# BAB IV
# PENUTUP

Dengan selesainya penyusunan tugas merancang ini, maka penulis dapat mengambil kesimpulan yang berhubungan dengan perencanaan kapal Ferry Ro – Ro 821,76 DWT sebagai sarana angkutan laut yang dapat menunjang perkembangan transportasi di Indonesia. Adapun kesimpulan penulisan tersebut adalah sebagai berikut :

1. Ringkasan spesifikasi teknis dari kapal Ferry Ro – Ro 821,76 DWT :
   - Panjang seluruhnya (Loa) = 90,79 m
   - Panjang antar garis tegak (Lpp) = 81,00 m
   - Lebar (B) = 15,60 m
   - Tinggi (H) = 3,75 m
   - Sarat air (T) = 5,00 m
   - Koefisien blok (Cb) = 0,56
   - Koefisien prismatik (Cp) = 0,574
   - Koefisien garis air (Cw) = 0,67364
   - Koefisien tengah kapal (Cm) = 0,975
   - Displasemen ($\Delta$) = 2719,899 ton
   - Volume ($\nabla$) = 2511,405 $m^3$
   - Jumlah anak buah kapal (ABK) = 30 orang
   - Alat penggerak yang digunakan :

     Jumlah Mesin : 2 (satu) buah
     Merk : Wartsila Vasa 22
     Tipe : In line 8R 22
     Daya : 1500 kW / 2040 HP
     Putaran mesin : 1.100 rpm
     Bore x Stroke : 220 mm x 240 / 260 mm

Diameter Propeller : 2,6 m

Jumlah daun : 4 (empat) buah

- Kecepatan dinas (Vs) = 16,00 Knot

2. Dalam rancangan, untuk dapat menentukan besarnya daya motor induk sebagai penggerak utama kapal, maka faktor kecepatan, daerah pelayaran serta dimensi dari kapal rancangan mempunyai pengaruh yang sangat besar

3. Dalam menentukan generator set didasarkan pada pembebanan penggunaan daya yang terbesar yaitu pada saat kapal melakukan manuver sebesar 1651,55 kW, dengan menggunakan 3 buah generator masing-masing berkapasitas 680 kW daya yang dibutuhkan dapat terpenuhi.

4. Dalam perancangan kamar mesin, tidak terlepas adanya asumsi – asumsi yang diberikan untuk mempermudah dalam perhitungan dengan tidak mengabaikan tanggung jawab secara teknis, ekonomis serta peraturan-peraturan yang ada sehingga hasil perhitungan dapat mendekati keadaan yang sebenarnya.

5. Tata letak mesin induk , mesin bantu serta permesinan lainnya diatur seefisien mungkin , hal ini untuk mempermudah dalam hal perawatan dan perbaikan peralatan yang ada dikamar mesin serta tata letaknya sangat berpengaruh pada stabilitas kapal.

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## DAFTAR NOTASI

- A : Luas bidang
- B : Lebar kapal
- b : Tinggi daun kemudi
- BHP : Brake horse power
- Cr : Gaya daun kemudi
- CR : Koefisien tahanan sisa
- CF : Koefisien tahanan gesek
- CA : Koefisien tahanan tambahan
- Cw : Faktor koreksi beban
- d : Diameter
- db : Diameter pipa ballast
- D : Diameter silinder mesin induk (bore)
- Dcl : Diameter efektif panjang rantai
- df : Diameter pipa discharge sistem pemadam kebakaran
- dpb : Diameter pipa bilga
- Dt : Diameter poros kemudi
- dw : Diameter tali tambat
- dz : Diameter cabang sistem bilga
- Fn : Froude number
- Ga : Berat jangkar
- g : Gravitasi
- H : Tinggi kapal, langkah torak mesin induk ( stroke )
- ha : Head statis total
- He : Head kerugian sistem ventilasi
- Hd : Head dinamis
- hi : Head total sistem
- hl : Head kerugian saluran, katup, dll

- HP : Daya kuda / horse power
- Ia : Ratio antara putaran motor dengan putaran cable lifter
- Icl : Panjang rantai untuk satu putaran cable lifter
- Iw : Ratio antara putaran motor dengan putaran cable lifter
- J : Kapasitas botol angin / start
- K : Koefisien hambatan untuk katup dan lifting
- L : Letak midship section, panjang pipa
- La : Panjang rantai yang menggantung
- lb : Lebar ruangan
- LCB : Letak titik tekan keatas terhadap midship section
- Lpp : Panjang kapal antara dua garis tegak
- Lwl : Panjang garis air
- Mcl : Torsi pada cable lifter
- Mm : Torsi poros motor
- N : Putaran mesin, putaran propeller
- Ncl : Putaran cable lifter
- Ne : Daya efektif pompa
- Nth : Daya kompresor
- Nw : : Putaran poros penggulung tali tambat
- Pa : Tekanan kerja maksimum botol udara start
- Pb : Tekanan kerja minimum botol udara start
- PC : Koefisien propulsi
- Pe : Tekanan udara luar
- Q : Kapasitas aliran fluida, kapasitas kompresor udara
- Qc : Kapasitas fan
- rz : Koreksi kerugian pada sistem transmisi
- $r_3$ : Koreksi karena perubahan B/T kapal terhadap B/T standard

- Rbr : Beban putus tali tambat
- Re : Reynold number
- Rpm : Putaran mesin per menit / rotation per minute
- RT : Tahanan total
- S : Jarak pelayaran, luas basah kapal
- SFOC : Pemakaian bahan bakar spesifik untuk mesin induk
- T : Sarat kapal
- t : Waktu
- Tcl : Gaya tarik untuk menarik dua jangkar
- Tmaks : Torsi maksimum daun kemudi
- Tmin : Torsi minimum daun kemudi
- Tw : Gaya tarik pada penggulung di capstan
- V : Volume ruangan, kecepatan
- Va : Kecepatan angkat rantai jangkar rata-rata
- Vb : Volume tangki ballast
- Vco : Volume bahan bakar motor bantu
- Vfo : Volume bahan bakar motor induk
- Vfw : Volume kebutuhan air tawar
- Vlost : Volume tangki minyak pelumas
- Vs : Kecepatan dinas kapal
- Vsil : Volume minyak pelumas silinder
- Vst : Volume tangki settling bahan bakar
- Vtfw : Volume tangki air tawar
- Vtfo : Volume tangki bahan bakar motor induk
- Vw : Kecepatan tarik tali tambat
- Wb : Berat air ballast
- Wco : Berat minyak pelumas mesin induk
- Wdo : Berat bahan bakar motor bantu
- Wfo : Berat bahan bakar motor induk
- Wfwd : Berat air tawar untuk makan dan minum

# LEMBAR PERBAIKAN

Nama : I.B. NUR LUKINTO UTAMA

NIM / NIRM : 95320006 / 953123743250006

Jurusan : Teknik Permesinan Kapal

Panitia penguji menyatakan bahwa mahasiswa yang bersangkutan telah diuji dan harus membuat perbaikan dan diselesaikan dalam waktu 1 (satu) bulan.

| No | Dosen penguji | Jenis perbaikan | Selesai diperbaiki | Paraf dosen penguji |
|---|---|---|---|---|
| | | | | |

Jakarta,...............

Mengetahui
Dekan FTK

Ketua Dosen Penguji

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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:

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

$c_1$ = factor for the ship type

= 1,0 in general

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

= 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.

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14 - 2



**Materials**

For materials for rudder stock, pintles, cou- bolts etc. see Rules for Material Volume V. ...ial material requirements are to be observed for ...e notations ES3 and ES4 as well as for the arctic ...otations Arc 1 - Arc 4.

In general materials having a minimum ...inal upper yield point $R_{eH}$ of less than 200 ...m² and a minimum tensile strength of less than ... N/mm² or more than 900 N/mm² shall not be ...d for rudder stocks, pintles, keys and bolts. The ...irements of this Section are based on a material's ...imum nominal upper yield point $R_{eH}$ of 235 ...m². If material is used having a $R_{eH}$ differing ...m 235 N/mm², the material factor $k_r$ is to be ...mined 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/mm²]. $R_{eH}$ is not to be taken greater than $0,7 \cdot R_m$ or 450 N/mm², whichever is less. $R_m$ = tensile strength of the material used.

Before significant reductions in rudder stock ...meter due to the application of steels with $R_{eH}$ ...eding 235 N/mm² are granted, the Society may ...uire the evaluation of the elastic rudder stock ...ections. Large deflections should be avoided in ...er to avoid excessive edge pressures in way of ...rings.

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

**Definitions**

$C_R$ = rudder force in [N]

$Q_R$ = rudder torque in [Nm]

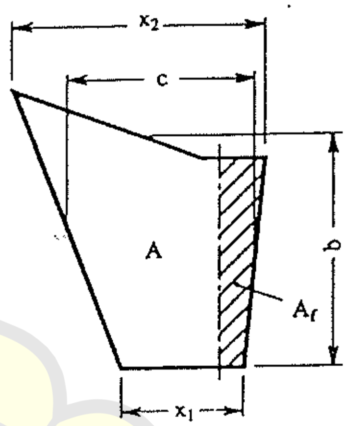
A = total movable area of the rudder in [m²]
For nozzle rudders, A is not to be taken less than 1,35 times the projected area of the nozzle;

$A_t$ = A + area of a rudder horn, if any, in [m²]

$A_f$ = portion of rudder area located ahead of the rudder stock axis in [m²]

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

$\Lambda$ = 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 $\kappa_2$ 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 ac-

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cording 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

$\kappa_1$ = coefficient, depending on the aspect ratio $\Lambda$

$\kappa_1$ = $(\Lambda + 2)/3$, where $\Lambda$ need not be taken greater than 2

$\kappa_2$ = coefficient, depending on the type of the rudder and the rudder profile according to Table 14.1.

$\kappa_3$ = coefficient, depending on the location of the rudder

$\kappa_3$ = 0,8 for rudders outside the propeller jet

$\kappa_3$ = 1,15 for rudders aft of the propeller nozzle

$\kappa_3$ = 1,0 elsewhere, including also rudders within the propeller jet

$\kappa_t$ = coefficient depending on the thrust coefficient $c_t$

$\kappa_t$ = 1,0 normally

In special cases for thrust coefficients $c_t > 1{,}0$ determination of $\kappa_t$ 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 | $\kappa_2$ ahead | $\kappa_2$ astern |
|---|---|---|
| NACA-00 series Gottinger 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]

$\alpha$ = 0,33 for ahead condition

$\alpha$ = 0,66 for astern condition (general)

$\alpha$ = 0,75 for astern condition (hollow profiles)

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

$\alpha$ = 0,25 for ahead condition

$\alpha$ = 0,55 for astern condition.

For high lift rudders $\alpha$ 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

$c_1 = A_1/b_1$

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$z = A_2/b_2$

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

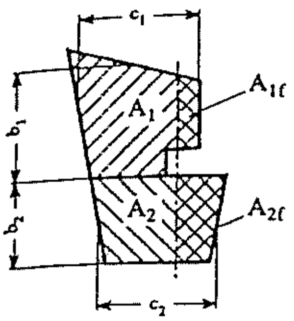


Fig. 14.2

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

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

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

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

for ahead condition

The greater value is to be taken.

## Scantlings of the Rudder Stock

### Rudder stock diameter

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

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

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

The related torsional stress is:

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

$k_r$ 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:

$$\sigma_v = \sqrt{\sigma_b^2 + 3\tau^2} \leq 118/k_r \quad [N/mm^2]$$

Bending stress:

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

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

Torsional stress:

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

$D_1$ = 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.*

# Section 18

# Equipment

## A. General

1. The equipment of anchors. chain cables, wires and ropes is to be determined from Table 18.2 in accordance with the equipment numeral Z.

*Guidance*

*1. The anchoring equipment required by this Section is intended of temporary mooring of a vessel within a harbour or sheltered area when the vessel 's awaiting berth, tide, etc.*

*2. The equipment is, therefore, not designed to ıold a ship off fully exposed coasts in rough weather ır to stop a ship which is moving or drifting. In this ondition the loads on the anchoring equipment ıcrease to such a degree that its components may e damaged or lost owing to the high energy forces enerated, particularly in large ships.*

*The anchoring equipment required by this ection is designed to hold a ship in good holding round in conditions such as to at dragging of the ıchor. In poor holding ground the holding power the anchors will be significantly reduced.*

*The equipment numeral formula for anoring equipment required under this Section is sed on an assumed current speed of 2,5 m/sec, ıd speed of 25 m/sec and a scope of chain cable ween 6 and 10, the scope being the ratio between gth of chain paid out and water depth.*

*It is assumed that under normal circumnces a ship will use only one bow anchor and ıin cable at a time.*

Every ship is to be equipped with at least one hor windlass.

ıdlass and chain stopper, if fitted, are to comply ı Volume III, Section 14, D.

For the substructures of windlasses and chain stoppers, see Section 10, B.5.

For the location of windlasses on tankers, see Section 24, A.9.

3. For ships having the navigation notation "L" (Small Coasting Service) affixed to their character of classification, the equipment may be determined as for one numeral range lower than required in accordance with the equipment numeral Z.

4. When determining the equipment for ships having the navigation notation "T" (Shallow Water Service) affixed to their character of classification, the provisions of Section 30, E. are to be observed.

5. When determining the equipment for tugs, Section 27, G. is to be observed.

When determining the equipment for fishing vessels, Section 28, D.8. is to be observed.

When determining the equipment of barges and pontoons, Section 31, G. is to be observed.

6. Ships build under survey of BKI and which are to have the mark stated in their Certificate and in the Register Book must be equipped with anchors and chain cables complying with the Rules for Materials and having been tested on approved machines in the presence of Surveyor.

7. For ships having three or more propellers, a reduction of the weight of the bower anchors and the chain cables may be considered.

## B. Equipment numeral

The equipment numeral is to be calculated as follows:

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$$Z = D^{2/3} + 2\,h\,B + \frac{A}{10}$$

D = moulded displacement in [ton] (in sea water having a density of 1,025 $t/m^3$) to the summer load waterline

h = effective height from the summer load waterline to the top of the uppermost house

$h = f_b + \sum h'$

$f_b$ = freeboard in [m], from the summer load waterline amidships

A = area in [$m^2$], in profile view of the hull, superstructures and houses, having a breadth greater then B/4, above the summer load waterline within the length L and up to the height h

$\sum h'$ = sum of height in [m] of superstructures and deckhouses, on the centreline of each tier having a breadth greater than B/4. Deck sheer, if any, is to be ignored. For the lowest tier, "h" is to be measured at centreline from the upper deck or from a notional deck line where there is local discontinuity in the upper deck.

Where a deckhouse having a breadth greater than B/4 is located above a deckhouse having a breadth of B/4 or less, the wide house is to be included and the narrow house ignored.

Screens of bulwarks 1,5 m or more in height are to be regarded as parts of houses when determining h and A, e.g. the area shown in Fig. 18.1 as $A_1$ is to be included in A. The height of the hatch coamings and that of any deck cargo, such as containers, may be disregarded when determining h and A.

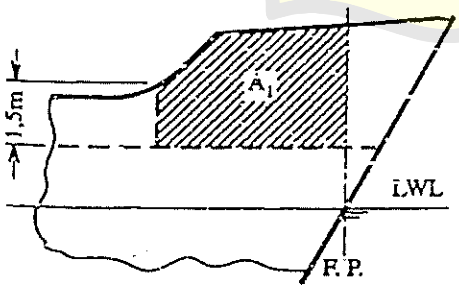


Fig. 18.1

C. Anchors

1. Two of the rule bower anchors are to be connected to their chain cables and positioned on board ready for use. Where in column 3 of table 18.2 three bower anchors are required the third anchor is intended as a spare bower anchor. Installation of the spare bower anchor on board is not required. Upon agreement by the owner the spare anchor may even be dispensed with.

*Guidance*

*National regulations concerning the provision of a spare anchor may need to be observed.*

2. Anchors must be of approved design. The mass of the heads of patent (ordinary stockless) anchors, including puns and fittings, is not to be less than 60 percent of the total mass of the anchor.

3. For stock anchors, the total mass of the anchor, including the stock, shall comply with the values in Table 18.2. The mass of the stock shall be 20 percent of this total mass.

4. The mass of each individual bower anchor may vary by up to 7 per cent above or below the required individual mass provided that the total mass of all the bower anchors is not less than the sum of the required individual masses.

5. Where special anchors approved as "High Holding Power Anchors" are used. the anchor mass may be 75 per cent of the anchor mass as per Table 18.2.

"High Holding Power Anchors" are anchors which are suitable for ship's use at any time and which do not require prior adjustment or special placement on the sea bed.

For approval as a "High Holding Power Anchor", satisfactory tests are to be made on various types of bottom and the anchor is to have a holding power at least twice that of a patent anchor ("Admiralty Standard Stockless") of the same mass. The weights of anchors to be tested should be representative of the full range of sizes intended to be manufactured. The tests are to be carried out on at least two sizes of anchors in association with the chain cables appropriate to the weight. The anchors to be tested and the standard stockless anchors should be of approx. the same mass.

The chain length used in the tests should be approx. 6 to 10 times the depth of water.

18 — 6



## Table 18.2 Anchor, Chain Cables and Ropes

| Equipment numeral Z | Stockless anchor | | | Stud link chain cables | | | | | | Recommended ropes | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Bower anchor | | Stream anchor | Bower anchors | | | | Stream wire or chain for stream anchor | | Towline | | Mooring ropes | | |
| | Number[1] | Mass per anchor | | Total length | Diameter $d_1$ | $d_2$ | $d_3$ | Length | Br. load[2] | Length | Br. load[2] | Number | Length | Br. load[2] |
| | | [kg] | | [m] | [mm] | [mm] | [mm] | [m] | [kN] | [m] | [kN] | | [m] | [kN] |
| 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 |
| up to - 50 | 2 | 120 | 40 | 165 | 12,5 | 12,5 | 12,5 | 80 | 65 | 180 | 100 | 3 | 80 | 35 |
| 50 - 70 | 2 | 180 | 60 | 220 | 14 | 12,5 | 12,5 | 80 | 65 | 180 | 100 | 3 | 80 | 35 |
| 70 - 90 | 2 | 240 | 80 | 220 | 16 | 14 | 14 | 85 | 75 | 180 | 100 | 3 | 100 | 40 |
| 90 - 110 | 2 | 300 | 100 | 247,5 | 17,5 | 16 | 16 | 85 | 80 | 180 | 100 | 3 | 110 | 40 |
| 110 - 130 | 2 | 360 | 120 | 247,5 | 19 | 17,5 | 17,5 | 90 | 90 | 180 | 100 | 3 | 110 | 45 |
| 130 - 150 | 2 | 420 | 140 | 275 | 20,5 | 17,5 | 17,5 | 90 | 100 | 180 | 100 | 3 | 120 | 50 |
| 150 - 175 | 2 | 480 | 165 | 275 | 22 | 19 | 19 | 90 | 110 | 180 | 100 | 3 | 120 | 55 |
| 175 - 205 | 2 | 570 | 190 | 302,5 | 24 | 20,5 | 20,5 | 90 | 120 | 180 | 110 | 3 | 120 | 60 |
| 205 - 240 | 3 | 660 | | 302,5 | 26 | 22 | 20,5 | | | 180 | 130 | 4 | 120 | 65 |
| 240 - 280 | 3 | 780 | | 330 | 28 | 24 | 22 | | | 180 | 150 | 4 | 120 | 70 |
| 280 - 320 | 3 | 900 | | 357,5 | 30 | 26 | 24 | | | 180 | 175 | 4 | 140 | 80 |
| 320 - 360 | 3 | 1020 | | 357,5 | 32 | 28 | 24 | | | 180 | 200 | 4 | 140 | 85 |
| 360 - 400 | 3 | 1140 | | 385 | 34 | 30 | 26 | | | 180 | 225 | 4 | 140 | 95 |
| 400 - 450 | 3 | 1290 | | 385 | 36 | 32 | 28 | | | 180 | 250 | 4 | 140 | 100 |
| 450 - 500 | 3 | 1440 | | 412,5 | 38 | 34 | 30 | | | 180 | 275 | 4 | 140 | 110 |
| 500 - 550 | 3 | 1590 | | 412,5 | 40 | 34 | 30 | | | 190 | 305 | 4 | 160 | 120 |
| 550 - 600 | 3 | 1740 | | 440 | 42 | 36 | 32 | | | 190 | 340 | 4 | 160 | 130 |
| 600 - 660 | 3 | 1920 | | 440 | 44 | 38 | 34 | | | 190 | 370 | 4 | 160 | 145 |
| 660 - 720 | 3 | 2100 | | 440 | 46 | 40 | 36 | | | 190 | 405 | 4 | 160 | 160 |
| 720 - 780 | 3 | 2280 | | 467,5 | 48 | 42 | 36 | | | 190 | 440 | 4 | 170 | 170 |
| 780 - 840 | 3 | 2460 | | 467,5 | 50 | 44 | 38 | | | 190 | 480 | 4 | 170 | 185 |
| 840 - 910 | 3 | 2640 | | 467,5 | 52 | 46 | 40 | | | 190 | 520 | 4 | 170 | 200 |
| 910 - 980 | 3 | 2850 | | 495 | 54 | 48 | 42 | | | 190 | 560 | 4 | 170 | 215 |
| 980 - 1060 | 3 | 3060 | | 495 | 56 | 50 | 44 | | | 200 | 600 | 4 | 180 | 230 |
| 1060 - 1140 | 3 | 3300 | | 495 | 58 | 50 | 46 | | | 200 | 645 | 4 | 180 | 250 |
| 1140 - 1220 | 3 | 3540 | | 522,5 | 60 | 52 | 46 | | | 200 | 690 | 4 | 180 | 270 |
| 1220 - 1300 | 3 | 3780 | | 522,5 | 62 | 54 | 48 | | | 200 | 740 | 4 | 180 | 285 |
| 1300 - 1390 | 3 | 4050 | | 522,5 | 64 | 56 | 50 | | | 200 | 785 | 4 | 180 | 305 |
| 1390 - 1480 | 3 | 4320 | | 550 | 66 | 58 | 50 | | | 200 | 835 | 4 | 180 | 325 |
| 1480 - 1570 | 3 | 4590 | | 550 | 68 | 60 | 52 | | | 220 | 890 | 5 | 190 | 325 |
| 1570 - 1670 | 3 | 4890 | | 550 | 70 | 62 | 54 | | | 220 | 940 | 5 | 190 | 335 |
| 1670 - 1790 | 3 | 5250 | | 577,5 | 73 | 64 | 56 | | | 220 | 1025 | 5 | 190 | 350 |
| 1790 - 1930 | 3 | 5610 | | 577,5 | 76 | 66 | 58 | | | 220 | 1110 | 5 | 190 | 375 |
| 1930 - 2080 | 3 | 6000 | | 577,5 | 78 | 68 | 60 | | | 220 | 1170 | 5 | 190 | 400 |
| 2080 - 2230 | 3 | 6450 | | 605 | 81 | 70 | 62 | | | 240 | 1260 | 5 | 200 | 425 |
| 2230 - 2380 | 3 | 6900 | | 605 | 84 | 73 | 64 | | | 240 | 1355 | 5 | 200 | 450 |
| 2380 - 2530 | 3 | 7350 | | 605 | 87 | 76 | 66 | | | 240 | 1455 | 5 | 200 | 480 |
| 2530 - 2700 | 3 | 7800 | | 632,5 | 90 | 78 | 68 | | | 260 | 1470 | 6 | 200 | 480 |
| 2700 - 2870 | 3 | 8300 | | 632,5 | 92 | 81 | 70 | | | 260 | 1470 | 6 | 200 | 490 |
| 2870 - 3040 | 3 | 8700 | | 632,5 | 95 | 84 | 73 | | | 260 | 1470 | 6 | 200 | 500 |
| 3040 - 3210 | 3 | 9300 | | 660 | 97 | 84 | 76 | | | 280 | 1470 | 6 | 200 | 520 |
| 3210 - 3400 | 3 | 9900 | | 660 | 100 | 87 | 78 | | | 280 | 1470 | 6 | 200 | 555 |
| 3400 - 3600 | 3 | 10500 | | 660 | 102 | 90 | 78 | | | 280 | 1470 | 6 | 200 | 590 |
| 3600 - 3800 | 3 | 11100 | | 687,5 | 105 | 92 | 81 | | | 300 | 1470 | 6 | 200 | 620 |
| 3800 - 4000 | 3 | 11700 | | 687,5 | 107 | 95 | 84 | | | 300 | 1470 | 6 | 200 | 650 |
| 4000 - 4200 | 3 | 12300 | | 687,5 | 111 | 97 | 87 | | | 300 | 1470 | 7 | 200 | 650 |
| 4200 - 4400 | 3 | 12900 | | 715 | 114 | 100 | 87 | | | 300 | 1470 | 7 | 200 | 660 |
| 4400 - 4600 | 3 | 13500 | | 715 | 117 | 102 | 90 | | | 300 | 1470 | 7 | 200 | 670 |
| 4600 - 4800 | 3 | 14100 | | 715 | 120 | 105 | 92 | | | 300 | 1470 | 7 | 200 | 680 |
| 4800 - 5000 | 3 | 14700 | | 742,5 | 122 | 107 | 95 | | | 300 | 1470 | 7 | 200 | 685 |
| 5000 - 5200 | 3 | 15400 | | 742,5 | 124 | 111 | 97 | | | 300 | 1470 | 8 | 200 | 685 |
| 5200 - 5500 | 3 | 16100 | | 742,5 | 127 | 111 | 97 | | | 300 | 1470 | 8 | 200 | 695 |
| 5500 - 5800 | 3 | 16900 | | 742,5 | 130 | 114 | 100 | | | 300 | 1470 | 8 | 200 | 705 |
| 5800 - 6100 | 3 | 17800 | | 742,5 | 132 | 117 | 102 | | | 300 | 1470 | 9 | 200 | 705 |
| 6100 - 6500 | 3 | 18800 | | 742,5 | | 120 | 107 | | | 300 | 1470 | 9 | 200 | 715 |
| 6500 - 6900 | 3 | 20000 | | 770 | | 124 | 111 | | | 300 | 1470 | 9 | 200 | 725 |
| 6900 - 7400 | 3 | 21500 | | 770 | | 127 | 114 | | | 300 | 1470 | 10 | 200 | 725 |
| 7400 - 7900 | 3 | 23000 | | 770 | | 132 | 117 | | | 300 | 1470 | 11 | 200 | 725 |
| 7900 - 8400 | 3 | 24500 | | 770 | | 137 | 122 | | | 300 | 1470 | 11 | 200 | 735 |
| 8400 - 8900 | 3 | 26000 | | 770 | | 142 | 127 | | | 300 | 1470 | 12 | 200 | 735 |
| 8900 - 9400 | 3 | 27500 | | 770 | | 147 | 132 | | | 300 | 1470 | 13 | 200 | 735 |
| 9400 - 10000 | 3 | 29000 | | 770 | | 152 | 132 | | | 300 | 1470 | 14 | 200 | 735 |
| 10000 - 10700 | 3 | 31000 | | 770 | | | 137 | | | 300 | 1470 | 15 | 200 | 735 |
| 10700 - 11500 | 3 | 33000 | | 770 | | | 142 | | | 300 | 1470 | 16 | 200 | 735 |
| 11500 - 12400 | 3 | 35500 | | 770 | | | 147 | | | 300 | 1470 | 17 | 200 | 735 |
| 12400 - 13400 | 3 | 38500 | | 770 | | | 152 | | | 300 | 1470 | 18 | 200 | 735 |
| 13400 - 14600 | 3 | 42000 | | 770 | | | 157 | | | 300 | 1470 | 19 | 200 | 735 |
| 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

## Table 18.2 Anchor, Chain Cables and Ropes

| No. for Reg. | Equipment numeral Z | Stockless anchor: Bower anchor: Number [1] | Stockless anchor: Bower anchor: Mass per anchor [kg] | Stockless anchor: Stream anchor [kg] | Stud link chain cables: Bower anchors: Total length [m] | Stud link chain cables: Bower anchors: Diameter $d_1$ [mm] | Stud link chain cables: Bower anchors: Diameter $d_2$ [mm] | Stud link chain cables: Bower anchors: Diameter $d_3$ [mm] | Stream wire or chain for stream anchor: Length [m] | Stream wire or chain for stream anchor: Br. load [2] [kN] | Recommended ropes: Towline: Length [m] | Recommended ropes: Towline: Br. load [2] [kN] | Recommended ropes: Mooring ropes: Number | Recommended ropes: Mooring ropes: Length [m] | Recommended ropes: Mooring ropes: Br. load [2] [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 | 710 | 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 | 84 | 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 | 685 |
| 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 | 725 |
| 159 | 7900 - 8400 | 3 | 24500 | | 770 | | 137 | 122 | | | 300 | 1470 | 11 | 200 | 735 |
| 160 | 8400 - 8900 | 3 | 26000 | | 770 | | 142 | 127 | | | 300 | 1470 | 12 | 200 | 735 |
| 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

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1.3 For spaces for independent tanks on tankers :cording to A.1.2. b) the diameters of the main ıd branch bilge lines are calculated as follows:

$$= 1{,}68 \cdot \sqrt{(B + H)\, l_2 - (b + h)\, l_{T2}} + 25 \ [\mathrm{mm}]$$

$$= 2{,}15 \cdot \sqrt{(B + H)\, l - (b + h)\, l_T} + 25 \ [\mathrm{mm}]$$

ere

| | |
|---|---|
| [mm] | Inside diameter of main bilge line |
| [mm] | Inside diameter of branch bilge line |
| [m] | Breadth of ship |
| [m] | Moulded depth of ship |
| [m] | Total length of cargo area |
| [m] | Length of watertight compartment |
| [m] | Maximum breadth of cargo tanks |
| [m] | Maximum depth of cargo tanks |
| [m] | Total length of all cargo tanks |
| [m] | Length of tanks in the watertight compartment. |

:apacity of each bilge pump is to be calculated ding to Section 11, N.3.1. At least two bilge ·s are to be provided.

When separate bilge pumps, e.g. ejectors rovided for compartments with watertight eads the pump capacity is to be evaluated as ied in 4.1.3 and is to be divided according the of the individual compartments. For each ırtment two bilge pumps are to be fitted of a ty of not less than 5 m³/h each.

Spaces for independent tanks are to be pro- vith sounding arrangements.

ballast or cooling water lines are fitted in for independent tanks bilge level alarms are rovided.

**Bilge pumping of cargo pump rooms and cofferdams in the cargo area**

Bilge pumping equipment is to be located in ;o area to serve the cargo pump rooms and ıms. A cargo pump may also be used as a mp. On oil tankers used exclusively for the : of flammable liquids with flash points 0 °C, cargo pump rooms and cofferdams connected to the engine room bilge system.

**4.2.2** Where a cargo pump is used as bilge pump, measures are to be taken, e.g. by fitting screw-down non-return valves, to ensure that cargo cannot enter the bilge system. Where the bilge line can be pressurized from the cargo system, an additional non-return valve is to be fitted.

**4.2.3** Means must be provided for pumping the bilges when special circumstances render the pump room inaccessible. The equipment necessary for this is to be capable of being operated from outside the pump room or from the pump room casing above the tank deck (freeboard deck).

**4.3 Ballast systems in the cargo area**

**4.3.1** Means for ballasting cargo tanks or permanent ballast tanks within the cargo area must be located in the cargo area and must be independent of piping systems forward and aft of the cofferdams.

**4.3.2** Ballast water pipes shall not pass through cargo oil tanks. Exemptions for short length of pipe may be approved by BKI on condition that the following is complied with :

a) Minimum wall thicknesses

| | | |
|---|---|---|
| up to | DN 50 mm | 6,3 mm |
| | DN 100 mm | 8,6 mm |
| | DN 125 mm | 9,5 mm |
| | DN 150 mm | 11,0 mm |
| | DN 200 mm and over | 12,5 mm |

b) Only completely welded pipes or equivalent are permitted

c) Where cargoes other than oil products are carried, relaxation from these Rules may be approved BKI.

**4.3.3** Ballast tank sounding and air pipes routed through cargo oil tanks are subject to para. 4.3.2 analogously.

**5. Ventilation and gas-freeing**

**5.1 Ventilation of cargo and ballast pump rooms in the cargo area**

**5.1.1** Pump rooms are to be provided with efficient means of ventilation. These systems may not be connected to the ventilation systems of other spaces in the ship.

**5.1.2** Pump rooms are to be ventilated by mechanically driven fans of the extraction type. Fresh air is to be induced into the pump room from above.

The exhaust duct is to be so installed that its suction opening is close to the bottom of the pump room.

## Appendix to Section 2

### Part C :

### Approximate Calculation of the Starting Air Supply

#### 1. Starting air for installations with reversible engines

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 as follows:

$$J = a \bullet \sqrt[3]{\frac{H}{D}} \bullet (z + b \bullet p_{e,e} \bullet n_A + 0{,}9) \bullet V_h \bullet c \qquad (13)$$

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) |
| $P_{e,perm}$ | [bar] | maximum permissible working pressure of the starting air receiver |
| z | [-] | number of cylinders |
| $P_{e,e}$ | [bar] | mean effective working pressure in cylinder at rated power |

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

- for two-stroke engines: a = 0,4714
- for four-stroke engines: a = 0,4190

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, where $P_{e,perm}$ = 30 bar

$$c = \frac{0{,}0584}{1 - e^{(0{,}11 - 0{,}05 \bullet l_A \bullet p_{e,perm})}}$$

where $p_{e,perm} \neq 30$ 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 $p_{e,perm}$ to the starting pressure $P_A$ , then the value of "c" shown in Fig. 2.14 is to be used.

The following values of $n_A$ are to be applied:

| | |
|---|---|
| $n_A = 0{,}06 \bullet n_o + 14$ | where $n_o \leq 1000$ |
| $n_A = 0{,}25 \bullet n_o - 176$ | where $n_o > 1000$ |
| $n_o$ [Rpm] | = rated speed |

#### 2. Starting air for installations with non reversible engines

For each non-reversible main engine driving a controllable pitch propeller or where starting without torque resistance is possible the calculated starting air supply may be reduced to 0,5 . J though not less than that needed for six start-up operations.

1.5.2 A combination of a non-return valve without shut-off mechanism and a shut-off valve may be recognized as equivalent with the Society's approval.

### 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 of which is to be a screw-down non-return valve.

One of such means of protection is to be fitted in each suction line.

1.6.2 For bilge connections outside machinery spaces, a combination of a non-return valve without shut-off and a remote-controlled shut-off valve may be recognized as equivalent.

1.6.3 The direct bilge suction and the emergency injection need only have one means of reverse-flow protection as specified in 1.5.1.

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

1.6.5 The discharge lines of oily water separators are to be fitted with a non-return valve 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.

### 2.2 Dry cargo and passenger ships

a) **main bilge pipes**

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

b) **branch bilge pipes**

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

where

$d_H$ [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:

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

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 2.2 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 bilge pumps

Each bilge pump must be capable of delivering:

$$Q = 5{,}75 \cdot 10 \qquad (7)$$

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 One bilge pump with a smaller capacity than that required according to formula (7) is acceptable provided that the other pump is designed for a correspondingly larger capacity. However, the capacity of the smaller bilge pump shall not be less than 85 % of the calculated capacity.

**3.4 Use of other pumps for bilge pumping**

3.4.1 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).

3.4.2 In the event of failure of one of the required bilge pumps, one pump each must be available for fire fighting and bilge pumping.

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

3.4.4 Bilge ejectors are acceptable as bilge pumping arrangements provided that there is an independent supply of driving water.

**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 suction 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 suction in accordance with 4.1.1 a).

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

The diameter of the direct suction may not be less than that of the main bilge pipe.

4.1.3 The diameter of the emergency bilge suction on steam ships in accordance with 4.1.1 c) is to be at least 2/3 of the diameter and on motor ships equal to the diameter of the cooling water pump suction line. Exceptions to this Rule require the approval of the Society. The emergency bilge suction must be connected to the cooling water pump suction line by means of a screw-down non-return valve.

This valve is to be provided with a plate with the notice :

**Emergency bilge valve!**
**To be opened in an emergency only!**

Emergency bilge valves and cooling water inlet valves must be capable of being operated from above the floor plates.

4.1.4 Engine control rooms and similar spaces as well as decks in engine rooms are to be provided with drains to the engine room bilge. A drain pipe which passes through a watertight bulkhead is to be fitted with a self-closing valve.

### 4.2 Shaft tunnel

A bilge suction is to be arranged at the after end of the shaft tunnel. Where the shape of the bottom or

---

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

### 6.3.2 Bilge lines

Valves and control lines are to be located as far as possible from the bottom and sides of the ship.

### 6.3.3 Ballast pipes

The requirements stated in 6.3.2 also apply here to the location of valves and control lines.

Where remote controlled valves are arranged inside the ballast tanks, the valves should always be located in the tank adjoining that to which they relate.

### 6.3.4 Fuel pipes

Remote controlled valves mounted on fuel tanks located above the double bottom must be capable of being closed from outside the compartment in which they are installed.

### 6.3.5 Cargo pipes

Where remote controlled valves are arranged inside cargo tanks, valves should always be fitted in the tank adjoining that to which they relate.

A direct arrangement of the remote controlled valves in the tanks concerned is allowed only if each tank is fitted with two suction lines each of which is provided with a remote controlled valve.

## 6.4 Control stands

6.4.1 The control devices of remote controlled valves are to be arranged together in one control stand.

6.4.2 The control devices are to be clearly and permanently identified and marked.

6.4.3 It must be recognized at the control stand weather the valves are open or closed.

In the case of bilge valves and valves for changeable tanks, the closed position is to be indicated by limit position indicators approved by the Society as well as by visual indicators at the control stand.

6.4.4 The control devices of valves for changeable tanks are to be interlocked to ensure that only the valve relating to the tank concerned can be operated. The same also applies to the valves of cargo holds and tanks in which dry cargo and ballast water are carried alternately.

6.4.5 On passenger ships, the control stand for remote controlled bilge valves is to be located outside the machinery spaces and above the bulkhead deck.

## 6.5 Power units

6.5.1 Power units are to be equipped with at least two independent sets for supplying power for remote controlled valves.

6.5.2 The energy required for the closing of valves which are not closed by spring power is to be supplied by a pressure accumulator.

6.5.3 Pneumatically operated valves can be supplied with air from the general compressed air system.

Where the quick-closing valves of fuel tanks are closed pneumatically, a separate pressure accumulator is to be provided. This is to be of adequate capacity and is to be located outside the engine room. Filling of this accumulator by a direct connection to the general compressed air system is allowed. A non-return valve is to be arranged in the filling connection of the pressure accumulator.

The accumulator is to be provided either with a pressure control device with a visual and acoustic alarm or with a hand-compressor as a second filling appliance.

The hand-compressor is to be located outside the engine room.

6.6 After installation on board, the entire system is to be subjected to an operational test.

## 7. Pumps

7.1 For materials and construction requirements the "Regulations for Construction and Testing of Pumps" of BKI are to be applied.

7.2 For the pumps listed below, a performance test is to be carried out in the manufacturer's works under the Society's supervision

- Bilge pumps/bilge ejectors
- Ballast pumps
- Sea cooling water pumps
- Fresh cooling water pumps
- Fire extinguishing pumps
- Emergency fire extinguishing pumps including drive units
- Condensate pumps
- Boiler feedwater pumps

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.

#### 4.10 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.

#### 4.11 Chain lockers

Chain lockers are to be drained by means of appropriate arrangements. They may not be drained to the fore peak.

### 5. Additional Rules for passenger vessels

5.1.1 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 Rules for Hull Construction, Volume II, Section 29.F.

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

5.1.2 Valve boxes and valves of the bilge system are to be installed in such a way that each compartment can be emptied by at least one pump in the event of ingress of water.

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.

5.1.3 Where only one common piping system is provided for all pumps, all the shutoff and changeover 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.

5.1.4 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.

#### 5.2 Bilge suctions

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

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

#### 5.3 Arrangement of bilge pumps

5.3.1 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 or 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.

5.3.2 The bilge pumps specified in 3.6 and their energy sources may not be located forward of the collision bulkhead.

#### 5.4 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.

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[1]) See SOLAS 1974, Chapter II-1, parts A, Regulation 5 and 18

6. **Additional Rules for tankers**

See Section 15, B.4.

7. **Bilge testing**

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

). **Equipment for the Treatment and Storage of Bilge Water and Fuel and Oil - Residues[1])**

. **Oily water separating equipment**

.1 Ships of 400 tons gross and above shall be tted with an oily water separator or a filter plant for ie separation of oil/water mixtures.

2 Ships of 10.000 tons gross and above shall : fitted, in addition to the equipment required in ıragraph 1.1, with an oil discharge monitoring and ›ntrol system or with a 15 ppm alarm system.

3 A sampling device is to be arranged in the scharge line of oily water separating equipent/filtering systems.

4 By-pass lines are not permitted for oilywater ›aratoring equipment/filtering systems.

**Discharge of fuel and oil residues**

A sludge tank is to be provided. For the fitgs and mountings of sludge tanks, see Section 10,

A self-priming pump is to be provided for dge discharge to reception facilities. The capacity he pump shall be such that the sludge tank can be ptied in a reasonable time.

A separate discharge line is to be provided discharge of fuel and oil residues to reception faties.

2.4 Where incinerating plants are used for fuel and oil residues, compliance is required with Section 9 and with the Regulations for the Design and Testing of Waste Incinerating Plants on Seagoing Ships.

P. **Ballast Systems**

1. **Ballast lines**

1.1 **Arrangement of piping - general**

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

1.1.2 Ships having very wide double bottom tanks are also to be provided with suction at the outer sides of the tanks. Where the length of the ballast water tanks exceeds 30 m, the Society may require suction 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.4.5 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.

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[1] With regard to the installation on ships of oily water separators, filter plants, oil collecting tanks, oil discharge lines and a monitoring and control system or a 15 ppm alarm device in the water outlet of oily water separators, compliance is required with the provisions of the International Convention for the Prevention of Pollution from Ships, 1973, (MARPOL) and the Protocol of 1978.
Form F.136 is to submitted for approval.

HARVALD

DALBO

# Bab

# Lima

# PENENTUAN TAHANAN KAPAL

## 5.1. PENDAHULUAN

Dalam membuat usulan awal untuk kapal baru atau melakukan studi transportasi, pertanyaan vital yang sering dihadapi pemilik kapal, arsitek kapal, politikus, ahli ekonomi, atau mahasiswa adalah besarnya daya yang diperlukan. Jawabannya dapat dicari dengan berbagai cara. Seperti halnya dalam perancangan awal kapal, ada tiga kelompok yang dapat dipilih :

Metode kapal pembanding
Metode statistik
Metode satu per satu

Jika memakai metode yang pertama maka harus dipilih suatu kapal pembanding. Kapal pembanding ini harus merupakan jenis yang sama dengan jenis kapal yang disyaratkan dalam usulan. Selain itu, ukuran utama dan kecepatan kapal pembanding tersebut harus tidak jauh berbeda dengan yang diharapkan untuk kapal yang akan diusulkan. Koefisien admiralty $A_c$ untuk kapal pembanding dihitung dengan memakai rumus

$$A_c = \frac{\Delta^{2/3} V^3}{P} \qquad (5.1.1)$$

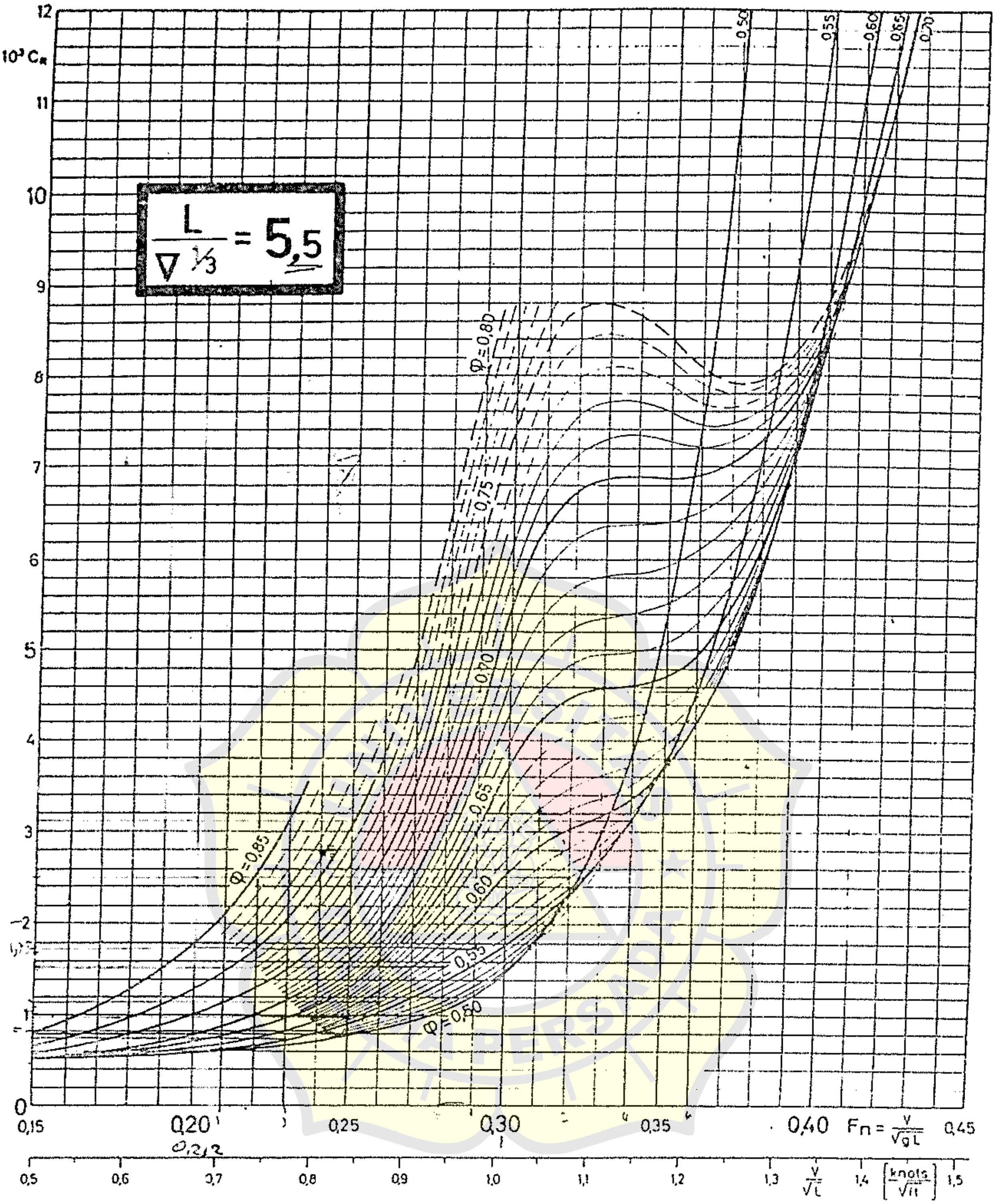
$P$ adalah daya yang diperlukan untuk menggerakkan kapal pada displasemen $\Delta$ dan kecepatan $V$. Kemudian daya $P_p$ untuk kapal yang diusulkan dapat dihitung dengan

$$P_p = \frac{\Delta_p^{2/3} V_p^3}{A_c} \qquad (5.1.2)$$

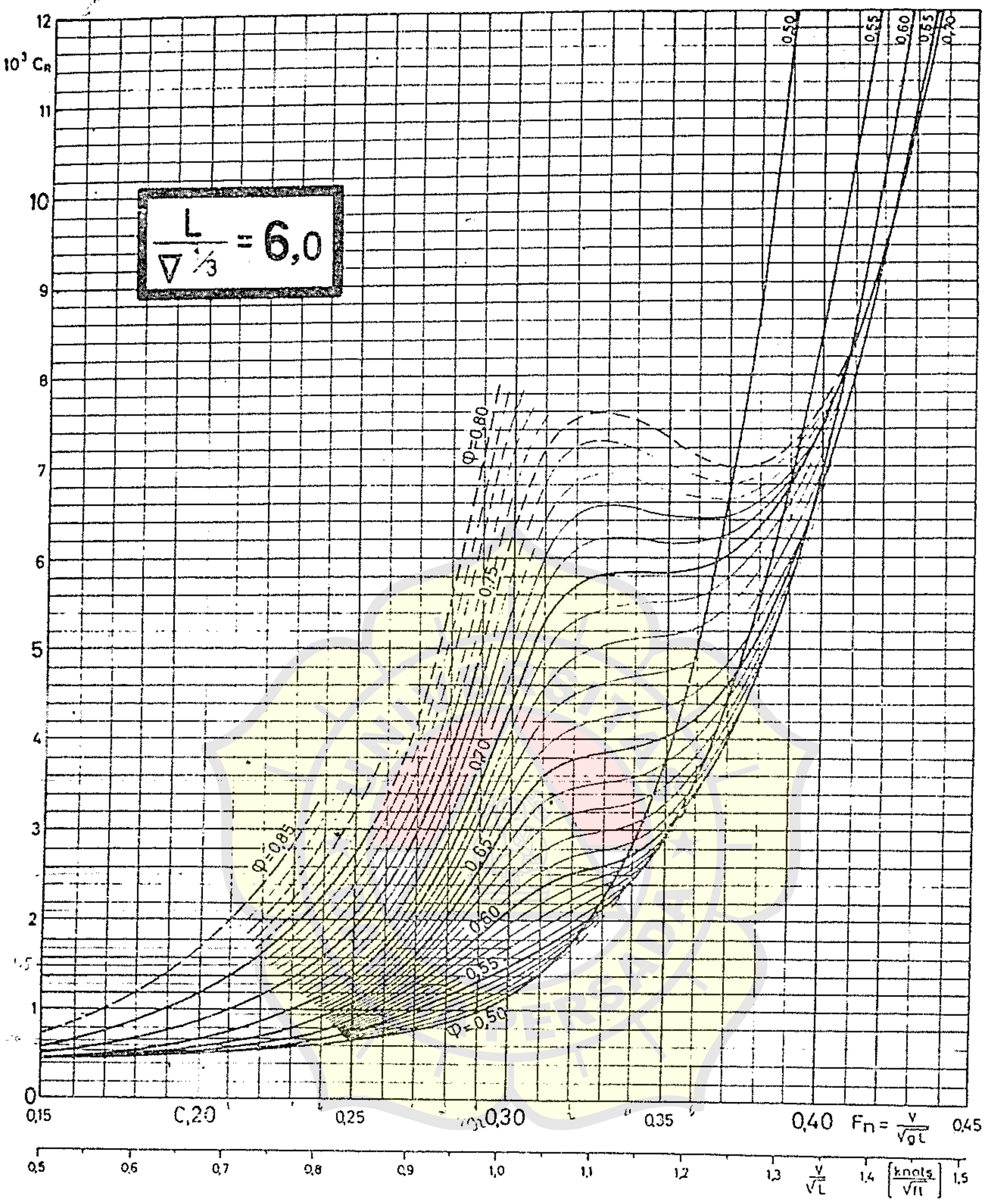
$\Delta_p$ dan $V_p$ masing-masing adalah displasemen dan kecepatan kapal yang diusulkan. Di sini daya yang diperlukan dianggap berbanding lurus dengan tahanan total kapal.

Bila memakai metode yang kedua maka data propulsi dari seperangkat kapal dikumpulkan dan dipelajari statistiknya. Hasilnya dapat diberikan berupa program untuk perhitungan atau seperangkat diagram yang menyatakan daya sebagai fungsi dari, mungkin, koefisien blok, displasemen, dan rasio panjang–displasemen. Seperangkat diagram semacam itu dapat dilihat di Bab 9.

Berbeda dengan kedua metode tadi, dalam metode yang ketiga tahanan kapal yang diusulkan itu sendirilah yang harus diketahui. Tahanan ini dapat diperkirakan dengan berbagai cara. Gagasan melakukan percobaan model di air untuk memperkirakan tahanan kapal berukuran penuh sebagaimana disebutkan di Bab 3 merupakan gagasan yang sudah timbul sejak lama, yaitu mulai dari sekitar tahun 1500 (Turnisi, 1953), namun demikian hingga tahun 1868 tidak ada metode yang dapat dipakai untuk mentransformasi data model ke kapal yang sebenarnya (Stoot, 1959). **William Froude** kemudian mengusulkan hukum perbandingannya dan menunjukkan cara pemakaiannya dalam praktek untuk memprakirakan tahanan kapal dari hasil model. Dalam bab ini akan diuraikan dan dibahas metode Froude dan metode yang paling akhir untuk menentukan tahanan kapal.




**Gambar 5.5.8.** Koefisien tahanan sisa terhadap rasio kecepatan–panjang untuk harga koefisien prismatik longitudinal yang berbeda-beda. $L/\nabla^{1/3} = 5,5$.




**Gambar 5.5.9.** Koefisien tahanan sisa terhadap rasio kecepatan-panjang untuk harga koefisien prismatik longitudinal yang berbeda-beda. $L/\nabla^{1/3} = 6,0$.

1. Semua data diacukan pada daerah (lingkup) model, dan tahanan model ($R_{Tm}$) ditentukan sebagai fungsi kecepatan.
2. Koefisien tahanan total spesifik model ($C_{Tm}$) ditentukan :

$$C_{Tm} = \frac{R_{Tm}}{\frac{1}{2}\rho V_m^2 S_m} \tag{5.5.5}$$

$\rho$ adalah massa jenis, $V_m$ kecepatan model, $S_m$ permukaan basah model (= panjang garis sisi rata-rata × panjang garis air).

3. Koefisien tahanan sisa spesifik ditentukan dari

$$C_R = C_{Tm} - C_{Fm} \tag{5.5.6}$$

$C_{Fm}$ adalah koefisien tahanan gesek spesifik. "Garis korelasi model – kapal ITTC 1957" dipakai untuk menentukan koefisien tahanan gesek.

$$C_F = \frac{0{,}075}{(\log_{10} R_n - 2)^2} \tag{5.5.7}$$

$R_n$ adalah angka Reynolds ($VL/\nu$, $\nu$ adalah koefisien viskositas kinematik dan $L$ panjang garis air). Dalam Gb. 5.5.4 diberikan kontur $C_F$ untuk berbagai harga $V$ dan $F_n$. Koordinat horizontal menunjukkan panjang model $L$. Diagram tersebut untuk $\nu = 1{,}139 \times 10^{-6}$ m s$^{-1}$, $\rho = 1000$ t/m$^3$, dan $T = 15$°C. Karena itu untuk memakai diagram tersebut dengan kondisi yang lain, yaitu massa jenis dan suhu yang lain, panjang kapal harus diubah dulu sebelum memakai diagram tersebut sebagai berikut

$$L_1 = \frac{1{,}139}{10^6 \nu} L \tag{5.5.8}$$

4. $C_R$ dinyatakan sebagai fungsi angka Froude

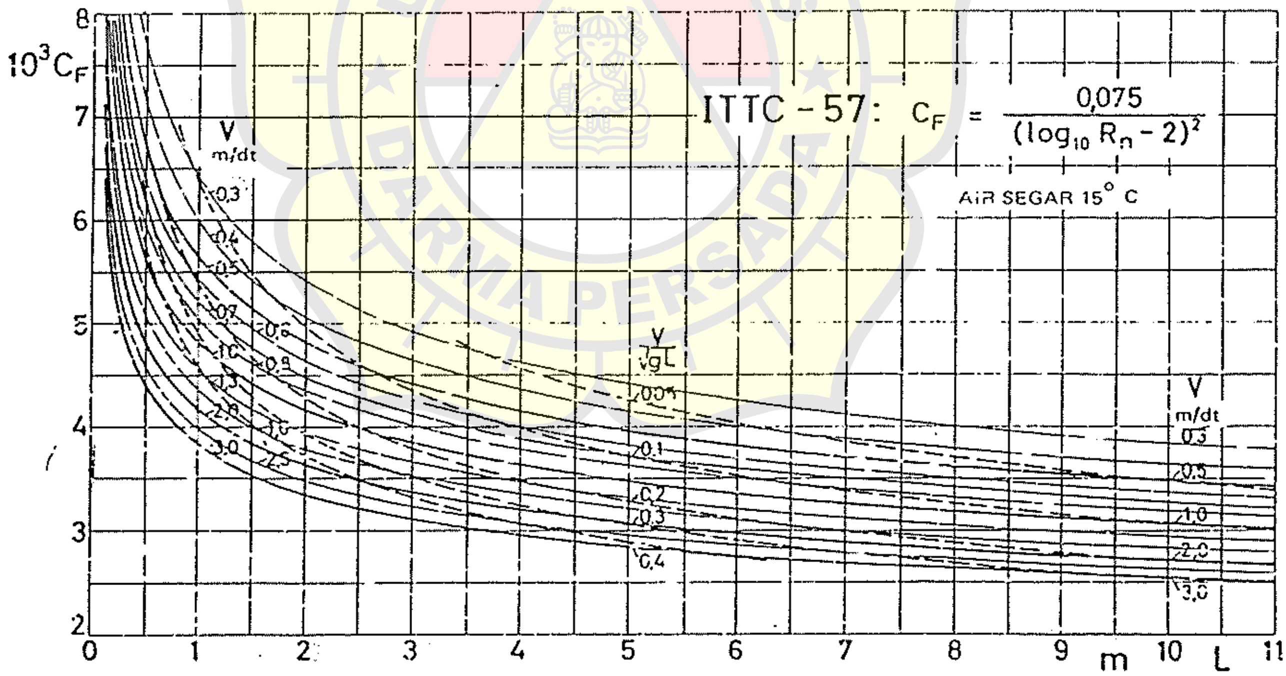
$$F_n = \frac{V}{\sqrt{gL}} \tag{5.5.9}$$

(rasio kecepatan – panjang $V/\sqrt{L}$, dalam hal ini $V$ diukur dalam knot dan $L$ dalam kaki, didapat dari subskala dalam diagram $C_R$).

5. Hasilnya dikelompokkan menurut rasio panjang – displasemen $L/\nabla^{1/3}$ dan koefisien prismatik $\varphi$ model. $V$ adalah volume displasemen dan

$$\varphi = \frac{\nabla}{LBT\beta} \tag{5.5.10}$$

$B$ adalah lebar, $T$ sarat, dan $\beta$ koefisien penampang melintang tengah kapal.




Gambar 5.5.4. Koefisien tahanan gesek $C_F$ (menurut ITTC 1957) sebagai fungsi dari panjang model kapal $L$ dan kecepatan $V$.

6. Diagram utama digambarkan untuk menyatakan kurva rata-rata $C_R$ untuk rasio lebar – sarat $B/T$ = 2,5. Diagram tersebut ditunjukkan di Gb. 5.5.5 – 5.5.13.

Dalam diagram tersebut kurva yang digambar dengan garis terputus-putus menunjukkan bahwa kurva tersebut didasarkan pada hasil percobaan yang sedikit jumlahnya atau diperoleh secara ekstrapolasi. Karena itu keraguan hasil di daerah kurva itu cukup besar. Selain itu, perlu diperhatikan pula bahwa di dan di dekat daerah kurva yang mempunyai punuk (tonjolan) yang menyolok, terutama jika kemiringannya menjadi negatif, tingkat ketidak pastiannya juga tinggi. Perubahan yang kecil saja dari bentuk badan kapal di dalam daerah tersebut dapat mempunyai pengaruh yang berarti pada harga $C_R$.

Perlu pula disebutkan di sini bahwa kurva tahanan tersebut berlaku untuk kapal yang mempunyai bentuk standar, yaitu letak titik benamnya standar, harga $B/T$ nya standar, bentuk penampangnya normal, buritannya merupakan buritan sendok (cruiser stern) yang moderat, dan linggi haluannya merupakan linggi haluan condong (raked stem).

Tahanan $R$ dan daya efektif $P_E$ untuk kapal baru dapat dihitung dengan memakai

$$R = C_T(\tfrac{1}{2}\rho V^2 S) \quad (\mathrm{N}) \qquad (5.5.11)$$

$$P_E = RV \quad (\mathrm{kW}) \qquad (5.5.12)$$

Dalam hal ini koefisien tahanan totalnya adalah

$$C_T = C_R + C_F + C_A \qquad (5.5.13)$$

$C_R$ = koefisien tahanan sisa. Untuk bentuk kapal yang "standar" dapat diambil dari diagram (Gb. 5.5.5 – 5.5.13)

$C_F$ = koefisien tahanan gesek dan dapat dihitung dengan memakai

$$C_F = \frac{0,075}{(\log_{10} R_n - 2)^2} \qquad (5.5.14)$$

atau dapat diambil dari Gb. 5.5.14. Dalam gambar ini kontur $C_F$ diberikan untuk berbagai harga $V$ yang berbeda. Koordinat $L$ horizontalnya adalah panjang kapal. Diagram tersebut berlaku untuk $\nu$ = 1,188 × $10^{-6}$ m.s$^{-1}$, $\rho$ = 1,025 t/m$^3$, dan $t$ = 15°C. Untuk kondisi yang lain, yaitu massa jenis dan suhu yang lain, sebelum memakai diagram tersebut panjang kapal harus diubah dulu sebagai berikut :

$$L_1 = \frac{1,188}{10^6 \nu} L \qquad (5.5.15)$$

$C_A$ = koefisien tahanan tambahan, yaitu koefisien kekasaran permukaan dan pengaruh skala pada hasil percobaan model. Dalam hal ini maka $C_A$ akan tergantung pada cara penentuan $C_R$ dan $C_F$.

Untuk kapal penarik, $R$ harus diganti dengan $R + F$. Dalam hal ini $F$ adalah gaya tarik tali penarik (tow rope pull).

Karena kapal pada umumnya berbeda dengan standar dengan tingkat perbedaan tertentu, lebih besar atau lebih kecil, maka harus dilakukan koreksi sebagai berikut.

*B/T*

Karena diagram tersebut dibuat berdasarkan rasio lebar – sarat

$$B/T = 2,5 \qquad (5.5.16)$$

maka harga $C_R$ untuk kapal yang mempunyai rasio lebar – sarat lebih besar atau lebih kecil daripada harga tersebut harus dikoreksi.

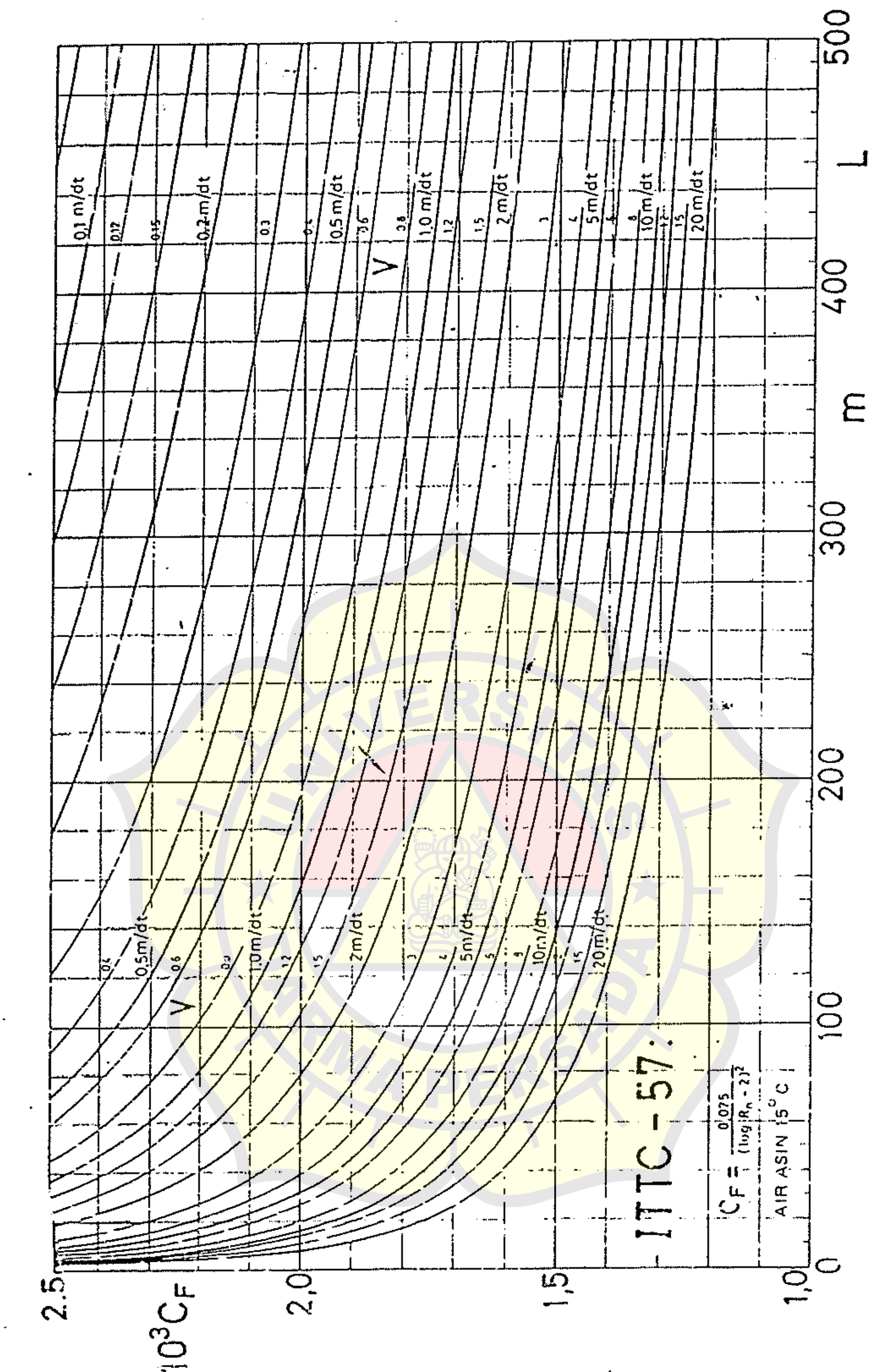
Berdasarkan hasil pemeriksaan materi pengujian yang ada saat ini maka disarankan untuk memakai rumus koreksi berikut ini :

$$10^3 C_R = 10^3 C_{R(B/T=2,5)} + 0.16(B/T - 2,5) \qquad (5.5.17)$$

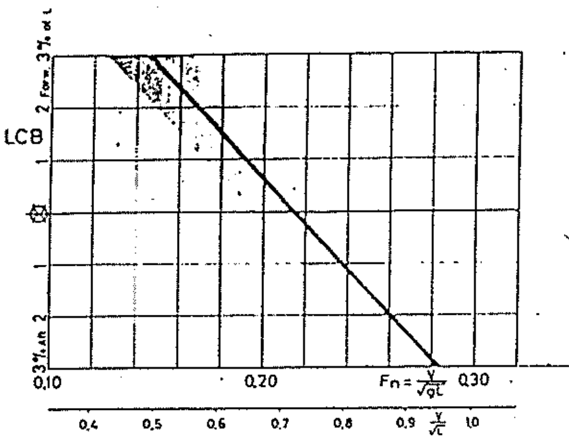
Koreksi ini dapat mempunyai harga yang negatif atau positif.

LCB

Semua kurva $C_R$ tersebut dimaksudkan untuk kapal yang letak titik benam longitudinalnya dekat dengan letak yang dewasa ini dipandang sebagai letak yang terbaik yang memungkinkan. Letak LCB yang optimum merupakan kuantitas yang masih agak meragukan, dan semua kepustakaan yang ada menunjukkan pendapat yang berbeda-beda sehingga memberikan gambaran yang agak membingungkan. Namun demikian, ketergantungan tahanan kapal pada LCB nampak jelas pada kecepatan yang tinggi. Sebagai upaya untuk mengatasi kerancuan tersebut maka semua informasi yang ada dikumpulkan dan diringkas pada Gb. 5.5.15. Namun ini harus dipandang sebagai LCB standar untuk metode itu saja.




Gambar 5.5.14. Koefisien tahanan gesek $C_F$ (menurut ITTC 1957) sebagai fungsi panjang kapal $L$ dan kecepatan $V$.




**Gambar 5.5.15.** LCB standar. Letak longitudinal titik benam yang dipandang terbaik.

Dalam hal ini, LCB standar tersebut didefinisikan sebagai fungsi linier angka Froude $F_n$. Karena tidak adanya ketergantungan yang pasti pada parameter lainnya yang tercatat maka LCB standar tersebut disajikan sebagai garis tunggal. Daerah yang diberi warna gelap di sekitar garis ini menunjukkan lingkup materi yang dikaji.

Sebagaimana disebutkan sebelumnya, karena letak LCB standar dianggap merupakan letak yang memberikan tahanan yang paling kecil maka letak yang lain pada prinsipnya akan memberikan tahanan yang lebih besar. Penambahan tahanan tersebut harus dicari dengan jalan mengalikan penyimpangan LCB dari standar, yaitu

$$\Delta \text{LCB} = \text{LCB} - \text{LCB}_{\text{standar}} (\text{LCB dalam } \%L) \tag{5.5.18}$$

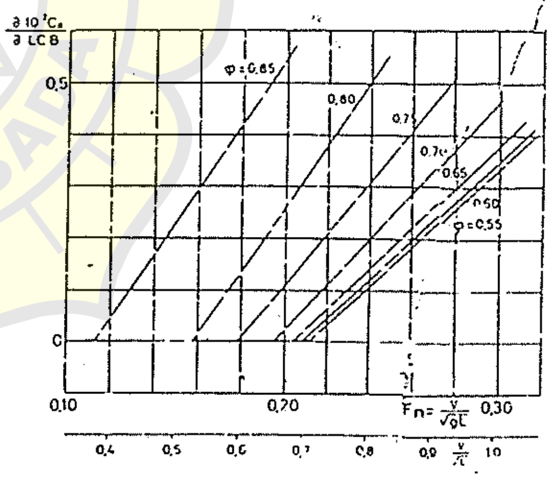
dengan faktor $\partial 10^3 C_R / \partial \text{LCB}$. Harga faktor ini dapat diperoleh dari Gb. 5.5.16, dan ini hanya berlaku untuk LCB yang berada di depan $\text{LCB}_{\text{standar}}$. Mengenai LCB yang berada di belakang $\text{LCB}_{\text{standar}}$, semua sumber yang ada mempunyai pendapat yang saling bertentangan. Namun demikian, karena kecenderungan terjadinya letak demikian itu sangat kecil maka pengabaian koreksi dalam hal itu tidak akan memberikan kesalahan yang berarti.

Dengan demikian maka koefisien tahanan sisa dengan koreksi tersebut untuk kapal yang mempunyai LCB di depan LCB standar adalah :

$$10^3 C_R = 10^3 C_{R(\text{standar})} + \frac{\partial 10^3 C_R}{\partial \text{LCB}} |\Delta \text{LCB}| \tag{5.5.19}$$

Bentuk badan kapal yang dilingkup dalam *Ship Resistance* adalah bentuk badan yang umum untuk jenis kapal niaga di sekitar tahun 1960 an, yaitu sampai dengan waktu diterbitkannya publikasi **Guldhammer** dan **Harvald** (1974). Bentuk badan kapal tersebut mempunyai buritan yang diletakkan tegak lurus di (berimpit dengan) sumbu tongkat kemudi (rudder stock) dan haluan yang tegak lurus di ujung depan garis air perancangan. Sejak tahun 1960 bentuk badan kapal telah mengalami pengembangan lebih lanjut, dan lebih bervariasi, misalnya berbagai bentuk haluan gembung yang telah dipakai secara luas. Rumus perhitungan tahanan yang diberikan di sini dapat dipakai baik untuk bentuk gembung modern atau yang lebih bervariasi maupun untuk bentuk tradisional, tetapi $L$ dan LCB harus mengikuti definisi yang lebih sesuai berikut ini. Panjang perhitungan $L$ didefinisikan sebagai panjang antara batas depan dan batas belakang displasemen, yaitu panjang terbesar dari bagian badan kapal yang berada di dalam air, dan ini adalah $L_{OS}$ menurut standar ITTC. Untuk kapal dengan bentuk tradisional tanpa gembung, panjang tersebut adalah panjang garis air.

LCB didefinisikan sebagai letak longitudinal titik benam, yaitu jarak antara titik ini dengan penampang tengah kapal, dan positif di belakang penampang tersebut. Midship section (penampang melintang tengah kapal, atau penampang tengah kapal, atau bidang tengah kapal, atau bidang tengah kapal) didefinisikan sebagai penampang melintang yang terletak sejauh 48,5%$L$ dari batas depan displasemen. $L$ adalah




**Gambar 5.5.16.** Koreksi koefisien tahanan sisa untuk LCB 1% di depan standar. Dengan demikian maka koreksi ini adalah $(\partial 10^3 C_R / \partial \text{LCB}) |\Delta \text{LCB}|$. $\Delta$LCB adalah jarak longitudinal antara LCB yang sebenarnya dengan LCB standar dalam persen $L$. Tidak ada koreksi untuk LCB yang terletak di belakang standar. Koreksi tersebut selalu positif.

panjang perhitungan menurut definisi tadi. Dengan demikian maka penampang tengah kapal menurut definisi ini adalah pertengahan antara kedua garis tegak bantu (auxuliary perpendiculars), $AP_1 - FP_1$; bandingkan di Gb. 5.5.17. Untuk bentuk normal, $AP_1 - FP_1$ ini akan sama dengan kedua garis tegak yang umum didefinisikan, $AP$ – $FP$.

## BENTUK BADAN KAPAL (BENTUK PENAMPANG MELINTANG DAN HALUAN)

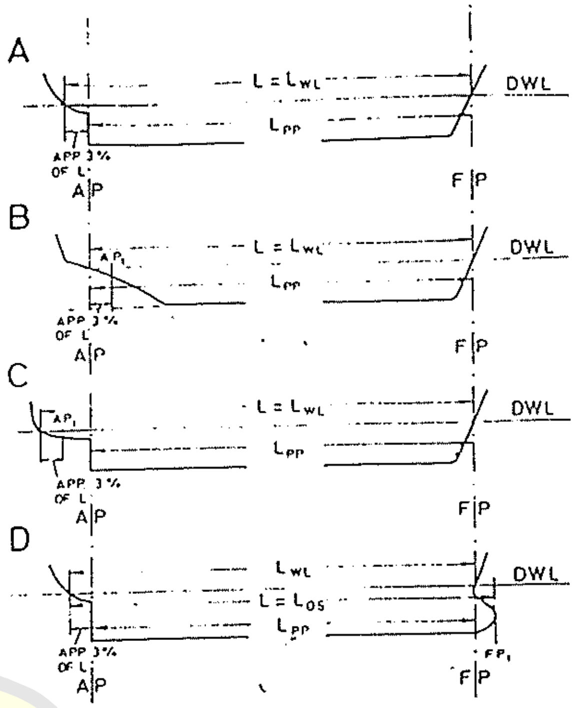
Sebagaimana disebutkan sebelumnya, kurva tahanan (yang diperoleh berdasarkan Gb. 5.5.5 – 5.5.13) dianggap berlaku untuk yang mempunyai bentuk "standar", yaitu penampangnya bukan yang benar-benar berbentuk U ataupun V. Karena itu, dalam menghitung daya efektif untuk perancangan awal umumnya tidak diperlukan koreksi untuk bentuk penampang badan kapal. Jika penampang tersebut merupakan penampang U atau V yang ekstrem maka harga $10^3 C_R$ dapat dikoreksi sebagai berikut : Koreksi $10^3 C_R$ untuk bentuk dari penampang

| | | |
|---|---|---|
| badan depan | ekstrem U −0,1 | ekstrem V +0,1 |
| badan belakang | ekstrem U +0,1 | ekstrem V −0,1 |

(5.5.20)

Koreksi ini berlaku untuk kecepatan $V/\sqrt{gL}$ dalam rentang 0,20 – 0,25. Selain itu, bentuk "standar" harus dipandang sebagai bentuk yang mempunyai garis yang dirancang dengan baik. Jika garis perancangan tersebut harus diubah untuk menyesuaikan kebutuhan operasional kapal, atau besarnya daya harus diberikan kelonggaran, maka disarankan agar $C_R$ dinaikkan sebesar 10% dan, untuk garis perancangan yang tidak optimal, mungkin sebesar 20% atau lebih.

Mengenai haluan, bentuk standar tersebut harus dipandang sebagai bentuk haluan kuno tanpa gembung. Untuk kapal dengan haluan gembung yang mempunyai harga $A_{BT}/A_X \geq 0{,}10$ ($A_{BT}$ adalah luas penampang haluan gembung di garis tegak depan dan $A_X$ adalah luas penampang tengah kapal) maka disarankan agar $10^3 C_R$ diberikan koreksi sebagai berikut :




Gambar 5.5.17. Definisi $L$ dan LCB. (a) Bentuk normal. Panjang buritan pada garis air umumnya 3% $L$. (b) Badan kapal tanpa linggi buritan (sternpost). AP umumnya diletakkan di ujung belakang DWL. Untuk koreksi LCB dipakai $AP_1$ 3% $L$ di depan ujung belakang garis air. (c) Badan kapal dengan panjang buritan yang ekstrem. Untuk koreksi LCB dipakai $AP_1$ 3% $L$ di depan ujung akhir garis air. $FP_1$ adalah batas depan displasemen.

Jika $A_{BT}/A_X$ = 0,10 maka bentuk haluan gembung akan tampak lebih menyolok. Koreksi untuk $0 < A_{BT}/A_X < 0{,}10$ dianggap berbanding lurus dengan ukuran gembung.

Koreksi ini hanya berlaku untuk kapal dalam kondisi bermuatan saja. Untuk kondisi balas maka koreksi karena adanya haluan gembung akan memberikan gambaran yang sebaliknya. Bentuk penuh ($\varphi > 0{,}70$) akan menunjukkan penurunan tahanan yang menyolok, harga koreksinya dua hingga tiga kali harga koreksi tersebut, sedangkan tahanan untuk bentuk ramping ($\varphi < 0{,}60$) umumnya akan cenderung naik.

| $F_n$ = 0,15 | 0,18 | 0,21 0,253 | 0,24 0,253 | 0,27 | 0,30 | 0,33 | 0,36 | $\varphi$ |
|---|---|---|---|---|---|---|---|---|
| | | +0,2 | 0 | −0,2 | −0,4 | −0,4 | −0,4 | 0,50 |
| | | +0,2 | 0 | −0,2 | −0,3 | −0,3 | −0,4 | 0,60 |
| | +0,2 | 0 | −0,2 | −0,3 | −0,3 | −0,3 [illegible] | [illegible] | 0,70 |
| +0,1 | 0 | −0,2 | | | | −0,3 0,58 | [illegible] | 0,80 |

(5.5.21)

ANGGOTA BADAN KAPAL

| | |
|---|---|
| Daun kemudi | Tidak ada koreksi bentuk standar sudah mencakup daun kemudi. |
| Lunas bilga (lunas sayap) | Tidak ada koreksi |
| Bos baling-baling | Untuk kapal penuh $C_R$ dinaikkan sebesar 3 – 5% (5.5.22) |
| Braket dan poros baling-baling | Untuk kapal ramping $C_R$ dinaikkan sebesar 5 – 8% |

TAHANAN TAMBAHAN

Pemberian koreksi pada $C_{FS}$ untuk kapal merupakan cara yang umum dilakukan dalam praktek dan sudah bertahun-tahun lamanya diterapkan untuk memperhitungkan pengaruh kekasaran permukaan kapal mengingat bahwa permukaan kapal tidak akan pernah semulus permukaan model, sekalipun kapal itu benar-benar baru dan catnyapun masih segar. Koefisien penambahan tahanan untuk korelasi model – kapal umumnya ditentukan sebesar $C_A = 0{,}0004$. Namun demikian, pengalaman lebih lanjut menunjukkan bahwa cara demikian itu tidak selalu benar. Karena itu, diusulkan koreksi untuk pengaruh kekasaran dan pengaruh sebagai berikut untuk kondisi pelayaran percobaan :

Untuk kapal dengan $L \leqq 100$ m, $10^3 C_A = 0{,}4$
$= 150$ m $\quad = 0{,}2$
$= 200$ m $\quad = 0$
$= 250$ m $\quad = -0{,}2$
$\geqq 300$ m $\quad = -0{,}3$

(5.5.23)

Beberapa pihak berpendapat bahwa koreksi yang diberikan di Bab 5, 5.2.4 lebih sesuai, yaitu,

| Displasemen | | |
|---|---|---|
| 1.000 t | $C_A = 0{,}6 \times 10^{-3}$ | |
| 10.000 t | $= 0{,}4 \times 10^{-3}$ | (5.5.24) |
| 100.000 t | $= 0$ | |
| 1.000.000 t | $= -0{,}6 \times 10^{-3}$ | |

Perlu disebutkan di sini bahwa koreksi untuk koefisien tahanan gesek ini masih agak meragukan.

ANGGOTA BADAN KAPAL

Koreksi $C_F$ untuk anggota badan kapal hanya dilakukan dengan jalan menaikkan $C_F$ sebanding dengan luas permukaan basah anggota badan begitu saja. Jadi,

$$C_{F'} = C_F \frac{S_1}{S} \tag{5.5.25}$$

$S$ adalah luas permukaan basah badan kapal dan $S_1$ adalah permukaan basah badan dan anggota badan kapal.

TAHANAN UDARA DAN TAHANAN KEMUDI

Tahanan udara dapat ditentukan dengan memakai data mengenai struktur yang berada di atas air dan data udara. Namun demikian, besarnya tahanan udara umumnya tidak terlalu penting, dan upaya yang harus dilakukan untuk mendapatkan hasil perhitungan yang tepat mungkin tidak memadai dengan pentingnya pengaruh udara tersebut. Karena itu, jika data mengenai angin dalam perancangan kapal tidak diketahui maka disarankan untuk mengoreksi $10^3 C_R$ sebagai berikut :

$$10^3 C_{AA} = 0{,}07 \tag{5.5.26}$$

Koreksi untuk tahanan kemudi mungkin sekitar

$$10^3 C_{AS} = 0{,}04 \tag{5.5.27}$$

tetapi tentu saja untuk kapal yang stabil dalam kondisi yang wajar koreksi tersebut dapat diabaikan.

Terlihat bahwa kedua koreksi tersebut kecil dan dalam perancangan awal koreksi ini umumnya sudah tercakup dalam tahanan tambahan.

KONDISI PELAYARAN DINAS

Tahanan dan daya efektif yang dihitung dengan memakai diagram yang diberikan di sini berlaku untuk kapal dalam kondisi pelayaran percobaan, yaitu, untuk kondisi ideal dari segi angin, gelombang, kedalaman air, dan kemulusan badan kapal. Untuk kondisi rata-rata pelayaran dinas harus diberikan kelonggaran tambahan pada tahanan dan daya efektif yang disebabkan oleh angin, laut, erosi, dan fouling pada badan kapal.

Tambahan kelonggaran ini sangat tergantung pada jalur pelayaran. Kelonggaran rata-rata untuk pelayaran dinas (kadang-kadang disebut sea margin atau service margin) untuk tahanan atau daya efektif diusulkan sebagai berikut :

Jalur pelayaran Atlantik Utara, ke Timur, untuk musim panas 15% dan musim dingin 20%

Jalur pelayaran Atlantik Utara, ke Barat, untuk musim panas 20% dan musim dingin 30% (5.5.28)

Jalur pelayaran Pasifik, 15 – 30%

Jalur pelayaran Atlantik Selatan dan Autralia, 12 – 18%

Jalur pelayaran Asia Timur, 15 – 20%

Tahanan total harus dihitung dengan memakai rumus

$$R_T = C_T(\tfrac{1}{2}\rho V^2 S) \qquad (5.5.29)$$

$S$ adalah luas permukaan basah badan kapal.

Banyak sekali metode untuk memperkirakan $S$. Dianjurkan untuk memakai salah satu dari dua metode berikut ini :

1. Publikasi FORMDATA I – V (Guldhammer, 1962, 1963, 1967, 1969, 1973) memuat data hidrostatik dari suatu seri yang sangat baik yang terdiri dari sejumlah bentuk kapal yang divariasikan secara sistematik. Permukaan basah semua bentuk memakai koefisien

$$\boxed{S} = \frac{S}{L(B + 2{,}5T)} \qquad (5.5.30)$$

Jika dari perancangan awal kapal bentuk badan kapal yang sebenarnya hampir sama dengan salah satu bentuk yang diberikan dalam FORMDATA tersebut maka akan diperoleh $S$ dengan kesalahan kurang dari 1%.

2. Permukaan basah untuk kapal niaga yang normal dapat dihitung dengan memakai rumus berikut ini (versi rumus Mumford) :

$$S = 1{,}025L_{PP}(\delta_{PP}B + 1{,}7T) \qquad (5.5.31)$$

Semua diagram $\boxed{S}$ dan rumus yang disertakan dalam FORMDATA berlaku untuk bentuk kapal yang buritan dan haluannya masing-masing terletak pada garis tegak belakang dan garis tegak depan. Hampir semua kapal mempunyai luas permukaan basah yang sesuai dengan asumsi tersebut, karena luas yang kurang dan luas yang lebih akan saling berimbang. Untuk kapal yang mempunyai juntaian (= bagian yang menggantung = overhang), atau lekukan (= bagian yang masuk = cutout), di dalam air yang besar maka hal tersebut harus diperhitungkan (diberikan kelonggaran).

Perhitungan tahanan dan daya efektif dapat dilakukan dengan prosedur seperti yang ditunjukkan dalam Contoh Formulir untuk Menghitung Daya Efektif (lihat halaman 132). Perhitungan dapat dilakukan dengan memakai komputer mini. Kini banyak arsitek kapal yang mempunyai program untuk perhitungan demikian itu.

Dalam tahap perancangan pertanyaan utama yang harus dituntaskan adalah jenis dan ukuran mesin (misalnya banyaknya dan ukuran silinder, jika memakai mesin disel). Tahanan harus ditentukan dengan tingkat kepastian yang memadai sehingga, atas dasar daya efektif $P_E$, daya poros akan dapat ditentukan dengan tingkat ketepatan yang cukup untuk dapat menjawab dengan aman pertanyaan vital tadi.

Di lain pihak, upaya untuk mencapai ketepatan yang melebihi dari yang diperlukan untuk menyelesaikan masalah tahanan tidak mempunyai arti yang besar. Tingkat ketidakpastian dalam faktor yang terlibat cukup tinggi, dan pembaca diingatkan untuk tidak membuang waktu untuk memburu ketepatan yang tersisa dengan perhitungan yang sifatnya hanya pendekatan.

Untuk kapal yang bertenaga mesin disel, merubah jumlah silinder, katakanlah dari 6 menjadi 7, atau dari 11 menjadi 12, akan berarti merubah daya masing-masing sebesar sekitar 17% atau 8%. Dengan memodifikasi tekanan efektif rata-rata dan jumlah kisaran maka akan dapat merubah luaran menerus (continuous output) sebesar sekitar 10%.

Mesin turbin mempunyai tingkatan menurut jenisnya.

Atas dasar pertimbangan tersebut barangkali tingkat ketepatan yang diperlukan dalam penentuan $P_E$ untuk perancangan awal kapal dapat ditentukan sebesar 1 hingga 5%. Ketepatan ini dapat dengan mudah diperoleh dengan memakai diagram dan formulir Perhitungan yang diberikan di sini.

Diagram dan rumus tersebut dapat pula dipakai dengan cara sebagai berikut. Setiap hasil yang diperoleh dari percobaan yang dilakukan sendiri oleh si arsitek kapal di tangki percobaan dicocokkan dengan diagram. Data ini kemudian dikoreksi dengan memakai rumus dan diagram tadi dan selanjutnya dipakai sebagai dasar materi untuk menentukan tahanan kapal baru yang akan diajukan dalam usulan. Sering bahwa dengan prosedur demikian ini dapat diperoleh hasil yang sangat baik.

Ref. no. 4



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, $\chi_r$, the torque, $M_{rs}$, in kg-m developed on the rudder head and the time, $\tau$, 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 $\tau$ 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 $\eta_{sg}$ 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 $2\alpha° = 70°$ | $2\alpha° = 64°$ |
|---|---|---|---|
| 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

$$M_m = \frac{M_{rs}}{i_{sg}\eta_{sg}} \text{ kg-m} \quad (312)$$

$$n_m = i_{sg} n_{rs} \text{ rpm} \quad (313)$$

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 $\omega_{rs}$ of the rudder stock can be calculated from the following formulas:

$$\omega_{rs} = \frac{\pi n_{rs}}{30} \text{ 1/sec} \quad (314)$$

$$\omega_{rs} = \frac{2\alpha°}{\tau}\frac{\pi}{180°} \text{ 1/sec} \quad (315)$$

where $\alpha°$ = maximum rudder angle from the middle-line plane.

It follows from formula (314) that

$$n_{rs} = \frac{30\omega_{rs}}{\pi} \text{ rpm} \quad (316)$$

Combining equations (315) and (316) we obtain

$$n_{rs} = \frac{30}{\pi}\frac{2\alpha°}{\tau}\frac{\pi}{180} = \frac{1}{3}\frac{\alpha°}{\tau} \text{ rpm} \quad (317)$$

Combining equations (313) and (317) we can write

$$i_{sg} = \frac{n_m}{n_{rs}} = \frac{n_m}{\frac{1}{3}\frac{\alpha°}{\tau}} = 3n_m\frac{\tau}{\alpha°} \quad (318)$$

Taking equations (314) and (315) into consideration, the power developed on the rudder stock is

$$N_{rs} = \frac{M_{rs}\omega_{rs}}{75} = \frac{M_{rs}}{75}\frac{2\alpha°}{\tau}\frac{\pi}{180°} = 4.65\frac{M_{rs}}{10^4}\frac{\alpha°}{\tau} \text{ metric hp} \quad (319)$$

$$N_{rs} = \frac{M_{rs}\omega_{rs}}{75} = \frac{M_{rs}}{75}\frac{\pi n_{rs}}{30} = 1.395\frac{M_{rs}n_{rs}}{10^3} \approx 1.4\frac{M_{rs}n_{rs}}{10^3} \text{ metric hp} \quad (320)$$

The shaft horse power of the steering engine motive unit will be

$$N_m = \frac{N_{rs}}{\eta_{sg}} = 4.65\frac{M_{rs}}{10^4\eta_{sg}}\frac{\alpha°}{\tau} \text{ metric hp} \quad (321)$$

$$\boxed{N_m = \frac{N_{rs}}{\eta_{sg}} = 1.4\frac{M_{rs}}{10^3\eta_{sg}}n_{rs}} \text{ metric hp} \quad (322)$$

The shaft horse power can also be determined from the shaft torque

Reference No. 4



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 4 | 360 |
| 3,000 | 0.25 | 750 |
| 4,500 | 0.2 | 900 |
| 7,000 | 0.167 | 1,165 |
| 12,000 | 0.156 | 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



$H$ = height of the side amidships, measured from the upper edge of the keel to the lower edge of the strength deck 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 $L$

$k_{1sp} = 1.5\frac{l_{st}}{L}$ 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 tabulated 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

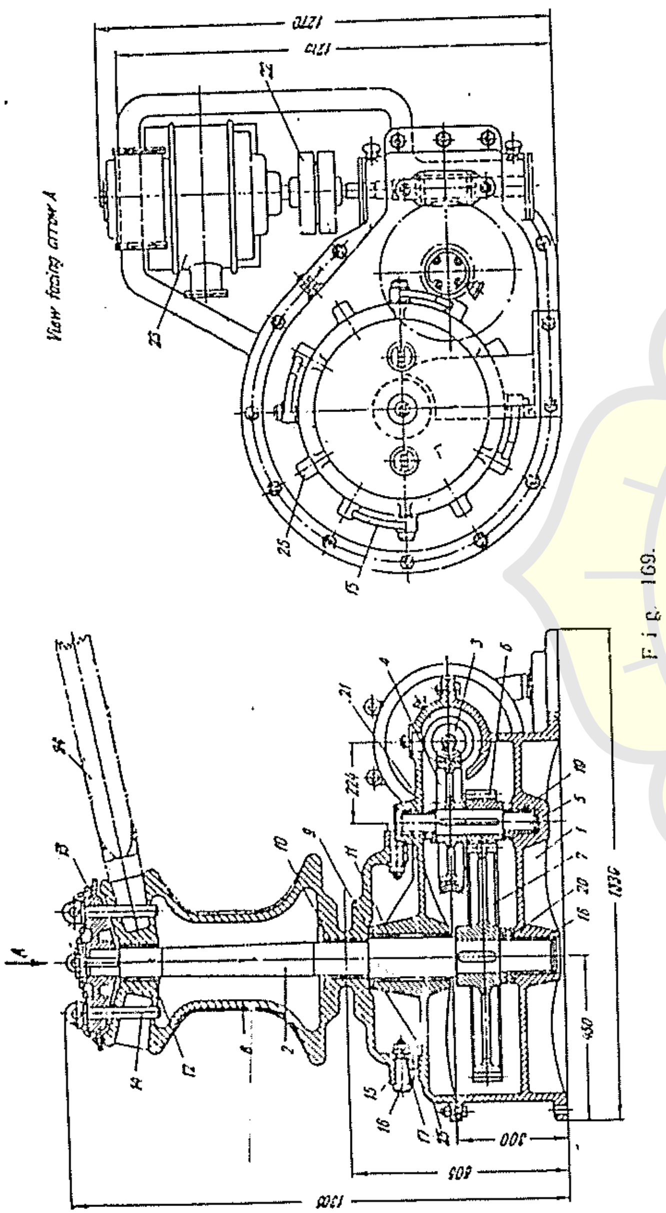


Fig. 169.



## 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
$\gamma_a$ = 7,750 = density of the material of the anchor, kg per cu m
$\gamma_w$ = 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

$$T_{cl} = 2f_h(G_a + p_aL_a)\left(1 - \frac{\gamma_w}{\gamma_a}\right) = 2\times 1.35\,(G_a + p_aL_a)\left(1 - \frac{1{,}025}{7{,}750}\right) = 2.35\,(G_a + p_aL_a) \text{ kg} \quad (383)$$

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

$$\left.\begin{aligned} &(a)\ p_{ao} = 0.023\,d_c^2 \text{ kg for open-link chain} \\ &(b)\ p_{as} = 0.0218\,d_c^2 \text{ kg for stud-link chain} \end{aligned}\right\} \quad (384)$$

According to the U.S.S.R. Shipping Register the aft 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.*

* In breaking away one anchor from the bottom

$$T_{cl} = 2G_a + 1.175\,(G_a + p_aL_a) \text{ kg}$$

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

$$V_f = V_e - D_1 = V_e - \frac{D}{6}$$

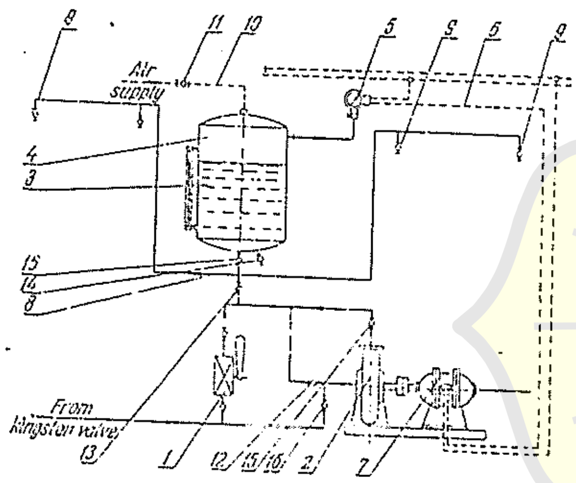



Fig. 189.

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.



(b) 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

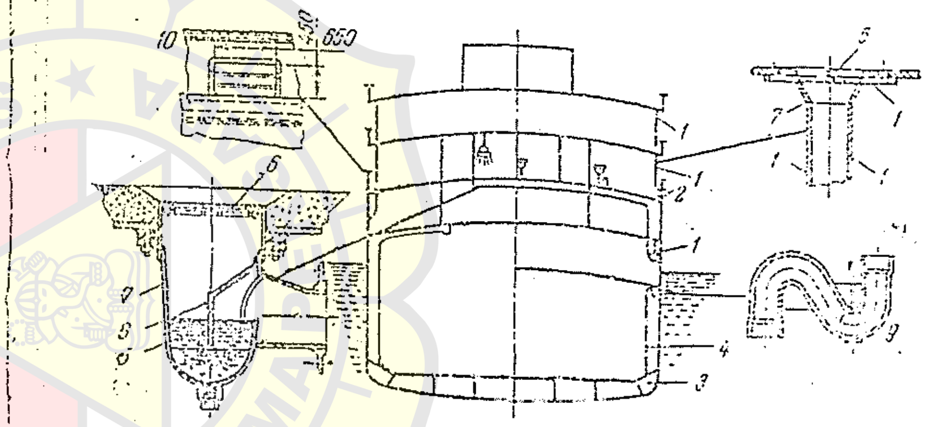


Fig. 190.

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.

Ref. no 6



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.

## 76. 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 for 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
- waiting time
(- according to owner's desire).

$P_{Bme}$ = 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).

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: $\gamma = 0.95\ t/m^3$

Required volume of storage tanks

$$V_{oil} = \frac{W}{\gamma} \quad [m^3]$$

Reference no. 6



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: $V_{diesel} = \frac{w_{diesel}}{\gamma_{diesel}}\ [m^3]$

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 $\gamma_{lubr} = 0.90\ t/m^3$ ; $V = \frac{w}{\gamma}\ (m^3)$

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 by cofferdams.

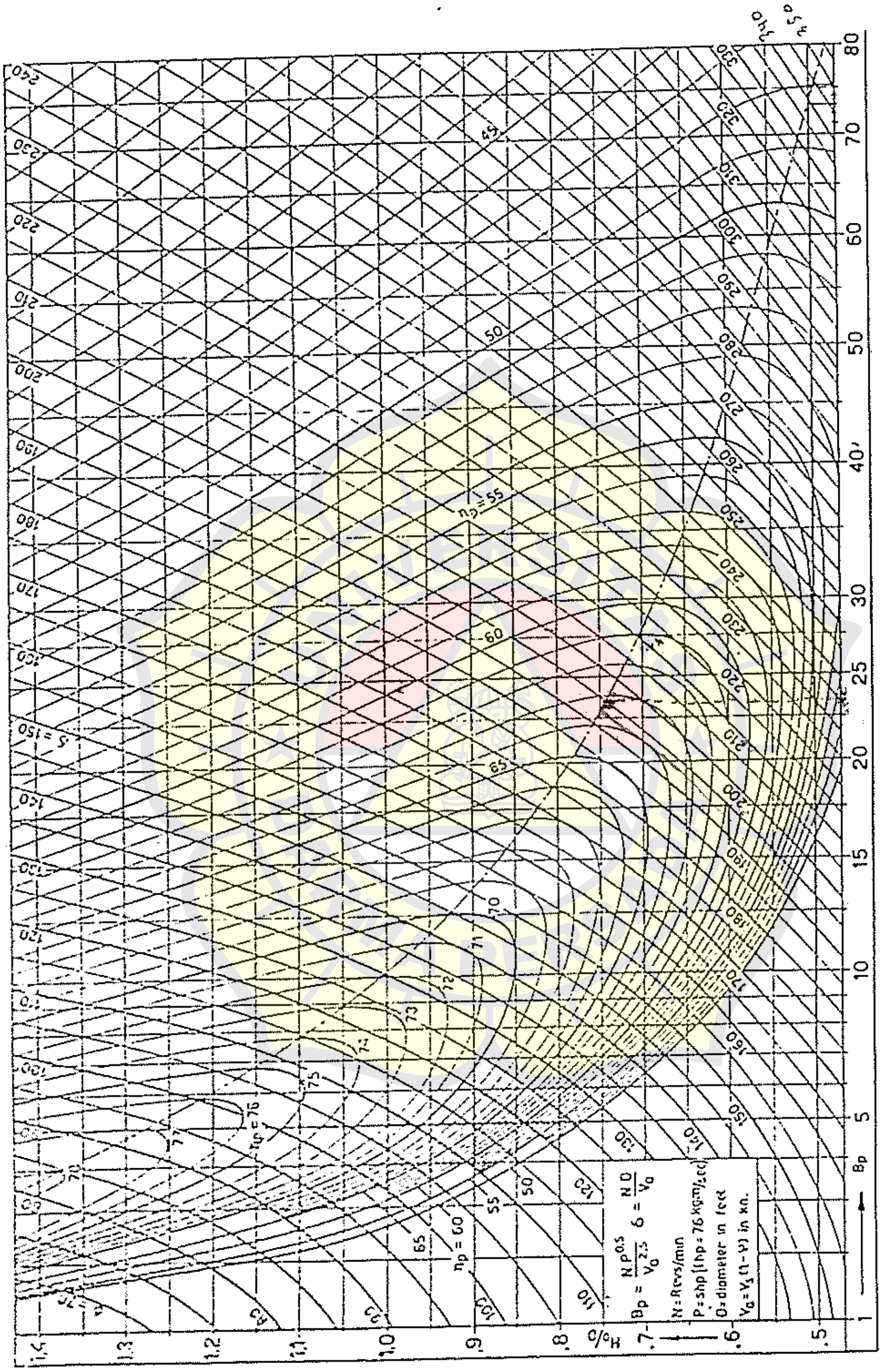



Fig. 3.15 Troost B.4 – 40 $Bp - \delta$ Chart

Ref no 6



b) 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 seaway; 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.

c) 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). |

c. Type and Location of Main Engine

is another part of the contract influencing ship design. (Ship weight, volume, fuel consumption). economy is determined by the choice of the main engine type, also

Ref. no. 6



by the quality of the crew (maintenance). The degree of possible automation depends on the personal quality as well. Sometimes the choice of the engine depends on the route because of maintenance and engine maker.

7.8. Crew Members

It depends on route, type of ship and on national rules. It is possible that the number of crew members of two equal ships is completely different, because one has an European crew and the other has an Asian crew. The rooms are devided in functions of the crew: deck worker, engine worker ... .

7.9. Outfit and Equipment

- Cargo gear, winches
- hatchway covers
- shifting equipment
- anchor winches.

7.10. Classification, Rules

have to be observed.

7.11. Restrictions of Dimensions

- Draught (because of port depth, kuala estuary trading, canals)
- breadth (canals, locks)
- length (locks, length of berth) berth = tempat membuang sauh,
- stability requirements.

7.12. Tonnage of Ships

Lit.: Results of the International Tonnage Conference London Hansa 1969, p. 1936.

The size of ships is resmi officially ditetapkan confirmed by tonnage. muatan Charge are dependent on tonnage, for example in ports, canals, for pilots ... . Most of the shipbuilding statistics are based on tonnage.

Tonnage unit: gross ton

$1 \text{ grt} = 100 \text{ cbf} = 2.83 \text{ m}^3$.

Ref. no 6



The new IMCO tonnage rules contain 7 rules being much easier than the former rules. The most important rules are no. 3 (gross tonnage) and no. 4 (net tonnage).

Gross tonnage

$GT = (0.2 + 0.02 \log_{10} \cdot V) \cdot V$

V = total volume of all closed rooms $[m^3]$

Net tonnage

$NT = (0.2 + 0.02 \cdot \log_{10} \cdot V_h) \cdot V_h \cdot \left(\frac{4 \cdot T}{3 \cdot D}\right)^2 + \left(1{,}25 \cdot \frac{GT + 10000}{10000}\right)\left(N_1 + \frac{N_2}{10}\right)$

$V_h$ = total volume of all holds in $[m^3]$

T = draught in [m] (midships)

D = depth in [m] "

$N_1$ = number of passengers in cabins with not more than 8 beds

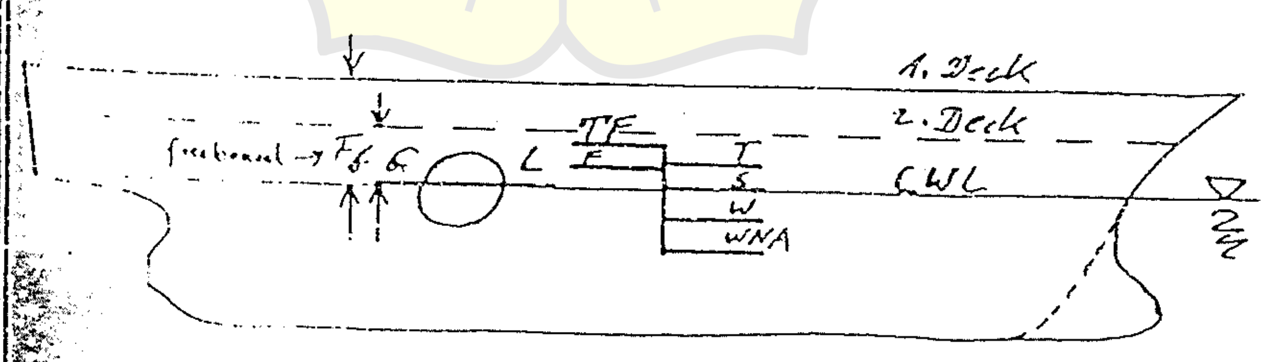
$N_2$ = number of other passengers.

Classification and notation of ship types according to their superstructure, freeboard and tonnage

Definition of freeboard:

Freeboard generally means the minimum distance from the water surface to the highest continuous deck measured at $L_{pp}/2$.

Ref.: International Freeboard Convention 1966.




Definition of superstructures:

Superstructures are erections on main deck the side walls of which have a distance of not more than 0.04 · B from the

WILBERT F. ST...
JEROLD W. JONES
... ndisian udara
(Terjemahan)
SUPRATMAN HARA



Daya untuk setiap kilowatt refrigerasi merupakan kebalikan dari koefisien prestasi, dan suatu sistem refrigerasi yang efisien akan memiliki nilai daya per-kilowatt refrigerasi yang rendah, tetapi mempunyai koefisien prestasi yang tinggi.

Contoh berikut ini menggambarkan perhitungan untuk menentukan prestasi daur kompresi uap standar.

**Contoh 10-1** Suatu daur kompresi-uap standar menghasilkan 50 kW refrigerasi dengan menggunakan refrigeran 22, bekerja pada suhu pengembunan 35°C dan suhu penguapan −10°C. Hitunglah (*a*) dampak refrigerasi dalam kilojoule per-kilogram, (*b*) laju pendauran refrigeran dalam kilogram per-detik, (*c*) daya yang dibutuhkan oleh kompresor dalam kilowatt, (*d*) koefisien prestasi, (*e*) laju alir volume yang diukur pada pipa hisap kompresor, (*f*) daya per kilowatt refrigerasi dan (*g*) suhu buang pada kompresor.

*Penyelesaian* Langkah pertama penyelesaian adalah menggambar diagram tekanan-entalpi (Gambar 10-12) dan menentukan dari Tabel A-6, Tabel A-7, dan Gambar A-4, entalpi-entalpi pada titik-titik penting. Nilai $h_1$ adalah entalpi uap jenuh pada −10°C, yaitu 401,6 kJ/kg.

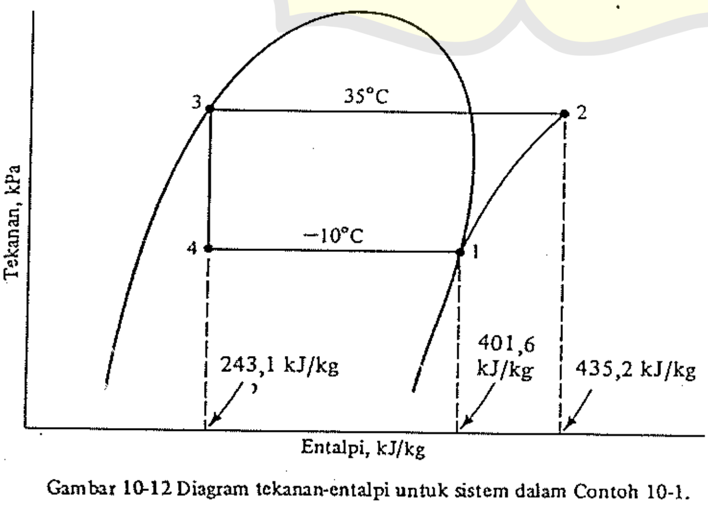
Untuk menemukan $h_2$ melalui garis entropi tetap geser titik 1 hingga mencapai tekanan jenuh yang sesuai dengan suhu 35°C. Tekanan pengembunan ini adalah 1354 kPa, dan nilai $h_2$ = 435,2 kJ/kg.

Nilai $h_3$ dan $h_4$ identik, dan sama dengan entalpi cairan jenuh pada 35°C, yaitu 243,1 kJ/kg. Sehingga

$$h_1 = 401{,}6 \text{ kJ/kg} \qquad h_2 = 435{,}2 \text{ kJ/kg}$$
$$h_3 = h_4 = 243{,}1 \text{ kJ/kg}$$

(*a*) Dampak refrigerasi:

$$h_1 - h_4 = 401{,}6 - 243{,}1 = 158{,}5 \text{ kJ/kg}$$




Gambar 10-12 Diagram tekanan-entalpi untuk sistem dalam Contoh 10-1.

(*b*) Laju pendauran refrigeran dapat dihitung dengan membagi kapasitas refrigerasi dengan dampak refrigerasi :

$$\text{Laju alir} = \frac{50 \text{ kW}}{158{,}5 \text{ kJ/kg}} = 0{,}315 \text{ kg/det}$$

(*c*) Daya yang dibutuhkan oleh kompresor adalah kerja kompresi per-kilogram dikalikan dengan laju aliran refrigeran

$$\text{Daya kompresor} = (0{,}315 \text{ kg/det})\,(435{,}2 - 401{,}6 \text{ kJ/kg}) = 10{,}6 \text{ kW}$$

(*d*) Koefisien prestasi adalah laju pendinginan dibagi dengan daya kompresor

$$\text{Koefisien prestasi} = \frac{50 \text{ kW}}{10{,}6 \text{ kW}} = 4{,}72$$

(*e*) Laju aliran pada seksi masuk kompresor memerlukan data volume spesifik refrigeran pada titik 1. Dari Tabel A-6 atau Gambar A-4 nilai ini 0,0654 $m^3$/kg, sehingga

$$\text{Laju aliran volume} = (0{,}315 \text{ kg/det})\,(0{,}0654 \text{ m}^3/\text{kg}) = 0{,}0206 \text{ m}^3/\text{det} = 20{,}6 \text{ L/det}$$

(*f*) Daya kompresor per kilowatt refrigerasi (yang merupakan kebalikan dari koefisien prestasi) adalah,

$$\text{Daya refrigerasi} = \frac{10{,}6 \text{ kW}}{50 \text{ kW}} = 0{,}212 \text{ kW/kW}$$

(*g*) Suhu buang kompresor adalah suhu uap panas-lanjut pada titik 2, yang dari Gambar A-4 didapatkan sebesar 57°C.

Semua sifat-sifat di dalam Contoh 10-1 dapat diambil dari Tabel A-6, kecuali $h_2$ dan $t_2$ yang berada di dalam daerah panas-lanjut. Sifat-sifat pada titik 2 dapat ditentukan baik dari diagram tekanan-entalpi, Gambar A-4, atau dari Tabel A-7. Tabel yang lebih lengkap tentang sifat uap panas-lanjut juga tersedia,[1] dan juga untuk refrigeran lainnya, dapat ditemukan. Sifat-sifat refrigeran pada titik 2 ditentukan dengan melakukan interpolasi pada Tabel A-7, pada tekanan dan entropi yang cocok.

**10-15 Penukar kalor (heat exchangers)** Beberapa sistem refrigerasi dilengkapi dengan penukar kalor jalur cair-ke-hisap (liquid-to-suction), yang menurunkan suhu (subcools) cairan dari kondenser dengan uap isap (suction vapor) yang datang dari evaporator. Susunannya diperlihatkan dalam Gambar 10-13*a*, dan diagram tekanan-entalpi yang bersangkutan dalam Gambar 10-13*b*.

Cairan jenuh pada titik 3 yang berasal dari kondensor didinginkan hingga titik 4 dengan cara bertukar kalor dengan uap pada titik 6 yang dipanaskan hingga mencapai titik 1. Dari keseimbangan kalor, $h_3 - h_4 = h_1 - h_6$. Dampak refrigerasinya dapat berbentuk $h_6 - h_5$ atau $h_1 - h_3$. Gambar 10-14 menunjukkan penampang terpotong penukar kalor jalur cair-hisap (liquid-to-suction heat exchanger).

Dibandingkan dengan daur kompresi uap standar, sistem yang menggunakan penukar kalor nampaknya lebih memiliki keuntungan yang jelas karena naiknya dampak refrigerasi. Kapasitas dan koefisien prestasi tampaknya dapatditingkatkan. Tetapi hal ini tidak sepenuhnya benar. Walaupun dampak refrigerasi dapat ditingkatkan, tetapi kompresi terdorong jauh masuk ke dalam daerah panas-lanjut, sehingga kerja kompresi akan lebih besar dibandingkan dengan yang dekat dengan garis uap-jenuh. Dari hal

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 |

$$\eta_v = \frac{Q_s}{Q_{th}}$$

di mana $Q_s$: Volume gas yang dihasilkan, pada kondisi tekanan dan temperatur isap ($m^3$/min)

$Q_{th}$: Perpindahan torak ($m^3$/min)

Besarnya efisiensi volumetris ini dapat dihitung secara teoritis berdasarkan volume gas yang dapat diisap secara efektif oleh kompres pada langkah isapnya, seperti telah diuraikan di atas. Dari perhitungan tersebut diperoleh rumus yang dapat ditulis sbb:

$$\eta_v \approx 1 - \varepsilon\left\{\left(\frac{P_d}{P_s}\right)^{1/n} - 1\right\} \tag{2.19}$$

di mana $\varepsilon$: $V_c/V_s$, volume sisa (clearance) relatip,

$P_d$: Tekanan keluar dari silinder tingkat pertama (kgf/cm$^2$ abs),

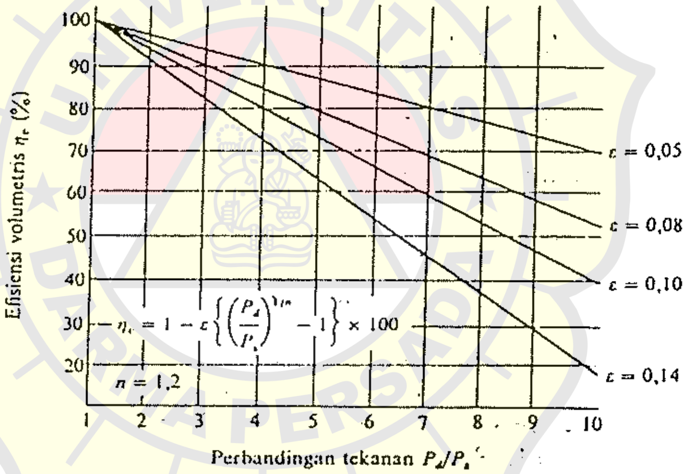
$P_s$: Tekanan isap dari silinder tingkat pertama (kgf/cm$^2$ abs).

$n$: Koefisien ekspansi gas yang tertinggal di dalam volume sisa; untuk udara, $n = 1{,}2$.

Tanda $\approx$ berarti "kira-kira sama dengan", karena rumus (2.19) diperoleh dari perhitungan teoritis. Adapun harga $\eta_v$ yang sesungguhnya adalah sedikit lebih kecil dari harga yang diperoleh dari rumus di atas karena adanya kebocoran melalui cincin torak dan katup-katup, serta tahanan pada katup-katup.

Dalam Gb. 2.11 diperlihatkan pengaruh $\varepsilon$ dan $P_d/P_s$ pada efisiensi volumetris $\eta_v$.

Sehubungan dengan hal-hal di atas dapat dimengerti jika efisiensi volumetris juga tergantung pada faktor-faktor rancangan kompresor seperti bentuk dan ukuran silinder, serta bentuk, ukuran, dan susunan katup-katup.




Gb. 2.11 Efisiensi volumetris dan perbandingan tekanan.

### 2.4.2 Efisiensi adiabatik keseluruhan

Efisiensi kompresor ditentukan oleh berbagai faktor seperti tahanan aerodinamik di dalam katup-katup, saluran-saluran, pipa-pipa, kerugian mekanis, efektivitas pendinginan, dll. Namun, menentukan secara tepat pengaruh masing-masing faktor tersebut adalah sangat sulit. Karena itu faktor-faktor ini digabungkan dalam efisiensi adiabatik keseluruhan.

Efisiensi adiabatik keseluruhan didefinisikan sebagai daya yang diperlukan untuk memampatkan gas dengan siklus adiabatik (menurut perhitungan teoritis), dibagi dengan daya yang sesungguhnya diperlukan oleh kompresor pada porosnya. Dalam rumus, efisiensi ini dapat ditulis sbb:

$$\eta_{ad} = \frac{L_{ad}}{L_s} \tag{2.20}$$

di mana $\eta_{ad}$: Efisiensi adiabatik keseluruhan (biasanya dinyatakan dalam %),
$L_{ad}$: Daya adiabatik teoritis (kW)
$L_s$: Daya yang masuk pada poros kompresor (kW).

Besarnya daya adiabatik teoritis dapat dihitung dengan rumus

$$L_{ad} = \frac{mk}{k-1}\frac{P_s Q_s}{6120}\left[\left(\frac{P_d}{P_s}\right)^{(k-1)/mk} - 1\right], \quad (\text{kW}) \tag{2.21a}$$

$P_s$: Tekanan isap tingkat pertama (kgf/m² abs)
$P_d$: Tekanan keluar dari tingkat terakhir (kgf/m² abs)
$Q_s$: Jumlah volume gas yang keluar dari tingkat terakhir (m³/min) dinyatakan pada kondisi tekan dan temperatur isap
$k$: $c_p/c_v$
$m$: Jumlah tingkat kompresi; lihat keterangan pada Pers. (2.16).

Jika dalam rumus ini dipakai satuan tekanan Pa maka Pers. (2.21) ditulis sebagai

$$L_{ad} = \frac{mk}{k-1}\frac{P_s Q_s}{60000}\left[\left(\frac{P_d}{P_s}\right)^{(k-1)/mk} - 1\right], \quad (\text{kW}) \tag{2.21b}$$

Dalam Tabel 2.7 diberikan harga-harga daya adiabatik teoritis yang diperlukan untuk mengkompresikan 1 m³/min udara dengan kondisi standar sebagai hasil perhitungan berdasarkan rumus di atas. Dari tabel terlihat bahwa daya yang diperlukan untuk kompresi 2 tingkat harganya lebih kecil dari pada kompresi 1 tingkat. Harga yang lebih rendah ini diperoleh pada kompresor 2 tingkat yang menggunakan pendingin antara (inter-cooler) di antara tingkat pertama dan tingkat ke dua. Penggunaan pendingin antara akan memperkecil kerja kompresi. Jika tidak digunakan pendingin antara, maka daya yang diperlukan untuk kompresi 2 tingkat adalah sama besarnya dengan daya untuk 1 tingkat, pada perbandingan tekanan yang sama.

Sebagai contoh, dari Tabel 2.7 terbaca bahwa untuk kompresi 1 tingkat sampai 7 kgf/cm² (g) atau 8,033 kgf/cm² abs, diperlukan daya sebesar 4,7074 kW. Ini diperoleh dari Pers. (2.21) dengan mengambil harga $k = 1,4$ dan $m = 1$. Daya sebesar 4,7074 kW tersebut juga akan diperlukan untuk kompresi 2 tingkat tanpa pendingin antara. Namun jika digunakan pendingin antara maka daya yang diperlukan menjadi sebesar 4,0227 kW. Harga ini dapat diperoleh dari Pers. (2.21a) jika diambil $k = 1,4$ dan $m = 2$.

Selanjutnya efisiensi adiabatik keseluruhan dapat dihitung menurut contoh sebagai berikut. Seandainya untuk sebuah kompressor 2 tingkat yang memampatkan udara menjadi 7 kgf/cm² (g) diperlukan daya poros sebesar 5,4 kW, maka dengan daya adiabatik teoritis sebesar 4,022 kW, kompresi ini mempunyai efisiensi adiabatik keseluruhan sebesar

$$\eta_{ad} = \frac{L_{ad}}{L_s} = \frac{4{,}022\ \text{kW}}{5{,}4\ \text{kW}} = 0{,}745 = 74{,}5\%$$

Tabel 2.7 Daya yang diperlukan untuk kompresi adiabatis teoritis.

| Tekanan ($kgf/cm^2$ (G)) | Kompresi 1-tingkat (kW) | Kompresi 2-tingkat (kW) | Tekanan ($kgf/cm^2$ (G)) | Kompresi 2-tingkat (kW) |
|---|---|---|---|---|
| 0,5 | 0,7053 | | 11 | 4,9639 |
| 1 | 1,2608 | | 12 | 5,1563 |
| 1,5 | 1,7256 | | 13 | 5,3365 |
| 2 | 2,1288 | | 14 | 5,5060 |
| 2,5 | 2,4869 | | 15 | 5,6661 |
| 3 | 2,8105 | | 16 | 5,8178 |
| 3,5 | 3,1065 | | 17 | 5,9621 |
| 4 | 3,3801 | 2,9994 | 18 | 6,0997 |
| 4,5 | 3,6348 | 3,2012 | 19 | 6,2313 |
| 5 | 3,8736 | 3,3879 | 20 | 6,3573 |
| 5,5 | 4,0987 | 3,5618 | 21 | 6,4783 |
| 6 | 4,3118 | 3,7247 | 22 | 6,5947 |
| 6,5 | 4,5143 | 3,8779 | 23 | 6,7068 |
| 7 | 4,7074 | 4,0227 | 24 | 6,8150 |
| 7,5 | 4,8922 | 4,1599 | 25 | 6,9195 |
| 8 | 5,0693 | 4,2904 | 26 | 7,0215 |
| 8,5 | 5,2396 | 4,4148 | 27 | 7,1195 |
| 9 | 5,4036 | 4,5338 | 28 | 7,1246 |
| 9,5 | 5,5619 | 4,6477 | 29 | 7,3069 |
| 10 | 5.7149 | 4,7572 | 30 | 7,3965 |

*Catatan*: Daya yang dinyatakan di atas adalah daya kompresi adiabatis teoritis untuk setiap $m^3$/menit udara bebas. 1 $kgf/cm^2$ = 0,0980665 MPa. G berarti tekanan lebih (gage)

Semakin tinggi efisiensi adiabatik keseluruhan sebuah kompresor, berarti semakin kecil daya poros yang diperlukan untuk perbandingan kompresi dan kapasitas yang sama. Namun setinggi-tinggi efisiensi ini, harganya tidak akan mencapai 100%.

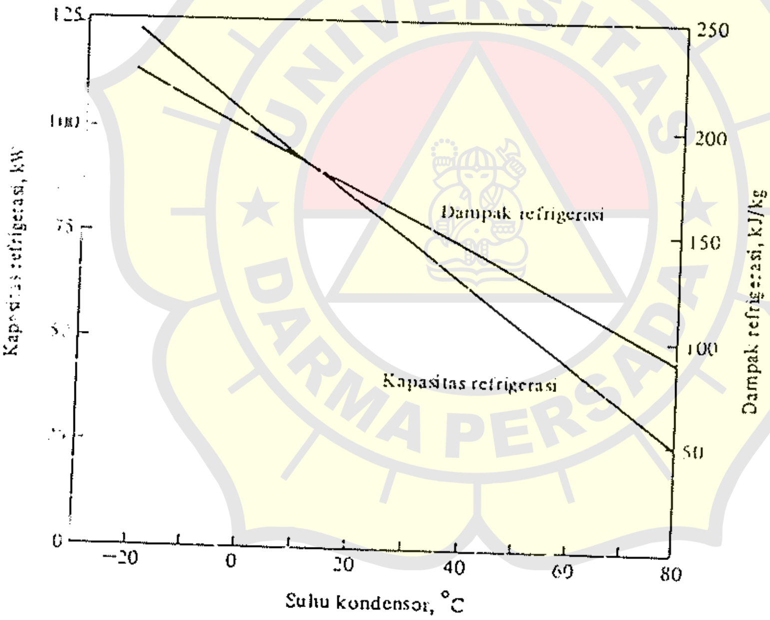
Selanjutnya, karena harga daya adiabatis teoritis untuk kompresor 1 tingkat berbeda dengan harga untuk kompresor 2 tingkat, maka memperbandingkan efisiensi kompresor harus dilakukan di antara yang sama jumlah tingkatnya.

Sebagai kesimpulan dapat dikemukakan bahwa efisiensi diabatik keseluruhan merupakan petunjuk bagi baik buruknya performansi dan ekonomi sebuah kompresor. Adapun efisiensi volumetris hanya merupakan suatu koefisien yang diperlukan oleh perencana kompresor dan tidak penting artinya bagi pemakai.

si volumetrik yang mempengaruhi laju alir massa, yang menunjukkan suatu penurunan akibat naiknya suhu kondensor. Gambar 11-10 menunjukkan penurunan tersebut yang progresif. Kapasitas refrigerasi adalah hasil kali antara dampak refrigerasi dan laju aliran massa, yang keduanya akan turun bila suhu kondensor naik. Jadi kapasitas refrigerasi turun agak lebih cepat karena naiknya suhu kondensor.

Karakteristik yang penting lagi adalah daya - yang diperlihatkan dalam Gambar 11-11. Daya kompresor adalah hasil perkalian antara kerja kompresi yang bersatuan kilojoule per-kilogram dan laju alir massa. Bila suhu kondensor naik, maka kerja kompresi dan laju alir massa menurun, sehingga daya naik mencapai puncak dan kemudian mulai turun. Sifat yang sama dengan daya ini, yaitu sebagai fungsi dari suhu evaporator, ditunjukkan dalam Gambar 11-6.

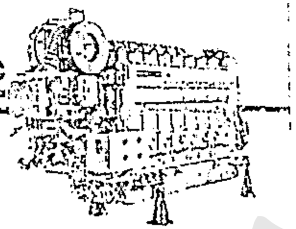
Beberapa penjelasan tentang arti dan sifat-sifat yang terdapat di dalam Gambar 11-9 hingga 11-11 adalah sebagai berikut: pencapaian puncak-puncak daya dapat terjadi dalam kompresor-kompresor nyata seperti juga pada kompresor ideal, tetapi hanya terjadi bila dilakukan pemompaan dari suhu-suhu rendah evaporator. Kompresi satu tingkat dari suhu penguapan $-20°C$ hingga suhu pengembunan $60°C$ yang menghasilkan puncak seperti pada Gambar 11-11, tidaklah umum. Dengan perbedaan suhu yang lebih sedikit antara kondensor dan evaporator, diperkirakan bila suhu kondensor naik, akan ada kenaikan daya pada kompresor, walaupun kenaikan tersebut mungkin hanya sedikit. Kapasitas refrigerasi selalu turun bila suhu kondensor naik. Karakteristik lain yang penting, tidak digambarkan dalam grafik, adalah *koefisien prestasi* (coefficient of performance) yang turun secara monoton bila suhu kondensor naik.




Gambar 11-10 Dampak refrigerasi dan kapasitas refrigerasi untuk kompresor ideal dengan refrigeran 22, volume sisa 4,5 persen, laju volume langkah 50 L/det, dan suhu evaporator $-20°C$.

Bertitik tolak dari daya dan efisiensi, diinginkan suhu kondensor yang rendah, jadi kondensor tersebut harus menggunakan udara atau air yang terdingin yang tersedia, mengalir secara maksimum dan ekonomis, serta permukaannya harus dijaga tetap bersih. Udara atau gas-gas yang tak dapat mengembun di dalam kondensor juga mengakibatkan tingginya tekanan kondensor tersebut.

WÄRTSILÄ VASA 22



## Wärtsilä Vasa 22

The Vasa 22 deri es from the pioneering Vasa 32. It too offers big engine qualities in a compact package with superb power/weight ratio backed up by fuel economy and long maintenance inte vals.

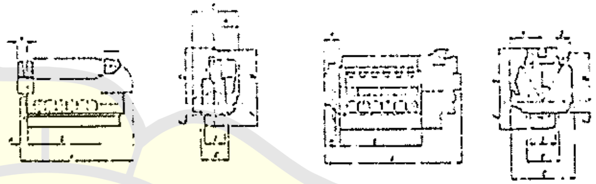
The broad outpu and speed range of the Vasa 22 make it an effective and powerful main engine for smaller vessels such as tugs and f shing boats. The engine also performs well as an auxiliary.

| MAIN DATA | |
|---|---|
| Cylinder bore | 220 mm |
| Piston stroke | 240/260 mm |
| Speed | 720–1200 rpm |
| Mean effective pressure | 22.8–19.1 bar |
| Piston speed | 6.2–9.6 m s |
| FUEL SPECIFICATION | |
| Fuel oil | 730 cSt/50 C / 7200 sR/100°F |
| ISO 8217, class F, RMF 55 | |

[illegible] Propulsion output [illegible]

| Output | Output in kW/BHP at | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 825 rpm | | 900 rpm | | 1000 rpm | | (1100 rpm) | | 1200 rpm* | |
| Engine type | kW | BHP | kW | BHP | kW | BHP | kW | BHP | kW | BHP |
| 4R22 | 600 | 820 | 650 | 890 | 710 | 970 | 750 | 1020 | – | – |
| 6R22 | 900 | 1220 | 975 | 1330 | 1065 | 1450 | 1125 | 1530 | – | – |
| (8R22) | 1200 | 1630 | 1300 | 1770 | 1420 | 1930 | (1500 | 2040) | – | – |
| 12V22 | - | - | 1740 | 2370 | 1930 | 2650 | 2030 | 2760 | 2100 | 2860 |
| 16V22 | - | - | 2320 | 3160 | 2600 | 3540 | 2710 | 3680 | 2800 | 3810 |

*When running on heavy fuel at 1200 rpm, max. viscosity 380 cSt/50°C



[illegible]

| Engine type | A | B | C | D | E | F | G |
|---|---|---|---|---|---|---|---|
| 4R22 | 2945 | 1665 | 1555 | 1300 | 355 | 810 | 1675 |
| 6R22 | 3625 | 1726 | 1555 | 1800 | 355 | 810 | 2355 |
| (8R22) | 4305 | 1790 | 1555 | 1800 | 355 | 810 | 3033 |
| 12V22 | 4099 | 1740 | 2156 | 1600 | 355 | 805 | 2655 |
| 16V22 | 4929 | 1850 | 2156 | 1600 | 400 | 755 | 3405 |

| Engine type | H | I | K | M | N | O | Weight |
|---|---|---|---|---|---|---|---|
| 4R22 | 242 | 670 | 900 | 815 | 370 | – | 7.7 |
| 6R22 | 242 | 670 | 900 | 815 | 370 | – | 9.5 |
| 8R22 | 242 | 670 | 900 | 815 | 370 | – | 11.5 |
| 12V22 | 142 | 834 | 1070 | 1065 | 490 | 565 | 15.0 |
| 16V22 | 142 | 760 | 1000 | 1065 | 490 | 565 | 21.2 |

[illegible]

| Output | Output at | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 720 rpm/ 60 Hz | | 750 rpm/ 50 Hz | | 900 rpm/ 60 Hz | | 1000 rpm/ 50 Hz | | 1200 rpm*/ 60 Hz | |
| Engine type | Eng. kW | Gen. kW | Eng. kW | Gen. kW | Eng. kW | Gen. kW | Eng. kW | Gen. kW | Eng. kW | Gen. kW |
| 4R22 | 540 | 510 | 560 | 530 | 650 | 620 | 710 | 670 | – | – |
| 6R22 | 810 | 770 | 840 | 800 | 975 | 930 | 1065 | 1010 | – | – |
| (8R22) | 1080 | 1030 | 1120 | 1060 | 1300 | 1230 | 1420 | 1350 | – | – |
| 12V22 | – | – | – | – | 1740 | 1650 | 1950 | 1850 | 2100 | 2000 |
| 16V22 | – | – | – | – | 2320 | 2200 | 2600 | 2470 | 2800 | 2360 |

*When running on heavy fuel at 1200 rpm, max. viscosity 380 cSt/50°C

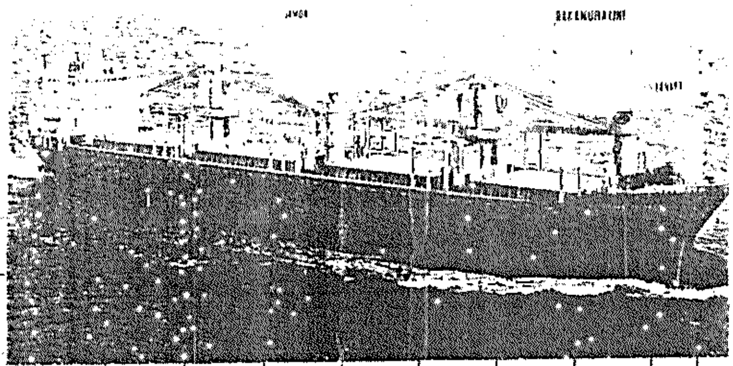
[illegible]

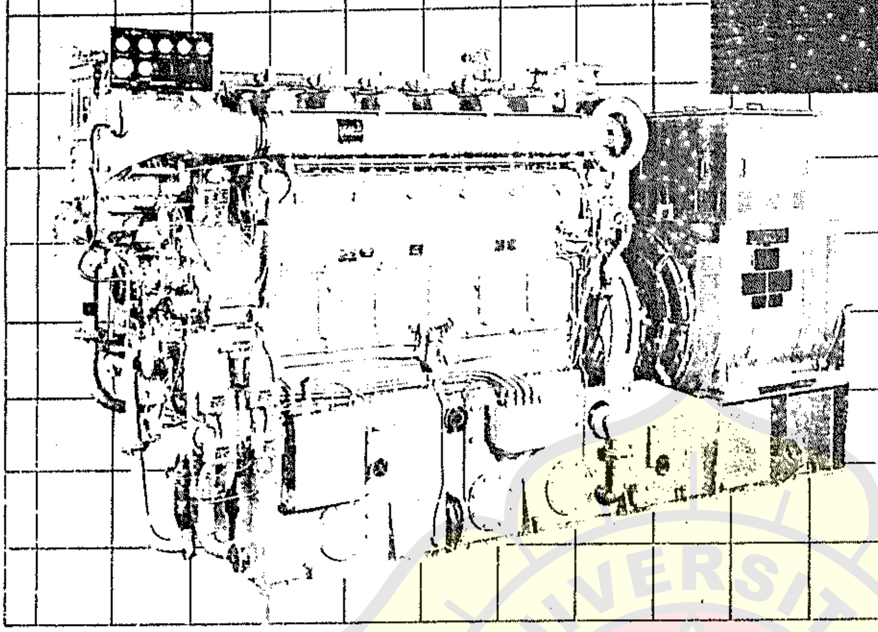
| Engine type | Length | Breadth | Height | Engine weight | Weight of gen. set |
|---|---|---|---|---|---|
| 4R22 | 4910 | 1565 | 2692 | 7.6 | 13.2 |
| 6R22 | 5700 | 1630 | 2720 | 9.3 | 17.. |
| (8R22) | 6030 | 1780 | 2790 | 11.3 | 20.9 |
| 12V22 | 6575 | 2065 | 2905 | 15.4 | 25.3 |
| 16V22 | 7750 | 2085 | 2905 | 20.3 | 31.1 |

# M200L

66

Engine output
441–662 kW (600–900 PS)





Depending on the specifications or options that have been chosen, your model may differ slightly from the one in the photograph

## Specifications

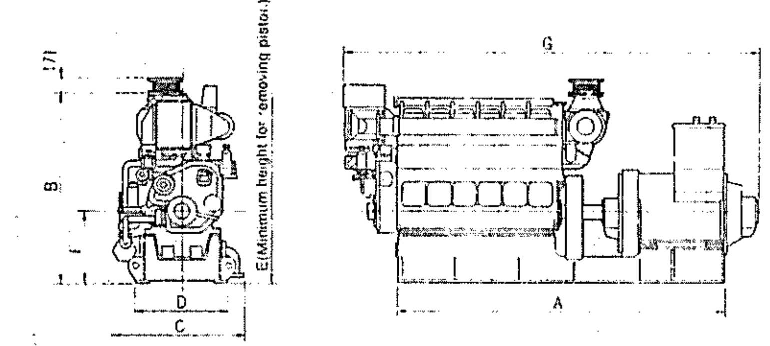
| Engine model | | M200L-UN (-UX) | | M200L-SN (-SX) | | M200L-EN (-EX) | | M200AL-UN | | M200AL-SN | | M200AL-EN | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Type | | Vertical water-cooled 4-cycle diesel engine | | | | | | | | | | | |
| No. of cylinders | | 6 | | | | | | | | | | | |
| Cylinder bore × stroke | mm | 200 × 260 | | | | | | | | | | | |
| Total displacement | ℓ | 49.01 | | | | | | | | | | | |
| Continuous rated output | kW (PS) | 441 (600) | | 485 (650) | | 552 (750) | | 530 (720) | | 610 (830) | | 662 (900) | |
| Engine speed | rpm | 720 | 750 | 720 | 750 | 720 | 750 | 900 | 1000 | 900 | 1000 | 900 | 1000 |
| Net mean effective pressure | MPa (kgf/cm²) | 1.500 (15.30) | 1.441 (14.69) | 1.650 (16.83) | 1.585 (16.16) | 1.876 (19.13) | 1.801 (18.36) | 1.441 (14.69) | 1.296 (13.22) | 1.661 (16.91) | 1.495 (15.24) | 1.801 (18.36) | 1.621 (16.53) |
| Generator capacity | kW | 400 | | 450 | | 500 | | 480 | | 560 | | 600 | |
| Combustion system | | Direct injection | | | | | | | | | | | |
| Starting system | | Compressed air | | | | | | | | | | | |
| External dimensions: Overall length | mm | 2919 | | 2923 | | 2977 | | 2919 | | 2977 | | 2977 | |
| External dimensions: Overall width | mm | 1120.5 | | | | | | | | | | | |
| External dimensions: Overall height | mm | 1844 | | 1880 | | 1883 | | 1844 | | 1883 | | 1883 | |
| Dry weight | kg | 5800 | | | | | | | | | | | |

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.



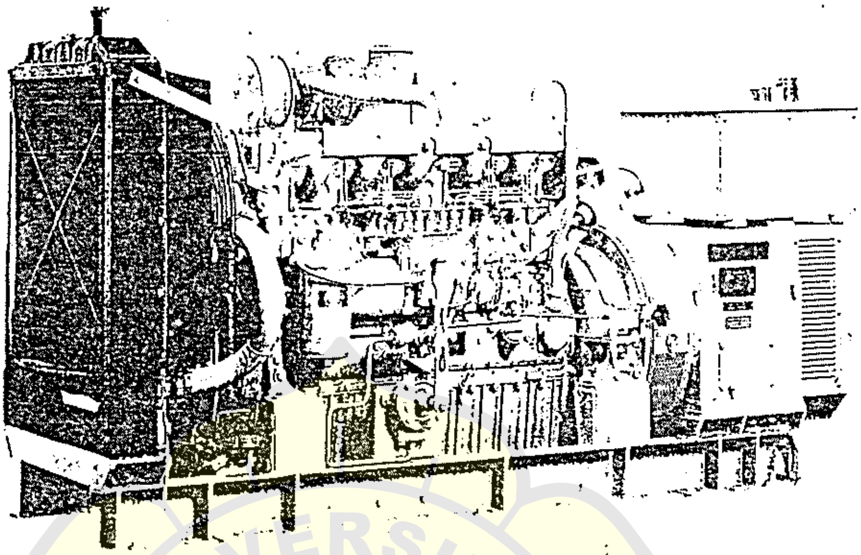

| Engine model | M200L-UN (-UX) | M200L-SN (-SX) M200AL-UN | M200L-EN (-EX) M200AL-SN M200AL-EN |
|---|---|---|---|
| A | 3650 | 3650 | 4000 |
| B | 2271 | 2271 | 2326 |
| C | 1520 | 1520 | 1520 |
| D | 1040 | 1040 | [illegible] |
| E | 2570 | 2570 | [illegible] |
| F | 950 | 950 | [illegible] |
| G | 4550 | [illegible] | [illegible] |
| [illegible] | 11000 | 11000 | [illegible] |

Note: Above data shows the case of [illegible]

Please confirm all specifications [illegible] on the separate [illegible]

# Emergency-Use Marine Power Generating Sets

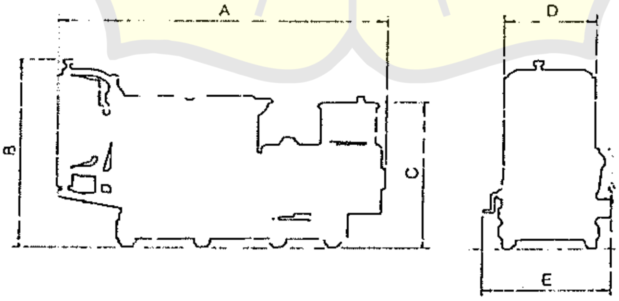
# YMGH Series



## pecifications

| Model | | | YMGH30 | YMGH40 | YMGH55 | YMGH80 | YMGH100 | YMGH120 | YMGH150 | YMGH200 | YMGH250 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| ator | Capacity | kVA | 30 | 40 | 55 | 80 | 100 | 120 | 150 | 200 | 25 |
| | Frequency | Hz | 60 | | | | | | | | |
| | Voltage | V | 450 | | | | | | | | |
| | Power factor | % | 80 | | | | | | | | |
| engine | Model | —— | 4TN100L-H | 4TN100L-HI | 4T112L-H | 4TN112L-TH | 6CHL-TH | 6CHL-HTH | 6HAL-H | 6HAL-TH | 6HAL-HTH |
| | Output | kW (PS) | 31.3 (42.5) | 40 (54) | 51 (70) | 72 (98) | 69 (121) | 107 (145) | 132 (180) | 177 (240) | 221 (300) |
| | Engine speed | rpm | 1800 | | | | | | | | |
| | Type | —— | Vertical 4-cycle diesel engine (radiator cooling) | | | | | | | | |
| | Combustion system | —— | Direct injection | | | | | | | | |
| | Starting system | —— | Remote (automatic) starting by starting motor | | | | | | | | |

Please confirm all specifications, etc. on the separate delivery specifications sheet




## GH-A Series Dimensions

(Unit: mm)

| el | A | B | C | D | E | Weight (kg) |
|---|---|---|---|---|---|---|
| 30A | 1640 | 955 | 330 | 645 | 815 | 735 |
| 40A | 1680 | 965 | 830 | 645 | 815 | 770 |
| 55A | 1920 | 1050 | 925 | 690 | 945 | 940 |
| 80A | 1990 | 1050 | 950 | 630 | 945 | 1120 |
| 00A | 2320 | 1310 | 1056 | 720 | 845 | 1530 |
| 20A | 2300 | 1310 | 1155 | 720 | 845 | 1640 |
| 50A | 2600 | 1460 | 12[illegible]0 | 720 | 1015 | 2[illegible]10 |
| 00A | 2670 | 1460 | 12[illegible]0 | 720 | 1015 | 2270 |
| 50A | 2725 | 1480 | 12[illegible]0 | 720 | 1015 | 2440 |

Generator manufacturer, Tokyo Denki Seizoh

## YMGH-B Series Dimensions

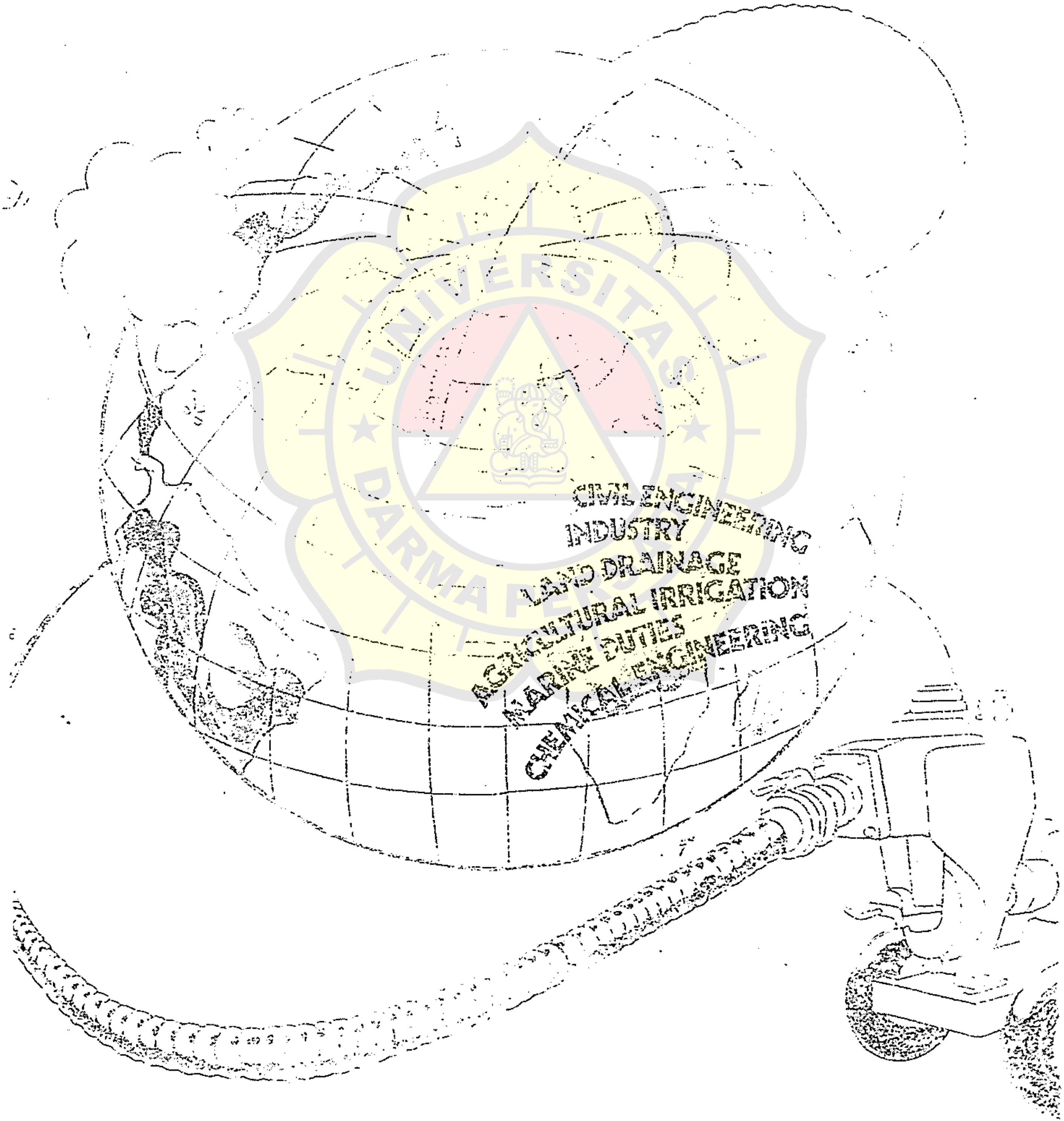
(Unit: mm)

| Model | A | B | C | D | E | Weight (kg) |
|---|---|---|---|---|---|---|
| YMGH 30B | 1590 | 965 | 905 | 645 | 815 | 765 |
| YMGH- 40B | 1630 | 965 | 905 | 645 | 815 | 780 |
| YMGH- 55B | 1800 | 1050 | 925 | 690 | 945 | 880 |
| YMGH- 80B | 1920 | 1050 | 925 | 690 | 945 | 1000 |
| YMGH-100B | 2290 | 1310 | 10[illegible]0 | 720 | 845 | 1490 |
| YMGH-120B | 2345 | 1310 | 1070 | 720 | 845 | 1440 |
| YMGH-150B | 2615 | 1460 | 1125 | 720 | 1015 | 1940 |
| YMGH-200B | 2705 | 1460 | 1205 | 720 | 1015 | 2190 |
| YMGH-250B | 2830 | 1480 | 1340 | 720 | 1015 | 2400 |

Generator manufacturer, Taiyo Denki

DESMI

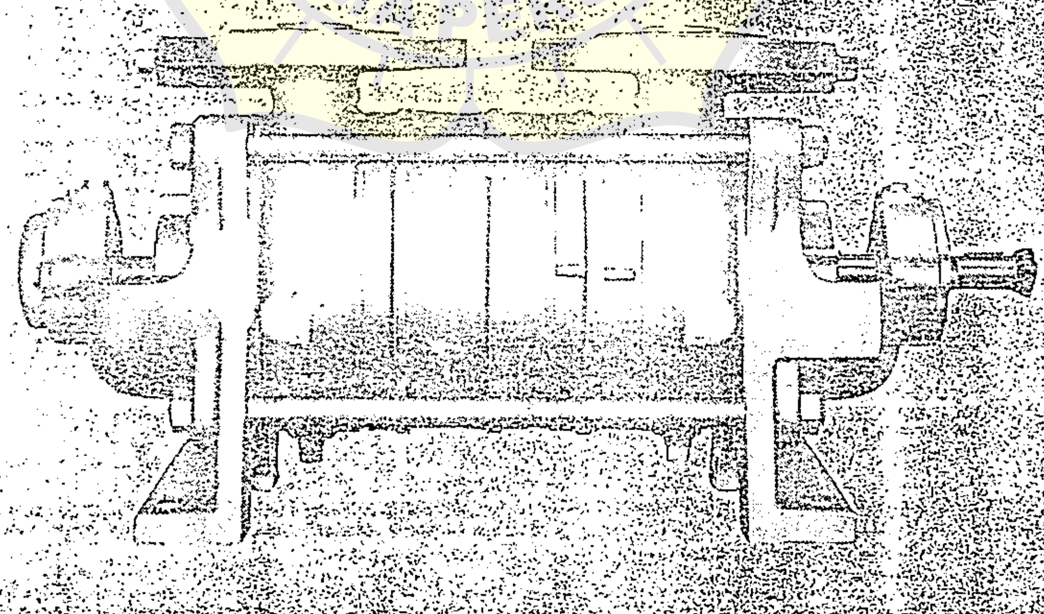
# the name in self-priming centrifugal pumps



SERO

# Self-Priming Centrifugal Pumps

– side channel pumps –
(patented)



## SON/SRN – SOB/SRB

(—Material DIN No. 1.4581)

B — oval flanges — SRN/SRB — round flanges

— 1450 RPM.

| Pump type | l/min. | 5 | 10 | 15 | 20 | 25 | 30 | 35 | 40 | 45 | 50 | 60 | 70 | 80 | 90 | 100 | 110 | 125 | 150 | 175 | 200 | 250 | 300 | 350 | 400 | 450 | 500 | 550 | 600 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | m³/h | 0,3 | 0,6 | 0,9 | 1,2 | 1,5 | 1,8 | 2,1 | 2,4 | 2,7 | 3,0 | 3,6 | 4,2 | 4,8 | 5,4 | 6,0 | 6,6 | 7,5 | 9,0 | 10,5 | 12,0 | 15,0 | 18,0 | 21,0 | 24,0 | 27,0 | 30,0 | 33,0 | 36,0 |
| | Connections | Total dynamic head (metres) recommended motor size (kW) | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| SON 101 | R 1" | 27 | 22 | 18 | 14 | 9 | 5 | | | | | | | | | | | | | | | | | | | | | | |
| SON 102 | R 1" | 50 | 42 | 33 | 25 | 17 | 9 | | | | | | | | | | | | | | | | | | | | | | |
| SON 103 SRN 103 | R 1" NW 25 | 73 | 61 | 49 | 37 | 25 | 13 | | | | | | | | | | | | | | | | | | | | | | |
| SON 111 | R 1" | 32 | 29 | 26 | 23 | 19 | 15 | 11 | 7 | | | | | | | | | | | | | | | | | | | | |
| SON 112 | R 1" | 60 | 54 | 48 | 42 | 35 | 28 | 20 | 13 | | | | | | | | | | | | | | | | | | | | |
| SON 113 SRN 113 | R 1" NW 25 | 88 | 79 | 70 | 61 | 51 | 41 | 30 | 19 | | | | | | | | | | | | | | | | | | | | |
| SRN 114 | NW 25 | 116 | 104 | 92 | 80 | 67 | 53 | 39 | 25 | | | | | | | | | | | | | | | | | | | | |
| SRN 115 | NW 25 | 144 | 129 | 114 | 100 | 83 | 66 | 49 | 31 | | | | | | | | | | | | | | | | | | | | |
| SRN 116 | NW 25 | 172 | 154 | 136 | 119 | 99 | 79 | 58 | 37 | | | | | | | | | | | | | | | | | | | | |
| SON 211 | R 1" | | | 38 | 36 | 33 | 30 | 26 | 22 | 17 | 13 | | | | | | | | | | | | | | | | | | |
| SON 212 | R 1" | | | 71 | 67 | 61 | 55 | 48 | 40 | 32 | 25 | | | | | | | | | | | | | | | | | | |
| ON 213 | R 1" | | | 104 | 98 | 89 | 80 | 70 | 58 | 47 | 37 | | | | | | | | | | | | | | | | | | |
| RN 214 | NW 32 | | | 137 | 129 | 117 | 105 | 91 | 76 | 62 | 49 | | | | | | | | | | | | | | | | | | |
| N 215 | NW 32 | | | 179 | 159 | 145 | 130 | 113 | 95 | 77 | 61 | | | | | | | | | | | | | | | | | | |
| N 216 | NW 32 | | | 203 | 190 | 173 | 155 | 135 | 113 | 92 | 73 | | | | | | | | | | | | | | | | | | |
| N 221 | R 1¼" | | | | 42 | 40 | 38 | 35 | 33 | 30 | 28 | 23 | 17 | 10 | | | | | | | | | | | | | | | |
| N 222 | R 1¼" | | | | 80 | 76 | 72 | 67 | 63 | 58 | 54 | 44 | 33 | 20 | | | | | | | | | | | | | | | |
| N 223 | R 1¼" | | | | 118 | 112 | 106 | 99 | 93 | 86 | 80 | 65 | 49 | 30 | | | | | | | | | | | | | | | |
| N 224 | NW 32 | | | | 156 | 148 | 140 | 131 | 123 | 114 | 106 | 86 | 65 | 40 | | | | | | | | | | | | | | | |
| N 225 | NW 32 | | | | 194 | 184 | 174 | 163 | 153 | 142 | 132 | 107 | 81 | 50 | | | | | | | | | | | | | | | |
| N 226 | NW 32 | | | | 232 | 220 | 208 | 195 | 183 | 170 | 158 | 128 | 97 | 60 | | | | | | | | | | | | | | | |
| N 331 | R 1½" | | | | | | | | | | 36 | 33 | 29 | 26 | 22 | 18 | 15 | 9 | | | | | | | | | | | |
| N 332 | R 1½" | | | | | | | | | | 68 | 62 | 54 | 48 | 41 | 34 | 28 | 17 | | | | | | | | | | | |
| ON 333 RN 333 | R 1½" NW 32 | | | | | | | | | | 100 | 91 | 80 | 70 | 60 | 50 | 41 | 25 | | | | | | | | | | | |
| RN 334 | NW 32 | | | | | | | | | | 132 | 120 | 106 | 95 | 79 | 66 | 54 | 33 | | | | | | | | | | | |
| RN 335 | NW 32 | | | | | | | | | | 164 | 140 | 132 | 115 | 98 | 82 | 66 | 41 | | | | | | | | | | | |
| RN 336 | NW 32 | | | | | | | | | | 196 | 176 | 157 | 138 | 118 | 98 | 79 | 49 | | | | | | | | | | | |
| N 441 | NW 40 | | | | | | | | | | | | | | | 37 | 34 | 30 | 23 | 16 | 9 | | | | | | | | |
| N 442 | NW 40 | | | | | | | | | | | | | | | 70 | 65 | 57 | [illegible] | 30 | 17 | | | | | | | | |
| N 443 | NW 40 | | | | | | | | | | | | | | | 103 | 96 | 94 | 64 | 44 | 25 | | | | | | | | |
| N 444 | NW 40 | | | | | | | | | | | | | | | 136 | 127 | 111 | 84 | 58 | 33 | | | | | | | | |
| N 445 | NW 40 | | | | | | | | | | | | | | | 170 | 157 | 136 | 105 | 72 | 41 | | | | | | | | |
| N 446 | NW 40 | | | | | | | | | | | | | | | 203 | 188 | 165 | 125 | 86 | 49 | | | | | | | | |
| N 551 | NW 50 | | | | | | | | | | | | | | | | | | | 39 | 33 | 25 | 18 | 10 | | | | | |
| N 552 | NW 50 | | | | | | | | | | | | | | | | | | | 71 | 62 | 46 | 32 | 18 | | | | | |
| N 553 | NW 50 | | | | | | | | | | | | | | | | | | | 104 | 91 | 67 | 47 | 26 | | | | | |
| N 554 | NW 50 | | | | | | | | | | | | | | | | | | | 137 | 120 | 88 | 61 | 34 | | | | | |
| N 661 | NW 65 | | | | | | | | | | | | | | | | | | | | | | 34 | 31 | 26 | 21 | 16 | 11 | 6 |
| N 662 | NW 65 | | | | | | | | | | | | | | | | | | | | | | 64 | 58 | 49 | 39 | 30 | 20 | 11 |
| N 663 | NW 65 | | | | | | | | | | | | | | | | | | | | | | 94 | 86 | 72 | 58 | 44 | 30 | 16 |

Total dynamic head = suction lift + discharge head + friction losses. Power requirement (kW) based on liquid density of 1 kg/dm³.
Tolerance: capacity and dynamic head ± 5% each. Power absorbed: see individual performance curves also note information on page 2.

Max. Casing Pressure:
SON, SOB 100 ... 330 : 16 bar
SRN, SRB 100 ... 550 : 25 bar
SRN 660 : 16 bar