Quiz NA104
Ship Resistance and Propulsion – Part 2
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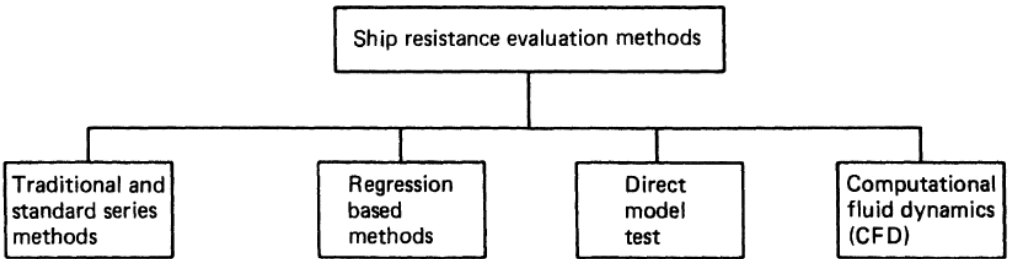
Problems

Problem 1

The following schematic shows the four types of ship resistance evaluation methods. One of these approaches began to be explored most notably with J. Scott's 1972 paper *A method for predicting trial performance of a single-screw merchant ships*, along with a corresponding paper for twin-screw vessels published in the same year. In the 1970s and 1980s, this line of research was reinforced by a series of papers published by J. Holtrop and his collaborators. The ship resistance assessment method in question is:



A) Traditional/standard series method.

B) Regression-based method.

C) Direct model test method.

D) CFD method.

Problem 2

True or false?

1.( ) The hull efficiency is given by the ratio of effective power to thrust power. As in the case of other ship efficiency elements, its value cannot be greater than unity.

2.( ) The relative rotative efficiency is generally assumed to be scale-independent.

3.( ) One of the components of a ship's propulsive efficiency is the open-water efficiency. This parameter can be subdivided in three contributions, namely,

$$\eta_O = \eta_A \eta_R \eta_F$$

Here, η_A is the ideal efficiency, based on axial momentum principles and allowing for a finite blade number, η_R accounts for losses due to fluid rotation induced by the propeller, and η_F accounts for losses due to blade friction drag. Experiments with single-screw models of seasoned propulsive devices, such as the Wageningen B-series, have revealed that friction losses are larger than axial and rotational losses.

4.( ) The evaluation of the propulsion test requires the resistance characteristics and the open-water characteristics of the stock propeller. In this regard, there are two approaches: the *thrust identity approach* and the *torque identity approach*. The standard used by the 1978 ITTC is the torque identity approach.

5.( ) According to the guidelines of the ITTC for ship models in a propulsion test, a self-propelled model will be free to surge, heave, roll and pitch, but sway and yaw motions are restricted.

6.( ) One objective in the design of hydrodynamic systems is the avoidance or at least the delay of cavitation. Cavitation numbers σ are used as a measure to quantify the likelihood of cavitation to occur. In its most popular formulation, the cavitation number is defined as the ratio of the difference between the static pressure in the flow and the vapor pressure, in the numerator, and the dynamic pressure, in the denominator. In this definition, the larger the cavitation number, the more likely cavitation is.

Problem 3 (Zubaly, 2009)

A newly designed bulk carrier 852 ft long displaces 100,000 tons at design draft. Resistance tests have shown that the effective power, including air and appendage allowances, is 12,860 horsepower at a design speed of 16.5 knots. Propulsion factors measured in self-propelled model tests are given below.

Wake fraction	0.325
Thrust deduction factor	0.178
Relative rotative efficiency	1.025

The propeller test revealed that at its design point the propeller's open-water efficiency is 0.685. Shaft transmission efficiency is estimated at 0.990, and a service power allowance of 20 percent has been chosen. From the data given, calculate the ship's propulsive efficiency and service shaft horsepower.

A) η_P = 77.59% and P_{SS} = 15,190 hp

B) η_P = 77.59% and P_{SS} = 18,230 hp

C) η_P = 84.66% and P_{SS} = 15,190 hp

D) η_P = 84.66% and P_{SS} = 18,230 hp

Problem 4 (Molland et al., 2017)

A thrust identity, self-propulsion test analysis was performed on the model of a single-screw ship. The model speed is 1.33 m/s, the propeller diameter D = 0.19 m, and the pitch ratio P/D = 0.80. The following data were measured.

Revolutions at the self-propulsion point	10.0 rps
Towed model resistance corresponding to the self-propulsion point	18.5 N
Thrust at self-propulsion point	22.42 N
Torque at self-propulsion point	0.58 N·m

The model propulsor is a Wageningen series B4.40 propeller with P/D = 0.80, for which the following relationships are suitable.

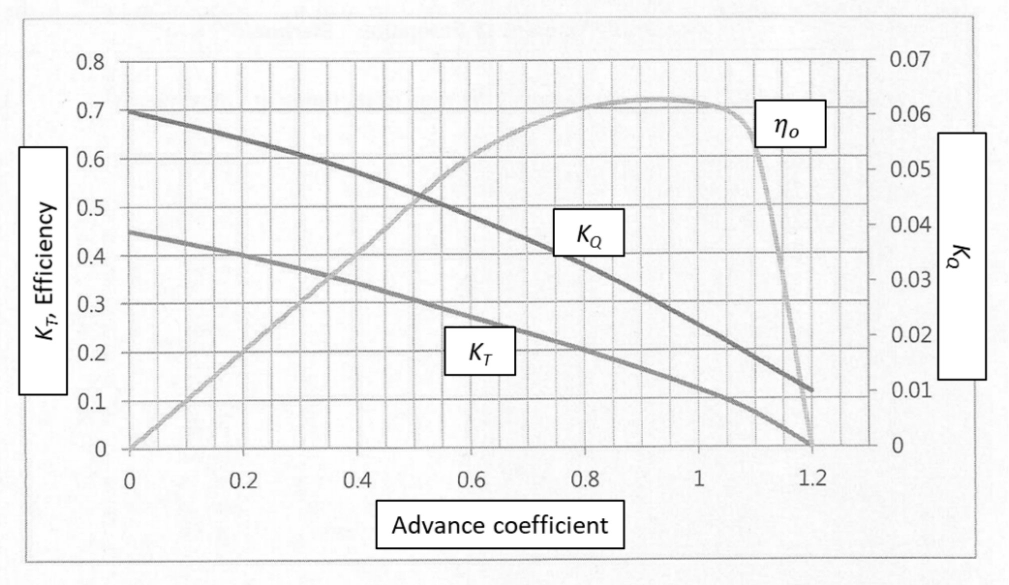
Thrust coefficient	Torque coefficient
$K_T = 0.320\left[1-\left(\dfrac{J}{0.90}\right)^{1.30}\right]$	$K_Q = 0.0360\left[1-\left(\dfrac{J}{0.98}\right)^{1.60}\right]$

True or false?

1.( ) The hull efficiency is greater than 1.25.
2.( ) The relative rotative efficiency is greater than unity.
3.( ) The open water efficiency is greater than 0.55.
4.( ) The quasi-propulsive efficiency is greater than 0.71.

Problem 5

A propeller with diameter of 21 ft is designed to provide 200 LTons thrust at 120 rpm in seawater (density = 1.99 slugs/ft^3). The propeller performance curves are provided below. The open water efficiency of the propeller and the speed of advance are:



A) η_0 = 56% and V_A = 15.5 ft/s

B) η_0 = 56% and V_A = 23.1 ft/s

C) η_0 = 70% and V_A = 15.5 ft/s

D) η_0 = 70% and V_A = 23.1 ft/s

Problem 6

A ship is designed to travel at 18 knots. The effective bare hull resistance at this speed is 300,000 lb. Hull efficiency is 0.99 and wake fraction is 0.31. The thrust required of the propeller at this speed is:

A) T = 150,000 lb

B) T = 235,000 lb

C) T = 325,000 lb

D) T = 430,000 lb

Problem 7

A ship has the following characteristics.

Underway Speed	18 knots
Shaft Horsepower	4500 hp
Propeller RPM	85 rpm
Propeller Diameter	19 ft
Apparent Slip Ratio	0.0
Wake Fraction	0.30

Determine the vessel's real slip ratio.

A) *Real slip* = 0.30

B) *Real slip* = 0.50

C) *Real slip* = 0.70

D) *Real slip* = 0.90

Problem 8A (Bertram, 2012, w/ permission)

Consider the propeller and ship described below.

Density of seawater	1025 kg/m^3
Kinematic viscosity of seawater	1.19×10^{-6} m^2/s
Kinematic viscosity of fresh water	1.14×10^{-6} m^2/s
Wake fraction	0.135
Propeller diameter	4.5 m
Advance number (open water)	0.833
Propeller rotations	3 s^{-1}
Thrust coefficient	0.1594
Relative rotative efficiency	0.95
Open water efficiency	0.684

The ship resistance at design speed is 580 kN. What is the thrust deduction factor?

A) $t = 0.0159$

B) $t = 0.0381$

C) $t = 0.0556$

D) $t = 0.0782$

Problem 8B

Estimate the quasi-propulsive coefficient for the vessel in question.

A) $QPC = 0.535$

B) $QPC = 0.655$

C) $QPC = 0.725$

D) $QPC = 0.845$

Problem 8C

For a 1:16 model of the ship a wake fraction $w = 0.19$ is measured in towing tank tests. What should be the propeller rpm at the model speed corresponding to the full-scale design speed?

A) $n_m = 311$ rpm

B) $n_m = 434$ rpm

C) $n_m = 554$ rpm

D) $n_m = 672$ rpm

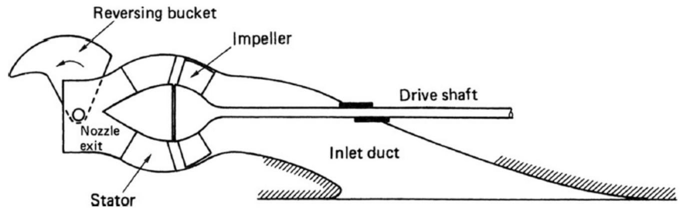
Problem 8D

Compare Reynolds numbers at $0.7R$ for the model and at full scale. The full scale propeller chord c at $r/R = 0.7$ is 2 m.

Problem 9

True or false?

1.( ) One of the advantages of waterjet propulsion systems is that, unlike exposed rotor-based devices such as propellers, the pump mechanism used to maintain the waterjet is free from cavitation. This is so regardless of the aerofoil forms used in the blade design of the pump impellers.



2.( ) In modern waterjet propulsion systems, the steering of the vessel can be performed directly by the waterjet and the accompanying deflector, and as such removes the need for rudders.

3.( ) One important consideration for hydrofoil vessels is the power needed to lift the craft to the foilborne condition. This is not a big concern when waterjet propulsion is used, as these devices have high efficiency at low speeds.

4.( ) Cavitation can also be a problem in hydrofoil vessels. Indeed, cavitation on foils designed for subcavitating conditions can be prevented by placing an upper limit on the vessel speed. If the vessel is to surpass this limit, supercavitating foils must be used. These devices are characterized by lift-to-drag ratios and lift coefficients lower than those of subcavitating foils.

5.( ) T-foils, trim tabs, and interceptors are used as parts of control systems for monohull and multihull semi-displacement vessels. Trim tabs and interceptors are situated at the transom stern, whereas T-foils are in the forward part of the vessel. Because the vertical ship motions are largest in the bow, it is an advantage that a heave and pitch damping device is placed close to the bow. The damping effect of T-foils, trim tabs, and interceptors increases with speed and would not be as efficient for conventional ships operating at slow speeds.

6.( ) With regard to high-speed vessels, it is known that a surface effect ship on cushion has a lower resistance than a similarly sized catamaran, can achieve a higher speed with less total power, and has better seakeeping characteristics in head sea conditions at moderate sea states.

Problem 10A

The resistance of a boat impelled by waterjet is 52,000 N, the boat speed is 65 knots and the jet efficiency is 60%. Calculate the waterjet speed.

A) V_J = 19.8 m/s

B) V_J = 34.4 m/s

C) V_J = 51.5 m/s

D) V_J = 77.9 m/s

Problem 10B

Determine the impeller diameter for the vessel considered in the previous problem.

A) D = 78 mm

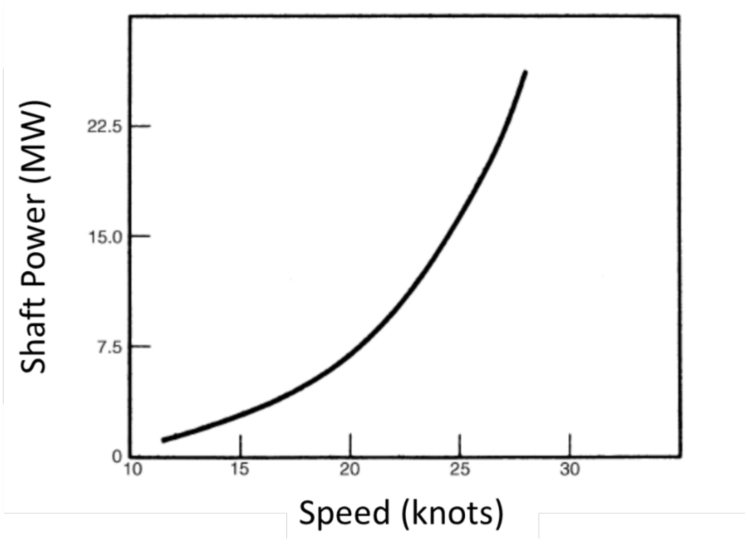
B) D = 101 mm

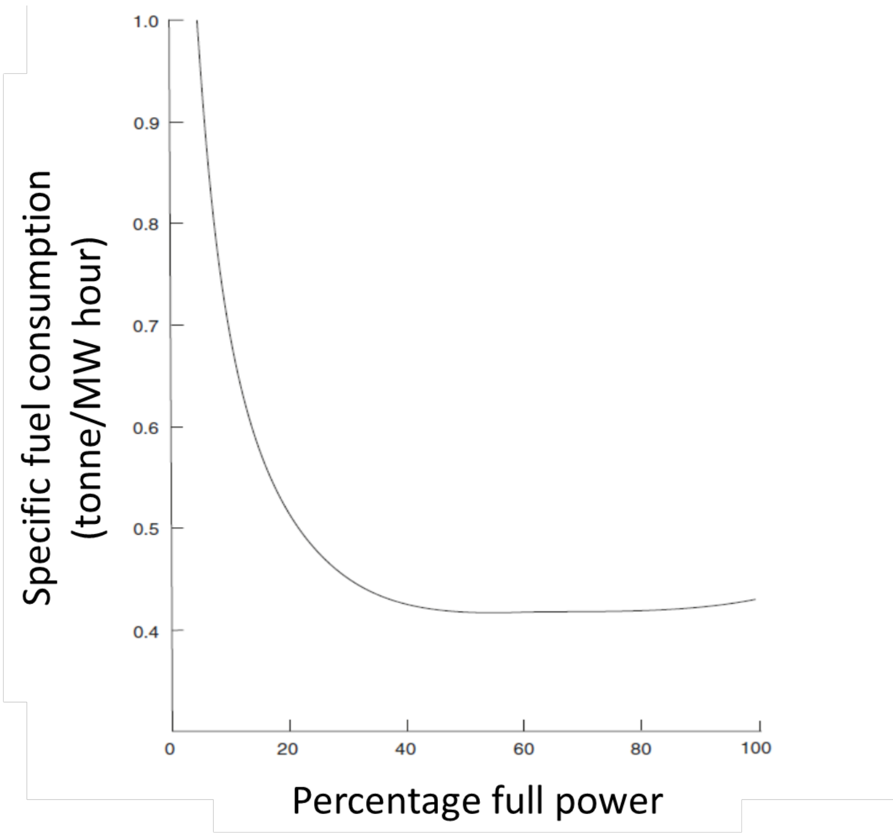
C) D = 134 mm

D) D = 170 mm

Problem 11 (Rawson & Tupper, 2001, w/ permission)

The shaft power speed curve for a given ship with a total installed power of 26.25 MW is shown in the first graph below. The specific fuel consumption for various percentages of fuel power are as shown in the second graph. Calculate the endurance for 1000 tonnef of fuel over a range of speeds and the weight of fuel required for 1000 miles endurance.





Solutions

P.1 ■ Solution

Scott's 1972 papers were a pioneering effort in the study of ship resistance and propulsion by dint of statistical methods. This line of research was then taken up by Holtrop and his collaborators in the mid-1970s and into the following decade, producing at least five important publications on the theme. Examples include 1977's *A statistical analysis of performance test results* and 1988's *Statistical re-analysis of resistance and propulsion data.* Holtrop's work traces the development of a power prediction method based on the regression analysis of random model and full-scale test data together with, most recently, the published results of the Series 64 high-speed displacement hull terms. In this latest version the regression analysis is now based on the results of over 300 model tests.

★ The correct answer is **B**.

P.2 ■ Solution

1. False. While it is true that hull efficiency is the ratio of effective power to thrust power, its value can in fact be greater than unity. Accordingly, η_H is not strictly a hydrodynamic efficiency term, but rather a hydrodynamic coefficient. The hull efficiency can be expressed solely in terms of the thrust deduction factor t and the wake fraction w; that is,

$$\eta_h = \frac{1-t}{1-w}$$

Depending on the dynamics of the propeller inflow and the ship wake, the numerator in the foregoing relation may be greater than the denominator, leading to $\eta_H > 1.0$. The hull efficiency for single-screw vessels takes values typically in the range from 1.05 to 1.10.

2. True. The relative rotative efficiency accounts for the differences in torque absorption characteristics of a propeller when operating at similar conditions in a mixed wake and open-water flows. It can be determined from resistance and propulsion model studies and, in general, is taken to be scale-independent. Its value usually ranges from 0.96 to 1.04. Indeed, of the three parameters obtained in typical propulsion tests – relative rotative efficiency, thrust deduction factor, and wake fraction – only the wake fraction is usually corrected in the transition from model to full scale.

3. False. Experiments and blade efficiency analyses have shown that *axial* losses prevail over the other two components of open water efficiency. For instance, in an investigation carried out by Molland et al. with the aid of blade element theory, the losses under certain settings were found to be 57% axial, 29% frictional, and 14% rotational. At a lower advance coefficient, the proportion of axial losses grew to 81%. Similarly, in a CFD study conducted by Lee et al., the axial loss was 70%, against 24% frictional loss and 6% rotational loss.

4. False. The ITTC uses the thrust identity approach. The assumption of thrust identity is preferred as it tends to provide more accurate performance predictions for the full scale vessel. For one, thrust is less affected by different model scales and its measurement is less susceptible to errors caused by friction in bearings and on propeller blades.

5. True. Indeed, a model in a propulsion test is allowed to perform the four aforementioned types of motion, but it is prevented from sway and yaw motions by guides at bow and stern. Typically, sinkage fore and aft is also recorded in a propulsion test.

6. False. Expressing the definition of cavitation number proposed in the statement, we write

$$\sigma = \frac{p_0 - p_V}{\frac{1}{2}\rho V^2}$$

Here, $p_o - p_V$ is the difference between static pressure in the flow and vapor pressure, and $(1/2)\rho V^2$ is the so-called dynamic pressure. In general, the larger the cavitation number, the *less* likely is cavitation. Indeed, a high static pressure p_o lowers the danger of cavitation (higher cavitation number σ), which can be achieved by deeply submerging a foil or propeller. High flow velocities, in turn, increases dynamic pressure, which increases the likelihood of cavitation (smaller cavitation number σ).

P.3 ■ Solution

The hull efficiency is given by

$$\eta_H = \frac{1-t}{1-w} = \frac{1-0.178}{1-0.325} = 1.218$$

The propulsive efficiency equals the product of hull efficiency, open-water efficiency, relative rotative efficiency, and shaft transmission efficiency. Thus,

$$\eta_P = \eta_H \eta_0 \eta_R \eta_S = 1.218 \times 0.685 \times 1.025 \times 0.990 = \boxed{84.66\%}$$

However, we know that the propulsive efficiency is the ratio of effective power to shaft power or, mathematically, $\eta_P = P_E/P_S$. Solving for P_S gives

$$\eta_P = \frac{P_E}{P_S} \rightarrow P_S = \frac{P_E}{\eta_P} = \frac{12{,}860}{0.8466} = 15{,}190 \text{ hp}$$

This is the trial shaft horsepower. In order to obtain the *service* shaft horsepower, we must include the service power allowance; that is,

$$P_{SS} = P_S \times 1.20 = 15{,}190 \times 1.20 = \boxed{18{,}230 \text{ hp}}$$

★ The correct answer is **D**.

Why "trial shaft horsepower"?

The trial shaft horsepower would be sufficient to propel the actual ship at its design speed only if the conditions of operation of the ship were similar to those of the model whose resistance was measured in the model test. Since model test conditions are ideal conditions, the power predicted is an ideal power. It is called the trial shaft power because the sea conditions that most nearly correspond to these ideal model test conditions are those present during the ship standardization trials, which are conducted before the ship is delivered to the owners. Measurements taken during ship trials are, in fact, the basis on which it is judged whether or not the ship's speed and power performance satisfies the contract between the builder and owner.

P.4 ■ Solution

1. False. To begin, we compute the advance ratio, the thrust coefficient, and the torque coefficient in the behind condition.

$$J_b = \frac{V_S}{nD} = \frac{1.33}{(10.0 \times 0.19)} = 0.7$$

$$K_{T,b} = \frac{T_b}{\rho n^2 D^4} = \frac{22.42}{1000 \times 10.0^2 \times 0.19^4} = 0.172$$

$$K_{Q,b} = \frac{Q_b}{\rho n^2 D^5} = \frac{0.58}{1000 \times 10.0^2 \times 0.19^5} = 0.0234$$

Using the equations we were given along with the thrust coefficient $K_{T,O} = K_{T,b} = 0.172$, we obtain the open-water advance ratio $J_O = 0.497$ and the open-water torque coefficient $K_{Q,O} = 0.0239$. We proceed to determine the wake fraction,

$$w_T = 1 - \frac{J_O}{J_b} = 1 - \frac{0.497}{0.700} = 0.290$$

and the thrust deduction factor,

$$t = 1 - \frac{R}{T_b} = 1 - \frac{18.5}{22.42} = 0.175$$

The hull efficiency follows as

$$\eta_H = \frac{1-t}{1-w_T} = \frac{1-0.175}{1-0.290} = 1.162$$

2. True. In view of the fact that the test is one of thrust identity, the relative rotative efficiency is determined as

$$\eta_R = \left(\frac{K_{Q,O}}{K_{Q,b}}\right)_{\text{thrust identity}} = \frac{0.0239}{0.0234} = 1.021$$

3. True. The open water efficiency is given by

$$\eta_O = \frac{J_O}{2\pi} \times \frac{K_{T,O}}{K_{Q,O}} = \frac{0.497}{2\pi} \times \frac{0.172}{0.0239} = 0.569$$

4. False. The quasi-propulsive coefficient is the product of the three foregoing terms,

$$QPC = \eta_H \times \eta_R \times \eta_O = 1.162 \times 1.021 \times 0.569 = 0.675$$

Another way to arrive at the same result is to apply the formula

$$\eta_D = QPC = \frac{RV}{2\pi nQ} = \frac{18.5 \times 1.33}{2\pi \times 10.0 \times 0.58} = 0.675$$

Following model tests, K_T, K_Q, and η_O can be scaled from model to full size by means of the pertaining correlations, or using the corrections available for propellers of the Wageningen B-series. Empirical relations are also available for scaling of the wake fraction.

P.5 ■ Solution

Using the data we were given, the thrust coefficient is calculated as

$$K_T = \frac{T}{\rho n^2 D^4} = \frac{(200 \times 2240)}{1.99 \times (120/60)^2 \times 21^4} = 0.289$$

Mapping this value onto the curves, we read the open water efficiency η_o = 0.56 and the advance coefficient J = 0.55. The speed of advance V_A can be established from the definition of advance coefficient, namely,

$$J = \frac{V_A}{nD} \rightarrow V_A = J \times n \times D$$

$$\therefore V_A = 0.55 \times (120/60) \times 21 = \boxed{23.1 \text{ ft/s}}$$

★ The correct answer is **B**.

P.6 ■ Solution

The ship speed is V_S = 18 × 1.689 = 30.4 ft/s. Knowing that w = 0.31 and V_S = 30.4 ft/s, the speed of advance can be obtained from the definition of wake fraction,

$$w = 1 - \frac{V_A}{V_S} \rightarrow 0.31 = 1 - \frac{V_A}{30.4}$$

$$\therefore V_A = 21 \text{ ft/s}$$

The thrust power is given by the product of hull efficiency and effective resistance, or

$$P = \eta_H \times P_E = \eta_H \times R_E \times V_S$$

Substituting the pertaining variables, we get

$$P = 0.99 \times 300{,}000 \times 30.4 = 9.03 \times 10^6 \text{ lb-ft/s}$$

or about 16,450 hp. Finally, the thrust T is determined to be

$$P = T \times V_A \to T = \frac{P}{V_A}$$

$$\therefore T = \frac{9.03 \times 10^6}{21} = \boxed{430{,}000 \text{ lb}}$$

★ The correct answer is **D**.

P.7 ■ Solution

In view of the fact that the apparent slip is zero, the pitch *P* is simply the ratio of ship speed to propeller rpm; that is,

$$P = \frac{V_S}{n} = \frac{(18 \times 1.689)}{\left(85 \times \frac{1}{60}\right)} = 21.5 \text{ ft}$$

The speed of advance V_A follows as

$$V_A = V_S \times (1 - w) = 18 \times (1 - 0.30) = 12.6 \text{ kn} = 21.3 \text{ m/s}$$

Finally, the real slip is determined as

$$\text{Real slip} = 1 - \frac{V_A}{P \times n} = 1 - \frac{21.3}{21.5 \times \left(85 \times \frac{1}{60}\right)} = \boxed{0.30}$$

★ The correct answer is **A**.

P.8 ■ Solution

Part A: The propeller thrust is given by

$$K_T = \frac{T}{\rho n^2 D^4} \to T = K_T \times \rho \times n^2 \times D^4$$

$$\therefore T = 0.1594 \times 1025 \times 3^2 \times 4.5^4 = 603 \text{ kN}$$

The thrust deduction factor can be obtained from the relation that associates thrust and resistance, namely,

$$T(1 - t) = R_T \to t = 1 - \frac{R_T}{T}$$

$$\therefore t = 1 - \frac{580}{603} = \boxed{0.0381}$$

★ The correct answer is **B**.

Part B: The inflow velocity is given by

$$V_A = V_S (1 - w) = 13 \times (1 - 0.135) = 11.2 \text{ m/s}$$

The hull efficiency is then

$$\eta_H = \frac{R_T \times V_S}{T \times V_A} = \frac{580 \times 13}{603 \times 11.2} = 1.116$$

The quasi-propulsive coefficient is the product of relative rotative efficiency, open-water efficiency, and hull efficiency; that is,

$$QPC = \eta_R \times \eta_O \times \eta_H = 0.95 \times 0.684 \times 1.116 = \boxed{0.725}$$

★ The correct answer is **C**.

Part C: By dint of similarity laws, the model speed is determined as

$$\frac{V_S}{V_m} = \sqrt{\lambda} \to V_m = \frac{V_S}{\sqrt{\lambda}}$$

$$\therefore V_m = \frac{13}{\sqrt{16}} = 3.25 \text{ m/s}$$

The inflow velocity at model scale is

$$V_{A,m} = V_m(1 - w_m) = 3.25 \times (1 - 0.19) = 2.63 \text{ m/s}$$

Noting that the advance number in model scale is the same as the advance number in model scale, we have

$$n_m = \frac{V_{A,m}}{J \times D_m} = \frac{V_{A,m}}{J \times (D_s/\lambda)}$$

$$\therefore n_m = \frac{2.63}{0.833 \times (4.5/16)} = 11.2 \text{ Hz} = \boxed{672 \text{ rpm}}$$

★ The correct answer is **D**.

Part D: The circumferential velocity at r/R = 0.7 is given by

$$V_R = \sqrt{V_A^2 + (0.7\pi n D)^2}$$

At full scale, then, we have

$$V_{R,S} = \sqrt{11.2^2 + (0.7\pi \times 3 \times 4.5)^2} = 31.7 \text{ m/s}$$

while at model scale,

$$V_{R,M} = \sqrt{2.63^2 + \left[0.7\pi \times 11.2 \times (4.5/16)\right]^2} = 7.41 \text{ m/s}$$

The Reynolds number at full scale is

$$\text{Re}_S = \frac{V_{R,S} c_S}{\nu_S} = \frac{31.7 \times 2}{(1.14 \times 10^{-6})} = 5.56 \times 10^7$$

while *Re* at model scale, in turn, is

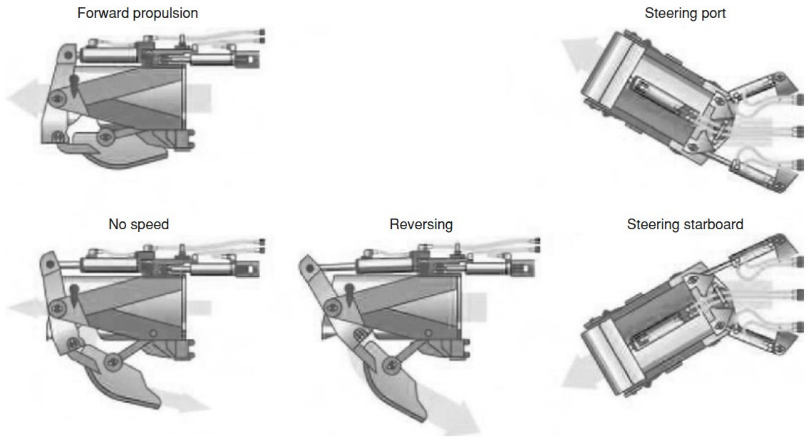
$$\text{Re}_m = \frac{V_{R,m} c_m}{\nu_m} = \frac{7.41 \times (2/16)}{(1.19 \times 10^{-6})} = 7.78 \times 10^5$$

The model scale Reynolds number is close to the point where laminar/turbulent transition is expected. Unless turbulence is stimulated, some contamination due to partially laminar flow may be expected.

P.9 ■ Solution

1. False. Waterjet propulsion systems are not always free from cavitation, and there are operating conditions where they may experience cavitation to a worrisome extent. Waterjet devices offer an additional dimension of choice to the propulsion solutions available for some vessels, and tend to be chosen when blade-based systems have been rejected for some reason – typically, reasons of efficiency, noise, immersion and draft, and, of course, cavitation. Carlton notes that, in the case of small vessels traveling at, say, 45 knots, one might expect that a conventional propeller would be fully cavitating, whereas a waterjet system would not. The potential for waterjet application, then, neglecting any small special purpose craft with particular requirements, is where conventional, trans-cavitating and super-cavitating propeller performance is beginning to fall off.

2. True. Earlier waterjet systems required use of rudders to perform certain maneuvers, but in modern systems the steering and reversing systems are integrated in the jet machinery. In a typical such device, deflector units are fitted to the outlet pipe which then direct the water flow and hence introduce turning forces by changing the direction of the jet momentum. The majority of waterjet units are fitted with either a steerable nozzle or deflectors of one form or another so as to provide a directional control of the propulsive device. The following figure illustrates how the waterjet direction of a Kamewa waterjet propulsion system by Rolls-Royce can be changed in order to stop, reverse, and steer the vessel.



3. False. It is true that having the vessel rise to foilborne conditions requires a substantial lift force, but it is a travesty to say that waterjet propulsion systems have high efficiency at low speeds. On the contrary, there is a drastic reduction in efficiency when a waterjet-impelled ship operates at low velocity. On the other hand, waterjets have excellent performance at fast conditions, offering nearly cavitation-free performance under circumstances that would be severely unstable for propeller-based systems. One of the main drawbacks of this system is the power required to lift water from the inlet to the nozzle level, especially when the nozzle is located above the waterline – i.e., in the case of foilborne craft.

4. True. Part of the statement is taken from Faltinsen's *Hydrodynamics of High-Speed Marine Vehicles*. Hydrofoils must be protected from cavitation, which can otherwise cause the material of the foil to be damaged and thereby reduce its lifting capability. The simplest measure to prevent a foil from undergoing cavitation is to reduce the speed of the vessel, often to no more than 50 knots. If this speed limit is to be exceeded, use of supercavitating foils becomes necessary. As mentioned in the statement, these components have much lower lift-to-drag ratios and lift coefficients than their subcavitating counterparts. The pressure distribution along the foil should be relatively flat in order to minimize cavitation, i.e., there should be no pronounced local pressure minima (suction peaks).

5. True. Indeed, the appendages described in the statement – T-foils, trim tabs (flaps) and interceptors – are particularly suitable for high speed vessels. It is noteworthy, however, that appendages such as T-foils often undergo cavitation and ventilation, which may be inevitable at the velocity range that characterizes catamarans and other high-speed marine vehicles. (Furthermore, since T-foils add drag to the vessel, it is an advantage that they can be retracted in calm water conditions. Retractable T-foils are also beneficial during operation in shallow water.)

6. True. This statement is an excerpt from Chapter 5 of Faltinsen's *Hydrodynamics of High-Speed Marine Vehicles*. A SES does possess inherent advantages when compared to a catamaran. On the other hand, problem areas for SESs include wear of skirts, loss of speed in waves, and the so-called cobblestone effect, a phenomenon that arises from compressible flow effects in the air cushion and causes unpleasant vertical accelerations in small sea states.

P.10 ■ Solution

Part A: The waterjet speed expressed in meters per second is V_S = 65×0.5144 = 33.4 m/s. Equipped with this quantity and the jet efficiency η_J = 0.60, the waterjet speed is determined as

$$\eta_J = \frac{2V_S}{V_J + V_S} \rightarrow V_J = \frac{2V_S}{\eta_J} - V_S$$

$$\therefore V_J = \frac{2\times 33.4}{0.60} - 33.4 = \boxed{77.9\text{ m/s}}$$

★ The correct answer is **D**.

Part B: The required thrust can be taken as equal to the vessel's resistance; that is,

$$T = \rho A V_J (V_J - V_S) = 52{,}000 \text{ N}$$

Noting that the flow rate $Q = AV_J$, we have

$$T = \rho \underbrace{AV_J}_{=Q} (V_J - V_S) \rightarrow Q = \frac{T}{\rho (V_J - V_S)}$$

$$\therefore Q = \frac{50{,}000}{1.025 \times (77.9 - 33.4)} = 1.10 \text{ m}^3/\text{s}$$

The cross-sectional area of the jet easily follows,

$$Q = 1.1 \rightarrow A = \frac{1.1}{V_J}$$

$$\therefore A = \frac{1.1}{77.9} = 0.0141 \text{ m}^2$$

so that

$$A = \frac{\pi \times D^2}{4} = 0.0141$$

$$\therefore D = \sqrt{\frac{4 \times 0.0141}{\pi}} = 0.134 \text{ m}$$

$$\therefore \boxed{D = 134 \text{ mm}}$$

★ The correct answer is **C**.

P.11 ■ Solution

At a speed of V knots, the distance traveled in 1 hour is V nautical miles. To travel 1000 miles takes 1000/V hrs. If the fuel consumption is S kg per MWh, the fuel required for 1000 miles at speed V is

$$S \times (\text{power}) \times \frac{1000}{V} \text{ kg}$$

Conversely, the number of hours steaming possible on 1000 tonnef of fuel is given by

$$\text{No. of hours} = \frac{1000 \times 10^3}{S[\text{power}]}$$

so that

$$\text{Distance traveled} = \frac{10^6 V}{S[\text{power}]} \text{ Nautical miles}$$

In view of these formulas and the graphs we were given, the following table is prepared.

V (kn)	Power (MW)	% full power	SFC (tonnef/MW-hour)	Tonnef of fuel per 1000 miles	Endurance for 1000 tonnef fuel
12	1.43	5.4	0.97	115	8700
14	2.26	8.6	0.74	119	8370
16	3.31	12.6	0.61	126	7920
18	4.8	18.3	0.52	139	7210
20	6.9	26.3	0.46	159	6300
22	9.74	37.1	0.43	190	5250
24	13.65	52.0	0.42	239	4190
26	18.74	71.4	0.42	303	3300
28	26.25	100.0	0.43	403	2480

As noted by Rawson & Tupper, there is not a lot of penalty in increasing the cruising or endurance speed from 12 to 16 knots. Indeed, economically, the lower salary bill would probably compensate for the increased fuel bill and the ship is a sounder economic proposition because of the increased mileage in a year.

Answer Summary

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

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