BAB VI
PENUTUP

V.1 Kesimpulan

Dari hasil perhitungan yang telah dilakukan pada kapal rancangan yaitu Kapal Ferry 1650 GT dengan dimensi kapal sebagai berikut :

Panjang keseluruhan (L_{OA})	: 115,97 m
Panjang antara garis tegak (L_{PP})	: 104,48 m
Lebar kapal (B)	: 17,58 m
Sarat kapal (T)	: 4,35 m
Kecepatan (V_s)	: 16,00 Knot
Gross Tonage (GT)	: 1650 Ton
Radius pelayaran	: 3600 mil
	: (20 kali trip x 15 mil laut x 2 hari)
Klasifikasi	: BKI

- ⇨ Untuk dapat menentukan besarnya daya motor induk sebagai penggerak utama kapal, maka factor kecepatan daerah pelayaran serta dimensi dari kapal mempunyai pengaruh sangat besar.
- ⇨ Di dalam perancangan kamar mesin, tidak terlepas dari adanya asumsi-asumsi yang diberikan untuk mempermudah dalam perhitungan dengan tidak mengabaikan tanggung secara teknis, ekonomis dan peraturan-peraturan yang ada, sehingga hasil perhitungan dapat mendekati keadaan yang sebenarnya.
- ⇨ Tata letak mesin induk, mesin bantu, maupun peralatan perlatan lain hendaknya diatur seefesien mungkin, hal ini untuk mempermudah dalam perawatan dan perbaikan peralatan yang ada di kamar mesin itu sendiri.
- ⇨ Peletakan permesinan berpengaruh pada stabilitas kapal.
- ⇨ Pemilihan mesin bantu tergantung dari jumlah daya yang harus disuplai pada kondisi operasi kapal yang berbeda beda.

V.2 Saran-saran

Kesempurnaan dari hasil penulisan adalah merupakan tujuan yang ingin dicapai penulis. Untuk itu penulis telah berusaha semaksimal mungkin dengan bantuan dan bimbingan dari dosen pembimbing.

Tetapi dalam hal ini penulis menyadari bahwa dalam penulisan masih banyak terdapat kesalahan dan kekurangan, maka dari itu penulis berharap adanya sumbangan pikiran untuk memperbaikan dalam tugas merancang ini.

Akhirnya, semoga ugas merancang ini dapat berguna bagi pembaca sekalian.

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Section 14

Rudder and Manoeuvring Arrangement

A. General

1. Manoeuvring arrangement

1.1 Each ship is to be provided with a manoeuvring arrangement which will guarantee sufficient manoeuvring capability.

1.2 The manoeuvring arrangement includes all parts from the rudder and steering gear to the steering position necessary for steering the ship.

1.3 Rudder stock, rudder coupling, rudder bearings and the rudder body are dealt with in this Section. The steering gear is to comply with Volume III, Section 14.

1.4 The steering gear compartment shall be readily accessible and, as far as practicable, separated from the machinery space. (See also Chapter II-1, Reg. 29.13 of SOLAS 74.)

Guidance

Concerning the use of non-magnetic material in the wheel house in way of a magnetic compass, the requirements of the national Administration concerned are to be observed.

1.5 For ice-strengthening see Section 15.

2. Structural details

2.1 Effective means are to be provided for supporting the weight of the rudder body without excessive bearing pressure, e.g. by a rudder carrier attached to the upper part of the rudder stock. The hull structure in way of the rudder carrier is to be suitably strengthened.

2.2 Suitable arrangements are to be provided to prevent the rudder from lifting.

2.3 The rudder stock is to be carried through the hull either enclosed in a watertight trunk, or glands are to be fitted above the deepest load waterline, to prevent water from entering the steering gear compartment and the lubricant from being washed away from the rudder carrier. If the top of the rudder trunk is below the deepest waterline two separate stuffing boxes are to be provided.

3. Size of rudder area

In order to achieve sufficient manoeuvring capability the size of the movable rudder area A is recommended to be not less than obtained from the following formula:

$$A = c_1 \cdot c_2 \cdot c_3 \cdot c_4 \cdot \frac{1{,}75 \cdot L \cdot T}{100} \qquad [m^2]$$

c_1 = factor for the ship type:

= 1,0 in general

= 0,9 for bulk carriers and tankers having a displacement of more than 50.000 ton

= 1,7 for tugs and trawlers

c_2 = factor for the rudder type:

= 1,0 in general

= 0,9 for semi-spade rudders

= 0,8 for double rudders (per rudder)

= 0,7 for high lift rudders

c_3 = factor for the rudder profile:

= 1,0 for NACA-profiles and plate rudder

= 0,8 for hollow profiles

c_4 = factor for the rudder arrangement:

= 1,0 for rudders in the propeller jet

= 1,5 for rudders outside the propeller jet

For semi-spade rudder 50% of the projected area of the rudder horn may be included into the rudder area A.

Materials

1 For materials for rudder stock, pintles, cou-ing bolts etc. see Rules for Material Volume V:)ecial material requirements are to be observed for c ice notations ES3 and ES4 as well as for the arctic c notations Arc 1- Arc 4.

2 In general materials having a minimum)minal upper yield point R_{eH} of less than 200 /mm^2 and a minimum tensile strength of less than)0 N/mm^2 or more than 900 N/mm^2 shall not be sed for rudder stocks, pintles, keys and bolts. The quirements of this Section are based on a material's inimum nominal upper yield point R_{eH} of 235 /mm^2. If material is used having a R_{eH} differing om 235 N/mm^2, the material factor k_r is to be etermined as follows:

$$k_r = \left[\frac{235}{R_{eH}}\right]^{0.75} \qquad \text{for } R_{eH} > 235 \text{ N/mm}^2$$

$$k_r = \frac{235}{R_{eH}} \qquad \text{for } R_{eH} < 235 \text{ N/mm}^2$$

R_{eH} = minimum nominal upper yield point of material used in [N/mm^2]. R_{eH} is not to be taken greater than $0{,}7 \cdot R_m$ or 450 N/mm^2, whichever is less. R_m = tensile strength of the material used.

.3 Before significant reductions in rudder stock iameter due to the application of steels with R_{eH} xceeding 235 N/mm^2 are granted, the Society may equire the evaluation of the elastic rudder stock eflections. Large deflections should be avoided in rder to avoid excessive edge pressures in way of earings.

.4 The permissible stresses given in E.1. are pplicable for ordinary hull structural steel. When igher tensile steels are used, higher values may be sed which will be fixed in each individual case.

. Definitions

C_R = rudder force in [N]

Q_R = rudder torque in [Nm]

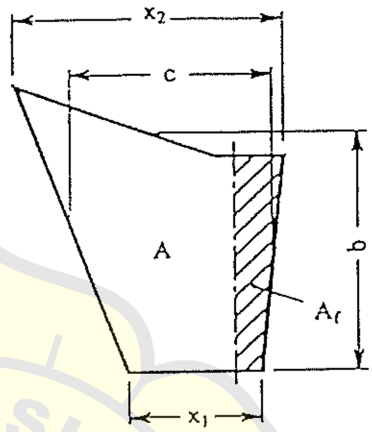
A = total movable area of the rudder in [m^2] For nozzle rudders, A is not to be taken less than 1,35 times the projected area of the nozzle;

A_t = A + area of a rudder horn, if any, in [m^2]

A_f = portion of rudder area located ahead of the rudder stock axis in [m^2]

b = mean height of rudder area in [m]

c = mean breadth of rudder area in [m] (see Fig. 14.1)



$$c = \frac{x_1 + x_2}{2} \qquad b = \frac{A}{c}$$

Fig. 14.1

Λ = aspect ratio of rudder area A_t

$\Lambda = b^2/A_t$

v_0 = ahead speed of ship in [kn] as defined in Section 1, H.5. if this speed is less than 10 kn, v_0 is to be taken as

$v_{min} = (v_0 + 20)/3$ [kn]

v_a = astern speed of ship in [kn]; if the astern speed $v_a \leq 0{,}4 \cdot v_0$ or 6 kn, whichever is less, determination of rudder force and torque for astern condition is not required. For greater astern speeds special evaluation of rudder force and torque as a function of the rudder angle may be required. If no limitations for the rudder angle at astern condition is stipulated, the factor κ_2 is not to be taken less than given in Table 14.1 for astern condition.

k = material factor according to Section 2, B.2.

B. Rudder Force and Torque

1. Rudder force and torque for normal rudders

1.1 The rudder force is to be determined according to the following formula:

$C_R = 132 \cdot A \cdot v^2 \cdot \kappa_1 \cdot \kappa_2 \cdot \kappa_3 \cdot \kappa_t$ [N]

v = v_0 for ahead condition

v = v_a for astern condition

κ_1 = coefficient, depending on the aspect ratio Λ

κ_1 = $(\Lambda + 2)/3$, where Λ need not be taken greater than 2

κ_2 = coefficient, depending on the type of the rudder and the rudder profile according to Table 14.1.

κ_3 = coefficient, depending on the location of the rudder

κ_3 = 0,8 for rudders outside the propeller jet

κ_3 = 1,15 for rudders aft of the propeller nozzle

κ_3 = 1,0 elsewhere, including also rudders within the propeller jet

κ_t = coefficient depending on the thrust coefficient c_t

κ_t = 1,0 normally

In special cases for thrust coefficients $c_t > 1,0$ determination of κ_t according to the following formula may be required:

$$\kappa_t = \frac{C_R\,(c_t)}{C_R\,(c_t = 1,0)}$$

Table 14.1

Profile/ type of rudder	κ_2 ahead	κ_2 astern
NACA-00 series Göttinger profiles	1,1	1,4
flat side profiles	1,1	1,4
hollow profiles	1,35	1,4
high lift rudders	1,7	to be specially considered; if not known: 1,7

1.2 The rudder torque is to be determined by the following formula:

$Q_R = C_R \cdot r$ [Nm]

$r = c\,(\alpha - k_b)$ [m]

α = 0,33 for ahead condition

α = 0,66 for astern condition (general)

α = 0,75 for astern condition (hollow profiles)

For parts of a rudder behind a fixed structure such as a rudder horn:

α = 0,25 for ahead condition

α = 0,55 for astern condition.

For high lift rudders α is to be specially considered. If not known, $\alpha = 0,4$ may be used for the ahead condition

k_b = balance factor as follows:

k_b = A_f/A

k_b = 0,08 for unbalanced rudders

r_{min} = $0,1 \cdot c$ [m] for ahead condition.

2. Rudder force and torque for rudder blades with cut-outs (semi-spade rudders)

2.1 The total rudder force C_R is to be calculated according to 1.1. The pressure distribution over the rudder area, upon which the determination of rudder torque and rudder blade strength is to be based, is to be derived as follows:

The rudder area may be divided into two rectangular or trapezoidal parts with areas A_1 and A_2 (see Fig. 14.2).

The resulting force of each part may be taken as:

$$C_{R1} = C_R \frac{A_1}{A} \text{ [N]}$$

$$C_{R2} = C_R \frac{A_2}{A} \text{ [N]}$$

2.2 The resulting torque of each part may be taken as:

$Q_{R1} = C_{R1} \cdot r_1$ [Nm]

$Q_{R2} = C_{R2} \cdot r_2$ [Nm]

$r_1 = c_1\,(\alpha - k_{b1})$ [m]

$r_2 = c_2\,(\alpha - k_{b2})$ [m]

$k_{b1} = A_{1f}/A_1$

$k_{b2} = A_{2f}/A_2$

A_{1f}, A_{2f} see Fig. 14.2

$C_1 = A_1/b_1$

3.2 Under strength decks in way of 0,6 L amidships, girders are to be fitted in alignment with longitudinal walls, which are to extend at least over three frame spacings beyond the end points of the longitudinal walls. The girders are to overlap with the longitudinal walls by at least two frame spacings.

4. Transverse structure of super-structures and deckhouses

The transverse structure of superstructures and deckhouses is to be sufficiently dimensioned by a suitable arrangement of end bulkheads, web frames, steel walls of cabins and casings, or by other measures.

5. Openings in closed superstructures

5.1 All access openings in end bulkheads of closed superstructures shall be fitted with weathertight doors permanently attached to the bulkhead, having the same strength as the bulkhead. The doors shall be so arranged that they can be operated from both sides of the bulkhead. The coaming heights of the access opening above the deck are to be determined according to LLC 66.

5.2 Any opening in a superstructure deck or in a deckhouse deck directly above the freeboard deck (deckhouse surrounding companionways), is to be protected by efficient weatherthight closures.

B. Side Plating and Decks of Non-Effective Superstructures

1. Side plating

1.1 The thickness of the side plating is not to be less than the greater of the following values:

$$t = 1{,}26 \cdot a \sqrt{p_s \cdot k} + t_K \quad [\text{mm}] \quad \text{or}$$

$$t = 0{,}8 \cdot t_2 \quad [\text{mm}]$$

t_2 see Section 6, B. 3.1.

1.2 The thickness of the side plating of upper tier superstructures may be reduced by 0,5 mm.

2. Deck plating

2.1 The thickness of deck plating is not to be less than the greater of the following values:

$$t = 1{,}26 \cdot a \sqrt{p \cdot k} + t_K \quad [\text{mm}]$$

$$t = (5{,}5 + 0{,}02\ L) \sqrt{k} \quad [\text{mm}]$$

p = p_{DA} or p_L, the greater value is to be taken.

L need not be taken greater than 200 m.

2.2 Where additional superstructures are arranged on non-effective superstructures located on the strength deck, the thickness required by 2.1 may be reduced by 10 percent.

2.3 Where plated decks are protected by sheathing, the thickness of the deck plating according to 2.1 and 2.2 may be reduced by t_K, however, it is not to be less than 5 mm.

Where a sheathing other than wood is used, attention is to be paid that the sheathing does not affect the steel. The sheathing is to be effectively fitted to the deck.

3. Deck beams, supporting deck structure, frames

3.1 The scantlings of the deck beams and the supporting deck structure are to be determined in accordance with Section 10.

3.2 The scantlings of superstructure frames are given in Section 9, A.3.

C. Superstructure End Bulkheads and Deckhouse Walls

1. General

The following requirements apply to superstructure end bulkheads and deckhouse walls forming the only protection for openings as per Regulation 18 of LLC 66 and for accommodations.

2. Definitions

The design load for determining the scantlings is:

$$p_A = n \cdot c\,(b \cdot f - z) \quad [\text{kN/m}^2]$$

$n = 20 + \frac{L}{12}$ for the lowest tier of unprotected fronts. The lowest tier is normally that tier which is directly situated above the uppermost continuous deck to which the Rule depth H is to be measured. However, where the actual distance exceeds the minimum non-corrected tabular freeboard by at least one standard superstructure height, this tier may be defined as the 2nd tier and the tier above as the 3rd tier.

$n = 10 + \frac{L}{12}$ for 2nd tier unprotected fronts

$n = 5 + \frac{L}{15}$ for 3rd tier and tiers above of unprotected fronts, for sides and protected fronts

$n = 7 + \frac{L}{100} - 8\frac{x}{L}$ for aft ends abaft amidships

for aft ends forward of amidships

L need not be taken greater than 300 m.

$$b = 1{,}0 + \left[\frac{\frac{x}{L} - 0{,}45}{C_B + 0{,}2}\right]^2 \qquad \text{for } \frac{x}{L} < 0{,}45$$

$$b = 10 + 1{,}5\left[\frac{\frac{x}{L} - 0{,}45}{C_B + 0{,}2}\right]^2 \qquad \text{for } \frac{x}{L} < 0{,}45$$

$0{,}60 \le C_B \le 0{,}80$; when determining scantlings of aft ends forward of amidships, C_B need not be taken less than 0,8.

x = distance in [m] between the bulkhead considered and aft end of the length L. When determining sides of a deckhouse, the deckhouse is to be subdivided into parts of approximately equal length, not exceeding 0,15 L each, and x is to be taken as the distance between aft end of the length L and the centre of each part considered.

$$f = 0{,}1\,L \cdot e^{-\frac{L}{300}} - \left[1 - \left[\frac{L}{150}\right]^2\right] \qquad \text{for } L < 150 \text{ m}$$

$$f = 0{,}1\,L \cdot e^{-\frac{L}{300}} \qquad \text{for } 150 \text{ m} \le L \le 300 \text{ m}$$

$$f = 11{,}04 \qquad \text{for } L > 300 \text{ m}$$

The factor f may be taken from Table 16.2.

z = vertical distance in [m] from the summer load line to the midpoint of stiffener span, or to the middle of the plate field.

$$c = 0{,}3 + 0{,}7\,\frac{b'}{B'}$$

b' = breadth of deckhouse at the position considered

B' = actual maximum breadth of ship on the exposed weather deck at the position considered.

b'/B' is not to be taken less than 0,25.

For exposed parts of machinery casings, c is not to be taken less than 1,0.

The design load p_A is not to be taken less than the minimum values given in Table 16.3.

a = spacing of stiffeners in [m]

ℓ = unsupported span in [m]; ℓ is to be taken as the superstructure height or deckhouse height respectively, however, not less than 2,0 m.

$C_2 \;=\; A_2/b_2$

b_1, b_2 = mean heights of the partial rudder areas A_1 and A_2 (see Fig. 14.2).

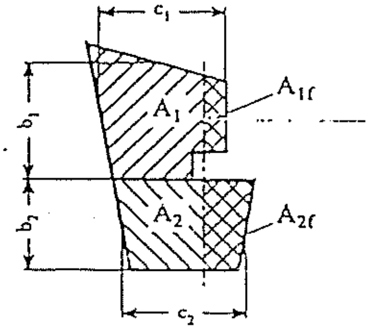


Fig. 14.2

2.3 The total rudder torque is to be determined according to the following formulae:

$$Q_R = Q_{R1} + Q_{R2} \quad [\text{Nm}] \quad \text{or}$$

$$Q_{Rmin} = C_R \cdot r_{1,2min} \quad [\text{Nm}]$$

$$r_{1,2min} = \frac{0,1}{A}\,(c_1 \cdot A_1 + c_2 \cdot A_2) \qquad [\text{m}]$$

for ahead condition

The greater value is to be taken.

C. Scantlings of the Rudder Stock

1. Rudder stock diameter

1.1 The diameter of the rudder stock for transmitting the torsional moment is not to be less than:

$$D_t = 4,2\,\sqrt[3]{Q_R \cdot k_r} \qquad [\text{mm}]$$

Q_R see B. 1.2 and B. 2.2 - 2.3.

The related torsional stress is:

$$\tau_t = \frac{68}{k_r} \qquad [\text{N/mm}^2]$$

k_r see A.4.2.

1.2 The diameter of the rudder stock determined according to 1.1 is decisive for the steering gear, the stoppers and the locking device.

1.3 In case of mechanical steering gear the diameter of the rudder stock in its upper part which is only intended for transmission of the torsional moment from the auxiliary steering gear may be 0,9 D_t. The length of the edge of the quadrangle for the auxiliary tiller must not be less than 0,77 D_t and the height not less than 0,8 D_t.

1.4 The rudder stock is to be secured against axial sliding. The degree of the permissible axial clearance depends on the construction of the steering engine and on the bearing.

2. Strengthening of rudder stock

2.1 If the rudder is so arranged that additional bending stresses occur in the rudder stock, the stock diameter has to be suitably increased. The increased diameter is, where applicable, decisive for the scantlings of the coupling.

For the increased rudder stock diameter the equivalent stress of bending and torsion is not to exceed the following value:

$$\sigma_v = \sqrt{\sigma_b^2 + 3\,\tau^2} \le 118/k_r \qquad [\text{N/mm}^2]$$

Bending stress:

$$\sigma_b = \frac{10,2 \cdot M_b}{D_1^3} \qquad [\text{N/mm}^2]$$

M_b = bending moment at the neck bearing in [Nm]

Torsional stress:

$$\tau = \frac{5,1 \cdot Q_R}{D_1^3} \qquad [\text{N/mm}^2]$$

D_1 = increased rudder stock diameter in [cm]

The increased rudder stock diameter may be determined by the following formula:

$$D_1 = D_t\,\sqrt[6]{1 + \frac{4}{3}\left[\frac{M_b}{Q_R}\right]^2}$$

Q_R see B.1.2 and B.2.2 - 2.3

D_t see 1.1.

Guidance

Where a double-piston steering gear is fitted, additional bending moments may be transmitted from the steering gear into the rudder stock. These additional bending moments are to be taken into account for determining the rudder stock diameter.

3. Analysis

3.1 General

The evaluation of bending moments, shear forces and support forces for the system rudder - rudder stock may be carried out for some basic rudder types as shown in Figs. 14.3 - 14.5 as outlined in 3.2. - 3.3.

3.2 Data for the analysis

ℓ_{10} - ℓ_{50} = lengths of the individual girders of the system in [m]

I_{10} - I_{50} = moments of inertia of these girders in [cm^4]

For rudders supported by a sole piece the length ℓ_{20} is the distance between lower edge of rudder body and centre of sole piece, and I_{20} is the moment of inertia of the pintle in the sole piece.

load on rudder body (general):

$$P_R = \frac{C_R}{\ell_{10} \cdot 10^3} \quad [\text{kN/m}]$$

load on semi-spade rudders:

$$P_{R10} = \frac{C_{R2}}{\ell_{10} \cdot 10^3} \quad [\text{kN/m}]$$

$$P_{R20} = \frac{C_{R1}}{\ell_{20} \cdot 10^3} \quad [\text{kN/m}]$$

C_R, C_{R1}, C_{R2} see B.1. and B.2.

Z = spring constant of support in the sole piece or rudder horn respectively

for the support in the sole piece (Fig. 14.3)

$$Z = \frac{6{,}18 \cdot I_{50}}{\ell_{50}^{\,3}} \quad [\text{kN/m}]$$

for the support in the rudder horn (Fig. 14.4)

$$Z = \frac{1}{f_b + f_t} \quad [\text{kN/m}]$$

f_b = unit displacement of rudder horn in [m] due to a unit force of 1 kN acting in the centre of support

$$f_b = 0{,}21 \frac{d^3}{I_n} \quad [\text{m/kN}] \text{ (guidance value)}$$

I_n = moment of inertia of rudder horn around the x-axis at d/2 in [cm^4] (see also Fig. 14.4)

f_t = unit displacement due to torsion

$$f_t = \frac{d \cdot e^2 \cdot \sum u_i/t_i}{3{,}14 \cdot 10^8 \cdot F_T^{\,2}} \quad [\text{m/kN}]$$

F_T = mean sectional area of rudder horn in [m^2]

u_i = breadth in [mm] of the individual plates forming the mean horn sectional area

t_i = plate thickness within the individual breadth u_i in [mm]

e, d = distances in [m] according to Fig. 14.4.

3.3 Moments and forces to be evaluated

3.3.1 The bending moment M_R and the shear force Q_1 in the rudder body, the bending moment M_b in the neck bearing and the support forces B_1, B_2, B_3 are to be evaluated.

The so evaluated moments and forces are to be used for the stress analyses required by 2. and E.1. of this Section and by Section 13, C.4. and C.5.

3.3.2 For spade rudders the moments and forces may be determined by the following formulae:

$$M_b = C_R \left[\ell_{20} + \frac{\ell_{10}(2x_1 + x_2)}{3(x_1 + x_2)} \right] \quad [\text{Nm}]$$

$$B_3 = \frac{M_b}{\ell_{30}} \quad [\text{N}]$$

$$B_2 = C_R + B_3 \quad [\text{N}]$$

4. Rudder trunk

Where the rudder stock is arranged in a trunk in such a way that the trunk is stressed by forces due to rudder action, the scantlings of the trunk are to be as such that the equivalent stress due to bending and shear does not exceed $0{,}35 \cdot R_{eH}$ of the material used.

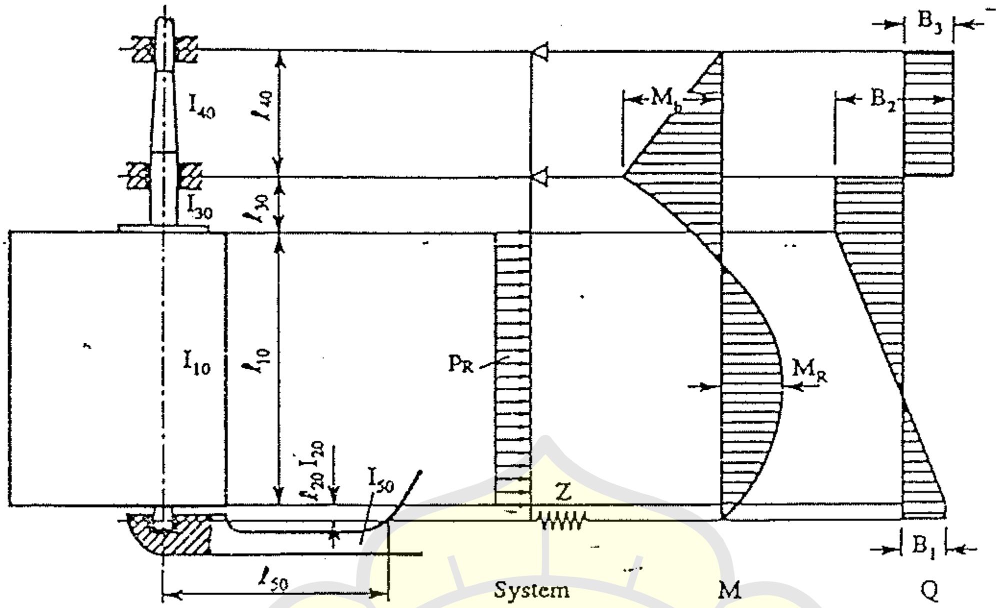


Fig. 14.3 Rudder supported by sole piece

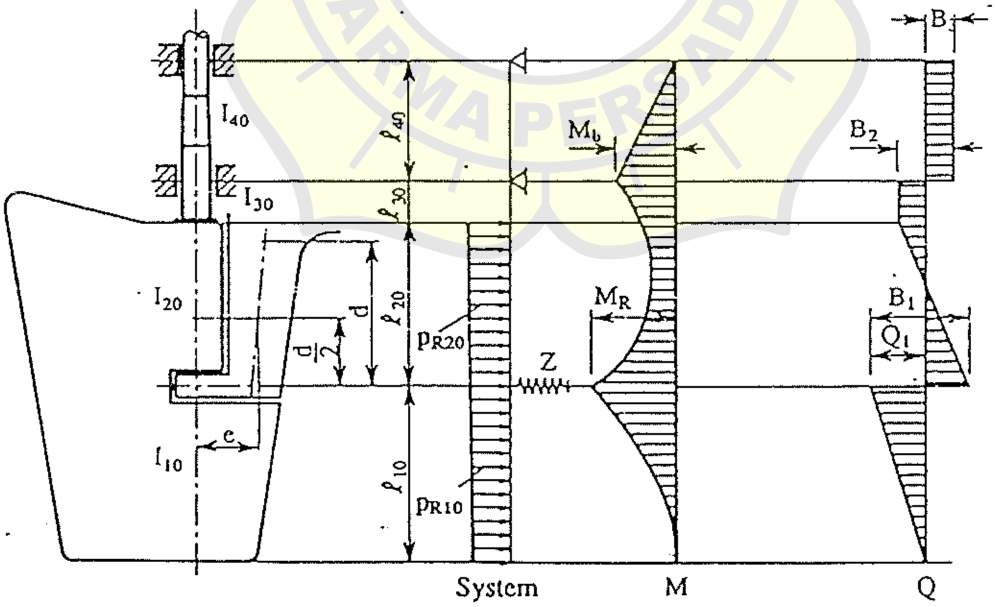


Fig. 14.4 Semi-spade rudder

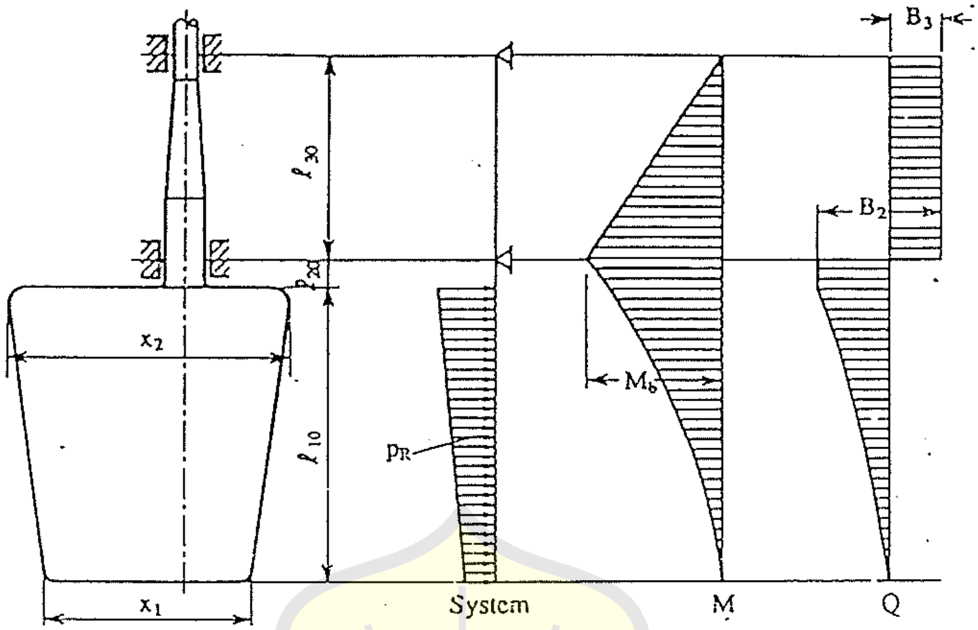


Fig. 14.5 Spade rudder

D. Rudder Couplings

1. General

1.1 The couplings are to be designed in such a way as to enable them to transmit the full torque of the rudder stock.

1.2 The distance of bolt axis from the edges of the flange is not to be less than 1,2 the diameter of the bolt. In horizontal couplings, at least 2 bolts are to be arranged forward of the stock axis.

1.3 The coupling bolts are to be fitted bolts. Their nuts are to be effectively locked, e.g., according to recognized standards.

1.4 For spade rudders horizontal couplings according to 2. are permissible only where the required thickness of the coupling flanges t_f is less than 50 mm, other wise cone couplings according to 4. are to be applied. For spade rudders of the high lift type, only cone couplings according to 4. are permitted.

2. Horizontal couplings

2.1 The diameter of coupling bolts is not to be less than:

$$d_b = 0{,}62\sqrt{\frac{D^3 \cdot k_b}{k_r \cdot n \cdot e}} \text{ [mm]}$$

D = rudder stock diameter according to C. in [mm]

n = total number of bolts, which is not to be less than 6

e = mean distance of the bolt axes from the centre of bolt system in [mm]

k_r = material factor for the rudder stock as given in A.4.2

k_b = material factor for the bolts analogue to A.4.2.

2.2 The thickness of the coupling flanges is not to be less than determined by the following formulae:

$$t_f = 0{,}62\sqrt{\frac{D^3 \cdot k_f}{k_r \cdot n \cdot e}} \text{ [mm]}$$

$$t_{fmin} = 0{,}9 \cdot d_b$$

k_f = material factor for the coupling flanges analogue to A.4.2.

The thickness of the coupling flanges clear of the bolt holes is not to be less than $0{,}65 \cdot t_f$.

The width of material outside the bolt holes is not to be less than $0{,}67 \cdot d_b$.

2.3 The coupling flanges are to be equipped with a fitted key according to recognized standards for relieving the bolts.

ed key may be dispensed with if the diameter olts is increased by 10%.

Horizontal coupling flanges should be forged :r with the rudder stock. If the flanges are to the rudder stock, the stock should have a ange[1]) with a diameter of 1,1 D (but not less + 20 mm) and with a thickness equal to the thickness (max. flange thickness + 5 mm).

For the connection of the coupling flanges ıe rudder body see also Section 19, B.4.4.

Vertical couplings

The diameter of the coupling bolts is not to ; than:

$$= \frac{0{,}81 \cdot D}{\sqrt{n}} \sqrt{\frac{k_b}{k_r}} \quad [mm]$$

k_r, n see 2.1, where n is not to be less than 8.

The first moment of area of the bolts about :ntre of the coupling is not to be less than:

$= 0{,}00043\ D^3 \quad [cm^3]$.

The thickness of the coupling flanges is not less than

$= d_b$ [mm]

width of material outside the bolt holes is not to ss than $0{,}67 \cdot d_b$.

Cone couplings

Cone couplings with key

1 Cone couplings should have a taper c on diaer of 1: 8 - 1:12.

$(d_0 - d_u)/\ell$ according Fig. 14.6.).

: cone shape should be very exact. The nut is to be :fully secured, e.g. as shown in Fig. 14.6.

[1] In special cases (e.g., for small stock diameter) weld flange may be dispensed with.

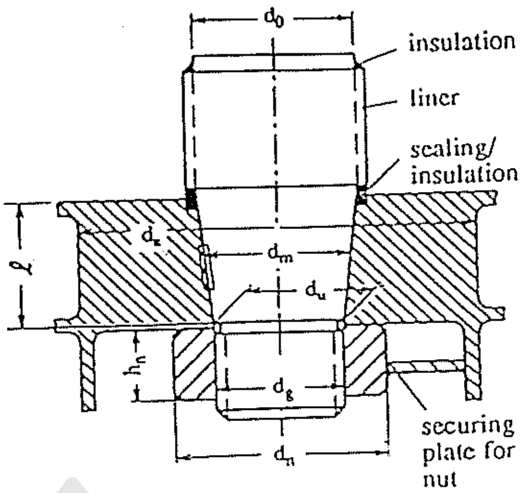


Fig. 14.6

4.1.2 The coupling length ℓ should, in general, not be less than $1{,}5 \cdot d_0$.

4.1.3 For couplings between stock and rudder a key is to be provided, the shear area of which is not to be less than:

$$a_s = \frac{16 \cdot Q_F}{d_k \cdot R_{eH1}} \quad [cm^2]$$

Q_F = design yield moment of rudder stock in [Nm] according to F.

d_k = diameter of the conical part of the rudder stock in [mm] at the key

R_{eH1} = minimum nominal upper yield point of the key material in [N/mm^2]

4.1.4 The effective surface area of the key (without rounded edges) between key and rudder stock or cone coupling, is not to be less than:

$$a_k = \frac{5 \cdot Q_F}{d_k \cdot R_{eH2}} \quad [cm^2]$$

R_{eH2} = minimum nominal upper yield point of the key, stock or coupling material in [N/mm^2], whichever is less.

4.1.5 The dimensions of the slugging nut are to be as follows:

- height:

  $h_n = 0{,}6 \cdot d_g$

- outer diameter (the greater value to be taken):

  $d_n = 1{,}2 \cdot d_u$ or $d_n = 1{,}5 \cdot d_g$

- external thread diameter:

$d_g = 0{,}65 \cdot d_0$

See Fig. 14.6.

4.1.6 It is to be proved that 50% of the design yield moment will be solely transmitted by friction in the cone couplings. This can be done by calculating the required push-up pressure and push-up length according to 4.2.3 for a torsional moment $Q'_F = 0{,}5 \cdot Q_F$

4.2 Cone couplings with special arrangements for mounting and dismounting the couplings

4.2.1 Where the stock diameter exceeds 200 mm the press fit is recommended to be effected by a hydraulic pressure connection. In such cases the cone should be more slender (c = 1:12 to = 1 : 20).

4.2.2 In case of hydraulic pressure connections the nut is to be effectively secured against the rudder stock or the pintle. A securing plate for securing the nut against the rudder body is not to be provided, see Fig. 14.7.

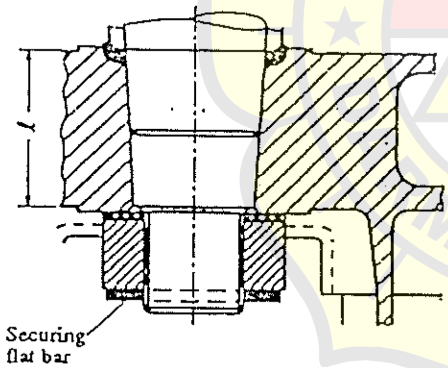


Fig. 14.7

4.2.3 For the safe transmission of the torsional moment by the coupling between rudder stock and rudder body the required push-up length and the push-up pressure are to be determined by the following formulae:

.1 required push-up pressure

$$p_{req1} = \frac{2 \cdot Q_F \cdot 10^3}{d_m^{\,2} \cdot \ell \cdot \pi \cdot \mu_0} \quad [\text{N/mm}^2]$$

or

$$p_{req2} = \frac{6 \cdot M_b \cdot 10^3}{\ell^2 \cdot d_m} \quad [\text{N/mm}^2]$$

Q_F = design yield moment of rudder stock according to F. in [Nm]

d_m = mean cone diameter in [mm]

ℓ = cone length in [mm]

μ_0 = 0,15 (frictional coefficient)

M_b = bending moment in the cone coupling (e.g. case of spade rudders) in [Nm].

The greater of the values p_{req1} or p_{req2} is to be taken.

It has to be proved that the required push-up pressure does not exceed the permissible surface pressure in the cone. The permissible surface pressure is to be determined by the following formula:

$$p_{perm} = \frac{0{,}81 \cdot R_{eH}(1 - \alpha^2)}{\sqrt{3 + \alpha^4}} \quad [\text{N/mm}^2]$$

R_{eH} = yield point in [N/mm^2] of the material of the gudgeon

α = d_m/d_a (see Fig 14.6)

The outer diameter of the gudgeon should not be less than:

d_a = $1{,}5 \cdot d_m$ [mm]

.2 required push-up length

$$\Delta\ell = \frac{p_{req} \cdot d_m}{E\left[\frac{1 - \alpha^2}{2}\right]c} + 0{,}8\,\frac{R_{tm}}{c} \quad [\text{mm}]$$

R_{tm} = mean roughness in [mm]

R_{tm} ≈ 0,01 mm

c = taper on diameter according to 4.2.1

E = Young's modulus ($2{,}06 \cdot 10^5$ N/mm^2)

A guidance figure for the minimum push-up length is:

$\Delta\ell_{min}$ = $d_m/150$ [mm].

This value is not to be greater than:

$$\Delta\ell = \frac{1{,}62 \cdot R_{eH} \cdot d_m}{\sqrt{3 + \alpha^4}\, E \cdot c} + 0{,}8\,\frac{R_{tm}}{c} \quad [\text{mm}]$$

Guidance

In case of hydraulic pressure connections the required push-up force P_e for the cone may be determined by the following formula:

$$P_e = P_{req} \cdot d_m \cdot \pi \cdot \ell \,(c/2 + 0{,}02) \qquad [N]$$

Where due to the fitting procedure a partial push-up effect caused by the rudder weight is given, this may be taken into account when fixing the required push-up length, subject to approval by BKI.

4.2.4 The required push-up pressure for pintle bearings is to be determined by the following formula:

$$p_{req} = 0{,}4\,\frac{B_1 \cdot d_0}{d_m^{\,2} \cdot \ell} \qquad [\text{N/mm}^2]$$

B_1 = supporting force in the pintle bearing in [N], see also Fig. 14.4

d_m, ℓ see 4.2.3

d_0 = pintle diameter in [mm] according to Fig. 14.6.

E. Rudder Body, Rudder Bearings

1. Strength of rudder body

1.1 The rudder body is to be stiffened by horizontal and vertical webs in such a manner that the rudder body will be effective as a beam. The rudder should be additionally stiffened at the aft edge.

1.2 The strength of the rudder body is to be proved by direct calculation according to C.3.

1.3 For rudder bodies without cut-outs the permissible stress are limited to:

bending stress due to M_R:

σ_b = 110 N/mm²

shear stress due to Q_1:

τ = 50 N/mm²

equivalent stress due to bending and shear:

$$\sigma_v = \sqrt{\sigma_b^{\,2} + 3\,\tau^2} = 120 \quad \text{N/mm}^2$$

M_R, Q_1 see C.3.3.

In case of openings in the rudder plating for access to cone coupling or pintle nut the permissible stresses according to 1.4 apply. Smaller permissible stress values may be required if the corner radii are less than $0.15 \cdot h$, where h = height of opening.

1.4 In rudder bodies with cut-outs (semi-spade rudders) the following stress values are not to be exceeded:

bending stress due to M_R :

σ_b = 90 N/mm²

shear stress due to Q_1 :

τ = 50 N/mm²

torsional stress due to M_t :

τ_t = 50 N/mm²

equivalent stress due to bending and shear and equivalent stress due to bending and torsion:

$$\sigma_{v1} = \sqrt{\sigma_b^{\,2} + 3\,\tau^2} = 120 \quad \text{N/mm}^2$$

$$\sigma_{v2} = \sqrt{\sigma_b^{\,2} + 3\,\tau^2} = 100 \quad \text{N/mm}^2$$

$$M_R = C_{R2} \cdot f_1 + B_1\,\frac{f_2}{2} \quad [\text{Nm}]$$

$$Q_1 = C_{R2} \quad [\text{N}]$$

f_1, f_2 see Fig. 14.8.

The torsional stress may be calculated in a simplified manner as follows:

$$\tau_t = \frac{M_t}{2 \cdot \ell \cdot h \cdot t} \quad [\text{N/mm}^2]$$

M_t = $C_{R2} \cdot e$ [Nm]

C_{R2} = partial rudder force in [N] of the partial rudder area A_2 below the cross section under consideration

e = lever for torsional moment in [m] (horizontal distance between the centroid of area A_2 and the centre line a-a of the effective cross sectional area under consideration, see Fig. 14.8. The centroid is to be assumed at $0{,}33 \cdot c_2$ aft of the forward edge of area A_2, where c = mean breadth of area A_2)

h, ℓ, t in [cm], see Fig. 14.8.

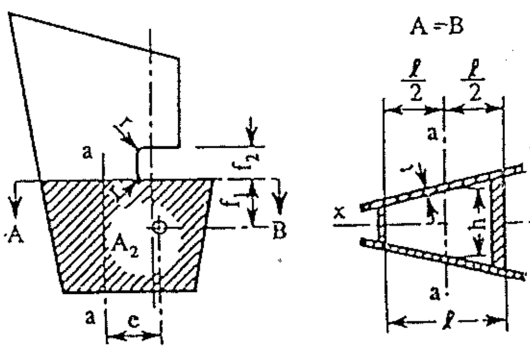


Fig. 14.8

The distance between the vertical webs should not exceed $1,2 \cdot h$.

The radii in the rudder plating are not to be less than 4 - 5 times the plate thickness, but in no case less than 50 mm.

1.5 It is recommended to keep the natural frequency of the fully immersed rudder at least 10% above the exciting frequency of the propeller (number of revolutions x number of blades).

2. Rudder plating

2.1 The thickness of the rudder plating is to be determined according to the following formula:

$$t = 1{,}74 \cdot a\sqrt{p_R \cdot k} + t_K \quad [\text{mm}]$$

$$P_R = 10 \cdot T + \frac{C_R}{10^3 \cdot A} \quad [\text{kN/m2}]$$

a = the smaller unsupported width of a plate panel in [m].

t_K = corrosion allowance, see Section 3, K.

The influence of the aspect ratio of the plate panels may be taken into account as given in Section 3, A.3.

The thickness shall, however, not be less than the thickness t_2 of the shell plating at the ends according to Section 6, B.3.

2.2 For connecting the side plating of the rudder to the webs tenon welding is not to be used. Where application of fillet welding is not practicable, the side plating is to be connected by means of slot welding to flat bars which are welded to the webs.

2.3 The thickness of the webs is not to be less than 70% of the thickness of the rudder plating according to 2.1, but not less than:

$$t_{min} = 8\sqrt{k} \quad [\text{mm}]$$

3. Transmitting of the rudder torque

3.1 For transmitting the rudder torque, the rudder plating according to 2.1 is to be increased by 25% in way of the coupling. A sufficient number of vertical webs is to be fitted in way of the coupling.

3.2 If the torque is transmitted by a prolonged shaft extended into the rudder, the latter must have the diameter D_t or D_1, whichever is greater, at the upper 10% of the intersection length. Downwards it may be tapered to 0,6 D_t, in spade rudders to 0,4 times the strengthened diameter, if sufficient support is provided for.

4. Rudder bearings

4.1 In way of bearings liners and bushes are to be fitted. Where in case of small ships bushes are not fitted, the rudder stock is to be suitably increased in diameter in way of bearings enabling the stock to be re-machined later.

4.2 An adequate lubrication is to be provided.

4.3 The bearing forces result from the direct calculation mentioned in C.3. As a first approximation the bearing force may be determined without taking account of the elastic supports. This can be done as follows:

.1 normal rudder with two supports:

The rudder force C_R is to be distributed to the supports according to their vertical distances from the centre of gravity of the rudder area.

.2 semi-spade rudders:

- support force in the rudder horn:

$$B_1 = C_R \cdot b/c \quad [\text{N}]$$

- support force in the neck bearing:

$$B_2 = C_R - B_1 \quad [\text{N}]$$

For b and c see Fig. 13.6 in Section 13.

4.4 The projected bearing surface A_b (bearing height x external diameter of liner) is not to be less than

$$A_b = \frac{B}{q} \quad [\text{mm}^2]$$

B = support force in [N]

q = permissible surface pressure according to Table 14.2

4.5 Stainless and wear resistant steels, bronze and hot-pressed bronze-graphite materials have a considerable difference in potential to non-alloyed steel. Respective preventive measures are required.

Table 14.2

Bearing material	q [N/mm²]
lignum vitae or synthetic material of little hardness [1]	2,5
white metal, oil lubricated	4,5
synthetic material with adequate hardness [1,2]	5,5
steel[3], bronze and hot-pressed bronze-graphite materials	7,0

1 Synthetic materials to be of approved type.
2 Indentation hardness test to be carried out at 23°C and with 50% moisture according to recognized standards, e.g. DIN 53456.
3 Stainless and wear resistant steel in an approved combination with stock liner.

Higher values may be taken for q if they are verified by test.

4.6 The bearing height shall be equal to the bearing diameter, however, is not to exceed 1,2 times the bearing diameter. Where the bearing depth is less than the bearing diameter, higher specific surface pressures may be allowed.

4.7 The wall thickness of pintle bearings in sole piece and rudder horn shall be approximately ¼ of the pintle diameter.

5. Pintles

5.1 Pintles are to have scantlings complying with the conditions given in 4.4 and 4.6. The pintle diameter is not to be less than:

$$d = 0{,}35\,\sqrt{B_1 \cdot k_r} \quad [\mathrm{mm}]$$

B_1 = support force in [N]

k_r see A.4.2.

5.2 The thickness of any liner or bush shall not be less than:

$$t = 0{,}01\,\sqrt{B_1} \quad [\mathrm{mm}]$$

t_{min} = 8 mm for metallic materials and synthetic material

t_{min} = 22 mm for lignum vitae

5.3 Where pintles are of conical shape, they are to comply with the following

taper on diameter 1: 8 to 1: 12
if keyed by slugging nut,

taper on diameter 1: 12 to 1: 20
if mounted with oil injection and hydraulic nut.

5.4 The pintles are to be arranged in such a manner as to prevent unintentional loosening and falling out.

For nuts and threads the requirements of D.4.1.5 and 4.2.2 apply accordingly.

6. Guidance values for bearing clearances

For metallic bearing material the bearing clearance should generally not be less than:

$$\frac{d_b}{1000} + 1{,}0 \quad [\mathrm{mm}] \text{ or } 1{,}5 \text{ mm as a minimum}$$

d_b = inner diameter of bush.

If non-metallic bearing material is applied, the bearing clearance is to be specially determined considering the material's swelling and thermal expansion properties.

F. Design Yield Moment of Rudder Stock

The design yield moment of the rudder stock is to be determined by the following formula:

$$Q_F = 0{,}02664\,\frac{D_t^3}{k_r} \quad [\mathrm{Nm}]$$

D_t = stock diameter in [mm] according to C.1.

Where the actual diameter D_{ta} is greater than the calculated diameter D_t, the diameter D_{ta} is to be used. However, D_{ta} need not be taken greater than 1,145 D_t.

G. Stopper, Locking Device

1. Stopper

The motions of quadrants or tillers are to be limited on either side by stoppers. The stoppers and their foundations connected to the ship's hull are to be of strong construction so that the yield point of the applied materials is not exceeded at the design yield moment of the rudder stock.

2. Locking device

Each steering gear is to be provided with a locking device in order to keep the rudder fixed at any position. This device as well as the foundation in the ship's hull are to be of strong construction so that the yield point of the applied materials is not exceeded at the design yield moment of the rudder stock as specified in F. Where the ship's speed exceeds 12 kn, the design yield moment need only be calculated for a stock diameter based on a speed v_0 = 12 kn.

3. Regarding stopper and locking device see also Volume III, Section 14.

H. Propeller Nozzles

1. General

1.1 The following requirements are applicable to propeller nozzles having an inner diameter of up to 5 m. Nozzles with larger diameters will be specially considered.

1.2 Special attention is to be given to the support of fixed nozzles at the hull structure.

2. Design pressure

The design pressure for propeller nozzles is to be determined by the following formula:

$p_d = c \cdot p_{d0}$ [kN/m²]

$p_{d0} = \varepsilon \dfrac{N}{A_p}$ [kN/m²]

N = maximum shaft power in [kW]

A_p = propeller disc area in [m]

$A_p = D^2 \dfrac{\pi}{4}$

D = propeller diameter in [m]

ε = factor according to the following formula:

$$\varepsilon = 0{,}21 - 2 \cdot 10^{-4} \frac{N}{A_p}$$

ε_{min} = 0,10

c = 1,0 in zone 2 (propeller zone),

c = 0,5 in zones 1 and 3

c = 0,35 in zone 4.

see Fig. 14.9

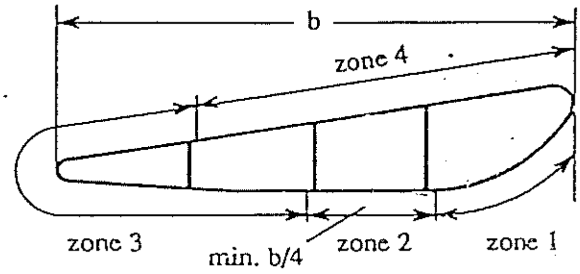


Fig. 14.9

3. Plate thickness

3.1 The thickness of the nozzle shell plating is not to be less than:

$t = 5 \cdot a \sqrt{p_d} + t_K$ [mm]

t_{min} = 7,5 mm

a = spacing of ring stiffeners in [m].

3.2 The web thickness of the internal stiffening rings shall not be less than the nozzle plating for zone 3, however, in no case be less than 7,5 mm.

4. Section modulus

The section modulus of the cross section shown in Fig. 14.9 around its neutral axis is not to be less than:

$W = n \cdot d^2 \cdot b \cdot v_0^2$ [cm³]

d = inner diameter of nozzle in [m]

b = length of nozzle in [m]

n = 1,0 for rudder nozzles

n = 0,7 for fixed nozzles.

5. Welding

The inner and outer nozzle shell plating is to be welded to the internal stiffening rings as far as practicable by double continuous welds. Plug welding is only permissible for the outer nozzle plating.

The tests are normally to be carried out from a tug, however, alternative shore based tests (e.g. with suitable winches) may be accepted.

Three tests are to be carried out for each anchor and type of bottom. The pull shall be measured by means of a dynamometer or recorded by a recording instrument. Measurements of pull based on rpm/bollard pull curve of the tug may be accepted. Testing by comparison with a previously approved HHP anchor may be accepted as a basis for approval.

The maximum of an anchor thus approved may be 10 times the mass of the large size anchor tested.

The dimensioning of the chain cable and of the windlass is to be based on the undiminished anchor mass according to the Tables.

6. Where stern anchor equipment is fitted, such equipment is to comply in all respects with the rules for anchor equipment. The mass of each stern anchor shall be at least 35 per cent of that of the bower anchors. The diameter of the chain cables is to be determined from the Tables in accordance with the anchor mass. Where a stern anchor windlass is fitted the requirements of Volume III, Section 14, are to be observed.

D. Chain Cables

1. The chain cable diameters given in the Tables apply to chain cables made of chain cable materials specified in the requirements of Volume V for the following grades:

Grade K 1	(ordinary quality)
Grade K 2	(special quality)
Grade K 3	(extra special quality)

2. Grade K 1 material used for chain cables in conjunction with "High Holding Power Anchors" must have a tensile strength R_m of not less than 400 N/mm^2.

3. Grade K 2 and K 3 chain cables must be purchased from the re-heat treated by recognized firms only.

4. The total length of chain given in the tables is to be divided in approximately equal parts between the two bower anchors.

5. Either stud link or short link chain cables may be used for stream anchors.

6. For connection of the anchor with the chain cable approved Kenter-type anchor shackles may be chosen in lieu of the common Dee-shackles. A forerunner with swivel is to be fitted between anchor and chain cable. In lieu of a forerunner with swivel an approved swivel shackle may be used. However, swivel shackles are not to be connected to the anchor shank unless specially approved. A sufficient number of suitable spare shackles are to be kept on board to facilitate fitting of the spare anchor at any time.

7. The attachment of the inboard ends of the chain cables to the ship's structure is to be provided with a mean suitable to permit, in case of emergency, an easy slipping of the chain cables to sea operable from an accessible position outside the chain locker.

The inboard ends of the chain cables are to be secured to the structures by a fastening able to withstand a force not less than 5% nor more than 30% of the rated breaking load of the chain cable.

E. Chain Locker

1. The chain locker is to be of capacity and depth adequate to provide an easy direct lead of the cables through the chain pipes and self-stowing of the cables. The chain locker is to be provided with an internal division so that the port and starboard chain cables may be fully and separately stowed.

2. The chain locker boundaries and their access openings are to be watertight as necessary to prevent accidental flooding of the chain locker from damaging essential auxiliaries or equipment or affecting the proper operation of the vessel.

3. Adequate drainage facilities of the chain locked are to be provided.

4. Where the chain locker boundaries are also tank boundaries their scantlings of stiffeners and plating are to be determined as for tanks in accordance with Section 12.

Where this is not the case the plate thickness is to be determined as for t_2 and the section modulus as for

W_2 in accordance with Section 12, B.2. and B.3. respectively. The distance from the load centre to the chain locker top is to be taken for calculating the load.

5. For the location of chain lockers on tankers Section 24, A.9 is to be observed.

F. Mooring Equipment

1. Ropes

1.1 The tow lines and mooring ropes specified in the tables and the contents of the following sub-paragraphs up to 1.6 are recommendations only, a compliance with which is not a condition of Class.

1.2 For tow lines and mooring lines, steel wire ropes as well as fibre ropes made of natural or synthetic fibres or wire ropes consisting of steel wire and fibre cores may be used. The breaking loads[1] specified in Table 18.2 are valid for wire ropes and ropes of natural fibre (Manila) only. Where ropes of synthetic fibre are used, the breaking load is to be increased above the table values. The extent of increase depends on the material quality.

The required diameters of synthetic fibre ropes used in lieu of steel wire ropes may be taken from Table 18.1.

1.3 Where the stream anchor is used in conjunction with a rope, this is to be a steel wire rope.

1.4 Wire ropes shall be of the following type:

- 144 wires (6 x 24) with 7 fibre cores for breaking loads of up to 500 kN
  type: Standard

- 216 wires (6 x 36) with 1 fibre core for breaking loads of more than 500 kN
  type: Standard.

Where wire ropes are stored on mooring winch drums, steel cored wire ropes may be used e.g.:

- 6 x 19 with 1 steel core
  type: Seale

- 6 x 36 with 1 steel core
  type: Warrington-Seale.

1.5 Regardless of the breaking load, recommended in Table 18.2, the diameter of fibre ropes should not be less than 20 mm.

1.6 The length of the individual mooring ropes may be up to 7 per cent less than that given in the table provided that the total length of all the wires and ropes is not less than the sum of the individual lengths.

Table 18.1

Steel wire ropes	Synthetic wire ropes Poly-amide [1])	Fibre ropes		
		Poly-amide	Poly-ester	Polypro-pylene
dia. [mm]	dia. [mm]	dia. [mm]	dia. [mm]	dia. [mm]
12	30	30	30	30
13	30	32	32	32
14	32	36	36	36
16	32	40	40	40
18	36	44	44	44
20	40	48	48	48
22	44	48	48	52
24	48	52	52	56
26	56	60	60	64
28	60	64	64	72
32	68	72	72	80
36	72	80	80	88
40	72	88	88	96

1) according to DIN 3068 or equivalent
2) Regular laid ropes of refined polyamide monofilaments and filament fibres.

Where mooring winches on large ships are located on one side of the ship, the lengths of mooring ropes should be increased accordingly.

For individual mooring lines with a breaking load above 500 kN the following alternatives may be applied:

.1 The breaking load of the individual mooring lines specified in Table 18.2 may be reduced with corresponding increase of the number of mooring lines, provided that the total breaking load of all lines aboard ship is not less than the rule value as per Table 18.2. No

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1) The term "Breaking Load" used throughout this Section means the "Nominal aggregate breaking load"

The tests are normally to be carried out from a tug, however, alternative shore based tests (e.g. with suitable winches) may be accepted.

Three tests are to be carried out for each anchor and type of bottom. The pull shall be measured by means of a dynamometer or recorded by a recording instrument. Measurements of pull based on rpm/bollard pull curve of the tug may be accepted. Testing by comparison with a previously approved HHP anchor may be accepted as a basis for approval.

The maximum of an anchor thus approved may be 10 times the mass of the large size anchor tested.

The dimensioning of the chain cable and of the windlass is to be based on the undiminished anchor mass according to the Tables.

6. Where stem anchor equipment is fitted, such equipment is to comply in all respects with the rules for anchor equipment. The mass of each stem anchor shall be at least 35 per cent of that of the bower anchors. The diameter of the chain cables is to be determined from the Tables in accordance with the anchor mass. Where a stem anchor windlass is fitted the requirements of Volume III, Section 14, are to be observed.

D. Chain Cables

1. The chain cable diameters given in the Tables apply to chain cables made of chain cable materials specified in the requirements of Volume V for the following grades:

Grade K 1	(ordinary quality)
Grade K 2	(special quality)
Grade K 3	(extra special quality)

2. Grade K 1 material used for chain cables in conjunction with "High Holding Power Anchors" must have a tensile strength R_m of not less than 400 N/mm^2.

3. Grade K 2 and K 3 chain cables must be purchased from the re-heat treated by recognized firms only.

4. The total length of chain given in the tables is to be divided in approximately equal parts between the two bower anchors.

5. Either stud link or short link chain cables may be used for stream anchors.

6. For connection of the anchor with the chain cable approved Kenter-type anchor shackles may be chosen in lieu of the common Dee-shackles. A forerunner with swivel is to be fitted between anchor and chain cable. In lieu of a forerunner with swivel an approved swivel shackle may be used. However, swivel shackles are not to be connected to the anchor shank unless specially approved. A sufficient number of suitable spare shackles are to be kept on board to facilitate fitting of the spare anchor at any time.

7. The attachment of the inboard ends of the chain cables to the ship's structure is to be provided with a mean suitable to permit, in case of emergency, an easy slipping of the chain cables to sea operable from an accessible position outside the chain locker.

The inboard ends of the chain cables are to be secured to the structures by a fastening able to withstand a force not less than 5% nor more than 30% of the rated breaking load of the chain cable.

E. Chain Locker

1. The chain locker is to be of capacity and depth adequate to provide an easy direct lead of the cables through the chain pipes and self-stowing of the cables. The chain locker is to be provided with an internal division so that the port and starboard chain cables may be fully and separately stowed.

2. The chain locker boundaries and their access openings are to be watertight as necessary to prevent accidental flooding of the chain locker from damaging essential auxiliaries or equipment or affecting the proper operation of the vessel.

3. Adequate drainage facilities of the chain locked are to be provided.

4. Where the chain locker boundaries are also tank boundaries their scantlings of stiffeners and plating are to be determined as for tanks in accordance with Section 12.

Where this is not the case the plate thickness is to be determined as for t_2 and the section modulus as for

Table 18.2 Anchor, Chain Cables and Ropes

No. for Reg.	Equipment numeral Z	Stockless anchor			Stud link chain cables				Stream wire or chain for stream anchor		Recommended ropes				
		Bower anchor		Stream anchor	Bower anchors						Towline		Mooring ropes		
		Number[1]	Mass per anchor		Total length	Diameter d_1	Diameter d_2	Diameter d_3	Length	Br. load[2]	Length	Br. load[2]	Number	Length	Br. load[2]
			[kg]		[m]	[mm]	[mm]	[mm]	[m]	[kN]	[m]	[kN]		[m]	[kN]
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
101	up to 50	2	120	40	165	12.5	12.5	12.5	80	65	180	100	3	80	35
102	50 - 70	2	180	60	220	14	12.5	12.5	80	65	180	100	3	80	35
103	70 - 90	2	240	80	220	16	14	14	85	75	180	100	3	100	40
104	90 - 110	2	300	100	247.5	17.5	16	16	85	80	180	100	3	110	40
105	110 - 130	2	360	120	247.5	19	17.5	17.5	90	90	180	100	3	110	45
106	130 - 150	2	420	140	275	20.5	17.5	17.5	90	100	180	100	3	120	50
107	150 - 175	2	480	165	275	22	19	19	90	110	180	100	3	120	55
108	175 - 205	2	570	190	302.5	24	20.5	20.5	90	120	180	110	3	120	60
109	205 - 240	3	660		302.5	26	22	20.5			180	130	4	120	65
110	240 - 280	3	780		330	28	24	22			180	150	4	120	70
111	280 - 320	3	900		357.5	30	26	24			180	175	4	140	80
112	320 - 360	3	1020		357.5	32	28	24			180	200	4	140	85
113	360 - 400	3	1140		385	34	30	26			180	225	4	140	95
114	400 - 450	3	1290		385	36	32	28			180	250	4	140	100
115	450 - 500	3	1440		412.5	38	34	30			180	275	4	140	110
116	500 - 550	3	1590		412.5	40	34	30			190	305	4	160	120
117	550 - 600	3	1740		440	42	36	32			190	340	4	160	130
118	600 - 660	3	1920		440	44	38	34			190	370	4	160	145
119	660 - 720	3	2100		440	46	40	36			190	405	4	160	160
120	720 - 780	3	2280		467.5	48	42	36			190	440	4	170	170
121	780 - 840	3	2460		467.5	50	44	38			190	480	4	170	185
122	840 - 910	3	2640		467.5	52	46	40			190	520	4	170	200
123	910 - 980	3	2850		495	54	48	42			190	560	4	170	215
124	980 - 1060	3	3060		495	56	50	44			200	600	4	180	230
125	1060 - 1140	3	3300		495	58	50	46			200	645	4	180	250
126	1140 - 1220	3	3540		522.5	60	52	46			200	690	4	180	270
127	1220 - 1300	3	3780		522.5	62	54	48			200	740	4	180	285
128	1300 - 1390	3	4050		522.5	64	56	50			200	785	4	180	305
129	1390 - 1480	3	4320		550	66	58	50			200	835	4	180	325
130	1480 - 1570	3	4590		550	68	60	52			220	890	5	190	325
131	1570 - 1670	3	4890		550	70	62	54			220	940	5	190	335
132	1670 - 1790	3	5250		577.5	73	64	56			220	1025	5	190	350
133	1790 - 1930	3	5610		577.5	76	66	58			220	1110	5	190	375
134	1930 - 2080	3	6000		577.5	78	68	60			220	1170	5	190	400
135	2080 - 2230	3	6450		605	81	70	62			240	1260	5	200	425
136	2230 - 2380	3	6900		605	84	73	64			240	1355	5	200	450
137	2380 - 2530	3	7350		605	87	76	66			240	1455	5	200	480
138	2530 - 2700	3	7800		632.5	90	78	68			260	1470	6	200	480
139	2700 - 2870	3	8300		632.5	92	81	70			260	1470	6	200	490
140	2870 - 3040	3	8700		632.5	95	84	73			260	1470	6	200	500
141	3040 - 3210	3	9300		660	97	[illegible]	76			280	1470	6	200	520
142	3210 - 3400	3	9900		660	100	87	78			280	1470	6	200	555
143	3400 - 3600	3	10500		660	102	90	78			280	1470	6	200	590
144	3600 - 3800	3	11100		687.5	105	92	81			300	1470	6	200	620
145	3800 - 4000	3	11700		687.5	107	95	84			300	1470	6	200	650
146	4000 - 4200	3	12300		687.5	111	97	87			300	1470	7	200	650
147	4200 - 4400	3	12900		715	114	100	87			300	1470	7	200	660
148	4400 - 4600	3	13500		715	117	102	90			300	1470	7	200	670
149	4600 - 4800	3	14100		715	120	105	92			300	1470	7	200	680
150	4800 - 5000	3	14700		742.5	122	107	95			300	1470	7	200	685
151	5000 - 5200	3	15400		742.5	124	111	97			300	1470	8	200	695
152	5200 - 5500	3	16100		742.5	127	111	97			300	1470	8	200	695
153	5500 - 5800	3	16900		742.5	130	114	100			300	1470	8	200	705
154	5800 - 6100	3	17800		742.5	132	117	102			300	1470	9	200	705
155	6100 - 6500	3	18800		742.5		120	107			300	1470	9	200	715
156	6500 - 6900	3	20000		770		124	111			300	1470	9	200	725
157	6900 - 7400	3	21500		770		127	114			300	1470	10	200	725
158	7400 - 7900	3	23000		770		132	117			300	1470	11	200	72[illegible]
159	7900 - 8400	3	24500		770		137	122			300	1470	11	200	73[illegible]
160	8400 - 8900	3	26000		770		142	127			300	1470	12	200	73[illegible]
161	8900 - 9400	3	27500		770		147	132			300	1470	13	200	735
162	9400 - 10000	3	29000		770		152	132			300	1470	14	200	735
163	10000 - 10700	3	31000		770			137			300	1470	15	200	735
164	10700 - 11500	3	33000		770			142			300	1470	16	200	735
165	11500 - 12400	3	35500		770			147			300	1470	17	200	735
166	12400 - 13400	3	38500		770			152			300	1470	18	200	735
167	13400 - 14600	3	42000		770			157			300	1470	19	200	735
168	14600 - 16000	3	46000		770			162			300	1470	21	200	735

d_1 = Chain diameter Grade K 1 (Ordinary quality)
d_2 = Chain diameter Grade K 2 (Special quality)
d_3 = Chain diameter Grade K 3 (Extra special quality)
See also D

1 see C.1.
2 see F.1.2

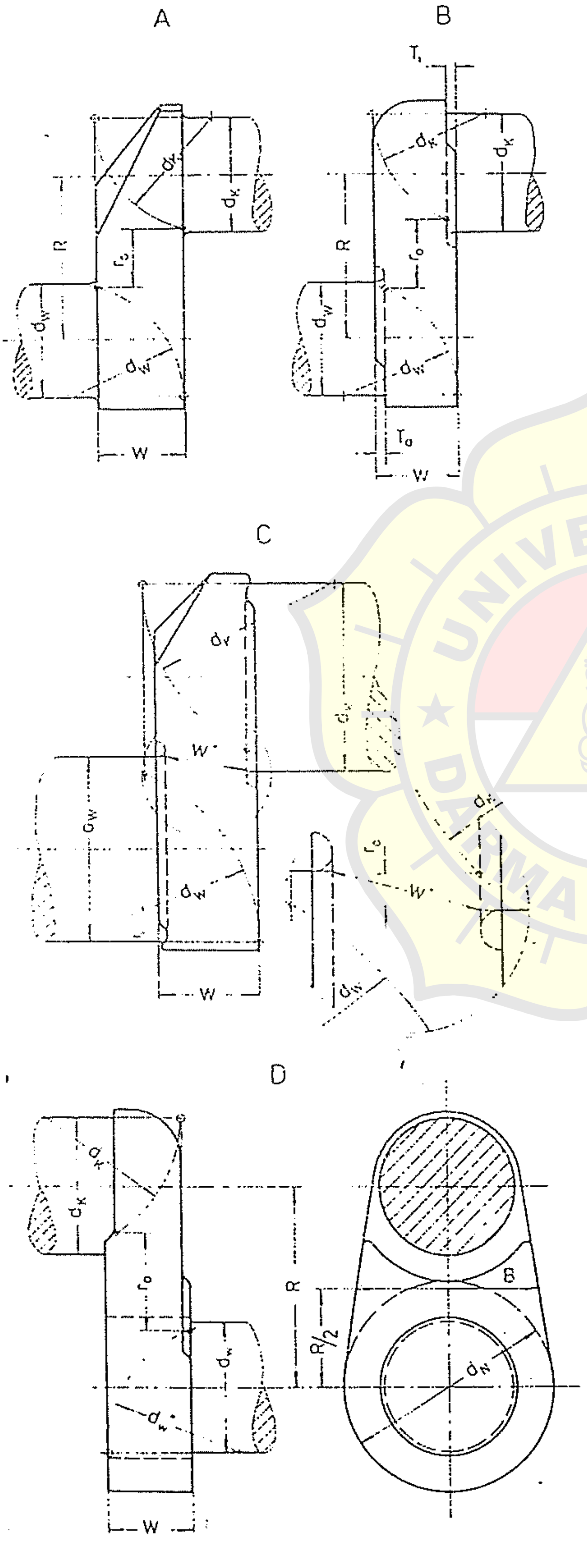


Fig. 2.11

$$r_o = 0,5\,(H + d_k + d_w) - W \left(\sqrt{\frac{2d_k}{W} - 1} + \sqrt{\frac{2d_w}{W} - 1} \right) \qquad (10)$$

In case of web undercut, W in formula (10) is to be replaced by:

$$W^* = 0,5\,(2 \cdot W - T_i - T_a) \qquad (11)$$

In the case of semi-built crankshafts in accordance with part D, the value d_W under the root sign only in formula (10) is to be replaced by:

$$d_w^* = 1/3\,(d_N - d_w) + d_w \qquad (12)$$

In case of web undercut, W* is also to be substituted for W in accordance with formula (11)

Where there is a positive pin/journal overlap ($s \geq 0$) according to part C, the value W in formula (10) is to be replaced by:

$$W^* = \sqrt{(W - T_i - T_a)^2 + \left[0,5\,(d_k + d_w - H)\right]^2} \qquad (13)$$

For the conventional designs, where

B/d_w = 1,37 to 1,51 in the case of solid-forged crankshafts, and

B/d_w = 1,51 to 1,63 in the case of semi-built crankshafts,

the influence of B in the normal calculation of r_O is already taken into account in the values of Δ_a in Fig. 2.9.

Where the values of B/d_W depart from the above (e.g. in the case of discs, oval webs etc.), the altered stiffening effect of B is to be allowed for by a fictitious web thickness W**, which is to be calculated by applying the following equations and is to be substituted for W in formula (10):

$$W^{**} = W^* \cdot \sqrt[3]{\frac{B}{d_w} - 0,44} \qquad \text{For solid-forged crankshafts} \qquad (14)$$

$$W^{**} = W^* \cdot \sqrt[3]{\frac{B}{d_w} - 0,57} \qquad \text{for semi-built crankshafts} \qquad (15)$$

Part C:

Approximate Calculation of the Starting Air Supply

1. Assuming an initial pressure of 30 bar and a final pressure of 9 bar in the starting air receivers, the preliminary calculation of the starting air supply for a reversible main engine may be performed follows:

$$J = a \cdot \sqrt[3]{\frac{D}{H}} \cdot (z + b \cdot p_{e,e} \cdot n_A + 0,9) \cdot V_h \cdot c \cdot d \qquad (16)$$

where

J	[dm³]	total capacity of the starting air receivers
D	[mm]	cylinder bore

H	[mm]	stroke
V_h	[dm³]	swept volume of one cylinder (in the case of double-acting engines, the swept volume of the upper portion of the cylinder)
$P_{e,mp}$	[bar]	maximum permissible working pressure of the starting air receiver
z	[-]	number of cylinders
$P_{e,e}$		mean effective working pressure in cylinder at rated power

The following values of "a" are to be used:

For two-stroke engines: a = 0,696
For four-stroke engines: a = 0,618

The following values of "b" are to be used:

For two-stroke engines: b = 0,059
For four-stroke engines: b = 0,056

The following values of "c" are to be used:

c = 1,0 For single-shaft propulsion plants where one engine acts on the shaft directly or via gears.

c = 2,0 For single-shaft propulsion plants where two identical engines act on the shaft via a gear transmission and cannot be coupled and uncoupled in service.

c = 1,5 For single-shaft propulsion plants where two identical engines act on the shaft via a gear transmission and couplings which can be engaged and disengaged in service.

c = 1,5 For two-shaft propulsion plants where each engine acts on the corresponding shaft directly or via gears.

c = 3,0 For two-shaft propulsion plants where two identical engines in each case act on the corresponding shaft via a gear transmission and cannot be coupled and uncoupled in service.

c = 2,0 For two-shaft propulsion plants where two identical engines in each case act on the corresponding shaft via a gear transmission and couplings which can be engaged and disengaged in service.

c = 3,0 For four-shaft propulsion plants where each engine acts on the corresponding shaft directly or via gears.

Where the arrangement of the main propulsion plant differs from the above, the value of "c" is to be agreed with the Society in each individual case.

For installations with electrical propeller drive, "c" is to be given the value specified in 2.2.

The following values of "d" are to be used:

d = 1, where $P_{e,mp}$ = 30 bar

$$d = \frac{0,0584}{1 - e^{(0,11 - 0,55 \cdot \ln P_{e,mp})}}$$

where $P_{e,mp} \neq$ 30 bar, if no pressure-reducing valve is fitted.

e [-] Euler's number (2,718....)

If a pressure-reducing valve is fitted, which reduces the pressure $P_{e,mp}$ to the starting pressure P_A, then the value of "d" shown in Fig. 2.12 is to be used.

The following values of n_A are to be applied:

$n_A = 0,06 \cdot n_o + 14$ where $n_o \leq 1000$
$n_A = 0,25 \cdot n_o - 176$ where $n_o > 1000$
n_o [min⁻¹] = rated speed

2. **Starting air supply for plants with non-reversing engines**

2.1 For each non-reversing main engine which drives a controllable pitch propeller or where starting is possible without resisting torque, the calculated supply of starting air may be reduced to 0,3 J, although it may not be less than that required for six starts.

2.2 Where diesel-electric propeller drive is installed, "c" in formula (16) is to be given the following values according to the number of generators n:

Table 2.14

n	1	2	3	4	5	6	7	8
c	0,30	0,60	0,84	1,08	1,26	1,38	1,44	1,50

This assumes prime movers having the same dimensions and the same number of cylinders. Where the dimensions and numbers of cylinders differ, the values of "c" are to be interpolated accordingly.

3. **Starting air supply for auxiliary engines on turbine ships**

The supply of starting air is to be calculated according to formula (16). The value of "c" to be used depends on the number of auxiliary engines:

c = 0,30 for 1 auxiliary engine
c = 0,45 for 2 auxiliary engines
c = 0,60 for 3 auxiliary engines
c = 0,75 for 4 auxiliary engines or over

For engines with different numbers of cylinders and/or main dimensions the values of "c" are to be interpolated accordingly.

Section 4

Main Shafting

A. General

1. Scope

The following Rules apply to standard and established types of main shafting. Novel designs require the Society's special approval.

In the case of ships with ice classes, the strengthening factors given in Section 13 are to be complied with. The Society reserves the right to call for propeller shaft dimensions in excess of those specified in this Section if the propeller arrangement results in increased bending stresses.

2. Documents for approval

General drawings of the entire shafting, from the main engine coupling flange to the propeller, and detail drawings of the shafts, couplings and other component parts transmitting the propelling engine torque, are each to be submitted to the Society in triplicate[1] for approval. The drawings must contain all the data necessary to enable the stresses to be evaluated.

B. Materials

1. Approved materials

Propeller, intermediate and thrust shafts together with flange and clamp couplings are to be made of forged steel; where appropriate, couplings may be made of cast steel. Rolled round steel may be used for plain, flangeless shafts.

In general, the tensile strength of steels used for shafting shall be between 400 N/mm² and 800 N/mm². However, for propeller shafts the value or R_m used for calculating the material factor C_W in accordance with formula (2) shall not be greater than 600 N/mm².

Where in special cases wrought copper alloys resistant to seawater are to be used for the shafting, the consent of BKI shall be obtained.

2. Testing of materials

All component parts of the shafting which assist in transmitting the torque from the ship's propulsion plant are subject to the Society's Rules for Materials and must be tested. This requirement also covers metal propeller shaft liners. Where propeller shafts running in seawater are protected against seawater penetration not by a metal liner but by plastic coatings, the coating technique used must be approved by the Society.

1) For Indonesian flag ships in quadruplicate, one for Government.

C. Shaft Dimensions

1. General

All parts of the shafting are to be dimensioned in accordance with the following formulae in compliance with the requirements relating to critical speeds set out in Section 16. The dimensions of the shafting shall be based on the total installed power. Where the geometry of a part is such that it cannot be dimensioned in accordance with these formulae, special evidence of the mechanical strength of the part or parts concerned is to be furnished to the Society.

2. Minimum diameter

The minimum shaft diameter is to be determined by applying formula (1).

$$d \geq F \cdot k \cdot \sqrt[3]{\frac{P_w}{n \cdot \left[1 - \left(\frac{d_i}{d_a}\right)^4\right]}} \cdot C_w \cdot C_{EW} \leq d_a \qquad (1)$$

d [mm] required outside diameter of shaft

d_i [mm] diameter of shaft bore, where present. If the bore in the shaft is $\leq 0{,}4 \cdot d$, the expression

$$1 - \left(\frac{d_i}{d_a}\right)^4 = 1{,}0 \text{ may be applied}$$

d_a [mm] actual shaft diameter

P_w [kW] rated power transmitted by shaft

n [min⁻¹] rated shaft speed

F [-] factor for the type of propulsion installation

a) Intermediate and thrust shafts

= 95
for turbine installations, engine installations with slip couplings and electric propulsion installations

= 100
for all other propulsion installations

b) Propeller shafts

= 100
for all types of installations

C_w [-] material factor

$$= \frac{560}{R_m + 160} \qquad (2)$$

R_m [N/mm²] tensile strength of the shaft material

C_{EW} [-] ice class strengthening factor in accordance with Section 13
= 1,0
for machinery with no ice class

k [-] factor for the type of shaft

k = 1,0 for intermediate shafts
with integral forged coupling flanges
or
with shrink-fitted keyless coupling flanges

k = 1,10 for intermediate shafts
where the coupling flanges are mounted on the ends of the shaft with the aid of keys. At a distance of at least 0,2 · d from the end of the keyway, such shafts can be reduced to a diameter corresponding to k = 1,0.

k = 1,10 for intermediate shafts
with radial holes whose diameter is not greater than 0,3 · d.

k = 1,10 for thrust shafts
near the plain bearings on either side of the thrust collar, or near the axial bearings where an antifriction bearing design is used.

k = 1,15 for intermediate shafts
designed as multi-splined shafts where d is the outside diameter of the splined shaft. Outside the splined section, the shafts can be reduced to a diameter corresponding to k = 1,0.

k = 1,20 for intermediate shafts
with longitudinal slots where the length and witdh of the slot do not exceed 1,17 · d and 0,25 · d respectively.

k = 1,22 for propeller shafts
from the area of the aft stern tube or shaft bracket bearing to the forward load-bearing face of the propeller boss subject to a minimum of 2,5 · d, if the propeller is shrink-fitted, without key, on to the tapered end of the propeller shaft using a method approved by the Society, or if the propeller is bolted to a flange forged on the propeller shaft, the propeller shaft runs in oil.

k = 1,26 for propeller shafts
in the area specified for k = 1,22, if the propeller is keyed to the tapered propeller shaft and the propeller shaft runs in oil, and also for water-lubricated propeller shafts which are protected against the penetration of seawater in accordance with D.3.2.

k = 1,40 for propeller shafts
in the area specified for k = 1,22, if the shaft inside the stern tube is lubricated with grease.

k = 1,15 for propeller shafts
forward portion of shafts to where they emerge from the stern tube. The portion of the propeller shaft located forward of the stern tube can be reduced to the size of the line shaft.

D. Design

1. General

Changes in diameter are to be effected by tapering or ample radiusing. For intermediate shafts, the radius at forged flanges is to be at least 0,08 · d [mm], that at the aft propeller shaft flange at least 0,125 · d [mm].

2. Shaft tapers and propeller nut threads

Keyways in the shaft taper for the propeller should be so designed that the forward end of the groove makes a gradual transition to the full shaft section. In addition, the forward end of the keyway should be spoon-shaped. The edges of the keyway at the surface of the shaft taper for the propeller may not be sharp. The forward end of the keyway must lie well within the seating of the propeller boss. Threaded holes to accommodate the securing screws for propeller keys should be located only in the aft half of the keyway (see Fig. 4.1).

In general, tapers for securing flange couplings should have a conicity of between 1 : 10 and 1 : 20. In the case of shaft tapers for propellers, the conicity must be between 1 : 10 and 1 : 15. Where the oil injection method is used to mount the propeller on its shaft, a degree of conicity between 1 : 15 and 1 : 20 is to be preferred.

The outside diameter of the threaded end for the propeller retaining nut should not be less than 60% of the calculated major taper diameter.

3. Propeller shaft protection

3.1 Sealing

Propeller shafts running in oil or grease are to be fitted with seals of proven efficiency and approved by BKI at the stern tube ends. The propeller boss seating is to be effectively protected against the ingress of seawater. The seal at the propeller can be dispensed with if the propeller shaft is made of corrosion-resistant material.

3.2 Shaft liners

3.2.1 Propeller shafts which are not made of corrosion-resistant material and which run in seawater are to be protected against contact with seawater by seawater-resistant metal liners or other liners approved by the Society and by seals of proven efficiency at the propeller.

3.2.2 The metal liners of propeller shafts running in sea-

Section 6

Propellers

A. General

1. Scope

These Rules apply to screw-propellers. Where a design is proposed to which the following Rules cannot be applied, special strength calculations are to be submitted to the Society. The tests to be carried out in each case are to be agreed with the Society. For the dimensions and materials of propellers for ships with ice classes, see Section 13

2. Documents for approval

Design drawings of propellers are to be submitted to the Society in triplicate[1] for examination. Drawings are to contain all the details necessary to verify compliance with the following Rules.

B. Materials

1. Approved materials

Propellers are to be made of seawater-resistant cast copper alloys or cast steel alloys with a minimum tensile strength of 440 N/mm², cf. Volume V Rules for Materials, Section 9, Part C, and Section 6, Part C and G. For the purpose of the following design Rules governing the thickness of the propeller blades, the requisite resistance to seawater of a cast copper alloy or cast steel alloy is considered to be achieved if the alloy used can be proved to withstand a fatigue test[2] under alternating bending stresses comprising 10^8 load cycles amounting to about 20% of the minimum tensile strength and carried out in a 3% NaCl solution, and if it can be proved that the fatigue strength under alternating bending stresses in natural seawater is not less than about 65% of the values established in 3% NaCl solution.

Grey cast iron with a minimum tensile strength of 250 N/mm² (GG 25 to DIN 1691) should be used only for spare propellers for short periods of service, and unalloyed or low alloy cast steels are primarily to be used only for navigating in icy waters, cf. Volume V.

2. Materials for blade retaining-bolts and studs

Unless protected against contact with seawater, the blade-retaining bolts or studs of assembled or controllable pitch propellers must likewise be made of materials which are resistant to seawater.

3. Novel materials.

Where it is proposed to use propeller materials whose serviceability is not attested by a sufficient period of practical experience, the Society must be provided with special proof of the suitability of such materials.

4. Material testing

The material of propellers and blade-retaining bolts or studs is to be tested in accordance with the Society's Rules for Materials.

C. Dimensions and Design of Propellers

1. Symbols and terms

A	[mm²]	Effective area of a shrink fit
B	[mm]	Developed blade width of cylindrical sections at radii 0,25 R, 0,35 R and 0,6 R
c	[-]	Coefficient for shrunk joints = 1,0 for engine and turbine gear transmissions = 1,2 for direct drives
C_{gr}, C_c	[-]	Ice class strengthening factors in accordance with Section 13 = 1,0 for machinery with no ice class
C_G	[-]	Size factor in accordance with formula (2)
C_{Dyn}	[-]	Dynamic factor in accordance with formula (3)
C_w	[-]	Characteristic value for propeller material as shown in Table 6.1 (corresponds to the minimum tensile strength R_m of the propeller material where this has been shown to possess sufficient fatigue strength under alternating bending stresses in accordance with paragraph B.1.)
C	[-]	Conicity of shaft ends $= \dfrac{\text{difference in taper diameter}}{\text{length of taper}}$
d	[mm]	Bolt-hole circle diameter of blade or propeller-fastening bolts
d_k	[mm]	Root diameter of blade or propeller-fastening bolts
D	[mm]	Diameter of propeller = 2 · R

1) For Indonesian flag ships in quadruplicate, one for Government.

2) Sufficient fatigue strength under alternating bending stresses must be proved by a method recognized by the Society. See also Section F.2. of the Society's "Regulations for the Determination of Dynamic Stresses on Propellers", December, 1971.

Table 6.1 Characteristic values C_w for propeller materials

Material	Description[1]	C_w
Cu 1	Cast manganese brass	480
Cu 2	Cast manganese nickel brass	500
Cu 3	Cast nickel aluminium bronze	660
Cu 4	Cast manganese aluminium bronze	620
Fe 1	Unalloyed cast steel	380
Fe 2	Low-alloy cast steel	380
Fe 3	Martensitic cast chrome steel 13/1-6	600
Fe 4	Martensitic-autenitic cast steel 17/4	600
Fe 5	Ferritic-austenitic cast steel 24/8	600
Fe 6	Austenitic cast steel 18/8-11	500
Fe 7	Grey cast iron	200

1)For the chemical composition of the alloys, see the Society's Rules for Materials and Regulations for the Assessment and Repair of Defects on Propellers.

d_m	[mm]	Mean taper diameter
e	[mm]	Blade rake to aft $= R \cdot \tan \epsilon$
E_r	[-]	Thrust stimulating factor in accordance with formula (5)
f, f_1, f_2, f_3	[-]	Factors in formulae (11), (2), (4) and (3)
F_M	[N]	Bolt load
H	[mm]	Propeller blade face pitch at radii 0,25 R, 0,35 R and 0,6 R
H_m	[mm]	Mean effective propeller pitch on blade face for pitch varying with the radius

$$= \frac{\Sigma (R \cdot B \cdot H)}{\Sigma (R \cdot B)}$$

in which R, B and H are to be substituted by values corresponding to the pitch at the various radii.

J	[-]	Degree of advance
k	[-]	Coefficient for various profile shapes in accordance with Table 6.2
k'	[-]	Coefficient calculated by applying formula (6) where use is made of profile shapes other than those given in Table 6.2
K_T	[-]	Thrust coefficient
l_M	[mm]	2/3 of the leading-edge component of the blade width at 0,9 R, but at least 1/4 of the total blade width at 0,9 R for propellers with heavily raked blades.
L	[mm]	Pull-up length when mounting propeller on taper
$L_{mech.}$	[mm]	Pull-up length at $t = 35°C$
$L_{temp.}$	[mm]	Temperature-related portion of pull-up length at $t < 35°C$
n_2	[min^{-1}]	Propeller speed in rev/min
P_w	[kW]	Shaft power
p	[N/mm^2]	Specific pressure in shrunk joint between propeller and shaft
Q	[N]	Peripheral force at mean taper diameter
S	[-]	Margin of safety against propeller slipping on taper = 2,8
t	[mm]	Maximum blade thickness of developed cylindrical section at radii 0,25 R, 0,35 R and 0,6 R
T	[N]	Propeller thrust
T_M	[Nm]	Impact moment
V_S	[kn]	Speed of ship
w	[-]	Wake factor
W_x	[mm^3]	Actual face modulus of developed cylindrical section referred to face blade pitch profiles about blade pitch line
Z	[-]	Total number of bolts used to retain one blade or propeller
z	[-]	Number of blades
α	[-]	Pitch angle of profile at radii 0,25 R, 0,35 R and 0,6 R

$$\alpha_{0,25} = \arctan \frac{1,27 \cdot H}{D}$$

$$\alpha_{0,35} = \arctan \frac{0,91 \cdot H}{D}$$

$$\alpha_{0,60} = \arctan \frac{0,53 \cdot H}{D}$$

α_A	[-]	Tightening factor for retaining bolts. and studs = 1,2 - 1,6 depending on the method of tightening used.
β_x	[-]	Factor for the section modulus of developed cylindrical section about blade pitch line for blade profiles in accordance with Table 6.2
β_x'	[-]	Factor for the section modulus of developed cylindrical section about blade pitch line for blade profiles other than those in Table 6.2
ϵ	[-]	Angle included by face generatix and normal
Θ	[-]	Half-conicity of shaft ends

$$= \frac{C}{2}$$

$$\frac{\sigma_{max}}{\sigma_m} = f_2 \cdot E_T + 1 \quad \text{where} \tag{4}$$

$$E_T = \frac{\sigma_{K_T}}{\sigma_J} \cdot \frac{J}{K_T} \tag{5}$$

$$= 4{,}3 \cdot 10^{-9} \; \frac{V_S \cdot n_2 \cdot (1 - w) \cdot D^3}{T}$$

f_2 = 0,4 - 0,6
for single-screw ships, the lower value applying to stern shapes with a wide propeller tip clearance and no rudder heel, and the larger value to sterns with little clearance and with rudder heel. Intermediate values are to be selected accordingly.

= 0,2
for twin-screw ships

f_3 = 0,2
for propeller materials which satisfy the requirements of B.1.

2.2 The blade thicknesses of controllable pitch propellers are to be determined at radii 0,35 R and 0,6 R by applying formula (1).

For the controllable pitch propellers of tugs, trawlers and special-duty ships with similar operating conditions, the diameter/pitch ratio D/H_m for the maximum static bollard pull is to be used in formula (1).

For other ships, the diameter/pitch ratio D/H_m applicable to open-water navigation can be used in formula (1).

2.3 The blade thicknesses calculated by applying formula (1) are minima for the finish-machined propellers.

2.4 The fillet radii at the transition from the face and the back of the blades to the propeller boss should correspond, in the case of three and four bladed propellers, to about 3,5% of the propeller diameter. For propellers with a larger number of blades the maximum fillet radii allowed by the propeller design should be aimed at, and the radii shall not in any case be made smaller than $0{,}4 \cdot t_{0,25}$.

2.5 For blades of special shape, special mechanical strength calculations are to be submitted to the Society as evidence that the propeller blades are adequately dimensioned.

For profile shapes other than those given in Table 6.2, the following condition applies:

where

$$k^1 = K \cdot \sqrt{\frac{\beta_x}{\beta_x{}^1}} \quad \text{and} \quad \beta_x{}^1 = \frac{W_x}{t^2 \cdot B} \tag{6}$$

Table 6.2 Values of k for various profile shapes

Profile shape	Values of k		
	0,25 R	0,35 R	0,6 R
Segmental Profiles with circular arced back, $\beta_x = 0{,}12$	73	62	44
Segmental profiles with parabolic back, $\beta_x = 0{,}11$	77	66	47
Blade profiles as for Wageningen B Series propellers where $\beta_{x\,0.25} = 0{,}10$, $\beta_{x\,0.35} = 0{,}11$, $\beta_{x\,0.6} = 0{,}12$	80	66	44

Notes :
The Society reserves the right to specify an increase in the values of k in the case of special propellers where the blade width B at 0,25 R is $< 4 \cdot t$.

D. Controllable Pitch Propellers

1. Documents for approval

In the case of controllable pitch propellers, besides the design drawings of blades and propeller boss, general and sectional drawings of the entire controllable pitch propeller installation are to be submitted to the Society in triplicate.[1]. Diagrams of control systems and pipework are to be accompanied by a functional description. For new designs and controllable pitch propellers which are to be installed for the first time on ships with a BKI class, a description of the controllable pitch propeller system is to be submitted at the same time.

2. Testing of materials

In addition to the material tests specified in B.4, the Society reserves the right to require component parts of the pitch-adjusting mechanism including, in particular, those which are not accessible for shipboard repairs, to be tested in accordance with the Rules for Materials. Piping subject to pressure above 10 bar is to be tested in accordance with Section 11.

3. Hydraulic control equipment

Where the pitch-control mechanism is operated hydraulically, two mutually independent, power-driven pump sets are to be fitted. For propulsion plants up to 200 kW, one power-driven pump set is sufficient pro-

1) For Indonesian flag ships in quadruplicate, one for Government.

vided that, in addition, a hand-operated pump is fitted for controlling the blade pitch and that this enables the blades to be moved from the ahead to the astern position in a short enough time.

4. Pitch control mechanism

For the pitch-control mechanism, proof is required that, when subjected to impact moments T_M as defined by formula (7), the individual components still have a safety factor of 1,5 with respect to the yield strength of the materials used.

$$T_M = \frac{0{,}65 \cdot 10^6 \, R_{p\,0,2} \cdot P_W \cdot I_M \cdot C_{EP}^2 \cdot C_G^2}{n_2 \cdot z \cdot C_W \cdot D} \qquad (7)$$

5. Blade retaining bolts and studs.

5.1 The root diameter of the bolts or studs used to attach blades is to be determined by applying formula (8):

$$d_k = 1{,}78 \cdot \sqrt{\frac{\alpha_A \cdot F_M}{R_{eH}}} \qquad (8)$$

$$F_M = \frac{0{,}28 \cdot 10^9 \cdot R_{p\,0,2} \cdot P_W \cdot C_{EP}^2 \cdot C_G^2}{n_2 \cdot z \cdot Z \cdot C_W \cdot d} \qquad (9)$$

5.2 The blade retaining bolts or studs are to be tightened in a controlled manner in such a way that the tension on the bolt or studs is about 60-70% of their yield strength.

The shank of blade retaining bolts or nuts may be designed with a minimum diameter equal to 0,9 times the root diameter of the thread. Blade retaining bolts must be secured against unintentional loosening.

6. Indicators

Controllable pitch propeller systems are to be provided with an engine room indicator showing the actual setting of the blades. Further blade position indicators are to be mounted on the bridge and in the engine room (see also Volume VII, and Volume IV, Section 8).

7. Failure of control system

Suitable devices are to be fitted to ensure that an alteration of the blade setting cannot overload the propulsion plant or cause it to stall.

Steps must be taken to ensure that, in the event of failure of the control system, the setting of the blades.

— does not change or
— assumes a final position slowly enough to allow the emergency control system to be put into operation.

8. Emergency control

Controllable pitch propeller systems must be equipped with means of emergency control enabling the controllable pitch propeller to remain in operation should the remote control system fail. It is recommended that a device be fitted which locks the propeller blades in the "ahead" setting.

E. Balancing and Testing

1. Balancing

The finished propeller and the blades of controllable pitch propellers are required to undergo static balancing.

2. Testing

The finished propeller is to be presented at the manufacturer's works to the BKI Surveyor for final inspection and verification of the dimensions. With regard to the assessment and the repair of defects on propellers, see the Society's Regulations for the Assessment and Repair of Defects on Propellers.

The Society reserves the right to require non-destructive tests to be conducted to detect surface cracks and casting defects.

In addition, controllabe pitch propeller systems are required to undergo pressure, tightness and operational tests.

F. Propeller Mounting

1. Tapered Mountings

1.1 Where the tapered joint between the shaft and the propeller is fitted with a key, the propeller is to be mounted on the tapered shaft in such a way that approximately the mean torque can be transmitted from the shaft to the propeller by the frictional bond. The "Reference data for the mounting length of keyed ships's propellers" issued by the Society can be used as a basis here. The propeller nut is to be secured in a suitable manner.

1.2 Where the tapered fit is effected by the hydraulic oil technique without the use of a key, the necessary pull-up distance on the tapered shaft is given by the expression.

$$L^{1)} = L_{mech.} + L_{temp.} \qquad (10)$$

where $L_{mech.}$ is determined according to the formulae of elasticity theory applied to shrunk joints for a specific pressure p [N/mm²] at the mean taper diameter found by applying formula (11) and for a water temperature of 35°C[2].

$$p = \frac{T}{A \cdot \Theta \cdot f}\left[\sqrt{1 + f\left(c^2 \cdot c_e^6 \cdot \frac{Q^2}{T^2} + 1\right)} - 1\right] \qquad (11)$$

1) Where appropriate, allowance is also to be made for surface smoothing when calculating L.

2) The von Mises's equivalent stress based on the maximum specific pressure p and the tangential stress in the bore of the propeller hub may not exceed 75% of the 0,2% proof stress or yield strength of the propeller material.

μ_o	[-]	Coefficient of static friction = 0,13 for hydraulic oil shrunk joints = 0,18 for dry shrunk joints
$R_{p\,0,2}$	[N/mm²]	0,2% proof stress of propeller material
R_{eH}	[N/mm²]	Yield strengths
R_m	[N/mm²]	Tensile strength of the material of fitted bolts or bolts
σ_{max}/σ_m	[-]	Ratio of maximum to mean stress at blade face

2. Calculation of blade thickness

2.1 At radii 0,25 R and 0,6 R, the blade thicknesses of solid propellers must, as a minimum requirement, comply with formula (1).

$$t = K_o \cdot k \cdot K_1 \cdot C_G \cdot C_{EP} \cdot C_{Dyn} \qquad (1)$$

$$K_o = 1 + \frac{e \cdot \cos\alpha}{H} + \frac{n_2}{15000}$$

k as in Table 6.2

$$K_1 = \sqrt{\frac{P_W \cdot 10^5 \cdot \left(2 \cdot \frac{D}{H_m} \cdot \cos\alpha + \sin\alpha\right)}{n_2 \cdot B \cdot z \cdot C_W \cdot \cos^2\varepsilon}}$$

C_G [-] Size factor

$$1,1 \geq \sqrt{\frac{f_1 + D}{12,2}} \geq 0,85 \qquad (2)$$

D to be inserted in [m]

f_1 = 7,2 for solid propellers

= 6,2 for separately cast blades of variable-pitch or built-up propellers

C_{EP} [-] Ice class strengthening factor in accordance with Section 13

= 1,0

for ships without ice class

C_{Dyn} [-] Dynamic factor

$$= \sqrt{\frac{(\sigma_{max}/\sigma_m - 1) + f_3}{0,5 + f_3}} \qquad (3)$$

for $\dfrac{\sigma_{max}}{\sigma_m} > 1,5$

σ_{max} / σ_m can be roughly calculated from the thrust-stimulating factor E_T according to formula (5). (For a more accurate calculation of σ_{max} / σ_m see the "Regulations for the Determination of Dynamic Stresses on Propellers, 1971".)

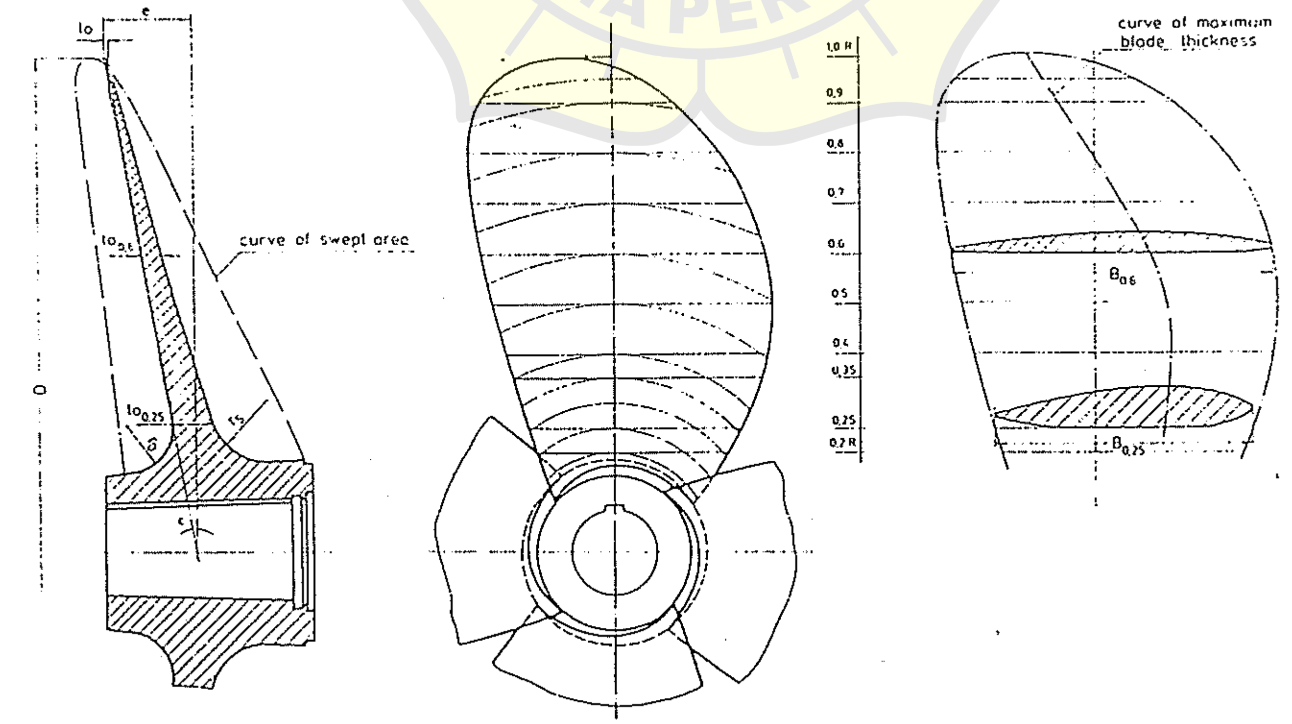


Fig. 6.1 Blade Sections

where $f = \left(\frac{\mu_0}{S \cdot \Theta}\right)^2 - 1 \geq 1{,}0$ (12)

$$L_{temp.} = \frac{d_m}{C} \cdot 6 \cdot 10^{-6} \cdot (35 - t) \qquad (13)$$

t [°C] The temperature at which the propeller is mounted.

$L_{temp.}$ applies only to propellers made of bronze and austenitic steel.

The tapers of propellers which are mounted on the propeller shaft with the aid of the hydraulic oil technique should not be more than 1 : 15 or less than 1 : 20.

The propeller nut must be strongly secured to the propeller shaft.

2. Flange connections

For propellers attached by flanges, the root diameter d_k of the retaining bolts or studs is to be determined by applying formula (8). In this formula, the bolt force F_M shall not be less than that defined by formula (14).

$$F_M = \frac{0{,}28 \cdot 10^9 \cdot R_{p\,0,2} \cdot P_W \cdot C_{EP}^2 \cdot C_G^2 \cdot S}{n_2 \cdot z \cdot Z \cdot d \cdot C_W} \qquad (14)$$

S [-] Safety factor

$$= 2 - 0{,}3\left(\frac{d_f'}{d_f}\right)^2 \geq 1{,}5$$

$\left(\frac{d_f'}{d_f}\right)$ [-] Ratio of the actual and required diameters of any set pins to formula (5) in Section 4.

connected in series, the intermediate pipe is to be fitted with a drain.

5.1.3 Each feedwater pump is to be fitted with a shutoff valve on the suction side and a screw-down non-return valve on the delivery side. The pipes are to be so arranged that each pump can supply each feedwater line.

5.2 Feedwater lines for auxiliary steam producers (auxiliary and exhaust gas boilers)

5.2.1 The provision of only one feedwater line for auxiliary- and exhaust gas boilers is sufficient if the preheaters and automatic regulating devices are fitted with by-pass lines.

5.2.2 The requirements in 5.1.2 are to apply as appropriate to the valves required to be fitted to the boiler inlet.

5.2.3 Continuous flow boilers need not be fitted with the valves required according to 5.1.2 provided that the heating of the boiler is automatically switched off should the feedwater supply fail and that the feedwater pump supplies only one boiler.

6. Boiler water circulating systems

6.1 Each forced-circulation boiler is to be equipped with two circulating pumps powered independently of each other. Failure of the circulating pump in operation is to be signalled by an alarm. The alarm may only be switched off is a circulating pump is started or when the boiler firing is shut down.

6.2 The provision of only one circulating pump for each boiler is sufficient if:

a) The boilers are heated only by gases whose tempe rature does not exceed 400°C or
b) a common stand-by circulating pump is provided which can be connected to any boiler or
c) the burners of oil or gas fired auxiliary boilers are so arranged that the are automatically shut off should the circulating pump fail and the heat stored in the boiler does not cause any unacceptable evaporation of the available water in the boiler.

7. Feedwater supply, evaporators

7.1 The feedwater supply is to be stored in several tanks.

7.2 One storage tank may be considered sufficient for auxiliary boiler units.

7.3 Two evaporators are to be provided for main steam producer units.

8. Condensate recirculation

8.1 The main condenser is to be equipped with two condensate pumps, each of which must be able to transfer the maximum volume of condensate produced.

8.2 The condensate of all heating systems used to heat oil (fuel, lubricating, cargo oil etc.) is to be led to condensate observation tanks. These tanks are to be fitted with air vents.

G. Oil Fuel Systems

1. Filling lines

The filling of oil fuels is to be effected by means of permanently installed lines either from the open deck or from filling stations located below deck which are to be isolated from other spaces.

Filling stations are to be so arranged that the filling can be performed from both sides of the ship without danger. This requirement is considered to be fulfilled where the filling line is extended to both sides of the ship. The filling lines are to be fitted with blind flanges on deck.

2. Tank filling lines

2.1 Filling lines of fuel tanks situated above the double bottom, which are led into the tanks below the tank top are to be fitted with remote controlled shutoff valves. It shall be possible to operate the remote controlled shutoff valves from a compartment accessible at all times and separate from the compartment where the fuel tanks are located.

2.2 Where filling lines are led through the tank top, a screw-down non-return valve at the tank top is sufficient.

Filling lines are to extend to the bottom of the tank.

Storage tank suction lines may also be used as filling lines.

3. Suction lines

3.1 Suction lines from tanks situated above the double bottom are to be fitted directly at the tank wall with remote controlled shutoff valves which can be closed from an adjacent compartment or deck above which is accessible at all times.

Service tanks up to 50 l capacity directly mounted on diesel engines need not be provided with remoted controlled shutoff valves.

3.2 The inlet connections of suction lines are to be arranged far enough from the drains in the tank so that water and impurities which have settled out will not enter the suctions.

4. Pipe layout

4.1 Fuel lines may not pass through tanks containing feedwater, drinking water, lubricating oil or thermal oil.

4.2 Fuel lines which pass through ballast tanks are to have an increased wall thickness according to Table 11.4.

4.3 Fuel lines may not be laid in the vicinity of boilers, turbines or equipment with high surface temperatures (over 220°C) or in way of electrical equipment.

The number of detachable pipe connections is to be limited.

Shutoff valves in fuel lines in the machinery spaces are to be operable from above the floor plates.

Glass and plastic components are not permitted in fuel systems.

5. Fuel transfer, feed and booster pumps

5.1 Fuel transfer, feed and booster pumps shall be designed for the proposed operating temperature of the medium pumped.

5.2 Arrangements for the transfer of oil fuel are to be arranged. The booster pumps may be used if they are suitable for this purpose.

5.3 At least two means of oil fuel transfer are to be provided for filling the day tanks.

Separators may serve as filling equipment.

5.4 Where a feed or booster pump is required to supply fuel to main or auxiliary engines, a standby pump shall be provided. Where, in the case of auxiliary engines, the pumps are attached to the engine, a standby pump may be dispensed with.

With the agreement of the Society, other arrangements may be approved.

5.5 For emergency shut-down devices, see Section 12, B.9.

6. Fuel spill lines of atmospheric stand-pipes are to be fitted with pressure loaded valves. The use of shutoff valves is not permitted. Where individual spill lines are connected to a common line each connection is to be fitted with a non-return valve.

7. Filters

7.1 The supply lines to the fuel injection pumps are to be fitted with duplex filters with a changeover cock or with automatic back-flushing filters fitted with a differential pressure control device.

The back-flushing intervals of the automatic back-flushing filters are to be controlled.

7.2 For auxiliary engines, simplex filters are sufficient.

7.3 Fuel transfer units are to be fitted with a simplex filter on the suction side.

8. Purifiers

8.1 Where a fuel purifier may exceptionally be used to purify lubricating oil the purifier supply and discharge lines are to be fitted with a change-over arrangement which prevents the possibility of fuel and lubricating oils being mixed. Unequal change-over joints or hoses of different length may be used for this purpose. Spectacle flanges are not considered as sufficient precaution.

Suitable equipment is also to be provided to prevent such mixing occuring over control and compression lines.

8.2 The sludge tanks of purifiers are to be fitted with a level alarm which ensures that the level in the sludge tank cannot interfere with the operation of the purifier.

9. Operation using heavy fuel oils

9.1 Heating of heavy fuel oil

9.1.1 The tank heating system shall be designed to meet operating requirements and to be compatible with the proposed grade to fuel oil.

For the tank heating system, see Section 10. B.5.

9.1.2 Heat tracing is to be arranged for pumps, filters and oil fuel lines as required.

9.1.3 Where heavy fuel oil is used, the injection valve cooling system is to be provided with additional means of heating so that the injection valves can be preheated.

9.2 Heavy fuel oil transfer

Settling tanks and day tanks for fuel are to be equipped with means of drainage.

9.3 Treatment of heavy fuel oil

9.3.1 Settling tanks

The capacity of the settling tanks shall be sufficient for at least one day's consumption of the plant. Settling tanks are to be provided with drains and with temperature measuring instruments in accordance with Section 10.B.5.3.

9.3.2 Heavy fuel oil cleaning for diesel engines

For cleaning of heavy fuels, purifiers or purifiers combined with automatic filters are to be provided.

9.3.3 Fuel oil blending and emulsifying equipments

Heavy fuel oil/diesel oil blending and emulsifying equipments require approval by the Society.

9.4 Heavy fuel oil day tanks

9.4.1 For the arrangement and equipment of the tanks, see Section 10.B.

9.4.2 The capacity of the day tanks shall be such that, should the treatment unit fail, operation can be maintained until treatment of the fuel can be resumed.

9.4.3 Where the overflow pipe of the day tank is terminated in a settling tank, suitable means shall be provided to ensure that no untreated heavy fuel oil can penetrate into the service tank.

9.5 Change-over arrangement diesel oil/heavy oil

9.5.1 The change-over arrangement of the fuel supply and return lines is to be interlocked so that faulty switching is excluded and to ensure reliable separation of the fuels.

Change-over valves devices are to be permitted

9.5.2 The change-over devices are to be accessible and

...ently marked. Their respective working position ... clearly indicated.

... Remote controlled change-over devices are to be ... with limit position indicators at the control ...

... Fuel supply through stand pipes

... Where the capacity of stand pipes exceeds 50 l, ... outlet pipe is to be fitted with a remote controlled ... closing valve operated from outside the engine ... Stand pipes are to be equipped with air/gas vents ... with self-closing connections for emptying and ... Stand pipes are to be fitted with a local ... temperature indicator.

... Atmospheric stand pipes (pressureless)

... regard to the arrangement and the maximum ... level in the service tanks, the stand pipes are to be ... located and arranged that a sufficient free space for ... circulation is available inside the stand pipes.

... Closed stand-pipes (pressurized systems)

... stand-pipes are to be designed as pressure vessels ... are to be fitted with the following equipment:

- ... non-return valve in the recirculating lines from the engines,
- ... automatic degasser,
- ... local pressure gauge,
- ... drain/emptying device, which is to be locked in the closed position.

... Fuel preheaters

... measures are to be taken to ensure that in the event of ... failure of the end preheater for the main propulsion ... operation can be temporarily maintained with fuel ... which does not require preheating (see Section 10, ...).

... Viscosity control

... Where main and auxiliary engines are operated ... heavy fuel oil, automatic viscosity control is to be ... provided.

... Viscosity regulators are to be fitted with a local ... temperature indicator.

... Local control devices

... following local control devices are to be fitted ... directly before the engine

- ... pressure gauge,
- ... temperature indicator.

... The heavy fuel system is to be effectively ... insulated as necessary.

... Lubricating Oil Systems

1. General requirements

1.1 Lubricating oil systems are to be constructed to ensure reliable lubrication over the whole range of speed and during run-down of the engines and to ensure adequate heat transfer.

1.2 Priming pumps

Where necessary, priming pumps are to be provided for supplying lubricating oil to the engines.

1.2 Emergency lubrication

A suitable emergency lubricating oil supply (e.g. gravity tank) is to be arranged to come automatically into use in the event of a failure of the supply from the pumps.

1.4 Lubricating oil treatment

The quipment necessary for adequate treatment of lubricating oil (purifiers, automatic back-flushing filters, filters and free-jet centrifuges) is to be provided.

2. Lubricating oil systems

2.1 Lubricating oil drain tanks and gravity tanks

2.1.1 For the capacity and location of these tanks see Section 10, C.

2.1.2 The suction connections of lubricating oil pumps are to be located as far as possible from drain pipes.

2.1.3 The gravity tank is to be fitted with an overflow pipe which leads to the drain tank. Arrangements are to be made for observing the flow of excess oil in the overflow pipe.

2.2 Pipe lines

Where lubricating oil lines must be led in the vicinity of hot machinery, e.g. superheated steam turbines, steel pipes which should be in one length and which are protected where necessary are to be used.

2.3 Filters

2.3.1 Lubricating oil filters are to be arranged in the pressures lines of the pumps.

2.3.2 The fineness and size of the filters are to be in accordance with the requirements of the engines.

2.3.3 The main filters are to be so arranged that they can be cleaned without stopping the engines. This requirement can be met by using change-over duplex filters or automatic back-flushing filters.

In the case of automatic back-flushing filters, the back-flushing intervals are to be controlled.

2.3.4 The fitting of a by-pass for change-over duplex filters is not permitted. A by-pass for automatic filters may be approved by the Society taking the requirements under 2.3.6 into consideration.

2.3.5 Where simplex filters are to be fitted after the main filters required according to 2.3.1 to 2.3.4, the simplex filters are to be provided with a by-pass and a pressure-difference alarm.

2.3.6 Where the fineness of the simplex filter (safety filter) is below that of the main filter, the by-pass may with approval of the Society be dispensed with provided that the safety filter may only take over the function of the main filter for a limited period of time.

2.3.7 Engines with rated output up to 150 kW which are supplied with lubricating oil from the engine oil sump may be fitted with simplex filters provided that a pressure alarm is arranged behind the filter and filter can be cleaned during operating. For this purpose, a by-pass with manually operated shutoff valves is to be provided.

2.3.8 For auxiliary engines simplex filters are sufficient.

2.4 Lubricating oil coolers

It is recommended that turbine and large engine plants be provided with more than one oil cooler.

2.5 Oil level indicators

Machines with their own oil charge are to be provided with a means of determining the oil level from outside during operation. This requirement also applies to reduction gears, thrust bearings and shaft bearings.

2.6 Purifiers

The requirements in G.8 apply as appropriate.

3. Lubricating oil pumps

3.1 Main engines

3.1.1 Main and independent stand-by pumps are to be arranged.

Main pumps driven by the main engines are to be so designed that the lubricating oil supply is ensured over the whole range of operation.

3.1.2 In multi-propeller plants and plants with more than one engine for the ranges of service "Unrestricted and Restricted International Service" which have independent lubricating oil systems, it may be deemed sufficient if the stand-by lubricating oil pump is capable of supplying one engine with oil and can be connected to each engine.

3.1.3 For multi-propeller plants and plants with more than one engine for "Coastal Service" and "Shallow Water Service", a complete spare lubricating oil pump may be carried on board instead of an independently driven stand-by pump provided that each main engine is fitted with an attached lubricating oil pump and the attached pumps are so arranged that they can be replaced with the means available on board.

3.2 Main turbine plant

3.2.1 Main and independent stand-by lubricating oil pumps are to be provided.

3.2.2 Emergency lubrication[1]

The lubricating oil supply to the main turbine plant for cooling the bearings during the run-down period is to be assured in the event of failure of the power supply. By means of suitable arrangements such as gravity tanks the supply of oil is also to be assured during starting of the emergency lubrication system.

3.3 Main reduction gearing (motor vessels)

3.3.1 Lubricating oil is to be supplied by a main pump and an independent stand-by pump.

3.3.2 Where a reduction gear has been approved by the Society to have adequate self-lubrication at 75% of the torque of the propelling engine, a stand-by lubricating oil pump for the reduction gear may be dispensed with up to a power ratio of

$$P/n \left[kW/min^{-1} \right] \leq 3,0$$

3.3.3 The regulations under 3.1.2 and 3.1.3 are to be applied for multi-propeller plants and plants with more than one engine.

3.4 Auxiliary machinery

3.4.1 Diesel generators

Where more than one diesel generator is available, stand-by pumps are not required.

Where only one diesel generator is available (e.g. on turbine-driven vessels where the diesel generator is needed for startup etc.) a complete spare pump is to be carried on board.

3.4.2 Auxiliary turbines

Turbogenerators and turbines used for driving important auxiliaries such as boiler feedwater pumps etc. are to be equipped with a main pump and an independent auxiliary pump. The auxiliary pump is to be designed to ensure a sufficient supply of lubricating oil during the startup and run-down operation.

J. Seawater Cooling Equipment

1. Sea connections, sea chests

1.1 At least two sea chests are to be provided. Wherever possible, the sea chests are to be arranged as low as possible on either side of the ship.

1.2 For service in shallow waters, it is recommended that an additional high seawater intake should be provided.

1.3 It is to be ensured that the total seawater supply for the engines can be taken from only one sea chest.

1.4 Each sea chest is to be provided with an effective vent. The following venting arrangements will be approved:

a) An air pipe of at least 32 mm ID which can be shut off and which extends above the bulkhead deck

1) See also Volume IV, Section 1, D.4.5.

b) Adequately dimensioned ventilation slots in the shell plating.

1.5 Steam or compressed air connections are to be provided for clearing the sea chest gratings. The steam or compressed air lines are to be fitted with shutoff valves fitted directly to the sea chest. Compressed air for blowing through sea chest gratings may exceed 2 bar only if the sea chest are constructed for higher pressures.

2. Special rules for ships with ice class

2.1 Sea connection, sea chests

2.1.1 At least one of the sea chests specified in 1.1 shall arranged as follows:

a) The sea inlet is to be located on the ship's centre line as far aft as possible.

b) In calculating the volume of the chest the following value shall be applied as a guide: about 1 m^3 for every 750 kW of the ship's engine output including the output of auxiliary engines necessary for the ship's service.

c) The chest shall be of sufficient height to allow ice to accumulate above the inlet pipe.

d) The seawater discharge line of the entire engine plant is to be connected to the top of the sea chest.

e) The free area of the strum holes shall be not less than four time the sectional area of the inlet-pipe.

2.1.2 As an alternative two smaller chests may be arranged as stated in 2.1.1.

2.2 All discharge valves shall be so arranged that the discharge of water at any draught will not be hindered by ice.

2.3 The sea chests shall be provided with a steam connection for de-icing and thawing.

2.4 In special cases, additional cooling water may be supplied to the engine plant from ballast water tanks with circulation cooling.

The system does not replace the requirement stated in 2.1.1.

2.5 A suction from the cooling water system is to be provided for the fire pumps.

3. Sea valves

3.1 Sea valves are to be so arranged that they can be operated from above the floor plates.

3.2 Discharge pipes for seawater cooling systems are to be fitted with a shutoff valve at the shell plating.

4. Strainer

The suction lines of the seawater pumps are to be fitted with strainers.

The strainers are to be so arranged that they can be cleaned during operation of the pumps.

Where cooling water is supplied by means of a scoop, strainers in the main seawater cooling line can be dispensed with.

5. Seawater cooling pumps

5.1 Diesel engine plants

5.1.1 Main propulsion plants are to be provided with main and stand-by cooling water pumps.

5.1.2 The main cooling water pump may be attached to the propulsion plant. It is to be ensured that the attached pump is of sufficient capacity for the cooling water required by main and auxiliary engines over the whole speed range of the propulsion plant.

The stand-by cooling water pump is to be driven independently of the main engine.

5.1.3 Main and stand-by cooling water pumps are each to be of sufficient capacity to meet the maximum cooling water requirements of the plant.

Alternatively, three cooling water pumps of the same capacity may be arranged, provided that two of the pumps are sufficient to supply the required cooling water for full load operation of the plant at design temperature. With this arrangement it is permissible for the second pump to be put into operation in the higher temperature range by means of a thermostat.

5.1.4 Ballast pumps or other suitable seawater pumps may be used as stand-by cooling water pumps.

5.1.5 Where cooling water is supplied by means of a scoop, the main and stand-by cooling water pumps are to be of a capacity which will ensure reliable operation of the plant under partial load conditions. The main cooling water pump is to be automatically started as soon as the speed falls below that required the operation of the scoop.

5.2 Steam turbine plants

5.2.1 Steam turbine plants are to be provided with a main and a stand-by cooling water pump.

The main cooling water pump is to be of sufficient capacity to supply the maximum cooling water requirements of the turbine plant. The capacity of the stand-by cooling water pump is to be such as to ensure reliable operation of the plant also during astern operation.

5.2.2 Where cooling water is supplied by means of a scoop, the main cooling water pump is to be of sufficient capacity for the cooling water requirements of the turbine plant under conditions of maximum astern output.

The main cooling water pump is to start automatically as soon as the speed falls below that required for the operation of the scoop.

5.3 Multi-propeller plants; plants with more than one main engine

For multi-propeller plants and plants with more than one engine, the stand-by pump has to supply sufficient cooling water for one plant and is to be capable of being connected to any plant.

5.4 Cooling water supply for auxiliary engines

Where a common cooling water pump is provided to serve more than one auxiliary engine, an independent stand-by cooling water pump with the same capacity is to be fitted. Independently operated cooling water pumps of the main engine plant may be used to supply cooling water to auxiliary engines while at sea, provided that the capacity of such pumps is sufficient to meet the additional cooling water requirement.

If each auxiliary engine is fitted with an attached cooling water pump, no stand-by cooling water pumps need be provided.

6. Cooling water supply in dock

It is recommended that a supply of cooling water, e.g. from a water ballast tank, should be available so that at least one diesel generator and, if necessary, the domestic refrigerating plant may be run when the ship in dock.

Cargo and container cooling systems shall conform to the requirements stated in Volume VIII Section 2,1.4.

K. Fresh Water Cooling Systems

1. General

1.1 Fresh water cooling systems are to be so arranged that the engines can be sufficiently cooled under all operating conditions.

1.2 Depending on the requirements of the engine plant, the following fresh water cooling systems are allowed:

a) a single cooling circuit for the entire plant.
b) separate cooling circuits for the main and auxiliary plant.
c) several independent cooling circuits for the main engine components which need cooling (e.g. cylinders, pistons and fuel valves) and for the auxiliary engines.
d) separate cooling circuits for various temperature ranges.

1.3 The cooling circuits are to be so arranged that, should one of the circuits fail, operation can be maintained by the other cooling circuits.

Where appropriate, the necessary change-over arrangements are to be provided for this purpose.

1.4 As far as possible, the temperature controls of main and auxiliary engines as well as of different circuits are to be independent of each other.

1.5 Where heat exchangers for fuel, lubricating oil or thermal oil are arranged in the cooling water circuits of cylinders or pistons, these systems are to be monitored for fuel and oil leakage.

1.6 Common cooling water systems for main and auxiliary plants are to be fitted with shutoff valves to enable repairs to be performed without taking the entire plant out of service.

2. Heat exchangers, coolers

2.1 The construction and equipment of heat exchangers and coolers are subject to the Rules of Section 8.

2.2 The coolers of cooling water systems, engines and equipment are to be constructed to ensure that the specified cooling water temperatures can be maintained under all operating conditions. Cooling water temperatures are to be adjusted to meet the requirements of engines and equipment.

2.3 Heat exchangers for auxiliary equipment in the main cooling water circuit are to be provided with bypasses if by this means it is possible, in the event of a failure of the heat exchanger, to keep the system in operation.

2.4 It is to be ensured that auxiliary machinery can be maintained in operation while repairing the main coolers. Necessary means are to be provided for changing over to other heat exchangers, machinery of equipment through which a temporary heat transfer can be achieved.

2.5 Shutoff valves are to be provided at the inlet and outlet of all heat exchangers.

2.6 Every heat exchanger and cooler is to be provided with a vent and a drain.

2.7 Keel coolers, chest coolers

2.7.1 Arrangement and construction drawings of keel and chest coolers are to be submitted for approval.

2.7.2 Permanent vents are to be provided at the top of keel coolers and chest coolers.

2.7.3 Keel coolers are to be fitted with pressure gauge connections at the fresh water inlet and outlet.

3. Expansion tanks

3.1 Expansion tanks are to be arranged at sufficient height for every cooling water circuit.

Different cooling circuits may only be connected to a common expansion tank if they do not interfere with each other. Care must be taken here to ensure that damage to or faults in one system cannot affect the other system.

3.2 Expansion tanks are to be fitted with filling connections, air pipes, water level indicators and a drain arrangement.

4. Fresh water cooling pumps

4.1 Main and stand-by cooling water pumps are to be provided for each fresh water cooling system.

4.2 Main cooling water pumps may be driven directly by the main or auxiliary engines which they are intended to cool provided that a sufficient supply of cooling water is assured under all operating conditions.

4.3 Stand-by cooling water pumps are to be driven independently of main engines.

4.4 Stand-by cooling water pumps are to have the same capacity as main cooling water pumps.

4.5 Main engines are to be fitted with at least one main and one stand-by cooling water pump. Where according to the construction of the engines more than one water cooling circuit is necessary, a stand-by pump is to be fitted for each main cooling water pump.

4.6 As an exception to 4.1, the Rules for seawater cooling pumps in J.5.4 are to be applied to the fresh water cooling pumps of important auxiliary engines.

4.7 A stand-by cooling water pump of a cooling water system may be used as a stand-by pump for another system provided that the necessary pipe connections are arranged. The shutoff valves in these connections are to be secured against unintended operation.

4.8 Equipment providing for emergency cooling from another system can be approved if the plant and system are suitable for this purpose.

4.9 For multi-propeller plants and plants with more than one engine, spare cooling water pumps ready for installation may be carried on board instead of permanently mounted stand-by cooling water pumps provided that,

a) the cooling water systems of each engine are independent or can be operated independently of the corresponding systems of the other engines and
b) the spare pumps can be installed with the means available on board.

In addition to a) and b), provision shall be made for one of the following arrangements:

c) the respective main cooling water pumps must be capable of being used interchangeably for the supply of cooling water,

   or

d) it must be possible to switch in a permanently installed pump to provide auxiliary cooling for each engine.

5. Temperature control

Cooling water circuits are to be provided with temperature controls in accordance with the requirements. Control devices whose failure may impair the functional reliability of the engine are to be equipped for manual operation.

6. Preheating for cooling water

Means are to be provided for preheating fresh cooling water.

7. Emergency generating units

Internal combustion engines driving emergency generating units are to be fitted with independent cooling systems. Such cooling systems are to be made proof against freezing.

L. Compressed Air Lines

1. General

1.1 Pressure lines connected to air compressors are to be fitted with non-return valves at the compressor outlet.

1.2 Efficient oil and water traps are to be provided in the filling lines of compressed air receivers.

1.3 Starting air lines may not be used as filling lines for air receivers.

1.4 The starting air line to each engine is to be fitted with a non-return valve and a drain.

1.5 Tyfons are to be connected to at least two compressed air receivers.

1.6 A safety valve is to be fitted behind each pressure-reducing valve.

1.7 Pressure water tanks and other tanks connected to the compressed air system are to be considered as pressure vessels and must comply with the requirements in Section 8 relating to the working pressure of the compressed air system.

2. For compressed air connections for blowing through sea chests refer to J.1.4.

3. For the compressed air supply to pneumatically operated valves refer to D.6.

4. Control air systems

Control air systems for essential consumers are to be provided with the necessary means of air treatment.

M. Exhaust Gas Lines

1. Pipe layout

1.1 Exhaust gas pipes from engines are to be installed separately from each other with regard to structural fire protection. The same applies to boiler exhaust gas pipes.

1.2 Account is to be taken of thermal expansion when laying out and suspending the lines.

1.3 Where exhaust gas lines discharge near water level, provisions are to be taken to prevent water from entering the engines.

2. Silencers

2.1 Engine exhaust pipes are to be fitted with effective silencers.

2.2 Silencers are to be provided with an inspection opening.

3. Water drains

Exhaust lines and silencers are to be provided with suitable drains of adequate size.

4. Insulation

4.1 Exhaust gas lines, silencers and exhaust gas boilers are to be effectively insulated to prevent the ignition of combustible materials on them.

4.2 Insulating materials must be incombustible.

4.3 Exhaust gas lines inside engine rooms are to be provided with a metal sheating or other approved type of hard sheathing.

5. For special Rules for tankers refer to Section 15, B.9.3.

N. Bilge Systems

1. Bilge lines

1.1 Layout of bilge lines

1.1.1 Bilge lines and bilge suctions are to be so arranged that the bilges can be completely pumped even under disadvantageous trim conditions.

1.1.2 Bilge suctions are normally to be located on both sides of the ship. For compartments located fore and aft in the ship, one bilge suction may be considered sufficient provided that it is capable of completely draining the relevant compartment.

1.1.3 Spaces located forward of the collision bulkhead and aft of the stern tube bulkhead and not connected to the general bilge system are to be drained by other suitable means of adequate capacity.

1.1.4 The required pipe thicknesses of bilge lines are to be in accordance with Table 11.4.

1.2 Pipes laid through tanks

1.2.1 Bilge pipes may not be led through tanks for lubricating oil, thermal oil, drinking water or feed-water.

1.2.2 Where bilge pipes are led through fuel tanks located above the double bottom and terminate in spaces which are not accessible during the voyage, an additional non-return valve is to be fitted in the bilge pipe where the pipe from the suction enters the fuel tank.

1.3 Bilge suctions and strums

1.3.1 Bilge suctions are to be so arranged as not to impede the cleaning of bilges and bilge wells. They are to be fitted with easily detachable, corrosion-resistant strums.

1.3.2 Emergency bilge suctions are to be arranged in such a manner that they are accesible, with free flow and at a suitable distance from the tank top or the ship's bottom.

1.3.3 For the size and design of bilge wells see Volume II, Construction Rules for Hull, Section 8, B.6.2.

1.4 Bilge valves

1.4.1 Valves in connecting pipes between the bilge and the seawater and ballast water system, as well as between the bilge connections of different compartments, are to be so arranged that even in the event of faulty operation or intermediate positions of the valves, penetration of seawater through the bilge system will be safely prevented.

1.4.2 Bilge discharge pipes are to be fitted with shutoff valves at the ship's side.

1.4.3 Bilge valves are to be arranged so as to be always accessible irrespective of the ballast and loading condition of the ship.

1.5 Reverse-flow protection

1.5.1 A screw-down non-return valve is recognized as reverse-flow protection.

1.5.2 For bilge connections outside engine rooms, a combination of a non-return valve without shutoff mechanism and a remote controlled shutoff valve may be recognized as equivalent.

1.6 Pipe connections

1.6.1 To prevent the penetration of ballast and seawater into the ship through the bilge system, two means of reverse-flow protection are to be fitted in the bilge connections. One such means of protection is to be fitted in each suction line.

1.6.2 The direct bilge suction and the emergency injection need only have one means of reverse-flow protection.

1.6.3 Where a direct seawater connection is arranged for attached bilge pumps to protect them against running dry, the bilge suctions are also to be fitted with two screw-down non-return valves.

1.6.4 Where the direct suction is connected to a centrifugal pump which can also be used for cooling water, ballast water or fire extinguishing, a screw-down non-return valve is to be fitted in the discharge pipe of the pump.

1.6.5. The discharge pipes of oily water separators are to be fitted with to two non-return valves at the ship's side.

2. Calculation of pipe diameters

2.1 The calculated values according to formulae (4) to (6) are to be rounded up to the next higher nominal diameter.

a) main bilge pipes

$$d_H = 1{,}68 \cdot \sqrt{(B + H) \cdot L} + 25 \text{ [mm]} \quad (4)$$

b) branch bilge pipes

$$d_z = 2{,}15 \cdot \sqrt{(B + H) \cdot l} + 25 \text{ [mm]} \quad (5)$$

d_H	[mm] =	calculated inside diameter of main bilge pipe
d_z	[mm] =	calculated inside diameter of branch bilge pipe
L	[m]	length of ship between perpendiculars
B	[m]	moulded breadth of ship
H	[m]	depth of ship to the bulkhead deck
l	[m]	length of the watertight compartment

2.3 Tankers

The diameter of the main bilge pipe in the engine rooms of tankers and bulk cargo/oil carriers is calculated using the formula:

$$d_H = 3{,}0 \cdot \sqrt{(B + H) \cdot l_1} + 35 \text{ [mm]} \quad (6)$$

where :

l_1 [m] = total length of spaces between cofferdam or pump-room bulkhead and stern tube bulkhead

Other terms as in formulae (4) and (5).

Branch bilge pipes are to be dimensioned in accordance with 2.2 b). For bilge installations for spaces in the cargo area of tankers and bulk cargo/oil carriers see Section 15.

2.4 Minimum diameter

The inside diameter of main and branch bilge pipes is not to be less than 50 mm. For ships under 25 m length, the diameter may be reduced to 40 mm.

2.5 Maximum diameter

The diameter of the main bilge line calculated according to 2.1 a) need not exceed ND 200.

2.6 Deviations

Where in individual cases formula (5) requires a greater bilge pipe diameter than that determined by formula (4), a greater pipe diameter than that according to formula (4) is not necessary.

3. Bilge pumps

3.1 Capacity of independent pumps

Each bilge pump must be capable of delivering:

$$Q = 5{,}75 \cdot 10^{-3} \cdot d_H^2 \quad [m^3/h] \quad (7)$$

where :

Q [m³/h] = minimum capacity

d_H [mm] calculated inside diameter of main bilge pipe

3.2 Where centrifugal pumps are used for bilge pumping, they must be self-priming or connected to an air extracting device.

3.3 Capacity of attached bilge pumps

Bilge pumps having a smaller capacity than that specified by formula (7) are acceptable provided that the independent pumps are designed for a correspondingly larger capacity.

3.4 Use of other pumps for bilge pumping

Ballast pumps, stand-by seawater cooling pumps and general service pumps may also be used as independent bilge pumps provided they are self-priming and of the required capacity according to formula (7).

3.4.1 Fuel and oil pumps may not be connected to the bilge system.

3.5 Number of bilge pumps for cargo ships

Cargo ships are to be provided with two independent, power bilge pumps. On ships up to 2000 tons gross, one of these pumps may be attached to the main engine.

On ships of less than 100 tons gross, one engine-driven bilge pump is sufficient. The second independent bilge pump may be a permanently installed manual bilge pump. The engine-driven bilge pump may be coupled to the main propulsion plant.

3.6 Number of bilge pumps for passenger ships

At least three bilge pumps are to be provided. One pump may be coupled to the main propulsion plant. Where the criterion numeral is 30[1] or more, a further bilge pump is to be provided.

4. Bilge pumping for various spaces

4.1 Machinery spaces

4.1.1 On ships of more than 100 tons gross, the bilges of every main machinery space must be capable of being pumped as follows:

a) Through the bilge suctions connected to the main bilge system,

b) through one direct suction connected to the largest independent bilge pump and

c) through an emergency bilge suction connected to the cooling water pump of the main propulsion plant or through another suitable emergency bilge system.

4.1.2 If the ship's propulsion plant is located in several spaces, a direct suction in accordance with 4.1.1 b) is to be provided in each watertight compartment in addition to branch bilge suctions in accordance with 4.1.1 a).

When the direct suctions are in use, it must be possible to pump simultaneously from the main bilge line by means of all the other bilge pumps.

1) See SOLAS 1974, Chapter II-1, part A, regulations 5 and 18.

[illegible] direct suction may not be less than [illegible] main bilge pipe.

[illegible] diameter of the emergency bilge suction on [illegible] accordance with 4.1.1 c) is to be at least [illegible] diameter and on motor ships equal to the [illegible] cooling water pump suction line. Ex[illegible] this Rule require the approval of the So[illegible] emergency bilge suction must be connected [illegible] cooling water pump suction line by means of a [illegible] non-return valve. This valve is to be [illegible] with a plate with the notice:

[illegible] bilge valve!
[illegible] opened in an emergency
[illegible]

[illegible] bilge valves and cooling water inlet valves [illegible] of being operated from above the floor [illegible]

[illegible] control rooms and similar spaces as well [illegible] rooms are to be provided with drains [illegible] room bilge. A drain pipe which passes through a watertight bulkhead is to be fitted with a self-closing valve

[illegible] Shaft tunnel

[illegible] is to be arranged at the after end of the [illegible] Where the shape of the bottom or the length of the tunnel requires, an additional bilge suction is to be provided at the forward end. Bilge valves for the shaft tunnel are to be arranged outside the tunnel in the engine room.

[illegible] Cargo holds

[illegible] Cargo holds with bilge gutterways are to be fitted with bilge suctions fore and aft. Cargo holds having a length under 30 m may be provided with only one bilge suction on each side.

[illegible] Cargo holds in which the tank decks extend to the ship's shell are to be provided with bilge wells of adequate size.

[illegible] ships with only one cargo hold, bilge wells of sufficient size are to be provided fore and aft.

[illegible] For cargo holds for the transport of dangerous goods, see Section 12.0.6.

4.4 Spaces which may be used for ballast water, oil or dry cargo

Where dry cargo holds are also intended for carrying ballast water or oils, the branch bilge pipes from these spaces are to be conneted to the ballast or cargo pipe system only by change-over valves.

The change-over valves must be so arranged that intermediate positioning does not connect the different piping systems. Change-over connections are to be such that the pipe not connected to the cargo hold is isolated from the blanked off side of the change-over connection.

4.5 Refrigerated cargo spaces

Refrigerated cargo spaces and thawing trays are to be provided with drains which cannot be shut off. Each drain pipe is to be fitted at its discharge end with a trap to prevent the transfer of heat and odours.

4.6 Spaces for transporting livestock

Spaces intended for the transport of livestock are to be additionally fitted with pumps or ejectors for discharging the waste overboard.

4.7 Fore and after peaks

Connection of the fore and after peaks to the general bilge system is not permitted. Where the peak tanks are not connected to the ballast system, separate means of pumping are to be provided.

Where the after peak terminates at the engine room, it may be drained to the engine room bilge through a pipe fitted with a shutoff valve. Similar emptying of the fore peak into an adjoining space is not permitted.

4.8 Spaces above peak tanks

These spaces may either be connected to the bilge system or be pumped by means of hand-operated bilge pumps. Spaces above the after peak may be drained to the shaft tunnel or machinery space, provided that the drain line is fitted with a self-closing shutoff valve at a clearly visible and easily accessible position. The drain pipes shall have an inside diameter of at least 40 mm.

4.9 Cofferdams, pipe tunnels and void spaces

Cofferdams, pipe tunnels and void spaces adjoining the ship's shell are to be connected to the bilge system.

4.10 Chain lockers

Chain lockers may be drained either though the bilge system or by means of manual pumps. They may not be drained to the fore peak.

5. Oily water separating equipment

Ships of 400 tons gross and over are to be provided with an oily water separator or filter plant for the separation of water/oil mixtures.

For outfitting with oily water separators, filter plants, oil collecting tanks and oil discharge pipes and with a system for monitoring and controlling the discharge of water from oily water separators, attention is drawn to the need to comply with the provisions of the International Convention of 1973 and of the Protocol of 1978 for the prevention of pollution from ships "MARPOL".

6. Additional Rules for passenger vessels

6.1.1 The arrangement of bilge pipes

— within 0,2 B of the ship's side measured at the level of the subdivision load line,
— in the double bottom lower than 460 mm above the

base line or

— below the horizontal level specified in Volume II, Section 29, F.2 (Construction Rules for Hull).

is permitted only if a non-return valve is fitted in the compartment in which the corresponding bilge suction is located.

Where parts of the bilge arrangement (pump with suction connections) are situated less than 0,2 B from the ship's shell, damage to one part of the arrangement must not result in the rest of the bilge arrangement being rendered inoperable.

6.1.2 Where only one common piping system is provided for all pumps, all the shutoff and change-over valves necessary for bilge pumping must be arranged for operating from above the bulkhead deck. Where an emergency bilge pumping system is provided in addition to the main bilge system, this is to be independent of the latter and must be so arranged as to permit pumping of any flooded compartment. In this case, only the shutoff and change-over valves of the emergency system need be capable of being operated from above the bulkhead deck.

6.1.3 Shutoff and change-over valves which must be capable of being operated from above the bulkhead deck should be clearly marked, accessible and fitted with a position indicator.

6.2 Bilge suctions

Bilge pumps in the machinery spaces must be provided with direct bilge suctions in these spaces, but not more than two direct suctions need be provided in any one space.

Bilge pumps located in other spaces are to have direct suctions to the space in which they are installed.

6.3 Arrangement of bilge pumps

Bilge pumps must be installed in separate watertight compartments which are to be so arranged that they are unlikely to be simultaneously flooded in the event of damage to the ship. Ships with a length of 91,5 m or over or having a criterion numeral of 30[1] or more are to have at least one bilge pump available in emergency cases. This requirement is satisfied if

a) one of the required pumps is a submersible emergency bilge pump connected to its own bilge system and powered from a source located above the bulkhead deck or

b) the pumps and their sources of power are distributed over the entire length of the ship the buoyancy of which in damaged condition is ascertained by calculation for each individual compartment or group of compartments, at least one pump being available in an undamaged compartment or

c) bilge pumps are installed above the bulkhead deck.

6.4 Passenger vessels for limited range of service

The range of bilge pumping for passenger vessels with limited range of service, e.g. navigation on shallow water service, can be agreed with BKI.

1). See Solas 1974, Chapter II-1, part A, regulations 5 and 18

7. Additional Rules for tankers

See Section 15, B.4.

8. Bilge testing

All bilge arrangements are to be tested under the Society's supervision.

O. Ballast Systems

1. Ballast lines

1.1 Arrangement of piping — general

1.1.1 Suctions in ballast water tanks are to be so arranged that the tanks can be emptied despite unfavourable conditions of trim and list.

1.1.2 Ships having very wide double bottom tanks are also to be provided with suctions at the outer sides of the tanks. Where the length of the ballast water tanks exceeds 30 m, the Society may require suctions to be provided in the forward part of the tanks.

1.2 Pipes passing through tanks

Ballast water pipes may not pass through drinking water, feedwater, thermal oil or lubricating oil tanks.

1.3 Piping systems

1.3.1 Where a tank is used alternately for ballast water and fuel (change-over tank), the suction in this tank is to be connected to the respective system by three-way cocks with L-type plugs, cocks with open bottom or change-over piston valves. These must be arranged so that there is no connection between the ballast water and the fuel systems when the valve or cock is in an intermediate position. Change-over pipe connections may be used instead of the above mentioned valves. Each change-over tank is to be individually connected to its respective system. For remotely controlled valves see D.6.

1.3.2 Where ballast water tanks may be used exceptionally as dry cargo holds, such tanks are also to be connected to the bilge system. The requirements specified in N.4.4 are applicable.

1.3.3 Where, on cargo ships, pipelines are led through the collision bulkhead below the freeboard deck, a shutoff valve is to be fitted directly at the collision bulkhead inside the fore peak.

It is to be capable of being remotely operated from above the freeboard deck.

Where the fore peak is immediately adjacent to a permanently accessible room (e.g. bow thruster room) which is separated from the cargo space, this shutoff valve may be mounted directly at the collision bulkhead inside this room without provision for remote control.

1.3.4 On passenger ships, only one pipeline may be led through the collision bulkhead below the free-board deck. The pipeline is to be fitted with a remote controlled shutoff inside the forepeak directly at the collision bulkhead. The remote control must be operated from above the freeboard deck. Where the forepeak is divided into two compartments, two pipelines may in exceptional cases be passed through the collision bulkhead below freeboard deck.

2. Ballast pumps

The number and capacity of the pumps must satisfy the vessel's operational requirements.

3. Additional Rules for passenger ships

3.1 Cross-flooding arrangements

As far as possible, anti-heeling arrangements for equalising unsymmetrical flooding should operate automatically. Where the arrangement does not operate automatically, any shutoff valves and other devices must be capable of being operated from above the bulkhead deck.

The cross-flooding arrangements must ensure that in case of flooding, equalisation is achieved within 15 minutes. Separation of the tanks is to be ensured during normal operation.

Anti-heeling arrangements for equalising unsymmetrical flooding are to be submitted to the Society for approval.

4. Additional Rules for tankers

See Section 15, B.4.

5. Operational testing

The ballast arrangement is to be subjected to operational testing under the Society's supervision.

P. Thermal Oil Systems

Thermal oil systems shall be installed in accordance with Section 7-II.

The pipelines, pumps and valves belonging to these systems are also subject to the following requirements.

1. Pumps

1.1 Circulating pumps

Two circulating pumps which are to be independent of each other are to be provided.

1.2 Transfer pumps

A transfer pump is to be installed for filling the expansion tank.

1.3 The pumps are to be so mounted that any oil leakage can be safely disposed of.

1.4 For emergency stopping see Section 12, B.9.

2. Valves

2.1 Only valves made of ductile materials may be used.

2.2 Valves shall be designed for a nominal pressure of PN 16.

2.3 Valves are to be mounted in accessible positions.

2.4 Non-return valves are to be fitted in the pressure lines of the pumps.

2.5 Valves in return pipes are to be secured in the open position.

3. Piping

3.1 Pipes in accordance with Table 11.1 are to be used.

3.2 The material of the sealing joints is to be suitable for permanent operation at the design temperature and resistant to the thermal oil.

3.3 Provision is to be made for thermal expansion by an appropriate pipe layout and the use of suitable compensators.

3.4 The pipe lines are to be preferably connected by means of welding. The number of detachable pipe connections is to be minimized.

3.5 The laying of pipes through accommodation, public or service spaces is not permitted.

3.6 Pipelines passing through cargo holds are to be installed in such a way that no damage can be caused.

3.7 Pipe penetrations through bulkheads and decks are to be insulated against conduction of heat.

3.8 The venting is to be so arranged that air/oil mixtures can be carried away without danger.

4. Pressure testing

See B.4.

5. Tightness and operational testing

After installation, the entire arrangement is to be subjected to tightness and operational testing under the supervision of the Society.

Q. Air, Overflow and Sounding Pipes

General

The laying of air, overflow and sounding pipes is permitted only in places where the laying of the corresponding piping system is also permitted (see Table 11.4).

1. Air and overflow pipes

1.1 Arrangement

1.1.1 All tanks, void spaces etc. are to be fitted at their highest position with air pipes which must normally terminate above the open deck.

1.1.2 Air and overflow pipes are to be laid vertically.

Air and overflow pipes terminating above the open deck are to be fitted with automatic shutoff devices of approved type. Proof is to be furnished that the penetration of water is prevented even in inclined positions.

1.1.3 Air and overflow pipes passing through cargo holds are to be protected against damage.

1.1.4 For the height above deck of air and overflow pipes see Volume II, Construction Rules for Hull, Section 21.

1.1.5 Air pipes from unheated leakage oil and lubricating oil tanks located above the double bottom may terminate at clearly visible positions in the engine room. Funnels and pipes are to be fitted to provide for safe drainage in the event of a possible overflow.

1.1.6 Air pipes from lubricating oil storage tanks which form part of the ship's shell are to terminate in the engine room casing above the freeboard deck, in passenger ships above the bulkhead deck. It must be ensured that no leaking oil can spread on to heated surfaces where it may ignite. For the air pipes of lubricating oil drain tanks see Section 10, C.

1.1.7 Wherever possible, the air pipes of feedwater and distillate tanks should not extend into the open. Where these tanks form part of the ship's shell the air pipes are to terminate within the engine room casing above the freeboard deck, in passenger ships above the bulkhead deck.

1.1.8 Air pipes for cofferdams and void spaces with bilge connections are to be extended above the open deck.

1.1.9 Overflow pipes from fuel tanks are to be led at a sufficient gradient into an overflow tank of sufficient capacity.

With closed fuel overflow systems proof is required that the resistance to flow of the overflow pipes at the chosen flow velocities cannot cause unacceptably high pressures in the tanks in the event of overflow.

The use of fuel supply tanks as overflow tanks is permitted but requires the fitting of an additional level alarm and of an air pipe with a cross-section 1,25 times that of the fuel bunker line.

Air pipes of fuel day tanks which are led above the open deck must be located at a sufficient height and protected against breaking seas.

Where fuel day tanks are fitted with change-over overflow pipes, the change-over devices are to be so arranged that the overflow is led to one of the storage tanks.

1.1.10 The overflow pipes of tanks used alternatively for oil fuel and ballast water must be capable of being separated from the fuel overflow system and led separately to atmosphere when the tanks are to be filled with ballast water.

1.1.11 Where the air and overflow pipes of several tanks situated at the ship's shell lead to a common line, the connections to this line are to be above the freeboard deck if possible but at least so high above the deepest load waterline that should a leakage occur in one tank due to damage to the hull or listing of the ship, fuel or water cannot flow into another tank.

The air and overflow pipes of lubricating oil and fuel tanks shall not be led to a common line.

1.1.12 For the connection to a common line of air and overflow pipes on ships with classification mark ⊞ or ⊞ , see D.9.

1.2 Number of air and overflow pipes

Tanks which extend from side to side of the ship must be fitted with an air pipe at each corner. At the narrow ends of double bottom tanks in the forward and after parts of the ship one air pipe is sufficient. Where several double bottom tanks are arranged adjacent to each other, one air pipe at the forward and after ends of the tanks is sufficient.

1.3 Determination of the pipe cross-sections

1.3.1 Regarding the cross-sectional areas of air and overflow pipes see Table 11.14.

The minimum diameter of air and overflow pipes shall not be smaller than 50 mm.

1.3.2 The clear cross-sectional are of air pipes on passenger ships with cross-flooding arrangement must be so large that the water can pass from one side of the ship to the other within 15 minutes. See also O.3.

1.4 The minimum thicknesses of air and overflow pipes are to comply with Tables 11.15 a and 11.15 b.

1.5 For pipe materials see B.

2. Sounding pipes

2.1 General

2.1.1 Sounding pipes are to be provided for tanks, cofferdams and void spaces with bilge connections and for bilges and bilge wells in spaces which are not accessible at all times.

On application, the provision of sounding pipes for bilge wells in permanently accessible spaces may be dispensed with.

2.1.2 Where the tanks are fitted with remote level indicators which are type-approved by BKI the arrangement of sounding pipes can be dispensed with.

2.1.3 As far as possible, sounding pipes are to be laid straight and are to extend as near as possible to the bottom of the tank.

2.1.4 Sounding pipes which terminate below the deepest load waterline are to be fitted with elfclosing shutoff devices. Such sounding pipes are only permissible in spaces which are accesible at all times.

All other sounding pipes are to be extended to the open deck. The sounding pipe openings must always be accessible and fitted with watertight closures.

2.1.5 Sounding pipes of tanks are to be provided close to the top of the tank with holes for equalising the pressure.

2.1.6 In cargo holds, a sounding pipe is to be fitted to each bilge well.

2.1.7 Sounding pipes passing through cargo holds are to be protected against damage.

2.2 Sounding pipes for fuel, lubricating oil and thermal oil tanks

2.2.1 Sounding pipes which terminate below the open deck are to be provided with self-closing devices as well as with self-closing test valves.

2.2.2 Sounding pipes shall not to be located in the vicinity of firing plants, machine components with high surface temperatures or electrical equipment.

2.2.3 Sounding pipes must not terminate in accommodation or service spaces.

2.2.4 Sounding pipes are not to be used as filling pipes.

2.3 Cross-sections of pipes

2.3.1 Sounding pipes shall have a nominal inside diameter of at least 32 mm.

2.3.2 The nominal diameters of sounding pipes which pass through refrigerated holds at temperatures below 0°C are to be increased.

2.4 The minimum wall thicknesses of sounding pipes are to be in accordance with Tables 11.15 a and 11.15 b.

2.5 For pipe materials see B.

R. Drinking Water Systems

1. Drinking water tanks

For the design and arrangement of drinking water tanks see Volume II, Hull Construction Rules, Section 12.

2. Drinking water tank connections

2.1 Filling connections are to be located sufficiently high above deck and are to be fitted with a closing device.

2.1.1 Filling connections are not to be fitted to air pipes.

2.2 Air/overflow pipes are to be extended above the open deck.

2.2.1 The upper openings of air pipes are to be fitted with automatic closing devices and are to be protected against the entry of insects.

2.3 Sounding pipes must terminate sufficiently high above deck.

3. Drinking water pipe lines

3.1 Drinking water pipe lines are not to be connected to pipe lines carrying other media.

3.2 Drinking water pipe lines are not to be laid through tanks which do not contain drinking water.

3.3 Drinking water supply to tanks which do not contain drinking water (e.g. expansion tanks of the fresh water cooling system) is to be made by means of an open funnel or with means of preventing flow-back.

4. Drinking water pumps

4.1 Separate drinking water pumps are to be provided for drinking water systems.

4.2 The pressure lines of the pumps of drinking water pressure tanks are to be fitted with screw-down non-return valves.

5. Drinking water generation

Where the distillate produced by the ship's own evaporator unit is used for the drinking water supply, the treatment of the distillate has to comply with the requirements of national health authorities.

S. Sanitary Systems

1. General arrangement

1.1 For scuppers and overboard discharges see Volume II, Hull Construction Rules, Section 21.

1.2 Sanitary discharge pipes located in cargo holds are to be specially protected. For minimum wall thicknesses see Tables 11.15 a and 11.15 b. Individual sanitary discharge pipes are to be connected to common discharge pipes.

1.3 The discharge lines of the sewage pumps are to be fitted with a storm valve and a gate valve which is to be fitted in the discharge line directly at the ship's side.

A second means of preventing backflow is to be arranged either in the suction or in the pressure line of sewage pumps.

1.4 Sewage tanks

1.4.1 Air pipes are to be extended above the open deck and are to be fitted with automatic closing devices.

1.4.2 A flushing connection is to be provided.

1.4.3 If, when the tank is full, sewage can flow out through the sanitary arrangements connected, the tank is to be fitted with a level alarm.

1.4.4 Ballast and bilge pumps may not be used for emptying sewage tanks.

2. Additional rules for ships with classification mark ⊞ or ⊞̇

2.1 On ships with the classification mark ⊞ or ⊞̇, the sanitary arrangements are to be situated above the bulkhead deck and their discharge pipes are to be so arranged that, in the event of damage, the undamaged compartments cannot be flooded by damaged discharge lines even if the ship inclines temporarily.

2.2 Sanitary discharge lines on ships with the classifi-

Table 11.14 Sections of air and overflow lines

Tank filling and overflow systems		Clear cross-section[1] of air and overflow pipes: A P	A O P	O P	D P[2]	Remarks
Filling system	Flooding	1/3 f per tank	—	—	—	—
	Pumping	—	A O P or O P with 1,25 f per tank		—	—
	Stand pipe[3] with overflow as pressure relief	1/3 f per tank	—	1,25 F to over-flow tank	—	cross-section of stand-pipe 1,25 F
Overflow system	Common over-flow chest	1/3 F at chest	—	1,25 f per tank	1,25 F	—
	Common overflow manifold	1/3 F at manifold	—	1,25 f per tank	1,25 F	cross-section of manifold 1,25 F
	Over-flow tank	Provided that the tank is fitted with a reliable alarm which operates when the tank is 1/2 to 3/4 full, AP = 70 mm I.D., AP = 1,25 F				

Explanatory notes:

- AP = air pipe
- AOP = air and overflow pipe
- OP = overflow pipe
- f = cross-section of tank filling pipe
- DP = drainage line from overflow manifolds or chest to overflow tank
- F = cross-section of main filling pipe

1) 1,25 f as the total cross-section is sufficient if it can be proved that the resistance to flow of the air and overflow pipes including their shutoff devices at the chosen flow velocities cannot cause unacceptably high pressures in the tanks in the event of overflow.

2) Drain pipes should be provided with means of observation. Sight glasses must be able to withstand a test pressure of at least 4 bar. Shutoff devices in drain pipes must be sealed in the open position.

3) Stand pipes must branch off from the filling lines or pump pressure line without a shutoff device. The pressure-relief loop is to be fitted with a vent pipe to the open air.

cation mark ⊞ or ⊕ which terminate below the bulkhead deck are to be connected to a waste water tank (sewage tank). Each watertight compartment shall normally be provided with such a tank.

2.3 Where discharge lines from several watertight compartments are connected to one tank, the compartments are to be separated from each other by remote controlled gate valves which are to be arranged at the watertight bulkheads. The gate valves must be capable of being operated from an accessible position above the bulkhead deck and are to be fitted with an indicator for the closed position.

T. Hose Assemblies and Compensators of Non Metallic Materials

Hose assemblies are hoses and hose fittings in the installed and tested condition.

Table 11.15a Choice of minimum wall thicknesses

Piping system or position of open pipe outlets	Location: Tanks with same media	Tanks with disparate media	Drain lines and scupper pipes: below freeboard deck or bulkhead deck, WITHOUT shutoff on ship's side	Drain lines and scupper pipes: below freeboard deck or bulkhead deck, WITH shutoff on ship's side	Drain lines and scupper pipes: above freeboard deck	Air, sounding and overflow pipes: above weather deck	Air, sounding and overflow pipes: below weather deck	Cargo holds	Machinery spaces
Air, overflow and sounding pipes	A	C	–	–	–	C	A	A	A
Scupper pipes from open deck	A	B	B	A1)	A	–	–	B	A
Discharge and scupper pipes leading directly overboard	A	B	B	A1)	–	–	–	B	A
Discharge pipes of pumps for sanitary systems	A	B	–	A1)	A	–	–	B	A

1) Group N minimum wall thicknesses to Table 11.5 may be used for drain pipes terminating in a non-watertight sewage treatment plant.

Table 11.15b Minimum wall thicknesses of air, overflow, sounding and sanitary pipes

Pipe O.D [mm]	Minimum wall thickness [mm] A	B	C
38 - 82,5	4,5	7,1	6,3
88,9	4,5	8	6,3
101,6 - 114,3	4,5	8	7,1
127 - 139,7	4,5	8,8	8
152,4	4,5	10	8,8
159-177,8	5	10	8,8
193,7	5,4	12,5	8,8
219,1	5,9	12,5	8,8
244,5 - 457,2	6,3	12,5	8,8

1. Scope

The following rules apply to hose assemblies in fuel, lubricating oil, hydraulic, compressed air, auxiliary steam and water systems and to compensators made of non-metallic materials.

For rubber compensators for bilge and ballast lines see D.1.3.

Rubber compensators are not permitted in the cargo lines of tankers.

Hose assemblies in CO_2 systems are subject to the "Regulations for the Type Testing of Hoses for Carbon Dioxide (CO_2) High-Pressure Fire Extinguishing Systems".

Hose assemblies in halon systems are subject to the "Regulations for the Type Testing of Hoses for Halon Fire Extinguishing Systems".

2. Requirements

2.1 Proof must be provided that hose assemblies and compensators including their connection fittings are suitable for the intended working media, pressures and temperatures.

2.2 Hose assemblies and compensators shall be selected for the maximum allowable working pressure of the system concerned. At least 5 bar is to be considered as the minimum working pressure. The bursting pressure is subject to the requirement in 2.3.

2.3 Hose assemblies and compensators shall be designed for a bursting pressure equal to at least four times the maximum permissible working pressure.

Compensators with nominal diameters of more than 400 mm are to be dimensioned for a bursting pressure of at least three times the maximum working pressure.

2.4 Hose assemblies and compensators in fuel, lubricating oil, hydraulic and seawater systems shall be flame-resistant.[1)]

3. Layout

3.1 The flexible hoses are not to be longer than required for the application. The bending radii are not to be less than specified by the manufactures. Flexible hoses are to be arranged in accessible position.

3.2 The flexible hoses are to be bound or pressed in the end fittings.

Water lines with working pressures ≤ 5 bar and supercharge air and scavenge air lines may be connected to flexible hoses by strong clips.

1) For the purpose of these Rules, a hose, or a hose provided with suitable flame protection, is considered to be "flame-resistant" if it is able at an internal pressure corresponding to the service pressure (of at least 5 bar) and with a through-flowing water temperature of approximately 80°C at the hose outlet to sustain for a period of at leat 30 minutes a temperatures of about 800°C and exposure to flames without the integrity of the hose being impaired.

4. Testing

4.1 Hose assemblies and components are to be subjected to a pressure test. The test pressure shall equal at least 2,0 times the maximum permissible working pressure.

4.2 Proof of the bursting pressure of each type of flexible hose is to be submitted to the Society.

4.3 The pressure testing of flame-resistant flexible hoses is to comply with the Regulations for the "Flame Resistance Test for Hoses and Compensator Made of Non-Metallic Materials".

5. Cargo hoses

5.1 The selection and testing of cargo hoses are subject to the Society's "Requirements and Testing of Cargo Hoses Made of Non-Metallic and Metallic Materials".

5.2 The cargo hoses of gas tankers are subject to the Society's Rules specified in Volume IX, Section 5.4.

5.3 The cargo hoses of chemical tankers are subject to the requirements specified in Section 15, D.2.2.3.

results from towing tests have been coordinated. The analysis of the collected basis material has been carried out in the following way:

1. All data have been referred to the model area, and the model resistance (R_{Tm}) has been determined as a function of speed.
2. The specific total resistance coefficient of the model (C_{Tm}) has been determined:

$$C_{Tm} = \frac{R_{Tm}}{\frac{1}{2}\rho V_m^2 S_m} \qquad (5.5.5)$$

where ρ is the mass density, V_m is velocity of model, S_m is wetted surface of model (= mean girth × length on waterline).

3. The specific residual resistance coefficient has been determined from

$$C_R = C_{Tm} - C_{Fm} \qquad (5.5.6)$$

where C_{Fm} is the specific frictional resistance coefficient. The "ITTC 1957 model–ship correlation line" has been used to determine the frictional resistance coefficient

$$C_F = \frac{0.075}{(\log_{10}R_n - 2)^2} \qquad (5.5.7)$$

where R_n is the Reynolds Number (VL/ν, where ν is coefficient of kinematic viscosity and L is the length on waterline). In Fig. 5.5.4 contours of C_F are given for different values of V and F_n. The abscissa is the length L of the model. The diagram corresponds to $\nu = 1.139 \times 10^{-6}$ m s^{-1}, $\rho = 1.000$ t/m^3, and $T = 15$°C. The diagram may therefore be used at other conditions, that is, other densities and temperatures, only if the length is altered before entering the diagram to

$$L_1 = \frac{1.139}{10^6\nu} L \qquad (5.5.8)$$

4. C_R has been expressed as a function of Froude number

$$F_n = \frac{V}{\sqrt{gL}} \qquad (5.5.9)$$

(the speed–length ratio $V/\sqrt{L}$, where V is measured in knots and L is in feet, is found as a subscale on the C_R diagrams).

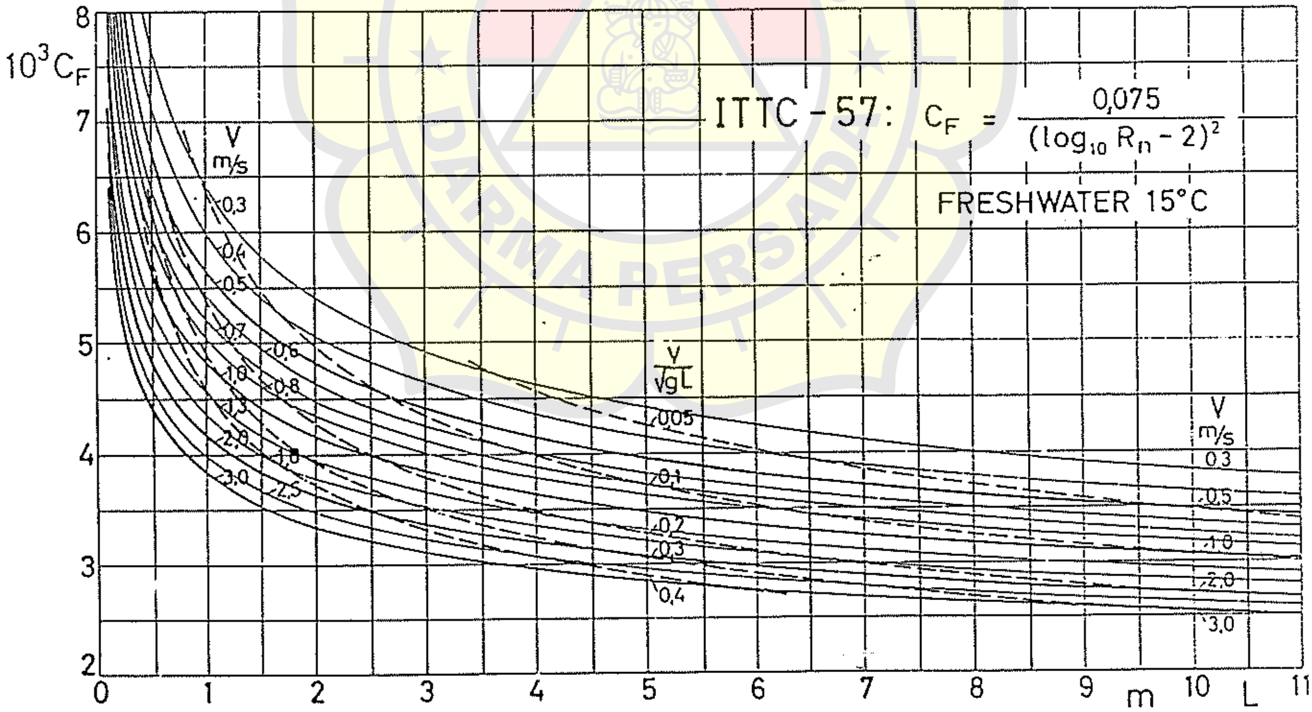


Figure 5.5.4. The frictional resistance coefficient C_F (according to ITTC 1957) as a function of ship–model length L and speed V.

5. The results have been arranged in groups according to length–displacement ratio $L/\nabla^{1/3}$ and the prismatic coefficient φ of the model. Here ∇ is the volumetric displacement and

$$\varphi = \frac{\nabla}{LBT\beta} \qquad (5.5.10)$$

where B is breadth, T is draught, and β is midship section area coefficient.

6. The main diagrams have been drawn giving the mean curves of C_R for the breadth–draught ratio $B/T = 2.5$. The diagrams are shown in Figs. 5.5.5–5.5.13.

In some places in the diagram the curves are dotted in order to indicate that they have been based either on very few test results or determined by extrapolation. The uncertainty is therefore comparatively great in these areas. Furthermore, it should be noted that the uncertainty is also great in and near the areas where the curves have pronounced humps, especially where the slope becomes negative. Small alterations in the hull form in these areas can considerably influence the C_R value.

It must also be mentioned that the resistance curves correspond to vessels with a standard form, that is, a standard position of the center of buoyancy, standard B/T, normally shaped sections, moderate cruiser stern, and raked stem.

The resistance R and the effective power P_E for a new ship can then be calculated by

$$R = C_T(\tfrac{1}{2}\rho V^2 S) \quad (\mathrm{N}) \qquad (5.5.11)$$

$$P_E = RV \quad (\mathrm{kW}) \qquad (5.5.12)$$

where the total ship resistance coefficient is

$$C_T = C_R + C_F + C_A \qquad (5.5.13)$$

where

C_R = residual resistance coefficient, which for the "standard" ship form can be taken from the diagrams (Figs. 5.5.5–5.5.13)

C_F = frictional resistance coefficient, which can be calculated by

$$C_F = \frac{0.075}{(\log_{10}R_n - 2)^2} \qquad (5.5.14)$$

or can be taken from Fig. 5.5.14 where contours of C_F are given from different values of V. The abscissa is the length L of the ship. The diagram corresponds to $\nu = 1.188 \times 10^{-6}$ m s^{-1}, $\rho = 1.025$ t/m^3, and $t = 15$°C. The diagram may therefore be used at other conditions, that is, other densities and temperatures, only if the length is altered before entering the diagram to:

$$L_1 = \frac{1.188}{10^6\nu} L \qquad (5.5.15)$$

C_A = incremental resistance coefficient, which is a coefficient correcting for roughness of the surface and scale effect on the results from the model experiments. In this way C_A will depend on the way in which C_R and C_F are fixed.

If the ship has to tow, R must be replaced by $R + F$, where F is the two-rope pull.

As ships are generally different from the standard to a greater or lesser extent, the following corrections should be taken into account, when the ship resistance of the ship and the environments had to be taken into account.

B/T

As the diagrams have been prepared for a breadth–draught ratio corresponding to

$$B/T = 2.5 \qquad (5.5.16)$$

a correction must be made if C_R is desired for a ship with a larger or smaller breadth–draught ratio.

Examination of the present test material has shown that the following correcting formula can be recommended:

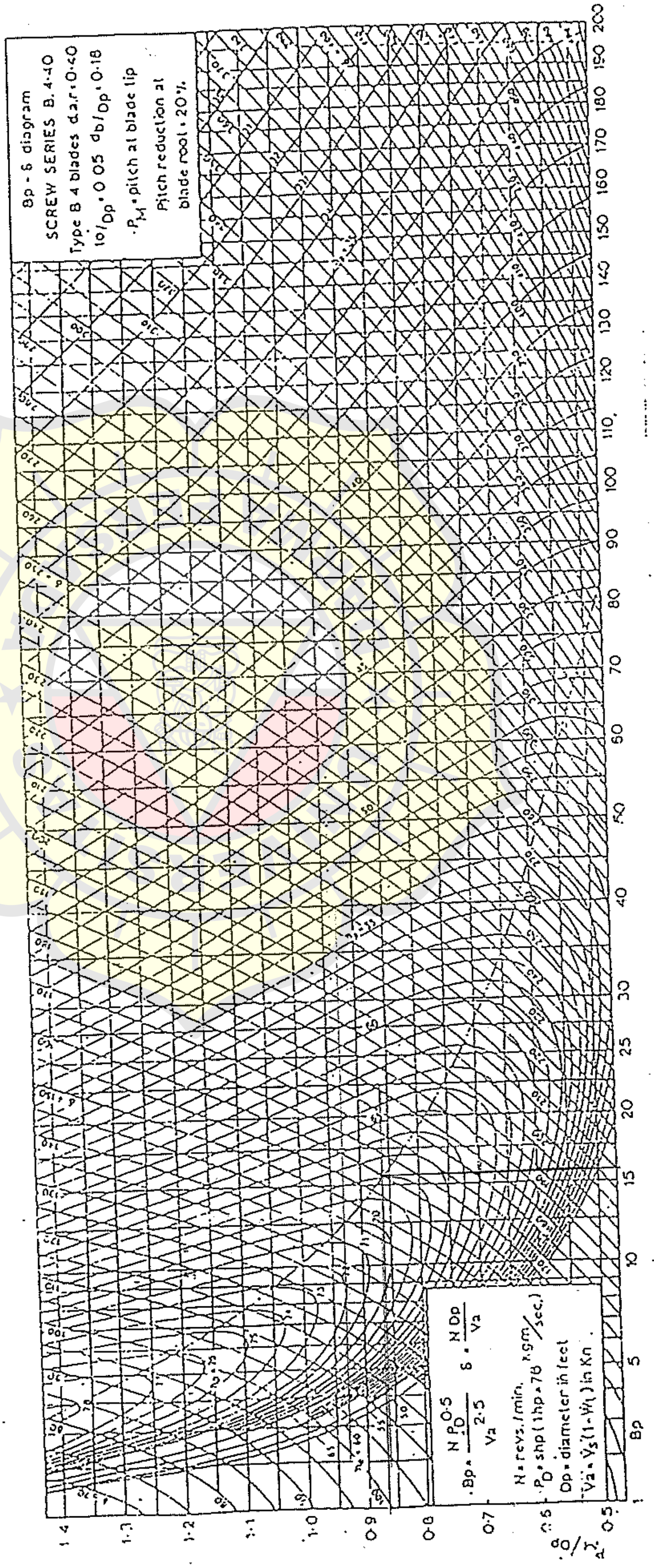
$$10^3 C_R = 10^3 C_{R(B/T=2.5)} + 0.16\,(B/T - 2.5) \qquad (5.517)$$

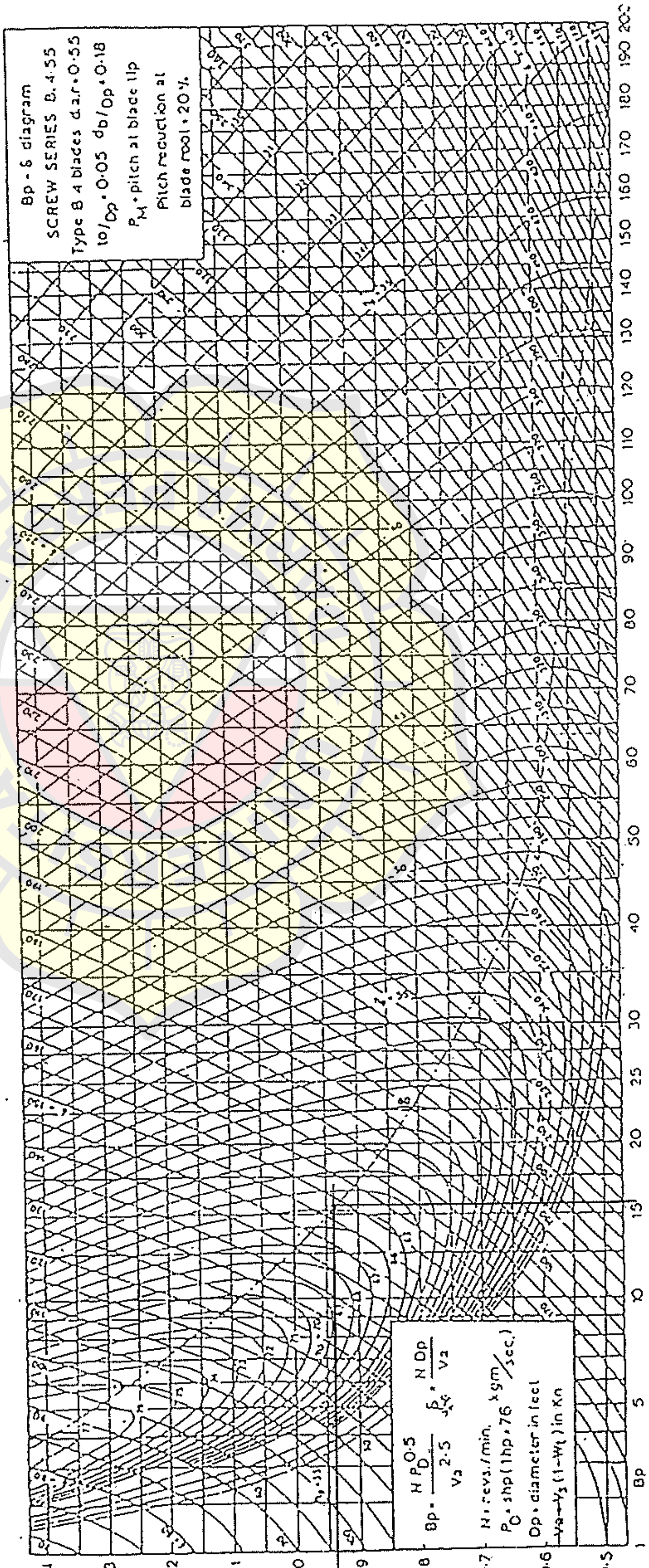
The correction may be positive as well as negative.

LCB

The C_R curves are intended to correspond to vessels with a longitudinal position of center of buoyancy (LCB) near to what is today considered the best possible position. The optimum LCB is a quantity that is in some doubt, and the available literature shows differences of opinion that make the picture rather confused. The dependence of ship

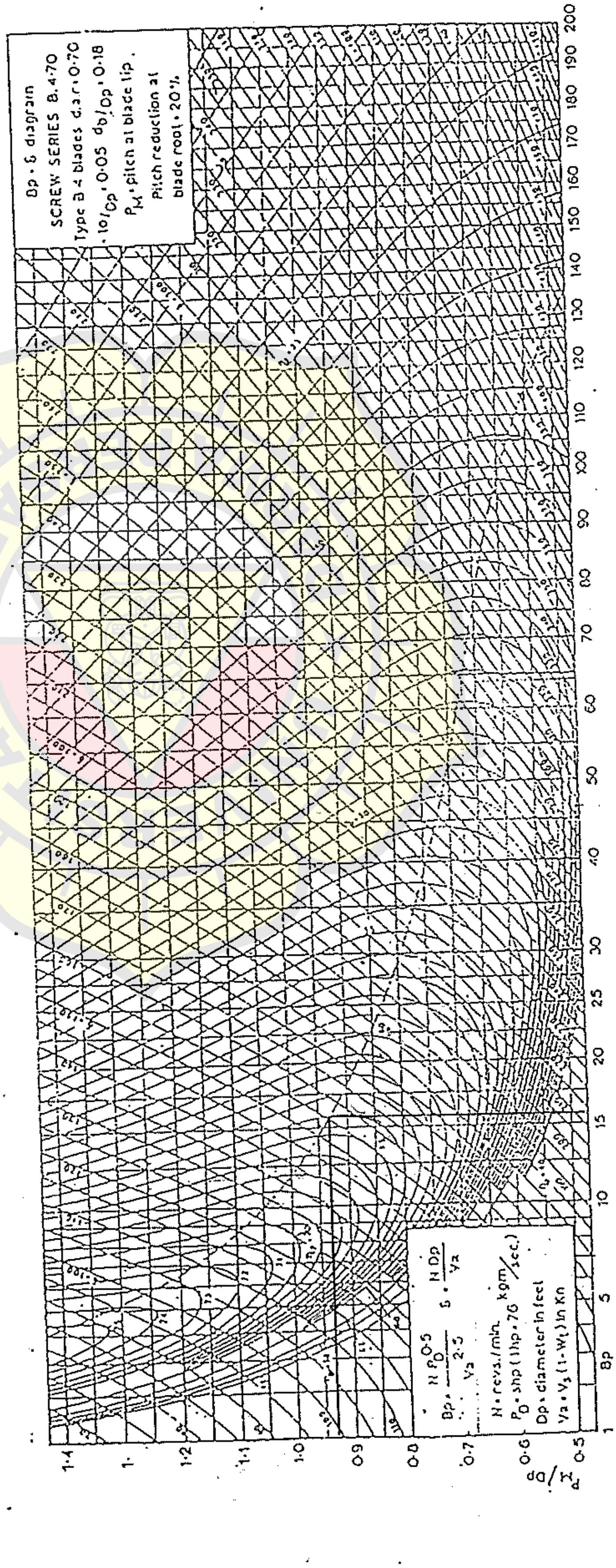
Lampiran 16. Diagram Bp δ Series B4-40





Lampiran 17. Diagram Bp δ Series B4-55

Lampiran 18. Diagram Bp δ Series B4-70



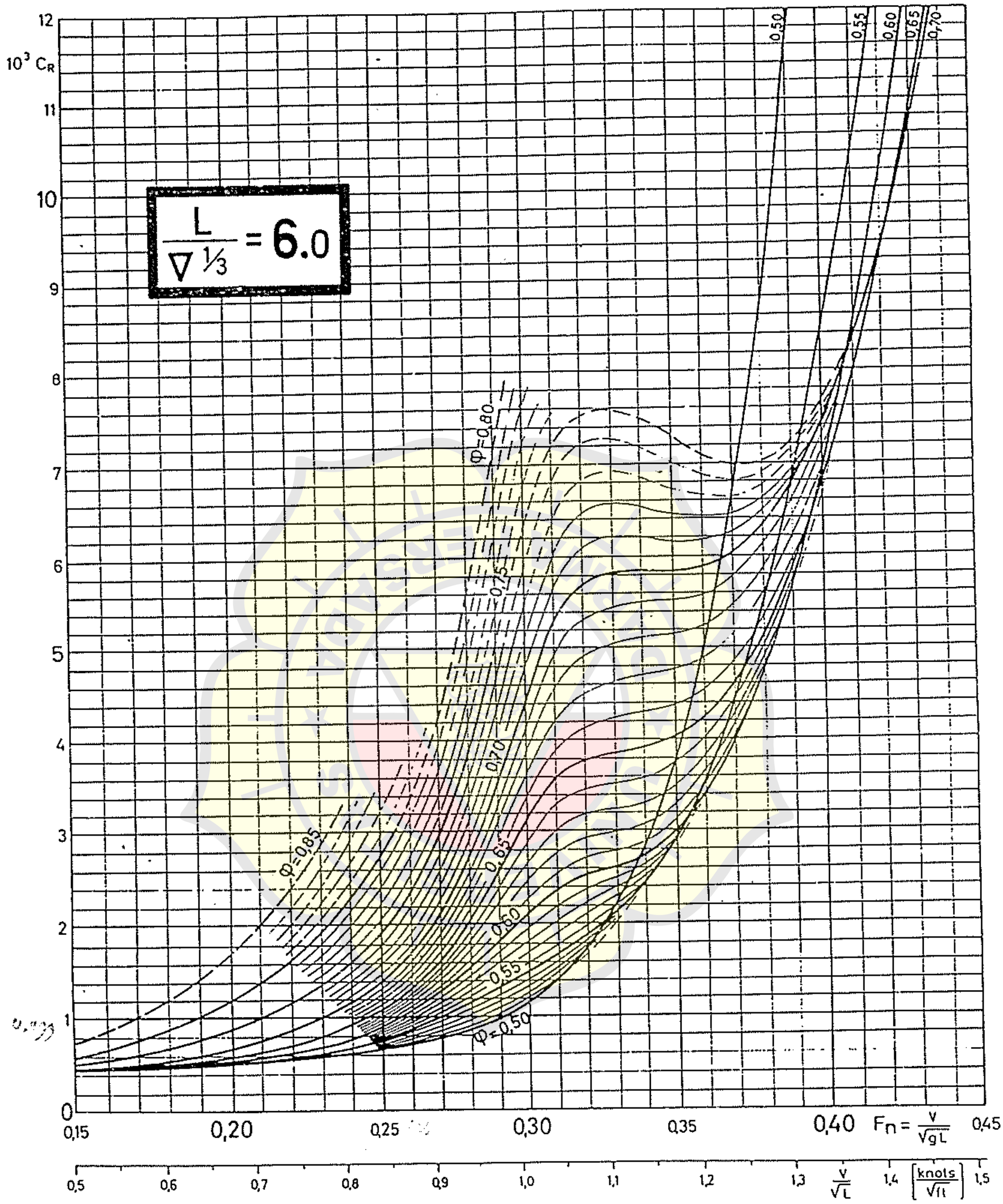


Figure 5.5.9. Residuary resistance coefficient versus speed–length ratio for different values of longitudinal prismatic coefficient. $L/\nabla^{1/3} = 6.0$.

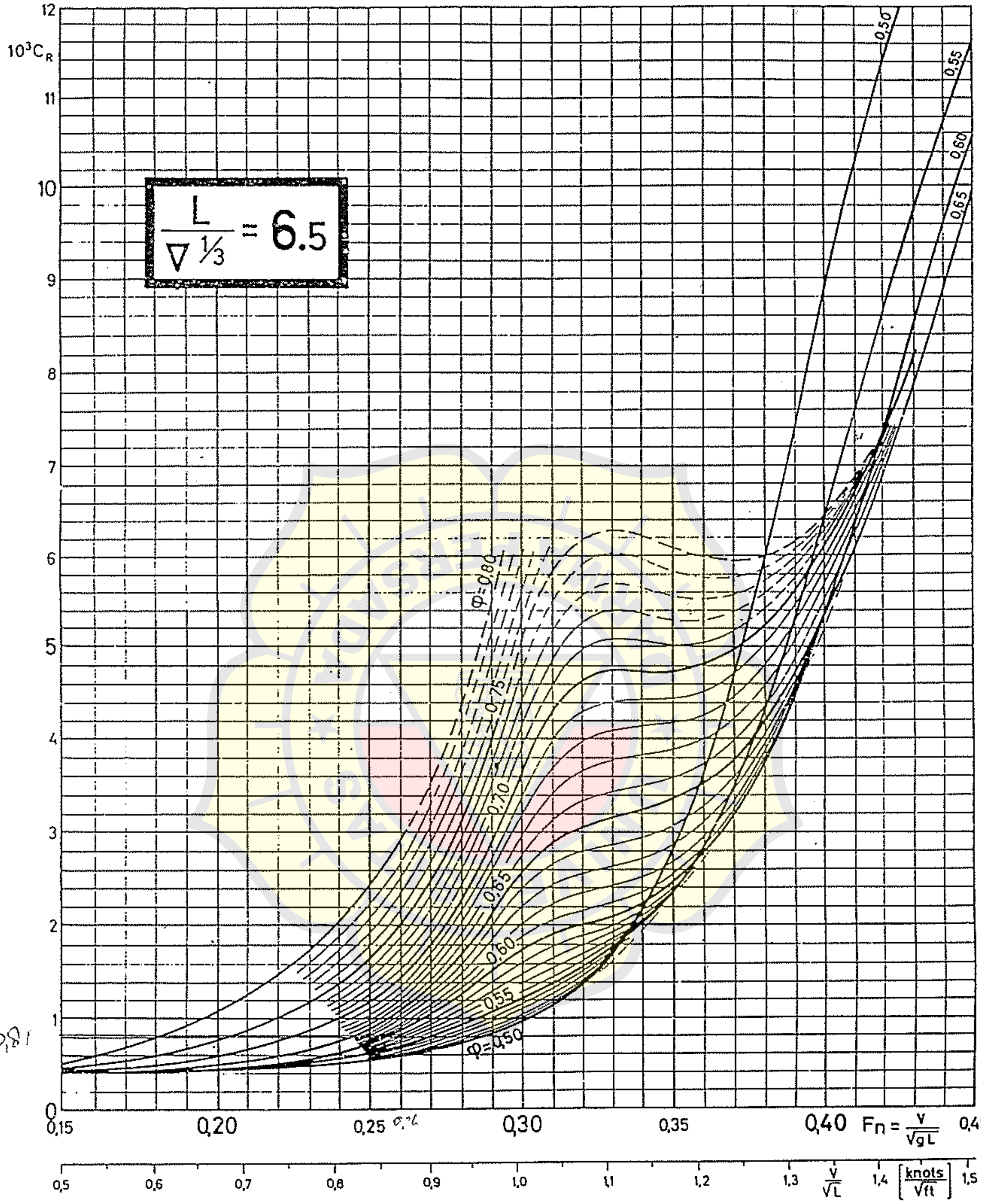


Figure 5.5.10. Residuary resistance coefficient versus speed–length ratio for different values of longitudinal prismatic coefficient. $L/\nabla^{1/3} = 6.5$.

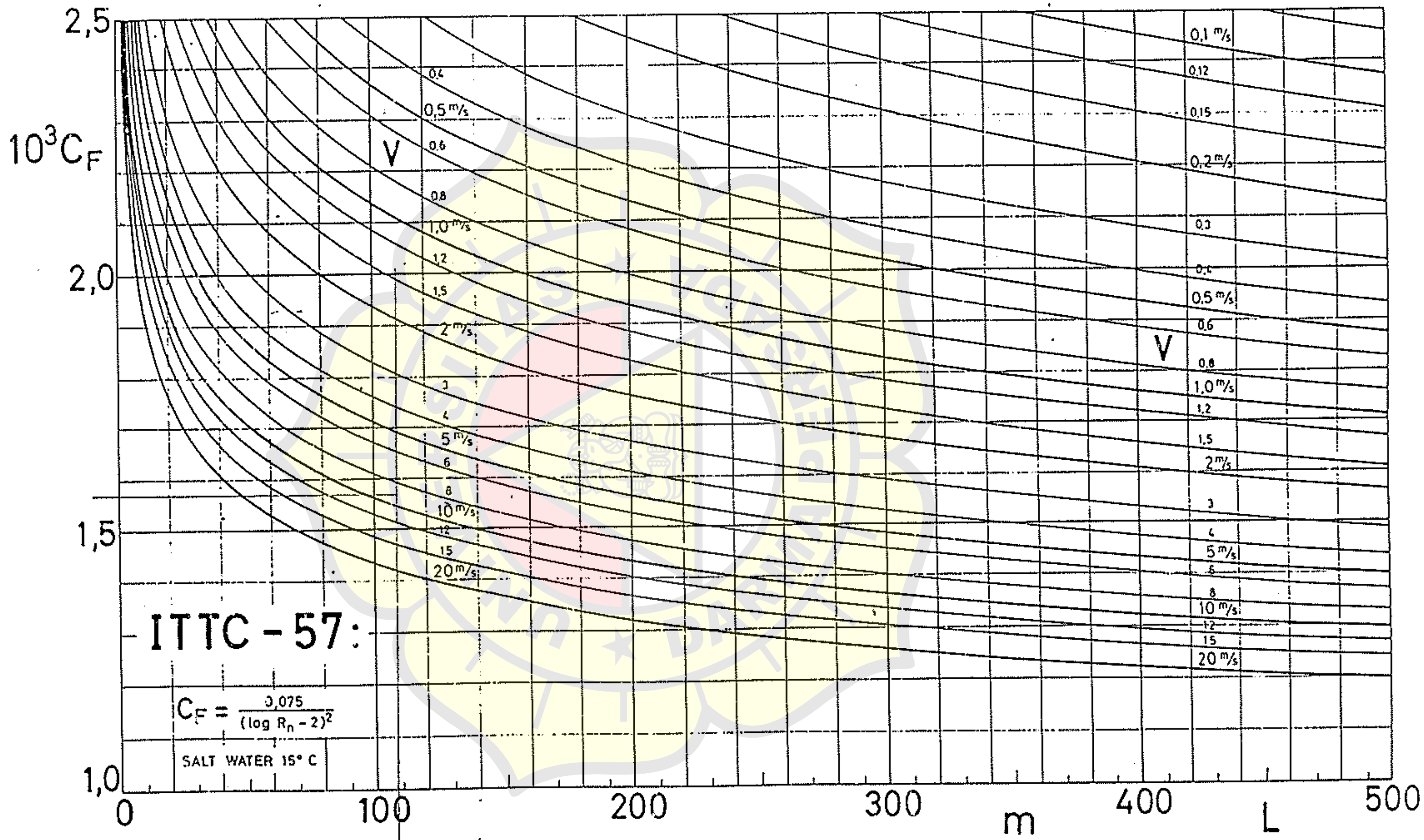


Figure 5.5.14. The frictional resistance coefficient C_F (according to ITTC 1957) as a function of ship length L and speed V.

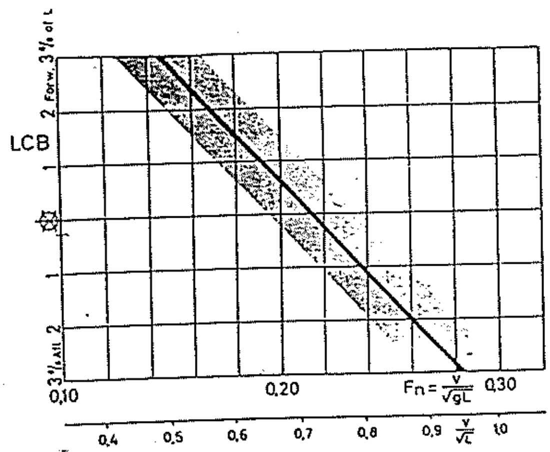


Figure 5.5.15. Standard LCB. The longitudinal position of the center of buoyancy that is considered the best possible.

resistance on LCB is, however, evident at higher speeds. In an attempt to make some order out of the confusion, the available information has been collected and condensed in the Fig. 5.5.15, which must be regarded as the standard LCB of the method.

The standard LCB has in this way been defined as a linear function on the Froude number F_n. As no safe dependency on other parameters have been recorded, the standard LCB is represented in the diagram by a single line, and the shaded area around this line illustrates the spread of the examined material.

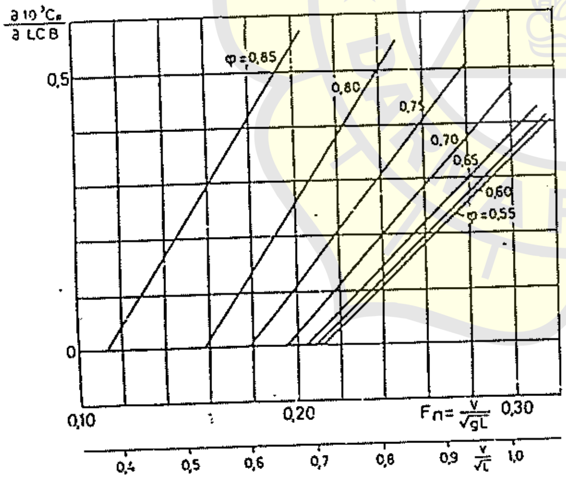


Figure 5.5.16. The correction of the residual resistance coefficient for LCB 1% forward of standard. The correction is thus $(\partial 10^3 C_R/\partial \mathrm{LCB})|\Delta \mathrm{LCB}|$, where ΔLCB is the longitudinal distance between actual and standard LCB in percent of L. There is no correction for LCB aft of standard. The correction is always positive.

As the standard position of LCB is, as mentioned earlier, assumed to give the smallest possible resistance, all other positions must in principle give resistances that are larger. The increase in resistance is to be found by multiplying the deviation of LCB from standard

$$\Delta \mathrm{LCB} = \mathrm{LCB} - \mathrm{LCB}_{\text{standard}} (\mathrm{LCB} \text{ in } \% \text{ of } L) \tag{5.5.18}$$

by a factor $\partial 10^3 C_R/\partial$LCB. The values of the factor may be obtained from the Fig. 5.5.16, which is valid for the case where LCB is forward of $\mathrm{LCB}_{\text{standard}}$. When LCB is aft of the $\mathrm{LCB}_{\text{standard}}$, the sources are very contradictory, and as the tendencies are very slight, no serious error will be introduced by neglecting the correction in such cases.

The corrected residual resistance coefficient for a ship with LCB forward of standard is consequently determined by:

$$10^3 C_R = 10^3 C_{R(\text{standard})} + \frac{\partial 10^3 C_R}{\partial \mathrm{LCB}} |\Delta \mathrm{LCB}| \tag{5.5.19}$$

The hull form dealt with in *Ship Resistance* is the hull form that was common for merchant ship types around 1960, that is, up to the time of publication of Guldhammer and Harvald (1974). This hull form has the aft perpendicularly placed in the axis of the rudder stock and the fore perpendicular in the fore end point of the design waterline. Since 1960 the hull forms have been developed further, and they have also become more varied, for instance, various bulbous bows have become widely used. The formulas given here for resistance calculation can be used for the modern and more varied bulb forms as well as for the traditional forms, provided the following more suitable definitions of L and LCB are used. The calculation length L is defined as the length between the fore and aft limits of the displacement, that is, the ultimate length of the submerged part of the hull, L_{OS} according to ITTC standard. For ships of traditional form with no bulb this length is exactly the waterline length.

LCB defines the longitudinal position of the center of buoyancy as the distance from this point to the midship section, positive aft of this section. The midship section is defined as the section at a distance of 48.5% of L from the fore limit of the displacement. L is the calculation length described above. The midship section thus defined is therefore the midpoint between the auxiliary perpendiculars $AP_1 - FP_1$; compare Fig. 5.5.17. $AP_1 - FP_1$ for a normal form will coincide with the perpendiculars defined in the usual way $AP - FP$.

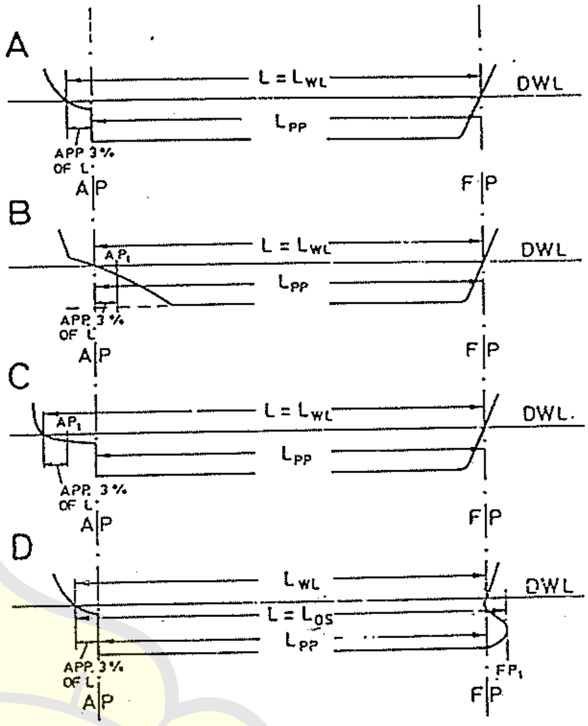


Figure 5.5.17. Definition of L and LCB. (a) Normal form. Length of the stern in the waterline is normally approximately 3% of L. (b) Hull with no sternpost. AP often placed in the endpoint of DWL. For LCB correction AP_1 3% of L before the endpoint of the waterline is used. (c) Hull with stern of extreme length. For the LCB correction AP_1 3% before the endpoint of the waterline is used. (d) Hull with bulbous bow. FP_1 is the fore limit of the displacement.

Hull Form (Shape of Sections and Bow)

As previously stated it is assumed that the resistance curve (deduced from Figs. 5.5.5–5.5.13) applies to a ship having a "standard" form, that is, the sections are neither distinctly U shaped nor V shaped. Therefore, in calculating the effective power of a preliminary ship design it should not normally be necessary to make a correction for shape of hull sections. If the sections are extremely U or V shaped, the 10^3C_R values may be corrected as follows: Corrections to 10^3C_R for shape of sections

Fore Body	Extreme U -0.1	Extreme V $+0.1$
After Body	Extreme U $+0.1$	Extreme V -0.1

(5.5.20)

These corrections cover the speed range $V/\sqrt{gL} = 0.20$–0.25. Furthermore, it must be considered that the "standard" form is a form with well-designed lines. If it is necessary to alter the lines due to the operating requirements of the ship, or allowance to the power must be made, it is recommended that C_R be increased by 10% and perhaps 20% or more for nonoptimal lines.

Concerning the bow, the standard form must be regarded as having an orthodox nonbulbous bow. For a vessel with bulbous bow having $A_{BT}/A_X \geqq 0.10$ (A_{BT} is the sectional area of the bulbous bow at the fore perpendicular and A_X is the area of the midship section) the following corrections to 10^3C_R are suggested:

With $A_{BT}/A_X = 0.10$ the bulbous bow is rather pronounced. For $0 < A_{BT}/A_X < 0.10$ the corrections are assumed to be proportional with size of bulb.

These corrections are valid for loaded conditions only. At ballast conditions the corrections due to bulbous bows will give an opposite picture. Full forms ($\varphi > 0.70$) will show a remarkable decrease in resistance, the corrections having two to three times these values, whereas the resistance for fine forms ($\varphi < 0.60$) generally will tend to increase.

$F_n = 0.15$	0.18	0.21	0.24	0.27	0.30	0.33	0.36	φ
		+0.2	0	−0.2	−0.4	−0.4	−0.4	0.50
		+0.2	0	−0.2	−0.3	−0.3		0.60
	+0.2	0	−0.2	−0.3	−0.3			0.70
+0.1	0	−0.2						0.80

(5.5.21)

Appendages

Rudders	No correction. The standard form is intended to include a rudder.	
Bilge keel	No correction.	(5.5.22)
Bossings	For full ships add 3–5% to C_R.	
Shaft brackets and shafts	For fine ships add 5–8% to C_R.	

Incremental Resistance

For many years it has been general practice to apply a correction to the C_{FS} for the ship, in order to include the effect of the roughness of the surface of the ship, which will never be "model-smooth" even when brand-new and freshly painted. This incremental resistance coefficient for model–ship correlation has very often been fixed at $C_A = 0.0004$. More recent experience has shown that this cannot be true in all cases. Therefore, the following correction for roughness and scale effect is proposed for the trial condition:

$$
\begin{aligned}
\text{For vessels with } L &\leqq 100 \text{ m}, & 10^3 C_A &= 0.4\\
&= 150 \text{ m} & &= 0.2\\
&= 200 \text{ m} & &= 0\\
&= 250 \text{ m} & &= -0.2\\
&\geqq 300 \text{ m} & &= -0.3
\end{aligned}
\tag{5.5.23}
$$

Some find the corrections given in Section 5.2.4 more suitable, that is,

Displacement			
1 000 t	$C_A =$	0.6×10^{-3}	
10 000 t		0.4×10^{-3}	(5.5.24)
100 000 t		0	
1 000 000 t		-0.6×10^{-3}	

It must be mentioned that these corrections of the frictional resistance coefficients are still rather uncertain.

Appendages

The correction of C_F for appendages is made by simply increasing C_F proportionally to the wetted surfaces of the appendages. Thus

$$C_{F'} = C_F \frac{S_1}{S} \tag{5.5.25}$$

where S is the wetted surface of the hull and S_1 is the wetted surface of the hull and appendages.

Air Resistance and Steering Resistance

The air resistance may be determined by use of data for the abovewater structure and the air. The magnitude of the air resistance is, however, very often of minor importance and the expenditure of effort in making an accurate calculation may not be justified. Therefore, in the absence of knowledge of the windage of a ship design it is suggested that $10^3 C_R$ be corrected by

$$10^3 C_{AA} = 0.07 \tag{5.5.26}$$

The correction for steering resistance may be about

$$10^3 C_{AS} = 0.04 \tag{5.5.27}$$

but may for course stable ships under favorable conditions be negligible.

It can be seen that both corrections are small and that for a preliminary design they may be assumed to be included in the incremental resistance.

The Service Condition

The resistance and the effective power calculated by use of the diagrams given here correspond to the values for a ship in the trial condition, that is, for ideal conditions as regards winds and waves, deep sheltered water, and smooth hull. For the mean service condition an extra allowance has to be made for the resistance and the effective power because of wind, sea, erosion, and fouling of the hull. This extra allowance is dependent on the shipping route. The following average service allowances (sometimes called sea margin or service margin) on the calculated resistance or effective power are proposed:

North Atlantic route, eastward, 15–20% in summer and winter, respectively
North Atlantic route, westward, 20–30% in summer and winter, respectively (5.5.28)
Pacific route, 15–30%
South Atlantic and Australian routes, 12–18%
East Asiatic route, 15–20%

The total resistance has to be calculated from

$$R_T = C_T(\tfrac{1}{2}\rho V^2 S) \qquad (5.5.29)$$

where S is the wetted surface of the hull.

Numerous methods for approximate determination of S exist. Use of one of the following two methods is recommended:

1. The publications FORMDATA I–V (Guldhammer, 1962, 1963, 1967, 1969, 1973) contain hydrostatic data for a comprehensive series of systematically varied ship forms. The wetted surface of these forms are mapped (volumes III–V) using the coefficient

$$\boxed{S} = \frac{S}{L(B + 2.5T)} \qquad (5.5.30)$$

If the actual form for the preliminary ship design largely coincides with one of the FORMDATA forms, an error of less than 1% in the determination of S will be obtained.

2. For normal merchant ship forms the wetted surface can be obtained from the following formula (a version of Mumford's formula):

$$S = 1.025L_{PP}(\delta_{PP}B + 1.7T) \qquad (5.5.31)$$

The FORMDATA $\boxed{S}$ diagrams and the preceding formula correspond to ship forms having a vertical stern and stem at the perpendiculars. Most ships will have a wetted surface corresponding to this assumption as the plus and minus areas will balance each other. For ships with a large underwater overhang or with large cutouts, these conditions ought to be allowed for in the calculations.

The calculations of the resistance and the effective power can be carried out as shown in Sample Form for the Calculation of Effective Power (see p. 132). The calculations can be performed using mini-computers. Many naval architects now have computer programs for such calculations.

In the design stage the main question to be settled is the type and size of the engine (e.g., number and dimensions of cylinders if diesel machinery). The determination of the resistance must be sufficiently exact so that, on the basis of effective power P_E, it is possible to determine the shaft power accurately enough to arrive at a safe solution to this vital question.

On the other hand, trying to attain greater accuracy than needed to solve this problem makes little sense. The uncertainty of the factors involved is considerable, and readers are warned against wasting time in attempting to squeeze the last ounce of accuracy out of a calculation that can only be an estimation.

In diesel-engined ships a change in the number of cylinders from, say, 6 to 7 or from 11 to 12 means that the power is changing by about 17% or 8%, respectively. By modifying the mean effective pressure and number of revolutions it is possible to vary the continuous output by about 10%.

Turbine manufacturers have corresponding steps between types.

On the basis of these considerations perhaps the required accuracy in the determination of P_E for a preliminary ship design can be fixed at 1 up to 5%. This accuracy will be easily obtained in many cases by using the diagrams and the calculation forms in this section.

The diagrams and the formulas can also be used in the following manner. Every time the naval architect has a result from his or her own towing experiments the results are pricked in on the diagrams. Then when making an estimation of the resistance for a proposal for a new ship, the naval architect uses his or her own data as basis material and uses the diagrams and the formulas in this section to correct the data. Often the results will be very good when using such a procedure.

Sekalipun masih banyak masalah yang belum dapat dipecahkan sepenuhnya mengenai pelaksanaan uji model di terowongan kavitasi, percobaan demikian itu dapat memberikan banyak informasi dan petunjuk mengenai berbagai pengaruh yang merusak dari kavitasi.

6.6.6. Kriteria untuk Pencegahan Kavitasi

Dalam menyiapkan proposal awal untuk kapal baru hal yang ingin diketahui oleh pihak arsitek kapal dalam tahap dini adalah ukuran utama dan karakteristik baling-baling. Baling-baling harus demikian rupa hingga tidak terjadi kavitasi yang merusak; karena itu, perlu adanya kriteria sederhana untuk memprakirakan terjadinya kavitasi. Kriteria demikian itu dapat didasarkan pada gaya dorong baling-baling rata-rata tiap satuan luas proyeksi permukaan daun dalam hubungannya dengan angka kavitasi, kadang-kadang angka kavitasi setempat. Burill (1943) memakai koefisien yang τ_c yang didefinisikan sebagai

$$\tau_c = \frac{T/A_p}{\frac{1}{2}\rho (V_R)^2} = \frac{T/A_p}{q_{0,7R}} \qquad (6.6.27)$$

T = gaya dorong baling-baling
A_p = luas proyeksi daun
ρ = massa jenis
V_R = kecepatan relatif air pada 0,7 jari-jari ujung R
$q_{0,7R}$ = tekanan dinamis pada 0,7 jari-jari ujung

Dalam diagram yang diberikan oleh Burill τ_c digambar berdasarkan angka kavitasi setempat pada 0,7 jari-jari :

$$\sigma_{0,7R} = \frac{p_0 - p_v}{q_{0,7R}} \qquad (6.6.28)$$

$p_0 - p_v$ = tekanan pada garis pusat baling-baling
p_0 = tekanan sekeliling yang absolut (absolute ambient pressure)
p_v = tekanan uap air

Tekanan absolut sekitar (sekeliling) nya pada garis pusat baling-baling adalah tekanan atmosfir ditambah dengan tekanan dari kolom air di atas poros baling-baling; ini berarti

$$p_0 = \text{atm} + \rho g(T - E + \zeta_A) \qquad (6.6.29)$$

ρ adalah massa jenis, g percepatan gravitasi, T sarat kapal, E tinggi letak poros dari garis dasar, dan ζ_A adalah amplitudo gelombang. ζ_A dapat dianggap sekitar $0,0075L$ atau dapat perkirakan dengan memakai diagram di Gb. 6.4.12 atau 6.4.13. L adalah panjang kapal.

Jika tekanan atmosfir sama dengan 101,3 kN/m² (atau kPa) (tekanan atmosfir standar pada permukaan laut) maka $p_0 - p_v$ pada 15°C menjadi

$$p_0 - p_v \approx 99,6 + 10,05(T - E + \zeta_A) \quad \text{(kPa)} \qquad (6.6.30)$$

p_v pada 15°C adalah sekitar 1,7 kPa. Variasi p_v terhadap suhu ditunjukkan di Gb. 6.6.7. Kurva tersebut dianggap berlaku baik untuk air tawar maupun untuk air laut.

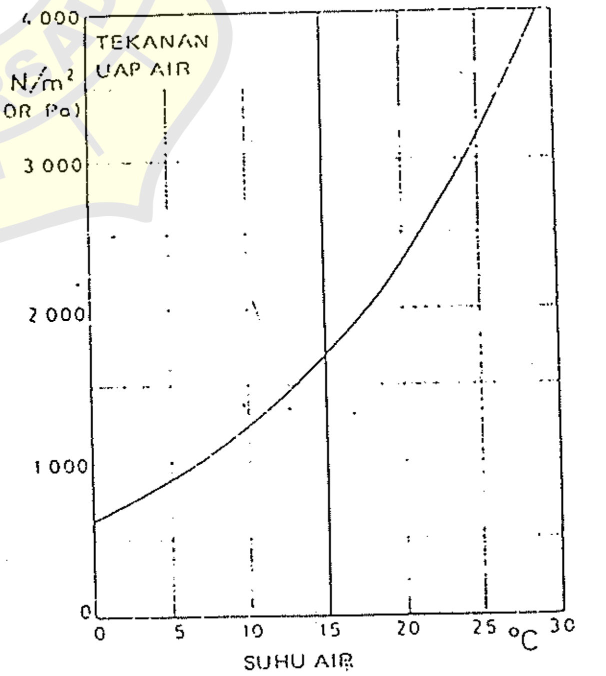
Kecepatan relatif air pada 0,7 jari-jari ujung adalah

$$V_R = \sqrt{V_A^2 + (0,7\,\pi D n)^2}$$

V_A = kecepatan maju baling-baling
D = garis-tengah baling-baling
n = laju kisaran

Luas proyeksi daun baling-baling A_p hampir sama dengan

$$A_p \approx A_D(1,067 - 0,229\,P/D) \qquad (6.6.31)$$



Gambar 6.6.7. Kurva tekanan uap air terhadap suhu.

$_D$ adalah luas kembang daun baling-baling; dalam erhitungan kasar luas ini dapat diganti dengan luas entang daun baling-baling A_E.

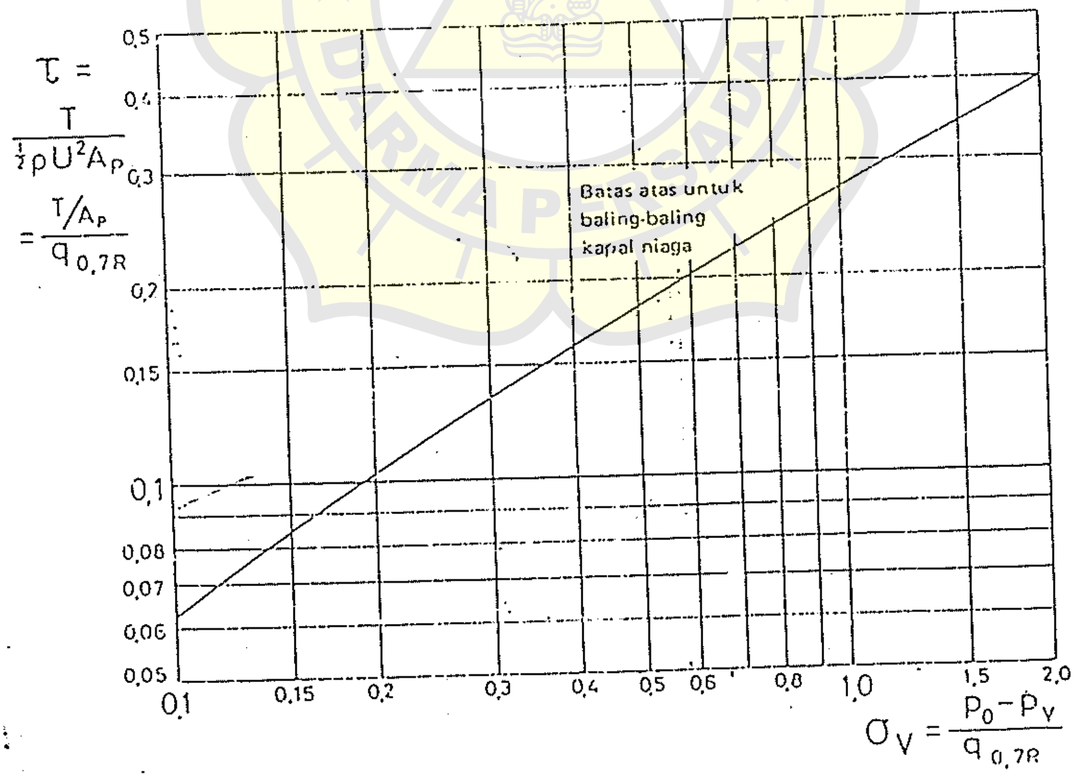
Gambar 6.6.8. menunjukkan salah satu kurva yang iajukan oleh Burrill (1943). Kurva tersebut merupa-an kurva "batas atas yang disarankan untuk baling-aling kapal niaga", yaitu berarti bahwa untuk ienghindari kavitasi yang berlebihan dan erosi dalam ondisi pelayaran rata-rata di laut maka baling-baling apal yang bersangkutan harus bekerja di bawah kurva ersebut.

Kriteria tersebut dapat pula dinyatakan dalam syarat ahwa luas bentang yang diperlukan harus tidak kurang ari

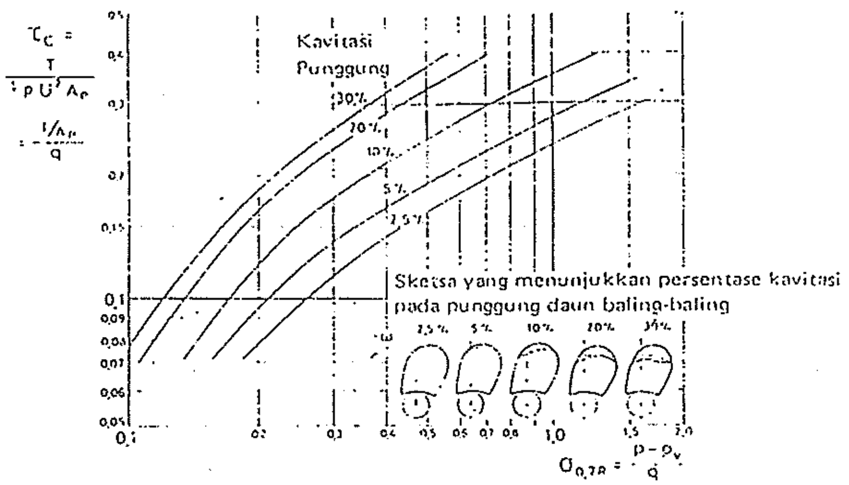
$$\left(\frac{A_E}{A_0}\right)_{nec} = \frac{T}{A_0(1{,}067 - 0{,}229P/D)(0{,}3\sigma_{0,7R}^{0,5} - 0{,}03)q_{0,7R}} \tag{6.6.32}$$

$_0$ adalah luas diskus baling-baling $(=\pi D^2/4)$. riteria ini sangat kasar. Van Manen memakai teori usaran untuk menghitung seri baling-baling berdaun dua, tiga, empat, dan lima dengan berbagai rasio luas daun dan dengan berbagai rasio langkah ulir. Hasilnya digambar dalam diagram (Manen, 1957b, Gb. 66 dan 67), yaitu seperti Gb. 6.6.8. Hasil tersebut menunjukkan ketergantungan kriteria kravitasi tersebut pada parameter tadi, terutama pada rasio langkah ulir.

Hasil yang diberikan di Gb. 6.6.9 adalah hasil dari baling-baling berdaun empat dengan rasio luas bentang 0,60 dari seri baling-baling kapal niaga yang diuji di suatu terowongan kavitasi (Burrill dan Emerson, 1962–1963) terhadap koefisien maju dan angka kavitasi dalam rentang yang luas. Dalam gambar tersebut diberikan garis untuk 2,5%, 5%, 10%, 20%, dan 30% kavitasi punggung yang timbul. Dari gambar tersebut terlihat bahwa garis batas atas untuk baling-baling kapal niaga yang ditunjukkan di Gb. 6.6.8. terletak sangat dekat dengan garis untuk 5% kavitasi punggung. Hasil pengamatan baling-baling menunjukkan bahwa jika baling-baling tersebut bekerja pada kondisi perancangan atau pada kondisi kerja yang sesuai dengan garis 5% maka baling-baling itu akan didapatkan dalam keadaan yang cukup bersih dan bebas erosi, barangkali bukan karena mengkilapkan permukaan logam tersebut setelah beberapa tahun bekerja. Karena satu dan lain hal mengusahakan agar mendapatkan luas daun yang sekecil mungkin lebih disukai daripada mendapatkan kelebihan luas daun yang besar.



Gambar 6.6.8. Diagram kavitasi (Burrill).



Gambar 6.6.9. Diagram kavitasi untuk seri model baling-baling berdaun empat untuk kapal niaga.

Untuk merancang baling-baling dengan memakai teori sirkulasi (lihat Bab 6, 6.7.5) perlu lebih dulu memilih garis tengah baling-baling, umumnya ditentukan dari diagram rancang (misalnya, Gb. 6.3.14). Karena itu untuk menghindari kavitasi diperlukan suatu kriteria yang agak umum dalam pemilihan luas daun. Diagram di Gb. 6.6.8 dapat dipakai sebagai pedoman demikian itu. Jika bentuk penampang daun telah diketahui maka distribusi tekanan di sekeliling penampang tersebut akan dapat dihitung (lihat Bab 2, 2.4 dan 2.6), atau mengukurnya di terowongan angin atau di terowongan air. Dengan memakai teori sirkulasi maka sudut insiden (angle of incidence) yang sebenarnya berikut pengurangan yang maksimum untuk tekanan pada punggung penampang dapat dicari. Tekanan yang dihitung tersebut kemudian dapat dibandingkan dengan tekanan statis $p_0 - p_v$ yang ada. Sudut insiden yang sebenarnya tergantung pada pola arus ikut di tempat bekerjanya baling-baling dan dalam satu kisaran baling-baling sudut tersebut akan berubah-ubah. Perhitungan tersebut harus dilakukan dengan memakai harga arus ikut mengeliling rata-rata pada setiap jari-jari tertentu. Dengan demikian maka kavitasi akan terjadi pada kisaran yang agak lebih rendah daripada yang dihitung, sehingga harus diberikan kelonggaran untuk itu. Sering bahwa setelah perhitungan selesai dilakukan kemudian dibuat model baling-balingnya dan dilakukan pengujian di terowongan kavitasi untuk memastikan tidak terjadinya pengaruh kavitasi yang merusak.

6.7. TEORI PERANCANGAN BALING-BALING

6.7.1. Pendahuluan

Telah banyak teori yang diajukan untuk menjelaskan cara sebuah baling-baling menghasilkan gaya dorong. Semua teori tersebut dikembangkan melalui pekerjaan yang sangat banyak, baik secara teoritis maupun memakai percobaan, yang dilakukan dalam cabang ilmu aerodinamika. Sekalipun demikian belum ada teori yang diajukan yang memperhitungkan semua faktor yang terlibat dalam aksi baling-baling. Selain itu, sekalipun konsep dari sebagian besar teori tersebut cukup sederhana matematiknya cukup rumit sehingga harus dipakai sejumlah anggapan tertentu untuk menyederhanakan masalahnya. Teori tersebut dapat diterapkan dalam praktek dengan memakai komputer, tetapi pemakaian teori yang akan diberikan berikut ini dan program komputer begitu saja tanpa memahaminya kadang-kadang dapat membuat malu yang besar. Karena itu, perancangan praktis baling-baling kapal yang cocok untuk kondisi yang diberikan masih sering tergantung pada hasil percobaan yang dilakukan secara sistematik dengan memakai model baling-baling; pemakaian pertimbangan yang baik merupakan hal yang hakiki. Di lain pihak, pengetahuan teoritis mengenai cara kerja baling-baling merupakan hal yang penting bagi pihak arsitek kapal untuk dapat menghasilkan rancang bangun baling-baling yang terbaik.

consideration all parasitic resistances in the boat's falls and in the various guide blocks through which the tackle fall runs to the winch heads.

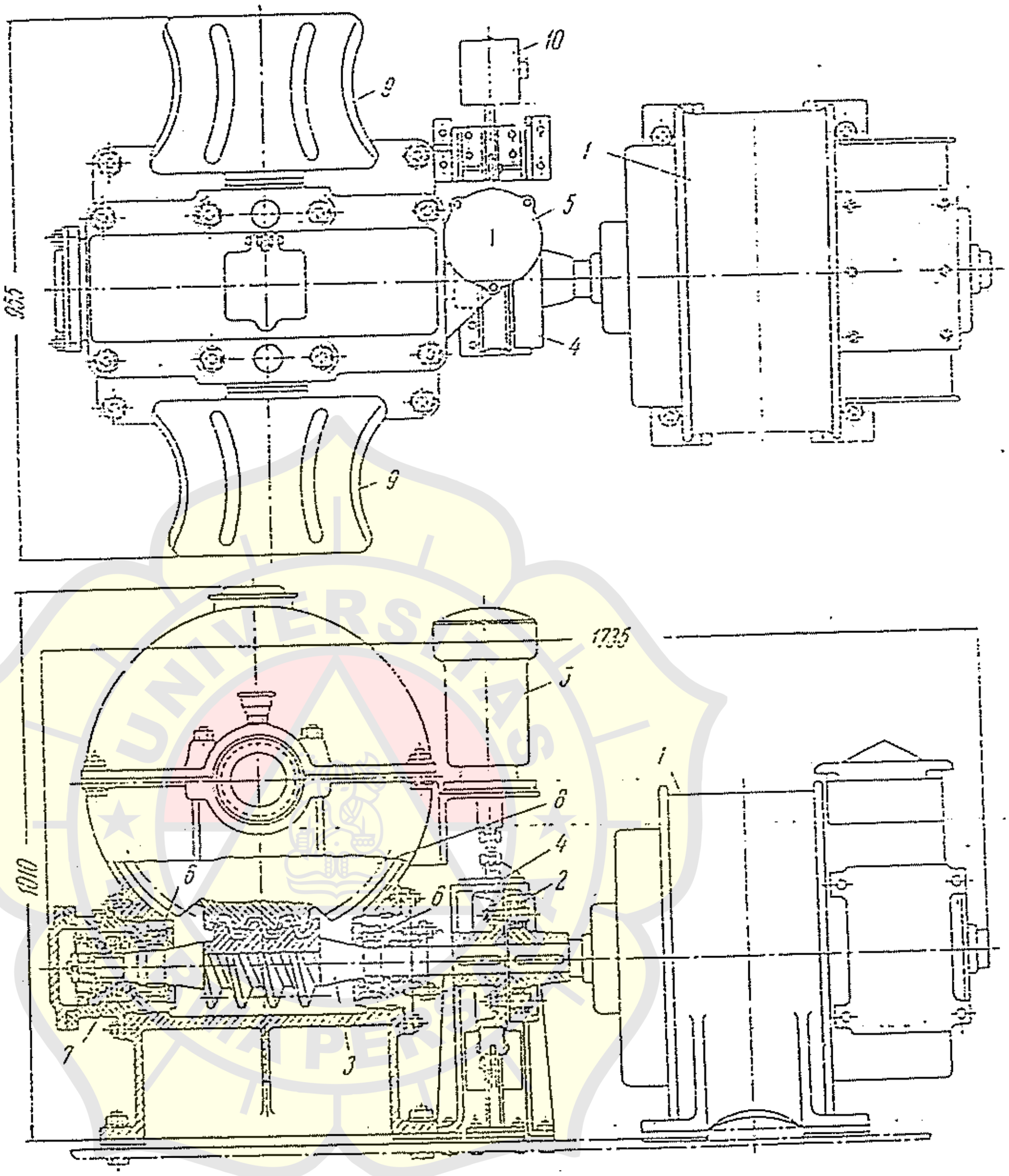


Fig. 184.

The design load Q acting on the falls of one davit is determined from the formula

$$Q = 0.5\,(Q_b + Q_p k_n) + Q_f$$

where Q_b = 570 to 2,175 = weight of the fully rigged boat, kg
Q_p = total weight of all persons allowed to embark (the weight of one person is approximately 75 kg; the number of persons in a boat may reach 78), kg
$Q_f = 0.05(Q_b + Q_p)$ = weight of the boat's falls, kg
k_n = 0.9 to 1.1 = coefficient of nonequal distribution of the movable load due to the weight of the persons in the boat.

The maximum tension of the fall at the winch head, after running over the maximum number of guide devices, is

$$T_{max} = \frac{0.5(Q_b + 1.1Q_p) + Q_f}{m\eta_f \eta_r \eta_s^a}$$

where m = total number of blocks in the boat's falls;

$$\eta_f = \frac{1}{m\varepsilon^m} \frac{\varepsilon^m - 1}{\varepsilon - 1} = \text{efficiency of the boat's falls}$$

ε = coefficient depending upon the ratio of the block diameter to the tackle fall diameter ($\varepsilon = 1.1$ for a hemp fall and $\varepsilon = 1.04$ to 1.06 for a steel wire rope)
η_r = 0.9 to 0.97 = efficiency of the davit guide roller
η_s = 0.9 to 0.97 = efficiency of the snatch-block
a = maximum number of blocks between the davit guide roller and the winch head.

The tension at the end of a rope that has run over the minimum number of blocks is

$$T_{min} = \frac{0.5(Q_b + 0.9Q_p) + Q_f}{m\eta_f \eta_r \eta_s^c}$$

where c = minimum number of blocks.

The diameter, d_f, of a hemp fall is selected according to the breaking strength $(T_{max} + T_{min})\,6 \leq R_{br}$ as a function of the boat length from Table 63 (U.S.S.R. Shipping Register).

Table 63

Boat length, m	Circumference of tackle fall, mm	Breaking strength, kg
8.25 to 9.14	95	6,100
7.62 to 8.25	89	5,400
7.35 to 7.62	83	4,600
6.72 to 7.35	73	3,900

The winch head diameter is

$$D_h = (5 \text{ to } 8)\, d_f$$

The speed, v_b, with which the boats are hoisted and lowered is assigned so that if the ship is rolling one of these operations can be carried out during the rolling period. This can be accomplished if the heaving-in speed is greater than the vertical component of the crest speed of waves running along the length of the ship. It has been established that the boat hoisting speed must be at least $v_b = 0.15$ m per sec under these conditions. The heaving-in speed of the tackle fall when single-sheave blocks are used must in this case be $v_f = 0.3$ m per sec.

The required winch head speed is found from the equation

$$\pi (D_h + d_f)\, n_h = 60 v_f$$

$$n_h = \frac{60 v_f}{\pi (D_h + d_f)} = 19.1 \frac{v_f}{D_h + d_f} \quad \text{rpm}$$

Assigning a motive unit speed (n_m = 500 to 1,600 rpm for electric motors and n_m = 200 to 380 rpm for steam engines), we can find the gearing ratio of the boat winch. Thus

$$i_{bw} = \frac{n_m}{n_h}$$

In designing nonreversible worm gearing the number of teeth on the worm wheel is taken in the range from 24 to 44. The pulling force on the winch head is equal to the sum of the pulling forces on the tackle falls:

$$T = T_{max} + T_{min}$$

Disregarding friction losses, the torque developed on the winch head shaft will be

$$M_h = \frac{T (D_h + d_f)}{2}$$

If the winch has an efficiency of η_{bw}, the torque and power on the motive unit shaft will be

$$M_{mb} = \frac{M_h}{\eta_{bw} i_{bw}} = \frac{T (D_h + d_f)}{2 \eta_{bw} i_{bw}}$$

and

$$N_e = \frac{M_{mb} n_m}{716.20} \quad \text{metric hp}$$

The cylinder diameter and indicated power of steam boat winches are determined from the same Posdyunin formulas used in determining these values for other steam-powered auxiliary machinery.

The required motor power rating for an electric winch can be calculated from the formula

$$N_e = \frac{(T_{max} + T_{min}) v_h}{75 \times 60 \eta_{bc}} \quad \text{metric hp}$$

6-5. Winch Operation

Before starting a winch it is necessary to make sure by inspection that it is in order and that no foreign objects hinder the moving parts of the winch. Then, warm up the steam supply line and the winch engine, apply lubricant to all friction surfaces, check the starting gear by turning it to the "hoist" and "lower" positions, check whether the speed-changing clutches engage and disengage properly and whether the braking gear is in order. After this, test the forward and reverse operation of the winch; if no knocking is heard and reversal is rapid and smooth, the winch is ready for regular operation. The winch is started and stopped and the shaft speed is changed in operation by opening and closing a stop valve or the starting valve. The winch is reversed either by shifting the links of the reversing gear or by operating the starting valve.

During winch operation it is necessary to see that lubricant is being properly fed, to check the temperature of the parts subjected to friction and to listen for knocking. As soon as abnormal noise is heard, stop the winch, find and eliminate the cause of the noise.

The load lowering speed should be regulated by applying the brake. Backsteam should be resorted to only when the load drops too fast even after applying the brake.

If hoisting operations are interrupted for short periods in winter, the winch should be run idle at low speed with open blow-off valves.

If the winch is not to be operated for a prolonged period in winter, it is necessary to drain the condensate from the cylinders and the live and exhaust steam lines.

Winch operation is prohibited if cracks are found in critical parts, if the motive mechanism, steam distribution or breaking gear is out of order or if the gears are excessively worn or some of the teeth are missing.

extreme left position. The relative positions of the main and pilot valves are shown in Fig. 26.

The arrangement of the main 2 and pilot 1 valves for various positions of the steam piston is shown schematically in Fig. 27.

The main piston valve is in the extreme left position, shown in Figs. 26 and 27a, because it is held there by live steam admitted

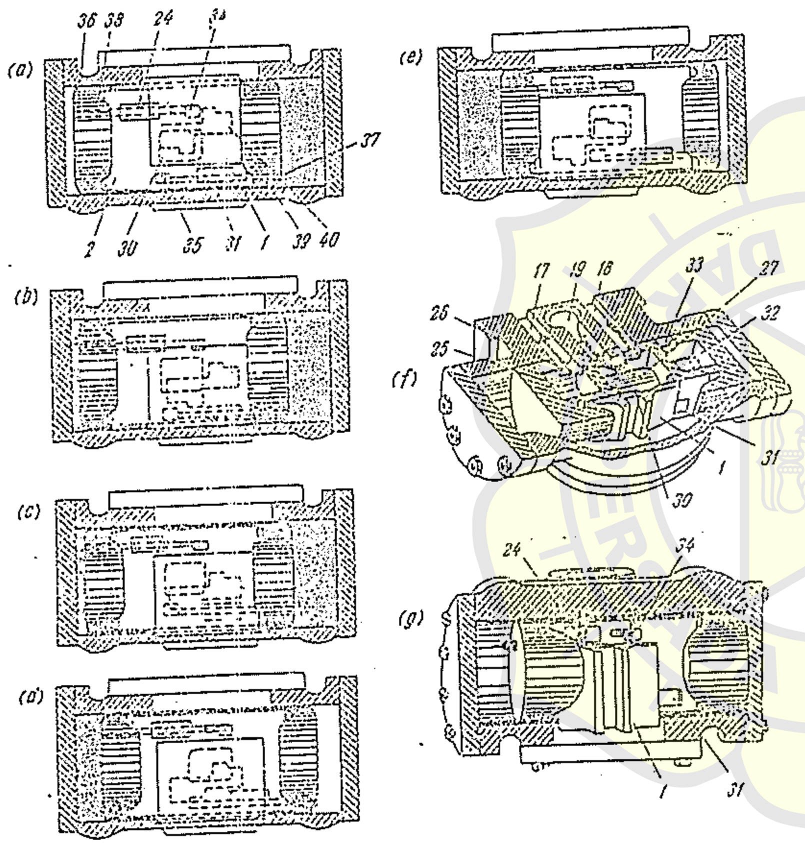


Fig. 27.

through port 35, passage 39 and port 37 into the right sleeve of the valve. Live steam is also admitted through ports 24 and 31. Steam passes through port 24, cavity 25 and port 26 where it is stopped by a blank wall (Fig. 26). The other port 31 admits steam through cavity 32, port 33 and passage 18 to the upper end of the steam cylinder, forcing the piston downwards. Spent steam is exhausted, at the same time, from the lower end through passage 17, port 28, cavity 27 and passage 19 to the exhaust line.

As the steam piston travels downwards it does not at first actuate the pilot valve. The latter begins to move when tappet 11 of rocker arm 7-4 runs up against nuts 13. This occurs when the piston has travelled about 0.6 of its stroke. At 0.75 of its stroke the piston puts the pilot in a position where it closes port 31. In this position ports 31 and 35 are closed so that live steam cut-off occurs. The main piston valve remains in its extreme left position while the piston continues its down stroke under the pressure of the expanding steam pulling rocker arm 7-4 and the pilot valve farther downwards.

In its downward movement the pilot valve opens port 34 (Fig. 27b) and live steam is admitted into the left sleeve of the main valve through passage 38 and port 36. The main valve will not shift to the right, however, until the pilot valve reaches the position shown in Fig. 27c, since the steam pressure is equal in the left and right sleeves. In the position of Fig. 27c recess 40 of the pilot valve connects port 35 to port 30 of middle cavity 27 which is connected to the steam exhaust line. At this, the steam pressure in the right sleeve drops and the main valve is shifted by the live steam to its extreme right position (Fig. 27d). Just before it reaches this position, port 35 moves away from recess 40 in the pilot valve, thereby cutting off steam exhaust from the right sleeve. The steam remaining in the sleeve is compressed by the moving main valve, forming a cushion and stopping the valve before it can strike the bottom of the sleeve (Fig. 27e).

As the main valve travels to the right (Fig. 27d) first passages 17 and 19 are cut off from cavity 27. This reduces steam exhaust from the lower end of the steam cylinder, the piston compresses the remaining steam and comes to a smooth stop. This compression of the steam continues until cavity 25 becomes connected to passage 17 through ports 24 and 26, thereby admitting live steam into the lower end of the steam cylinder (Fig. 27e, f and g). Since at this time the upper end of the cylinder is connected with the exhaust line through passage 18, ports 28 and 29 and the middle cavity 27, the piston of the steam cylinder starts its up stroke. If the pump is stopped in a position where the pilot valve closes the steam admission ports it cannot be subsequently started by merely turning on the steam. In such cases it will be necessary to turn the sleeves by their handles 21 so that shaped slots 22 and 23 on the sleeves register with ports 28 and 33.

Let us assume that the pump must be started with the pilot valve in the position shown in Fig. 26. This can be done if the right sleeve is turned so that its slot 23 registers with port 33 of the main valve. This admits live steam through slit 41 (Fig. 26) into the space maximum pressure, p_f kg per sq m, then the amount of liquid pumped is

$$V_p = V_e - V_f = D_1 \quad \text{cu m}$$

This equation can be solved for V_e and V_f:

$$V_e = V_f + D_1 = V_f + \frac{D}{6}$$

and

$$V_f = V_e - D_1 = V_e - \frac{D}{6}$$

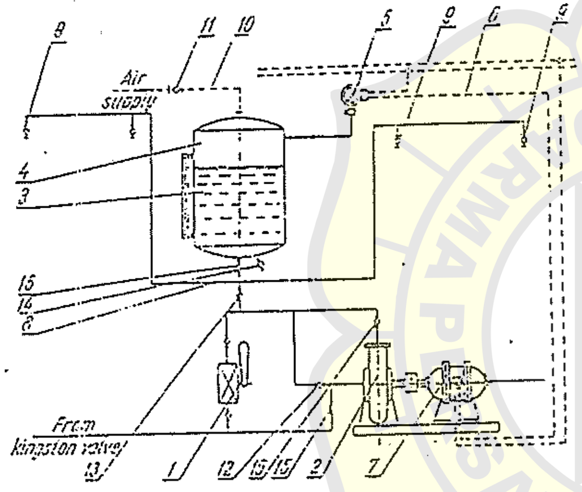


Fig. 189.

The equation of state for the air in the air cushion can be written as

$$V_e p_e = V_f p_f = \left(V_f + \frac{D}{6}\right) p_e = \left(V_e - \frac{D}{6}\right) p_f$$

Therefore the minimum and maximum volumes of the air are

$$V_f = \frac{Dp_e}{6(p_f - p_e)} \quad \text{and} \quad V_e = \frac{Dp_f}{6(p_f - p_e)}$$

Denoting by V_0 the volume of liquid remaining in the tank at the lowest level, we find that the volume of the pneumatic tank is

$$V_t = V_0 + V_e = V_0 + \frac{Dp_f}{6(p_f - p_e)}$$

Such tanks may also be used in the drinking and washing water systems.

(D) SANITARY AND SCUPPER SYSTEMS

The sanitary and scupper systems serve to remove water from the deck and also to dispose of used water from baths, laundries, refreshment bars, galleys, storerooms, etc. Water is drained from the decks through scuppers and their pipes which range from 50 to 100 mm in diameter.

The diagram in Fig. 190 shows how water is removed through scupper pipes *1* from the upper decks and compartment decks. From each deck water runs down to the next lower deck through scupper

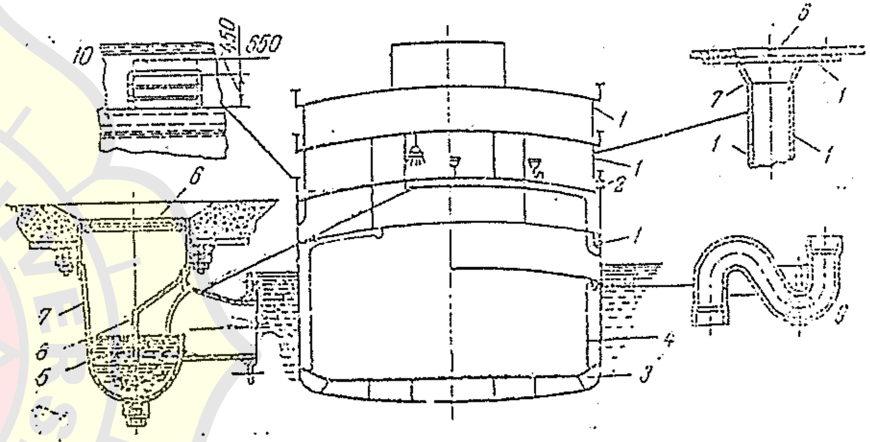


Fig. 190.

pipes until it reaches the last open deck above the load waterline from where it is discharged overboard through deck scuppers *2*. Large amounts of water drain from open decks through freeing ports *10* installed in the bulwarks.

Water is drained from decks located lower than the load waterline through scupper pipes *4* into bilge courses *3* or into dirty water tanks arranged in the double-bottom or side spaces from where it is discharged overboard by pumps.

Scuppers *7* with grates *6*, cowls *8* and sumps *5* avoid clogging of the scupper pipes. S-traps *9* are provided in scupper pipes which drain water from closed compartments to prevent the odour of the sewage spaces from getting into the compartments.

Shipside outlets of scupper pipes serving closed compartments are fitted with swing-check valves to exclude sea water in rough weather.

Sanitary pipelines made of galvanized pipe must be laid with a grade of at least 0.05 to ensure reliable water flow.

(1) provisioned boats of the proper type, size and quantity as stipulated by the Shipping Register;

(2) a device driven by hand and by machinery powered with motive units for rapidly and safely swinging out and lowering boats to set them afloat and for hoisting them aboard the ship;

(3) gear for stowing the boats in cruising order.

The cubic capacity of the boats depend upon their purposes and is calculated from the expression 0.6 LBH,

where L = maximum length of the boat at the height of the gunwale, m

B = maximum breadth of the boat as measured between the external surfaces, m

H = height of the boat as measured from the garboard strake at the upper false keel to the upper edge of the rubber at the middle of the boat's length, m.

Boats may be made of wood, plastics or metals. The latter are built in the same manner as steel ships or are pressworked as two halves which are then joined along the centre line.

Wooden boats are made of oak, ash, elm, larch, pine or chestnut. Metal and wooden boats are of about the same weight.

A disadvantage of steel boats is their rapid corrosion, of wooden boats—damage by woodworms.

Hand-operated machinery is usually installed to hoist boats weighing up to 510 kg.

Heavy boats, weighing about 510 kg and more, are hoisted by power-operated winches with nonreversible worm gearing.

Steam and electric boat winches have found the most extensive application. Steam winches are less economical than the electric type due to the steam condensation losses in the steam lines and steam engines. Moreover, it is necessary to warm up the steam engine before using the winch while electric winches are always ready for action. The steam engines of boat winches may be designed to operate either with or without steam expansion. In the former case the steam cut-off value ranges from 0.4 to 0.8.

Fig. 184 illustrates a boat winch powered by electric motor *1* whose shaft is linked to the shaft of worm *3* by the flexible coupling *2*. The half-coupling of the worm shaft has a brake drum on which brake band *4* is fitted. This band is linked through a lever system with the electromagnetic braking device *5*. The worm shaft runs in ball bearings *6* while axial forces are carried by the double ball thrust bearing *7*.

Worm *3* drives the worm wheel *8* on whose shaft identical winch heads *9* are mounted for winding the pendant of the boat's falls.

The required pulling force of the winch heads is the sum of the weights of the boats with rigging and the passengers, taking into

to 20 for a rope heaving-in speed of 0.4 to 0.8 m per sec. The engine is reversed by the starting valve. Load lifting capacities of ash hoisting winches range from 100 to 400 kg.

The construction of a steam-and-hand ash hoisting winch is shown in Fig. 183.

Worm *8* is mounted on the shaft *6* of a single-expansion two-cylinder vertical steam engine. This worm meshes with worm wheel *9* mounted on shaft *10* of the winch barrel *7*. The barrel is freely mounted on the shaft together with ring *11* which has a lug *12*. Bolt *13*, located between projections *14* of the barrel, passes through a slot in lug *12*. Ring *11* and ratchet wheel *15* are secured on the shaft by a single key *16*. Pawl *17* of ratchet wheel *15* is pivoted on a pin between lugs *29* of the barrel. This pawl is disengaged from the wheel and held in the disengaged position by hook *18* when the winch is to be driven by the steam engine. Flat spring *30* presses the pawl into engagement with the teeth of the ratchet wheel for the hand drive. Worm *19*, also keyed on shaft *10*, meshes with worm wheel *20* on vertical shaft *21* whose lower end is threaded and carries the nut *22*. Lever *24*, which actuates the differential valve *25*, is mounted on trunnions *28*. The engine is started by shifting the differential valve *25* from its central position, turning lever *24* about trunnions *28*.

The movement of the pistons is transmitted through piston rods *5* and connecting rods *4* to crank disk *26* and further through worm *8*, worm wheel *9*, shaft *10* and ring *11* to the barrel *7*. At the same time, worm *19* rotates vertical shaft *21* through worm wheel *20*.

Axial movement of shaft *21* is restricted so that upon its rotation, nut *22* will travel vertically. Its action, through trunnions *23*, on lever *24* returns the differential valve *25* to the central position. Here we have the same servomotor principle that is employed in steering gear.

If ashes are to be hoisted by hand, bolt *13* with its nut is shifted along the slot of lug *12* towards the centre to bring it out of engagement with the projections *14* of barrel *7*. The pawl *17* is placed into the position shown in the illustration and crank handle *27* is secured to the winch barrel. The winch is driven with this handle.

The same formulas are used in ash hoisting winch design as for other types of winches.

6-4. Boat Winches

Boat-handling gear is used for:

(1) communications between the ship and shore when the ship is riding at anchor in a roadstead;

(2) saving the passengers and crew in case of a shipwreck.

The boat-handling gear must include:

determining these values for other steam-powered and ... ery.

The required motor power rating for an electric winch can be calculated from the formula

$$N_e = \frac{(T_{max} + T_{min})\,v_b}{75 \times 60\,\eta_{bw}} \quad \text{metric hp}$$

6-5. Winch Operation

Before starting a winch it is necessary to make sure by inspection that it is in order and that no foreign objects hinder the moving parts of the winch. Then, warm up the steam supply line and the winch engine, apply lubricant to all friction surfaces, check the starting gear by turning it to the "hoist" and "lower" positions, check whether the speed-changing clutches engage and disengage properly and whether the braking gear is in order. After this, test the forward and reverse operation of the winch; if no knocking is heard and reversal is rapid and smooth, the winch is ready for regular operation. The winch is started and stopped and the shaft speed is changed in operation by opening and closing a stop valve or the starting valve. The winch is reversed either by shifting the links of the reversing gear or by operating the starting valve.

During winch operation it is necessary to see that lubricant is being properly fed, to check the temperature of the parts subjected to friction and to listen for knocking. As soon as abnormal noise is heard, stop the winch; find and eliminate the cause of the noise.

The load lowering speed should be regulated by applying the brake. Backsteam should be resorted to only when the load drops too fast even after applying the brake.

If hoisting operations are interrupted for short periods in winter, the winch should be run idle at low speed with open blow-off valves.

If the winch is not to be operated for a prolonged period in winter, it is necessary to drain the condensate from the cylinders and the live and exhaust steam lines.

Winch operation is prohibited if cracks are found in critical parts, if the motive mechanism, steam distribution or breaking gear is out of order or if the gears are excessively worn or some of the teeth are missing.

$$n_h \quad v_f \quad n_h \quad d_f$$

The speed, v_b, with which the boat ... is assigned so that if the ship is rolling one of these operations can be carried out during the rolling period. This can be accomplished if the heaving-in speed is greater than the vertical component of the crest speed of waves running along the length of the ship. It has been established that the boat hoisting speed must be at least $v_b = 0.15$ m per sec under these conditions. The heaving-in speed of the tackle fall when single-sheave blocks are used must in this case be $v_f = 0.3$ m per sec.

The required winch head speed is found from the equation

$$\pi (D_h + d_f)\, n_h = 60 v_f$$

$$n_h = \frac{60 v_f}{\pi (D_h + d_f)} = 19.1\,\frac{v_f}{D_h + d_f} \quad \text{rpm}$$

Assigning a motive unit speed ($n_m = 500$ to 1,600 rpm for electric motors and $n_m = 200$ to 380 rpm for steam engines), we can find the gearing ratio of the boat winch. Thus

$$i_{bw} = \frac{n_m}{n_h}$$

In designing nonreversible worm gearing the number of teeth on the worm wheel is taken in the range from 24 to 44. The pulling force on the winch head is equal to the sum of the pulling forces on the tackle falls:

$$T = T_{max} + T_{min}$$

Disregarding friction losses, the torque developed on the winch head shaft will be

$$M_h = \frac{T (D_h + d_f)}{2}$$

If the winch has an efficiency of η_{bw}, the torque and power on the motive unit shaft will be

$$M_{mb} = \frac{M_h}{\eta_{bw} i_{bw}} = \frac{T (D_h + d_f)}{2 \eta_{bw} i_{bw}}$$

and

$$N_e = \frac{M_{mb} n_m}{716.20} \quad \text{metric hp}$$

The cylinder diameter and indicated power of steam boat winches are determined from the same Posdyunin formulas used in difference in pressures in the chambers will cause the vanes to turn clockwise.

As soon as the helmsman stops turning the wheel the pressure in the system drops, valve *41* is returned to its central position by spring *44* and the rudder comes to rest.

In cases when the rudder is operated by emergency steering facilities (quadrants, rudder tackle, etc.), compression of the liquid in the chambers is prevented by opening the relief-bypass valve *33* by its spindle *45*.

The interaction of the parts of this steering gear for counter-clockwise rotation of the rudder can be followed out in Fig. 158.

4-4. Determining the Principal Data Required in the Design of Steam and Electric Steering Gears

The main initial data required to determine the principal dimensions of steering gears are the rudder characteristic, χ_r, the torque, M_{rs}, in kg-m developed on the rudder head and the time, τ, required to put over the rudder.

The time required to put the rudder from hard-over to hard-over, depending upon the purpose of the ship and used in steering gear design, is listed in Table 47. It should not exceed the standards established by the U. S. S. R. Shipping Register.

The time that elapses before the steering engine reaches its rated speed, which we shall call the starting time, must be taken into consideration by reducing the time τ for putting the rudder from hard-over to hard-over by 1.5 to 2 seconds.

If we denote the gearing ratio between the rudder stock and steering engine shaft as i_{sg}, the overall efficiency of the steering gear as η_{sg} and the speed at which the rudder stock turns,

Table 47

Type of ship	Time required to put rudder from hard-over to hard-over, sec	Speed of rudder movement, deg/sec, for rudder angle of $2\alpha^\circ = 70^\circ$	Speed of rudder movement, deg/sec, for rudder angle of $2\alpha^\circ = 64^\circ$
Ice breakers	15	4.66	4.25
Sea-going craft and transport ships	25 to 30	2.8 to 2.34	2.56 to 2.13
Towboats	20 to 25	3.5 to 2.8	3.2 to 2.56
River craft	40 to 45	1.75 to 1.55	1.6 to 1.44

expressed in rpm, as n_{rs}, then the torque developed on the steering engine shaft and its speed, n_m rpm, will be

$$M_m = \frac{M_{rs}}{i_{sg}\eta_{sg}} \text{ kg-m} \qquad (312)$$

$$n_m = i_{sg} n_{rs} \text{ rpm} \qquad (313)$$

where n_m = 100 to 350 rpm for steam engines
n_m = 300 to 1,800 rpm for electric motors.

The angular velocity of rotation ω_{rs} of the rudder stock can be calculated from the following formulas:

$$\omega_{rs} = \frac{\pi n_{rs}}{30} \text{ 1/sec} \qquad (314)$$

$$\omega_{rs} = \frac{2\alpha^\circ}{\tau} \frac{\pi}{180^\circ} \text{ 1/sec} \qquad (315)$$

where α° = maximum rudder angle from the middle-line plane.

It follows from formula (314) that

$$n_{rs} = \frac{30\omega_{rs}}{\pi} \text{ rpm} \qquad (316)$$

Combining equations (315) and (316) we obtain

$$n_{rs} = \frac{30}{\pi} \frac{2\alpha^\circ}{\tau} \frac{\pi}{180} = \frac{1}{3} \frac{\alpha^\circ}{\tau} \text{ rpm} \qquad (317)$$

Combining equations (313) and (317) we can write

$$i_{sg} = \frac{n_m}{n_{rs}} = \frac{n_m}{\frac{1}{3}\frac{\alpha^\circ}{\tau}} = 3n_m \frac{\tau}{\alpha^\circ} \qquad (318)$$

Taking equations (314) and (315) into consideration, the power developed on the rudder stock is

$$N_{rs} = \frac{M_{rs}\omega_{rs}}{75} = \frac{M_{rs}}{75} \frac{2\alpha^\circ}{\tau} \frac{\pi}{180^\circ} = 4.65 \frac{M_{rs}}{10^4} \frac{\alpha^\circ}{\tau} \text{ metric hp} \qquad (319)$$

$$N_{rs} = \frac{M_{rs}\omega_{rs}}{75} = \frac{M_{rs}}{75} \frac{\pi n_{rs}}{30} = 1.395 \frac{M_{rs} n_{rs}}{10^3} \approx 1.4 \frac{M_{rs} n_{rs}}{10^3} \text{ metric hp} \qquad (320)$$

The shaft horse power of the steering engine motive unit will be

$$N_m = \frac{N_{rs}}{\eta_{sg}} = 4.65 \frac{M_{rs}}{10^4 \eta_{sg}} \frac{\alpha^\circ}{\tau} \text{ metric hp} \qquad (321)$$

$$N_m = \frac{N_{rs}}{\eta_{sg}} = 1.4 \frac{M_{rs}}{10^3 \eta_{sg}} n_{rs} \text{ metric hp} \qquad (322)$$

The shaft horse power can also be determined from the shaft torque

The power required to drive a fan is found from the formula

$$N_m = \frac{Q_a H}{75\eta_f 3{,}600} \text{ hp}$$

The overall efficiency of a fan is made up of the following efficiencies:

1. Hydraulic efficiency, which takes into consideration the loss of head in the fan

$$\eta_h = \frac{H}{H+\Delta H} = \frac{H}{H_t} = 0.7 \text{ to } 0.85$$

where ΔH = loss of head in the fan.

2. Hydraulic friction efficiency which takes into account the losses due to the friction of the impeller shrouds against the fluid being transferred

$$\eta_{fr} = \frac{N_{fr}}{N_a} = \frac{\beta 10^{-6}\rho D_2^2 u_2^3}{N_a}$$

where N_{fr} = power lost in overcoming fluid friction

$\beta = (5 \text{ to } 15)\left(1+5\frac{b_2}{D_2}\right)$ = coefficient obtained from data compiled by the Central Institute of Aero- and Hydrodynamics

b_2 = width of the impeller at air outlet

D_2 = impeller diameter at air outlet

For backward-curved vanes—$\eta_{fr} \cong 0.6$ to 0.75

For forward-curved vanes—$\eta_r \cong 0.75$ to 0.9.

3. Mechanical efficiency which takes into account the losses due to mechanical friction

$$\eta_m = \frac{N_a - \Delta N_{mf}}{N_a} \approx 0.95 \text{ to } 0.99$$

where ΔN_{mf} = power lost in overcoming mechanical friction. The overall efficiency of a fan is thus

$$\eta_f = \eta_h \eta_{fr} \eta_m = 0.4 \text{ to } 0.75 \qquad (279)$$

The overall efficiency of an axial fan may reach $\eta_f \approx 0.84$.

2-2. Design and Selection of Fans

Strictly aerodynamical calculations in fan design do not, as a rule, ensure results in subsequent tests that comply with the initial design data.

More accurate results may be achieved by designing a fan similar to one which has already been built, tested and modified to obtain the most favourable aerodynamic and design features.

This type of fan design is carried out by the similarity method using aerodynamic diagrams and dimensionless characteristics which we will consider in the following.

The initial data for fan design comprise: the total head, H, consisting of the static, H_{st}, and dynamic, H_{dyn}, heads, capacity, Q_s, and the rotational speed, n, at maximum efficiency. Thus

$$H = H_{st} + H_{dyn} = H_{st} + \frac{v^2}{2g} \times 10^{-3} \text{ mmH}_2\text{O} \qquad (280)$$

where v = mean velocity in the discharge connection of the fan. On the basis of the discharge per second, Q_s, head, H, and speed, n, we next determine the specific velocity of the fan.

The specific velocity of a fan is a value that relates the air discharge, Q_a, cu m per sec, the total head, H mmH$_2$O, and the impeller speed, n, at maximum efficiency:

$$n_s = \frac{n\sqrt{Q_a}}{\sqrt[4]{H^3}} \qquad (281)$$

It is evident that the ratio of the capacities of a series of geometrically similar fans of identical design can be expressed by the dimensionless discharge coefficient $\overline{Q}_k$. Therefore

$$\overline{Q}_k = \frac{Q_s}{Fu_2}$$

and

$$Q_s = \overline{Q}_k F u_2 = \overline{Q}_k \frac{\pi D_2^2}{4} u_2 \text{ cu m per sec}$$

where F = area of the impeller, sq m.

D_2 = outside diameter of the impeller, m.

The peripheral speed at the outlet circumference of the impeller is found from the formula

$$u_2 = \frac{\pi D_2 n}{60} \text{ m per sec}$$

The pressure developed by a series of geometrically similar fans can be characterized by the pressure coefficient, $\overline{H}_k$:

$$\overline{H}_k = \frac{H}{u_2^2\rho} \text{ — for the total head, and}$$

$$\overline{H}_{kst} = \frac{H_{st}}{u_2^2\rho} \text{ — for the static head.}$$

Whence, if we know $\overline{H}_k$ from the characteristics of pilot models, we can determine

$$H = \overline{H}_k u_2^2 \rho \text{ mmH}_2\text{O}$$

$$H_{st} = \overline{H}_{kst} u_2^2 \rho \text{ mmH}_2\text{O} \qquad (282)$$

5-3. Determining the Principal Dimensions of Anchoring and Warping Machinery

The initial data used to determine the principal dimensions of anchoring machinery are the required pull of the cable lifter and the speed at which the anchor is weighed from the anchorage depth, which is equal to the distance from the hawse hole to the bottom.

It is advisable to determine the pull on the cable lifter so as to ensure that one anchor will be brought in at a speed of at least 12 m per min from the anchorage depth which is taken equal to:

80 m if each anchor weighs 1,000 kg or less

90 m if the anchor weighs from 1,500 to 3,000 kg

100 m if the anchor weighs from 3,000 to 6,000 kg.

The following notation will be used to derive the formulas for determining the pull on the cable lifter:

G_a = weight of the anchor, kg

p_a = weight per running metre of the chain cable, kg

L_a = length of the suspended cable, m

γ_a = 7,750 = density of the material of the anchor, kg per cu m

γ_w = 1,025 = density of sea water, kg per cu m

f_h = 1.28 to 1.35 = a factor taking into account the friction losses in the hawse hole and stopper.

The required pull of the cable lifter to hoist two anchors is

$$T_{cl} = 2f_h(G_a + p_aL_a)\left(1 - \frac{\gamma_w}{\gamma_a}\right) = 2\times 1.35\,(G_a + p_aL_a)\left(1 - \frac{1,025}{7,750}\right) = 2.35\,(G_a + p_aL_a)\ \ \text{kg} \tag{383}$$

In hoisting one anchor*

$$T_{cl} = 1.175\,(G_a + p_aL_a)\ \ \text{kg}$$

The following empirical formulas can be derived from a comparison of the weights of anchors and the size of their chains as stipulated by the U.S.S.R. Shipping Register, as well as the U.S.S.R. Standard on anchor chain:

'The chain bar size $d_c \approx \sqrt{G_a}$ mm. The weight per running metre of anchor chain is

$$\left.\begin{aligned}&(a)\ p_{ao} = 0.023\,d_c^2\ \ \text{kg for open-link chain}\\ &(b)\ p_{as} = 0.0218\,d_c^2\ \ \text{kg for stud-link chain}\end{aligned}\right\} \tag{384}$$

According to the U.S.S.R. Shipping Register the all anchoring arrangement, usually consisting of a capstan, must break away the anchor and heave it in at a speed of at least 9 m per min.'

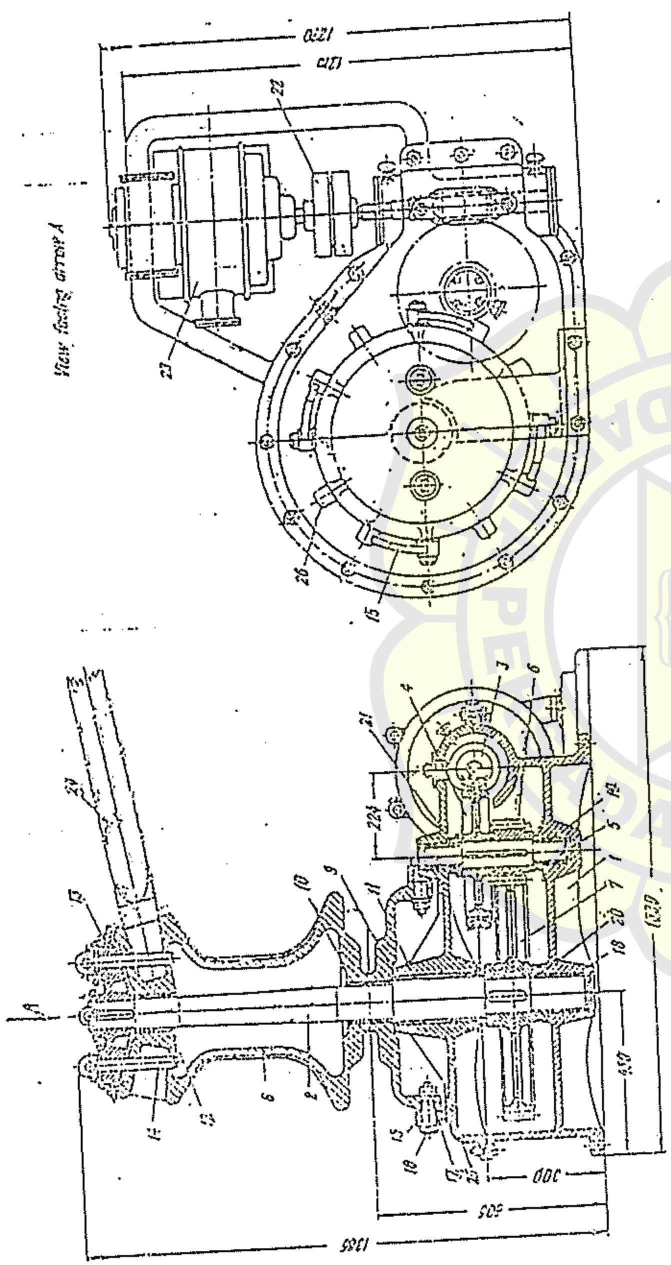


Fig. 169

* In breaking away one anchor from the bottom

$$T_{cl} = 2G_a + 1.175\,(G_a + p_aL_a)\ \ \text{kg}$$

If a windlass serves both for handling the anchor and for warping, the pull of the warp ends must not exceed

$$T_w = \frac{R_{br}}{6} \qquad (385)$$

where R_{br} = breaking strength of the warping hawser.

The speed at which a capstan barrel heaves in a warping hawser can be taken from Table 58 which has been compiled from the manufacturing specifications for capstans worked out by the Central Marine Research Institute of the U.S.S.R.

Table 58

Pull of the capstan barrel, kg	Hawser heaving-in speed, m per sec	Useful power, kg-m/sec
1,200	0.3	360
3,000	0.25	750
4,500	0.2	900
7,000	0.167	1,165
12,000	0.150	1,800

The speed at which a warping hawser is heaved in by a windlass is not limited by the values in Table 58, and usually is equal to about 0.4 m per sec.

It has been stated previously that the same machinery is employed both for anchoring and warping purposes. It follows that windlasses and capstans must be designed so as to ensure normal operation of both the anchoring and warping arrangements.

As a rule, anchoring and warping capstans and windlasses are designed to ensure the proper operation of the anchoring arrangement, and then a check is made to see whether they provide for the required pull and heaving-in speed of the warping hawsers.

The number of anchors, their weight, the size of the anchor chain cables, the circumference of warping hawsers and towing ropes, and their length are determined from the tables of the pertinent regulations of the Shipping Register. To find these values it is necessary to calculate the rigging characteristic of the anchoring and warping arrangement:

$$X = L(B + H) + \Sigma \chi_i \qquad (386)$$

where L = length of the ship at the summer load line, m
B = maximum breadth between the outer edges of the ship's hull, m
... edge of the keel to the lower edge of the sheer strake stringer, m
$\Sigma \chi_i$ = correction factor taking into account the sail effect of the superstructures.

Correction factors for the sail effect of the superstructures having a height h_i and length l_i consist of:

(a) correction factors for the superstructures of the forecastle, poop and midships, each having a length l_{sp} and height h_{sp}:

$$\chi_{sp} = k_{1sp} \Sigma l_{sp} h_{sp}$$

where k_{1sp} = 0.75 if the total length of the superstructures is equal to or less than 0.5 L

$k_{1sp} = 1.5 \frac{l_{sp}}{L}$ if the total length of the superstructures exceeds 0.5 L

(b) correction factors for the deck houses, each having a height h_{dh} and length l_{dh}:

$$\chi_{dh} = k_{dh} \Sigma l_{dh} h_{dh}$$

where k_{dh} = 0.5 if the deck house has a length l_{dh} equal to or less than 0.5 L

$k_{dh} = \frac{l_{dh}}{L}$ if the length, l_{dh}, of the deck house exceeds 0.5 L.

If the breadth, b_{dh}, of the deck house exceeds its length, l_{dh}, then the product $b_{dh} h_{dh}$ is substituted into the equation in place of $l_{dh} h_{dh}$. Thus

$$\chi_{dh} = k_{dh} \Sigma b_{dh} h_{dh}$$

(c) correction factor for the quarter deck having a length l_q and height h_q:

$$\chi_q = l_q h_q$$

Data on the anchoring and warping arrangements are listed in Tables 59 and 60. The weight of each anchor is found by dividing their total weight by the number of anchors. The separate anchors may be lighter than the specified values by 7.5 per cent. The lengths of the anchor chain cables are given in the table on the assumption that the average length of each shot is 25 m. The cable length does not include the lengths of the chain slip, joining shackles, connecting links and short pieces of shots with swivels. If the tabular cable length comprises an odd number of shots, then the length of the starboard anchor chain cable is taken one shot longer than the port cable.

A section taken through the central plane of the usual five-snug cable lifter (Fig. 170) perpendicular to the shaft will be a regular

(a) for windlasses and capstans of bower anchors:

$$n_{cl} = \frac{60\,v_a}{0.04\,d_c} = \frac{60\times0.2}{0.04\,d_c} = \frac{300}{d_c} \quad \text{rpm}$$

(b) for the stern anchoring capstan:

$$n_{cl} = \frac{9}{0.04\,d_c} = \frac{225}{d_c} \quad \text{rpm}$$

The efficiency of the anchoring arrangement is $\eta_a = 0.7$ to 0.85 for mechanisms with spur gearing and $\eta_a = 0.65$ to 0.75 for mechanisms

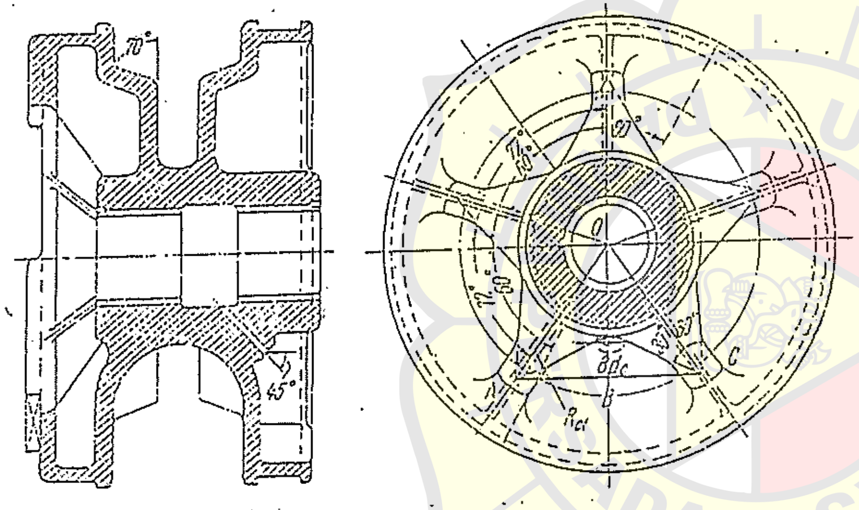


Fig. 170.

with worm gearing. It is the product of the efficiencies of the transmissions and shaft bearings in the gear train (Fig. 171):

$$\eta_a = \eta_{cl}\eta_{sh}^{a}\eta_{ps}^{c}\eta_{wg}$$

where η_{cl}, η_{sh}, η_{ps}, η_{wg} = efficiencies of the cable lifter, shaft bearings, pairs of spur gears and worm gearing

a and c = number of shaft bearings and pairs of spur gears.

The torque on the cable lifter is

$$M_{cl} = \frac{T_{cl}D_{cl}}{2\eta_{cl}} \quad \text{kg-m}$$

where $\eta_{cl} = 0.9$ to 0.92 = efficiency of the cable lifter.

Denoting the engine shaft speed as n_m, the gearing ratio of the mechanism (Table 61) is

$$i_a = \frac{n_m}{n_{cl}}$$

Table 61

Anchor handling gear	Motive unit shaft speed n_m, rpm	Gearing ratio of mechanism, i_a
Hand-operated capstans		4 to 40
Steam capstans	180 to 320	18 to 60
Electric capstans	800 to 1450	110 to 200
Hand-powered windlasses		9 to 15
Steam windlasses	90 to 270	6 to 30
Electric windlasses	720 to 1560	105 to [illegible]

The torque developed on the shaft of the motive unit is

$$M_m = \frac{M_{cl}}{i_a\eta_a} \quad \text{kg-m}$$

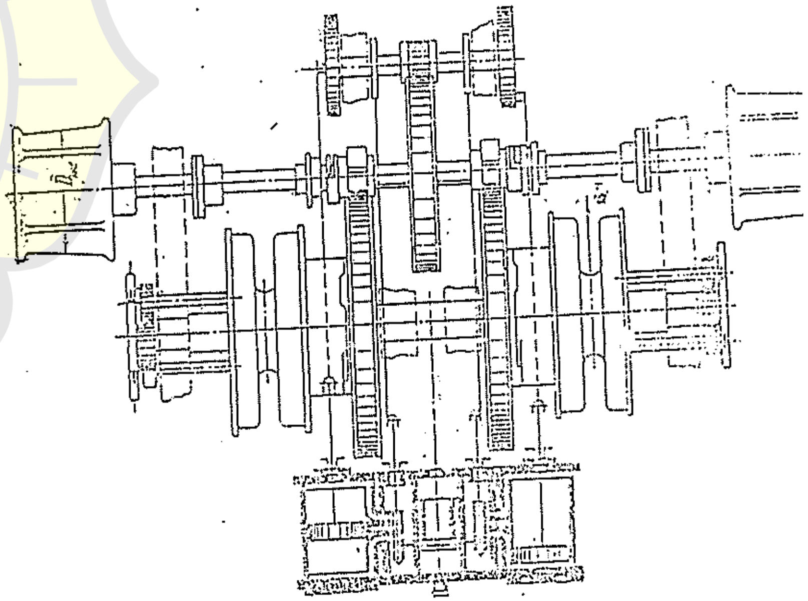


Fig. 171.

The mean shaft power of the motive unit should be

$$N_e = \frac{M_m n_m}{716.20} \quad \text{metric hp}$$

The mean indicated power is

$$N_{im} = \frac{N_e}{\eta_m}$$

The cylinder diameter of the steam engine, according to Posdyunin's formula which is based on the conditions for starting from a dead stop, is

$$D_{ca} = 1.37 \sqrt[3]{\frac{M_m}{\psi_a \eta_m (\alpha_i k_i p_{ls} - p_{ss})}} \quad \text{cm} \tag{389}$$

where M_m = torque developed on the shaft of the engine, kg-cm
ψ_a = 0.85 to 1.7 = cylinder ratio, i.e., $S : D_{ca}$.

The value of $(\alpha_i k_i p_{ls} - p_{ss})$ is approximately from 10 to 15 per cent lower than that taken for a steering engine, due to longer distance from the anchoring mechanism to the steam supply, resulting in higher condensation losses in the pipelines. The other values in the formula are to be within the same limits as for steam steering engines.

The indicated power N_{ia} required to start the engine from rest and the coefficient of reserve power are

$$N_{ia} = \frac{\psi_a D_{ca}^3 (\alpha_i k_i p_{ls} - p_{ss}) n_m}{143,300} \quad \text{metric hp} \tag{390}$$

$$\psi_{res} = \frac{N_{ia}}{N_{im}}.$$

The steam consumption of the engine driving the anchoring arrangement is

$$G_{ia} = g_{ia} N_{ia} \quad \text{kg per hour}$$

where g_{ia} = specific steam consumption, kg per ihp-hr (the same values are taken as for a steam steering engine).

If need arises to determine the pull on the cable lifter from data measured on the anchoring mechanism, formula (390) can be used.

Solving Posdyunin's formula (389) for the torque developed on the shaft of the steam engine we can write

$$M_m = \left(\frac{D_{ca}}{1.37}\right)^3 \eta_m \psi_a (\alpha_i k_i p_{ls} - p_{ss}) \quad \text{kg-cm}$$

On the other hand, if i_a is the total gearing ratio of the transmission in the anchoring mechanism, then

$$M_m = \frac{M_{cl}}{\eta_a i_a} = \frac{T_{cl} D_{cl}}{2 \eta_a i_a} \quad \text{kg-cm}$$

Combining the last two equations we obtain

$$T_{cl} = \frac{2 M_m \eta_a i_a}{D_{cl}} = 2 \left(\frac{D_{ca}}{1.37}\right)^3 \frac{\eta_m \psi_a (\alpha_i k_i p_{ls} - p_{ss}) \eta_a i_a}{D_{cl}} =$$
$$= 0.78 \frac{D_{ca}^3}{D_{cl}} \eta_m \psi_a (\alpha_i k_i p_{ls} - p_{ss}) \eta_a i_a \quad \text{kg}$$

The diameter of the warp ends is taken equal to

(a) $D_{we} = (5 \text{ to } 8)\, d_w$ for hemp ropes (391)

(b) $D_{we} = (15 \text{ to } 20)\, d_w$ for steel ropes (392)

where d_w = diameter of the warping hawser.

Warp end diameters determined from the diameter of hemp ropes will be suitable for steel ropes as well.

Denoting the hawser heaving-in speed as v_w m per sec we can find the speed of the warping shaft from the length of hawser heaved in per minute. Thus

$$n_w = \frac{60 v_w}{\pi (D_{we} + d_w)} = 19.1 \frac{v_w}{D_{we} + d_w} \quad \text{rpm} \tag{393}$$

where v_w = hawser heaving-in speed, m per sec, is to be assigned according to the pull of the warp end (Table 55).

The gearing ratio between the warping shaft and the shaft of the motive unit is

$$i_w = \frac{n_m}{n_w}$$

The pulling force developed on the warp end is

$$T_{we} = \frac{M_{we}}{\frac{1}{2}(D_{we} + d_w)} = \frac{2 M_m \eta_w i_w}{D_{we} + d_w} \leqslant \frac{R_{br}}{6} \tag{394}$$

where M_{we} = torque developed on the warp end
η_w = efficiency of the transmission between the warping and motive unit shafts.

If n_m rpm is the speed of the motive unit shaft, the speed at which the hawser is heaved in will be

$$v_w = \frac{\pi (D_{we} + d_w) n_m}{60 i_w} \quad \text{m per sec} \tag{395}$$

6-5. Winch Operation

Before starting a winch it is necessary to make sure by inspection that it is in order and that no foreign objects hinder the moving parts of the winch. Then, warm up the steam supply line and the winch engine, apply lubricant to all friction surfaces, check the starting gear by turning it to the "hoist" and "lower" positions, check whether the speed-changing clutches engage and disengage properly and whether the braking gear is in order. After this, test the forward and reverse operation of the winch; if no knocking is heard and reversal is rapid and smooth, the winch is ready for regular operation. The winch is started and stopped and the shaft speed is changed in operation by opening and closing a stop valve or the starting valve. The winch is reversed either by shifting the links of the reversing gear or by operating the starting valve.

During winch operation it is necessary to see that lubricant is being properly fed, to check the temperature of the parts subjected to friction and to listen for knocking. As soon as abnormal noise is heard, stop the winch, find and eliminate the cause of the noise.

The load lowering speed should be regulated by applying the brake. Backsteam should be resorted to only when the load drops too fast even after applying the brake.

If hoisting operations are interrupted for short periods in winter, the winch should be run idle at low speed with open blow-off valves.

If the winch is not to be operated for a prolonged period in winter, it is necessary to drain the condensate from the cylinders and the live and exhaust steam lines.

Winch operation is prohibited if cracks are found in critical parts, if the motive mechanism, steam distribution or breaking gear is out of order or if the gears are excessively worn or some of the teeth are missing.

crest speed of waves running along the length of the ship. It has been established that the boat hoisting speed must be at least v = 0.15 m per sec under these conditions. The heaving-in speed of the tackle fall when single-sheave blocks are used must in this case be v_f = 0.3 m per sec.

The required winch head speed is found from the equation

$$\pi (D_h + d_f)\, n_h = 60 v_f$$

$$n_h = \frac{60 v_f}{\pi (D_h + d_f)} = 19.1 \frac{v_f}{D_h + d_f} \quad \text{rpm}$$

Assigning a motive unit speed (n_m = 500 to 1,600 rpm for electric motors and n_m = 200 to 380 rpm for steam engines), we can find the gearing ratio of the boat winch. Thus

$$i_{bw} = \frac{n_m}{n_h}$$

In designing nonreversible worm gearing the number of teeth on the worm wheel is taken in the range from 24 to 44. The pulling force on the winch head is equal to the sum of the pulling force of the tackle falls:

$$T = T_{max} + T_{min}$$

Disregarding friction losses, the torque developed on the winch head shaft will be

$$M_h = \frac{T (D_h + d_f)}{2}$$

If the winch has an efficiency of η_{bw}, the torque and power on the motive unit shaft will be

$$M_{mb} = \frac{M_h}{\eta_{bw} i_{bw}} = \frac{T (D_h + d_f)}{2 \eta_{bw} i_{bw}}$$

and

$$N_e = \frac{M_{mb} n_m}{716.20} \quad \text{metric hp}$$

The cylinder diameter and indicated power of steam boat winches are determined from the same Posdyunin formulas used of gravity are not yet exactly known in the early project stage. If the model does not accomplish the required speed the designer has to alter the hull. This alteration, however, is possible in the early project stage only. If the trial speed in ballast condition corresponds to the model trial speed in ballast, it can be assumed that service speed in loaded condition is attained, too.

Service speed of a ship is smaller than trial speed because of:

- increase of resistance by wind more than Beaufort 2
- increase of resistance by seaway
- increase of resistance by fouling on shell plating.

In general

$$V_{trial} \approx 1.06 \cdot V_{service}$$ (this corresponds to a power margin of about 20 - 25%).

The propeller is designed for 85% ... 90% of the driving power, at 100% of revolutions.

7.6. Consumables and tanks

There are some more special requirements in ship design: Capacities of

- consumables
- provisions
- ballast.

a) consumables are (depending on type of engine plant, time one round trip, number of crew members):

- fuel oil

$$W_{fuel\ oil}\ [t] = P_{Bme} \cdot b_{me} + P_{ae} \cdot b_{ae} \cdot \frac{S}{V_{serv}} \cdot 10^{-6} \cdot [1.3 \ldots 1.5]$$

last brackets for reserve:

- fuel rests in tanks
- seaway
- wind

P_{bme} = break horsepower of the main engine [KW]

b_{me} = specific fuel oil consumption main engine [g/KW·h]

P_{ae} = total power of auxiliary engines [KW]

b_{ae} = specific fuel oil consumption auxiliary engines [g/KW·h]

s = operating range [sm]

V_{serv} = speed [kn]

1 KW = 0.736 PS (BHP).

Power and heat units
1 HP (HORSEPOWER)
= 550 foot pounds per second
= 0.746 kilowatt
= 1,014 PS (or Cheval Vapeur)
1 PS (PFERDESTAERKE) or CV (CHEVAL VAPEUR)
= 542 foot pounds per second
= 0.986 hp.
= 0.736 kw

1 LONG TON = 1,01605 ...
1 SHORT TON = 0,907185 T

1 kW (kilowatt)
= 1000 watts
= 1340 hp
= 1359 PS or CV
= 737 foot pounds per se...

Motors:

Specific fuel oil consumption:

for two-stroke engines b = 205 ... 211 [g/KW·h]

for four-stroke engines with cylinder power more than 300 KW

b = 196 ... 209 [g/KW·h]

for full power: addition 5%

for diesel fuel: reduction 5% (dependent on heating value of diesel fuel)

For steam turbines:

Standard circulation without furnace gas reheat

livesteam: 64 ... 82 bar at 513 ... 538°C

b = 278 ... 286 [g/KW·h].

with furnace gas reheat

livesteam: 80 ... 110 bar at 513 ... 538°C

b = 252 ... 265 [g/KW·h]

For gas turbines:

Gasoline and light crude oils

b = 299 ... 312 [g/KW·h]

Specific weight of heavy fuel oil: γ = 0.95 t/m^3

Required volume of storage tanks

$$V_{oil} = \frac{W}{\gamma} \ [m^3]$$

1 kw = 0,736 PS

Additions to the volume

- 2% for double bottom tanks
- 1 ... 2% for top tanks and deep tanks
- 2% for thermal expansion, i.e. 98% filled only.

Diesel oil

used for auxiliary engines and for the main engine during estuary trading.

$$w_{diesel} = (0.1 \ldots 0.2) \cdot w_{heavy\ fuel\ oil}$$

specific weight $\gamma_{diesel} = 0.85\ t/m^3$

Volume: $V_{diesel} = \dfrac{w_{diesel}}{\gamma_{diesel}}$ $[m^3]$

additions see fuel oil!

Lubrication oil

In general ships have about 30 ... 50 t lubrication oil, because otherwise the tanks will get too small. (According to owner's desire!).

$$w_{lubr.} = P_{Bme} \cdot b_{me} \cdot \frac{S}{V_{serv}} + \text{addition}$$

b = 0.8 ... 1.2 [g/KW·h] diesel engine two stroke
b = 1.2 ... 1.6 [g/KW·h] diesel engine four stroke
b = 0.14 [g/KW·h] turbines and gearboxes

specific weight $\gamma_{lubr} = 0.90\ t/m^3$; $V = \dfrac{w}{\gamma}$ (m^3)

$W = t/m^3 \quad m^3$

Fresh water

- drinking water 10 ... 20 kg/pers · day
- washing water 60 kg/pers · day without bathing room
  up to 200 kg/pers · day with bathing room
- boiler feed water 0.14 kg/KW·h plus first filling.

additions to the tank volume: 3 ... 4% for special coatings in case of fresh water.

Fresh water tanks have to be separated from all other tanks.

ii) Ballast capacity used for

- trim (immersion of propeller; resistance)
- providing of sufficient stability (at the end of the voyage)
- heeling (heavy lift vessels; RoRo-vessels; container ships, because of container guides)
- longitudinal strength (bulker, tanker)
- immersion of ship (tanker, to avoid heavy motions in sea-way; therefore light or heavy ballast).

Ballast capacity to be provided depending on ship type and on desires of the owner: between 10% and 50% of deadweight.

Additions to required ballast tank volumina are larger at the ends of the ship.

- +5% lower fore peak tank
- +3% upper fore peak tank
- +2% double bottom tank.

The new IMCO-rules recommend segregated ballast tanks to avoid pollution. Cargo oil tanks are separated from the ballast tank system. The economy decreases and more tank capacity is needed.

Sounding/ullage tables delivered by yard.

Provisions/persons/luggage

Weight of provisions	3 ... 5 kg/pers · day
weight of persons	75 kg (crew and passengers)
weight of luggage	20 kg/pers (short distance)
	60 kg/pers (long distance passenger and crew).

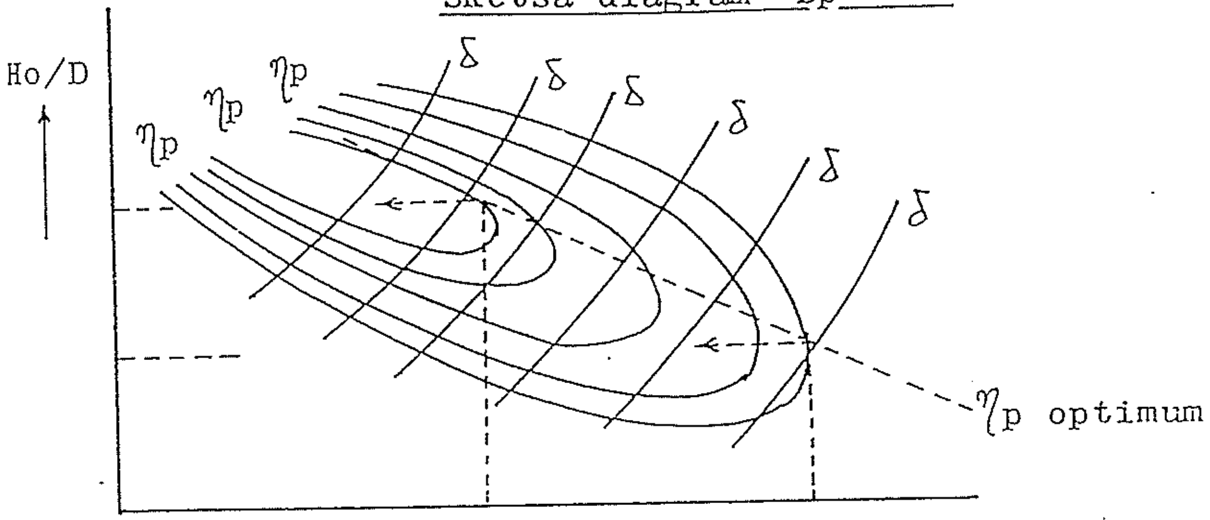
...and Location of Main Engine

...nother part of the contract influencing ship design.

...p weight, volume, fuel consumption).

...omy is determined by the choice of the main engine type, also

Sketsa diagram Bp - δ.



Pemakaian Bp- δ Diagram dari Baling2 Type B-Series.

Data yang diperlukan untuk perencanaan baling baling dengan memakai diagram Bp- δ .

Baling-baling Troost series type B adalah :

1. Kecepatan kapal Vs dalam knots.
2. Besarnya tenaga kuda ditempat dimana baling2 berada P (= 1 HP Inggris = 76 kg.m/dt)
3. Besarnya perputaran baling2 N dalam rpm. pada keadaan kecepatan Vs dan tenaga P diatas.
4. Harga diameter maximum dari baling2 yang diperkenankan sehubungan dengan sarat air kapal serta ukuran dan bentuk stern frame serta posisi dari sumbu poros baling2, (dalam feet).

Selanjutnya beberapa koreksi harus diadakan sesuai petunjuk yakni untuk memperbaiki ketelitian dalam perhitungan yaitu sebagai berikut :

1. Koreksi karena adanya pengaruh skala (scale effect) Rpm dari baling2 harus dikoreksi sehubungan adanya pengaruh skala dalam komponen2 dari Propulsive effi ciency (yaitu: wake, thrust deduction, dan propeller efficiency). Wake fraction untuk kapal2 yang baru selesai dibangun dan catnya masih segar dan sangat licin (trial condition) ternyata menurut pe-

nelitian harganya lebih kecil dari model parafin yang dipergunakan dalam towing test ditangki percobaan, karena harga itu besarnya turun bilamana angka Reynolds bertambah besar. Walaupun kapal2 baru yang baru selesai dicat itu mempunyai permukaan yang lebih kasar terhadap permukaan model parafin, tetapi penambahan besarnya harga wake fraction akibat kekasaran permukaan tidak lebih besar dari pengurangan harga wake fraction akibat bertambah besarnya angka Reynolds untuk kapal sebenarnya.

Berhubung data yang tersedia adalah hasil dari model kapal dari parafin yang permukaannya sangat licin (yaitu data harga2 wake, thrust deduction dan effisiensi baling2), sedangkan yang direncanakan adalah baling2 untuk kapal yang sebenarnya, maka perbedaan harga wake dimuka menyebabkan rpm dari baling2 yang direncanakan harus dikoreksi dengan mengurangi harganya.

Hal itu adalah agar supaya "propeller behind the ship" bekerja pada putaran yang dikehendaki dapat sama dengan yang dipakai pada percobaan yang menghasilkan diagram Bp- δ yang dipilih dalam perencanaan itu. Penurunan perputaran baling2 akibat adanya fouling kapal juga harus diperhitungkan.

Sampai saat ini NSMB mempergunakan harga2 koreksi untuk rpm sebagai berikut :

Kapal2 ber-baling2 tunggal:

Untuk service condition: - 2 %
Untuk trial condition : - 3 %

<u>Kapal2 ber-baling2 ganda</u> :

Untuk service condition: - 1 %
Untuk trial condition : - 2 %

2. <u>Koreksi tenaga</u> :

Koreksi ini adalah untuk memperhitungkan adanya kerugian2 gesekan pada stuffing dan bantalan2 lainnya dsb. pada shafting arrangement kapal.

Baik untuk kapal2 ber-baling2 ganda maupun tunggal besarnya koreksi untuk masing2 poros baling baling adalah :

Kamar mesin dibelakang : - 3 %

Kamar mesin ditengah : - 5 %

Untuk kapal2 perang destroyer, cruiser dsb. biasanya untuk kerugian gesekan pada shaftingnya diperkirakan hanya perlu koreksi - 1%.

Perlu diingat bahwa perhitungan P pada Bp-δ adalah memakai H.P. Inggris yaitu = 76 kg.m/detik, sehingga kapal data P yang diberikan adalah dalam metric maka perlu adanya koreksi penyesuaian.

3. Koreksi air tawar menjadi air laut :

Bp - δ maupun diagram Bu - δ semuanya adalah hasil percobaan yang dilaksanakan ditangki percobaan memakai air tawar. Bilamana baling2 yang direncanakan adalah untuk kapal laut, maka perlu diadakan koreksi pada harga P untuk penyesuaian air tawar terhadap air laut tersebut.

Besarnya koreksi adalah :

$$P \times \frac{1.000}{1,025}$$

4. Koreksi harga δ :

Baik Bp - δ maupun diagram Bu - δ adalah dihasilkan dari open water tests dimana model baling2 bekerja pada kondisi terbuka atau open condition.

Karena baling2 yang direncanakan adalah nantinya bekerja pada behind condition, maka perlu adanya koreksi sebagai berikut :

Kapal ber-baling2 tunggal :

Untuk C_b besar : - 4% s/d - 5%

Untuk C_b kecil : - 2%

Kapal ber-baling2 ganda: - 2% s/d - 4%

Adapun prosedur perencanaannya adalah sbb. :

a). Hitung harga load factor Bp :

$$Bp = \frac{N\ P^{1/2}}{Va^{2 1/2}}$$

Adakan koreksi2 seperti telah diterangkan di muka.

b). Dengan memakai Bp - δ diagram pada harga Bp yg telah didapat dari perhitungan diatas, maka dapatlah diketahui besarnya harga optimum advance coefficient δ yaitu dimana mempunyai harga ηp maximum (digaris putus2 pada Bp - δ diagram). Hitung beberapa harga untuk beberapa harga Fa/F.

c). Adakan koreksi untuk harga δ optimum dengan cara dan data yang telah diterangkan diatas.

d). Hitung besarnya D dari harga-harga δ yang telah dikoreksi yakni;

$D = \frac{\delta \,.\, Va}{N}$ dan baca pada diagram harga pitch ratio H/D pada harga δ yang telah dikoreksi tadi.

e). Periksa pada harga Fa/F bagaimana terhadap bahaya kavitasi. Bilamana perlu tentukan harga D dan H/D dengan cara interpolasi untuk harga Fa/F yang diingini.

f). Adakan pemeriksaan terhadap kekuatan baling2 ya itu apakah tebal daun yang direncanakan dengan meniru type yang dipilih sudah cukup kuat dan memenuhi persyaratan kekuatan dan Klasifikasi (Caranya akan diterangkan di paragrap kemudian).

3. Analisa Sebuah Baling2 pada Kondisi Penarikan/Beban Berlebihan.

Berikut ini adalah cara untuk dapat membuat estimasi besarnya gaya tarik tali atau tow rope force dari sebuah kapal pada kondisi penarikan (towing)ataupun pada kondisi beban berlebihan (overload condition) dimana ditentukan kecepatan kapal pada kondisi itu dan baling2 yang dipergunakannya adalah baling2 type B-series yang telah diketahui pula.

Seperti diketahui, pada saat kapal menarik kapal lain ataupun adanya beban lain yang melebihi dari keadaannya bila mana kapal tersebut bebas (free running) maka tahanan kapal (dengan beban tambahannya) akan bertambah dan walaupun mesin induk sebagai mesin penggerak kapal sudah dengan putaran max. yang dapat dicapai dengan kondisi tersebut, kecepatan kapal Vs jelas akan lebih rendah dari kecepatan kapal pada keadaan bebas.

Jadi, dengan demikian harga Va pun akan turun. Untuk mesin2 internal combustion engines, putaran mesin akan turun walaupun sudah digas penuh.

Sedangkan untuk bollard pull test kapal tunda, biasanya percobaan dengan maximum continuous rating (MCR) tidak lebih dari satu jam. Adapun cara perhitungannya mencari Tow Rope Force pada suatu kecepatan yang ditentukan adalah sebagai berikut :

1) Hitung harga koeffisien kecepatan; $J = \frac{Va}{n\ D}$ pada

Engine Data

Engine Power

The table contains data regarding the engine power, speed and specific fuel oil consumption of the engines of the MC Programme.

Engine power is specified in both BHP and kW, in rounded figures, for each cylinder number and layout points L_1, L_2, L_3 and L_4:

L_1 designates nominal maximum continuous rating (nominal MCR), at 100% engine power and 100% engine speed.

L_2, L_3 and L_4 designate layout points at the other three corners of the layout area, chosen for easy reference.

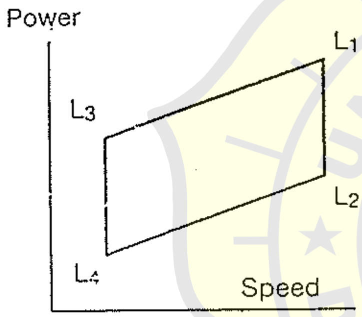


Fig. 1.01: Layout diagram for engine power and speed

Overload corresponds to 110% of the power at MCR, and may be permitted for a limited period of one hour every 12 hours.

The engine power figures given in the tables remain valid up to tropical conditions at sea level, ie.:

Blower inlet temperature . . . . 45 °C
Blower inlet pressure . . . . 1000 mbar
Seawater temperature . . . . 32 °C

Specific fuel oil consumption (SFOC)

Specific fuel oil consumption values refer to brake power, and the following reference conditions:

ISO 3046/1-1986:
Blower inlet temperature . . . . 25 °C
Blower inlet pressure . . . . 1000 mbar
Charge air coolant temperature . . . . 25 °C
Fuel oil lower calorific value . . . . 42,700 kJ/kg
(10,200 kcal/kg)

Although the engine will develop the power specified up to tropical ambient conditions, the specific fuel oil consumption varies with ambient conditions and fuel oil lower calorific value. For calculation of these changes, see section 2.

SFOC guarantee

The figures given in this project guide represent the values obtained when the engine and turbocharger are matched with a view to obtaining the lowest possible SFOC values and fulfilling the IMO NO_x emission limitations.

The Specific Fuel Oil Consumption (SFOC) is guaranteed for one engine load (power-speed combination), this being the one in which the engine is optimised.

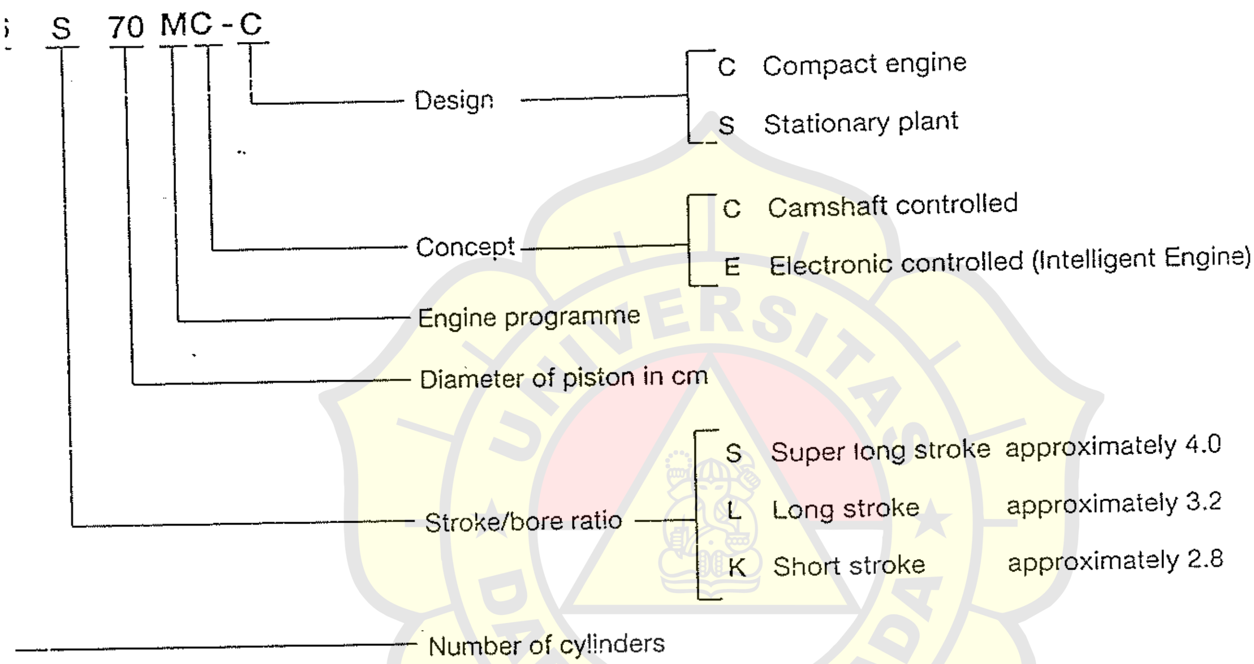
The guarantee is given with a margin of 5%.

As SFOC and NO_x are interrelated parameters, an engine offered without fulfilling the IMO NO_x limitations is subject to a tolerance of only 3% of the SFOC.

Lubricating oil data

The cylinder oil consumption figures stated in the tables are valid under normal conditions.
During running-in periods and under special conditions, feed rates of up to 1.5 times the stated values should be used.

he engine types of the MC programme are
lentified by the following letters and figures



178 34 39-1.0

ig. 1.02: Engine type designation

E K 98 S 26 K9 98 S 24 S9 90 S 31 L9 90 S 29 K 90 S 25 Fig.

Engine type	Layout point	Engine speed r/min	Mean effective pressure bar	Power kW / BHP — Number of cylinders: 4	5	6	7	8	9	10	11	12
S42MC	L_1	136	19.5	4320 5880	5400 7350	6480 8820	7560 10290	8640 11760	9720 13230	10800 14700	11880 16170	12960 17640
Bore 420 mm	L_2	136	15.6	3460 4700	4325 5875	5190 7050	6055 8225	6920 9400	7785 10575	8650 11750	9515 12925	10380 14100
Stroke 1764 mm	L_3	115	19.5	3660 4960	4575 6200	5490 7440	6405 8680	7320 9920	8235 11160	9150 12400	10065 13640	10980 14880
	L_4	115	15.6	2920 3980	3650 4975	4380 5970	5110 6965	5840 7960	6570 8955	7300 9950	8030 10945	8760 11940
L42MC	L_1	176	18.0	3980 5420	4975 6775	5970 8130	6965 9485	7960 10840	8955 12195	9950 13550	10945 14905	11940 16260
Bore 420 mm	L_2	176	11.5	2540 3460	3175 4345	3810 5190	4445 6055	5080 6920	5715 7785	6350 8650	6985 9515	7620 10380
Stroke 1360 mm	L_3	132	18.0	2980 4060	3725 5075	4470 6090	5215 7105	5960 8120	6705 9135	7450 10150	8195 11165	8940 12180
	L_4	132	11.5	1920 2600	2400 3250	2880 3900	3360 4550	3840 5200	4320 5850	4800 6500	5280 7150	5760 7800
S35MC	L_1	173	19.1	2960 4040	3700 5050	4440 6060	5180 7070	5920 8080	6660 9090	7400 10100	8140 11110	8880 12120
Bore 350 mm	L_2	173	15.3	2380 3220	2975 4025	3570 4830	4165 5635	4760 6440	5355 7245	5950 8050	6545 8855	7140 9660
Stroke 1400 mm	L_3	147	19.1	2520 3420	3150 4275	3780 5130	4410 5985	5040 6840	5670 7695	6300 8550	6930 9405	7560 10260
	L_4	147	15.3	2020 2740	2525 3425	3030 4110	3535 4795	4040 5480	4545 6165	5050 6850	5555 7535	6060 8220
L35MC	L_1	210	18.4	2600 3520	3250 4400	3900 5280	4550 6160	5200 7040	5850 7920	6500 8800	7150 9680	7800 10560
Bore 350 mm	L_2	210	14.7	2080 2820	2600 3525	3120 4230	3640 4935	4160 5640	4680 6345	5200 7050	5720 7755	6240 8460
Stroke 1050 mm	L_3	178	18.4	2200 3000	2750 3750	3000 4500	3850 5250	4400 6000	4950 6750	5500 7500	6050 8250	6600 9000
	L_4	178	14.7	1760 2400	2200 3000	2640 3600	3080 4200	3520 4800	3960 5400	4400 6600	4840 6600	5280 7200
S26MC	L_1	250	18.5	1600 2180	2000 2725	2400 3270	2800 3815	3200 4360	3600 4905	4000 5450	4400 5995	4800 6540
Bore 260 mm	L_2	250	14.8	1280 1740	1600 2175	1920 2610	2240 3045	2560 3480	2880 3915	3200 4350	3520 4785	3840 5220
Stroke 980 mm	L_3	212	18.5	1360 1860	1700 2325	2040 2790	2380 3255	2720 3720	3060 4185	3400 4650	3740 5115	4080 5580
	L_4	212	14.8	1100 1480	1375 1850	1650 2220	1925 2590	2200 2960	2475 3330	2750 3700	3025 4070	3300 4440

178 46 78-9.0

Fig. 1.03e: Power and speed

At load layout point		Specific fuel oil consumption g/kWh g/BHPh, with conventional turbochargers: 100%	80%	Lubricating oil consumption: System oil, Approx. kg/cyl. 24h	Cylinder oil, g/kWh g/BHPh
L42MC	L_1	177 130	174 129	3-4	0.8-1.2 0.6-0.9
	L_2	165 121	163 120		
	L_3	177 130	174 129		
	L_4	165 121	163 120		
S35MC	L_1	178 131	177 130	2-3	0.95-1.5 0.7-1.1
	L_2	173 127	171 126		
	L_3	178 131	177 130		
	L_4	173 127	171 126		
L35MC	L_1	177 130	175 129	2-3	0.8-1.2 0.6-0.9
	L_2	171 126	170 125		
	L_3	177 130	175 129		
	L_4	171 126	170 125		
S26MC	L_1	179 132	178 131	1.5-3	0.95-1.5 0.7-1.1
	L_2	174 128	173 127		
	L_3	179 132	178 131		
	L_4	174 128	173 127		

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Fig. 1.05f: Fuel and lubricating oil consumption

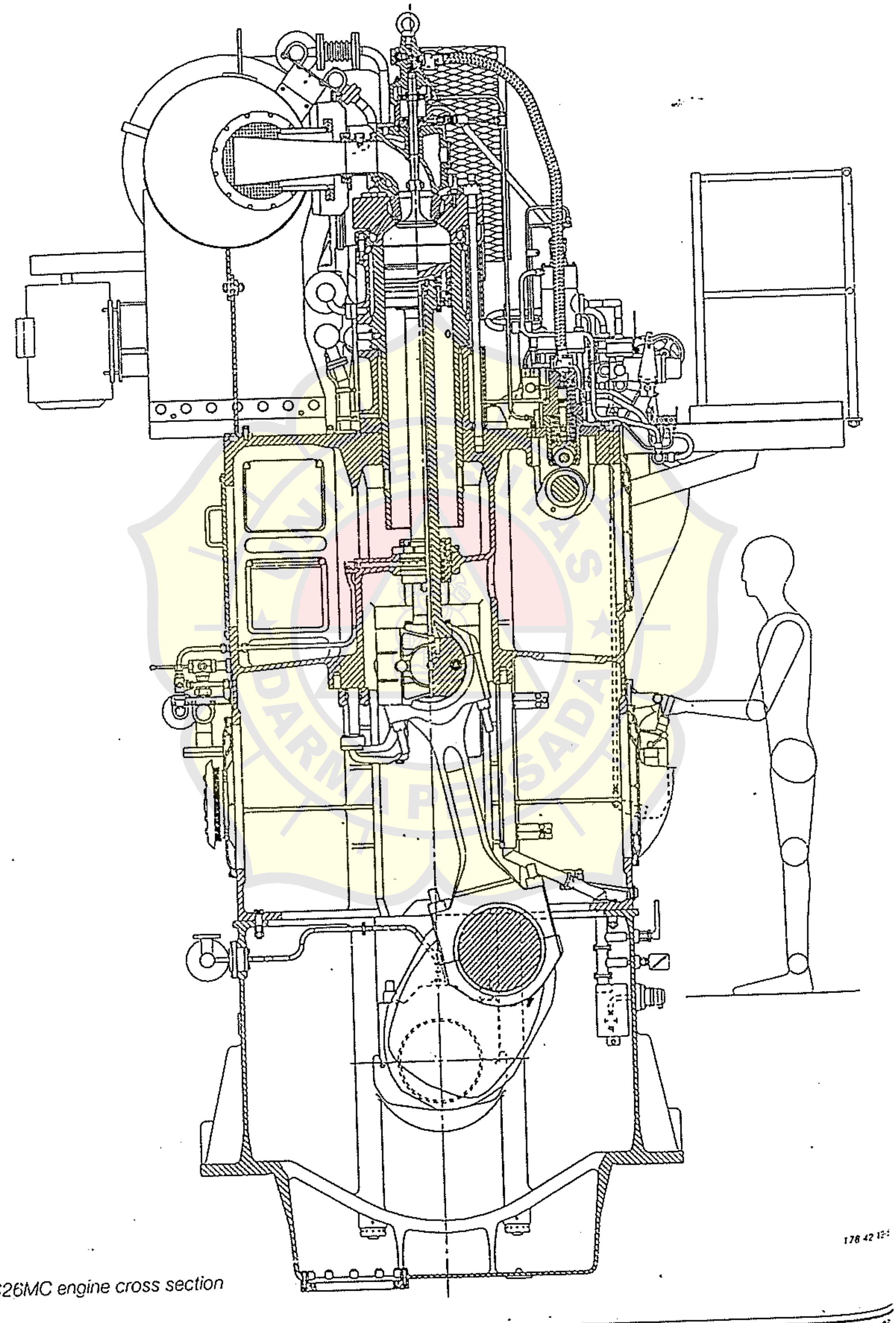

Fig. 1.11: S26MC engine cross section

Engine type	Number of cylinders 4	5	6	7	8	9	10	11	12
S70MC-C	1 x 80-B11	1 x 85-B11	1 x 85-B11	1 x 85-B12	2 x 80-B11	–	–	–	–
S70MC	1 x 80-B11	1 x 80-B12	1 x 85-B11	1 x 85-B12	2 x 80-B11	–	–	–	–
L70MC	n.a.	n.a.	n.a.	n.a.	n.a.	–	–	–	–
S60MC-C	1 x 77-B11	1 x 80-B11	1 x 80-B12	1 x 85-B11	1 x 85-B11	–	–	–	–
S60MC	1 x 77-B11	1 x 77-B12	1 x 80-B11	1 x 80-B12	1 x 85-B11	–	–	–	–
L60MC	1 x 77-B11	1 x 77-B12	1 x 80-B11	1 x 80-B12	1 x 85-B11	–	–	–	–
S50MC-C	1 x 73-B11	1 x 77-B11	1 x 77-B11	1 x 77-B12	1 x 80-B11	–	–	–	–
S50MC	1 x 73-B11	1 x 73-B12	1 x 77-B11	1 x 77-B12	1 x 80-B11	–	–	–	–
L50MC	1 x 73-B11	1 x 73-B12	1 x 77-B11	1 x 77-B11	1 x 77-B12	–	–	–	–
S46MC-C	1 x 73-B11	1 x 73-B11	1 x 77-B11	1 x 77-B11	1 x 77-B12	–	–	–	–
S42MC	1 x 69-A10	1 x 73-B11	1 x 73-B11	1 x 73-B12	1 x 77-B11	1 x 77-B11	2 x 73-B11	2 x 73-B11	2 x 73-B11
L42MC	1 x 69-A10	1 x 73-B11	1 x 73-B11	1 x 73-B12	1 x 73-B12	1 x 77-B11	2 x 73-B11	2 x 73-B11	2 x 73-B11
S35MC	1 x 65-A10	1 x 69-A10	1 x 69-A10	1 x 73-B11	1 x 73-B11	1 x 73-B11	2 x 69-A10	2 x 69-A10	2 x 69-A10
L35MC	1 x 65-A10	1 x 65-A10	1 x 69-A10	1 x 69-A10	1 x 73-B11	1 x 73-B11	2 x 65-A10	2 x 65-A10	2 x 69-A10
S26MC	1xTPS57D*	1xTPS57D*	1 x 61-A10	1 x 61-A10	1 x 65-A10	1 x 65-A10	2 x TPS57D*	2 x 61-A10	2 x 61-A10

All turbochargers in this table are of the TPL-type.

* For 4 and 5 cylinder S26MC the full designation is listed in the table.

n.a. Not applicable

– Not included in the production programme

Example of a full designation: 6S70MC-C requires 1 x TPL85-B11 at nominal MCR.

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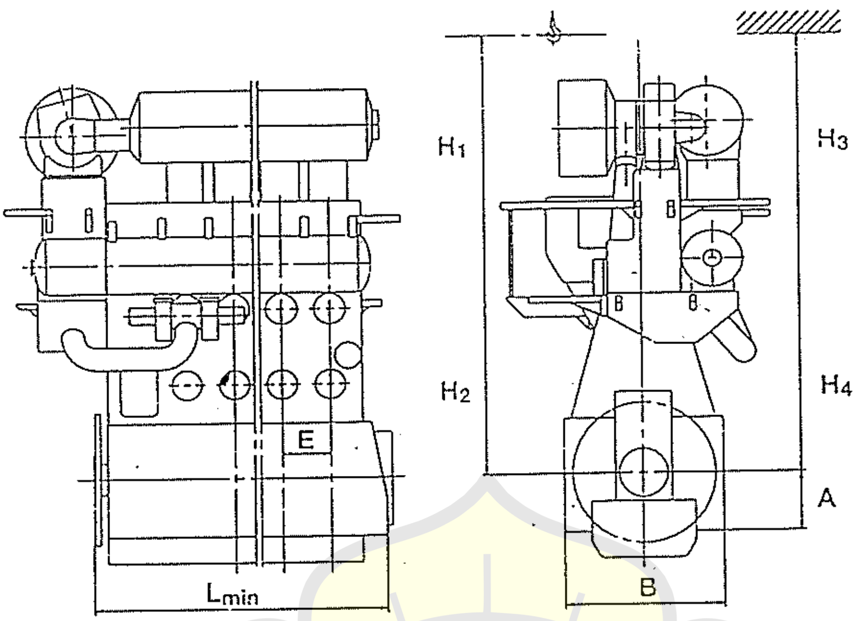
Fig. 3.07: ABB conventional turbochargers, type TPL, for engines with nominal rating (L_1) complying with IMO's NO_x emission limits



	S50-C	S50	L50	S46-C	S42	L42	S35	L35	S26
					Dimensions in mm				
A	1085	1085	944	986	900	690	650	550	420
B	3150	2950	2710	2924	2670	2460	2200	1980	1880
E	850	890	890	782	748	748	600	600	490
H1	8950	8800	7825	8600	8050	6700	6425	5200	4825
H2	8375	8250	7325	8075	7525	6250	6050	4850	4725
H3	8150	8100	7400	7850	7300	6350	5925	5025	4525
H4							5850	4825	4500
					Lmin				
4 cyl.	4739	5730	5615	4357	4240	4661	3480	3445	2975
5 cyl.	5589	6620	6505	5139	4988	5409	4080	4045	3465
6 cyl.	6439	7510	7395	5921	5736	6157	4680	4645	3955
7 cyl.	7289	8400	8285	6703	6484	6905	5280	5245	4445
8 cyl.	8139	9290	9175	7485	7232	7653	5880	5845	4935
9 cyl.					7980	8401	6480	6445	5425
10 cyl.					9476	9897	7080	7645	6405
11 cyl.					10224	10645	8280	8245	6895
12 cyl.					10972	11393	8880	8845	7385
					Dry masses in tons				
4 cyl.	155	171	163	133	109	95	57	50	32
5 cyl.	181	195	188	153	125	110	65	58	37
6 cyl.	207	225	215	171	143	125	75	67	42
7 cyl.	238	255	249	197	160	143	84	75	48
8 cyl.	273	288	276	217	176	158	93	83	53
9 cyl.					195	176	103	92	58
10 cyl.					232	210	122	108	68
11 cyl.					249	229	132	118	74
12 cyl.					269	244	141	126	79

The distances H_1 and H_2 are from the centre of the crankshaft to the crane hook. The distances H_3 and H_4 for the double jib crane are from the centre of the crankshaft to the lower edge of the deck beam.

E - Cylinder distance H_1 - Vertical lift H_2 - Tilted lift H_3 - Electrical double jib crane H_4 Manual double jib crane

178 87 19-8.0

Fig. 5.01b: Space requirements and masses

		Cyl.		4	5	6	7	8	9	10	11	12
	Nominal MCR at 250 r/min	kW		1600	2000	2400	2800	3200	3600	4000	4400	4800
Pumps	Fuel oil circulating pump	m³/h		1.5	1.8	2.0	2.4	2.7	3.0	3.3	3.6	3.9
	Fuel oil supply pump	m³/h		0.4	0.5	0.6	0.7	0.8	0.9	1.1	1.2	1.3
	Jacket cooling water pump	m³/h	1)	16	20	24	28	32	36	40	44	48
			2)	16	20	24	28	32	36	40	44	48
			3)	24	28	25	29	34	38	55	47	51
			4)	16	20	24	28	32	36	40	44	48
	Central cooling water pump*	m³/h	1)	70	88	105	125	140	160	175	190	210
			2)	71	88	105	125	140	160	175	195	210
			3)	73	90	105	125	140	155	180	190	210
			4)	71	88	105	125	140	155	175	190	210
	Seawater pump*	m³/h	1)	52	66	79	92	105	120	130	145	160
			2)	53	66	79	92	105	120	130	145	155
			3)	56	68	78	91	105	115	135	145	155
			4)	53	66	78	91	105	115	130	145	155
	Lubricating oil pump*	m³/h	1)	49	57	65	72	84	94	99	105	115
			2)	51	58	66	73	83	93	100	105	115
			3)	48	55	63	70	80	90	95	100	110
			4)	50	57	65	72	82	92	99	105	115
	Booster pump f. exh. valve actator	m³/h		n.a.	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.	n.a.
Coolers	Scavenge air cooler Heat dissipation approx.	kW		560	710	850	990	1130	1270	1410	1550	1690
	Central cooling water	m³/h		45	56	68	79	90	101	112	123	134
	Lubricating oil cooler Heat dissipation approx.*	kW	1)	220	275	350	400	460	510	550	600	700
			2)	230	280	340	390	450	500	580	630	680
			3)	200	250	300	350	400	450	500	550	600
			4)	225	275	325	375	425	475	550	600	650
	Lubricating oil*	m³/h		See above "Lubricating oil pump"								
	Central cooling water	m³/h	1)	25	34	37	46	50	59	63	67	76
			2)	25	34	37	46	50	59	63	72	76
			3)	25	34	37	46	50	54	68	67	76
			4)	25	34	37	46	50	54	63	67	76
	Jacket water cooler Heat dissipation approx.	kW	1)	310	385	460	540	620	690	770	850	920
			2)	310	385	460	540	620	690	770	850	920
			3)	395	470	485	560	650	720	940	890	970
			4)	310	385	460	540	620	690	770	850	920
	Jacket cooling water	m³/h		See above "Jacket cooling water"								
	Central cooling water	m³/h		See above "Central cooling water quantity" for lube oil cooler								
	Central cooler Heat dissipation approx.*	kW	1)	1090	1370	1660	1930	2210	2470	2730	3000	3310
			2)	1100	1380	1650	1920	2200	2460	2760	3030	3290
			3)	1160	1430	1640	1900	2180	2440	2850	2990	3260
			4)	1100	1370	1640	1910	2180	2440	2730	3000	3260
	Central cooling water*	m³/h		See above "Central cooling water pump"								
	Seawater*	m³/h		See above "Seawater cooling pump"								
	Fuel oil heater	kW		39	47	52	63	71	79	87	94	100
	Exhaust gas flow at 260 °C**	kg/h		12400	15600	18700	21800	24900	28000	31100	34200	37300
	Air consumption of engine	kg/s		3.4	4.2	5.1	5.9	6.8	7.6	8.4	9.3	10.1

178 42 76-5.1

Fig. 6.04z: List of capacities, S26MC with ***central cooling system*** *stated at the nominal MCR power (L_1) for engines complying with IMO's NO_x emission limitations*

Starting air system: 30 bar (gauge)

Cylinder No.		4	5	6	7	8	9	10	11	12
S42MC										
Reversible engine										
Receiver volume (12 starts)	m^3	2 x 3.0	2 x 3.0	2 x 3.0	2 x 3.0	2 x 3.5	2 x 3.5	2 x 3.5	2 x 3.5	2 x 3.5
Compressor capacity, total	m^3/h	180	180	180	180	210	210	210	210	210
Non-reversible engine										
Receiver volume (6 starts)	m^3	2 x 2.0	2 x 2.0	2 x 2.0	2 x 2.0	2 x 2.5	2 x 2.5	2 x 2.5	2 x 2.5	2 x 2.5
Compressor capacity, total	m^3/h	120	120	120	120	150	150	150	150	150
L42MC										
Reversible engine										
Receiver volume (12 starts)	m^3	2 x 2.0	2 x 2.0	2 x 2.0	2 x 2.0	2 x 2.5	2 x 2.5	2 x 2.5	2 x 2.5	2 x 2.5
Compressor capacity, total	m^3/h	120	120	120	120	150	150	150	150	150
Non-reversible engine										
Receiver volume (6 starts)	m^3	2 x 1.5	2 x 1.5	2 x 1.5	2 x 1.5	2 x 1.5	2 x 1.5	2 x 1.5	2 x 1.5	2 x 1.5
Compressor capacity, total	m^3/h	90	90	90	90	90	90	90	90	90
S35MC										
Reversible engine										
Receiver volume (12 starts)	m^3	2 x 1.0	2 x 1.0	2 x 1.0	2 x 1.0	2 x 1.5	2 x 1.5	2 x 1.5	2 x 1.5	2 x 1.5
Compressor capacity, total	m^3/h	60	60	60	60	90	90	90	90	90
Non-reversible engine										
Receiver volume (6 starts)	m^3	2 x 0.5	2 x 0.5	2 x 0.5	2 x 0.5	2 x 1.0	2 x 1.0	2 x 1.0	2 x 1.0	2 x 1.0
Compressor capacity, total	m^3/h	30	30	30	30	60	60	60	60	60
L35MC										
Reversible engine										
Receiver volume (12 starts)	m^3	2 x 1.0	2 x 1.0	2 x 1.0	2 x 1.0	2 x 1.5	2 x 1.5	2 x 1.5	2 x 1.5	2 x 1.5
Compressor capacity, total	m^3/h	60	60	60	60	90	90	90	90	90
Non-reversible engine										
Receiver volume (6 starts)	m^3	2 x 0.5	2 x 0.5	2 x 0.5	2 x 0.5	2 x 1.0	2 x 1.0	2 x 1.0	2 x 1.0	2 x 1.0
Compressor capacity, total	m^3/h	30	30	30	30	60	60	60	60	60
S26MC										
Reversible engine										
Receiver volume (12 starts)	m^3	2 x 0.9	2 x 0.9	2 x 1.0	2 x 1.0	2 x 1.0	2 x 1.0	2 x 1.0	2 x 1.0	2 x 1.0
Compressor capacity, total	m^3/h	54	54	60	60	60	60	60	60	60
Non-reversible engine										
Receiver volume (6 starts)	m^3	2 x 0.4	2 x 0.4	2 x 0.4	2 x 0.4	2 x 0.5	2 x 0.5	2 x 0.5	2 x 0.5	2 x 0.5
Compressor capacity, total	m^3/h	24	24	24	24	30	30	30	30	30

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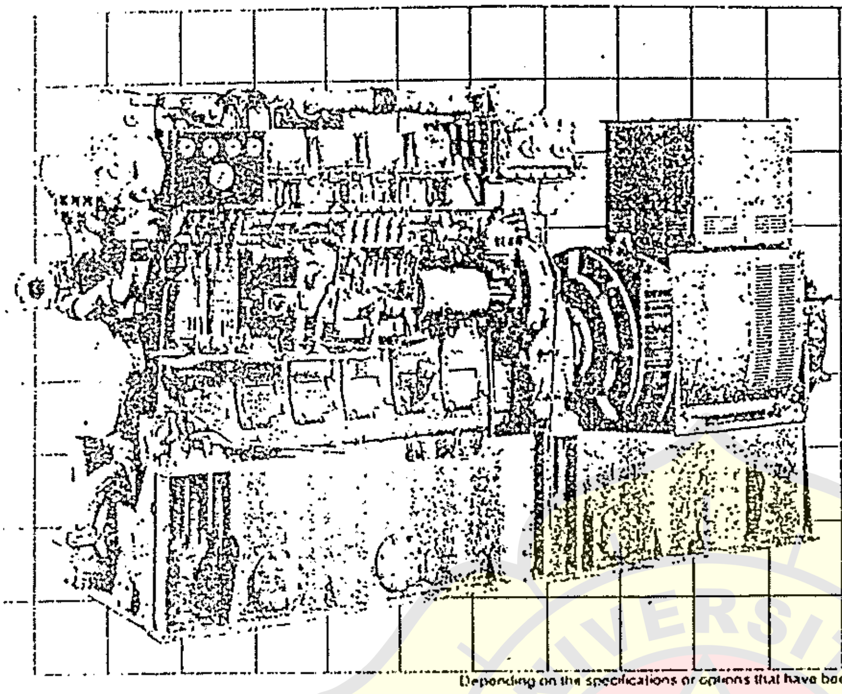
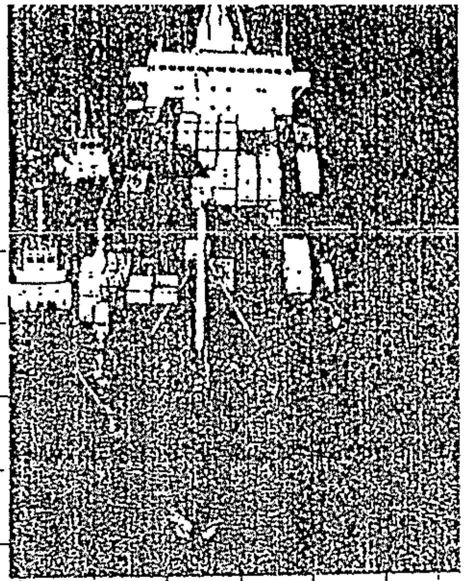
Fig. 6.01.05d: Capacities of starting air receivers and compressors for main engine

6NY16L

Engine output
200 441 kW (272-600 PS)





Depending on the specifications or options that have been chosen, your model may differ slightly from the one in the photograph

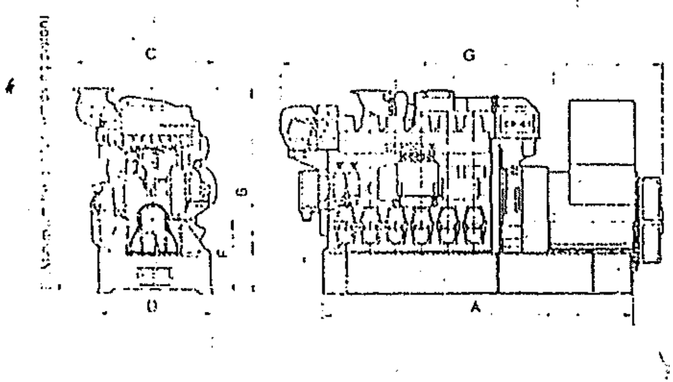
Specifications

Engine model		6NY16L-HN		6NY16L-DN		6NY16L-UN		6NY16L-SN		6NY16L-EN	
Type		Vertical water-cooled 4-cycle diesel engine									
No. of cylinders		6									
Cylinder bore × stroke	mm	160 × 200									
Total displacement	ℓ	24.13									
Continuous rated output	kW (PS)	200 (272)	265 (360)	235 (320)	310 (421)	270 (367)	355 (483)	310 (421)	400 (544)	353 (480)	441 (600)
Engine speed	rpm	1000	1200	1000	1200	1000	1200	1000	1200	1000	1200
Net mean effective pressure	MPa (kgf/cm²)	0.995 (10.15)	1.097 (11.19)	1.171 (11.94)	1.283 (13.09)	1.343 (13.69)	1.472 (15.01)	1.540 (15.70)	1.658 (16.91)	1.756 (17.90)	1.829 (18.65)
Generator capacity	kW	180	240	200	280	240	320	280	360	320	400
Combustion system		Direct Injection									
Starting system		Compressed air									
External dimensions: Overall length	mm	1996									
External dimensions: Overall width	mm	1085									
External dimensions: Overall height	mm	1532									
Dry weight	kg	2800									

The engine dry weight may differ depending upon the specifications and attached accessories

Dimensions (Units: mm)

The dimensions and weights for the diesel engine generator sets are simply reference values. The values may differ for different generator manufacturers.



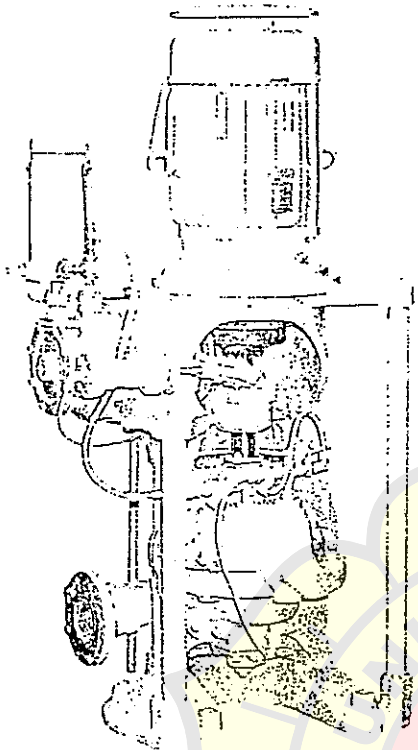
Engine model	6NY16L-HN	6NY16L-DN	6NY16L-UN	6NY16L-SN	6NY16L-EN
A	2530	2530	2530	2530	2530
B	1613	1613	1613	1613	1613
C	1136	1136	1136	1136	1136
D	940	940	940	940	940
E	1725	1725	1725	1725	1725
F	600	600	600	600	600
G	2991	2991	2991	2991	2991
Dry weight of generating equipment (kg)	5500	5500	5500	5500	5500

Please confirm all specifications, etc. on the separate delivery specifications sheet.

FIRE & G.S.PUMP
BILGE & BALLAST PUMP

VSN SERIES

VERTICAL TWO STAGE SINGLE SUCTION PUMP



APPLICATION

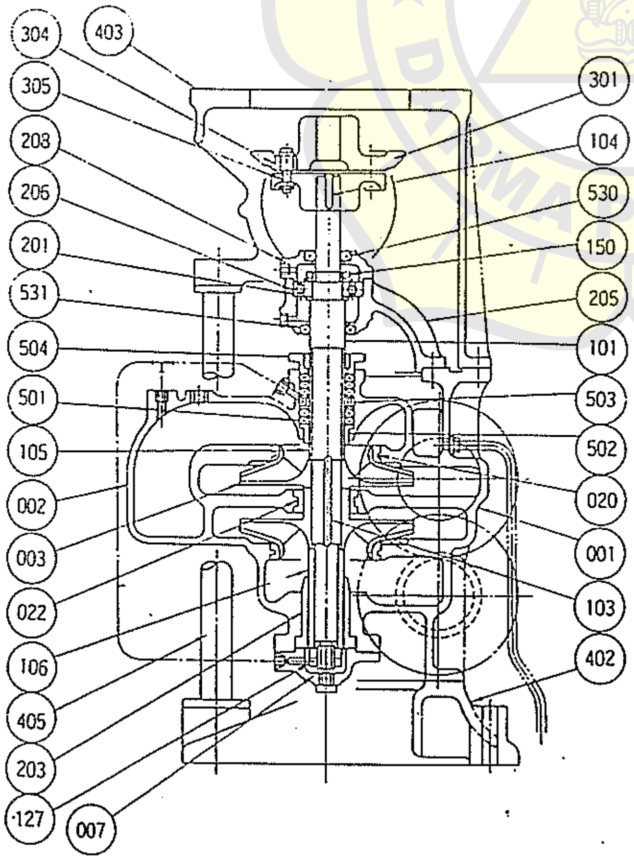
消防兼雑用水、ビルジ兼バラスト、冷却清水、冷却海水

Fire & General Service, Bilge & Ballast, Cooling Fresh Water, Cooling Sea Water.

CONSTRUCTION

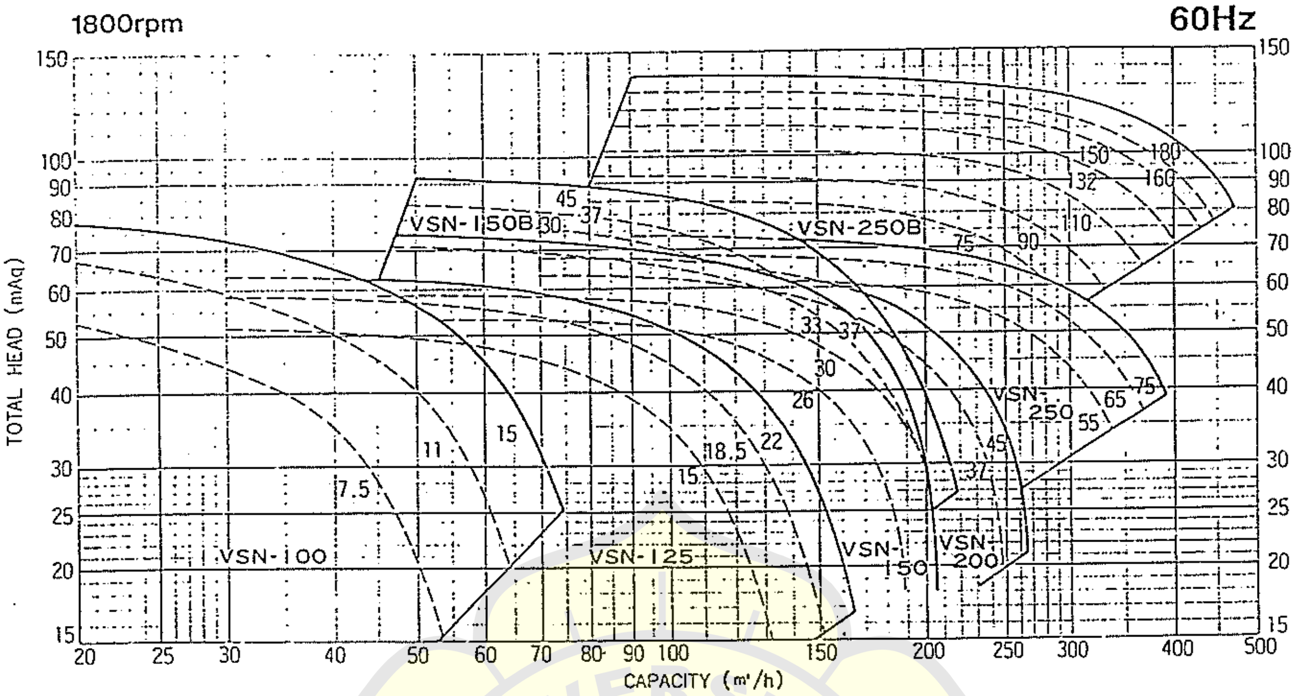
ケーシングは、２分割形で吸込、吐出口をリヤーケーシングに設けており、配管ラインを除去することなく分解、点検が可能です。真空ポンプと連動されます。

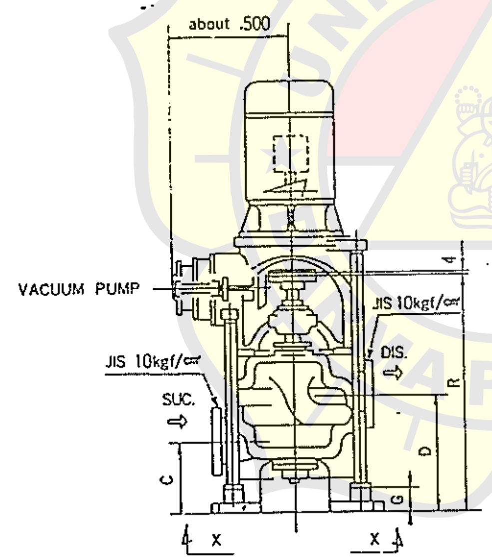
The pump casing of the type VSN is vertically split type. Suction inlet and discharge outlet are arranged on both sides of rear casing, so that the pump can be dismonted and inspected without removing the connection pipe. The pump can be used as an automatic self-prming pump by fitting a vaccum pump (type: N-20) or an air ejecter.

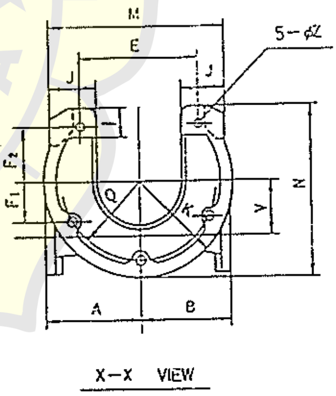


NAME & MATERIAL

PART NO.	NAME	REQ. NO.	SEA WATER MATERIAL	SEA WATER JIS	FRESH WATER MATERIAL	FRESH WATER JIS
001	CASING	1	BRONZE CASTING	BC3	CAST IRON	FC20
002	CASING COVER	1	BRONZE CASTING	BC3	CAST IRON	FC20
003	IMPELLER	2	PHOSPHOR BRONZE	PBC2	PHOSPHOR BRONZE	PBC2
007	BOTTOM COVER	1	BRONZE CASTING	BC3	CAST IRON	FC20
020	CASING RING	2	BRONZE CASTING	BC3	BRONZE CASTING	BC3
022	STAGE BUSH	1	BRONZE CASTING	BC3	BRONZE CASTING	BC3
101	SHAFT	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
103	IMPELLER KEY	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
104	COUPLING KEY	1	CARBON STEEL	S45C-D	CARBON STEEL	S45C-D
105	SLEEVE	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
106	SLEEVE	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
127	SLEEVE NUT	2	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
150	BEARING NUT	1	MILD STEEL	SS41	MILD STEEL	SS41
201	BALL BEARING	1	BEARING STEEL	SUJ2	BEARING STEEL	SUJ2
203	BOTTOM METAL	1	LEAD BRONZE	LBC4	LEAD BRONZE	LBC4
205	BEARING HOUSING	1	CAST IRON	FC20	CAST IRON	FC20
206	HOUSING COVER	1	CAST IRON	FC20	CAST IRON	FC20
208	BEARING COVER	1	CAST IRON	FC20	CAST IRON	FC20
301	COUPLING	1	CAST IRON	FC20	CAST IRON	FC20
304	COUPLING BUSH		RUBBER	NBR	RUBBER	NBR
305	COUPLING BOLT & NUT		MILD STEEL	SS41	MILD STEEL	SS41
402	BED	1	CAST IRON	FC20	CAST IRON	FC20
403	MOTOR FRAME	1	CAST IRON	FC20	CAST IRON	FC20
405	SUPPORT	2	BRONZE GAS PIPE	SGP	STEEL GAS PIPE	SGP
501	GLAND PACKING	5	CARBONIZED FIBER	NBR	CARBONIZED FIBER	NBR
502	NECK BUSH	1	BRONZE CASTING	BC3	BRONZE CASTING	BC3
503	SEAL RING	1	BRONZE CASTING	BC3	BRONZE CASTING	BC3
504	GLAND	1	BRONZE CASTING	BC3	BRONZE CASTING	BC3
530	OIL SEAL	1	RUBBER	NBR	RUBBER	NBR
531	OIL SEAL	1	RUBBER	NBR	RUBBER	NBR







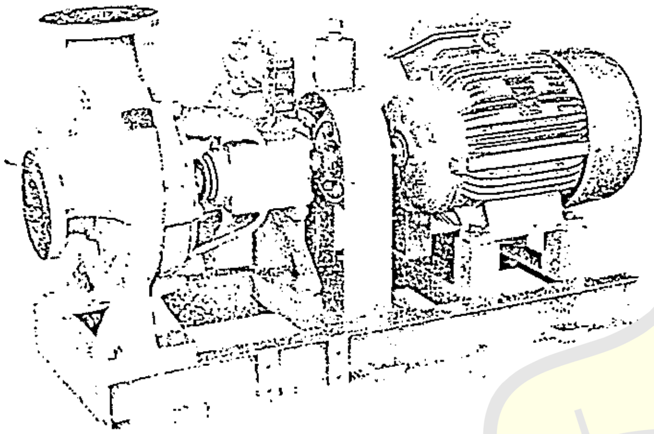
DIMENSION

TYPE	BORE		DIMENSION (mm)																
	SUC.	DIS.	A	B	C	D	E	F_1	F_2	G	I	J	K	M	N	Q	R	V	Z
VSN-100	100	100	300	270	220	380	368	130	184	80	80	175	310	590	550	260	846	170	24
VSN-125	125	125	290	320	262	442	354	125	177	90	76	140	300	550	525	250	970	175	26
VSN-150	150	150	370	350	245	428	424	150	212	90	120	215	360	750	663	300	976	210	28
VSN-150B	150	150	350	330	262	442	424	150	212	90	120	215	360	750	663	300	972	200	28
VSN-200	200	200	370	350	245	428	424	150	212	90	120	215	360	750	663	300	876	210	28
VSN-250	250	250	400	430	290	552	425	150	212	90	120	227	360	750	663	300	955	210	28
VSN-250B	250	250	430	430	290	450	452	160	226	80	113	245	365	750	668	320	1095	330	28

FIRE & G.S.PUMP

HORIZONTAL ONE STAGE SINGLE SUCTION PUMP

EHCN SERIES



APPLICATION

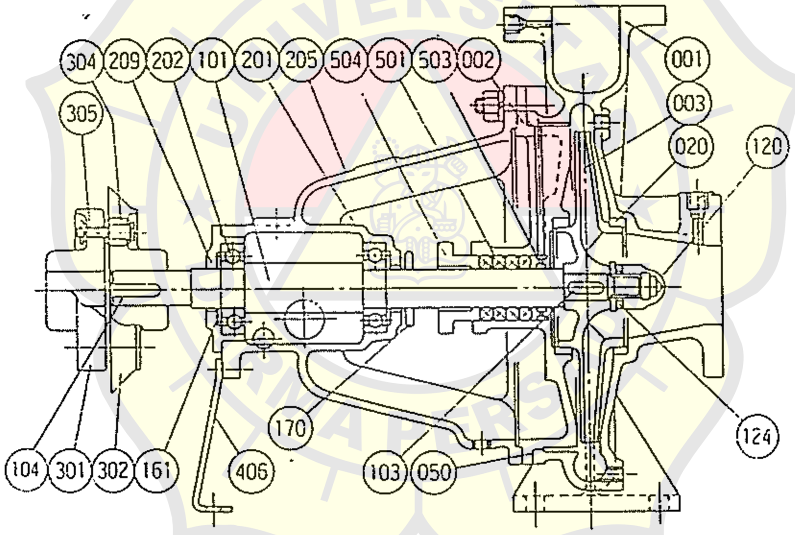
消防兼雑用水、ビルジ兼バラスト

Fire & General Service, Bilge & Ballast.

CONSTRUCTION

このポンプは、真空ポンプ(NVP-20形)と連動して使用します。
軸封はグランドパッキンを標準とします。

In this purpose, the pump is used as an automatic self-priming pump, by fitting a vaccum pump (type: NVP-20).
The gland-packing is applied as a standard method of shaft sealing.

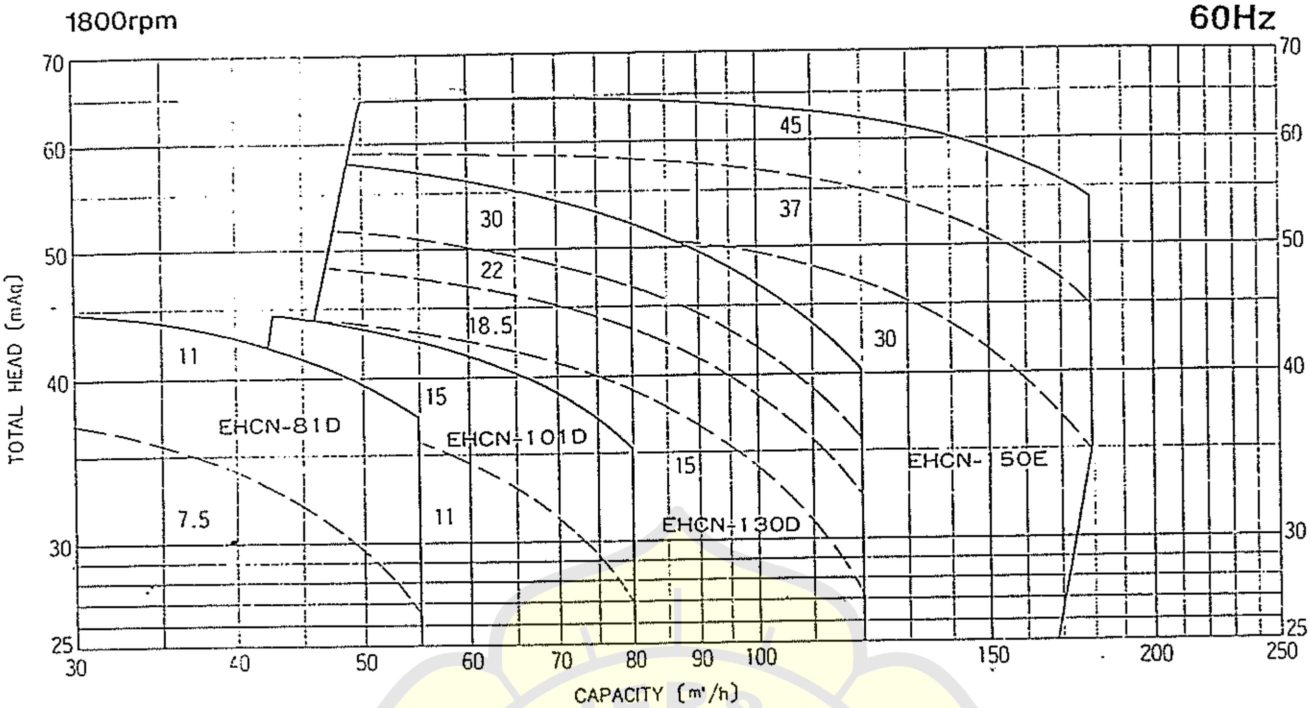


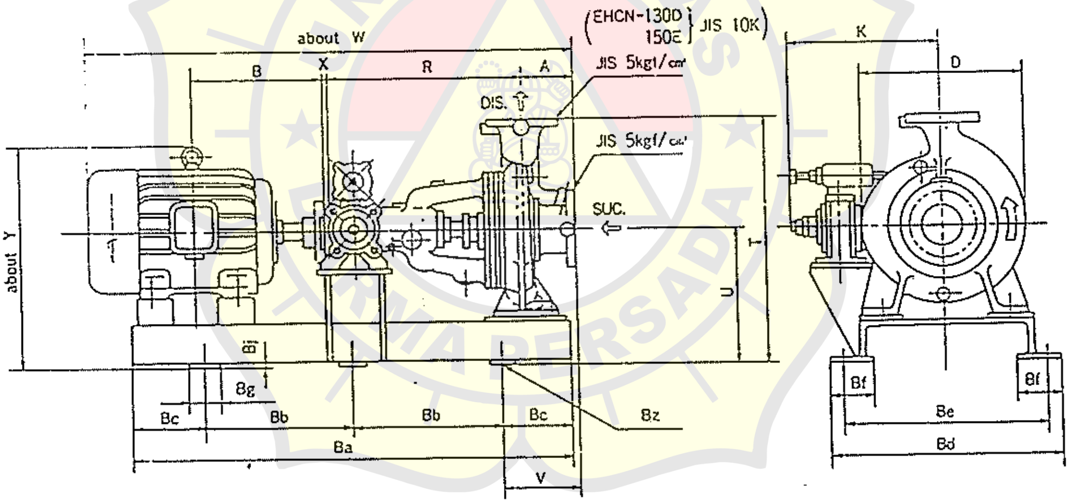
NAME & MATERIAL

PART NO.	NAME	REQ. NO.	SEA WATER MATERIAL	SEA WATER JIS	FRESH WATER MATERIAL	FRESH WATER JIS
001	CASING	1	BRONZE CASTING	BC3	CAST IRON	FC20
002	CASING COVER	1	BRONZE CASTING	BC3	CAST IRON	FC20
003	IMPELLER	1	PHOSPHOR BRONZE	PBC2	PHOSPHOR BRONZE	PBC2
020	CASING RING	2	BRONZE CASTING	BC3	BRONZE CASTING	BC3
050	O-RING	1	RUBBER	NBR	RUBBER	NBR
101	SHAFT	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
103	IMPELLER KEY	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
104	COUPLING KEY	1	CARBON STEEL	S45C-D	CARBON STEEL	S45C-D
120	IMPELLER NUT	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
124	IMPELLER WASHER	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
161	RETAINING	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304

PART NO.	NAME	REQ. NO.	SEA WATER MATERIAL	SEA WATER JIS	FRESH WATER MATERIAL	FRESH WATER JIS
170	FLINGER	1	RUBBER	NBR	RUBBER	NBR
201	BALL BEARING	1	BEARING STEEL	SUJ2	BEARING STEEL	SUJ2
202	BALL BEARING	1	BEARING STEEL	SUJ2	BEARING STEEL	SUJ2
205	BEARING HOUSING	1	CAST IRON	FC20	CAST IRON	FC20
209	BEARING COVER	1	CAST IRON	FC20	CAST IRON	FC20
301	COUPLING	1	CAST IRON	FC20	CAST IRON	FC20
302	COUPLING	1	CAST IRON	FC20	CAST IRON	FC20
406	SUPPORT	1	MILD STEEL	SS41	MILD STEEL	SS41
501	GLAND PACKING	1	CARBONIZED FIBER		CARBONIZED FIBER	
503	LANTERN RING	1	BRONZE CASTING	BC3	BRONZE CASTING	BC3
504	GLAND	1	BRONZE CASTING	BC3	BRONZE CASTING	BC3

TAIKO KIKAI IND. CO., LTD.





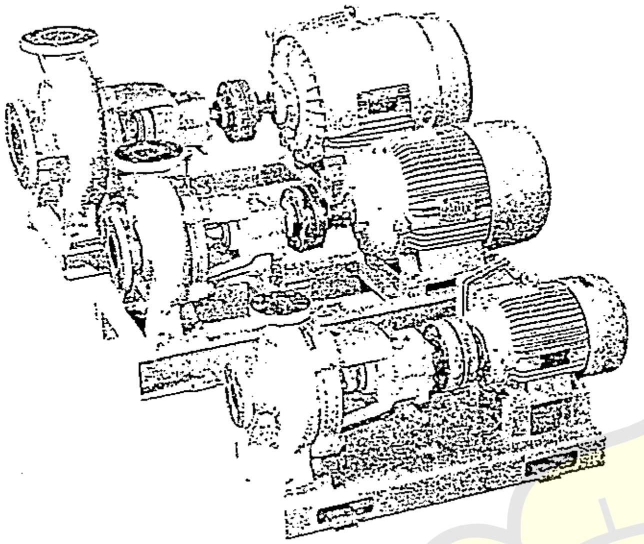
DIMENTION

TYPE	MOTOR kw	MOTOR r/m	BORE SUC.	BORE DIS.	A	B	D	K	R	T	U	V	W	X	Y	Ba	Bb	Bc	Bd	Be	Bf	Bg	Bj	Bz
EHCN-81D	7.5	1800	80	80	125	258	413	206	470	625	345	185	1086	3	502	1000	350	150	470	430	65	60	12	6-φ19
	11					323		228					1199		534	1100	400							
EHCN-101D	11	1800	100	100	125	323	420	228	470	685	370	185	1237	3	584	1100	400	150	470	430	65	60	12	6-φ19
	15					345							1220											
EHCN-130D	15	1800	125	100	140	345	455	263	545	685	370	200	1308	3	590	1100	400	150	470	430	65	60	12	6-φ19
	18.5					351.5		296					1345		610									
	22					370.5		323					1382			1200	450							
	30					395.5		363					1450											
EHCN-150E	30	1800	150	125	140	395.5	576	363	530	865	465	170	1450	4	645	1300	500	150	550	500	65	60	30	6-φ24
	37					432		382					1500		690									
	45					444.5							1520		690									

F.W. & SANITARY PUMP

EHC SERIES

HORIZONTAL SINGLE SUCTION PUMP



APPLICATION

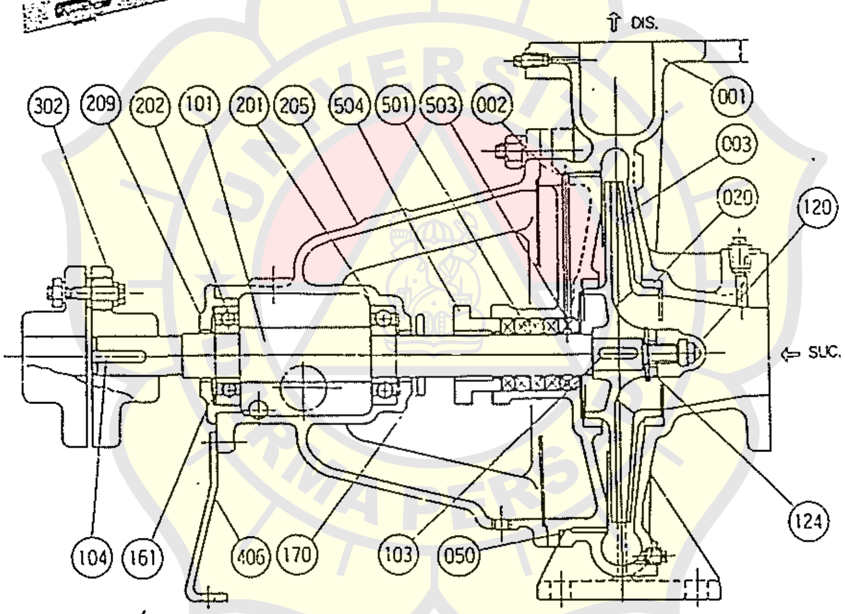
清水、サニタリー、冷却水

Fresh Water, Sanitary, Cooling Water.

CONSTRUCTION

このポンプは、横形非自吸式です。
軸封はグランドパッキン式が標準です。

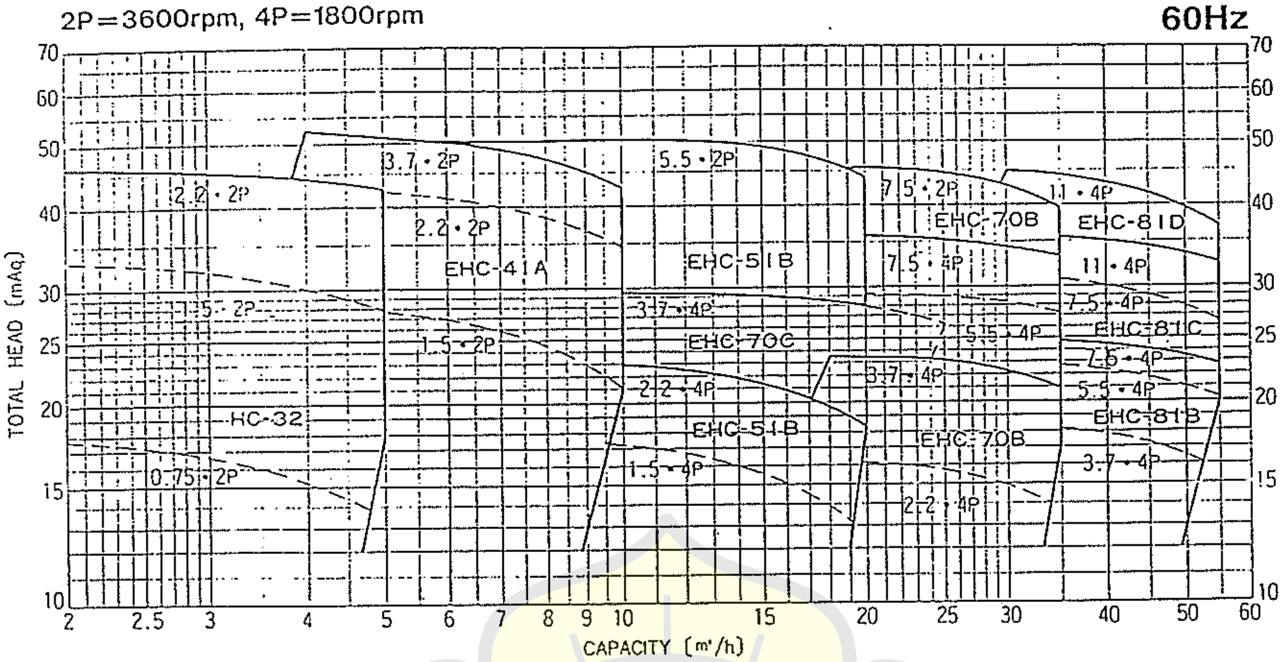
The EHC-type is non self-priming.
The gland-packing is applied as a standard method of shaft sealing.

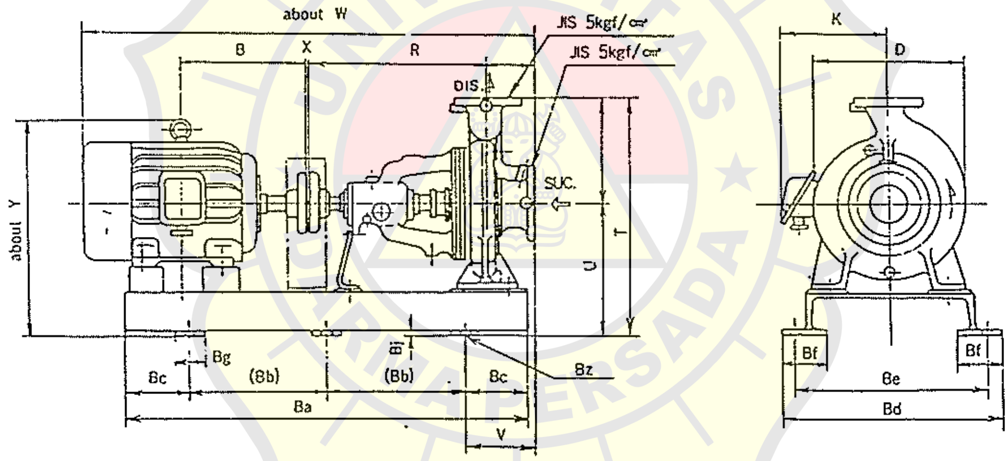


NAME & MATERIAL

PART NO.	NAME	REQ. NO.	SEA WATER MATERIAL	SEA WATER JIS	FRESH WATER MATERIAL	FRESH WATER JIS
001	CASING	1	BRONZE CASTING	BC3	CAST IRON	FC20
002	CASING COVER	1	BRONZE CASTING	BC3	CAST IRON	FC20
003	IMPELLER	1	PHOSPHOR BRONZE	PBC2	PHOSPHOR BRONZE	PBC2
020	CASING RING	2	BRONZE CASTING	BC3	BRONZE CASTING	BC3
050	O-RING	1	RUBBER	NBR	RUBBER	NBR
101	SHAFT	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
103	IMPELLER KEY	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
104	COUPLING KEY	1	CARBON STEEL	S45C-D	CARBON STEEL	S45C-D
120	IMPELLER NUT	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
124	IMPELLER WASHER	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
161	RETAINING RING	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304

PART NO.	NAME	REQ. NO.	SEA WATER MATERIAL	SEA WATER JIS	FRESH WATER MATERIAL	FRESH WATER JIS
170	FLINGER	1	RUBBER	NBR	RUBBER	NBR
201	BALL BEARING	1	BEARING STEEL	SUJ2	BEARING STEEL	SUJ2
202	BALL BEARING	1	BEARING STEEL	SUJ2	BEARING STEEL	SUJ2
205	BEARING HOUSING	1	CAST IRON	FC20	CAST IRON	FC20
209	BEARING COVER	1	CAST IRON	FC20	CAST IRON	FC20
302	COUPLING	1	CAST IRON	FC20	CAST IRON	FC20
406	SUPPORT	1	MILD STEEL	SS41	MILD STEEL	SS41
501	GLAND PACKING	4	ASBESTOS		ASBESTOS	
503	LANTERN RING	1	BRONZE CASTING	BC3	BRONZE CASTING	BC3
504	GLAND	1	BRONZE CASTING	BC3	BRONZE CASTING	BC3





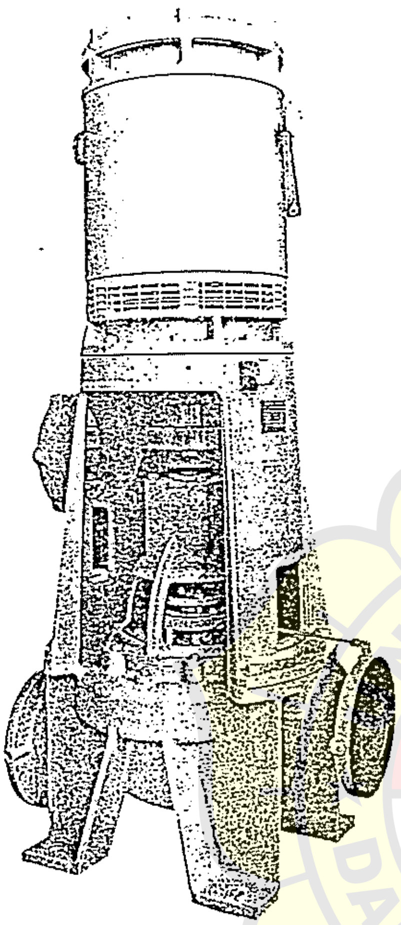
DIMENTION

TYPE	MOTOR kw	MOTOR r/m	BORE SUC.	BORE DIS.	A	B	D	K	R	T	U	V	W	X	Y	Ba	Bb	Bc	Bd	Be	Bf	Bg	Bj	Bz
	0.75					140		139					610		273									
HC-32	1.5	3600	32	32	80	169	270	148	260	330	190	180	665	3	284	500	350	75	275	240	55	50	12	4-φ15
	2.2					183		159					696		299	600	400	100						
	1.5					169		143					618		334									
EHC-41A	2.2	3600	40	40	80	183	207	159	360	400	240	145	793	3	349	700	450	125	325	290	55	50	12	4-φ15
	3.7					200		169					817		388	800	500	150						
EHC-41B	1.5	1800	40	40	80	169	260	148	360	455	275	145	762	3	368	700	450	125	325	290	55	50	12	4-φ15
	1.5	1800				169		148					782		368	700	450	125						
EHC-51B	2.2		50	50	100	183	268	159	360	450	270	165	813	3	379				325	290	55	50	12	4-φ15
	5.5	3600				239		206					913		438	800	500	150						
	2.2	1800				183		159					813		379	700	450	125	325	290	55	50		4-φ15
EHC-70B	3.7		65	65	100	200	286	169	360	470	270	165	937	3	418	800	500	150					12	
	7.5	3600				258		206					951		438	900	550	175	390	350	65	60		4-φ15
	3.7					200		169					837		438									
EHC-70C	5.5	1800	65	65	100	239	338	206	360	515	290	165	906	3	458	800	500	150	325	290	55	50	12	4-φ15
	7.5					258							951			900	550	175	390	350	65	60		4-φ15
	3.7					200							937		438									
EHC-81B	5.5	1800	80	80	100	239	326	206	360	515	290	165	906	3	458	800	500	150	325	290	55	50	12	4-φ15
	7.5					258							951			900	550	175	390	350	65	66		4-φ15
EHC-81C	7.5	1800	80	80	100	258	364	206	470	570	320	160	1054	3	502	1000	350	150	470	430	65	60	12	6-φ19
	11					323		228					1068		534	1100	400							
EHC-81D	7.5	1800	80	80	125	258	413	206	470	625	345	185	1086	3	502	1000	350	150	470	430	65	60	12	6-φ19
	11					323		226					1199		534	1100	400							

COOLING WATER PUMP

VERTICAL ONE STAGE SINGLE SUCTION PUMP

EVC SERIES



APPLICATION

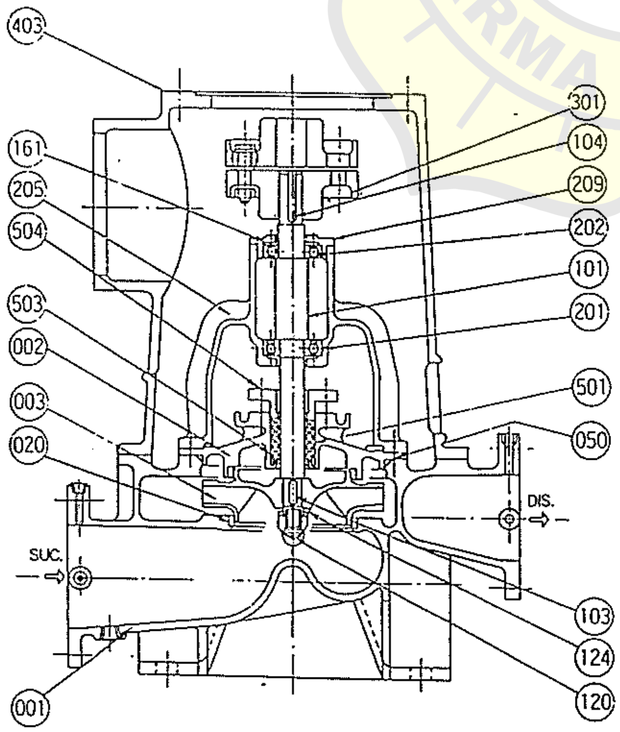
冷却清水、冷却海水、バラスト、海水サービス

Cooling Fresh Water, Cooling Sea Water, Ballast, Sea Water Service.

CONSTRUCTION

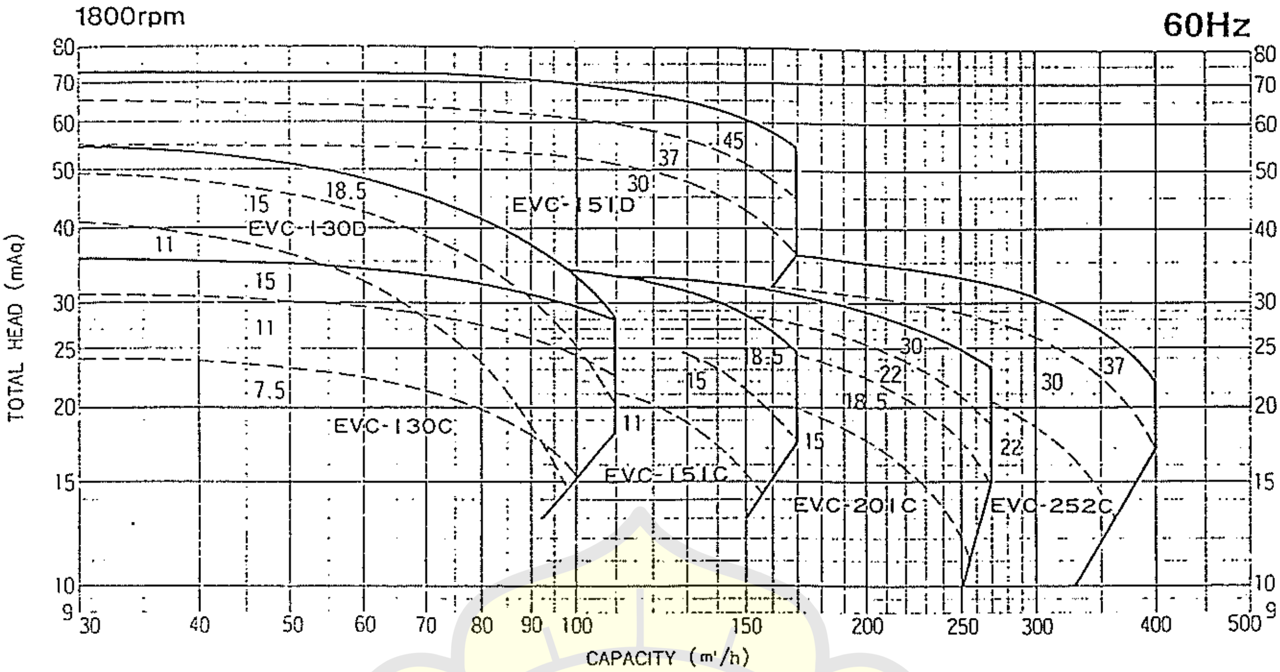
EVC形 ポンプは、立一体形ケーシング構造です。軸には、独立した２個の軸受を持ち、ドライ運転も問題ありません。真空ポンプと連動できます。

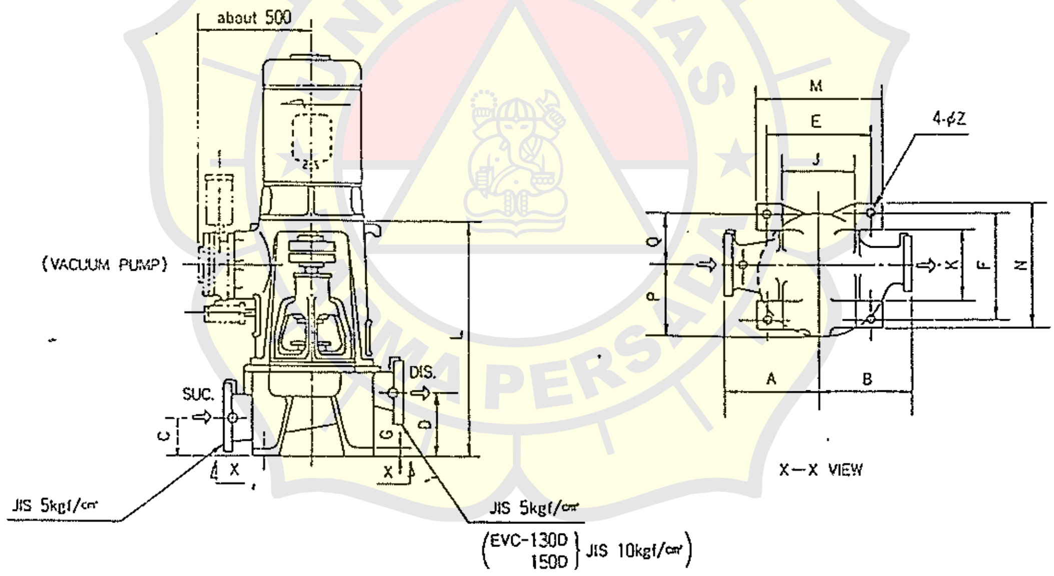
The pump casing of the EVC type is a vertical solid type. There is neither load to the motor bearing nor problem in case of the dry-operation, because pump shaft has two independent bearings. The pump can be used an automitic self-priming pump by fitting a vaccum pump, (type: N-20A)



NAME & MATERIAL

PART NO.	NAME	REQ. NO.	SEA WATER MATERIAL	SEA WATER JIS	FRESH WATER MATERIAL	FRESH WATER JIS
001	CASING	1	BRONZE CASTING	BC3	CAST IRON	FC20
002	CASING COVER	1	BRONZE CASTING	BC3	CAST IRON	FC20
003	IMPELLER	1	PHOSPHOR BRONZE	PBC2	PHOSPHOR BRONZE	PBC2
020	CASING RING	2	BRONZE CASTING	BC3	BRONZE CASTING	BC3
050	O-RING	1	RUBBER	NBR	RUBBER	NBR
101	SHAFT	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
103	IMPELLER KEY	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
104	COUPLING KEY	1	CARBON STEEL	S45C-D	CARBON STEEL	S45C-D
120	IMPELLER NUT	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
124	IMPELLER WASHER	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
161	RETAINING RING	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
201	BALL BEARING	1	BEARING STEEL	SUJ2	BEARING STEEL	SUJ2
202	BALL BEARING	1	BEARING STEEL	SUJ2	BEARING STEEL	SUJ2
205	BEARING HOUSING	1	CAST IRON	FC20	CAST IRON	FC20
209	BEARING COVER	1	CAST IRON	FC20	CAST IRON	FC20
301	COUPLING	1	CAST IRON	FC20	CAST IRON	FC20
403	FRAME	1	CAST IRON	FC20	CAST IRON	FC20
501	GLAND PACKING	4	CARBONIZED FIBER		CARBONIZED FIBER	
503	LANTERN RING	1	BRONZE CASTING	BC3	BRONZE CASTING	BC3
504	GLAND	1	BRONZE CASTING	BC3	BRONZE CASTING	BC3





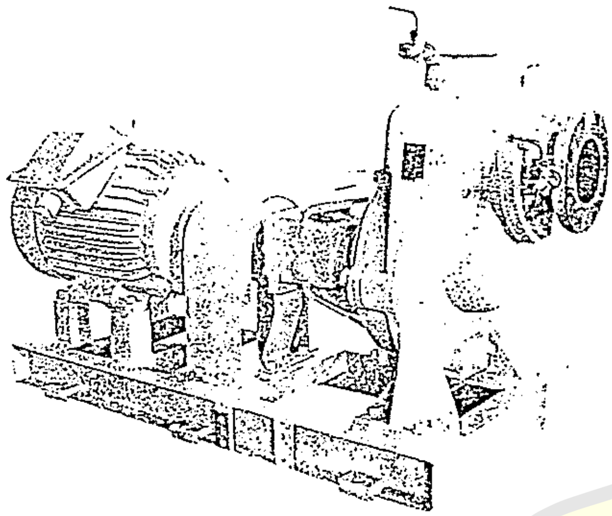
DIMENSION

TYPE	BORE		DIMENSION (mm)														
	SUC.	DIS.	A	B	C	D	E	F	G	J	K	L	M	N	P	Q	Z
EVC-130C	125	125	300	315	140	150	340	320	15	220	240	764	420	400	250	180	28
EVC-130D	125	100	345	345	150	190	380	380	25	250	250	882	450	450	305	225	28
EVC-151C	150	150	315	335	140	205	360	370	20	240	250	804	440	450	250	180	28
EVC-151D	150	125	345	345	150	190	380	380	23	250	250	980	450	450	290	290	28
EVC-201C	200	200	335	335	190	285	380	390	20	260	270	897	460	470	260	200	28
EVC-252C	250	250	380	400	215	285	420	420	25	260	280	1020	540	500	310	210	28

COOLING WATER PUMP

EHS SERIES

HORIZONTAL SINGLE SUCTION SELF-PRIMING PUMP



APPLICATION

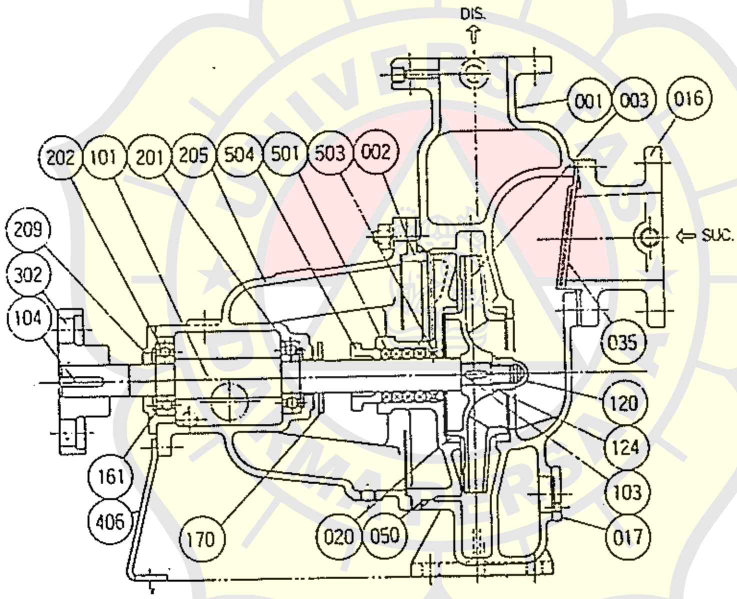
冷却清水、冷却海水、ビルジ、バラスト、雑用水

Cooling Fresh Water, Cooling Sea Water, Bilge, Ballast, General Service.

CONSTRUCTION

このポンプは、横形自吸式です。
軸封はグランドパッキン式が標準で、要望によりメカニカルシール式も製作します。

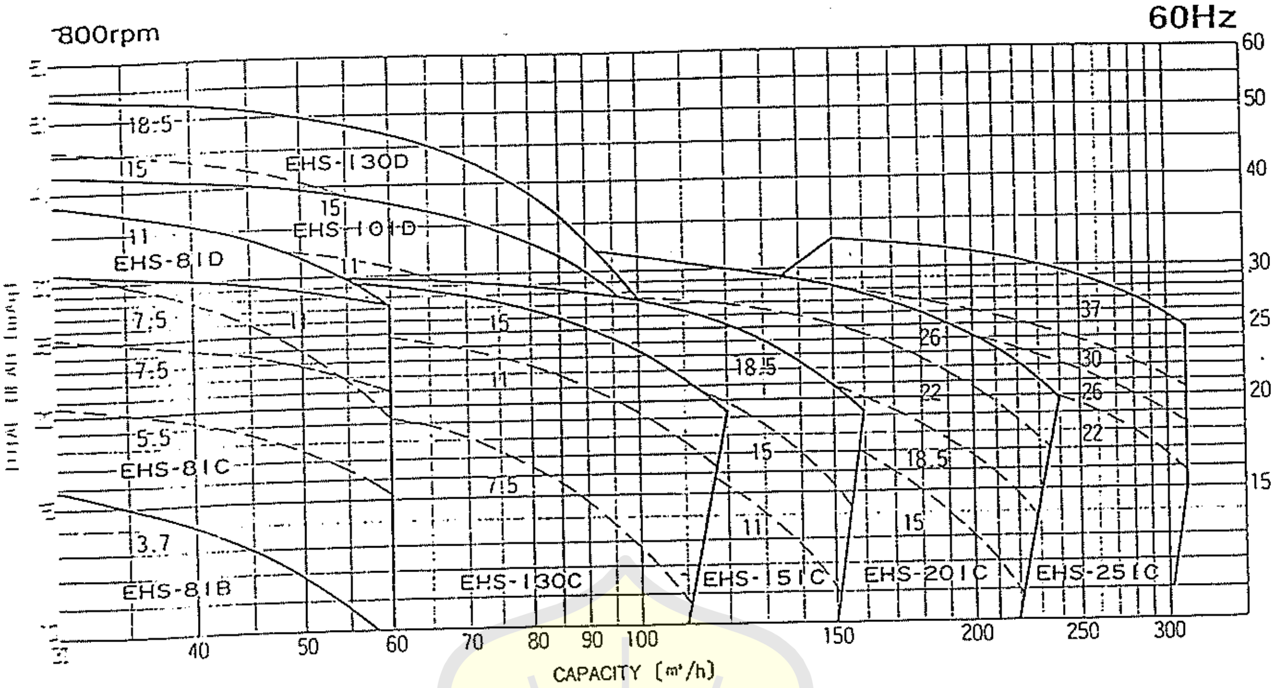
The EHS-type is horizontal self-priming pump.
The gland-packing is adopted as a standard method of shaft sealing.

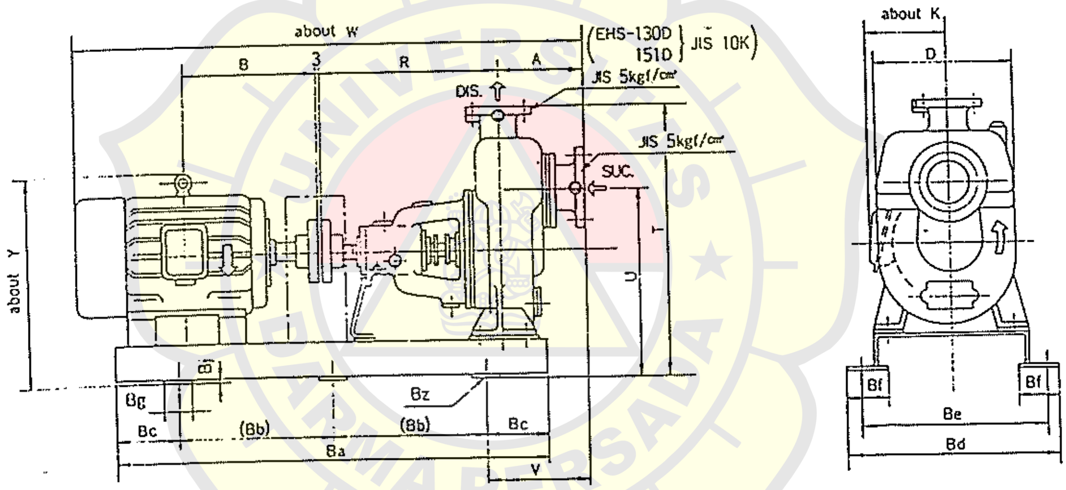


NAME & MATERIAL

PART NO.	NAME	REQ. NO.	SEA WATER MATERIAL	SEA WATER JIS	FRESH WATER MATERIAL	FRESH WATER JIS
001	CASING	1	BRONZE CASTING	BC3	CAST IRON	FC20
002	CASING COVER	1	BRONZE CASTING	BC3	CAST IRON	FC20
003	IMPELLER	1	PHOSPHOR BRONZE	PBC2	PHOSPHOR BRONZE	PBC2
016	SUCTION COVER	1	BRONZE CASTING	BC3	CAST IRON	FC20
017	DRAIN COVER	1	BRONZE CASTING	BC3	CAST IRON	FC20
020	CASING RING	2	BRONZE CASTING	BC3	BRONZE CASTING	BC3
035	CHECK VALVE	1	RUBBER	NBR	RUBBER	NBR
050	O-RING	1	RUBBER	NBR	RUBBER	NBR
101	SHAFT	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
103	IMPELLER KEY	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
104	COUPLING KEY	1	CABBON STEEL	S45CD	CARBON STEEL	S45CD
120	IMPELLER NUT	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304

PART NO.	NAME	REQ. NO.	SEA WATER MATERIAL	SEA WATER JIS	FRESH WATER MATERIAL	FRESH WATER JIS
124	IMPELEER WASHER	1	STAINLESS STEEL	SUS304	STAINLESS STEEL	SUS304
161	RETAINING RING	1	SPRING STEEL	SUP6	SPRING STEEL	SUP6
170	FLINGER	1	RUBBER	NBR	RUBBER	NBR
201	BALL BEARING	1	BEARING STEEL	SUJ2	BEARING STEEL	SUJ2
202	BALL BEARING	1	BEARING STEEL	SUJ2	BEARING STEEL	SUJ2
205	BEARING HOUSING	1	CAST IRON	FC20	CAST IRON	FC20
209	BEARING COVER	1	CAST IRON	FC20	CAST IRON	FC20
302	COUPLING	1SET	CAST IRON	FC20	CAST IRON	FC20
406	SUPRORT	1	MILD STEEL	SS41	MILD STEEL	SS41
501	GLAND PACKING	4	SEMI-METALLIC		SEMI-METALLIC	
503	LANTERN RING	1	BRONZE CASTING	BC3	BRONZE CASTING	BC3
504	GLAND	1	BRONZE CASTING	BC3	BRONZE CASTING	BC3





DIMENSION

TYPE	MOTOR		BORE		DIMENSION (mm)																		
	kw	r/m	SUC.	DIS.	A	B	D	K	R	T	U	V	W	Y	Ba	Bb	Bc	Bd	Be	Bf	Bg	Bi	Bz
EHS-81B	3.7	1800	80	80	230	200	280	200	365	585	430	300	990	455	800	500	150	390	350	65	60	12	4-ϕ15
EHS-81C	5.5 7.5 11	1800	80	80	205	239 258 323	340	210 315	470	670	470	265	1130 1170 1280	505 640	1000 1100	350 400	150	470	430	65	60	12	6-ϕ19
EHS-81D	7.5 11	1800	80	80	215	258 323	385	210 315	470	720	520	275	1180 1290	530 665	1000 1100	350 400	150	470	430	65	60	12	6-ϕ19
EHS-101D	11 15	1800	100	100	215	323 345	384	228 263	470	720	520	275	1283 1327	560	1100	400	150	470	430	65	60	12	6-ϕ19
EHS-130C	7.5 11 15	1800	125	125	225	258 323 345	358	260 228 263	470	700	495	285	1060 1293 1337	560	1100	400	150	470	430	65	60	12	6-ϕ19
EHS-130D	15 18.5	1800	125	100	280	345 351.5	416	275 315	575	770	545	340	1480 1520	535 555	1300	500	150	440	410	50	60	25	6-ϕ19
EHS-151C	11 15 18.5	1800	150	150	285	323 345 352	355	275 263 296	470	790	570	385	1190 1397 1413	560 560 595	1100	400	150	470	430	65	60	12	6-ϕ19
EHS-201C	15 18.5 22 26	1800	200	200	325	345 352 371 396	400	275 296 355	530	955	655	365	1190 1453 1491 1615	575 595 620 635	1100 1200	400 450	150	470	430	65	60	12	6-ϕ19
EHS-251C	22 26 30 37	1800	250	250	335	370.5 396 432	500	315 355 395	530	955	655	365	1350 1646 1695	605 625 680	1300	500	150	600	550	65	60	12	6-ϕ24

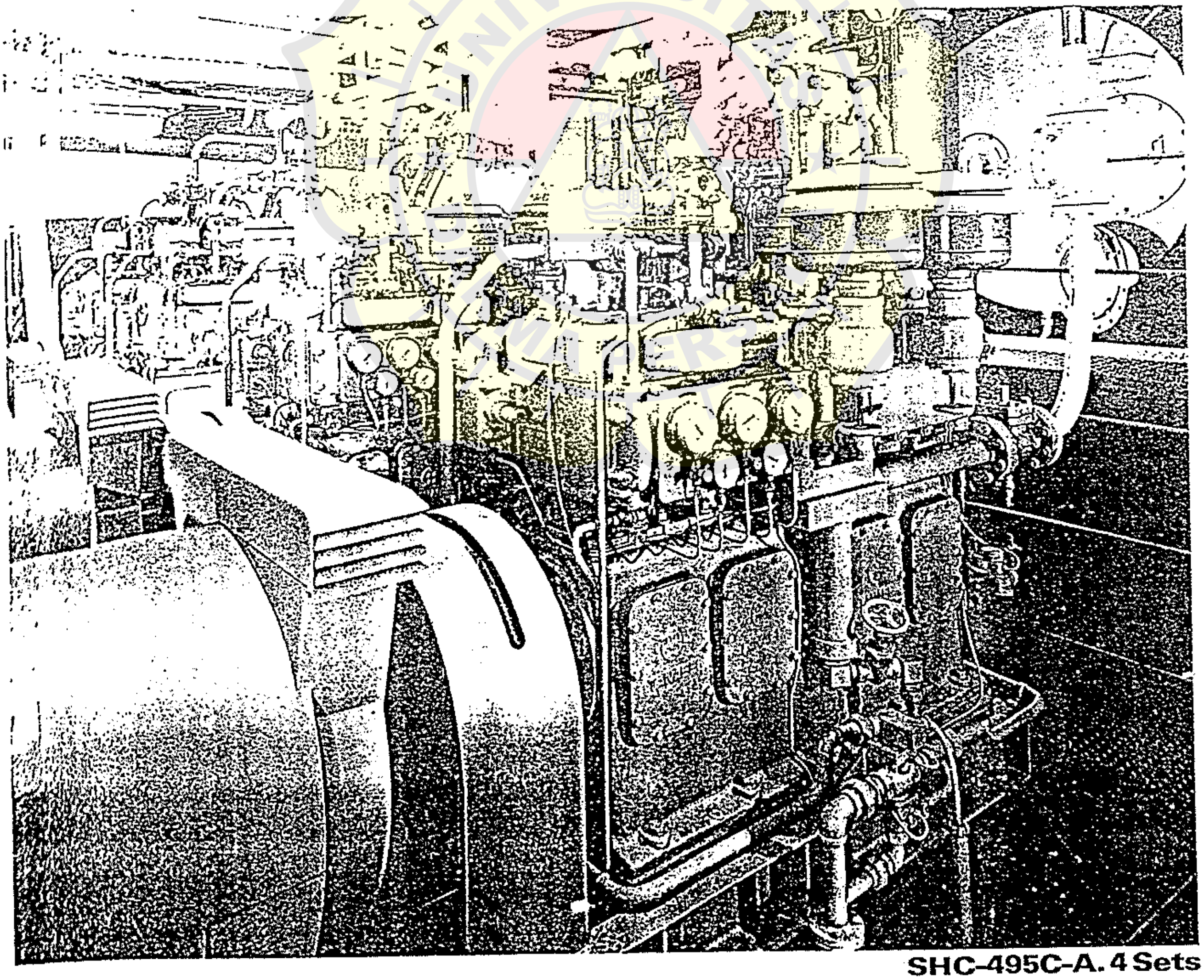
TANABE
MARINE COMPRESSORS

TANABE PNEUMATIC MACHINERY CO., LTD.
SENRIOKA near OSAKA, JAPAN

TANABE
MARINE COMPRESSORS

STARTING COMPRESSORS / AUTOMATIC CONTROL COMPRESSORS
GENERAL SERVICE COMPRESSORS / SOOT BLOWER'S COMPRESSORS
LIQUID GAS TRANSFER COMPRESSORS /

TANABE has been manufacturing marine compressors for more than 50 years. This long span has brought a thorough experience and exact knowledge of all requirements needed to meet marine use. TANABE marine compressors in air capacities from 5 to 600m³/hr at 25-30kg/cm² are ever-increasingly being installed in ships of all kinds from fishing craft, cargo vessel, container ship, oil tanker, bulk carrier and liquid gas tanker, to war ship.
Specification and Data on the different series are supplied in this bulletin. Fuller information will be gladly furnished by the specialist in our works for your approval.



SHC-495C-A. 4 Sets

TANABE'S LONG BROAD HARD EXPERIENCES ARE NOW AT YOUR SERVICE

ADVANTAGES

Advanced engineering features insure highest fficiency and endurance.

Balanced simple design conforms to all marime compressed air requirement.

Certified by all registers of shipping (LR, AB, BV, NV, GL and NK) through long experience in production.

Dual profit by low maintenance cost together with reliable performance.

SPECIFICATIONS

'alvas:
'anabe patented "Union Valve" of special alloy steel, in vhich suction and delivery valve work through one spring ommon to both.
arge valve area affords low lift and high volumetric efciency.
ipecially designed "concentric type plate valve" for high peed rotation. Practically trouble-free, long life, assuring eliable smooth operation in utmost severe working conitions.

:rankcase, Cylinder Block and Cylinder Head:
\re of close grained cast iron. They are separately mahined and assembled into one, ensuring perfect alignment f total motion work. For high-speed models, hardening rocess (Tuftride) of bore surface is applied.

:rankshaft:
s of forged steel with the journals finally ground finish to ine limits.

:onnectingrods:
\re of forged steel or special cast steel of high quality havng ample strength against any load and speed.

Aain Bearings:
\re precision type roller bearing correspond to Internationl standard. Close alignment is ensured by suitable iolder and lubricated properly to certify high durability.

:rank Pin Bearings:
\re of high grade whitemetal lined on bronze halves with djusting shims. Ample bearing area and well designed oil grooves afford smooth running.

Pistons:
Are of close grained cast iron or aluminium alloy having superior characteristics against high temperature. They are fitted with an adequate number of compression and oil scraper Rings.

Piston Pin:
Are of case-hardened steel, precision ground all over and fixed on piston body by snap rings, - full-floating type.

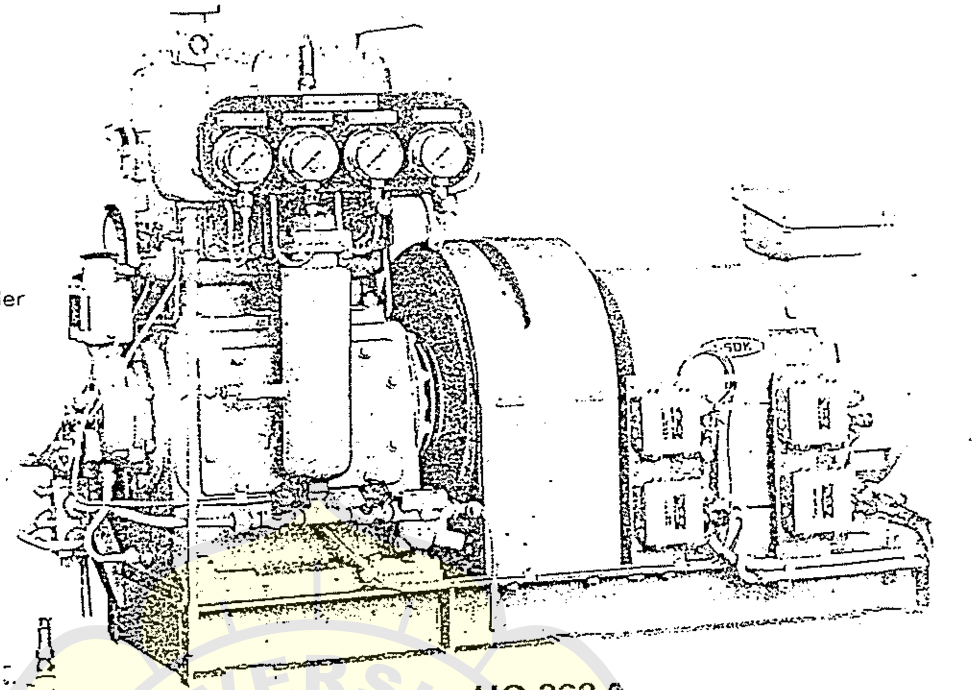
Cooling:
Ample capacity of cooling water jackets and efficient air coolers with copper or cuplo-nickel tube. Gear or centrifugal pumps are furnished for perfect water circulation.
Air cooling is attained by high speed axial fan, and multifinned intercooler and aftercooler keep compression performance in perfect state.

Drive:
Various types of driving are available for diesel and electric prime-mover.
Directly coupled through flexible coupling and manual, pneumatic or magnetic friction clutch.
V belt driving is also applied in case of speed reduction.

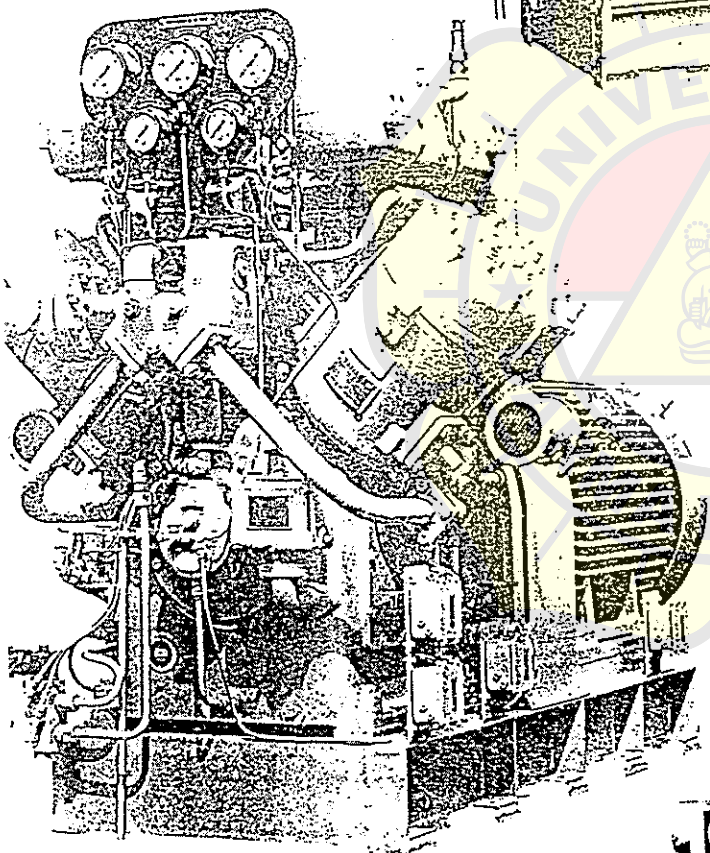
Lubrication:
Totally enclosed force-feed lubrication system by gear or plunger type oil pump. Pressure to all bearings.
Precision type oil filter, pressure regulating valve, and gauge assure reliable operation.

Control & Regulation:
Mechanical unloader of suction shut-off and suction valve open up types promise smooth continuous controlling operation. They are actuated both pneumatic and oil pressure system accompanied with pressure regulator.
Automatic control by pressure switch, magnetic valve and variable timer relay.
Emergency stop and alarm equipment are also available.
They are designed and arranged to meet the requirement of all sorts of modern automation systems in ships.

Single and double cylinder
2-stage water cooled
starting compressor



HC-260 A

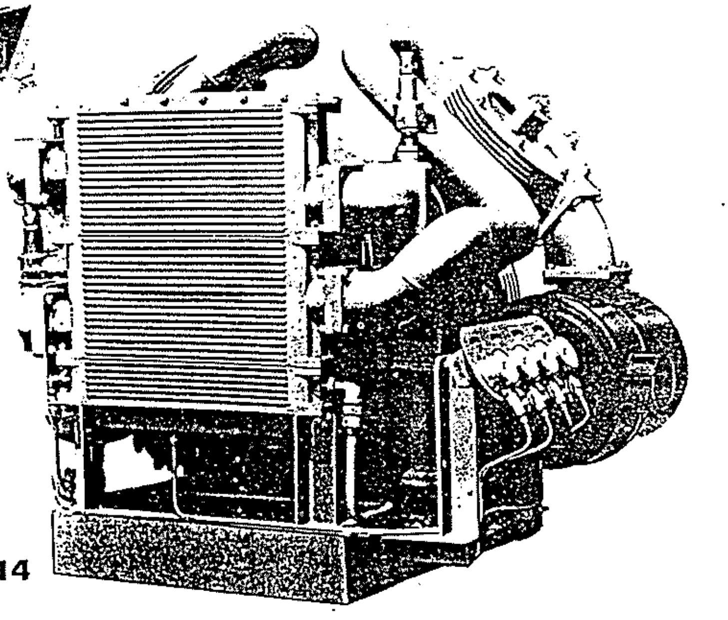


V-type four cylinder
2-stage water cooled
starting compressor

VH-470 D

V-type two and four cylinder
3-stage air cooled
starting compressor

VLHH-114



STARTING COMPRESSOR (Vertical 2-stage Water-cooled)

Model No.	Speed (r.p.m)	25 kg/cm^2			30 kg/cm^2		
		m^3/hr FA	PS	Motor(KW)	m^3/hr FA	PS	Motor(KW)
HC-54A	720	38	11	11	37	11.5	11
	900	47	14	11	46	14.5	15
HC-65A	720	68	19.5	18.5	66	20	18.5
	900	85	24.5	18.5	82	26	22
HC-65AS	1200	105	28.5	22	100	30.5	25
HC-234A	900	136	37	30	132	41	33
HC-265A	720	135	38	30	132	40	33
	900	170	49	37	164	52	40
HC-275A	720	195	51.5	40	190	54.5	45
	900	240	63.5	50	230	67	55
HC-277A	720	260	67	55	250	70	55
	900	310	82.5	65	300	87	70

STARTING COMPRESSOR (V-type 2-stage Water-cooled)

Model No.	Speed (r.p.m)	25 kg/cm^2			30 kg/cm^2		
		m^3/hr FA	PS	Motor(KW)	m^3/hr FA	PS	Motor(KW)
VH-475D	720	390	103	80	380	109	85
	900	480	127	100	460	134	110
VH-477D	720	520	134	110	500	140	110
	900	620	165	125	600	174	132

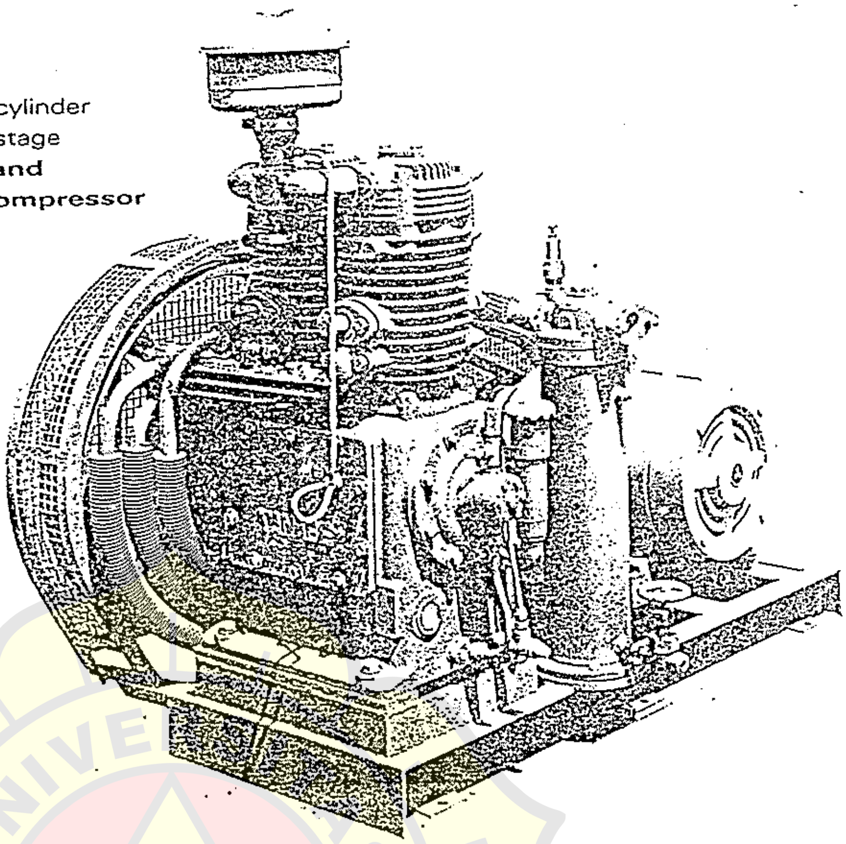
STARTING COMPRESSOR (Vertical 2-stage Water-cooled)

Model No.	Speed (r.p.m)	25 kg/cm^2			30 kg/cm^2		
		m^3/hr FA	PS	Motor(KW)	m^3/hr FA	PS	Motor(KW)
SHC-295C-A	720	285	72	55	280	76	65
	900	360	94	75	350	98	80
SHC-495C-A	720	570	144	110	560	152	125
	900	720	188	150	700	196	150

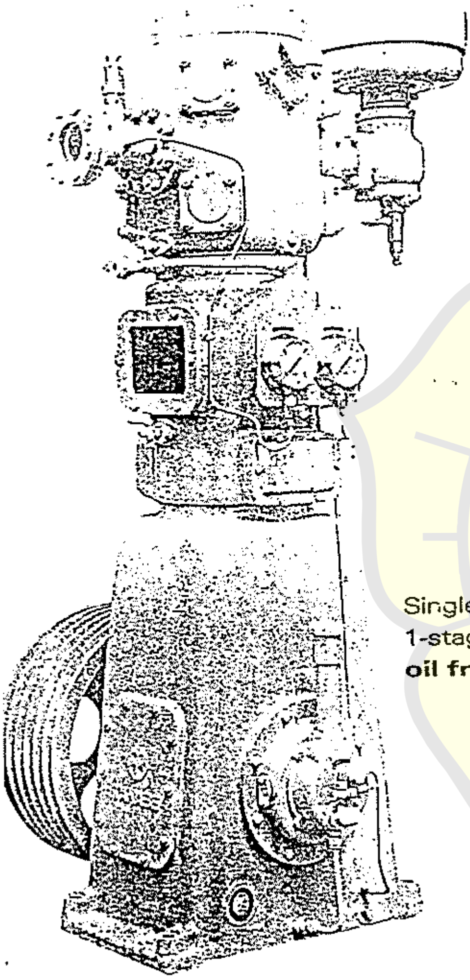
STARTING COMPRESSOR (V-type 3-stage Air-cooled)

Model No.	Speed (r.p.m)	25 kg/cm^2			30 kg/cm^2		
		m^3/hr FA	PS	Motor(KW)	m^3/hr FA	PS	Motor(KW)
VLHH-64	900	78	18	15	75	19.5	19
	1200	105	24	19	100	26	22
VLHH-74	900	115	27	22	110	28.5	25
	1200	155	36	30	150	38	30
VLHH-94	900	170	38	30	165	42	37
	1200	230	52	40	220	56	45
VLHH-114	900	260	57	45	250	62	50
	1200	350	78	60	340	83	65
VLHH-2114	720	410	94	70	400	198	75
	900	520	118	90	500	124	95

Single, double, four cylinder
vertical or V-type 2-stage
air cooled **control and**
general service compressor

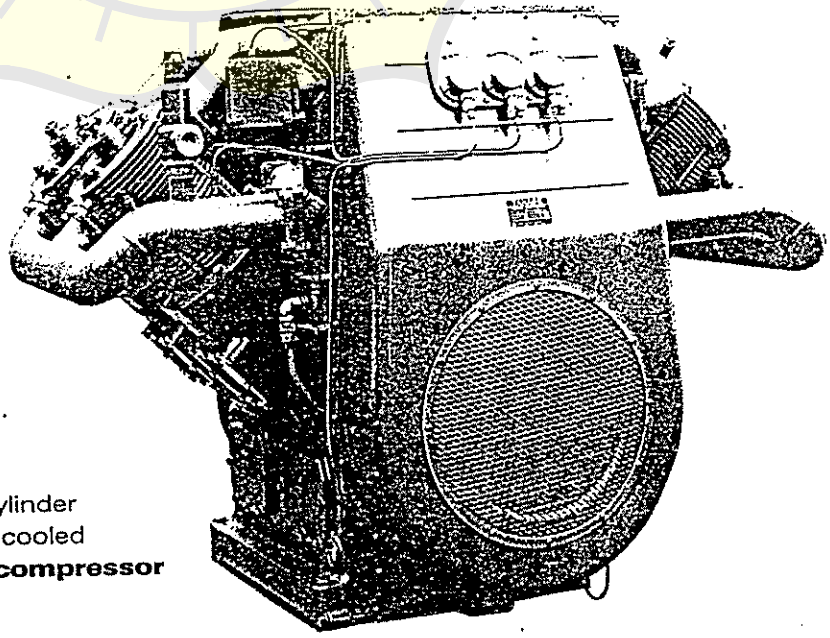


LHC-254A



Single, double and four cylinder
1-stage water cooled
oil free control compressor

OS-97A



Double and four cylinder
V-type 2-stage air cooled
oil free control compressor

VLHOS-2114

CONTROL and GENERAL SERVICE COMPRESSOR
(Vertical V-type 2-stage Air-cooled)

Model No.	Speed (r.p.m)	7 kg/cm²			9 kg/cm²		
		m³/hr FA	PS	Motor (KW)	m³/hr FA	PS	Motor (KW)
LHC-54A	720	40	6.7	5.5	38	7.2	5.5
	900	53	8.5	7.5	50	9.2	7.5
LHC-254A	720	80	13.5	11	76	14.5	11
	900	106	17.0	15	100	18.5	15
VLH-64	1200	95	16	15	90	18	15
	1500	120	20	19	115	22	19
VLH-74	1200	145	24	19	140	26	22
	1500	180	30	25	175	32	25
VLH-94	900	165	27	22	155	30	25
	1200	220	35	30	210	40	33
VLH-114	900	225	32	25	217	35	25
	1200	330	48	37	320	52	40
VLH-2114	900	450	64	50	435	70	55
	1200	660	94	75	640	103	80

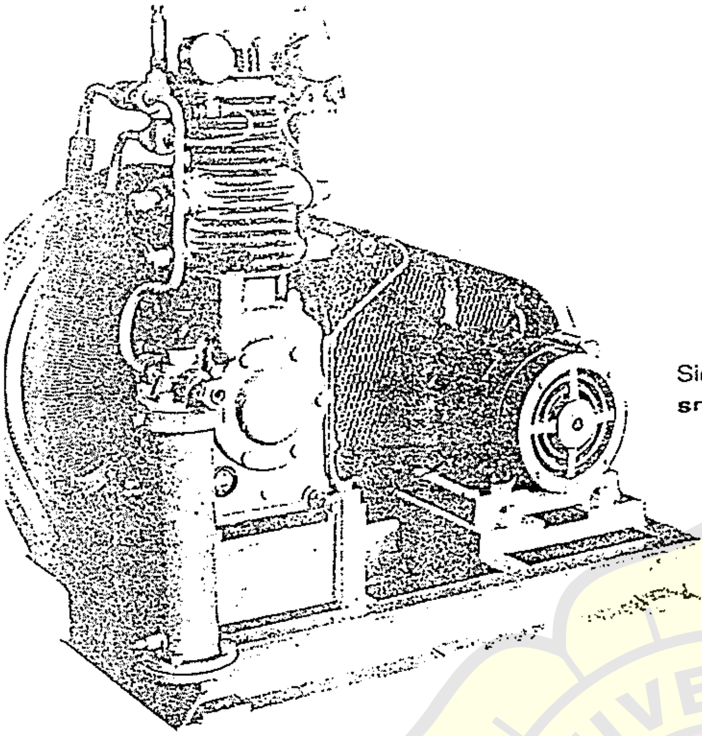
CONTROL COMPRESSOR (Vertical 1-stage Water-cooled Oil Free)

Model No.	Speed (r.p.m)	4 kg/cm²			7 kg/cm²		
		m³/hr FA	PS	Motor (KW)	m³/hr FA	PS	Motor (KW)
OS-54A	600	65	8.3	7.5	52	9.5	7.5
	720	78	9.9	7.5	62	11.5	11
OS-254B	600	130	17	15	104	19	15
	720	156	20.4	15	125	23	19
OS-265B	580	175	26	19	141	29	22
	660	201	29.5	22	160	33	25
OS-97A	560	290	43	25	250	49.5	40
	660	340	51	37	295	58	40
OS-297B	560	580	85	40	500	98	45
	660	680	100	75	590	116	90

CONTROL COMPRESSOR (V-type 2-stage Air-cooled Oil Free)

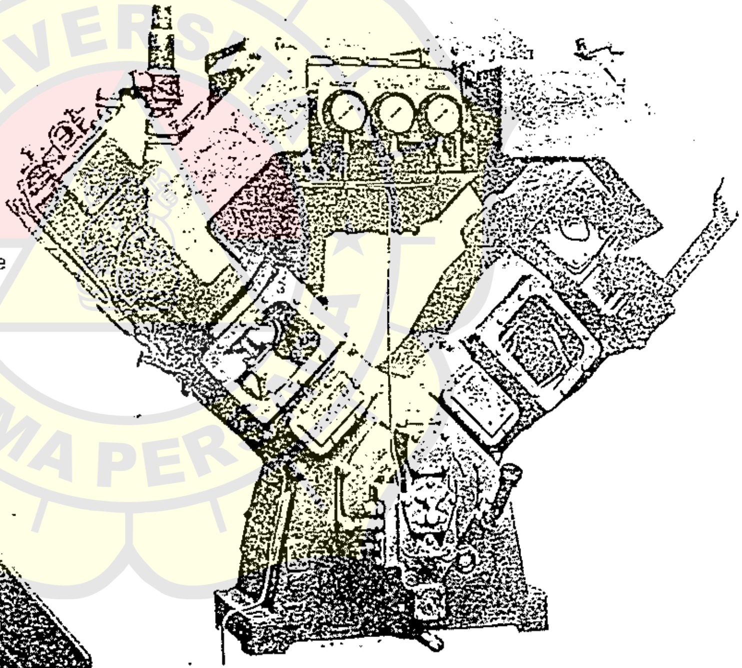
Model No.	Speed (r.p.m)	7 kg/cm²			9 kg/cm²		
		m³/hr FA	PS	Motor (KW)	m³/hr FA	PS	Motor (KW)
VLHOS-64	780	66	11	11	62	12	11
VHOS-64	1000	85	13.5	11	80	14	11
VLHOS-74	780	98	15.5	15	94	16	15
VHOS-74	1000	125	19.5	19	120	20	19
VLHOS-94	780	150	22	19	140	23.5	19
VHOS-94	1000	190	27	22	180	30	25
VLHOS-114	780	220	32	25	210	35	27
VHOS-114	1000	280	41	33	270	45	37
VLHOS-2114	780	440	64	50	420	70	55
VHOS-2114	1000	560	82	75	540	90	75

VLHOS: Air cooled VHOS: Water cooled



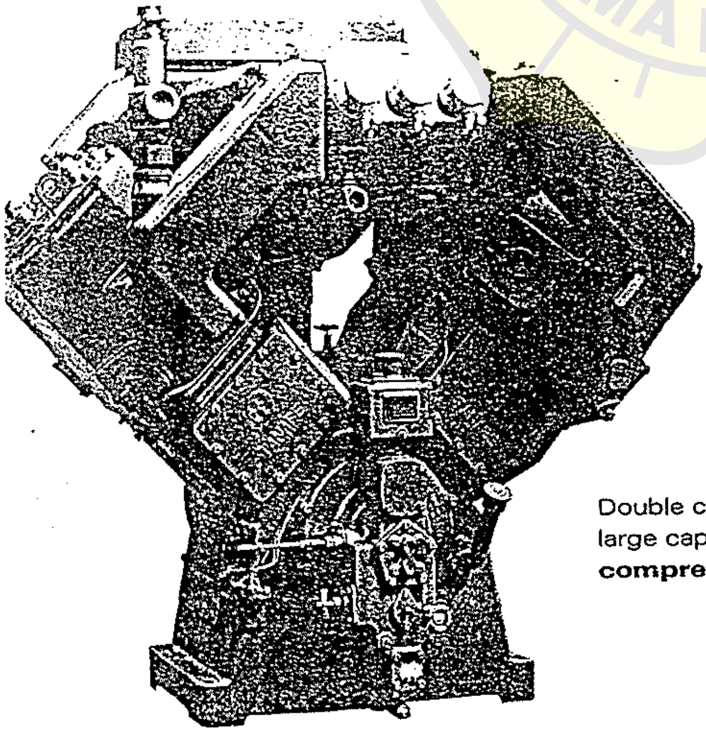
LSHC-30

Single cylinder 2-stage air cooled
small starting compressor



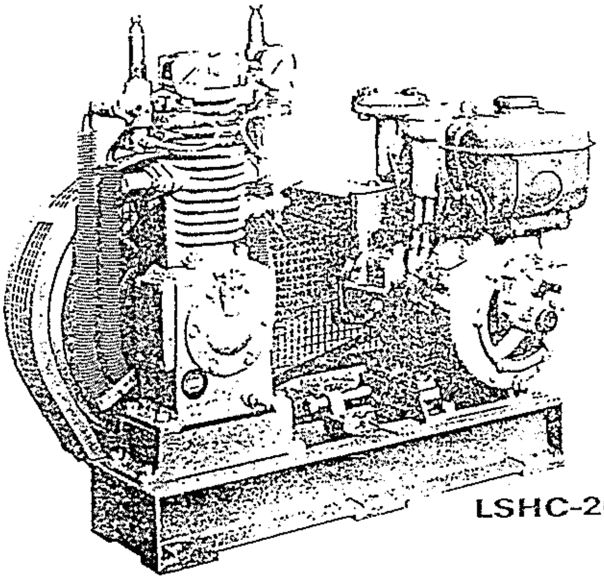
VHOS-145

Double cylinder double acting 2-stage
large capacity water cooled
oil free compressor



VH-165

Double cylinder double acting 2-stage
large capacity **general service**
compressor



LSHC-20B

Single cylinder 2-stage air-cooled
emergency starting compressor

EMERGENCY AND SMALL STARTING COMPRESSOR
(Vertical 2-stage Air and Water-cooled)

Model No.	Speed (r.p.m)	15 kg/cm²		25～30 kg/cm²	
		m³/hr FA	PS	m³/hr FA	PS
LSHC-20B	900	4.7	1.4	4.3	1.6
	1000	5.2	1.5	5.0	1.7
LSHC-30A	900	13.6	4.8	12.8	5.3
	1000	14.8	5.3	13.8	5.8
LSHC-40A	900	20.4	7.2	19.4	8.0
	1000	22.3	7.9	21.2	8.9
SHC-30C	900	13.8	4.7	13.0	5.2
	1000	15.0	5.2	14.0	5.7

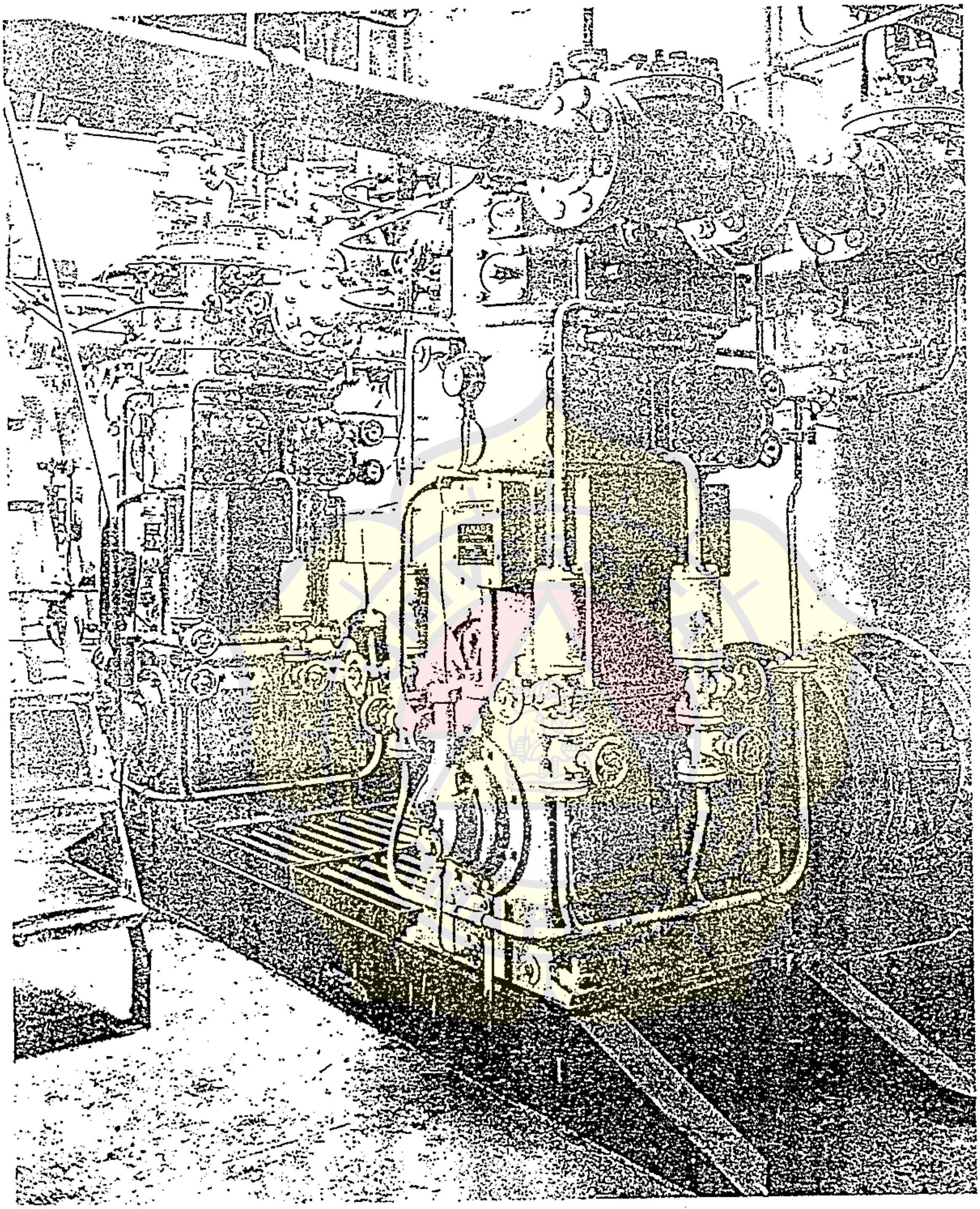
LSHC: Air cooled SHC: Water cooled

GENERAL SERVICE COMPRESSOR (V-type 2-stage Water-cooled)

Model No.	Speed (r.p.m)	7 kg/cm²			9 kg/cm²		
		m³/hr FA	PS	Motor (KW)	m³/hr FA	PS	Motor (KW)
VH-114	900	225	32	25	217	35	25
	1200	300	48	37	320	52	37
VH-2114	900	450	64	50	435	70	55
	1200	600	94	75	640	103	75
VH-145	900	1050	140	110	1034	139	110
	1000	1050	140	110	1035	139	110
VH-165	900	1410	185	150	1344	187	150
	1000	1410	185	150	1362	187	150

GENERAL SERVICE COMPRESSOR
(V-type 2-stage Water-cooled Oil Free)

Model No.	Speed (r.p.m)	7 kg/cm²			9 kg/cm²		
		m³/hr FA	PS	Motor (KW)	m³/hr FA	PS	Motor (KW)
VHOS-145	600	780	95	75	750	105	80
	720	930	110	90	900	125	95
VHOS-165	600	970	125	95	950	135	110
	720	1170	150	120	1140	160	125



LIQUID GAS TRANSFER COMPRESSOR (LPG COMPRESSOR)
(Vertical 1-stage Water-cooled Oil Free)

Model No.	Speed (r.p.m)	Gas Pressure (Max.) kg/cm^2	Pressure Difference kg/cm^2	Capaeity m^3/hr	PS
GOS-54A	400−600	18	1.5−3.0	50− 83	7.5− 15
GOS-254B	400−600	18	1.5−3.0	100−165	15− 30
GOS-265B	400−600	18	1.5−3.0	180−260	25− 50
GOS-97A	350−550	18	1.5−3.0	250−400	40− 75
GOS-297B	350−550	18	1.5−3.0	500−800	80−150

TANABE
MARINE COMPRESSORS

EXPLANATION of the CHART

1) The model No. of compressors shown in following charts are indicated by newly revised No. to which improved modifications are applied.
   (Example)

previous No.	revised No.
HC-65	HC-65A
VHC-475C	VH-475D

   A and D show modification symbols.
   The contents of principal modification and improvement are as follows:-
   - Range of operating speed (rpm) was increased for about 20%.
   - To ensure high speed performance, LP & HP valves were improved to high-speed type, and each stage cooler capacity was amplified.
   - Total balancing was fully improved.
   - Cylinder lubrication system was modified to direct oil forcing into cylinder wall, and application of wall hardening process was put into practice.

   Remarks:
   These new models are manufactured by the application of ISO Metric thread and screw except taper thread pipe jointings.

2) The delivery Free air Volume of Emergency, Control and Main air compressor is indicated by the volume measured according to charging test method.

3) The delivery Free air Volume of Control Oil-free compressor is indicated by the volume measured according to orifice metering method prescribe in JIS (Japan Industrial Standard); - mostly corresponds to DIN.

4) In case of direct-coupled Main air compressor, following considerations are paid in its performance.
   i. The delivery Free air Volume at each speed is indicated by that which involves estimated motor slippage. Therefore, the air compressor having 50m³/hr capacity at nominal speed of 900 rpm, it also fulfils that capacity at actual speed of about 875 rpm.
   ii. The power required at each speed and pressure is calculated according to the synchronized speed of electric motor.

SERVICE NETWORK FOR "TANABE" AIR COMPRESSORS

a) JAPAN

Kobe/Osaka ....
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Telex: 523-3537 TANABE J
Cable: TAPCO SUITA (OSAKA)

Tanabe International, Ltd.
15-24, Naganohigashi, Suita-City, Osaka
Cable: TICOSA SENRI OSAKA
Telex: 5286221 TAINTL J

Tokyo/Yokohama ....
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Tokyo Branch
6, 1-chome, Nihonbashi, Muromachi, Chuo-ku, Tokyo
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Telex: 222-2170 TAPCOT J

Tanabe Compressor Co., Ltd. Tokyo Works
785, Yukizato, Nakano, Ebina-City, Kanagawa-Pref.
Phone: 0462-31-1701
Telex: 3872-559 TACOMP J

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Phone: 212-966-2000
Telex: RCA 235514 ARGO UR

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Netherland.............
D. Van De Wetering B.V.
Bunschotenweg 134, Rotterdam-3022, Netherland
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Cable: DEVANDEW ROTTERDAM

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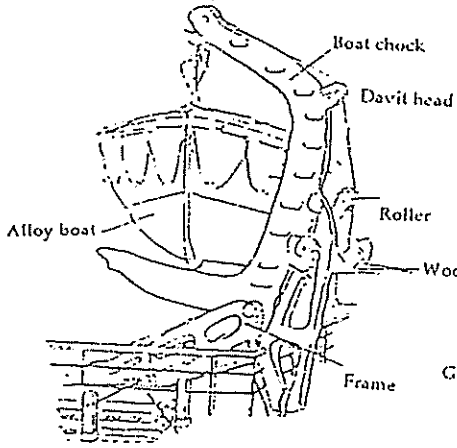
Argo International Corp.
117 Noorderlaan, Antwerp, Belgium
Phone: 031-42 2432
Telex: 32530 TANGHE B

STANDART UKURAN SEKOCI OLEH BOT (BOARD OF TRADE) ENGLAND

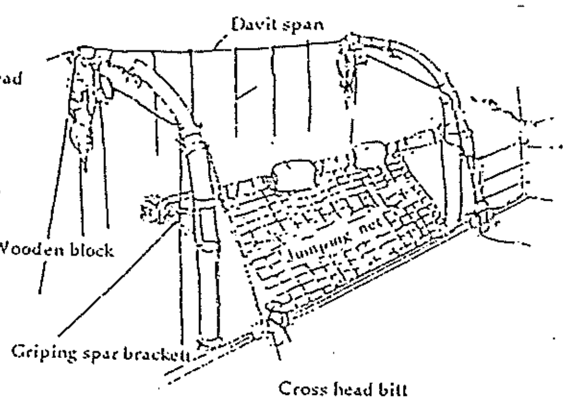
Tabel II

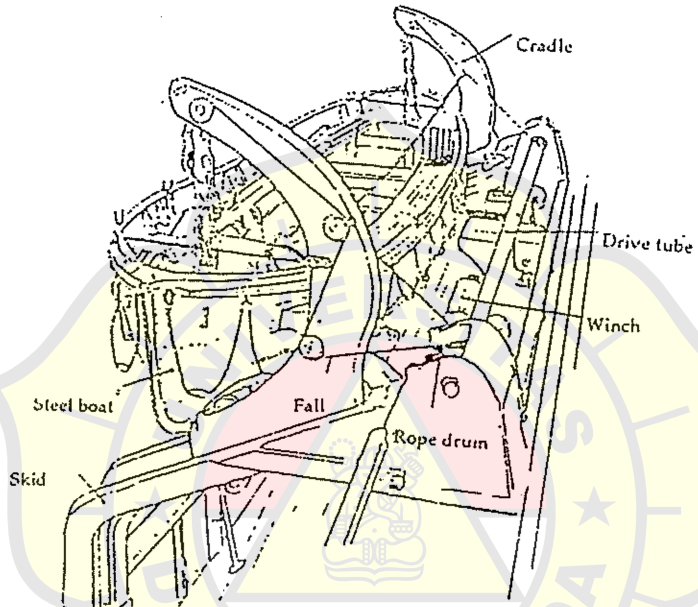
L. B. H (m)	L. B. H (ft3)	Kapasitas (ft3)	Jumlah orang	berat sekoci (kg)	Berat Orang (kg)	berat perlengkapan (kg)	Total berat (kg)
9,4 x 2,74 x 1 x 1,114	30 x 9 x 3,75	607	60	2205	4500	356	7061
8,84 x 2,74 x 1,10	29 x 8,75 x 3,60	545	54	1976	4050	356	6382
8,53 x 2,59 x 1,07	28 x 8,50 x 3,50	500	50	1824	3750	330	5894
8,23 x 2,51 x 1,04	27 x 8,25 x 3,40	454	45	1646	3376	330	5351
7,92 x 2,44 x 0,99	26 x 8,00 x 3,25	405	40	473	3000	305	4778
7,62 x 2,36 x 0,96	25 x 7,75 x 3,15	366	36	1326	2700	305	4331
7,31 x 22,9 x 0,91	24 x 7,50 x 3,00	324	32	1180	2400	254	3843
7,01 x 2,29 x 0,88	23 x 7,50 x 2,90	300	30	1087	2250	254	3591
6,71 x 2,21 x 0,84	22 x 7,25 x 2,75	236	26	955	1950	229	3134
6,40 x 2,13 x 0,82	21 x 7,00 x 2,70	238	23	864	1725	229	2818
6,10 x 2,06 x 0,79	20 x 6,75 x 2,60	210	21	762	1575	203	2540
5,79 x 1,98 x 0,76	19 x 6,50 x 2,50	182	18	650	1350	178	2178
5,49 x 1,90 x 0,73	18 x 6,25 x 2,40	162	16	590	1200	152	1942
5,18 x 1,83 x 0,715	17 x 6,00 x 2,30	143	14	508	1050	152	1710
4,88 x 1,75 x 0,70	16 x 5,75 x 2,30	127	12	475	900	127	1484

Gravity type davit

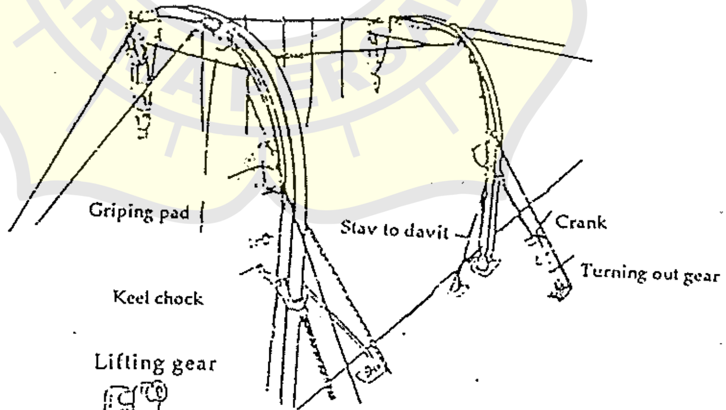


Radial davit

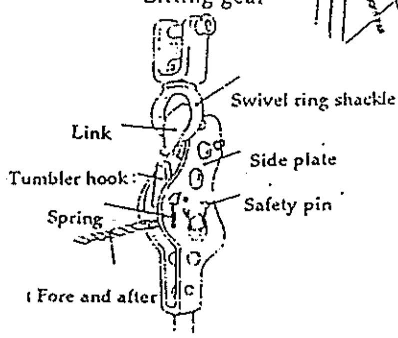




Luffing type davit



Lifting gear



Gambar