National Exams December 2012

98-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. This is a CLOSED BOOK EXAM.
   Any Casio or Sharp approved calculator is permitted.
3. FIVE (5) questions constitute a complete exam paper.
   The first five questions as they appear in the answer book will be marked.
4. Each question is of equal value.
5. Most questions require an answer in essay format. Clarity and organization of the answer are important.
6. Note Data Sheet (11 pages in total) also included with exam paper.

1. It is required to design a tail shaft for a ship having the following particulars:

Shaft power	8280 KW
Propeller weight	120 KN
Steady thrust	1025 KN
Shaft Speed	110 RPM
Min. yield tensile strength	210 MPa
Endurance limit	180 MPa

Consider a 5% variation in the torque acting on the shaft. Use the ABS rules to find an estimate for the shaft diameter, then use the Navy method to check if this estimate is appropriate. Assume that the propeller is keyed to the shaft and that the stern tube bearings are oil lubricated. The shaft is made of carbon steel with specific weight of 76.5 KN/m^3. The specific weight of sea water is 10.55 KN/m^3.

2. It is required to select a centrifugal pump for a system aboard a certain ship having the following characteristics:

Piping length	21.0 meters
Piping diameter	0.1 meters
Equivalent piping length for secondary losses	2.0 meters
Static head	3.5 meters
Coefficient of friction	0.02

Two pumps are available whose characteristics are given as

Pump A:

Q (m^3/s)	0	0.006	0.012	0.018	0.024	0.03	0.036
H (m)	22.6	21.9	20.3	17.7	14.2	9.7	3.9
η_o	0	32	74	86	85	66	28

Pump B:

Q (m^3/s)	0	0.006	0.012	0.018	0.024	0.03	0.036
H (m)	16.2	13.6	11.9	11.6	10.7	9.0	6.4
η_o	0	14	22	60	80	80	60

Select the more suitable pump for this duty and justify your selection. Determine the discharge, the head, and the required input power for the pump.

3. A gas turbine propulsion system is modelled for the purpose of studying its longitudinal vibration characteristics as shown in Figure 1. Simplify the system to a single mass spring system to determine a first approximation to the fundamental natural frequency. Then apply Holzer's method to the actual system to determine the first natural frequency of the system (perform only two iterations). Also, find the corresponding mode shape.

The system has the following characteristics:

M (propeller)	3987.0 kg
M_1	1164.0 kg
M_2	2565.0 kg
M_3	1338.0 kg
K_1	7.6 H 10^8 N/m
K_2	1.16$10^8$ N/m
K_3	5.0 H 10^7N/m
K_4	5.0 H 10^7N/m

4. A 60.0 cm dia. Impeller centrifugal pump running at 750 rpm, has the following characteristics:

Q (m^3/s)	0.00	0.12	0.23	0.35	0.47	0.58	0.70	0.82	0.93
H (m)	40.0	40.6	40.4	39.3	38.0	33.6	25.6	14.5	0
η (%)	0	41	60	74	83	83	74	51	0

a. The pump is used to pump water from one reservoir to another, the difference between the water levels in the two reservoirs is 17.0 m. The pipeline is 45.0 cm in diameter and 130.0 m long. The coefficient of friction of the pipeline is 0.02 and contains 2 gate valves (K= 0.2) and ten 90° bends (K= 0.35). Determine the discharge of the pump and power absorbed.

b. If a geometrically similar pump having an impeller diameter of 50.0 cm is used running at 900 rpm, determine the discharge of the new pump and the power absorbed.

c. Comment on which pump you choose for the task.

5. a) Examine the balancing of a six cylinder reciprocating engine with angular crank positions of 0, 240, 120, 120, 240, 360 degrees. All cylinders are similar and each two consecutive cylinders are separated by a distance of 0.3 m. Calculate the magnitude of the unbalanced couple when the piston of the first cylinder is in the upper dead center.

b) Discuss briefly the following:

1. The difference between the maximum continuous rating and the continuous service rating for a diesel engine.

2. The difference between the mean indicated pressure and the mean effective pressure for a diesel engine.
3. The effect of the ambient conditions on the performance of a gas turbine.

6. Use Holzer's method to calculate the natural frequency of the first torsional vibration mode and the corresponding mode shape for the system shown in Figure 2.

N_1	=	380 rpm	N_2	=	120 rpm
$J_1 = J_2 = J_6$	=	250.0 $Kg.m^2$			
J_4 =		3600.0 $Kg.m^2$	$J_3 = J_7$	=	65.0 $Kg.m^2$
J_5 =		120.0 $Kg.m^2$			
J_8	=	4750.0 $Kg.m^2$	J_9	=	8400.0 $Kg.m^2$
K_1	=	21.5 E6 N.m/rad	K_2	=	14.5 E6 N.m/rad
K_3	=	3.6 E6 N.m/rad	K_4	=	10.8 E6 N.m/rad
K_5	=	72.0 E6 N.m/rad	K_6	=	180.0 E6 N.m/rad

7. A counter flow heat exchanger was designed to cool oil using sea water. The design overall heat transfer coefficient was 930.0 W/ m^2. °C and the heat transfer area is 3.33 m^2. The heat exchanger was tested after sometime to check its performance, the following data was obtained from the test:

Description	Mass flow rate (kg/s)	Specific heat (J/kg. °C)	Inlet Temperatures, °C	Outlet Temperatures, °C
Oil	2.0	2330	147	107
Water	1.0	4180	27	-

a. Determine the percentage change in the overall heat transfer coefficient as a result of fouling.
b. What is the outlet temperature of the cooling water?
c. Calculate the effectiveness of the heat exchanger.

d. Determine the NTU for this heat exchanger.

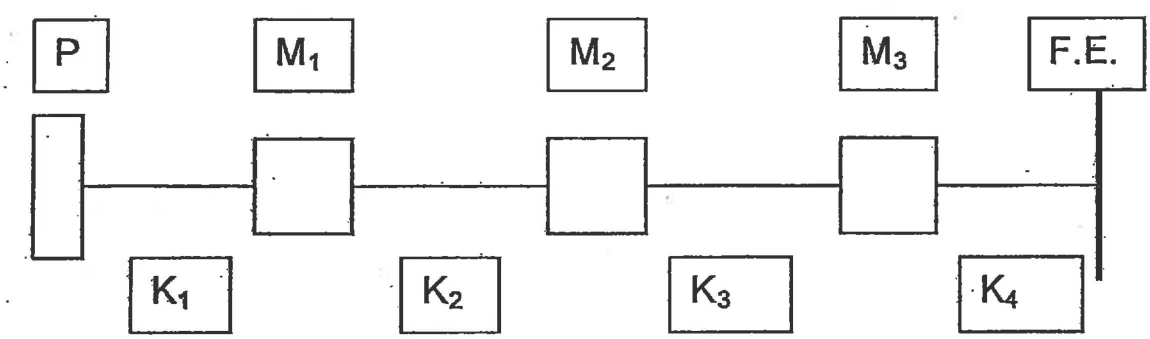


Fig. 1

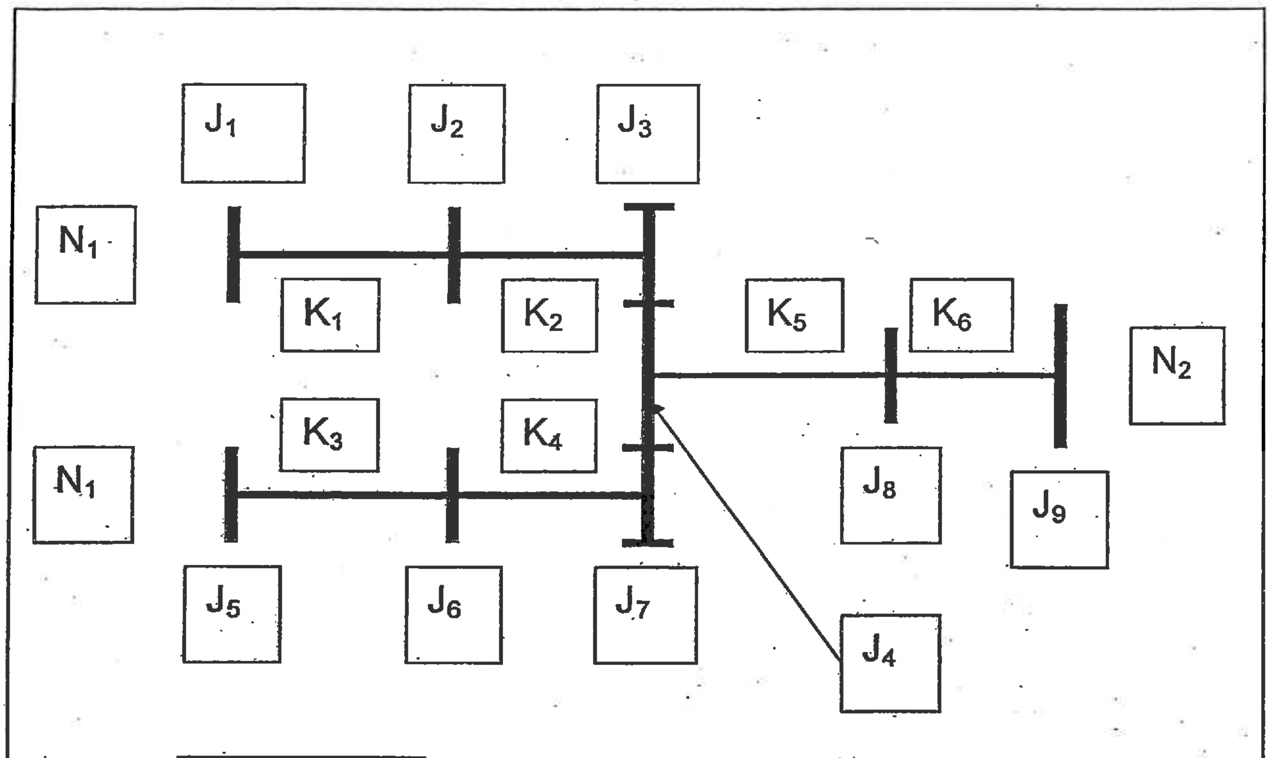


Fig. 2

Data Sheet

Three moment equation:

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

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

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

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:

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

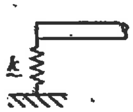
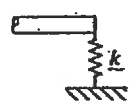
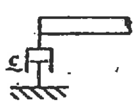
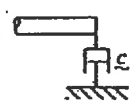
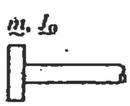
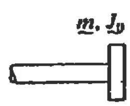
Torsional Vibration of a Shaft:

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

Losses in pipe fittings

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

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

Boundary conditions for the transverse vibration of a beam

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

Boundary conditions for a bar in longitudinal vibration.

Boundary condition	At left end ($x = 0$)	At right end ($x = l$)
Fixed end	$\theta(0, t) = 0$ $x = 0$	$\theta(l, t) = 0$ $x = l$
Free end	$\frac{\partial \theta}{\partial x}(0, t) = 0$ $x = 0$	$\frac{\partial \theta}{\partial x}(l, t) = 0$ $x = l$
End torsional spring (spring constant = $\underline{k}_t$)	k_t $GJ\frac{\partial \theta}{\partial x}(0, t) = \underline{k}_t\theta(0, t)$ $x = 0$	k_t $GJ\frac{\partial \theta}{\partial x}(l, t) = -\underline{k}_t\theta(l, t)$ $x = l$
End torsional damper (damping constant = $\underline{c}_t$)	$\underline{c}_t$ $GJ\frac{\partial \theta}{\partial x}(0, t) = \underline{c}_t\frac{\partial \theta}{\partial t}(0, t)$ $x = 0$	$\underline{c}_t$ $GJ\frac{\partial \theta}{\partial x}(l, t) = -\underline{c}_t\frac{\partial \theta}{\partial t}(l, t)$ $x = l$
End inertia (inertia = $\underline{J}_0$)	J_0 $GJ\frac{\partial \theta}{\partial x}(0, t) = \underline{J}_0\frac{\partial^2 \theta}{\partial t^2}(0, t)$ $x = 0$	J_0 $GJ\frac{\partial \theta}{\partial x}(l, t) = -\underline{J}_0\frac{\partial^2 \theta}{\partial t^2}(l, t)$ $x = l$

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

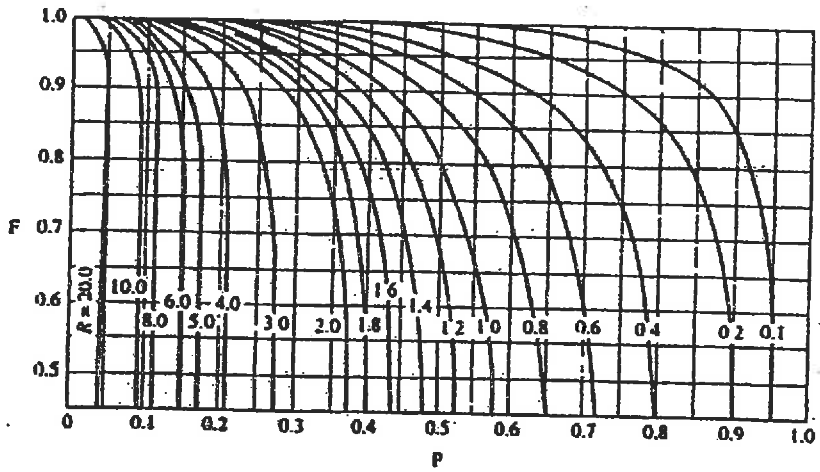


Figure 7.11 Correction-factor plot for exchanger with one shell pass and two, four, or any multiple of two tube passes

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.

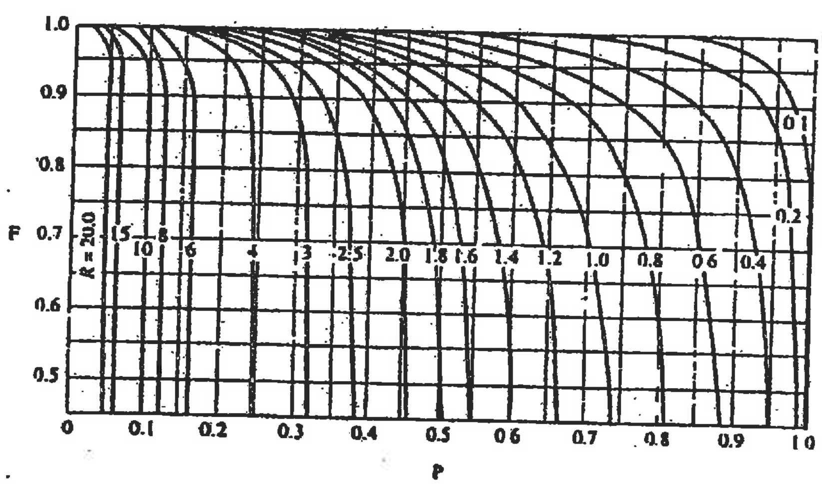


Figure 7.12 Correction-factor plot for exchanger with two shell passes and four, or any multiple of four tube passes

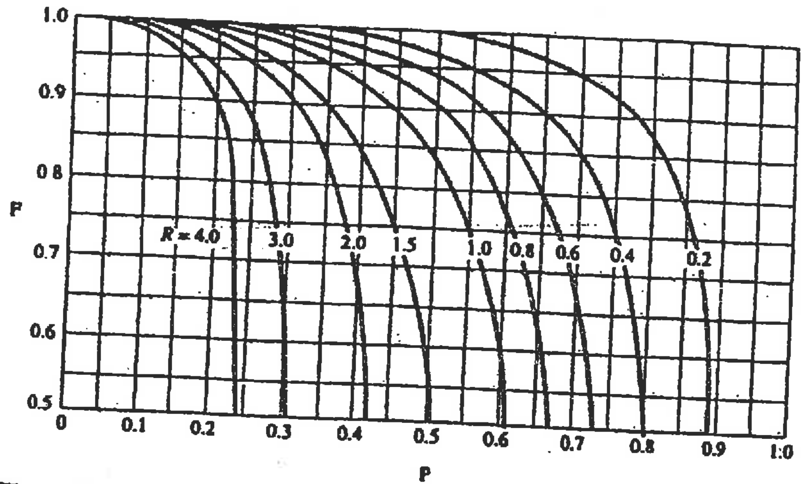


Figure 7.13 Correction-factor plot for single-pass cross-flow exchanger one fluid mixed, the other unmixed

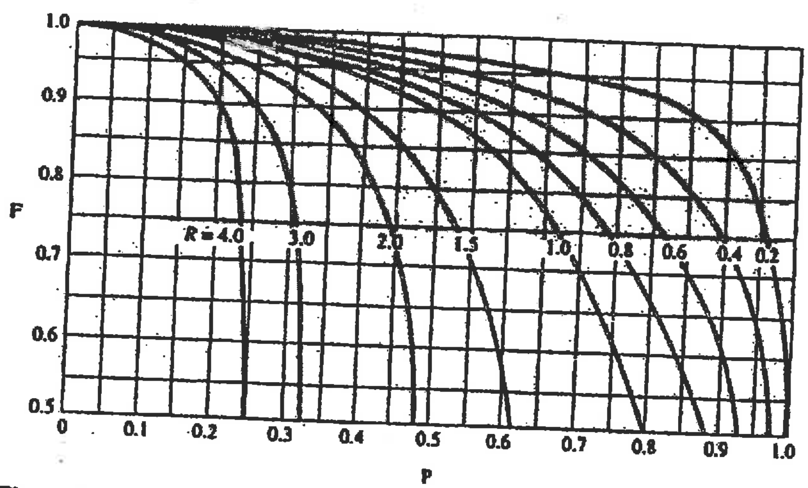
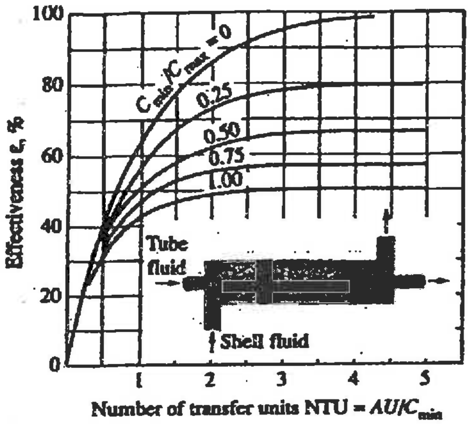


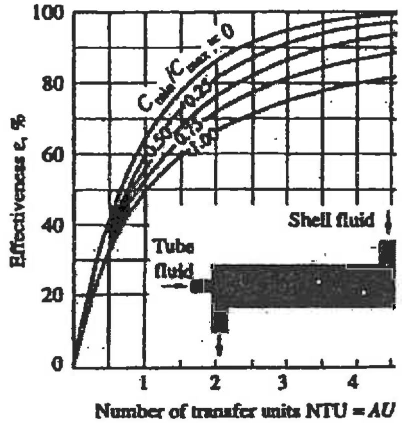
Figure 7.14 Correction-factor plot for single-pass cross-flow exchanger both fluids unmixed

FIGURE 10-26

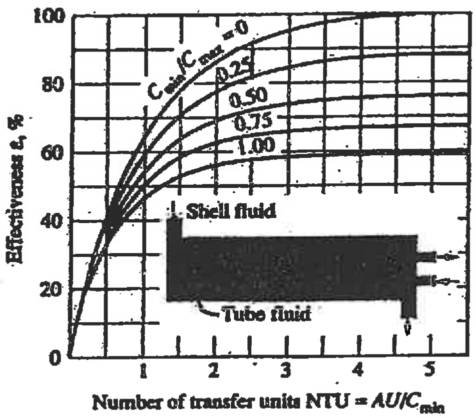
Effectiveness for heat exchangers (from Kays and London, Ref. 7).



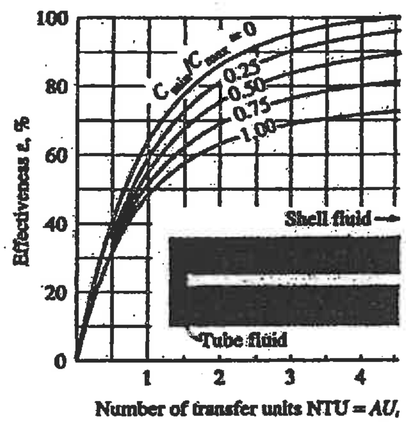
(*a*) Parallel-flow



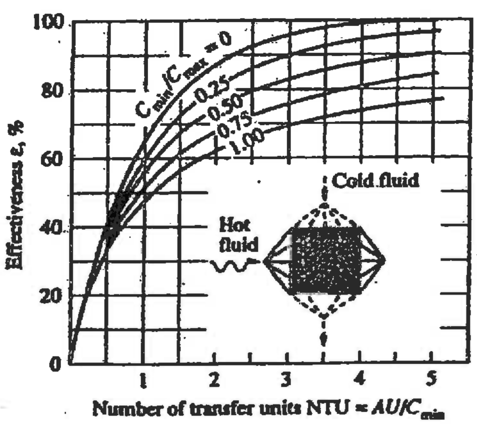
(*b*) Counter-flow



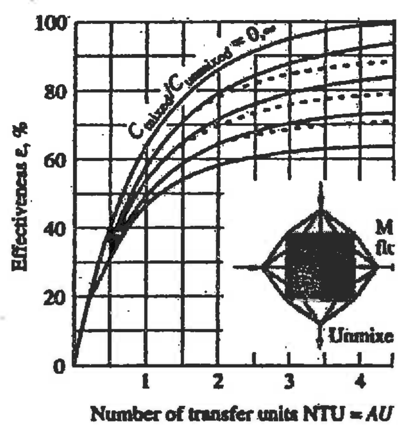
(*c*) One-shell pass and 2, 4, 6, tube passes



(*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)

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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

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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.)

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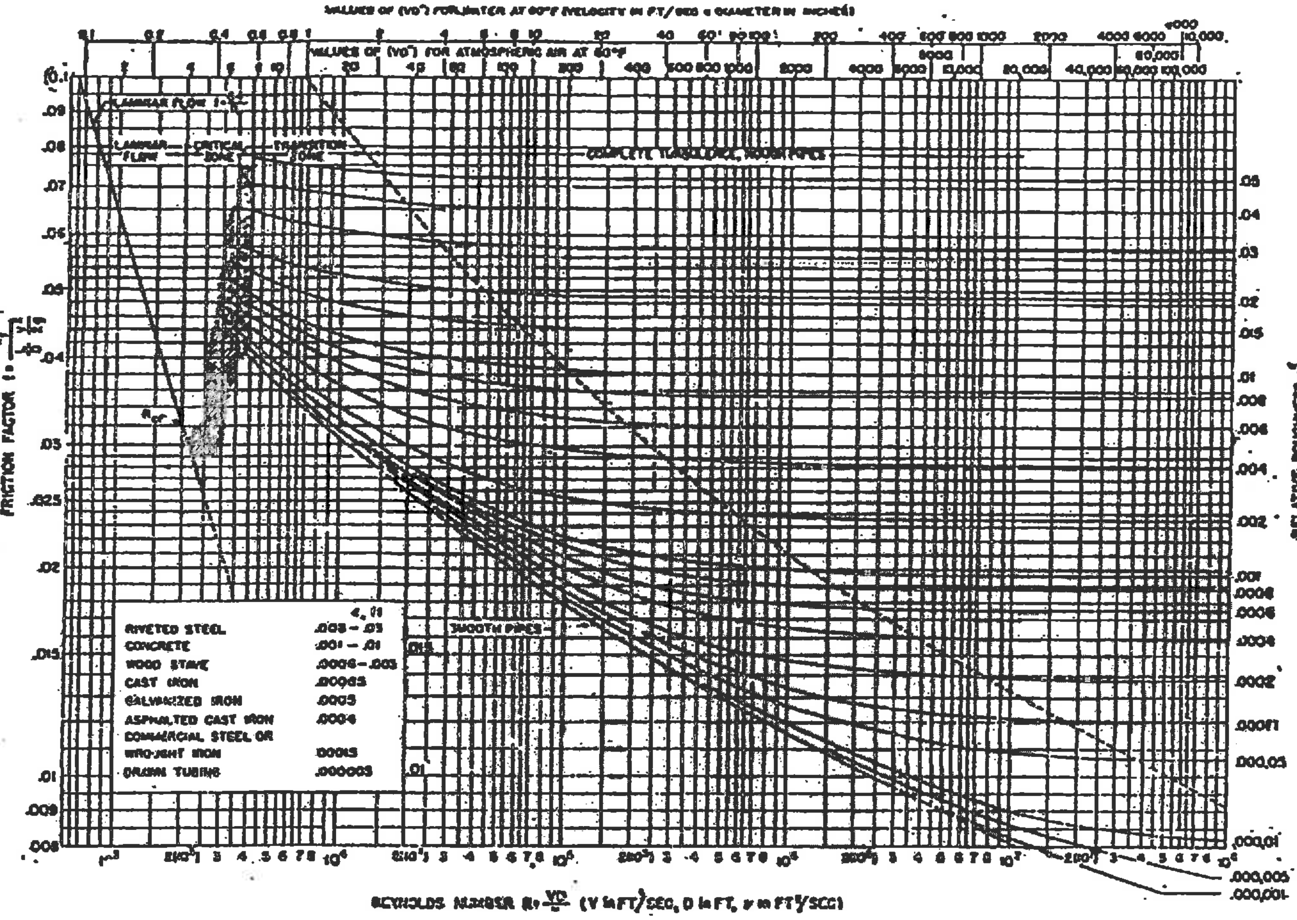

Fig. 7-4. Moody diagram. (This diagram, reproduced on a larger scale, is in an envelope attached to the back cover.)