayant la résistance de vagues minima pour une valeur de $\frac{V}{\sqrt{gL}}$ donnée. Il est donc possible et je crois utile, chaque fois que l'on parle d'une forme de carène nouvelle, d'indiquer ce que donnerait la carène TAYLOR de même longueur, de même volume et de même stabilité. On aurait ainsi un coefficient T_W (en général voisin de l'unité) variable avec le degré $\frac{V}{\sqrt{gL}}$ et qu'il serait utile de porter en courbe en même temps que les résultats de la carène nouvelle.

Souvent, on rapproche sur un même dessin la courbe de résistance de la carène proposée et la courbe d'*une* carène TAYLOR. Ce n'est pas ce que je propose de faire, car le coefficient T_W est le rapport de la résistance de la carène proposée à la carène de TAYLOR optima pour chaque $\frac{V}{\sqrt{gL}}$ c'est-à-dire à une carène de repère qui n'est pas la même pour toutes les vitesses.

L'adjonction de la courbe des T_W sur la planche des résultats d'essais ne nécessite pas un grand travail. Elle a l'avantage de montrer immédiatement la qualité de la carène proposée par rapport à l'étalon, et de montrer quels sont les $\frac{V}{\sqrt{gL}}$ pour lesquels la nouvelle forme est avantageuse.

Nous avons dit que T_W est voisin de 1.

Des comparaisons faites pendant plusieurs années nous ont montré qu'il était rare de trouver des T_W inférieurs à 0,95.

De l'autre coté de l'unité, un T_W supérieur à 1 n'est pas forcément l'indice d'une carène mal dessinée; la valeur de T_W indique en effet, pour des carènes bien dessinées, le sacrifice qu'il a fallu consentir sur la puissance propulsive du fait que d'autres considérations que celle de faible résistance à la marche, interviennent dans le choix de la forme de carène d'un navire réel.

Ma proposition comprend donc 2 parties:

1° Ajouter aux planches d'essai une courbe de comparaison avec une famille-étalon, la même pour toutes les comparaisons.

2° Choisir comme étalon celui qui, à égalité de stabilité donne pour chaque $\frac{V}{\sqrt{gL}}$ le minimum de résistance de vagues d'après TAYLOR.

Cette proposition suppose que la résistance d'une carène n'est examinée que pour une seule vitesse. Elle n'est donc qu'une étape préparatoire pour la comparaison des carènes qui, ou bien doivent être utilisées à deux vitesses, ou bien doivent être utilisées à deux deplacements.

Pour la délégation Française,

BARRILLON.

MISCELLANEOUS NOTES

BY E. V. TELFER, D. SC., PH. D.

(Written Contribution)

PRESENTATION

Data for world consumption should first clearly demonstrate its own probable accuracy by simple inspection. Since frictional resistance preponderates the presentation of research data must be in terms of $\frac{R}{\varrho/_2 \, Av^2}$ to a base of Reynold's number. The turbulent frictional resistance line given by the Schlichting (Göttingen) formula should be shown on the same diagram, and also, in the interregnum, the plank resistance line hitherto used by the particular tank. No correction of any kind to be made in this presentation a temperature correction is entirely avoided by the use of Reynold's number.

For convenience the speed base may be calibrated for Froude number. The original base *must* be Reynold's number to verify accuracy of data.

Accompanying these data a table of actual resistance speeds and water temperature must be given.

MODEL SELF-PROPULSION

I agree in principle with Prof. Horn that the model screw should have the same thrust loading as the ship. It must also have, however, the same feed speed as the ship. Thus, if this speed, owing to wake scale effect, is one knot higher in the ship than in the model, then for ship self-propulsion at 10 knots, the model must be towed at 11 knots with the model thrust unloaded by the 11 knots frictional correction plus the difference in model resistance between 11 knots and 10 knots. Even if this technique were correctly mastered there remains the propeller efficiency scale effect, the thrust deduction scale effect and the uncertainty of the frictional correction itself.

The Teddington method of using model self-propulsion without any correction whilst scientifically crude, yet has the distinct advantage of simplicity. The method assumes that the loss in *model* propeller efficiency due to greater slip is balanced by the loss in *ship* hull efficiency due to decreased wake and increased thrust deduction, this hull efficiency loss being itself reduced by the gain in ship propeller efficiency.

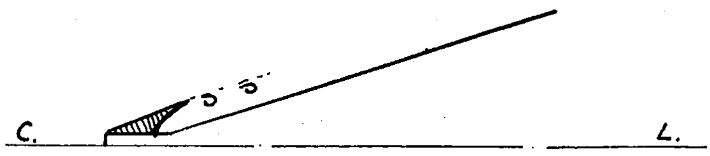
It may be argued that Teddington have no right to make this assumption. The assumption moreover cannot be equally correct for centre and wing screws but as a matter of experiment routine it is justifiable, since the correction to the model results should, in our present state of knowledge, not be made on the model itself but by separate scale effect calculations. The technique of these requires development and geometrically similar selfpropelled models will be the basis of this technique. The variation in model thrust and model T_c instead of model resistance thus requires study to a base of Reynold's number. The thrust extrapolator will probably differ from the resistance extrapolator but these will of course be functionally related.

To use the thrust extrapolator to predict ship torque, the propeller polar recommended by Prof. Barrillon is of exceptional convenience. The peculiarity of this polar is that uni-directional scale effects have quite subordinate influence on the polar, thus if T_c (i.e. $\frac{T}{\varrho n^2 D^4}$) is the thrust extrapolator then the corresponding Q_c can be obtained from the polar. The whole technique of the self-propelled model requires new development. As a fundamental principle however the model must be propelled at its own thrust loading and no attempt made to correct the model as is current practice everywhere except Teddington.

MODEL TURBULENCE

All models *m st* be entirely turbulent. Each tank should attack this subject by every manner of means on some agreed standard model. This model should be of the size generally used by the particular tank and will thus vary in size between the various tanks.

One suggestion which may be worthy of trial is to attempt to generate a turbulent frictional belt right from the bow. The device should therefore have the correct initial rate of growth of the turbulent belt and should be shaped to produce the most intense eddying in its wake. This device can be called the turbulent stem and experiments should be made to test its efficiency. Its expected shape is shown in enlarged sketch.



Proposed turbulent stem

MATHEMATICAL FORMS

The method of form delineation suggested by Pavlenko and developed by Shigemitsu for model experiments should be more generally adopted in preference to mathematical forms of the Taylor or Weinblum type.

SYMBOLS

All special symbols which are not generally typographically available are inadmissible for international adoption. For this reason, the British practice of encircling symbols such as C and P etc. are automatically ruled out. For the same reason Greek symbols should be used as sparingly as possible. The symbol ζ recommended by Hamburg and Berlin for force coefficients and efficiency factors will lead to many typographical errors. Few practioners even know this symbol. The definition of a coefficient should be indicated by a suffix c to the Roman symbol of the particular fence i.e. resistance coefficient $\frac{R}{\varrho/_2\,Av^2}$ is Rc etc.

Any symbol which cannot be accurately and simply reproduced by the shipbuilding technical press of the world is unsuitable for international adoption.

PRESENTATION OF HULL DESIGN DATA

The presentation of resistance data intended to demonstrate the internal behaviour of a systematically varied series of models is a different problem from that of the presentation of the results of a single model. The basis results of such a series must still be given in the research presentations, but in addition *to facilitate the practioners assimilation of the lessons to be learned for the ship,* the resistance needs to be expressed as a function of the displacement.

The purest non-dimensional presentation.is pounds per ton to a base of speed-length ratio. The one disadvantage of this system is that it does not clearly reveal the incidence of wave making phenomena. For this reason we are forced to incorporate a speed-squared term in the resistance coefficient and thus must commence with a resistance coefficient of the form $\frac{R\,(L)}{\Delta\,v^2}$, or, using effective horse power instead of resistance, the coefficient becomes $\frac{(EHP)\,(L)}{\Delta\,v^3}$. Now if the non-dimensional speed base used is $\frac{v}{\sqrt{L}}$ then the power coefficient must be $\frac{EHP\,.\,L}{\Delta\,v^3}$, since speeds graded in a length base must also have resistance coefficients similarly graded. If, however, the speed is graded on a displacement base (i.e. $\frac{v}{\Delta^{1/6}}$) then the resistance coefficient must be

$$\frac{EHP}{v\,.\,\Delta}\left(\frac{\Delta^{1/6}}{v}\right)^2 = \frac{EHP}{\Delta^{2/3}\,.\,v^3}.$$

It is entirely erroneous to use a resistance-displacement coefficient in conjunction with a speed-length ratio and vice versa. I demonstrated this at length in discussing A. L. Ayre's 1927 N.E.C. paper. The error involved is that the least $\frac{EHP}{\Delta^{2/3}\,.\,v^3}$ at given $\frac{v}{\sqrt{L}}$, does not necessary give the least pounds per ton or the most economical model, since the influence of $L/\Delta^{1/3}$ may entirely change the comparison.

To secure international agreement, as it appears fairly certain that speed-length ratio is in much commoner use than speed-displacement ratio, it follows that the power coefficient to be used is $\frac{EHP\,.\,L}{\Delta\,.\,v^3}$. This coefficient avoids fractional powers and thus has the advantage of simple *numerical* calculation. For practical convenience the coefficient should be $\frac{10 \times EHP\,.\,L}{\Delta\,v^3}$. It should be noted that Froude in using C almost invariably also used K. This combination is correct. The C—P combination is incorrect *presentationnally* and should not be prolonged. As a matter of interest however it is useful to enquire as to the

correct power coefficient to be used in conjunction with $\textcircled{P}$.
Since $\textcircled{P} = k \frac{v}{\sqrt{L \varphi}}$ it follows that the power coefficient must be $\frac{EHP}{v \, . \, \Delta}\left(\frac{\sqrt{L \varphi}}{v}\right)^2 = \frac{EHP \, . \, L \, . \, \varphi}{\Delta \, v^3}$ but $\Delta = M.L.\varphi$ where M is the midship area. Power coefficient $= \frac{EHP.L.\varphi}{M.L.\varphi. \, v^3} = \frac{EHP}{Mv^3}$. This finding is of extreme interest, as it shows that the newest speed-length ratio *must* be associated with the oldest power coefficient since $\frac{EHP}{Mv^3}$ is merely the inverse of the French midship section rule $\frac{M \, . \, v^3}{EHP}$.

The alternative use of $\frac{EHP}{Mv^3}$ with $\frac{v}{\sqrt{L \varphi}}$ is not without considerable possibility and should be accepted as a permissible alternative.

Summarizing therefore acceptable presentations for the design selection problem of model resistance data, we have

1) $\frac{10 \, EHP \, . \, L}{\Delta \, v^3}$ and $\frac{v}{\sqrt{L}}$

2) $\frac{EHP}{M \, . \, v^3}$ and $\frac{v}{\sqrt{L \, . \, \varphi}}$

3) $\textcircled{C} = \frac{427{,}1 \, EHP.}{\Delta^{2/3} \, . \, v^3}$ and $\textcircled{k} = .5834 \, \frac{v}{\Delta^{1/6}}$

DECISIONS.

In connection with the paper ,,Remarks and proposals in Tankwork'' from Ned. Scheepsbouwkundig Proefstation, Wageningen, the following decisions were taken:

1. The decisions regard the method of execution and publication for all scientific model experiment work.

2. The decisions are in force till the next meeting in 1934.

3. *On the dimensions and finish of models.*
a. The dimensions of the model should be sufficiently large to ensure that the turbulent boundary layer is formed over the major portion of the length of hull, so that the fitting of a trip wire or the use of a local roughening on certain portions of the entrance can be dispensed with. In this connection the ,,Committee of Four''[1]) will try to fix minimum Reynold's numbers for different cases.
b. All paraffine models should have any pores filled so as to produce a reasonably smooth surface all over in all cases.

4. *On the determination of length and wetted surface.*
a. For vessels with cruiser stern the length on waterline should be used. For vessels with raised stern the length is to be taken to the afterside of the rudderpost.
b. All Delegates agree in adopting the mean girth multiplied by the length for the wetted surface.

5. *Physical constants.*
The ,,Committee of Four'' accepts the table of Lyle and Hosking for the kinematic viscosity of fresh water.

6. *Froude's method of calculation.*
The Delegates agree in adopting Froude's method of calculation using R. E. Froude's frictional formula. The exact frictional coefficients will be fixed by the ,,Committee of Four'' by replotting the original coefficients.

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[1]) G. S. Baker, Teddington. E. G. Barrillon, Paris. G. Kempf, Hamburg. L. Troost, Wageningen.

The „Committee of Four" adopts the proposals made on page 124, viz.: that all *model* results should be corrected to a standard temperature of 15° C = 59° F by a correction of 0,43% of the frictional resistance per 1° C or 0,24% per 1° F. (No correction will be made for varying temperature of the sea on trial.)

7. *On the representation of ship model tests.*

a. The Delegates agree in the representation of model particulars in the manner presented in plate I and II of the Wageningen paper.

b. In general the Delegates agree in giving the resistance experiment data for an isolated model in one or more of the „constant" forms as proposed on page 107 (plate III), using as a base one or the other of the „constant" forms also there given and stating the length of model and temperature of water.

It is recommended to mark the resistance curve in dotted line, when Reynold's numbers are lower than the minimum values fixed by the „Committee of Four".

8. *On the publication of screw propeller tests.*

The Delegates accept the complete chapter II as the method of publication for all scientific work.

Instead of the developed area the Delegates agree in using the expanded area, taken outside of the bossline, as shown on the central part of plate IV.

9. *Symbols and terminology.*

It is left to the „Committee of Four" to settle the terminology for defining certain coefficients and symbols.

10. If in future publications are made it is sufficient to state that they are in agreement with the principles laid down in The Hague 1933.

JULY 14th, 1933.