Montogue

Quiz NA103
Ship Resistance and Propulsion – Part 1

Lucas Montogue

Problems

Problem 1A

A ship of 100 m length has a speed of 12 kn. Determine the corresponding speed for a similar 5 m model.

A) V_m = 2.68 kn

B) V_m = 5.34 kn

C) V_m = 7.21 kn

D) V_m = 9.32 kn

Problem 1B

A ship of 12,000 t displacement has a speed of 15 kn. Determine the corresponding speed for a similar ship of displacement 2400 t using geometric and Froude similarity.

A) V_2 = 6.51 kn

B) V_2 = 7.33 kn

C) V_2 = 11.5 kn

D) V_2 = 14.8 kn

Problem 2

Explain why dynamic similarity cannot be achieved in model tests. Which dimensionless parameter is usually kept constant in ship similarity experiments? Why is this quantity maintained constant to the detriment of other dimensionless parameters?

Problem 3

Regarding the discussion of different models of frictional resistance of a ship, which of the following is *true*?

A) Despite its age, Froude's theory for flow over a flat plate continues to be an accurate model for some designs, especially long bulk carriers, tankers, and other vessels with bulky shape.

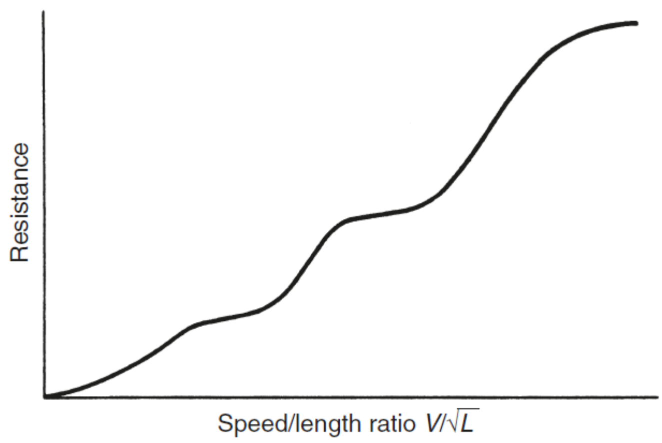
B) The 1957 ITTC formula, now considered a mainstay in ship powering calculations, was derived from the empirical formulas obtained by Froude and his son R.E. Froude in the late 19th and early 20th centuries.

C) One of the advantages of the Schoenherr approach over preceding resistance formulations – such as Froude's – is that it possesses a better theoretical foundation, having been derived from von Karman's theory for flow across a flat plate.

D) The most serious alternative to the most widely used formulas is the Grigson approach, which provides a single efficient expression for friction factor as a function of Reynolds number only.

Problem 4

The curve of wave-making resistance of a typical vessel usually presents "humps" and "hollows" as the speed-length ratio increases. This occurs because of



A) the variations in surface roughness of the ship. Indeed, the humps and hollows in question become more and more pronounced with time as the wetted surface undergoes corrosion and fouling.
B) problems with powering and performance of the vessel. The existence of humps and hollows in the resistance curve indicate that the propulsion system is faltering and operating in an intermittent fashion. The problem is particularly evident for vessels with twin-screw propulsion systems.
C) The interference patterns created by the bow wave system, the stern wave system, and other waves the ship encounters in its path. Constructive interference leads to the formation of humps in the wave resistance curve, while destructive interference is associated with the formation of hollows.
D) The fact that the metacentric height is far too low, causing the ship to become unstable and operate in less-than-optimal conditions. This leads to the formation of a wave-making resistance curve characterized by peaks and valleys, depending on the speed of the vessel and the value of GM.

Problem 5

True or false?

1.( ) Froude-Rankine momentum theory is exceedingly limited to design a modern, aerofoil-based propeller, as it does not model rotative and viscous losses, and hence does not provide important design parameters such as the propeller ideal efficiency.

2.( ) According to Froude-Rankine momentum theory, for a given thrust and speed of advance, increasing propeller area causes the ideal propeller efficiency to increase as well.

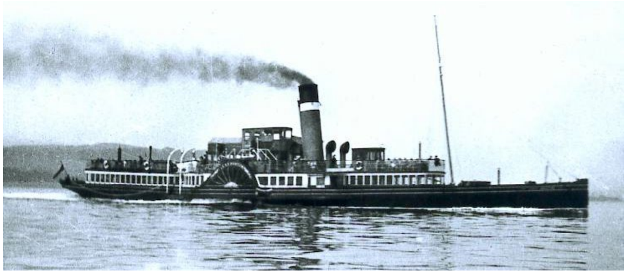
3.( ) Use of a bulbous bow reduces the wave-making resistance of a ship, but this comes at the cost of an increased frictional resistance due to the additional wetted area. Bulbous bows are not suitable for all vessels, as demonstrated by several unsuccessful attempts to use bulbous bows on sailing yachts.

4.( ) Drag due to roughness is in fact a type of separation drag that occurs behind each individual item of roughness. Despite being associated with poor hydrodynamic performance, turbulent boundary layers are accompanied with a thin laminar sublayer close to the surface which can smooth out the surface by flowing round small roughness without flow separation. Roughness only increases drag if it is large enough to project through the sublayer.

5.( ) In order to model the appendage resistance of a vessel, the ITTC recommends associating the drag coefficient of a model to that of a ship by means of an expression of the form

$$C_{D,\text{ship}} = \beta C_{D,\text{model}}$$

where β is a factor that generally varies from 0.5 to 1.0. One of the advantages of this approach, relative to other formulations of appendage drag, is that it is based on a large number of scaling models. One such source of data is the set of measurements obtained by the British Ship Research Association for the Lucy Ashton vessel (shown below), which, albeit having been developed in the 1950s, has been proven time and again to be a highly consistent set of appendage drag measurements.



Problem 6

A 5500 tonnef destroyer develops a total power of 41.4 MW at 30 knots. Assuming that the effective power is 50 percent of the total power, calculate the resistance of the naked hull.

A) R= 380 kN

B) R = 650 kN

C) R = 990 kN

D) R = 1340 kN

Problem 7 (Bertram, 2012, w/ permission)

A 3-m diameter screw propeller is required to produce a thrust of 730 kN at a speed of 23.8 kN in seawater. Use momentum theory to estimate the minimum power which must be supplied to the propeller.

A) P = 8.22 MW

B) P = 11.3 MW

C) P = 14.2 MW

D) P = 17.1 MW

Problem 8 (Bertram, 2012, w/ permission)

A ship with L = 122 m, B = 19.80 m, T = 7.33 m, and Δ = 8700 t has the following power requirements.

V (kn)	16	17	18	19	20
P (kW)	2420	3010	3740	4620	5710

Estimate the power requirements for a similar ship with 16,250 t displacement and speed of 19.5 kn.

A) P = 5090 kW

B) P = 6080 kW

C) P = 7150 kW

D) P = 8200 kW

Problem 9 (Tupper, 2005)

Consider a hypothetical vessel with the following characteristics.

Ship length	140 m
Beam	19 m
Draft	8.5 m
Speed	17 knots
Block coefficient	0.65
Midship area coefficient	0.98
Wetted surface area	3250 m^2
Density of seawater	1.025 g/cm^3
Kinematic viscosity of seawater	1.19×10^{-6} m^2/s

Tests on a geometrically similar model with length of 5.6 m, ran at a corresponding speed, gave a total resistance of 34.5 N in fresh water of density 1 g/cm^3 and kinematic viscosity 1.14×10^{-6} m^2/s. Estimate the resistance of the full-scale ship. Use the following equation to provide an allowance for roughness and other phenomena not accounted for elsewhere. Use $k_s = 150\times10^{-6}$ as an estimate of sand grain height.

$$\Delta C_F = \left[105\left(\frac{k_S}{L}\right)^{1/3} - 0.64\right]\times10^{-3}$$

A) $R_{T,S}$ = 408 kN

B) $R_{T,S}$ = 613 kN

C) $R_{T,S}$ = 805 kN

D) $R_{T,S}$ = 1020 kN

Problem 10 (Patterson & Ridley, 2014)

A 5 m long model, scale 1:20, is used to determine the resistance of a ship. The model is tested at a speed of 1.5 m/s. The f coefficient for the model, that is, the friction coefficient in Froude's formulation of resistance, is 1.670 for the model and 1.408 for the ship. The wetted surface area of the model is 5.4 m^2, and its measured total resistance is 26.1 N. Determine the total resistance of the ship at the same Froude number.

A) $R_{T,S}$ = 77 kN

B) $R_{T,S}$ = 157 kN

C) $R_{T,S}$ = 240 kN

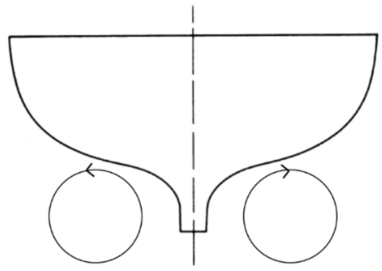
D) $R_{T,S}$ = 321 kN

Problem 11

True or false?

1.( ) Over the years, there have been a number of standard series of propellers tested in many different establishments around the world. The Wageningen B-screw series, developed in the Netherlands, is the most widely used propeller series for design and analysis purposes. It was originally conceived for fixed-pitch, non-ducted propellers.

2.( ) Propellers can be designed to turn in either direction when producing an ahead thrust. In a twin-screw ship, the starboard propeller is normally right-handed and the port propeller left-handed – that is, they turn as illustrated below.



3.( ) Controllable pitch propellers grew in popularity in the last fifty years, having gone from a small proportion of devices manufactured to a very expressive market share of about 35 percent in the early 2010s. One of the main advantages of these devices, relatively to their fixed pitch counterparts, is their superior maneuverability. Indeed, if the blade rotation is high enough, the propeller can produce astern thrust by itself, so that there is no need for a reversing gearbox.

4.( ) Furthermore, CPPs use simpler control mechanisms in the hub region than do their fixed pitch counterparts, thus allowing the engineer to employ a smaller boss and thereby optimize cavitation performance at the root of the blades.

5.( ) In a podded propeller, the propulsor system is attached to a streamlined body of revolution (pod) by a vertical strut extending downward from the hull of the ship. One of the advantages is the saved space within the hull, which would otherwise be reserved for part of the propulsion machinery.

6.( ) On the other hand, the presence of the pod near the propeller disturbs the inflow of water coming towards it, disrupting the wake and causing greater noise, hull vibration and, ultimately, cavitation.

7.( ) Contra-rotating propulsion systems have greater efficiency than do single-screw propellers, but the overall gain in power is counteracted by greater mechanical losses in shafting.

8.( ) Contra-rotating propulsion systems have the hydrodynamic advantage of recovering part of the slipstream rotational energy which would otherwise be lost to a conventional single-screw system. In order to maximize efficiency, both propellers are usually designed with the same diameter and number of blades.

Problem **12**

A propeller absorbs a delivered power of 7500 kW in open water at 120 rpm, the speed of advance being 12.0 knots. Using the B_P-δ diagram given in the Additional Information section, determine the optimum diameter of the propeller and the propeller thrust.

A) D = 3.2 m and T = 405 kN

B) D = 3.2 m and T = 721 kN

C) D = 5.8 m and T = 405 kN

D) D = 5.8 m and T = 721 kN

Problem **13A** (Molland et al., 2017)

A cargo vessel has L = 120 m, B = 20 m, and T = 9 m. For a given power, the vessel travels at 10 knots in deep water. Determine the speed loss when the vessel is traveling at the same power in water of infinite breadth and depth h = 13 m.

Hint: use the Lackenby formula,

$$\frac{\Delta V}{V} = 0.1242\left(\frac{A_M}{h^2} - 0.05\right) + 1 - \left[\tanh\left(\frac{gh}{V^2}\right)\right]^{0.5}$$

A) ΔV = 0.79 kn

B) ΔV = 1.26 kn

C) ΔV = 1.75 kn

D) ΔV = 2.18 kn

Problem **13B**

Suppose now that the ship is located in a river of breadth of 200 m and depth of water of h = 13 m when traveling at the same power as in deep water at 8 knots. Determine the speed loss.

Hint: use the approximation

$$\frac{V_h}{V_\infty} = 1 - 0.09 \times \left(\frac{\sqrt{A_M}}{R_H}\right)^{1.5}$$

A) ΔV = 0.58 kn

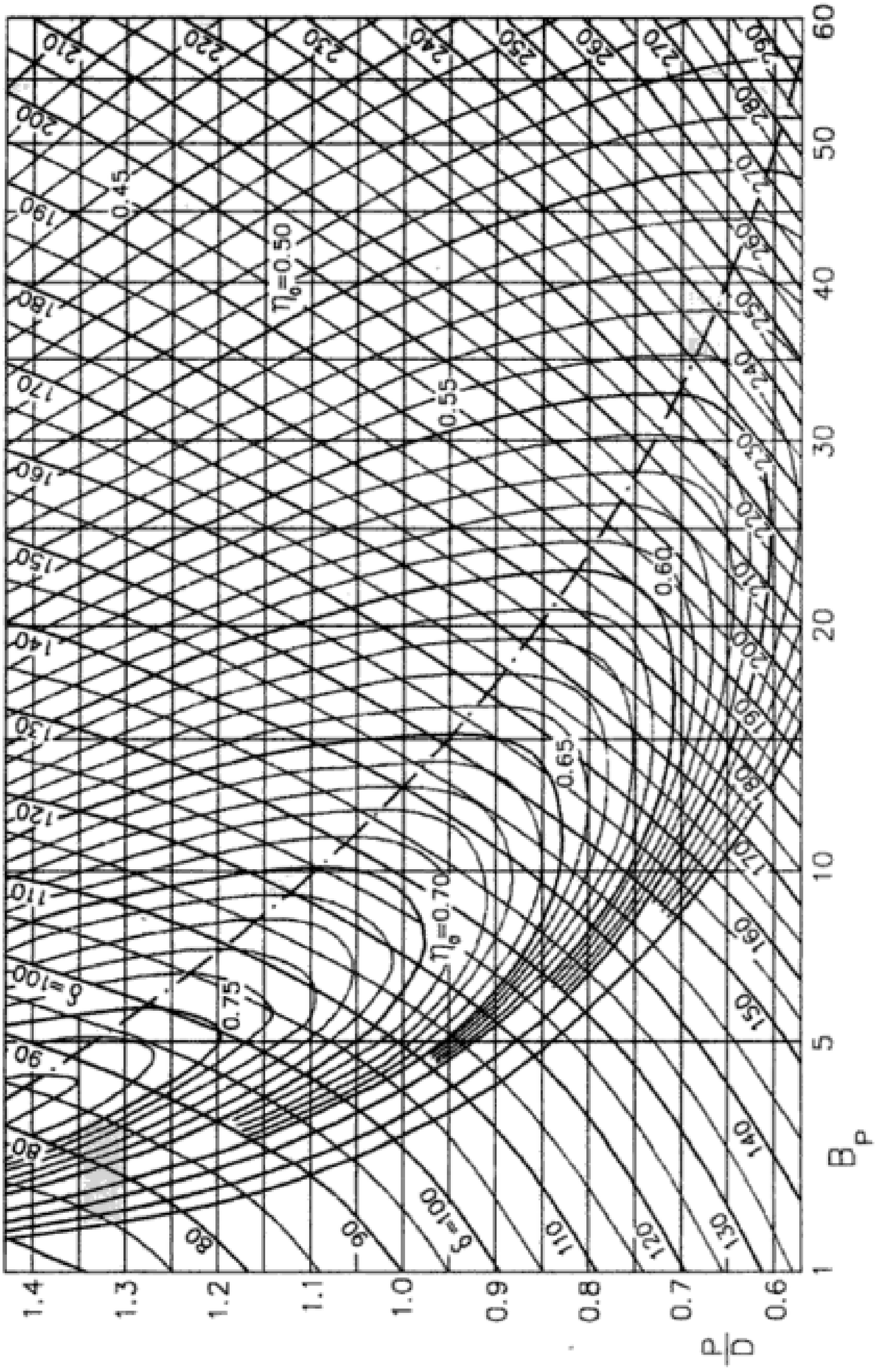
B) ΔV = 1.07 kn

C) ΔV = 1.59 kn

D) ΔV = 2.16 kn

Additional Information

Figure 1 Propeller design chart.



Solutions

P.1 ■ Solution

Part A: The velocity of the model is easily determined with Froude similarity.

$$\frac{V_m}{\sqrt{gL_m}} = \frac{V_S}{\sqrt{gL_S}} \rightarrow V_m = V_S \times \sqrt{\frac{L_m}{L_S}}$$

$$\therefore V_m = 12 \times \sqrt{\frac{5}{100}} = \boxed{2.68 \text{ kn}}$$

★ The correct answer is **A**.

Part B: Let us denote the ship with 12,000 t displacement with a subscript 1, and the ship with a 7000 t displacement using a subscript 2. The lengths *L* of the ships correlate to the displacements Δ in accordance with geometric similarity, namely,

$$\frac{L_1^3}{L_2^3} = \frac{\Delta_1}{\Delta_2} \rightarrow \frac{L_1}{L_2} = \frac{\sqrt[3]{\Delta_1}}{\sqrt[3]{\Delta_2}}$$

The speeds follow from Froude similarity,

$$\frac{V_1}{\sqrt{gL_1}} = \frac{V_2}{\sqrt{gL_2}} \rightarrow V_2 = \sqrt{\frac{L_2}{L_1}} \times V_1$$

$$\therefore V_2 = \sqrt{\frac{L_2}{L_1}} \times V_1 = \left(\frac{\Delta_2}{\Delta_1}\right)^{1/6} \times V_1 = \left(\frac{2400}{12,000}\right)^{1/6} \times 15 = \boxed{11.5 \text{ kn}}$$

★ The correct answer is **C**.

P.2 ■ Solution

For two geometrically similar forms, complete dynamic similarity can only be achieved if the Froude number and the Reynolds number are both equal for the two bodies. This would require $V/(gL)^{0.5}$ and VL/ν to be the same for model and ship. This cannot be achieved for two vessels of different size running in the same fluid. As a result, dynamic similarity cannot be achieved and the engineer has to choose which dimensionless parameter will be held constant. Suppose the Reynolds number were chosen. In this case, we'd have

$$\text{Re}_1 = \text{Re}_2 \rightarrow \frac{V_1 L_1}{\nu_1} = \frac{V_2 L_2}{\nu_2}$$

$$\therefore \frac{V_1}{V_2} = \cancel{\frac{\nu_1}{\nu_2}} \times \frac{L_2}{L_1}$$

$$\therefore \frac{V_1}{V_2} = \frac{L_2}{L_1}$$

where we have cancelled out kinematic viscosity because the fluid medium is water in both cases. (Truthfully, however, the fluid is generally dock water for the model and seawater for the vessel, but even so the difference in kinematic viscosity can be neglected for this simple thought experiment.) If the model scale is, say, $L_2/L_1 = 25$, then $V_1/V_2 = 25$, i.e., the model test would be run at a speed 25 times greater than that of the ship. This is highly impractical. Suppose, instead, that the Froude number were kept unchanged. Mathematically, we'd have

$$\text{Fr}_1 = \text{Fr}_2 \rightarrow \frac{V_1}{\sqrt{gL_1}} = \frac{V_2}{\sqrt{gL_2}}$$

$$\therefore \frac{V_1}{V_2} = \sqrt{\frac{L_1}{L_2}}$$

In our example, $L_1/L_2 = 1/25$ and, consequently, $V_1/V_2 = 1/5$. That is, the model would be tested at a speed of one fifth the speed of the real vessel, which is a feasible solution. In summary, dynamic similarity cannot be achieved because both the Reynolds and Froude numbers would have to be the same for model and ship. Since only one parameter can be held constant, the Froude number is the usual choice.

P.3 ■ Solution

The accuracy of Froude's approach is lost *especially* in the case of long vessels, particularly those with length greater than 500 ft (152 m) or more; in these cases, the Froude approach may overestimate the power of the vessel by as much as 15%. The 1957 ITTC formula was devised from the work carried out by Schoenherr in the 1920s, and as such is not based on the Froude model. The Grigson formula is indeed an accurate alternative to the ITTC and Schoenherr formulas, but it is not cast as a single expression – rather, the equation to be used depends on the range of Reynolds number being considered.

★ The correct answer is **C**.

P.4 ■ Solution

The peculiar shape of wave-making resistance curves is not related to surface roughness, powering system performance, or stability conditions. Rather, it is the result of wave interference. The divergent component of the wave system derived from the bow and the stern generally does not exhibit any strong interference characteristics. This is not the case, however, with the transverse wave systems created by the vessel, since these can show a strong interference behavior. Accordingly, if the bow and stern wave systems interact such that they are in phase, a reinforcement of the transverse wave patterns at the stern and large waves are formed in that region, leading to greater wave-making resistance and the formation of a 'hump' in the wave resistance curve. In a similar fashion, if there is destructive interference between the waves, the wave-making resistance faced by the vessel is reduced and this is manifested as a 'hollow' in the wave resistance curve. Usually, a vessel is designed to operate in conditions associated with a hollow segment of the curve, although other considerations may override this design principle.

★ The correct answer is **C**.

P.5 ■ Solution

1. False. While it is true that momentum theory does not account for rotative and viscous losses (after all, it substitutes the complex geometry of the propeller with a simplistic actuator disk), it must be noted that the ideal efficiency is one of the few useful parameters that this approach *does* provide.

2. True. As per momentum theory, a propeller's ideal efficiency is given by

$$\eta_i = \frac{2}{1+\sqrt{1+C_T}}$$

where C_T is the thrust coefficient, which is such that

$$C_T = \frac{T}{\frac{1}{2}\rho A_0 V_A^2}$$

It can be appreciated, then, that increasing the propeller's cross-sectional area A_0 will cause the thrust coefficient to reduce, which in turn raises the ideal efficiency η_i. This is about as much as momentum theory offers in regard to propeller geometry as possible. Momentum theory does not consider matters such as propeller pitch, blade geometry, and number of blades.

3. True. As in the case of other hydrodynamic adaptations, endowing a ship with a bulbous bow is a double-edged sword: on the one hand, the hydrodynamic shape reduces the overall wave-making resistance; on the other hand, the additional wetted area implies a greater frictional resistance. For use of this structure to be viable, then, the penalty in friction must be smaller than the ensuing decrease in residuary resistance. Bulbous bows tend to work best with vessels at moderate Froude numbers, with U- and V-shaped sections forward and high block coefficients.

4. True. Indeed, turbulent boundary layers coexist with a thin laminar sublayer located close to the surface. This layer smooths out a rough surface, preventing it from disturbing the flow that surrounds the hull. Consequently, roughness will only add to overall drag if the roughness of the hull material is capable of projecting outward from the viscous sublayer.

5. False. The statement has two questionable points: first, there is a small amount of actual data on appendage scaling, and there is no evidence that the ITTC approach is based on more data than other methods of computation of appendage drag; second, the Lucy Ashton measurements have not always proven to be a reliable set of data, with Molland & Hudson noting that *the resulting information from the* Lucy Ashton *and other such tests tends to be inconclusive and the data have been reanalyzed many times over the years.*

P.6 ■ Solution

The effective power is calculated as

$$P_E = 0.50 \times 41.4 = 20.7 \text{ MW}$$

Recalling that power equals force times velocity, we have

$$P_E = R \times V \rightarrow \left(20.7 \times 10^6\right) = R \times \left(30 \times 0.5144\right)$$

$$\therefore \boxed{R = 1340 \text{ kN}}$$

★ The correct answer is **D**.

P.7 ■ Solution

The speed of the ship is $V_S = 23.8 \times 0.5144 = 12.2$ m/s and the area of the propeller is $A_0 = (\pi/4) \times 3^2 = 7.07$ m^2. The thrust coefficient is computed as

$$C_{Th} = \frac{T}{\frac{1}{2}\rho V_S^2 A_0} = \frac{730{,}000}{0.5 \times 1025 \times 12.2^2 \times 7.07} = 1.354$$

The ideal efficiency of the propeller is

$$\eta_i = \frac{2}{1 + \sqrt{1 + C_{Th}}} = \frac{2}{1 + \sqrt{1 + 1.354}} = 0.789$$

Lastly, the power that must be supplied to the propeller is

$$P = \frac{T \times V_S}{\eta_i} = \frac{730 \times 12.2}{0.789} = \boxed{11.3 \text{ MW}}$$

★ The correct answer is **B**.

P.8 ■ Solution

We convert the speed of the new ship of 19.5 kn to the corresponding speed for the original ship, following Froude and geometric similarity,

$$\frac{V_1}{\sqrt{gL_1}} = \frac{V_2}{\sqrt{gL_2}} \rightarrow V_1 = V_2\sqrt{\frac{L_1}{L_2}} \rightarrow V_1 = V_2\left(\frac{\Delta_1}{\Delta_2}\right)^{1/6}$$

$$\therefore V_1 = 19.5 \times \left(\frac{8700}{16{,}250}\right)^{1/6} = 17.6 \text{ kn}$$

For this speed, the power for the original ship is linearly interpolated as follows,

$$P_1 = (17.6 - 17) \times \frac{(3740 - 3010)}{18 - 17} + 3010 = 3448 \text{ kW}$$

Scaling to the new ship, we get

$$P = P_1 \times \left(\frac{\Delta_{\text{new}}}{\Delta_{\text{old}}}\right)^{3.5/3} = 3448 \times \left(\frac{16{,}250}{8700}\right)^{3.5/3} = \boxed{7150 \text{ kW}}$$

Bertram notes that the exponent above occurs because power is scaled with speed V^5 and area A, which gives for the speed with Froude similarity a scale with length scale $\lambda^{1.5}$ and for area λ^2, thus amounting to $\lambda^{3.5}$. If we use the displacement instead of length, we divide by λ^3, hence the exponent 3.5/3.

★ The correct answer is **C.**

P.9 ■ Solution

From similarity, the speed of the model is determined as

$$V_M = 17 \times \left(\frac{5.6}{140}\right)^{0.5} = 3.4 \text{ kn} = 1.75 \text{ m/s}$$

The wetted surface of the model, in turn, is

$$S_M = 3250 \times \left(\frac{5.6}{140}\right)^2 = 5.2 \text{ m}^2$$

The speed of the ship is V_S = 17×0.5144 = 8.74 m/s. The Reynolds number for the model is

$$\text{Re}_M = \frac{V_M L_M}{\nu_{\text{fw}}} = \frac{1.75 \times 5.6}{\left(1.14 \times 10^{-6}\right)} = 8.60 \times 10^6$$

while that of the ship is

$$\text{Re}_S = \frac{8.74 \times 140}{\left(1.19 \times 10^{-6}\right)} = 1.03 \times 10^9$$

We can then determine the friction coefficient from the ITTC-1957 correlation line, which is given by

$$C_F = \frac{0.075}{\left(\log_{10} \text{Re} - 2\right)^2}$$

Applying this relation for model and ship, we have respectively

$$C_{F,M} = \frac{0.075}{\left[\log_{10}\left(8.60 \times 10^6\right) - 2\right]^2} = 0.00308$$

and

$$C_{F,S} = \frac{0.075}{\left[\log_{10}\left(1.03 \times 10^9\right) - 2\right]^2} = 0.00153$$

The coefficient of total resistance for the model follows as

$$C_{T,M} = \frac{R_{T,M}}{\frac{1}{2}\rho_{\text{fw}} V_M^2 S_M} = \frac{34.5}{0.5 \times 1000 \times 1.75^2 \times 5.2} = 0.00433$$

The coefficient of residual resistance, which should be the same for model and ship, is given by the difference

$$C_{R,M} = C_{R,S} = 0.00433 - 0.00308 = 0.00125$$

The coefficient of total resistance for the ship is then

$$C_{T,S} = C_{F,S} + C_{M,S} = 0.00153 + 0.00125 = 0.00278$$

We should provide an allowance for roughness and other phenomena not accounted for by other means. With k_s = 150×10^{-6} m, the allowance in question is determined with the correlation coefficient

$$\Delta C_F = \left[105\left(\frac{k_S}{L}\right)^{1/3} - 0.64\right] \times 10^{-3} = \left[105\left(\frac{150 \times 10^{-6}}{140}\right)^{1/3} - 0.64\right] \times 10^{-3} = 0.000434$$

The updated coefficient of total resistance is then

$$C_{T,S} = 0.00278 + \Delta C_F = 0.00321$$

Lastly, the total resistance is calculated as

$$R_{T,S} = \frac{1}{2}\rho_{\text{sw}} V_S^2 S_S \times C_{T,S} = 0.5 \times 1025 \times 8.74^2 \times 3250 \times 0.00321 \approx \boxed{408 \text{ kN}}$$

★ The correct answer is **A**.

P.10 ■ Solution

In view of the scale of 1:20, the length of the ship is L_S = 20×5 = 100 m and the wetted area is S_S = 20²×5.4 = 2160 m². The speed of the ship is determined from similarity of Froude numbers,

$$\frac{V_M}{\sqrt{gL_M}} = \frac{V_S}{\sqrt{gL_S}} \rightarrow V_S = 1.5\times\sqrt{\frac{100}{5}} = 6.71\text{ m/s}$$

The frictional resistance of the model, $R_{F,M}$, is determined with the following equation,

$$R_{F,M} = f_M S_M V_M^{1.825} = 1.670\times 5.4\times 1.5^{1.825} = 18.9\text{ N}$$

The residual resistance of the model is the difference between total resistance and frictional resistance; that is,

$$R_{R,M} = R_{T,M} - R_{F,M} = 26.1 - 18.9 = 7.2\text{ N}$$

The residual resistance of the ship, $R_{R,S}$, can be obtained with Froude's law of similarity,

$$\frac{R_{R,S}}{R_{R,M}} = \frac{\Delta_S}{\Delta_M} \rightarrow R_{R,S} = \left(\frac{L_S}{L_M}\right)^3 \times \left(\frac{\rho_{\text{seawater}}}{\rho_{\text{fresh water}}}\right)\times R_{R,M}$$

$$\therefore R_{R,S} = 20^3 \times \frac{1.025}{1.0}\times 7.2 = 59.0\text{ kN}$$

We proceed to calculate the frictional resistance of the ship, $R_{F,S}$,

$$R_{F,S} = f_S S_S V_S^{1.825} = 1.408\times 2160\times 6.71^{1.825} = 98.1\text{ kN}$$

Finally, the total resistance of the ship is determined as

$$R_{T,S} = R_{R,S} + R_{F,S} = 59.0 + 98.1 = \boxed{157\text{ kN}}$$

★ The correct answer is **B**.

P.11 ■ Solution

1. True. Indeed, the original Wageningen B-series, whose development began with the work of Troost in the late 1940s, was originally intended for fixed-pitch, non-ducted devices. Since then, researchers at MARIN in Wageningen have devised standard model series for other types of propulsors, including ducted propellers and contra-rotating propellers.

2. True. If a propeller turns clockwise when viewed from aft, it is said to be *right-handed*; if anti-clockwise, it is said to be *left-handed*. Indeed, in a twin-screw ship the starboard propeller is normally right-handed and the port propeller left-handed, and the devices operate as illustrated in the foregoing illustration. The propellers are then said to be *outward turning*. One of the advantages of this setup relatively to inward turning propellers is the smaller susceptibility to cavitation.

3. True. Indeed, the adjustable blade profile that is characteristic of CPPs allows the shipmaster to change and redirect lift forces and produce astern thrust without the need for a reversing gearbox. Maneuvering can be faster as the blades' angle can be varied more rapidly than can the shaft revolutions (but there will be an optimum rate of change to produce maximum acceleration or deceleration).

4. False. Truthfully, CPPs tend to have larger, not smaller, hub diameters than do equivalent fixed pitch propellers. This is so because the hub must house the pitch control equipment, which is inherently bulkier and more complex than the transmission machinery of a FPP. According to Carlton, a controllable pitch propeller hub has a diameter in the range 0.24 – 0.32*D*, but for some applications may rise to as much as 0.4*D* or even half the propeller diameter; fixed pitch devices, in contrast, usually have hub diameters no larger than 0.25*D*. In addition to the undesirable added weight and the need for additional material, large boss diameters may give rise to complex hydrodynamic problems, often cavitation related.

5. True. Indeed, the most immediate advantage of a podded propeller is the space saved in the hull as opposed to a traditional propeller system.

6. False. On the contrary, the presence of the propulsive mechanisms under water reduces overall noise and vibration when compared to shaftline-based equipment. What's more, if the propeller is mounted at the leading edge of the pod, a clean and uniform wake is encountered in most cases. The absence of rudders, stern thrusters and other additional mechanisms also contributes to the clean wake.

7. True. Indeed, use of a CRP may increase the overall efficiency of the propulsion system substantially, thus allowing more power to be transmitted for a given propeller radius. The benefits are offset, however, by the additional power required for shafting, which is substantially greater than that of a single-screw propeller solution, not to mention its inherently more complicated mechanical installation and maintenance.

8. False. In order for harvest maximum energy from the incoming flow, contra-rotating propellers are designed with different size and number of blades. Indeed, in marine applications of contra-rotating propulsion, it is normal for the aftermost propeller to have a smaller diameter than the forward propeller and, in this way, better accommodate the slipstream contraction effects. Having propellers with different sizes also helps to prevent problems with cavitation. The blade numbers of the forward and aft propellers are usually different, often four and five for the forward and aft propellers, respectively. The effectiveness of CRPs is a matter of debate, with reported figures of additional fuel efficiency ranging from 6 to as much as 20%. In a more recent assessment of rotational energy losses, other investigators obtained a much more modest range of 3 to 6%.

P.12 ■ Solution

We have P_D = 7500 kW, n = 120 rpm, and V_A = 12.0 kn = 6.17 m/s. In order for us to use the B_P diagram, the delivered power must be converted to British units. Thus, we take P_D = 9812 hp and calculate $B_P = nP_D^{0.5}/V_A^{2.5} = 120\times9812^{0.5}/12.0^{2.5}$ = 23.8. From the optimum efficiency line in the diagram, one obtains P/D = 0.798, δ = 190.5, and η_0 = 0.593. The diameter of the propeller is then

$$D = \frac{V_A\delta}{n} = \frac{12.0\times190.5}{120} = 19.05\text{ ft} = \boxed{5.8\text{ m}}$$

while the propeller thrust is calculated as

$$T = \frac{P_D\eta_0}{V_A} = \frac{7500\times0.593}{6.17} = \boxed{721\text{ kN}}$$

★ The correct answer is **D**.

P.13 ■ Solution

Part A: The length-based Froude number is calculated as

$$Fr = \frac{V}{\sqrt{gL}} = \frac{(10\times0.5144)}{\sqrt{9.81\times120}} = 0.150$$

while the depth-based Froude number is computed as

$$Fr_h = \frac{V}{\sqrt{gh}} = \frac{(10\times0.5144)}{\sqrt{9.81\times13}} = 0.456$$

The maximum cross-sectional area of the hull is A_M = 20×9 = 180 m^2. Substituting this quantity and other parameters into the Lackenby formula, we find that

$$\frac{\Delta V}{V} = 0.1242\left(\frac{A_M}{h^2} - 0.05\right) + 1 - \left[\tanh\left(\frac{gh}{V^2}\right)\right]^{0.5}$$

$$\therefore \frac{\Delta V}{V} = 0.1242\left(\frac{180}{13^2} - 0.05\right) + 1 - \left[\tanh\left(\frac{1}{0.456}\right)^2\right]^{0.5} = 0.126$$

$$\therefore \Delta V = 0.126 \times V = 0.126 \times 10 = \boxed{1.26 \text{ kn}}$$

That is, the restricted depth leads to a decrease in velocity of 1.26 knots, or about 12.5 percent, in the velocity of the craft.

★ The correct answer is **B**.

Part B: The wetted girth is of the hull is $p = B + 2T = 20 + 2\times9 = 38$ m. The hydraulic radius for a rectangular channel such as the present one is given by

$$R_H = \frac{bh - A_M}{b + 2h + p} = \frac{200 \times 13 - 180}{200 + 2 \times 13 + 38} = 9.17 \text{ m}$$

The ratio of the speed in deep water V_∞ to the speed in restricted conditions V_h is calculated as

$$\frac{V_h}{V_\infty} = 1 - 0.09 \times \left(\frac{\sqrt{A_M}}{R_H}\right)^{1.5}$$

$$\therefore \frac{V_h}{V_\infty} = 1 - 0.09 \times \left(\frac{\sqrt{180}}{9.17}\right)^{1.5} = 0.841$$

$$\therefore V_h = 0.841 \times V_\infty = 0.841 \times 10 = 8.41 \text{ kn}$$

The speed of the vessel under conditions of restricted depth and breadth is 8.41 knots, indicating a loss of speed of 1.59 knots, or approximately 16 percent.

★ The correct answer is **C**.

Answer Summary

Problem 1	**1A**	A
	1B	C
Problem 2		Open-ended pb.
Problem 3		C
Problem 4		C
Problem 5		T/F
Problem 6		D
Problem 7		B
Problem 8		C
Problem 9		A
Problem 10		B
Problem 11		T/F
Problem 12		D
Problem 13	**13A**	B
	13B	C

References

- BERTRAM, V. (2012). *Practical Ship Hydrodynamics*. 2nd edition. Oxford: Butterworth-Heinemann.
- CARLTON, J. (2012) *Marine Propellers and Propulsion*. 3rd edition. Oxford: Butterworth-Heinemann.
- GHOSE, J. and R. GOKAM. (2015). *Basic Ship Propulsion*. New Delhi: KW Publishers.
- GILLMER, T. and JOHNSON, B. (1982). *Introduction to Naval Architecture*. London: E. & F.N. Spon.
- MOLLAND, A., TURNOCK, S. and HUDSON, D. (2017). *Ship Resistance and Propulsion*. 2nd edition. Cambridge: Cambridge University Press.

- PATTERSON, C. and RIDLEY, J. (2014). *Ship Stability, Powering and Resistance*. London: Bloomsbury.
- TUPPER, E. (2005). *Introduction to Naval Architecture.* 4th edition. Oxford: Elsevier.

Got any questions related to this quiz? We can help!
Send a message to contact@montogue.com and we'll
answer your question as soon as possible.