# National Exams May 2017

## 16-Nav-B2, Marine Engineering

**3 hours duration**

**NOTES:**

1. If doubt exists as to the interpretation of any question, the candidate is urged to submit with the answer paper, a clear statement of any assumptions made.

2. Candidates may use one of two calculators, the Casio or Sharp approved models. This is a closed book exam.

3. Any five questions constitute a complete paper. Only the first five questions as they appear in your answer book will be marked.

4. All questions are of equal value.

1. Calculate the diameter of a propeller shaft for a ship having the following particulars:

| | |
|---|---|
| Shaft power = | 7350 KW |
| Propeller weigh = | 200.00 KN |
| Steady thrust = | 800.00 KN |
| Shaft speed = | 95 rpm |
| Min. yield tensile strength = | 207.00 MN / $m^2$ |
| Endurance limit = | 186.00 MN / $m^2$ |

Take the specific weight for steel is 76.5 KN/$m^3$. Consider a 5% variation in the torque acting on the shaft. Outline any assumptions you make in determining the bending moment on the shaft. Assume the shaft material to have an Ultimate Strength of 415.0 N/$mm^2$.

2. Calculate the moments and reactions at the bearings for the propeller shaft sketched shown in Fig. 1. Propeller weight is 350.00 KN and the shaft diameter is 535.00 mm. Assume that the shaft is simply supported at bearing D. Take the specific weight for steel is 76.5 KN/$m^3$.

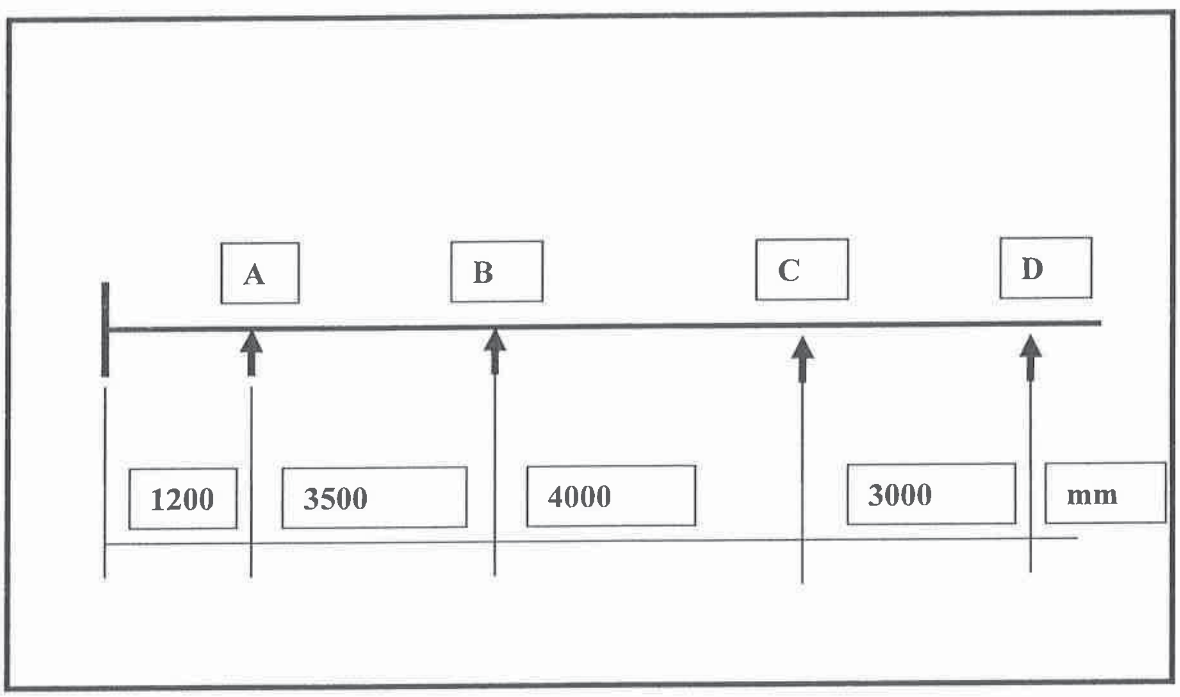



Figure 1

3. A centrifugal pump running at 1000 rpm has the following characteristics:

| | | | | | |
|---|---|---|---|---|---|
| Q (l/s) | 10 | 15 | 20 | 25 | 27 |
| H (m) | 25 | 24 | 22.5 | 21.2 | 20.4 |
| $\eta_o$ % | 65.8 | 70 | 73.5 | 69.6 | 66 |

Draw the operating characteristics of the pump and determine its specific speed. If the pump is driven by a 7.35 KW motor, determine the maximum discharge that can be delivered by the pump.

This pump has been selected to serve a fire fighting line. Two pumps are to be connected in series to a pipeline having the following characteristics:

| | | |
|---|---|---|
| Length | = | 200.0 meters |
| Diameter | = | 100.0 mm |
| Static Head | = | 8.0 meters |
| Coefficient of friction | = | 0.02 |
| Nozzle diameter | = | 50.0 mm |
| Nozzle coefficient of velocity | = | 0.97 |

Determine the operating characteristics of the system and the power required. Is this a good choice?

4. Determine the heat transfer surface area required for a heat exchanger constructed from a 0.0254 m (o.d) tube to cool 6.93 kg/s of water from 65.6°C to 39.4°C, using 6.30 kg/s of water available at 10.00°C. Assume that the overall coefficient of heat transfer based on the outer-tube area is 568.00 $W/m^2.K$ and consider each of the following arrangements:

a) Parallel-flow tube and shell.
b) Counterflow tube and shell.

5.a) Describe briefly a method for propeller shaft alignment.

5.b) The arrangement of cranks in an 8-cylinder in-line engine is such that the cylinders are separated by a distance 180.00 mm in the axial direction and the angular positions of the cranks are given by 0, 270.0, 90.0, 180.0, 135.0, 225.0, 45.0, and 315.0. The crank length, connecting rod length, engine speed, and the reciprocating weight of each cylinder is 120.0 mm, 500.0 mm, 1500 rpm, and 150.0 N/cylinder, respectively. Find the primary and secondary unbalanced forces and moments when the piston is at the upper dead center.

6.Discuss briefly the following:

a. the effect of the ambient conditions on the performance of a gas turbine.
b. the effect of the number of propeller blades on the vertical bending moment and thrust variation of the propeller shaft of a single screw ship.
c. the difference between the mean effective pressure and the mean indicated pressure of a diesel engine.

7.The first natural torsional vibration for a 4-cylinder diesel engine connected to a propeller through a gear box is suspected to be between 300.0 rad/sec and 400.0 rad/sec. Use Holzer's method to calculate the first natural frequency. The system is modeled as shown in Figure 2. The characteristics of the system are as follows:

| | | |
|---|---|---|
| $J = 0.75$ kg.m$^2$ | $J_G = 1.6$ kg.m$^2$ | $J_P = 0.14$ kg.m$^2$ |
| $K = 4.2$ E5 N/rad | $K_I = 2.6$ E5 N/rad | |

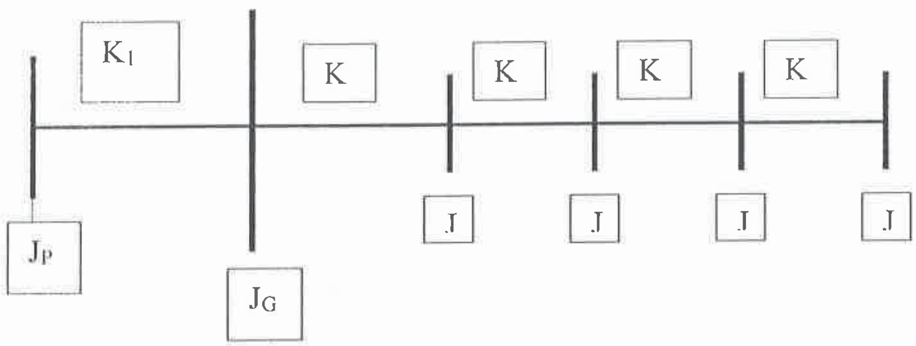



Figure 2

Data Sheet

Three moment equation:

$$M_{n-1}\left(\frac{l_n}{I_n}\right) + 2M_n\left[\frac{l_n}{I_n} + \frac{l_{n+1}}{I_{n+1}}\right] + M_{n+1}\left(\frac{l_{n+1}}{I_{n+1}}\right)$$

$$= -\left[w_n\left(\frac{l_n^3}{I_n}\right) + w_{n+1}\left(\frac{l_{n+1}^3}{I_{n+1}}\right)\right]/4 - 6E[\beta_n - \beta_{n+1}]$$

$$\beta_n = \frac{\delta_{n-1} - \delta_n}{l_n}$$

Longitudinal Vibration of a Shaft:

$$C^2 \frac{\partial^2 u}{\partial x^2} = \frac{\partial^2 u}{\partial t^2}$$

Transverse Vibration of a Shaft:

$$EI \frac{\partial^4 y}{\partial x^4} + \rho A \frac{\partial^2 y}{\partial t^2} = 0$$

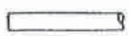
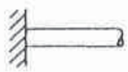
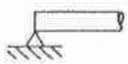
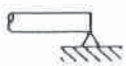
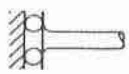
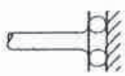
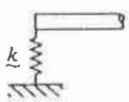
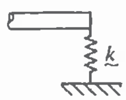
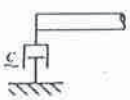
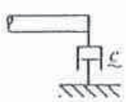
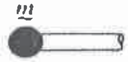
Torsional Vibration of a Shaft:

$$C^2 \frac{\partial^2 \theta}{\partial x^2} = \frac{\partial^2 \theta}{\partial t^2}$$

Losses in pipe fittings

$$h = K\left(\frac{V^2}{2\,g}\right)$$

| Fitting | K |
|---|---|
| Globe valve | 10 |
| Return bend | 2.2 |
| 90° elbow | 0.9 |
| 45° elbow | 0.4 |
| Sharp pipe entry or exit | 0.5 |

| Boundary condition | At left end ($x = 0$) | At right end ($x = l$) |
|---|---|---|
| Free end (bending moment = 0, shear force = 0) |  $x = 0$ <br> $EI\frac{\partial^2 w}{\partial x^2}(0,t) = 0$ <br> $\frac{\partial}{\partial x}\left(EI\frac{\partial^2 w}{\partial x^2}\right)\Big\vert_{(0,t)} = 0$ |  $x = l$ <br> $EI\frac{\partial^2 w}{\partial x^2}(l,t) = 0$ <br> $\frac{\partial}{\partial x}\left(EI\frac{\partial^2 w}{\partial x^2}\right)\Big\vert_{(l,t)} = 0$ |
| Fixed end (deflection = 0, slope = 0) |  $x = 0$ <br> $w(0,t) = 0$ <br> $\frac{\partial w}{\partial x}(0,t) = 0$ |  $x = l$ <br> $w(l,t) = 0$ <br> $\frac{\partial w}{\partial x}(l,t) = 0$ |
| Simply supported end (deflection = 0, bending moment = 0) |  $x = 0$ <br> $w(0,t) = 0$ <br> $EI\frac{\partial^2 w}{\partial x^2}(0,t) = 0$ |  $x = l$ <br> $w(l,t) = 0$ <br> $EI\frac{\partial^2 w}{\partial x^2}(l,t) = 0$ |
| Sliding end (slope = 0, shear force = 0) |  $x = 0$ <br> $\frac{\partial w}{\partial x}(0,t) = 0$ <br> $\frac{\partial}{\partial x}\left(EI\frac{\partial^2 w}{\partial x^2}\right)\Big\vert_{(0,t)} = 0$ |  $x = l$ <br> $\frac{\partial w}{\partial x}(l,t) = 0$ <br> $\frac{\partial}{\partial x}\left(EI\frac{\partial^2 w}{\partial x^2}\right)\Big\vert_{(l,t)} = 0$ |
| End spring (spring constant = $\underset{\sim}{k}$) |  $\underset{\sim}{k}$ $x = 0$ <br> $\frac{\partial}{\partial x}\left(EI\frac{\partial^2 w}{\partial x^2}\right)\Big\vert_{(0,t)} = -\underset{\sim}{k}\, w(0,t)$ <br> $EI\frac{\partial^2 w}{\partial x^2}(0,t) = 0$ |  $\underset{\sim}{k}$ $x = l$ <br> $\frac{\partial}{\partial x}\left(EI\frac{\partial^2 w}{\partial x^2}\right)\Big\vert_{(l,t)} = +\underset{\sim}{k}\, w(l,t)$ <br> $EI\frac{\partial^2 w}{\partial x^2}(l,t) = 0$ |
| End damper (damping constant = $\underset{\sim}{c}$) |  $\underset{\sim}{c}$ $x = 0$ <br> $\frac{\partial}{\partial x}\left(EI\frac{\partial^2 w}{\partial x^2}\right)\Big\vert_{(0,t)} = -\underset{\sim}{c}\frac{\partial w}{\partial t}(0,t)$ <br> $EI\frac{\partial^2 w}{\partial x^2}(0,t) = 0$ |  $\underset{\sim}{c}$ $x = l$ <br> $\frac{\partial}{\partial x}\left(EI\frac{\partial^2 w}{\partial x^2}\right)\Big\vert_{(l,t)} = +\underset{\sim}{c}\frac{\partial w}{\partial t}(l,t)$ <br> $EI\frac{\partial^2 w}{\partial x^2}(l,t) = 0$ |
| End mass (mass = $\underset{\sim}{m}$ with negligible moment of inertia) | $\underset{\sim}{m}$  $x = 0$ <br> $\frac{\partial}{\partial x}\left(EI\frac{\partial^2 w}{\partial x^2}\right)\Big\vert_{(0,t)} = -\underset{\sim}{m}\frac{\partial^2 w}{\partial t^2}(0,t)$ <br> $EI\frac{\partial^2 w}{\partial x^2}(0,t) = 0$ | $\underset{\sim}{m}$  $x = l$ <br> $\frac{\partial}{\partial x}\left(EI\frac{\partial^2 w}{\partial x^2}\right)\Big\vert_{(l,t)} = +\underset{\sim}{m}\frac{\partial^2 w}{\partial t^2}(l,t)$ <br> $EI\frac{\partial^2 w}{\partial x^2}(l,t) = 0$ |
| End mass with moment of inertia (mass = $\underset{\sim}{m}$, moment of inertia = $\underset{\sim}{J_0}$) | $\underset{\sim}{m}, \underset{\sim}{J_0}$  $x = 0$ <br> $EI\frac{\partial^2 w}{\partial x^2}(0,t) = \underset{\sim}{J_0}\frac{\partial^3 w}{\partial x \partial t^2}(0,t)$ <br> $\frac{\partial}{\partial x}\left(EI\frac{\partial^2 w}{\partial x^2}\right)\Big\vert_{(0,t)} = \underset{\sim}{m}\frac{\partial^2 w}{\partial t^2}(0,t)$ | $\underset{\sim}{m}, \underset{\sim}{J_0}$  $x = l$ <br> $EI\frac{\partial^2 w}{\partial x^2}(l,t) = -\underset{\sim}{J_0}\frac{\partial^3 w}{\partial x \partial t^2}(l,t)$ <br> $\frac{\partial}{\partial x}\left(EI\frac{\partial^2 w}{\partial x^2}\right)\Big\vert_{(l,t)} = -\underset{\sim}{m}\frac{\partial^2 w}{\partial t^2}(l,t)$ |

Boundary conditions for the transverse vibration of a beam

| Boundary condition | At left end ($x = 0$) | At right end ($x = l$) |
|---|---|---|
| Fixed end | $u(0, t) = 0$ <br> $x = 0$ | $u(l, t) = 0$ <br> $x = l$ |
| Free end | $\frac{\partial u}{\partial x}(0, t) = 0$ <br> $x = 0$ | $\frac{\partial u}{\partial x}(l, t) = 0$ <br> $x = l$ |
| End spring (spring constant = $k$) | $k$ <br> $AE\frac{\partial u}{\partial x}(0, t) = k u(0, t)$ <br> $x = 0$ | $k$ <br> $AE\frac{\partial u}{\partial x}(l, t) = -k u(l, t)$ <br> $x = l$ |
| End damper (damping constant = $c$) | $c$ <br> $AE\frac{\partial u}{\partial x}(0, t) = c\frac{\partial u}{\partial t}(0, t)$ <br> $x = 0$ | $c$ <br> $AE\frac{\partial u}{\partial x}(l, t) = -c\frac{\partial u}{\partial t}(l, t)$ <br> $x = l$ |
| End mass (mass = $m$) | $m$ <br> $AE\frac{\partial u}{\partial x}(0, t) = m\frac{\partial^2 u}{\partial t^2}(0, t)$ <br> $x = 0$ | $m$ <br> $AE\frac{\partial u}{\partial x}(l, t) = -m\frac{\partial^2 u}{\partial t^2}(l, t)$ <br> $x = l$ |

Boundary conditions for a bar in longitudinal vibration.

| Boundary condition | At left end ($x = 0$) | At right end ($x = l$) |
|---|---|---|
| Fixed end | $\theta(0, t) = 0$ <br> $x = 0$ | $\theta(l, t) = 0$ <br> $x = l$ |
| Free end | $\frac{\partial \theta}{\partial x}(0, t) = 0$ <br> $x = 0$ | $\frac{\partial \theta}{\partial x}(l, t) = 0$ <br> $x = l$ |
| End torsional spring (spring constant = $k$) | $k_t$ <br> $GJ\frac{\partial \theta}{\partial x}(0, t) = k_t\theta(0, t)$ <br> $x = 0$ | $k_t$ <br> $GJ\frac{\partial \theta}{\partial x}(l, t) = -k_t\theta(l, t)$ <br> $x = l$ |
| End torsional damper (damping constant = $c$) | $c_t$ <br> $GJ\frac{\partial \theta}{\partial x}(0, t) = c_t\frac{\partial \theta}{\partial t}(0, t)$ <br> $x = 0$ | $c_t$ <br> $GJ\frac{\partial \theta}{\partial x}(l, t) = -c_t\frac{\partial \theta}{\partial t}(l, t)$ <br> $x = l$ |
| End inertia (inertia = $J_0$) | $J_0$ <br> $GJ\frac{\partial \theta}{\partial x}(0, t) = J_0\frac{\partial^2 \theta}{\partial t^2}(0, t)$ <br> $x = 0$ | $J_0$ <br> $GJ\frac{\partial \theta}{\partial x}(l, t) = -J_0\frac{\partial^2 \theta}{\partial t^2}(l, t)$ <br> $x = l$ |

Boundary conditions for a shaft (rod) subjected to torsional vibration.

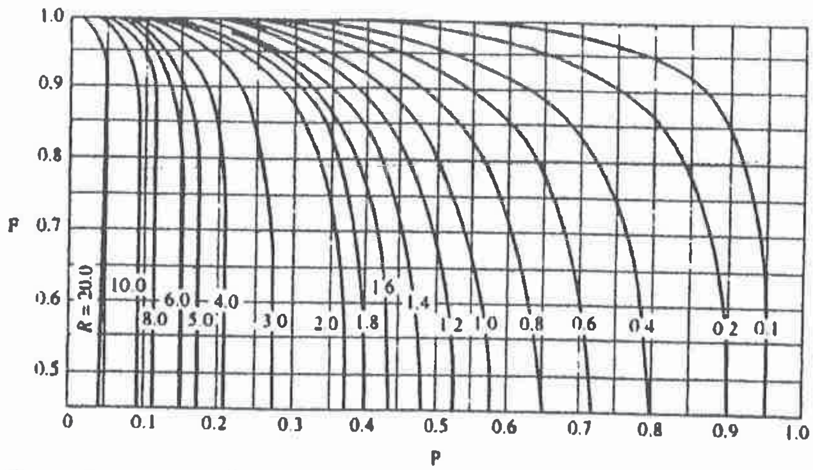



**Figure 7.11** Correction-factor plot for exchanger with one shell pass and two, four, or any multiple of two tube passes

Figs. 7.11–7.14.[11] The first two charts are associated with shell-and-tube heat exchangers and the last two charts with cross-flow heat exchangers. Among these, the shell-and-tube types are inherently heavy and are considered for stationary applications, while the cross-flow types are inherently light and are considered for mobile applications. Here we illustrate the use of the correction factor in terms of an example.

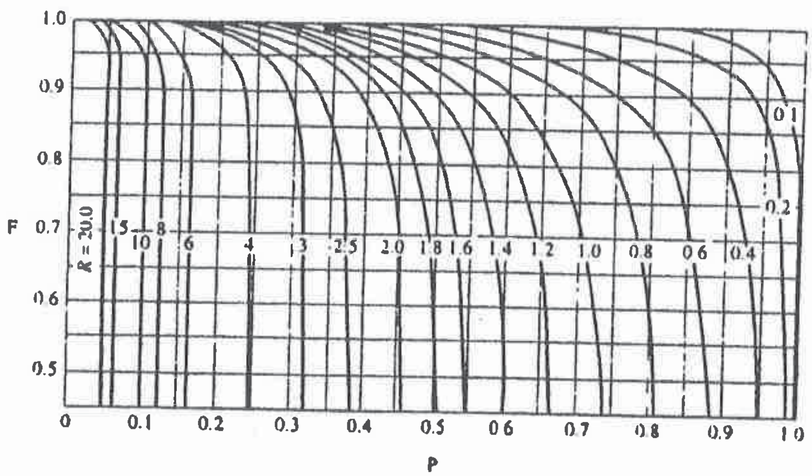



**Figure 7.12** Correction-factor plot for exchanger with two shell passes and four, or any multiple of four tube passes

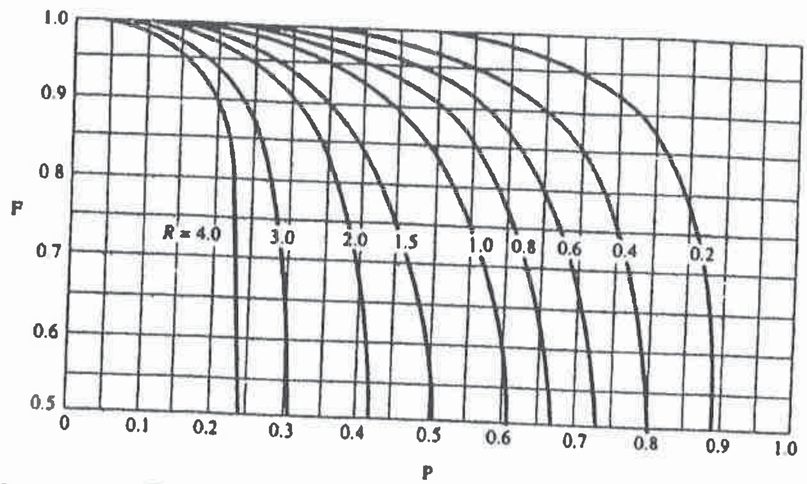



**Figure 7.13** Correction-factor plot for single-pass cross-flow exchanger one fluid mixed, the other unmixed

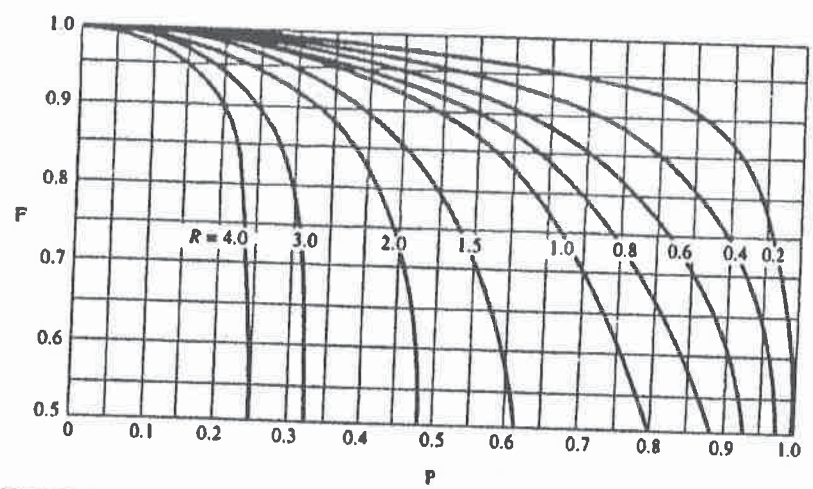
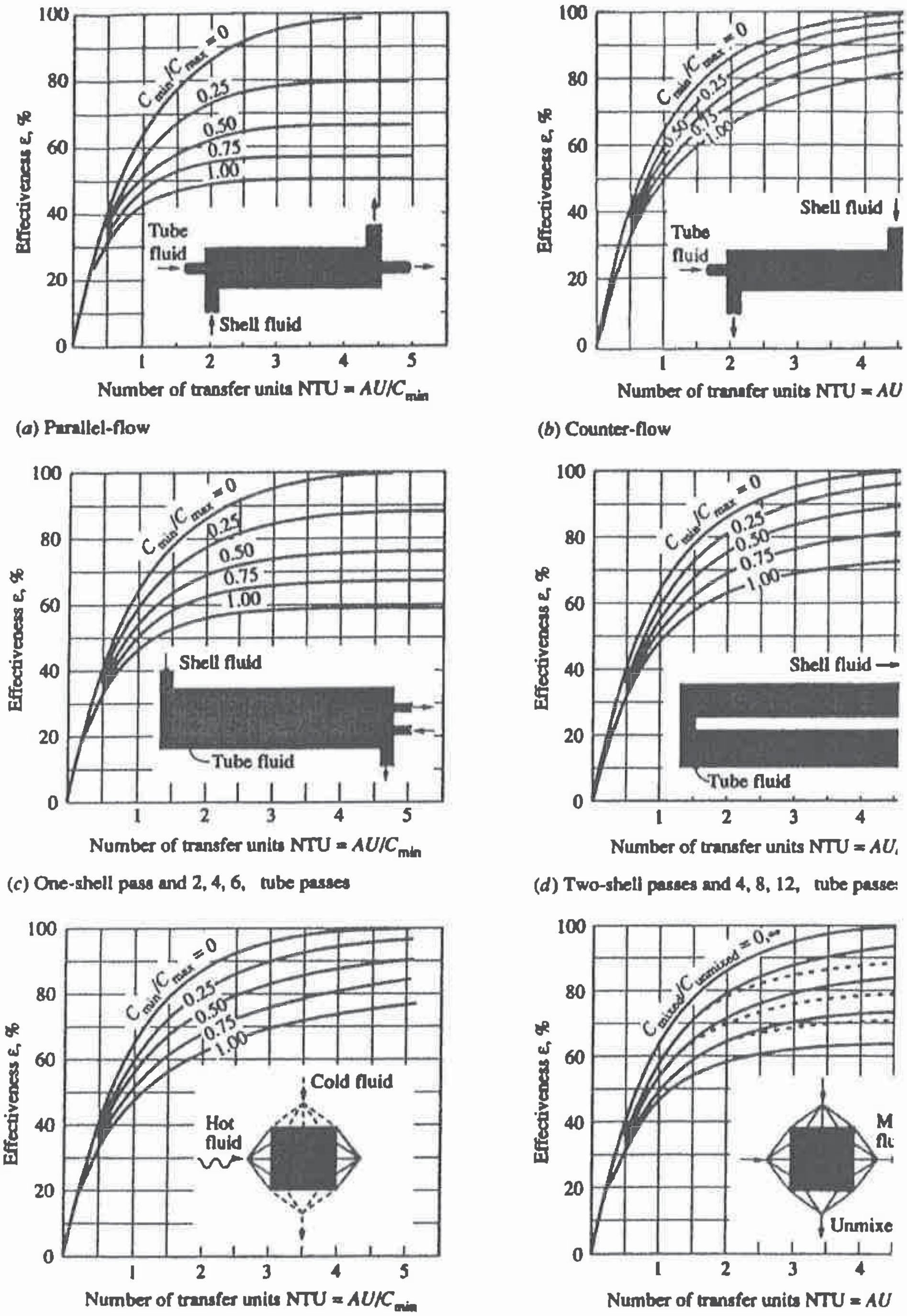



**Figure 7.14** Correction-factor plot for single-pass cross-flow exchanger both fluids unmixed

**FIGURE 10-26**

Effectiveness for heat exchangers (from Kays and London, Ref. 7).




(*a*) Parallel-flow

(*b*) Counter-flow

(*c*) One-shell pass and 2, 4, 6, tube passes

(*d*) Two-shell passes and 4, 8, 12, tube passe:

(*e*) Cross-flow with both fluids unmixed

(*f*) Cross-flow with one fluid mixed and the other unmixed

### 3.7 Material Tests

#### 3.7.1 General

Materials for all torque-transmitting parts, including shafts, clutches, couplings, coupling bolts and keys are to be tested in the presence of the Surveyor. The materials are to meet the specifications of 2-3-7/5. 2-3-7/7 and 2-3-8/1 or other specifications approved in connection with the design.

#### 3.7.2 Alternative Test Requirements

*3.7.2(a) 375 kW (500 hp) or less.* Materials for parts transmitting 375 kW (500 hp) or less may be accepted by the Surveyor based on verification of manufacturer's certification and witnessed hardness check.

*3.7.2(b) Coupling bolts.* Coupling bolts manufactured and marked to a recognized standard will not require material testing.

#### 3.7.3 Inspections and Nondestructive Tests

Shafting and couplings are to be surface examined by the Surveyor.

Forgings for tail shafts 455 mm (18 in.) and over in finished diameter are to be ultrasonically examined in accordance with 2-3-7/1.13.2. Tail shafts in the finished machine condition are to be subjected to magnetic particle, dye penetrant or other nondestructive examinations. They are to be free of linear discontinuities greater than 3.2 mm ($^1/_8$ in.), except that in the following locations the shafts are to be free of all linear discontinuities:

*3.7.3(a) Tapered tail shafts*: the forward one-third length of the taper, including the forward end of any keyway and an equal length of the parallel part of the shaft immediately forward of the taper.

*3.7.3(b) Flanged tail shafts*: the flange fillet area.

## 5 Design and Construction

### 5.1 Shaft Diameters

The minimum diameter of propulsion shafting is to be determined by the following equation:

$$D = 100K \cdot \sqrt[3]{\frac{H}{R}\left(\frac{c_1}{U + c_2}\right)}$$

where

| | | |
|---|---|---|
| $D$ | = | required solid shaft diameter, except hollow shaft; mm (in.) |
| $H$ | = | power at rated speed; kW (PS, hp) (1 PS = 735 W; 1 hp = 746 W) |
| $K$ | = | shaft design factor, see 4-3-2/Table 1 or 4-3-2/Table 2 |
| $R$ | = | rated speed rpm |
| $U$ | = | minimum specified ultimate tensile strength of shaft material (regardless of the actual minimum specified tensile strength of the material, the value of $U$ used in these calculations is not to exceed that indicated in 4-3-2/Table 3; N/mm$^2$ (kgf/mm$^2$, psi) |

$c_1$ and $c_2$ are given below:

| | *SI units* | *MKS units* | *US units* |
|---|---|---|---|
| $c_1$ | 560 | 41.95 | 3.695 |
| $c_2$ | 160 | 16.3 | 23180 |

## TABLE 1
## Shaft Design Factors $K$ and $C_K$ for Line Shafts and Thrust Shafts *(2006)*

| *Factor* | *Propulsion drives* | *Design features* (1) | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | *Integral flange* | *Shrink fit coupling* | *Key-ways* (2) | *Radial holes, transverse holes* (3) | *Longitudinal slots* (4) | *On both sides of thrust collars* | *In way of axial bearings used as thrust bearings* | *Straight sections* |
| $K$ | Type A | 0.95 | 0.95 | 1.045 | 1.045 | 1.14 | 1.045 | 1.045 | 0.95 |
| | Type B | 1.0 | 1.0 | 1.1 | 1.1 | 1.2 | 1.1 | 1.1 | 1.0 |
| $C_K$ | | 1.0 | 1.0 | 0.6 | 0.5 | 0.3 | 0.85 | 0.85 | 1.0 |

Type A: Turbine drives; electric drives; diesel drive through slip couplings (electric or hydraulic).

Type B: All other diesel drives.

*Notes*

1 Geometric features other than those listed will be specially considered.

2 After a length of not less than $0.2D$ from the end of the keyway, the shaft diameter may be reduced to the diameter calculated for straight sections.

Fillet radii in the transverse section of the keyway are not to be less than $0.0125D$.

3 Diameter of bore not more than $0.3D$.

4 Length of the slot not more than $1.4D$, width of slot not more than $0.2D$, whereby $D$ is calculated with $K = 1.0$.

## TABLE 2
## Shaft Design Factors $K$ and $C_K$ for Tail Shafts and Stern Tube Shafts (1) *(2006)*

| *Factor* | *Propulsion drive* | *Stern tube configuration* | *Tail shafts: propeller attachment method* (2) | | | *Stern tube shafts* (7, 8) |
|---|---|---|---|---|---|---|
| | | | *Keyed* (3) | *Keyless attachment by shrink fit* (4) | *Flanged* (5) | |
| $K$ | All | Oil lubricated bearings | 1.26 | 1.22 | 1.22 | 1.15 |
| | All | Water lubricated bearings: continuous shaft liners or equivalent (see 4-3-2/5.17.6) | 1.26 | 1.22 | 1.22 | 1.15 |
| | All | Water lubricated bearings: non-continuous shaft liners (6) | 1.29 | 1.25 | 1.25 | 1.18 |
| $C_K$ | | | 0.55 | 0.55 | 0.55 | 0.8 |

## TABLE 2 (continued) Shaft Design Factors $K$ and $C_K$ for Tail Shafts and Stern Tube Shafts [1] *(2006)*

*Notes*

1 Tail shaft may be reduced to stern tube shaft diameter forward of the bearing supporting the propeller, and the stern tube shaft reduced to line shaft diameter inboard of the forward stern tube seal.

2 Other attachments are subject to special consideration.

3 Fillet radii in the transverse section at the bottom of the keyway are not to be less than 0.0125D.

4 See also 4-3-2/5.11 and 4-3-3/5.15.2.

5 For flange fillet radii and flange thickness, see 4-3-2/5.19.3.

6 For Great Lakes Service, $K$ factor corresponding to continuous liner configuration may be used.

7 $K$ factor applies to shafting between the forward edge of the propeller-end bearing and the inboard stern tube seal.

8 Where keyed couplings are fitted on stern tube shaft, the shaft diameters are to be increased by 10% in way of the coupling. See Note 2 of 4-3-2/Table 1.

## TABLE 3 Maximum Values of $U$ to be Used in Shaft Calculations *(1 July 2006)*

| | *SI units* $N/mm^2$ | *MKS units* $kgf/mm^2$ | *US units* *psi* |
|---|---|---|---|
| 1. For all alloy steel shafts except tail shafts and tube shafts stated in 3 and 4 below. | 800 | 81.5 | 116,000 |
| 2. For all carbon and carbon-manganese shafts except tail shafts and tube shafts stated in 3 and 4 below. | 760 | 77.5 | 110,200 |
| 3. For tail shafts and tube shafts in oil lubricated bearings or in saltwater lubricated bearings but fitted with continuous liner or equivalent (see 4-3-2/5.17.6). | 600 | 61.2 | 87,000 |
| 4. For tail shafts and tube shafts in saltwater lubricated bearings fitted with non-continuous liners. | 415 | 42.2 | 60,000 |

### 5.3 Hollow Shafts

For hollow shafts where the bore exceeds 40% of the outside diameter, the minimum outside shaft diameter is not to be less than that determined through successive approximation utilizing the following equation:

$$D_o = D\sqrt[3]{\frac{1}{[1-(D_i/D_o)^4]}}$$

where

$D_o$ = required outer diameter of shaft; mm (in.)

$D$ = solid shaft diameter required by 4-3-2/5.1; mm (in.)

$D_i$ = actual inner diameter of shaft; mm (in.)

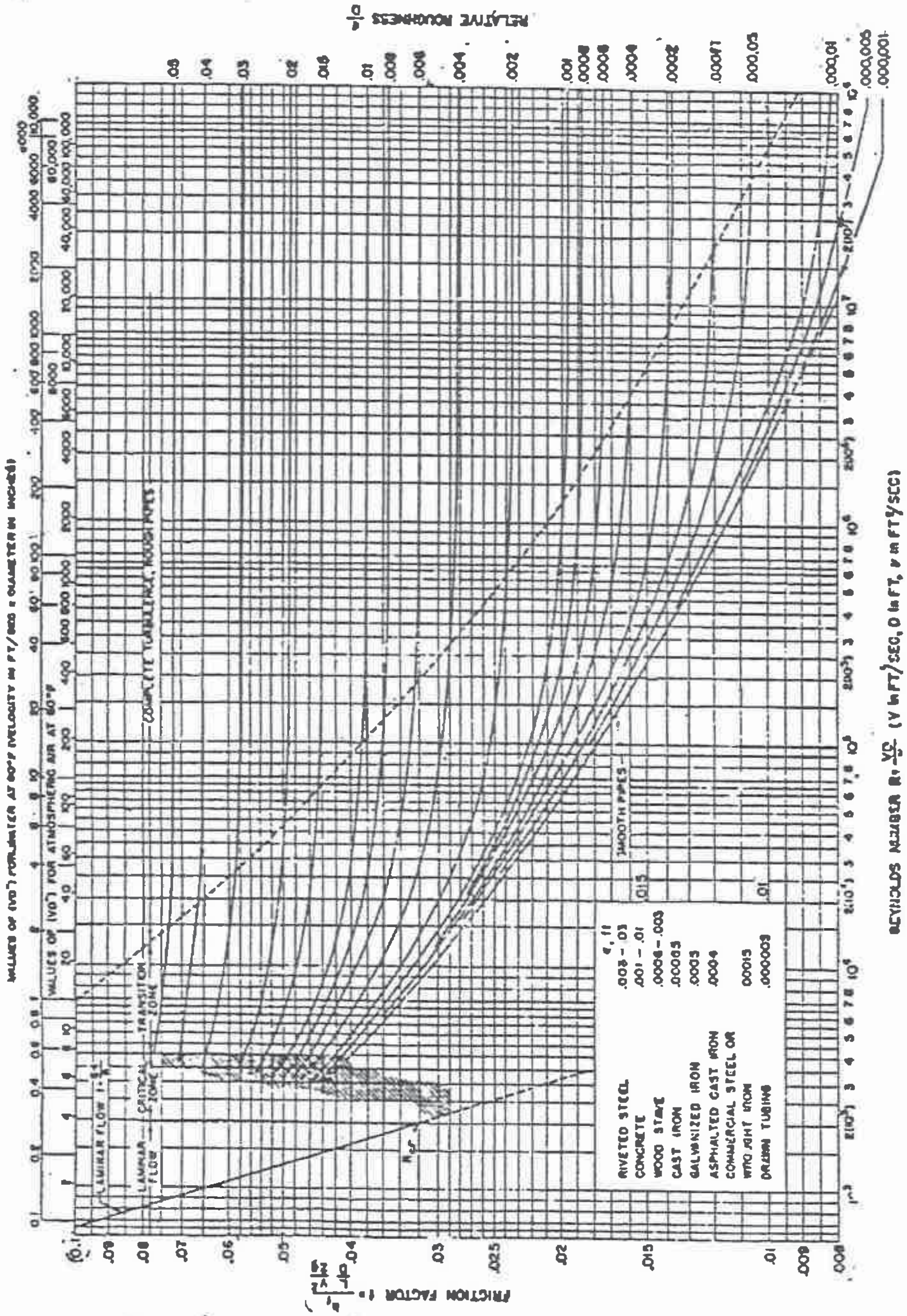



Fig. 7-4. Moody diagram. (This diagram, reproduced on a larger scale, is in an envelope attached to the back cover.)