in induced drag due to sweep is nearly completely compensated for by the favorable influence of sweep on wave resistance.

At yaw angle larger than about 5 degrees the flow along the aftbody usually separates and the resistance associated with yaw angle increases markedly. Even on sailing yachts, with block coefficients around 0.4, this occurs because the flow on the windward side of the hull forward of the rudder separates.

The immersed hull of a heeled ship will be asymmetrical, with the leeside of the vessel being considerably bluffer. This invariably leads to an increment in the wavemaking resistance. In many cases also the viscous resistance increases because of added wetted area or because of more unfavorable boundary layer development leading to flow separation, or both. In yachts with long bow and stern overhangs, this increase in resistance is compensated for to a certain extent because of the increase in effective wavemaking length of the hull as the hull heels. A typical result for such a hull is shown in Fig. 56 which gives the resistance in kN for the J-class yacht *Rainbow*, as measured at MARIN for various angles of heel. At speeds between 6 and about 9 knots it can be seen that the increase in resistance with heel angle is marginal due to the increase in wavemaking length.

Section 6
The Uses of Models for Determining Ship Resistance

6.1 Historical. Because of the complicated nature of ship resistance it was natural that early recourse should have been made to experiments, and it is recorded that Leonardo da Vinci (1452-1519) carried out tests on three models of ships having different fore-and-aft distributions of displacement (Tursini, 1953). The next known use of models to investigate ship resistance were qualitative experiments made by Samuel Fortrey (1622-1681), who used small wooden models towed in a tank by falling weights (Baker, 1937). From this time onwards there was a steady growth of interest in model experiment work (Todd, 1951). Colonel Beaufoy, under the auspices of the Society for the Improvement of Naval Architecture, founded in London in 1791, carried out between nine and ten thousand towing experiments between 1791 and 1798 in the Greenland Dock, using models of geometrical shape and flat planks (Beaufoy, 1834). Benjamin Franklin was probably the first American to make model experiments, in 1764, to verify observations he had made in Holland that resistance to motion increased in shallow water (Rumble, 1955).

Throughout this period this method of gravity towing was universally used, and William Froude made his first model experiments in 1863 in a large rainwater tank using the same type of towing mechanism. He soon became dissatisfied with the limitations of these experiments and turned his mind to the use of a larger tank, making proposals to the British Admiralty in 1868, which were accepted, and a new tank was completed near his home in Torquay in 1871 (W. Froude, 1955). This tank had a length of 84.7 m (277.8 ft), a width at the water surface of 11 m (36 ft) and a depth of water along the centerline of 3.05 m (10 ft). It was equipped with a mechanically propelled towing carriage to tow the models, in place of a gravitational device, and because of this and its size may be considered as the forerunner of the tanks so common today. At this time Froude was already 61 years of age, having put forward his law of comparison in 1868 and shown how it could be used in practice to predict ship resistance from model results.

D. W. Taylor graduated from the U.S. Naval Academy in 1885 and from there went to the Royal Naval College, Greenwich, England, where he became aware of the work done by Froude. On his appointment to the Navy Department in Washington in 1894 he advocated the building of a towing tank for the U.S. Navy. As a result, the Experimental Model Basin (EMB) was built in the Washington Navy Yard. It had a length of 143.3 m (470 ft), a breadth on the water surface of 12.8 m (42 ft) and a centerline depth of 3.06 m (10 ft), the towing carriage having a top speed of 7.7 m/sec (25.2 ft/sec) (EMB, 1925). The Basin was opened in 1900, and Taylor remained in charge of it for some 14 years, during which time much work of great value to naval architects everywhere was conducted under his inspiration and guidance.

At the end of the 19th century there were perhaps five model experiment tanks in the world. Now they number about 125 and are regarded as a necessary and important adjunct to the shipbuilding industry of every maritime nation.

6.2 Modern Facilities. There is a considerable number of small tanks generally associated with educational and research establishments, using models 1 to 2 m in length, engaged in measuring resistance in smooth water and motions, resistance, and loss of speed in waves. In some of these tanks the towing force is still provided by a falling weight, suitably geared, and which is constant during a run, the technique being to measure the speed attained for a given towing force, i.e., for a given resistance. A few facilities have monorail carriages.

The larger tanks in general employ mechanically or electrically-driven towing carriages, use models 4 to 10 or more meters in length (32.8 ft), and conduct propulsion as well as resistance tests on ship models and various other bodies. Typical dimensions of these larger tanks are 250 m (820 ft) long, 10 m (33 ft) wide, and 5 m (16 ft) deep. For investigations in shallow water, some establishments have adopted a basin in excess of 20 m (66 ft) wide and a (variable) depth of up to about 1.5 m (5 ft). For investigations involving high-speed craft, the tank needs to be extra long. In that case, often relatively narrow basins are employed, typically 4 m wide and 4 m deep, (13 ft) and the speed of the towing carriages needs to be well in excess of the maximum speed of about 10 m/sec (33 ft/sec) normally employed in other tanks. Resistance tests are carried out with the model attached to the carriage. Besides measuring the so-called towing force, the rise (or sinkage) of the model at the forward and aft perpendicular is measured and observations of the waves along the model are made.

After the first resistance tests, many models are modified and further experiments made. This is often done on the basis of observations of the flow around the hull by means of a paint-smear technique and/or wool-tufts, which set themselves in the flow lines. Wool-tufts secured to pins set normal to the model surface enable the flow to be explored at points some distance out from the hull. Photographs of the underwater hull form are then taken for analysis of the flow pattern (Fig. 57). Such observations provide useful information for judging the quality of the hull and appendages, often leading to design modifications.

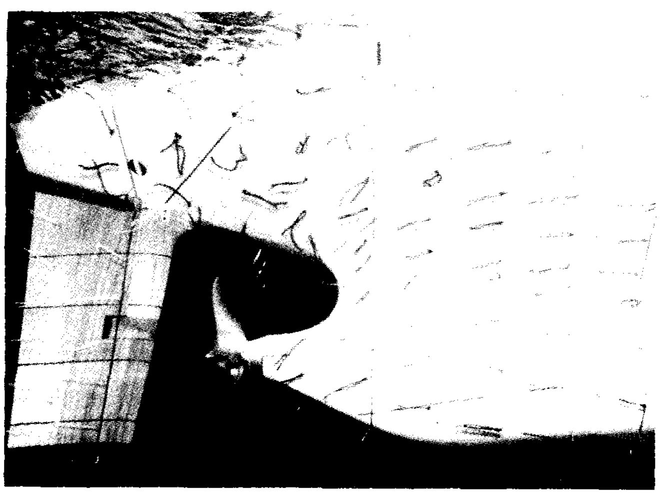

Fig. 57 Model in MARIN towing tank showing stream-line flow

6.3 Model Testing Techniques. The accepted basis of predicting ship resistance from that of a model still rests on the assumption made by Froude that the total resistance can be divided into frictional and residuary components, as set out in Section 2.3. The residuary resistance coefficient C_R is assumed to be the same for model and ship at the same value of the Froude number $Fn = V/\sqrt{gL}$, but the frictional resistance coefficient C_F is a function of Reynolds number and therefore bears the major responsibility for correct extrapolation. The "jump" in going from model to ship is very large; in the case of a 125-m ship and a 5-m model, used as an example in Section 2.3, the speeds of the ship and model were, respectively, 25 and 5 knots, so that the values of Reynolds number, proportional to the product VL, would be in the ratio of 1 to 125. (This is not always realized in looking at experiment plots, because it is general to use a base of log Rn, which greatly reduces the apparent degree of extrapolation.) The measurement of the model resistance must therefore be extremely accurate to minimize errors in the extension to the ship. Constancy of speed of the towing carriage is a most important basic requirement, as is the accuracy of the towing dynamometer.

The models must be made to close tolerances, the surface correctly finished and the models properly ballasted and trimmed. The choice of model length is governed by several considerations. The larger the model the more accurately it can be made and the larger are the forces to be measured, both features leading to greater accuracy in the measurement of resistance. However, the bigger the model the more expensive it is to build and handle, the larger are the facilities and instruments necessary, and some compromise in size must be reached. If the model is too large for a particular basin, interference from the walls and bottom will increase the resistance. There is still no real agreement on the proper assessment of this interference effect. Broadly speaking, the model should not have a length greater than the depth of water or than half the width of the basin in order to avoid interference with the wave resistance. The midship cross-sectional area of the model should not exceed about 1/200 of that of the basin in order to avoid setting up appreciable return flow in the water around the model, the so-called blockage effect. However, in cases where wavemaking is small, larger models can be used and corrections made for the remaining blockage effect (Comstock, et al, 1942, Telfer, 1953-4, Hughes, 1957, Emerson, 1957 Hughes, 1961, and Kim, 1962). In certain high-speed models care must also be taken to avoid the critical speed $V = \sqrt{gh}$, in the basin which would result in the formation of a wave of translation, as already described. A model run at any speed above about 0.7 of this value will give a resistance different from that appropriate to deep water.

Another precaution that must be taken in all model testing is to ensure that the flow over the model is fully turbulent, since flow around the full-scale ship is turbulent. In discussing the frictional resistance of smooth planks, it was shown that two regimes of flow are possible, laminar and turbulent, the latter giving a much higher specific resistance than the former at the same value of Reynolds number. Also, with increasing Reynolds number, the regime changes, but

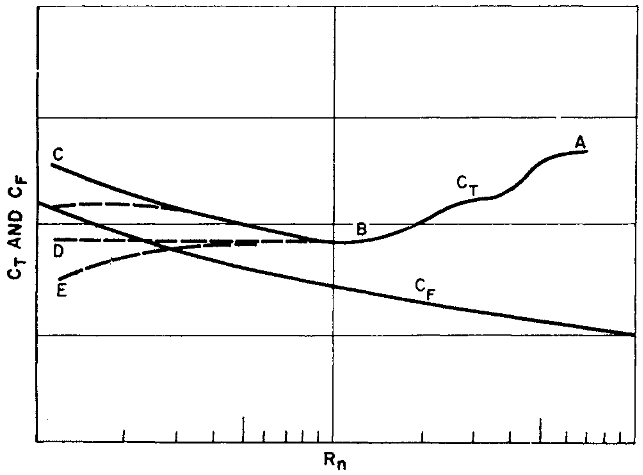

Fig. 58 Effect of laminar flow on resistance curves

the transition curve depends on the individual circumstances (Fig. 2).

The resistance curves of models show the same character of transition at low values of Reynolds number, and the resulting low resistance coefficients if scaled up to the ship on the basis of any of the turbulent friction lines would give much too low ship resistance values. It is therefore necessary to avoid this situation.

The presence of laminar flow can usually be detected from the shape of the resistance curve (Fig. 58). At low values of the Froude number, where the wavemaking resistance is vanishingly small, the C_T curve should run in more or less parallel to the curve of skin friction coefficient C_F, as *ABC*. A curve which falls away in this region or even becomes horizontal, such as *ABD* or *ABE*, is at once suspect as being subject to partial laminar flow. To investigate this, Prohaska's method can be used. For low speeds the resulting value of $1 + k$ should approach a constant value.

Some typical values of Reynolds numbers for ships and models are given in Table 6 as lending a quantitative meaning to the problem. Results of plank experiments, such as the plot shown in Fig. 3, indicate that laminar-flow effects occur up to Reynolds numbers of the order of 5×10^6. On this basis the models required for the 125 m and 300 m ships considered in Table 6, would have to have minimum lengths of about 6 and 4 m, respectively, to avoid serious laminar flow at the speeds in question. Experiments with ship models have shown, however, that without special devices to stimulate turbulence, as are discussed later, even these sizes are sometimes inadequate. The persistence of laminar flow has been found to depend to a great extent on the pressure gradient along the entrance (which is absent in the plank) and on the factors which affect this, such as shape of stem profile, half-angle of entrance on the load waterline and the shape of the entrance area curve. When these features combine to give a negative pressure gradient just abaft the bow, with consequent increasing velocity, the flow is stable and laminar flow tends to persist over considerable areas, sometimes as far as to the forward shoulder, where the pressure gradient becomes constant or positive (Allan, et al, 1949).

The practical answer to the problem is to deliberately "trip" the laminar flow by some kind of roughness near the bow. Perhaps the first reference to such a practice is the use of a 0.025 mm diameter trip wire at the nose of a spheroid, 0.61 m long and 0.15 m diameter (2×0.492 ft) tested in air, which completely altered the character of the resistance curve (ARC, 1922).

Trip-wires some 0.9 mm diameter (0.035 in.) placed around the hull at a station 5 percent of the forward perpendicular were used in the Berlin tank as early as 1925 and came into general use there around 1933, and it is now standard practice to use such a wire or other equivalent device in most model basins. Among these other devices are struts towed ahead of the model and sandstrips, studs or pins on the hull itself. The stimulating device, if attached to the model, increases the resistance because of its own parasitic drag. If it is placed too near the stem, there is a danger of the laminar flow reestablishing itself if the pressure gradient is favorable, while when placed in the usual position, 5 percent aft of the stem, it leaves the laminar flow, if it exists, undisturbed over the first part of the length up to the stimulator. In this case the resistance of this portion of the surface will be less than the turbulent resistance desired. It is usual to assume that this defect in resistance balances the additional parasitic drag of the wire or studs. The strut has the advantage of being unattached to the model, so that its drag does not come into the measurement of model resistance, and if it successfully stimulates turbulence in the water there will be no area of laminar flow on the model. However, experiments with fine, high-speed

Table 6—Typical Reynolds Numbers

Scale		Length in m	Speed in m/sec.	Rn at 15 deg C
125-m ship at 10 knots				
1/1	(ship)	125	5.144	540.9×10^6
1/10		12.5	1.627	17.10×10^6
1/15		8.33	1.328	9.31×10^6
1/20		6.25	1.150	6.05×10^6
1/25		5.00	1.029	4.33×10^6
1/50		2.50	0.728	1.53×10^6
1/100		1.25	0.514	0.54×10^6
300-m ship at 30 knots				
1/1	(ship)	300	15.432	3894.3×10^6
1/30		10.0	2.817	23.70×10^6
1/50		6.0	2.182	11.01×10^6
1/75		4.0	1.782	6.00×10^6
1/100		3.0	1.543	3.89×10^6
1/125		2.4	1.380	2.79×10^6
1/250		1.2	0.976	0.99×10^6

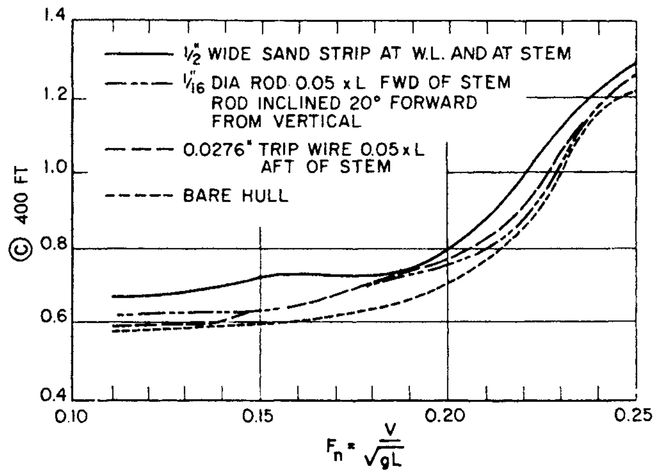

Fig. 59 Resistance curves showing effects of turbulence stimulation

models which showed no signs of laminar flow when unstimulated, have shown reduced resistance with a strut ahead, presumably due to the strut wake. The use of studs, some 3 mm in diameter and 2.5 mm high (0.12 × 0.098 in.) spaced 25 mm (1 in.) apart along a line parallel to the stem contour, was proposed by Hughes et al (1951). The distance of the line of studs from the stem was controlled by the half-angle of entrance on the waterline. The object is to ensure that the stud drag approximately balances the reduction in resistance of the laminar flow area forward. For models 5 to 6 m in length, the studs appeared to be more effective than the trip-wire, stimulating turbulent flow at an earlier speed and giving greater coverage, and they also maintained turbulent flow to lower speeds, an important factor in experiments designed to investigate methods of extrapolation.

The magnitude of the problem depends upon the size of model and type of ship. Stimulation shows little or no effect in 6 m models of high-speed warships and merchant ships of block coefficient 0.65 and below. In the fuller types of merchant ships, the effect of stimulation appears to depend to a considerable extent upon the type of bow. A heavily raked stem and pronounced V-sections seem to favor the persistence of laminar flow, while a vertical stem and U type sections seem to feel the effect much less. A typical example of the former type of hull is the *Liberty* ship, and some results of experiments on a model of this design are shown in Fig. 59. At the service speed the spread in resistance between the bare model without stimulation and the highest resistance obtained with stimulation is of the order of 20 percent.

Without turbulence stimulation, therefore, even comparative model tests may be misleading, to say nothing of the errors in ship estimates. Two models of the same design, one with V and the other with U-sections forward could give results apparently showing the former to have considerably less resistance, whereas some or all of this difference could be due simply to a greater area of laminar flow. Again, as the LCB is moved forward, the fore end becomes fuller, pressure gradients are altered, and changes in resistance attributed to the change in shape may, in truth, be purely a stimulation effect. For these reasons it is now almost universal practice to use stimulating devices on any model which is even remotely likely to suffer from laminar flow. In this connection it is well to bear in mind that many of the methodical series tests of the past were run without any stimulation devices and, indeed, before the need was even recognized. For the type of ship which is prone to such trouble, these early series results and those of individual models also should be treated with caution. Some indications of the probable magnitude of the correction necessary have been given by Dawson, et al (1949).

If self-propelled model experiments are to follow the resistance tests, as is usual, the size of the model propellers has also to be considered when choosing the scale for the hull model (Section 5. Chapter VI).

The model when completed is ballasted and trimmed to the required displacement and waterline, and attached to the resistance dynamometer of the towing carriage. A clear statement as to whether molded or total displacement is meant, should be included (see Chapter I). The model is free to take up any sinkage, rise or trim that may be dictated by the water forces, but any yawing motion is prevented by guides.

Table 7—Principal Particulars of Model and Ship

	Ship	Model
Scale, $1/\lambda$	1/1	1/21.667
Length on waterline, m, L_{WL}	260.0	12.000
Length between perpendiculars, m, L_{PP}	262.0	12.092
Beam, m, B	42.00	1.938
Draft, m, T	10.640	0.4911
Displacement volume (mld), m^3, ∇	86266	8.481
Block coefficient, C_B	0.7425	0.7425
Wetted surface, m^2, S	12898.9	27.476

Table 8—Model Experiment Results

Model speed, m/sec. V_M	Ship speed, knots V_k	Measured model resistance, N R_{TM}
0.7736	7.000	32.05
0.8842	8.000	40.89
0.9947	9.000	50.91
1.1052	10.000	62.21
1.2157	11.000	74.86
1.3262	12.000	89.17
1.4368	13.000	104.60
1.5473	14.000	121.18
1.6578	15.000	138.76

Basin water temperature, deg C = 16.2 (61 deg F)

On any given test run, the carriage is driven at the desired constant speed and records are taken of speed, resistance and trim of the model, and often the wave profile along the hull is photographed as an aid to subsequent understanding of the results. For any given displacement and trim condition, a number of test runs are made at different speeds and a curve of resistance against speed obtained.

Before proceeding to propulsion tests, a number of other experiments are often made, to determine the best line for bilge keels, the flow around the afterbody to settle the best alignment for bossings, shaft brackets and rudders and, in some cases, the flow over the whole form, as previously discussed.

6.4 Calculation of Effective Power. The estimation of ship resistance and effective power from model tests is carried out on the basis of the Froude assumption as set out in Section 2.3, with refinements based on increased understanding of resistance.

In 1978 the ITTC Performance Committee advocated

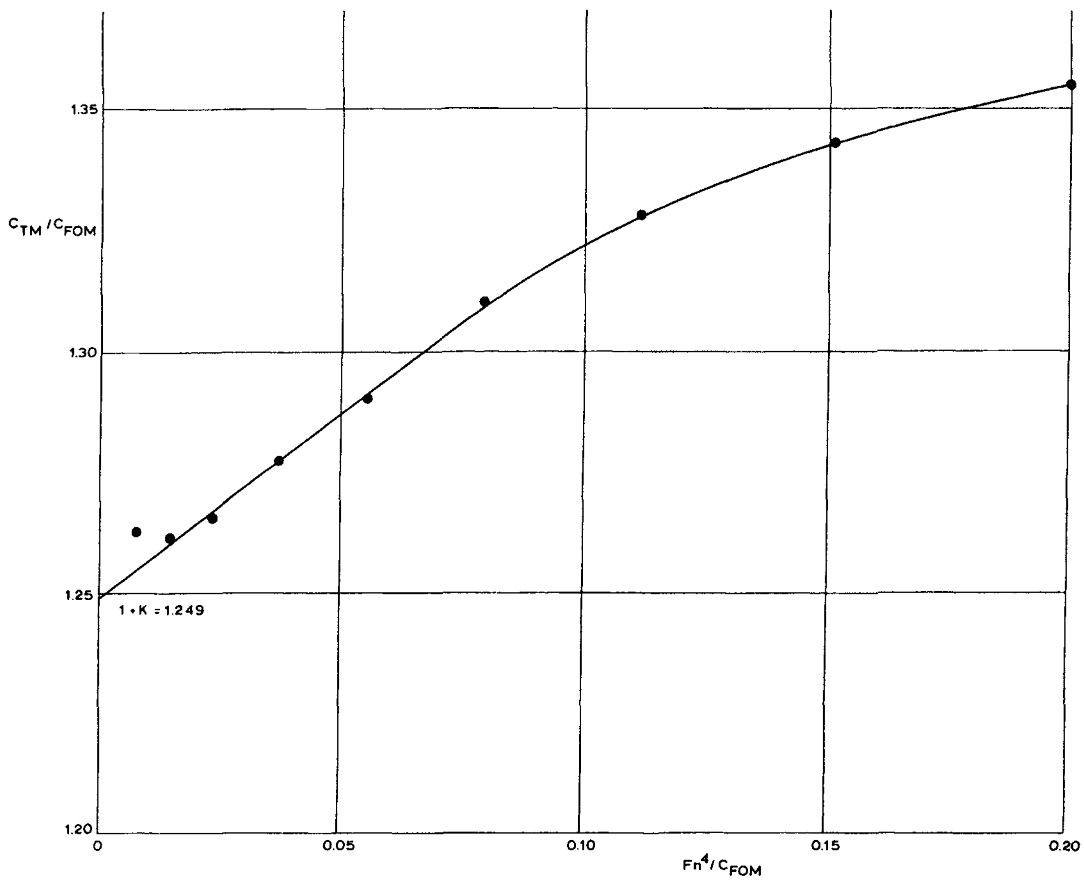

Fig. 60 Plot of Prohaska method to derive the effect of hull form on viscous resistance, based on data in Table 9.

Table 9—Values Required to Find $1 + k$ According to Prohaska

V_M	Fn	$Fn^4 \times 10^6$	$Rn \times 10^{-6}$	$C_{TM} \times 10^3$	ITTC-1957 $C_{FOM} \times 10^3$	Fn^4/C_{FOM}	C_{TM}/C_{FOM}
0.7736	0.0713	2.586	8.409	3.901	3.092	0.0084	1.262
0.8842	0.0815	4.414	9.611	3.810	3.021	0.0146	1.261
0.9947	0.0917	7.069	10.812	3.748	2.960	0.0239	1.266
1.2157	0.1221	15.773	13.215	3.690	2.860	0.0552	1.290
1.3262	0.1223	22.338	14.419	3.693	2.818	0.0793	1.310
1.4368	0.1324	30.775	15.618	3.691	2.780	0.1107	1.328
1.5473	0.1426	41.391	16.819	3.687	2.746	0.1507	1.343
1.6578	0.1528	54.543	18.020	3.678	2.715	0.2009	1.355

The values of C_{FOM} adopted in Table 9 are those according to the ITTC-1957 friction line which, together with the ATTC (Schoenherr) friction line, are the most widely used. The values of C_F according to both these friction lines are given in Table 12.

Table 10—Values of Kinematic Viscosity ν for Fresh and Salt Water Adopted by the ITTC in 1963; Salinity of Salt Water is 3.5 Percent

Temperature deg C*	Kinematic viscosity of fresh water ν, $\frac{m^2}{sec} \times 10^6$	Kinematic viscosity of salt water ν_S, $\frac{m^2}{sec} \times 10^6$
0	1.78667	1.82844
1	1.72701	1.76915
2	1.67040	1.71306
3	1.61655	1.65988
4	1.56557	1.60940
5	1.51698	1.56142
6	1.47070	1.51584
7	1.42667	1.47242
8	1.38471	1.43102
9	1.34463	1.39152
10	1.30641	1.35383
11	1.26988	1.31773
12	1.23495	1.28324
13	1.20159	1.25028
14	1.16964	1.21862
15	1.13902	1.18831
16	1.10966	1.15916
17	1.08155	1.13125
18	1.05456	1.10438
19	1.02865	1.07854
20	1.00374	1.05372
21	0.97984	1.02981
22	0.95682	1.00678
23	0.93471	0.98457
24	0.91340	0.96315
25	0.89292	0.94252
26	0.87313	0.92255
27	0.85409	0.90331
28	0.83572	0.88470
29	0.81798	0.86671
30	0.80091	0.84931

For other salinities, interpolate linearly.

the use of the form-factor approach (see Section 3.6) in determining the effective power from model tests. Calculations have been made to illustrate this modified method in detail and to show the differences in the final effective power which result from using this approach relative to the two-dimensional frictional resistance formulations (see Section 3.5), for both ATTC and ITTC (1957) friction lines. The calculation is for a 12-m (39.36 ft) model of the 120,000-m^3 methane carriers *Castor* and *Nestor* (Muntjewerf, et al, 1983). The main particulars of model and ship have been given in Table 7.

(*a*) *Calculation of resistance and effective power by three-dimension extrapolation procedure.* Following the Prohaska (1966) proposal (Section 3.6), the value of $1 + k$ can be determined from the gradient of the C_{TM}/C_{FOM} vs. $c\text{Fn}^4/C_{FOM}$ curve for $\text{Fn} \rightarrow 0$. For this particular case $c = 1$ has been taken, so as to fit the C_{TM} curve as well as possible. The values of the quantities required to find the form factor value are given in Table 9. The Froude number and Reynolds number given in Table 9 have been calculated as follows:

$$\text{Fn} = \frac{V_M}{\sqrt{gL_{WL_M}}}; \text{ for 14 knots}$$

$$\text{Fn} = \frac{1.5473}{\sqrt{9.81 \times 12}} = 0.1426$$

$$\text{Rn} = \frac{V_M L_{WL_M}}{\nu}; \text{ for 14 knots}$$

$$\text{Rn} = \frac{1.5473 \times 12}{1.1040 \times 10^{-6}} = 1.6819 \times 10^7$$

Table 11—Values of Mass Density ρ for Fresh and Salt Water Adopted by the ITTC in 1963; Salinity of Salt Water is 3.5 Percent

Temperature deg C*	Density of fresh water, ρ, kg/m^3 or 1000 kg/L	Density of salt water, ρ_S, kg/m^3 or 1000 kg/L
0	999.8	1028.0
1	999.8	1027.9
2	999.9	1027.8
3	999.9	1027.8
4	999.9	1027.7
5	999.9	1027.6
6	999.9	1027.4
7	999.8	1027.3
8	999.8	1027.1
9	999.7	1027.0
10	999.6	1026.9
11	999.5	1026.7
12	999.4	1026.6
13	999.3	1026.3
14	999.1	1026.1
15	999.0	1025.9
16	998.9	1025.7
17	998.7	1025.4
18	998.5	1025.2
19	998.3	1025.0
20	998.1	1024.7
21	997.9	1024.4
22	997.7	1024.1
23	997.4	1023.8
24	997.2	1023.5
25	996.9	1023.2
26	996.7	1022.9
27	996.4	1022.6
28	996.2	1022.3
29	995.9	1022.0
30	995.6	1021.7

* deg F = 9/5 C + 32

The value of the kinematic viscosity ν for fresh and salt water at the basin water temperature of 16.2 deg C (61 deg F) can be determined from Table 10.

C_{TM} is calculated on the basis of the measured total resistance, which for 14 knots (model speed, 1.5473 m/sec) was found to be 121.18 N:

$$C_{TM} = \frac{R_{TM}}{\frac{1}{2}\rho V_M^2 S_M}$$
$$= \frac{121.18}{0.5 \times 999.25 \times (1.5473)^2 \times 27.476}$$
$$= 0.003687$$

The value of the mass density, ρ (kg/m^3), for fresh and salt water at the basin water temperature of 16.2 deg C (61 F) can be determined from Table 11. From the plot in Fig. 60 the value of $1 + k$ is found to be 1.249, which is the value of C_{TM}/C_{FOM} when $\text{Fn}^4/C_{FOM} = 0$.

To calculate the resistance and effective power for a smooth hull at a full-scale speed of 14 knots, for example, the following procedure is adopted. proceeding as in Section 3.6 and Fig. 5:

$$C_{TS} = C_{VS} + C_{WS} \qquad (57)$$

where $C_{VS} = (1 + k)\, C_{FOS}$

and $C_{WS} = C_{WM} = C_{TM} - C_{VM}$

in which $C_{VM} = (1 + k)\, C_{FOM}$

Following this procedure it has implicitly been assumed that $1 + k$ is independent of Rn. Following Tanaka (1979), the form factor may be defined as:

$$k = a + b\, C_F^{-0.5} + c\, C_F^{-1.0} + d\, C_F^{-1.5}$$

where a, b, c, d depend on the ship form. a is related to equivalence with a two dimensional body, b to equivalence with a body of revolution, c is related to separation and d is related to base drag. Thereby it is evident that the form factor may depend on Rn. However no quantitative information on this dependency is known. With $\text{Rn}_M = 1.6819 \times 10^7$, $C_{FOM} = 2.746 \times 10^{-3}$ for the ITTC-1957 friction line and $C_{FOM} = 2.700 \times 10^{-3}$ for the ATTC friction line.

Thus $C_{VM} = 1.249 \times 0.002746 = 0.003422$ (ITTC) and $C_{VM} = 1.249 \times 0.002700 = 0.003372$ (ATTC).

The corresponding value of $C_{TM} = 0.003687$ (see Table 9).

Thus $C_{WS} = C_{WM} = 0.003687 - 0.003422 = 0.000265$ (ITTC) and
$C_{WS} = 0.003687 - 0.003372 = 0.000315$ (ATTC).

It has been agreed by the ITTC and ATTC that for published work all ship results shall be given for a standard temperature of 15 deg C (59 deg F). The corresponding value of the kinematic viscosity for salt water is 1.1883×10^{-6} m^2/sec. (see Table 10). The full scale Reynolds number for 14 knots is then,

$$\text{Rn}_S = \frac{V \times L_{WL_S}}{\nu_S} = \frac{14 \times 0.5144 \times 260}{1.1883 \times 10^{-6}} = 1.576 \times 10^9$$

Table 12—Values of C_F According to the ITTC-1957 and ATTC Friction Lines

Rn	$C_F \times 10^3$ (ITTC)	$C_F \times 10^3$ (ATTC)
1×10^5	8.333	7.179
2	6.882	6.137
3	6.203	5.623
4	5.780	5.294
5	5.482	5.057
6	5.254	4.875
7	5.073	4.727
8	4.923	4.605
9	4.797	4.500
1×10^6	4.688	4.409
2	4.054	3.872
3	3.741	3.600
4	3.541	3.423
5	3.397	3.294
6	3.285	3.193
7	3.195	3.112
8	3.120	3.044
9	3.056	2.985
1×10^7	3.000	2.934
2	2.669	2.628
3	2.500	2.470
4	2.390	2.365
5	2.309	2.289
6	2.246	2.229
7	2.195	2.180
8	2.162	2.138
9	2.115	2.103
1×10^8	2.083	2.072
2	1.889	1.884
3	1.788	1.784
4	1.721	1.719
5	1.671	1.670
6	1.632	1.632
7	1.601	1.600
8	1.574	1.574
9	1.551	1.551
1×10^9	1.531	1.531
2	1.407	1.408
3	1.342	1.342
4	1.298	1.299
5	1.265	1.266
6	1.240	1.240
7	1.219	1.219
8	1.201	1.201
9	1.185	1.186
1×10^{10}	1.172	1.172

$$C_F = \frac{0.075}{(\log_{10}\text{Rn} - 2)^2} \qquad \text{(ITTC-1957)}$$

$$\frac{0.242}{\sqrt{C_F}} = \log_{10}(\text{Rn} \times C_F) \qquad \text{(ATTC)}$$

$$\text{Rn} = \frac{VL}{\nu}$$

At this value of the Reynolds number, the value of the friction coefficient (from the formula given in Table 12) $C_{FOS} = 0.001448$. Note that at this value of the Reynolds number the ITTC and ATTC values of C_{FO} are equal.

It follows that $C_{VS} = 1.249 \times 0.001448 = 0.001809$; thus $C_{TS} = 0.001809 + 0.000265 = 0.002074$ (ITTC) and $C_{TS} = 0.001809 + 0.000315 = 0.002124$ (ATTC).

Finally it is necessary to add a correlation allowance to the smooth-ship resistance to obtain the resistance of the actual ship, an allowance which has been given the symbol C_A, the A standing for "additional" resistance, in analogy with C_F for frictional, C_R for residuary, and so on. The significance to be attached to C_A is discussed next.

The value of C_A depends upon a number of things. From the differences which can arise in the predicted ship P_E from identical model results by using different extrapolation methods, these latter will be one of the prime factors influencing C_A. In order to build up a standard method of deriving such allowances for use in future design work, it is necessary for each establishment to use one method of extrapolation, and the resulting values of C_A will apply only to that method. C_A covers not only roughness, but also differences due to extrapolation methods, together with scale effects in such processes, as well as in all the factors making up the propulsive efficiency. The need for such a factor arises from our lack of exact knowledge in the foregoing fields.

The part of C_A due to hull roughness can be attributed to a number of causes:

1. Structural roughness, resulting from the method of construction of the shell—whether riveted, welded, or partly of each kind. Other contributors are valve openings, scoops, damage-control valves, waviness of plating between frames, and so on.

2. Paint roughness. Smooth, hard-drying paints will in general cover up some of the structural roughness, such as welding beads and rivet points, and may reduce the resistance below that of an unpainted shell. Paints of rough texture or applied badly, leaving "runs" on the surface, on the other hand, can increase resistance greatly.

3. Corrosion resistance, resulting from breakdown of the paint film in service, causing corrosion and erosion of the shell plating. This can be controlled to some extent by frequent docking, cleaning and painting and the use of cathodic protection, but there is a general long-term deterioration, as evidenced by full-scale trials carried out at intervals over a number of years. Modern methods of grit or shot-blasting may help to restore new-ship quality to the shell, but they are expensive treatments and must be weighed against possible savings in fuel and future maintenance costs.

4. Fouling resistance, caused by marine organisms depositing shell, grass, and so forth. One of the chief factors influencing this type of resistance is the percentage of time spent at sea. Atlantic liners foul very little. Cargo ships spending perhaps half their time in port foul more rapidly, and are affected also by the ports they visit and the time of year. The problem of fouling is much less important now because of the greater efficiency of modern antifouling paints.

Considerable progress in the area of hull roughness has been made by BMT (Lackenby, 1962). Measurements of shell roughness were made on a number of new ships prior to trial, records 0.762 m long (2.5 ft) being taken at some 50 points on each ship. The maximum amplitude of roughness was measured over each 0.05 m length of every record, and the mean of these taken to represent the average roughness of the hull.

The average value for 68 ships was 0.0188 cm (7 mil), 25 of them having values between 0.0165 cm and 0.0191 cm. The greatest roughness was twice the average, the least one half of the average. A statistical analysis of the BMT trial results for new ships indicated that paint roughness was the most significant factor in explaining the variation in C_A-values.

Lackenby (1962) has stated that an increase in average roughness of 0.0025 cm would increase the resistance of a large, new single-screw ship by about 2½ percent

On this basis, the roughest ship, with a value of 0.0366 cm (14 mil) or 0.0178 cm above the average, would have about 17 percent greater resistance, and the smoothest, with a value of 0.0089 cm would have some 8 percent less resistance than the average ship. The spread on new ships due only to variation in roughness could therefore be as much as 25 percent. In the ships tested by BMT, differences of as much as 20 percent have been found between sister ships, and this suggests again that paint surface is probably the major cause of roughness and increased resistance.

However, hull roughness is by no means the only factor affecting the correlation of ship resistance as predicted from the model and as deduced from full-scale trials on the ship. From what has been said in Section 3 it is clear that for a given model result the predicted ship resistance will depend on many other things; i.e., the size of the model, adequate stimulation of turbulence, corrections where necessary for tank wall and bottom effects on model resistance and, above all, upon the method of extrapolation adopted, as illustrated in the present example.

Recent analyses are aimed at determining the influence of hull roughness on the value of C_A, which has led to proposals to no longer use an overall value of C_A but instead to use separate values to account for the specific effects. Thus $C_A = \delta C_f' + \delta C_r$ where $\delta C_f'$ is the correlation allowance for roughness effects and δC_R is a correction for phenomena not accounted for elsewhere.

Bowden, et al (1974) proposed a formula for C_A:

$$10^3 \times C_A = 105 \left(\frac{k_s}{L}\right)^{0.333} - 0.64 \qquad (58)$$

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where k_s is the mean apparent amplitude of the surface roughness over a 50mm wavelength (1.96 in.). In this relation L should not exceed 400m (1312 ft). The above formula has been established from an analysis of thrust measurements taken during ship trials. The equation is based on the measured roughness and should be used in conjunction with a form factor method and the 1957 ITTC line.

The above relation also includes phenomena which are not due to surface roughness. Therefore the increase due to roughness alone was calculated following Yokoo (1966) and Sasajima, et al (1955):

$$\delta C_f' = \delta C_{FM}(k_s, V)\left\{\frac{C_{FS}(V, L_S)}{C_{FM}(V, L_M)}\right\}^2 \quad (59)$$

where δC_{FM} is the resistance increase due to roughness for a flat plate with a standard length L_M. For δC_{FM} the following relation was found (ITTC, 1981):

$$\delta C_{FM} = 1.867\ \delta C_{FM}^*(k_s)\left[1 - \left(1 - \left(\frac{V}{20}\right)^{1.9}\right)^{3.5}\right] \quad (60)$$

where $\delta C_{FM}x^*(k_s)$ is a roughness parameter depending on the type of surface. When $\delta C_{FM}(k_s)$ was put equal to 0.3×10^{-3}, the following equation for δC_R could be established:

$$10^3 \times \delta C_R = 5.725\ L^{-0.11} - 3$$

Holtrop et al (1978) have analyzed 108 full-scale trial measurements of newly built ships without bulbs, in good condition, and compared the results with corresponding model test values. They derived the following statistical relation for the model-ship correlation factor C_A to be used in conjunction with the form factor extrapolation procedure, using the ITTC line, $C_A = 0.006\ (L_{WLS} + 100)^{-0.16} - 0.00205$ which is valid for $T_F/L_{WL} > 0.04$, where T_F is the draft at the fore perpendicular. This relation for C_A reflects MARIN's experience; model test results from other towing tanks may require different correlation allowances, for reasons which have been discussed above. In the present case $C_A = 0.006(360)^{-0.16} - 0.00205 = 0.000290$.

The final value of $C_{TS} = 0.002074 + 0.000290 = 0.002364$ (ITTC) and $C_{TS} = 0.002124 + 0.000290 = 0.002414$ (ATTC). The correlation allowance is taken the same if the ATTC line is used, in order to show the differences using the two lines. The ship resistance in salt water of 15 deg C (59 deg F) is given by:

$$\begin{aligned} R_{TS} &= \tfrac{1}{2}\rho S_S V_S^2 \times C_{TS} \\ &= 0.5 \times 1.0259 \times 12898.9 \times (14 \times 0.5144)^2 \times 0.002364 \\ &= 811.210 \text{ kN (ITTC) and} \\ R_{TS} &= 828.367 \text{ kN (ATTC)} \end{aligned}$$

The effective power P_{ES} is:

$$\begin{aligned} P_{ES} &= R_{TS} \times V_S \text{ kW} = 5842 \text{ kW (ITTC)} \quad (61) \\ &= 5966 \text{ kW (ATTC)} \end{aligned}$$

(b) Calculation of resistance and effective power by two-dimensional extrapolation procedure using ITTC (1957) and ATTC friction coefficients. In the case of two-dimensional extrapolation (see Section 3.5 and Fig. 5), the following procedure is adopted:

$$C_{TS} = C_{FOS} + C_{RS}$$

where

$$C_{RS} = C_{RM} = C_{TM} - C_{FOM}$$

Again for the example considered before with $Rn_M = 1.6819 \times 10^7$, $C_{FOM} = 0.002746$ for the ITTC 1957 friction line and $C_{FOM} = 0.002700$ for the ATTC friction line (see Table 12). With $C_{TM} = 0.003687$, $C_{RS} = C_{RM}$ follows from

$$\begin{aligned} C_{RS} = C_{RM} &= 0.003687 - 0.002746 \\ &= 0.000941 \text{ for the ITTC friction line} \\ \text{and } C_{RS} = C_{RM} &= 0.003687 - 0.002700 \\ &= 0.000987 \text{ for the ATTC friction line.} \end{aligned}$$

The full-scale Reynolds number value is $Rn_S = 1.576 \times 10^9$ for which the ITTC and ATTC friction coefficients are equivalent, and equal to 0.001448 (see previous example). Hence

$$\begin{aligned} C_{TS} &= 0.001448 + 0.000941 \\ &= 0.002389 \quad \text{(ITTC)} \\ \text{and } C_{TS} &= 0.001448 + 0.000987 \\ &= 0.002435 \quad \text{(ATTC)} \end{aligned}$$

The value of the correlation allowance factor C_A to be used in conjunction with two-dimensional extrapolation procedures has not been analyzed as thoroughly as has been for the three-dimensional case. Keller (1973), however, has given overall values for C_A to be used in conjunction with two-dimensional extrapolation procedures adopting the ITTC friction coefficients, Table 13. These values again reflect MARIN's experience. They may also be used in conjunction with the ATTC friction coefficients provided that the Reynolds number at which the model tests are carried out, exceeds approximately 1×10^7. In that case differences between both friction lines are small, as is shown.

Table 13—Values of the Model-Ship Correlation Allowance C_A According to Keller (1973)

Length of ship L_{WL}	Value of correlation allowance C_A
50 – 150 m	+0.0004 to +0.00035
150 – 210 m	+0.0002
210 – 260 m	+0.0001
260 – 300 m	+0
300 – 350 m	−0.0001
350 – 450 m	−0.00025

On adopting these values in the current example, for

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which L_{WL} = 260 m, it follows that $C_A \approx 0$. The resistance and effective power then become:

$$R_{TS} = \tfrac{1}{2}\rho S_S V_S^2 \times C_{TS}$$
$$= 0.5 \times 1.0259 \times 12898.9 \times (14 \times 0.5144)^2 \times 0.002389$$
$$= 819.789 \text{ kN} \quad \text{(ITTC)}$$
$$\text{and } R_{TS} = 835.574 \text{ kN} \quad \text{(ATTC)}$$
$$P_{ES} = 5904 \text{ kW} \quad \text{(ITTC)}$$
$$\text{and } P_{ES} = 6017 \text{ kW} \quad \text{(ATTC)}$$

(c) Comparison between R_{TS} and P_{ES} calculated by the different methods. The values of resistance and effective power calculated by the different methods are summarized in Table 14.

In this case the three-dimensional extrapolation method with the correlation allowance value according to Holtrop corresponds satisfactorily with the values derived from the two-dimensional method with the correlation allowance of 0.

When not accounting for the correlation allowance in the three-dimensional method the respective R_{TS} and P_{ES} values are some 15 percent smaller. The differences between the ITTC-1957 and ATTC friction values only lead to about 2 percent differences in the respective resistance and effective power values for both the three dimensional and the two-dimensional procedures. These differences are small only because the adopted example concerns a model test with a 12 m (39 ft) long model. If a 2 m (6.5 ft) model had been used, these differences would have been nearly 10 percent because the large differences in the respective C_{FOM} values occur for Reynolds number values less than 1×10^7. Below this value the ITTC line has a steeper slope and hence greater values for the same Rn. On using models with a length of 7 meters (23 ft) or more, resistance predictions for the ship will not often be influenced to a great extent whether the ATTC or ITTC line is used.

The accuracy of both the two-dimensional and the three-dimensional extrapolation techniques depend to a large extent on the information available at individual model basins relative to the value of the correlation allowance to be adopted.

Table 14—Calculated Values of Resistance and Effective Power

Method	R_{TS} with allowance (in kN)	R_{TS} without allowance (in kN)	P_{ES} with allowance (in kW)	P_{ES} without allowance (in kW)
3D—ITTC-1957	811.2	711.7	5842	5125
3D—ATTC	828.4	728.9	5966	5249
2D—ITTC-1957	—	819.8	—	5904
2D—ATTC	—	835.6	—	6017

Section 7
Methods of Presenting Model Resistance Data

7.1 General. The most useful method of presenting model resistance data depends upon the particular purpose for which they are to be used. There is no unanimity of opinion in the matter, and the ITTC Committee on the presentation of data (now the Information Committee) has not recommended any generally acceptable method.

Two points may be made in this respect:

(a) It is desirable that the original model data be given, including measured speed and resistance, water temperature, method of turbulence stimulation, cross-sectional area of the tank, model dimensions and displacement and any other relevant information. The user can then convert them to any desired form. This policy has been followed by The Society of Naval Architects and Marine Engineers in its Model Resistance Data Sheets (undated).

(b) If the data are presented in coefficient form, these latter should be nondimensional, so that they will have the same numerical value in any consistent system of units. Unfortunately, this practice has not been followed in the past, with the result that the naval architect should be familiar with a number of the more commonly used presentations.

7.2 The C_T-Rn Presentation. In research problems concerned with the separation of resistance into its components, methods of extrapolation to the ship, model-ship correlation allowances and the like, the resistance coefficient (Section 2.3)

$$C_T = \frac{R_T}{\tfrac{1}{2}\rho S V^2}$$

is usually used, plotted to a base of the logarithm of Reynolds number Rn = VL/ν.

Curves of this kind have been used in earlier sections of this chapter. In any consistent system of units, both C_T and Rn are nondimensional.

7.3 Design Presentations. For design purposes, a method is desired which will show the relative merits of different ship forms.

Ships are usually designed to carry a given displacement at a specified speed. C_T is not suitable for such cases, since it is based on wetted surface and not on displacement, and can lead to misleading presentations. An obvious merit criterion is the resistance per