Subjects 2 and 4
Skin Friction and Turbulence Stimulation

Chairman: Prof. L. Troost.
Reporter: Dr. F. H. Todd.
Secretary: Dr. G. Hughes.

Prof. L. Troost (Introductory words).

There is a definite reason why I think that this meeting of to-day is a very important one. Personally, I have been in the tankery business —which is one of the main businesses of I. T. T. C.— from the beginning, for 24 years, and until now we have been struggling with the spectre of not having an international language which is understood and spoken by all of us and which would connect the scientific work of people all over the world in such a way that the correlation of this work by everybody would be possible. Personally, I believe that the atmosphere has never been as favourable as it is to-day to reach an international agreement about the extrapolator for skin friction that we will use, in the future as an engineering line.

There is before us a proposal of the Skin Friction Committee and this proposal is looked upon by many of us as a workable proposal to get definite agreement in this matter I need not dwell upon how important this agreement would be.

I hope that these 24 years will be crowned to-day and if possible this morning, with a general opinion about the merits of the proposal of the Committee. I therefore call upon you for co-operation in this matter; if possible, arrange your own personal opinions, and adopt one of the proposals of the Committee. I think if we do not do this, the subject would lose the interest of many of us and we might be the laugh of the profession if we do not come so far.

It is therefore my policy this morning to start to consider every contribution we have before us. There are 20 and you all will be called upon to take the floor for a few minutes to summarize shortly your papers and to tell us how you feel and how the summary of your papers is related to this important issue. As soon as we have got a picture of the general opinion we will go further into the technicality of the tankery and then we will go into the scientific points raised in several of the important papers which we have to hand as well as into an informal discussion where everybody may express his personal opinion in this matter. However, please help me in backing the issue for the international adoption of a proposal, preferably one of the proposals of the Committee.

Dr. F. H. Todd (Introductory remarks).

There has been much activity in the field of skin friction resistance since the Conference last met in Scandinavia in 1954. The members of the Committee have endeavoured to follow these developments and to keep them in mind when preparing the report for this Conference. The report has been circulated to all members in advance of this meeting, and in Appendix I it gives a brief resume of some of the latest developments, although no claim is made that this is in any way a complete coverage of the subject.

At the 1954 Conference the Skin Friction Committee was given the task of carrying out the recommendation embodied in the following resolution, passed by the Conference at its last session (1):

> "3. The Conference is of the opinion that agreement on a friction formulation based on Reynolds number and adequate for practical ship design purposes is urgently required.
>
> To meet this requirement the Skin Friction Committee should consider all relevant data and make a definite recommendation for adoption not later than the next Conference."

This task has been the Committee's major preoccupation over the last two and a half years. The plan of action was determined at a meeting of the

(1) Proceedings of the 7th International Conference on Ship Hydrdynamics, p. 326. Publication No. 34 of the Swedish State Shipbuilding Experimental Tank, Goteborg, 1955.

Committee held in Copenhagen in 1954, immediately following the conclusion of the Conference, and a statement of the philosophy to be followed by the Committee was subsequently drawn up. A great deal of analysis work has been carried out on available geosim data using a variety of existing or proposed formulations. A three day meeting of the Committee was held in London in the Summer of 1956 to discuss the results of the work and to prepare the Report for submission to this Conference.

This Report has been circulated to all members. It contains the philosophy statement mentioned above, and also the opinions collected from many towing tanks all over the world in reply to questions from the Committee members. These opinions are set out in some detail in Appendix II of the Report, since they were an important consideration in the minds of the members in coming to the final conclusions.

Many difficulties have been encountered in the analysis, due to the extraneous effects of turbulence stimulation or the lack thereof, the possible presence of tank wall effects, accuracy of measurement and so on, which have in many cases obscured the main issue.

Under the circumstances, the Committee came to the conclusion that the Schoenherr or some closely similar line was the most acceptable for values of Reynolds number greater than 10^7, with a line of somewhat steeper slope below this point. The members were unable to come to an unanimous recommendation, however, on an exact specification of the desired line, and in the Report have proposed two solutions either of which they feel is acceptable as a single-line, engineering solution. They feel that it is a matter for the whole Conference to decide which, if either, of these lines should be adopted at this time.

Figure 1 shows the suggested lines in comparison with some (*) of the existing formulations.

Since the meeting of the Committee in London at which the recommendation was drawn up, the American Towing Tank Conference has held its 1956 meeting in September in Washington. The subject of skin friction resistance and the problem of model-ship extrapolation was then discussed at considerable length. The following resolution, proposed by the A. T. T. C. Skin Friction Committee under the Chairmanship of Dr. L. Landweber, was carried unanimously:

"The Committee has reviewed the present state of knowledge of the subject and in view of the great amount of work going on in this field feels that the time is inopportune to make any departure from the decision to use the 1947, A. T. T. C. line.

It is also recommended that in the future the latter designation be used rather than the name Schoenherr line."

As Chairman of the International Committee I would at this time like to pay tribute to the work of all the members in helping to carry out the, at times, very onerous analysis, to Dr. Telfer and Mr. Lap for analysing the geosim data in accordance with their own methods and to Mr. Hinterthan of the Taylor Model Basin for his work in collecting all the geosim data and analysing it by the Schoenherr formulation.

Committee Report

Introduction.

From time to time the Conference has considered the problem of predicting ship resistance from that of similar models, and in particular the method of extrapolation and the frictional coefficients to be used. Prior to 1948 the European towing tanks in general used the method and coefficients due to the Froudes, while in America, after a period covering a number of changes, agreement had finally been reached to use the Schoenherr formulation.

At the London Conference in 1948 the delegates agreed "that results may be published using either Froude or Schoenherr coefficients" (1). At the same time the Skin Friction Committee was appointed "to survey the problem of skin friction in general, and in particular to recommend what further research should be carried out to establish the minimum turbulent friction line for both model and ship use...".

No change in procedure was proposed at the Washington or Scandinavian Conferences in 1951 and 1954 (2, 3). At the latter meeting, however, there was general agreement that the time had come to take another look at the problem in view of the work which had been carried out in the intervening years. The 1954 Conference therefore made the following recommendations:

> "1. The Conference agrees that a proper correlation of model and ship resistance must take account of the effects of three-dimensional flow, and other causes of variable extrapolation. As much new data as possible should be obtained and old data re-examined in the light of recent proposals to determine three-dimensional or form effects.
>
> 2. In view of work recently completed and in progress in many related fields, the Conference considers that it is not desirable to attempt to reach a decision at this Conference as to a lasting skin friction formulation.

(*) Identical with Fig. 1 of Dr. Todd's formal discussion.

3. The Conference is of the opinion that agreement on a friction formulation based on Reynolds number and adequate for practical ship design purposes is urgently required.

To meet this requirement the Skin Friction Committee should consider all relevant data and make a definite recommendation for adoption not later than the next Conference.

4. The Conference is of the opinion that in the interval before the next Conference work should also proceed generally on the problems of roughness, turbulence stimulation and tank boundary effects on the lines indicated in the report of the Skin Friction Committee."

Work of the Committee since 1954.

At the conclusion of the Scandinavian Conference, the Committee held a meeting in Copenhagen in order to discuss the procedure necessary to carry out its mission as requested in the above recommendations. A program was agreed upon and a great deal of analysis has been carried out since that time. The results were discussed at a meeting of the Committee held in London in July and August, 1956. The members finally agreed upon certain recommendations which are set forth at the end of this report.

The members have also continued to survey the skin friction field throughout the period, and to keep the Chairman informed of developments in their respective geographical areas. A brief review of these is presented as Appendix I to this report.

The philosophy governing the Committee's work.

Following the end of the Scandinavian Conference, the Committee felt it necessary to set down in some detail a philosophy to be followed as a guide to the members in carrying out its mission.

In this respect, the Committee concluded that the discussion at the Scandinavian Conference in 1954 indicated:

(i) a stronger disposition than in 1948 and 1951 to abandon the Froude coefficients;

(ii) a reluctance to take only this step, as it would, in effect, entail the universal use of the Schoenherr line at a time when a better "engineering" formulation "adequate for practical ship design purposes" might perhaps be near at hand; and

(iii) a strong feeling that, although the next major advance in the subject should be to "take account of the effects of three-dimensional flow and other causes of variable extrapolation", agreement as to generally acceptable ways of doing this was unlikely to be reached for some years.

In view of (iii), a better "engineering" formulation, as referred to in (ii), has in effect been defined as a single-line function of Reynolds number, to be used in the same way as any other single-line formulation.

The Committee concluded that the opinions expressed by the delegates to the 1954 Conference pointed to the following specifications for such a "Conference Line":

(A) It must produce, on the average, better correlation among geosim models of a variety of forms at different scales than does the Schoenherr line; and

(B) it must produce lower smooth-ship resistances in order to avoid the small or negative roughness allowances found for modern all-welded hulls when the Schoenherr line is used for prediction of ship resistance.

With regard to A, it is possible, *in principle,* to determine from geosim data whether a new line can be found which produces better correlation among model sizes, on the average, than does the Schoenherr line.

In practice, this is extremely difficult because of

(a) deficicencies of test accuracy, per se;

(b) uncertainties as to the degree of turbulent flow achieved and the effects of tank boundaries on the measured resistance; and

(c) the possibility that, because of three-dimensional effects, all forms will not fit the same single-line function equally well.

With regard to B, it is not possible to establish accurately how much reduction in smooth-ship resistance prediction is desirable over the ship range of Reynolds numbers, because of

(a) the lack of any absolute measure of the effect of surface roughness on resistance;

(b) the uncertainties of ship resistance figures determined by direct towing trials;

(c) the uncertainties of ship resistance values deduced from full-scale thrust and power measurements because of possible scale effect on thrust deduction coefficient, propeller efficiency and other component factors of the propulsive efficiency; and

(d) the fact that for any particular full scale trial the "roughness allowance" deduced from a comparison of actual and predicted power depends to a large degree upon the method of extrapolation used.

The Committee therefore regards any "Conference Line" which can be proposed for adoption in the near future as essentially an arbitrary or compromise line.

To aid in reaching a decision as to the desirable features of such a Conference Line, the following steps were taken by the Committee:

(1) The available geosim data were assembled in tabular form by the Taylor Model Basin.

(2) These data were analyzed using the single line method and employing friction formulations proposed by Hughes, Lap-Troost, Schoenherr and Telfer, in order to see if reliable guidance could be obtained as to slopes in the model region which would, on the average, improve geosim correlation in accordance with specification A. The Schoenherr analysis was varied out at T. M. B. in close consultation with E. T. T., while Hughes, Lap and Telfer each carried out their own analysis.

(3) The results of a considerable amount of new geosim work recently carried out at N. P. L. were also presented and discussed.

(a) Available ship data were assembled to help in the selection of the level and slope of any proposed line in the ship region, in order to fulfil specification B.

They were discussed at considerable length at the Committee meeting held in London on 30 and 31 July, and 1 August, 1956. During the first two days meetings, Dr. Telfer and Mr. Lap were present by invitation of the Committee in order to present the results of their analyses, and Dr. Telfer remained as an observer on the last day.

The Chairman had circulated all the North American towing tanks in order to obtain their views on the question of the choice of a new single-line formulation, Dr. Kempf and Professor Lunde had made similar enquiries in Europe, and Dr. Hughes presented the results of a meeting of interested British delegates held in London on 3 July.

This information is set out in somewhat abbreviated form in Appendix II and was taken into account by the Committee in coming to its decisions.

As a result of its deliberations the Committee decided upon the following recommendations for submission to the 8th International Towing Tank Conference in Madrid in 1957.

Recommendations.

1. A review of the existing plank, geosim and ship data show that for model-ship extrapolation a line somewhat steeper than the Schoenherr line is indicated.

2. Despite this very positive indication, we find considerable difficulty at a period of such intense research in this field in agreeing upon a single-line formulation quantifying the desired steepness.

3. Nevertheless, we are conscious of the instruction given to the Committee by the 1954 Conference to make a definite recommendation for adoption not later than the next Conference. To try to give effect to this instruction, we have very carefully considered two suggestions. In the first of these, the present Schoenherr values of C_F would be retained for Reynolds numbers above 10^7, while they would be progressively increased at lower Reynolds numbers to give an increase of about 0.0004 at 10^6. In the second, the value of the Schoenherr C_F would be approximately retained at 10^7 and a new line drawn with a steeper slope, giving a value of C_F about 0.0004 higher at 10^6 and 0.0001 lower at 10^9 than the Schoenherr line.

4. If the Conference believes that the time is opportune for making a change in the friction formulation, the Committee feels that these two suggestions are both reasonable and that the choice between them should be made by the Conference.

REFERENCES

1. Proceedings of the 5th International Conference of Ship Tank Superintendents. London, 1948, p. 113.
2. Proceedings of the 6th International Conference of Ship Tank Superintendents. Washington, 1951, page 10.
3. Proceedings of the 7th International Conference on Ship Hydrodynamics. Scandinavia, 1954, p. 326.

F. H. TODD.
Chairman International Committee for Subjects 2-4.

APPENDIX I

REVIEW OF DEVELOPMENTS IN THE FIELD OF SKIN FRICTION RESISTANCE SINCE 1954.

A great deal of work has been carried out since 1954 upon the subject of skin friction resistance in the experimental, theoretical and analytical fields. This Appendix contains a brief summary of some of the principal contributions.

France

Towing trials have been carried out on a new mine-sweeper 42.96 metres in length (1). The ship was first run over the measured mile under her own power and the usual meaurements taken. She was then towed by a sister ship and the resistance measured. Model tests were also made. A comparison between model and ship resistance can thus be made and values of the so-called "roughness coefficient" ΔC_F obtained. In the paper the comparison is made between the actual ship results and those predicted from the model in two ways:

(1) Using the Schoenherr line and the Froude assumption that the residuary resistance coefficient C_R is the same for model and ship at the same value of $V/\sqrt{L}$.

(2) Using the Schoenherr line, but assuming that C_R can be divided into two parts, C_W due to wave-making resistance and C_V due to viscous form drag. C_W is assumed to be the same for model and ship at the same value of $V/\sqrt{L}$, but C_V is assumed to vary with Reynolds number, R_n. For any given hull shape, C_V is assumed to be a constant percentage of the basic plank friction C_F, and the ratio C_V/C_F has been called the "form factor".

The results can also be used to make a prediction in still a third way.

(3) Assuming C_W is constant and that C_V is proportional to C_F, as in (2), but using the Hughes two-dimensional turbulent line instead of the Schoenherr line.

The resulting values of ΔC_F are listed below, and it will be seen that there is little difference between the values obtained from methods (2) and (3), but that these values are considerably greater than those found from method (1).

Speed in knots	ΔC_F using method (1)	(2)	(3)
4	+ 0.00030	+ 0.00078	+ 0.00086
6	+ 0.00015	+ 0.00060	+ 0.00064
8	+ 0.00030	+ 0.00072	+ 0.00072
10	+ 0.00050	+ 0.00088	+ 0.00088
12	+ 0.00052	+ 0.00090	+ 0.00090
Value of Form factor.	—	0,36	0.46

The Schoenherr smooth C_F value at 6 knots is 0.00205, so that at this speed the C_F in column (1) represents an increase in skin friction resistance of 7.3 %, whereas that in colum (2) amounts to 29.2 %. While there is thus apparently only a relatively small gain to be expected by further attention to the smoothnes of the hull if we compare ship and model by method (1), if the comparison is made by methods (2) or (3) the possibilities appear much higher.

Germany

There have been no further geosim series tested since 1951, when results for pontoon geosims were published by Kempf and Karhan for Reynolds numbers up to $\log_{10} R_n = 8.4$.

Great Britain

A considerable amount of new experiment work has been carried out at the National Physical Laboratory on tests with ship model geosim series, pontoon tests with rough surfaces and tests with 6" diameter submerged cylinders.

In the geosim tests, four series of models were used covering a range of block coefficient. Each series was chosen to link up with tests which had already been made in other establishments: (1) the Victory model series, block coefficient 0.688, originally tested at the N. S. M. B. in Wageningen, (2) A fast steamer series, block coefficient 0.546, originally tested at H. S. V. A., (3) a cruiser series, block coefficient 0.52, also from H. S. V. A. and (4) a 0.75 block coefficient single screw merchant ship series (initially a B. S. R. A. form) tested first at S.N.T.H., Trondheim. In each case the N. P. L. series of models was tested in both the N. P. L. tanks. The complete results of this work are to be found in references (2), (3) and (4). The latter also includes a summary of the N. P. L. work of recent years on the viscous resistance of plane surfaces and ship models. The forms chosen for this investigation had already been the subject of extensive geosim testing in other tanks larger than the two N. P. L. tanks, so that the combined tests cover as large a range as possible of Reynolds number and of blockage. Except in the case of the Victory model series, the analysis to determine the viscous correlating slope was made according to a method developed by Hughes (3). This method depends on tests being made in different sizes of tank, and the author states that from a series of geosim tests made in this way the correct viscous correlating slope can be determined with practical accuracy without requiring a close determination of blockage effects. This development was made after the first paper (2) had been published, but the conclusions from the Victory model series, though less definite, are nevertheless in broad agreement with the findings from the later work.

Hughes concluded from his analysis that there was no evidence of any appreciable or consistent departure from Froude's Law of Comparison for the scaling up of wave resistance over the range of Froude number covered by these series. In this statement, wave resistance is to be defined as the resistance in excess of the viscous resistance corresponding to the line to which the model results "run-in" at low Froude number. Assuming that the results of any particular geosim series can be correlated by the use of a "viscous correlating line" of the form

$$r \times 0.066 (\log R_n - 2.03)^{-2}$$

Hughes also deduced from the analysis the followbers up $\log_{10} R_n = 8.4$.

Series	C_b	C_m	$\frac{L}{B}$	$\frac{B}{d}$	$\frac{L}{d}$	$\frac{\nabla 1/3}{L}$	Max F_n	Range of log R_n	Max. a/A %	Mean r for range of Froude number
Victory	0.688	0.988	7.04	2.18	15.32	0.186	0.27	6.0-7.2-7.9 *	2.94	1.27
	0.658	0.985	7.04	2.77	19.52	0.169	0.27	6.0-7.2-7.9 *	2.37	1.27
Fast steamer	0.546	0.969	8.30	3.42	28.35	0.1323	0.34	6.1-7.5	1.23	1.26
	0.504	0.954	8.30	4.79	39.7	0.1151	0.34	6.1-7.5	0.86	1.34
Cruiser	0.520	0.859	10.35	3.10	32.1	0.1161	0.48	6.0-7.7	0.79	1.24
	0.494	0.819	9.78	4.01	39.2	0.1088	0.50	6.0-7.7	0.58	1.34
B. S. R. A.	0.748	0.987	7.27	2.12	15.4	0.188	0.22	6.2-7.1	2.74	1.39
	0.731	0.989	7.27	2.62	19.0	0.174	0.22	6.2-7.1	2.20	1.41
	0.706	0.978	7.27	3.44	25.0	0.157	0.22	6.2-7.1	1.67	1.35

NOTE.—The length used is L_{BP} except for the fast steamer and cruiser, where the length on the waterline at each draught is used.

* Including D. C. Endert Jr.

The pontoons used originally for smooth surface friction investigations at N. P. L. were made the basis for a series of tests of different kinds of roughness, such as transverse plate edges, transverse welding beads, rivet points on transverse edges, frames and fore-and-aft seams and simulated paint roughness (5). Measurements were made of the total resistance and the velocity distribution in the boundary layer. The results showed that the calculated reductions in resistance due to the elimination of structural roughnesses were in good agreement with conclusions based on ship trial data. They also emphasized the marked variation in the effect of structural roughness on large ships as compared with small ships, and indicated that the resistance of a modern flush-welded ship with a good paint finish on top of clean bare steel may be considerably above that of a perfectly smooth surface. This work is continuing.

The resistance tests at N. P. L. using 6" diameter fully submerged cylinders of various lengths have been planned for some time, but have not made the expected progress due to pressure of other work. It is hoped that some results may be available when the Conference meets.

A programme of research was commenced in 1956 at Saunders-Roe, Cowes, using three models of the "Lucy Ashton" and three of the Victory ship. The models are to be tested in both the available tanks in order to provide data in the small model range on both scale and blockage effects. The Lucy Ashton models are of scales 1/30, 1/20 and 1/16 and the Victory models of 1/80, 1/50 and 1/36. The blockage values range from 0.0163 to 0.0018 for the former series, and from 0.0536 to 0.0041 for the latter. Flow observation tests are also being made to determine the nature of the boundary layer flow and the effectiveness of the stimulators.

Holland

Work has continued at the Netherlands Ship Model Basin on the extensive geosim tests on models of the Victory ship. An analysis of the resistance and thrust measurements on a number of models was presented in 1954 (6). The tests covered 9 models ranging in scale from 1/160 to 1/18, and also a model "boat" of scale 1/6, the latter being 72 feet in length. The results were analysed using the Schoenherr line in the standard manner and also using the Hughes and the Lap-Troost ship form frictional drag curves. The authors concluded that the resistance tests did not as yet give any direct indication as to the correctness of any particular method of extrapolation, the Schoenherr mean line being somewhat too flat and the Hughes and Lap-Troost methods giving too steep an extrapolator. In the analysis of the thrust deduction fraction from the self-propelled results, a considerable scale effect was found, the value of "t" increasing with increase in Reynolds number i. e. with size of model.

The theoretical formulae for pipe flow have been examined by Lap to see if resistance formulae for turbulent flow along flat plates could thereby be derived (7). The author concluded that it is not satisfactory to apply plate-friction coefficients to ship forms, and derived a new formula applicable to pipes, plates and ship forms:

$$K\sqrt{2}/\sqrt{C_F} = ln\ (R_n\sqrt{C_F}/A) + C$$

Where K = a universal constant, 0.4144;
C = 2.366 (*)
and log A = 1.000 for pipes,
= 1.980 for flat plates,
= 2.10 to 2.50 for ship forms.

(*) The constant C was wrongly given as 6.547 in the original Report. Its correct value is 2.366 as given above. See the foot-note on page 102 of these Proceedings (Editors' note).

Scandinavia

No new experimental work has been carried out since the last Conference in 1954. All the Scandinavian tanks use the Froude method and coefficients for extrapolation from model to ship.

As regards allowances for trial conditions on ships with newly-painted, all-welded hulls, the Trondheim tank uses — 5 % for super-tankers, 0 % for large vessels and + 10 % for small coasters.

Dr. Telfer has continued his investigations into the skin friction problem, and has analysed the geosim data collected by T. M. B. and made the results available to the Committee. In the search for a single-line, engineering solution, asked for by the 1954 Conference, Dr. Telfer has proposed the following formula:

$$C_F = 0.070 \, (\log R_n - 2.12)^{-2}$$

Spain.

Some new geosim tests corresponding to a few characteristic ship forms are being undertaken, as a continuation of those on tanker "Tina Onasis" carried out before the last Conference. Different types of turbulence stimulators are being tried.

A general review of the different friction formulae and correlation proposals is under consideration and will be presented at the Conference.

United States

A review of the present knowledge of skin friction was given by Todd in 1955 (8). After a brief history of the problem of predicting ship resistance from model tests, a summary of current proposals was presented in order to provide a basis for discussion at the forthcoming I. T. T. C. in 1957. Specifically, a correlation line is being sought which will properly correlate all sizes of models with one another and with full-scale results and prevent, in addition, such anomalies as negative roughness allowances on the actual ship.

A general analysis of the viscous resistance of different shapes has been developed from a hydrodynamic viewpoint by Hoerner (9). The viscous resistance coefficient of bare hulls is considered to be directly proportional to flat-plate frictional resistance coefficients, either smooth or rough. There are two components which are related to ship geometry: a frictional resistance which is larger than that of an equivalent flat plate due to a higher average velocity along the hull, and a pressure resistance due to viscous effects at the stern which prevent the formation of an after stagnation point. The resistance coefficient of rough surfaces when plotted against Reynolds number is stated to follow curves with slopes which are between zero and that for a smooth surfece. It is further stated that ships may attain a constant viscous resistance coefficient above speeds of 40 knots. Appendages are believed to show a resistance coefficient which increases slowly with Reynolds number in a manner stated to be similar to single roughness elements.

In order to extend the geosim analysis of the "Lucy Ashton" to lower Reynolds numbers, two smaller models were tested at MIT for correlation with the B. S. R. A. results (10). Best overall correlation was achieved with the Schoenherr line plus 6 % and the Lap-Troost line with log A = 2.034. However, for lower speeds correlation was somewhat better with the Lap-Troost and for the higher speeds with the Schoenherr line. In all cases a roughness allowance resulted for the full-scale vessel.

The problem of turbulence stimulation is yet to be completely solved. Some of the frustrating difficulties encountered on a particularly difficult model were described by Ridgely-Nevitt (11). It was finally found that raising the temperature of the water in the towing tank at Webb Institute above 72° F and so increasing the Reynolds number permitted satisfactory stimulation with a trip wire. Below 72° F only one configuration of sand strips gave a fairly satisfactory stimulation.

More fundamental investigation into the mechanics of transition is in progress at the National Bureau of Standards by Schubauer and Klebanoff (12). Artificially-produced turbulence spots are being used to observe their development into turbulent flow throughout the whole boundary layer. Transition is considered to start from perturbations of the laminar flow which develop into turbulence spots, which then grow in a fanwise manner.

Similar investigations into the mechanics of transition are being conducted by Hama at the University of Maryland with a small towing facility (13). It is observed that the two-dimensional, discrete vortex line, which results from the amplified perturbation wave, has a strong tendency in shear flows to form a three-dimensional vortex loop with a marked transverse wave length. The formation and development of the vortex loop are found to be the essential features preceding the origination of a turbulence spot and to be the guiding principle of transition, not only in boundary layers but also in wakes and jets. From these observations a very efficient turbulence stimulator was developed consisting of triangular patches to promote three-dimensionality of the laminar flow.

Additional analytical studies into the velocity laws of turbulent boundary layers have been conducted by Coles and separately by Landweber. Coles has made a comprehensive analytical analysis of the law of the wall and of the velocity-defect law (14, 15). The velocity-defect law is generalized into a new so-called law of the wake which is applicable also to boundary layers with pressure gradients and to three-dimensional boundary layers. Landweber modified the velocity-defect law to include the effect of the history of the flow (16). The application of this new velocity-defect law to the region of overlap with the law of the wall results in a general power law relation which includes the usual logarithmic law as a special case.

Hama has made a detailed study of the velocity laws of turbulent boundary layers for both smooth and rough flat plates (17). It was experimentally confirmed that the velocity-defect law is independent of roughness for flat plates as well as for pipes. The effects of roughness on the law of the wall for flat plates is found to be similar to that for pipes. From this it is suggested that pipe flow be used to study roughnesses arising from industrial processes.

Experimental studies of regular roughnesses have been performed by Ambrose at the University of Tennessee (18). Both cylindrical holes and projections were studied in pipe flow. The depression-type rougness failed to produce a constant resistance coefficient at the highest Reynolds numbers tested. On the other hand, the projection-type roughness and combinations of projections and depressions appeared to approach a constant resistance coefficient at high Reynolds number.

A considerable effort is being devoted to frictional resistance research at the David Taylor Model Basin. Granville has made a comprehensive study of methods of extrapolating the viscous resistance of ship forms from model to full scale (19). The skin friction of flat plates was discussed on the basis of the similarity laws of the velocity profile of turbulent boundary layers. A general formula for the resistance of flat plates was derived which provides a common basis for the form of the Schoenherr, Lap-Troost and Hughes formulae. The current proposals of Hughes, Lap-Troost, Telfer, Kempf and Karhan and Townsend were critically examined. Granville also considered the available test data, and arrived at the following conclusions:

(1) The basic Froude hypothesis concerning the separation of viscous and wave-making resistance is justified within the accuracy of the test data.

(2) The data are not accurate enough to establish the best procedure of extrapolating form or viscous resistance.

(3) It is imperative that a full-scale ship with a hydrodynamically smooth hull be tested.

(4) Additional research is needed into basin wall effects on the large models and turbulence stimulation on the small models.

(5) The use of a flat-plate line as a reference line for form or viscous resistance still seems a fruitful method of enquiry; and

(6) The concept of the form or viscous resistance coefficient having a fixed part and a part proportional to the flat-plate resistance coefficient is inclusive enough to encompass any probable variations.

Analytical methods have been developed for calculating the frictional resistance coefficients and velocity profile shape parameters from flat plate similarity laws instead of pipe similarity laws as has hitherto been the case. The use of similarity laws has, furthermore, the great advantage of permitting proper laboratory measurements of the frictional resistance of randomly rough surfaces like paint coatings to be extrapolated to full scale conditions. The details of the use of similarity laws for roughness studies will be published shortly.

The experimental program to be followed is to correlate local skin friction measurement with the velocity distribution of the boundary layer of the rough surface. Measurements of this nature will be made in the low turbulence wind tunnel. A floating element dynamometer has been designed and is now under construction. The towed cylinder being readied for transverse curvature tests will also be used for roughness studies at high Reynolds numbers.

In order to improve the present resistance prediction method, it is desirable to separate the wave and viscous resistances associated with surface flows. The theoretical derivation is based upon the extrapolation of Betz's method in separating the induced and viscous drags of airplane wings. The method utilizes pressure and velocity surveys in a plane normal to the flow direction and in the narrow wake region directly behind the body. The instrument for the experimental investigation was designed in accordance with the requirements of the theoretical analysis. It consists primarily of a traversing mechanism carrying two vertical wake survey rakes. A total of four rakes (3 ft. long) have been made, two for total head and static pressure measurement and two for flow angle measurements. In order to determine the region of integration for the measured quantities, provisions were made for measuring the height of the water surface at any point along the traverse. The capacitance wire system has been adopted. The velocity effect which is due to water-pile-up in front of the wire would give a false recording of the actual wave height; consequently the system was improved by using a reference probe, identical with the measuring wire, but located outside the wake region. The velocity effect would be cancelled since both probes would be affected equally. Consistent readings were obtained up to about seven knots and its accuracy of wave height measurement was about 1/16 in. Due to the difficulties in operating the air-filled manometer experienced from the past, it is planned to use pressure transducers with suitable manifold system. Such plans are being developed and a manifold will be constructed.

As part of the program of developing a submarine of least resistance, a study is being made of the viscous drag of deeply submerged bodies of revolution by means of the boundary layer concept. Theoretical methods are being developed and refined for calculation of the growth of boundary layer so that the drag of the proposed bodies of revolution can be accurately predicted from their geometry. The experimental investigation has been coordinated with the theoretical approach. Experiments have been carried out by using the existing 7 ½ ft. ALBACORE model in the subsonic wind tunnel of the Aerodynamics Laboratory. The model consists of a body of revolution with a removable deck, conning tower, and tail control surfaces, so that tests were made both on the basic form of a body of revolution and on the actual configuration of the ALBACORE. The measurements consisted of boundary layer surveys and local skin friction measurements by surface pitot tube at various angular locations and for a range of speeds. In addition, similar measurements are planned for the full scale ship, which will provide extremely valuable information, not only at full scale ship Reynolds number but also in extrapolating from model to full-scale conditions.

Systematic measurements are being made of turbulent boundary layers in adverse pressure gradients to develop more accurate methods of predicting boundary layer growth and turbulent separation. The application of a second order approximation based on boundary layer thickness to the boundary layer equation yields a generalized momentum equation in which an extra term of Reynolds normal stress in the direction of flow is introduced. Further experimental evidence is desirable to ascertain the significance of this stress in the mechanism of turbulent separation. The necessary measurements for this study include precise boundary layer velocity profile measurements, local turbulent skin friction measurements and the Reynolds stress components in the boundary layer. For precise velocity profile measurements, the design of a micro-manometer which was originally developed by the Douglas Aircraft Company and can measure 0.0002" of water, is now being modified. The surface tube technique for measuring local skin friction which was originally used in zero pressure gradients by Preston in England has been extended for use in adverse pressure gradients. The details of the use of such techniques have been given by Hsu (20).

In predicting the frictional resistance of surface vessels or submerged bodies from that of models, knowledge of transition from laminar to turbulent flow plays an important role. Efforts have been made to develop proper turbulence stimulators so that a predetermined position of turbulent boundary layer on any model can be assured. The results of chemical transition indicator technique as developed for the investigation of tack-type internal turbulence stimulators in the Model Basin have led to the following conclusions:

a) the technique permits vivid differentiation between those surface areas subjected to laminar flow and those subjected to turbulent flows,

(b) the difficult and time consuming nature of both the execution of a test and analysis of results makes this technique impractical for the detailed test program originally initiated.

Tests will be conducted on flat plates with and without pressure gradients, and on bodies of revolution in the low turbulence wind tunnel through the use of the hot-wire anemometer technique.

Knowledge of the transverse curvature effect in frictional resistance would enable us to apply a correction to the results of resistance tests on small models having small radii of curvature. To study this problem a submerged, neutrally buoyant cylinder has been buit to be towed in the direction of its axis. The cylinder is made of aluminum, 6 in. in diameter, and is made up of 10 ft. sections to give a variable length up to 150 ft. Shear force will be measured in a one-foot test section, supported by internal flexures, which can be inserted at any station along the length. A satisfactory rubber seal for the one foot dynamometer section has been developed and the entire test equipment is now complete.

In connection with the work of the Skin Friction Committee, the Taylor Model Basin undertook the collection of available geosim series data from all sources and, in cooperation with the Experimental Towing Tank of Stevens Institute, its analysis according to the Schoenherr friction formulation. The results of this work were presented to the A. T. T. C. in September, 1956 (21). They indicated that more careful consideration should be given to the methods of testing models, in order to ensure agreement between different tanks, and that in using the results to try to determine an "engineering" line solution to the extrapolation problem special care should be paid to the stimulating devices used and to the possible influence of wall effect. The results suggest a friction line somewhat steeper than the Schoenherr line in the lower range of Reynolds number below 10^7. The desirable trend at higher Reynolds numbers can only be determined by a consideration of full scale ship data.

Hughes has also analysed the data collected by Taylor Model Basin. On the various geosim sheets new analysis lines were drawn based on the Hughes formulation

$$r \times C_F = r \times 0.066\ (\log R_n - 2.03)^{-2}$$

In the first instance these lines were drawn for the value of r which gave "position" agreement with the low Froude number "run-in". This was not well defined in some cases, but the values of r used ranged between 1.20 and 1.30 in most cases. A considered assessment was then made of the correct mean correlation slope for the whole range of Froude number in each case, after allowing for any possible effects of laminar flow and tank boundary interference. While it was not possible to assess this slope with a high degree of accuracy, Hughes concluded that in general it is not less than given by the value of r for "position" agreement at the low Froude number run-in. That is, values of r rather greater than within the range 1.20 to 1.30 appeared to be required for these geosim series in general. The composite plots prepared by TMB were also considered from a somewhat different approach. Hughes states that the *upper* envelope of a series of plots all conforming to the same viscous slope cannot be steeper than this slope; hence the *average* correct viscous slope for all the series included in these composite plots is not less than given by the upper envelope. For the complete composite plot the approximate upper envelope has a slope corresponding to about $r = 1.30$, and for the restricted composite plot about $r = 1.20$ to 1.30.

There is broad agreement between the general average and range of the values of r deduced by Hughes from the Taylor Model Basin data sheets and his own tests on the Victory, Liberty and other series.

Experiments with small models of destroyers, the Lucy Ashton, Victory and Liberty ships have been carried out at Webb Institute of Naval Architecture (22). From these tests a friction line has been proposed having the formula

$$C_F = \frac{0.80}{(\log_{10} R_n)^{2.86}}$$

This lies above the Schoenherr line for Reynolds numbers less than 10^8 and has a steeper slope.

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APPENDIX II

REPLIES TO ENQUIRIES REQUESTING VIEWS OF VARIONS TOWING TANKS.

The members of the Skin Friction Committee have canvassed the many towing tanks in their own geographical areas and requested information as to the current practice regarding ship prediction techniques and also the views of the staffs as to the most desirable procedures both for the immediate future and for the long term solution of this problem.

These views were available to the Committee during its deliberations and undoubtedly influenced its recommendations. It is considered important, therefore, that they should be available to members of the Conference before the meeting in Madrid. They are set out below in a somewhat abridged form, but it is believed that no essential points have been omitted.

Opinions of North American Towing Tanks.

1. *Letter from Dr. Todd, dated 6 June 1956.*

"Dear ,

As Chairman of the International Skin Friction Committee, I am holding a meeting in London on 30 and 31 July and 1 August, to discuss the work of the Committee, the latest developments in this field, and, in a preliminary way, the report the Committee will present to the Madrid Conference next year.

As you will be aware from the Proceedings of the Seventh International Towing Tank Conference in Scandinavia in 1954, the members on that occasion passed the following resolution (p. 326):

> '3. The Conference is of the opinion that agreement on a friction formulation based on Reynolds number and adequate for practical ship design purposes is urgently required. To meet this requirement, the Skin Friction Committee should consider all relevant data and make a definite recommendation not later than the next Conference.'

In order that Dr. Davidson and I may correctly represent the opinions of the American and Canadian tanks at the London meeting, I should appreciate having your comments on the above resolution.

In particular, we should like to know whether you favor retention of the Schoenherr line for the time being, or whether you feel the time has come to adopt a new line which would have universal acceptance and so eliminate the present division of presentation of results on the Froude and Schoenherr coefficients.

In the latter case, we should like to have your opinions on the specifications you think such a line should meet.

signed/ F. H. TODD."

2. *Brief Summary of Replies.*

(a) *Michigan Tank (Profesor Baier)*:

1. The Schultz-Grunow formulation is slightly better than Schoenherr's, but the difference does not warrant any change, as other factors are more important.
2. Agreement should be reached on a definition of "effective length" for use in Froude and Reynolds number.
3. An obliquity factor should be used in calculating wetted surface.
4. ΔC_F should not have a constant value, but should decrease to zero on small boats. Suggested values for roughness allowance are:

C_F Smooth Schoenherr	0.0035	0.0030	0.0025	0.0020	0.0015	0.0010
% allowance						
riveted	0	5.8	12.0	18.7	26.7	—
welded	—	0	6.3	12.9	20.1	30.0
ΔC_F						
riveted	0	0.00017	0.00030	0.00037	0.0004	—
welded	-	0	0.00016	0.00026	0.00030	—

(b) *Newport News Hydraulics Laboratory (Mr. Hancock)*.

1. In favor of the action expressed in the resolutions of I. T. T. C., 1954.
2. Content to leave choice of line to delegates at London meeting.

(c) *National Research Council Laboratory, Ottawa (Mr. Turner)*:

1. Agrees with I. T. T. C. resolution, 1954—it is confusing to have two skin friction formulations in use.
2. Problem is to determine which formulation exactly expresses the frictional drag on both model and prototype.
3. At present we do not know enough about the problem to formulate an expression which will give an exact answer.
4. Schoenherr line is not perfect, but has given good results and has been used for a long time. It should not be discarded without very serious consideration, for it is the standard upon which a vast amount of information now rests, and only then when some very much improved formulation is obtained.
5. Favors retaining the Schoenherr formulation until the ultimate goal, the formulation of which is scientifically correct, has been reached.

(d) *Webb Institute of Naval Architecture (Profesor Ridgely-Nevitt)*:

1. Schoenherr line should be abandoned at earliest possible date.

2. No method dealing with resistances at non-wavemaking speeds (such as Hughes or Lap-Troost) should be considered, because of resulting turbulence stimulation difficulties.

3. The Telfer extrapolator is entirely unsatisfactory. We know of no physical reason why friction should conveniently plot in a linear fashion.

4. Feel a perfectly satisfactory formulation of the single line type can be developed which will tie together recent fully turbulent geosim series. The slope should be higher than the Schoenherr line and the ship prediction (not necessarily the skin friction value) should be lower than present predictions based on Schoenherr.

5. The Webb Tank would be happy to adopt any formula meeting the above requirements. If no international agreement is reached, Webb will have to abandon Schoenherr because small models with fully turbulent flow result in too high a ship prediction.

6. Webb Institute suggests a formula

$$C_F = \frac{0.80}{(\log_{10} R_n)^{2.86}}$$

This line:

(i) Has a slope which correlates various sized models of the same ship, and

(ii) gives numerical values of the same order of magnitude as those determined from recent fricction plane experiments having Length/Draft values in the vicinity of 10.

(e) *M. I. T. (Prof. Troost and Prof. Abkowitz)*:

1. As far as geosim analysis at present available goes, the time has not yet come to adopt a new line which would have universal acceptance.

2. There are certain areas needing extended study before adopting a new "engineering" formulation for the time being, e. g.:

(i) Measurement of pressure distributions on model and ship hulls, and

(ii) Change of 3-dimensional "hull facfors" when proceeding from non-wavemaking through moderate wavemaking $V/\sqrt{L}$ ratios in large as well as restricted water depth.

3. Do not believe there is sufficient experimental evidence to defend (the proposition) that there is one unique formulation at present available to give definitely improved results for ship forms at all reasonable values of $V/\sqrt{L}$ and water depths as compared with the Schoenherr line.

4. In view of the very important practical consequences of adopting (even for a limited number of years) a new "engineering" line, it is our definite opinion for the time being it will be wise policy to favor retention of the Schoenherr line (with freedom to apply the Froude coefficients for those who wish it) at least until a possible 1960 Conference.

5. We need the period 1956-1960 to study and evaluate new work completed and in hand.

6. The adoption for the time being of a new friction formulation might at this time be felt by many conference members to be an over-hurried decision, possibly unduly influenced by national or personal sensibilities, and doing more harm than good for the profession as a whole. No disaster of any kind can arise from a retention of presently existing international agreements for another period of three years, which we believe is needed for consolidation of widely varying opinions on this presently "too hot" issue.

(f) *Iowa Institute (Dr. Landweber)*:

1. Presently in favor of retaining the Schoenherr line for the time being, but would be willing to accept slight adjustments, as indicated by the geosim analyses, in order to obtain international agreement on a uniform procedure.

2. Agrees with specifications for such a line as set out by the International Committee in its statement of philosophy.

3. Does not have much hope that the geosim studies will prove or disprove any particular friction line or set of lines. We need rather to learn the causes of variations between tests of the same model at various towing tanks, to investigate in detail the flow conditions which result in anomalous values of C_T for many small models (separation, stimulation, etc.) and to devise and carry out a set of experiments in which the friction, form and wave drags are separately measured on a series of models.

Until such a rational program of research is carried out I doubt that available data would show another line or lines to be preferable to Schoenherr's.

(g) *University of California (Dr. Schade)*:

1. I do not favor retention of the Schoenherr line for the time being.

2. I feel the time has come to adopt a new line which would have universal acceptance.

3. The most important specification such a line should meet is that of acceptance by all tanks concerned; in fact in my opinion this is the only really important specification.

(h) *E. T. T., Stevens Institute (Dr. Davidson)*:

No reply was received from E. T. T. in letter form, but Dr. Davidson stated at the meeting in London that they would prefer to keep Schoenherr, although they would not object to a slight steepening at Reynolds numbers below 10^7.

(i) *Taylor Model Basin (Dr. Todd)*:

1. The Schoenherr line should be used for all values of Reynold number above $R_n = 5 \times 10^6$. Below this no objection would be raised to a progressive steepening of the line to better fit the smaller models of the geosim series.
2. The line should be defined by means of an explicit formula, of polynomial type, for use on large electronic computers.
3. The range of its use should be clearly defined.
4. It should be called a "model-ship correlation line", rather than a skin friction line.
5. The Schoenherr formulation is presently used together with the Froude assumption of constancy of C_R. This latter assumption is almost certainly in error, and of recent years other assumptions have been proposed for the breakdown of model resistance into its component parts. At the opposite end of the scale to the Froude assumption is that proposed by Hughes, in which all the so-called viscous drag coefficient is assumed to vary with Reynolds number. The truth is certainly somewhere between these two limits, and until more certain knowledge is obtained the Taylor Model Basin would be strongly averse to forsaking the Schoenherr line for any new arbitrary engineering line, which would be a compromise line and one which might be changed again in a few years. More especially is this the case because of the vast amount of data presently analysed by the current Schoenherr method and the many model-ship correlations now available, which enable sufficiently correct ship predictions to be made.

Opinions of British Tanks.

Extracts from Minutes of an Informal Meeting of British Delegates, held in London on Tuesday, 3 July, 1956.

Present:

Dr. J. F. Allan, Chairman, N. P. L.
Mr. K. C. Barnaby, Messrs. Thorneycroft.
Prof. L. C. Burrill, King's College, Newcastle.
Dr. J. F. C. Conn, B. S. R. A.
Mr. W. Crago, Saunders-Roe, Ltd.
Mr. J. M. Ferguson, John Brown & Co.
Dr. R. W. L. Gawn, A. E. W., Haslar.
Dr. G. Hughes, N. P. L.
Mr. D. I. Moor, Messrs. Vickers, St. Albans.
Mr. S. J. Palmer, R. C. N. C., Greenwich.
Dr. S. L. Smith, B. S. R. A.
Dr. E. V. Telfer, University, Trondheim.
Mr. W. P. Walker, Denny Bros., Dumbarton.
Mr. H. Lackenby, B. S. R. A., Acting Secretary.

"The Chairman reminded the delegates that at the last informal meeting on 24 February, 1955, it was agreed that the Hughes line + 10 % would be acceptable in this country as an average extrapolator for model results. At that time Dr. Allan informed the Chairman of the International Skin Friction Committee of this view of the British delegates and so far there had been no reply. As mentioned earlier, the Skin Friction Committee is meeting in London on 1 August next and Professor Telfer and Mr. Lap are attending preliminary meetings on the previous two days.

Dr. Hughes explained the work of the Skin Friction Committee and the geosim analysis.

Professor Telfer proposed a formula of the type

$$C_F = \frac{\text{constant}}{(\log R_n - A)^2}$$

where the constant = 0.07 approximately and $A = 2.12$.

It was pointed out both by Dr. Hughes and the Chairman that a form factor of 20 % above the Hughes line was much more acceptable for average merchant ship work than 10 %.

It was finally agreed that there was now sufficient evidence for the Skin Friction Committee to come to a definite practical decision in this matter. It was also agreed that the name to be given to the new formulation should be impersonal, such as 'I. T. T. C. line' or something similar."

In the covering letter to the above minutes, the Chairman wrote:

"... As mentioned at that meeting, there is a great deal of evidence to show that for the average run of merchant ship work a line in the region of Hughes + 20 % is much nearer correct than Hughes + 10%. Since the meeting Hughes has been giving this matter some thought and has suggested a formula

$$C_F = \frac{0.080}{(\log R_n - 2.0)^2}$$

which lies very close to Hughes + 20 %, and has the added advantage that the constants are simple round numbers.

In my opinion, this formula would provide a very happy solution for an I. T. T. C. line.

signed/ J. F. Allan."

Opinions of Continental European Towing Tanks.

1. *Letter from Dr. Kempf, dated Istanbul, May, 1956.*

"Dear :

Dr. Todd, as Chairman, has proposed to hold a meeting of the Skin Friction Committee of the I. T. T. C. at the end of July in London. He has asked the members (Dr. Davidson, Dr. Hughes, Dr. Kempf, Dr. Lunde) to explore the prevailing opinions of the other tank leaders of their geographical area.

The main object will be the discussion of a suitable line and formula for turbulent friction of a smooth flat suface in view of a proposal to adopt such a line as a standard for the transformation from model to ship. Therefore I beg you to answer the following questions:

1. What basis for turbulent friction of a smooth flat surface and what method of transformation from model to ship do you use in your tank?

2. How do you allow for the roughness of a new painted all-welded ship skin on trial condition at Beaufort 2?

3. What basis and method of transformation from model to ship would you prefer to adopt as a standard regarding the existing experimental results and theoretical considerations?

4. What arguments have guided you to the above preference?

Sincerely yours,

signed/ G. Kempf."

2. *Brief Summary of Replies.*

(a) *Hamburg Tank (Dr. Lerbs):*

1. Use Schoenherr curve as base line and predict ship results by Froude method, using Froude's as well as Schoenherr's friction values.

2. Using Schoenherr line, the following allowances are made:

$$\Delta C_F = 0.0002 \text{ for } R < 10^9 \quad ; \text{ and}$$
$$\Delta C_F = 0.00015 \text{ for } R > 10^9$$

3. Froude method using Schoenherr line plus allowances is the most reliable. Results are also analysed using the Froude coefficients for comparison with earlier tests.

4. Analysis of former data (using the Froude method and the Schoenherr line) have given results in the past in very satisfactory agreement with trial data, and also in comparisons between results in Hamburg and in other continental tanks.

(b) *Vienna Tank (Dr. Kretschmer):*

1. Would prefer to use Schoenherr but customers prefer Froude because of accumulated results derived in this way.

2. If use Schoenherr, would adopt a value of

$$\Delta C_F = + 0.0003.$$

3. Hughes method in its complete form.

4. Gives better correlation between smooth models and smooth ship and therefore a better division of resistance into its components. Moreover, having correct prediction for a smooth hull gives opportunity of more correct assessment or roughness. This latter requires more precise definition. We have to accept modern knowledge but it will be difficult to convince our customers of its use because of all the accumulated data.

(c) *Berlin University (Dr. Horn):*

1. No answer—no tank!

2. ΔC_F should not be less than $+$ 0.00025. Smaller additions are a sign that the viscous line used in extrapolation is not correct.

3 and 4. Dr. Horn prefers lines with a physical basis, i. e. 2-dimensional line as a base with individual form addition. Thus Schoenherr and Prandtl-Schlichting lines are not correct. It is not possible from normal model testing with one model to get the form addition for a given ship form with accuracy. Believes we must have systematic tests for this purpose and present is not the time to change procedure to some other friction line (such as Hughes) in association with individual form factors.

(d) *Rome Tank (Dr. Castagnetto):*

1. Use Froude method and Froude coefficients.

2. No allowances are made—this question is left to the shipbuilders.

3. and 4. For new-painted, all-welded ships, would prefer straight Schoenherr with no allowances. It would be preferable to supply customers with both old and new style predictions for a time.

(e) *N. S. M. B. Wageningen (Dr. van Lammeren):*

1. Use Froude method and coefficients because it is still the official method and all past data is based on it. For scientific investigations, mostly choose Lap-Troost method.

2. For new-painted, all-welded hulls, make following additions (calculated according to Froude):

0-5 % for big tankers,
7 % for average cargo ships, and
12 % for coasters, tugs, etc.

3. If have to change, prefer to go to straight Lap-Troost method.

4. (i) There is certain theoretical evidence for the Lap-Troost method in agreement with experimental results.

(ii) Formula belongs to a family of theoretical formulae which are fairly generally accepted in other fields of hydrodynamics.

(iii) Formula shows close relationship with velocity distribution in boundary layer.

(iv) Only theoretical formulae of this type, as, for instance, Lap-Troost, make physically correct treatment of the roughness problem possible.

(v) Method can be used for shallow water work. Results of experiments with geosims in shallow water will be published this year.

(vi) Three-dimensional methods of this kind, if applied to a ship with and without appendages, in general result in scale effect on appendages being completely taken into account.

(f) *Berlin Tank (West) (Dr. Schuster)*:

1. Schultz-Grunow line used as basic line, predictions being made by Froude method.

2. $\Delta C_F = +\ 0.0002$ for all-welded hull,
$\Delta C_F = +\ 0.0003$ for partially riveted hull.
$\Delta C_F = +\ 0.0004$ for all-riveted hulls.

3. From theoretical considerations, prefers Hughes' method—but since we have not yet, clear ideas as to the values of the form factors, we have no option but use existing methods. Preference is to remain with Schultz-Grunow or Schoenherr.

4. Transformation by Schultz-Grunow or Schoenherr is simple, in widespread use and gives good agreement with trials.

(g) *Berlin Tank (East) (Dr. Gutsche)*:

1. Schoenherr.

2. $\Delta C_F = +\ 0.0002$.

3. and 4. Too little experience in field of comparison between model and ship to give opinion.

(h) *Duisburg Tank (Dr. Graff)*:

1. Froude friction coefficients used.

2. Add + 0.0002 to Schoenherr.

3. Until international agreement is reached, prefer to continue use of Froude coefficients, but basically prepared to adopt any other method based on Reynolds number, as, for instance, the Schoenherr line.

4. With new, all-welded ships, the use of Froude or Schoenherr coefficients with $\Delta C_F = +\ 0.0002$ gives good correlation with trials. Prepared to make a change to any other method if it has probability of some permanence in time.

(i) *Madrid Tank (Dr. Acevedo)*:

1. In normal routine tests, calculations are made according to W. Froude's conventional method, using the R. E. Froude frictional coefficients for model and for ship (15° C, 59° F).

In some research work methods due to Prandtl-Schlichting, Schoenherr, Schultz-Grunow and Hughes have been used.

2. No systematic method. No separate allowances are given for roughness, wind, scale effect or tank wall interference.

Allowance for trial prediction is some times made. A few years ago this was 15 %, more recently reduced to 10-12 % for newly painted, all-welded ships of medium size. For super-tankers negative allowances of 5 to 6 % have been found.

All the above figures apply to calculations according to the W. Froude conventional method using R. E. Froude friction coefficients.

3. Anxious to depart from the old and irrational R. E. Froude coefficients and from the extrapolation method of W. Froude. Dr. Acevedo has no definite opinion on what the solution should be, and even doubts whether the time is right for adopting one, solidly founded in fact, and which would not be merely conventional and perhaps quickly become out of date.

4. Dr. Acevedo has no conclusive preference, but only arguments as to why he has none.

Personally, he inclines towards extrapolation based on constant form factor. However, from a practical point of view, the prior work of determining the form factor is a handicap in comparison with the easier Froude method.

Hughes' line is possibly truer than Prandtl-Schlichting or Karman-Schoenherr in the model range, but too low in the ship range.

It is desirable that the Skin Friction Committee after its next meeting inform us as soon as possible on studies and progress made since the Scandinavian meeting.

(j) *Zagreb Tank, Yugoslavia (Prof. Silovic)*:

1. At present only small models, 1.5 to 1.8 metres in length, are used, the calculations being made using the Froude friction coefficients.

2. Not yet sufficient experience to decide on any allowance.

3. Froude method gives too high a prediction when using small models. Decided to use Schoenherr line with $\Delta C_F = +\ 0.0003$ allowance, but would consider Hughes or Lap methods for future use.

4. Results of Victory model tests with different forms of stimulation.

Introductory Remarks on Techniques of Measuring Model Resistance

by Prof. Dr. Ing. G. Kempf (*).

One can hardly deny that the accuracy of predicting the performance of a ship, based on the measured model resistance, has reached a critical point, that for many years there has not been significant progress, and with which we are not yet completely satisfied. Considering only the main reasons for this statement, two causes may be offered:

(1) Regarding the ship, it is the actual skin roughness and its equivalent value in terms of resistance which is difficult to evaluate accurately.

(2) Regarding the model, it is the continuing lack of agreement between measurements of resistance for a model, with a technically smooth surface in unlimited smooth water and fully developed turbulent flow, made at different times or places, which is of concern to us. Only this resistance can be the basis for a sound prediction of the ship performance in the same condition.

Many resistance tests of models have clearly shown that the results do not sufficiently agree when the same models are tested at different times or in different tanks. The possible reasons are so manifold and some of them probably unknown that they have offered for many years a vast field for fruitless discussions. Therefore, it is not intended to discuss here these causes, but rather how this situation of uncertainty can be overcome so as to get better agreement between the measurements of resistance for the same model.

As a mechanical basis, the techniques of measuring model resistance may first be described.

The different techniques which are known and in use are:

(1) The model is towed by a certain constant force, for example by a falling weight on a cord passing over a pulley. The resistance, equal to the falling weight, is known and the speed attained is measured.

(2) The model is towed at a certain constant speed and the resistance corresponding to this speed is measured.

In this case, the model is either:

(a) towed with a cord at constant speed, or;

(b) attached to a carriage which runs with constant speed. The model is then either:

(i) rigidly attached to the carriage and its resistance is measured electrically, or

(ii) it is more or less "softly" attached to the carriage by a spring-balance or by a pendulum-balance and its resistance is measured mechanically, and one must put up with a certain period of oscillation of the system (model + balance), till a state of equilibrium has been reached.

(3) Sometimes a combination of (1) and (2) is used: The model is towed by a cord carrying a certain constant weight and passing over a pulley attached to the carriage. The carriage runs at constant speed. The model is at first accelerated with the carriage, but left free when the speed of the carriage has become constant. Then the model, independent of the carriage, will soon reach its own constant speed under the load of its towing-weight. The difference between the speeds is measured.

With all these techniques it is possible by careful measuring to keep the mechanical error in the resistance within $\pm$ 1 %, provided that the error in measuring the speed remains at least within $\pm$ ½%. Greater differences in the measurements of resistance are not so much mechanical as physical in cause, and these causes are manifold.

In such a situation the method of mass-statistics offers itself as a means of analysis.

The proposal of Ing. General Barrillon, which the reporter strongly endorses, may therefore be submitted to the Conference for consideration:

Every tank which can afford to do so should procure a standard model or two of a material which will retain its shape and its smooth surface. It may be of lacquered wood, polished metal or some other similar material.

The resistance of this model should be measured over a whole year, every fortnight and at different times of the day. All physical qualities of the surroundings should be carefully registered. In addition it is recommended that any changes in the measured resistance be noted when:

(1) the model is run head-on in the opposite direction through the tank (influence of current).

(2) the acceleration is changed (separation of flow at the rear).

(3) the position along the tank at which measurements are made is changed (reflection of the wave).

(4) the method of measuring is changed.

(5) the interval of time between the tests is altered.

Moreover, it is recommended that two standard models be used of two block coefficients: one less

(*) Former Director Hamburg Tank, not in attendance.

than 0.6 and the other more than 0.8. The fine model will be free of separation of flow, the full one will suffer from it to a greater or less degree with changing acceleration and with temperature of water. Comparing the different changes in resistance of the two models under the same conditions will show sparately the influence of separation of flow.

This proposal to measure the resistance of standard models over a whole year under different physical conditions will be a first step towards recognising some of the causes of differences in the measured resistances and finally to avoid or to eliminate them.

The second step would be collecting the results from the various tanks, their evaluation and coordination by a special committee and the preparation of a report.

A third step may be advisable later to exchange the models between different tanks and repeat the programme.

Within three years our knowledge as well as the accuracy of measuring model resistance could be improved, by eliminating some faults, for the benefit of all tanks and the prediction of ship performance, resulting moreover in an increasing trust in tank work by the ship owners.

Formal Discussion

Dr. F. H. Todd (Written contribution (*).

The proper correlation of the resistance of ship models of different sizes and the reliable prediction of actual ship resistance from model results lie at the very foundation of our work. The first step towards the solution of these problems was taken by William Froude when he proposed to divide the total model resistance into two parts, one due to wave making, which he called residuary resistance, and the other due to frictional resistance, or, in coefficient form

$$C_T = C_R + C_F \qquad (1)$$

He showed that the wave-making resistance would scale directly as the displacement at "corresponding speeds" i. e. at the same value of $V/\sqrt{L}$, C_R would be the same for any size of geometrically similar hull. To determine the frictional drag he made the assumption that the frictional resistance of the curved surface of the model or ship would be the same as that of a smooth plank having the same length and wetted surface, and gave frictional coefficients for different lengths of plank.

This whole technique, as exemplified by equation (1), may be called the "Froude assumption". It has stood the test of time so well that it is still used by all experiment tanks and, however imperfect, still remains the basis of all methods of ship power predictions from the results of model tests. The two methods at present agreed upon by this Conference for use in all published work are based on this assumption, one using the Froude friction coefficients, the other the American Towing Conference 1947 line, originally derived by Schoenherr.

What of our present knowledge of the subject, and the chances of agreement on a common formulation?

In 1954 the Skin Friction Committee in its report stated that it believed that a proper correlation of model and ship resistance must take account of the effects of 3-dimensional flow, and that a suitable basis from which to develop a method for such correlation would be the smooth turbulent friction line for 2-dimensional flow. The experiments considered by Froude, Schoenherr and others were made on planks of a variety of sizes and shapes, geometrical similarity being ignored, and the results suffered in varying degree from lack of turbulence stimulation, aspect ratio and edge effects, and surface wave-making. In the last few years Hughes has published results of careful experiments on planks and pontoons covering a great range of size, and by extrapolation has derived a 2-dimensional line for turbulent flow friction over a smooth surface (1). This is compared with the A. T. T. C. 1947 line in figure 1. The Hughes line is higher and steeper for values of log R_n below 5.65 and lower for all values of R_n above this point, although in the ship region it is somewhat less steep than the A. T. T. C. line. It should be remembered that the Hughes 2-dimensional flow line is derived from 3-dimensional data and so depends upon the method of extrapolation used. That it is not necessarily a unique line may be concluded from the fact that Wieghardt, using Hughes' data but a different method of extrapolation, has derived a different two-dimensional line.

The frictional coefficients represented by the Froude, Schoenherr, Hughes and similar lines are derived from measurements of total drag. Another method of obtaining a friction formulation is by measuring the velocity distribution in the boundary layer, so determining the shearing force at the surface and then the resistance by subsequent integration. Using this method, Landweber has given results which, over the practical range, coincide almost exactly with the A. T. T. C. 1947 line (2). Coles, on the other hand, found a line much lower than the A. T. T. C. at low Reynolds numbers, but with much less slope, so that it almost coincided with the A. T. T. C. line at the highest ship values of Reynolds number (3). Townsend, using a theoretical approach, proposed a line about midway between Hughes and A. T. T. C., and having less slope than the latter (4).

Some of these lines are shown in figure 1. At the moment all that can be said is that there is no absolute agreement between any of these lines, the slopes differing to an appreciable extent, and the maximum difference in C_F at $R_n = 10^{10}$ being some 12 %.

(*) This contribution was presented by Mr. J. Hadler, since it states the opinion of the Taylor Model Basin and Dr. Todd, at the time of the Conference, has already left his post of Technical Director of that Basin and had become a British delegate. (Editors' note).

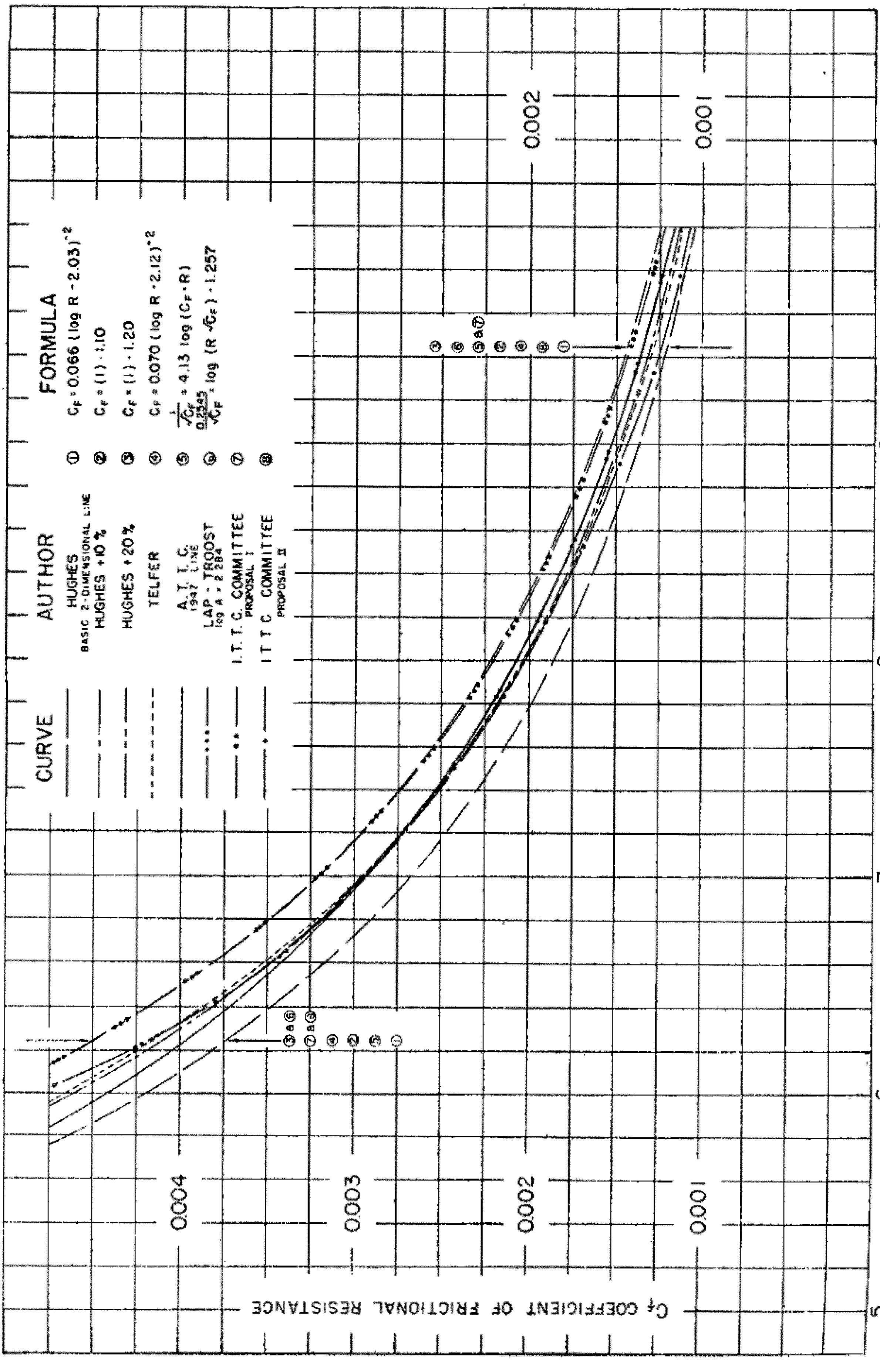

CURVE
AUTHOR
FORMULA
HUGHES BASIC 2-DIMENSIONAL LINE ① $C_F = 0.066\,(\log R - 2.03)^{-2}$
HUGHES +10% ② $C_F = (1) \cdot 1.10$
HUGHES +20% ③ $C_F = (1) \cdot 1.20$
TELFER ④ $C_F = 0.070\,(\log R - 2.12)^{-2}$
A.T.T.C. 1947 LINE ⑤ $\frac{1}{\sqrt{C_F}} = 4.13 \log (C_F \cdot R)$
LAP - TROOST log A = 2.284 ⑥ $\frac{0.2545}{\sqrt{C_F}} = \log (R\sqrt{C_F}) - 1.257$
I.T.T.C. COMMITTEE PROPOSAL I ⑦
I T T C COMMITTEE PROPOSAL II ⑧
0.004
0.003
0.002
0.001
5
6
7
8
9
10
LOG_{10} R
C_f COEFFICIENT OF FRICTIONAL RESISTANCE

Fig. 1

However, the search for a 2-dimensional flow line is only part of our problem.

Indeed, there are some who do not believe that this line is an essential preliminary to the extrapolation problem, but rather that each hull form requires a separate extrapolator. Telfer has advocated this approach since 1927, believing that the specific frictional resistance for geometrically similar forms (geosims) is a linear function of $R_n^{-1/3}$ (5). Troost and Lap have published a proposed method which includes a coefficient A dependent upon the hull form in any given case (6).

It is evident, of course, that the Froude assumption is only a first approach to the extrapolation problem. The frictional resistance of the actual curved hull surface cannot, in general, be equal to that of the equivalent plank, for the local velocities of the flow are different and the paths traversed by the streamlines are longer. Wigley has made estimates of the increase in skin friction resistance on some mathematical forms from this cause, and found that it could amount to some 7 % even on fine, slender models (7). This increase in resistance, being frictional in nature, might be expected to scale with Reynolds number at a similar rate to the "plank" skin friction. Also, the transverse curvature of a ship form has an effect upon resistance. As shown by Landweber, the increase depends upon the absolute curvature, and is therefore greater on a small model than on a large model of exactly similar shape (8). In addition, some resistance is present due to eddymaking and flow separation, and this may be of significant amount on a full model. The extent of separation on the model will depend on the state of turbulence in the water as well as on the size of model and its shape. The eddy-making resistance itself will scale with Froude number and not with Reynolds.

The wave-making resistance is essentially a high Froude number or speed effect, and at low speeds with most models becomes negligible. Thus in carefully conducted experiments, where care is taken with regard to turbulence stimulation, steady speed and good surface finish, a point A (figure 2) can usually be found below which the curve of C_T is "sensibly" parallel to the two-dimensional friction line. This is called by Hughes the "low Froude number run-in" point.

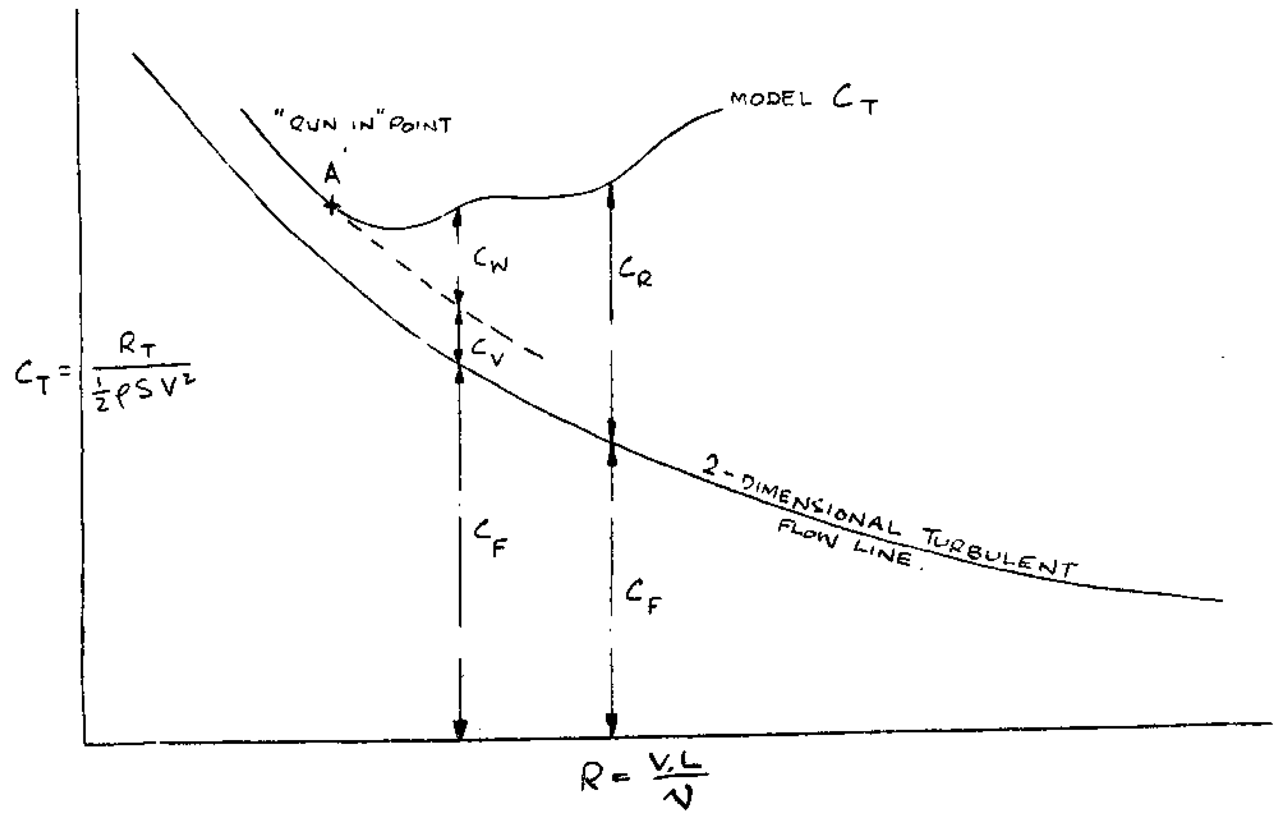
We can take our analysis a step further now by writing

$$C_T = C_F + C_V + C_W \qquad (2)$$

where

C_F = resistance of equivalente plank in 2-dimensional flow,
C_V = form drag due to shape of hull, and
C_W = wave-making resistance.

FIG 2 COMPONENTS OF C_T



As will be obvious, C_V (see figure 2) is made up of components due to additional skin friction caused by curvature effects, both in the fore-and-aft and transverse directions, to separation of flow and to eddymaking. These components cannot be separated in any clear cutway, at least with our present knowledge, since the separation point may change with speed, and the frictional and wavemaking resistances affect one another as speed and so wave formation change.

We are thus led to realise what a complicated problem this model-ship correlation really presents. Our immediate task is to derive a method for scaling the 3-dimensional viscous form drag represented by C_V in figure 2.

The Froude assumption means, in effect, that we scale all the form resistance represented by C_V with the wave-making resistance C_W, thereby assuming that $(C_V + C_W) = C_R$, remains unaltered in passing from model to ship at the same value of the Froude number $V/\sqrt{gL}$. In view of the fact that some of the resistance represented by C_V is undoubtedly of a skin friction nature scaling with Reynolds number, this assumption cannot be correct, however convenient it may prove to be in practice.

The other extreme would be to scale C_V directly with Reynolds number, on the assumption that C_V is a constant percentage of C_F for any given form, leaving only C_W to scale with Froude number. This has, in effect, been proposed by Hughes in association with his new 2-dimensional flow friction line (figure 3). The "form factor" C_V/C_F is determined from the "run-in" of the C_T curve for the model at low Froude numbers, and is assumed to apply directly to the ship. The value of the form factor is a function of the model shape, fullness, length to draft ratio, etc.

Hughes' assumption goes to the opposite extreme from Froude's, and it is evident that the truth lies somewhere between them, but there is not sufficient knowledge or data available today to take the analysis any further. It is probably true to say that most experiment tanks would desire an extended period of trial of this method before they would be willing to endorse it for universal adoption.

When in the future we eventually reach the state where we can with certainty predict the resistance of any model or ship from the results of experiments on another geometrically similar model, we shall still have a further problem. The ship results so predicted will be for a "smooth" ship, and the necessary allowances must be made for the "roughness" of the actual hull surface. This roughness is partly due to structural roughness, caused by plate edges, rivet points, weeds and shell fittings, partly to paint surface characteristics and, for ships in service, to marine fouling. This roughness allowance expressed in coefficient form is usually given the symbol ΔC_F.

Knowledge of ΔC_F can be derived in a number of ways. The results of experiments on different paints on a 21 ft. aluminum plank run at Taylor Model Basin are shown in figure 4. In general the paint roughness gives rise to a constant addition to the smooth C_F curve rather than to a constant total value of C_F, as found in sand roughness. This same

FIG. 3 DERIVATION OF ΔC_F FOR FRENCH MINESWEEPER "ALDEBARAN" USING HUGHES METHOD.

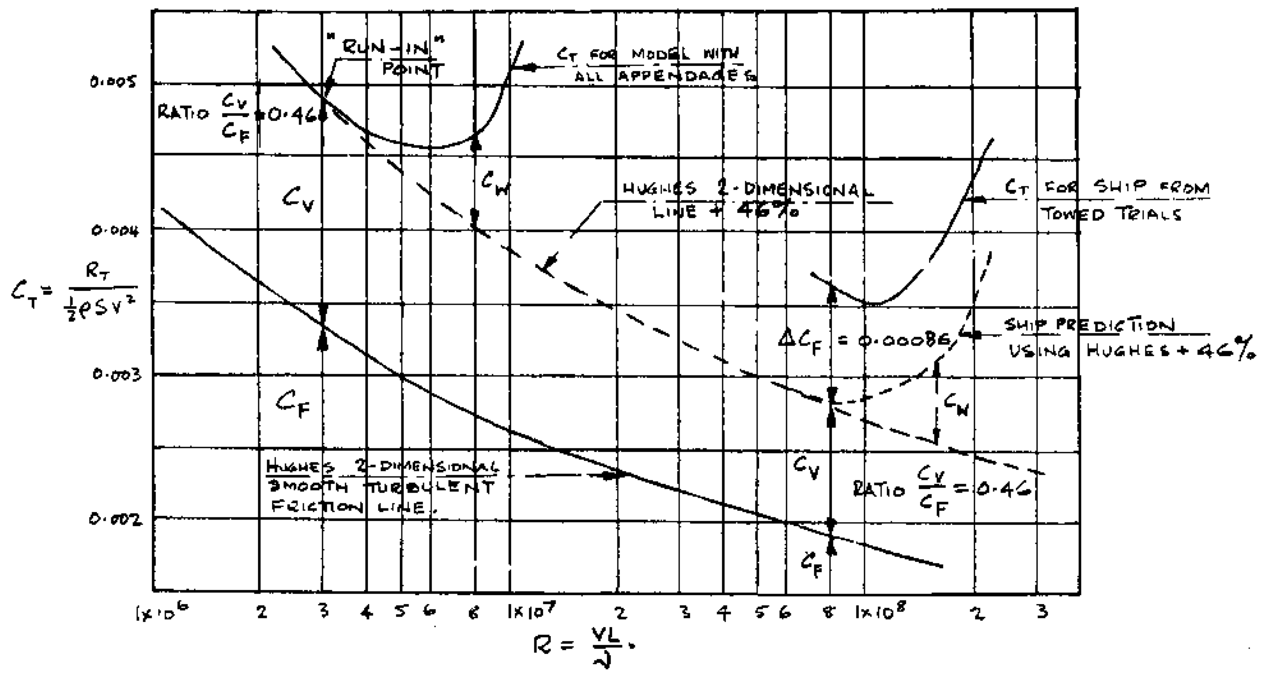
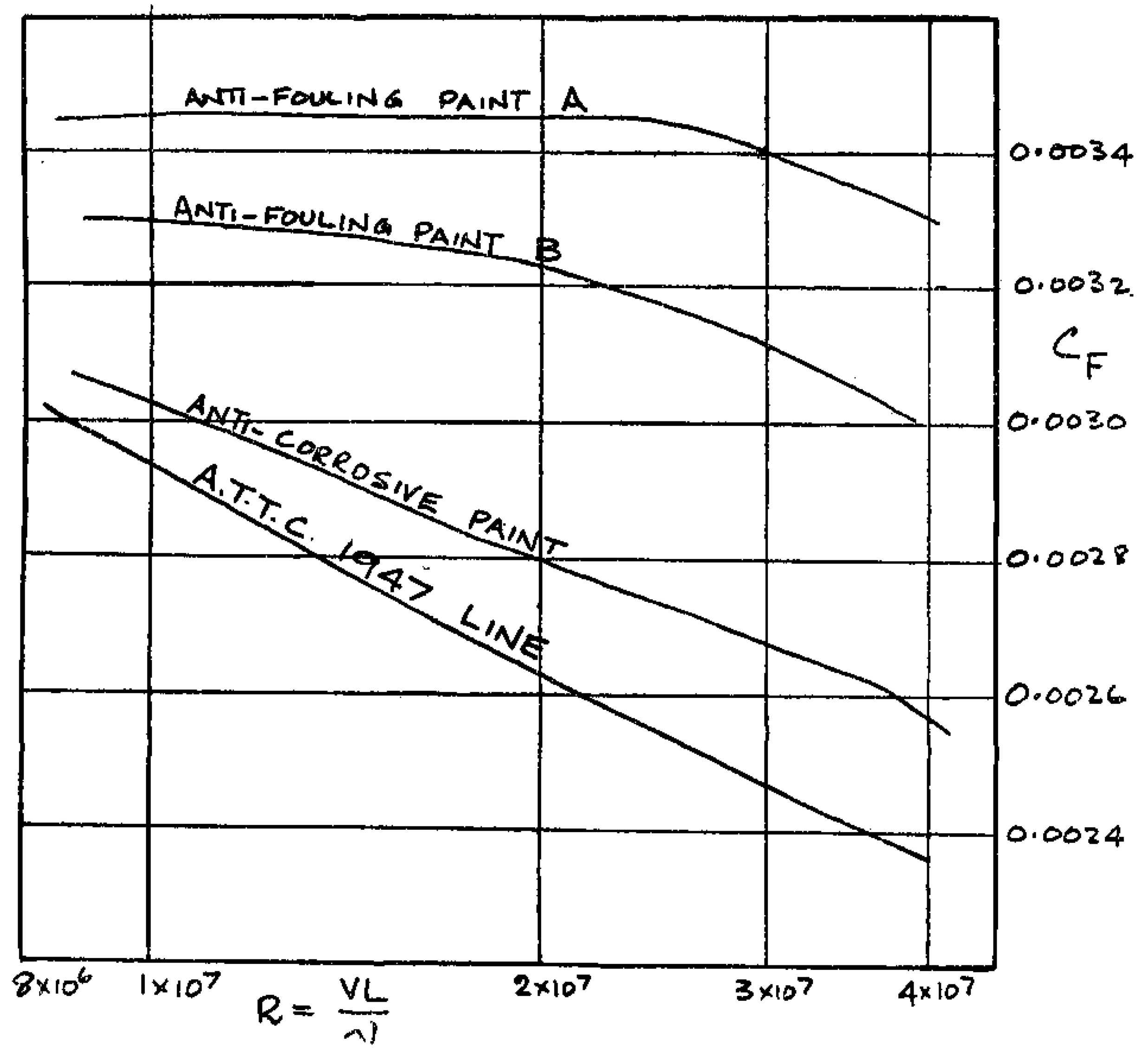


FIG. 4. FRICTIONAL RESISTANCE COEFFICIENTS FOR PAINTS — T.M.B. PLANK TESTS.



approximate constancy of ΔC_F has also been found in many full scale trial results of U. S. Navy surface ships, ΔC_F varying between + 0.0001 and + 0.0010, or about the same range as found in the plank tests (9).

Values of ΔC_F can also be derived from full scale trial results, and this indeed is the final test of our extrapolation methods. The ideal way of obtaining these full scale values of ΔC_F would be to measure the actual resistance of the ship. There are many intrinsic difficulties in doing so, and the most successful trials of this kind to date are undoubtedly those of the "Lucy Ashton" carried out by the British Shipbuilding Research Association.

As an alternative to towing trials, some information can be obtained from full scale power trials, especially when thrust is measured. If we measure the thrust deduction coefficient t on the model and assume there is no scale effect on t in passing to the full sized ship, then the resistance of the ship is

$$R_T = T\,(1 - t)$$

If we also assume that the residuary resistance coefficient C_R of the ship and model are the same, then the frictional resistance of the ship can be found from

$$R_F = R_T - R_R$$

We may well question the validity of these assumptions. Until recently no full scale measurements of t have been made, but the Wageningen Victory ship programa suggests that there may be appreciable scale effect—for example, the value of t increased from 0.22 for a 19.33 ft. model to 0.27 for

TABLE I

SHIP AND TRIAL CHARACTERISTICS

Type of ship	C-3 Cargo	C-4 Cargo Mariner	30,000 DWT tw. scr. Pass.—Cargo	28,000 DWT sgl. scr. tanker Light	28,000 DWT sgl. scr. tanker Heavy	26,000 DWT sgl. scr. tanker	"Lucy Ashton" Ex-"Clyde" paddlewheel steamer	French Naval wood minesweeper
L_{WL}	469.5'	523.3'	650.0'	608.0'		612'	190.5'	27.95'
Beam at L_{WL}	69.5'	76.0' mld.	89.0'	84.0'		82.5'	21.0' mld.	140.95'
Block Coeff.	.6540	.607	.599	.748		.752	.685	.480
Prismatic Coeff.	.6659	.619	.616	.754		.755	.705	.566
⊗ Area Coeff.	.9821	.981	.972	.993		.995	.972	.850
Hull Seams	flush welded	flush welded	lap riveted	one seam lap riveted		bilge strake only, lap riveted	sharp	faired smooth
Hull Butts	flush welded	flush welded	flush welded	flush welded		flush welded	flush	flush smooth
Bottom Paint	hot plastic	commercial	commercial AF	commercial		Red Hand copper	red oxide	alum. French Navy AF
Days out of dock	34	not available	3	3	2	8	4 3	8 to 23
Mean draft on trial	23.04'	23.99'	26.54'	24.50'	33.19'	31'-7 3/8"	5'-4"	7.054'
Trim on trial	4'-7" by stern	3.26' by stern	10" by stern	8.0' by stern	even keel	0	even keel	1.116' by stern
Displ. on trial, tons	13700	16453	26068	25945	36289	34278	390	378
Sea temp. °F	61	54.2	44	58	56	46	59	59

a 74.1 ft. model (10). Recent full scale power and towing trials on the USS *Albacore* have indicated a similar variation of *t*, the rate of increase from model to ship relative to change in Reynolds number being almost the same as in the Victory model tests.

The assumption that C_R is constant in going from model to ship at the same $V/\sqrt{L}$ value is also open to question, since as indicated earlier in this discussion some of the components of C_R should show a Reynolds number effect.

If the ship measurements do not include thrust, a comparison can still be made from the S. H. P. but this introduces still further assumptions.

Subject to all the above doubts and uncertainties, the value of C_F for the ship can be obtained and compared with the "smooth" ship value obtained from the model using one or other of the extrapolation methods proposed. The difference is the so called "roughness allowance coefficient", ΔC_F. It must be emphasised that the values of ΔC_F so obtained will depend to a great extent on the actual method of extrapolation used, and are to this extent subjective rather than objective.

Results of such an analysis are shown in figure 5, using the A. T. T. C. 1947 line and the Froude assumption. Particulars of the vessels are shown in Table 1. The average values of ΔC_F range from $+ 0.00015$ to $+ 0.00100$ and for each trial there is a considerable variation of ΔC_F over the speed range. For the U. S. merchant ships, welded construction with some riveted shell seams, painted with brands of commercial paint, the ΔC_F values vary between $+ 0.00015$ and $+ 0.0004$, the increase in frictional resistance varying from 10 to 28 % and in total resistance from 5 to 12 % (11). Similar correlations between model predictions and ship trials, also using the A. T. T. C. line, were given for a large number of British-built merchant ships in 1954 (12). For clean vessels with a shell 50 % riveted and 50 % welded, the average value of ΔC_F for 7 ships was $+ 0.00026$. For all-welded shell, 5 ships gave a negative allowance averaging $- 0.00010$ and 3 others gave positive values averaging $+ 0.00020$. The overall average for this all-welded group was practically zero. Figure 6 shows that in these cases also there was considerable variation in ΔC_F for any individual ship as the speed varied.

At first sight we might assume that these very small and even negative values of ΔC_F indicated that all-welded ship hulls with commercial-type paint had reached almost the limit of smoothness. It is known that to maintain hydraulic smoothness the roughness must decrease in actual size as we go to higher Reynolds numbers, which would imply that the ship hulls must be smoother than the models and planks on which the extrapolation methods are based. Anyone with a knowledge of actual hulls

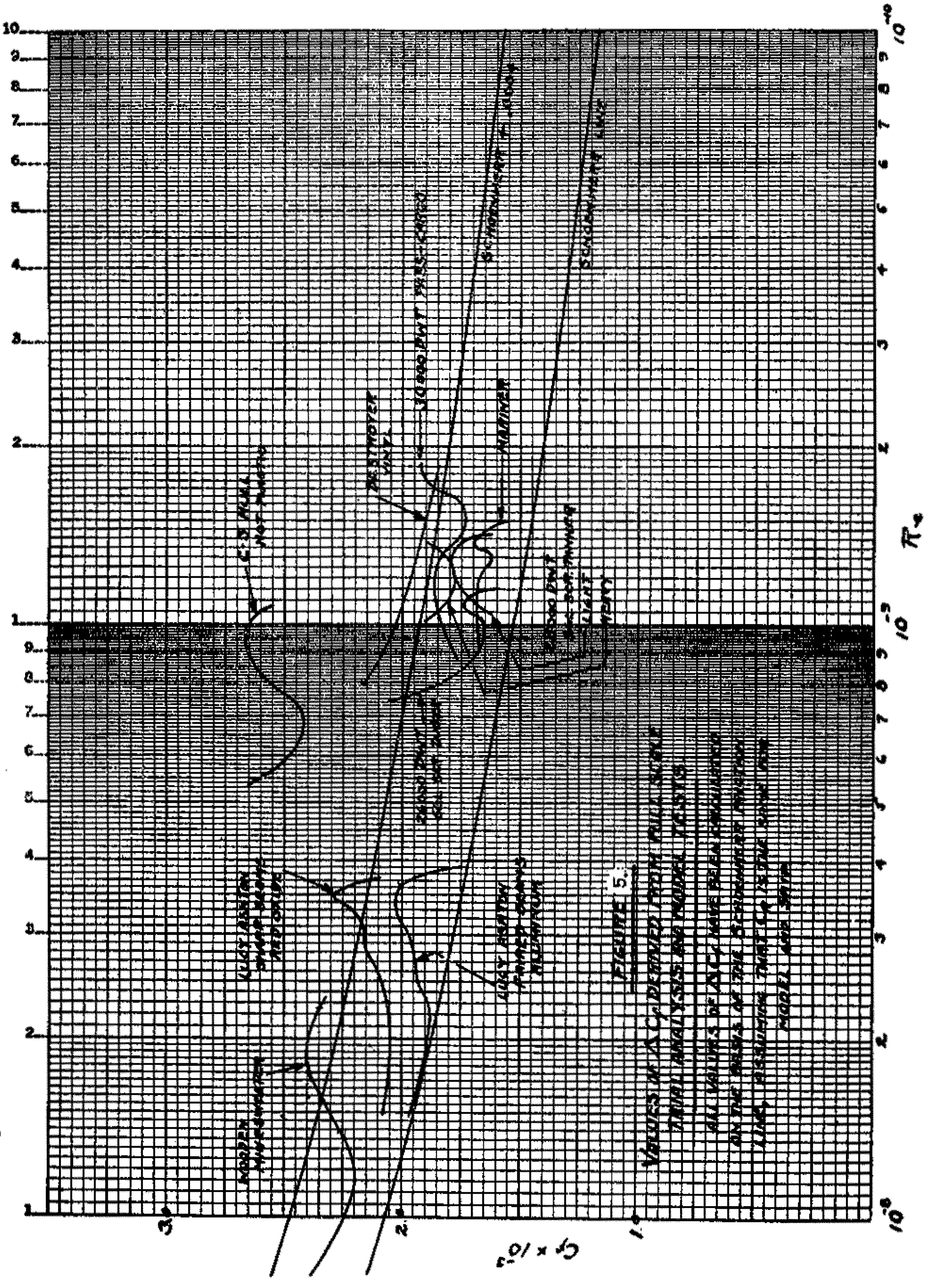
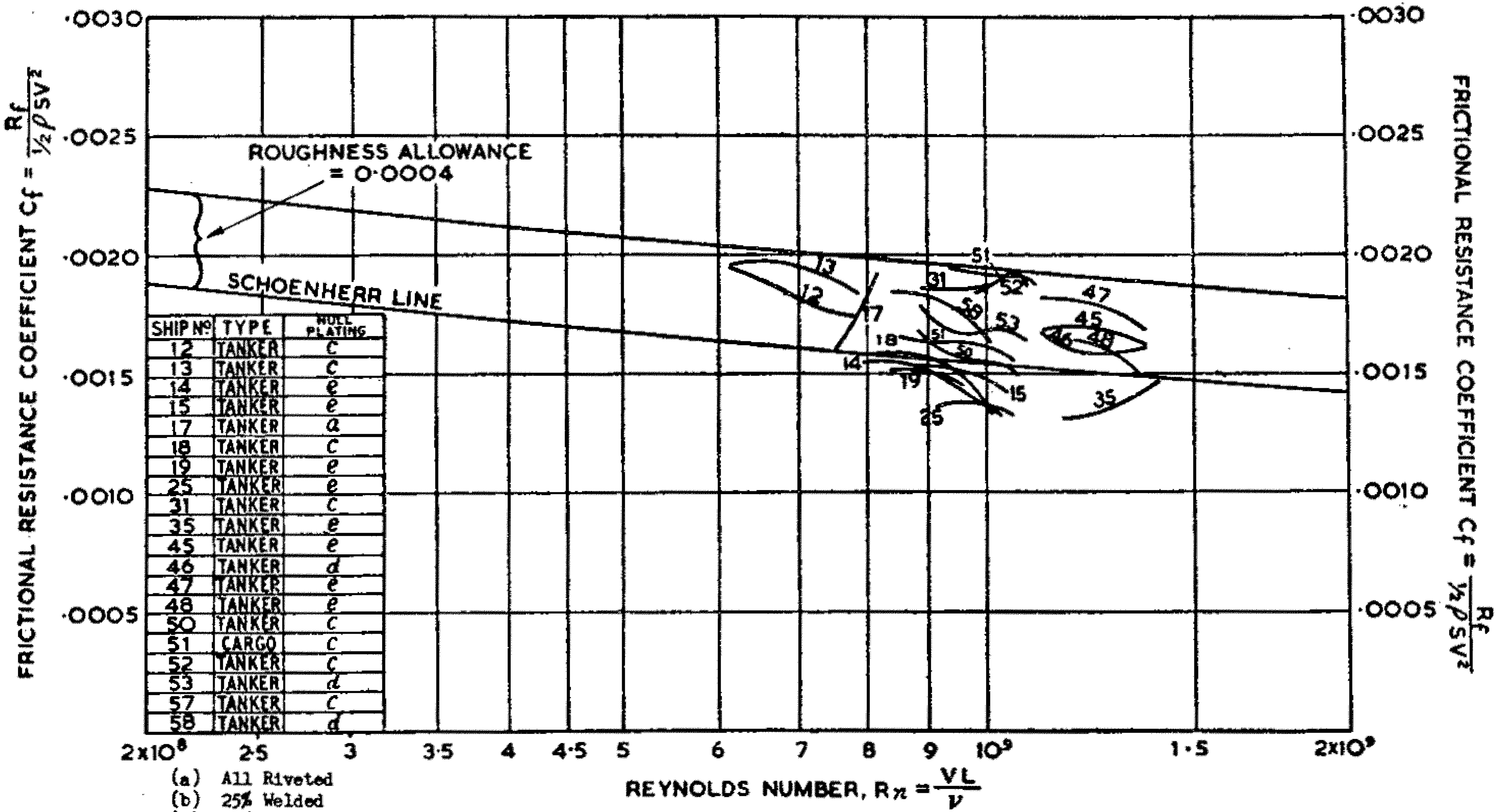


Figure 5

Figure 6



SHIP Nº	TYPE	HULL PLATING
12	TANKER	c
13	TANKER	c
14	TANKER	e
15	TANKER	e
17	TANKER	a
18	TANKER	c
19	TANKER	e
25	TANKER	e
31	TANKER	c
35	TANKER	e
45	TANKER	e
46	TANKER	d
47	TANKER	e
48	TANKER	e
50	TANKER	c
51	CARGO	c
52	TANKER	c
53	TANKER	d
57	TANKER	c
58	TANKER	d

(a) All Riveted
(b) 25% Welded
(c) 50% Welded
(d) 75% Welded
(e) 100% Welded

SHIP-MODEL CORRELATION (Less than 20 days out of dock)

C_f values necessary to give agreement between ehp deduced from power measurements and Schoenherr ehp.

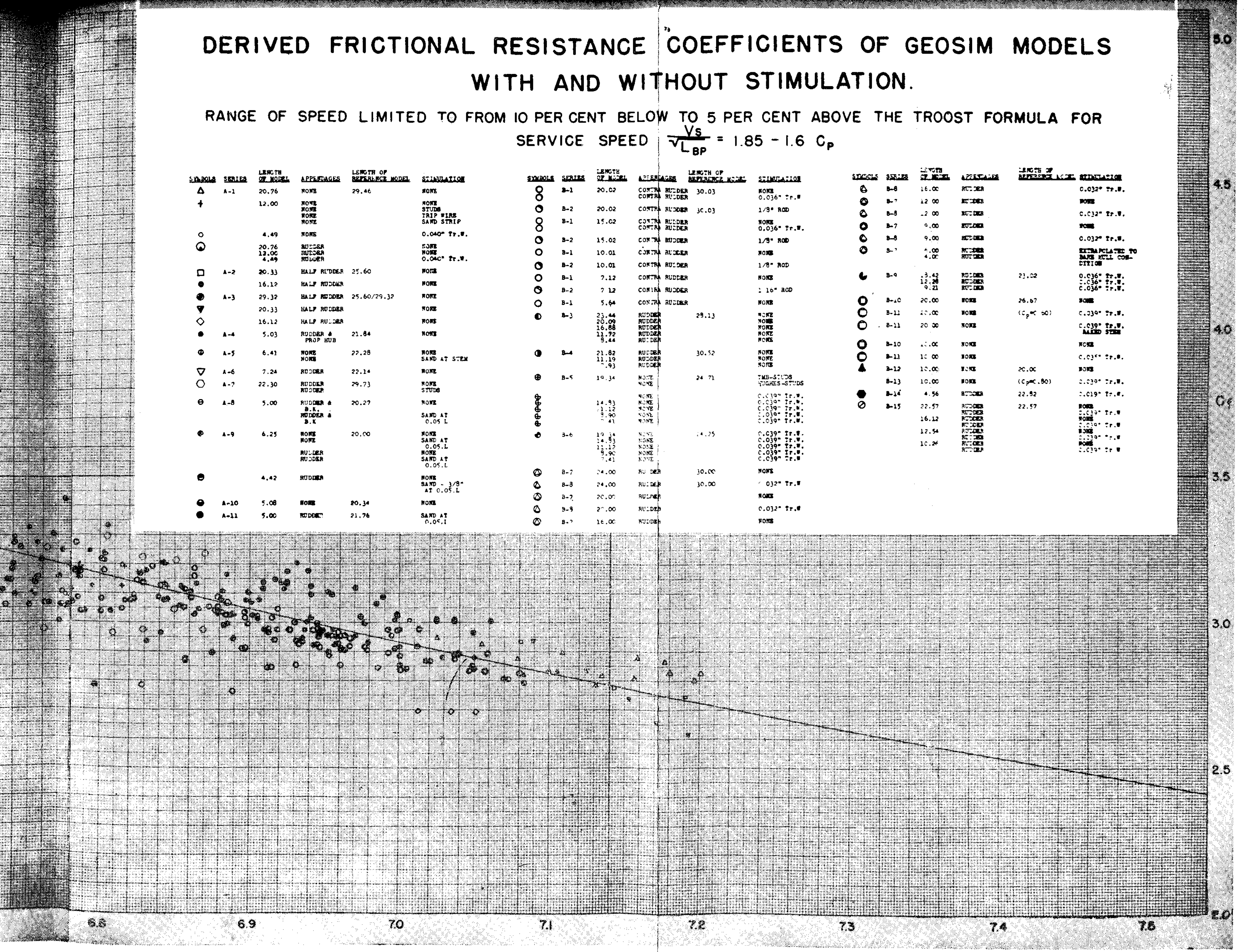
DERIVED FRICTIONAL RESISTANCE COEFFICIENTS OF GEOSIM MODELS WITH AND WITHOUT STIMULATION.

RANGE OF SPEED LIMITED TO FROM 10 PER CENT BELOW TO 5 PER CENT ABOVE THE TROOST FORMULA FOR SERVICE SPEED $\frac{V_S}{\sqrt{L_{BP}}} = 1.85 - 1.6\ C_P$

SYMBOLS	SERIES	LENGTH OF MODEL	APPENDAGES	LENGTH OF REFERENCE MODEL	STIMULATION
△	A-1	20.76	NONE	29.46	NONE
+		12.00	NONE NONE NONE NONE		NONE STUDS TRIP WIRE SAND STRIP
○		4.49	NONE		0.040" Tr.W.
◒		20.76 12.00 4.49	RUDDER RUDDER RUDDER		NONE NONE 0.040" Tr.W.
□	A-2	20.33	HALF RUDDER	25.60	NONE
●		16.12	HALF RUDDER		NONE
●	A-3	29.32	HALF RUDDER	25.60/29.32	NONE
▼		20.33	HALF RUDDER		NONE
◇		16.12	HALF RUDDER		NONE
●	A-4	5.03	RUDDER & PROP HUB	21.84	NONE
⦶	A-5	6.41	NONE NONE	22.28	NONE SAND AT STEM
▽	A-6	7.24	RUDDER	22.14	NONE
○	A-7	22.30	RUDDER RUDDER	29.73	NONE STUDS
⊖	A-8	5.00	RUDDER & B.K. RUDDER & B.K	20.27	NONE SAND AT 0.05 L
⊕	A-9	6.25	NONE NONE RUDDER RUDDER	20.00	NONE SAND AT 0.05.L NONE SAND AT 0.05.L
●		4.42	RUDDER		NONE SAND - 3/8" AT 0.05.L
●	A-10	5.08	NONE	20.34	NONE
●	A-11	5.00	RUDDER	21.76	SAND AT 0.05.L

SYMBOLS	SERIES	LENGTH OF MODEL	APPENDAGES	LENGTH OF REFERENCE MODEL	STIMULATION
○ ○	B-1	20.02	CONTRA RUDDER CONTRA RUDDER	30.03	NONE 0.036" Tr.W
○	B-2	20.02	CONTRA RUDDER	30.03	1/8" ROD
○ ○	B-1	15.02	CONTRA RUDDER CONTRA RUDDER		NONE 0.036" Tr.W.
○	B-2	15.02	CONTRA RUDDER		1/8" ROD
○	B-1	10.01	CONTRA RUDDER		NONE
○	B-2	10.01	CONTRA RUDDER		1/8" ROD
○	B-1	7.12	CONTRA RUDDER		NONE
○	B-2	7.12	CONTRA RUDDER		1/16" ROD
○	B-1	5.64	CONTRA RUDDER		NONE
◐	B-3	23.44 20.09 16.88 11.72 9.44	RUDDER RUDDER RUDDER RUDDER RUDDER	29.13	NONE NONE NONE NONE NONE
◑	B-4	21.82 11.19 7.93	RUDDER RUDDER RUDDER	30.52	NONE NONE NONE
⊕	B-5	19.34	NONE NONE	24.71	TMB-STUDS HUGHES-STUDS
⊕ ⊕ ⊕ ⊕ ⊕		14.93 11.12 9.90 7.41	NONE NONE NONE NONE NONE		0.039" Tr.W. 0.039" Tr.W. 0.039" Tr.W. 0.039" Tr.W. 0.039" Tr.W.
●	B-6	19.34 14.93 11.12 9.90 7.41	NONE NONE NONE NONE NONE	24.25	0.039" Tr.W. 0.039" Tr.W. 0.039" Tr.W. 0.039" Tr.W. 0.039" Tr.W
○	B-7	24.00	RUDDER	30.00	NONE
○	B-8	24.00	RUDDER	30.00	0.032" Tr.W
○	B-7	20.00	RUDDER		NONE
○	B-8	20.00	RUDDER		0.032" Tr.W
○	B-7	16.00	RUDDER		NONE

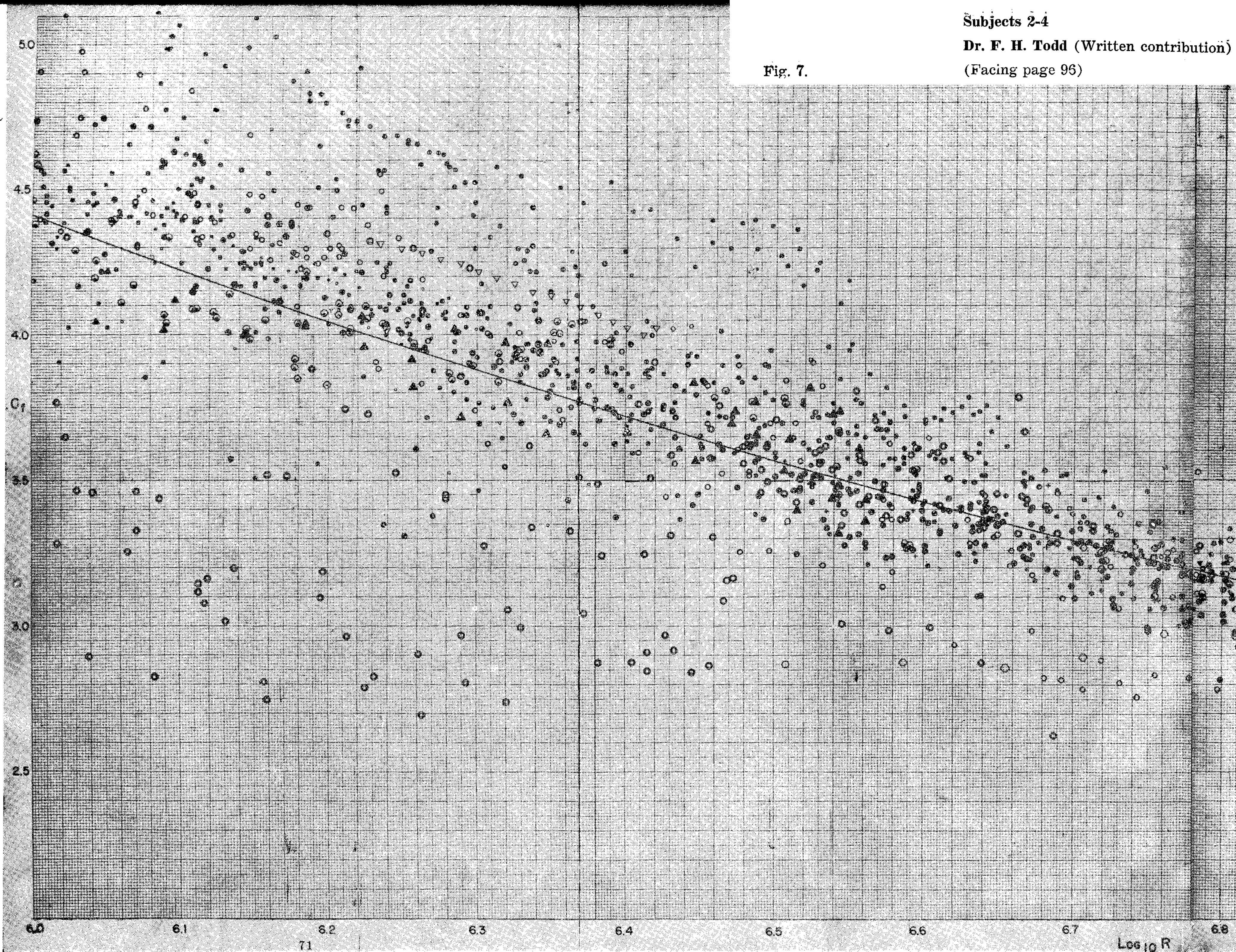
SYMBOLS	SERIES	LENGTH OF MODEL	APPENDAGES	LENGTH OF REFERENCE MODEL	STIMULATION
○	B-8	16.00	RUDDER		0.032" Tr.W.
○	B-7	12.00	RUDDER		NONE
○	B-8	12.00	RUDDER		0.032" Tr.W.
○	B-7	9.00	RUDDER		NONE
○	B-8	9.00	RUDDER		0.032" Tr.W.
○	B-7	6.00 4.00	RUDDER RUDDER		EXTRAPOLATED TO BARE HULL CONDITION
◔	B-9	13.42 12.28 9.21	RUDDER RUDDER RUDDER	21.02	0.036" Tr.W. 0.036" Tr.W. 0.036" Tr.W.
○	B-10	20.00	NONE	26.67	NONE
○	B-11	20.00	NONE	(C_P=0.60)	0.039" Tr.W.
○	B-11	20.00	NONE		0.039" Tr.W. RAKED STEM
○	B-10	10.00	NONE		NONE
○	B-11	10.00	NONE		0.039" Tr.W.
▲	B-12	10.00	NONE	20.00	NONE
	B-13	10.00	NONE	(C_P=0.80)	0.039" Tr.W.
⬢	B-14	4.56	RUDDER	22.52	0.019" Tr.W.
⊘	B-15	22.57 16.12 12.54 10.24	RUDDER RUDDER RUDDER RUDDER RUDDER RUDDER RUDDER RUDDER	22.57	NONE 0.039" Tr.W NONE 0.039" Tr.W NONE 0.039" Tr.W NONE 0.039" Tr.W



Dr. F. H. Todd (Written contribution)

(Facing page 96)

Fig. 7.

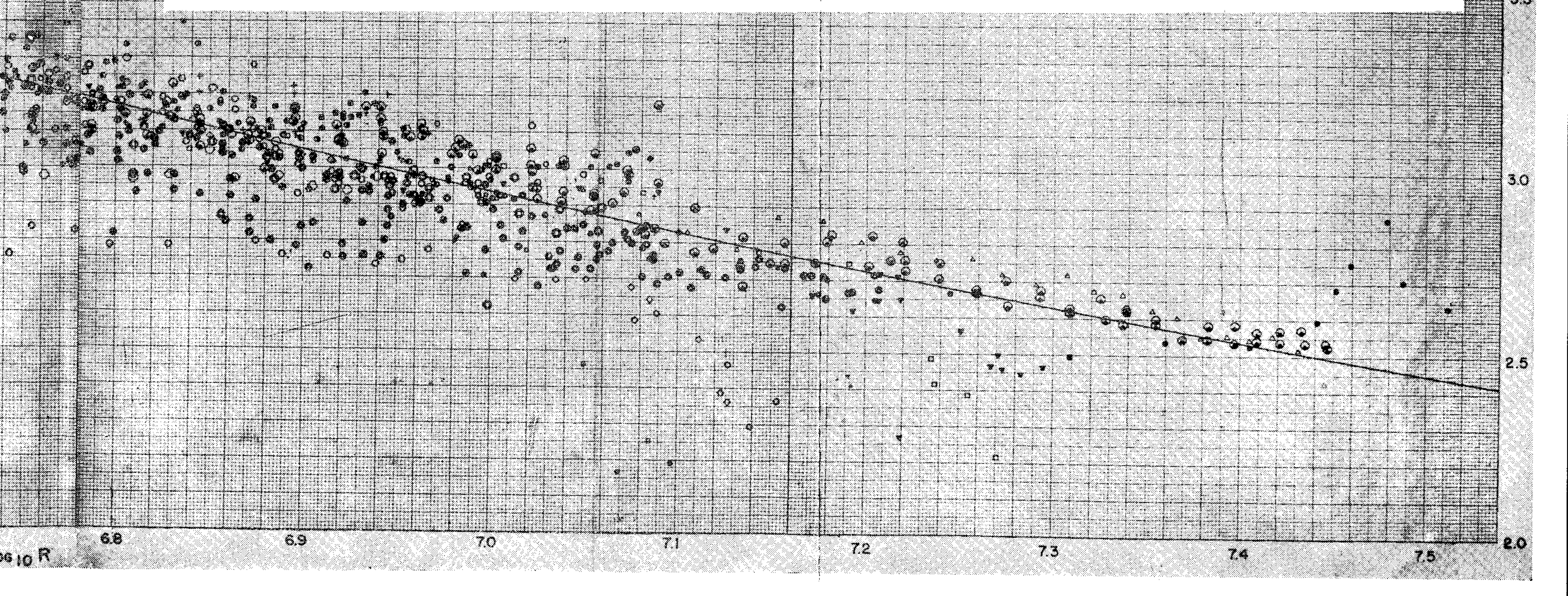


DERIVED FRICTIONAL RESISTANCE COEFFICIENTS OF GEOSIM MODELS

SYMBOL	SERIES	LENGTH OF MODEL	APPENDAGES	LENGTH OF REFERENCE MODEL	STIMULATION
△	A - 1	20.76	NONE	29.46	NONE
+	.....	12.00	NONE	.....	NONE
	.....	.....	....	.....	STUDS
	.....	.....	....	.....	TRIP WIRE
	.....	.....	.....	.....	SAND STRIP
○	.....	4.49	NONE	.....	0.040" TRIP WIRE
◬	.....	20.76	RUDDER	.....	NONE
	.....	12.00	RUDDER	.....	NONE
	.....	4.49	RUDDER	.....	0.040" TRIP WIRE
□	A - 2	20.33	1/2 RUDDER	25.60	NONE
●	.....	16.12	1/2 RUDDER	.....	NONE
●	A - 3	29.32	1/2 RUDDER	25.60/29.32	NONE
▼	.....	20.33	1/2 RUDDER	.....	NONE
◇	.....	16.12	1/2 RUDDER	.....	NONE
•	A - 4	5.03	RUDDER AND PROP. HUB	21.84	NONE
⦶	A - 5	6.41	NONE	22.28	NONE
	.....	6.41	NONE	22.28	SAND 1/2" AT STEM
▽	A - 6	7.24	RUDDER	22.14	NONE
⬡	A - 7	22.30	RUDDER	29.73	NONE
	.....	22.30	RUDDER	29.73	STUDS
⊖	A - 8	5.00	RUDDER & BILGE KEEL	20.27	NONE
	.....	5.00	RUDDER & BILGE KEEL	20.27	SAND 1/2" AT 0.05·L
⊗	A - 9	6.25	NONE	20.00	NONE
	.....	6.25	NONE	20.00	SAND 1/2" AT 0.05·L
	.....	....	RUDDER	.....	NONE
	.....	....	RUDDER	.....	SAND 1/2" AT 0.05·L
⊖	.....	4.42	RUDDER	.....	NONE
	.....	4.42	RUDDER	.....	SAND 3/8" AT 0.05·L
⊖	A - 10	5.08	NONE	20.34	NONE
⊗	A - 11	5.00	RUDDER	21.76	SAND 1/2" AT 0.05·L

SYMBOL	SERIES	LENGTH OF MODEL	APPENDAGES	LENGTH OF REFERENCE MODEL	STIMULATION
○	B - 1	20.02	CONTRA RUDDER	30.03	NONE
	.....	20.02	CONTRA RUDDER	30.03	0.036" TRIP WIRE
○	B - 2	20.02	CONTRA RUDDER	30.03	1/8" ROD
	B - 1	15.02	CONTRA RUDDER	.....	NONE
	.....	.....	....	.....	0.036" TRIP WIRE
	B - 2	15.02	CONTRA RUDDER	.....	1/8" ROD
	B - 1	10.01	CONTRA RUDDER	.....	NONE
	B - 2	10.01	CONTRA RUDDER	.....	1/8" ROD
	B - 1	7.12	CONTRA RUDDER	.....	NONE
	B - 2	7.12	CONTRA RUDDER	.....	1/16" ROD
	B - 1	5.64	CONTRA RUDDER	.....	NONE
◐	B - 3	23.44	RUDDER	28.13	NONE
	.....	20.00	RUDDER	.....	NONE
	.....	16.88	RUDDER	.....	NONE
	.....	11.72	RUDDER	.....	NONE
	.....	8.44	RUDDER	.....	NONE
◑	B - 4	21.82	RUDDER	30.52	NONE
	.....	11.19	RUDDER	.....	NONE
	.....	7.93	RUDDER	.....	NONE
⊕	B - 5	19.34	NONE	24.71	0.039" TRIP WIRE
	.....	.....	....	.....	T M B - STUDS
	.....	.....	....	.....	HUGHES - STUDS
	.....	14.83	NONE	.....	0.039" TRIP WIRE
	.....	11.12	NONE	.....	0.039" TRIP WIRE
	.....	8.90	NONE	.....	0.039" TRIP WIRE
	.....	7.41	NONE	.....	0.039" TRIP WIRE
⊕	B - 6	19.34	NONE	24.25	0.039" TRIP WIRE
	.....	14.83	NONE	.....	0.039" TRIP WIRE
	.....	11.12	NONE	.....	0.039" TRIP WIRE
	.....	8.90	NONE	.....	0.039" TRIP WIRE
	.....	7.41	NONE	.....	0.039" TRIP WIRE

SYMBOL	SERIES	LENGTH OF MODEL	APPENDAGES	LENGTH OF REFERENCE MODEL	STIMULATION
⊖	B - 7	24.00	RUDDER	30.00	NONE
⊘	B - 8	24.00	RUDDER	30.00	0.032" TRIP WIRE
	B - 7	20.00	RUDDER	.....	NONE
	B - 8	20.00	RUDDER	.....	0.032" TRIP WIRE
	B - 7	16.00	RUDDER	.....	NONE
	B - 8	16.00	RUDDER	.....	0.032" TRIP WIRE
	B - 7	12.00	RUDDER	.....	NONE
	B - 8	12.00	RUDDER	.....	0.032" TRIP WIRE
	B - 7	9.00	RUDDER	.....	NONE
	B - 8	9.00	RUDDER	.....	0.032" TRIP WIRE
	B - 7	6.00	RUDDER	.....	EXTRAPOLATED TO BARE HULL CONDITION.
	B - 7	4.00	RUDDER	.....	EXTRAPOLATED TO BARE HULL CONDITION.
●	B - 9	15.42	RUDDER	23.02	0.036" TRIP WIRE
	.....	12.28	RUDDER	.....	0.036" TRIP WIRE
	.....	9.21	RUDDER	....	0.036" TRIP WIRE
O	B - 10	20.00	NONE	26.67 (C_p=0.60)	NONE
O	B - 11	20.00	NONE	26.67 (C_p=0.60)	0.039" TRIP WIRE
	B - 11	20.00	NONE	....	0.039" TRIP WIRE (RAKED STEM)
	B - 10	10.00	NONE	....	NONE
	B - 11	10.00	NONE	....	0.039" TRIP WIRE
▲	B - 12	10.00	NONE	20.00 (C_p=0.80)	NONE
	B - 13	10.00	NONE	20.00 (C_p=0.80)	0.039" TRIP WIRE
⬢	B - 14	4.56	RUDDER	22.82	0.019" TRIP WIRE
⊘	B - 15	22.57	RUDDER	22.57	NONE
	......	.....	......	.....	0.039" TRIP WIRE
	......	16.12	......	.....	NONE
	......	.....	......	.....	0.039" TRIP WIRE
	......	12.54	......	.....	NONE
	......	.....	......	.....	0.039" TRIP WIRE
	......	10.26	......	....	NONE



Subjects 2-4

Dr. F. H. Todd (Written contribution)

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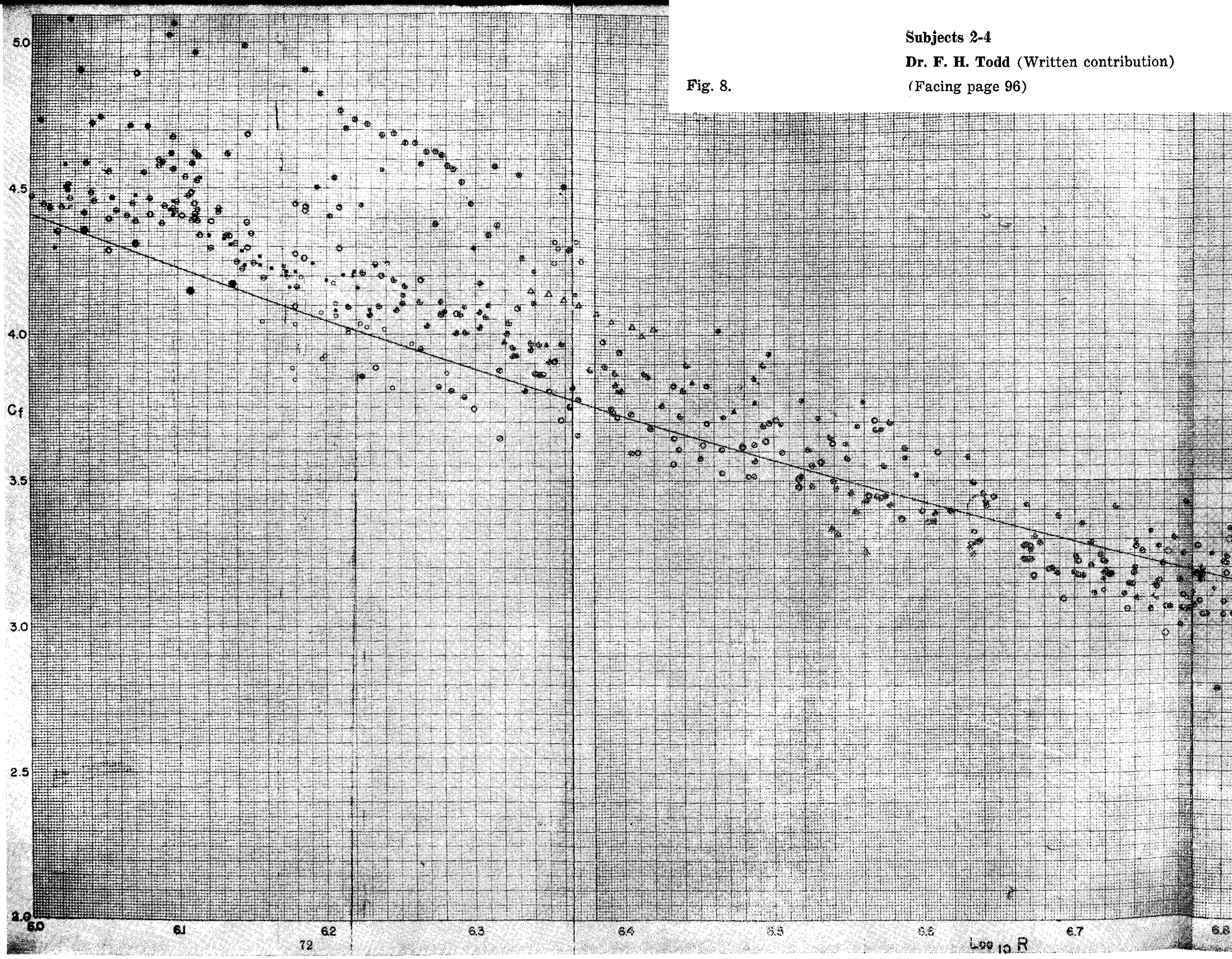
Fig. 8.



would certainly not subscribe to such a conclusion. The truth is that ΔC_F, apart from possible errors from scale effect and other assumptions, depends to a large degree on the method of extrapolation used, as stated above. Results of some recent trials with French minesweepers illustrate this fact (13). Taking the results at a speed of 6 knots, and using first the Froude assumption with the A. T. T. C. line, and then the Hughes proposal that all the resistance represented by C_V scales with C_F, using in turn the A. T. T. C. line and then the Hughes line, the values of ΔC_F are + 0.00015, + 0.00060 and + 0.00064 respectively, the form factors C_V/C_F being 0.36 and 0.46 in the two latter cases. The first two values of ΔC_F represent increases of 7.3 % and 29.2 % above the smooth A. T. T. C. line. While there is thus apparently only a relatively small gain to be expected by further attention to the smoothness of the hull if we compare ship and model on the Froude assumption, if the comparison is made on the basis that C_V/C_F is constant for a given shape of hull, then the possibilities of improvement appear much higher.

We thus see that ΔC_F, while it must include any necessary allowance for roughness, is essentially a correlating factor to balance the predicted "smooth" ship resistance against that of the actual ship. For a successful method of prediction in practice, it is necessary for each establishment to compare actual ship and model prediction data, however these latter may be calculated, and so to build up sufficient knowledge of the values of this correlating factor ΔC_F for different types of ship and conditions of surface that estimates of power for new designs may be made with confidence.

What of the future?

The I. T. T. C. at its meeting in Scandinavia asked its Skin Friction Committee to prepare a proposal for consideration at this Conference for a "frictional formulation based on Reynolds number" which would be "adequate for practical ship design purposes".

The first requirement rules out the use of the Froude coefficients. The other friction formulation at present accepted by the Conference for the publication of model data is the A. T. T. C. 1947 line, originally due to Schoenherr. A perusal of the discussion at the Conference in Scandinavia suggests that opinion there was divided between those who, in view of the large amount of research presently in progress in this field favoured the continued use of the A. T. T. C. line until more knowledge and experience had been gained of the new proposed methods, and those who believed that the time was still far distant when any finality would be reached and that some new single line could be found now which would be an improvement on the A. T. T. C. line and by inference still be based upon the Froude assumption.

What must be looked for in any such line?

It must be confessed that at present we cannot with certainty predict the resistance of one model from the results of another model on a different scale, where no question of roughness obscures the issue. This, then, would seem to be *one requirement we must look for in any new line—that it must lead to better correlation between families of geosims than does any existing line.*

It has been stated in the past by several experimenters that the values of C_V for geosim models, analysed by using the A. T. T. C. line, generally become smaller with increasing size. If this is so, and the extrapolation method is to be based upon the Froude assumption ($C_R = C_V + C_W =$ constant), then a line somewhat steeper than the A. T. T. C. line would be required over the range of small model Reynolds numbers—say for R_n ranging from 5×10^5 to 5×10^6. For example, it has been indicated that the "Lucy Ashton" geosim models can be reasonably well correlated by using the A. T. T. C. line with an addition of 8 % over the entire range of R_n—this has the effect, of course, of increasing the slope of the basic A. T. T. C. line. On the other hand, Murray has shown that the geosim pairs for a number of models run at the Stevens Institute Tank and at Taylor Model Basin correlate very well using the original A. T. T. C. line (14). Discussing the results of geosim tests on models of the Victory Ship, ranging in size from 2.728 feet to 72.75 feet length, van Lammeren has stated that the "Schoenherr mean line is in agreement with the results of the whole model family" and that "William Froude's original assumption that the specific residuary resistance in the non-wavemaking region is constant seems, therefore, to be justified for the Victory model family" (15). An analysis by Gertler of a large number of resistance experiments on a family of bodies of revolution, covering a wide range of length-diameter ratios, and run deeply submerged so as to be free of wave making, has shown that the resistance curves are parallel to the A. T. T. C. line up to Reynolds numbers of the order of 2.6×10^7 (16).

It will be seen, therefore, that there is a considerable amount of evidence to support the A. T. T. C. line as a practical one for use in the correlation of geosim models if we are looking for a single line extrapolator.

If we waive this latter requirement, then we have to consider the proposals which have been made on the basis that each model shape requires a unique extrapolator. The principal advocates of this approach are Telfer, Lap, Troost and Hughes.

Telfer has worked for many years on this problem, and believes that the results of any geosim series can be correlated by a straight line when C_T is protted against $R_n^{-1/3}$. Lap-Troost, in their formulation, state that the correlation can be obtained by variations in the abscissa when C_T is plotted against log R_n, a term in the formula being of the form of log A, where A is a coefficient—depending upon the actual hull form. Hughes advocates the use of form factors, C_V/C_F, which will be constant for any given form.

In theory, it should be possible to distinguish between these different proposals by applying them in turn to the existing geosim data. *In practice,* this is extremely difficult. At the low $V/\sqrt{L}$ speeds at which wavemaking is absent, it is difficult to measure the resistance accurately in the first place, and therefore to determine the correct slope of the model curves in this region—what Hughes calls the "run-in" point. At the lower end of the curve, also, the results are often suspect for lack of adequate turbulence stimulation, while at the higher end interference effects between the model and the sides and bottom of the tank enter into the problem. Thus in general the results over only a very limited speed range for any one model are available for such correlation analysis.

Despite these practical difficulties, the International Committee has made a determined effort to glean any information possible from existing geosim data. Most of the material available has been tabulated at Taylor Model Basin, and a cooperative effort made to analyse it by the various proposed methods. At the request of the Committee, Hughes, Lap and Telfer undertook to do their own analysis by their respective methods, and will doubtless present any results they may wish to the Conference. At Taylor Model Basin the analysis was confined to the use of the A. T. T. C. line, and the results were presented in detail by Hinterthan to the A. T. T. C. in Washington in 1956 (17).

The analysis was carried out in the same way as that done by Murray (14)—the C_R values for the largest model of the series were assumed to be the most nearly correct, and then substracted, at the appropiate $V/\sqrt{L}$ values, from the C_T values for the smaller models, so obtaining values of C_F for these smaller models. The analysis covered some 26 geosim series (including pairs of models). The results showing all the derived spots are shown in fig. 7 (*). There is evidence that many of the models were affected by laminar flow, and in order to eliminate some of this effect and also of wall effect at high speeds, a second plot was made to include only the speed range from 10 percent below to 5 percent above the service speed, the latter being defined by Troost's formula (18),

$$\frac{V_s}{\sqrt{L_{BP}}} = 1.85 - 1.6\ C_p \qquad (3)$$

This plot is shown in fig. 8 (*). The results of this analysis indicate that greater care is necessary in the testing of the models to eliminate various inaccuracies in measurement, turbulence problems and wall effects. In general, the overall picture suggests the need for a correlation line somewhat steeper than the A. T. T. C. line over the lower range of Reynolds numbers, say below $\log_{10} R_n = 6.6$.

Over the ship range of Reynolds numbers, no basic experimental data are available to determine the smooth turbulent curve. As has been pointed out above, full scale trial data only lead to values of an allowance factor $\Delta\ C_F$, which must include not only the undoubted effects of roughness and fouling on the ship, but also all the errors in the scaling laws applied to the model data in order to obtain the smooth ship prediction. If we use the A. T. T. C. line for extrapolation to the ship and the Froude assumption of constancy of C_R, it has been shown above that trials of modern, all-welded ships lead to small, zero or even negative values of $\Delta\ C_F$. This would at first suggest that either the ship hulls are considerably smoother, in the absolute sense, than the surfaces of the models, or that the A. T. T. C. line is too high in this range of Reynolds number. The former is untenable, and the latter is only necessarily true if the Froude assumption holds. If, in fact, the value of C_R is less at ship values of R_n than at the model vlaues for a given hull form, as suggested by Hughes and Lap, then the same trial data would lead to considerably increased values of $\Delta\ C_F$, as shown above for the case of the French minesweepers. At this time we have no means of separating these various effects, and whatever method of extrapolation is used to predict the *smooth* ship resistance from that of the *smooth* model, the final step to the *actual* ship can only be taken by the use of an empirical allowance of the type $\Delta\ C_F$. If this factor is called, for convenience, a "roughness allowance", it is well to remember that the name covers a multitude of sins in the form of our ignorance of the fundamentals of the problem.

At present, the results of all model resistance tests published in reports and papers are either based upon the Froude or A. T. T. C. formulations, and the labour entailed in converting one to the other is very great. Often it is never done, and large amounts of information are lost to the busy naval architect dealing with resistance problems.

In view of the many areas in which our knowledge is still so imperfect, and of the large amount of research presently being carried out in the fields of

(*) Enclosed in the pocket attached to the rear cover of the book.

laminar and turbulent boundary layers, roughness codification and geosim analyses, both by theoretical and experimental methods, no finality is in sight at this time. However, it would be a major step forward if all establishments could agree on a single procedure, so that at least new published data would be at once comparable. At the same time, if the proposed procedure is not one of the two already accepted, then it will need to have some strong claim to temporal permanence for a number of years ahead before the different experiment tanks are likely to view with enthusiasm the change to a new system and the consequent difficulties of assimilating the results into their already considerable archives.

The opinions of the various towing tanks as given in Appendix II of the Skin Friction Committee's report show that there is a substantial feeling that the time is not yet here to make any change, and that of the two present methods the A. T. T. C. 1947 line is acceptable to many establishments. This feeling, so far as North America is concerned, was reflected by the action of the American Towing Tank Conference at its meeting in September, 1956. After much discussion, ranging over all aspects of the subject, the Conference unanimously adopted a resolution put forward by its Skin Friction Committee, under the Chairmanship of Dr. L. Landweber, to the effect that "the Committee has reviewed the present state of knowledge of the subject and in view of the great amount of work going on in this field feels that the time is inopportune to make any departure from the decision to use the 1947 A. T. T. C. line".

The Taylor Model Basin is in full agreement with the above resolution, and no change in our present practice is desired at this time. The whole problem has been carefully reviewed at the Basin over the last three years and the results of many geosim tests studied. The general conclusions from this survey were well set out by Granville in a recent paper (19), where he stated that:

> "1. The basic Froude hypothesis concerning the separation of viscous and wavemaking resistances is justified.
>
> 2. The data are not accurate enough to establish the best procedure of extrapolating forma or viscous resistance.
>
> 3. It is imperative that a full-scale ship with a hydrodynamically smooth hull be tested.
>
> 4. Additional research is needed into basin-wall effects on the large models and the stimulation of turbulent flow on the small models; the position of transition on the model to be experimentally determined so as to eliminate any doubt on this score.
>
> The use of a flat-plate line as a reference line for form or viscous resistance still seems a fruitful method of inquiry. The concept of the form—or viscous—resistance coefficient is inclusive enough to encompass any probable variations. Finally, more accuracy is needed in the experimental procedures before any final decision as to the correct method of extrapolating model results to ship predictions can be reached."

The Taylor Model Basin has been using the A. T. T. C. line since its adoption in 1947. The surface ship models tested for resistance and propulsion are normally between 20 and 30 feet in length. The ship results are predicted using the A. T. T. C. line, and from the full-scale trials values of ΔC_F have been obtained for a great variety of types of ships having different types of structural roughness and different paint surfaces. Bearing in mind that ΔC_F is in fact a "correlating factor" and not a roughness allowance, we are well satisfied with the present system as an engineering method, though fully aware of its deficiencies in the fundamental sense. We have accumulated a vast amount of data over the last ten years, and have no desire to re-analyse these and obtain new "correlating factors" for future use unless assured of no further change for a number of years. In view of the known limitations of the Froude assumption and the use of a single line extrapolator and all the new work in progress on form effect, any such assurance seems impossible today. Also, any method involving the use of form factors must necessarily go through a trial period before being adopted universally. We would require considerable experience with any system using the method of "run-in" values of C_F to determine the form factor before being willing to change to such a system.

At the same time, the Model Basin is aware of the difficulties of the extrapolation of results from small models, and would be prepared to go along with some steepening of the extrapolation line in this range of Reynolds numbers should the Conference so recommend. In this respect, we would therefore favour the first of the proposals made by the Committee in its report—namely, the retention of the A. T. T. C. line above $R_n = 10^7$ and a steeper line below this point. This would leave most of our model-ship correlations unaltered, and assist the problems of the small basins. Actually, we would prefer to keep the A. T. T. C. line for all values of R_n above 5×10^6, since some 10 % of our model results lie in the range between $R_n = 6 \times 10^6$ and $R_n = 10^7$.

In brief, the opinions of Taylor Model Basin may be stated as follows:

1. We prefer to continue using the A.T.T.C. 1947 line for the present.
2. If the Conference should decide to adopt the first of the Committee's proposals to use the A. T. T. C. line above $R_n = 10^7$—we would abide by such a decision in the interests of international agreements.
3. We would agree to any reasonable steepening of the A. T. T. C. line below $R_n = 10^7$. The amount suggested by the Committee seems reasonable, but this should be largely a question for the small model users to decide.
4. The formula for any new line should preferable be an explicit one, for use on high-speed, electronic computers.
5. The name should be "the I. T. T. C. 1957 model-ship correlation line", and there should be no implication that it professes to be a skin friction resistance line, either for flat plates in 2-dimensional flow or for any average hull form with an average form factor included.

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Mr. J. Hadler (Oral remarks).

I should just like, at this time, to clarify the position of the David Taylor Model Basin on the line to be used for Skin Friction extrapolation. Our position is essentially that expressed by Dr. Todd in Appendix II of the Committee's report. We feel that the suggestion of proposal 1 is preferable to us on the assumption that the small model basins wish to steepen the low end of the fricction line. But it should be expressed by an explicit formula for ease in using a high speed computing machine:

$$\frac{1}{\sqrt{c_F}} = A + B \lg R_n + C (\lg R_n)^2 + D (\lg R_n)^3$$

$$A = -\ 64.7493$$
$$B = +\ 26.1766$$
$$C = -\ \ 2.9036$$
$$D = +\ \ 0.12333$$

Dr. F. H. Todd (Oral remarks).

My statement is very brief. If this Conference finds itself prepared to accept proposal number 1 of the Skin Friction Committee or some line closely approximating to it for use in all published work, then the British Delegates are prepared to support such a position.

In addition, we feel that any such line should not be associated with anyone's name but as, in the case of the American Towing Tank Conference, it should be known simply as a Conference line.

Capt. M. L. Acevedo (Written contribution).

1.—Introduction.

Model-ship correlation, the fundamental problem in Model-Towing Tanks, has deserved particular and increasing attention during the past few years. Among the variety of reasons for this growing interest, the most pressing has undoubtedly been that the allowances found between ship trial results and their corresponding model predictions have been too small (or, at times, even negative) so that they have allowed no room for a reasonable roughness effect of the actual ship. Probably this abnormal occurrence, repeatedly registered in sea trials of recent years, is mainly a direct consequence of an improper estimation of the viscous resistance in the present Model Tank procedures, the revision of which has, therefore, become absolutely necessary.

Such an important question as the viscous resistance naturally has been one of the most prevalent considerations since the early Conferences of Ship Tank Superintendents held before World War Two [1a] to [1e] (''); and it was again included in the agenda of the post War Conferences [2] to [4]. Before the War, however, the question which was primarily discussed was that of two-dimensional frictional resistance. Although the influence of form on this resistance was mentioned at the earliest Conferences and a special section was even devoted to it at the Berlin meeting (1937), attention was preponderantly focussed on the substitution of R. E. Froude's formula and frictional coefficients by others more rational according to the formulation of the mechanical similitude.

Then, there was a rather quiet period during which the question of basic frictional resistance seemed to be settled, since the two or three best qualified formulae had led to final values which practically coincided. In such circumstances, the decision of departing from R. E. Froude was mainly delayed by the weight of the archives and respect for tradition. And, of course, no hastening fact, like the one existing at present, was pressing for the change at that time. Nevertheless, some Tanks did decide to change, and this was undoubtedly an important step, at least as a formal reconciliation with the law of similitude.

To make a decision at present, however, is much more difficult. On the one hand, new basic frictional formulae have been proposed, among them that of Hughes, which, supported by an ample experimental basis, gives values of frictional coefficient c_f appreciably lower than any previous ones. On the other hand, the question of resistances due to viscosity is no longer considered separately, but in connection with the correlation problem as a whole. And, finally, new correlation methods have been proposed to substitute W. Froude's old correlation. Evidently, the problem has become much more complex and the decision consequently more difficult. But, at the same time, there is no doubt that by considering the whole problem, we are on the way to a more perfect solution of it.

Present contribution is a critical review of the diverse questions involved, establishing comparisons in general, but concrete, terms, between the different proposed basic friction formulae and correlation methods. As a result of this analysis and by taking into account some physical reasons generally admitted in connection with these questions, various provisional conclusions are reached.

* * *

In presenting the question as it stands to-day, we can sum it up by saying that two successive steps have been scheduled for its solution:

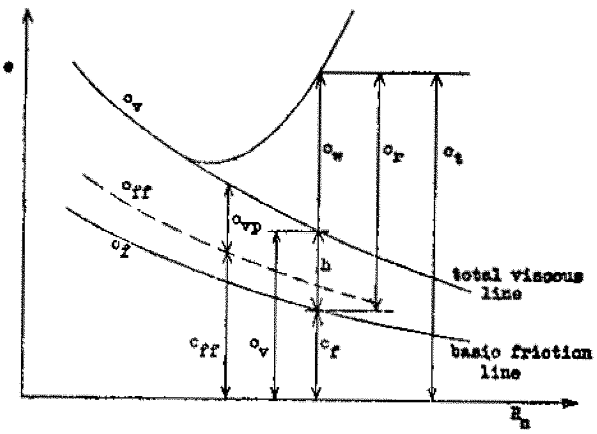


Fig. 1.

(1) Adoption of a basic line c_f representing the tangential frictional resistance of the smooth, flat plate, in the case of two-dimensional turbulent flow (fig. 1).

(2) Proceeding from the above basic c_f to the total viscous resistance line c_v in the case of three-dimensional turbulent flow, as it occurs for a body, by introducing:

(a) The influence of the body-form on the tangential frictional resistance. The basic tangential frictional resistance coefficient c_f then becomes the body tangential frictional resistance coefficiente c_{ff} ('').

(b) The viscous pressure resistance c_{vp} arising around a body, as a consequence of the potential field alteration caused, according to present knowledge, by:

('') Numbers in [ ] refer to the list of references at the of this contribution.

('') The symbol c_{ff} is used in this contribution to distinguish the tangential frictional resistance of a body, in which form influence is included, from the flat plate basic frictional resistance c_f, in which that influence does not exist.

(b1) Thickness of the boundary-layer.

(b2) Flow separation, when this exists.

We can write $c_v = c_{ff} + c_{vp}$.
Ratio $r = c_v/c_f$ is the form factor.

It is evident that, if steps (1) and (2) were overcome and moreover the assumption were admitted that total resistance of a surface body is the addition of two parts governed respectively by the laws of Froude and Reynolds, it would be possible to correlate model and ship resistance by means of the formula (*)

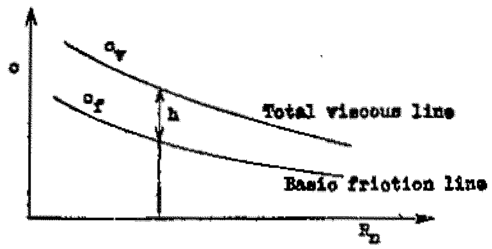
$$c_t = c'_t - c'_v + c_v = c'_t - r' c'_f + r c_f$$

This relation, interactions excepted, is theoretically correct, since the part $c'_t - c'_v = c'_t - r'c'_f$ which is scaled according to Froude's law, is exclusively that part of the model resistance which is governed by this law.

(*) An accent on the symbol (c'_t) is used to denote the model range when a distinction between this range and the ship range (c_t) is advisable.

TABLE 1

THE DIFFERENT MODES OF VARIATION OF ADDITIVE h, FORM FACTOR r, AND CORRELATOR m



$c_v = c_f + h$ $\quad$ h (additive) $= c_v - c_f$ $\quad$ $h = c_f\,(r - 1)$

$c_v = r\,c_f$ $\quad$ r (form factor) $= c_v/c_f$ $\quad$ $r = 1 + h/c_f$

$\partial c_v/\partial R_n = m\,(\partial c_f/\partial R_n)$ $\quad$ m (correlator) $= (\partial c_v/\partial R_n)/(\partial c_f/\partial R_n)$

When R_n is increasing				Existing correlation proposals	
Geometry of c_v in respect to c_f	Additive h is	h/c_f is	Form factor r is	Author (")	R_n varying from 0 to ∞ r varies from
converging ($m > 1$)	decreasing	decreasing	decreasing	Lap-Troost (*)	∞ to 1
		constant	constant ($r = m$)	Hughes	remains constant
		increasing	increasing	Landweber (x)	m to ∞ m to $m + n/b$
parallel ($m = 1$)	constant	increasing	increasing	W. Froude (&)	1 to ∞ 1 to $1 + h/b$
diverging ($m < 1$)	increasing	increasing	increasing	(**)	—

(") Telfer's correlation is not included in this Table, since it does not imply particular sign for the variation of h or r. Leaving aside linearizacion, Telfer's correlation is, indeed, an objective estimation of facts. However, if the nature of these facts is considered, the normal place for this correlation would oscillate, as would Landweber's, between W. Froude's and Hughes' correlations.

(*) Lap-Troost's correlation implies that lines c_v and c_f are plotted upon log R_n. In this case, line c_v is the result of a horizontal shift of line c_f, the value of the shift depending on the hull form.

(x) Landweber's correlation implies a form factor $r = m + n/c_f$. Variation of r from m to ∞ occurs if a friction formula is used in which $c_f = 0$ when $R_n = \infty$ (f. i., Hughes' formula); variation from m to $m + n/b$, if the friction formula used is of type $c_f = b + e/R_n$ (f. i., Telfer's formula).

(&) With W. Froude's correlation r varies from 1 to ∞, or from 1 to $1 + h/b$, according to whether the friction formula used is, respectively, of the first or the second type mentioned in the above note.

(**) Could divergence of c_v in respect to c_f, as R_n increases, correspond to cases in which an extraordinary flow separation exists?

Designating by h the difference $c_v - c_f$, amounting to $h = (c_{ff} - c_f) + c_{vp}$ (Fig. 1), we have

$$r = 1 + \frac{h}{c_f} \qquad h = c_f\,(r - 1)$$

If, when increasing R_n, lines c_v and c_f converge proportionally to c_f, that is, if h/c_f remains constant, the form factor r will be constant.

A stronger convergence would indicate a decreasing factor r. On the contrary, a slighter one, and with more reason still the parallelism or divergence between lines c_v and c_f, would reveal the existence of an increasing form factor r.

Parallelism, which corresponds to the case of a constant additive h, leads us to the following correlation:

$$c_t = c'_t - (c'_f + h) + (c_f + h) = c'_t - (c'_f - c_f)$$

The additive h is eliminated from the final result and there only remains the basic frictional coefficients. This is the W. Froude correlation.

These different cases are resumed in Table 1. This Table is discussed later, when discussing the various correlation proposals.

2.—The basic friction line.

The basic friction line is the basis in respect to which the form factor r is measured; it also constitutes the pattern which guides the extrapolation from model to ship.

Let us consider the correlation formula written under the form

$$c_t = c'_t - c'_v + c_v \quad \text{where} \quad c_v = r\,c_f$$

Evidently, difference $c'_t - c'_v = c'_w$ may be established directly from experimentation, without any consideration of basic line or form factor; a short extrapolation of c'_v beyond the non wave-making zone pertaining to the model considered, or the experimenting of another slightly bigger model, should indeed be sufficient to know c'_v in all the required model range. Nevertheless, consideration of the basic line is indispensable in order to obtain the last term $c_v = r\,c_f$ of the above formula.

Before Hughes evolved his basic frictional line, different formulae had been proposed to represent the same physical effect. Restraining our references to the modern formulae, the following should be mentioned:

Prandtl-Schlichting (1932) [5] ...... $c_f = \dfrac{0.455}{(\log R_n)^{2.58}}$

~~Prandtl-Schlichting~~ (1932) [5] .... $\dfrac{0.242}{\sqrt{c_f}} = \log\,(R_n\,c_f)$
Karman-Schoenherr

Schultz-Grünow (1940) [7] .... $c_f = \dfrac{0.427}{(\log R_n - 0.407)^{2.64}}$

Nikuradse (1946) [8] .... $c_f = \dfrac{0.02666}{R_n^{0.139}}$

Kempf-Karhan (1951) [9] .... $c_f = \dfrac{0.055}{R_n^{0.182}}$

Lap-Troost (1952) ..... [10] [11] (*) $\dfrac{0.2545}{\sqrt{c_f}} = -0.9526 + \log\,(R_n\sqrt{c_f})$

Hughes (1954) [12] .... $c_f = \dfrac{0.066}{(\log R_n - 2.03)^2}$

All these formulae are frequently referred to, except that of Nikuradse, which, to my knowledge, has not yet been mentioned in ship circles. Nikuradse's previous work is well-known and has been widely used (precisely as experimental basis of several of the other formulae quoted above), but it was mainly on pipes, while that on which this new formula is based, refers directly to plates.

In Table 2 all these formulae are numerically compared, and in Fig. 2, Hughes' line is compared with the others; the two suggestions of the Skin Friction Committee Report have also been included. In Table 3, Schoenherr's and Hughes' formulae are particularly compared. All these comparisons reveal a rather unsatisfactory situation. It can be seen, for instance, that Hughes' line runs substantially lower than the others, excepting Nikuradse's and Kempf's lines, which in the model and ship ranges, respectively, come somewhat near to it, though in slope they differ (in opposite sense) considerably from that of Hughes. Although the convenience of having one basic line only, is obvious, we realize that a unanimous decision will not be reached too easily. It could, however, be brought about by the presentation of some conclusive facts.

In this respect, it should be recognised that the experimentation supporting Hughes' formula was undoubtedly most careful, much more exhaustive than that on which the other formulae mentioned were based, and that, moreover, it dealt directly with the friction of plates in water, and not of pipes (in many cases considered in air-flow), as occurred in the majority of the other cases. Hughes particu-

(*) Readers of the second reference (TINA 1956) should be warned that Constant 6.547 in the formula at the top of page 160 in that publication is wrong, the correct value being 2.366. The Author is grateful to Mr. Lap for having subsequently given him the correct value.

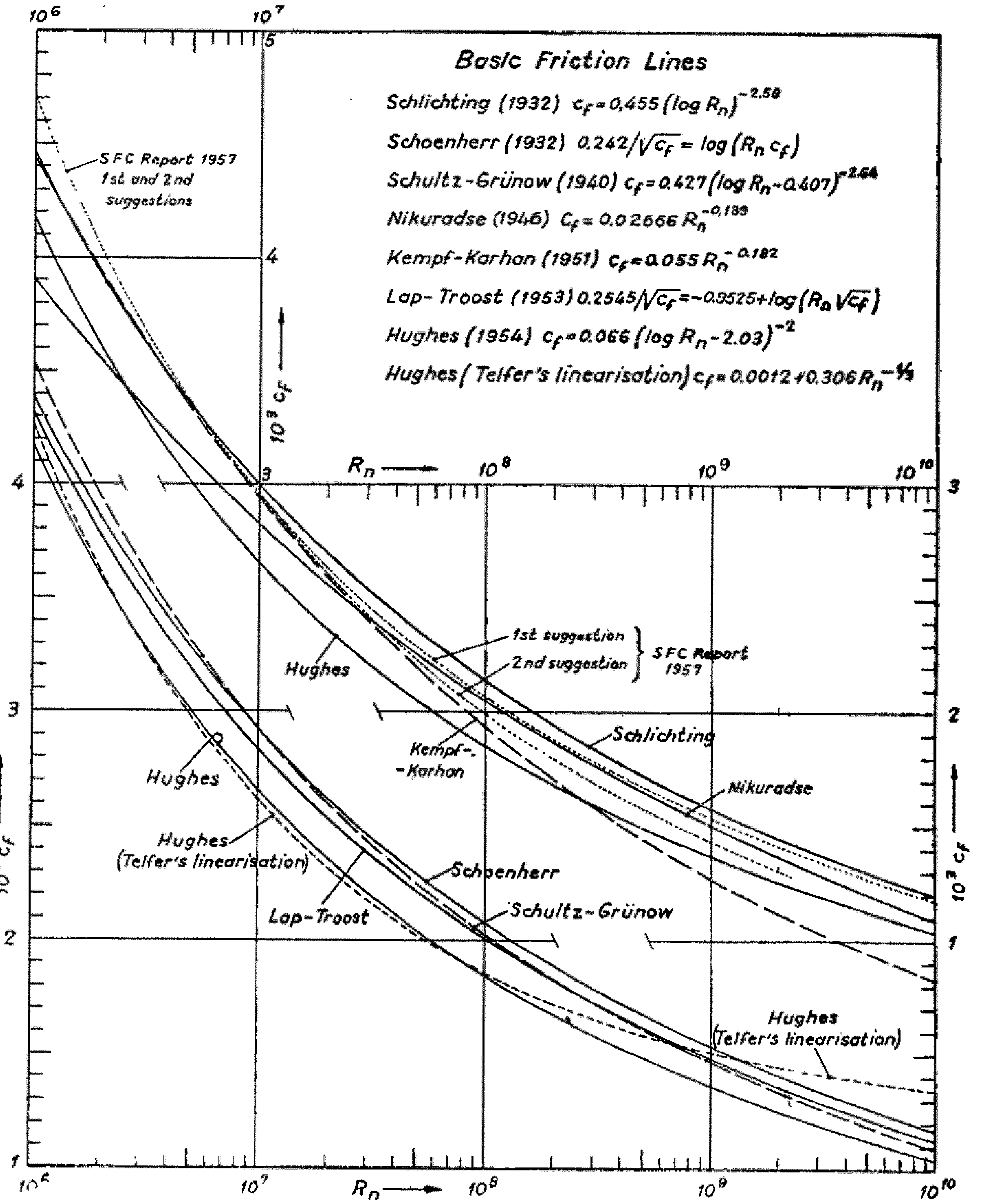


Fig. 2.—Comparison of different basic friction lines c_f.

larly emphasizes that his formula interprets the two-dimensional flow in the case of plates with much more accuracy than the previous ones, and denies the existence of laminar flow influence in some of his tests, as certain of his critics have alleged.

Against the formula of Hughes stands, particularly, the fact that those of Schoenherr and Schlichting (which practically coincide with each other) present a long record of use in Aeronautics, been found in practical agreement with experimental checking, both in wind tunnel and in flight. It has also been argued that Hughes' formula lacks theoretical basis, while theory has contributed to the establishment of some of the ones mentioned above. But, if it is true that several of these formulae have been obtained after an important intervention of theoretical reasonings, owing to the sim-

TABLE 2

DIFFERENT FORMULAE FOR THE BASIC FRICTIONAL RESISTANCE COEFFICIENT C_F, SMOOTH FLAT PLATE IN TURBULENT FLOW

	R_n	1932 Schlichting $\frac{0.455}{(\log R_n)^{2.58}}$	1932 Schoenherr $\frac{0.242}{\sqrt{c_f}} = \log (R_n\, c_f)$	1940 Schultz-Grünow $\frac{0.427}{(\log R_n - 0.407)^{2.64}}$	1946 Nikuradse $\frac{0.02666}{R_n^{0.139}}$
Model range	10^5	7.156×10^{-3}	7.179×10^{-3}	7.630×10^{-3}	25.381×10^{-3}
	10^6	4.471	4.409	4.536	3.907
	5×10^6	3.364	3.294	3.327	3.124
	10^7	3.004	2.934	2.938	2.837
Ship range	10^8	2.128	2.072	2.024	2.060
	10^9	1.571	1.531	1.460	1.496
	10^{10}	1.197	1.172	1.092	1.086

	R_n	1951 Kempf-Karhan $\frac{0.055}{R_n^{0.182}}$	1952 Lap-Troost $\frac{0.2545}{\sqrt{c_f}} = -0.9526 + \log (R_n \sqrt{c_f})$	1954 Hughes $\frac{0.066}{(\log R_n - 2.03)^2}$	Hughes (Telfer's linearization) $0.0012 + \frac{0.306}{R_n^{1/3}}$
Model range	10^5	6.766×10^{-3}	7.290×10^{-3}	7.482×10^{-3}	7.793×10^{-3}
	10^6	4.450	4.332	4.188	4.260
	5×10^6	3.320	3.198	3.027	2.990
	10^7	2.927	2.839	2.672	2.620
Ship range	10^8	1.925	1.995	1.852	1.859
	10^9	1.266	1.472	1.359	1.506
	10^{10}	0.832	1.129	1.039	1.342

	R_n	1957 Webb's Inst. $\frac{0.80}{(\log R_n)^{2.86}}$	1957 SFC's Report First proposal	1957 SFC's Report Second proposal	1957 Telfer $\frac{0.070}{(\log R_n - 2.12)^2}$
Model range	10^5	8.017×10^{-3}	—	—	8.439×10^{-3}
	10^6	4.759	4.809×10^{-3}	4.809×10^{-3}	4.650
	5×10^6	3.472	3.368 (approx.)	3.368 (approx.)	3.339
	10^7	3.063	2.934	2.934	2.939
Ship range	10^8	2.090	2.072	1.989 (approx.)	2.025
	10^9	1.493	1.531	1.431	1.479
	10^{10}	1.104	1.172	—	1.127

TABLE 3

COMPARISON BETWEEN C_F SCHOENHERR AND C_F HUGHES

	R_n	$10^3 \times$ c_fSchoenherr	$10^3 \times$ c_fHughes	$10^3 \times$ (c_fSch - c_fHug)	$100 \times \frac{c_f\text{Sch} - c_f\text{Hug}}{c_f\text{Hug}}$
Model range	10^5	7.179	7.482	— 0.303	— 4.22
	2.2356×10^5	5.990	5.990	crossing point	
	10^6	4.409	4.188	0.221	5.28
	5×10^6	3.294	3.027	0.267	8.82
	10^7	2.934	2.672	0.262	9.80
Ship range	10^8	2.072	1.852	0.220	11.88
	10^9	1.531	1.359	0.172	12.66
	10^{10}	1.172	1.039	0.133	12.80

plifications and assumptions introduced in their deduction (some of them as more or less subjective where results calculated according to them have conceptions of the phenomena), the value of the final formulae as trustworthy interpretations of the friction mechanism is rather questionable. Moreover, some of them (Schlichting's, Schultz-Grünow's, etc.) are, as is well-known, merely interpolation formulae; their logarithmical structure is quite empirical and no tie exists betaween it and the logarithm operator which at some stages of the deduction, arises as mathematical expression of some physical reasoning (e. g., Prandtl's lineal relation between mixing length and wall distance).

However, a weak point in the establishment of Hughes' formula is, undoubtedly, the fact that, owing to the relatively small dimensions of the tank where the experiments were carried out, some of his results are affected by wall interference. Corrections of interference effects, where wave-making exists, are at present rather uncertain. [13], [14] and sixth reference of [41].

All these basic lines are discussed later, together with the correlation methods.

3.—Form factor and model-ship correlation.

The form factor is defined and measured in respect to the basic friction line. Therefore, as long as there are so many formulae which may be taken as basic lines, the form factor and the model-ship correlation will evidently retain a purely conventional value, depending on the basic friction line to which they are linked.

The liberty of relating a given experimental viscous line c_v to one or another basic line c_f may not be absolute, but restricted, if a certain law or property is attributed to the correlation, for instance, if we admit independence from R_n of the form factor. In this case, indeed, if values c_v were found to be proportional, for instance, to Hughes' c_f, any other basic line parallel to Hughes' should be rejected, whether Hughes' is the right one or not. This is what might more or less happen when considering Hughes' and Schoenherr's lines, which, at different levels, run nearly parallel for quite a stretch; if r is constant in respect to Hughes', it will decrease in respect to Schoenherr's, when R_n increases.

As long as no sufficient statistical information is available, the form factor should be obtained in each case. Experimentation (in the non wave-making zone, geosim tests) seems to be at present the only way to get it. Although subject to practical limitations, the experimentation should cover an extra zone as ample as possible beyond the usual model range, since, on the one hand, the value of the form factor in the usual model range is to be found; and, on the other, its scaling law to the ship size is to be investigated, unless constancy of r be a priori admitted.

As regards the model range, it seems possible to work out an average value of the form factor with the approximation required for practical uses. In our previous Report on the geosim tests of the *Tina Onassis* [4], [15], in which a constant form factor was assumed, it was found that a relatively important variation of r produces only very little change in the final result P_p.

Regarding the obtention of the form factor in the ship range, evidently the experimentation in the model range, however wide it may be, is too narrow and far removed, and therefore cannot offer, without theoretical help, a sound basis on which the scaling law of r could be found. Although a discrimination of the viscous resistance components in the model range could give some light on the internal structure

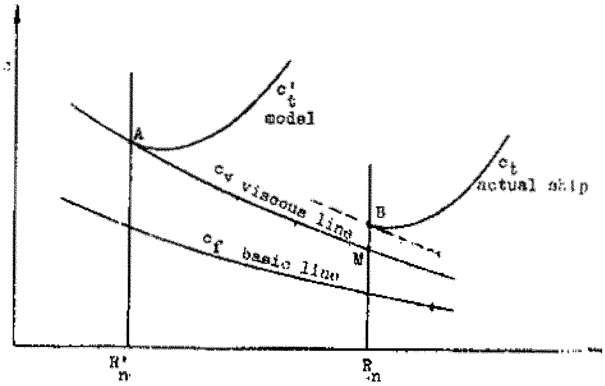


Fig. 3.

of c'_v, the knowledge of what really occurs in the ship range could only be reached by experimenting in this zone. But experiments at the full scale, [16] to [21], apart from their inherent practical difficulties, also involve, as far as correlation is concerned, the intrinsic difficulty which roughness of the actual ship implies.

Fig. 3 clearly illustrates the question: A is the end of the viscous line in the model range (the "run-in" point). B is the corresponding point at the actual ship; B implies a resistance test at the full scale, or the conversion into ship resistance, of the ship propulsion power; difficulties in obtaining point B are obvious. From A to M runs the hypothetical viscous line, which, model smoothness being assumed, is determined by the scale effects on the viscous phenomena; for a surface hull this line corresponds to the vanishing Froude Numbers (non wave-making zone). The vertical gap MB includes those resistance effects which, existing in the actual ship but not in the model, should be added to M in order to reach B. Roughness resistance is the main of these effects; all the others (wind resistance, etc.) are generally minor effects which may be approximately estimated, so that, having subtracted them, we could consider gap MB as practically reduced to the roughness effect. Undoubtedly, the knowledge of point M would be a definite help in removing the uncertainty prevailing in the zone between A and B. But in spite of the progress made in our knowledge of roughness [22] to [24], it is not yet possible, by the direct and exact estimation of the ship roughness which would be neccesary to get down from B to M, to situate M exactly where it is in reality. On the other hand, proceeding from A towards M, a model-ship correlation or extrapolation method leads us to a certain M; correctness of this point and that of the consequently resulting roughness effect MB are unknown, since both depend on the correlation method employed; if, for instance, this combines, as in Hughes' proposal, a basid line c_f and a viscous line c_v, any error picked up from A to M may be due to errors existing: either in line c_f (the law of constant form factor being physically true), or in line c_v (basic line c_f being true), or in both lines c_f and c_v.

In spite of this imprecision, there is no doubt that consideration of margin MB is what has brought home the urgent necessity of revising ship-model correlation methods. For while a reasonable reduction of this margin could be interpreted as reflecting that of ship roughness as a consequence of welded hulls and of a better finish and painting, a margin leaving no room for a reasonable physical roughness of the actual ship (according to Horn, Δc_f should not be inferior to $+ 0.00025$) is not admissible; and even less that it could become negative, as has been found repeatedly, particularly for certain types of ships, in sea trials of recent years [25] to [30].

4.—The different correlation proposals (")

Two of the correlation proposals are, as already said:

(1) That of a constante additive h, $c_v = c_f + h$
(2) That of a constant multiplier or form factor r, $c_v = r\ c_f$

Since $r = 1 + h/c_f$ and $h = c_f\ (r - 1)$, proposal (1) implies an increasing form factor r, and proposal (2) a decreasing additive h, when R_n increases, even though dependence on R_n, of r in the first case, and that of h in the second, are merely consequences of that of c_f (Fig. 1).

(1) *The constant additive (W. Froude)*:

As said before, correlation according to a constant additive h leads to the correlation formula

$$c_t = c'_t - (c'_f - c_f)$$

(") Presentation here of the different correlation proposals is not chronological but rather according to a rational order.

The additive h is eliminated from the final result and there only remain the basic frictional coefficients. This is the W. Froude correlation, in which $h = c_r$ (residuary resistance coefficiente) $- c_w$. Therefore, if the constancy of h is admitted, errors resulting from using Froude's correlation are to be imputed to the use of inappropriate basic frictional coefficients. It should be observed that the level of the basic line has no influence on result c_t which is only influenced by the slope of that line in so far as it determines decrement $c'_f - c_f$. This shows that, when W. Froude's correlation is used, one may well be unaware of applying a basic line situated at a wrong level as regards the physical meaning of the line.

W. Froude's correlation may be used in association with any basic frictional line (Schoenherr's, etc.). Traditionally, it has been applied in association with R. E. Froude's friction coefficients, which are not a function of Reynolds Number. R. E. Froude's friction coefficients, written under the form $R_f/\frac{1}{2}\rho V^2 S$, read

$$c_f \text{ (R. E. Froude)} = 0.02014 \text{ "O" } F_n^{\ -0.175}$$

where 0.02014 is a dimensionless numerical value, and "O" is also a dimensionless value, depending on length and temperature. Evidently, R. E. Froude's friction coefficients do not produce a single friction line.

Against the correlation according to a constant additive it has often been argued from a general physical point of view. Nevertheless, constancy of h does not imply that the different viscous phenomena included in h are independent from R_n, but only that the sum of their resistance effects (within which + and — signs may be found) turns out to be so; therefore, the existence, along a more or less restricted range, of a practically constant additive h might be admitted, at least as possible, and not generally and necessarily rejected on theoretical grounds.

It should be added, however, that, as explained below, according to the at present available data, a decreasing h seems to be most probable. Model and ship ranges are to be considered.

(a) As regards the model range:

Irrespective of the basic line to which it is associated, W. Froude's correlation when applied to a geosims family shows frequently that test results of a certain geosim cannot exactly be predicted from results of another geosim of the same family, but of a different size. A more satisfactory correlation has been tried for in such cases by slightly varying the slope of the basic line concerned; for instance, as regards Schoenherr's line, a closer correlation among geosims has been obtained in some cases by making the basic line somewhat steeper. But, evidently, this could only be justified if the constant additive were the correct correlation method, and only the slope of the basic line were wrong. If this is not the case, varying the slope really amounts to despoiling the basic line of its physical content, merely for the practical convenience of using W. Froude's correlation; so that success, even as to the practical end pursued, is achieved only partially, no general solution valid for any ship-form being reached.

(b) As regards the ship range:

A direct comparison, more general than that possible by considering particular ships, in given on Figs. 4, 5, 6 and 7.

Fig. 4 shows the different ship predictions c_t which result from using W. Froude's correlation associated to the different basic frictional formulae quoted in Section 2 (''). All extrapolation lines radiate from the same model test spot ($R'_n = 10^7$, $c'_t = 5.0 \times 10^{-3}$); this spot corresponds to a case in which frictional resistance amounts about to 60 % of total resistance. Extrapolation lines according to Hughes' correlation (for $r = 1.20$ and $r = 1.30$) are also represented. Discrepancies among extrapolation lines, apart from intrinsic causes, depend on the ship values R_n at which they are considered; due to the, in general, divergent character of the lines, the greater R_n, the greater the discrepancy, as much in absolute values, as in percentaje of c_t. If, keeping to the same value $R'_n = 10^7$, model spot c'_t were taken at another height, the picture, quite unaltered, would vertically shift by the same distance as spot c'_t; discrepancies, although invariable in absolute values, would then become greater or smaller in percentage of c_t, according to whether c'_t had been lowered or raised, i. e., shifted in the direction of a relatively greater frictional resistance or of a greater wave resistance. The Table included in the figure allows an easy consideration of the picture for different types of ships, at various speeds.

Figs. 5, 6 and 7 show the same as Fig. 4, but, extrapolation lines radiate from model spots situated at lower R'_n ranges (5×10^6, 10^6 and 10^5). Values c'_t corresponding to these spots were taken on the same Schoenherr extrapolation line passing through the model spot of Fig. 4.

The picture mutations observed when Figs. 4, 5, 6 and 7 are successively examined, are the consequence of the different local slopes of the basic lines, existing at the four model R_n ranges considerad. At an equal fluid and temperature, they represent the influence of the model size on the extrapolation picture. Thus, Figs. 4, 5, 6 and 7 convey a general impression of the different basic formulae, as far as their slopes are concerned, when they are associated with W. Froude's correlation. In Tables 3a and 3b the numerical values may be compared.

While for a restricted group of basic formulae (mainly Schoenherr's, Lap-Troost's and Hughes') differences in ship predictions c_t are not important and little influenced by the scale variation, at least at the usual moder R_n ranges, for quite a number of other formulae differences are certainly appreciable and greatly influenced by changes of the model size. They become considerable at the smallest $R'_n = 10^5$ (Fig. 7).

Since in Figs. 4, 5 and 6 Lap-Troost's line runs somewhat higher than Schoenherr's, and Hughes' even more, the finding of unacceptable allowances (too small or even negative values, in which no room for a positive reasonable roughness allowance exists) in quite a number of cases when using Schoenherr's basic line, would also occur, aggravated, if the other two formulae were used. Schultz-Grünow's extrapolation line, which, due to the greater steepness of its basic line, runs lower than the three mentioned above, leaves a somewhat greater allowance; however, from the reasonable balance of the whole picture it should not be concluded that Schultz-Grünow's line represents a vindication of W. Froude's correlation.

In summary, the unsatisfactory correlation of geosims in the model range and the unreasonable roughness allowances resulting in the ship range, whatever the basic line used, clearly indicate the general incorrectness of W. Froude's correlation method. If instead of a constant additive h (probably main source of error), an adequate form factor r were introduced, a certain convergence of lines c_v and c_f would be obtained, and correlation would adhere closer to real facts.

The above figures also show that if correlation according to a form factor is correct, errors resulting from the use of W. Froude's method would be greatly amplified by using small models. From these figures, it can also be clearly understood why too small and even negative roughness allowances have been usually found when comparing big tanker trial results with model predictions calculated according to W. Froude's correlation (three influences concur in such cases: important slope of the extrapolation line resulting from a considerable form factor r; lower location of model spot as corresponding to a considerable percentage of viscous resistance; and high ship R_n range).

('') Although Lap-Troost's and Hughes' basic friction lines have their own correlation, for the sake of generality, they are presented in association with W. Froude's correlation, in Figs. 4, 5, 6 and 7.

Type	L m	V knots	F_n	R_n (*)
Small fisher	30	9-11	0.27-0.33	1.17-1.43 x 10^8
Trawler	50	9-13	0.21-0.30	1.95-2.81 "
Coaster	60	10-13	0.21-0.28	2.60-3.37 "
Small cargo	90	10-13-16	0.17-0.23-0.28	3.89-5.06-6.22 "
Normal cargo	120	12-15-17	0.18-0.23-0.25	5.06-7.78-8.81 "
Pass. and c. liner	180	15-18-21	0.18-0.22-0.26	1.17-1.40-1.63 x 10^9
Big tanker	200	13-15-17	0.15-0.17-0.20	1.13-1.29-1.47 "
Supertanker	280	14-16-18	0.14-0.16-0.18	1.69-1.94-2.18 "
Atlantic liner	300	28-30-33	0.27-0.28-0.31	3.63-3.89-4.28 "
Destroyer	110	25-30-40	0.39-0.47-0.63	1.19-1.43-1.90 "
Cruiser	210	20-30-35	0.23-0.34-0.40	1.82-2.72-3.18 "
Aircraft carrier	300	25-30-35	0.24-0.28-0.33	3.24-3.89-4.52 "

(*) salt water 15°C $\nu = 1.191 \times 10^{-6}\ m^2 s^{-1}$

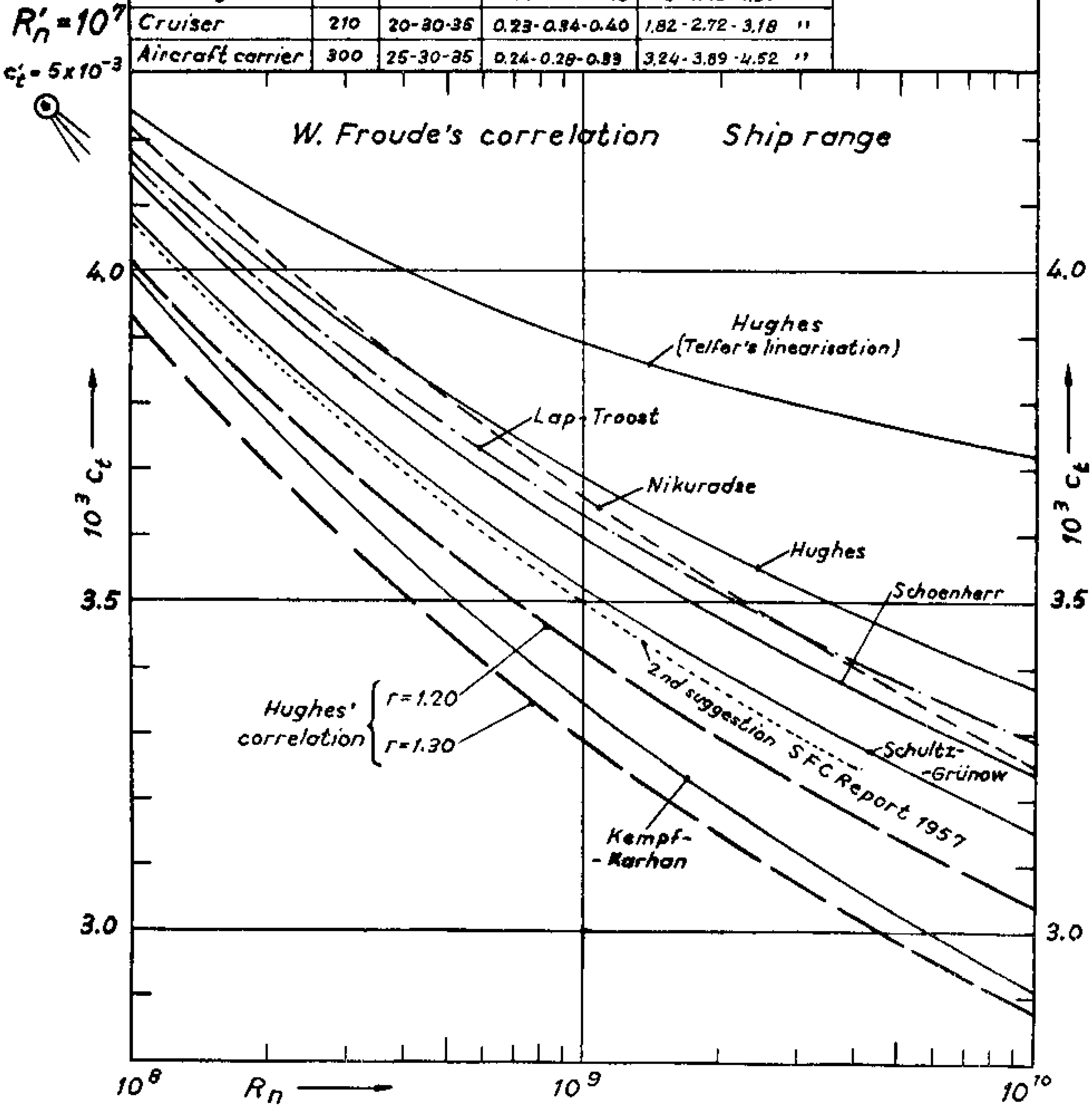


Fig. 4.—W. Froude's correlation, ship range.—Model spot $R'_n = 10^7$.

Comparison of ship predictions c_t by using different basic friction formulae. All extrapolation lines radiate from the same model spot $R'_n = 10^7$, $c'_t = 5 \times 10^{-3}$. Dotted line corresponds to the modified Schoenherr line according to the SFC Report, second suggestion(1957); first suggestion coincides in this case with Schoenherr's original line. Comparison in extended to Hughes' correlation for r = 1.20 and r = 1.30. SFC's 2nd suggestion pratically coincides with Hughes' correlation for r = 1.145.

Type	L m	V knots	F_n	R_n (*)
Small fisher	30	9-11	0.27-0.33	1.17-1.43 x10⁸
Trawler	50	9-13	0.21-0.30	1.95-2.81 "
Coaster	60	10-13	0.21-0.28	2.60-3.37 "
Small cargo	90	10-13-16	0.17-0.23-0.28	3.89-5.06-6.22 "
Normal cargo	120	12-15-17	0.18-0.23-0.25	5.06-7.78-8.81 "
Pass. and c. liner	180	15-18-21	0.18-0.22-0.26	1.17-1.40-1.63 x10⁹
Big tanker	200	13-15-17	0.15-0.17-0.20	1.13-1.29-1.47 "
Supertanker	280	14-16-18	0.14-0.16-0.18	1.69-1.94-2.18 "
Atlantic liner	300	28-30-33	0.27-0.28-0.31	3.63-3.89-4.28 "
Destroyer	110	25-30-40	0.39-0.47-0.63	1.19-1.43-1.90 "
Cruiser	210	20-30-35	0.23-0.34-0.40	1.82-2.72-3.18 "
Aircraft carrier	300	25-30-35	0.24-0.28-0.33	3.24-3.89-4.52 "

(*): salt water 15°C $\nu = 1.191 \times 10^{-6}\ m^2 s^{-1}$

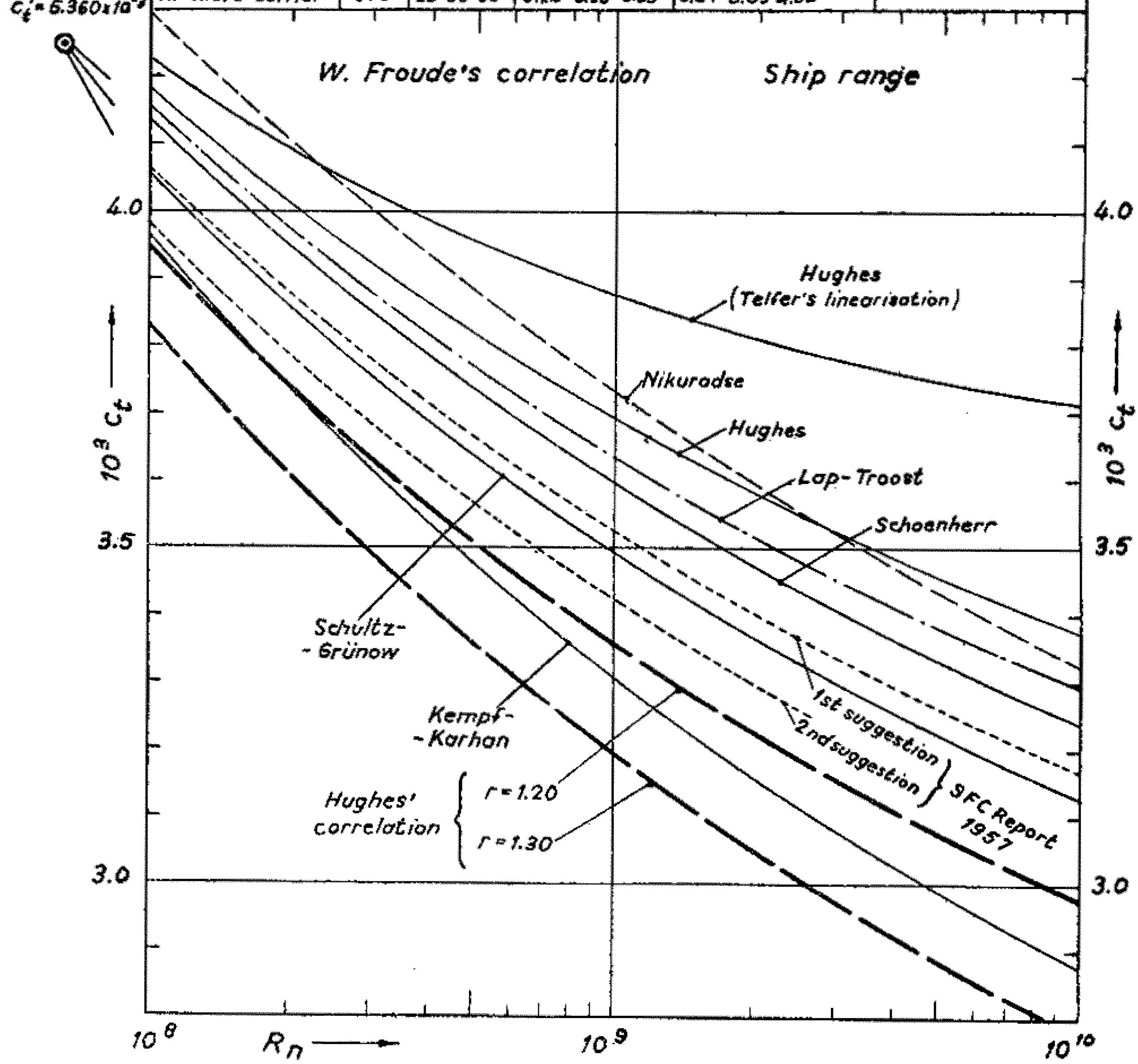


Fig. 5.—W. Froude's correlation, ship range.—Model spot $R'_n = 5 \times 10^6$.

Comparison of ship predictions c_t by using different basic friction formulae. All extrapolation lines radiate from the same model spot $R'_n = 5 \times 10^6$, $c'_t = 5.360 \times 10^{-3}$. Dotted lines correspond to the modified Schoenherr line according to the SFC Report, first and second suggestions (1957).

Comparison in extended to Hughes' correlation for r = 1.20 and r = 1.30. SFC's 1st suggestion practically coincides with Hughes' correlation for r = 1.161.

Type	L m	V knots	F_n	R_n (*)	(*)-salt water 15°C $\nu = 1.191 \times 10^{-6} m^2 s^{-1}$
Small fisher	30	9-11	0.27-0.33	$1{,}17-1{,}43 \times 10^8$	
Trawler	50	9-13	0.21-0.30	1,95-2,81 "	
Coaster	60	10-13	0.21-0.28	2,60-3,37 "	
Small cargo	90	10-13-16	0,17-0.23-0.28	3,89-5,06-6,22 "	
Normal cargo	120	12-15-17	0.18-0.23-0.25	5.06-7.78-8,81 "	
Pass. and c. liner	180	15-18-21	0.18-0.22-0.26	$1{,}17-1{,}40-1{,}63 \times 10^9$	
Big tanker	200	13-15-17	0.15-0.17-0.20	1,13-1,29-1,47 "	
Supertanker	280	14-16-18	0.14-0.16-0.18	1,69-1,94-2,18 "	
Atlantic liner	300	28-30-33	0.27-0.28-0.31	3,63-3,89-4,28 "	
Destroyer	110	25-30-40	0.39-0.47-0.63	1,19-1,43-1,90 "	
Cruiser	210	20-30-35	0.23-0.34-0.40	1,82-2,72-3,18 "	
Aircraft carrier	300	25-30-35	0.24-0.28-0.33	3,24-3,89-4,52 "	

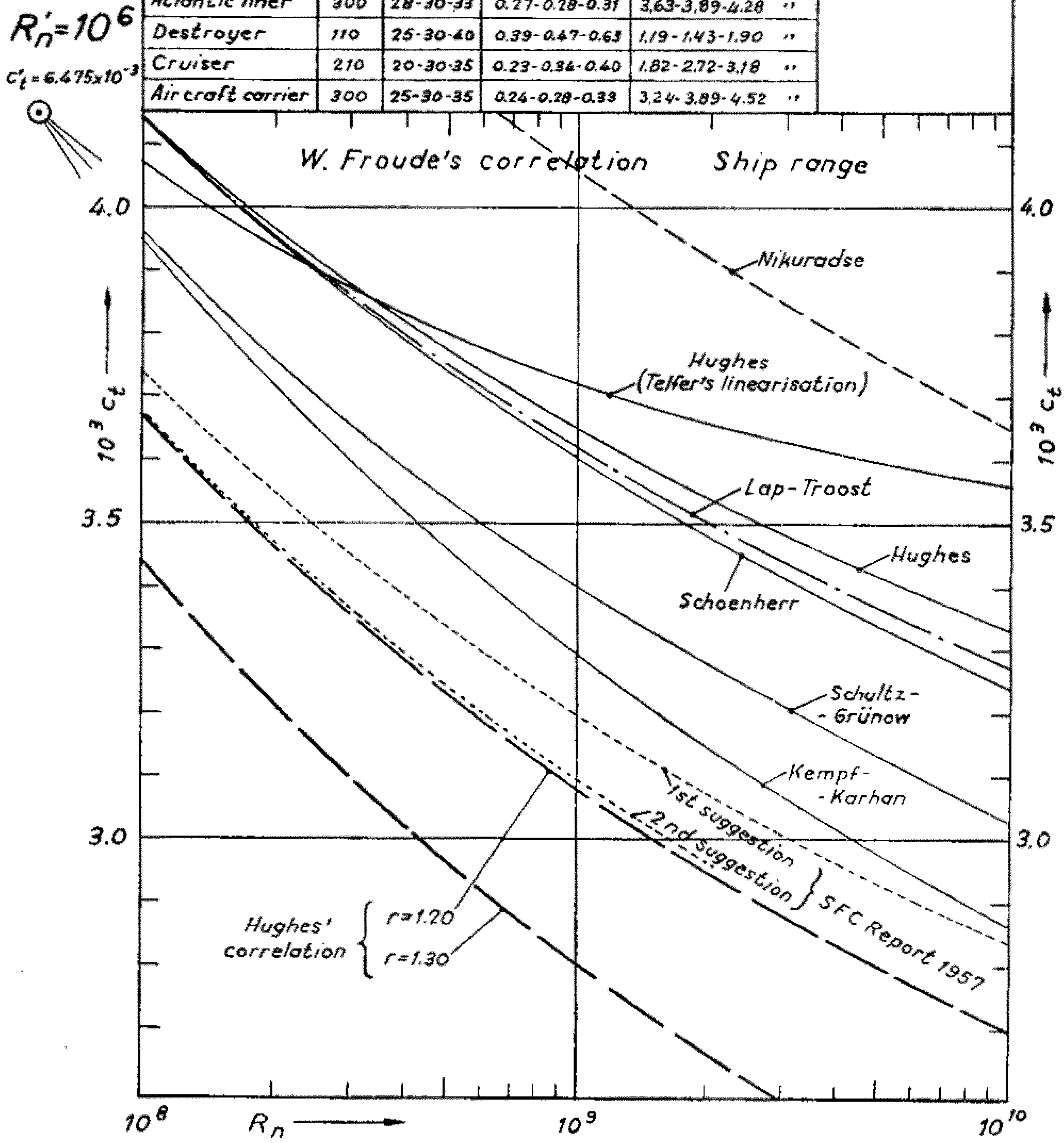


Fig. 6.—W. Froude's correlation, ship range.—Model spot $R'_n = 10^6$.
Comparison of ship predictions c_t by using different basic friction formulae. All extrapolation lines radiate from the same model spot $R'_n = 10^6$, $c'_t = 6.475 \times 10^{-3}$. Dotted lines correspond to the modified Schoenherr line according to the SFC Report, first and second suggestions (1957.
Comparison in extended to Hughes' correlation for r = 1.20 and r = 1.30.—SFC's 1st suggestion practically coincides with Hughes' correlation for r = 1.159; SFC's 2nd suggestion practically coincides with r = 1.194.

Type	L m	V knots	F_n	R_n (*)
Small fisher	30	9 - 11	0.27 - 0.33	1.17 - 1.43 × 10^8
Trowler	50	9 - 13	0.21 - 0.30	1.95 - 2.81 "
Coaster	60	10 - 13	0.21 - 0.28	2.60 - 3.37 "
Small cargo	90	10 - 13 - 16	0.17 - 0.23 - 0.28	3.89 - 5.06 - 6.22 "
Normal cargo	120	12 - 15 - 17	0.18 - 0.23 - 0.25	5.06 - 7.78 - 8.81 "
Pass. and c. liner	180	15 - 18 - 21	0.18 - 0.22 - 0.26	1.17 - 1.40 - 1.63 × 10^9
Big tanker	200	13 - 15 - 17	0.15 - 0.17 - 0.20	1.13 - 1.29 - 1.47 "
Supertanker	280	14 - 16 - 18	0.14 - 0.16 - 0.18	1.69 - 1.94 - 2.18 "
Atlantic liner	300	28 - 30 - 33	0.27 - 0.28 - 0.31	3.63 - 3.89 - 4.28 "
Destroyer	110	25 - 30 - 40	0.39 - 0.47 - 0.63	1.19 - 1.43 - 1.90 "
Cruiser	210	20 - 30 - 35	0.23 - 0.34 - 0.40	1.82 - 2.72 - 3.18 "
Aircraft carrier	300	25 - 30 - 35	0.24 - 0.28 - 0.33	3.24 - 3.89 - 4.52 "

(*)-salt water 15°C $\nu = 1.191 \times 10^{-6} m^2 s^{-1}$

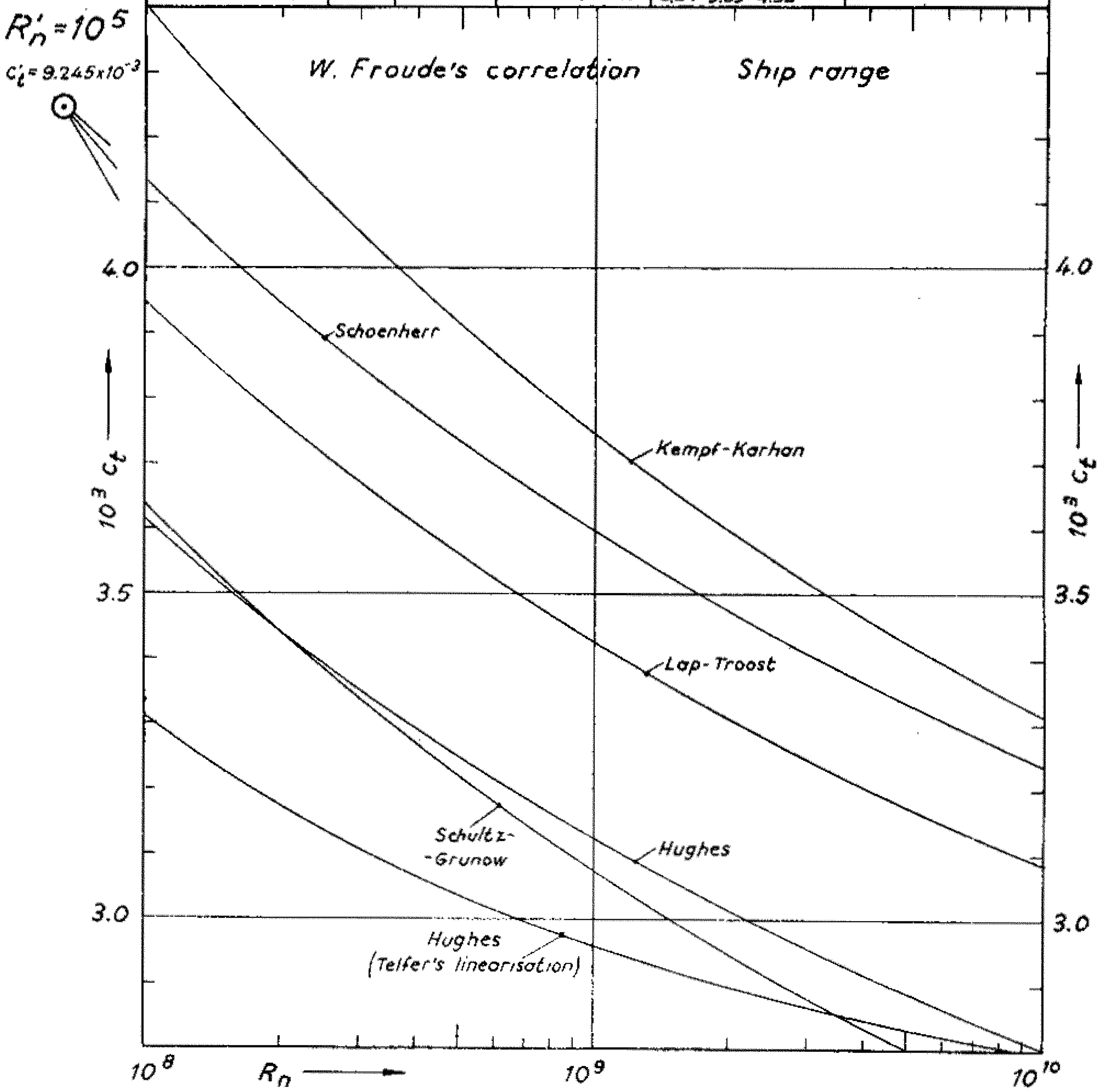


Fig. 7.—W. Froude's correlation, ship range.—Model spot $R'_n = 10^5$.
Comparison of ship predictions c_t by using different basic friction formulae. All extrapolation lines radiate from the same model spot $R'_n = 10^5$, $c'_t = 9.245 \times 10^{-3}$. (SFC's 1st and 2nd scuggestions do not indicate values of c_t for the model range $R''_n = 10^6$).
Hughes' correlation for $r = 1.20$ and $r = 1.30$ fall lower than the limits of the figure.

TABLE 3a

W. FROUDE'S CORRELATION.—COMPARISON OF PPREDICTIONS C_T BY USING DIFFERENT BASIC FRICTION FORMULAE

All extrapolation lines radiate from the same model spot which is mentioned in each case.
Basic friction formulae are the same and taken in the same sequence as in Table 2.

	R_n	$10^3 c_t$ obtained by using the basic friction formula of					
		Schlichting	Schoenherr	Schultz--Grünow	Nikuradse	Kempf--Karhan	Lap--Troost
		Model spot: $R'_n = 10^7$, $c'_t = 5.000 \times 10^{-3}$					
Model range	10^5						
	10^6						
	5×10^6						
	10^7	5.000	5.000	5.000	5.000	5.000	5.000
Ship range	10^8	4.124	4.138	4.086	4.223	3.998	4.156
	10^9	3.567	3.597	3.522	3.659	3.339	3.633
	10^{10}	3.193	3.238	3.154	3.249	2.905	3.290
		Model spot: $R'_n = 5 \times 10^6$, $c'_t = 5.360 \times 10^{-3}$					
Model range	10^5						
	10^6						
	5×10^6	5.360	5.360	5.360	5.360	5.360	5.360
	10^7	5.000	5.000	4.971	5.073	4.967	5.001
Ship range	10^8	4.124	4.138	4.057	4.296	3.965	4.157
	10^9	3.567	3.597	3.493	3.732	3.306	3.634
	10^{10}	3.193	3.238	3.125	3.322	2.872	3.291
		Model spot: $R'_n = 10^6$, $c'_t = 6.475 \times 10^{-3}$					
Model range	10^5						
	10^6	6.475	6.475	6.475	6.475	6.475	6.475
	5×10^6	5.368	5.360	5.266	5.692	5.345	5.341
	10^7	5.008	5.000	4.877	5.405	4.952	4.982
Ship range	10^8	4.132	4.138	3.963	4.628	3.950	4.138
	10^9	3.575	3.597	3.399	4.064	3.291	3.615
	10^{10}	3.201	3.238	3.031	3.654	2.857	3.272
		Model spot: $R'_n = 10^5$, $c'_t = 9.245 \times 10^{-3}$					
Model range	10^5	9.245	9.245	9.245	9.245	9.245	9.245
	10^6	6.560	6.475	6.151	7.771	6.929	6.287
	5×10^6	5.453	5.360	4.942	6.988	5.799	5.153
	10^7	5.093	5.000	4.553	6.701	5.406	4.794
Ship range	10^8	4.217	4.138	3.639	5.924	4.404	3.950
	10^9	3.660	3.597	3.075	5.360	3.745	3.427
	10^{10}	3.286	3.238	2.707	4.950	3.311	3.084

TABLE 3b

W. FROUDE'S CORRELATION.—COMPARISON OF PREDICTION C_T BY USING DIFFERENT BASIC FRICTION FORMULAE

All extrapolation lines radiate from the same model spot which is mentioned in each case. Basic friction formulae are the same and taken in the same sequence as in Table 2.

	R_n	$10^3\, c_t$ obtained by using the basic friction formula of					
		Hughes	Hughes (Telfer's linearization)	Webb's Institute	SFC Report 1st proposal	SFC Report 2nd proposal	Telfer (1957)
		Model spot: $R'_n = 10^7$, $c'_t = 5.000 \times 10^{-3}$					
Model range	10^5						
	10^6						
	5×10^6						
	10^7	5.000	5.000	5.000	5.000	5.000	5.000
Ship range	10^8	4.180	4.239	4.027	4.138	4.055	4.086
	10^9	3.687	3.886	3.430	3.597	3.497	3.540
	10^{10}	3.367	3.722	3.041	3.238		3.188
		Model spot: $R'_n = 5 \times 10^6$, $c'_t = 5.360 \times 10^{-3}$					
Model range	10^5						
	10^6						
	5×10^6	5.360	5.360	5.360	5.360	5.360	5.360
	10^7	5.005	4.990	4.951	4.926	4.926	4.960
Ship range	10^8	4.185	4.229	3.978	4.064	3.981	4.046
	10^9	3.692	3.876	3.381	3.523	3.423	3.500
	10^{10}	3.372	3.712	2.992	3.164		3.148
		Model spot: $R'_n = 10^6$, $c'_t = 6.475 \times 10^{-3}$					
Model range	10^5						
	10^6	6.475	6.475	6.475	6.475	6.475	6.475
	5×10^6	5.314	5.205	5.188	5.034	5.034	5.164
	10^7	4.959	4.835	4.779	4.600	4.600	4.755
Ship range	10^8	4.139	4.074	3.806	3.738	3.655	3.850
	10^9	3.646	3.721	3.209	3.197	3.097	3.304
	10^{10}	3.326	3.557	2.820	2.838		2.952
		Model spot: $R'_n = 10^5$, $c'_t = 9.245 \times 10^{-3}$					
Model range	10^5	9.245	9.245	9.245			9.245
	10^6	5.951	5.712	5.987			5.456
	5×10^6	4.790	4.442	4.700			4.145
	10^7	4.435	4.072	4.291			3.745
Ship range	10^8	3.615	3.311	3.318			2.831
	10^9	3.122	2.958	2.721			2.286
	10^{10}	2.802	2.794	2.332			1.933

(2) *The constant form factor* (*Hughes*):

Correlation according to a constant form factor $r = c_v/c_f$ leads us to the correlation formula

$$c_t = c'_t - (c'_v - c_v) = c'_t - r\,(c'_f - c_f)$$

In this formula, the correlating term $r(c'_f - c_f)$ is directly dependent on the level of the basic friction line at the model range (where experimental determination of r is possible), and on the slope of that line when proceeding from model to ship size. Not only the right slope, but also the right level of the basic friction line are then required, in contrast to what happens in the case of W. Froude's correlation.

Since r is by nature superior to 1, it is evident that calculations made according to this correlation will lead to lower values of ship resistance than those obtained by using the previous correlation formula $c_t = c'_t - (c'_f - c_f)$, which did not contain a form factor. Other than lower values could only be obtained, if the earlier correlation were associated to a friction decrement $c'_f - c_f$ bigger enough, in comparison with that used with the new correlation, to compensate the absence of r, which is not real.

Correlation of this type, which is that proposed by Hughes [12], has not been used until now in Naval Architecture, although the idea of a form factor is already found, implicitly or explicitly, in the early contributions to the correlation problem. The proposal in favour of a factor $r = c_v/c_f$ independent from R_n and dependent only on the body form, is based on theoretical work emanating chiefly from the Aeronautical sector. Among other contributors to this subject, Pretsch [31], Squire [32], Young [33] and Scholz [34], may be mentioned. The latter arrives at the simple formula, for the case of two-dimensional bodies:

$$r = 1 + 3\,\frac{\Delta V}{V}$$

which he contends is approximately applicable to bodies of revolution. ΔV is the over-velocity (average value) existing along the body, and, naturally, as long as the formula applied, ratio $\Delta V/V$ is independent from R_n and depends only on the body form. Although quite simple in form, the formula is not as easy to apply, since determination of ratio $\Delta V/V$ is a rather laborious matter. Values of $\Delta V/V$ are known for some usual aerofoil profiles (see, for instance [35]). For the case of surface ship hulls, Horn [36] proposed years ago an approximate experimental method based on the sinkage of the models during their run. This proposal by Horn and his subsequent ones to convert over-velocity into an additional resistance, constitute really one of the first attempts to determine the form factor in an analogous sense as it is now considered.

Ratio $\Delta V/V$, and consequently r, are considerably greater in two-dimensional bodies than in bodies of revolution. In Fig. 8, which has been deduced from Scholz' results, values of form factor r are given for two-dimensional bodies and for bodies of revolution, having as constant parallel section and as meridian section, respectively, the same NACA profile.

Full lines correspond to the bodies derived from the NACA 35, which is a symmetric profile with its maximum thickness at 50 % of the chord, and having an area ratio $\alpha = 0.7262$. The body of revolution derived from this profile has the following fullness coefficients

$$\delta = 0.472,\ \beta = \pi/4 = 0.785,\ \varphi = 0.601 \quad (*)$$

Fig. 8 shows that values of r are appreciably higher in the two-dimensional body than in the body of revolution; in both cases r increases as thickness ratio s/l increases (**).

Dashed lines on the same Fig. 8 correspond to the NACA 63, also a symmetric profile, but having its maximum thickness at 30 % of the chord. The lines show in the case of the two-dimensional body, about the same numerical values and the same slope as above. However, in the case of the body of revolution, values and slope are lower for the NACA 63 than for the NACA 35; this seems to indicate that, while form factor in the two-dimensional bodies here considered is chiefly a question of body proportions and little of the form itself, in the case of the three-dimensional bodies, both proportions and the form itself (in this case, length of the run), have important influence. Considered in connection with ships, this observation might be interesting, although it should be remarked that the cases considered by Scholz were free of flow separation.

The proposal of a constant form factor is not generally valid, since the independence of r from R_n is conditioned by:

(a) Non-existence of dynamical lifting effects.
(b) Turbulent flow existing practically around the whole of the body. This implies a minimum laminar area at the bow and stable transition points.

(*) Between the prismatic coefficient φ of a body of revolution and the area coefficient α of its meridian section, there exists the relation $\varphi/\alpha = 4\ y/s$ (where y is the ordinate upon the axis, of the centre of gravity of the half section). This relation shows that, in normal cases ($4\ y/s = 0.81$ to 0.83), a substantial sharpening of the form is implied in proceeding from the two-dimensional to the body of revolution; this is quite in line with the great difference in form factor existing between the two cases. The above relation also shows that, unless the value of α is very high, the resulting body of revolution is necessarily rather slender.

(**) Schlichting's basic line was used by Scholz.

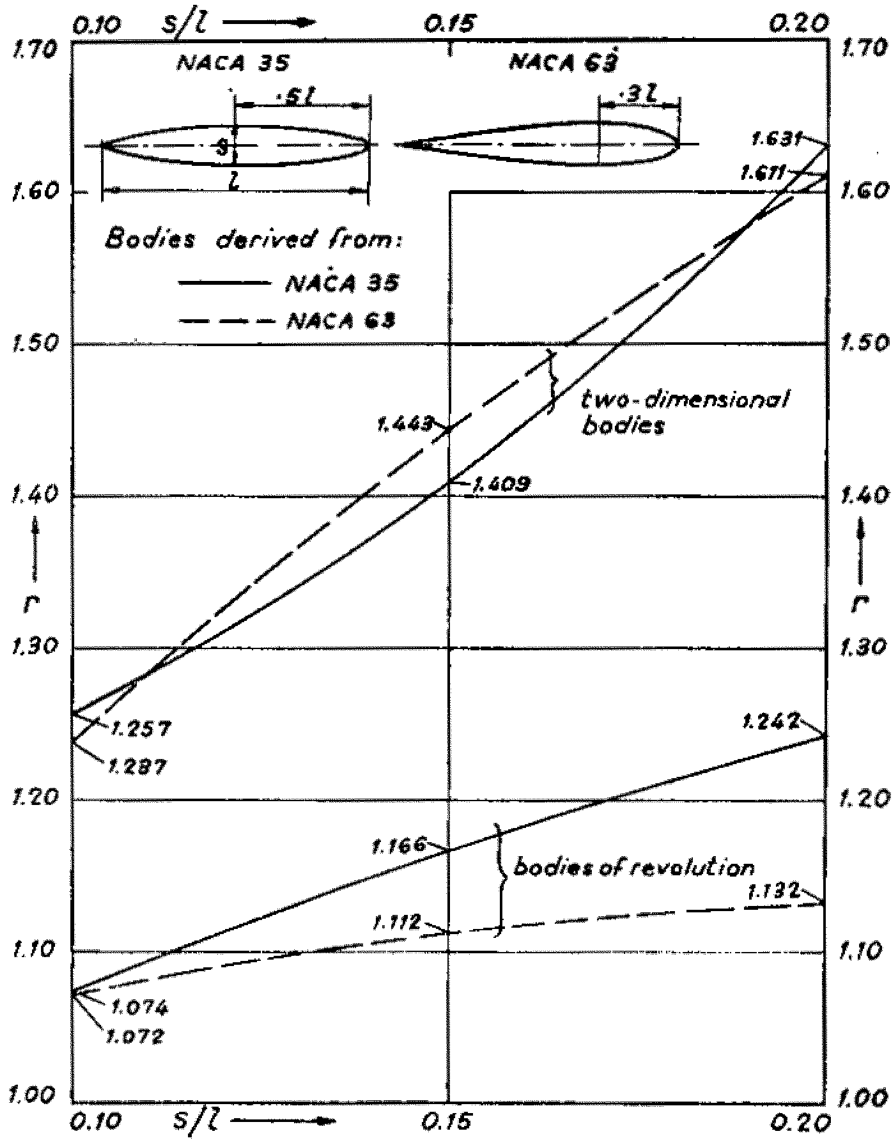


Fig. 8.—Values of form factor r, according to Scholz, for two-dimensional bodies and bodies of revolution, having as constant parallel section and as meridian section, respectively, the same NACA profile.

(c) Non-existence of flow separation. In this case, the viscous pressure resistance c_{vp} is only due to the alteration of the potential field caused by the boundary-layer thickness.

Excepting submarines, the forms of which may, more or less, approach bodies of revolution, it is evident that this particular case of three-dimensional body differs by far from that of the present-day conventional forms of normal surface ships. On the other hand, although some parts of these (for instance, vertical walls of large tankers, at maximum draught) would probably behave not very differently from a two-dimensional body, undoubtedly ship forms cannot be classified under either of the two types of bodies to which Scholz' formula applies. However, if no quantitative agreement with results of Fig. 8 is to be expected in the case of ship hulls, but only roughly intermediate values, a more approximate agreement with the formula's character (independence of r from R_n) might be hoped for, were the above conditions (a) to (c) to be fulfilled.

Condition (a), if small high-speed crafts are excepted, is practically realized by all the present hulls of normal surface ships, in rectilineal course. Also by submarines at zero incidence; lifting effects possibly developed at some of their appendages, should be considered separately.

Condition (b) is realized at the full scale. At the model scale, it can be admitted that, in a great majority of cases, using models of 6 to 7 m in length, provided with adequate turbulence stimulators [37], [38] and [39], it is also realized, at least at the speed ranges of normal routine tests. However, with certain hull forms (f. i., those having a full entrance), and, generally, when using small models, fulfilment of condition (b) is very doubtful, especially at the lowest R_n ranges.

Condition (c) is probably that about which it is most difficult to say whether it is realized or not, along the whole R_n range extending from model to ship. It should be recognised that our present knowledge of the extent of the areas suffering from flow separation is rather uncertain, and even still worse known is the amount of the corresponding resistance effect. When it is said that important flow separation is to be expected on certain hull forms, this is, for the moment, rather a surmise than something founded on proved and measured facts; the existence of important zones suffering from flow separation should lead us to exclude the forms concerned (among which those of big tankers might be typical examples) from the correlation based on a constant form factor.

Nevertheless, owing to the improvement now brought to ship lines, it might be that flow separation, when existing, is restricted to small zones and the consequent resistance effect is of little importance, except in cases of high block coefficients and full forms sharply shouldered at the after body. And even in these cases, it should be observed that flow separation, as a cause of resistance, manifests itself chiefly by two simultaneous effects of opposite signs:

- one additive, eddy-making, which increases the viscous pressure resistance;
- the other subtractive, virtual reduction of the active frictional area, which diminishes the tangential friction resistance.

Although the first effect generally prevails, undoubtedly its value is somewhat counterbalanced by the second.

For all these reasons, it is possible that in quite a number of cases resistance effect due to flow separation is small in comparison with the other viscous resistance effects, and that no appreciable raising of value c_v arises from it, or at least in any degree capable of resulting in a variation of form factor r which could be felt in the final value of the ship resistance.

The above considerations apply mainly to the full scale and to big models. With smaller models and at low R_n, the situation as regards condition (c) may become much more doubtful, especially in the case of certain full forms. If, as is frequent in such cases, there is a certain persistency of the laminar flow, this will probably influence the flow separation in two ways:

- first, as points of separation then become labil and show a tendency to go fore, the area of flow separation is abnormally extended beyond that which would be *naturally* inherent in the form, at the scale considered;
- second, since the additional pressure resistance due to flow separation may now be masked by the lessening effect of the laminar flow on the frictional resistance, a particularly confused situation may be created.

Quite misleading points can result from experimenting under such abnormal and inconsistent circumstances; one might happen on an apparently fair, but fallacious, c_v line, and even on a line having a constant form factor r, although conditions (b) and (c), which are required for the constancy of r, were both unfulfilled.

For these reasons it is not advisable to try and obtain further information by extending geosim tests to the range of very low R_n values, through the experimenting of very small models.

The constant form factor as a "Merit-Parameter":

The analysis of resistances which the form factor implies, that is: separation of the viscous resistance from the total resistance, and estimation of the former in respect to a certain basis or standard (basic friction line), confers to form factor r the character of a merit-parameter.

The independence of r from R_n, if confirmed, might upset the traditional use of R. E. Froude's merit-parameter Ⓒ since, in that case, a discrimination on a more rational basis than that used up to now would be possible, by separating form characteristics from those of size, as follows:

(a) Form characteristics.—These should be considered through two parameters, both independent from R_n, i. e., from the scale:

one ($r = c_v/c_f$) concerning the viscous resistance resulting from the three-dimensional flow. This is a function of the body form,

the other (c_w) concerning the gravity resistance resulting from the existence of a free surface in the fluid. This is a function both of body form and F_n. The greater F_n, the greater its importance, and it vanishes when $F_n = 0$. It also diminishes in bodies immersed near the free surfacce, when the immersion increases.

(b) Size characteristics.—These should be considered through the basic frictional coefficient c_f, which is a function of R_n, i. e., of V and L; and through the values S, V, etc., which are required when proceeding to the absolute values.

We believe this aspect of the form factor is most interesting. It would undoubtedly be worthy of further study. However, we would but suggest it for the moment.

(3) *The combination of a constant multiplier and a constant additive (Landweber)*:

As said when treating the correlation according to a constant form factor, flow separation is probably, in cases of ship-forms, the main cause responsible for departures from the constancy of form factor r.

Although scaling of flow separation is rather unknown, it is generally admitted that, with flow separation, the rate at which h decreases when R_n increases becomes somewhat slower than that of c_f. For these cases Landweber suggested [40] an intermediate correlation between Hughes' and W. Froude's, by adding to the proportional term a constant positive one, as follows

$$c_v = m\,c_f + n$$

where m and n are constants independent from R_n, and only dependent on the ship-form.

According to this formula, a part of the viscous resistances integrating difference $c_v - c_f$ is established through a constant multiplier $(m - 1)$ applied to c_f, and the other through a constant additive n. Proportional part $m\,c_f$ corresponds to the viscous resistances of a hypothetical body free of flow separation; constant additional part n is the resistance effect due to flow separation at the actual body. According to the relative importance of the latter, correlation will be nearer either to W. Froude's or to Hughes'.

An increasing form factor r as R_n increases, is then obtained

$$r = m + \frac{n}{c_f}$$

In Table 1, the place for this correlation would oscillate between Hughes' and W. Froude's correlations.

(4) *The linearization or constant slope (Telfer)*:

Telfer was the first who proposed, as early as 30 years ago [41], to correlate model and ship by means of families of geosims. Since then, Telfer has contributed with successive papers to the correlation problem [41]. Telfer's method may be resumed as follows:

According to W. Froude's first assumption, the iso-F_n lines corresponding to a family of geosims should constitute a parallel system. Experience shows that, generally, this parallelism is indeed practically realized.

Telfer suggested that such a parallel system which shows, with R_n or log R_n as basis, a more or less curved shape, could be transformed into a rectilineal or linearized one, if an adequate $f(R_n)$ was chosen as basis. The linearizing function which Telfer advocates is $X = R_n^{-1/3}$. The linearized iso-F_n lines then have equations of the form

$$c_t = b + e\,X$$

While b (c_t at $X = 0$), has a different value for each iso-F_n line according to the height of the F_n concerned, slope e is a form parameter characteristic of the geosims network as a whole, since it has the same value for all the iso-F_n lines of a family, and varies from one family to another.

Among the iso-F_n lines of a given geosims family, there is that which we now call viscous line c_v ($F_n = 0$). Using this line under its linearized equation, the correlation formula reads

$$c_t = c'_t - (c'_v - c_v) = c'_t - e(X' - X)$$

The final result of this formula shows that, neither any basic friction line, nor any viscous line, are required, and only slope e is to be known, to correlate model and ship. In attention to the role which slope e plays in the correlation, Telfer gave it the name of "extrapolator". Value e should be determined by means of geosim tests; however, since knowledge of viscous line c_v is unnecessary, the complete geosims network is not required, and an adequate part of it may be sufficient. For ordinary merchant ships, values of extrapolator e stand, according to Telfer, within the range 360 to 370 $\times$ 10^{-3}.

Nothing is fore-ordained in Telfer's correlation as regards modes of variation of additive h or of form factor r. Their determination requires the obtention of at least one point of viscous line c_v, in order to establish the level of this line

The constancy either of additive h or of form factor r implies of course that basic line c_f also admits a linearization, i. e., that $c_f = b_o + e_o\,X$ (Fig. 9).

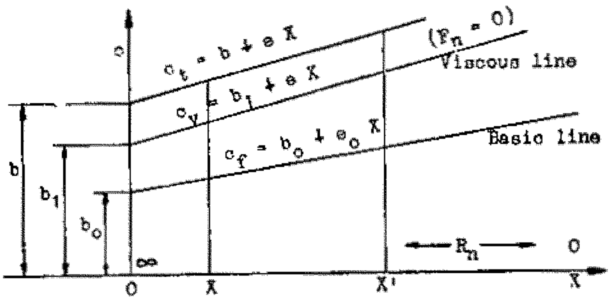


Fig. 9.—Telfer's correlation.

Additive h will then be constant when $e = e_o$. As to the form factor, the following hyperbolic law is found

$$r = \frac{e}{e_o} + \frac{b_o\,(b_1/b_o - e/e_o)}{b_o + e_o\,X}$$

This is, evidently, a particular case of the preceding correlation. According to whether $b_1/b_0 \gtreqless e/e_0$, we shall have an increasing, constant or decreasing form factor r, as R_n increases (Fig. 10). When R_n

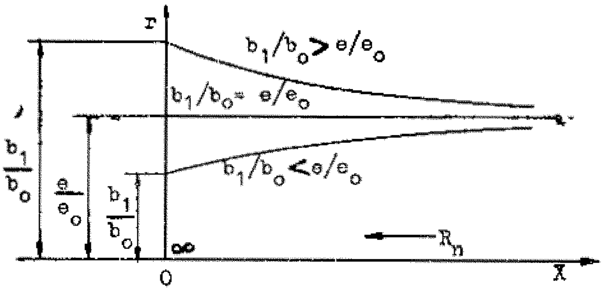


Fig. 10.—Form factor according to Telfer's linearization.

varies from 0 to ∞, form factor r varies between finite limits, from e/e_0 to b_1/b_0, respectively. On account of the nature of r, linearization should, therefore, produce a ratio $e/e_0 \geqslant 1$; this excludes the case of divergency (last line of Table 1) for Telfer's correlation.

Summing up, Telfer's method is a consequence of:

(a) The mutual parallelism of the iso-F_n lines.

(b) The existence of an adequate function $f(R_n)$ allowing linearization of the iso-F_n lines, in the model range.

(c) The assumption that this linearization, i.e., the constancy of slope e, continues up to the ship range.

Obviously, objections to Telfer's method will mainly refer to points (b) and (c).

As regards (b), we think that, although no convincing basis to Telfer's linearization is found in our present-day theoretical knowledge, from a practical point of view we see no reason for not using it, if in practice it is arrived at with sufficient approximation. In this connection it may be interesting to remark that, according to some of the newest trends, a rather approximate linearization should be expected in the model range. Hughes's basic line allows, indeed, in the model range, a close linearization ($10^3\, c_f = 1.2 + 306\, X$) as regards numerical values, although somewhat less as to where the slope is concerned (Fig. 2).

As regards (c), on the contrary, these new trends seem rather to indicate how doubtful it is that linearization, although realized in the model range, may continue up to the ship range, especially if this range reaches a certain high level of R_n. Fig. 2 shows the important differences in values and, especially, in slope, which, in the range of high R_n, exist between Hughes' original basic line and its linearization. Since errors in Hughes' original line, large enough to cover such differences are not probable, the linearization of the basic line in this range should not be accepted. Consequently, the linearization of c_v line and, therefore, of the geosims network can hardly be admitted either.

Objections to assumption (c) seem to us, even leaving aside their theoretical force, the most serious ones which, on account of their probable practical importance, may be brought forward against Telfer's method. From a general point of view, Telfer's linearization by means of function X may also be found somewhat misleading in the ship range, due to the considerable contraction applied to this range.

(5) *The constant horizontal shift (Lap-Troost)*:

Lap-Troost's general formula [10], [11], arrived at for pipes, plates and ship-forms, reads

$$\frac{0.2545}{\sqrt{c_v}} = 1.0275 + \log\,(R_n\,\sqrt{c_v}/A)$$

where:

$\log A = 1.00$ for pipes.
$= 1.98$ for flat plates (basic friction line).
$= 2.10$ to 2.50 for ship-forms (to be experimentally determined).

According to the structure of the above formula, if two values, $\log A_1$ and $\log A_2$, are introduced in it, two lines c_{v1} and c_{v2} are produced, of which, when both are plotted upon a basis $\log R_n$, the second is the result of applying to the former the horizontal shift $\log A_2 - \log A_1$.

This provides an easy method of correlating model and ship, since viscous line c_v corresponding to a certain ship-form (i. e., to a certain value of $\log A$) may simply be obtained by applying to basic friction line c_f, the horizontal shift $\log A - 1.98$ (Fig. 11).

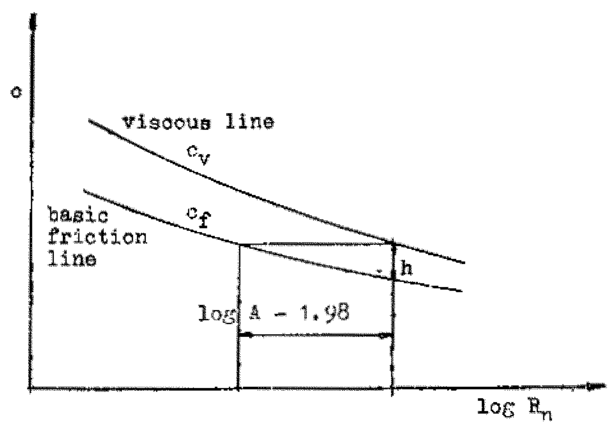


Fig. 11.—Lap-Troost's correlation.

If the above relation between c_v and c_f is compared with those established by means of a constant additive h or a constant form factor r, as done respectively in correlations (1) and (2), it is evident that,

Type	L m	V knots	F_n	R_n (*)
Small fisher	30	9 - 11	0.27 - 0.33	1.17 - 1.43 x 10^8
Trawler	50	9 - 13	0.21 - 0.30	1.95 - 2.81 "
Coaster	60	10 - 13	0.21 - 0.28	2.60 - 3.37 "
Small cargo	90	10 - 13 - 16	0.17 - 0.23 - 0.28	3.89 - 5.06 - 6.22 "
Normal cargo	120	12 - 15 - 17	0.18 - 0.23 - 0.25	5.06 - 7.78 - 8.81 "
Pass. and c. liner	180	15 - 18 - 21	0.18 - 0.22 - 0.26	1.17 - 1.40 - 1.63 x 10^9
Big tanker	200	13 - 15 - 17	0.15 - 0.17 - 0.20	1.13 - 1.29 - 1.47 "
Supertanker	280	14 - 16 - 18	0.14 - 0.16 - 0.18	1.69 - 1.94 - 2.18 "
Atlantic liner	300	28 - 30 - 33	0.27 - 0.28 - 0.31	3.63 - 3.89 - 4.28 "
Destroyer	110	25 - 30 - 40	0.39 - 0.47 - 0.63	1.19 - 1.43 - 1.90 "
Cruiser	210	20 - 30 - 35	0.23 - 0.34 - 0.40	1.82 - 2.72 - 3.18 "
Aircraft carrier	300	25 - 30 - 35	0.24 - 0.28 - 0.33	3.24 - 3.89 - 4.52 "

(*) salt water 15 °C $\nu = 1.191 \times 10^{-6}\ m^2 s^{-1}$

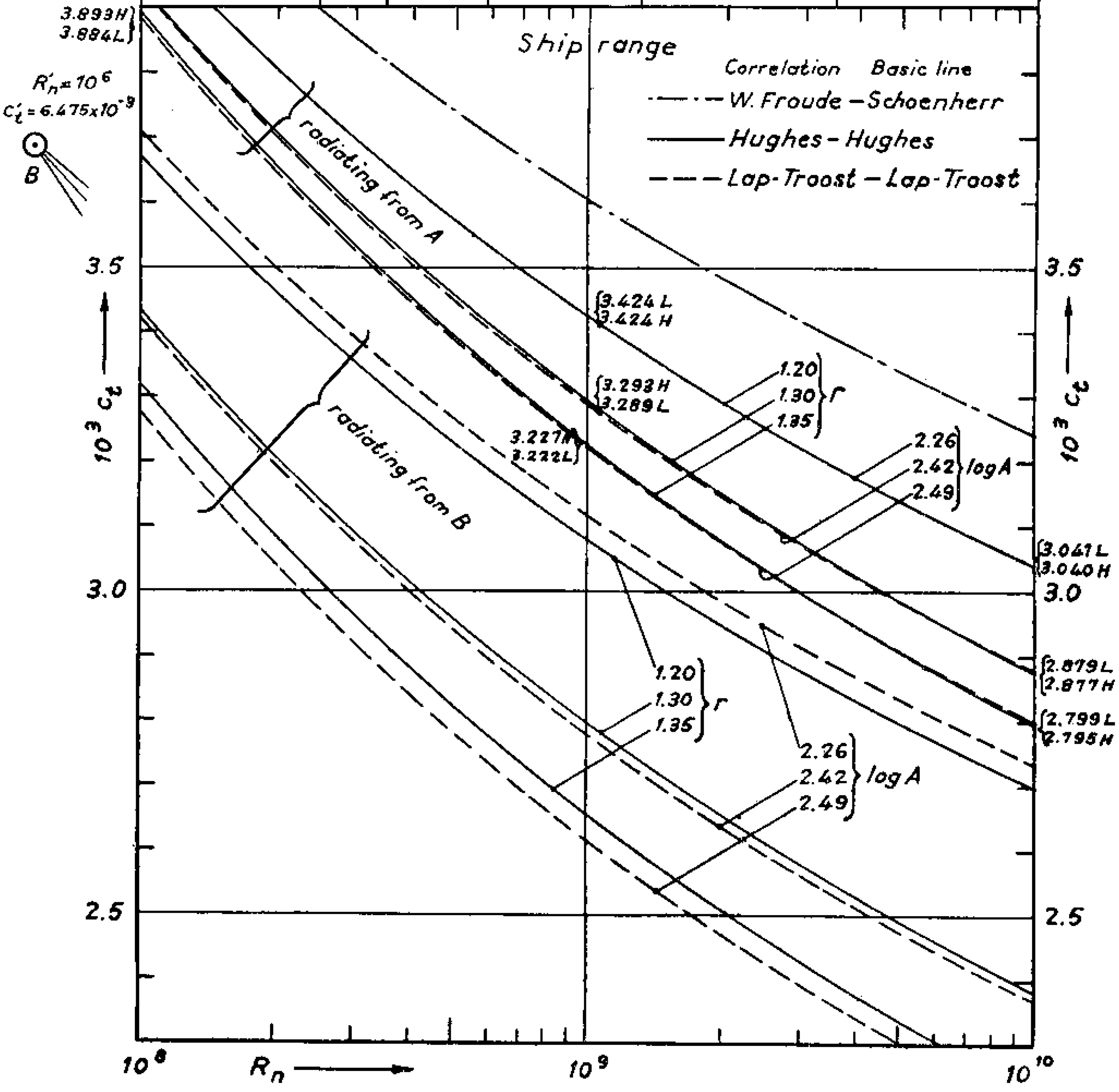


Fig. 12.—Comparison of ship predictions c_t according to Hughes' and Lap-Troost's correlations, for different values of form factor r and parameter log A, respectively. Two model spots A and B, situated at ranges $R'_n = 10^7$ and $R'_n = 10^6$, respectively, are considered.—Comparison is extended to W. Froude-Schoenherr's correlation.

while in these two we are directly attributing a quite definite and simple physical law to the viscous resistances involved in difference $c_v - c_f$, it would be difficult to explain in a physical sense the final result arrived at with Lap-Troost's correlation: that $c_v - c_f$ is equal to increase Δc_f resulting from a constant decrease $\Delta \log R_n = \log A - 1.98$, the value of which depends on ship-form. Evidently, the theoretical reasonings supporting Lap-Troost's formula remain implicit and do not become apparent in this final result. As a distinctive difference, it should also be remarked that, while correlations based on the constant additive or on the form factor constancy may be employed in association with any basic line, Lap-Troost's correlation implies association exclusively with its own basic line.

The decreasing slope of Lap-Troost's basic friction line, as $\log R_n$ increases,

$$\partial c_f / \partial (\log R_n) = - c_f / (0.2172 + 0.12725/\sqrt{c_f})$$

determines, for a constant horizontal shift, a decreasing h, that is, the convergency of lines c_v and c_f, as R_n increases. As, moreover, the rate at which h decreases is faster than that of c_f, Lap-Troost's correlation implies a decreasing form factor r. When R_n varies from 0 to ∞, form factor r varies from ∞ to 1. Lap-Troost's correlation is, therefore, in respect to W. Froude's, a departure more important than that of Hughes. In respect to Hughes', Lap-Troost's correlation departs in the sense opposed to that, which, by considering the flow separation, seems to be physically most probable. In Table 1, the place for this correlation is in the top line.

Although Hughes's and Lap-Troost's correlations obey different principles, within the limited range in which ship-model correlation takes place, differences resulting from using one or the other method are not important. Fig 12, which is similar to Figs. 4 to 7 mentioned before, shows that ship predictions c_t arrived at by following Hughes' and Lap-Troost's correlations may be made practically to coincide (lines radiating from model spot A) if an adequate correspondence between values of form factor r and of $\log A$ is provided. This correspondence is, however, somewhat dependent on the model size; Fig. 12 shows in what measure the closeness reached for the range $R'_n = 10^7$ is modified when changing to range $R'_n = 10^6$ (lines radiating from model spot B). Fig. 12 serves also to appreciate the important influence which changes of the model size have on the differences between these two correlations and that of W. Froude.

5.—FORM FACTOR AND CORRELATION FACTOR.—THEIR DETERMINATION USING THE SLOPES OF THE iso-F_n LINES.

Form factor $r = c_v/c_f$ of surface ships can be obtained from experiments in the absence of waves, either by testing surface models at very low F_n values, or submerged double-models. However, neither of these two methods provides an easy way of covering with certainty the ample range of R_n which is required, specially when the aim is to try and check a certain law attributed to factor r (for instance, its constancy), and even less when this aim is the discovery of the law or trend.

(a) As regards the lowest model R_n-range, the difficulties of hydrodynamic nature (laminar flow, instability of flow separation, etc.) arising in this range have been examined and discussed at length in Section 4; it is therefore not necessary to mention them now, nor to indicate how fallacious points may fortuitously be obtained at such a low range. Besides, measurements of very small resistances may be, relatively, lacking in precision.

(b) Regarding the highest model R_n-range, this can be reached, when using surface models at low F_n values, only with big models, since the required model length is determined by the formula

$$L' = (\nu' \sqrt{g})^{2/3} (R_n/F_n)^{2/3}, \quad \text{for} \quad 15^\circ\,\mathrm{C}(59^\circ\,\mathrm{F}) \quad :$$
$$L' \,(\text{in } m) = 0.5098\,(10^{-6}\, R_n/F_n)^{2/3}$$

If, for instance, $R_n = 10^7$ and $F_n = 0.08$, we have $L' = 12.75$ m.

However, practical reasons limit the model size, and, moreover, the use of big models can give rise to tank wall interferences. In a somewhat lesser degree, similar difficulties, to which must be added the uncorrectness and inconveniences inherent to the method, are encountered when testing submerged double-models.

* * *

Differing from the above, the method based on the examination of the slopes of the iso-F_n lines —which I recommended at the last Conference in Scandinacia— [4], [15], provides another way of arriving at the correlation. Since it allows for the formation of waves in the model tests, it practically suppresses the doubtful experimenting in the non wave-making zone and by passes most of the causes of error involved therein. This method may be resumed as follows.

Ship-model correlation, from a global and rather external point of view, may be considered and treated as a question of slope. Referring to Fig. 13, it is evident that the correlation problem, i. e., that of proceeding from spot A in the model range to spot B

in the ship range, will be solved as soon as the slope of the iso-F_n line connecting A with B is known over the whole range from model to ship. We may intend to determine the slope of this line in relation to the slope of another line, which, being known over the

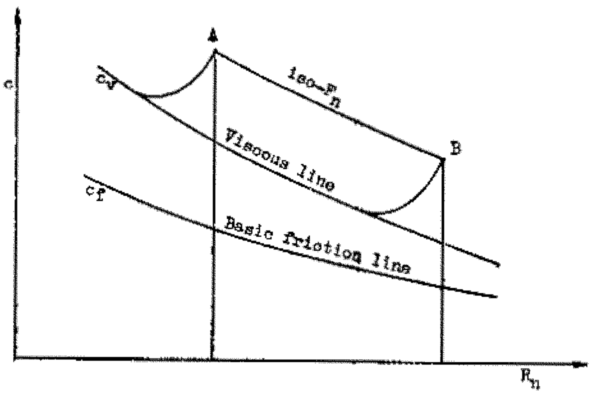


Fig. 13.

above mentioned whole range, may serve as a guide for the extrapolation.

Let us take basic friction line c_f as such an auxiliary line. Designating by m the ratio of the slopes of both lines, at the same R_n, we have

$$\frac{\partial c_t}{\partial R_n} = m \frac{\partial c_f}{\partial R_n}$$

Since $c_t = c_v + c_w$, if it is assumed that $\partial c_w / \partial R_n = 0$, we have

$$\frac{\partial c_v}{\partial R_n} = m \frac{\partial c_f}{\partial R_n}$$

This is a partial differential equation, the integration of which requires the knowledge of function $m(R_n)$. Attempts can be made in a geosim network to find simple functions $m(R_n)$, which may fit and interpret, within reasonable approximation, the experimental data; exploration should be realized in respect to different basic friction lines. Once a function $m(R_n)$ has been obtained, integration of the above partial differential equation can be undertaken. Evidently, the outlined process is, in general, not an easy one. However, there is one case in which the solution is immediate: when m is found to be independent from R_n. In this case—which, on account of some of the general physical reasons given in the preceding Sections, should certainly be expected to occur—the above equation can be integrated, and the following general expression is obtained

$$c_v = m\,c_f + n$$

where n is a constant of integration, independent from R_n and, like m, only dependent on the ship form.

Then, a form factor r results

$$r = m + \frac{n}{c_f}$$

and the correlation formula reads

$$c_t = c'_t - (c'_v - c_v) = c'_t - m\,(c'_f - c_f)$$

This formula shows that, in this case, no complete knowledge of form factor r is needed to perform the correlation from model to ship, but only a part of it —that of slopes ratio m. Slopes ratio m might, therefore, be named *"correlation factor"* or, simply, *"correlator"*.

Linearization procedures such as Telfer's and Granville's [42], may be used in the geosim analysis (").

Fig. 14 shows, in a manner similar to Figs. 4 to 7, predictions c_t resulting from using Hughes', Schoenherr's and Schultz-Grünow's basic friction formulae, in association to two values of correlation factor m (1.25 and 1.35). All extrapolation lines radiate from the same model spot ($R'_n = 10^7$, $c'_t = 5 \times 10^{-3}$) already considered in Fig. 4. Extrapolation line according to W. Froude's correlation associated to Schoenherr's basic line is also given as a reference. Fig. 14 shows that predictions c_t, with low values relatively to predictions according to W. Froude-Schoenherr's correlation, are mainly due to the presence of m and less to the use of one or the other basic formula. For an equal m, predictions c_t according to Hughes' basic formula are higher than the other two (*).

While correlation factor m is an immediate result from the slopes analysis, knowledge of additional constant n requires the obtention of some of the points of viscous line c_v. Once line $m\,c_f$ has been established from the slopes analysis, constant n is the difference between this line and the experimental spots of line c_v. Knowledge of n is evidently a further

(") Telfer's extrapolator e may be considered as a simplification of correlation factor m, in which slopes ratio becomes unnecessary, since extrapolation lines are, according to Telfer's linearization, straight lines.

(*) Comparisons at an equal m are not, of course, entirely correct, since different values m result in general from relating the same geosims network to different basic lines. Thus, values m obtained in respect to Hughes' basic line are somewhat greater than those in respect to Schultz-Grünow's basic line, according to the different slopes which each of these two lines have in the model range.

It should also be observed that constancy of m in respect to these lines is not simultaneously possible, since ratio between the slopes of these basic lines is not constant either, but it slowly augments according to the following rate:

$\log R_n$	6.0	6.2	6.4	6.6	6.8	7.0
$\dfrac{(\partial c_f/\partial R_n)_{S\text{-}G}}{(\partial c_f/\partial R_n)_H}$	1.015	1.035	1.053	1.068	1.082	1.094

Type	L m	V knots	F_n	R_n (*)
Small fisher	30	9 - 11	0.27-0.33	1.17 - 1.43 x 10^8
Trawler	50	9 - 13	0.21- 0.30	1.95 - 2.81 "
Coaster	60	10-13	0.21- 0.28	2.60 - 3.37 "
Small cargo	90	10 - 13 - 16	0.17 - 0.23 - 0.28	3.89 - 5.06 - 6.22 "
Normal cargo	120	12 - 15 - 17	0.18 - 0.23 - 0.25	5.06 - 7.78 - 8.81 "
Pass. and c. liner	180	15 - 18 - 21	0.18 - 0.22 - 0.26	1.17 - 1.40 - 1.63 x 10^9
Big tanker	200	13 - 15 - 17	0.15 - 0.17 - 0.20	1.13 - 1.29 - 1.47 "
Supertanker	280	14 - 16 - 18	0.14 - 0.16 - 0.18	1.69 - 1.94 - 2.18 "
Atlantic liner	300	28 - 30 - 33	0.27 - 0.28 - 0.31	3.63 - 3.89 - 4.28 "
Destroyer	110	25 - 30 - 40	0.39 - 0.47 - 0.63	1.19 - 1.43 - 1.90 "
Cruiser	210	20 - 30 - 35	0.23 - 0.34 - 0.40	1.82 - 2.72 - 3.18 "
Aircraft carrier	300	25 - 30 - 35	0.24 - 0.28 - 0.33	3.24 - 3.89 - 4.52 "

(*) salt water 15 °C $\nu = 1.191 \times 10^{-6}\ m^2 s^{-1}$

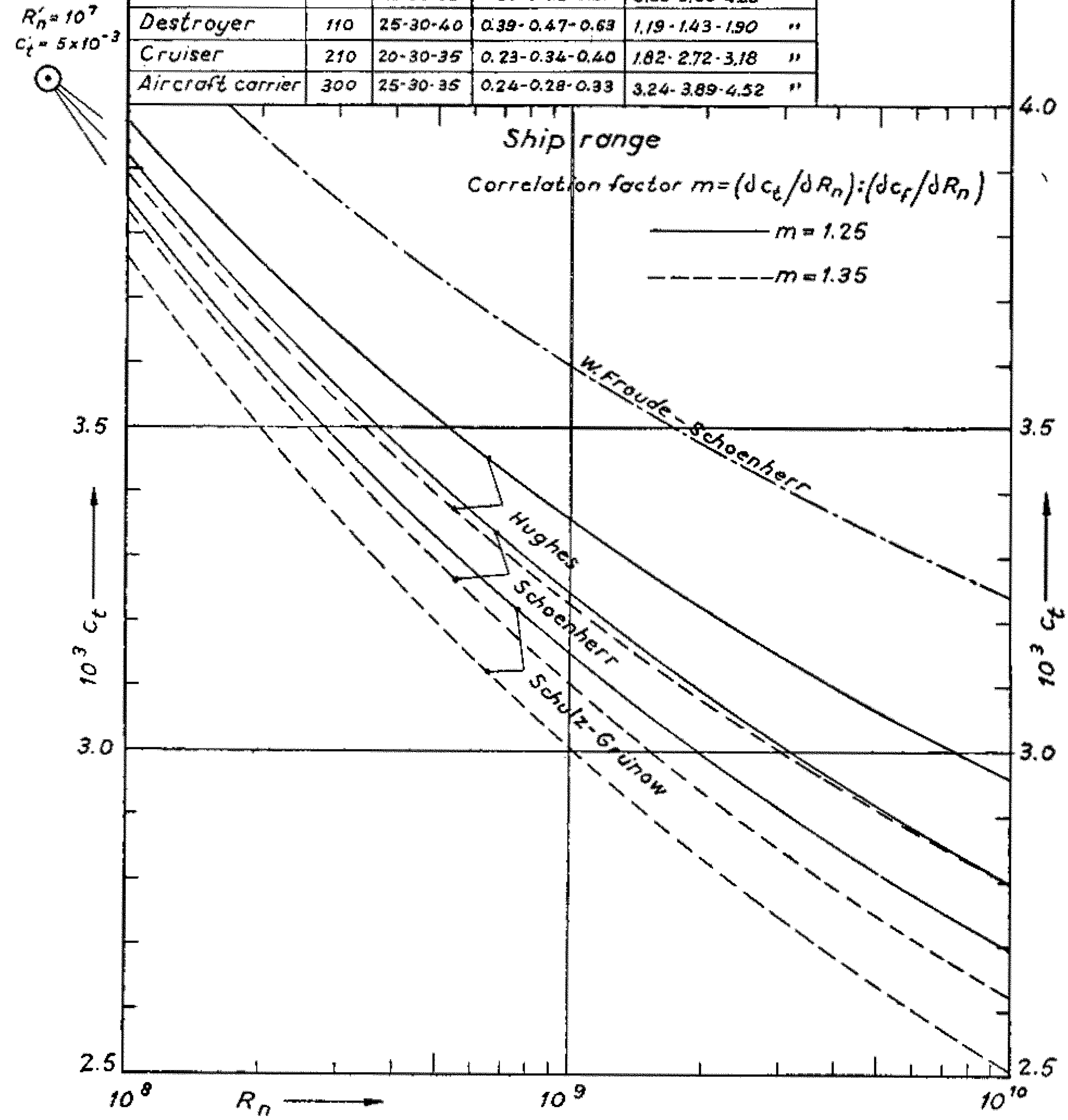


Fig. 14.—Correlation according to a constant correlation factor. Ship predictions c_t resulting from Hughes', Schoenherr's and Schultz-Grünow's basic friction lines, are compared. Comparison is extended to W. Froude-Schoenherr's correlation. All extrapolation lines radiate from model spot $R'_n = 10^7$, $c'_t = 5.0 \times 10^{-3}$.

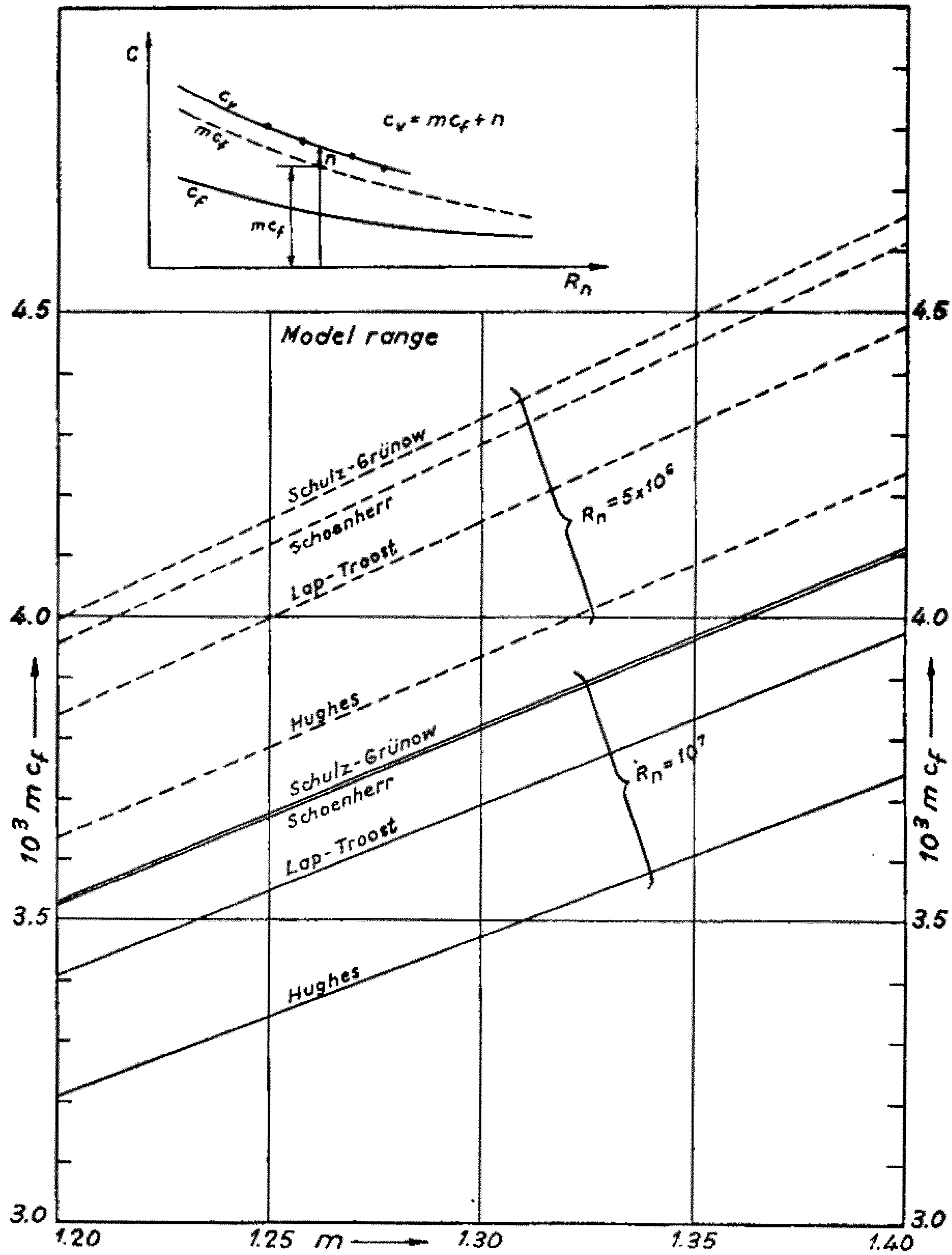


Fig. 16.—Comparison of heights of line mc_f resulting from using different basic friction formulae, at equal values of m. Two model ranges R'_n (6×10^6 and 10^7) are considered. Correlation factor $m = (\partial c_v/\partial R_n) : (\partial c_f/\partial R_n)$.

stage of the viscous resistances analysis, somewhat apart from the mere correlation. Consideration of form factor r as a "merit-parameter" may now be refined by considering splitting $r = m + n/c_f$.

Three cases are to be distinguished, as far as n is concerned:

(a) $n = 0$:

In this case, the form factor is constant and its value $r = m$. This is the Hughes correlation. In this case, form factor and correlator are the same thing.

(b) $n > 0$:

In this case, the expression

$$c_v = m\, c_f + n$$

which is arrived at, coincides with that suggested by Landweber, as an intermediate correlation between Hughes' and W. Froude's.

As said before, proportional part $m\, c_f$, and constant part n, might be interpreted as being, respectively, the viscous resistance of a hypothetical body free of flow separation, and the additional resistance due to the flow separation eventually occuring at an actual body (Fig. 15).

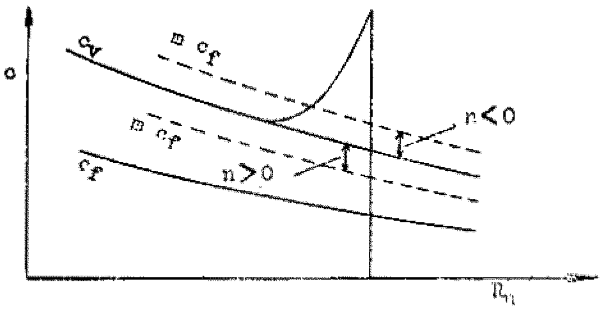


Fig. 15.

An increasing form factor r, as R_n increases,

$$r = m + n/c_f$$

exists in this case. In Table 1, this correlation comes on the third line, i. e., between Hughes' and W. Froude's correlations. Form factor r is greater than correlator m.

(c) $n < 0$:

Circumstances in this case are similar to (b), only line $m\, c_f$ is situated at a higher level than c_v, instead of lower (Fig. 15).

A decreasing form factor r, as R_n increases,

$$r = m - n/c_f$$

results in this case. Accordingly, its place in Table 1 is on the first line. Form factor r is smaller than correlator m.

Evidently, if terms $m\, c_f$ and n are physically interpreted as in case $n > 0$, which at first sight seems logical enough, the case $n < 0$ means that a subtractive resistance effect due to flow separation, exists. Although, as already said, two contradictory resistance effects are implied in the flow separation phenomena, the above sounds, certainly, strange. Whenever found, a revision of the whole process which has led to it should therefore be undertaken.

Different incorrectnesses could be responsible. For instance, the occurrence of $n < 0$ may well be a consequence of using a basic line which is wrongly situated at a too high level. The obtention of n, therefore, supplies an implicit control of the correctness of the basic line, as far as its level in the model range is concerned. It may be that Hughes' basic line, the level of which is manifestly lower than those of the others, is the only correct one in such respect. Fig. 16 shows, for the model ranges $R'_n = 5 \times 10^6$ and 10^7, the heights of line $m\, c_f$, corresponding to various basic formulae (''). Differences in respect to Hughes are certainly important. Therefore, if line $m\, c_f$ obtained by using Hughes' basic line should result very close to viscous line c_v, as has already been found in various cases, the other basic lines would most probably lead to values $n < 0$.

The occurence of a negative n could also be a consequence of having determined viscous line c_v through experimental spots which were affected by laminar flow.

6.—Resume and provisional conclusions.

(1) W. Froude's first fundamental assumption, that total resistance may be separated into two additive parts, respectively governed by Froude's and Reynolds' laws-or, in other words, the parallelism of the iso-F_n lines in a network of geosims— is practically realized in a great majority of cases. Exceptions, however, should sometimes be made for restricted zones in which either contingent causes (laminar flow, tank wall interference effects, etc.) or the mutual influences of viscosity and waves may become more or less apparent.

(2) W. Froude's correlation, based on his second assumption (constant residuary resistance), does not always correlate geosims results satisfactorily; it also leads at times, in the ship range, to predictions which are unacceptably high since they leave no margin for a reasonable roughness allowance of the actual ship. Both failures occur in a greater or lesser degree, whatever the basic line used. This seems to indicate that the constant additive $h = c_v - c_f$ is the main source of error, and that a

('') Foot-note to page 121, referring to comparisons at an equal m, is also applicable here.

certain convergence of lines c_v and c_f, different according to the ship-form, would be nearer to the truth.

Attempts to fit geosim results more satisfactorily according to W. Froude, or to obtain, also by means of this correlation, reasonably lower ship predictions, by sligthtly varying the slope of the basic line, could be justified if constant additive h were the correct correlation and only the slope of the basic line were wrong. However, since this is not the case, varying the slope of the basic line really amounts to despoiling it of its physical content, merely for the practical convenience of keeping W. Froude's correlation. The result is that no general solution valid for any ship-form is reached, while even the practical aim pursued is only partially achieved.

(3) Correlation by means of a constant form factor r, as proposed by Hughes, applies, according to theoretical reasonings, to bodies of revolution and (with higher numerical values of r) to two-dimensional bodies. In both cases turbulence, the absence of lifting effects and of flow separation are required.

The present-day conventional forms of normal surface ships cannot be classified under either of the two types of bodies mentioned above and, therefore, no agreement with one or the other type as regards the numerical values of r should be expected in the case of ship-forms. However, a certain agreement with its character (independence of r from R_n) might be hoped for, were requirements about turbulence, the absence of lifting effects and of flow separation, to be fulfilled. The importance in this connection, of securing turbulent flow in the model testing is emphasized.

(4) Flow separation is probably, in cases of ship-forms, the main factor responsible for departures from the form factor constancy. Although the scaling of flow separation is largely unknown, it is at present generally admitted that, when flow separation exists, h decreases as R_n increases at a somewhat slower rate than c_f. Consequently, departures from the form factor constancy due to flow separation should lead to an increasing form factor, as R_n increases.

(5) In summing up paragraphs under (3) and (4), it could be said that the correlation according to a constant form factor, as proposed by Hughes; or, eventually, the more general correlation which combines a constant multiplier and a constant additive, as Landweber suggested, is in our opinion, a logical and simple interpretation of the real facts, and probably closer to the truth than all the other proposals.

(6) Although Lap-Troost's correlation obeys different principles than Hughes' correlation, it is possible, within the limited range in which ship-model correlation takes place, to provide an adequate correspondence between values of log A and form factor r, so that ship predictions arrived at by one and the other correlation practically coincide. The correspondence between log A and r is, however, not a general one, but somewhat dependent on the model size. From a physical point of view, it should be said that the direction in which Lap-Troost's correlation departs from Hughes' is opposed (a decreasing form factor) to that which seems most probable when considering flow separation. It might also be remarked that, while the other correlations considered may be used in association with any basic line, Lap-Troost's correlation implies association with its own basic line.

(7) Telfer's linearization by means of function $X = R_n^{-1/3}$ is probably approximately realized in the model range, at least as to the numerical values concerned, although somewhat less as regards the slope. However, the assumption that this linearization continues up to the ship range appears to be rather doubtful, especially if this range reaches a certain high level of R_n.

(8) Considered from a rather external point of view, model-ship correlation is a question of slope, the correlation problem will, indeed, be solved as soon as the slope of the iso-F_n lines is known over the whole range from model to ship. This slope may be determined in relation to the slope of another line known all the way through.

Taking basic friction line c_f as such an auxiliary line, if m is the ratio of slopes $(\partial c_t/\partial R_n)/(\partial c_f/\partial R_n)$ at the same R_n, and $\partial c_w/\partial R_n = 0$ is assumed, the correlation problem consists in solving the partial differential equation $\partial c_v/\partial R_n = m(\partial c_f/\partial R_n)$. Attempts can be made in a geosim network to find simple functions $m(R_n)$, which may fit and interpret, within reasonable approximation, the experimental data; explorations should be realized in respect to different basic friction lines.

(9) Although the process outlined in (8) is, in general, not an easy one, there is one case —when m is found to be independent from R_n— in which the solution is immediate. In this case, the above partial differential equation has the solution $c_v = m\, c_f + n$, where n is, like m, independent from R_n and only dependent on the ship-form. According to the correlation formula, which now reads

$$c_t = c'_t - m(c'_f - c_f)$$

knowledge only of m is required to perform the correlation from model to ship.

Value of m ("*correlation factor*" or "*correlator*") is a direct result from the slopes analysis.

(10) Knowledge of n, which requires that some points of viscous line c_v be obtained, is a further stage of the viscous resistances analysis, separate from the mere correlation. Three cases are to be distinguished as far as n is concerned:

$n = 0$, $c_v = m\,c_f$, $r = m$; this is the Hughes correlation.

$n > 0$, $c_v = m\,c_f + n$, $r = m + n/c_f > m$; this coincides with Landweber's suggestion. A physical interpretation in this case is that $m\,c_f$ represents the viscous resistance of a hypothetical body free of flow separation, and n represents the additional resistance due to the flow separation eventually occurring at an actual body.

$n < 0$, $c_v = m\,c_f - n$, $r = m - n/c_f < m$; if the same physical interpretation as in the precedent case is given, the case $n < 0$ seems to be unnatural, since it implies a subtractive resistance effect due to flow separation. A revision of the whole process which has led to $n < 0$ should therefore be undertaken. Various incorrectnesses could be responsible; for instance, a basic line which is wrongly situated at a too high level; having determined viscous line c_v through experimental spots affected by laminar flow, etc. The obtainment of n therefore supplies an implicit check of the correctness of the process applied.

(11) On account of some general theoretical reasons and because of the physical interpretation given in the above to the solution $c_v = m\,c_f + n$, constancy of slopes ratio m should certainly be expected to occur. Therefore, when exploring in respect to different basic friction lines, as indicated in (8), the finding of a vertical convergence of lines c_v and c_f as R_n increases, might certainly be considered as a primary requirement to be fulfilled by the correct basic line, as far as its slope is concerned.

(12) Taking into account the great insecurity involved in determining the slope of viscous line c_v from the experimental spots obtained in the non wave-making zone, it seems highly advisable to determine correlation factor m through the slopes analysis of the iso-F_n lines (preferably the rather high ones, provided they are free of tank wall interferences, or these effects have been duly corrected (")). Experimental spots of viscous line c_f should be used exclusively to establish the level of this line; this is easier than to determine the slope, and any error is of lesser consequence, since n does not intervene in the model-ship correlation.

(") Since the question of tank wall interferences is of little importance in El Pardo Tank on account of its large cross-section, this question has not been considered in the present contribution.

(13) According to (10), in normal cases, form factor r may be equal or greater than correlator m. An undue identification of both factors may well have fortuitously taken place in the past, as a consequence of more or less forcing the obtainment of a constant form factor r. Of course, the application of a smaller correlator m will produce ship predictions c_t somewhat higher than would be obtained by applying Hughes' pure proposal.

(14) Since the irregular phenomenon of flow separation has been excluded from m, to establish a systematic dependence of m on the ship-form particulars, should undoubtedly be easier than establishing that of r, in which flow separation effects are involved.

To obtain such a systematical dependence seems to be the most immediate task which should be faced. A systematical programme of geosim tests could be arranged by the Skin Friction Committee, and its realization shared among the various Towing Tanks.

In carrying out such a programme of geosim tests, the following considerations, among others, should be taken into account:

(a) Experiments should be so arranged that a direct determination of the iso-F_n lines is achieved, that is, models should be run exactly at the speeds corresponding to the contemplated iso-F_n lines; spots between these lines may be disregarded.

Until now this has not been done: instead, iso-F_n lines have generally been established by the crossing of the relevant ordinates with the resistance curves of normal model tests. It is well known, however, that such crossing points are very imprecise owing to the sharp angle at which the crossing occurs, and, moreover, while iso-F_n lines present a slight, gradual curvature, resistancce curves c_t are rather irregular as they contain the wave interference. Consideration of the latter is not particularly relevant to the aspect of the correlation which we are now considering.

If one proceeds as suggested above, the conversion to a certain temperature of points which were run at a different one —a process frequently applied— is thus avoided. Such a temperature correction involves calculations according to a method the correctness of which we are discussing.

(b) To run a greater number of models at different scales, than up to now, is absolutely necessary. It is, indeed, quite illusory to try to determine with any certainty an iso-F_n line by 3 spots, or, even 4, which has been the number most generally available.

(c) It is possible by taking advantage of the annual variation of the water temperature, to get more than one point on each F_n line, by employing only a single model. In this case, even though precautions have been taken to prevent any deformity in the model, it should be re-milled shortly before testing.

(d) As proposed in (12), greater use than previously, should be made of the higher iso-F_n lines.

(e) It is, undoubtedly, a handicap that iso-F_n lines have to be determined by testing different models. This makes it necessary to be extremely careful to reach the highest possible uniformity in the models and testing conditions: the material and the finish of the models (especially as regards surface pores), the length of time that the model is immersed in water from its completion until the testing, turbulence devices, etc.

It is probable that by giving greater attention to the above considerations, more satisfactory geosim results could be rendered than those obtained up to now. A revision along such lines, of all our previous geosim tests is now under way in El Pardo Tank.

It would be very helpful to the Towing Tanks engaged in this particular work, and for a better coordination among them, if the Skin Friction Committee in arranging the aforementioned geosim programme, would include some instructions, similar to those above (a) to (e), giving some details particularly connected with such special tests. The measurement of the water temperature could be the subject of an additional instruction. A suitably graduated table of kinematic viscosity of fresh and salt water versus temperature, should also be prepared by the Skin Friction Committee and proposed for general use, once approved by the Conference.

(15) Coming now again to the question of the basic friction line, two remarks should still be made:

As regards the slope:

When using W. Froude's correlation, Hughes' basic line leads to ship predictions c_t somewhat higher than those obtained by using Schultz-Grünow's, Schoenherr's, or Lap-Troost's basic lines in the case of the model ranges of Figs. 4, 5 and 6; for very small model ranges as in Fig. 7, Hughes' line predictions c_t are lower than the other two.

When Hughes' or Landwber's correlations are used, ship predictions c_t according to Hughes' basic line are also somewhat higher than those resulting from using Schultz-Grünow's or Schoenherr's basic lines, for an equal value of correlator m (Fig. 14).

As regards the level:

It may well be that Hughes' basic line on account of its lower level, relatively to the others, is the only correct one in order to prevent a negative n, as explained in (10). Fig. 16 shows, for the model ranges $R'_n = 5 \times 10^6$ and 10^7, the heights of line $m\,c_f$, corresponding to various basic formulae. Differences in respect to Hughes' are certainly important. Therefore, if line $m\,c_f$ obtained by using Hughes' basic line should result very close to viscous line c_v, as has already been found in various cases, the other basic lines would most probably lead to negative values of n.

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Capt. M. L. Acevedo (Final conclusions) (*).

(a) In principle, our opinion is quite inclined to the correlation by means of a constant correlator $m = (\partial c_t/\partial R_n)/(\partial c_f/\partial R_n)$.

Although in quite a number of cases, possibly, correlator m and form factor $r = c_v/c_f$ have practically the same value, we prefer to speak of correlation according to a constant m than according to a constant r, since the former is more general and the second is a particular case of it.

(b) It is considered most urgent to start with a programme of systematic geosim tests. In carrying out this programme, explorations in respect to Hughes' basic line seem to be particularly interesting in view of what has been stated, about the constancy of m, convergence of lines c_v and c_f, and prevention of a negative n.

(c) As regards the proposals suggested as provisional solutions to improve W. Froude's correlation for the time being, i. e.:

(*) These final conclusions having been written just before the Conference, it was possible to include in them some comments on Dr. Hughes' second contribution "Additional statement".

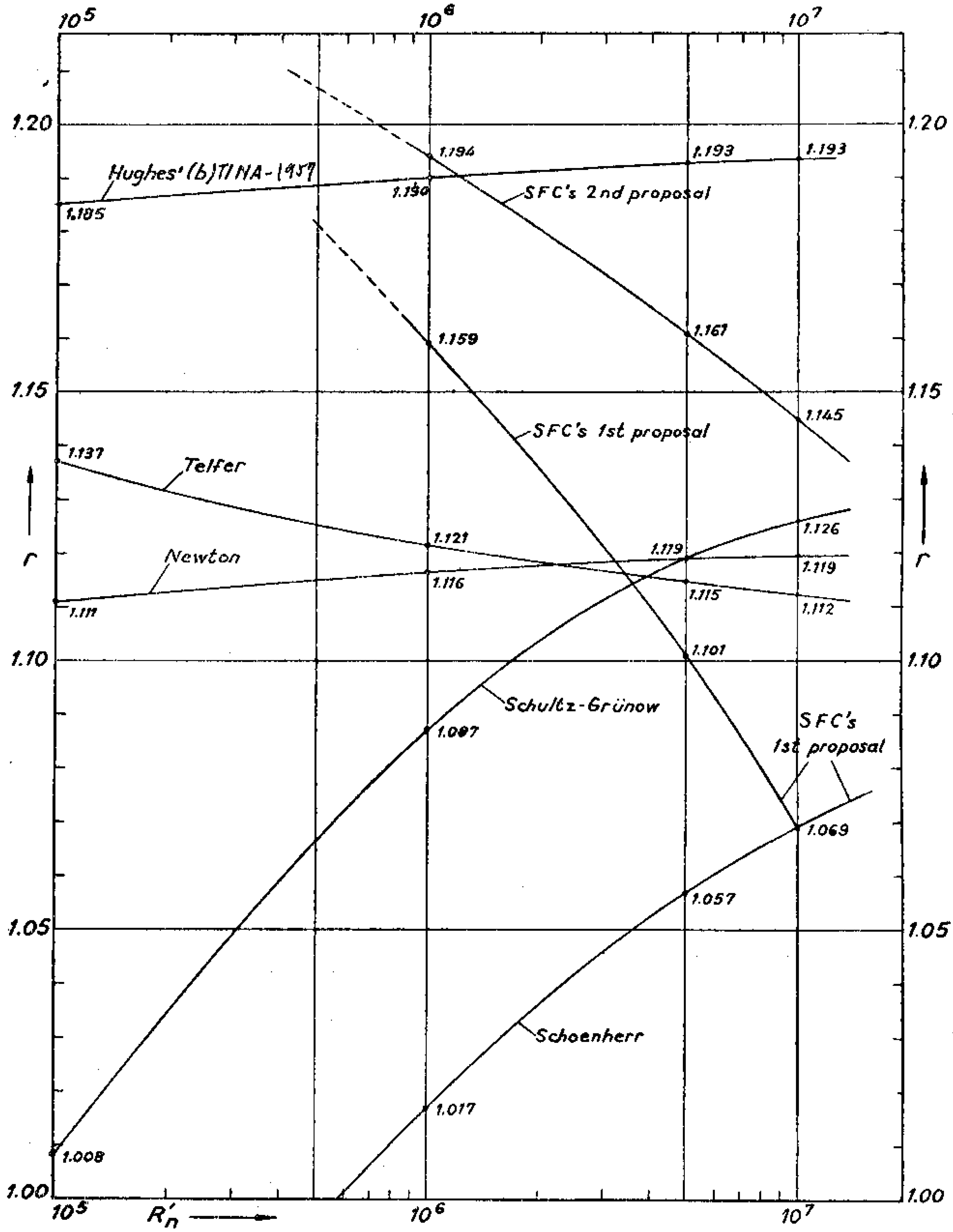

Fig. 17.—Values r to be used as a constant form factor in association with Hughes'basic line, in order to obtain in the ship range ($R_n = 10^8$ to 10^{10}) practically the same predictions c_t which are obtained by using W. Froude's correlation in association with the basis friction lines mentioned on each curve.—Values r are plotted as function of the model range R'_n from which model-ship extrapolation starts.

Skin Friction Committee's Report, 1st and 2nd suggestions,
Hughes, proposal (b), INA, $c_f = 0.080/(\log R_n - 2)^2$
Newton, $c_f = 0.075/(\log R_n - 2)^2$
Telfer, $c_f = 0.070/(\log R_n - 2.12)^2$

a comparison of all of them is shown in Fig. 17. Schoenherr's and Schultz-Grünow's original lines have also been considered in this figure.

The form in which this figure is presented, allows for a mutual comparison of the different suggestions, and of all of them in respect to Hughes' correlation and Hughes' basic line. The figure shows that, while Hughes', Telfer's and Newton's proposals may be considered as the result of multiplying Hughes' original basic line by a practically constant correlator (approximately 1.19 for Hughes' and 1.12 for Telfer's and Newton's), the factors involved in the two proposals of the SFC. vary appreciably when different model ranges are considered, increasing as R_n decreases, i. e., as the model size is reduced. A similar tendency to the latter, but in the opposite direction, is shown by Schoenherr's and Schultz-Grünow's original basic lines.

If we confide a certain trust in the future of the correlation by means of a correlator independent of R_n and only depending on the ship-form, all the above proposals have the incorrectness of transferring to the final allowance, in a greater or lesser degree according to the ship type, a part of the influence of ship-form, which ought to have been accounted for by the correlator itself. Moreover, SFC's proposals (as well as Schultz-Grünow's and Schoenherr's original lines), present the illogical occurrence that the correlator varies when the model size is changed, which, as stated above, does not practically occur with the three last proposals. From this aspect, we consider that Hughes' (b), Telfer's and Newton's proposals are more acceptable than either of the two proposed in the Report of the SFC. Of course, not any of the newly proposed lines may be used in the viscous analysis indicated in point (10) of our Resumé.

Furthermore, when model range equals or is greater than $R_n = 10^7$ (the range corresponding to the model size normally used in large Tanks), the first SFC proposal differs in no way from Schoenherr's original line. It is also noted that for this normal model Reynolds range (about 10^7), the value which would be achieved by using the new proposals, would be practically the same as those obtained by employing Schultz-Grünow's line.

Finally, it is observed that the probability of finding negative allowances becomes less with the two SFC proposals, and greater with Schoenherr's and Schultz-Grünow's original lines, respectively, when the model size is reduced. And that, with Hughes' (b), Telfer's and Newton's proposals, changes of the model size leave such a probability practically unaffected.

(d) As a general position of ours, consequent on our views, we would really not feel very happy in adopting an "engineering line", such as these new proposals would amount to, in effect.

Since, however, present knowledge is not mature enough to permit, with the completeness required for practical use in Towing Tanks, the adoption of a correlation by means of a constant correlator (which seems, in my opinion at least, to be the correlation of the future), we are prepared, for sake of the mutual co-ordination, to add our agreement to the opinion of the majority of the Conference, if this were in favour of adopting a definite proposal based on the lines of those mentioned in (c) above.

In this case, our view would favour Hughes' proposal (b), which is equivalent to the inclusion of a correlator practically independent of R'_n, and having a more normal average value than Telfer's and Newton's proposals (which two are practically the same).

(e) If such a proposal is adopted, it should clearly be stated that its adoption is quite provisional, and that it will not interfere with the future adoption of a more scientific solution, the pursuit of which should be continued with the greatest possible effort.

Capt. M. L. Acevedo (Written contribution to Subject 4: Turbulence Stimulation).

Trip wires have been used as turbulence stimulators in ship-model tests ever since the late Dr. Weitbrecht introduced them, about 25 years ago, as a standard practice in the Berliner Tank. Other turbulence-producing devices (sand strips, studs, etc.) have been in use just as long or even more. But in spite of the many years elapsed since then, during which valuable papers and contributions on the subject have been produced (1), (2), (3), turbulence stimulation in ship-model tests cannot yet be considered as a solved problem.

This contribution is an attempt to assess the efficiency of the trip wire device, in the light of a new transition criterion due to Fage (4). It supplies a way of ascertaining the ranges within which, and the location at which, trip wires may be used efficiently.

According to Fage, transition in the boundary-layer from laminar to turbulent flow will be produced by a surface wire, at the wire itself, in a zone where the flow would be laminar without it, if, and only when,

$$\frac{u\,d}{\nu} \geqslant 400$$

where:

u = velocity in the laminar boundary-layer at wire location (without the wire), at a distance from the wall. d
d = wire diameter.
ν = kinematic viscosity of the fluid.

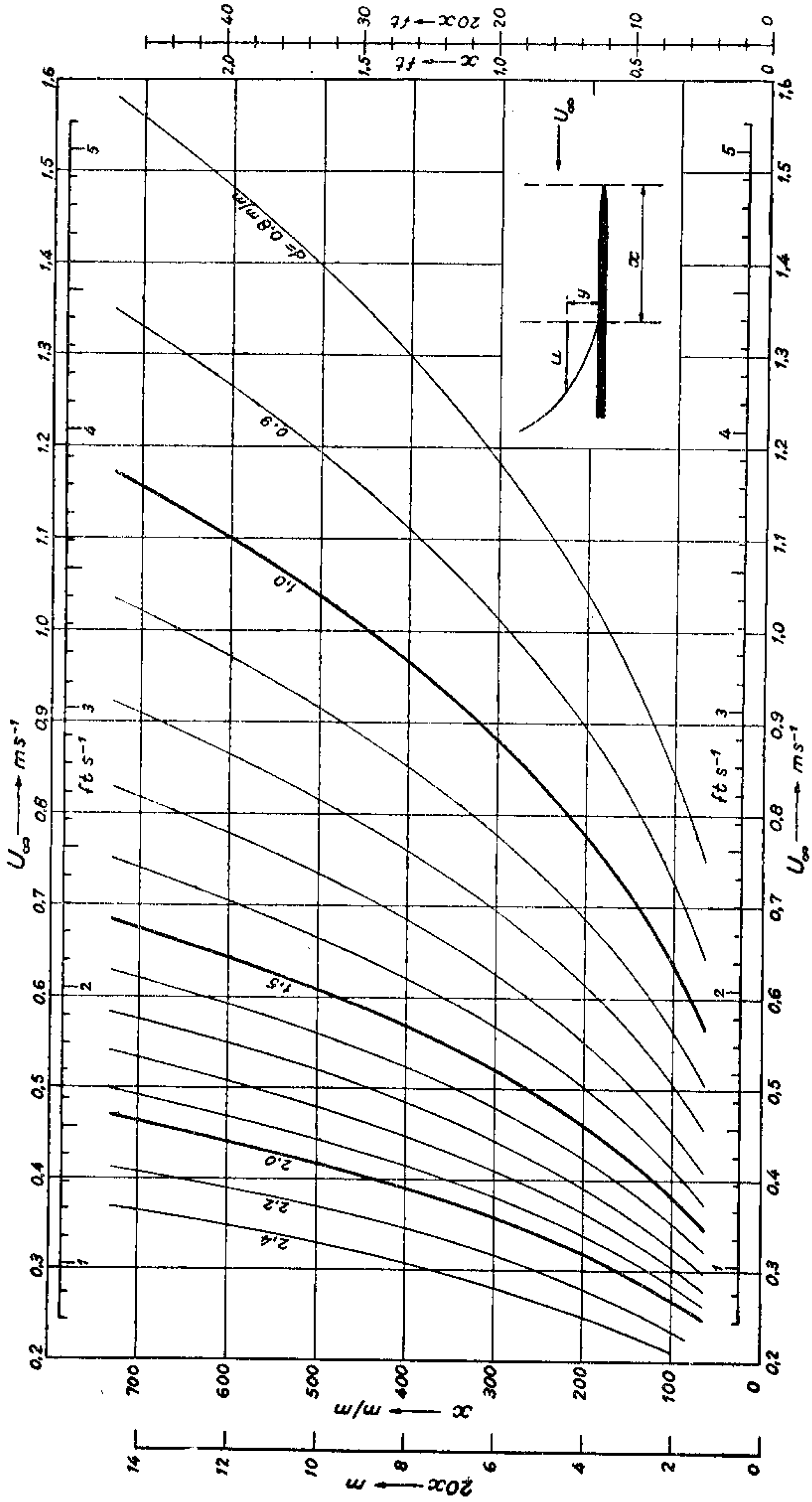

Fig. 1.—Maximum abscissa x, admissible for the wire location in accordance with Fage's criterion. Temperature 15° C, kinematic viscosity = 1.144×10^{-6} m^2 s^{-1}

Fage's criterion, which adopts the form of a Reynolds Number, shows that, depending on the wire diameter d, a minimum inflow velocity u is required in order that the trip wire may act as an effective turbulence stimulator, i. e., that the flow, arriving laminar at the device, become turbulent at the device itself.

The laminar boundary-layer theory makes it possible to establish a relation between the transverse velocity distribution in the boundary-layer

$$\frac{u}{U_\infty}$$

and the dimensionless co-ordinate

$$\eta = y \sqrt{\frac{U_\infty}{\nu x}}$$

where:

U_∞ = undisturbed velocity of the laminar flow (velocity at which the model is run in the case of a model test).
u = velocity at point (x y) of the boundary-layer.

By introducing in the above expressions the minimum inflow velocities u resulting from Fage's criterion and the corresponding ordinates $y = d$, an abscissa x will be obtained, which will be the maximum distance from the entrance edge, at which the trip wire of d diameter may be located if, according to Fage, it is to be an effective turbulence stimulator of the flow U_∞ under consideration.

Using Howarth's computation (6) of function $u/U_\infty = f(\eta)$, the calculation outlined above has been pursued for different wire diameters d and test velocities U_∞ The value

$$\nu = 1.144 \times 10^{-6}\ \mathrm{m^2\ s^{-1}}\ (1.231 \times 10^{-5}\ \mathrm{ft^2\ s^{-1}})$$

corresponding to a temperature of 15° C (59° F) has been taken.

The results of this calculation have been plotted in fig. 1. This shows the maximum abscissa x at which the wire may be located, in function of velocity U_∞, for different wire diameters d. The scale 20 x, on the left of this diagram, relates model length to trip wire locations when, as usually done, the trip wire is located at Section 19.

In order to furnish direct information concerning this particular case, a second diagram (Fig. 2) has been drawn, in which the values of d upon U_∞ , for model lengths 2, 4, 6, 8 and 10 metres are given.

Conclusions.

Fig. 1 in general, and Fig. 2 in the particular case of a trip wire located at Section 19, allow a thorough examination, from different points of view, of the question under study.

Among others, the following conclusions may be drawn:

(1) For a given wire diameter d, the value of maximum abscissa x diminishes as velocity U_∞ decreases. For each d, a limit of U_∞ exists, under which the trip wire ceases to have effect as a turbulence stimulator. The greater d, the lower this limit.

(2) Trip wires as usually applied (0.9 to 1.0 mm diameter, at Section 19) may be considered as satisfactory for velocities of advance higher than 0.8 to 0.9 m s^{-1} approximately. This type of device will therefore be generally satisfactory for normal routine tests, at least in the range of the service speeds. However, with certain hull forms, e. g., those having a full entrance, extra precautions may be necessary.

(3) At velocities lower than about 0.8 to 0.9 m s^{-1}, the mentioned usual device is no longer satisfactory. It should be regarded as definitely insufficient at the lowest velocity ranges used to determine the form factor in the non wave-making zone. Velocities between 0.3 and 0.4 m s^{-1}, for instance, will require trip wires of 2 mm diameter and over.

(4) With usual trip wire location at Section 19, at a constant speed of advance, the greater the model length, the greater the wire diameter should be.

(5) The contrary occurs when tests are made at a constant Froude Number; the shorter the model, the thicker the wire should be. This is clearly shown by the transversal iso-F_n curves of Fig. 2. The considerable slope increase of these curves, as lower F_n values are considered, should be noted.

(6) Since the calculations of this report are based on Fage's lowest figure $u\ d/\nu = 400$, the wire diameters resulting from Figs. 1 and 2 are minimum values. It might therefore be a reasonable precaution to increase them slightly in practice.

(7) As our calculations are based on a temperature of 15° C (59° F), results shown in Figs. 1 and 2 should be corrected for other temperatures.

(8) Since Howarth's estimation of function $u/U_\infty = f(\eta)$ refers to the case of the laminar flow along a flat plate at zero incidence, results shown in Figs. 1 and 2 do not include the influence of the potential flow existing around a ship model. The nature of the factors involved makes it impossible to conclude generally, whether, as compared to the case of the plate, the influence of this potential flow on the inflow velocity at the wire, would be additive or subtractive, that is, in the sense of favouring or retarding the transition at the wire. With full forms having a great angle of entrance (one of the cases in which laminar persistency is most to be feared), and with usual model lengths and trip wire location, conditions at the place of the wire would probably be unfavourable, since the wire would likely find

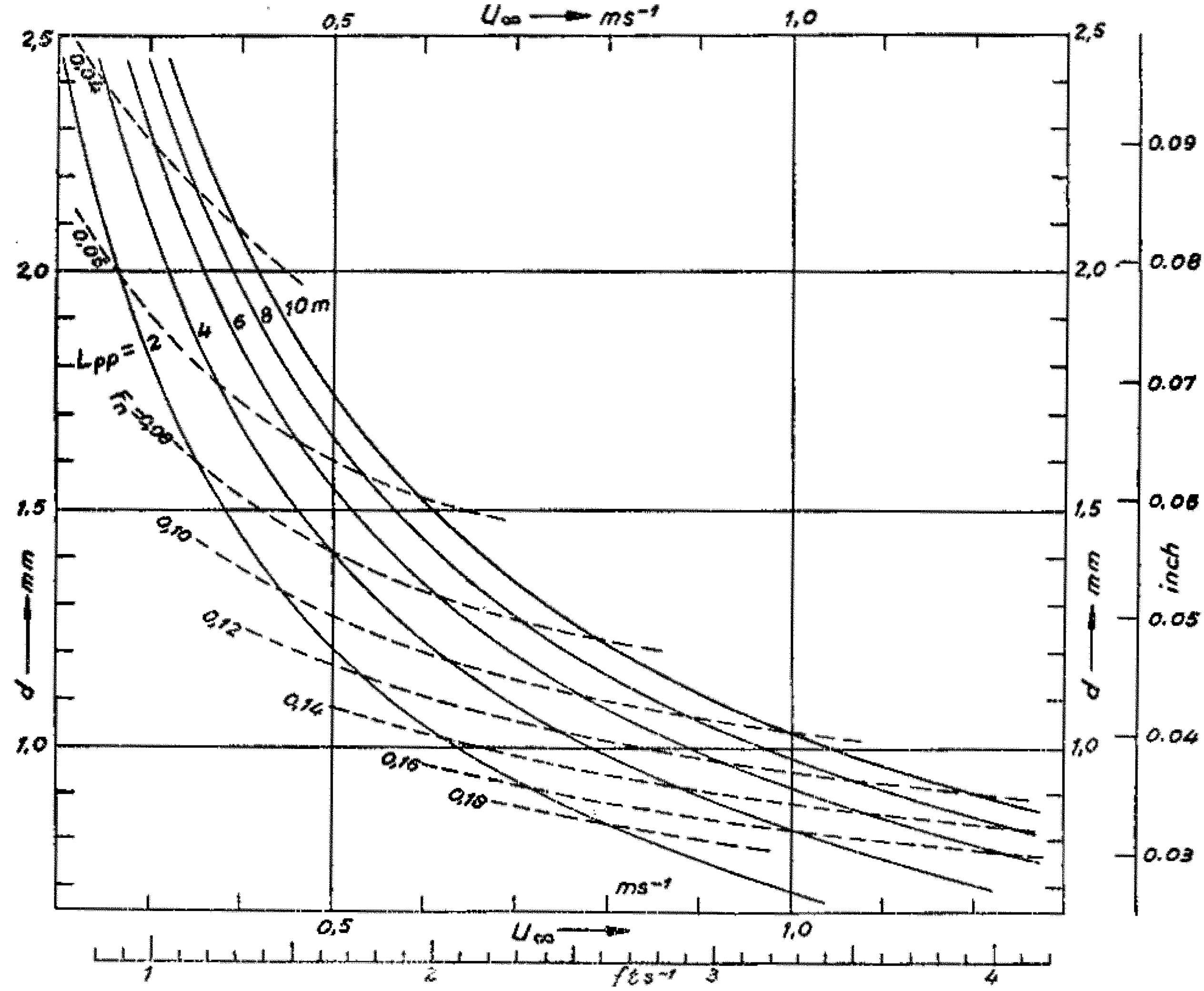

Fig. 2. Wire diameters d, according to Fage's criterion, for different model lengths, in the case of wire location at Section 19. Temperature 15° C, kinematic viscosity = 1.144 × 10⁻⁶ m² s⁻¹

itself in a potential subvelocity zone. Although the main influence exercised by the potential flow on the transition process is to be found in the consequences resulting from the longitudinal velocity gradient, rather than in the small possible alteration of the inflow velocity at the wire, this influence should also be considered, as far as possible.

(9) Fage's criterion extended to the full scale shows how improbable it is that any laminar flow may exist at normal service speeds in the full size, however small the ship roughness and even disregarding the possible initial turbulence of sea water.

(10) The recommendation in favour of wires somewhat thicker than those used at present, arrived at in this report, is an agreement with Weinblum's information (7) regarding both the practice followed at the Berlin Tank and some of Chenowitz' investigations in Leningrad.

(11) The experiments (such as published) on which Fage's criterion is based, may be considered as somewhat too limited. In any case, pursuance of these experiments seems highly desirable in view of the practical form which, as this contribution shows, such a criterion can take for its application to ship-model tests.

(12) Finally, it is suggested to try square section wires instead of the normal round ones. The simple device of square cutting the leading edge in order to stimulate turbulence has been used by Hughes (8), apparently with success, in his experiments of plates.

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Prof. G. S. J. Aertssen (Written contribution).

As stated in the report, the Froude assumption still remains the basis of all ship-model correlation work, the frictional resistance being calculated from the Froude friction coefficients or from the Schoenherr line. It offers no doubt that our knowledge on ship hydrodynamics has been improved so far that there is a need for a better approach to the correlation problem. The Hughes method is atrractive but it is felt that the amount of correlation work of measured-mile trials based on this method should be made more extensive before the method could be definitely adopted.

The writer has studied with great attention the last Bristol I. N. A. paper of Dr. Hughes and his three proposals. Perhaps the greatest value of his paper lies in the last paragraph where Hughes emphasizes that his "proposals are made for application to model results which have first been corrected to what might be called the model standard". The three corrections to be made are for the laminar flow area, the drag of the turbulence stimulator, and the tank boundary interference. For any ship-model correlation they should not be ignored. Rather than adopting an I. T. T. C. line it seems more urgent to come to a standardization of the model technique and the correlation work.

The proposals of Barrillon and Kempf regarding repeated resistance tests of the same model in various tanks merit thorough consideration and the writer wonders whether these tanks would not include propulsion tests in this programme of research. Moreover, it would be very interesting to complete this programme with a correlation work on full scale. Resistance tests on ships would undoubtedly be very useful in this scheme and from a scientifical point of view they are needed. Froude based his correlation work on them. But they are expensive and the resistance tests with a Victory ship are in their scheme period. Full scale propulsion trials would be, at this stage of research, an engineering solution. The measured-mile trials and the ship-model correlation work on a series of sister ships would be standardized as well as the model technique.

The writer has a strong feeling that the discrepancies in the trial results between sister ships are not bigger than those of the model results between different tanks. Therefore it is recommended to go to a measured mile with a series of sister ships, newly built and equipped with torsion-meters and thrustmeters. Perhaps could different measured miles, different torsionmeters and thrustmeters and even different observer teams be taken into consideration. For the torsionmeters the writer suggests calibrated Siemens-Ford and Maihak; for the thrustmeters Michell and Kingsbury equipment. Only trials in loaded condition would be considered and they would be conducted and their results analysed according to the Code of B. S. R. A. or S. N. A. M. E.

Roughness measurements would be carried out on each ship:

(i) Sample plates would be treated and painted in the same way and simultaneously with the hull of the vessel and examined in the Talysurf machine;

(ii) The roughness of the hull would be directly measured with a less accurate instrument.

Pitot traverses could complete the description of roughness but this requires the instalment of a pitometer log with extensible rodmeter.

A survey of the results of measured-mile trials carried out in loaded condition on several Belgian ships for the last 10 years, is given in this contribution. All these ships were equipped with torsionmeters. They are of modern construction, either fully welded or with butts welded and seams riveted. The ship-model correlation was worked out according to the Froude law of similarity and the Froude friction coefficients without roughness allowance. The excess of ships results above the tank predicted results was not bigger than 10 per cent (in both directions). If the results of the Victory ship *Tervaete* are disregarded —this ship was 8 years old— and if account is taken of a fouling effect of 9.2 per cent for the newly-built *Jadotville* on the Antwerp-Congo trade (this fouling effect for 6 months' service has been ascertained for the Antwerp-Congo trade on the newly-built *Lubumbashi*), the maximum excess of ship results above Froude tank predicted results (in both directions) is brought back to 8 per cent.

Ship's name Model Tank Scale	LBP B D (ft)	Trim (ft)	C_b C_p LCB (% LBP)	Prop. D Pm BAR No. blad.	Rudder type	Hull Plat.	Meas. mile	Ship results % excess above Froude predict. tank results: V knots	dhp	dhp corrected to clean ship	Thrust
Tervaete NPL 1/24	436.5 62.0 26.2	5.3 A 2.2 A	0.678 0.686 1.7 F	20.5 22.5 0.50 4	Contra-rudder	Fully welded	Pitot Pitot	16.0 17.0	+ 6.9 + 9.9	clean 8 years old	— —
Lubumbashi NPL 1/24	446.2 61.4 25.8	0.3 A	0.699 0.709 1.2 F	17.6 14.7 0.46 4	Stream-lined Oertz type	Butts welded seams rivet.	Polp. id. id. id.	15.0 15.5 16.0 16.5	— 7.7 — 6.7 — 6.5 — 4.7	clean newly built	— 5.5 — 4.0 — 3.0 — 2.2
Lufira NPL 1/24	446.2 61.4 25.5	1.0 A	0.699 0.709 1.2 F	17.2 15.4 0.57 4	Stream-lined Oertz type	Butts welded seams rivet.	Polp. id. id.	15.5 16.0 16.5	— 7.7 — 7.5 — 6.9	clean newly built	— — —
Jadotville NSMB 1/26	511.8 69.9 25.9	0.5 A	0.670 0.688 0.44 F	19.7 18.2 0.52 4	Stream-lined Oertz type	Butts welded seams: bott. riv. side weld.	Polp. id. id.	16.0 17.0 18.0	+ 14.9 + 13.7 + 11.3	+ 5.2 + 4.1 + 2.0	6 months' service fouling 9.2 %
Elisabeth NSMB 1/27.5	628.5 82.7 31.3	Level	0.765 0.771 1.4 F	20.3 16.1 0.46 4	Stream-lined Simplex type	Fully welded	Polp. id.	16.5	+ 1.0	clean newly built	—

Except for m. v. *Lufira*, which was equipped with the torsionmeter of her sister ship *Lubumbashi*, all the torsionmeters were calibrated on the shaft. The trials were carried out with a wind strength not higher than 18 knots. The *Tervaete* correlation is based on sea trials where speed was measured only by means of a pitot log.

Dr. L. Kretschmer (Written contribution).

It would be of great importance if all establishments could agree to one friction line based on Reynolds number. But it is necessary that also all tanks in future take the values from this line for the correlation of model data and prediction of the ship resistance. The procedure may be a single one for routine works.

The Hughes method is attractive, but we think that is not the time to adopt it. There is not experience enough for using the "run-in" values of C_F to determine the form factors. It is difficult to find the "run-in" point.

The Vienna Tank would prefer the first of the Committee's proposals. We would agree to the steepening of the A. T. T. C. line below $R_n = 10^7$. We have found that then the measured resistance of the models in our usual size of about 20 feet at lower speeds is smaller than the calculated model friction resistance. But we are aware of the difficulties of the extrapolation of results from small models and so assist the problems of the small basins. The most of our model results lie in the range between $R_n = 5 \times 10^6$ and $R_n = 10^7$.

The formula for the new line should preferable be an explicit one.

The name should be impersonal, such as "I.T.T.C. 1957 line" or similar.

The proposals of Prof. Barillon and Prof. Kempf to measure the resistance of a standard model in all establishments over a whole year, every fortnight and at different times of the day, seems very informative. This model should be in all basins of the same shape and material. The size should be so, that the blockage value is nearly the same in all tanks.

The second proposal to test two standard models of two block coefficients would also give good information over the differences in the various tanks. It is necessary to find the causes of differences in measured resistances and avoid or eliminate them.

Mr. C. Wigley (*) (Written contribution).

The resistances throughout are supposed expressed in non-dimensional form, and by the same symbols as those used by Dr. Todd, though the definitions of these have been changed a little for clarity. Thus:

C_T = Total resistance.
C_F = Resistance of plank of same area, length and speed.
C_V = Form resistance, i. e., the difference between C_T and C_F at very low speeds.
C_W = Total wave resistance.
C_R = Total residuary resistance = $C_T - C_F$.

In addition, the following symbols are introduced:

C_{WP} = Wave resistance in a perfect fluid.
C_{WF} = Decrease in wave resistance due to viscosity of water.

The assumption usually now made to enable a prediction of the ship resistance from that of the model is that

$$C_T = C_F + C_V + C_W \qquad (1)$$

or

$$C_R = C_V + C_W$$

It is also assumed that C_R is independent of the scale.

Dr. Todd has dealt at length with the change in C_V with scale and concludes that C_V must decrease as the scale increases, but probably not so fast as does C_F.

If C_W is independent of the scale, then it must be equal to C_{WP}, the wave resistance in a perfect fluid. For, if the scale is indefinitely increased, the wave resistance must tend to C_{WP}. Therefore, in our notation, equations (1) are better written in the form

$$C_T = C_F + C_V + C_{WP} \qquad (1a)$$

or

$$C_R = C_V + C_{WP}$$

But is is know from the comparison of the calculated wave resistance with results of model experiments that the wave resistance of a model is not the same as that in a perfect fluid, but is less by a quantity depending on the scale, which we denote here by C_{WF}, so that the wave resistance of a model is $C_{WP} - C_{WF}$.

The equation giving C_T will then be, instead of (1a):

$$C_T = C_F + C_V + C_{WP} - C_{WF} \qquad (2)$$

and

$$C_R = C_V + C_{WP} - C_{WF}$$

The quantity C_{WF} vanishes at low speeds, being necessarily less than C_{WP}, but it also vanishes at speed-length ratios greater than 1.5.

It is possible to obtain a direct experimental check on the respective validities of equations (1a) and (2) by testing a model moving astern as well as ahead. For, if accented quantities refer to the motion astern, evidently $C'_F = C_F$ and $C'_{WP} = C_{WP}$.

Hence, if equation (1a) is true, the difference in resistance in the two directions of motion will be given by $C'_V - C_V$.

Recalling the law of variation of C_V, this will give

(*) Not in attendance. His contribution was presented by Mr. R. N. Newton.

a practically constant difference in resistance over the range of speed of a model experiment. But if equation (2) is used, the difference in resistance for the two directions of motion will be

$$C'_T - C_T - C'_{WF} + C_{WF}$$

This would give a practically constant difference in resistance at low speeds, rising then to a maximum difference at a speed-length ratio of about 0.9 and then decreasing to the original constant value at a speed-length ratio of 1.5.

This difference has been measured for three models [1] (") and [2], and repeated with a quarter-scale model for one of the three [3]. The results of the experiments on the two scales have been compared [4]. In all cases the type of difference described in the last paragraph, derived from equation (2), has been confirmed.

The experimental evidence therefore supports equation (2), but this is of little use for calculation at present because we do not know the law of variation of C_{WF} with scale. When the scale is sufficiently large, C_{WF} must tend to zero; also the experiments mentioned above show that $C'_{WF} - C_{WF}$ increases some 40 % to 50 % when a model of 4 ft length is compared with one 16 ft long.

The further information which we need here is the wave resistance for a ship, and a method of finding this has been suggested by Guilloton [5] using a method of calculating the wave resistance based on measurements of the wave profile round the ship.

In spite of our lack of actual knowledge, it is now possible to make an estimate of the order of the error involved in assuming that C_W is independent of the scale. At very slow speed-length ratios, the error would cause an overestimate of C_T for the ship, but probably not more than 1 or 2 %. As the speed is increased, this would change to an underestimate of C_T which might amount to a percentaje of 5 to 7 % at a speed-length ratio of 0.8 to 0.9. Thereafter, the percentaje error would diminish to a speed-length ratio of 1.5, after which the error would change sign again and take about the same value as at very low speeds.

The practical conclusions to be derived from the above analysis are that it is possible that no universal method of correlation between model and ship exists of the accuracy which is needed for the estimate of a ship's power. Also that even if one does exist, we have not enough knowledge at present to find it.

Hence the only method of calculating the power for a ship is to use any convenient method without regard to its accuracy, and then to add a percentage derived from comparison of trial and model results for a similar ship (Dr. Todd's ΔC_F) which will also take into account many other factors than those due to incorrect correlation.

(") Numbers in square brackets refer to the list of references at the end of this note.

It follows that all the disputes as to the proper curve of C_F to be used are not of much practical importance; the important feature of such a curve is its convenience in use, and the present writer would support the use of the A. T. T. C. line (Schoenherr) for a new Tank.

Nevertheless, the change in method due to any change of the curve in an existing Tank would be so confusing that the writer would no support any change from the Froude method where that has been in use for many years.

Comparative tables between the results of the use of the Froude method and the A. T. T. C. line for a number of model and ship lengths at various speeds, would be very useful, if they could be compactly arranged.

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A physical explanation of the so-called "blockage coefficient".

It is said that an increase of resistance (called the "blockage effect") is experienced with models in Tanks even at low speeds.

When the speed is low enough to neglect effects of wave-making, this cannot arise from direct interference, since the flow at the walls and bottom of a Tank will not be affected by viscosity, so that the effect on the resistance must be the same as in a perfect fluid, and therefore is zero.

The speed at which wave-making effects disappear can be calculated by Strettensky's method [1] and [2].

But there is another explanation of such an increase of resistance in a Tank, for it is a matter of experience that after some hours of test work an oscillation is built up in a Tank consisting in a piling up of the water at alternate ends of the Tank with a node at the mid-point of its length. The period of this oscillation is very slow, from 30 secs upwards, and the amplitude may be of the order of 0.1 inches.

The complete theory of such oscillations will be found in Lamb's Hydrodynamics, Chapter 8.

The energy contained in such a wave must be derived from the model in the first place, and it can easily be seen that the order of its magnitude is such as would cause a small increase in the resistance. It is, therefore, very probable that this is the physical explanation of the so-called "blockage effect". The easiest way to confirm or disprove this would be to run a model in an undisturbed Tank, and leave it at rest at the far end of the Tank for some 10 minutes, meanwhile taking a continuous record of the water height near one of the ends of the Tank. It would probaby be necessary to measure to an accuracy of about 1/1000 inch, but this should not be a difficult matter.

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Remarks on Dr. Kempf's "Introductory Remarks on Techniques of Measuring Model Resistance".

The present writer is wholeheartedly in support of Dr. Kempf's suggestion that standard models should be tested at frequent intervals in each Tank. This was done by Froude many years ago and the re-introduction of the practice should certainly lead to an increase in accuracy.

For the greatest advantage the results should be evaluated and co-ordinated by a special Committee as Dr. Kempf proposes.

Each Tank should supply to that Committee all the relevant information, including all the actual measurements taken before any analysis thereof.

Mr. R. N. Newton (Written contribution).

1. The Skin Friction Committee are to be highly commended for their efforts in the past three years to bring this difficult and, to say the least, debatable subject, nearer to an acceptable solution, and for their clear report. The review of developments since the last Conference, given in Appendix I, the frank exposition of views expressed by the numerous Ship Tank Authorities, given in Appendix II, and the excellent summary of the development of the subject provided by Dr. Todd in his formal remarks are as valuable as they are commendable.

2. The Committee are to be more strongly congratulated on being able to present to the Conference, in the light of the considerable divergence of views on the subject and yet still in accordance with their instructions, two judicial recommendations, viz. recommendations 3 and 4. These leave the choice of when a change should be made to a single line friction formulation, and a choice between two alternatives which they have carefully considered, to the present Conference.

3. On behalf of the Admiralty Experiment Works, Haslar, and with a full knowledge of the views of our late Superintendent, Dr. Gawn, I would like to dwell upon the philosophy rather than the technicalities of these recomendations.

4. The mere fact that the subject has been debated so openly since 1933 is proof of the general desire to adopt a common skin friction formulation acceptable to all authorities in the interests of international interchange of information and development in ship hydrodynamics generally. The fact that it has taken such a long time for the stage to be reached before logical and practical recommendations can be made is proof that the search for a theoretically sound formulation is still a long way from completion. From its inception the Froude method has been admitted as purely empirical and although several physically sound concepts have been postulated, their originators have generally admitted to them not being theoretically sound in the mathematical sense. Schoenherr is a typical and laudable example of this.

5. Furthermore, one must not lose sight of the necessity for adding, on a purely empirical basis, a correction of up to 30 per cent of the value of the coefficient for a smooth surface to obtain that for the full scale ship. This, in our present stage of knowledge, largely if not completely nullifies any advantages this theoretical background may be claimed to possess. It is also pertinent to quote from the notes of Professor Troost at the Fifth Conference in 1948: "From a statistical investigation of the results of a large number of models it appears that a difference of 4 per cent in frictional resistance affects the normal service speeds of ships by less than 1 per cent. In most cases it takes 5 to 8 per cent (ad more) of the frictional resistance to affect the speed by 1 per cent."

6. The debate of several years has in fact centred around the validity of several smooth surface friction lines which differ from a mean line through them of no more than 3 per cent at any part of the model or ship range of Reynolds' number. No doubt further research, especially into the mechanics of turbulence, will establish a more accurate smooth surface formula combining consistency with modern physical concepts and a sound theoretical basis. Nobody can deny the need for continuing this work but, in the meantime, it is considered more important to press on with research in the full scale field to fulfil the obvious need to establish at least a reasonably accurate method of assessing the so-called "roughness coefficient", or ΔC_F. One has only to study the replies to Kempf's questionnaire to various Towing Tanks in Appendix II of the Committee's report to realise how widely opinions differ on the value of ΔC_F to be allowed above the smooth friction line. They actually vary from .00015 to .0004 for all riveted hulls, corresponding to from 7 ½ per

cent to 20 per cent on the smooth friction formulation for a 400-ft. ship at 15 knots. For an 800-ft. ship at 30 knots the corresponding figures are 10 per cent and 30 per cent.

7. The urgent need for full scale trials to determine a realistic value of ΔC_F for different conditions of hull, in different but reasonable trial conditions of wind and sea, using different types of torsionmeters and thrustmeters and at the same time carrying out local velocity surveys is at once apparent. Professor Aertssen's suggestion in his formal remarks that this should be done with a series of sister ships and Granville's conclusion (3) on Appendix II of the report are therefore strongly supported. The results of such full scale tests would go a long way towards solving not only the all important ΔC_F problem but also some of those being faced in Subjects 1 and 5.

8. It is hoped that these brief comments will place the problem of deciding upon a smooth skin friction line in its correct perspective and leave no doubt as to the need to reach agreement on a formulation without further delay. It is considered the time is propitious to do so, on the clear understanding that the line selected be regarded, in the words of the Committee, "as essentially an arbitrary or compromise line", chosen in the interests of International interchange of information and development, so that results by one Establishment can be readily and correctly interpreted by others.

9. There is little to choose between the two suggested I. T. T. C. lines in the model range but at higher Reynolds' numbers, applicable to full scale ships, Alternative I is preferred because its slope conforms more closely with that given by Froude extrapolation. Both lines, however, require careful definition by a table of values of C_F against R_n. In the interests of simplicity it would be advantageous to define the line by a simple formula amenable to rapid calculation and the following is attractive

$$1/\sqrt{C_F} = 3.65 \log (R_n/100) \qquad \text{(A)}$$

The values of C_F given by this formula are compared with those taken from the alternative lines suggested by the Committee in the following table:

Log R_n	$10^6 \times C_F$				
	By Formula A	From I. T. T. C. Proposal I	Percentage Deviation From A	From I. T. T. C. Proposal II	Percentage Deviation From A
10	1173	1173	0.0	1073	— 8.5
9.5	1334	1335	+ 0.1	1235	— 7.4
9	1532	1532	0.0	1444	— 5.7
8.5	1777	1774	— 0.2	1704	— 4.1
8	2085	2074	— 0.5	2024	— 2.9
7.5	2481	2452	— 1.2	2427	— 2.2
7	3002	2937	— 2.2	2937	— 2.2
6.5	3707	3670	— 1.0	3670	— 1.0
6	4691	4813	+ 2.6	4813	+ 2.6

10. It will be seen that the maximum variation of C_F given by the Formula A from the I. T. T. C. Proposal I for $R_n = 10^6$ and upwards, is 2 ½ per cent and for $R_n > 10^7$ the deviation is negligible. Whilst appreciating Dr. Todd's view that any new line should preferably be suitable for use on high-speed electronic computers, one cannot refute the advantages of such a simple formula as the one suggested.

11. In conclusion I would like to say that these formal remarks have deliberately been produced as an impartial and practical view of the situation. Arguments in favour of, or in criticism of, any or several of the formulations which have been postulated could have been produced but are deemed to be less likely to contribute to the general agreement in the objective.

Prof. Dr. Ing. F. Horn (Written contribution).

1. With respect to the viscous line, I am still favouring, in principle, a conception which takes account, as far as possible, of the proper physical fundamentals, i. e., a conception with which an individual viscous form resistance is added to the basic two-dimensional frictional plate resistance. I firmly believe that such conception will, in more or less near future, be developed to sufficient fitness as to be realized in practical tank work. At present, it could only be recommended if it were already developed to such a rather definite shape which could be relied upon for a long time. Unfortunately, in spite of the manifold and intense efforts which have been made since the last Conference and deserve much appreciation, such a definite condition

has, as far as I can judge from the present reports, not yet been attained. Therefore, I am quite aware that for the time being a provisional procedure will be needed by way of a compromise and that for the latter a so-called single-line conception will come into question only. Since, evidently, such compromise cannot have the nature of mere plate resistance, but must, in some mean way, account for the viscous form resistance, it will have to be based, with due consideration of theoretical aspects, mainly on experience and experiment.

2. Under these circumstances, it might perhaps seem to be most reasonable to keep to the present condition until a definite procedure has been found which satisfies theoretical and practical demands; in other words, to abide in the meanwhile—whilst excluding the further use of the R. E. Froude friction coefficients—by the A. T. T. C. line, as in fact is strongly recommended by several authorities.

Personally, I feel rather strong objections against such a procedure, because, in my opinion, the A. T. T. C. line has, in the course of time, proved to agree too little with reality and is frequently leading to predictions of full scale resistance which are evidently in error. I consider it, the longer the more, as being prejudicious to the credit of towing tanks when it can happen that with such predictions driving power is overestimated up to about 10 % and when, as is shown by Fig. 6 of the contribution of Dr. Todd, in numerous instances negative values of roughness allowance $\Delta\zeta_f$ are established. No doubt, Dr. Todd is right in stating that the so-called roughness allowance is actually a correlation term which accounts for all gaps of our knowledge of real hydrodynamic conditions. But, besides roughness itself, very probably the trace of the viscous line is mainly concerned in $\Delta\zeta_f$, and since proper roughness allowance can never become negative, apparent negative values of $\Delta\zeta_f$ point strongly to an incorrectness of the viscous line which has been used.

At present, it is not yet possible to estimate the magnitude of this incorrectness with sufficient exactness, since our knowledge on the quantitative effect of roughness is still mainly restricted to that of sand roughness which cannot be applied to ship surface roughness. But from the following it can be gathered that ship surface roughness effect is greater than has hitherto been assumed. The wellknown fact that it is much more difficult to attain hydrodynamical smoothness in full scale than in model has anew been illustrated by the results of new NPL—tests by which it has been established that the resistance of a modern flush-welded ship with a good paint finish on top of clean bare steel may be considerably above that of a perfectly smooth surface. This result points to the probability of the real roughness additions $\Delta\zeta_f$ being much higher than according to the usual normal figures. There from it follows that the discrepancy which is demonstrated by negative $\Delta\zeta_f$ values is further enlarged and that the steepness of a true viscous line will be the more superior to that of the A.T.T.C. line. From the above reasons I consider it as being a main point of further research to get reliable quantitative data of the resistance effect of different ship surface roughnesses, included most perfect smoothness, by tests apt to approach full scale Reynolds numbers as far as possible. Perhaps new systematic pontoon tests, the like of those which were formerly made by Dr. Kempf, with very long pontoons up to high towing speeds, could serve this purpose.

In any case, even a provisional viscous single line ought to be such as to preclude negative $\Delta\zeta_f$ values.

3. Further, the provisionad viscous single line should, of course, be sufficiently supported by geosim test results. In this way very ample and intense research work has been done since the last Conference. Unfortunately, the difficulties which arise when trying to pick out from geosim results that special line which will show best mean adaptation to these results prove to be extremely great, as may, for instance, become evident from the most ample collection of data of existing geosim results which is contained in Fig. 7 and 8 of Dr. Todd's contribution. No doubt, a certain general tendency can be established from these figures, i. e., of a mean single line needing, in the range below $R_n = 10^7$, a rather steeper slope than has the A. T. T. C. line. But no criterion of sufficient reliability can be deduced as to which other single line formulation will in the mean suit best, since just those formulations which come primarily into question show but little differences in slope in the said range. There is to be added that rather numerous geosim test spots still suffer from uncertainties due to laminar influences and tank wall interference (*).

In order to improve the usefullness of geosim tests, the arguments and propositions which are contained in the present review of Capt. Acevedo should be given due consideration. Especially, it seems to be essential that of establishing the respective correlator m which has been introduced by Acevedo the iso-F_n lines within the higher range of F_n should be mainly taken use of, because this range will be least affected by laminar influences. But since, of course, the use of this correlator presumes the knowledge of a corresponding basic line, further thorough research work is needed for advancing, by way of geosim tests, a future definite settling of the whole problem. I endorse, therefore, the urgent proposal of Capt. Acevedo according to which the Skin Friction

(*) By the way, tank wall interference could, on my opinion, with sufficient approximation rather simply be eliminated along the analytical lines which are contained in recent years work of Dr. Schuster, especially in his publication in "Schiffstechnik", Vol. 3 (1955-56), p. 93.

Committee is called to start a *systematic* geosim programme. Such a programme in which, from obvious reasons, also tests with deeply submerged double models ought to be included—should have the main object of going into the three-dimensional problem and of establishing, on systematical lines, the viscous form resistance factors, for the main different types of ship form, in their dependency from the essential form factors.

Since, evidently, within such a programme also the condition of vanishing wave resistance is to be included in order to establish the position of the respective viscous line, reliable results will only be achieved when laminar influences at low model speeds can be eliminated safely. This means still a difficult problem, since all efforts with respect to safe turbulence stimulation at such low speeds have so far not proved to be fully successful. On my opinion, the method of towing a thin rod in front of the model—which method, though known since a rather long time, seems to have hitherto not gained sufficient study and application in shipbuilbing towing tanks, but has now anew been drawn attention to by rather remarkable arguments of Gamlin—ought to be applied somewhat systematically in some geosim series. It might be expected that the results when compared with those gained from usual stimulation tests will prove to be more consistent than the latter, and such comparisons could perhaps yield also useful information on the amount of alteration in model resistance due to the model lying in the wake of the rod. (It seems to be not quite unlikely that the resistance alteration proves to be so small that it could practically be neglected altogether.)

Moreover, the study of acoustic means of turbulence stimulation should not lost sight of.

4. Since a rather long time will be needed for all such future researches as were pointed to in the above with respect to geosim tests, and since for the time being existing geosim results cannot give sufficient information on the present question of the best provisional single line, with that choice, on my opinion, the first treated argument of reasonable ΔC_F values and, especially, of eliminating negative values proves to be predominant, provided that the line to be chosen corresponds in model range to the general mean tendency which is contained in existing geosim results.

As to the both lines which the Skin Friction Committee would be willing to accept, the first one would, on my opinion, frequently not lead to reasonable ΔC_F values, since, when compared with the A. T. T. C. line, alteration would take place only at Reynolds numbers below 10^7 and would, therefore, practically become effective with smaller models only. But hitherto also the ship predictions which are derived from tests with model sizes usual in large tanks (about 6 m length) prove frequently not to be satisfactory. In this respect, the second line would be more favourable, through probably in overall slope not yet quite sufficient. In view of a more systematical way, I should prefer a line which corresponds to a form factor r equal to about 1.15 (or a little higher) within the formulation of Dr. Hughes. I do believe that with such a compromise mean line ship-model correlation and, consequentely, ship predictions from model tests, in connexion with reasonable ΔC_F values according to the individual ship surface condition, would be greatly improved compared with the present state and could stand for a rather long time.

If, however, the Conference would, from tactical reasons, decide to keep, for the time being, still to the A. T. T. C. line, such decision ought to be clearly coupled with the establishment of that line not coming into question within a later definite solution.

Prof. Dr. Ing. F. Horn (Oral remarks, translated from German).

I may add that Dr. Hughes' formulation with a form factor 1.15 gives about the same curve as those proposed by the British gentlemen, which corresdons to a form factor 1.13.

Your attention is also called to the fact that this line practically coincides with the Schultz-Grunow one, which is already in use in some Tanks.

Mr. H. B. Lindgren and Mr. E. Bjärne (Written contribution) (*)

1. INTRODUCTION

The desirability of being able to make a consistent comparison between experimental results obtained in different model experimental tanks led to a large number of towing tank establishments throughout the world carrying out careful tests on models of the Victory ship prior to the Seventh International Conference on Ship Hydrodynamics in Scandinavia in 1954. A summary of the test results obtained by the various establishments is given in [1] (**).

The Swedish State Shipbuilding Experimental Tank (SSPA) contributed to this comparison by carrying out tests on a model in the scale of 1:24.

Among the Decisions and Recommendations put forward by the Seventh International Conference on Ship Hydrodynamics [1] were the following:

a) Each establishment should re-examine its experimental technique in the light of the Victory ship comparative tests.

b) Each establishment should consider self-propulsion test by alternative methods. Particular

(*) Public. no. 40 SSPA.

(**) Numbers in brackets refer to the list of references at the end of this contribution.

attention should be given to accuracy of model propeller construction as well as experiment technique to ensure the refinement of accuracy necessary for the investigation of the small scale effects concerned.

c) In carrying out self-propulsion experiments for research purposes the thrust loading should be extended over the range between model and ship self-propulsion for at least two speeds of advance.

Since completing the above mentioned tests with a model in the scale of 1:24, new more modern apparatus for measuring propeller thrust and torque have been brought into use at SSPA. In view of this innovation and the above recommendations, it was decided at SSPA that a new 1/24th scale model should be made and tested. Since it was also considered desirable to make a further study of the wall-effect in the tank, the plans were extended to include tests with a family of five geometrically similar models of different sizes.

Both resistance tests and self-propulsion tests with this family of models have now been completed. This report deals mainly with the results of the resistance tests. The results of the self-propulsion tests are briefly summarized in Appendix 1, but it is expected that it will be possible to give a more detailed study of these and the results of the open water propeller tests at the 1957 International Towing Tank Conference.

Tests with similar families of models have been carried out both at NSMB, Wageningen, and at NPL, Teddington, and the results were published [2, 3] while the SSPA tests were in progress. Hughes has made a comparative analysis of these tests [3] and he has also evolved a method of determining the slope of the friction line with the aid of results of resistance tests on a family of models carried out in different experimental tanks [4]. This method is illustrated in Appendix 2.

2. Symbols

The symbols have been chosen in accordance with the nomenclature adopted by the Sixth International Conference of Ship Tank Superintendents as a tentative standard.

A_M = immersed midship section area
A_T = cross section area of tank
$C_1 = \dfrac{\nabla^{2/3} V^3}{P_E}$ (m², Metric knots and HP)
$C_2 = \dfrac{\nabla^{2/3} V^3}{P_S}$ (m², Metric knots and HP)
$C_T = \dfrac{R}{\rho/2 \cdot S V^2}$ = total resistance coefficient
C_V = viscous resistance coefficient
C_{FH}, C_{FS} = friction resistance coefficiente according to Hughes' basic line and Schoenherr's mean line, respectively
k, n, p = form influence factors
n = rate of revolution (revs. per unit time)
L = length on waterline
P_E = effective power
P_S = shaft power (at tail end of shaft)
R = resistance (total)
$R_n = \dfrac{VL}{\nu}$ = Reynolds number
S = wetted surface area (including wetted surface area of rudder and bossing if any)
T = propeller thrust
$t = \dfrac{T-R}{T}$ = thrust deduction factor
V = speed
w = wake fraction (Taylor)
∇ = volumetric displacement
ρ = density of water (102.0 kg sec.²/m⁴ for fresh water) (104.5 kg sec.²/m⁴ for sea water)
ν = kinematic viscosity of water [1])
$\eta = \dfrac{P_E}{P_S}$ = propulsive efficiency

For g (acceleration due to gravity) the value 9.81 m/sec.² has been used.

3. Ship Models Tested

The models represented a Victory ship, the main particulars of which are given in Table 1.

Table 1

Length on waterline, L = 135.31 m
Length between perpendiculars, L_{PP} = 133.05 m
Breadth, B = 18.90 m
Draught, T = 8.53 m
Wetted surface area, S = 3698 m²
Displacement, ∇ = 14745 m³
L/B = 7.16
L_{PP}/B = 7.04
B/T = 2.22
$L/\nabla^{1/3}$ = 5.52

$$\delta = \frac{\nabla}{LBT} = 0.676$$
$$\delta_{PP} = \frac{\nabla}{L_{PP}BT} = 0.687$$
$$\beta = \frac{A_M}{BT} = 0.990$$
$$\Phi = \frac{\nabla}{A_M L} = 0.683$$

Longitudinal centre of buoyancy = 0.18 % of L_{PP} forward of $L_{PP}/2$.
Half angle of entrance on waterline = 12 degrees.
Parallel middle body = 15 % of L_{PP}.

1) For ν; see [5] p. 6-7.

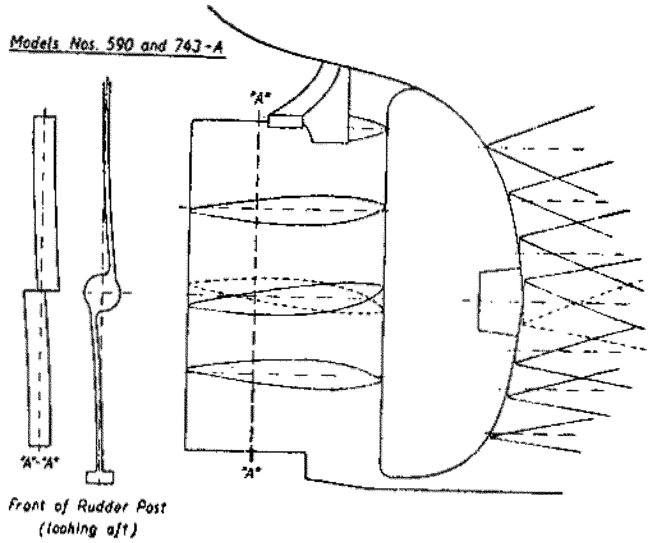

Fig. 1 a.

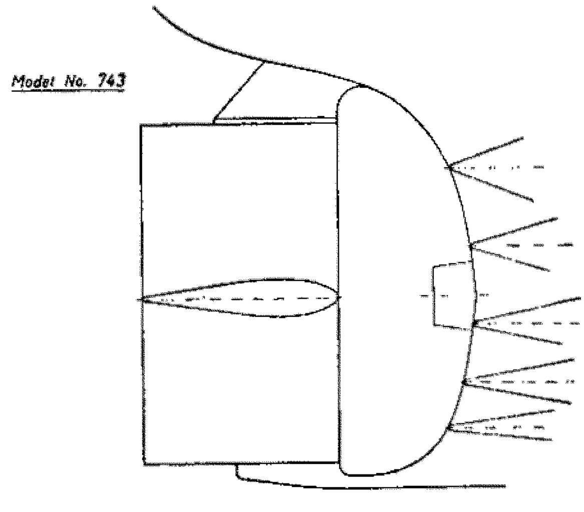

Fig. 1 b.

Six models, in all, have been tested. The oldest of these, Model No. 590 (scale 1:24) was tested in 1953 and was used in the original comparison between different towing tank establishments. As stated in the Introduction, it was considered desirable, for several reasons, to repeat these experiments and a new model (Number 743-A) in the scale of 1:24 was therefore made.

Models Nos. 590 and 743-A were each fitted with a contra-stern and a contra-rudder as shown in Fig. 1 a and as fitted on the fullscale ship. It was however considered that models of such special form were not entirely suitable for the investigations in question and the contra-stern and contra-rudder were therefore replaced by a normally shaped stern and a normal, symmetrical rudder, as shown in Fig. 1 b. Figs. 2 and 3 show the body plan and end contours of the thus modified Victory ship.

In addition to the two 1/24th scale models, four other models in the scales 1:17, 1:20, 1:28 and 1:32 were made and tested. The main particulars of all the different models are given in Table 2.

Table 2

	Units					
Model Scale.	—	1:17	1:20	1:24	1:28	1:32
Model No.....	—	755	754	743	753	778
L..............	m	7.959	6.766	5.638	4.833	4.228
L_{PP}	—	7.826	6.653	5.544	4.752	4.158
∇	m^3	3.001	1.843	1.067	0.672	0.450
S	m^2	12.80	9.25	6.42	4.72	3.61
Blockage, A_M/A_T......	%	1.12	0.81	0.56	0.41	0.32

All the models were made of paraffin wax and were fitted with a 1 mm tripwire round a section 5 % of L_{PP} abaft the FP.

The models were fitted with rudders in all these experiments. The Victory ship models, which were subjected to resistance tests at NPL [2] and NSMB [3], were without rudders and the NPL and NSMB tests also differed from the SSPA tests in the draught at which the models were run. Therefore, for the purpose of enabling comparisons between the various series of tests, the SSPA programme was extended by running two of the models, Nos. 743 and 755, whithout rudder and at a draught corresponding to that employed at NSMB and NPL. These experiments are further discussed in Appendix 2.

4. Tets and Results

Resistance tests were carried out over a wide speed range with all the models. A special pendulum apparatus, which is described in [5], was used for measuring small values of resistance (up to about 1 kg). The normal Gebers' type resistance dynamometer was used for measuring greater forces.

Comprehensive self-propulsion tests with varied thrust loading were also carried out with the four larger models and the propellers were similarly tested in open water. As mentioned in the Introduction, the results of these experiments are being analysed separately and will be presented at the Eighth International Towing Tank Conference. The results of the normal self-propulsion tests carried out in accordance with SSPA practice with the friction correction based on Froude's friction coefficients, are summarised in Appendix 1.

Results of Resistance Tests.

The primary results of the resistance tests with Models Nos. 743, 753-755 and 778 are given in tabular form in Appendix 3.

An analysis of the results shows that at very low speeds there is considerable scatter, presumably due to insufficient turbulence stimulation and inaccu-

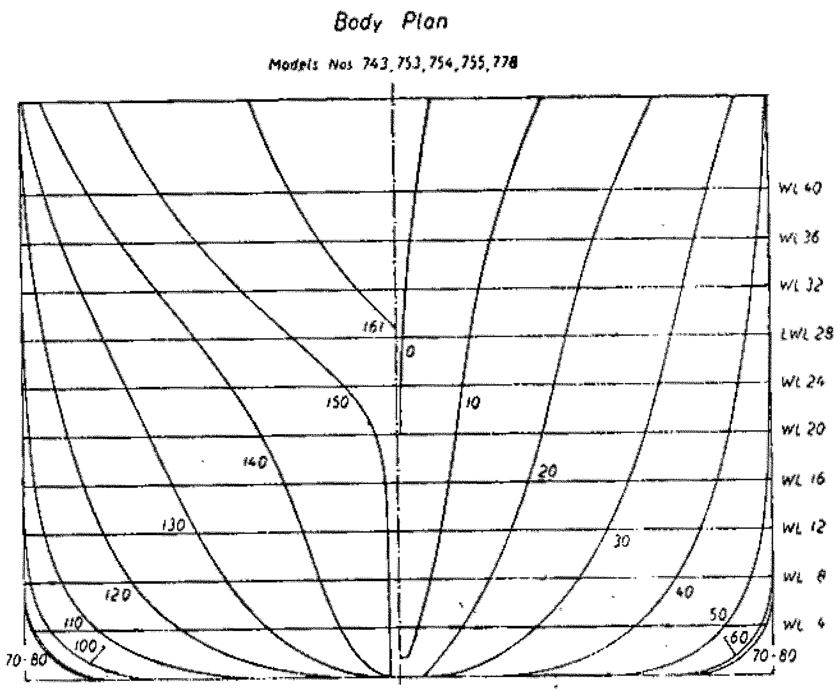

Fig. 2.

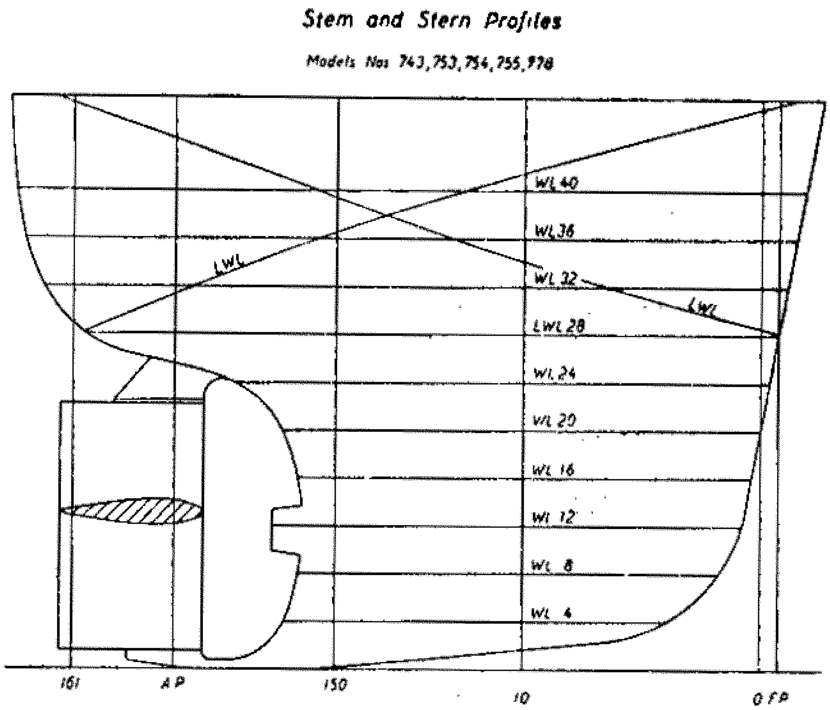

Fig. 3.

racies of measurement. In subsequent development of the results, therefore, all records obtained at speeds lower than that corresponding to a Reynolds number of 4.5×10^6 for the largest model have been ignored. The corresponding limit for the smallest model was set at 2.4×10^6 and for the intermediate models the limits were linearly interpolated. These limits were chosen empirically.

The results are illustrated in Fig. 4. In order to facilitate a comparison of the results, the resistance values have been corrected in this case to a common temperature of 15.3° C. This temperature was chosen because many of the tests were carried out at 15.3° C. and the corrections were thereby minimised. The corrections, which are of secondary importance, were based on a friction line somewhat steeper than

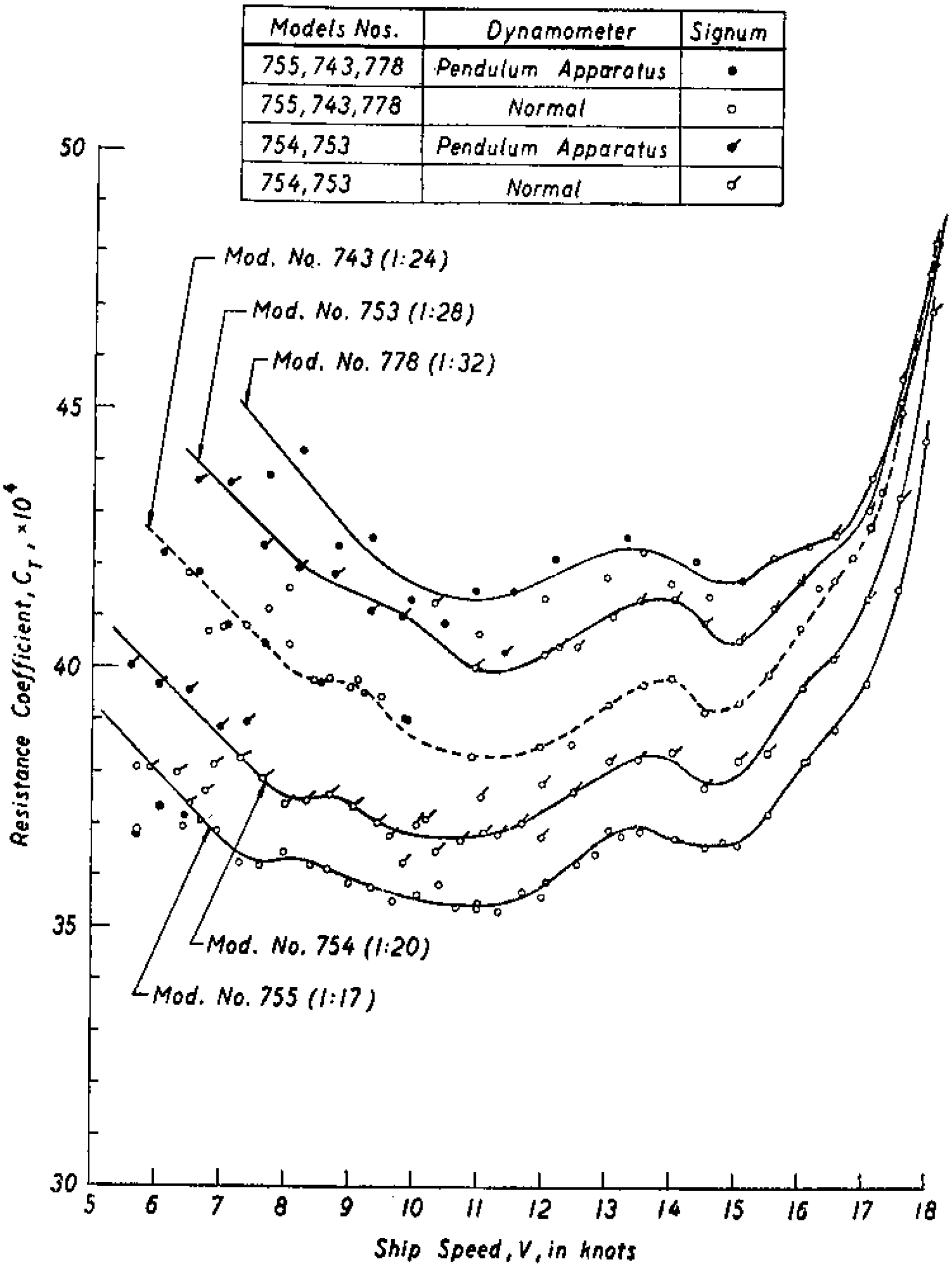

Fig. 4.—Model Resistance Coefficients.

the Schoenherr line (Schoenherr + 17 % form addition, see Section 5, Fig. 9).

Fig. 5 shows values of effective power and C_1 calculated by the conventional method on the Froude basis [1]) from the results of resistance tests with Models Nos. 743, 753-755 and 778. These curves have been derived from the measured values obtained in Test Series 1-5 (see Appendix 3) and not from the faired curves shown in Fig. 4. The differences between the curves amount to as much as 5 or 6 % and thus can hardly be attributed to inaccuracies of measurement but rather to defects in the method of calculation.

1) The frictional resistance has been calculated using the formulae decided upon at the Tank Superintendents Conference in Paris in 1935.

Fig. 6 gives the results of similar calculations for Models Nos. 590 and 743-A (both with contra-sterns and contra-rudders) and Model No. 743. The agreement between the results obtained with the old model (No. 590) and the new model (No. 743-A) is good and the effect of the alterations to the sternframe and rudder (No. 743) was negligible.

A more detailed analysis of the experimental results follows in Section 5.

5. Analysis of the Test Results

General.

The resistance of a ship or a ship model in water is normally divided into a wave-making component,

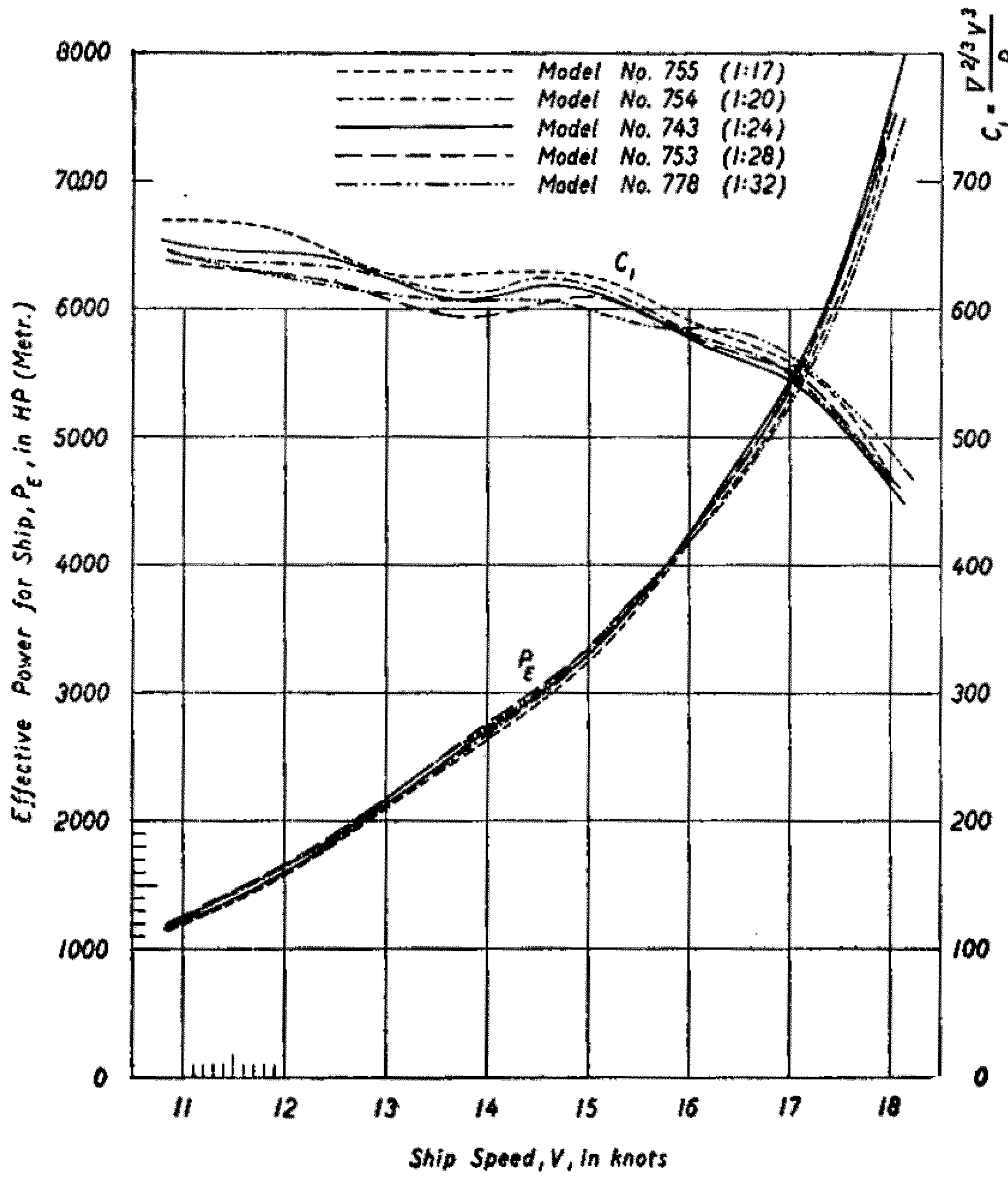

Fig. 5.—Resistance Test Results according to Froude.

which is assumed to conform to Froude's well-known law of comparisons, and a frictional component.

The frictional component can in turn be divided into two parts, of which one is taken as being equal to the resistance of an infinitely thin flat plate of the same length and wetted surface area as the ship or model. The other part represents the form effect.

The questions of the correct formulation of the plate friction line and the character of the form effect have promoted considerable discussion during recent years. The Froude formulation for skin friction which is employed at many experimental establishments is now considered out of date. Furthermore, the Froude procedure of relating the form effect to the wave-making resistance coefficient, which assumes that it follows the law of comparisons, is considered not wholly correct. Several proposals for alternative solutions to the problem are being discussed at the moment.

The main purpose of this analysis of the tests with Victory ship models was to derive a formulatión giving the best correlation on converting the various experimental results to full scale. Unfortunately, however, it is not possible in an analysis of this type, to disregard the blockage effect of the tank. (The blockage is defined as the relation between the midship section area of the model A_M and the cross section area of the tank A_T). Unsatisfactory correlation may be due to errors in the assumed skin friction coefficients, incorrect calculation of the form effect and blockage effect. These various ef-

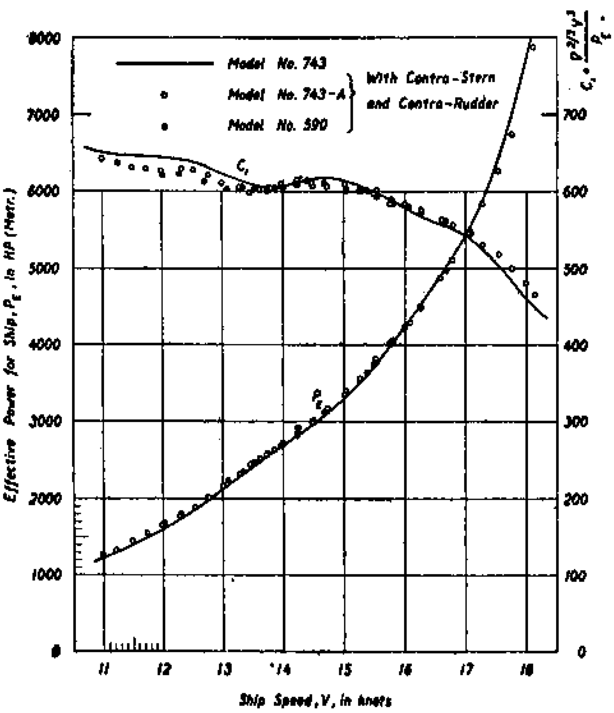

Fig. 6.

fects cannot be separated by analysing the results of tests with a family of models in only one tank.

Hughes, however, has suggested a method of separating the blockage effect from the others, and thus obtaining the correct slope of the three-dimensional fricction line, by comparing the results of tests with a family of models carried out in tanks of different cross section area [4]. An attempt to apply this method is shown in Appendix 2. Unfortunately, though, errors of measurement, small differences in the shape and surface of the models, turbulence stimulation and other differences between the different tanks have combined to produce so great an effect that no positive conclusions can be drawn from this analysis.

The resistance coefficient, C_T, for each model is plotted on a base of Reynolds number, R_n, in Fig. 7. Points of equal speed in knots have been marked on the curves. The friction line proposed by Hughes for two-dimensional flow, represented by the equation

$$C_{FH} = 0.066\,(-2.03 + \log R_n)^{-2} \qquad (1)$$

and Schoenherr's well-known mean line, expressed by the formula

$$0.242 \;/\; \sqrt{C_{FS}} = \log (R_n\, C_{FS}) \qquad (2)$$

are both shown in Fig. 7.

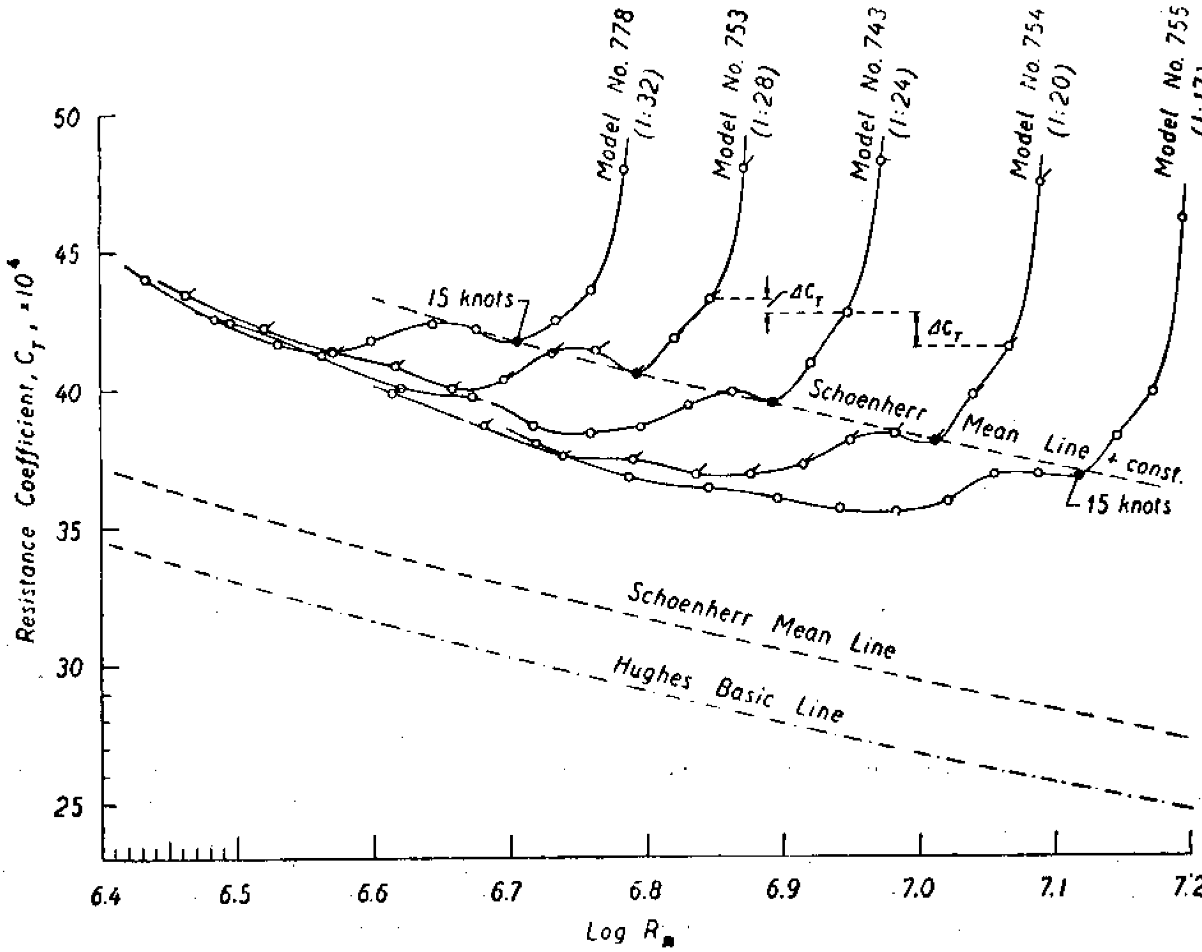

Fig. 7.—Model Resistance Coefficient.

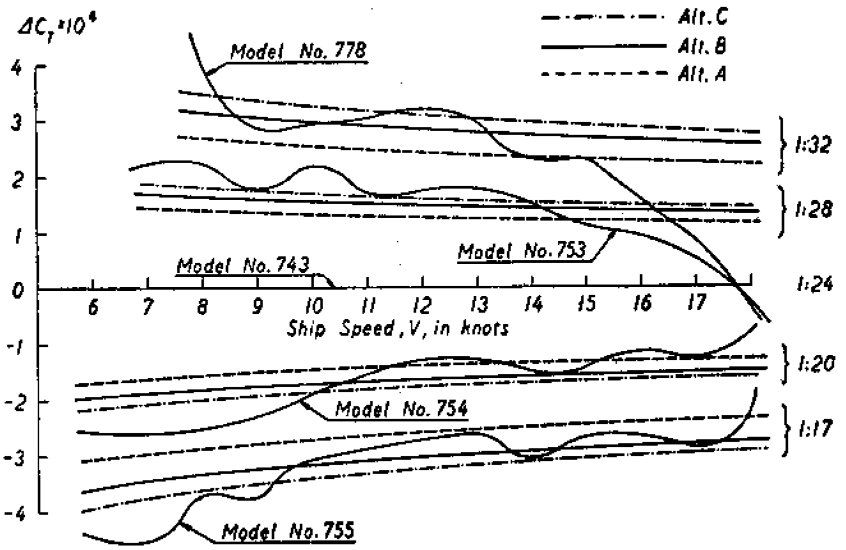

Fig. 8.—Model Resistance Coefficient Difference Curves.

In the following, an attempt will be made to determine which of three different methods suggested for calculating the frictional resistance gives the best correlation, i. e. most closely approaches the ideal of giving the same prediction for ship resistance independent of the scale of the model used as a basis.

Landweber [6] has proposed that the total viscous resistance coefficiente, C_V (the three-dimensional friction line), should be expressed as

$$C_V = C_F + nC_F + p \quad (3)$$

where the constants n and p are independent of Reynolds number.

Three suggested simplifications of (3) were examined, namely:

A) $C_V = C_{FS} + p$ .... (4)

B) $C_V = C_{FS} + nC_{FS}$ .... (5)

where C_{FS} is as used in equation (2)

C) $C_V = C_{FH} + kC_{FH}$ .... (6)

where C_{FH} is given by equation (1).

Alternative C represents the line with the most slope and alternative A that with the least slope of the proposed formulations (apart from Froude's friction lines, which are now considered obsolete).

Hypotheses on the slope of the friction line can be examined by calculating the differences in C_T values obtained at equal values of Froude number in experiments with models of different scales.

Fig. 8 has been derived by calculating the differences, ΔC_T (see fig. 7), for the various models, relative to Model No. 743, at constant ship's speed (i. e. at constant values of Froude number). The values of ΔC_T obtained experimentally were thus compared with values of ΔC_T calculated on the basis of the above three hypotheses. The constant n was assumed to have a value of 0.17 (see Fig. 9) and the value of k was taken at 0.27 as proposed by Hughes [4].

As is evident from Fig. 8, it is difficult to conclude that any of the hypotheses A — C is the correct one. At high speeds, there is considerable divergence from all the lines, which indicates that blockage effect is present. The result is analogous with that shown by Hughes.

Since the blockage effect is shown to be dependent upon speed, an attempt has been made to determine the best slope for the friction line at different speeds. For this purpose a low speed range, covering the non-wave-making range, and a normal speed range were examined. In view of the reasonable blockage values, the blockage effect can probably be neglected in the non-wavemaking speed range.

Low Speed Range.

A number of attempts have been made to determine, by means of model tests in the non-wave-making speed range, the correct friction line for three-dimensional bodies. It has been shown possible (within reasonable limits) to derive points on the friction line in this manner but it is considerably more difficult to obtain the slope of the friction line by means of such experiments on account of the very limited speed range which can be used for the analysis. At low speeds it is difficult to prevent laminar flow and at the higher speeds the resisitance begins to be affected by wave-making.

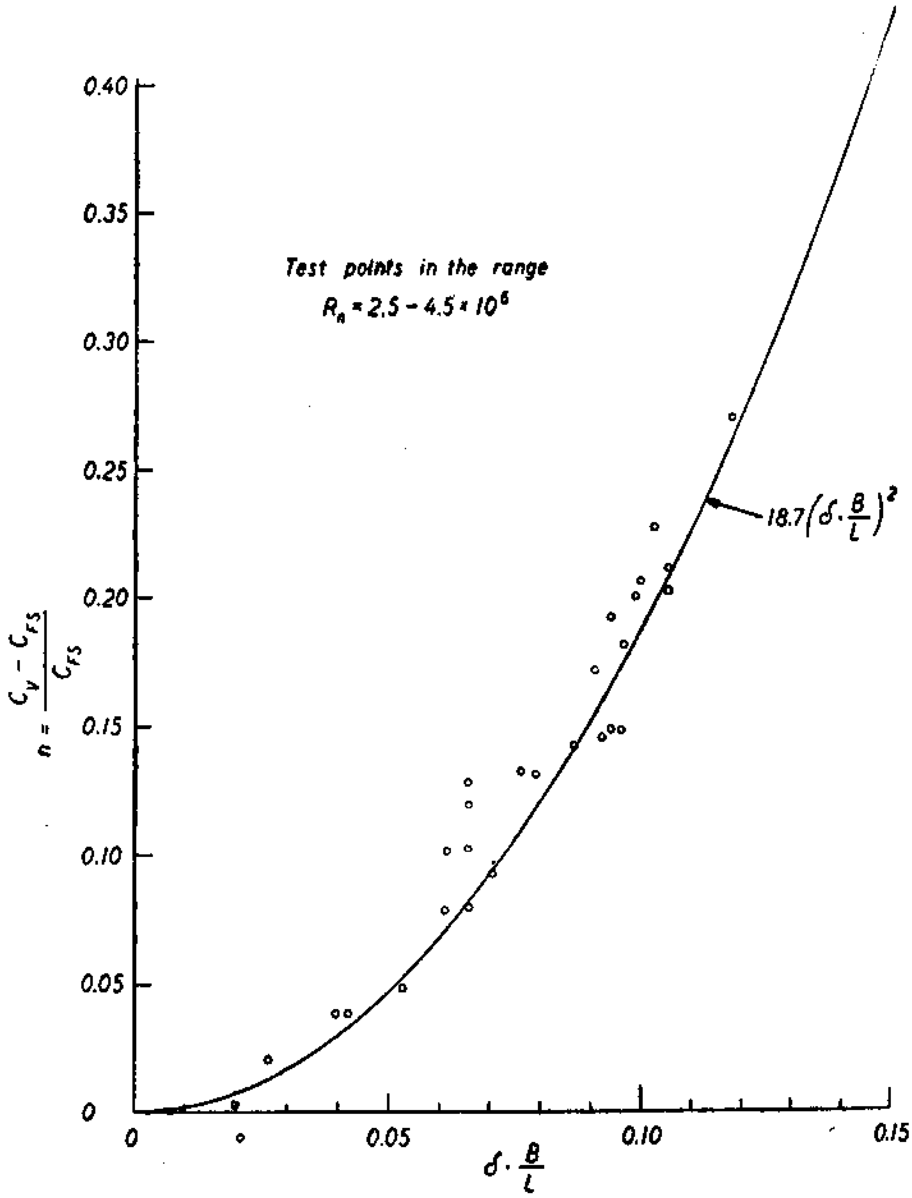

Fig. 9.—Percentage Addition to Schoenherr Meanline according to Granville.

On the basis of results of tests in the non wave-making speed range given in [7], Granville [8] derived a curve for the form influence factor n in equation (5). This curve is shown in Fig. 9. A number of new tests in the non wave-making range have been carried out at SSPA. The SSPA results (obtained in the range $R_n = 2.5 - 4.0 \times 10^6$) have been plotted in Fig. 9 and show reasonably good agreement with the curve.

It was considered that it might also be possible to verify the correct slope of the friction line by means of tests with geometrically similar models. In Fig. 10, all the experimental spots obtained at speeds below the equivalent of 8 knots (apart from those obtained at extremely low Reynolds numbers, which were ignored in accordance with the principle mentioned earlier), have been plotted on a base of Reynolds number.

The line with the most slope, alternative C (Hughes), gives the best agreement in the speed range considered, but all the friction lines given by the above hypotheses A — C (equ. 4-6) appear to be too flat. An attempt was made to derive a new line by the least squares method, but the wide scatter of the spots adversely affected its reliability. It was also apparent that the line largely depends on the number of points included in the calculations, i. e. on the speed range which could be taken as the non-wave-making range for each model.

Normal Speed Range.

In the normal speed range, 11-18 knots, the slope of the hypothetical lines can be verified by calculating the differences between the C_T values obtained

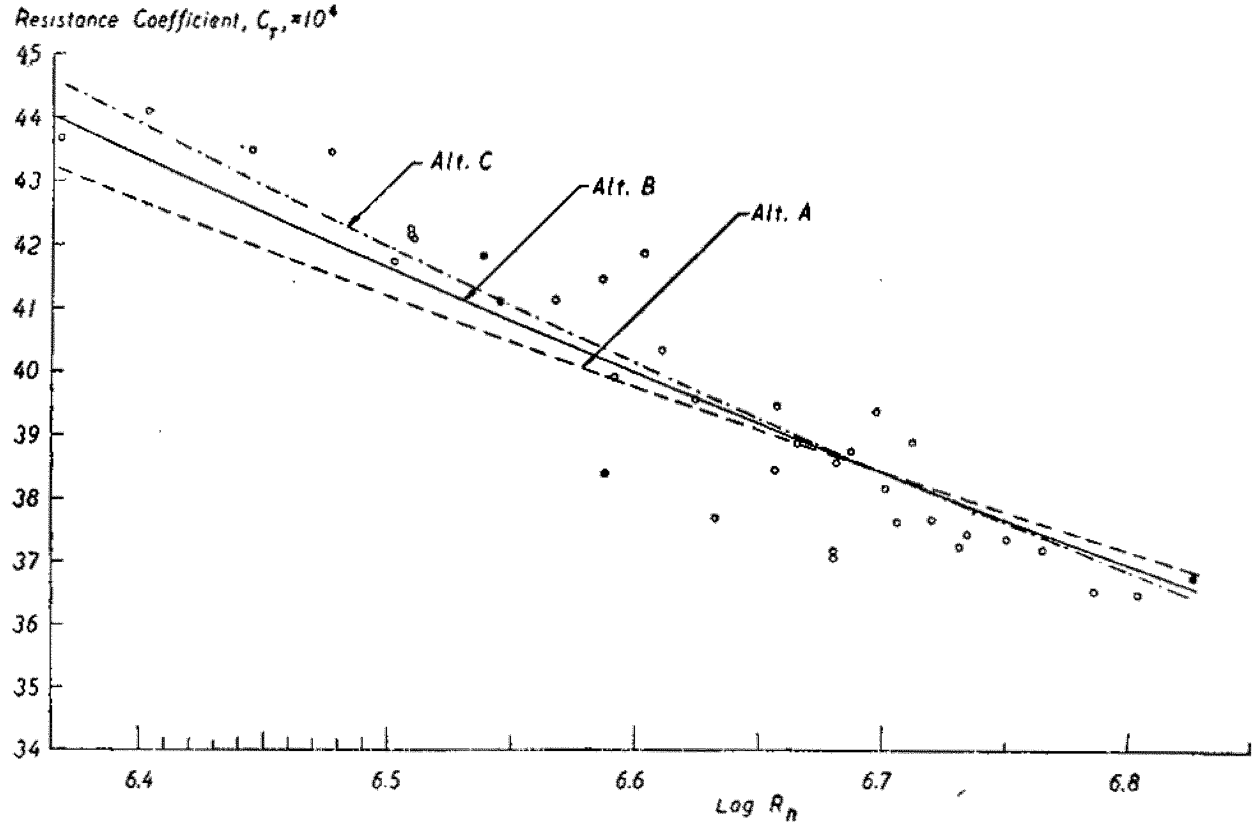

Fig. 10.—Model Resistance in the Low Speeu Range.

from model tests and lines drawn through suitable points parallel to the alternative friction lines. The point chosen as suitable in this case was that representing the mean values of C_T and R_n for the different models at constant Froude number (i. e. constant ship's speed). Only the hypotheses A and C (see above) were considered, since these represented the extremes of slope.

Fig. 11 shows the differences obtained relative to the friction line given by alternative C (Hughes), while Fig. 12 shows the differences relative to the line given by alternative A (Schoenherr). The differences are expressed as percentages of the imaginary parallel friction lines.

It is evident from these diagrams that alternative A gives the best correlation. Except at the extreme

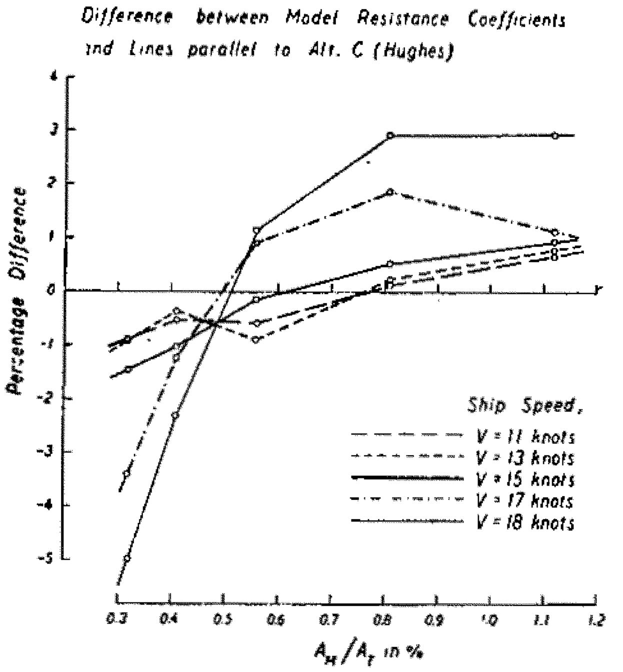

Fig. 11.

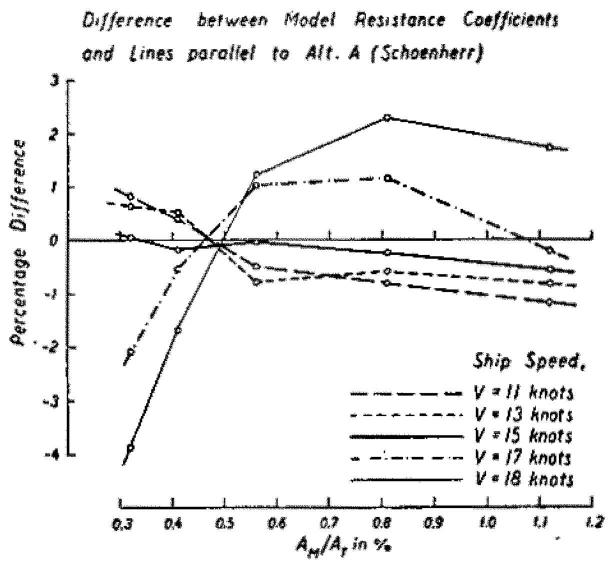

Fig. 12.

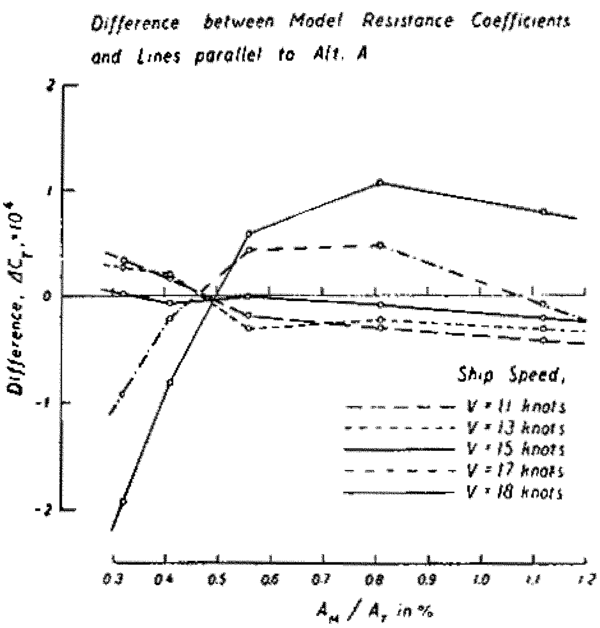

Fig. 13.

speeds, 17 and 18 knots, the differences do not exceed $\pm$ 1 %. Thus, in the normal speed range, the method satisfied the requirements that the same prediction of ship resistance should be obtained regardless of the model scale. Fig. 13 shows the absolute differences in C_T for alternative *A*.

At high speeds (17-18 knots) the slope of the difference lines indicates a blockage effect.

As shown by Fig. 11, all the difference lines based on alternative *C* (Hughes) are sloped. The slope of the lines increases with increasing speed and this indicates that there is blockage effect even at the lower speeds.

Fig. 14 shows the P_E and C_1 curves calculated on the basis of hipothesis *A* (Schoenherr). In this case, a roughness allowance of 0.0004 was added to all the values of resistance coefficient C_T. The divergence between the curves is clearly less than in Fig. 5 (Froude).

6. Summary

Resistance tests covering a wide range of speeds have been carried out with five geometrically similar Victory ship models. The experimental results have been described and analysed mainly with a view to verifying different theories about the form of the friction line for ship-shaped bodies and also in order to investigate the question of blockage effect in experiments with large models.

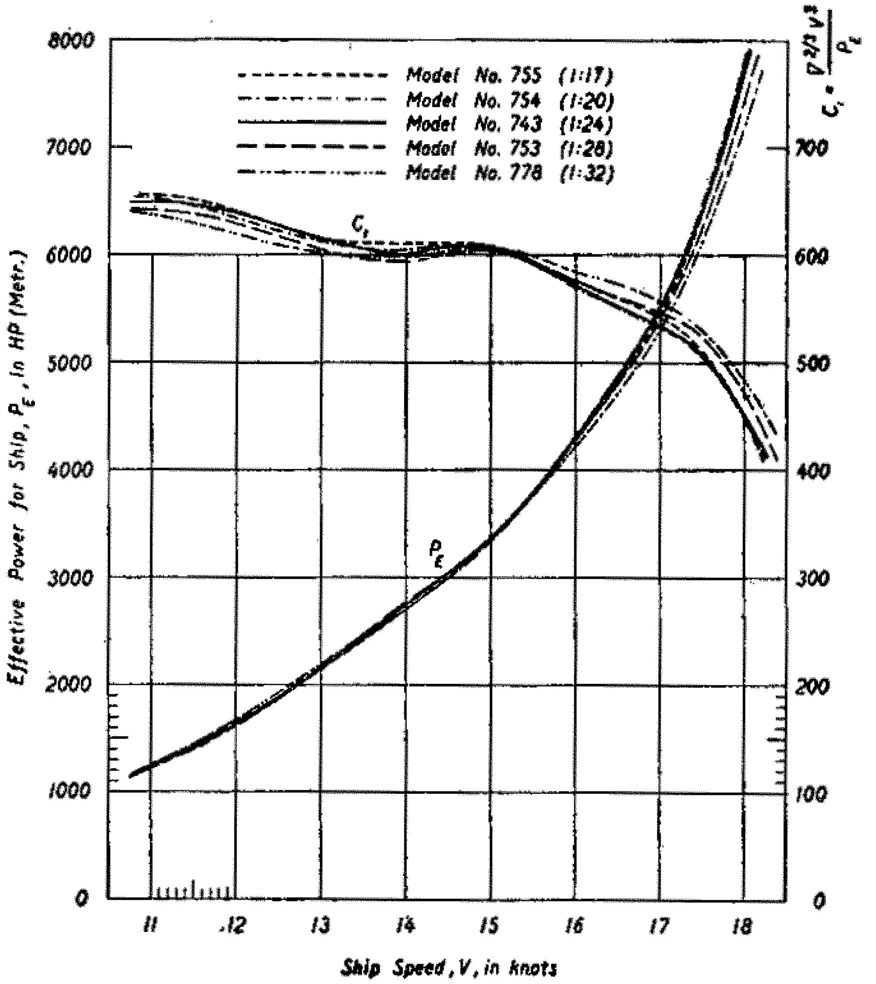

Fig. 14.—Resistance Test-Results according to Shoenherr.

In connection with the form of the friction line, hypotheses A — *C* as tabulated below, have been examined.

Alternative hypotheses	3-Dimensional Line C_V	Composition of Line	
		Plate Line C_F	Form Allowance
A	$C_V = C_{FS} + p$	Schoenherr	Constant
B	$C_V = C_{FS} + nC_{FS}$	Schoenherr	Percentage
C	$C_V = C_{FH} + kC_{FH}$	Hughes	Percentage

The following conclusions may be drawn from the experimental results:

1. The requirement that the same prediction of ship resistance should be obtained whatever the scale of the model, is best satisfied in the normal speed range (11-18 knots) by hypothesis *A* (Schoenherr + constant form allowance).

2. If hypothesis *A* is accepted, the blockage effect only becomes noticeable at very high speeds (about 18 knots).

3. One fact which does not support acceptance of hypothesis *A* is that in the non-wave-making speed range, the slope of a mean line drawn through the experimental points is such that it most nearly coincides with a line based on hypothesis *C* (Hughes). It is difficult, however, to be certain about this, since the experimental points are considerably scattered in the low speed range.

4. If hypothesis *C* is correct, the results are influenced by blockage effect over the whole speed range. The good correlation given by hypothesis *A* must then be explained by supposing the error in the slope of the friction line to be largely balanced by blockage effect.

Finally, it can be said that there seems to be little possibility of obtaining a correct solution to the combined problem of three dimensional friction line —form effect— blockage effect by means of experiments with geometrically similar models. However, the proposed "image" method of van Lammeren, van Manen and Lap [2] and the previously mentioned Hughes method of using the experimental results from tanks of different cross section may prove valuable in obtaining a definite solution to the problem.

It seems that it will be a considerable time before any such solution is found and the experimental results discussed herein indicate that until that time it would be resonable to use Schoenherr's mean line with a constant form allowance. This means that the form allowance, in accordance with the Froude procedure, can be included with the wave-making resistance in a residual resistance component which obeys Froude's law of comparisons. The advantages of this method are:

1. It is based on a friction line giving one value at any value of Reynolds number (unlike Froude's friction lines).

2. It is simple to use and involves no knowledge of the absolute value of the form addition in the non-wave-making speed range (unlike Hughes' method).

3. With models of the sizes normally employed, it appears as if the blockage effect can within practical limits be ignored (unlike with Hughes' method).

4. The method has been used with good results for many years at a large number of experimental establishments, where empirical correction factors (for roughness, etc.) have also been worked out.

One disadvantage of this method is that if, as suggested by some investigators, it is based on too flat a friction line (compare also Hughes' line), the calculated resistance will be too high in the case of the smooth ship. The empirically deduced roughness addition is then correspondingly underestimated.

7. List of References.

[1] Nordström, H. F.; Edstrand, Hans: "Seventh International Conference on Ship Hydrodynamics". *Publication No. 34 of the Swedish State Shipbuilding Experimental Tank*, Gothenburg, 1954.

[2] Van Lammeren, W. P. A.; van Manen, J. D.; Lap, A. J. W.: "Scale Effect Experiments on Victory Ship and Models. Part I", *Trans. INA*, 1955.

[3] Hughes, G.: "Viscous and Interference Effects Deduced from NSMB and NPL Victory Model Tests", *Trans. INA*, 1956.

[4] Hughes, G.: "The Effect of Model and Tank Size in Two Series of Resistance Tests", *Trans. INA*, 1956.

[5] Nordström, H. F.; Edstrand, Hans: "Model Tests with Turbulence Producing Devices". *Publication No. 18 of the Swedish State Shipbuilding Experimental Tank*, Gothenburg, 1951.

[6] Landweber, L.: "Sixth International Conference of Ship Tank Superintendents", Comments on Subject 2—Skin Friction, *Published by SNAME*, 1953.

[7] Nordström, H. F.; Edstrand, Hans; Lindgren, Hans: "On the Influence of Form upon the Skin Friction Resistance", *Publication No. 31 of the Swedish State Shipbuilding Experimental Tank*, Gothenburg, 1954.

[8] Granville, P. S.: "The Viscous Resistance of Surface Vessels and the Skin Fricction of Flat Plates", *Trans. SNAME*, 1956.

APPENDIX 1

Self-Propulsion Tests.

Self-propulsión tests with differents degrees of thrust loading were carried out with all the models and open water tests were similarly carried out with the model propellers. The results of these tests are being analysed separately and, as previously mentioned, will be presented at the 1957 International Towing Tank Conference.

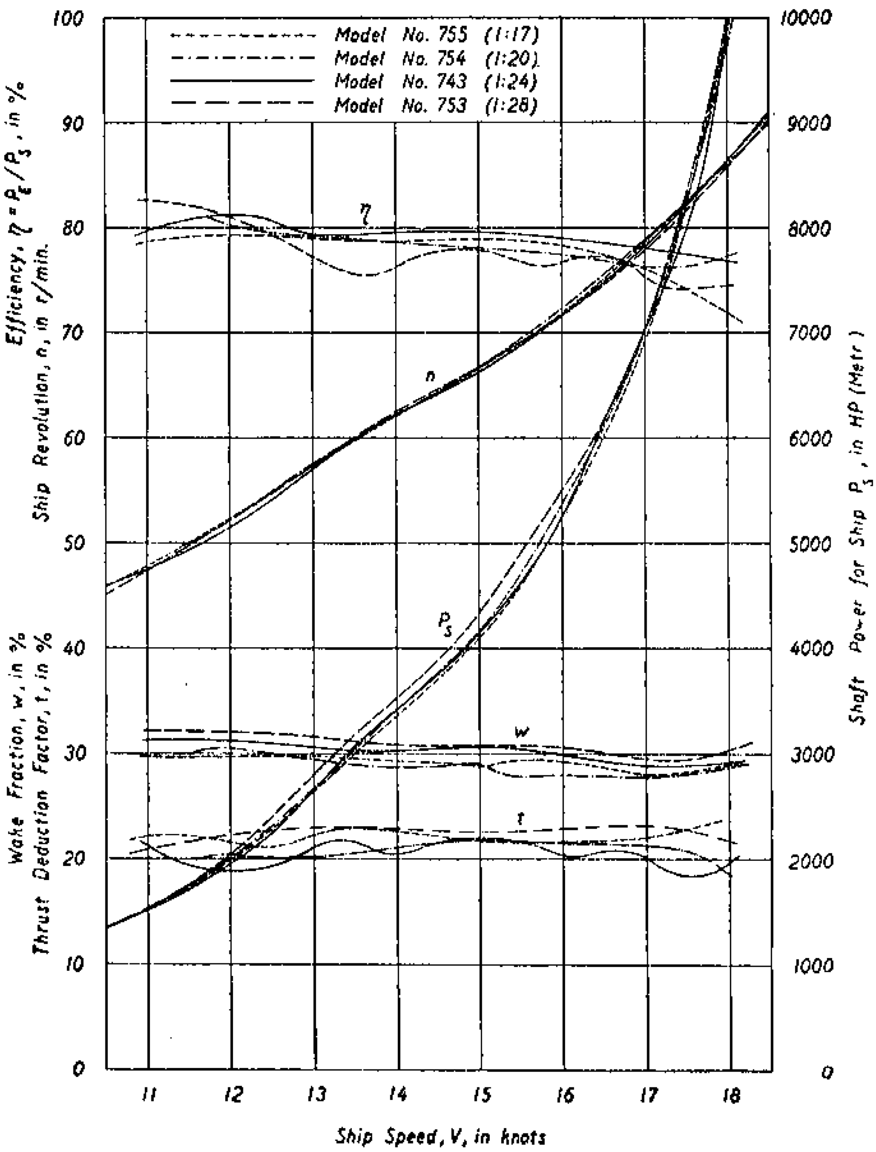

Fig. 15.—Seef-Propulsion Test Results according to Froude.

The results of the self-propulsion tests with Models Nos. 755 (1:17), 754 (1:20), 743 (1:24) and 753 (1:28) are summarised here in Fig. 15. These curves have been derived in accordance with the practice normally adopted at SSPA, i. e. from self-propulsion tests carried out according to the so called Continental method (Gebers), the skin friction correction, based on the Froude frictional coefficients, being applied as a towing force.

No corrections for scale effects, air resistance, hull condition, etc., have been applied in converting the measured values to ship scale.

Wake fractions have been calculated in the usual way, using the propeller as a wake integrator. Values of wake fraction were worked out on the basis of thrust identity, with the aid of curves of the results from the open water propeller tests.

The main particulars of the propellers used in these tests were as follows (in ship scale):

Number of blades = 4.
Diameter D = 6.25 m.
Pitch P = 6.98 m (max.).
Pitch ratio P/D = 1.117.

APPENDIX 2

Comparison with Other Victory Series.

Experiments with geometrically similar Victory models have been carried out both at NSMB, Wageningen and at NPL, Teddington. In these experiments the models were run without rudders and at a draught which differed to some extent from that used at SSPA.

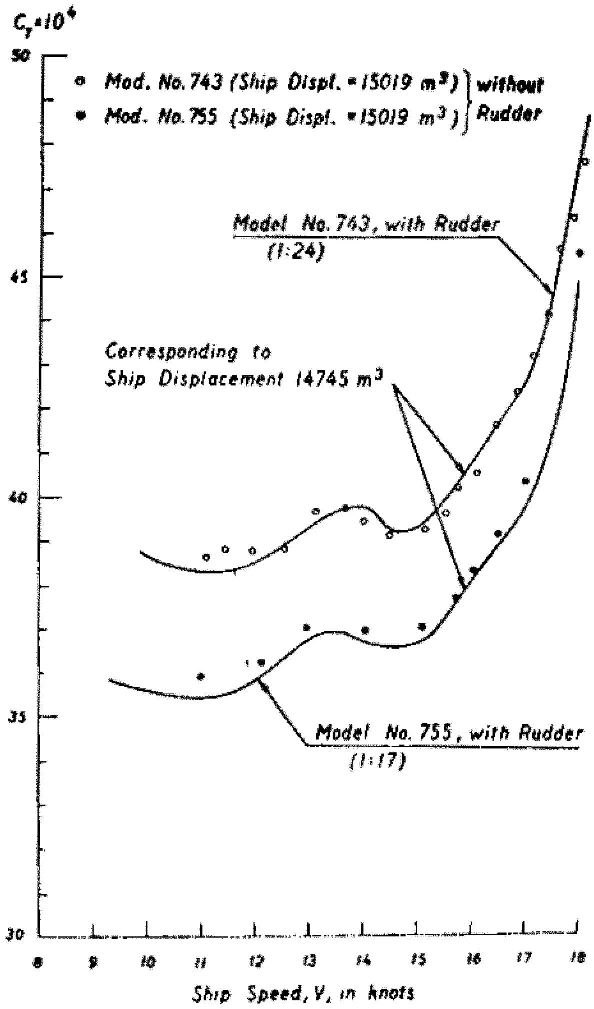

Fig. 16.

In order therefore to be able to compare the various results, two of the SSPA models (Nos. 743 and 755) were run without rudders and at a draught corresponding to 8.69 m (displacement = 15 019 metres³). As may be seen from Fig. 16, the resistance coefficients calculated from the results of these new tests showed only insignificant differences compared with those obtained earlier. In the case of Model No. 743 (1:24) the differences appear to be accidental in character, while the resistance coefficients for Model No. 755 are generally, though insignificantly, increased in the new tests. In spite of these small differences, the experimental results obtained previously have been used in the comparison with the NSMB and NPL results.

In experiments with families of similar models in testing tanks of different cross section area, the resistance coefficients obtained at the same Froude number can be compared. The differences between these resistance coefficients and those derived at the appropriate Reynolds number from the correct "three-dimensional" friction line should be the same within practical limits. Various hypotheses regarding the correct slope of the friction line can thus be investigated by means of such comparisons. This method was proposed by Hughes, who explains its theoretical basis in [4].

An attempt has been made at SSPA to apply the Hughes method to the Victory model series. The corrected experimental results from NSMB and NPL, as tabulated in [3] were used for this purpose. In the case of the NPL results, only those obtained from the tests in No. 2 tank, where plates were employed as turbulence stimulators, were considered. The SSPA results were corrected for tripwire resistance and laminar area according to the method explained in [3].

The assumed friction line was of the form proposed by Hughes (see also equ. (6)), namely

$$C_V = (1 + k)\, C_{FH}$$

where C_{FH} follows the Hughes basic line, equ. (1). Various values of k were tried, viz. 0, 0.10 and 0.27 (corresponding, according to Hughes, to $r = 1.00$, 1.10 and 1.27), representing different slopes of the friction line. Using $k = 0$ gives a line of approximately the same slope as alternative A, while $k = 0.27$ gives a line similar to alternative C (see Section 5).

Figs. 17-19 show the differences, ΔC_T, between the values of resistance coefficient, C_T, obtained experimentally at the various establishments and the C_V values calculated from the above equation.

The correct slope of the friction line, according to Hughes, is that which gives the best agreement between the ΔC_T curves derived from the various tanks at the same speed. A comparison between Figs. 17-19 indicates that the closest agreement is achieved with a k value of about 0.10 (Fig. 18).

Since the cross sectional areas of the NSMB and SSPA tanks are of the same order, there is less possibility of drawing definite conclusions from a comparison between results obtained at these establishments than from a comparison between NPL and SSPA values.

In Fig. 20, the differences between the above mentioned ΔC_T curves from SSPA and NPL are plotted to a base of k at three different blockage values ($A_M/A_T = 0.50$, 0.70 and 0.90 %). In this case, the k-value corresponding to the best friction line is indicated by the intersection of the ΔC_T-difference lines with the k-axis. Generally, this k-value lies between 0.10 and 0.20, i. e. lower than the value of 0.27 proposed by Hughes. However, there is appreciable scatter and presumably the results are influenced to some degree by errors of measurement, small differences in model form and surface condition, different turbulence stimulation and other unknown factors.

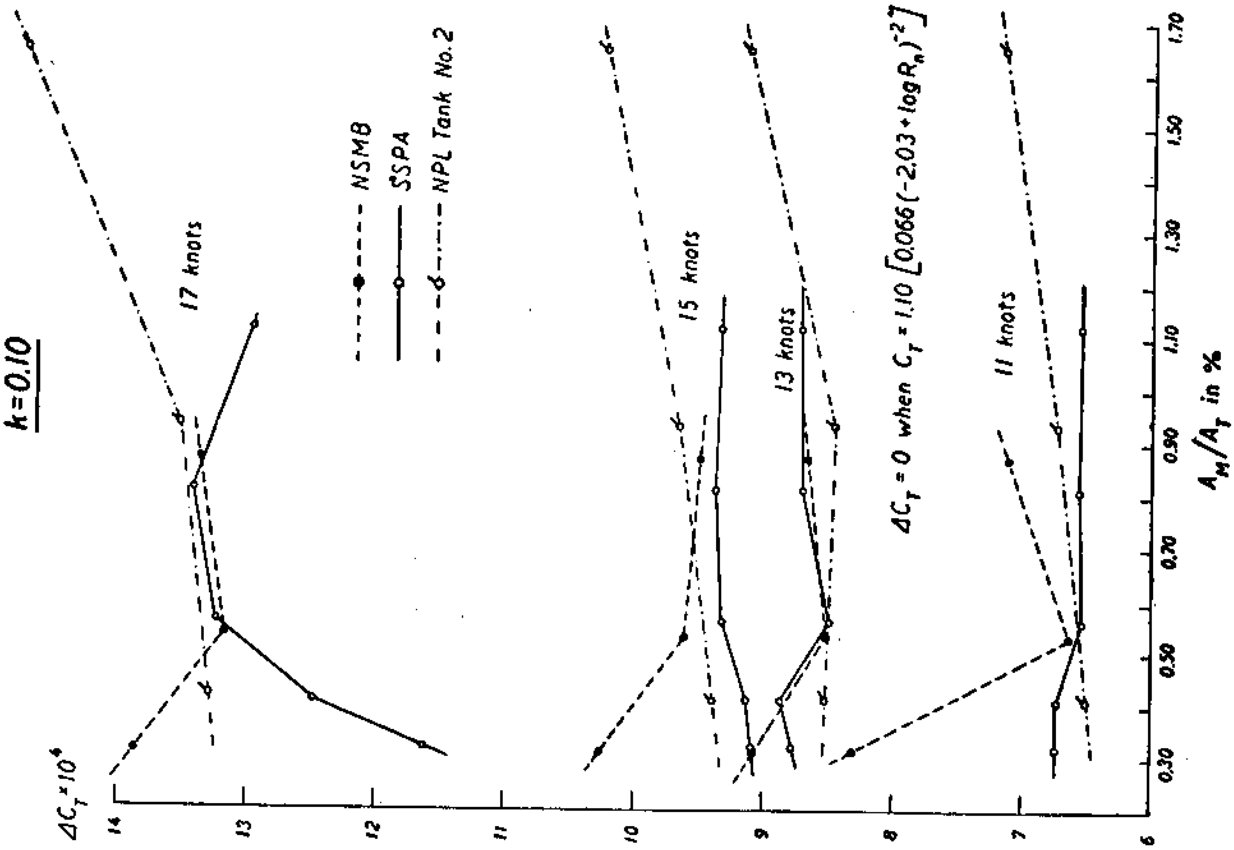
k=0.10
ΔC_T × 10^4
17 knots
15 knots
13 knots
11 knots
NSMB
SSPA
NPL Tank No. 2
ΔC_T = 0 when C_T = 1.10 [0.066(-2.03 + log R_n)^-2]
A_M/A_T in %
0.30
0.50
0.70
0.90
1.10
1.30
1.50
1.70
6
7
8
9
10
11
12
13
14

Fig. 18.

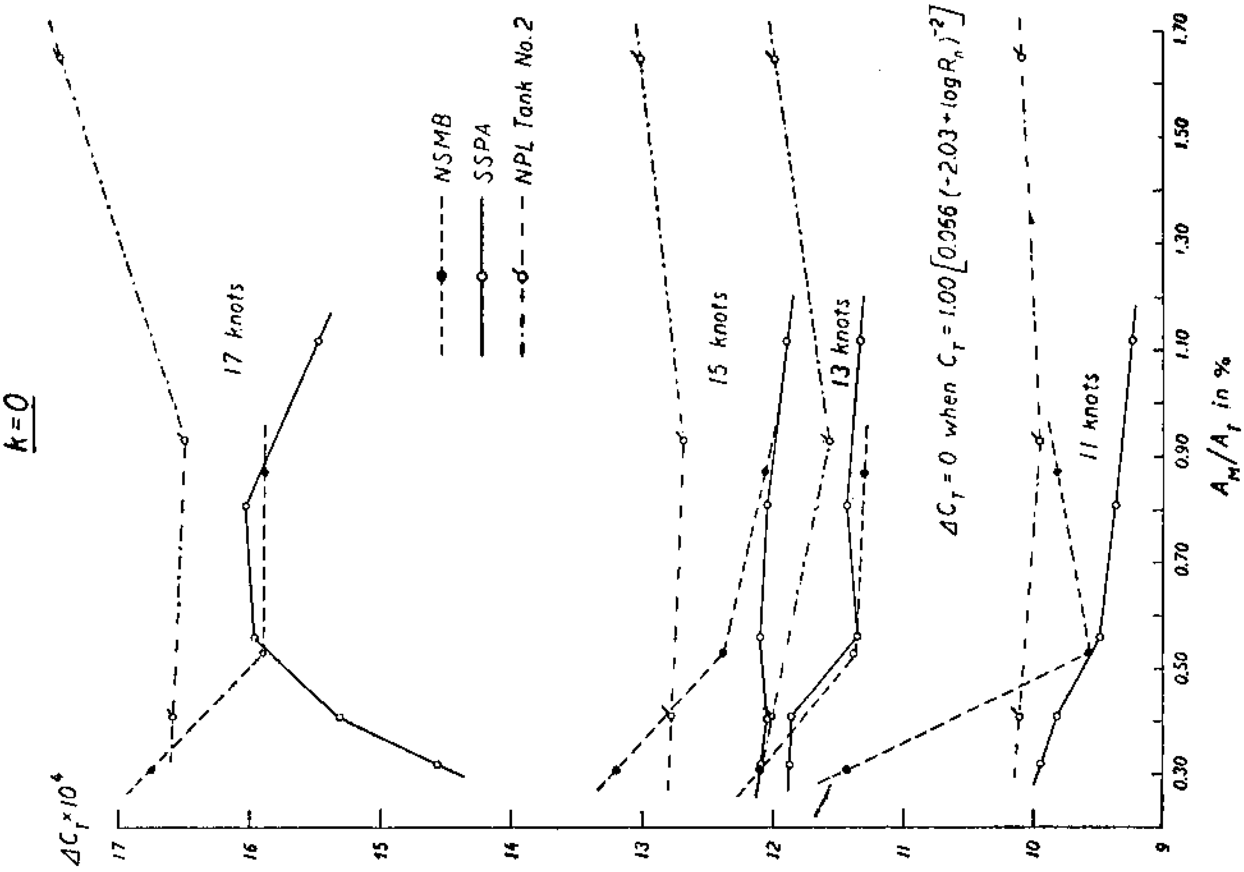
k=0
ΔC_T × 10^4
17 knots
15 knots
13 knots
11 knots
NSMB
SSPA
NPL Tank No. 2
ΔC_T = 0 when C_T = 1.00 [0.066(-2.03 + log R_n)^-2]
A_M/A_t in %
0.30
0.50
0.70
0.90
1.10
1.30
1.50
1.70
9
10
11
12
13
14
15
16
17

Fig. 17.

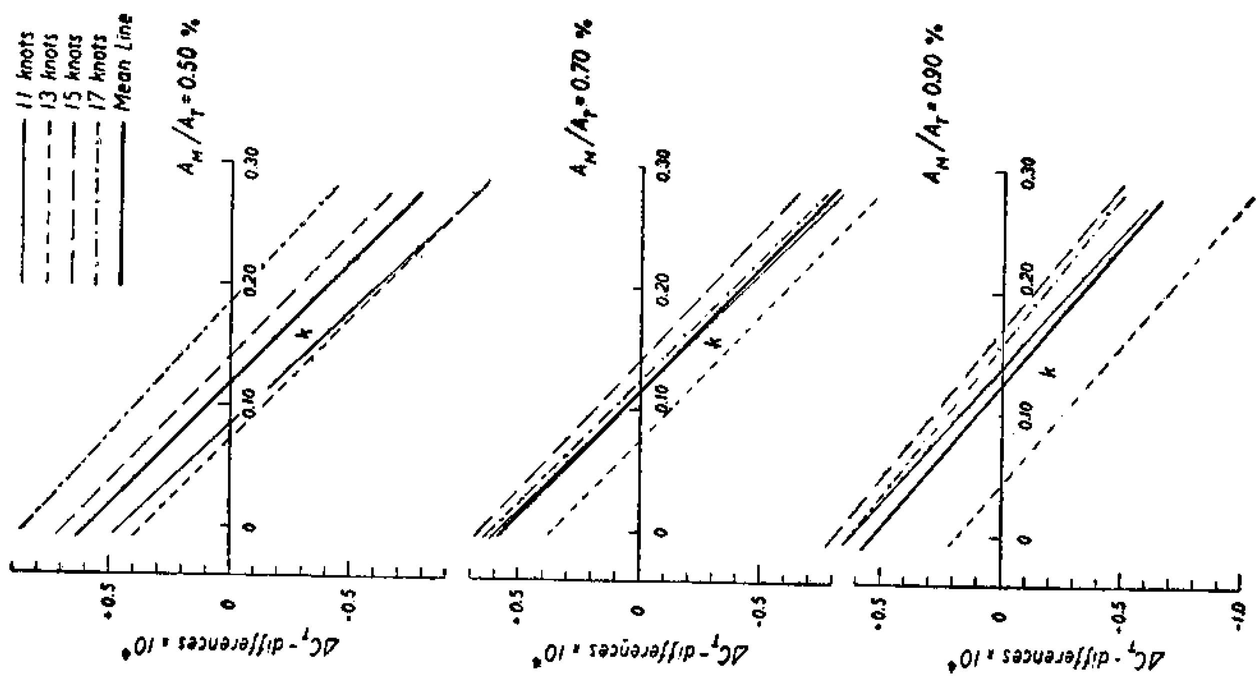
11 knots
13 knots
15 knots
17 knots
Mean Line
$A_M/A_T = 0.50$ %
$A_M/A_T = 0.70$ %
$A_M/A_T = 0.90$ %
ΔC_T - differences × 10^4
k

Fig. 20.

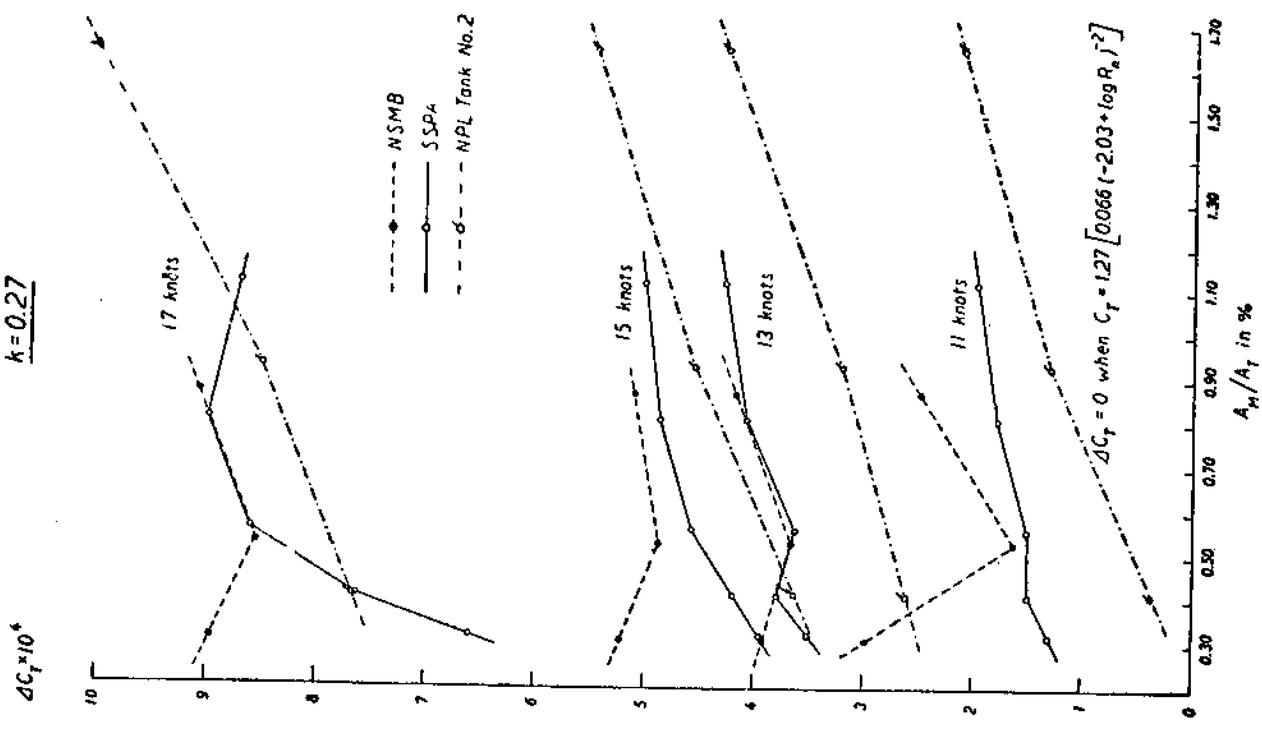
k=0.27
$\Delta C_T \times 10^4$
NSMB
SSPA
NPL Tank No.2
17 knots
15 knots
13 knots
11 knots
$\Delta C_T = 0$ when $C_T = 1.27\left[0.066(-2.03+\log R_n)^{-2}\right]$
A_M/A_T in %

Fig. 19.

APPENDIX 3

Model No. 755. Scale 1:17

Series 1
Normal Dynamometer

Date 2nd October, 1956
Watertemp. = 15,3° C

V m/sec.	$10^{-6} . R_n$	R kg	$10^4 . C_T$
1.372	9.641	4.36	35.50
1.495	10.51	5.20	35.65
1.562	10.98	5.78	36.29
1.624	11.41	6.36	36.95
1.686	11,85	6.85	36.91
1.754	12.33	7.39	36.80
1.813	12.74	7.86	36.63
1.875	13.18	8.42	36.69
1.934	13.59	9.10	37.27
2.003	14.08	10.04	38.33
2.063	14.50	10.82	38.94
2.122	14.91	11.71	39.83
2.182	15.33	12.94	41.64
2.234	15.70	14.51	44.54
2.008	14.11	10.09	38.33
1.847	12.98	8.18	36.74
1.598	11.23	6.08	36.48
1.649	11.59	6.54	36.84

Series 9
Pendulum Apparatus

Date 3rd October, 1956
Watertemp. = 15,3° C

V m/sec.	$10^{-6} . R_n$	R kg	$10^4 . C_T$
0.112	0.7870	0.037	45.35
0.158	1.110	0.074	45.35
0.212	1.490	0.123	37.76
0.255	1.792	0.179	42.19
0.310	2.178	0.263	41.93
0.356	2.502	0.341	41.23
0.409	2.874	0.437	40.02
0.455	3.197	0.541	40.05
0.502	3.528	0.647	39.33
0.556	3.907	0.793	39.31
0.602	4.230	0.950	40.15
0.658	4.624	1.099	38.88
0.709	4.982	1.285	39.16
0.585	4.111	0.893	39.99
0.532	3.738	0.730	39.53

Series 6
Normal Dynamometer

Date 29th November, 1956
Watertemp. = 13,7° C

V m/sec.	$10^{-6} . R_n$	R kg	$10^4 . C_T$
0.626	4.217	0.94	36.75
0.665	4.480	1.08	37.41
0.710	4.783	1.22	37.07
0.755	5.086	1.40	37.63
0.804	5.417	1.58	37.44
0.835	5.625	1.70	37.35

Date 30th November, 1956
Watertemp. = 13,6° C

V m/sec.	$10^{-6} . R_n$	R kg	$10^4 . C_T$
0.712	4.785	1.23	37.18
0.801	5.383	1.56	37.24
0.866	5.820	1.82	37.18
0.909	6.108	1.97	36.52
0.948	6.371	2.14	36.48
0.996	6.693	2.38	36.75
1.047	7.036	2.61	36.48
1.080	7.258	2.77	36.40
1.122	7.540	2.97	36.14
1.164	7.822	3.19	36.06
1.208	8.118	3.41	35.80
1.253	8.420	3.68	35.91
1.296	8.709	3.96	36.11
1.330	8.938	4.12	35.68
1.370	9.206	4.37	35.66
1.411	9.482	4.63	35.62
1.457	9.791	4.99	36.00
1.505	10.11	5.35	36.19

Model No. 754. Scale 1:20

Series 2
Normal Dynamometer

Date 26th July, 1956
Watertemp. = 18.2° C

V m/sec.	$10^{-6} \cdot R_n$	R kg	$10^4 \cdot C_T$
1.268	8.156	2.81	37.08
1.377	8.857	3.34	37.37
1.433	9.217	3.60	37.20
1.497	9.629	4.00	37.86
1.548	9.957	4.28	37.88
1.610	10.36	4.65	38.05
1.668	10.73	4.90	37.35
1.729	11.12	5.34	37.90
1.781	11.46	5.69	38.05
1.842	11.85	6.29	39.32
1.899	12.21	6.78	39.87
1.958	12.59	7.42	41.04
2.011	12.93	8.20	43.01
2.068	13.30	9.40	46.62

Series 7
Normal Dynamometer

Date 4th December, 1956
Watertemp. = 13.6°

V m/sec.	$10^{-6} \cdot R_n$	R kg	$10^4 \cdot C_T$
0.568	3.245	0.58	38.14
0.624	3.565	0.71	38.67
0.677	3.868	0.83	38.41
0.750	4.285	1.00	37.71
0.793	4.530	1.14	38.45
0.839	4.793	1.28	38.56
0.879	5.022	1.39	38.16
0.919	5.250	1.50	37.67
0.957	5.467	1.63	37.75
0.999	5.707	1.78	37.84
1.040	5.942	1.92	37.63
1.082	6.181	2.06	37.31
1.107	6.324	2.14	37.05
1.151	6.576	2.33	37.29
1.189	6.793	2.45	36.76
1.233	7.044	2.65	36.97
1.273	7.273	2.84	37.16
1.299	7.421	2.95	37.10
1.340	7.655	3.16	37.31
1.377	7.867	3.31	37.03
1.130	6.456	2.20	36.54
1.169	6.678	2.41	37.39
0.725	4.142	0.95	38.33
0.777	4.439	1.08	37.94

Series 10
Pendulum Apparatus

Date 24th August, 1956
Watertemp. = 15.9° C

V m/sec.	$10^{-6} \cdot R_n$	R kg	$10^4 \cdot C_T$
0.289	1.754	0.155	39.37
0.339	2.057	0.214	39.49
0.395	2.397	0.294	39.98
0.443	2.688	0.360	38.92
0.489	2.967	0.442	39.22
0.543	3.295	0.551	39.64
0.592	3.592	0.666	40.30
0.643	3.902	0.778	39.92
0.693	4.205	0.896	39.58
0.748	4.539	1.041	39.47
0.802	4.867	1.175	38.75
0.850	5.158	1.324	38.88
0.474	2.876	0.418	39.45
0.371	2.251	0.257	39.62
0.248	1.505	0.122	42.08

Model No. 743 Scale 1:24

Series 3 — Normal Dynamometer — Date 27th June, 1956, Watertemp. = 16.6° C

V m/sec.	$10^{-6} \cdot R_n$	R kg	$10^4 \cdot C_T$
1.143	5.883	1.63	38.11
1.253	6.449	1.97	38.33
1.305	6.717	2.14	38.39
1.365	7.026	2.39	39.18
1.420	7.309	2.61	39.55
1.468	7.556	2.80	39.67
1.522	7.834	2.96	39.03
1.578	8.122	3.20	39.24
1.627	8.374	3.45	39.79
1.678	8.637	3.75	40.68
1.733	8.920	4.09	41.60
1.791	9.218	4.48	42.66
1.841	9.476	4.98	44.86
1.893	9.743	5.65	48.16
1.706	8.781	3.95	41.44
1.763	9.074	4.28	42.05
1.808	9.306	4.64	43.34

Series 8 — Pendulum Apparatus

Date	V m/sec.	$10^{-6} \cdot R_n$	R kg	$10^4 \cdot C_T$
Date 30th November, 1956, Watertemp. = 13.6° C	0.515	2.451	0.376	43,31
	0.564	2.685	0.449	43.12
	0.597	2.842	0.498	42.66
	0.628	2.989	0.550	42.60
	0.666	3.170	0.606	41.72
Date 1st December, 1956, Watertemp. = 13.6° C	0.624	2.970	0.535	41.96
	0.679	3.232	0.636	42.15
	0.737	3.508	0.731	41.11
	0.775	3.689	0.809	41.14
	0.810	3.856	0.891	41.47
	0.842	4.008	0.972	41.87
	0.883	4.203	1.024	40.10
	0.908	4.322	1.083	40.13
	0.954	4.541	1.193	40.04
	0.994	4.731	1.286	39.76
	1.032	4.912	1.372	39.34
	0.944	4.493	1.166	39.95
	0.843	4.013	0.949	40.77
	0.711	3.384	0.679	41.02

Series 11 — Pendulum Apparatus — Date 22nd August, 1956, Watertemp. = 16.0° C

V m/sec.	$10^{-6} \cdot R_n$	R kg	$10^4 \cdot C_T$
0.300	1.521	0.120	40.71
0.336	1.703	0.153	41.38
0.393	1.992	0.214	42.33
0.439	2.225	0.266	42.15
0.488	2.474	0.335	42.97
0.538	2.727	0.407	42.94
0.594	3.011	0.490	42.42
0.638	3.234	0.561	42.08
0.695	3.523	0.660	41.72
0.804	4.075	0.854	40.34
0.895	4.537	1.039	39.61
0.966	4.897	1.205	39.43
1.037	5.257	1.370	38.91
0.743	3.766	0.736	40.71

Model No. 758 Scale 1:28

Date 10th August, 1956
Watertemp. = 16.9° C

Series 4 Normal Dynamometer			
V m/sec.	$10^{-6} . R_n$	R kg	$10^4 . C_T$
1.062	4.722	1.08	39.78
1.166	5.184	1.31	40.03
1.216	5.406	1.43	40.20
1.269	5.642	1.58	40.78
1.311	5.829	1.70	41.11
1.363	6.060	1.84	41.15
1.408	6.260	1.94	40.70
1.460	6.491	2.07	40.36
1.514	6.731	2.26	40.99
1.551	6.896	2.40	41.49
1.605	7.136	2.63	42.44
1.655	7.358	2.83	42.94
1.705	7.580	3.18	45.48
1.745	7.758	3.49	47.64

Date 23rd August, 1956
Watertemp. = 16.9° C

Series 12 Pendulum Apparatus			
V m/sec.	$10^{-6} . R_n$	R kg	$10^4 . C_T$
0.287	1.247	0.089	45.02
0.392	1.703	0.159	42.98
0.443	1.925	0.206	43.65
0.488	2.120	0.247	43.11
0.540	2.346	0.306	43.61
0.590	2.564	0.362	43.23
0.642	2.789	0.431	43.48
0.691	3.002	0.499	43.44
0.743	3.228	0.561	42.24
0.795	3.454	0.636	41.82
0.849	3.689	0.723	41.69
0.903	3.924	0.804	40.99
0.999	4.341	0.988	41.15
1.105	4.801	1.181	40.20
1.188	5.162	1.370	40.36
0.950	4.128	0.888	40.90

Model No. 778 Scale 1:32

Date 17th November, 1956
Watertemp. = 14.2° C

Series 5 Normal Dynamometer			
V m/sec.	$10^{-6} . R_n$	R kg	$10^4 . C_T$
0.998	3.620	0.75	40.89
1.090	3.953	0.91	41.59
1.177	4.269	1.07	41.95
1.228	4.454	1.18	42.49
1.268	4.599	1.24	41.87
1.322	4.795	1.34	41.63
1.370	4.969	1.45	41.95
1.413	5.125	1.56	42.42
1.463	5.306	1.68	42.63
1.502	5.448	1.78	42.84
1.552	5.629	1.95	43.96
1.592	5.774	2.12	45.43
1.642	5.956	2.41	48.54
1.142	4.142	0.99	41.22
0.998	3.620	0.75	40.89

Date 21st November, 1956.
Watertemp. = 14.0° C

Series 13 Pendulum Apparatus			
V m/sec.	$10^{-6} . R_n$	R kg	$10^4 . C_T$
0.200	0.7216	0.034	45.61
0.296	1.068	0.073	44.94
0.401	1.447	0.126	42.55
0.495	1.786	0.196	43.44
0.547	1.974	0.240	43.55
0.596	2.150	0.292	44.64
0.646	2.331	0.337	43.85
0.701	2.529	0.399	44.09
0.748	2.699	0.458	44.45
0.798	2.879	0.500	42.64
0.847	3.056	0.565	42.77
0.901	3.251	0.621	41.54
0.949	3.424	0.682	41.12
0.993	3.583	0.758	41.75
1.105	3.987	0.953	42.38
1.046	3.774	0.841	41.74
1.206	4.351	1.147	42.82
1.303	4.701	1.324	42.34
0.446	1.609	0.153	41.77

Mr. H. B. Lindgren (Oral remarks).

Percentage differences between C_T values from the tests of the SSPA Victory series and lines parallel with the Committee's alternative 1 have been plotted as a function of the blockage for different speeds in the normal speed range (Fig. 21). The requirement that the same prediction of ship resistance should be obtaine regardless of model scale is quite satisfactorily fulfilled. Most of the differences can be said to fall within the accuracy of the measurements. It is not possible to judge between alternatives 1 and 2 in this way, due to the fact that they only differ at higher Reynolds numbers.

A line found to give good correlation according to the above method is not necessarily —as all of you know— although it can be useful for practical purposes, a correctly sloped line. That the slope seems to be correct can for instance depend upon the fact that a greater slope of the true line is balanced by the blockage influence. Dr. Hughes' method of getting the correct slope of the viscous line by comparing geosim series from different tanks seems in all the series investigated, including the SSPA Victory series comparison, to verify this theory astonishingly well although I am afraid that he sometimes draws more definite conclusions than the quality of the test results given reason for.

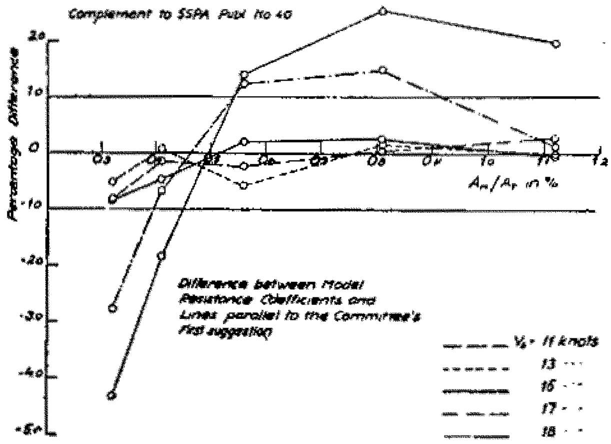

Fig. 21.

The work that goes on in this field and the results obtained by, for instance, Dr. Hughes and Mr. Lap, let us surmise that we have not found the final solution of the problems yet. It seems also that there will be a considerable time before any such solution is reached. Until then it would be reasonable to use Schoenherr's line or preferably a slightly modified line according to alternative 1 of the Committee with a constant form allowance. An agreement on this will be welcomed by the Gothenburg Tank.

Prof. L. Landweber, Mr. T. T. Siao (*) (Written contribution).

Introduction.

In a recent paper (1)** it was indicated that there appeared to be a need for generalizing the well-known logarithmic law of turbulent boundary layers and such a generalization was derived on the basis of a suggestion due to Townsend (2). It was found that the logarithmic formulas constitute only one member of a family of possible formulas, among which that one which best fits the boundary-layer data should be selected.

A previous study of data on flat-plate boundary layers (3) on the basis of the logarithmic law had resulted in mean curves which it was stated could be considered only as tentative because of the wide scatter of the data. It was then suggested that the method of analysis there illustrated be reapplied to more precise data when they become available.

The characteristics of the boundary-layer on a flat plate in zero pressure gradient were subsequently measured in the wind tunnel at the Iowa Institute of Hydraulic Research. An important feature of the new data is the independent determination of the shear stress at the wall by means of Preston's method (4) employing a calibrated stagnation tube. While the experimental value of the momentum thickness reliably gives the total surface shear, its derivative can be considered as only a poor measurement of the local surface-shear stress. The reason is that the measuring of the local slopes of a curve drawn through experimental points cannot be very accurate. Because of the precautions and care taken in obtaining this set of data, the authors believe that it is sufficiently precise to serve as a basis for the critical analysis of boundary-layer laws. Other sets of flat-plate boundary-layer data have not been included in this analysis because it was found in such a previous study (3) that the dispersion of these data is large and consequently might confuse the desired comparison between the two laws. Consequently only the new data will be presented and analyzed in accordance with both the logarithmic law and its generalization.

The analysis based upon the logarithmic laws has been refined in one practical aspect. The necessity for selecting a value for the boundary-layer thickness at each velocity profile, a value which is inherently ill-defined because of its asymptotic nature, has been avoided by following a suggestion made in the Appendix of the previous work (3) to use a natural length scale defined by the boundary-layer laws themselves. It will be seen that this modification does not affect the formal nature of the resulting expressions for the boundary-layer characteristics, although the numerical values of some of the constants appearing in the equations are greatly altered.

The analyses according to the logarithmic law and its generalization, which will hereafter be called the power law, appear to fit the data equally well in the narrow range of Reynolds numbers encompassed by the tests. The resulting curves for the coefficients of shear stress and drag versus Reynolds number deviate appreciably from each other at higher Reynolds numbers, the value predicted by the logarithmic law exceeding that from the power law by about 10 percent at a Reynolds number of 10^9. Thus comparison with two-dimensional flat-plate data at higher Reynolds numbers should indicate which of the boundary-layer laws is preferable.

* From Iowa Institute of Hydraulic Research, not in attendance.

** Numbers in parentheses refer to the Bibliography at the end of the paper.

Nomenclature.

a, b, B, C — constants.
c_0, c_1, c_2, c_3 — constants defined by definite integrals
c_f — frictional-resistance coefficient.
c_τ — shear-stress coefficient.
d — a constant.
D — pitot-tube diameter.
f, f_0 — inner-law function $f(y^*)$, $f_0 = f(y_0^*)$.
F, F_1 — outer-law function $F(\zeta)$; $F_1 = F(\zeta_1)$.
g — a function, $g(\sigma)$.
H — shape parameter, δ_1/δ_2.
k, K — constants.
L — length scale of boundary layer.
n — an exponent.
p — pressure.
p — pressure outside boundary layer.
p_t — total pressure, $p + \frac{1}{2}\rho u^2$
r, r_i — outer and inner radii of pitot tube.
$R_x, R_\delta, R_{\delta_1}, R_{\delta_2}$ — Reynolds numbers based on indicated linear dimensions; e. g. $R_x = Ux/\nu$.
u — x-component of mean velocity.
u' — x-component of fluctuating part of velocity.
u_τ — shear velocity, $\sqrt{\tau_0/\rho}$.
U — incident velocity.
x — distance from leading edge in direction of flow.
y — normal distance from plate.
y^* — the dimensionless quantity yu_τ/ν.
y_0, y_1 — initial and final ordinates of linear logarithmic relation.
y_0^* — $y_0 u_\tau/\nu$.
y' — corrected value of y, $y' = y + \varepsilon$.
$\alpha, \alpha_0, \alpha_1$ — constants.
β, β_0, β_1 — constants.
γ — a constant.
δ — boundary-layer thickness.
δ_1, δ_2 — displacement and momentum thicknesses.
ε — correction to ordinate of pitot tube.
$\zeta, \zeta_1, \zeta_\delta$ — $\zeta = y/L$, $\zeta_1 = y_1/L$, $\zeta_\delta = \delta/L$.
η — $e^{\sigma/k}$
λ — a constant.
μ, ν — dynamic and kinematic viscosities.
ρ — density of fluid.
σ — dimensionless parameter, U/u_τ
τ_0 — shear stress at the wall.

Preston-s Method and Calibration of Stagnation Tube.

Suppose that a stagnation tube, the diameter of which is small compared to the thickness of the inner region of the boundary-layer, is placed in contact with the wall. Then the dynamical pressure p exerted on the tip of the tube depends on ρ, ν, τ_0, the outer radius r and the inner radius r_i of the tube. It can then be shown by dimensional analysis that the parameter $p_t r^2/\rho\nu^2$ depends on the parameters $\tau_0 r^2/\rho\nu^2$ and r/r_i.

Three tubes of 0.0277, 0.0416, and 0.0576 inch in outside diameter with the inner-outer diameter ratios 0.505, 0.577, and 0.722, respectively, were calibrated in a uniform flow created in a brass pipe 2.063 inches in inside diameter. The tube was placed in contact with the inner wall of the pipe at the outlet section, and the total pressure was read. The shear stress at the wall was calculated from the measurement of the pressure drop for a 5-foot stretch of the pipe. The constancy of the gradient in the axial direction was checked by measuring the pressure at an additional point remote from the stretch. The results obtained are shown in Table 1 and plotted in Fig. 2. Later, in measuring the shear stress on the plate, the tube with outside diameter 0.0416 inch was used since the response of the smaller tube was too sluggish.

Experimental Arrangement and Data.

The test portion of the wind tunnel at the Iowa Institute of Hydraulic Research is 24 feet long, 5 feet wide, and 3 feet 8 inches high, as shown in Fig. 3. The flat plate for test was placed in the horizontal central plane of the tunnel. It was composed chiefly of a main $^1/_4$-inch glass plate, 3 feet wide and 10 feet long, followed by another glass plate 3 feet wide and 4 feet long, and two flanking aluminum plates, 1 foot wide and 14 feet long. The plate started at the 10-foot section of the test section. As the head form of the plate a wooden piece of the forward half of a 4-to-1 elliptical cross section, 3 inches vertical and 6 inches horizontal, was smoothly connected to the leading edge of the flat plate. The plate was aligned by means of a surveying level and its straightness was achieved by adjusting the heights of the intermediate supports underneath and along the centerline of the plate.

The adjustable part of the tunnel ceiling was hinged to the fixed part one foot ahead of the leading edge of the plate and extended dowstream for the remaining length of the test portion. The clearance height at the leading edge of the plate was 1 foot 10 inches.

The tunnel was run at an average speed of 93.2 feet per second outside the boundary layer. The initial intensity of turbulence $\sqrt{\overline{u'^2}}/U$ had the measured value 0.002.

To measure the mean velocity u, a stagnation tube 0.0416 inch in diameter was used. The static pressure was indicated by a pitot tube $^1/_8$-inch in diameter placed outside the boundary layer and at a distance of 4 inches above the plate. The two tubes were connected to a sensitive differential gage which gave readings to 0.001 inch of alcohol. The contact of the tube with the plate was made by sighting the tube and its image in the glass plate through a magnifying telescope. The tube was moved downwards until the illuminated gap between the tube and its image just vanished. The vertical distance above the plate was thereafter indicated by a deformation dial graduated to 0.001 inch. Another stagnation tube was placed 4 inches above the plate to indicate the velocity of approach U.

The data obtained from the test are given in Tables 2 and 3. The indicated values of y correspond to

TABLE 1 CALIBRATION DATA OF STAGNATION TUBES IN CONTACT WITH WALL.

$r/r_i = 0.505$, $2r = 0.0277''$		$r/r_i = 0.577$, $2r = 0.0416''$		$r/r_i = 0.722$, $2r = 0.0576''$	
$\frac{p_t r^2}{\rho \nu^2} \times 10^{-4}$	$\frac{\tau_0 r^2}{\rho \nu^2} \times 10^{-3}$	$\frac{p_t r^2}{\rho \nu^2} \times 10^{-4}$	$\frac{\tau_0 r^2}{\rho \nu^2} \times 10^{-3}$	$\frac{p_t r^2}{\rho \nu^2} \times 10^{-4}$	$\frac{\tau_0 r^2}{\rho \nu^2} \times 10^{-3}$
0.219	0.0378	0.433	0.0816	1.147	0.1653
0.391	0.0753	1.156	0.1609	2.66	0.327
0.655	0.1120	1.952	0.241	4.44	0.488
1.572	0.221	3.60	0.399	10.22	0.963
2.98	0.360	6.44	0.636	18.08	1.579
4.82	0.530	10.31	0.949	28.1	2.32
6.69	0.695	14.23	1.266	37.5	3.04
10.47	1.027	18.14	1.576	57.5	4.49
14.36	1.386	28.5	2.36	77.9	5.90
18.26	1.663	38.9	3.13	97.6	7.26
22.06	1.970	49.4	3.89	118.5	8.61
		63.4	4.85		

the position of the geometric center of the face of the pitot. It has been shown (5), however, that the total pressure registered by a pitot in a transverse total-pressure gradient should be associated not with the geometric center but with a point displaced toward the region of higher velocity by an amount given approximately by $\varepsilon = 0.18\,D$, where D is the outer diameter of the pitot. The corrected values of the ordinates, $y' = y + 0.0075$ in inches, are given in Table 2 and were used in the analyses of the data.

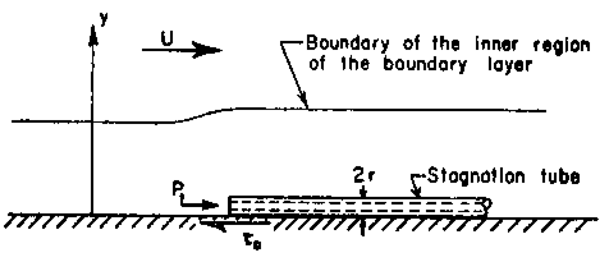

Fig. 1.—Stagnation tube in contact with wall.

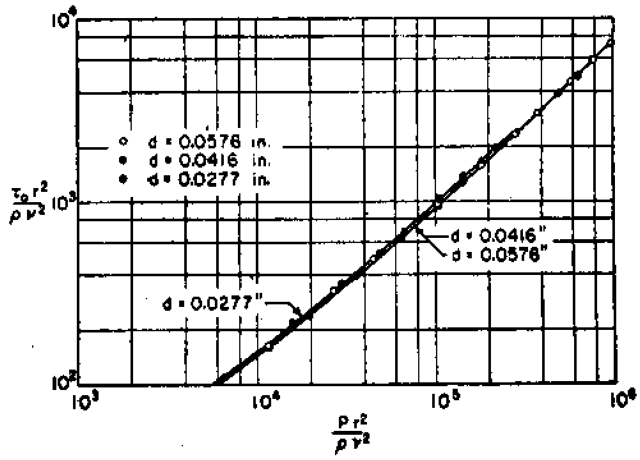

Fig. 2.—Calibration curves of stagnation tubes.

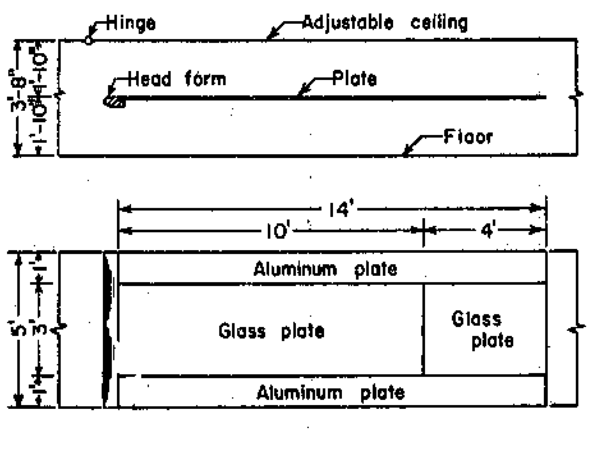

Fig. 3.—Main experimental arrangement.

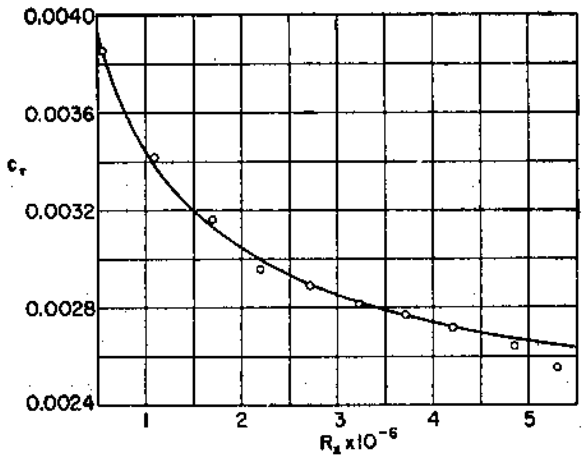

Fig. 5.—Experimental c_τ curves.

TABLE 2 EXPERIMENTAL VALUES OF u/U.

y'	$x = 1'$	2'	3'	4'	5'	6'	7'	8'	9'	9.95'
.0283	0.672	0.628	0.603	0.583	0.572	0.564	0.558	0.550	0.554	0.533
.0375	0.698	0.651	0.622	0.604	0.593	0.583	0.575	0.567	0.563	0.554
.0575	0.741	0.686	0.659	0.639	0.627	0.617	0.605	0.600	0.596	0.588
.0775	0.778	0.713	0.686	0.667	0.652	0.643	0.631	0.624	0.620	0.613
.1075	0.922	0.750	0.719	0.698	0.681	0.671	0.660	0.650	0.647	0.640
.1575	0.888	0.804	0.764	0.738	0.721	0.708	0.695	0.688	0.682	0.677
.2075	0.942	0.846	0.801	0.772	0.750	0.737	0.722	0.712	0.707	0.702
.2575	0.979	0.886	0.835	0.800	0.776	0.761	0.745	0.735	0.729	0.723
.3075	0.996	0.919	0.866	0.828	0.800	0.781	0.765	0.752	0.747	0.738
.3575	0.999	0.951	0.894	0.853	0.822					
.4075		0.975	0.919	0.876	0.843	0.820	0.802	0.786	0.778	0.770
.4575		0.990	0.942	0.897						
.5075		0.997	0.961	0.918	0.881	0.856	0.834	0.816		
.5575		1.000	0.978							
.6075			0.988	0.952	0.915	0.887	0.863	0.843	0.832	0.819
.7075			0.998	0.978	0.944	0.916	0.889			
.8075			0.999	0.994	0.969	0.941	0.916	0.894	0.877	0.863
.9075				0.999	0.985	0.963				
1.008				1.000	0.993	0.979	0.956	0.936	0.920	0.902
1.108					0.997					
1.208					0.999	0.996	0.983	0.968	0.952	0.934
1.408						1.000	0.997	0.991	0.976	0.963
1.608							1.000	0.998	0.992	0.982
1.808								1.000	0.998	0.996
2.008									1.000	0.999
2.208										1.000

TABLE 3 EXPERIMENTAL DATA OF THE PLATE FLOW

x	1	2	3	4	5	6	7	8	9	9.95
U	92.78	92.56	96.02	93.79	93.39	93.06	92.66	92.43	93.02	92.49
$\rho \times 10^{+3}$	2.245	2.244	2.255	2.245	2.237	2.232	2.222	2.218	2.238	2.233
$\nu \times 10^{+4}$	1.715	1.722	1.708	1.715	1.732	1.740	1.755	1.761	1.725	1.738
$R \times 10^{-6}$	0.5410	1.075	1.687	2.187	2.696	3.209	3.696	4.109	4.852	5.294
R_δ	15770	24640	37470	45560	53910	62390	70400	78740	89900	97570
R_{δ_1}	2150	3420	4720	5752	6828	7757	8822	9698	10850	11740
R_{δ_2}	1492	2459	3450	4215	5080	5823	6546	7247	8165	8846
$\tau_0 \times 10^{+2}$	3.722	3.287	3.287	2.919	2.864	2.722	2.643	2.569	2.558	2.441
$c_\tau \times 10^{+3}$	3.852	3.419	3.162	2.956	2.891	2.816	2.771	2.712	2.642	2.555
$c_f \times 10^{+3}$	5.516	4.575	4.090	3.855	3.769	3.629	3.542	3.452	3.366	3.342
σ	22.78	24.18	25.15	26.01	26.30	26.65	26.87	27.16	27.51	27.98
$2P/\rho U^2$	0.5236	0.5135	0.5144	0.5098	0.5117	0.5167	0.5163	0.5168	0.5176	0.5234

Quantities with dimensions in Table 3 are referred to foot-pound-second units. The mean-velocity measurements given in Table 2 are plotted in Fig. 4, and the c_τ -volues given in Table 3 are plotted in Fig. 5.

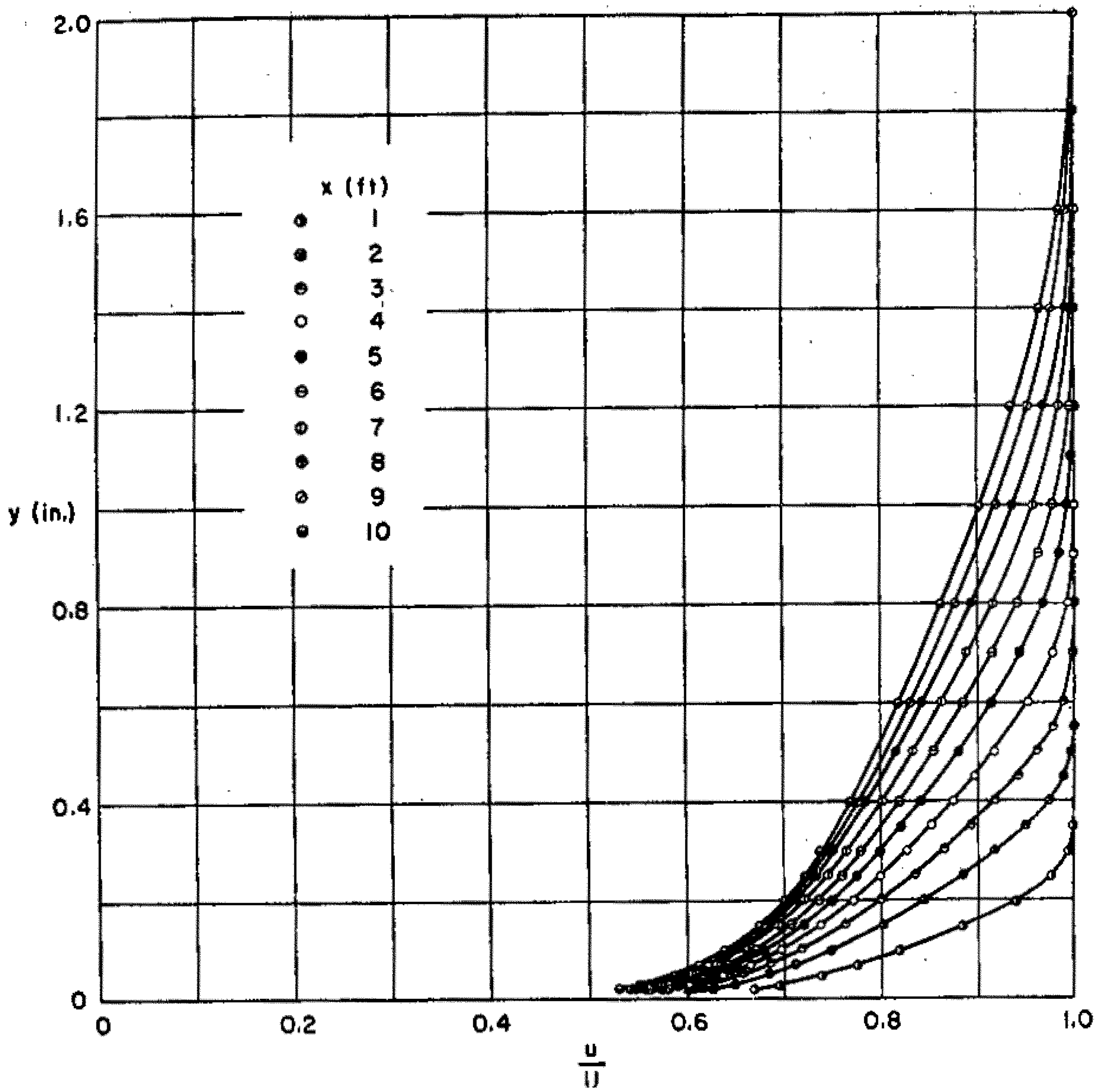

Fig. 4.—Experimental u/U profiles.

Analysis According to the Logarithmic Law.

In the neighborhood of the wall, it will be assumed that the boundary-layer velocity profile satisfies the inner law

$$\frac{u}{u_\tau} = f(y^*) \quad ; \quad y^* = \frac{yu_\tau}{\nu} \quad , \quad u_\tau = \sqrt{\frac{\tau_0}{\rho}} \qquad [1]$$

In a range overlapping the inner law and extending to the outer edge of the boundary layer, it will be assumed that the velocity profile obeys the outer law

$$\frac{U - u}{u_\tau} = F(\zeta) \quad , \quad \zeta = \frac{y}{L} \qquad [2]$$

where L is an unspecified length scale. This is usually taken as the boundary-layer thickness δ, although, because δ is ill-defined, Rotta (6) has used the displacement thickness δ_1 for this purpose. It will be shown that the boundary-layer laws themselves suggest a natural value for L.

Addition of Eqs. [1] and [2] gives

$$\sigma = f(y^*) + F(\zeta) \qquad [3]$$

where

$$\sigma = \frac{U}{u_\tau} = \sqrt{\frac{2}{c_\tau}} \quad , \quad c_\tau = \frac{2\tau_0}{\rho U^2} \qquad [4]$$

Differentiating Eq. [3] with respect to y and then multiplying by y now yields

$$y^* \frac{df}{dy^*} + \zeta \frac{dF}{d\zeta} = 0$$

whence, since y^* and ζ may be considered as independent variables, one has

$$y^* \frac{df}{dy^*} = -\zeta \frac{dF}{d\zeta} = k \qquad [5]$$

where k is a constant, and hence integration gives

$$f(y^*) = a + k \ln y^* \qquad [6]$$

$$F(\zeta) = b - k \ln \zeta \qquad [7]$$

in the overlapping range. Substituting Eqs. [6] and [7] into Eq [3] gives

$$\sigma = a + b + k \ln \eta \quad , \quad \eta = \frac{Lu_\tau}{\nu}$$

or, if the length scale is now chosen so as to make $b = -a$, one obtains

$$\eta = e^{\sigma/k} \quad , \quad L = \frac{\nu}{u_\tau} e^{\sigma/k} \qquad [8]$$

and

$$\zeta = y^* e^{-\sigma/k} \qquad [9]$$

It can radily be verified that the equations of the boundary-layer characteristics previously derived (3) are unaltered in form by the new choice of length scale. For the sake convenience, these equations are collected here:

$$c_\tau = \frac{2}{\sigma^2} \qquad [10]$$

$$R_\delta = \zeta_\delta \sigma \eta \qquad [11]$$

$$R_{\delta_1} = \beta\eta + \lambda \qquad [12]$$

$$R_{\delta_2} = \eta \left(\beta - \frac{\gamma}{\sigma}\right) - \lambda - \frac{\alpha}{\sigma} \qquad [13]$$

$$R_x = \eta\,[\beta\sigma^2 - (\gamma + 2\beta k)\,\sigma + 2k\,(\gamma + \beta k)] + \alpha\sigma + C \qquad [14]$$

$$c_f = 2R_{\delta_2}/R_x \qquad [15]$$

where

$\zeta_\delta = \delta/L$ is obtained from $F(\zeta_\delta) = 0$

$$\eta = e^{\sigma/k}$$

$$\alpha = c_1 - y_0^*\,[(f_0 - k)^2 + k^2]$$

$$\beta = \zeta_1\,(F_1 + k) + c_2$$

$$\gamma = \zeta_1\,[(F_1 + k)^2 + k^2] + c_3$$

$$\lambda = y_0^*\,(f_0 - k) - c_0$$

y_0^* is the initial value of y^* for which Eq. [6] is valid

ζ_1 is the terminal value of ζ for which Eq. [7] is valid

$$f_0 = f(y_0^*) \quad , \quad F_1 = F(\zeta_1)$$

$$c_0 = \int_0^{y_0^*} f\,dy^* \quad , \quad c_1 = \int_0^{y_0^*} f^2\,dy^*$$

$$c_2 = \int_{\zeta_1}^{\zeta_\delta} F\,d\zeta \quad , \quad c_3 = \int_{\zeta_1}^{\zeta_\delta} F^2\,d\zeta$$

The velocity profiles are plotted according to the inner law in Fig. 6 and the outer law in Fig. 7. The value $k = 2.36$ obtained from the slope of the linear portion of the curve in Fig. 6 was used to compute the values of ζ for Fig. 7. Test results for only the five sections from $x = 5$ to 8 feet, with Reynolds numbers from 2.7 to 4.7×10^6 were used in Fig. 7. It was considered that equilibrium turbulent velocity profiles were not yet established at positions farther upstream and that effects due to pressure gradients were appearing at positions farther downstream.

The mean straight lines drawn through a range of the data in Figs. 6 and 7 correspond to the choices

$$a = -b = 5.70 \quad , \quad k = 2.63$$

in Eqs. [6] and [7]. It appears that the data could be better fitted by a curve of non-zero curvature, as is attempted in the analysis according to the power law. Tillman (7) and Townsend (8) also found it difficult to fit a truly straight part to the inner-law plot. The lower limit y_0^* of the linear logarithmic range was taken at $y_0^* = 30$, the value found in the previous work (3).

The inner-law function for values of y^* less than 30 was fitted by the curve

$$\frac{u}{u_\tau} = y^* + 0.000962 y^{*3} - 0.00862 y^{*2.5} \qquad [16]$$

in which the coefficients and the exponent 2.5 have been chosen so as to satisfy conditions at $y^* = 0$ and $y^* = 30$ and to agree with Laufer's data (9). The graph of the curve is shown in Fig. 8.

Thus the logarithmic similarity laws given by the present experimental data may be summarized as follows:

(A) $$\frac{u}{u_\tau} = f(y^*) = y^* + 0{,}000962 y^{*3} - 0.00862 y^{*2.5} \,, \quad 0 < y^* < 30$$

(B) $$\begin{cases} f(y^*) = 5.70 + 2.36 \ln y^* & 30 < y^* < 0.004\, e^{\frac{\sigma}{2.36}} \\ F(\zeta) = -5.70 - 2.36 \ln \zeta \,, & 30\, e^{-\frac{\sigma}{2.36}} < \zeta < 0.004 \end{cases}$$

(C) $$\frac{U-u}{u_\tau} = F(\zeta)$$

as given by the curve in Fig. 7 for $0.004 < \zeta < 0.028$. The inner law is applicable to ranges (A) and (B), the outer law to ranges (B) and (C). From the fo-

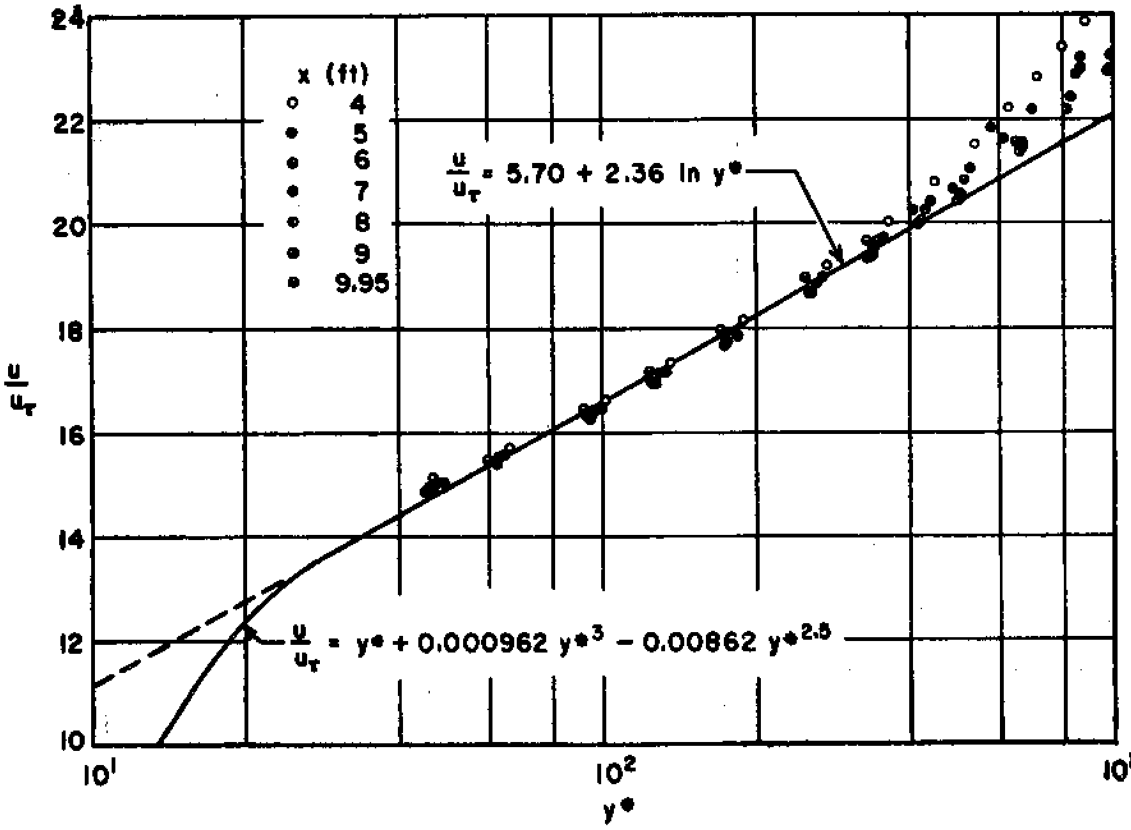

Fig. 6.—Inner logarithmic law.

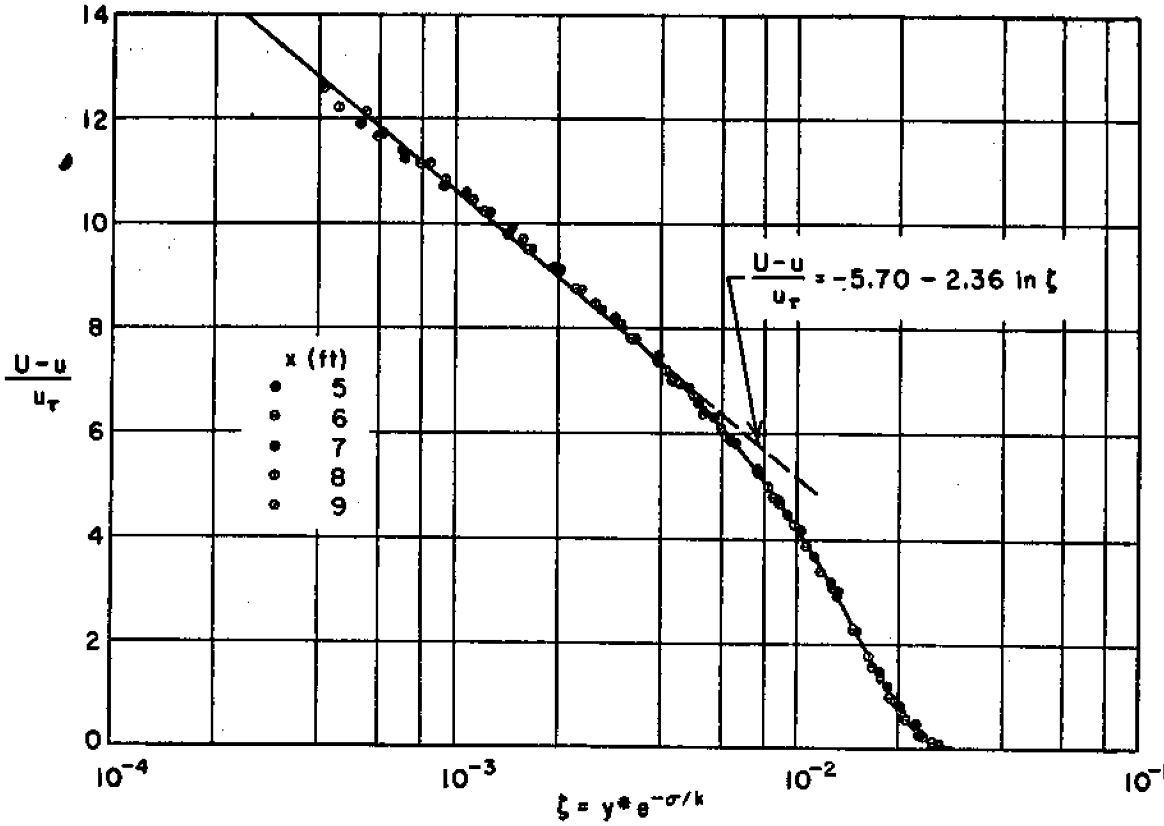

Fig. 7. Outer law.

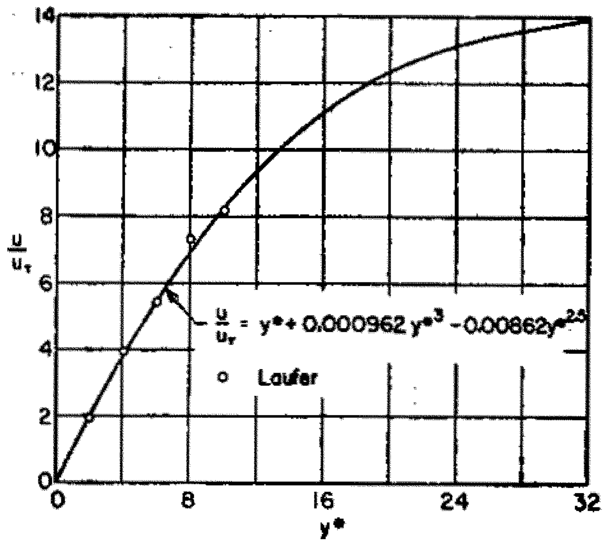

Fig. 8.—Velocity distribution for laminar sublayer.

regoing limits for the various ranges it is seen that

$$y_0^* = 30 \quad , \quad \zeta_1 = 0.004 \quad , \quad \zeta_\delta = 0.028 \ .$$

The requirement that the inner and outer laws overlap in range (B) may be expressed as the inequality

$$30 < 0.004\, e^{\frac{\sigma}{2.36}} \quad , \quad \text{or} \quad \sigma > 21$$

This gives limiting value of σ below which the logarithmic similarity laws cannot be applied.

The foregoing values of the constants obtained from the graphs and formulas for the similarity laws and of the definite integrals c_0 , c_1 , c_2 and c_3 derived from these graphs are collected in the following table. The values of the constants α , β , γ and λ , which occur in the equations of the boundary-layer characteristics are also given in the table 4.

TABLE 4 CONSTANTS FOR LOGARITHMIC SIMILARITY LAWS

$a = -b = 5.70$	$f_0 = 13.73$	$c_3 = 0.2559$
$k = 2.36$	$F_1 = 7.33$	$\alpha = -967$
$y_0^* = 30$	$c_0 = 281.1$	$\beta = 0.0975$
$\zeta_1 = 0.004$	$c_1 = 3071$	$\gamma = 0.6538$
$\zeta_\delta = 0.028$	$c_2 = 0.0588$	$\lambda = 60.03$

Substitution of these values of the constants into Eqs. [10] to [14] permits the values of the various quantities to be computed as functions of σ. The results are given in Table 5.

Curves of the values of H and c_τ against R_{δ_1} and R_{δ_2} , given in Table 5, as well as the values computed from the experimental data, are shown in Figs. 9, 10, and 11. These will be compared with the corresponding results from the power law in the following section.

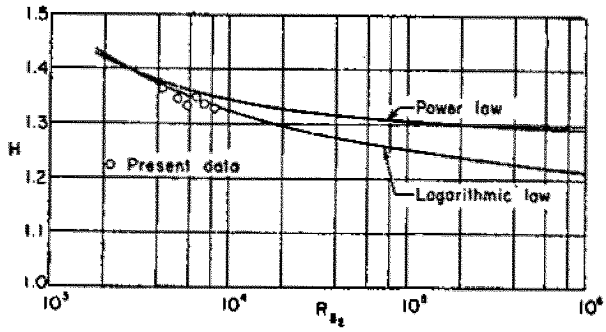

Fig.—9.—Comparison of computed curves for shape parameter H with experimental data.

The analytical expression for R_x , Eq. [14], contains a constant C whose value is a measure of the length preceding the transition from a laminar to a turbulent boundary-layer. Comparison of the curve of c_τ against $R_x - C$, obtained from Table 5, and the experimentally obtained values, shown in Fig. 11, indicates that the value $C = -2.5 \times 10^5$ should be selected in order to make the computed curve

TABLE 5 COMPUTED VALUES OF BOUNDARY-LAYER CHARACTERISTICS ON BASIS OF LOGARITHMIC SIMILARITY LAWS

σ	$10^3 \times c_\tau$	$10^{-4} \times \eta$	$10^{-6} \times R_\delta$	$10^{-6} \times R_{\delta_1}$	$10^{-6} \times R_{\delta_2}$	$H = \frac{\delta_1}{\delta_2}$	$10^5 \times (R_x - C)$	$10^5 \times R_x$	$10^3 \times c_f$
22	4.132	1.112	0.0684	0.0114	0.00738	1.544	2.770	0.270	
24	3.472	2.585	0.174	0.0258	0.01798	1.434	8.447	5.947	6.05
26	2.959	6.084	0.443	0.0599	0.0438	1.367	24.76	22.26	3.94
28	2.551	14.20	1.118	0.1390	0.1050	1.323	69.89	67.39	3.11
30	2.222	33.18	2.787	0.3240	0.2509	1.291	193.8	191.3	2.623
32	1.953	77.4	6.935	0.7550	0.5964	1.266	528.7	526.2	2.267
34	1.730	180.4	17.17	1.765	1.413	1.249	1425	1422	1.987
36	1.543	421.5	43.49	4.116	3.342	1.232	3811	3808	1.755
38	1.385	983.0	104.6	9.590	7.893	1.215	10087	10084	1.565
40	1.250	229.6	257.2	22.39	18.64	1.201	26542	26539	1.405

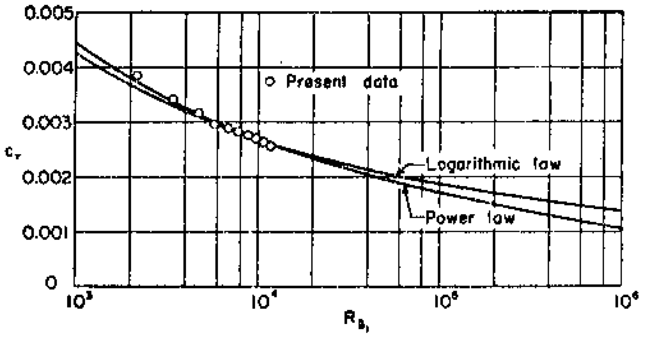

Fig. 10.—Comparison of computed curves with experimental data for c_τ versus R_{δ_1}.

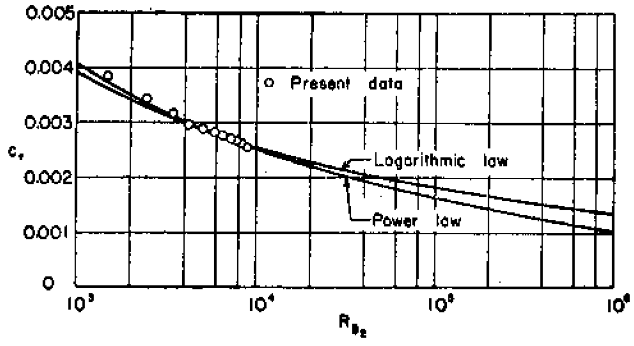

Fig. 11.—Comparison of computed curves with experimental data for c_τ versus R_{δ_2}.

agree with the data. The values of R_x and c_f given in Table 5 were computed on the basis of this value of C. Graphs of c_f versus R_x for various values of C are given in Fig. 12. Also shown in the figure are the values of c_f computed from the experimental data from the formula $c_f = 2R_{\delta 2}/R_x$. The values corresponding to $x = 1, 2, 3$, and 4 ft are included to show the effect of transition. These results will be discussed after those from the power law have also been obtained.

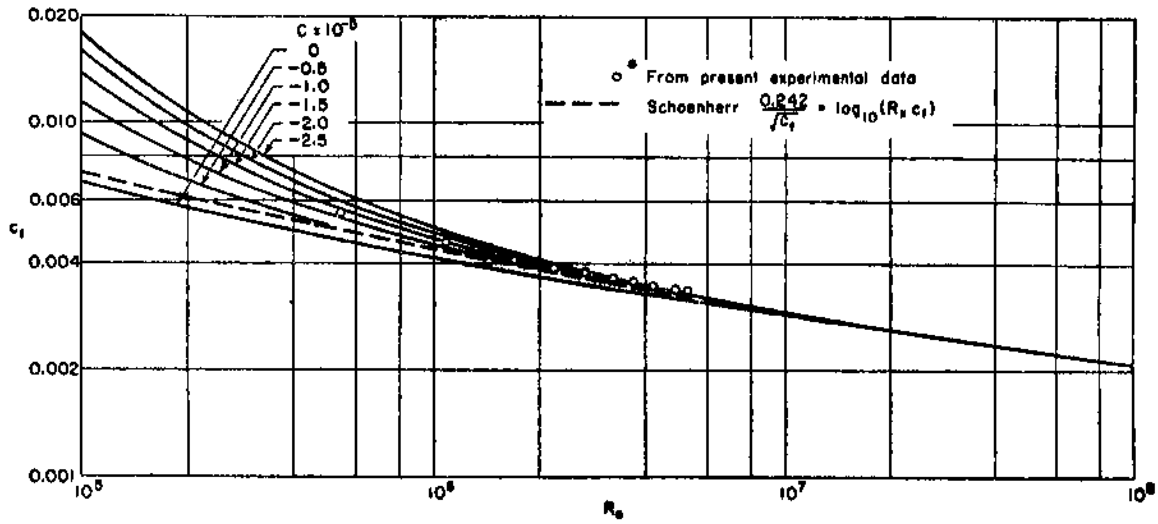

Fig. 12.—Comparison of curves of total frictional resistance coefficients computed from logarithmic law with experimental data and Schoenherr's curve.

Analysis According to the Power Law.

It has been shown (1) that if the outer law, Eq. [3], is generalized by replacing u_τ by u_{τ_1} where u_{τ_1} is the value of u_τ at an upstream position $x = x_1$, thereby taking into account, to a certain degree, the influence of the previous history of the flow, and if the corresponding values of σ are assumed to be functionally related, viz.,

$$\sigma_1 = \sigma\, g(\sigma) \quad , \quad \sigma_1 = \frac{U}{u_{\tau_1}} \qquad [17]$$

then the assumption that there exists a region in which the inner and outer laws overlap leads to a family of boundary-layer laws among which the logarithmic law is a special case. It is found that the law depends upon the value of an index n; if $n = 0$, the logarithmic relations are obtained, if $n \neq 0$, the resulting relations resemble power laws. The latter will be assumed to be the case in the present section.

It can then be derived, by purely functional analysis, that

$$g(\sigma) = \frac{B}{\sigma - d} \qquad [18]$$

where B and d are constants, and that the length scale L appearing in the modified form of the outer law, Eq. [2], is given by

$$\frac{Lu_\tau}{\nu} = (\sigma - d)^{1/n} \qquad [19]$$

Hence

$$\zeta = \frac{y}{L} = \frac{y^*}{(\sigma - d)^{1/n}} \qquad [20]$$

But, from Eq. [18],

$$\frac{U-u}{u_{\tau_1}} = \frac{U-u}{u_\tau} \; \frac{B}{\sigma - d} = B \, \frac{\sigma - u/u_\tau}{\sigma - d}$$

Hence the modified form of the outer law may be expressed as

$$\frac{\sigma - u/u_\tau}{\sigma - d} = F(\zeta) \qquad [21]$$

where the constant B has been incorporated into the function $F(\zeta)$. In the overlapping range the inner and outer laws assume the form

$$\frac{u}{u_\tau} = K y^{*n} + d \qquad [22]$$

$$\frac{\sigma - u/u_\tau}{\sigma - d} = 1 - K\zeta^n \qquad [23]$$

where K is another constant.

The equations of the boundary-layer characteristics, derived on the basis of the present assuptions (1), are as follows:

$$R_0 = \zeta_\delta \, \sigma \, (\sigma - d)^{1/n} \qquad [24]$$

$$R_{\delta_1} = \alpha_0 + \alpha_1 (\sigma - d)^{1 + 1/n} \qquad [25]$$

$$R_{\delta_2} = \frac{1}{\sigma} \Big[\beta_0 - \alpha_0 (\sigma - d) + d\alpha_1 (\sigma - d)^{1 + 1/n} + \beta_1 (\sigma - d)^{2 + 1/n} \Big]$$

$$R_x = -(\alpha_0 d + \beta_0)(\sigma - d) + d^2 \alpha_1 (\sigma - d)^{1 + 1/n} + d\left(\frac{\alpha_1}{2n+1} + \beta_1\right)(\sigma - d)^{2 + 1/n} + \frac{n+1}{3n+1} \beta_1 (\sigma - d)^{3 + 1/n} + C \qquad [27]$$

where

$$\alpha_0 = dy_0^* - c_0 + \frac{K}{n+1} y_0^{*n+1}$$

$$\alpha_1 = \zeta_1 - \frac{A\zeta_1^{n+1}}{n+1} + c_2$$

$$\beta_0 = \frac{K^2 y_0^{*2n+1}}{2n+1} + \frac{K d y_0^{*n+1}}{n+1} + dc_0 - c_1$$

$$\beta_1 = \frac{K\zeta_1^{n+1}}{n+1} - \frac{K^2 \zeta_1^{2n+1}}{2n+1} + c_2 - c_3$$

$\zeta_\delta = \delta/L$ is obtained from $F(\zeta_\delta) = 0$.

y^*_0 is the initial value of y^* for which Eq. [22] is valid.

ζ_1 is the terminal value of ζ for which Eq. [23] is valid

c_0, c_1, c_2, c_3 have formally the same definitions as for the logarithmic case.

In order to plot the velocity profiles according to the inner and outer laws it is first necessary to choose a value for n. After several trials, $n = 0.08$ was selected as the value wich gives best agreement with the present data. The velocity profiles are plotted according to the inner law in Fig. 13 and the outer law in Fig. 14. The value $d = -10$ obtained from the linear portion of the curve in Fig. 13 was used to compute the values of ζ for Fig. 14. The mean straight lines drawn through a range of the data in Figs. 6 and 7 correspond to the choices

$$n = 0.08 \quad , \quad d = -10 \quad , \quad K = 18.4$$

It will be assumed that for values of y^* less than 30 the inner law is given by the same curve as was used in the logarithmic case. Thus the power laws given by the present experimental data may be summarized as follows:

(A) $$\frac{u}{u_\tau} = y^* + 0.000962\, y^{*3} - 0.00862\, y^{*2.5} \quad ,$$

$$0 < y^* < y_0^*$$

(B) $$\begin{cases} \dfrac{u}{u_\tau} = 18.4\, y^{*.08} - 10 \quad , \quad y_0^* < y^* < \zeta_1 (\sigma + 10)^{1/n} \\[2ex] \dfrac{\sigma - u/u}{\sigma + 10} = 1 - 18.4\,\zeta \quad , \quad \dfrac{y_0^*}{(\sigma + 10)^{1/n}} < \zeta < \zeta_1 \end{cases}$$

(C) $$\frac{\sigma - u/u_\tau}{\sigma + 10} = F(\zeta)$$

as given by the curve in Fig. 7 for $\zeta_1 < \zeta < \zeta_\delta$ where $y_0^* = 30$, $\zeta_1 = 8.29 \times 10^{-18}$, $\zeta_\delta = 70.2 \times 10^{-18}$.

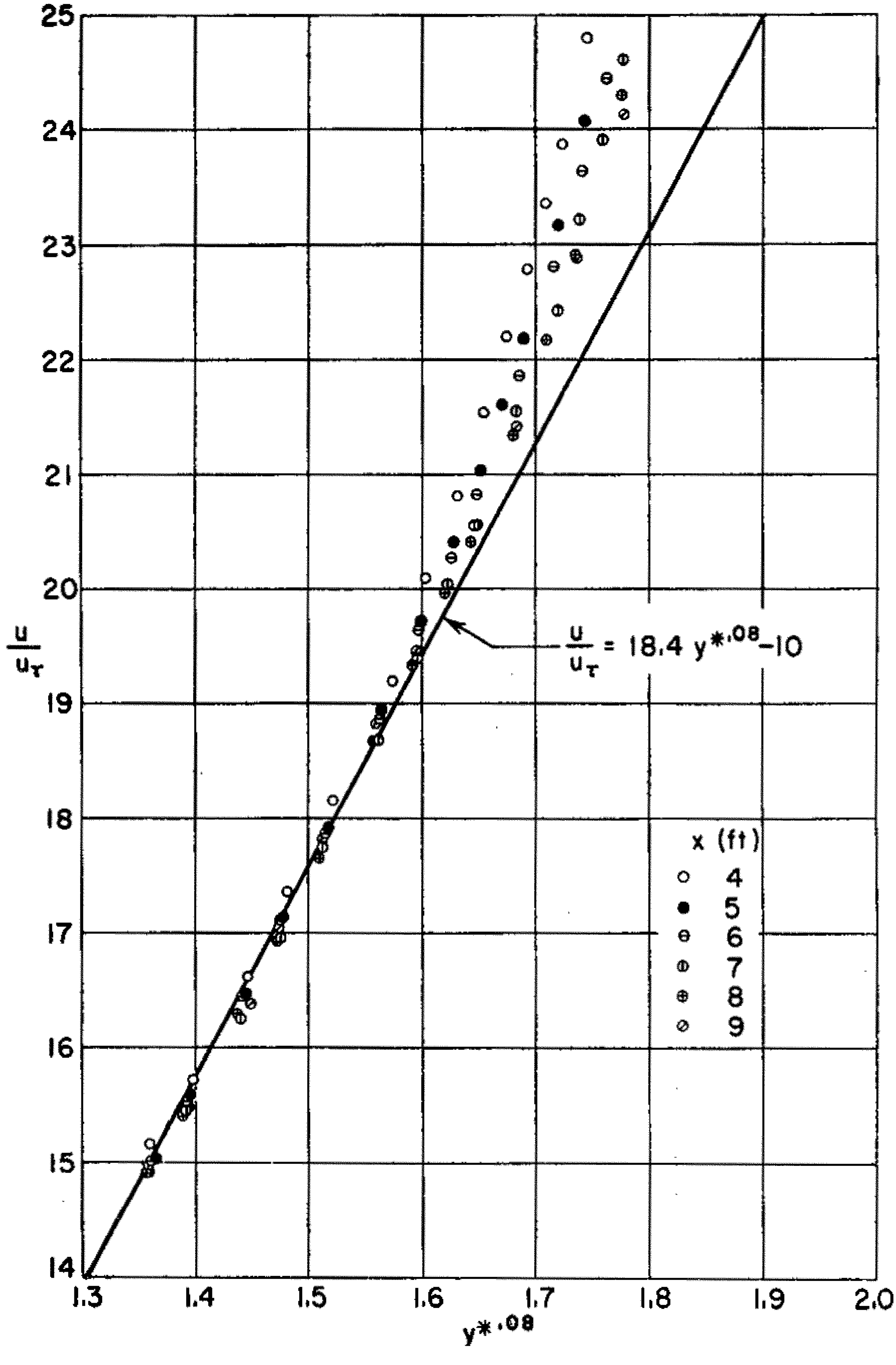

Fig. 13.—Inner power law.

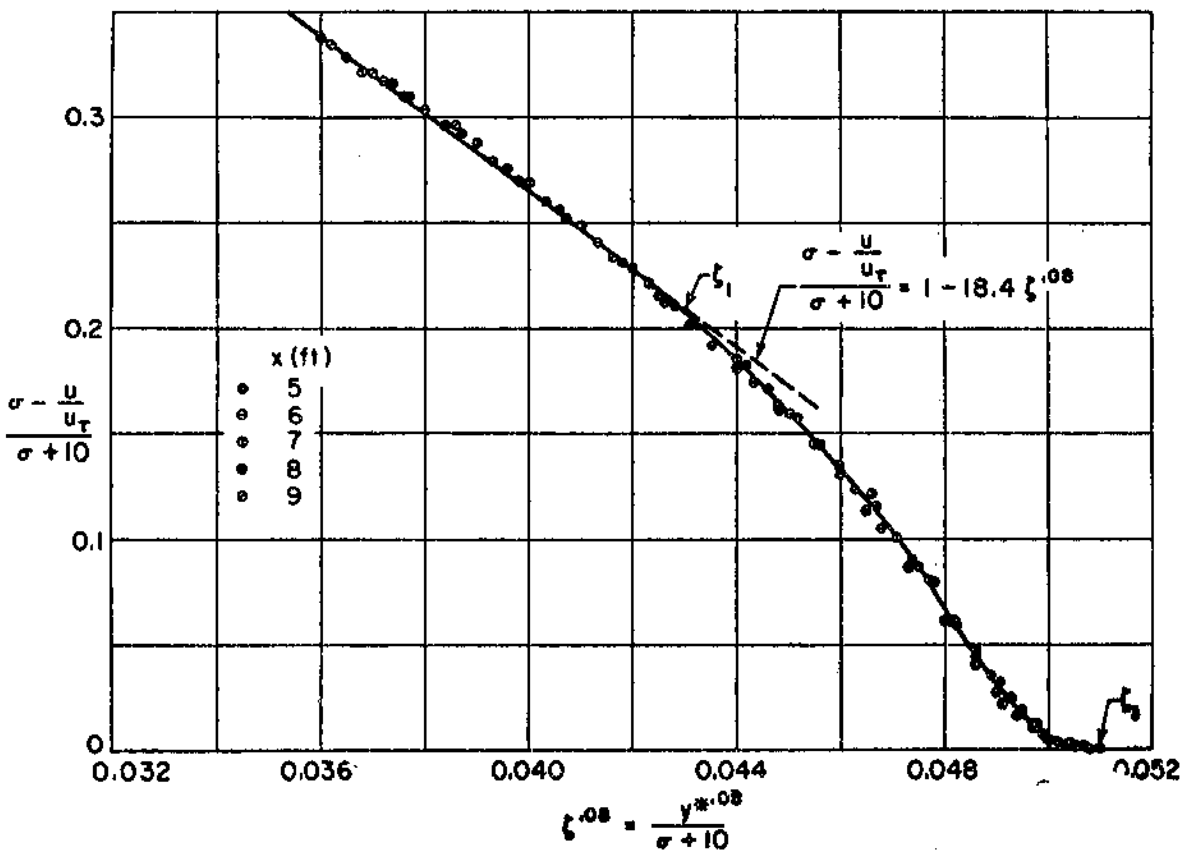

Fig. 14.—Outer power law.

The requirement that the inner and outer laws overlap in range (B) gives the inequality

$$\sigma > \left(\frac{y_0^*}{\zeta_1}\right)^n - 10 = 20.5$$

This gives the limiting value of σ below which the power laws cannot be applied.

The values of the constants obtained from the graphs of the similarity laws, of the definite integrals c_0, c_1, c_2, c_3, and of the constants appearing in the equations of the boundary-layer characteristics are given in Table 6.

TABLE 6 CONSTANTS FOR POWER SIMILARITY LAWS

$n = 0.08$	$c_2 = 3.804 \times 10^{-18}$	$\alpha_0 = 90.13$
$d = -10$	$c_3 = 0.4662 \times 10^{-18}$	$\alpha_1 = 6.02 \times 10^{-18}$
$K = 18.4$	$y_0^* = 30$	$\beta_0 = 2495$
$c_0 = 281.1$	$\zeta_1 = 8.29 \times 10^{-18}$	$\beta_1 = 4.937 \times 10^{-18}$
$c_1 = 3071$	$\zeta_0 = 70.2 \times 10^{-18}$	

TABLE 7 COMPUTED VALUES OF BOUNDARY-LAYER

Characteristics on Basis of Power Similarity Laws

σ	$10^3 \times c_\tau$	$10^{-5} \times R_\delta$	$10^{-5} \times R_{\delta_1}$	$10^{-5} \times R_{\delta_2}$	$H = \frac{\delta_1}{\delta_2}$	$10^{-6} \times R_x - C$	$10^3 \times c_f$: $C = 0$	$C = -10^6$	$C = -2 \times 10^6$
20	0.005	0.0409	0.00618	0.00374	1.652	0.7825	9.559		
22	4.132	0.1006	0.01349	0.00909	1.485	3.15	5.771	8.456	15.81
24	3.472	0.2345	0.02952	0.02080	1.419	9.634	4.318	4.818	5.449
26	2.959	0.520	0.06318	0.04604	1.372	25.22	3.651	3.802	3.966
28	2.551	1.096	0.1290	0.09596	1.344	62.60	3.066	3.116	3.167
30	2.222	2.235	0.2574	0.1938	1.328	144.9	2.675	2.694	2.712
32	1.953	4.367	0.4943	0.3750	1.318	322.2	2.328	2.335	2.342
34	1.730	8.299	0.9259	0.7050	1.313	678.5	2.078	2.081	2.084
36	1.543	15.36	1.691	1.296	1.305	1399	1.853	1.854	1.855
38	1.385	25.43	3.005	2.311	1.300	2875	1.608	1.608	1.609
40	1.250	48.52	5.225	4.031	1.296	5424	1.486	1.487	1.487

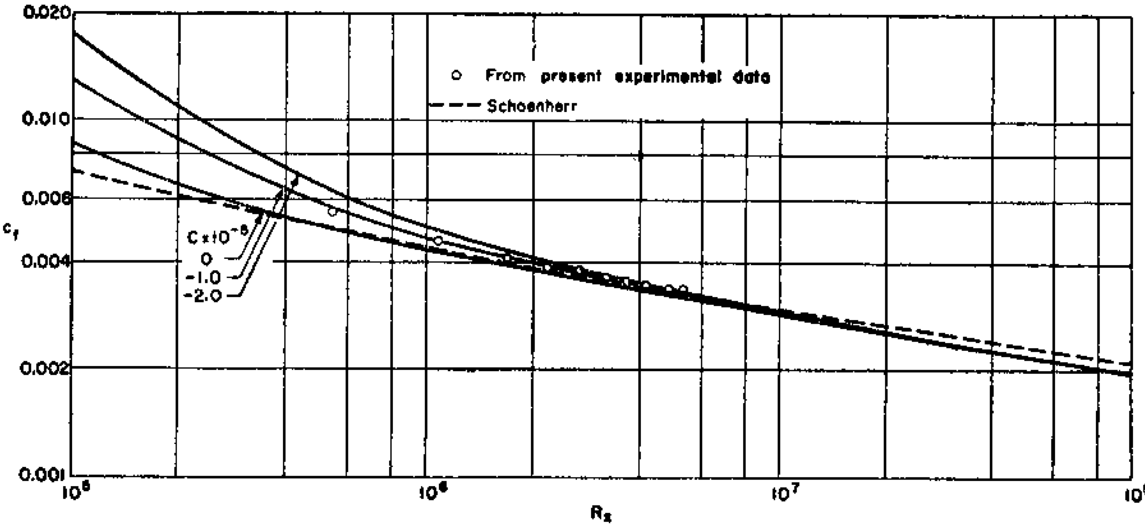

Fig. 15.—Comparison of curves of total frictional resistance coefficients computed from power law with experimental data and Schoenherr's curve.

Substitution of these values of the constants into Eqs. [24] to [27] permits the values of the various quantities to be computed as functions of σ. The results are given in Table 7.

Curves of the values of H and c_τ against R_{δ_1} and R_{δ_2} from Table 7 are shown in Figs. 9, 10, and 11, and those for c_f in Fig. 15. It is seen in Fig. 9 that the logarithmic law is in better agreement with the data for H versus R_{δ_2} than the power law, although the deviation between the laws in the neighborhood of the data is only about one percent. At larger values of R_{δ_2} the desviation between the laws becomes greater, amounting to 6 percent when $R_{\delta_2} = 10^6$. Figures 10 and 11, however, show that the curves of c_T versus R_{δ_1} and R_{δ_2} agree equally well with the data, especially for the values of x from 5 to 9 feet on which the analyses have been based. At larger Reynolds numbers the predictions from the two laws differ considerably. The present analysis indicates the value of c_T at $R_{\delta_2} = 10^6$ for the logarithmic law exceeds that for the power law by about 35 percent. It has been pointed out (1) that such a large difference is to be expected at high Reynolds numbers because of the greatly differing analytical nature of the laws.

Comparison between the experimental data and Schoenherr's curve for c_f versus R_x in Figs. 12 or 15 shows that the data exceed the curve by 10 percent at $x = 1$ foot ($R_x = 5.4 \times 10^5$) and by about 3 percent from $x = 3$ feet to $x = 10$ feet ($R_x = 1.7 \times \times 10^6$ to 5.3×10^6). This difference may be at least partly resolved by correcting the momentum equation for the effects of the turbulent fluctuations and the slight three-dimensionality of the flow because of boundary-layer growth along the side walls of the wind tunnel. It was shown in the discussion of Reference (3) that, because of the turbulent fluctuations, the value of c_f, which was computed from the data from the formula $c_f = 2R_{\delta_2}/R_x$, should be reduced by about 1.5 percent. An analysis of the effect of the three-dimensionality of the flow, following a method due to Kehl (10), indicates that an additional reduction of 1.5 percent is plausible. Thus it appears that the agreement between the present data and Schoenherr's curve is excellent. It is also observed in Fig. 12 that Schoenherr's curve practically coincides with the curves for c_f predicted by the logarithmic law at Reynolds numbers greater than about 10^7. At lower Reynolds numbers, however, Schoenherr's curve cuts across the family of logarithmic curves, so that, if the latter are correct, the predictions from Schoenherr's curve would tend to be low in the range of Reynolds numbers from 10^5 to 10^6. Similar remarks apply to the comparison between Schoenherr's curve and those computed from the power law in the lower range of Reynolds numbers. In the upper range, the friction coefficients from Schoenherr's curve exceed those from the power-law curves, beginning at about $R_x = 5 \times \times 10^6$, and amounting to about 7.5 percent at $R_x = 10^8$. Since Schoenherr's curve essentially coincides with the asymptote of the logarithmic-law curves, this shows that the latter exceed the power-law curves by the same amount in the upper range of Reynolds numbers. In the lower range, $R_x = 10^6$ to 10^6, the predictions from the two laws are practically coincident at suitable values of C.

The trend of the data for c_τ and c_f at the low values of x ($x = 1$ to 4 feet) is interesting. On the basis of either Fig. 10 or Fig. 11 one would conclude that the boundary layer was already turbulent at these positions. This is also indicated by the comparison between the data and Schoenherr's curve in Figs. 12 and 15. Yet the velocity profiles at these positions, shown in Fig. 4, had not yet attained the equilibrium state to which similarity laws could be applied. This phenomenon is not unexpected, since it is well known that boundary layers are sensitive to their previous history and adjust themselves slowly to an equilibrium state after a disturbance. Thus, although transition occurred ahead of the one-foot position on the plate in the present experiment, the velocity profiles did not become suitable for similarity analysis until at least $x = 5$ feet.

Conclusions.

It appears possible to analyze the mean velocity profile on a flat plate in zero pressure gradient equally well by either the logarithmic law or the power law. The predictions from the two laws agree reasonably well for Reynolds numbers R_x less than 10^7, but deviate increasingly and significantly at higher Reynolds numbers. Thus data at higher Reynolds numbers than those for the presently available ones are needed to distinguish between the two laws.

Schoenherr's curve is in excellent agreement with the present data and the prediction on the basis of the logarithmic law at Reynolds numbers greater than 2×10^6. At lower Reynolds numbers the indications by the data and by either law are that Schoenherr's curve is too low.

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Dr. L. Landweber (Oral remarks):

In continuation of an earlier work published in 1950 in which I had analysed the basis of the work of many the two-dimensional friction lines, I have determined it would be desirable to obtain new data. We repeated the analysis, and in the interim we have also been able to carry through an exploitation of a theory by Townsend which has been presented at the previous international Conference of this body. And therefore I took the opportunity to carry through an analysis of these data with two methods.

The interesting result is that one can equally well carry through such an analysis by either theory, and the point I wish to emphasize is that we have available not only the well-known logarithmic theory, but also a whole family of possible theories of which the first one may be presented only as a special case. The practical application of these results is that one may find certain evidence which is not direct evidence, since we are dealing with two-dimensional friction lines.

But whatever evidence is available indicates that if we accept the logarithmic law there is a certain confirmation for suggestion 1 of the Friction Committee and that if we feel that the generalization of the law is true, then there is a confirmation or an indication for proposal 2 of the Committee. I would like to leave this audience with the thought that these are only indications because the entire work concerned only two-dimensional friction lines.

Prof. Dr. K. Wieghardt (Written contribution).

Introduction.

To study the effect of transverse curvature on frictional resistance the boundary layer along a smooth cylinder (diameter $2r_o = 150$ mm) and a square prism with the same "diameter" (side length $a = 2r_o = 150$ mm) has been investigated in a windtunnel. The bodies were 3,5 m long with the last 50 cm sticking out of the test section; a slotted wall jet with 1 m diameter and 3,5 m length; the wind speed was $U = 30$ m/s. The cylinder had a hemispherical head; correspondingly the head of the prism was the surface of intersection of the two half cylinders over opposite sides of the prism's front. This form of head was chosen to ensure that turbulent transition happened at the end of the head where the static pressure increases rapidly.

The static pressure was found constant for $x > 0.8$ m (i. e. $x/r > 11$), with x = distance from the end of the head. Since the boundary layers around the bodies proved to be symmetrical, tests were made only in a single plane through the axis of the cylinder and in various sections perpendicular to one side plane of the prism. The total frictional resistance of a body of length $x + r$ was computed by the measured momentum loss as follows:

a) *Cylinder*:

Momentum loss

$$D_o = \rho U^2 \, 2\pi r_o d_2 (x)$$

with

$$d_2 (x) = \int_0^\infty \frac{u}{U} \left(1 - \frac{u}{U}\right) \left(1 + \frac{y}{r}\right) dy \, .$$

ρ = desity of air = 0.121 Kg s²/m⁴ at about 25° C
u = velocity in the boundary layer.
y = wall distance.
r_o = radius of cylinder = 75 mm.
Wetted surface = S_o = head + shaft = $2\pi r_o (r_o + x)$

$$c_F = \frac{D_o}{S_o \rho U^2/2} = 2d_2/(x + r_o)$$

Reynolds' number = $R = U (x + r_o)/\nu$
$\nu = 0.164$ cm²/s at about 25° C.

It is interesting to relate the drag also to the volume of the body:

$$V_o = 2\pi r_o^3 \left(\frac{x}{2r_o} + \frac{1}{3}\right) \; ; \quad \text{then}$$

$$c_{V_{ol}} = \frac{D_o}{V_o^{2/3} \rho U^2/2} = 3.69 \, \frac{d_2}{r_o} \left(\frac{x}{2r_o} + \frac{1}{3}\right)^{-2/3}$$

with $3.69 = 4 \, (\pi/4)^{1/3}$.

b) *Square prism*:

Since the cross flow is small —except in the neighbourhood of the head— the pitot head gives the velocity component u in the boundary layer. Perpendicular to a vertical side plane $u(y)$ was measured in various horizontal planes $z = 0$, ± 20 , ± 40 , ± 60 , ± 70 and ± 75 mm; here z denotes the vertical distance from the horizontal plane of symmetry and again y = distance from the wall. For each section the momentum loss thickness was integrated:

$$\delta_2 (x \, , \, z) = \int_0^\infty \frac{u}{U} \left(1 - \frac{u}{U}\right) dy$$

and averaged over the side plane:

$$\overline{\delta}_2 (x) = \frac{1}{a} \int_{-a/2}^{+a/2} \delta_2 (x \, , z) \; dz \, .$$

To complete the total momentum loss area at x, a correction at the edges has to be added which, however, amounts only to 1 to 2 % since the boundary layer in the test range was small compared with the body dimensions. We assume that round the corners of the prism the momentum loss area can be filled up by quadrants with δ_2 as radius. Then the total momentum loss over each side of the prism is

$$a\overline{\overline{\delta}}_2 (x) = a\overline{\delta}_2 + \frac{\pi}{4} \, \delta_2^2 \, .$$

and

$$D_p = \rho U^2 \, 8 r_p \overline{\overline{\delta}}_2 \, ,$$

with

$$r_p = a/2 = 75 \text{ mm.}$$

The surface of the prism and its head is

$$S_p = 8 r_p \, (r_p + x) \; ,$$

and therefore

$$c_F = \frac{D_p}{S_p \, \rho U^2/2} = 2\overline{\overline{\delta}}_2/(r_p + x) \quad , \quad R = U \, (r_p + x)/\nu .$$

The volume is

$$V_p = 8 r_p^3 \left(\frac{x}{2r_p} + \frac{1}{3}\right)$$

and

$$C_{vol} = \frac{D_p}{V_p^{2/3} \rho U^2/2} = 4 \, \frac{\overline{\overline{\delta}}_2}{r_p} \left(\frac{x}{2r_p} + \frac{1}{3}\right)^{-2/3} .$$

Results.

a) *Square prism.*

Near the head the turbulent boundary layer is very thin at the edges and thick in the middle of the side planes (Fig. 1). Further downstream the distribution is smoothed out but a certain regular waviness of δ_2 over z remains up to the last sections. The corresponding curves for the local skin friction

$\tau_0/\overline{\tau_0}$ in Fig. 2 are more straightened out and less spectacular. The local skin friction τ_0 was guessed by the logarithmic part of the velocity profile. Near the head, for low x-values the cross flow in the boundary layer has certainly falsified the values for δ_2 and τ_0, nevertheless, Figs. 1 and 2 will give an idea of the flow in this region too.

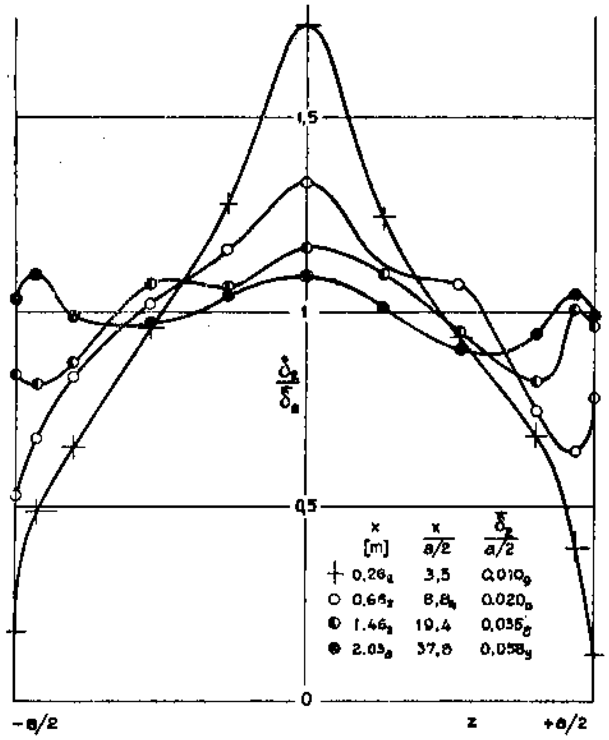

Fig. 1.—Momentum Loss Thickness over the Side Plane of the Square Prism.

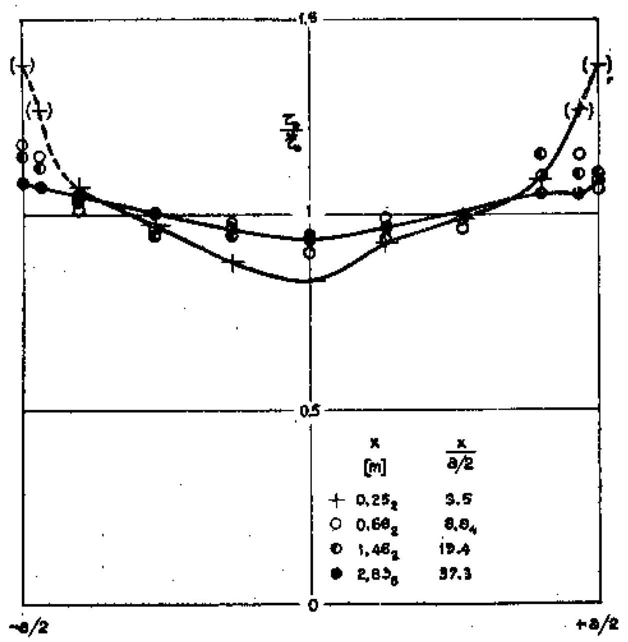

Fig. 2.—Local Skin Friction over the Side Plane of the Square Prism.

In Fig. 3 crosses denote the c_F-values of the prisms with various lengths calculated by the momentum loss. (Because of the pressure field of the head, only values for log $U(x + r)/\nu > 6.2$ have a sound physical meaning.) They fall between the Schoenherr- and the Hughes-Line just as it is usual for modern tests of the flat plate resistance.

b) *Cylinder.*

The C_F-values for cylinders are marked by circles in Fig. 3. All these points are above the Schoenherr-Line and about 8 to 11 % higher than those for the prisms. I. e. at the same diameter of cylinder and prism the transverse curvature of the cylinder increases the frictional drag more than do the edges of the square prism.

A theoretical value for the effect of transverse curvature on frictional resistance has been given by L. Landweber [1]:

$$D_{Cylinder}/D_{Plate} = 1 + 0{,}0124 \frac{x}{r_o} R^{-0.2}$$

In our case; $r_o = 75$ mm , $U = 30$ m/s , $\nu = 0{,}154$ cm²/s , and this gives for $x = 3$ m with $R = 6.0 \times 10^6$ only

$$D_{Cylinder}/D_{Plate} = 1.022 \;.$$

For practical purposes it is useful to refer the drag to the volume V instead of the surface. Fig. 3 shows that $c_{Vol} = D/V^{2/3}\ \tfrac{1}{2}\ \rho U^2$ is the same for cylinder and prisms of equal diameter and length, and its value increases only slightly with the length of the body. Now, by definition

$$c_{F_o}/c_{F_p} = (4/\pi)^{1/3} \frac{(x/r_o + 2/3)^{2/3}}{(x/r_o + 1)} \times$$

$$\times \frac{(x/r_p + 1)}{(x/r_p + 2/3)^{2/3}} \times \frac{c_{Vol\ o}}{c_{Vol\ p}}$$

$$\approx (4/\pi)^{1/3}\ (r_o/r_p)^{1/3} \frac{c_{Vol\ o}}{c_{Vol\ p}} \quad \text{for} \quad x/r \gg 1 .$$

Therefore —in our case with $r_o = r_p$— the experimental fact $c_{Vol\ o} \approx c_{Vol\ p}$ goes together with $c_{Fo}/c_{Fp} \approx (4/\pi)^{1/3} \approx 1.083$.

In general, c_{Fo} or c_{Fp} will be functions not only of R but also of x/r_o or x/r_p characterizing the dependency of the ratio boundary layer thickness to body radius: δ/r. The influence of both of the variables can be found only by series of further tests on bodies of various diameters at different speeds. Still, it does not seem likely that ships with arc form are preferable to usual ones with regard to the frictional drag. Of course, the gap between the Reynolds' num-

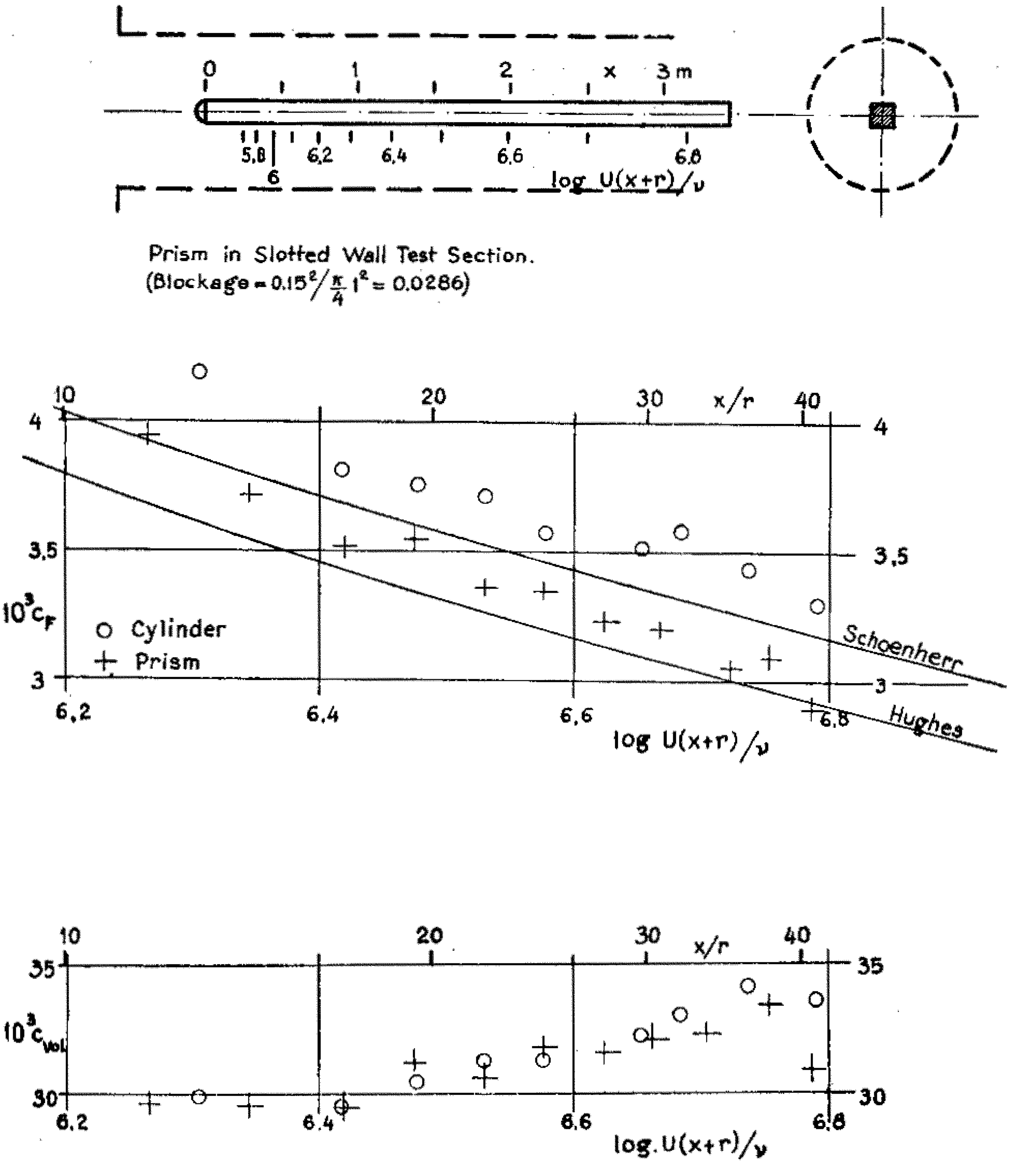

Fig. 3.—Drag Coefficients of Cylinders and Prisms with various Length. (Diameter $2\,r_o = 2\,r_p = 150$ mm. Wind Speed $U = 30$ m/s.).

bers on a ship and in the test range is too high for a definitive answer. On the other hand, the difference in the other variable δ/r is not so large. With $\delta/r \approx 0{,}37\, R^{-1/5}\, x/r$ the value of δ/r on a ship e. g. with $x/r = 12$ and $R = 10^8$ is about 0,1, and in the windtunnel at $x = 1$ m with $r = 7{,}5$ cm, $U = 30$ m/s and $R = 2 \times 10^6$ we have $\delta/r = 0.27$.

Incidentally, there is an analogy with the *laminar* boundary layer; J. C. Cooke [2] finds for laminar flow along general cylinders (without corners) that the drag of an elliptic cylinder is less than that of a circular cylinder of the same perimeter and surface.

Further details of the tests will be given in the diploma paper by W. Kleuters. Unfortunately, we cannot increase the Reynolds number in the present Hamburg windtunnel. On the other hand, G. Kempf (3) had measured the towing resistance of long floating cylinders at high Reynolds numbers; however, because of the unknown form drag of the fore and after body his results and ours are not directly comparable. Therefore, we are expecting with great interest the test on submerged cylinders at TMB which Dr. Tood (4) mentions in his report.

References.

[1] L. Landweber: Effect of Transverse Curvature on Frictional Resistance TMB, Report 689 (1949).
[2] J. C. Cooke: The Flow of Fluid along Cylinders. Quart. Journ. of Mech and Appl. Math. X (1957), p. 312.
[3] G. Kempf: Flächenwiderstand. Werft Reederei Hafen 5 (1924), p. 521.
[4] F. H. Todd: Report of I. T. T. C. Skin Friction Committee, 1957, Appendix I.

Prof. Dr. K. Wieghardt (Oral remarks, translated from German).

Since I am not a practitioner, it is not my intention to show any preference on the two proposals of the Committee. My intention is to present a remark. Capt. Acevedo quotes, among others, a Nikuradse line. This line has been worked by Nikuradse and published in a not easily obtainable paper during the war.

Too much confidence should not be put on this line for high R_n, since it is based on similarity considerations for speed profiles, which do not coincide with other results.

Dr. S. L. Smith (Written contribution).

(i) *Skin Friction and Turbulence Stimulation.*

In the first place, I should like to compliment Dr. Todd and his Committee on a commendably concise and lucid report. Their task was to consider all relevant data over the past three years and to make a definite recommendation for a friction formulation based on Reynolds Number, adequate for practical ship design purposes. This, I think, they have very largely accomplished.

The first specification for such a line was that it must produce on the average better correlation among geosin models of a variety of forms than does the Schoenherr Line. This was in accordance with the wishes of the last Conference and seems eminently reasonable to me because the ability to be able to predict the resistance of one size of model from that of another would appear to be an essential prerequisite to undertaking the more onerous and ambitious task of predicting the resistance of a full size ship.

Like the Committee, we at B.S.R.A. are fully aware of the difficulties of determining reliable extrapolator slopes from geosim data. You can have models too small and you can have them too big also and with the bigger ones tank boundary interference is the bugbear.

As regards the small models, the Committee report refers to the difficulties of turbulence stimulation and I have no doubt this is a very important consideration. I should like to raise another point, however, not mentioned in the report, namely, the effects of possible separation in the afterbody, especially on the smaller models. This is known to be a viscous effect controlled by Reynolds Number, the tendency to separate being greater with small Reynolds Number, that is, the smaller the model. If separation is present, the resistance of the smaller models might well be exaggerated on that account.

The second specification for the Conference Line was that it must produce lower smooth ship predictions in order to avoid the so-called "negative roughness allowances" sometimes found in modern-all-welded hulls in conjunction with the Schoenherr Line. Y think a word of warning is required here, for a possible explanation of these apparently anomalous allowances may well be due to scale effect on propulsion factors which are generally ignored in ship-model correlation work. In the "Lucy Ashton" trials where the fullscale resistance was correctly measured—perhaps for the first time—no negative roughness allowances were found by any method of extrapolation, even for the smoothest hull surface tested, that is, when the seams were faired off and the hull was coated with the smoother aluminium paint.

In view of these unresolved difficulties I am inclined to agree with the Committee that any Conference Line that can be proposed now would be essentially an arbitrary or compromise line. We feel, however, that a change now to a Reynolds Number based formulation on the lines suggested in the report would go a very long way indeed to overcoming the disadvantages inherent in the empirical methods at present in use.

To como now to the recommendations themselves:

(i) We agree that the evidence suggest that a somewhat steeper line than the Schoenherr Line is required and more especially at Reynolds Numbers below 10^7.

(ii) Of the two suggestions made in Recommendation 3, like other British delegates, we are inclined to favour the first, that is, where the Schoenherr Line is retained above $R_n = 10^7$ and progressively steepened below this such that at 10^6 the ordinate is increased by about 0.0004. The reasons for this I shall mention shortly.

Either of the two suggestions made would give roughly, the same smooth ship prediction as some other modern formulations suggested in recent years, the slope over the relevant range being of much the same order. The fact that there are differences in the magnitude of the ordinate between some of them is not really important as far as skin friction correction is concerned.

The reason why we would prefer the first suggestion made in Recommendation 3 is that retaining the

Schoenherr Line above 10^7 would mean that extrapolations already made by the Schoenherr Line for models tests carried out in the neighbourhood of $R_n = 10^7$ (and there are quite a number) would be largely unaffected.

Although this may not be a perfect and complete solution to the problem, we feel that a change now to such a Reynolds Number based line would lead to much more realistic smooth ship predictions than the present empirical methods. In time this should lead to much more realistic allowances for hull surface roughness and other factors which affect the correlation of ship and model results. When we have such a breakdown of the ship-model correlation into its real rather than empirical components we should then be in a much better position to know where to look for improvement. We should also overcome the present rather embarassing situation where the magnitude of the ship prediction varies according to the scale of the model.

Eventually, of course, we shall have to come to grip with the magnitude of the skin-friction formulation as well as its slope if we are to understand completely the physics of ship resistance. The clarification of the rather conflicting evidence on this is, I feel, a matter to which attention should now be directed.

Finally, there is also the question of taking into account in the extrapolation problem the form or three dimensional effect. As far as this is concerned, I agree with the statement made in the report that the time is perhaps not yet ripe for a definite decision on this. Clearly more work is required. Arising from this, if we can come to an agreement on the compromise line referred to earlier I think we ought to make it quite clear that it should be used in the usual manner without any attempt to take variable form extrapolation into account. I would also agree with Dr. Todd's suggestion that the agreed formulation should be known as the "I.T.T.C. 1957 Ship Model Correlation Line" os perphaps more concisely—the "I.T.T.C. 1957 Line".

Dr. E. Castagneto (Written contribution).

I wish to express my comments on the subject from the point of view of customers of ship model basins, especially those of the Rome Tank.

To tell the truth, in our first decisions (see Paris Conference 1933) we took care to state that they only concerned the method of execution and publication for all scientific experimental work on models, but, it is clear that to day, an agreement on a new friction formulation must have a practical aim, adequate, as said under point three of VIIth I.T.T.C., for practical design purposes.

* * *

The testing and correlating method the Rome Tank has been using since its beginning, is as follows:

a) Ship models in painted wood, as big as possible: normally six meters long;

b) Coefficients and Froude method of calculation of ship resistance;

c) Appendage-resistance included in total model resistance, according to a single law of correlation;

d) Propulsion tests at ship - propulsion point;

e) No extra allowance introduced by the tank for predicting the ship trial performances.

Any extra correlating factor on model tests needs, of course, ascertainments and judgments on building systems, structural finish of the ship and propeller, good shape of superstructures, quality of paint and its application; all of which are beyond the basin's authority, and involve economical conflicting interests from which the ship-model basins, must be completely removed.

As stated in the Rome Basin constitution "its activity is based on model tests, and correlation to full-scale ship, is a scientific one: therefore it cannot hold the Institute to any responsibility or guarantee".

Of course that doesn't mean that our Tank disregards all these matters, but it defines precisely the field of the different liabilities.

During a period of about 30 years, we have tested some 850 models and collected tank-records. At the same time, shipbuilders were busy collecting sea-records. Now, thanks to the results obtained from ship-models, shipyards know what to do about any new ship.

Consequentely our customers, althought they didn't claim openly to be against any change at all, obvioulsy are not too enthusiastic about it.

In this connection an improvement, turbulence stimulation for instance, is not by itself an immediate gain, from the point of view of shipbuilders, because a new correlating factor between model and ship has to be checked.

And justly, Italian customers, for a long time, have asked me to supply them the results with and without turbulence stimulating devices, before accepting the new testing procedure unconditionally.

There is no need to say that so far, all ship basins, except the American ones, use about the same correlating method as Rome Tank.

Hence the explanation for the attitude of Italian shipbuilders.

Commercial relations of Italian shipbuilders with U.S. shipowners were few untill now, and it isn't without meaning the fact that, in those few cases, the above mentioned owners didn't request the use of the 1947 A.T.T.C. line, but instead requested skin friction Froude coefficients for models, and Tideman ones, for ships; that is to say, the method employed at Washington Tank before 1947.

Of course even ten years is a short period for this matter.

* * *

According to shipbuilders, changes might be justified by a better obtainment of these objectives:

A) Inmediate and safe comparison of effective and shaft horse powers deduced from two different models tested in the same tank;

B) Immediate and safe comparison between testing results obtained from different tanks or publications;

C) A more safe prediction of ship performances according to ones own and other experiences.

Besides differences in measuring systems, these goals need:

1.° Agreement between measurements of resistance, torque, and revolutions on the same model at different time or places;

2.° The same testing and correlating method;

3.° The same flexible allowances for hull, wind, sea, etc. conditions.

1.° Point one is of course the most important for tank technicians, because it is under their own exclusive.

I concur with Dr. Kempf's introductory remarks on techniques of measuring model resistance and proposals. I'd like to add that Rome Tank has already started on it: since last summer, once a month, we have been repeating towing tests with the same model.

I would suggest choosing, as far as possible, the same drawing in all the tanks: in the study of a new question, it is wise to lower the number of unknown quantities.

I find the comparison of results obtained in different basins always most instructive.

As far as time intervals between one test and another I would suggest the following: one day, two days, one week, two weeks, one month, one month, one month, etc.

According to my experience, changes in resistance, if changes there are, are more frequent in the beginning with a newly built model. It is a question of time, as we said in previous meetings, but without reaching a very definite conclusion.

We must also consider it is time consuming to test two models every fortnight, for three years. Basins are all very busy.

2.° Point two needs agreement on: a) minimum turbulent skin friction line; b) ΔC_f allowances; c) method of correlating appendage-resistance; d) self propulsion tests at model or at ship-propulsion point.

The Skin Friction Committee confines its proposals to minimum skin friction line. Now: what shall we do about ΔC_f? and what skin friction correction shall we choose in propulsion tests? Shall we use a single ΔC_f value also with very small scale ratios as 1:2, or will each tank act according to its own judgment for each experiment, each model? Or shall we choose such ΔC_f to reach about the same results as with the Froude coefficients?

In my opinion an agreement on ΔC_f can't be avoided, even if $\Delta C_f = 0$; otherwise we shall reach the opposite aim we are looking for. We must also take into account the remarks already pointed out concerning basins and shipbuilders' responsibilities.

As to appendage-resistance Iwould prefer, for practical reasons, a single extrapolator: not ever can appendage-resistance be measured with sufficient certainty (sometimes it is a negative one) and not every model is tested in bare condition and with appendages.

3.° If we now choose a new skin friction formulation, Italian customers will ask me to supply them, for a time (two or three years) with both old and new style predictions to gain experience. Certainly they would like to be helped with it. If the new line approaches the 1947 A.T.T.C. line, some more notices on "correlating factors" used by Washington Tank, as said on page 98 of the formal discussion by Dr. Todd, would be greatly appreciated, at least as far as merchant ships are concerned.

Dr. E. Castagneto (Oral remarks).

Regarding, the proposals of the Friction Committee, I agree with the first one, but in our opinion the agreement on point 2 is necessary for the practical designer purposes as stated in the conclusions of the previous Committee. Otherwise there will be no reason to change from the point of view of cooperation and we shall have an opposite aim than the one we are looking for.

Dr. H. Edstrand (Oral remarks introducing R. L. Gamlin's contribution).

In recent years the knowledge and literature concerning boundary layer theory has grown considerably and at the Swedish State Shibuilding Experimental Tank (SSPA) it was felt that it would be an advantage if an overall picture was obtained of the work which had been done and the method used in the investigations in this field. (In agreement with item 1 of the Recommendations of the Scandinavian Conference). To this end, Mr. Gamlin was instructed to examine all the pertinent literature, outline the problems involved and make suggestions for tests to be carried out at SSPA.

As stated in the preface, it was originally intended to be an internal report for the staff of SSPA but later, as it was felt the material was of considerable general interest, it was decided to publish it as a contribution to this Conference.

I ought to mention that Mr. Gamlin's work in its preliminary form was completed more than a year ago and that Mr. Gamlin, of course, could not have included many important papers published since concerning this matter.

Mr. R. L. Gamlin (*) (Written contribution).

1. 0. *Introduction.*

Much effort has been expended in recent years in an attempt to determine the correct relationship between the skin friction coefficient and Reynolds number corresponding to two dimensional flow over a flat plate at zero incidence with an all turbulent boundary layer. From the naval architects point of view it is necessary to know this relationship accurately since it has to be extrapolated from Reynolds numbers of the order 10^7, corresponding to ship model tests, to those of order 10^9 or 10^{10}, for its use as an invariable basis in the estimation of the frictional resistance of full-ships.

One of the attempts to arrive at the required "friction line" by an analysis of "flat-plate" data was made by K. E. Schoenherr [1] (**), who related an analysis of previous friction test results and the results of plate tests he carried out himself, to a theory by Th. von Kármán. This work was published in 1932 and was seriously considered by naval architects for the purpose mentioned above. Subsequently other attempts were made by various people, but the difference between their results and Schoenherr's was not thought sufficient to justify the adoption of another relationship. Then in 1952, G. Hughes [2], published some preliminary results of an extensive series of very carefully conducted plane-surface friction tests, followed in 1954 by the publication of a paper [3], which presented the final results of the whole investigation and concluded a comprehensive piece of work spread over a period of six years. On the basis of these results, Hughes proposed a two-dimensional line which did differ appreciably from Schoenherr's.

Undoubtedly many of the discrepancies can be referred to the methods used to obtain these lines and the sources of error involved. It therefore became obvious that a considerable amount of further work had to be carried out to clarify the situation.

This paper presents an examination of problems involved in any investigation to determine the two-dimensional turbulent friction line.

The report outlines, to begin with the problems associated with the determination of the two-dimensional turbulent skin frictional relationship with Reynolds number, for a smooth plane surface in zero pressure gradient. It then goes on to present the available up-to-date data on the mechanics of boundary layer transition and stimulation. Finally, certain tests are suggested, based on the experience and results of previous investigators, which should assist in the determination of the true two-dimensional turbulent skin friction line.

(*) From the Swedish State Shipbuilding Tank, not in attendance.

(**) Numbers in brackets refer to the list of references.

2. 0. *General.*

The problems involved in any investigation to determine the two-dimensional turbulent friction line are mainly three.

Two concern the elimination of form (or thickness) effects and edge (or three-dimensional) influence, and the third is to obtain consistent control and knowledge of the position at which the boundary layer flow changes from a laminar to a turbulent nature.

2. 1. *Form Effects.*

These are the effects that arise due to the thickness of the plate and, in general, are two-fold. They both give increments to the resistance, one being due to the resultant force in the longitudinal direction resulting from the pressure distribution, and the other being due to the waves caused by the proximity of a free fluid surface. The latter increment can sometimes be eliminated by testing the plate in a horizontal submerged condition, providing that the depth of submersion is sufficient. However, this method has several disadvantages compared with that of testing in a vertical position with only one longitudinal edge submerged. These are:

(a) an increased edge effect (see section 2. 2.)

(b) to obtain the effect of increasing plate breadth, new models must be made. In the case of the vertical plate, only an increase in depth of immersion is necessary

(c) the model supports must enter the water thus causing an additional resistance which must be accurately allowed for.

These may also have a slight wave making resistance. Such is the accuracy required of the results in this work, that any detail in any way affecting resistance must be accurately allowed for. Generally this necessitates several corrections, so that it is essential to adopt a testing technique which is as simple as possible, to minimise the number of possible sources of error.

2. 2. *Edge Effect.*

This is due to the three-dimensional character of the plate, and the precise nature of the longitudinal edge flow is not yet fully understood. It is caused, however, by the interaction of the free stream flow with the suddenly terminated boundary layer of the plate surfaces. The velocity of the latter, going from approximately free stream velocity at its outer extent, to zero on the plate surface, suddenly meeting the free stream velocity along the longitudinal edge of the plate, causes an influence region *around* this edge. It will have a similar velocity distribution to a normal boundary layer inasmuch as the velocity must go from zero at the edge of the plate to free stream velocity at some distance away from the edge. This influence region must also extend inboard of the edge as the original boundary layer velocity distribution will be modified somewhat. Whether this edge effect is sensibly confined only to a local region adjacent to the edge or whether it is felt over the whole plate has not yet been conclusively determined, although the small amount of experimental data available shows that the edge influence extends at least over a considerable portion of the plate, if not as is indicated in one case (see [4] Fig. 5) over the whole plate.

2. 3. *Transition Point.*

The relationship between the friction coefficient and Reynolds nomber is required for turbulent flow. This strictly requires turbulent flow to start instantaneously from the leading edge of the plate, and is generally thought to be impossible to obtain in practice. Normally, boundary layer flow starts in a laminar condition, where the skin friction coefficient is a different function of R_n than for turbulent flow, and passes through a transition stage of partially turbulent flow before becoming fully turbulent. Thus the only way to obtain friction coefficient values that can be said to apply for "all turbulent flow", is to cause transition to occur so close to the leading edge that, strictly speaking, the ratio of the friction resistance of this leading part of the plate in *turbulent* flow, to the resistance of the whole plate in turbulent flow, is negligible. This, however, presupposes knowledge of the results that are trying to be obtained, so that in practice it must be assumed that the difference between the resistance of the leading edge portion in the actual laminar flow and in the hypothetical turbulent flow, is sufficiently small that the ratio of the resistance of the leading edge in *laminar* flow to that of the whole plate with laminar leading edge, can be used to indicate when the effect of the leading edge flow is negligible.

In practice it has been found that for "normal" flow conditions and in a zero pressure gradient, natural transition is most likely to commence within the range of "local" or "transition Reynolds numbers" from 10^5 to 10^6, based on free stream velocity and distance from model leading edge to position of initiation of transition. These values, however, are tentative as they are very sensitive to free stream turbulence and model surface roughness. For example, the transition *) R_n on a model in a wind tunnel with a very smooth airstream has been as high as 3 x 10^6. The above range of transition R_n refers to a normal polished smooth surface, e. g. as obtained on rolled metal sheet, plastic sheet, varnished wood etc., and a stream turbulence **) of about 0. 2 to 1. 0 per cent, i. e. as found in wind tunnel with an usual amount of turbulence. As the degree of turbulence in ship towing tanks is probably considerably less than this ***), it is to be expected that natural transition would be delayed to rather high transition Reynolds numbers. However, if the model R_n is sufficiently high, ie. 10^7 or higher, the above range of R_{ntr} can normally be used to give an approximate indication of where the onset of natural transition would occur. Accepting that transition starts betwenn $R_{ntr} = 10^5$ and 10^6, the higher the model R_n, the better is the relative definition of the region lying between these values. For example, transition on a flat plate in a stream with the order of turbulence of abut 0. 2 to 1. 0 %, at a Reynolds number of 10^7, would probably occur between 1 % and 10 % of its length from the leading edge. For the same plate at a R_n of 10^8, transition would probably occur between 0. 1 % and 1 %.

To reduce the region of laminar flow to a minimum it is necessary to cause transition prematurely, ie. in front of the position of "natural" transition. For this to happen, the boundary layer must be disturbed or "stimulated" in some manner, upstream of the position where natural transition occurs. Various methods have been tried, the simpler and consequently most usual, being to place a fine wire, a row of studs or a strip of rough surfaced material across the stream, attached to, and in the boundary layer of the model at the position where transition is desired. The employment of this form of stimulation, however, introduces other problems. The most obvious of these is the determination of the stimulator's own resistance so that it can be accurately allowed for when determining the frictional resistance of the plate. A more subtle problem is that concerned with stimulator effectiveness and arises because the ability of a particular type and shape of stimulator to cause turbulent flow from

*) The term "transition" refers to the beginning of transition unless otherwise stated in the text.

**) Expressed as the percentage ratio of the root mean square value of the longitudinal turbulent velocity component, to the free stream velocity.

***) From some preliminary work done at the David Taylor Model Basin [8] the order of turbulence was found to be 0.3 %, one minute after a run with a full form model at a speed of 2 knots, which decreased exponentially to 0.1 % after 13 minutes. This latter figure is comparable with that for a low turbulence wind tunnel.

directly behind itself, would appear to depend on two factors. These are:

(a) Size of stimulator.

(b) Local R_n of stimulator based on distance from leading edge plate.

As the local R_n of the stimulator decreases, so the size of stimulator required to cause turbulent flow from immediately behind it, increases. If the disturbance caused by the stimulator is insufficient for this purpose, fully developed turbulent flow will not be realised until some position further dowstream, the intermediate distance being occupied by transitional or even laminar as well as transitional flow. On the other hand if the required disturbance corresponding to a certain local R_n is considerably exceeded, there are some experimental results which have been thought to indicate the existence of another phenomenon which has been termed "over-stimulation". If this phenomenon does exist, it is characterised by an increase in frictional resistance above that due to normal turbulent flow- if it does not, there remain increments in resistance in certain experimental results that are very difficult to explain. These were found in [3], see p. 317, where over-stimulation was offered as one of the possible causes, and the phenomenon is mentioned again in [9], p. 18, when referring to the results of [10]. It is possible that this increase in resistance is associated with the formation of eddies in the boundary layer behind the stimulator instead of the desired homogenous turbulent flow.

Generally, when friction tests with flat plates are being made, the velocity as well as the length of plate is varied in order that as large a range of plate R_n 's as possible will be covered. Thus, as the usual practice is to attach a stimulator at a fixed position along the plate, the local R_n of the stimulator in such cases will vary during the tests, and the risk of the above troubles is incurred at the lower or (and) higher end/s of the speed range. Other methods of stimulation have been tried with varying degrees of success and these will be referred to, briefly, later on in this report. All of these seem to have their own particular disadvantages and as yet no optimum method of stimulation has been suggested.

From what has been said above regarding the effectiveness of certain types of stimulators, it will be obvious that the main problem is to know how far infront of the point of natural transition a particular type and size of stimulator can be placed, while still obtaining transition directly behind it at all test speeds. For all types of stimulation it is necessary to know this "gain" in transition point. Very little detailed work has been done on this, although the doubts raised about the validity of skin friction results in many instances are associated with the extent to which laminar flow prevailed in the tests, ie. where transition actually occurred.

Another problem, and one of a very fundamental nature, is that of determining whether the type of flow caused by the stimulator is the same as that of natural turbulent flow ie. turbulent flow resulting from natural transition. If not, whether the difference in the nature of the flow has any significant effect on the skin friction resistance. It has been assumed in the past, in the absence of any data on this question, that flow induced by artificial stimulation was the same as natural turbulent flow, and only recently has any detailed information been published on this.

It will be obvious from the above outline of the problems involved in determining the relationship between the two-dimensional turbulent skin friction coefficient and Reynolds number, for zero pressure gradient, that even when two-dimensional conditions are obtained and thickness effects eliminated, the remaining problems concerned with obtaining effectively all turbulent flow over the plate, ie. of obtaining the earliest possible transition and knowing with certainty where it occurs, are still considerable and that the whole problem of artificial stimulation is very involved. Much work remains to be done on stimulation before the process is really understood and before a more scientific treatment can be given to the problem of causing premature transition. This full understanding is partially obstructed by the fact that the complete mechanism of natural transition is still not fully understood, although much attention has been given to it, especially in recent years when considerable progress has been made. Until our knowledge of natural transition is complete, artificial stimulation can never be fully understood. Because of this, natural transition is dealt with in more detail in the next part of this paper where the most recent information on this subject is summarised. This is followed by a section on stimulation.

In case it is thought that the amount of consideration given to transition and stimulation is disproportionate to their importance, it should perhaps be pointed out that the problems associated with these matters are present in all of the several different experimental methods of obtaining turbulent skin friction data, and that if any progress is to be made in obtaining turbulent skin friction values at low R_n, then the employment of some form of stimulation, with its attendant problems, cannot be avoided.

3.0. *Natural Transition.*

As a result of recent experiments, the nature of the transition region between laminar and "fully developed" turbulent flow can be pictured more clearly. Only since the application of "hot-wire" measuring techniques to the investigation of boundary layer flow characteristics has this been possible.

A review of the ideas of how transition is caused, as held up to 1948, is given in [11]. It is summarised below.

Ever since Reynolds discovered the change of character of flow in pipes at some distance downstream from the entry, this distance decreasing as velocity increased, the origin of transition has been thought to be intimately connected with instability of laminar flow. There were two fundamentally different theories put forward to explain the nature of this instability, however. One was that of Tollmien and Schlichting who thought that laminar flow was an unstable medium for very small disturbances present in the flow, caused, for example, by surface roughness. It was assumed that these, when amplified, caused transition. The other was that put forward by G. I. Taylor, who proposed that instability was caused by finite external disturbances in the form of free stream turbulence. He thought that transition was a result of momentary separation of the laminar boundary layer in regions of adverse pressure gradient associated with turbulence in the free stream. Early experimental work confirmed Taylor's theory, no evidence being found of amplified disturbances leading to transition, which occurred suddenly and intermittenly. Later work, however, showed that for a certain flow condition the Tollmien-Schlichting theory was true. This latter discovery was made by Schubauer and Skramstad in 1940, in a wind tunnel with an air flow of very lowturbulence level, and an account of their work is given in [12]. Thus both theories were found to be true and there are, in fact, two ways in which transition can be initiated. The governing factor is the amount of free stream turbulence. For a flat plate, if this is less than about 0.1 %, the Tollmien-Schlichting theory applies and transition is immediately preceded by large disturbances in the laminar boundary layer which are present as a result of the amplification of very small disturbances. When the turbulence is greater than about 0.1 %, as, for example, in ordinary low-speed wind tunnels, transition occurs more in accordance with the Taylor theory *). This fact explained why the small disturbance mechanism of transition had not been obtained earlier. In all previous cases the free stream turbulence had been too great.

From the very small amount of data available, as has been stated earlier (see section 2.3) the order of turbulence in a towing tank can quite normally be as low as 0.1 %, so that it is to be expected that natural transition on models in a tank can happen in either of the above ways. However, bearing in mind that a negative pressure gradient, as present on a ship's bows, retards the position of transition, it is possible that in the case of a ship model the Taylor mechanism would apply down to a lower value of free stream turbulence than for a flat plate.

3.1. *Verification of the Tollmien-Schlichting Theory.*

Schubauer and Skramstad [12] discovered, while investigating transition on a flat plate in a very low turbulence air-stream with the use of hot-wire anemometers, that when the amount of free stream turbulence was low, small disturbances occurring in the laminar boundary layer were amplified until they were sufficiently large to cause turbulence. In these experiments the output of the hot-wire was led to a cathode-ray oscillograph, the trace being recorded photographically with a moving-film camera. The vertical scale on the oscillograph represents the longitudinal velocity fluctuation component of the disturbances. Figs. 1 and 2, taken from [12], show typical records of the development of laminar boundary layer disturbances. Fig. 1 shows how the small initial fluctuations gradually develop into large irregular ones which are followed by "bursts" of very high frequency, these being the trace-characteristic of turbulence. It will be seen that the change from large laminar flow fluctuations to turbulence occurs instantaneously. These bursts of turbulence occur very rarely at the beginning of the transition region, but become more and more frequent with distance into the region, until, at the end of it, only the very high frequency disturbances are present. This is then the fully developed turbulent flow show in the last record of Fig. 1. Fig 2 show how the presence of a negative pressure gradient delays transition by delaying the development of the disturbances. Reasons why the amplification of a disturbance of rather pure frequency, as seen in Figs. 1 and 2, should result from random initial disturbances, are given in [12].

Since it is of relevance to the mechanism of transition caused by the amplification of small disturbances, another interesting outcome of the investigations reported in [12] can be briefly mentioned. This is illustrated by Fig. 3, in which the frequency of disturbance is plotted against Reynolds number based on boundary layer displacement thickness, ie. $R = U\,\delta^*/\nu$, where δ^* is the boundary layer displacement thickness.

Laminar boundary layer stability theory indicates that for two-dimensional disturbances, only those within a certain frequency range are amplified, those outside this range being damped. These frequency boundaries according to Lin and Schlicht-

*) There is still some doubt that local separation of the laminar boundary layer takes place in all cases where transition is attributed to external disturbances in the form of free stream turbulence, see [13].

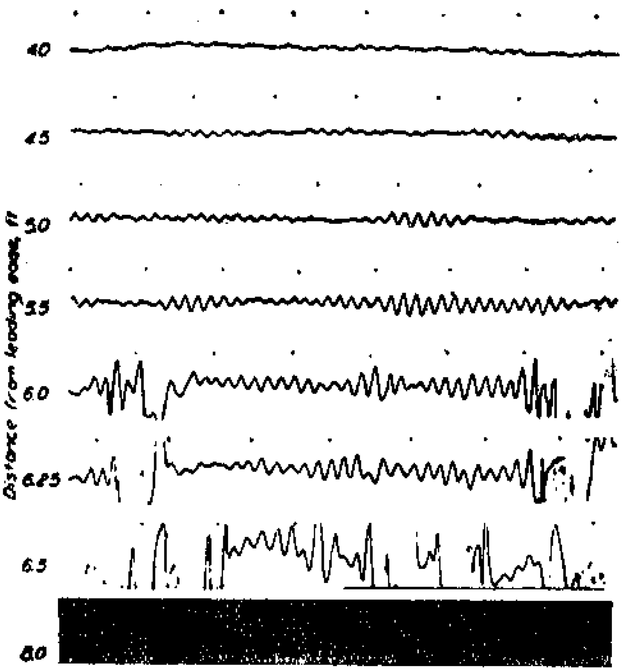

Fig. 1 (from ref. 12).
Oscillograms of u-fluctuations showing laminar boundary-layer oscillations in boundary layer of flat plate. Distance from surface, 0.023 inch; $U_0 = 80$ ft/sec; time interval between dots, 1/30 sec.

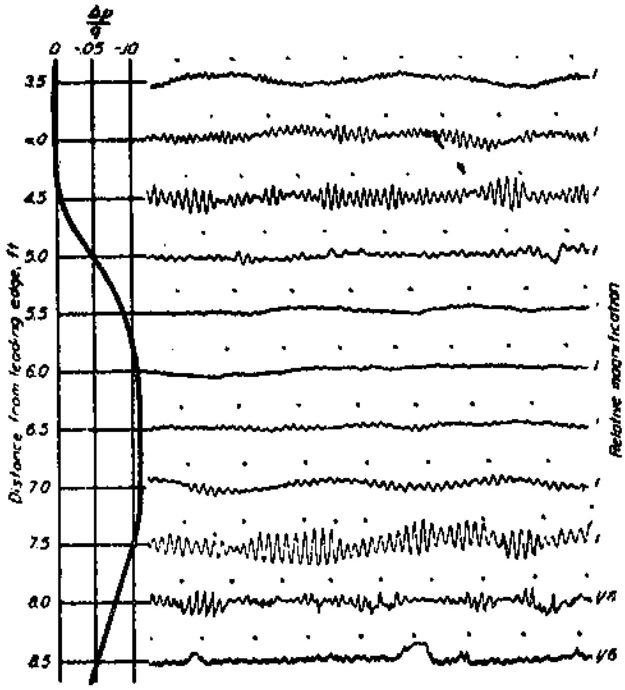

Fig. 2 (from ref. 12).
Oscillograms of u-fluctuations showing effect of pressure gradient on boundary-layer oscillations. Distance from surface, 0.021 inch; $U_0 = 95$ ft/sec; time interval between dots, 1/30 sec.

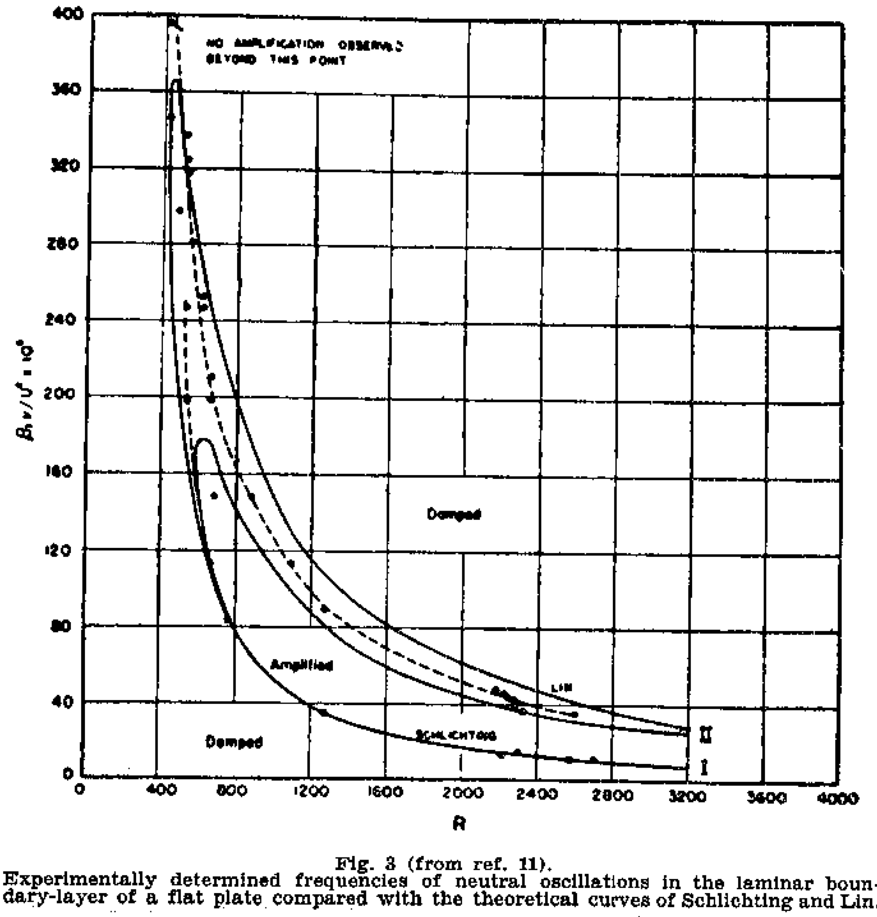

Fig. 3 (from ref. 11).
Experimentally determined frequencies of neutral oscillations in the laminar boundary-layer of a flat plate compared with the theoretical curves of Schlichting and Lin.

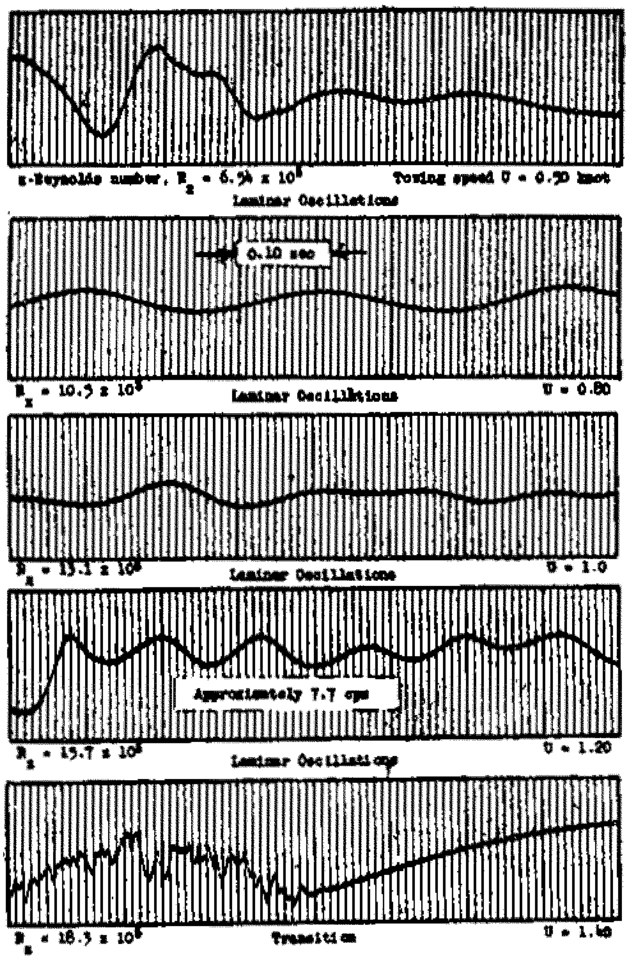

Fig. 4 (from ref. 14).
Oscillograms of Oscillations in the Laminar Boundary Layer at 7.75 ft from bow and 1 ft above the keel of tanker model 4083.

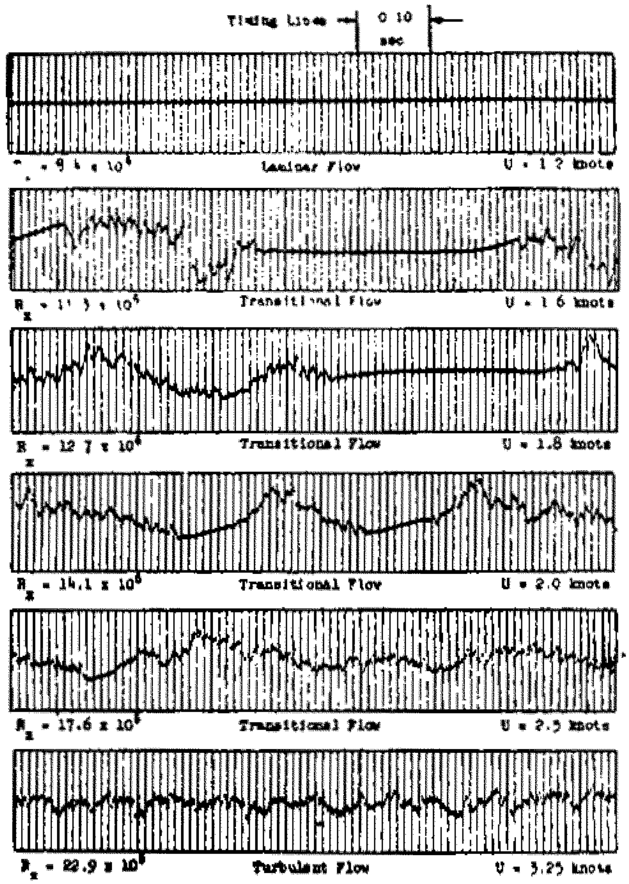

Fig. 5 (from ref. 14).
Characteristic Oscillograms of the development of the turbulent boundary layer at hot wire 17 without artificial turbulence stimulation.

ing are as shown in Fig. 3. Schubauer and Skramstad introduced artificial two-dimensional disturbances of known frequency into the laminar boundary layer, and investigated whether these were subsequently amplified or damped. The results are shown plotted in Fig. 3, where it is seen that a very good agreement between experiment and theory was obtained. Since the Reynolds number increases with distance along a plate, it will be seen that disturbances with certain frequencies can be amplified as they pass down-stream for part of their journey, but then they become damped. This is discussed in detail in [12]. The case of the three-dimensional disturbance has still to be treated.

A similar sequence of events leading to natural transition and eventually fully developed turbulent flow, as found by Schubauer and Skramstad [12], on a flat plate in air with a very low turbulence level, has been found in a towing tank on a ship hull by Breslin and Macovsky [14], while using hot-wire equipment in stimulator investigations. This is shown in their records of oscillograph traces which are presented as Figs. 4 and 5 of this report. Fig. 4 shows the laminar disturbances and the occurrence of turbulence characterised by the very high frequency oscillation in the last record. Fig. 5 shows how the bursts of turbulence grow in frequency of occurrence with increase in local Reynolds number, the last trace showing fully developed turbulent how the bursts of turbulence grow in frequency of obtaining the sequence of events leading to fully turbulent flow, as used in [12] and [14], was that in the former, measurements were made for a constant velocity at various stations along the plate, while in the latter, recordings were made at one position for various speeds. However, since natural transition characteristics are a function of local Reynolds number, if, in two different tests with the same order of stream turbulence, a variatión in R_{nl} is obtained by changing the characteristic length only in one case, and velocity only in the other, as was done in the above instances, the qualitative results in both cases should be the same, as in fact they were. The oscillograph traces of Figs. 1, 2, 4 and 5 allow certain conclusions to be drawn with regard to the way in which transition occurrs, but they do not, in themselves, permit a unique physical picture to be drawn of what happens in the transition region. The fact that the change from highly disturbed laminar flow to turbulent flow occurrs in an instant of time is obvious from the traces. It is equally obvious that the bursts of turbulence occur more and more frequently with depth of penetration into the transition region, but the traces can have more than one interpretation with regard to what happens in this region. The concept that has been most commonly held is that transition occurred abruptly along an irregular line which separated the laminar flow in front of it from the turbulent flow behind and that this line surged upstream and downstream, defining the transition region by the limits of its movement. This concept is in agreement with the evidence of the traces if the transition line surges up and down the transition region entering and leaving different parts of the region with a frequency corresponding to the frequency of occurrance of the turbulent bursts in the various parts. Another possibility, which would still be in accordance with the oscillograph records, is that local breakdown of the laminar flow occurred at various points in the transition region where the laminar disturbances were sufficiently large, producing spots of turbulence which spread as they passed downstream. As the likelihood of laminar flow breakdown increases with penetration of the transition region, due to the increasing size of the disturbances and because of the spreading of the spots, it would be expected that the proportion of turbulent flow increases with penetration of the transition region. This would account for the increasing number and length of turbulent bursts as shown by Fig. 1. It could be determined whether or not the former of the above two concepts is the true one by making two simultaneous hot-wire recordings in the transition region, the second one being taken at a position some distance away from, and exactly behind, the first one. If there was ever any evidence of laminar flow at the downstream position when there was turbulence at the upstream one then the first of the concepts cannot be true. Unfortunately no such records seem to have been made until very recently, when Schubauer and Klebanoff [13] made a thorough investigation to determine the physical nature of the transition region. Their results showed that the first of the above concepts of the transition region was incorrect and as a result of their finding put forward the second one which they state to be the correct one. This very recent work will be discussed in more detail later on, as it also includes some information on transition with high free stream turbulence and transition behind stimulators.

3.2. *On the Taylor Concept of Transition.*

It would appear that there remains considerable work to be done before an understanding is obtained of the mechanics of transition in the presence of relatively high free stream turbulence ($> \sim 0.2$ %). No correspondingly thorough work to that for the verification of the Tollmien-Schlichting theory, referred to in the preceding section, seems to have been carried out. There is no unanimous opinion whether or not transition to turbulent flow is necessarily preceded by local separation of the laminar boundary layer or even if transition occurs in this way at all. Dryden [11], states, apparently on the basis of the work of Schubauer and Skramstad [12], that when the stream turbulence is greater than about 0.2 %, the turbulence is the controlling factor of transition and that the mechanism is that of the Taylor theory, which assumes momentary local separation. The only evidence of [12] which could support the idea of local separation causing turbulence, however, was that turbulence, in some instances, generally appeared in the low velocity part of the disturbance oscillograph traces, but even this feature had another possible explanation, see p. 29 [12]. In another instance a slight flow reversal at the surface was indicated but there was no evidence of transition having occurred. Thus it would appear that the mere presence of local separation is not a sufficient condition for transition. The Taylor theory does not claim this, however, and it has not been proved that if free stream turbulence is sufficiently large the pressure gradients associated with the turbulence cannot induce transition as a result of causing local separation. In the apparent absence of more positive information on this question, it is felt that whether or not transition can occur in strict accordance with Taylor's concept has still to be conclusively determined, as have also the conditions under which it applies, if and when it does, and the alternative mechanism if and when it does not.

Irrespective of the way in which transition occurrs, it has been shown by Schubauer and Skramstad [12] that for values of free stream turbulence greater than about 0.1 %, an increase in stream turbulence reduces the local Reynolds number at which transition commences. The precise variation and the effect of turbulence on transition region length for a flat plate in a wind tunnel, as found by the above authors, is shown in Fig. 6, which is taken from their paper. Il will be noticed that the length of the transition region was independent of the amount of free stream turbulence below a turbulence value of 0.1 % and between 0.16 % and the limit investigated, 0.34 %. It was thought in [12], that the independence of the commencement of the point of transition on local Reynolds number, with reduction in turbulence be-

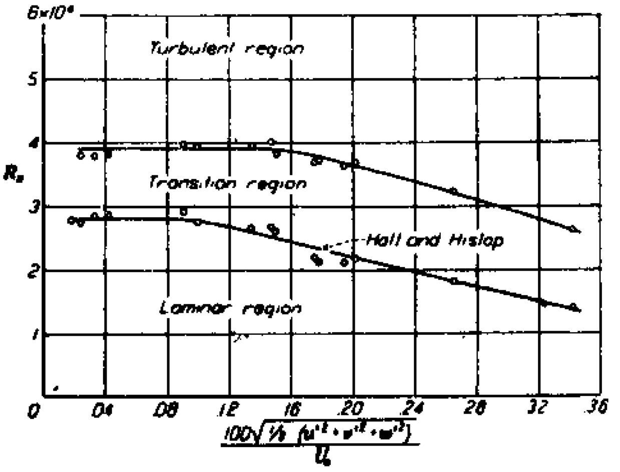

Fig. 6 (from ref. 12).

Effect of turbulence on 2-Reynolds number of transition. Flat plate; zero pressure gradient.

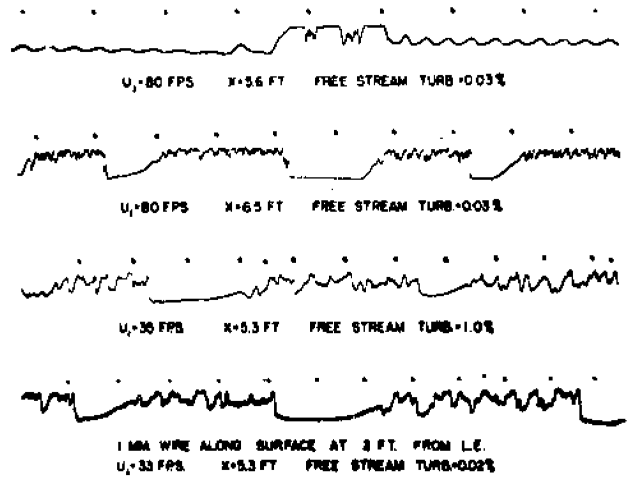

Fig. 8 (from ref. 13).

Velocity fluctuations in transition region. Time interval between dots 1/60 sec. Oscillograms of u-fluctuation in transition regions with hot-wire probe 0.013 inch from surface.

low a value of about 0.80 %, could be attributed to the effect of tunnel operating noise, as an analysis of the noise spectrum in the tunnel when the total turbulence in the free stream was 0.03 %, showed that the apparent turbulence due to the noise could account for nearly all the hot-wire output. A more detailed explanation is given on p. 30 of [12]. The authors infer that had the tests been carried out under quiet conditions it could be expected that as zero turbulence was neared, the curves of Fig. 6 would have increased their slope with decrease in turbulence, instead of levelling off as they actually do, since, in the limiting case when no disturbances are present, transition should theoretically not occur, and so as turbulence $\rightarrow 0$, both curves in Fig. 6 should approach a Reynolds number of infinity. It might be possible to verify this by performing similar tests in a ship towing tank, where for the same Reynolds numbers as obtained in the wind tunnel, the noise would be much less.

One of the curves just discussed above, that showing the variation of transition initiation-point Reynolds number with free stream turbulence, has been extended by Macovsky and Breslin [15] with other experimental results. All the results define a unique line, as shown in Fig. 7, taken from [15]. From the slope of this line at the lower Reynolds numbers, it would seem that transition can commence at Reynolds numbers lower than that found to be the limiting Reynolds number in accordance with laminar boundary layer stability theory. The curve does not indicate any limit below which transition cannot be caused, providing sufficient turbulence is present in the free stream, but since the last experimental point was at $R_{ntr} = 10^5$, this indication must await the confirmation of further experimental results before it can be relied on as being fact.

3. 3 *Results of Recent Work.*

The foregoing outlines the extent of knowledge of the mechanics of transition up to the year 1950.

In 1955, Schubauer and Klebanoff [13], published their results of a careful investigation to obtain a better insight into the process of transition and to ascertain the physical nature of the transition region. The flow over a flat plate was investigated in a very low turbulence level tunnel ($\sim$ 0. 015 % at 30 ft./sec., $\sim$ 0. 040 % at 110 ft./sec.) using hot-wire measuring techniques, the plate and tunnel being the same as those used in [12]. Velocity fluctuations in the boundary layer were recorded as an oscillogram trace in a similar way to that of [12]. A summary of the relevant results is given below.

Similar records to those obtained in the transition region in other, earlier, investigations with low free stream turbulence were obtained by Schubauer and Klebanoff. Typical examples are shown in the first two traces of Fig. 8 *), which is reproduced from their paper. The first trace, from near the start of transition, show the oscillations in the laminar boundary layer followed by a turbulent burst, which is followed in turn by laminar flow again. The second trace, from further into the transition region, shows the increased occurrence of the number of turbulent bursts. The third record shows the resulting trace when the free stream turbulence was in-

*) The sequence of events in all the traces reproduced from [13] is from right to left unless otherwise stated. It is of importance to note this, especially in Fig. 12.

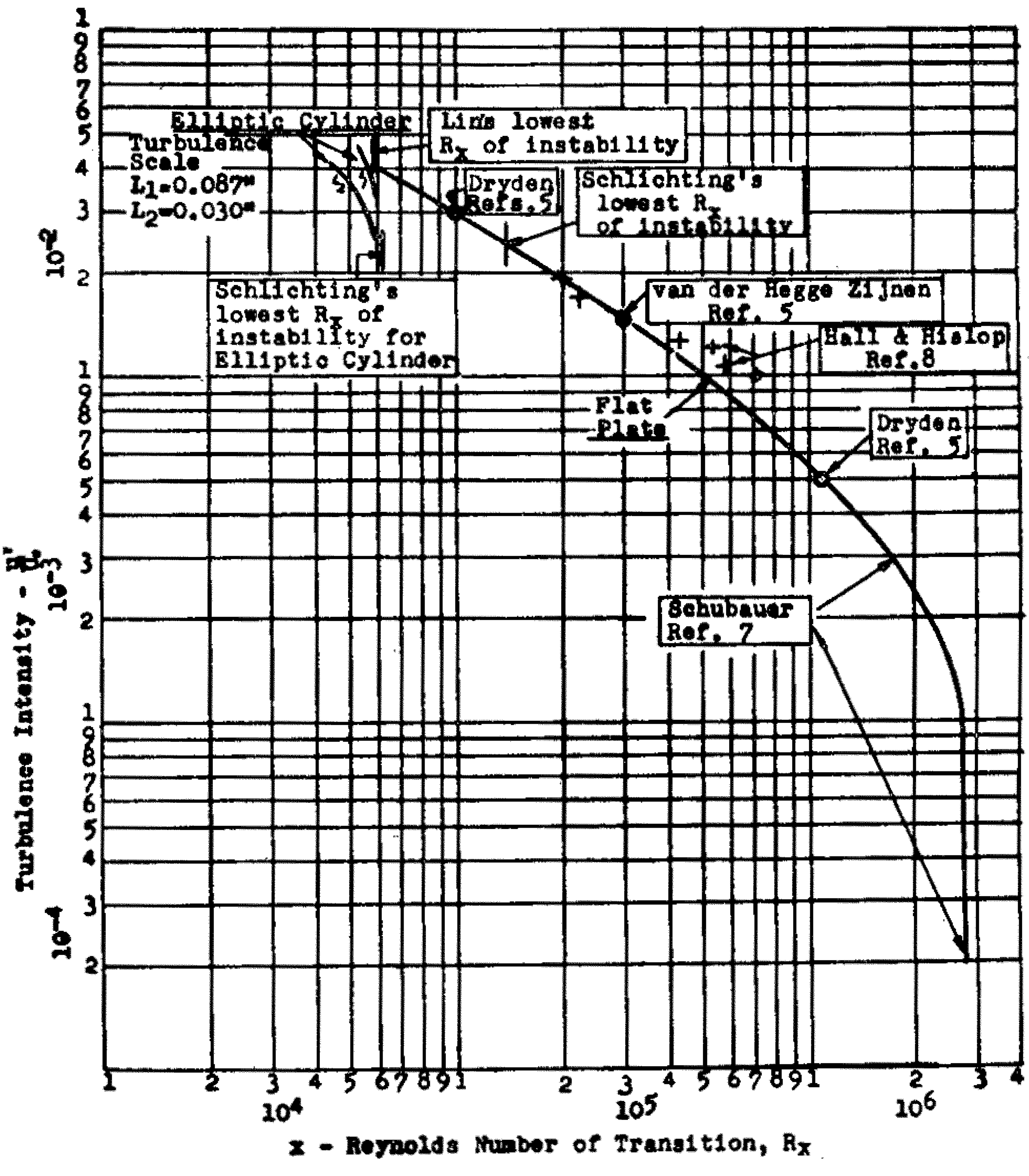

Fig. 7 (from ref. 15).

Experimental data showing effect of free stream turbulence on position of transition for a flat plate and elliptical cylinder.
Data for elliptic cylinders taken at length Reynolds Number $R = 1.18 \times 10^5$.

creased to 1.0 %, when, according to Dryden [11], transition should occur in accordance with Taylors's ideas. The trace has similar general characteristics to that obtained when free stream turbulence was low, the only difference being the nature of the trace of the turbulent bursts. With low turbulence in the stream, turbulence in the boundary layer was characterised by a small amplitude, high-frequency record. With high stream turbulence, the turbulent part of the trace has a large irregular amplitude and a lower frequency. In both cases the turbulent bursts were interspaced by laminar flow.

Whether the free stream turbulence was high or low, in all the cases investigated the flow was laminar for some distance from the leading edge but contained low-frequency velocity fluctuations generally consisting of regular amplified disturbances showing on the trace as a sinusoidal wave form when

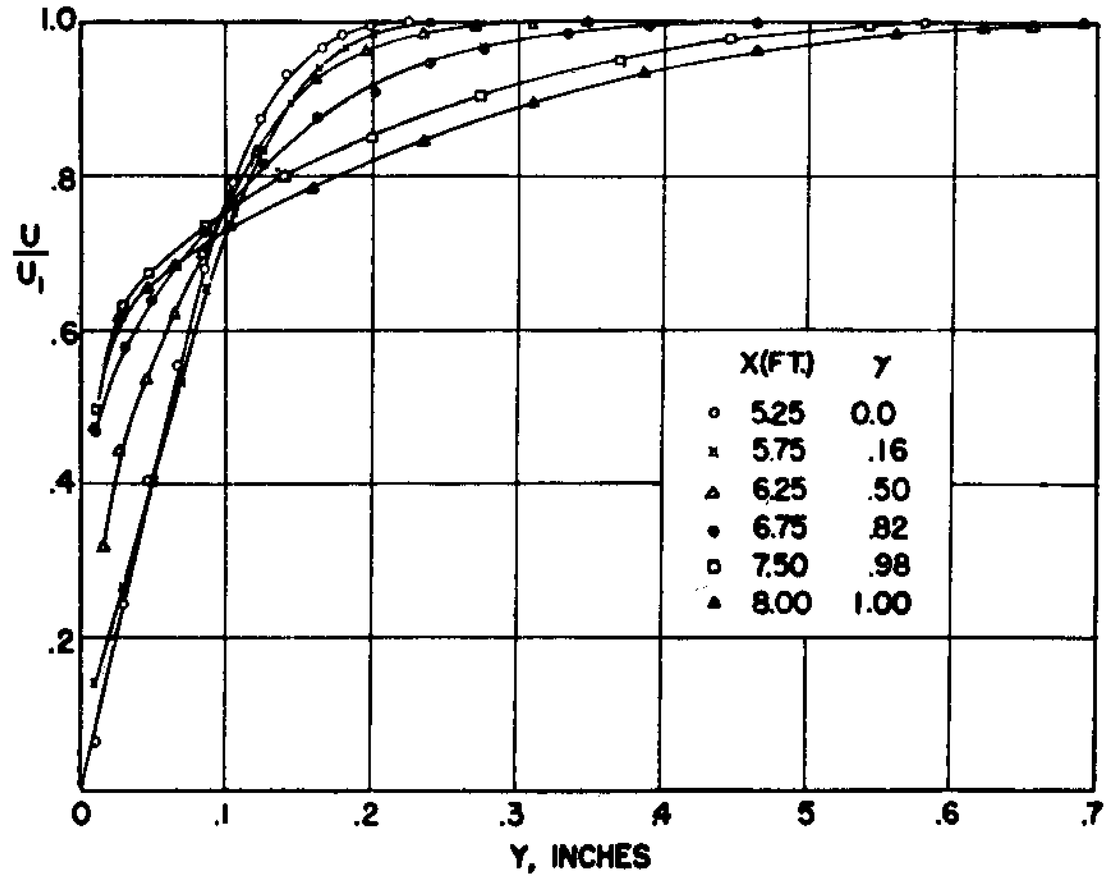

Fig. 9 (from ref. 13).
Mean-velocity profiles through transition region, $U_1 = 80$ ft/sec. Free stream turbulence 0,03 %.

the stream turbulence was low, or fluctuations of a less regular nature when the disturbance level was high. The laminar flow was followed by the transition region (which, in the cases investigated, was usually about 2 ft. long), in which increasing amounts of turbulence were observed, and finally by the completely turbulent flow. Throughout the transition region it was found that the mean characteristics of the boundary layer change gradually from those characterising laminar flow to those characterising fully developed turbulent flow. This is illustrated, for the case of low stream turbulence, by Fig. 9, taken from [13], which shows the gradual change in velocity distribution across the boundary layer with penetration of the transition region. The velocity measurements were made with a pitot tube in the normal way. The velocity distributions up to the commencement of the transition region, defined as the point nearest the leading edge at which turbulence could be found, (5.25 ft. from the leading edge in this case), were of the type characteristic of laminar boundary layer flow in a zero pressure gradient, i. e. the Blasius type. The form then changed gradually with distance into the transition region as shown in Fig. 9, until it acquired the type characteristic of fully developed turbulent flow at the end of the region (8 ft. from the leading edge in this case).

Another interesting feature of the transition region, which again shows the gradual change in its character, was that found from a statistical analysis of the frequency of occurrence of the turbulent bursts at various stations along the transition region. From many records such as are shown in Fig. 8, the proportion of total time that the flow was turbulent, defined as an intermittency factor, was determined for various points in the region and plotted against length of region. This was done for several cases in which the conditions leading to transition were varied. The length of the transition region was different in each case but it was found that the resulting distribution curves were all similar. When reduced to a common base as described on page "4 p. 4" of [13], it was found that the distributions in all cases could be represented by the same curve. This curve was a Gaussian integral curve and the good fit of this curve to the intermittency distributions is shown on Fig. 2 of [13]. Thus, for a zero pressure gradient, it was found that the transition

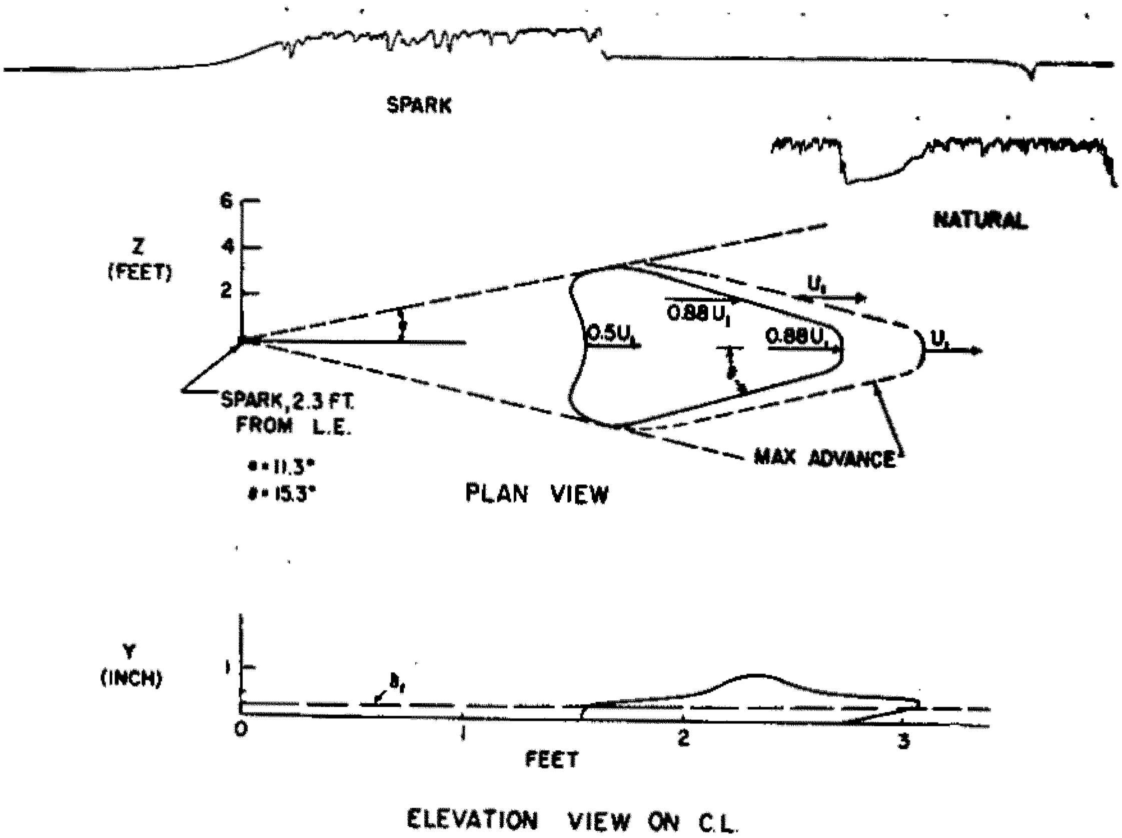

Fig. 10 (from ref. 13).

Turbulent spot initiated by electric spark between needle electrode and surface. Oscillograms with 1/60-second timing dots shown above, time progression right to left, upper showing spark discharge on right and spot passage on left, lower showing natural transition, U_1 = free stream velocity.

regions were statistically similar, whether long or short and whether the disturbances were strong or weak, irrespective of their introduction from either the free stream or a roughness element on the surface.

Schubauer and Klebanoff spent considerable time attempeting to find evidence of momentary separation in transition regions to support Taylor's concept of how transition is caused. No evidence could be found in any part of a transition region; so it would appear that this concept of transition is not a generally valid one.

Before the investigations of [13] had been made, some other experimenters had found, on rare occasions, evidence which led to the opinion that turbulent patches could exist in the laminar part of the boundary layer. Schubauer and Klebanoff decided to try and investigate this phenomenon more fully. It is known that if a roughness particle exists on a surface in a region of laminar boundary layer flow, and the free stream velocity is sufficiently high, transition occurrs at the particle and a wedge shaped region of turbulent flow extends downstream. This has been observed on aerofoils when particles of dirt or other similar surface irregularities have caused these turbulent wedges. However, if the particle producing such a wedge is suddenly removed the turbulence will recede downstream followed by laminar flow. If a particle or equivalent disturbance exists for only a brief instant of time, a turbulent spot is produced which passes downstream. Such a spot can be caused by a spark discharge across the boundary layer thickness. The above experimenters used this method to produce turbulent spots whose growth and movement were investigated with the aid of hot-wire anemometers. Their results are shown in Fig. 10, reproduced from [13]. It was seen that the trace recording the passage of an artificially produced turbulent spot had exactly the same characteristics as those typifying the turbulent bursts of a natural transition region (see upper and lower traces respectively, of Fig. 10). This discovery alone was not proof that the transition region comprised a region of turbulent spot production, although this was suspected. To ascertain if this was so further tests were made using two hot-wires, one mounted close to, and directly behind, the other. It was found that turbulence appeared first at the leading wire then

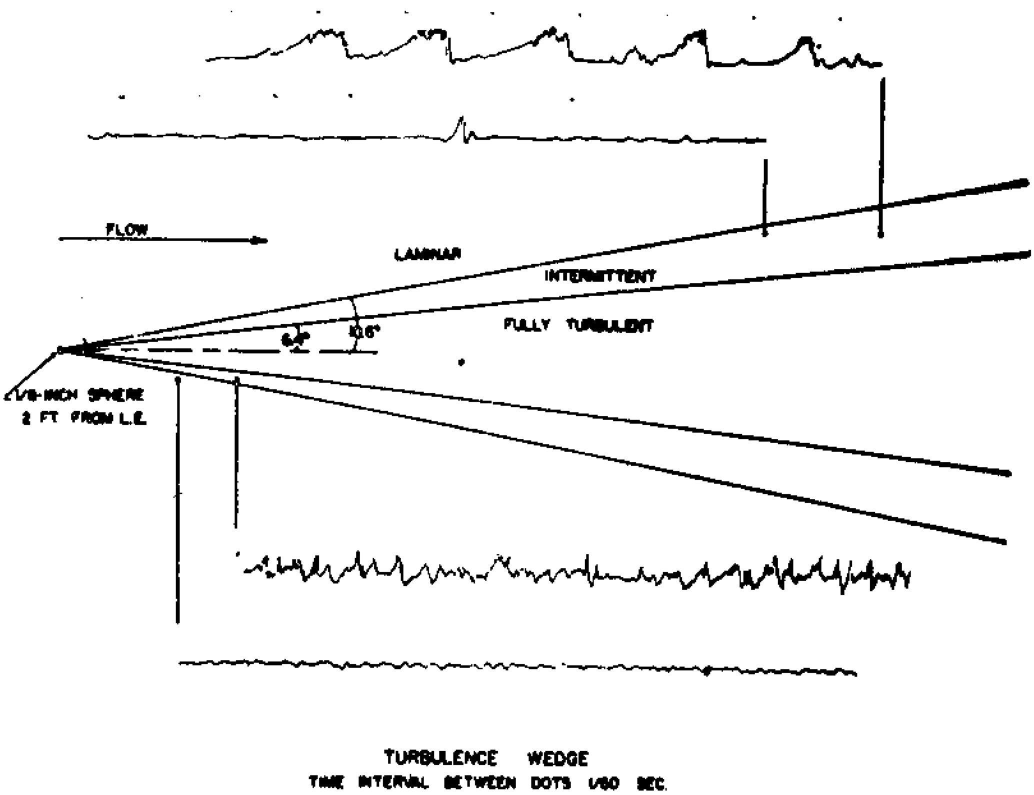

Fig. 11 (from ref. 13).

Turbulence wedge produced by three-dimensional roughness element (1/8-inch sphere) on surface. Free-stream velocity 80 ft/sec.

shortly afterwards at the rear one. The turbulence left the leading wire first and, again shortly afterwards at the rear one. This series of events can be seen from Fig. 8 of [13]. From the evidence of many such similar records, all showing the same feature, it was finally concluded that the transition region is a region in which local breakdown of the laminar flow is caused by sufficiently large disturbances in the laminar boundary layer, resulting in the formation of turbulent spots which grow and merge as they pass downstream *) in accordance with the data given in Fig. 10 of this report, and as described more fully in [13].

Another interesting discovery of [13] can be briefly mentioned. All the records made in the transition region showed that turbulent patches always exhibited the following two features:

(a) an abrupt velocity increase at their commencement, and

(b) a slow exponential-like velocity decrease at their end.

It was further noticed that deep in a transition region where the flow was turbulent most of the time, oscillations in the intermediate laminar flow were conspicuously absent. This slow velocity decrease at the end of a turbulence patch has been termed the "recovery trail" by Schubauer and Klebanoff who found that following the passage of such a recovery trail the boundary layer remained in a state of absolute calm before the disturbed condition set in again. In this calm period transition would not occur. Thus it appeared as if the recovery trail had a calming effect on the flow. It was decided to investigate this phenomenon. An artifically produced patch of turbulence was made to pass over a natural transition region. Hot-wire records were made at several stations along the region with and without spot passage. The test conditions were as follows: free stream turbulence 0.03 %, free stream velocity 80 ft./sec., beginning of transition 5. 5 ft. from leading edge, spark 0. 25 ft. from leading edge, The estimated width of turbulence spot at 5. 5 ft. position was 2 ft. The results are shown in Fig. 12, reproduced from [13]. The upper trace, at each station, is of the normal natural flow and the lower is from the natural flow while being passed over by the artificially produced recovery trail. The lower trace at the 5. 5 ft. position shows the calmed condition follo-

*) This is in agreement with a concept of the transition region put forward by H. W. Emmons in 1951.

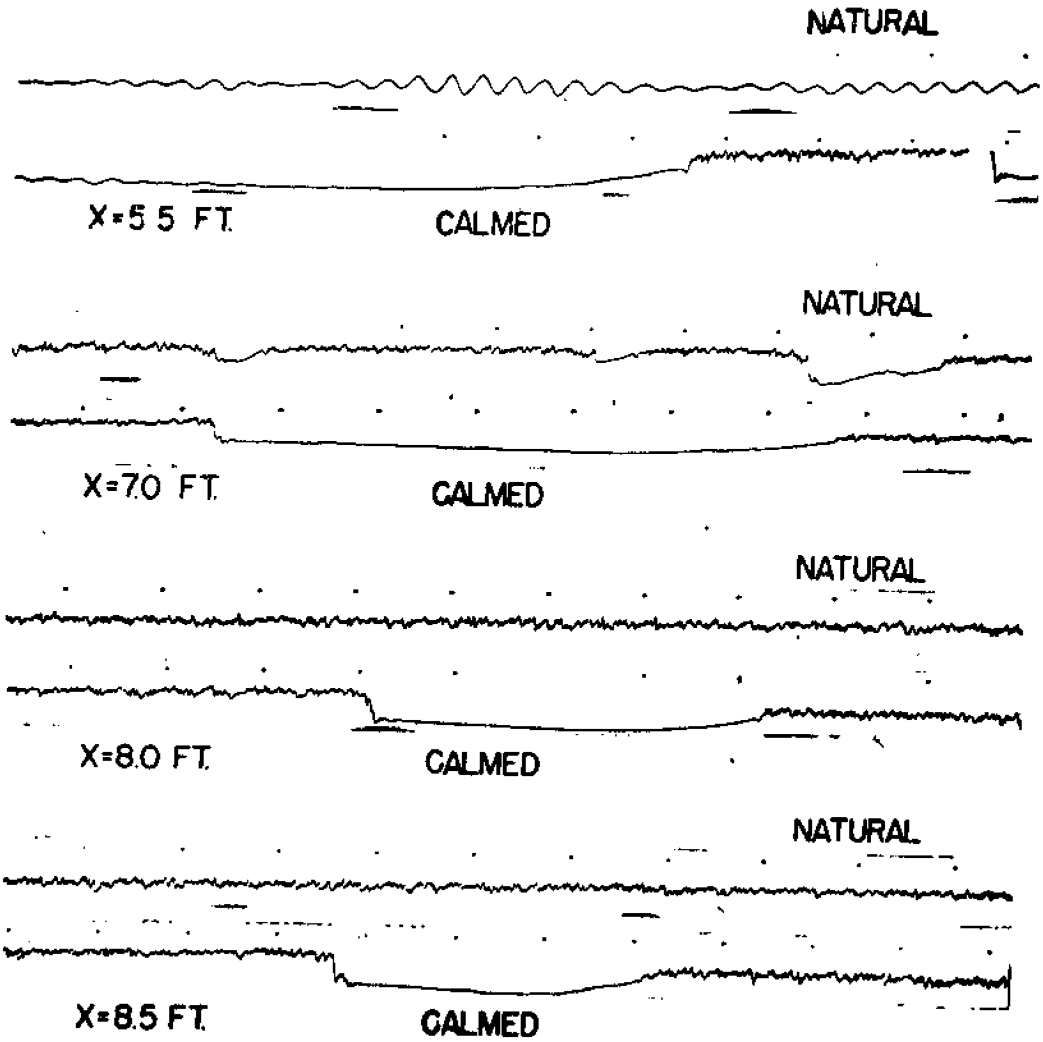

Fig. 12 (from ref. 13).

Oscillograms comparing natural and calmed condition. Time progression from right to left, 1/60 sec. between timing dots. Free-stream velocity 80 ft/sec.

wing the recovery trail as compared with the normal flow exhibiting laminar boundary layer disturbances. The traces at 7.0 ft. show that although the natural flow is nearly fully developed turbulent flow, the recovery trail brought a region of laminar flow to this station, thus producing a condition that would have occurred at a more upstream station in the normal flow. The remaining two pairs of traces show how laminar flow is brought to the 8.0 and 8.5 ft. positions, although the normal flow has been of a fully developed nature for some distance upstream of these positions. Arguments to explain this phenomenon and details of other experiments to further investigate it are given in [13]. From this work it was seen that the onset of fully developed turbulent flow could be *delayed* by the introduction of a turbulence spot into the laminar boundary layer, due to its effect of extending the transition region. The full implications of this discovery have still to be determined, but it is logical to suppose that if turbulence patches are produced by some means at the right frequency, the onset of the fully turbulent flow condition could be considerably delayed.

3.4. *Resumé of Natural Transition Knowledge.*

On the basis of recent work it can be said with fair certainty that the nature of the transition region occurring naturally in a zero pressure gradient on a hydraulically smooth surface is as given below.

It is, in general, a region of random local or spot breakdown of the laminar boundary layer caused by sufficiently large disturbances in the laminar layer. These disturbances make themselves evident as low-frequency velocity fluctuations which, when free stream turbulence is low ($< \sim 0.1$ %), are generally of a sinusoidal nature of rather pure frequency and result from the amplification of initial small random disturbances. When the free stream turbulence is of a higher orden ($> \sim 0.1$ %), the fluctuations are of a less regular nature. The turbulent spots resulting from the local breakdown of the laminar flow grow, at a fairly definite rate, as they pass downstream and merge with other growing spots. As the leading edge of a spot travelling downstream travels at a faster rate than the trailing edge of the spot ahead of it, spots eventually telescope into one another. When the lateral and longitudinal merging has developed to the extent that no intermediate laminar flow remains, the flow is then described as fully turbulent.

The whole process of laminar disturbance growth, local breakdown and growth of turbulent spot, is moving downstream as it occurrs.

From the above description of the transition region it will be realised that due to the randomness of breakdown and consequent merging processes, the beginning and end of the region will vary slightly with time and it can therefore be only strictly defined statistically.

It appears that natural transition in a zero pressure gradient does not normally occur in accordance with the Taylor concept ie. as a result of local separation of the laminar boundary layer, at any rate, not for a stream turbulence of < 1.0 %. It is known, however, that if conditions are such as to cause the whole laminar boundary layer to separate, e. g., in the presence of a strong adverse pressure gradient, it can reattach as a turbulent layer.

When stream turbulence is sufficiently great, transition can apparently occur at lower local Reynolds numbers than that found to be the limit below which all small disturbances in the laminar layer are damped.

A reasonable conception has been obtained of the sequence of events leading up to the formation of a turbulence path and of its subsequent growth. The process of and the reasons for, the actual change from highly disturbed laminar flow to turbulence remains so far, an unsolved mystery.

For further reading on transition, see [16] and [17].

4. 0. *Artificial Transition-Stimulation.*

4. 1. *Reasons for Stimulation.*

As was stated in section 2. 3, estimulation is necessary to cause fully turbulent flow to commence as near the leading edge of a model as is possible. This is due to the fact that boundary layer flow normally starts in a laminar condition and then passes through a transition stage, as described in the preceding section of this report, before becoming fully turbulent. Now it is wellknown that the relationship between skin friction coefficient and Reynolds number, which is required for all turbulent flow, is different in laminar and transitional flow to what it is in turbulent flow, as is shown in Fig. 21. 2, p. 439 of [18], and Fig. 8 of [1]. Because the lengths of flat surfaces tested are limited, for obvious practical reasons, it is necessary to reduce the length of the leading edge regions of laminar and transitional flow to a minimum, or to such an extent that their effect on the frictional resistance of the plate in "all-turbulent flow" *) is negligible. The only known means of doing this are by either testing at sufficiently high velocities or "stimulation", - the term given to the action of artificially stimulating or disturbing the boundary layer upstream of the point where naturally formed fully turbulent flow commences, so as to cause it prematurely. However, skin friction values are required at low, as well as high velocities, to obtain results over a sufficiently large range of Reynolds number, and it is therefore necessary to resort to the employment of stimulation. There are many ways of providing the necessary disturbance for stimulation and the more well known ones are mentioned below.

4. 2. *Problems of Stimulation.*

The first problem is, obviously, to know what is required of a stimulator in a particular case, to ensure that fully turbulent flow commences sufficiently near the leading edge so that the error in friction resistance caused by the presence of laminar and transitional flow comes within the permissible experimental error of the test results. That is, it is required to know what is the permissible length from the leading edge to the commencement of fully turbulent flow. This will define the minimum requirement of the stimulation and should be able to be estimated with reasonable accuracy by "trial and error", assuming conditions of laminar flow ahead of the position considered and fully turbulent flow behind, using the total skin friction curves as they exist at present. That is, for a given plate at a particular velocity, the ratio of the laminar skin friction of a leading edge portion of the plate to the total skn friction of the plate in mainly turbulent flow, as read from the Schoenherr line for example **), is determined for a series of different distances from the leading edge. The maximum distance at which this ratio becomes suffciently small as to be within permissible experimental error, is the

*) See section 2.3 for assumption used in practice.
**) This involves the assumption mentioned in section 2.3 of this report.

most downstream position that a "stimulator" *), placed in laminar flow, can be positioned to have the desired effect. However, if the stimulator is placed at this position, it must be ascertained that it actually does produce turbulent flow immediately behind it under all test conditions. If laminar or transitional flow should be produced downstream of the stimulation position, the distance determined above obviously does not apply and the maximum distance that the stimulator can be from the leading edge in this case will be less than that determined in the above way. The above method of estimation obviously only applies for stimulation positions up to the position of natural transition commencement, but in general, the distance estimated as above should be appreciably less than that from the leading edge to the position of natural transition **). If it is not, stimulation is not so necessary, unless it is used merely to shorten the length of the transition region.

It should perhaps be mentioned that if a greater percentage of laminar flow was tolerated than that indicated by the above method, and a correction made for it by substracting the friction of the laminar leading edge portion, the resulting turbulent flow data could only give information on the integration of local turbulent skin friction values between the range of Reynolds numbers defined by the local R_n of the end of the laminar region and the R_n of the whole plate. This information is of relatively little use until the effect of the leading edge becomes negligible, which is the case considered previously.

Having determined the minimum demands required of a stimulator the next problem is obviously to obtain a stimulator capable of supplying the demands. Very little detailed information is available on this, and up to the present, the most usual method of stimulation has been that of placing some form of "roughness element" at a constant position near the leading edge of the model and "hoping for the best". The efficiency of the stimulator is judged by the nearness of the resulting experimental values of skin friction to a line purporting to represent results for effectively all turbulent flow, but which itself is not thought to be absolutely correct. This state of affairs is indicative of the difficulty of the problems concerned with stimulation. As explained in section 2. 3, during a series of tests in which velocity varies, the demand from a stimulator at a constant position, usually adopted for simplicity of experimental technique, will vary. This may involve the troubles, mentioned in section 2. 3, of partial or over-stimulation. The answer is to vary either the power or the position of stimulation, even if this latter solution is troublesome, but this is not of much use unless an idea can be obtained of what is qualitatively happening in the boundary layer at the same time. Thus until a way of "seeing" the effect of the stimulator can be found, the capabilities of a particular type of stimulator cannot be accurately assessed However, as a guide to the required change of position of stimulation, it should be varied so as to keep the local R_n of the position constant, since transition phenomena are a function of local Reynolds number. This would have the additional advantage, theoretically, that the friction resistance of the laminar leading edge portion of the plate would be constant. The use of hot-wire techniques should be able to provide the required insight into stimulation effects but involves the use of expensive and elaborate test equipment. Another possible method, though not nearly so precise, is that used to determine the length of a transition region by Burgers and van der Hegge Zijnen mentioned in [7], p. 325. This is a very simple method and is known in aeronautics as the "surface pitot technique" *). The method does not seem to have been used very much in hydrodynamics. It involves finding the velocity gradient distribution at the plate surface along the plate in the region where transition occurs. This can be done by traversing a pitot tube along the surface. The method relies on the fact that in both laminar and turbulent flow the velocity gradient at the wall decreases with distance dowstream but in transition flow increases. This is shown in Fig. 13 where velocity gradient α is plotted against distance downstream from the leading edge for various free stream velocities U_1. These are the results of Burgers and van der Hegge Zijnen obtai-

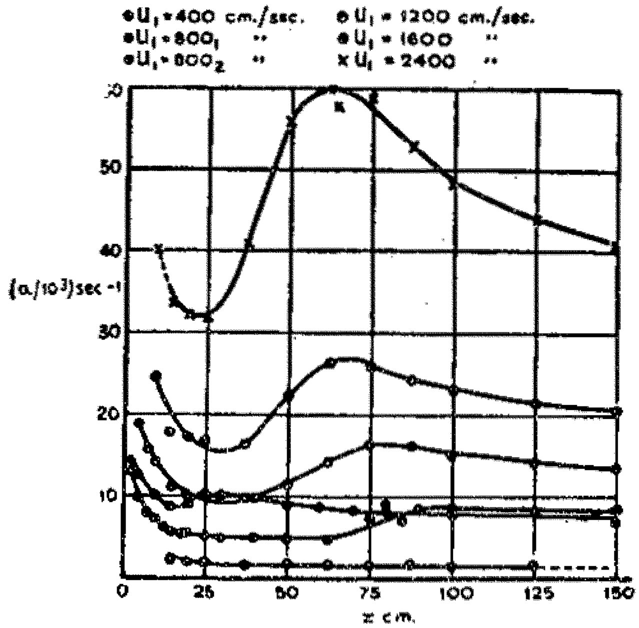

Fig. 13 (from ref. 7).

*) This term is used for convenience and refers to any source of stimulation whether it be a small body or merely a source of energy.

**) See section 2.3.

*) This technique is explained fully in [19].

ned in a wind tunnel on a glass plate. The figure is reproduced from [7].

Another general problem of stimulation is to determine whether or not the flow produced by stimulation is the same as natural flow and, if it is not, whether or not the difference has any significant effect on the skin friction resistance. Because of the obvious difficulties of finding this out, it is usually taken for granted that stimulated flow has the same character as natural flow, although evidence of over-stimulation seems to testify that this is not always so. Again, before this can be investigated fully, some means of knowing in detail what goes on in the boundary layer must be found. Only from the results of hot-wire test has any light been shed on this problem. This will be dealt with in seccion 4.4.

The above outlines, in the author's opinion, the main ganeral problems of stimulation. Other problems exist but these are associated with the particular type of stimulator used and will be mentioned in the following section.

4. 3. *Methods of Stimulation.*

The methods of stimulation can be divided into two main groups, - those that employ the disturbance caused by a physical body, usually placed in the boundary layer, and those that use the disturbance caused by a pure source of energy.

In the first group the disturbance is usually supplied by a fine "trip" wire, a row of studs of suitable size and spacing, or a rough strip of material such as sand paper. The stimulator is placed parallel to and near the leading edge of the model. The advantages of this type of stimulator lie in its simplicity. The disadvantages are that being a physical body, it has a resistance itself which has to be known accurately and allowed for and that its stimulation property is uncontrollable and purely dependent on the velocity of the test, being constant for a given velocity. This type is also more likely to produce "unnatural" flow (causing over-stimulation?) than a pure source of energy. Because of its simplicity, ease of use and cheapness, despite its disagvantages this type of stimulator is the one most commonly used and consequently about which most is known with regard to its effectiveness. Even so, this knowledge is slight.

The other group, those employing some sort of energy disturbance, comprise methods of stimulation based on the effects of creating large free stream turbulence sound waves, vibration, an electrical discharge (spark) etc. Even less information exists about this group than for the previous group because of the relative newness and consequent rarity in the use of these methods. Thus their potentialities have not yet been determined. The advantages of this type of stimulation are that they involve no corrections to the measured plate resistance and that there exists the possibility to vary the amount of stimulation exactly in accordance with the requirements of a test. It is also possible that this type can be more effective. The disadvantages are the generally more complex and costly equipment required to provide the source of energy.

4. 4. *Empirical Data on Stimulation.*

As has been stated previously, very little detailed data exists on the effects of estimulation and what there is, is of an uncorrelated nature. The available data is considered below.

To begin with, there is the question of whether or not the type of flow resulting from stimulation is the same as natural flow. This must obviously be determined for each individual case. It is possible that homogeneous natural flow may be produced by a stimulator rod, for example, at certain speeds, whereas at higher speeds discrete eddies could be formed in the flow. This information can at present be only supplied by hot-wire investigations. The only data that can be found on this question is that shown in Figs. 8 and 11 of this report which are reproduced from [13]. Fig. 8, indicates that the type of flow produced at a distance of 3. 3 ft. behind a 1 mm trip wire placed along the surface of a plate at 2 ft. from the leading edge in a low turbulance airstream (0.02 %) at a wind speed of 33 ft./sec. had the same characteristics as the natural flow occurring at the same position at approximately the same speed with a free stream turbulence of 1. 0 %. Thus in this case the flow produced by a stimulator in a low turbulence stream was similar to the natural flow occurring in a high turbulence stream. As is seen from the difference in character of the turbulent bursts, this is different from the flow produced in a low turbulence stream. Thus the effect of the stimulator was to effectively increase the stream turbulence and presumably have the same effect on the point of commencement of fully turbulent flow as an increase in free stream turbulence. Hence it would appear that stimulation can be caused as effectively by increasing free stream turbulence, a relatively simple procedure, as by using a wire stimulator on a model and incurring all the associated problems. However, this conclusion must await further confirmation. It should be noted that in the above case the flow was found to be laminar *downstream* of the trip wire, which emphasises the necessity for determining positively, in experiments, the points of initial and final transition. The conclusion arrived at above may only apply when disturbances are caused in the laminar boundary layer, ie. at higher Reynolds numbers, when transitional or fully turbulent flow commences immediately behind the stimulator its effect may not be as concluded above.

Another piece of information supplied by the results of [13] is shown in Fig. 11 of this report. One type of stimulator is that formed by a row of studs or "roughness elements". Fig. 11 sheds some light on the performance of one such element - in this case a 1/8 in. sphere positioned 2 ft. from the leading edge of a plate in a windstream of 80 ft./sec. If, for a given velocity, the size of a roughness element placed in a laminar stream is sufficient, transition can commence at the stimulator. Fig. 11 shows the results of hot-wire investigations into the nature of transition behind the above sphere. The traces record the type of flow fluctuations occurring in various parts of the transition region. The fully turbulent flow was found to have the same character as natural fully turbulent flow, as found in the case of the trip wire. When the boundaries of the transition region are linear the angles are, as near as can be determined, always the same. Is was previously thought that there was only one boundary which had a half angle of about 10°, but the results of [13] show that there are in fact two which include a transition region as shown in Fig. 11. The fully turbulent core was found to have a half angle of approximately 6.5° and to increase in width and always be much wider than the wake of the roughness element. This more rapid widening has been termed "transverse contamination". The above angles can be of use in estimating the length of transition region expected from a given element spacing. It is obvious that if this type of stimulator is used, which is probably more effective ie. it can produce fully turbulent flow at a lower R_{nt}, than the trip wire because of the three-dimensional disturbance produced (as compared with the two-dimensional disturbance of the wire), the spacing must be quite close to ensure a small transition legth. It should be noted, in connection with the way in which transition is caused by this stimulator configuration, that when the velocity corresponding to a particular size of stimulator was not sufficient the above boundaries were not straight, or consequently the angle constant, but initially curved outward, approaching the angle asymptotically. It is also known that in the above case the wedge may not begin until some distance downstream of the stimulator.

The other information available is of a more general nature and concerns the effectiveness of various types of stimulators. It is summarised below.

The most detailed data is given in [14] which deals with the hot-wire results of an investigation to determine the effectiveness of various types of stimulators in producing turbulent flow on ship hulls. The types of stimulator investigated were a sand strip and trip wire placed on the bow and a rod towed in front of the model. Of these the latter was by far the most effective *) in that it produced turbulent flow at the lowest local Reynolds number. Of the other two the trip wire was slightly more effective. The only problem connected with the rod was that it gave the model a reduced effective speed, which was not known accurately, due to the model being in the wake of the rod. A very interesting result of the rod experiments was that at a sufficient model speed (3.5 knots) fully turbulent flow was found to commence right at the model "leading edge". This rod was 1/8 in. diam, and towed at 48 diams. in front of the model, this distance having been found to be the most effective. Further remarks on the use of rods as stimulators are given in [15].

Some work has been done on acoustic methods of stimulation on the basis of the concept of transition in accordance with the stability theory of laminar flow, mentioned in section 3.1 of this report. This is dealt with in [22]. No empirical results are yet obtainable.

4.5. *Concluding Remarks.*

From the above account of the state of affairs of stimulation at the present time, it will be clear that very much work remains to be done before it can be employed really successfully. The understanding and knowledge of this process has been delayed by the absence of a method which enables a detailed insight into the mechanics of stimulation to be obtained. With the advent of the application of hot-wire techniques to the study of stimulation, this insight is gradually being obtained but until this is much more complete the need for the adoption of one of the simpler methods that exist for determining where transition actually occurs in all model tests in emphasized.

5.0. *Suggestions for Further Tests at SSPA.*

5.1. *General.*

Because of the limited time and financial means available for "flat-plate" skin friction investigations it would be too ambitious a programme to attempt to determine sufficient skin friction values for a complete turbulent friction line. It is the author's opinion that only spot tests should be attempted to provide a check on the correct level of this line in the light of previous investigators' experience and results. Therefore the tests suggested below are of a simple nature and fall into three categories, viz.

(a) Stimulator tests to check the applicability of a method of determining the extent and position of the transition region and to investigate the effectiveness of a suggested stimulation method.

(b) Total resistance tests to check the edge effect

*) This finding was also endorsed by the results of [21].

conclusion from the results of Allan and Cutland [5], to obtain more information on the extent of edge effect and to determine effects of curvature and possibilities of testing thin curved plates.

(c) Pitot tube investigation incorporating the findings of the tests of categories (a) and (b), to determine two-dimensional skin friction values in turbulent flow.

In all the tests dealt with above the thickness should be a minimum consistent with negligible distortion. All the edges should be as finely tapered as possible and the surfaces as smooth as possible.

5. 2. *Stimulator Investigations.*

If possible, stimulator investigations should be carried out with hot-wire equipment. The special application to tests in water is given in the appendix to [14].

If this equipment is unobtainable, the method of [19], referred to in section 4. 2. of this report, should be tried.

It was seen in section 4. 4. that the most effective form of stimulation of the well known types was that obtained by a rod placed at a fixed position in front of the model. The only disadvantage was that the effective speed of the model was altered by an unknown amount. To obviate this it is suggested that a screen or lattice of such rods, extending outside the width of the plate being tested, should be towed in front of the model. This will give a modified velocity field as in a wind tunnel, instead of the narrow wake produced by one rod. The screen or grid should be placed at such a distance ahead of the plate that a uniform velocity field is produced across the plate. Pitot tube neasurements should be made between the grill and model to determine the modifield velocity. A suggested mesh is of 4 mm diam.. rods at a spacing of 4 cm. Mesh dimensions should be investigated, however, to find the most suitable. These initial tests could be performed with the small plate suggested in the next section. For convenience of pitot tube traversing, the plate should be towed in the horizontal plane at a submersion that causes negligible wave formation. The position and extent of the transition region should be found by one of the above methods for the range of tests speeds covered in the main tests suggested in section 5. 4. If these tests are a success and fully turbulent flow commences sufficiantly near the leading edge, as is found to be necessary for the main tests, this form of stimulation should be used for the subsequent tests. If not, the same investigations should be carried out with a trip wire to determine its most effective position ie. the position nearest the leading edge at which transition can be caused, and this method of stimulation used for subsequent tests.

5. 3. *Exploratory Tests.*

These are partly tests on a small flat plate to investigate the extent of the edge effect that is shown to exist by the results of Hughes' tests, and to try and verify the conclusion drawn from the results of Allan and Cutlands' tests [5] that the maximum length/breadth ratio of a flat plate, with one edge immersed, on which *any* effectively two-dimensional flow can exist is approx. 7.

The other tests in this section are to investigate the possibility of testing curved, thin plates, with a radius of curvature so large compared with its length that negligible "pipeflow" effects are obtained, as a means of obtaining two-dimensional flow skin friction values.

It is suggested that a plate 1 1/2 m × 1 m should be suspended in a vertical plane, to obtain the advantages mentioned in section 2.1, and tested at various immersions with both the long edge and short edge vertical in turn. A range of immersions should be covered in each case to give an l/b ratio variation from about 10 to the minimum practicable at equal intervals and total resistance measurements made at each configuration. When two-dimensional flow is obtained the increments in resistance should be equal.

For curved plate tests, thin sheet should be used owing to the increased anti-bending stability introduced by the curvature. Flat sheet should be flexed and welded to form a series of cylindrical shells. A suggested length is 1 m. A variation in diameter of from 1/2 m upwards should be tested to determine the maximum curvature (minimum diameter) at which an increase in diameter has no appreciable effect on total resistance. If this comes within practicable limits this configuration provides a possible solution to the problem of obtaining two-dimensional flow conditions. A theoretical analysis of the influence of transverse curvature on frictional resistance is given in [23].

5. 4. *Main Tests.*

Having determined a suitable stimulation technique and the required l/b ratio for obtaining a region of two-dimensional flow at the centre of a plate, pitot tube measurements should be made to determine local and total skin friction values. It is suggested that a plate at least 3 m long should be used (4 m if possible) and tested in the horizontal plane, supported at the corners (and mid edge-length if necessary), for convenience of pitot traversing. The breadth will be determined by the l/b ratio determined from the exploratory tests. The l/b ratio to be used will be actually half that found from these tests because of the two immersed edges, ie. if the previous value of 7 concluded is correct, an l/b ratio of 3.5 should be used to determine the minimum breadth of plate. As large a speed range as possible should be covered and, in the author's opinion, it is essential to determine the position and extent of the transition region in each run. The importance of this cannot be overemphasized.

Appendix.

Summary of factors which can affect resistance measuments to be borne in mind when conducting "plane" friction tests.

Whether using vertical planes partly submerged, pontoons of very shallow draft or completely submerged planes, it is impossible to avoid the measured resistance being affected by some of the following factors:

(a) Form resistance.

(b) Wave resistance (a consequence of form).

(c) Lack of absolute symmetry of the plane, resulting in side forces and possible further distortion if the plane is structurally weak.

(d) Change of wetted surface area.

(e) Change of trim.

(f) Resistance of the turbulence stimulator itself (if fitted).

(g) Incomplete turbulence stimulation (whether a turbulence stimulator is fitted or not).

(h) "Over-stimulation". (Due to large stimulators, required for turbulence stimulation at low speed, being used at high speed). If small stimulators, eg. rough strip, are used, these are very efficient at the relatively high speeds but can give incomplete turbulence stimulation at the lower speeds.

(i) Interference from supports.

(j) Air resistance of above-water structure.

Some of these effects are always present, eg. air resistance, form resistance and the resistance of the stimulator (when fiitted). These can be measured or estimated with fair accuracy and the results corrected accordingly. At low actual speeds of advance complete turbulence stimulation is difficult to achieve unless the stimulator is relatively large. This is not desirable because of the possibility of over-stimulation at higher speeds. At moderate or high speeds of advance, wave resistance (being a result of form effect) and/or excess resistance due to distortion of the plane (if the latter is made thin to minimise form influence) cannot be avoided except with completely submerged planes, and with these, interference from underwater supports is inevitable.

The obtainment of truly corrected results, having considered the above effects, is further complicated by the "edge effect" problem, thus making deduction of true two-dimensional resistance values extremely difficult.

Nomenclature.

R_n = Reynolds number $= Ul/\nu$

U = Free stream velocity

ν = Kinematic viscosity of fluid

C_F = Coefficient of frictional resistance = $\dfrac{\text{Frictional Resistance}}{\frac{1}{2}\rho U^2 A}$

ρ = Mass density of fluid

A = Wetted surface area of body

R_{ntr} = Transition Reynolds number $= Ul_{tr}/\nu$

l_{tr} = Length from leading edge of model to point of transition initiation

R_{nl} = Local Reynolds number $= Ul/\nu$

l = Length from leading edge of model to position under consideration

N. B. The nomenclature used in the diagrams is that of the reference from which they were taken, unless otherwise stated.

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Mr. A. Emerson (Written contribution).

The use of geosims to study the effect of scale on model resistance experiments is limited because the large model results require a substantial blockage correction and the small model results require a substantial turbulence stimulation correction. Experiments have been made at King's College to examine these two corrections.

1. *Blockage.*—The Lucy Ashton model tests (1) were made in N. P. L. No. 1 Tank, which has a breadth (30′) and normal water depth (11.5′) two and a half times as large as the corresponding dimensions of the King's College Tank (12′ × 4,5′). Thus by testing a 9′ Lucy Ashton model at King's College, a blockage area ratio — m, of 0.042 is obtained instead of the 0.007 at N. P. L. This avoids the possible errors introduced when blockage is assessed by tests of different size models in the same tank. Tests were made over the full range with a 9′ Lucy Ashton model, over a considerable portion of the range with a 12′ model, and a few low speed values were obtained with a 16′ model. For the 16′ model, there is insufficient length of run to ensure steady resistance values.

Using the method given by Schuster (2) as a basis, it will be seen that the shallow water wave effect is negligibly small, if, h being the water depth, $v/\sqrt{gh}$ is 0.5 or less.

It is convenient to present all the results in a single diagram and this has been achieved in Fig. 1 by plotting $C_T = R/\frac{1}{2}\,\rho v^2\,S$ corrected by Schoenherr coefficients to 59° F, against $v/\sqrt{gl}$. The well defined curve for the 16′ model in N. P. L. No. 1 Tank has been used to give a standard resistance curve and the viscous resistance coefficient taken, as in the original Lucy Ashton paper by Conn, Lackenby and Walker, as Schoenherr + 8 % is compared with each set of experimental results. The C_T values plotted are corrected for trip wire resistance using the method given by Allan and Hughes. (3) Some of the results of N. P. L. tests without trip wire have been plotted at the higher speeds to help the definition of the curves.

The blockage effect $\delta v/v$, the increase in speed necessary to make the experimental C_T curve coincide with the standard C_T curve is given in Table 1.

In the Table, v is the measured speed, $v + \delta v$ the equivalent unrestricted water speed and m the ratio of model midship section area to tank cross section area. The calculated values are obtained from Schuster's equation 8,

$$\frac{\delta v}{v} = \frac{m}{1 - F_h^2 - m}$$

where $F_h = v/\sqrt{gh}$ and h is the depth of water. The ratio of measured value $\delta v/v(m)$ to calculated value $\delta v/v(c)$ is about unity at low speeds, but the resistance curves are ill defined in the region. Over the range $v/\sqrt{gl}$ 0.16 to 0.32, the ratio is about 1.6 Having regard to the fact that this is a ratio of small differences to the calculated values, the results are very consistent. The measured values are related to the 16′ result at N. P. L. and a comparison with a true "zero" blockage would increase the 1.6 ratio to 1.8 or 1.9.

The calculated δv is a mean value. As the changes in velocity due to blockage from zero at bow and stern to a maximum amidship, are of the same sign as the velocity changes caused by the model in unrestricted water, it is not unreasonable to find that the effective measured δv_m is greater than the calculated mean value, particularly in the range where wave making resistance is important.

Incidentally the low value of C_T on the 9′ model of the maximum wave interference hump, $v/\sqrt{gl} \sim 0.3$ may indicate a tendency of importance when model results are extrapolated to full scale.

2. *Turbulence stimulation.*—Ink stream observations on a 6.25′ Lucy Ashton model appeared to indicate laminar flow at the fore end, over about 8 % of the total wetted surface at $v/\sqrt{gl} = 0.16$, over about 5 % at $v/\sqrt{gl} = 0.22$, and 1 ½ % at $v/\sqrt{gl} = 0.26$.

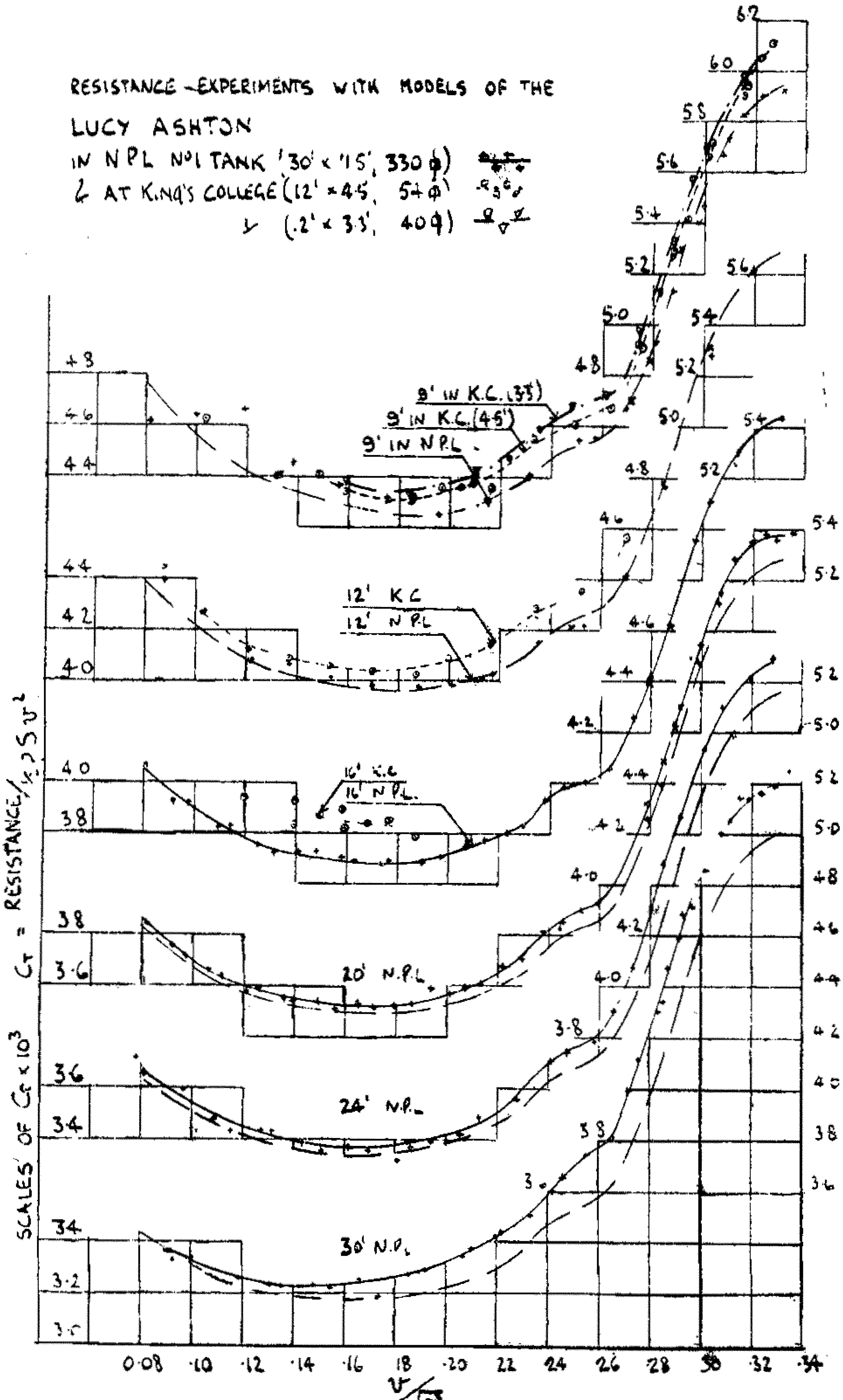

Fig. 1.—Blockage effect.

Table 1. BLOCKAGE EFFECT

$\frac{v}{\sqrt{gl}}$	0.12	.16	.20	.24	.28	.30	.32
30' Model at N. P. L. $m = 0.0077$							
$R_n \times 10^6$	9.11	12.15	15.18	18.22	21.26	22.77	24.30
C_T meas.	3.255	3.240	3.320	3.585	4.230	4.830	5.145
Standard	3.217	3.175	3.219	3.477	3.973	4.582	4.943
$\delta v/v$ meas.	.006	.010	.014	.014	.016	.016	.016
$\delta v/v$ calc.	.008	.008	.008	.009	.010	.011	.011
Meas. δv/Calc. δv	0.7	1.2	1.6	1.8	1.6	1.5	1.5
24' Model at N. P. L. $m = 0.0049$							
$R_n \times 10^6$	6.520	8.692	10.87	13.04	15.22	16.31	17.39
C_T meas.	3.440	3.380	3.420	3.700	4.295	4.905	5.220
Standard	3.402	3.345	3.381	3.628	4.124	4.730	5.091
$\delta v/v$ meas.	.006	.006	.006	.0085	.0105	.010	.010
$\delta v/v$ calc.	.005	.005	.006	.006	.006	.006	.0065
Meas. δv/Calc. δv	1.2	1.2	1.0	1.4	1.7	1.7	1.6
20' Model at N. P. L. $m = 0.0034$							
$R_n \times 10^6$	4.975	6.610	8.263	9.918	11.57	12.39	13.22
C_T meas.	3.605	3.525	3.565	3.825	4.350	4.960	5.330
Standard	3.564	3.498	3.524	3.759	4.257	4.862	5.220
$\delta v/v$ meas.	.003	.004	.005	.007	.0055	.006	.008
$\delta v/v$ calc.	.0035	.004	.004	.004	.004	.004	.004
Meas. δv/Calc. δv	0.9	1.0	1.2	1.7	1.6	1.6	2.0
16' Model at *N. P. L.* $m = 0.0022$							
$R_n \times 10^6$	3.547	4.730	5.913	7.096	8.278	8.869	9.462
C_T meas.	3.775	3.694	3.712	3.943	4.426	5.031	5.386
Standard							
$\delta v/v$ meas.		Assumed zero					
$\delta v/v$ calc.	.002						.0025
16, Model at *K. C.* $m = 0.0133$							
C_T meas.	3.945	3.850					
$\delta v/v$ meas.	.024	.021					
$\delta v/v$ calc.	.016	.016					
Meas. δv/Calc. δv	1.5	1.3					

TABLE I. cont'd. BLOCKAGE EFFECT

$\frac{v}{\sqrt{gl}}$	0.12	.16	.20	.24	.28	.30	.32
12' Model at N. P. L. $m = 0.0012$							
$R_n \times 10^6$	2.306	3.076	3.846	4.611	5.383	5.765	6.152
C_T meas.	4.120	4.000	as standarrd				
$\delta v/v$ meas.	4.069	3.975	3.976	4.194	4.673	5.272	5.624
$\delta v/v$ calc.	0.001						
Meas. δv/Calc. δv							
12' Model at K. C. $m = 0.0075$							
C_T meas.	4.135	4.045	4.075	4.320			
$\delta v/v$ meas.	0.0085	.0095	.0115	.0125			
$\delta v/v$ calc.	.008	.008	.0085	.009			
Meas. δv/Calc. δv	1.1	1.1	1.4	1.4			
9' Model at N. P. L. $m = 0.0007$							
$R_n \times 10^6$	1.497	1.996	2.494	2.994	3.494	3.741	3.991
C_T meas.	?	4.350		standard		5.490	
Standard	4.411	4.294	4.273	4.479	4.945	5.541	5.887
$\delta v/v$ meas.							
$\delta v/v$ calc.	0.0007						
Meas. δv/Calc. δv							
9' Model at K. C. $m = 0.0042$							
C_T meas.		4.350	4.350	4.560	5.070	5.435	6.010
$\delta v/v$ meas.		0.007	0.009	0.007	0.008	0.004	0.009
$\delta v/v$ calc.		.004	.005	.005	.005	.005	.005
Meas. δv/Calc. δv		1.7	1.8	1.4	1.6	0.8	1.8
9' Model at K. C. (reduced depth) $m = 0.0057$							
C_T meas.		4.375	4.375	4.610	5.105	5.680	6.025
$\delta v/v$ meas.		0.010	.011	.013	.010	.007	.010
$\delta v/v$ calc.		.006	.006	.007	.007	.007	.008
Meas. δv/Calc. δv		1.6	1.8	1.9	1.4	0.9	1.2
6.25' Model at K. C.							
C_T meas.		4.540	4.565	4.740	5.180	5.770	6.090
Standard		4.738	4.697	4.884	5.340	5.922	6.260

When a 0.033″ trip wire was fitted at 9 ½ Station (5 % l from the fore perpendicular), the laminar streams appeared to become turbulent at the trip wire at the lowest speed observed, $v/\sqrt{gl}=0.16$, except for one stream near the keel, which persisted about 2″ abaft the wire at that speed.

Investigation of the nature of the boundary layer over the forward part of the model was also made by Mr. R. L. Townsin, using a hot wire technique which he describes briefly in Appendix 2. For the model fitted with an 0.033″ trip wire, at station 9 ½, the hot wire probe was traversed round a section 1 ½″ behind the trip wire. At very low speeds $v/\sqrt{gl} = 0.09$, the boundary layer was laminar round most of the section. With increased speed varying amounts of transitional flow were recorded and at $v/\sqrt{gl} = 0.12$, the records consisted entirely of turbulent oscillations. Traverses at 10 % and 15 % from the stem showed that the flow remained turbulent as it moved aft over this part of the model.

Results of resistance experiments made with the 6.25′ model, naked (without trip wire), with the 0.033″ trip wire and with an 0.067″ trip wire are shown in the upper part of Fig. 2 as values of C_T ($=$ Resistance/½ $\rho S v^2$) to base of $v/\sqrt{gl}$. The values have been corrected to 59° F using Schoenherr coefficients, and allowance has been made for trip wire resistance, following the method given by Allan and Hughes. The temperature correction is small and the amount of trip wire correction is shown in Fig. 2.

The results are reasonably consistent. The two different trip wire sizes give the same corrected C_T curve. If allowance for observed laminar area is added to the naked model results—approximately 0.030×10^{-3} for each 1 % laminar area—they approach the other values and some of the difference will be due to transitional flow.

The standard curve which fits the other Lucy Ashton geosims reasonably well is also shown in Fig. 2 and it will be seen that the 6.25′ model results lie about 0.150×10^{-3} below the standard curve. This difference is not due to inadequate turbulence at the bow. The possible area of laminar flow forward of the trip wire position on any of the models is about ½ %—making a total possible differential of 0.020×10^{-3}. The trip wire correction for the 0.067″ trip wire is questionable, because the diameter is of the same order as the boundary layer thickness. But this does not affect the 0.033″ trip wire result.

Adjustments to the basic friction line would improve the 6.25′ comparison and spoil the 9′ model comparison. (It is interesting to observe that if only the 6.25′ model and the 30′ model had been tested, a comparison based on Schoenherr with no form correction would have appeared satisfactory.)

It is clearly desirable to investigate this model result more closely, to find out what flow differences occur between the 9′ and 6.25′ models.

The estimation of trip wire resistance is still unresolved. The bottom half of Fig. 2 showing the results obtained with the 9′ Lucy Ashton with and without trip wire. The speed at which transition to turbulence was forward of the trip wire position is indicated on the diagram. Above this speed the two curves—with and without tripe wire should coincide. The result suggests that the correction is underestimated—the approximation used by Breslin and Macovsky give values roughly three times as large, and this difference fairly represents the possible error in estimating stimulation drag. When transition takes place forward of the trip wire the estimate is invalid in any case.

3. *General conclusions.*

1. For standardisation of model presentation it is suggested that a blockage correction based on Schuster's analysis would reduce errors due to tank size.

2. For models 16′ and larger, the present practice of roughly equating possible laminar flow effect forward of the trip wire with trip wire drag is as accurate as present data allows. Small models require more elaborate examination.

3. Taking the Lucy Ashton results presented in the original paper as typical, it is unsound to base conclusion on values and slopes derived from experiments at very low speeds where the wave making resistance is negligibly small.

References.

(1) B. S. R. A. Resistance Experiments on the Lucy Ashton. J. F. C. Conn, H. Lackenby, W. P. Wal-
(2) Beitrag zur Frage der Kanalkorrektur bei Modelversuchen. S. Schuster. Schiffstechnik. Vol. 3, 1955-56.
(3) Turbulence Stimulation on Ship Models. J. F. Allan and G. Hughes. Soc. Naval Architects and Marine Engineers. Vol. 59, 1951.

APPENDIX 1

Model data.

The Lucy Ashton models were fully described in the original I. N. A. paper in 1953.

Corrections for a temperature less than 59° F were made by finding the Reynolds' Number at the same speed, but using the 59° F coefficient of kinematic viscosity (1.2285×10^{-5}) taking the difference in Schoenherr skin friction coefficient at the original Reynold's Number and the 59° F Reynold's Number and substracting this difference from the experimental C_T coefficient.

The trip wire corrections at $v/\sqrt{gl} = 0.22$ were 0.025, 0.013, 0.008 $\times 10^{-3}$ for the 9′, 12′ and 16′ models respectively.

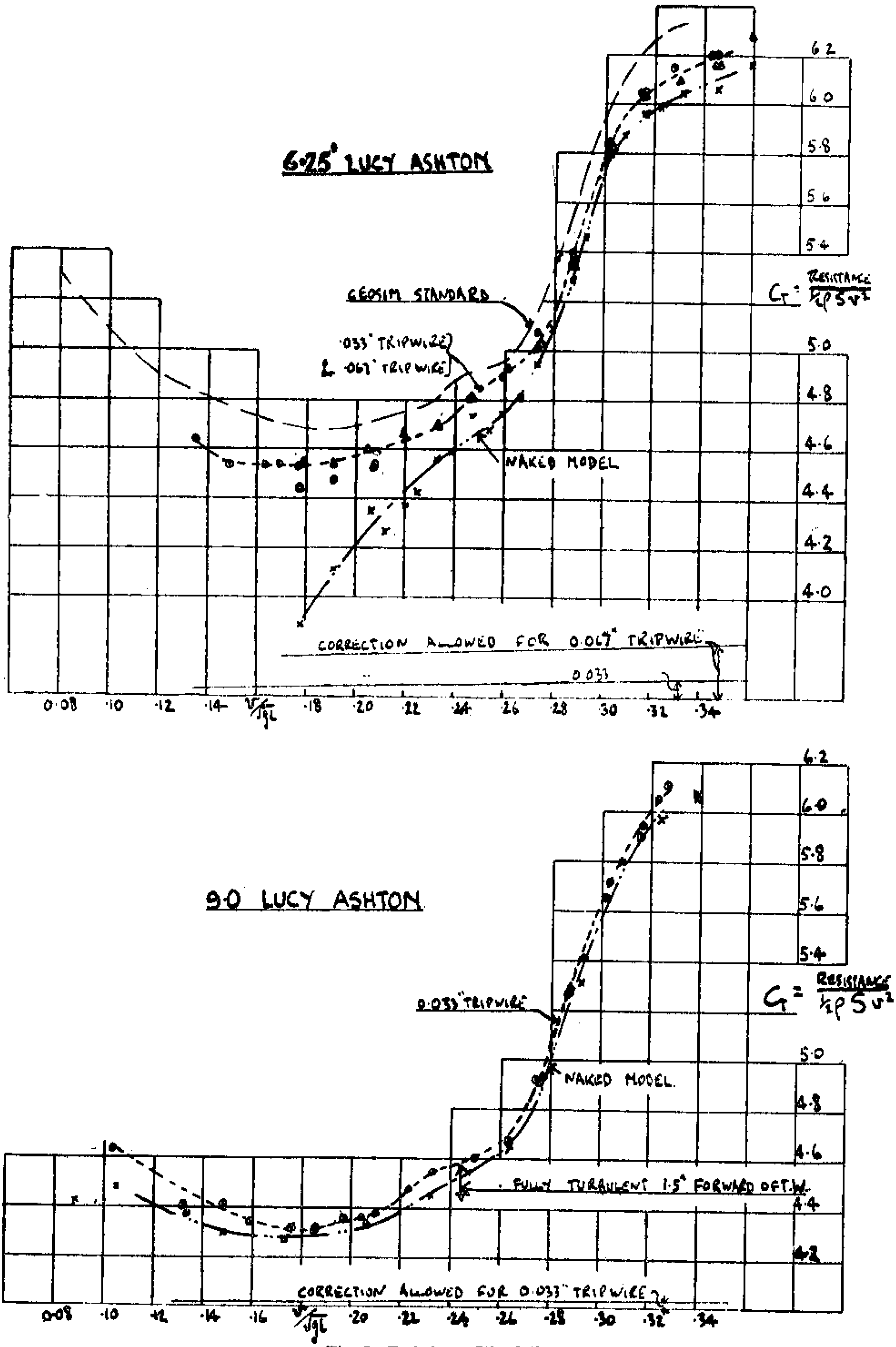

Fig. 2.—Turbulence Stimulation.

For the 6.25′ model, the displacement was 30.1 lbs. and the wetted surface 4.69 sq. ft. The trip wire correction at $v/\sqrt{gl} = 0.22$ was 0.045×10^{-3} (the 0.033″ trip wire) and 0.177×10^{-3} (for the 0.067″ trip wire).

APPENDIX 2

R. L. Towsin. *Boundary layer examination using a hot wire technique.*

Towards the conclusion of the experiments described in this paper, a successful pilot model of a hot wire instrument was developed which was intended for use as a turbulence detecting device. The simple circuit diagram and probe are shown in Fig. A. All the apparatus was kept at the control end of the towing tank, and connected to the hot wire probe by means of a light coaxial cable. The sensing wire itself was held between the ends of two prongs projecting from one end of a copper tube. By means of clamps on the tube, the hot wire could be inserted into the boundary layer at any point on a model's surface. If a current was passed through the sensing wire so that it became sufficiently hot, and if the wire was sufficiently fine, it responded very rapidly to the rate at which heat was taken from it. Thus, in a steady fluid flow the temperature of the wire and hence its resistance and hence the potential across it, in the circuit shown, remained steady. If, however, the flow was fluctuating, as was the case with turbulence, the potential followed the fluctuations. It remained to record the potential in order to examine the state of turbulence in the boundary layer. In these experiments, the fluctuations were watched continuously on a cathode ray oscilloscope and recorded selectively by a pen recorder. A specimen record is shown in Fig. B. The pilot model of the instrument has shown that with the substitution of a very thin sensing film sputtered onto the end of a probe, in place of the fragile wire, and by using a well designed universal head to position the probe,

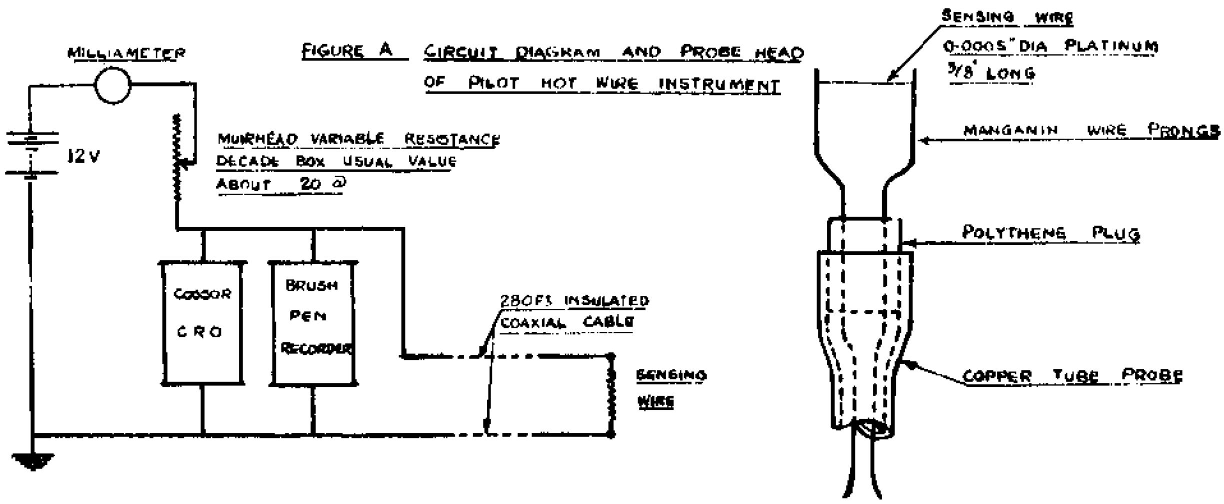

Fig. A

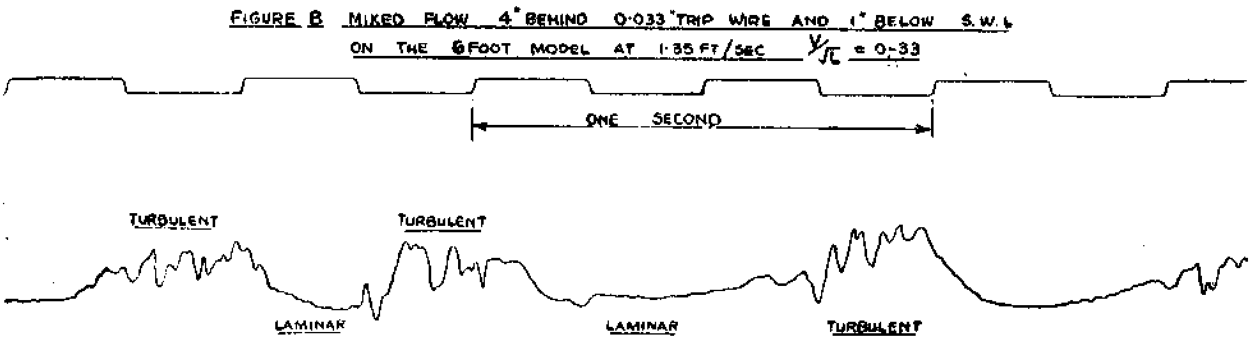

Fig. B.

a thorough examination of the boundary layer of a model can be made in less time and with less inconvenience than is the case with ink stream and chemical paint techniques.

It is here suggested that certainly with small models and preferably with all models the boundary layer flow should be examined with an instrument of the type described, as a matter of routine. The work of Schubauer, Skramstad and Klebanoff (Ref. 4 and others) on a flat plate in a wind tunnel and of Breslin and Macovsky (Ref. 5) on a tanker model, clearly shows how useful the hot wire technique can be. The actual use of the technique has convinced the writer that it is just as important to be certain of the nature of the flow over a model, as it is to be sure of its drag and that the hot wire instrument is the only method of gaining certitude. Techniques involving the emmision of an ink stream into the boundary layer are unsatisfactory in so far as they only define completely laminar paths, the ink disappearing in both fully turbulent and mixed flow regions; hence, for example, it is possible to draw the conclusion that a turbulence stimulator is fully effective because an ink stream disappears when it passes over it, whereas in fact there may be a considerable area of mixed flow behind the stimulator.

In addition to its use as a turbulence detecting device a hot wire instrument can be used to measure the amplitudes and frequencies of turbulence. For example, in the tests described here it was noticed that in these positions where the flow was turbulent without the trip wire the scale of the turbulence very considerably increased when the trip wire was put in position. This suggests the possibility that the flow behind a trip wire may not be the same as if it had started as a turbulent boundary layer from the leading edge without stimulation, as is usually implicitly assumed. The artificially increased violence of turbulence with the trip wire in position might be indicative of an artificially high turbulent drag behind trip wires.

References.

Schubauer, G. B., and Klebanoff, P. S.: "Contributions on the Mechanics of Boundary Layer Transition". Symposium on Boundary Layer Effects in Aerodynamic N. P. L., 1955, H. M. S. O.

Breslin, J. P., and Macovsky, M. S.: "Effects of Turbulence Stimulators on the Boundary Layer and Resistance of a Ship Model as Detected by Hot Wires". D. T. M. B. Rep. 724, 1950.

Dr. K. M. Kinoshita (Written contribution).

1. This report deals with the results of a series of specially planned standardization trials of four sister vessels, with perfectly same lines, powered with Diesel engines of the same type and the same power, built at the two different shipyards of the Hitachi Shipbuilding & Engineering Co., Ltd. The envisaged aim was to investigate the relation between waviness of the under-water shell plating and roughness coefficient Δc_f of the actual vessels.

2. Principal particulars of the four vessels are as follows:

Lp	138.000 m.
B	18.800 m.
D	11.850 m.
d	8.850 m.
C_b	0.734

Main Engine		Hitachi B & W 574 VTBF-160 × 1
Revolutions		115 r. p. m.
Normal Output		6,250 BHP
Propeller	dia.	5.350 m.
	pitch ratio	0.780
Percentage of welding used		95 %

3. For "A" and "B" ships among the four sister vessels, which were built at the "P" shipyard, the speed trials were made on the course of the "X" Island. For "C" and "D" ships built at the "Q" shipyard, the speed trials were made on the course of the "Y" Island.

4. The following items were measured at the sea trials.

1) Speed for ground (calculated from the measured time between mile posts).
2) Speed for water (measured by Shiba-log).
3) Revolutions per minute of the propeller.
4) B. H. P. of the main engine.
5) S. H. P. (measured by the two Hitachi Zosen type torsionmeters at two different positions on the intermediate shaft simultaneously).
6) Relative speed and direction of the wind. (measured electrically by the Koshin-Vane attached to the out-rigger of the foremast continuously).
7) The absolute value of the tidal current on the trial course during 24 hours including the time of the speed trial. ... for "B" and "D" ships only (recorded by Ono-currentmeter and Eckmann-currentmeters continuously).
8) The amount of the concave or the convex of the panels surrounded by the floors and longitudinals of the bottom plating due to welding (measured at the time of docking before the speed trial).

5. Sea conditions at the time of trials were very calm for the four vessels. Standardized results of the speed trials for the four vessels are shown in Fig. 1.

6. For "B" and "D" ships, the continuous measurement of the tidal current on the trial course was made. The component of the tidal speed on the trial course measured by the currentmeters, and the speed

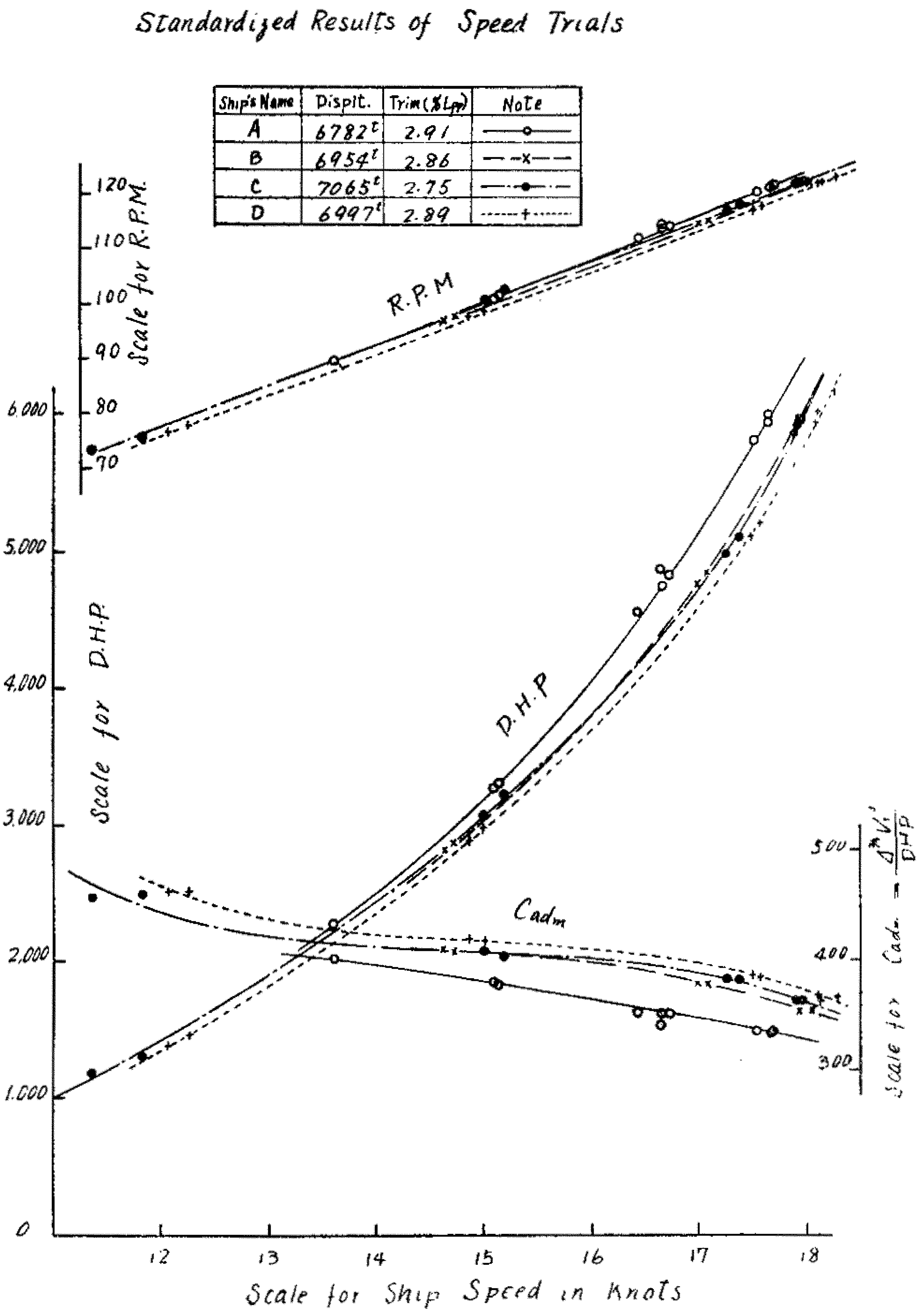
Standardized Results of Speed Trials
Ship's Name | Dispt. | Trim (%Lpp) | Note
A | 6782t | 2.91
B | 6954t | 2.86
C | 7065t | 2.75
D | 6997t | 2.89
Scale for R.P.M.
120
110
100
90
80
70
R.P.M
Scale for D.H.P
6,000
5,000
4,000
3,000
2,000
1,000
0
D.H.P
Cadm
500
400
300
Scale for Cadm = Δ^{2/3}V^3/DHP
12
13
14
15
16
17
18
Scale for Ship Speed in Knots

Fig. 1.

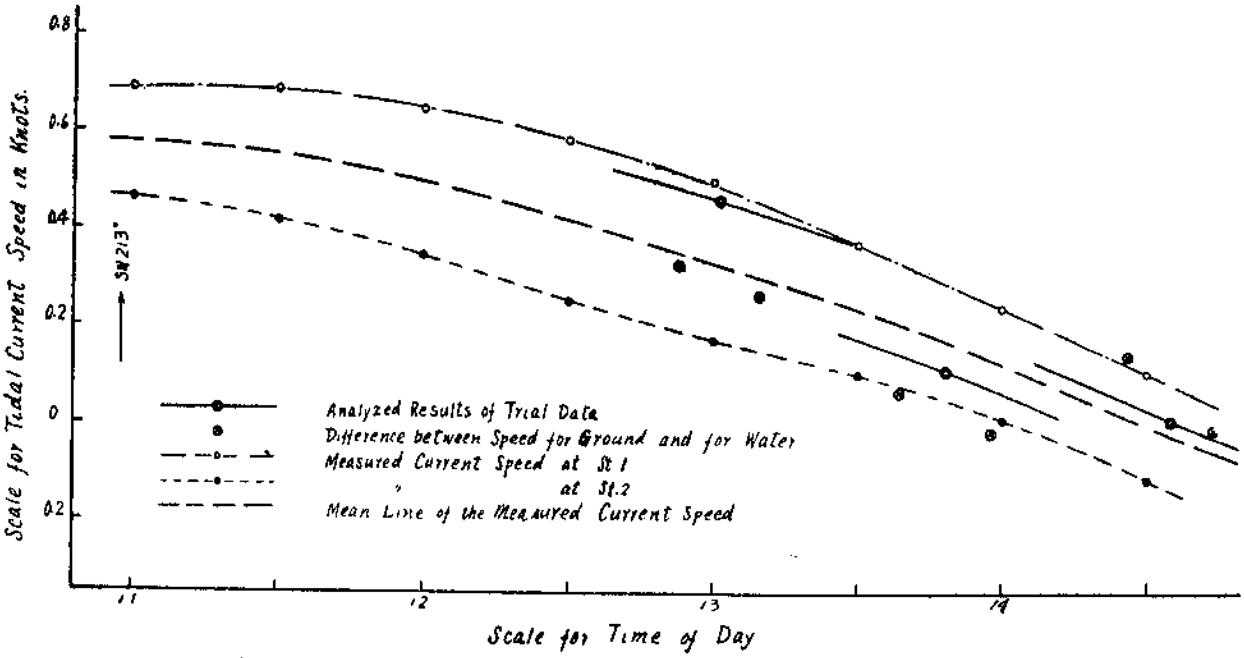

Fig. 2.—The Tide on Trial Course off the "X" Island March 11, 1957.

for water measured by the Shiba-log are compared with the calculated values from the standardization analysis respectively, as shown in Table 1, Figs. 2 and 3.

7. As the waviness of the under-water shell plating, the amount of the concave or the convex of the outer surface of the bottom plating at the centre of every panel surrounded by the transverse floors and the longitudinals was measured. As an example, measured results of the "D" vessel are shown in Fig. 4. Measured data were analysed as follows. All panels were divided into several groups, according to frame space and thickness of shell platings.

For each group with the same space and thickness,

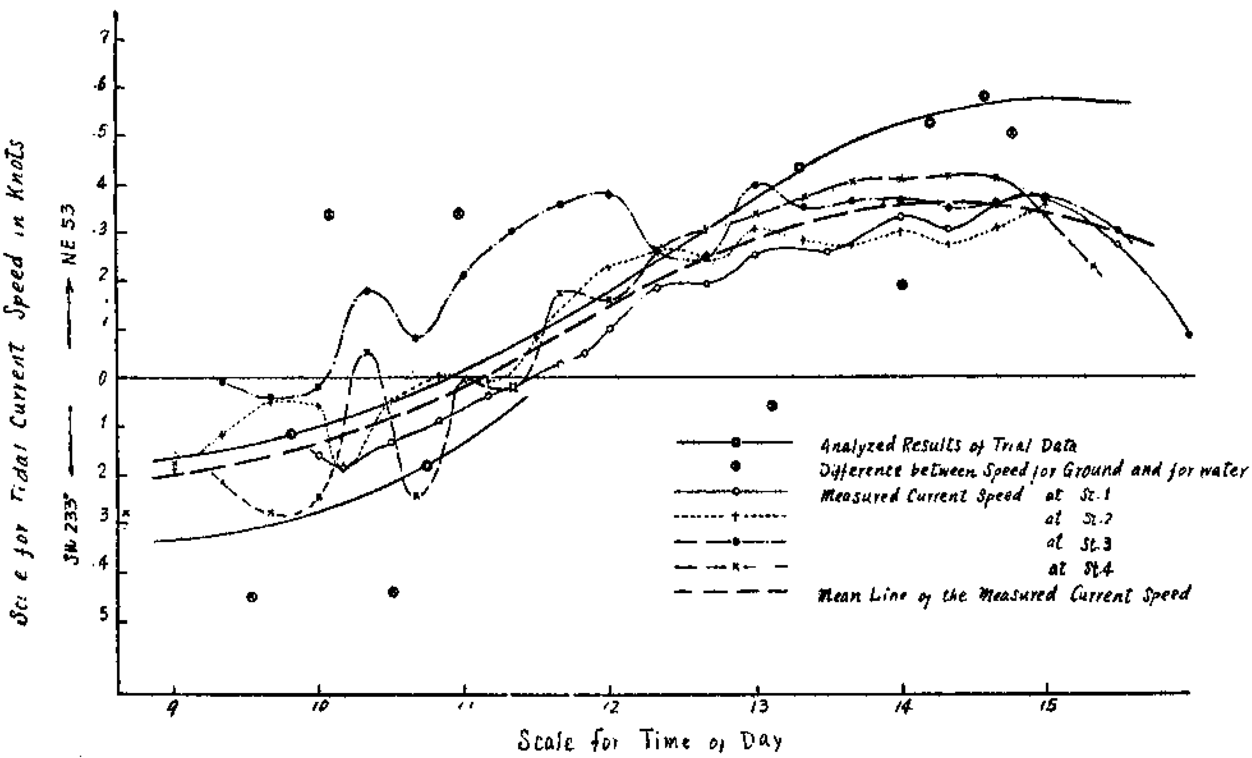

Fig. 3.—The Tide on Trial Course off the "Y" Island March 23, 1957.

TABLE 1

Comparison of Speed for no Wind and Still Water by each Method.

Name of Ship	No. of Run	Course	Ground Speed V_s	Standardizing Analysis: Correction for Wind ΔV_s	Standardizing Analysis: Analyzed Curr. Speed V_{sc}	Standardizing Analysis: V_{so}	Measuring Current: Mean Current Speed V'_{sc}	Measuring Current: V'_{so}	Shiba - log: Water Speed V''_s	Shiba - log: V''_{so}	Shiba - log: Current Speed $V''_{sc}=V_s-V''_s$	Comparison of each Results: $V_{so}-V_{so}'$	Comparison of each Results: $V_{so}-V_{so}''$	Comparison of each Results: $\frac{V_{so}-V_{so}'}{V_{so}}$ %	Comparison of each Results: $\frac{V_{so}-V_{so}''}{V_{so}}$ %
B	1	SW 213°	18.411	—	−0.480	17.93	−0.346	18.07	18.09	18.09	−0.32	−0.14	−0.16	−0.78	−0.89
	2	NE 33°	17.619	—	+0.430	18.05	−0.295	17.91	17.88	17.88	−0.26	+0.14	+0.17	+0.78	+0.94
	3	SW 213°	17.252	−0.024	−0.140	17.09	−0.202	17.03	17.19	17.17	−0.06	+0.06	−0.08	+0.35	−0.47
	4	NE 33°	16.845	+0.088	−0.067	17.00	−0.130	17.06	16.83	16.92	+0.02	−0.06	+0.08	−0.35	+0.47
	5	SW 213°	14.823	−0.037	−0.047	14.74	−0.010	14.78	14.68	14.64	−0.14	−0.04	+0.10	−0.27	+0.68
	6	NE 33°	14.580	+0.083	+0.029	14.63	+0.060	14.60	14.57	14.65	+0.01	+0.03	−0.02	+0.21	−0.14
D	1	SW 233°	12.52	−0.12	−0.14	12.26	−0.17	12.23	12.07	11.95	−0.45	+0.03	+0.31	+0.24	+2.53
	2	NE 53°	11.66	+0.32	−0.09	12.07	−0.13	12.11	11.32	11.64	+0.34	−0.04	+0.43	−0.33	+3.56
	3	SW 233°	15.24	−0.01	−0.22	15.01	−0.08	15.15	14.80	14.79	−0.44	−0.14	+0.22	−0.93	+1.47
	4	NE 53°	14.72	+0.02	−0.14	14.88	−0.03	14.77	14.38	14.40	+0.34	+0.11	+0.48	+0.74	+3.22
	5	SW 233°	17.25	−0.08	+0.40	17.57	+0.30	17.47	17.19	17.11	−0.06	+0.10	+0.46	+0.57	+2.62
	6	NE 53°	17.92	+0.04	+0.46	17.50	+0.33	17.63	16.95	16.99	+0.97	−0.13	+0.51	−0.74	+2.91
	7	SW 233°	17.77	—	+0.52	18.29	+0.36	18.13	17.96	17.96	+0.19	+0.16	+0.33	+0.87	+1.80
	8	NE 53°	18.65	—	+0.55	18.10	+0.36	18.29	17.84	17.84	+0.81	−0.19	+0.26	−1.05	+1.44
	9	SW 233°	17.55	—	+0.57	18.12	+0.35	17.90	18.05	18.05	+0.50	+0.22	+0.07	+1.21	+0.39

1. Speed is represented in Knots.
2. Correction for wind ΔV_s is also applied for other two cases.
3. In representation of current direction .(+) means NE, and (—) SW.

the frequency percentage were plotted on the base of the relative waviness, the ratio of the amount of the concave or the convex at the centre of the panel to the length of the frame spacing, as shown in Fig. 5.

The mean relative waviness Kn (n : 1, 2, ..., 5) of each group is defined by the following formula.

$$Kn = \frac{\Sigma\, am}{\Sigma\, m} = \frac{\Sigma\, am}{Mn}$$

where m denotes the percentage frequency of the case of relative waviness a is measured.

The total mean relative waviness K of each vessel is defined by the following formula,

$$K = \frac{\Sigma\, Kn\, Mn}{\Sigma\, Mn}$$

In the above summation, the groups with the thickness of 15.5 mm. and the frame space of 2.400 m and 3.200 m were omitted for the sake of simplicity as the panels belonging to their groups were few in number.

The results of the above-mentioned calculations are given in Table 2.

8. Roughness of the painted surface of each vessel could not be measured actually, but it seemed almost same for four vessels, because the same kind of oil paint and vinyl paint were used as the bottom coating and boot topping coating respectively for all vessels.

The number of days until the speed trials after last docking are shown as follows:

Ship name	days
"A"	22
"B"	13
"C"	9
"D"	7

9. Model experiments were made at the Transportation Technical Research Institute and the towing tank of Osaka Univ. at the strictly same conditions for the cases of sea trials of the four vessels.

Schoenherr C_f was used in calculating the frictional resistance of models.

10. The obtained C_f of the four actual vessels are shown in Fig. 6. Mean values of roughness coefficient ΔC_f to be added to the Schoenherr line are obtained as follows:

	"A" ship	"B" ship	"C" ship	"D" ship
ΔCf	0.00040	0.00018	0.00014	0.00010

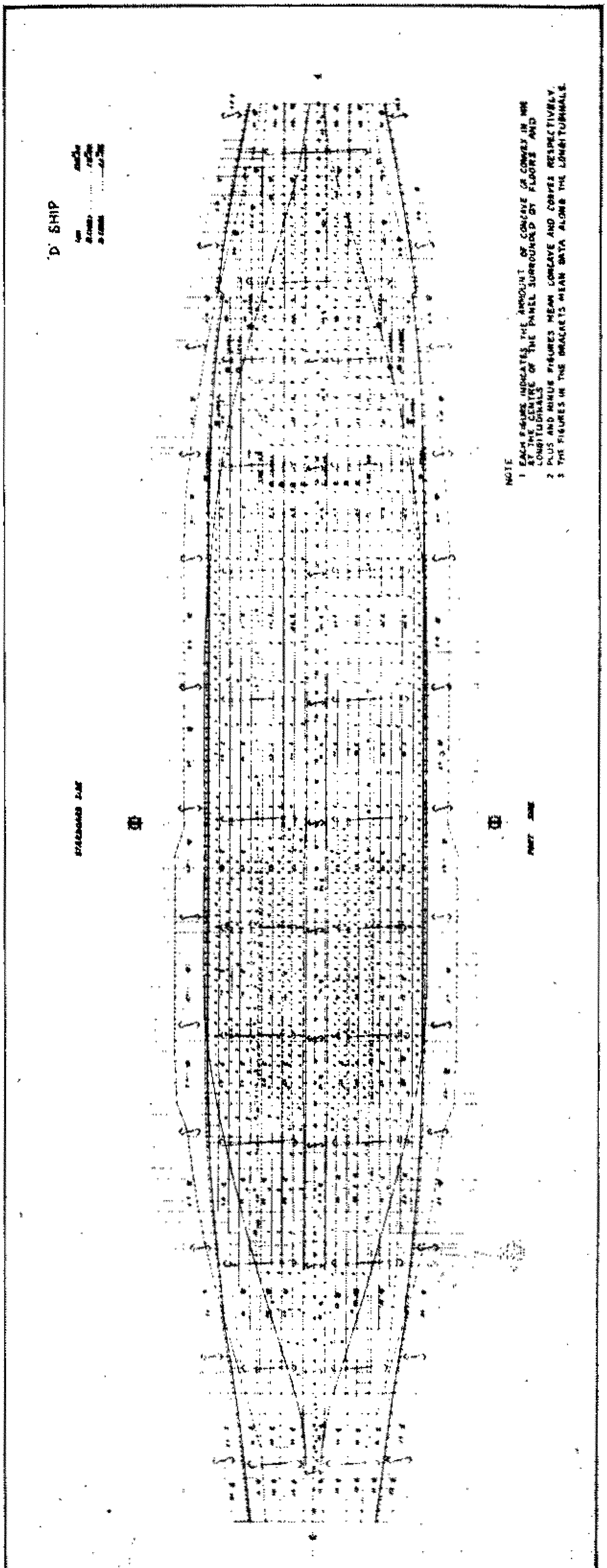
'D' SHIP
STARBOARD SIDE
PORT SIDE
NOTE
1 EACH FIGURE INDICATES THE AMOUNT OF CONCAVE OR CONVEX IN MM AT THE CENTRE OF THE PANEL SURROUNDED BY FLOORS AND LONGITUDINALS
2 PLUS AND MINUS FIGURES MEAN CONCAVE AND CONVEX RESPECTIVELY.
3 THE FIGURES IN THE BRACKETS MEAN DATA ALONG THE LONGITUDINALS

Fig. 4

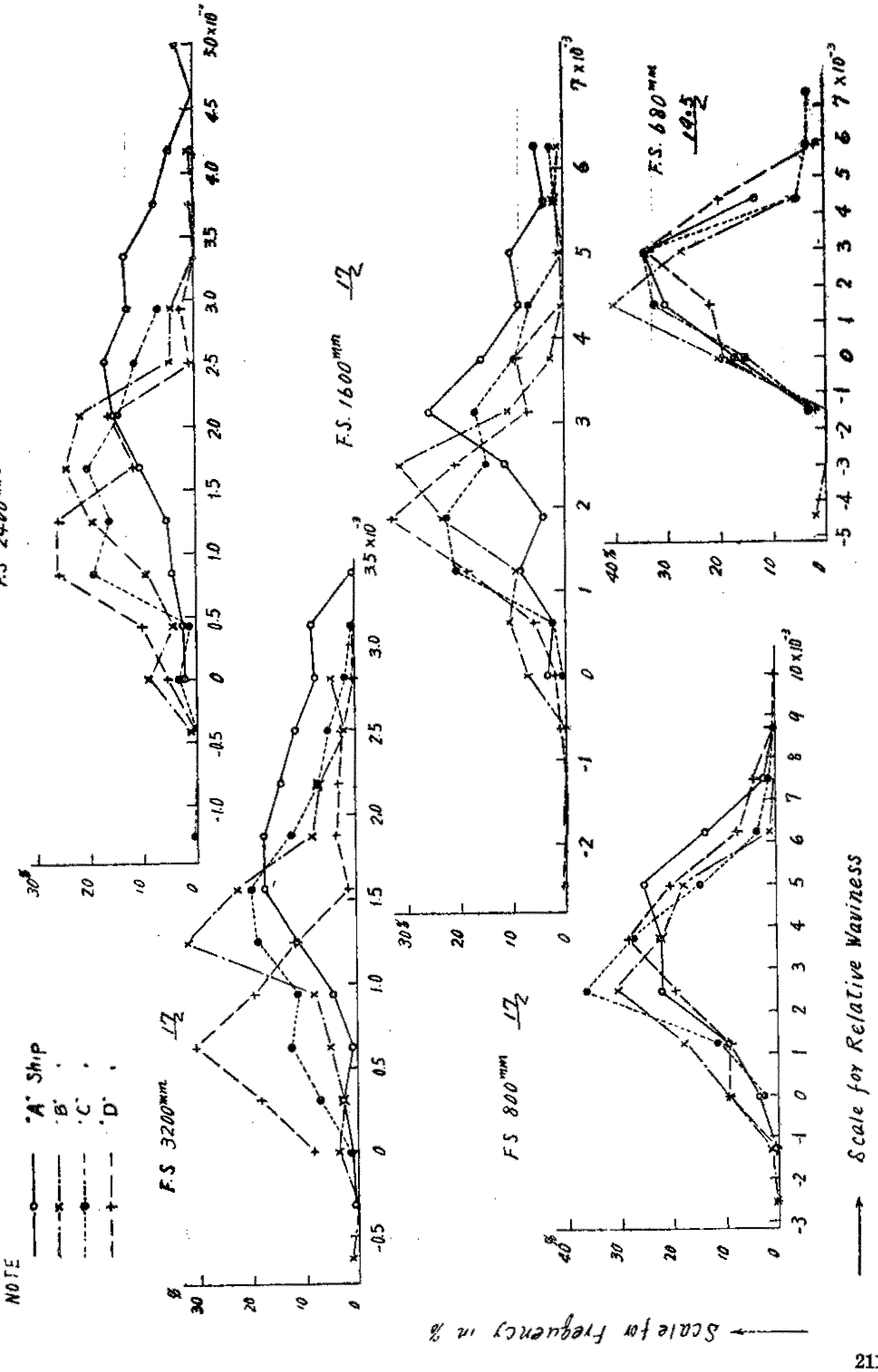
NOTE
'A' Ship
'B'
'C'
'D'
F.S 2400mm
F.S. 1600mm
F.S 3200mm
F.S 800mm
F.S. 680mm
Scale for Relative Waviness
Scale for Frequency in %

Fig. 5.

Table 2.—MEAN RELATIVE WAVINESS

Group / Ship's Name	1 F. S. 3.200 $t = 17$ K_1	2 F. S. 2.400 $t = 17$ K_2	3 F. S. 3.600 $t = 17$ K_3	4 F. S. 800 $t = 17$ K_4	5 F. S. 680 $t = 19.5$ K_5	Total mean rel. waviness K
A	0.00191	0.00250	0.00330	0.00383	0.00201	0.00311
B	0.00136	0.00149	0.00208	0.00267	0.00174	0.00214
C	0.00136	0.00164	0.00245	0.00329	0.00220	0.00250
D	0.00080	0.00123	0.00198	0.00371	0.00229	0.00252

11. The frictional coefficients of the four vessels were plotted in the group of curves in Fig. 7, which were obtained by Nikuradse for the various values of equivalent sand roughness. The values of equivalent sand roughness (Ka/L) of the four vessels which were obtained from the results plotted in Fig. 7 are as follows:

	"A" ship	"B" ship	"C" ship	"D" ship
$Ka/L =$	5.00×10^{-7}	2.86×10^{-7}	2.50×10^{-7}	2.00×10^{-7}

12. The effect of total mean relative waviness K on the values of ΔC_f and the values of equivalent sand roughness Ka/L are summerized in Fig. 8.

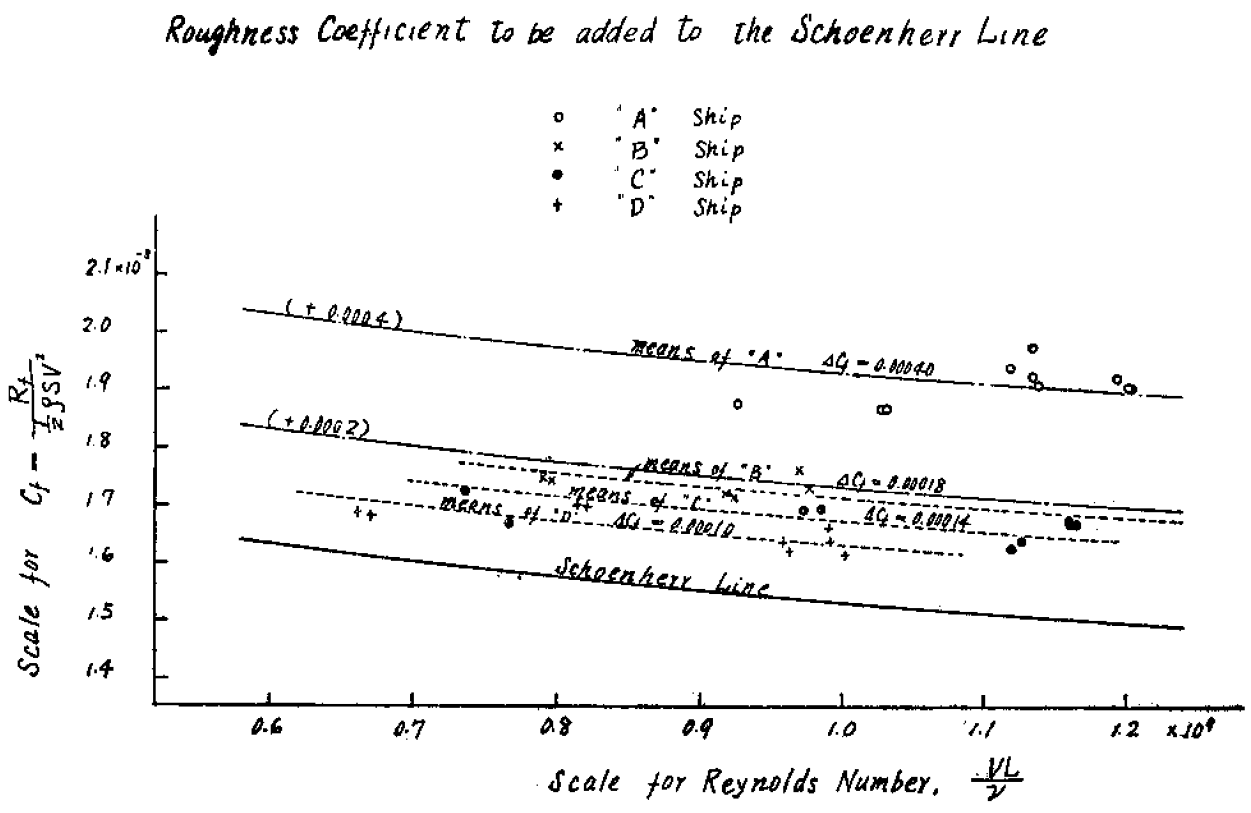

Fig. 6.

Correlation of Roughness Allowances with Equivalent Sand Roughness

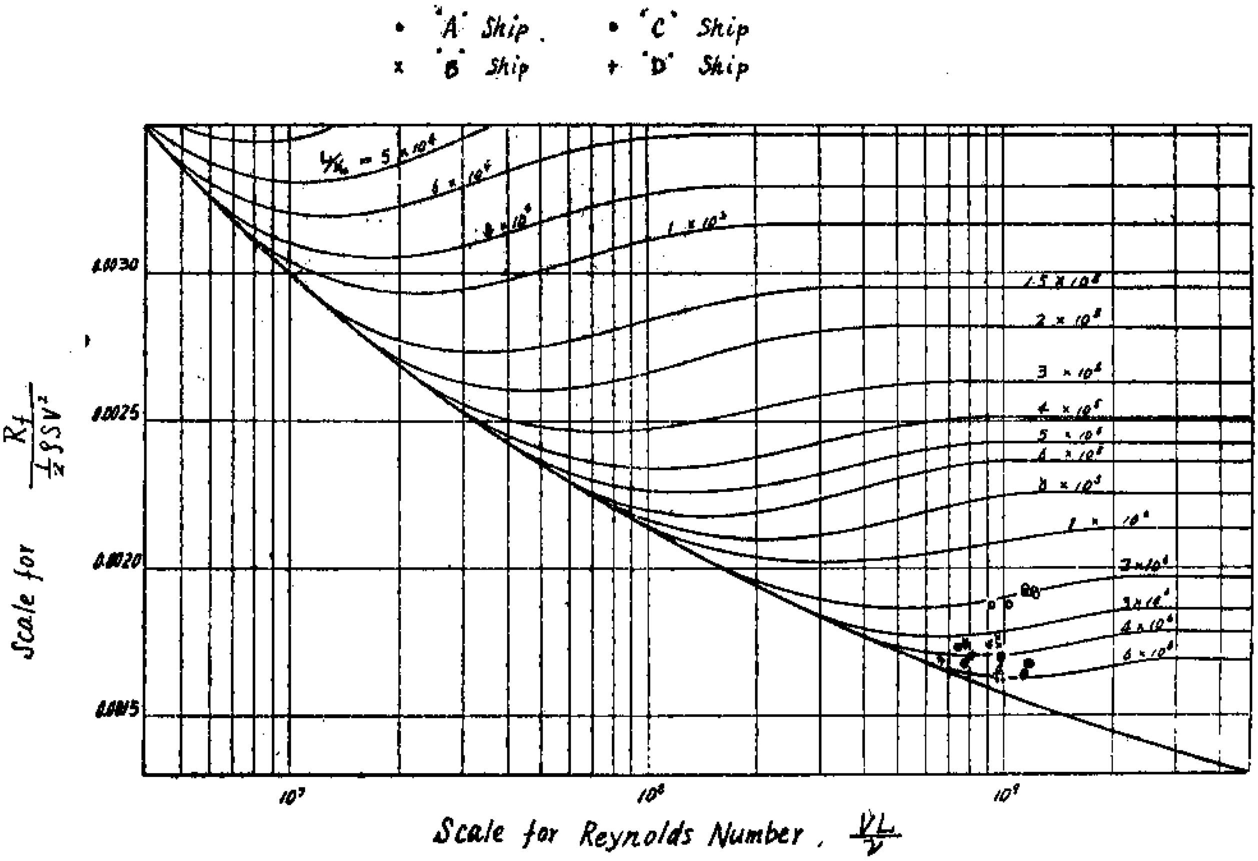

Fig. 7.

Correlation of Relative Waviness, Schoenherr ΔC_f and Equivalent Sand Roughness

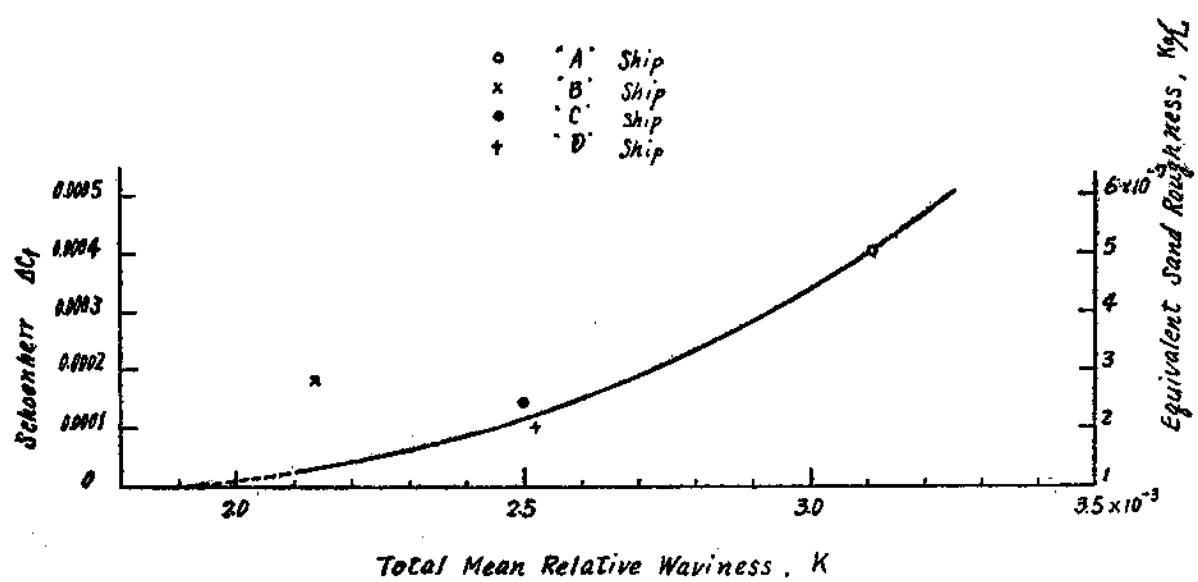

Fig. 8.

Dr. K. M. Kinoshita (Oral remarks).

The only point I wish to make is that, as far as the sister vessels at the same trial condition are concerned, the difference between the values of the so-called roughness coefficients Δc_f of the two sister vessels, depends only upon the conditions of the surface of the underwater shell plating, and never depends upon the formula of frictional resistance of hydrodynamically smooth surface applied. The absolute values of Δc_f may depend entirely upon the frictional formula, but the difference between the two values of the sister vessels does not change.

So, in Fig. 8 on the last page of my report, the abscissa can change its position perpendicularly to itself, according to the frictional formula for the hydrodynamically smooth flat surface used. If we neglect the effect of rivet seams, still remain in few lines, and furthermore, if we assume that the newly painted flat surface is nearly equal in its frictional character to the hydrodynamically smooth surface, Δc_f becomes zero at $K \sim 1.9 \times 10^{-3}$, as far as the present Schoenherr line is used.

Roughly speaking, it means that the limiting values δ of the amount of concave or convex at centre of the panel for the bottom plating of a vessel become as follows:

Frame space (f. s.) mm	δ for $\Delta c_f = 0$ at $R_n \sim 10^9$ mm
680	1.3
800	1.5
1000	3.0
2400	4.5
3200	6.0

Our scientific common sense will, of course, deny these values, as they seem to be too large.

I think that the abscissa in Fig. 8 must be lowered down to the acceptable position, by making some proper correction to the present Schoenherr line, with the steepening tendency as in the Proposal I or II of the Committee.

This is only a result of a tentative analysis which has been made to find out the correlation between the relative values of Δc_f and the surface conditions of the actual welded vessels.

And I know, of course, that the data obtained and shown in Fig. 8 are too small in number to deduce at once a definite conclusion.

A large number of supertankers and high speed cargo vessels are now under construction in Japan, and their sea trials are being carried out now, and will be carried out in the near future. I have a plan to continue my present work for some of these vessels. Many data of actual vessels for high Reynolds numbers above 10^9, will be obtained in the next few years.

So, the Towing Tank Committee of Japan is of the opinion (sorry to say, just the same as at the Scandinavian Conference) that it is not yet desirable to attempt to reach any decision at this Conference, as to a lasting skin friction formulation.

Prof. T. Y. Wu, Mr. M. S. Plesset (*) (Written contribution).

We wish to express our comments on the subject from the viewpoint of making efforts along a new approach for determining the total ship resistance rather than seeking possible improvements of the method based on the Froude assumption or choosing a skin-friction line which would be satisfactory to everyone. Although this attempt may sound slightly too ambitious, it is our hope that this approach toward solving the problem would lead to a more systematic investigation by eliminating, one by one, the relevant physical factors whose direct or indirect effects on the value of ship resistance are so far not well understood. At least it is hoped that a better understanding can be achieved the physical problem as a whole.

According to the present conventional practice in naval technology, the total ship resistance coefficient C_T is decomposed into the following components

$$C_T = C_F + C_V + C_W \qquad [1]$$

where C_F is taken to be the fully turbulent skin frictional resistance of an equivalent plank in two-dimensional flow, C_V is defined to be the form drag due to the ship-hull form, determined by the Hughes "low Froude number run in point", and C_W is the wave-making resistance. Under Froude's assumption, C_F is a function of the Reynolds number R_n $(= VL/\nu)$ alone and $(C_V + C_W)$ depends only on the Froude number F_n $(= V/\sqrt{gL})$; the scaling between the model and a geometrically similar (geosim) ship is then carried out according to this rule. In the final application of the test results by using a "smooth" model for the prediction of the ship resistance, the "roughness" of the actual ship hull is accounted for by introducing an aditional term ΔC_F, the socalled "roughness allowance coefficient". That Froude's assumption is crude is quite obvious for the following reasons.

(i) Without using a purposely installed device, such as a trip wire, the boundary layer over the hull in general starts with a laminar flow from the bow, then becomes fully turbulent after a transition stage which takes place in a region with the local R_n lying approximately in the range 5×10^5 and 3×10^6

(*) From California Institute of Technology, Pasadena, not in attendance.

provided the residual turbulence in the free stream is reasonably low. The skin friction in these two regimes is governed by different functions of R_n. The assumption of taking the boundary layer wholly turbulent is perhaps not serious in practice, for example, for ship R_n greater than 10^7. However, it is important to note that, when the correlation between the model and a geosim ship is carried out at the same F_n, as is usually done in practice, the ratio of the length of the laminar region to the hull length varies with R_n because the R_n for transition is certainly a fixed number. This effects is apparently not taken into account in the C_F as defined above.

(ii) In the above decomposition, the effect of the hull's curvature, both in the longitudinal (from bow to stern) and tranverse directions, on the skin resistance is neglected. This three-dimensional curvature effect, which again depends largely on R_n, may indeed be small, but may not be negligible. Although this effect may be included in C_V, its dependence on R_n is nevertheless lost in Froude's assumption.

(iii) If the above decomposition is expected to represent the actual value of ship resistance, then admittedly C_V is assigned a very complicated role, for it must include such components as those due to the correction factor caused by laminar-turbulent transition, to the aditional friction caused by curvature effects, to eddy-making resistance caused by separation of the flow, and to the "interaction" between the viscous effect and the gravity effect. On the other hand, either Froude's assumption or Hughes' assumption, which is to scale C_V directly with R_n, would narrow the capacity of the role of C_V down to a very limited one. The conjecture that all the above effects may be properly represented by the one parameter C_V after such one-sided scaling (i. e., scaling C_V with either F_n or R_n seems undoubtedly too optimistic. The incapability of representing these effects by one parameter C_V may well provide some clue as to the lack of good agreement between measurements of resistance of a model made at different times or places. Incidently, it also happened in the early days of aeronautical researches that inconsistent drag measurements were reported from different laboratories before the separation phenomenon was made well known.

(iv) It is true that when correlation of the model and the ship is made at the same F_n, and hence $(C_V + C_W)$ remains unaltered under Froude's assumption, the full scale friction resistance is equal to the C_F of the model multiplied by a scale factor n^{-a} (with n = shipscale / modelscale and $0< a <2$) which is less than unity for $n >1$. Thus the prediction of C_F of the ship will not be affected too critically by the above decomposition. This perhaps gives one reason for the practical success of Froude's assumption, despite its physical incorrectness. But when a higher accuracy of prediction is required, it seems more fruitful to consider this whole problem from another viewpoint rather than to choose a universal friction line to accommodate all other effects.

Following this reasoning, we would like to propose a new method which emphasizes the measurement of some detailed information of the flow instead of the conventional practice of measuring the total resistance. It may be pointed out here that in this method the skin friction of an equivalent plank in two-dimensional flow still plays a significant role, except that now the transition phenomenon and the curvature effect are included in the definition of the friction resistance. Let us write

$$C_T = C_F{}^* + C_V{}^* + C_W + C_i \qquad [2]$$

in which the diferrent terms will be made specific below.

The quantity $C_F{}^*$ is taken to be the actual friction resistance on the model and may be determined as follows. The effect of the longitudinal curvature is calculated by using E. Truckenbrodt's method [1]. This method, as already applied in aerodynamics, uses essentially the energy integral instead of the momentum integral in determining the momentum thickness (see, e. g., also Ref. 2, Ch. 22). The measured quantity in the experiments is the pressure distribution, $p(x)$, over the hull, from which the velocity distribution of the ideal, potential flow past the edge of the boundary layer, $V(x)$, can be evaluated from the Bernouilli equation. This process has the advantage of omitting the possible inaccuracy in the theoretical pressure distribution. With both the curvature effect and the transition phenomenon taken into account, the given quantities ($V(x)$ and R_n then enable one to calculate $C_F{}^*$ in terms of the turbulent skin friction coefficient for an equivalent plank, C_F. The effect of the lateral curvature may be derived from a mathematical investigation by C. B. Millikan [3] in which the boundary layer theories of Prandtl and von Karman have been extended to apply to bodies of revolution. The analysis involves an approximate solution to the fundamental differential equation and some plausible assumptions for the turbulent portion of the boundary layer. According to Millikan's theory, the skin friction for a cigar shaped model should have been from 5 % to 6 % in excess of the flat plate value.

The second term $C_V{}^*$ is the actual form drag caused by the viscous effect, whether or not the flow is separated to form wakes. The rise of form drag in the presence of flow separation is easy to see. Even in the absence of flow separation the form drag should still be attributed to the skin friction; the reason for the existence of drag in this case lies in the fact that the boundary layer displaces the external potential flow by a distance equal to the displacement thickness and the pressure distribution around the hull is thereby changed even in the absence of sepration. With flow separation, C_V* then includes also the resistance due to the formation of the eddy wake. In the latter case, the problem presumably may be solved with the use of free streamline theory. This presents indeed another research problem. However, it is believed that in the general case of well-streamlined hull, the resistance due to eddy making is probable very small. It may be noted that C_V* defined here is quite different from C_V given previously.

The third term C_W is the resistance of wave making, whose definition is actually very clear after C_F* and C_V* are made explicit. For a theoretical counter check C_W may be calculated by using Michell's thin ship theory with the hull form now being given by the actual hull plus the displacement thickness of the boundary layer and the wake, if any.

The last term in (2), C_I, stands for the resistance due to the "interaction" between the viscous (R_n) and the gravity (F_n) effect. In order to help us to understand more of this interaction, we wish to propose a problem for further research, which would be of academic as well as practical interest. Consider a plank, being towed, as in Froude's original experiment, and running into a train of free surface waves. The problem is that of investigating the interaction between the gravity wave and the boundary layer flow. This problem is in a way parallel to the problem of the shock wave and boundary layer interaction in compressible fluid mechanics.

In short, we are of the opinion that before a better understanding of all these physical factors is obtained, the time is perhaps not ripe enough in the present state of the art to endorse a single friction line for universal adoption because it is intrinsically an open question whether any such friction line is able to cover all the general cases. However, we are quite confident that with the combined efforts of all of us, the picture will become brighter and perhaps some day, a satisfactory solution would come into existence and would be automatically acceptable to everyone.

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Dr. G. Hughes (Written contribution).

I wish first to make some comments on the results of the new Victory model work by Lindgren and Bjärne contained in the SSPA publication No. 40 (1).

The results in Fig. 4 suggest that in this work the trip wire was generally effective from fairly low values of Froude number. The authors did not apply any corrections for the trip wire resistance nor for the laminar flow area, but the relative values of the resistance coefficients for the differente model scales are not significantly affected thereby at low Froude number. In Fig. 10 a comparison of the low Froude number results with three assumed viscous lines is shown, and the authors state that all of these lines appear to be too flat. The steepest of the three lines corresponds to $r = 1.27$ [r = ratio to the basic line $0.066 (\log R_n - 2.03)^{-2}$]. This feature is perhaps better illustrated in Fig. 1a herewith, which is a reproduction of part of Fig. 7 with the line $r = 1.27$ added. This shows that this line is in approximate "position" agreement with the results for 7 to 8 knots but is clearly not quite steep enough for good correlation. It will be appreciated that any corrections for blockage effect to these results will tend to steepen the correlation further.

In Appendix 2 the authors have applied the method recently developed at NPL (2) for finding the correct correlating slope (which makes use of the results of tests in tanks of different size) to a comparison of the SSPA results and those obtained in NPL n. 2 tank (3). For this purpose they have corrected all the SSPA results for trip wire resistance and for laminar flow area on the assumption that the flow was laminar ahead of the trip wire position. They conclude (Fig. 20) that a mean value of $k = 0.12$, or $r = 1.12$, is required. On the basis of the analysis this appears to be a fair result. If correct, it conflicts seriously with the indications from the low Froude number results, for which the value of r for slope correlation is > 1.27. These two results are mutually acceptable only if there is a peculiar departure from Froude's Law of Comparison for the wave resistance or if there is a considerable interaction effect between the viscous and wave components. No evidence of either such effect has been observed in other detailed geosim analyses (4). However, the anomaly is removed by the following explanation.

From the Victory model analyses carried out at NPL (3) it was suspected that turbulent flow began naturally ahead of the trip wire position in the NSMB tests with the larger models (scales 1/18, 1/23 and 1/30). This suspicion was confirmed for the model of scale 1/18 by special NPL tests in which the effects of studs and trip wire were directly compared. However, because of the uncertainty on this matter for scale 1/23 and 1/30 the full application of the "tank difference" method of analysis was not made to the combined NPL and NSMB results, but it was concluded (4) that if the results for the three NSMB models in question were all corrected on the assumption that turbulent flow had begun naturally ahead of the trip wire position, a value of r close to 1.27 would have resulted from the slope analysis. In the new SSPA work there appears to be independent confirmation of the correctness of this assumption. This is in Fig. 4 of the SSPA publication (1). The relevant curves are reproduced in Fig. 1b herewith, and for comparison some NPL results for tests in which studs or plates were used are shown in Fig. 1c. The curves in Fig. 1b each show a peculiar "step" with little difference in slope on either side of the step. These steps are not due to wave resistance, since if so they should appear in Fig. 1c also; moreover, the Froude number at which they appear progressively changes with the model scale. The explanation offered is that ahead of the trip wire the flow was laminar below the step, and turbulent above it. The step represents the extra resistance due to the difference between laminar and turbulent on part of the area ahead of the trip wire and due to the greater resistance of the trip wire in the turbulent boundary layer. In Fig. 1c there is no evidence of any definite "step", though there is some indication of a gradual slight change of "position" in the same speed range. This is consistent with a gradual reduction of the laminar flow area with increase of speed but without any change of the resistance of the studs or plates, these being placed near to the bow profile so that it is unlikely that turbulent flow would occur ahead of them. It is of interest also to note that these "steps" were not consistently in evidence in the NSMB results. They show in the SSPA results apparently because the conditions of test (such as degree of turbulence in the tank water, or finish of the models at the bow profile) favoured easier or earlier transition to turbulent flow. This view is supported by Fig. 1a herewith; the steepness of slope at low Froude number shown here was not obtained in the NSMB results.

The average "step" in Fig. 1a is about $\Delta C_t = 0.75 \times 10^{-4}$, or approximately 2 % of the average model C_t. This agrees closely with the corresponding difference deduced from the NPL direct comparison of results with trip wire and with studs referred to previously (3), further supporting the explanation given above. As already noted, for the analysis of the combined SSPA, and NPL results given in Appendix 2 and Figs. 17 to 20 of the SSPA publication (1), the SSPA results were corrected on the assumption of laminar flow ahead of the trip wire. From the above it seems that the correction should be at least 0.75×10^{-4} less than was assumed. The effect of the adjusted correction on Fig. 20 is seen in Fig. 1d herewith. The mean value of k is increased from 0.12 to 0.28, or r from 1.12 to 1.28. This result is now entirely consistent with the low Froude number correlation (Fig. 1a). Therefore it seems highly probable that the true mean correlation slope for the Victory series is rather greater than given by $r = 1.27$ as was previously deduced (3, 4). It may be noted also that to agree with the above deductions the low Froude number results in Fig. 1a (SSPA Fig. 7 (1)) should be corrected by about $+ 0.60 \times \times 10^{-4}$ (see previous paper (3)). The mean "position" value of r for low Froude number would then be about 1.29.

Table I is reproduced from the recent paper (4) read in July at the INA summer meetings. It shows the results of the several geosim analyses to which reference is made above. Except for the Victory series the values of r given in Col. 4 for the mean slope correlation were obtained from the detailed "tank difference" method of analysis. For the Victory series the values of r were obtained from a more general analysis and were included in the table as being probably about right. It is now shown as above from the SSPA results that the inclusion of these values for the Victory series was justified, and that the Victory series now falls firmly "into line" with the other geosim analyses. The combined evidence from the various geosim analyses is that in all cases the correct correlating slope for the range of the data is not less than and usually a little greater than given by plane friction formulation when applied at the low Froude position of the specific resistance curve.

On the basis of the above evidence it is believed that a practical solution to the problem of model correlation and the prediction of smooth ship resistance is provided by the use of the plane friction formulation itself. The argument in favour of a "three-dimensional" solution and for this particular solution is fully set out in the July paper. The reasons for this preference may be reproduced here: (i) The geosim analyses show that for all practical purposes the same correlating slope may be used at all values of Froude number; (ii) at low Froude number where the resistance is entirely viscous the correct correlating line must agree in position with the model resistance curve, hence from (i) this same line is applicable at all values of Froude number; (iii) the plane friction formulation provides a variation of slope with position which is generally consistent with a large amount of data, and, while not giving precisely the correct correlation for all models, is likely to provide an acceptable differential for form effect; (iv) the conception that form effect should be a constant percentage independent of

Reynolds number has some theoretical support (e. g. the work of Young and of Scholz); (v) it is important for the future study of the effect of form on resistance that the breakdown of the total resistance into the viscous and wave components should be *quantitatively* correct, and this can only be so if the division is made at the position of the resistance curve at low Froude number.

A three-dimensional solution is undoubtedly a more complete solution than a "single-line" formulation. On the other hand, it will not be disputed that a "single-line" formulation can serve very well for the purpose of correlating model results at different scales and for ship predictions. The chief drawback to a single-line solution is that it does not fit in with condition (v) above. Therefore, if a single-line formulation is adopted, in order to make analyses of viscous and wave resistance "conform" to this formulation and still be of the right order quantitatively, it will be necessary to add or subtract a constant intercept to or from the wave component to bring its value to zero at low Froude number. The value of this intercept will vary with each model. This process amounts to admitting the importance of the three-dimensional effect in one section of the work while circumventing it in the other. This cannot be regarded as a happy solution.

However, if this Conference decides to adopt a single-line formulation it would appear from the geosim analyses that this should be of not less average slope than represented by $r = 1.20$ in the formulation $r \times 0.066 (\log R_n - 2.03)^{-2}$. This lower "limit" of slope value is also supported by the analysis which was made at NPL of the geosim data collected by TMB for the Skin Friction Committee. This analysis is referred to near the end of Appendix I of the Skin Friction Committee's Report. It has not been considered necessary to put the details before this Conference because this analysis is of only limited value due to many results being affected by laminar flow and blockage to largely unknown extents. The composite plots referred to are shown in Figs. 7 and 8 of Dr. Todd's formal discussion. It is easy to show that the *average* correct viscous slope for all the series included in such plots cannot be less than given by the *upper* envelope of the plot. Though not well-defined the upper envelope of Dr. Todd's Fig. 7 corresponds to a value of r not less than 1.30, and for Fig. 8 to between 1.20 and 1.30. Thus from a somewhat crude analysis it is seen that there is broad agreement with the findings from the more precise analyses of which the results are given in Table 1 of the present discussion.

The two proposals of the Skin Friction Committe are both lower in "position" than the line $r = 1.20$, while in slope they may be compared by the values of r which give the same "over-all" slope or the same reduction of ordinate, for various ranges of log R_n, as follows:

CORRESPONDING VALUE OF r FOR SAME "OVER-ALL" SLOPE

Range of log R_n	Committee's First Proposal	Committee's Second Proposal
6.0 to 7.0	1.237	1.237
7.0 to 9.0	1.07	1.145
6.0 to 9.0	1.16	1.195

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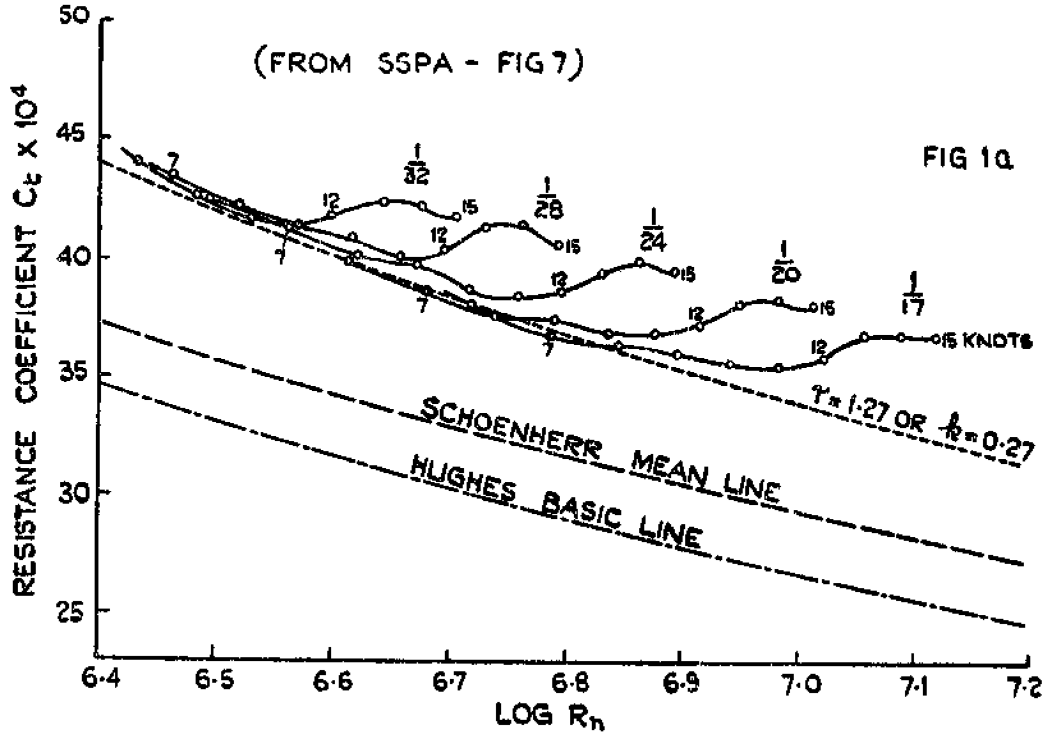
Table 1. VALUES OF r OBTAINED FROM GEOSIM ANALYSES

Series	Block coefficient at deepest draught	Draught	Value of r for mean slope correlation (a)	Maximum F_n	Range of log R_n	Value of r for position at low values F_n (b)	(a)/(b)
Victory	0.688	Load Light	1.27 1.27	0.27	6.0 to 7.2	1.27 1.27	1.0 1.0
Fast steamer	0.546	I II	1.27 1.34	0.34	6.1 to 7.5	1.18 1.21	1.08 1.11
Cruiser	0.520	I II	1.24 1.33	0.48	6.0 to 7.7	1.21 1.20	1.03 1.11
0.75 Block coefficient	0.748	LWL WL I WL II	1.40 1.36 1.35	0.21	6.2 to 7.1	1.30 1.28 1.26	1.08 1.06 1.07



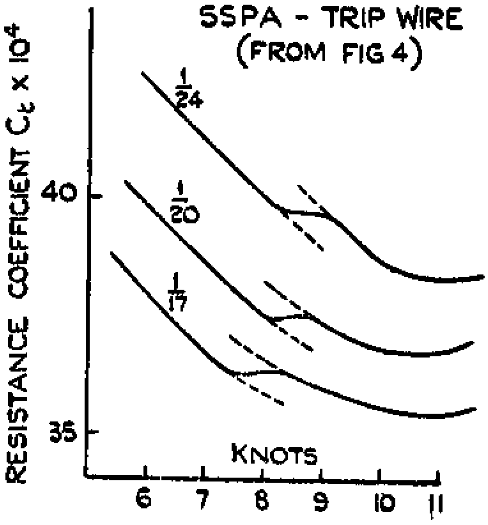


FIG 1b

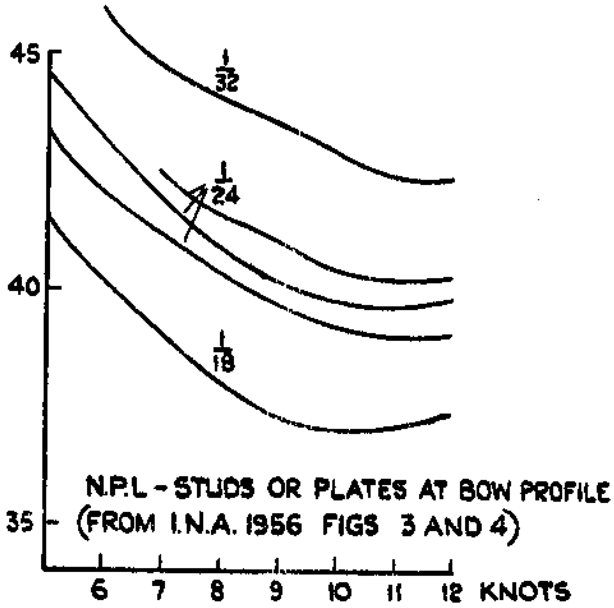


FIG 1c

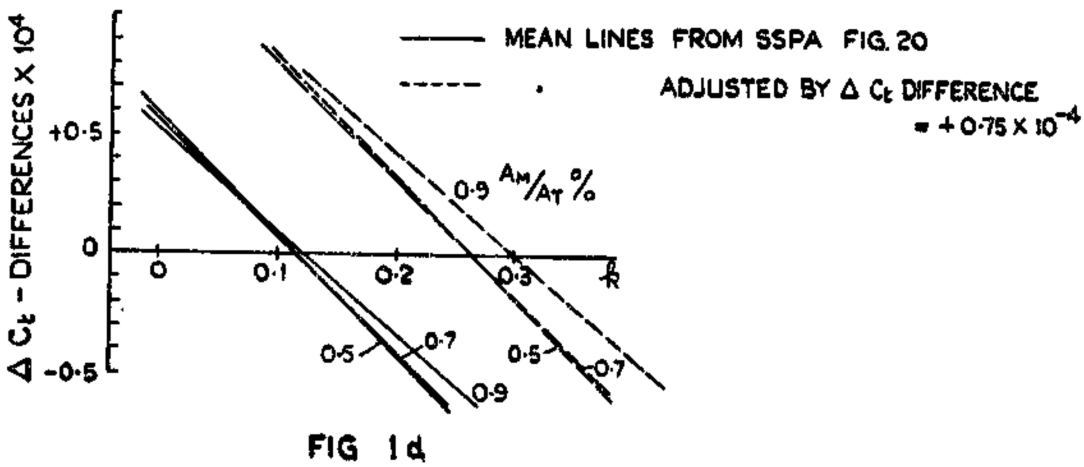


FIG 1d

Dr. G. Hughes (Additional statement).

The foregoing contribution in my name was prepared some months ago (*) with the object of presenting to the Conference the latest position so far as the NPL data and analyses are concerned and to give a considered opinion on this subject. It will be appreciated that this is an individual and personal opinion.

Since that time the NPL has suffered a most severe loss by the death of Dr. Allan, and it has fallen to me to state the official position of NPL concerning the proposals of the Skin Friction Committee.

Two meetings of the British delegates have been held this year, one in February at which Dr. Allan presided, and one in July after his death. As a result of these discussions it may be said that the British delegates are unanimously prepared to support the adoption of an agreed single line skin friction formulation, for the purpose of international exchange of data and development in this field. It may also be said that, of the two lines suggested by the Skin Friction Committee in Recommendation No. 3 they collectively prefer the first proposal or any other approximating to it on which agreement could be reached. It should be placed on record that Dr. Allan supported this point of view.

If the Skin Friction Committee's first proposal is adhered to rigorously it is practically impossible to find a simple formula to fit this proposal over the whole of the model and ship range. Mr. Newton points out the desirability of having a simple formula and his proposal is welcomed as practical and realistic. It is preferable to express his formula explicitly as

$$C_f = \frac{0.075}{(\log R_n - 2)^2}$$

His table shows that this formula gives an excellent mean of the Committee's proposal. Moreover it is a line of smoothly changing curvature, and since it fits in closely (**) with the plane friction formulation

$$C_f = \frac{r \times 0.066}{(\log R_n - 2.03)^2}$$

it gives a line which, *at its own level or position*, fits the *slope* of modern data very well.

(*) An early copy of the SSPA paper was sent to NPL by Dr. Edstrand before its general release.

(**) The difference due to the change of coefficient from 2.0 to 2.03 is very small.

The value of r to give agreement with Mr. Newton's formula is closely 1.13. The Skin Friction Committee's second proposal is also closely represented in mean *slope*, but not in *position*, by a value $r \div 1.20$. It may be said, therefore, that the Committee's proposals, so far as the effect on model to ship correlation and prediction is concerned, are both embraced within the range of formulae covered by $r \div 1.13$ to 1.20, or

$$C_f = \frac{0.075 \text{ to } 0.080}{(\log R_n - 2)^2}$$

To conclude, while a personal preference, in relation to a single line formulation, is, as previously stated, for r to be not less than 1.20, NPL is prepared to support, for the purpose of international exchange of data, any line which has a simple formula within the above range, or which is of close similarity to the above.

Dr. G. Hughes (Oral remarks).

Dr. Todd has already expressed the opinion of the Ship Division as to the agreement that should be reached regarding the adoption of a line. I can say, as regards researches made during the last six years, that we had a large amount of data which when put together showed a surprising amount of agreement on the general slope of the correlation over a very large range. I would personally like to see that if a single line is adopted by the Conference, it would be something that fits in with the slope that has been shown.

Prof. W. P. A. van Lammeren (Oral remarks).

Briefly, the opinion of the Netherland's ship model basin may be stated as follows:

1. We are in favour of a theoretically supported plate line.
2. Though it is definitely not an ideal solution from a fundamental point of view, we would agree to the adoption of the ATTC. 1947 line.
3. If the Conference is of the opinion that the ATTC. 1947 line should be steepened below $R_n = 10^7$ by an amount as suggested by the Skin Friction Committee, we would agree with this solution, although we consider it to be a great loss that this solution has no fundamental significance and is therefore inapplicable as a basis for a three-dimensional extrapolation system.

Dr. T. S. Raghuram and Mr. S. D. Nigam (*)
(Written Contribution).

With regard to recommendation (1) of the report of the Committee on Skin friction and Turbulence stimulation, it is felt that the model results in the range of Reynolds number 10^6-10^7 and below are influenced by laminar and mixed flow conditions. Therefore it is necessary to develop a new experimental technique to change the flow conditions around the model to fully turbulent flow, before a definite recommendation of this kind can be made.

The usual techniques for creating turbulent flow, namely, trip wire, sand strip and studs seem to be only partially effective at low Reynolds number. The stimulation of turbulence by the trip wire is dependent on the generation and break up of the two-dimensional columnar vortex behind it into three-dimensional vortex loops which is an essential feature preceding the creation of turbulent spots near the top of the loops and near the outer edge of the boundary layer. At low Reynolds numbers the two-dimensional columnar vortex is so weak that it is destroyed completely by the action of viscosity as soon as it is generated, and hence the flow remains laminar in character. This eliminates trip wire as an efficient turbulence stimulator at low Reynolds numbers and therefore more attention should be paid to develop techniques using sound waves, ultrasonic vibration and electric discharge for the creation of turbulence.

An interesting experiment by Fales (**) on a moving plate has revealed a remarkable phenomenon caused neither by high speed nor by a trip wire but merely by decelerating the plate. A deceleration of the plate from steady motion tends to distort the boundary layer and causes a momentary change of shape in the velocity gradient within the boundary layer and establishes a point of inflexion in the gradient curve. This makes the boundary layer unstable and the flow becomes turbulent even at small Reynolds number (during and after the deceleration period). This technique is being applied for creating fully turbulent flow conditions around a model of Lucy Ashton in the Ship Model Tank at the Indian Institute of Technology, Kharagpur, and the results of such a series of experiments will be published soon. The difficulties associated with the application of this simple technique in the model field have been overcome in the case of 6" models in a tank 9' $\times$ $\times$ 2,6" $\times$ 1' and work on bigger models is in progress.

(*) From Indian Institute of Technology, Kharagpur, not in attendance.

(**) Fales, E. N.: "A new laboratory technique for investigation of the origin of fluid turbulence". *Journal of the Franklin Institute,* Vol. 259, 6 (1955), pp. 491-516.

Mr. Y. V. Krivtzov (*) (Written contribution).

Up-to-date the Kryloff Institute has not taken part in International Towing Tank Conferences, as well as the other Institutes and Towing Tanks of the Soviet Union.

However the specialists of the Kryloff Institute have studied and discussed with great interest and care the publications of the works of previous Towing Tank Conferences and especially useful activity of technical committees.

The studying of the materials on subjects 2 and 4, kindly sent to us by the Organizing Committee of the 8-th International Towing Tank Conference permits us to make the following remarks.

During the past years in the practice of our Institute there were often the cases of essential disagreement between the full-scale trial results and those of the propulsion calculations on the base of model test data when using Froude frictional coefficients for estimating model and ship skin friction. The analysis of full-scale trial data, as well as the test results of geosim models, made us to come to the conclusion about the necessity of introducing essential allowances when calculating and in particular the allowances for the roughness of the actual hull surface. Accordingly, the Institute had to solve the question of further use of Froude frictional coefficients as the latters, in the Institute's opinion, permit more or less satisfactorily to estimate skin friction resistance of the models of the length 5-7 m only and decrease friction of the models of smaller length greatly. These coefficients do not also permit to make correct estimates of skin friction resistance of ships of the length over 120-150 m. Especially it concerns high-speed ships. In addition, the use of Froude coefficients had impeded the comparison of predicted ship propulsion data with the conclusions of scientific investigations being carried out, as a rule, on the base of the assumption of frictional coefficient depending on Reynolds number. Already in 1932, all this made the Kryloff Institute refuse to use Froude frictional coefficients and begin to calculate model and ship friction according to Prandtl-Schlichting formula with additional allowances for the roughness of the actual hull surface.

Of course, Prandtl-Schlichting formula has certain limitations. Yet in our opinion, the same limitations are inherent to Schoenherr formula. The reliable prediction of ship resistance from test results of the models with the length 5-7 m according to the first and second formulae practically gives rather close values. In comparison with Schlichting formula when using Schoenherr one model residuary resistance "increases" approximately as much as the ship friction evaluation "decreases".

(*) From Kryloff Institute, Leningrad, not in attendance. His contribution was received one day after the Conferente.

By the moment of publicating the recommendations on this Subject of the 6-th International Towing Tank Conference the Institute had already accumulated considerable experience on comparative model tests and full-scale trials and derived certain allowances for the roughness of hull surface.

Taking all mentioned above into consideration, we did not find to be expedient to refuse to calculate friction according to Prandtl-Schlichting formula and to accept Schoenherr one. Thus, up-to-date the Institute has been using Prandtl-Schlichting formula as well as the other large towing tanks of the Soviet Union.

The Institute has often carried out the experiments on geosim models (12 series) covering the range of model lengths from 0,7 to 9,0 m and ending for separate series both by self-propulsion model of the length from 15 to 27 m and by towing the full-scale ships of the length 45-49 m. In addition, the Institute has accumulated a great deal of material about full-scale ship propulsive trials.

In the towing tanks of the Institute geosim model tests had been carried out in the wide range of Reynolds numbers from 5.10^5 to 8.10^7. For seagoing models and full-scale ships maximum Reynolds numbers were about 3.10^9.

The experience of testing geosims allows the Institute to agree, in particular, to the recommendation of the committee about the increasement (in comparison with Schoenherr line) of frictional coefficients for the range of Reynolds number 3.10^6-1.10^5 and below. We consider however that the steep of the extrapolator in this range proposed by the Committee is excessive. Thus in our turn we may recommend the steep for the extrapolator at $R_n < 3.10^6$ wich is roughly analogous to Schultz-Grunow line. At the mean values of Reynolds numbers 10^6-10^7, in our opinion, the line should lie slightly above the Schoenherr one and approach to the Prandtl-Schlichting line. At high Reynolds numbers ($R_n > 10^8$) we do not see any reasons for the lowering of extrapolator according to Schoenherr, as it is recommended by the Committee. Even when using frictional coefficients according to the Prandtl-Schlichting line the Institute has always received the positive allowance for roughness though all ships being built in the Soviet Union, as a rule, are completely welded. All said above naturally refer to a newly painted surface hull and ship trials in good weather conditions (seaway and wind force do not exceed three).

In practice of our Institute there has never been such large scattering in allowances for roughness as it was indicated, for example, in published investigations of David Taylor Model Basin and in the report of Dr. F. H. Todd at the Conference (Formal discussion).

The methods of treatment of geosim model data and reliable prediction of a full-scale ship resistance proposed lately by Lap and Troost and also by Hughes which take into consideration the extrapolation on the base of three-dimensional flow and the method previously proposed by Telfer cannot be regarded to be universal at present time. In our opinion they cannot serve as a basis for developing a common extrapolator. In general we consider our knowledge to be insufficient at present time to state quite confidently the frictional extrapolator for the whole practically necessary range of Reynolds numbers from 1.10^5 to 1.10^9, which should be based theoretically, well confirmed by the experiments and corresponding to the physical essence of the effect.

Therefore until the development of more reliable relations for defining skin friction resistance we do not consider the calculating of skin friction according to other formulae to be of special necessity.

The development of a method of reliable prediction of ship resistance from model tests which should be more perfect and illuminate more completely the physical essence of the effect still remains the most important problem.

It will take however much work for its solution due to interconnection of separate resistance components and accordingly the difficulties of their division and studying in parts.

Therefore undoubtedly we shall be forced to conduct our work on the base of Froude method for considerable period of time. Then in addition to special question about the correction of frictional extrapolator for flat surface three main problems arise before towing tanks:

1) Transition from flat surface friction to three-dimensional body one.

2) Definition of form resistance in order to evaluate this component from residuary resistance, and

3) Transition from friction resistance of a ship with technically smooth surface to ships with actual rough skin.

The solutions of the first and second problems are the most difficult. Especially for this aim the combined, theoretical and experimental work of large towing tanks according to a common programme using modern experimental equipment is urgently necessary. It seems to us that the Committee should pay special attention to the planning of this part of work. In addition to pure theoretical and experimental physical methods of investigating in solution of the two problems both special methods of testing models (for example, Föttinger's method) and special ways of reliable prediction of model test results (the methods of Telfer, Lap and Troost, Hughes and similar ones) should be used. None of these methods, as it is, cannot lead to positive solution of the problem without studying physical essence of the question. It is not out of place to remind here that the method of estimating the form resistance component by means of defining resistance at "the low Froude number run-in" point similar to Hughes' one had been suggested by Pavlenko (Pavlenko G.

E. "Methodics of river ship model tests", 1930) and often checked in the Institute. Unfortunately, this method even being combined with tests of duplicated models according to Föttinger's method does not permit in many cases to receive correct single valued results due to the reasons pointed out by Dr. F. H. Todd in the report of formal discussion. Since 1937 the Institute has been also studying the question about the dependence of form resistance coefficient upon Reynolds numbers. However it is still necessary to continue verification of this question, especially at high Reynolds numbers.

In our opinion the solution of the third problem is much easier in spite of the difficulty to carry out full-scale ship trials with certain accuracy.

Perhaps up to now on the base of analysis of model tests and full-scale ship trials all the towing tanks had included without any division total difference between the resistance calculation of smooth hull and actual ship resistance in the assumption of allowance for roughness. Meanwhile, this allowance consists of three parts principally different:

1) Proper "allowance for roughness", taking into consideration the transition from technically smooth surface ship to real hull,

2) Tolerance on a model and full-scale experiment and

3) Tolerance connected with scale effect and conventionality of the Froude method. This latter is essentially a measure of our "lack of knowledge".

Similar division of this summary allowance into parts is urgently necessary. In view of this experiment tanks must pay special attention to the studying of the first part of allowance—proper "allowance for roughness". Considering the use of the conception about "equivalent" sand roughness according to Nikuradse to be too conventional and less perspective, the Institute has made an attempt to develop the method of hydrodynamical model test of resistance of plates with arbitrary total surface roughness. With the aid of this method the results of resistance tests of rough plates received at the towing tank of the Institute were evaluated in full-scale Reynolds numbers. Thus, for example, for "the equivalent" plate of the length 100 m at $R_n = 10^9$ having the surface roughness similar to roughness of a hull painted in accordance with the painting scheme used in the Soviet Union, the value of allowance for roughness is equal to $0{,}2.10^{-3}$. The additional local resistance due to seam and butt welded joints is, in our opinion, from $0{,}05.10^{-3}$ to $0{,}1.10^{-3}$ for modern ships.

Just the same value of allowance is connected with blind holes in the hull of a surface ship. It is stated that the influence of plating corrugation whichs is tolerated by actual rules for ship construction on resistance increasement is practically negligible.

It should be also noted that for a certain ship the value of summary roughness resistance coefficients is slightly changed in the wide range of speed.

After the definition of the value of the first part of allowance for proper roughness and taking into consideration that the second part of allowance due to tolerances on model test and full-scale ship trial can be estimated with sufficient accuracy in each separate case, one can perform the quantitative evaluation of the third part of allowance connected with scale effect and conventionality of the method of calculation.

There are some reasons to suppose that this part of allowance is not great when model experiments are being carried out appropriately (using modern reliable means of turbulence of flow on the model) as well as full-scale trials with certain accuracy.

It has been already stated above that the Institute considers our knowledge to be yet insufficient at present time for the development of a single well-grounded friction extrapolator. Therefore the Institute considers that there are no reasons to refuse to calculate friction resisitance according to Prandtl-Schlichting formula being used by the Institute.

However, if the Conference acknowledges expedient to develop a single friction extrapolator, the latter will have to be developed as a temporary "engineering" instrument on the base of two-dimensional flow.

Such a single extrapolator might be used by towing tanks in their research work and reports of the International Conferences and Committees. However, the introduction of a single extrapolator should not exclude the right for all experiment tanks to use the other methods of extrapolation which are the most convenient for them in the course of their current work.

Informal Discussion

Dr. K. S. M. Davidson.

I am surprised there is no more support for proposal 2. We should be quite clear of what we are doing.

If you adopt proposal 1, the increasing values are being adopted. I don't think there is a consistent opinion among tank users that an increase is necessary. I think some increase is in principle desirable; I don't think it is necessary. From my point of view I would prefer not to change.

I would like to suggest that we do not attempt in this session to reach an agreement; that should be left.

Prof. Dr. Ing. H. W. Lerbs.

I only want to state the position of the Hamburg Tank. I will be very brief because our position has been expressed previously. We would prefer the proposal 1 of the Committee, alhough we are aware that this proposal represents an engineering solution without any scientific basis, and so it might be considered only as a temporary one.

Prof. L. C. Burrill.

I want to speak as a practitioner and not as an experimentator or tank superintendent. I may say I think as a practitioner of some years standing and therefore representing the views of those who use the tank. I was one of those who wanted to see the adoption of the Schoenherr line or possibly a slight modification of the Schoenherr line in the light of data which have gone into our hands since Schoenherr has formulated his line. The reason for that was that I felt it was about time to recognize that the Reynolds number base was a more scientific approach to the extrapolation.

The other reason was the need—and I believe the absolute need— for an internationaly agreed standard procedure in relation to model experiments. Therefore my position is unchanged and as the Committee has examined the data in the light of geosim data, I would fully support the adoption of the number 1 proposal of the Skin Friction Committee.

Mr. G. A. Firsov (Translated from Russian).

1. A lot of friction lines have been obtained under the same conditions, all of them resulting from independent experiments. The whole bunch of these lines can be considered statistically. Then, one can obtain a curve with the least quadratic error. When we do this we obtain a curve very close to the Schoenherr line. Therefore, in Russia, such a curve is very popular, or that of Schlichting, which lies very close to Schoenrr's.

2. When frictional resistance is important and we have to deal with a fine ship, we must consider the distribution of wetted surface along the length of the ship.

3. Very thorough investigations in Russia on the effect of body shape upon frictional resistance have lead to the conclusion that for ships of L/B equal to 8 or 9 the increase of frictional resistance does not exceed 2-3 per cent above that of a smooth plank.

Mr. H. Lackenby.

Like Mr. Newton and Mr. Hadler, I have been trying to find explicit formulae. My formulae are simpler tham Mr. Hadler's and Mr. Newton's My approximation to the first proposal of the Committee is:

$$10^3 C_F = 0.804 + 124 R_n^{-1/4}$$

and to the second proposal:

$$10^3 C_F = 0.703 + 128 R_n^{-1/4}$$

The accuracy of these formulae is about 2 %. Finally I am in favour of proposal 1.

Mr. S. T. Mathews.

I should like to express muy appreciation for the great work which has been done. I should also like to say that we would prefer to retain the ATTC. 1947 line. However, we would agree on any line approximating this. But proposal 1 would be the more suitable.

Prof. E. V. Telfer.

I want to tell you that most of the different outlooks on the proposals made in connection with extrapolation are really not correct.

Prof. C. W. Prohaska.

It has been said that we would be the laughing stock of our profession. I am not quite sure that this is true. We might be the laughing stock of the profession if we agree on the single line. The single line might be the most practical solution and I would not be against that. But I think it should be expressely stated that it is an intermediate solution and it should be recommended that the studies be pursued.

Dr. Hughes has given quite a good experimental procedure, but it is lacking in theoretical background. I should like to persuade some hydrodynamist to take up this question.

There is one thing I should like to tell you, and that is that the influence of viscosity must be composed of at least two parts.

It is easy to prove that with the Hughes system you cannot get a constant value. If you plot the relation between the total coefficient and the local coefficient, you will not get the same values at different Reynolds numbers.

Dr. I. S. Schuster (Translated from German).

I want to present the opinion of the Berlin Tank. We are in favour of Committee's proposal 2 and against number 1. The line based on proposal 2 has a statistical background and fits naturally with reality. Certainly, there is a lack of physical basis for this line, but, as Prof. Horn has already said, that does not matter so much, since it is identical with that of Dr. Hughes, which has a sound physical basis.

Prof. Horn proposes the application of a factor 1.15 on the two-dimensional Hughes basic line. I would prefer 1.1 as a more suitable factor, but I don't want to discuss this point.

I want to add that proposal 2 lies very close to Schultz-Grünow line. If this line is based on tests performed many years ago and has since them given good results in its application, what is the reason to propose a new line, when an old and well grounded one exists?

I also want to add that Schultz-Grünow and Hughes lines have very handy explicit forms. They have no additive correction and are easily developed in series; therefore, they can be easily employed in computors. Finally, attention is called to the fact that Schultz-Grünow, Hughes and proposal 2 of the Committee just about agree with one another except in the range of low R_n.

Summing up, the Berliner Tank fully agrees with Prof. Horn's proposal, the content of which practically coincides with that of Dr. Hughes'. This means a recognition of the Committee's proposal 2.

Capt. M. L. Acevedo (Translated from Spanish).

I only want to sum up in a few words the position of El Pardo Tank in connection with the adoption of a new line. This position, according to the statements contained in my written contributions (particularly those made in the Final Conclusions), is as follows:

If a new line is to be adopted, we would prefer Hughes' proposal (b). In terms of figure 17 (page 129), this proposal and those of Telfer and Newton include a correlator, the value of which is very little influenced by the variation of the model size. If Hughes' proposal is preferred, it is because this proposal has, in comparison with the other two, the advantage of including a more normal average value of the correlator (approximately 1.19 instead of 1.12, the latter being too low, in our opinion, to always prevent negative or unreasonably low roughness allowances).

With reference to the proposals of the Committee, I have to say that both proposals may be considered, also in terms of figure 17, as including a correlator, the value of which appreciably depends on the model size, an occurence which it seems should not be. Furthermore, when model range reaches $R'_n = 10^7$ (the normal range in large Tanks), the first Committee proposal becomes identical with Schoenherr's line and in such a case no improvement in respect to present ship predictions will be obtained by adopting the new line. Figure 17 also shows clearly the discontinuity at $R'_n = 10^7$ which is involved in the first Committee proposal.

Ing. General E. G. Barrillon.

I think that grouping subject 2 and subject 4 is not a good change in the old methods of the Conference. I propose that in future subject 2 and subject 4 should be separated. The aim of Committee 2 would be to fix the best line for models only and permit intermodel correlation. The aim of Committee 4 would be to fix the best line for ships only. If both Committee 2 and 4 working independently find the same line, this line should be adopted.

In this meeting no change is desirable.

I should like to reemphasize what Dr. Davidson has said. However, I agree with proposal 1, in principle, but not in detail, and I believe the values should be increased. I believe that perhaps the best solution below $lg\,R_n = 6.5$ will be something like 10 %, which divides the increase of the proposal 1 by a factor of 2.

I should like to ask the Committee, when it gets to details, to look into the accuracy and revaluate on this basis.

Prof. M. A. Abkowitz.

The Skin Friction Committee in its Proposal 1 for the ITTC friction line has indicated a much increased slope (over Schoenherr) in the curve for the small model range of log Reynolds number 5.5 to 6.5. I assume that the increased slope in this Reynolds number region has been proposed to provide a satisfactory friction line for the towing tanks which use small models (about 5 ft in length). Our experience with small models in the M. I. T. tank indicates that when turbulence is stimulated by pins, sandstrips, or trip wire, the parasitic drag of these devices must be taken into account. Also, in the small model range, whereas the fuller bows may present some problems in turbulence stimulation, the fuller sterns may lead to separation in the laminar or partly laminar state. Separation in the laminar state tends to increase the resistance. Hence, if the friction curve is derived from geosim tests of fuller models at small Reynolds numbers, the slope may be greater tham that derived from geosim tests of fine models or planks. At the M. I. T. tank, results for each model at the small Reynolds numbers are individually analyzed, keeping in mind the parasitic drag of the stimulator and possibilities of laminar separation; thus avoiding a set routine or standard procedure of systematically taking measured results and subtracting a given friction coefficient.

Since the ITTC friction line still represents more or less a "plank" line, the slope of the line at the smaller Reynolds numbers should parallel the curve derived from geosim tests on the finer models and this according to our experience is about midway between the Schoenherr line and Proposal 1 of the Committee.

I therefore would like to recommend to the Committee that they consider altering Proposal 1 to provide a somewhat reduced slope at a log Reynolds number below 6.5 and that, where the small Reynolds number range joins the larger Reynolds number range, the slope of the line be continuous.

Dr. L. Landweber.

Two old, but unsolved problems in connection with the prediction of ship resistance, are urgently in need of solution: the problems of turbulence stimulation and friction extrapolation. In connection with the former a promising rational study of the effectiveness of three-dimensional roughness elements, in the form of studs, as turbulence stimulators, was undertaken at the David Taylor Model Basin, and I would like to urge the Taylor Model Basin to report upon the conclusions of this research, even if negative. In connection with the extrapolation of the frictional resistance of smooth surfaces there are needed laboratory data on friction coefficients at much higher Reynolds numbers than are now available. Such data would permit discrimination between competing extrapolation curves and theories. An incompleted project at the Taylor Model Basin to tow a long circular cylinder and measure the shear stresses along the length, offers the possibility of obtaining such data at high Reynolds numbers. I would like to stress the importance of this investigation and to express the hope that the results of it will not be long in forthcoming.

The Chairman, Prof. L. Troost.

We will now proceed to see if there is any delegate who wants to discuss turbulence stimulation at this time.

As to the proposal of Dr. Kempf representing some ideas on the testing of standard models in each tank, I think we all can endorse a reglamentation and a study of the subject. I propose that the meeting of the Skin Friction Committee should make a sort of reglamentation of such proceedings and each tank can see for itself how far it goes. Does anybody object?

Prof. E. V. Telfer.

I think it is an excellent idea, but I merely want to make a further suggestion because I think, as it stands now, it is not sufficient.

You should agree on a standard form and have it as a smooth form but also have another form exactly the same but a rough form.

I make that suggestion because in what Dr. Kempf is proposing it will be quite unable to distinguish between changes in resistance due to viscosity differences, or turbulence or laminar differences, from those differences which are due to pure instrumental technique.

If you have the same form repeated in rough, then because we know that such a rough form will have a constant viscous resistance independent of viscosity changes, you will be able to distinguish that your variations belong to the instrumental part of your work.

I think you have to have those two parallel checks going on, to distinguish between water technique and instrumental technique.

Dr. J. Breslin.

As representative of a small towing tank I am interested in arguing for the adoption of a more uniform method of turbulence on small models. It would be very valuable to have the results of a study available to us and I would like to urge that if they could find the time, to push that programme to completion.

Ing. General E. G. Barrillon (Translated from French).

It is possible to have hulls with the same wetted surface, the same form and the same length. They would be parent hulls but not geosims.

For such a family of hulls the theoretical wave resistances can be computed with only one calculation. And for them any of the actual friction formulae will give the same results.

If comparative tests in several basins are to be undertaken, it would be preferable to test such families of hulls and not to test geosims.

From my point of view, (a) comparative tests are necessary and in order to reach some results it is advisable to restrain ourselves to the simple problem: the towing resistance; (b) in each basin the model —that should remain always the same— ought to be periodically tested, as has already been done with the "Iris" model.

Mr. R. N. Newton.

I would also like to remark on Dr. Kempf's contribution on techniques of measuring model resistance in which he suggests that every tank which can afford to do so should procure a standard model, or two, which would be periodically tested for the purpose of correcting resistance results on other models. Delegates will be aware of the fact that this has been the practice in the last seventy years at Haslar, but they are possibly not aware of the large variations which can and do occur and which cannot be clearly ascribed to any known cause. Taking the Froude ship tank at A. E. W. as an example, the skin friction of the standard "Iris" model varies generally by $\pm 1\frac{1}{2}$ per cent. This variation has no relation to the season of the year or the temperature of the tank water nor has it yet been ascribed directly to animal growth or other "inclusions". It occurs whether or not the water is "purified" by addition of copper sulphate.

The frequency of the variation is equally unpredictable and there have been occasions on which its magnitude has reached very large proportions. In 1925 the resistence of the "Iris" model dropped by as much as 10 per cent and returned to its average value inside two months. Similarly in 1917 there was a drop of about 5 per cent over a period of three months.

In the light of this long experience and the magnitude of these figures, the essential need for the use of standard models cannot be refuted, and I agree with Prof. Barrillon that the reasons for the inconsistent results must be sought.

Regarding Dr. Telfer's suggestion, I want only to say that it is difficult to keep a model smooth; it would be more difficult to keep one "rough".

Dr. K. S. M. Davidson.

In regard to standard models, I think that there is only one way to deal with them, and it is to have some individual go round to different tanks and see what small differences there are.

Dr. Ing. F. Gutsche (Translated from German).

The proposal of Dr. Kempf to measure the resistance of standard models over a whole year or more is to be welcomed. He himself pointed out that all physical accompanying circumstances of those experiments should be communicated in the final report.

For the sake of a really accurate examination of those results the data should be given, firstly without any correction, and secondly, with that correction which is believed to be the best one for comparison.

I think it is well worth stating that for the data firstly mentioned in any case no correction should be made.