SMK3363 SHIP RESISTANCE AND PROPULSION

Assignment 1:

Drag, Boundary Layer and Hull Roughness on Ship hull surface

NAME: FIRDAUS BIN MAHAMAD

I/C NUMBER: 870821-02-05475

LECTURER: IR. DR MOHAMAND FAUZI

Faculty of Mechanical Engineering

Department Of Marine Technology

UNIVERSITY TEKNOLOGI MALAYSIA

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CHAPTER 1

INTRODUCTION

1.1 Background

One of the most important considerations for a naval architect is the powering requirement for a ship. Once the hull form has been decided upon, it is necessary to determine the amount of engine power that will enable the ship to meet its operational requirements. Knowing the power required to propel a ship enables the naval architect to select a propulsion plant, determine the amount of fuel storage required, and refine the ship's center of gravity estimate.

Throughout history, naval architects have endeavored to increase the speed of ships. Increased speed would enable a warship to close with its opponent, or conversely, to escape from an attack. Increased speed enables merchant vessels to reach port sooner and maximize profit for its owner.

Until the early 1800's, wind was the force used to propel ships through the water and ships could only go as fast as the wind would propel them. Additionally, because ships were constructed of wood, the structural limitations of wooden hull configurations drove hull designs to primarily meet the structural needs while hydrodynamics was only a secondary concern. With the advent of steam propulsion in the early 1800's, naval architects realized that ship speeds were no longer constrained by the wind and research began into the power required to propel a hull through the water using this new propulsion medium.

Testing of full-scale ships and models determined that the power required to propel a ship through the water was directly related to the amount of resistance a hull experiences when moving through the water.

As the resistance of full scale ship cannot be measured directly, our knowledge about the ships resistance has to be gathered from model tests. The measured calm water resistance is usually decomposed into various components, although all these components usually interact and most of them cannot be measured individually. The concept of resistance decomposition help in designing the hull form as the designer can focus on how to influence individual resistance components.

The resistance of a ship at a given speed is the force required to tow the ship at that speed in smooth water assuming no interference from the towing ship. This total resistance is made up of different components which interact with each other in an extremely complicated way. In order to deal with the question more simply, it is usual to consider the total resistance as being made up of two main components. These components are related to different laws of similitude which make the methods for model-ship extrapolation rather complicated. In this subject some important aspects on resistance and flow behavior are described.

1.2 Drag (Resistance)

The behavior of the total resistance can be roughly divided into two main components viz.

1. The drag generated due to viscous phenomena of the medium in which the ship (model) will run. It is mentioned viscous resistance or friction R_F.
2. The drag generated due to the running ship (model) close to the water surface. A wave will be generated due to the disturbance of the water level in consequence of the ship's action. This type of resistance is mentioned wave making resistance or residuary resistance R_R.

The total resistance R_T (sum of 1 and 2) is the force which is necessary to pull the ship model with a constant speed through the water. This force and speed can be simply measured. We then find value for, what we call model condition. This force has to be calculated to a ship (full size) condition, if we do a test with a ship model.

In formula: $R_T = R_F + R_R$

In former times well- known founders of experimental techniques in the domain of marine engineering and hydrodynamics already established that, if compared with full size results, the conversion of model test results into those of the full size ship were not realistic. The problems are concentrated in the behavior of the flow around the model which differed much from the real ship's flow.

William Froude was very successful into deriving physical, justified calculation methods for extrapolating model results into those for full size ships. He succeeded to calculate the frictional resistance for model and ship, whilst the wave drag was calculated in a different way. Reynolds pointed out the behavior of a flow related to the speed, the length of the running object and the viscosity of the medium (Reynolds' law: Flow characteristics are quite similar for equal Reynolds number R_N)

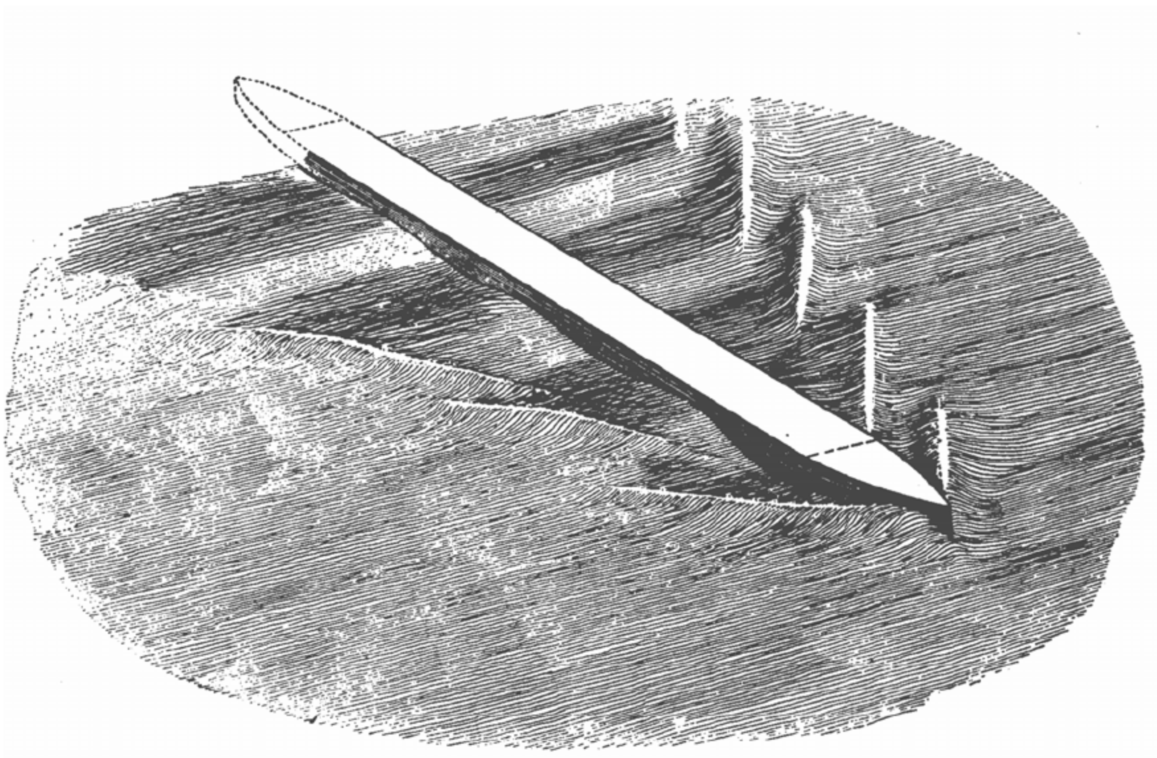

Figure 1.1 Froude's sketch of a characteristic wave train for ships

Froude himself proved the behavior of the wave resistance and based this aspect on speed, length of the object and the gravity. (Froude's Law: Wave making characteristics are quite similar for equal Froude number F_N)

The problem is that with constant medium (water) Reynolds' law for friction resistance and Froude's law for wave making resistance cannot be applied. Because, for the same corresponding conditions (for ship and model) the

- Flow behavior as well the
- Wave behavior

Have to be similarly scaled.

We have stated before that only at equal Reynolds number friction powers will correspond to each other. In our example that is not the case. From measurements it can be pointed out that at various R_N a characteristic curve of frictional coefficients can be generated.

Due to not fulfilling Reynolds' law the difference in Reynolds number caused a not to be neglected difference of viscous (friction) power. This difference in power has to be corrected in the result of the resistance tests. This correction has been dominated Friction deduction correction and is the consequence of not fulfilling Reynolds' law.

1.2.1 Total Hull Resistance (R_T)

As a ship moves through calm water, the ship experiences a force acting opposite to its direction of motion. This force is the water's resistance to the motion of the ship, which is referred to as "total hull resistance" (R_T). It is this resistance force that is used to calculate a ship's effective horsepower. A ship's calm water resistance is a function of many factors, including ship speed, hull form (draft, beam, length, wetted surface area), and water temperature.

For the ship operator planning a voyage, getting from Point A to Point B in a shortest amount of time (high speed) requires a lot more power than traveling the same distance at a slower speed. This increase in power is felt directly in the amount of fuel burned during the transit. A ship's fuel consumption curve is similar in shape to its horsepower and total resistance curves. Voyage planning requires careful attention to transit speed and fuel consumption rates to ensure that the ship arrives at its destination with an adequate supply of fuel onboard. The U.S. Navy generally requires that ships arrive with no less than 50 percent fuel onboard as a reserve.

1.2.2 Components of Total Hull Resistance

As a ship moves through calm water, there are many factors that combine to form the total resistance force acting on the hull. The principle factors affecting ship resistance are the friction and viscous effects of water acting on the hull, the energy required to create and maintain the ship's characteristic bow and stern waves, and the resistance that air provides to ship motion. In mathematical terms, total resistance can be written as:

$$R_T = R_V + R_W + R_{AA}$$

Where: R_T = total hull resistance

R_V = viscous (friction) resistance

R_W = wave making resistance

R_{AA} = resistance caused by calm air

Other factors affecting total hull resistance will also be presented. Figure 1.2 shows how the magnitude of each component of resistance varies with ship speed. At low speeds viscous resistance dominates, and at high speeds the total resistance curve turns upward dramatically as wave making resistance begins to dominate.

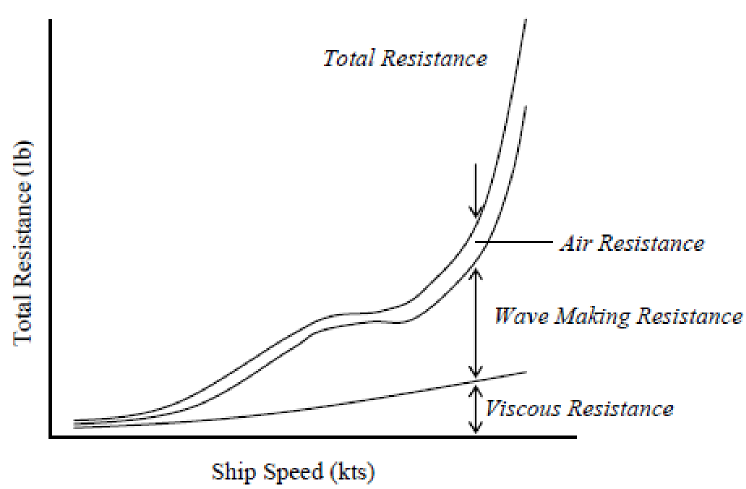

Figure 1.2 Components of Hull Resistance

1.2.2.1 Dimensionless Coefficients

Naval architects, as well as all engineers and scientists, use dimensionless coefficients to describe the performance of a system or to compare different systems to each other. Automotive engineers use a “drag coefficient” to describe the performance of a car. Aviators use the “Mach number” to compare the speed of an aircraft to the speed of sound. Naval architects use many dimensionless coefficients to describe the design and performance of a ship’s hull. Dimensionless coefficients allow the naval architect to compare model test data to full-scale ship data, or to compare the performance of several ship types.

The field of ship resistance and propulsion makes extensive use of standard dimensionless coefficients. The derivation of these standard coefficients is accomplished through dimensional analysis.

1.2.2.1.1 Dimensionless Resistance and Velocity

Just as total hull resistance is the sum of viscous, wave making, and air resistance, we can write an equation for total resistance in terms of dimensionless coefficients.

$$C_T = C_V + C_W$$

Where: C_T = coefficient of total hull resistance

C_V = coefficient of viscous resistance

C_W = coefficient of wave making resistance

Note that air resistance is not represented in dimensionless form. This is because the dimensionless form of resistance is a product of model testing, and most models do not have superstructures. Model tests are usually used to determine the performance of the hull and do not include the superstructure.

Since total hull resistance is a function of hull form, ship speed, and water properties, the coefficient of total hull resistance is also a function of hull form, ship speed, and water properties. The coefficient of total hull resistance is found from the following equation:

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

Where: R_T = total hull resistance (lb)

ρ = water density (lb-s^2/ft^4)

V = velocity (ft/s)

S = wetted surface area of the underwater hull (ft^2)

Naval architects also use a dimensionless form of velocity called the "Froude number" (F_n), named in honor of William Froude (1810-1878), one of the pioneers in ship model testing.

$$F_n = \frac{V}{\sqrt{gL}}$$

Where: V = velocity (ft/s)

g = acceleration of gravity (ft/s^2)

L = length of ship or model (ft)

Another common, although not dimensionless, way of expressing velocity is through the speed-to-length ratio. This ratio is similar to the Froude number except that the gravity term is omitted.

$$\text{speed} - \text{to} - \text{length ratio} = \frac{\mathrm{V}}{\sqrt{\mathrm{L}}}$$

Many times the velocity term in the above ratio is expressed in knots (1 knot = 1.688 ft/s). Care should be taken when using this ratio to ensure what units of the velocity term are correct. An example of using dimensionless coefficients to present data is shown in Figure 1.3, a plot comparing C_T and ship speed.

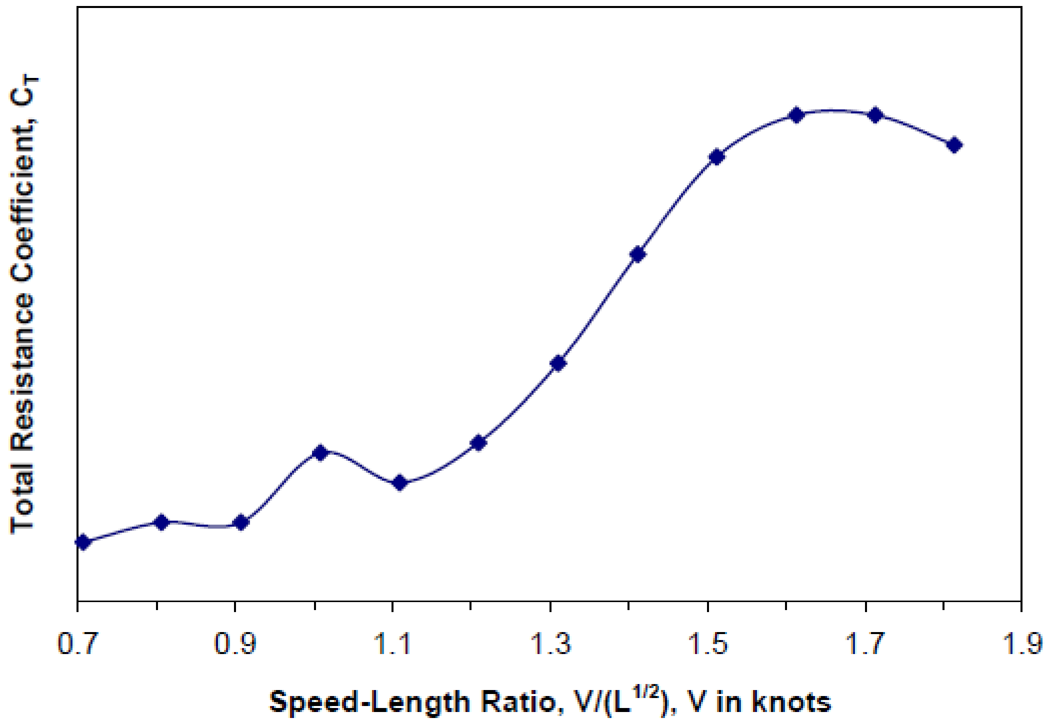

Figure 1.3 Typical relationships between C_T and speed to length ratio

1.3 Boundary layer

The flow along a deeply immersed behaves itself depending on different characteristics. These phenomena are rather strongly related to the transverse speed or better to the number Reynolds number. We will discuss some importance aspects on the flow behavior around the ship model. When streamlined body moves in a straight horizontal line at constant speed, deeply immersed in a real fluid two significant type of flow occur. Near the hull in a so-called boundary layer, the flow acts a force on the body against the motion. Because this fluid is in direct contact with the surface of the body, it is carried along with the surface, and the fluid in the close vicinity is set in motion in the same direction as that in which the body moving. This result in layers of water, which gets gradually thicker from the bow to the stern and in which the velocity varies from that of the body at its surface to that appropriate to the potential flow pattern at the outer edge of the layer. The boundary layer flow leaves the body behind in the form of frictional wake which moves in the same direction as the body.

The body continually enters undisturbed water and accelerates it to maintain the boundary layer. This phenomenon represents a continual drain of energy.

If the body is rather blunt at the aft end, the flow may leave the surface at some point- called separation point- giving rise to a pattern of eddies which is a further drain of energy.

The flow of fluid around a body can be divided into two general types of flow: laminar flow and turbulent flow. A typical flow pattern around a ship's hull showing laminar and turbulent flow is shown in Figure 1.4

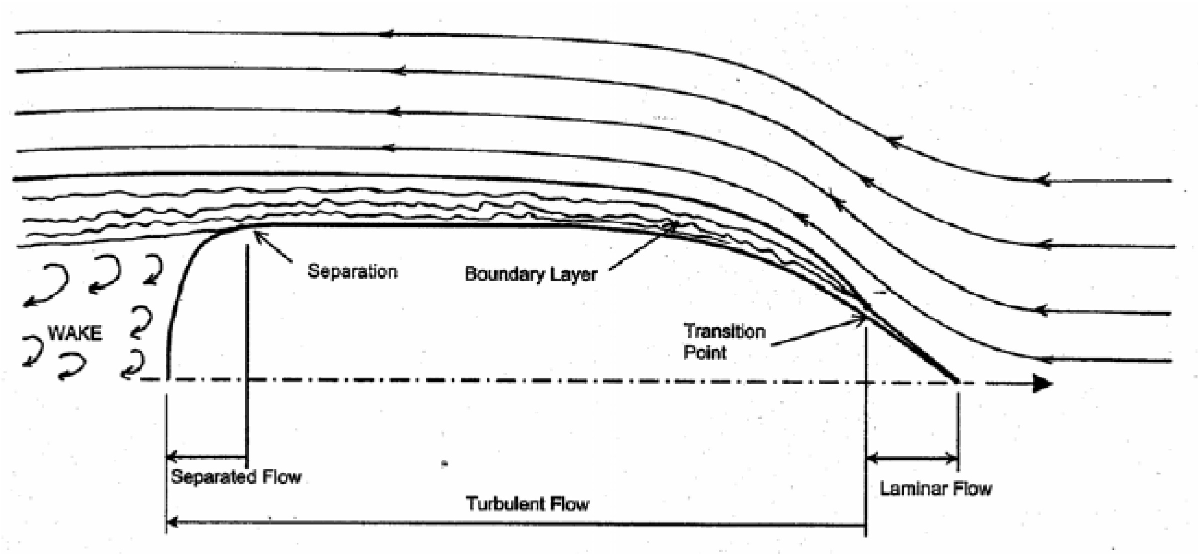

Figure 1.4 Typical water flow pattern around a ship's hull

Laminar flow is characterized by fluid flowing along smooth lines in an orderly fashion with a minimal amount of frictional resistance. For a typical ship, laminar flow exists for only a very small distance along the hull. As water flows along the hull, the laminar flow begins to break down and become chaotic and well mixed. This chaotic behavior is referred to as turbulent flow and the transition from laminar to turbulent flow occurs at the transition point shown in Figure 1.4.

Turbulent flow is characterized by the development of a layer of water along the hull moving with the ship along its direction of travel. This layer of water is referred to as the "boundary layer." Water molecules closest to the ship are carried along with the ship at the ship's velocity. Moving away from the hull, the velocity of water particles in the boundary layer becomes less, until at the outer edge of the boundary layer velocity is nearly that of the surrounding ocean. Formation of the boundary layer begins at the transition point and the thickness of the boundary layer increases along the length of the hull as the flow becomes more and more turbulent. For a ship underway, the boundary layer can be seen as the frothy white band of water next to the hull. Careful observation of this band will reveal the turbulent nature of the boundary layer, and perhaps you can see some of the water actually moving with the ship. As ship speed increases, the thickness of the boundary layer will increase, and the transition point between laminar and turbulent flow moves closer to the bow, thereby causing an increase in frictional resistance as speed increases.

Mathematically, laminar and turbulent flow can be described using the dimensionless coefficient known as the Reynolds Number in honor of Sir Osborne Reynolds' (1883) contribution to the study of hydrodynamics. For a ship, the Reynolds Number is calculated using the equation below:

$$R_{n=}\ \frac{LV}{\upsilon}$$

Where: R_n is the Reynolds number

L = length (ft)

V = velocity (ft/sec)

ν= kinematic viscosity of water (ft^2/sec)

For external flow over flat plates (or ship hulls), typical Reynolds number magnitudes are as follows:

Laminar flow: $R_n < 5 \times 10^5$

Turbulent flow: $R_n > 1 \times 10^6$

Values of R_n between these numbers represent transition from laminar to turbulent flow.

1.3.1 Separation Resistance

Figure 1.4 shows that at some point along the hull, the boundary layer separates from the hull. Flow separation usually occurs near the stern where the hull's curvature is too great for the boundary layer to remain attached to the hull. The space between the smooth flowing water and the hull is filled with eddies as shown if Figure 1.4. This region of eddies is known as the ship's wake, and due to viscous effects, the wake is pulled along with the ship, thus increasing the ship's resistance. The resistance due to flow separation from the hull is sometimes referred to as "separation resistance". The flow separation point is a function of hull design and ship speed. A hull that has smooth lines into the stern will have a separation point that is farther aft and tends to have a narrower wake with less separation resistance than a hull that has discontinuities that cause the flow to become separated from the hull. For naval vessels with transom sterns, the separation point is at the stern.

1.3.2 Viscous Pressure Drag

Figure 1.5 shows a body submerged in an ideal (in viscid) fluid. As the fluid flows around the body, there is a pressure distribution normal to the body. In the forward section of the hull there is a component of pressure resisting motion, and in the aft section of the body there is a component of pressure assisting motion. In an ideal fluid these pressure forces are equal and the body experiences no resistance.

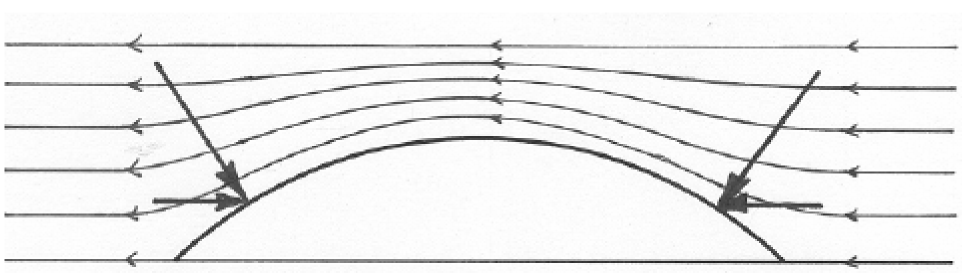

Figure 1.5 Ideal flows around a submerged body

However, water is not an ideal fluid, and therefore some differences in the flow around a body exist. Figure 1e shows a hull submerged in water. Note how the turbulent boundary layer has developed along the hull producing a wake similar to that shown in Figure 1.4. In the forward portion of the hull pressure forces act normal to the surface; however, in the aft portion of the hull the boundary layer reduces the forward acting component of pressure. This reduction in the forward acting component results in a net resistance force due to pressure acting on the hull. This increase in resistance due to pressure is called "viscous pressure drag" or "form drag", and is sometimes also referred to as the normal component of viscous resistance.

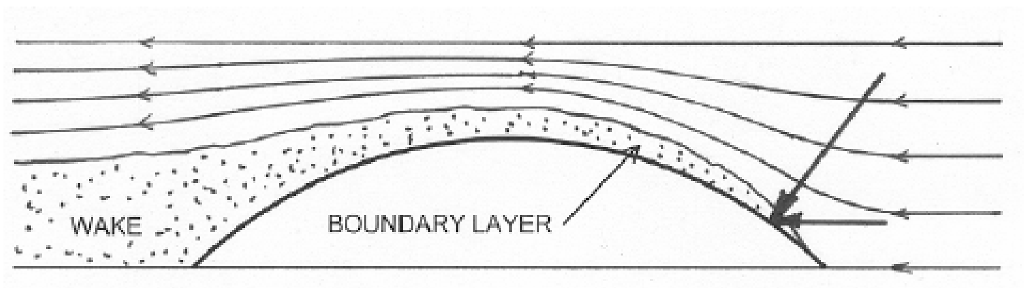

Figure 1.6 Flow around a body submerged in water

As you might expect, from looking at Figure 1.6, the shape of a ship's hull can influence the magnitude of viscous pressure drag. As you may expect, ships that are short in length with wide beams (a low length to beam ratio) will have greater form drag than those with a larger length to beam ratio. Also, ships that are fuller near the bow (e.g. bulk oil tanker) will have greater form drag than ships with fine bows (e.g. destroyer).

1.4 Hull Roughness on Ship Hull Surface

1.4.1 Relation of hull form to resistance

In research problems concerned with the separation of resistance into its components, methods of extrapolation to the ship, model–ship correlation allowance and the like, the total resistance coefficient

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

is usually used, plotted to a base of the logarithm of Reynolds number $R_n = V L / \nu$. Curves of this kind have been used in earlier sections. In any consistent system of units, both C_T and R_n are dimensionless.

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 unit displacement weight, R_T/W, which is nondimensional when R_T and W are expressed in the same units. This ratio is the basis of a number of presentations, which differ principally with regards to the speed coefficient used as the base.

Since lower resistance implies lower fuel costs, minimization of ship resistance is clearly a consideration in the design spiral. A new ship is usually required to carry certain deadweight at a particular speed, and the designer then estimates the probable displacement and principal dimensions. The latter are usually subject to restrictions not associated with resistance and propulsion. Length is expensive in first cost, is limited by docking and navigation restrictions, while added length increases scantlings, equipment and manning scales. From a resistance point of view, greater length for a given displacement will reduce the wave making resistance but increase the frictional resistance, so that longer lengths will be beneficial in ships running at high speeds and vice–versa. Longer lengths are also generally beneficial for behavior in rough seas.

An increase in draft, *T*, is generally beneficial for resistance, and is a cheap dimension in terms of cost. However, it may be limited by depths of harbors, canals, or rivers.

The beam, *B*, is one of the governing factors in ensuring adequate stability, and a minimum value of *B/T* is generally necessary on this account. An increase in *B* will increase the resistance unless it is accompanied by a corresponding finer hull. In cases of low speed ships however, a small reduction in length and a compensating increase in beam, because of the resulting decrease in wetted surface, may result in little or no increase in resistance. This results in a cheaper ship and also meets the need for increased stability in ships with large superstructures. This idea has been exploited in a number of large tankers.

The minimum wetted surface for a given displacement is also sensitive to the *B/T* ratio, the optimum value of which is about 2.25 for a block coefficient of 0.80 and about 3.0 at 0.50. However, the penalty for normal departures from these values is not very great. The effects of changes in *B/T* on wave making resistance can be studied from model experiment results. Generally, stability considerations and limiting drafts usually preclude values below 2.25 for full ships and 2.5 or even more for fine, higher speed hull forms.

While such considerations may be of guidance to naval architects in the choice of dimensions, they must meet many other demands, and will be influenced to a large extent by their knowledge of the particulars of existing successful ships. The process of design is essentially an iterative one, in which the various elements are changed until a proper balance is attained. In order to do this, parametric surveys have to be made on the effects of changes in dimensions, hull form, machinery types, and coefficients of form.

An approximate relation between the block coefficient *CB* and the Froude number F_n can be expressed by

$$\frac{V}{\sqrt{gL}} = 0.595(1.08 - C_B)$$

For trial speed, and

$$\frac{V}{\sqrt{gL}} = 0.595(1.05 - C_B)$$

for service speed. A similar formula for the sustained sea speed in terms of the prismatic coefficient *CP*, is

$$Vs/\sqrt{g}L = 0.55 - 0.48C_P$$

where the trial speed is taken as

$$V_T = 1.06V_S$$

The above relationships are intended as rough guides to the designer and do not take the place of a careful analysis, model experiments, and comparison of alternative designs. Relations between speed length ratio $Vs/\sqrt{L}$ (*V* in knots, *L* in feet), prismatic coefficient C_P, and displacement length ratio $W/(0.01L)^3$ (*W* in tons, *L* in feet) are shown in Figure 1.7. The underwater volume of the hull is denoted by ∇, so that there is no confusion with the speed *V*. The curves of this figure are based upon data from a variety of sources, and result in two pairs of empirical curves which define two "design lanes". These apply to merchant and combatant vessels of customary form, and not to special types such as fishing vessels and tugs.

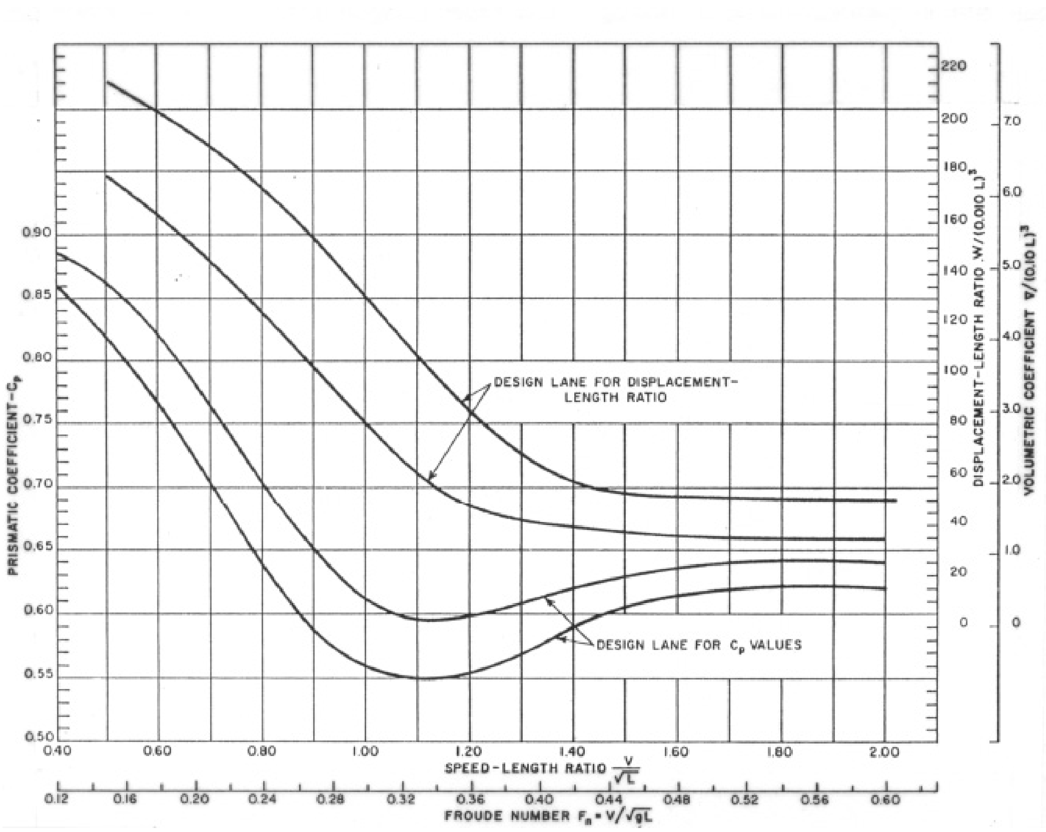

Figure 1.7 Design lanes for prismatic coefficient and displacement length ratio.

The load waterplane coefficient C_{WP} decreases with increasing fullness, its value depending also to a considerable extent upon the type of transverse sections. For Series 60 it is related to the C_P by the approximate formula

$$C_{WP} = 0.18 + 0.86C_P$$

In general, C_{WP} will depend also on stability requirements and seakeeping.

In full ships considerable parallel body can be worked in with advantage, and the entrance can be short, the run being long and fine to minimize flow separation and form resistance. As C_P decreases, so does parallel body, and the entrance is made longer to reduce the increase in wave making resistance, the LCB moving aft in consequence. Most of the reduction in C_P is thus accomplished by fining the entrance, the change in the coefficient of the run being much less.

In designing a new ship, systematic series of data for comparisons among a number of choices of hull form and proportions are available in the technical literature. Such a well known standard series is the Taylor series developed by Admiral Taylor in the 1930's in DTRC (Experimental Model Basin, EMB, at the time). The original parent hull was patterned after a British cruiser with the scary name *Leviathan*. The sectional area curves and body lines for the other models were derived from the parent partly by mathematical means. The lines of the parent form are shown in Figure 1.8. The midship section coefficient was 0.925. The prismatic coefficients of the fore and aft bodies were equal, and the LCB was always amidships. The quantities varied were C_P, B/T, and $W/(L/100)^3$, the midship section coefficient C_M remaining constant. The ranges of the variables covered in the Taylor standard series are (dimensionless or British units):

$$C_P = 0.48 \text{ to } 0.86$$

$$B/T = 2.25, 3.00, \text{ and } 3.75$$

$$W/(L/100)^3 = 20 \text{ to } 250$$

$$\nabla/L^3 = 0.70 \text{ to } 8.75 \times 10^{-3}$$

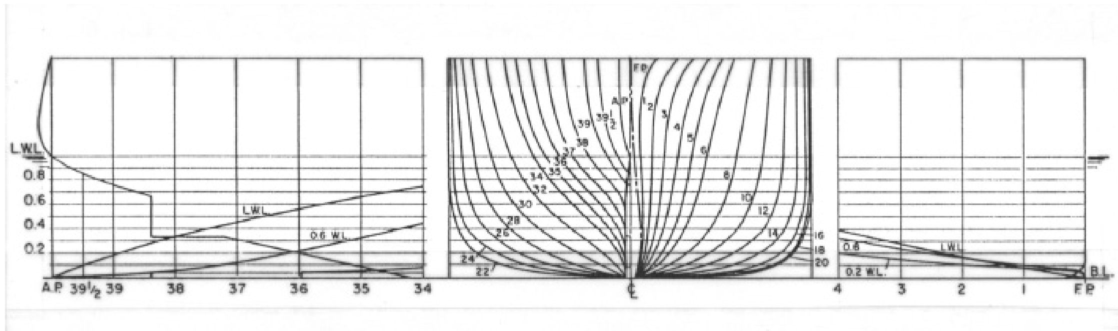

Figure 1.8 Lines for the parent form of Taylor standard series.

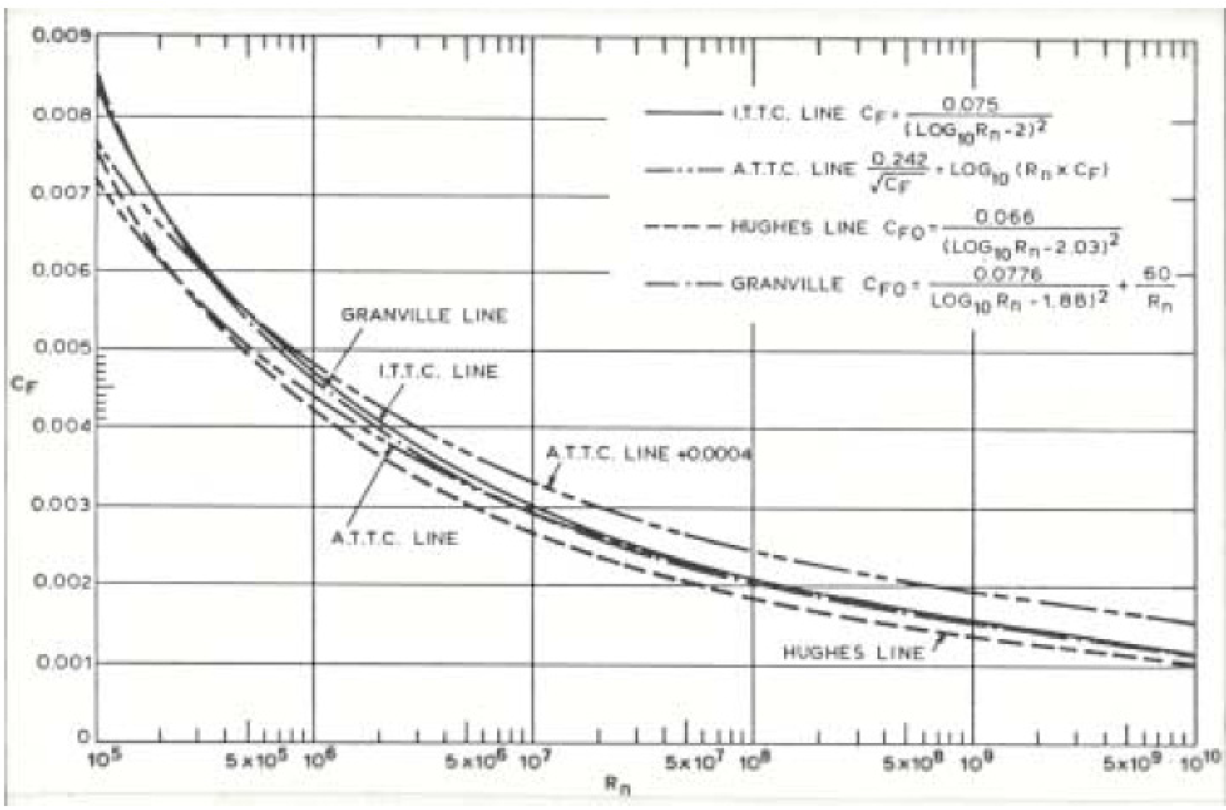

Figure 1.9 Standard skin friction lines.

The design charts give contours of the residual resistance coefficient C_R against $Vs/\sqrt{gL}$ for various values of ∇L^3 each chart being for a particular value of C_P and B/T, and a typical set is shown in Figure 1.10. In conjunction with frictional resistance coefficients (Figure 1.9) and an appropriate allowance coefficient, they can be used to provide design estimates of the total ship resistance.

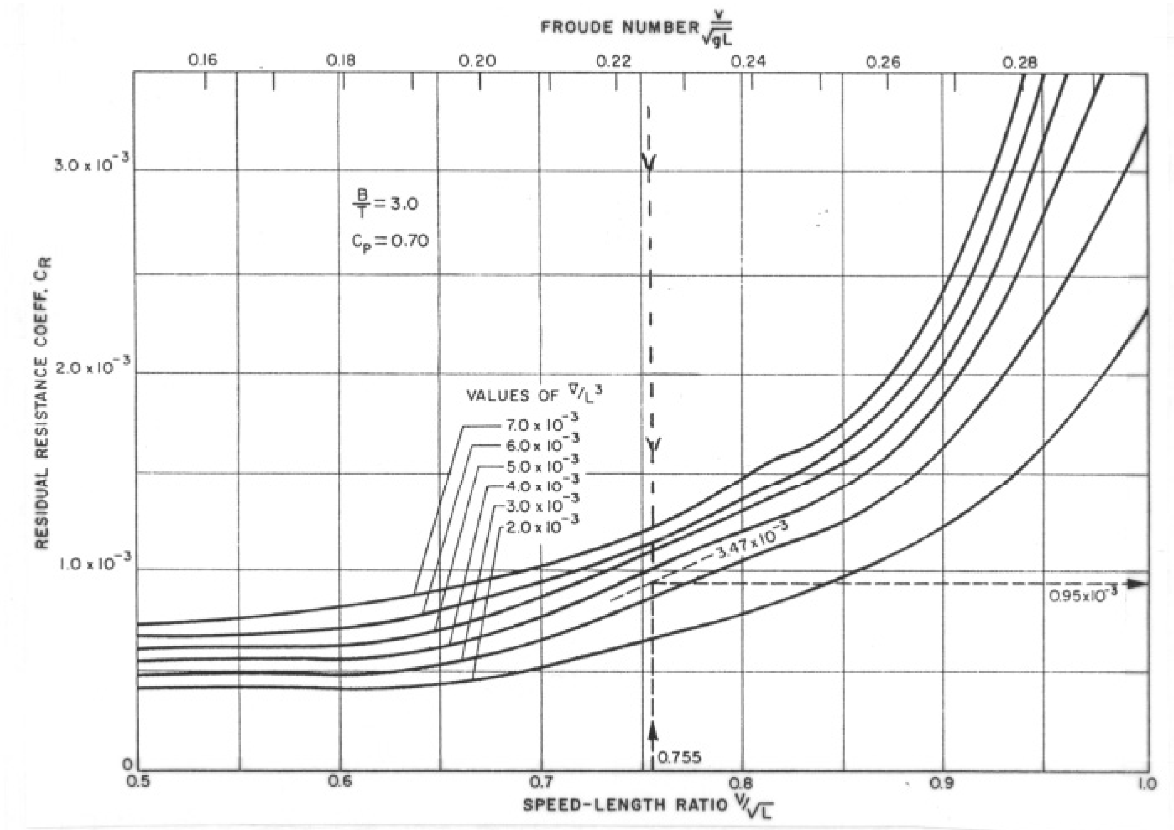

Figure 1.10 Typical Taylor standard series contours.

In using the Taylor series results it should be borne in mind that the models have a deep cruiser stern suitable for a twin screw propulsion arrangement. Also the LCB location has not being optimized but is stationed at amidships. Other systematic ship resistance series include the following:

a. Series 60: Very popular series developed by the Society of Naval Architects and Marine Engineers in cooperation with the ATTC. It is based on a single screw merchant parent and includes data on LCB variations, trim effects, and some propulsive data. It has a narrower range than Taylor series. Many other results have been developed for this series.

b. BSRA Series: This has resulted from a long series of tests with single screw merchant type hulls. It was developed by the British Ship Research Association in the 1960's. A comparison between Taylor, Series 60, and BSRA is shown in Table 1.1. Here $C_\Delta = W/(0.01L)^3$ and L_P is the length of the parallel midbody as percent of the length between perpendiculars

c. SSPA Series: Developed by the Swedish State Shipbuilding Experimental Tank in the 1950's and includes data for high speed, twin screw cargo liners; fast, single screw cargo shipmodels; tankers; and single screw cargo ships.

d. NPL Series: Developed by the National Physical Laboratory, England, it contains data for coaster models and high speed displacement crafts.

e. Form data Series: Developed in Denmark fairly recently (in the 1970's), it is an attempt to combine the previous main series, Taylor, 60, SSPA, and NPL.

	Taylor	Series 60	BSRA
Variables	$C_P, L/B, B/T, C_\Delta$	$C_B, L/B, B/T, C_\Delta$, LCB	$C_B, B/T, C_\Delta$, LCB, L_p
$V/\sqrt{L}$	0.5 to 2.0	0.4 to 1.0	0.4 to 0.85
C_B	0.4396 to 0.8018	0.60 to 0.80	0.65 to 0.85
L/B	3.92 to 17.25	5.5 to 8.5	6.89 and 7.27
B/T	2.25 to 3.75	2.5 to 3.5	2.12 to 3.95
C_Δ	26.5 to 221.6	68 to 302	114 to 385
LCF	0	−2.5 to 3.5	−0.5 to 4.05
L_p	0	0	0 to 50
propulsion	none	η_D, w, t, η_R	w, t, η_R

Table 1.1 Range of applicability of resistance standard series

CHAPTER 2

MAIN TEST

2.1 Theory behind Ship Modeling and Tank Testing

Tow tank testing of a ship model is the traditional method of determining a ship's total hull resistance and its EHP curves. In this method, a model of the ship's hull is built and towed in a towing tank, measuring hull resistance at various speeds. The model results are then scaled up to predict full-scale hull resistance and EHP.

In order for model test results and full-scale ship predictions to be useable, two fundamental relationships between the model and ship must be met: geometric and dynamic similarity.

2.1.1 Geometric Similarity

Geometric similarity is obtained when all characteristic dimensions of the model are directly proportional to the ship's dimensions. The model is then a scaled version of the real ship – a very accurately scaled version of the ship. The ratio of the length of the ship to the length of the model is typically used to define the scaling factor (_).

$$\text{Scale factor} = \lambda = L_S\ (\text{ft}) / L_M\ (\text{ft})$$

where: L_S = length of the ship

L_M = length of the model

From this it follows logically that the ratio of areas is equal to the scale factor squared and the ratio of volumes is equal to the cube of the scale factor. The characteristic area of importance for modeling is the wetted surface area of the underwater hull (S), and the characteristic volume of importance is the underwater volume of the ship (∇). These relationships are shown below:

$$\lambda^2 = \frac{S_S\ (ft^2)}{S_M(ft^2)}$$

$$\lambda^3 = \frac{\nabla_S\ (ft^3)}{\nabla_M(ft^3)}$$

2.1.2 Dynamic Similarity

In addition to geometric similarity between model and full-scale ship, there must also be dynamic similarity between the model and its environment and the full-scale ship and its environment. Dynamic similarity means that the velocities, accelerations, and forces associated with fluid flow around both the model and full-scale ship have scaled magnitudes and identical directions at corresponding locations along the hull. The model must behave in exactly the same manner as the full-scale ship.

Unfortunately, it is physically impossible to achieve true dynamic similarity between the model and full-scale ship. Resistance is a function of velocity, water and air pressure, kinematic viscosity of water (ν), air and water density, and the acceleration due to gravity. It is impossible to scale gravity (think of a model having a scale ratio of 36 … now, try to establish a lab environment whose acceleration of gravity is $1/36^{th}$ of 32.17 ft/sec^2). Similarly, it is impossible to scale water and its properties. Anyone who has seen Hollywood movies with ships at sea can appreciate this; the large globs of water "spray" coming from a model are not seen on a full-scale ship. Two fluids that come close to being scale versions of water are gasoline and liquid mercury; both of which pose serious health and safety issues.

So, if true dynamic similarity cannot be achieved, how can towing tanks exist, let alone produce meaningful results? The answer lies in achieving partial dynamic similarity between model and ship and Froude's "Law of Comparison", also referred to as the "Law of Corresponding Speeds".

2.1.3 The Law of Comparison and Tow Tank Testing

In previous sections of this chapter, we discussed ship resistance and ship performance in terms of dimensionless coefficients:

$$C_T = C_V + C_W$$

where: C_T = coefficient of total hull resistance

C_V = coefficient of viscous resistance

C_W = wave making coefficient

In an ideal world when comparing a geometrically similar ship and model, the coefficients of total resistance, viscous resistance, and wave making resistance would be equal. However, due to the viscous effects of water, this is not possible. The question is how to effectively take model data and calculate a coefficient of total hull resistance for the full-scale ship. This question was answered by Froude through his research on ship performance.

After many towing tank tests, Froude noticed that the wave pattern produced by a geometrically similar model and ship looked the same when the model and ship were traveling at the same speed to square root of length ratio. This is the Law of Corresponding Speeds, and is written as:

$$\frac{V_S}{\sqrt{L_S}} = \frac{V_M}{\sqrt{L_M}}$$

where: V_S = ship velocity (ft/s)

V_M = model velocity (ft/s)

L_S = ship length (ft)

L_M = model length (ft)

Because the wave patterns of the model and ship were similar using this relationship, Froude determined that it would be correct to use the same value of wave making coefficient (C_W) for both the model and ship when operating under these conditions, and therefore partial dynamic similarity between model and ship could be obtained. This can be summarized in the following mathematical relationships:

$$C_{WS} = C_{WM}$$

If,

$$V_V = \frac{V_S}{\sqrt{L_S}}\sqrt{L_M} = \lambda^{-1/2} V_S$$

The purpose of towing tank testing is to tow the model at speeds that correspond to full-scale ship speeds, measure the model's resistance and determine the model's coefficient of wave making resistance. Knowing that the coefficient of wave making resistance of the model and full-scale ship are equal, one can easily determine the coefficient of total hull resistance for the ship. Once the full-scale resistance coefficient is known, the total hull resistance and EHP for the ship are calculated.

To summarize, resistance testing of a model in a towing tank utilizes the following generalized procedure:

- Determine the full-scale ship speed range for the test: minimum ship speed to a desired maximum speed.
- Determine towing speeds for the model using the Law of Comparison.
- Tow the model at each speed, recording the total hull resistance of the model.
- Determine the coefficient of total hull resistance for the model at each speed.
- Determine the coefficient of viscous resistance for the model at each speed.
- Calculate the wave making coefficient of the model at each speed
- $C_{WS} = C_{WM}$
- Determine the coefficient of viscous resistance for the ship at ship speeds corresponding to model towing speeds.
- Determine the coefficient of total hull resistance for the ship at each speed.

- Determine the total hull resistance of the ship for each speed.
- Determine and plot the effective horsepower of the ship at each speed

Once the full-scale EHP curve is known, a similar shaft horsepower curve can be determined based on the assumed propulsive coefficient. The bottom line of EHP testing in the towing tank is to determine the amount of shaft horsepower that must be installed in the full scale ship in order to drive at its maximum speed. Once the maximum shaft horsepower is determined, the physical size and weight of the ship's propulsion plant can be resolved as well as the fuel storage requirements based on the expected steaming range (miles) of the ship. These factors are important in estimating the location of the ship's center of gravity as well as the design of the ship's structure.

2.2 Resistance test

Resistance test determine the resistance of the ship without propeller (and often also without other appendages; sometimes resistance test are performed for both the 'naked' hull and the hull wit appendages). Propulsion tests are performed with an operating propeller and other relevant appendages. A problem is that the forces on appendages are largely driven by viscosity effects with small to negligible gravity effects. As Reynolds similarity is violated, the force cannot be scaled easily to full scale. For ships with large and unusual appendages, the margins of errors in prediction are thus much larger than for usual hulls where experience helps in making appropriate corrections.

The models are towed by weight and wires (Figure 2.1). The main towing force comes from the main weight G_1. The weight G_2 is used for fine tuning:

$R_T = G_1 \pm G_2 \sin \alpha$

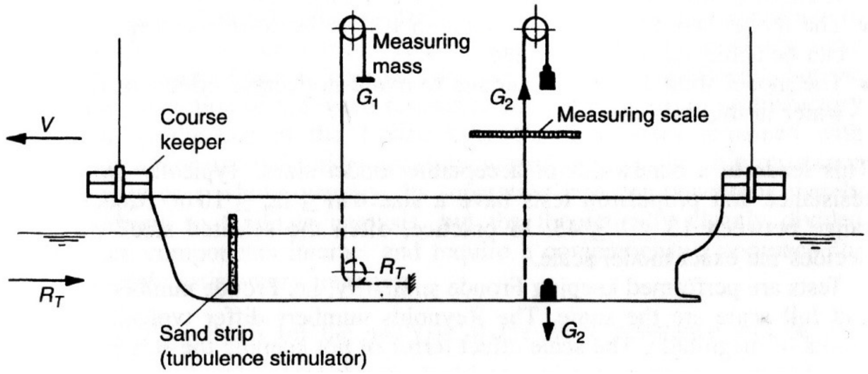

Figure 2.1 Experimental set- up for resistance test

The sign positive if the vertical wire moves aft. The angle α is determined indirectly by measuring the distance on the length scale. Alternatively, modern experimental techniques also use strain gauges as these do not tend to oscillate as the wire-weight systems.

The model test gives the resistance (and power) for towing tank conditions:

- (usually) sufficiently deep water
- no seaway
- no wing
- fresh water at room temperature

This model resistance has to be converted for a prediction of the full-scale ship.

To do this conversion several methods are outlined in the following chapters, namely:

- Method ITTC 1957
- Method of Hughes/ Prohaska
- Method ITTC 1978
- Geosim method of Telfer

CHAPTER 3

CONCLUSION

The total resistance can be divided into two main components viz the viscous and wave drag. Each component is subjected to its own law of scaling, viz. Reynolds and Froude's law respectively. For practical reason both laws cannot be applied simultaneously and consequently corrections have to employed for model ship extrapolation. The flow behavior has to be recognized in order to prevent inconvenient scale effects. Laminar and turbulent flow are the main parameters for the generated amount of viscous resistance whilst the hull shapes with the given speed are criteria for the wave making resistance. The description of the flow just behind the vessel is defined as the difference between the ship's speed and speed acting locally in the plane of the absented propeller (nominal wake).

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