

## BAB IV

### PENUTUP

#### IV.1. Kesimpulan

Dari perencanaan yang telah dilakukan terhadap kapal Semi Container 3650 DWT dengan ukuran kapal :

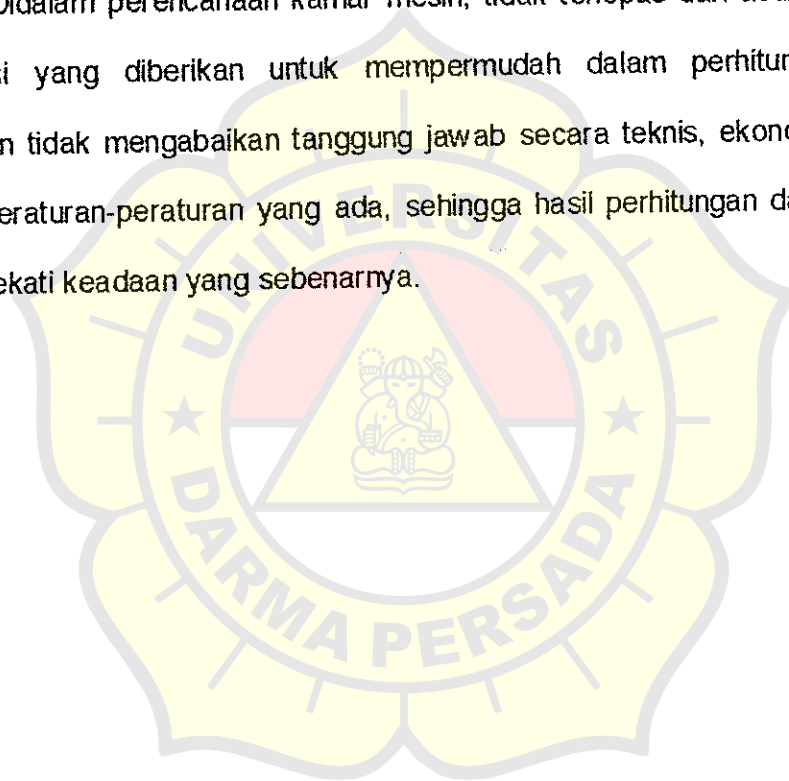
- Panjang antara garis tegak (  $L_{pp}$  ) : 92,15 M
- Lebar kapal (  $B$  ) : 16,50 M
- Tinggi kapal (  $H$  ) : 7,80 M
- Sarat air (  $T$  ) : 5,40 M
- Koefisien blok (  $C_b$  ) : 0,647
- Kecepatan (  $V_s$  ) : 12 Knot
- Klasifikasi : Biro Klasifikasi Indonesia
- Jarak Pelayaran : 2000 Mil

Maka dapat diambil kesimpulan, sebagai berikut :

- Besarnya Daya Continuous Rating (MCR) yang diperlukan agar kapal dapat mencapai kecepatan 12 knot adalah 1840 kW.
- Motor penggerak utama dipilih mesin diesel empat langkah dengan Merk : Wartsila, Model Nohab 25 MC (8V25), Daya 1840 kW dan putaran 900 rpm dan mesin dipasang di belakang kapal.

- Permesinan geladak dihitung berdasarkan referensi dari Biro Klasifikasi Indonesia (BKI), Marine Auxilliary Machinery System, Khetagurov dan Marine Engineering Society in Japan, Machinery Out Fitting Design Manual Vol. 1.

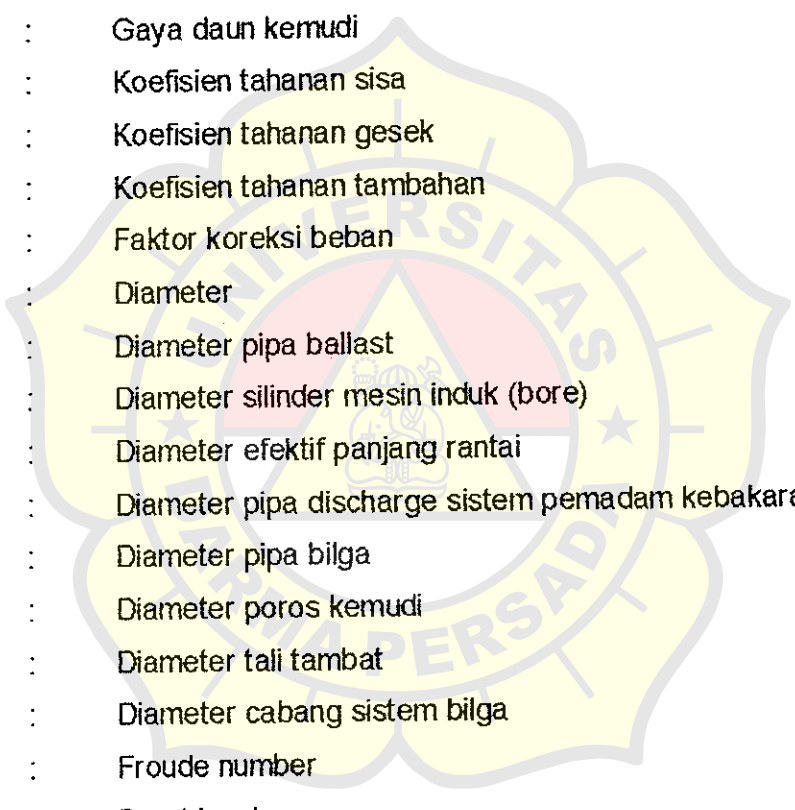
Didalam perencanaan kamar mesin, tidak terlepas dari asumsi-asumsi yang diberikan untuk mempermudah dalam perhitungan dengan tidak mengabaikan tanggung jawab secara teknis, ekonomis dan peraturan-peraturan yang ada, sehingga hasil perhitungan dapat mendekati keadaan yang sebenarnya.



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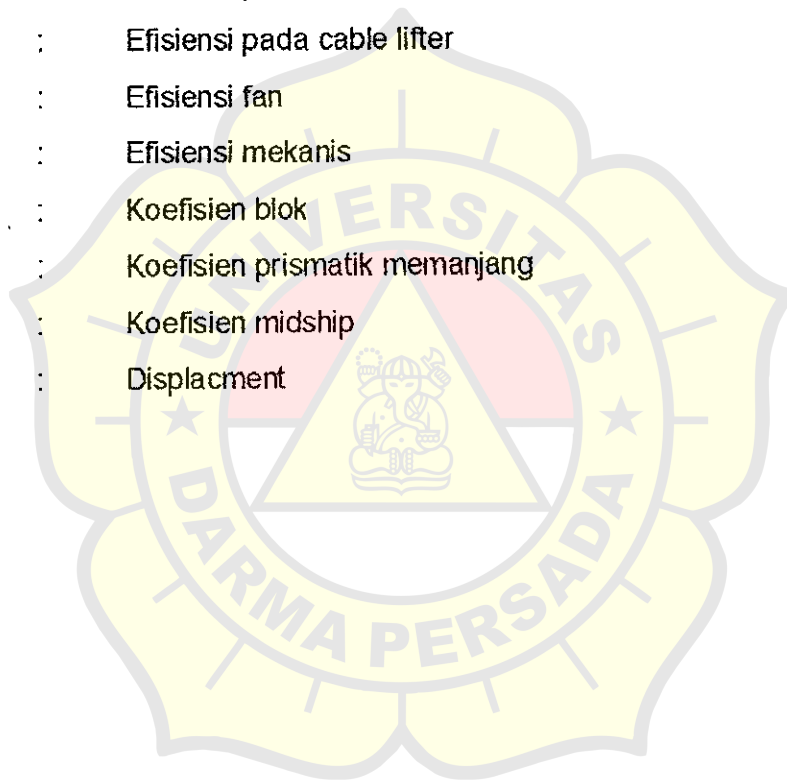


▪ A	:	Luas bidang
▪ B	:	Lebar kapal
▪ b	:	Tinggi daun kemudi
▪ BHP	:	Brake horse power
▪ Cr	:	Gaya daun kemudi
▪ CR	:	Koefisien tahanan sisa
▪ CF	:	Koefisien tahanan gesek
▪ CA	:	Koefisien tahanan tambahan
▪ Cw	:	Faktor koreksi beban
▪ d	:	Diameter
▪ db	:	Diameter pipa ballast
▪ D	:	Diameter silinder 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
- $I_a$  : Ratio antara putaran motor dengan putaran cable lifter
- $I_{cl}$  : Panjang rantai untuk satu putaran cable lifter
- $I_w$  : Ratio antara putaran motor dengan putaran cable lifter
- J : Kapasitas botol angin / udara start
- K : Koefisien hambatan untuk katup dan lifting
- L : Letak midship section, panjang pipa
- $L_a$  : Panjang rantai yang menggantung
- $I_b$  : Lebar ruangan
- LCB : Letak titik tekan keatas terhadap midship section
- $L_{pp}$  : Panjang kapal antara dua garis tegak
- $L_{wl}$  : Panjang garis air
- $M_{cl}$  : Torsi pada cable lifter
- $M_m$  : Torsi poros motor
- N : Putaran mesin, putaran propeller
- $N_{cl}$  : Putaran cable lifter
- $N_e$  : Daya efektif pompa
- $N_{th}$  : Daya kompresor
- $N_w$  : Putaran poros penggulung tali tambat
- $P_a$  : Tekanan kerja maksimum botol udara start
- $P_b$  : Tekanan kerja minimum botol udara start
- PC : Koefisien propulsi
- $P_e$  : Tekanan udara luar
- Q : Kapasitas aliran fluida, kapasitas kompresor udara
- $Q_c$  : Kapasitas fan
- $r_z$  : 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
- Vdo : Volume bahan bakar motor bantu
- Vfo : Volume bahan bakar motor induk
- Vfw : Volume kebutuhan air tawar
- Vmost : 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
- Wfww : Berat air untuk cuci dan mandi

- $W_{fwc}$  : Berat air untuk pendingin motor
- $Z_c$  : Jumlah ABK
- $\lambda$  : Koefisien gesek pada saluran
- $\nu$  : Viskositas kinematis fluida
- $\lambda$  : Berat jenis fluida
- $\alpha$  : Sudut putar kemudi
- $\eta_{cl}$  : Efisiensi peralatan mesin jangkar
- $\eta_h$  : Efisiensi pada cable lifter
- $\eta_m$  : Efisiensi fan
- $\eta$  : Efisiensi mekanis
- $\delta$  : Koefisien blok
- $\varphi$  : Koefisien prismatic memanjang
- $\beta$  : Koefisien midship
- $\Delta$  : Displacement



**LEMBAR PERBAIKAN**

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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

Mengetahui  
Dekan FTK

Jakarta,.....  
Ketua Dosen Penguji

( ) ( )



The suction lift, or simply lift, is the loss of head required to overcome resistance in the suction line of the pumping plant; it is measured in mH<sub>2</sub>O.

The useful power of a pump is the energy increment in the flow of liquid passing through the pump in unit time and is expressed in horsepower or kilowatts. Thus,

$$N_u = \frac{QH}{60 \times 60 \times 75} \text{ hp} = \frac{QH}{60 \times 60 \times 102} \text{ kW}$$

$$N_u = \frac{Q \gamma H}{60 \times 60 \times 75} \text{ hp} = \frac{Q \gamma H}{60 \times 60 \times 102} \text{ kW}$$

where  $H$  = the actual head created by the pump, mH<sub>2</sub>O.

The mechanical efficiency,  $\eta_m$ , of a pump determines the loss in energy in its operation and enables the required power input  $N$  to be calculated:

$$N = \frac{N_u}{\eta_m} \text{ hp (kW)}$$

### 1-3. Pump Classification According to Purpose and Principle of Operation

#### (A) PUMP CLASSIFICATION ACCORDING TO PURPOSE

In accordance with their purpose, shipboard pumps can be divided into three groups:

1. General service pumps whose function is to ensure the seaworthiness of the ship and to provide for the domestic needs of the crew and passengers, and also to maintain the necessary sanitary conditions on board.
    2. Pumps of the shipboard systems, designed to serve the main and auxiliary systems, and to facilitate the maintenance of normal conditions for their operation.
    3. Special-purpose pumps in tankers, trawlers, ice-breakers, life-saving ships and dredgers.
- General service pumps include:
- (1) bilge pumps,
  - (2) sanitary pumps,
  - (3) fire pumps,
  - (4) emergency pumps.

#### Bilge Pumps

Bilge pumps include ballast and drainage pumps.

Ballast pumps. The purpose of these pumps is to take in liquid ballast from overboard, to fill the ballast tanks and peaks, to empty

them of ballast and to transfer the ballast from certain tanks and peaks to others.

The capacity of ballast pumps must be sufficient to enable the tanks they serve to be emptied within 4 to 10 hours, depending upon the size of the ship.

Tanks that hold from 20 to 265 cu m of ballast are usually pumped dry within 1 to 3.5 hours; those holding from 360 to 1300 cu m are emptied within 4 to 5.5 hours.

The required capacity of a ballast pump can be determined from the formula

$$Q_b = 0.2825 d_i v_b \text{ cu m per hr} \quad (11)$$

where  $d_i$  = diameter, in cm, of the inlet pipe of the largest ballast tank. According to the regulations of the U.S.S.R. Shipping Register this value is to be taken from Table 3, depending upon the tank capacity in tons.

$v_b = 2$  to  $2.5$  = velocity of water flow in the input line of the pump, m per sec.

Table 3

Tank capacity, tons	Inside diameter of pipe and fittings, mm	Tank capacity, tons	Inside diameter of pipe and fittings, mm
Up to 20	60	265 to 360	125
20 to 40	73	360 to 480	140
40 to 75	89	480 to 620	150
75 to 120	90	620 to 800	160
120 to 190	100	800 to 1000	175
190 to 265	113	1000 to 1300	200

At a water velocity  $v_b = 2$  m per sec (recommended by the U.S.S.R. Shipping Register) the required ballast pump capacity will be

$$Q_b = 0.565 d_i^2 \text{ cu m per hr.} \quad (12)$$

Because of water leakage, this calculated capacity must be increased by 5 or 10 per cent. Ballast pump capacities range from 60 to 300 cu m per hour. The number of ballast pumps is not stipulated by the regulations of the U.S.S.R. Shipping Register.

Any pump of suitable capacity in a shipboard installation, except drinking-water pumps, can be employed for ballasting operations if the ballast tanks are not used to store liquid fuel. In the latter case, the use of standby cooling pumps of the internal combustion engines and the fire pumps for ballasting duty is prohibited.

Self-contained ballast pumps must be installed on oil tankers to serve the fore ballast tank.

- (1) fans of service and living compartments, designed to provide induced ventilation in these spaces;
  - (2) cargo hold fans, designed for ventilating the holds of dry-store bulk carriers, tankers and refrigerated cargo vessels, as well as refrigerated provision chambers;
  - (3) boiler plant fans, designed to produce artificial draught for the steam boilers;
  - (4) coal bunker fans.
- Depending upon the way they are installed fans are classified as:
- (1) supply fans in which the fan discharge is connected with the spaces being served;
  - (2) exhaust fans in which the fan inlet is connected to the spaces being served;
  - (3) ceiling fans, designed to produce air movement in the spaces without providing exchange.

As regards the pressure they develop, fans are divided into:

- (1) low-pressure fans developing a head up to 100 mmH<sub>2</sub>O;
  - (2) medium-pressure fans developing a head up to 300 mmH<sub>2</sub>O;
  - (3) high-pressure fans developing a head up to 1,500 mmH<sub>2</sub>O.
- According to the mechanical composition of the gas they handle, there are:

- (1) fans for delivering pure gases;
- (2) dust fans designed for delivering gases polluted by mechanical impurities.

The specific velocity,  $n_s$ , of a fan is a value relating the air discharge,  $Q$  cu m per hour, full head,  $H$  mmH<sub>2</sub>O, at normal atmospheric conditions and the fan wheel speed,  $n$  rpm, at the highest efficiency:

$$n_s = \frac{n \sqrt{Q}}{\sqrt{H^3}}$$

Gas passing through the fan is compressed to only the slightest degree and is therefore assumed to be an incompressible fluid. In this case all the main principles in the theory and operation of centrifugal pumps are valid for fans as well.

The capacity of a fan for ventilating definite service quarters must be sufficient to maintain the chemical composition, humidity and temperature of the air within the requirements of sanitary regulations.

The unwholesomeness, or contamination, of the air in a room or compartment due to the presence of people is usually estimated by the carbon dioxide content, which increases with an increase of harmful impurities in the air. The carbon dioxide content of the air must not exceed 0.1 to 0.15 per cent by volume.

The fan capacity, required to maintain a stipulated chemical composition of the air in a compartment is

$$Q_{ch} = V_r \frac{V_{rc}}{V_{mr} - V_{ca}} \quad \text{cu m per hour} \quad (273)$$

where  $V_{rc}$  = volume of carbon dioxide produced per cu m of the given room, litres per cu m

$V_{mr}$  = volume of the room, cu m

$V_{ca} \approx 1$  = the maximum carbon dioxide content per cu m of the given room, litres per cu m

$V_{ca} \approx 0.3$  = carbon dioxide content per cu m of sea air entering the room, litres per cu m.

The volume of air required to maintain the prescribed temperature in a room is

$$Q_t = \frac{Q_r}{c_a(t_r - t_{ra}) \gamma_{ra}} = \frac{Q_r}{c_a(t_r - t_{ra}) \frac{\gamma_0}{1 + \alpha t_r}} = \frac{Q_r(1 + \alpha t_r)}{c_a(t_r - t_{ra}) \gamma_0} \quad (274)$$

where  $c_a \approx 0.24$  = mean heat capacity of air, kcal per kg °C

$t_r$  = given temperature of the room, °C

$t_{ra}$  = temperature of the fresh air entering the room, °C

$Q_r$  = amount of heat entering the room, kcal per hour

$\gamma_{ra}$  = density of the fresh air entering the room, kg per cu m

$\gamma_0 \approx 1.29$  = density of dry air at 0°C and a pressure of 760 mmHg, kg per cu m

$\alpha = \frac{1}{273}$  = coefficient of volumetric expansion of air.

The amount of external air required to maintain the relative humidity in a room is

$$Q_{hu} = \frac{100 D_{hu}}{\psi_r d_r - \psi_{ra} d_{ra}} \quad \text{cu m per hour} \quad (275)$$

where  $D_{hu}$  = amount of moisture entering the room, g per hour

$d_r$  and  $d_{ra}$  = absolute humidity of saturated air at the room temperature,  $t_r$ , and at the temperature,  $t_{ra}$ , of the

entering air, g per cu m (see Table 38)

$\psi_r$  and  $\psi_{ra}$  = relative humidity of the air in the room and of the entering air, per cent.

Data on the relative humidity and temperature of the outside air depending upon the locality in which the ship is operating, and the permissible values for various accommodations are listed in Table 39.

The amount of carbon dioxide, heat and vapour produced by persons in a room can be calculated from the data of Table 40.

Table 42

Compartment	Number of air renewals per hour for	
	Pleenum ventilation	Exhaust ventilation
Passengers', officers' and crew accommodations . . . . .	10 to 15	—
Public rooms (staterooms, dining-saloons, etc.) . . . . .	15 to 20	10 to 15
Smoking rooms . . . . .	—	15 to 20
Gymnasiums . . . . .	15	20
Swimming pools . . . . .	—	20
Russian baths . . . . .	—	10 to 20
Galleys . . . . .	5 to 10	40 to 60
Provision rooms without cooling facilities . . . . .	5 to 10	10 to 15
Bathrooms, toilets and laundries . . . . .	5	15 to 20
Sick bays . . . . .	5 to 10	10 to 20
Baggage rooms . . . . .	—	20
Deck refreshment bars . . . . .	10 to 15	25 to 30
Upper deck passageways . . . . .	—	6
Middle deck passageways . . . . .	—	7
Lower deck passageways . . . . .	—	8
Engine and boiler rooms . . . . .	30	35

$P_{st}$  = 760 mmHg, relative humidity of  $\phi_{st}$  = 50 per cent and density  $\gamma_{st}$  = 1.2 kg per cu m. The capacity of the fan determined for air in a given state, having a pressure  $P_a$ , volume  $Q_a$  and temperature  $t_a$ , can be converted to the standard air capacity by using formula (276) which is derived from the equation

$$\frac{P_{st} Q_{st}}{1 + \alpha t_{st}} = \frac{P_a Q_a}{1 + \alpha t_a}$$

whence

$$Q_{st} = \frac{(1 + \alpha t_{st}) P_a Q_a}{P_{st} (1 + \alpha t_a)} = Q_a \frac{\left( \frac{1 + \frac{1}{273} 20}{1 + \frac{1}{273} t_a} \right) \frac{P_a}{760}}{\frac{293}{760} + t_a \frac{P_a}{760}} \text{ cu m per hour} \quad (276)$$

The theoretical head developed by the fan is expressed in mm of water column:

$$H_{l\infty} = \frac{1}{g} (C_{2u} u_2 - C_{1u} u_1) = \frac{1,000}{g} \frac{\gamma_{air}}{\gamma_{wat}} (C_{2u} u_2 - C_{1u} u_1) = \rho (C_{2u} u_2 - C_{1u} u_1) \text{ mmH}_2\text{O} \quad (277)$$

where  $\gamma_{air}$  = density of air, kg per cu m

$\gamma_{wat}$  = 1,000 = density of water, kg per cu m

$\rho$  = mass density of air, kg-sec<sup>2</sup> per m<sup>4</sup>

Upon radial entry of the air onto the fan impeller vanes

$$H_{l\infty} = \rho C_{2u} u_2 \text{ or mmH}_2\text{O}$$

Taking into account the effect of having a finite number of impeller vanes on the developed head by the factor  $\sigma$  and for the losses of head in the fan by the hydraulic efficiency  $\eta_h$ , we obtain the actual head

$$H = H_{l\infty} \sigma \eta_h = \sigma \rho C_{2u} u_2^2 \eta_h = \sigma \rho \frac{C_{2u}^2}{u_2} u_2 u_2^2 \eta_h = \sigma \rho \phi u_2^2 \eta_h = \rho \psi u_2^2 \text{ mmH}_2\text{O} \quad (278)$$

where  $\phi = \frac{C_{2u}}{u_2}$  = eddy current factor

$\psi = \sigma \phi \eta_h$  = head factor taken equal to: 0.8 to 1.1 for forward-curved vanes; 0.6 to 0.8 for radial, or straight, vanes; 0.5 to 0.7 for backward-curved vanes.

The fan head required to accommodate a given ventilating system depends upon the resistance and characteristic curves of the latter.

The permissible maximum peripheral speeds (tip speeds) of an impeller, based upon fan design and strength considerations, are listed in Table 43. The table also lists the most widely used inlet and outlet angles of the vanes.

Table 43

Type of fan	Peripheral speed, m per sec	Inlet angle	Outlet angle
Low-pressure . . . . .	30 to 40	95 to 105	15 to 25
Medium-pressure . . . . .	40 to 50	125 to 130	30 to 35
High-pressure . . . . .	50 to 90	140 to 145	40 to 45

Backward curved vanes are rarely employed and then only for low-pressure fans. The number of vanes is usually assigned so as to facilitate laying out and may be equal to 4, 6, 8, 12, 16, 24, 32 or 48.

The power required to drive a fan is found from the formula

$$N_m = \frac{Q_a H}{75 \eta_f 3,600} \text{ hp}$$

The overall efficiency of a fan is made up of the following efficiencies:

1. Hydraulic efficiency, which takes into consideration the loss of head in the fan

$$\eta_h = \frac{H}{H + \Delta H} = \frac{H}{H_f} = 0.7 \text{ to } 0.85$$

where  $\Delta H$  = loss of head in the fan.

2. Hydraulic friction efficiency which takes into account the losses due to the friction of the impeller shrouds against the fluid being transferred:

$$\eta_{fr} = \frac{N_{fr}}{N_a} = \frac{\beta 10^{-6} \rho D_2^3 u_2^3}{N_a}$$

where  $N_{fr}$  = power lost in overcoming fluid friction

$\beta$  = (5 to 15)  $(1 + 5 \frac{b_2}{D_2})$  = coefficient obtained from data compiled by the Central Institute of Aero- and Hydrodynamics

$b_2$  = width of the impeller at air outlet

$D_2$  = impeller diameter at air outlet

For backward-curved vanes— $\eta_{fr} \approx 0.6$  to 0.75

For forward-curved vanes— $\eta_{fr} \approx 0.75$  to 0.9.

3. Mechanical efficiency which takes into account the losses due to mechanical friction

$$\eta_m = \frac{N_a - \Delta N_{mf}}{N_a} \approx 0.95 \text{ to } 0.99$$

where  $\Delta N_{mf}$  = power lost in overcoming mechanical friction. The overall efficiency of a fan is thus

$$\eta_f = \eta_h \eta_{fr} \eta_m = 0.4 \text{ to } 0.75 \quad (279)$$

The overall efficiency of an axial fan may reach  $\eta_f \approx 0.84$ .

## 2-2. Design and Selection of Fans

Strictly aerodynamical calculations in fan design do not, as a rule, ensure results in subsequent tests that comply with the initial design data.

More accurate results may be achieved by designing a fan similar to one which has already been built, tested and modified to obtain the most favourable aerodynamic and design features.

This type of fan design is carried out by the similarity method using aerodynamic diagrams and dimensionless characteristics which we will consider in the following.

The initial data for fan design comprise: the total head,  $H$ , consisting of the static,  $H_{st}$ , and dynamic,  $H_{dyn}$ , heads, capacity,  $Q_s$ , and the rotational speed,  $n$ , at maximum efficiency. Thus

$$H = H_{st} + H_{dyn} = H_{st} + \frac{v^2}{2g} \times 10^{-3} \text{ mmH}_2\text{O} \quad (280)$$

where  $v$  = mean velocity in the discharge connection of the fan. On the basis of the discharge per second,  $Q_s$ , head,  $H$ , and speed,  $n$ , we next determine the specific velocity of the fan.

The specific velocity of a fan is a value that relates the air discharge,  $Q_{st}$ , cu m per sec, the total head,  $H$  mmH<sub>2</sub>O, and the impeller speed,  $n$ , at maximum efficiency:

$$n_s = \frac{n \sqrt{Q_s}}{\sqrt{H^3}} \quad (281)$$

It is evident that the ratio of the capacities of a series of geometrically similar fans of identical design can be expressed by the dimensionless discharge coefficient  $\bar{Q}_k$ . Therefore

$$\bar{Q}_k = \frac{Q_s}{F u_2}$$

and

$$Q_s = \bar{Q}_k F u_2 = \bar{Q}_k \frac{\pi D_2^2}{4} u_2 \text{ cu m per sec}$$

where  $F$  = area of the impeller, sq m

$D_2$  = outside diameter of the impeller, m.

The peripheral speed at the outlet circumference of the impeller is found from the formula

$$u_2 = \frac{\pi D_2 n}{60} \text{ m per sec}$$

The pressure developed by a series of geometrically similar fans can be characterized by the pressure coefficient,  $\bar{H}_k$ :

$$\bar{H}_k = \frac{H}{u_2^2 \rho} \text{--- for the total head, and}$$

$$\bar{H}_{kst} = \frac{H_{kst}}{u_2^2 \rho} \text{--- for the static head.}$$

Whence, if we know  $\bar{H}_k$  from the characteristics of pilot models, we can determine

$$\begin{aligned} H &= \bar{H}_k u_2^2 \rho \text{ mmH}_2\text{O} \\ H_{kst} &= \bar{H}_{kst} u_2^2 \rho \text{ mmH}_2\text{O} \end{aligned} \quad (282)$$

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  $i_{sr}$ , 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, sec	Speed of rudder movement, deg/sec, for rudder angle of	
		20° = 70°	20° = 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

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

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

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:

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

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

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

It follows from formula (314) that

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

Combining equations (315) and (316) we obtain

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

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

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

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

$$N_{rs} = \frac{M_{rs} \omega_{rs}}{75} = \frac{M_{rs} \cdot 2\alpha^\circ}{75} \frac{\pi}{180^\circ} = 4.65 \frac{M_{rs} \alpha^\circ}{10^3 \tau} \quad \text{metric hp} \quad (319)$$

$$N_{rs} = \frac{M_{rs} \omega_{rs}}{75} = \frac{M_{rs}}{75} \frac{1000 n_{rs}}{30} = 1.395 \frac{M_{rs} n_{rs}}{10^3} \approx 1.4 \frac{M_{rs} n_{rs}}{10^3} \quad \text{metric hp} \quad (320)$$

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

$$N_m = \frac{N_{rs}}{\eta_{sg}} = 4.65 \frac{M_{rs} \alpha^\circ}{10^3 \eta_{sg} \tau} \quad \text{metric hp} \quad (321)$$

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

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

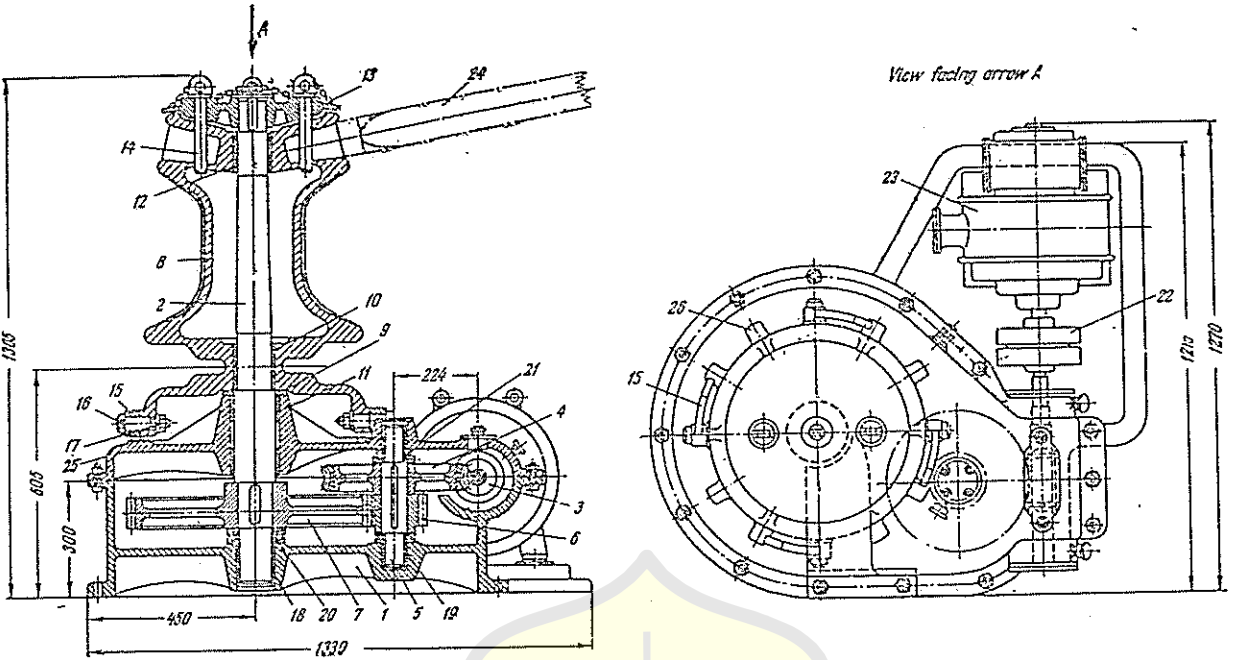


Fig. 169.

### 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

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

In hoisting one anchor

$$T_{cl} = 1.175(G_a + P_a L_a) \text{ kg}$$

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

- (a)  $P_{ao} = 0.023d_c^2$  kg for open-link chain
- (b)  $P_{as} = 0.0218d_c^2$  kg for stud-link chain

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

$$T_{cl} = 50G_a + 1.175(G_a + P_a L_a) \text{ kg}$$

If a windlass serves both for handling the anchor and for warping, the pull of the warp ends must not exceed

$$T_w = \frac{R_{br}}{6} \quad (385)$$

where  $R_{br}$  = breaking strength of the warping hawser.

The speed at which a capstan barrel heaves in a warping hawser can be taken from Table 58 which has been compiled from the manufacturing specifications for capstans worked out by the Central Marine Research Institute of the U.S.S.R.

Table 58

Pull of the capstan barrel, kg	Hawser heaving-in speed, m per sec	Useful power, kg-m/sec
1,200	0.3	360
3,000	0.25	750
4,500	0.2	900
7,000	0.167	1,165
12,000	0.150	1,800

The speed at which a warping hawser is heaved in by a windlass is not limited by the values in Table 58, and usually is equal to about 0.4 m per sec.

It has been stated previously that the same machinery is employed both for anchoring and warping purposes. It follows that windlasses and capstans must be designed so as to ensure normal operation of both the anchoring and warping arrangements.

As a rule, anchoring and warping capstans and windlasses are designed to ensure the proper operation of the anchoring arrangement, and then a check is made to see whether they provide for the required pull and heaving-in speed of the warping hawsers.

The number of anchors, their weight, the size of the anchor chain cables, the circumference of warping hawsers and towing ropes, and their length are determined from the tables of the pertinent regulations of the Shipping Register. To find these values it is necessary to calculate the rigging characteristic of the anchoring and warping arrangement:

$$X = L(B + H) + \Sigma \chi_i \quad (386)$$

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  $h_{sp}$ :

$$\chi_{so} = k_{isp} \Sigma l_{sp} h_{sp}$$

where  $k_{isp}$  = 0.75 if the total length of the superstructures is equal to or less than 0.5  $L$

$k_{isp}$  = 1.5  $\frac{l_{sp}}{L}$  if the total length of the superstructures exceeds 0.5  $L$

(b) correction factors for the deck houses, each having a height  $h_{dh}$  and length  $l_{dh}$ :

$$\chi_{dh} = k_{adh} \Sigma l_{dh} h_{dh}$$

where  $k_{adh}$  = 0.5 if the deck house has a length  $l_{dh}$  equal to or less than 0.5  $L$

$k_{adh}$  =  $\frac{l_{dh}}{L}$  if the length,  $l_{dh}$ , of the deck house exceeds 0.5  $L$ .

If the breadth,  $b_{adh}$ , of the deck house exceeds its length,  $l_{dh}$ , then the product  $b_{adh} h_{adh}$  is substituted into the equation in place of  $l_{dh} h_{dh}$ . Thus

$$\chi_{adh} = k_{adh} \Sigma b_{adh} h_{adh}$$

(c) correction factor for the quarter deck having a length  $l_q$  and height  $h_q$ :

$$\chi_q = l_q h_q$$

Data on the anchoring and warping arrangements are listed in Tables 59 and 60. The weight of each anchor is found by dividing their total weight by the number of anchors. The separate anchors may be lighter than the specified values by 7.5 per cent. The lengths of the anchor chain cables are given in the table on the assumption that the average length of each shot is 25 m. The cable length does not include the lengths of the chain slip, joining shackles, connecting links and short pieces of shots with swivels. If the tabular cable length comprises an odd number of shots, then the length of the starboard anchor chain cable is taken one shot longer than the port cable.

A section, taken through the central plane of the usual five-sung cable lifter (Fig. 170) perpendicular to the shaft will be a regular

Table 59  
Self-Propelled Transport Ships with an Unlimited Region of Navigation

No.	Characteristic X	Anchors		Chain cable for bower anchors		Chain or steel rope for the stream anchor		
		Bower	Stream	Total length of two cables, m	Anchor size, mm	Length, m	Anchor chain size, mm	Diameter of steel rope, mm
1	50	2	25	100	12	50	—	8.8
2	75	2	25	125	13	50	—	8.8
3	100	2	50	125	15	50	—	8.8
4	150	2	50	150	16	50	—	11
5	200	2	50	175	17	75	—	11
6	250	2	75	200	18	75	—	13
7	300	2	75	225	19	75	13	13
8	350	2	100	250	20	75	14	15.5
9	400	2	100	275	21	75	14	15.5
10	450	2	125	300	22	100	15	17.5
11	500	2	150	300	24	100	16	17.5
12	550	2	175	325	25	100	16	17.5
13	600	3	200	350	27	100	17	17.5
14	650	3	225	350	28	100	18	19.5
15	700	3	250	375	29	100	18	20.5
16	750	3	250	375	30	100	19	20.5
17	800	3	250	375	31	125	19	20.5
18	850	3	275	375	32	125	20	22
19	900	3	300	375	33	125	21	24
20	950	3	300	400	34	125	21	24
21	1000	3	320	400	36	125	22	24
22	1100	3	350	400	37	125	23	26
23	1200	3	375	420	38	150	25	26
24	1300	3	410	450	40	150	25	28
25	1400	3	425	450	41	150	25	28
26	1500	3	450	450	42	150	26	28
27	1600	3	475	450	43	150	26	28
28	1700	3	525	450	45	150	28	30
29	1850	3	550	450	46	150	28	30
30	2000	3	575	450	46	150	29	31.5
31	2150	3	600	475	48	175	29	31.5
32	2300	3	650	500	49	175	29	32.5
33	2500	3	675	500	50	175	29	32.5
34	2700	3	750	500	52	175	30	33.5

5-3. Dimensions of Anchoring and Winding Machinery 405

Continued

No.	Characteristic X	Anchors		Chain cable for bower anchors		Chain or steel rope for the stream anchor		
		Bower	Stream	Total length of two cables, m	Anchor size, mm	Length, m	Anchor chain size, mm	Diameter of steel rope, mm
35	3060	3	1000	500	53	200	31	33.5
36	3300	3	1000	500	55	200	31	33.5
37	3600	3	9750	525	57	200	33	34.5
38	3900	3	10500	550	59	225	33	34.5
39	4200	3	11000	550	61	225	34	37
40	4500	3	11500	550	62	225	35	37
41	4800	3	12900	550	65	225	36	—
42	5100	3	13500	550	67	250	37	—
43	5400	3	14500	575	66	250	37	—
44	5800	3	15000	600	70	250	40	—
45	6200	3	15800	620	72	250	40	—
46	6600	3	16300	620	74	275	43	—
47	7000	3	17600	600	76	275	43	—
48	7400	3	18000	600	77	275	44	—
49	7800	3	19500	600	80	275	46	—
50	8200	3	20300	600	82	275	48	—
51	8600	3	21000	600	83	275	49	—
52	9000	3	22000	600	85	275	50	—
53	9500	3	23000	600	87	275	50	—

Note: Two bower anchors with a total weight of at least 2/3 of the tabular value are sufficient for ships navigating in the Caspian Sea and having a characteristic of five or larger.

pentagon. If the bar size of the anchor chain cable is denoted as  $d_c$ , mm, then the chain pitch equal to  $8d_c$  is to be accommodated along one side AC of the pentagon. Thus, since  $AB=BC=4d_c$ , it is evident from triangle OBC that the effective diameter of the cable lifter is

$$D_{cl} = 2R_{cl} = 2 \frac{4d_c}{\sin \alpha} = 13.6d_c \text{ mm} = 0.0136d_c \text{ m} \quad (337)$$

The length of anchor chain cable heaved in in one revolution of the cable lifter is

$$L_c = 5l_c = 5 \times 3d_c = 40d_c \text{ mm} = 0.04d_c \text{ m} \quad (338)$$

where  $d_c$  = chain bar size, mm.



Table 60

Characteristic	Towing rope			Warping hawsers							
	Length, m	Circumference of hemp rope, mm	Diameter of steel rope, mm	Total length, m	Number of ropes	Circumference of hemp rope, mm	Diameter of steel rope, mm	Total length, m	Number of ropes	Circumference of hemp rope, mm	Diameter of steel rope, mm
50	50	75	—	50	1	65	—	—	—	—	—
75	50	90	11	50	1	65	—	—	—	—	—
100	75	90	11	75	1	65	8.5	—	—	—	—
150	75	100	12	75	1	75	8.5	—	—	—	—
200	100	100	12	100	2	75	9.5	—	—	—	—
250	100	125	15	140	2	100	12	—	—	—	—
300	110	125	15	160	2	100	12	—	—	—	—
350	110	150	17.5	160	2	100	12	—	—	—	—
400	135	150	17.5	180	2	125	15	—	—	—	—
450	135	150	17.5	180	2	125	15	80	1	100	12
500	135	150	17.5	200	2	125	15	80	1	100	12
550	135	175	19.5	200	2	125	15	85	1	100	12
600	135	175	19.5	220	2	150	17.5	90	1	100	12
650	135	175	19.5	240	2	150	17.5	90	1	100	12
700	150	200	21.5	240	2	150	17.5	90	1	100	12
750	150	200	21.5	360	4	150	17.5	90	1	125	15
800	150	200	21.5	360	4	150	17.5	90	1	125	15
850	175	200	21.5	360	4	150	17.5	90	1	125	15
900	175	225	24	360	4	175	19.5	120	2	125	15
950	175	225	24	360	4	175	19.5	120	2	125	15
1000	175	225	24	360	4	175	19.5	120	2	150	17.5
1100	175	225	24	360	4	175	19.5	140	2	150	17.5
1200	190	250	26	360	4	175	19.5	140	2	150	17.5
1300	190	250	26	400	4	200	21.5	150	2	150	17.5
1400	190	275	28	400	4	200	21.5	150	2	150	17.5
1500	190	275	28	480	4	200	21.5	150	2	150	17.5
1600	200	300	30	480	4	200	21.5	180	2	150	17.5
1700	200	300	30	480	4	200	21.5	180	2	150	17.5
1850	200	325	32.5	540	4	200	21.5	180	2	175	19.5
2000	200	350	34.5	540	4	200	21.5	180	2	175	19.5
2150	200	350	34.5	540	4	200	21.5	180	2	175	19.5
2300	220	350	34.5	540	4	225	24	180	2	175	19.5
2500	220	350	34.5	640	4	225	24	200	2	175	19.5

Continued

Characteristic	Towing rope			Warping hawsers							
	Length, m	Circumference of hemp rope, mm	Diameter of steel rope, mm	Total length, m	Number of ropes	Circumference of hemp rope, mm	Diameter of steel rope, mm	Total length, m	Number of ropes	Circumference of hemp rope, mm	Diameter of steel rope, mm
2700	220	350	34.5	640	4	225	24	200	2	200	21.5
3000	220	350	34.5	640	4	225	24	200	2	200	21.5
3300	240	375	35	640	4	250	26	200	2	200	21.5
3600	240	375	35	640	4	250	26	200	2	200	21.5
3900	240	400	43.5	640	4	250	26	200	2	200	21.5
4200	240	400	43.5	640	4	250	26	200	2	225	24
4300	240	425	48.5	720	4	250	26	200	2	225	24
4800	240	425	48.5	720	4	250	26	200	2	225	24
5100	240	425	48.5	720	4	275	28	240	2	225	24
5400	240	425	48.5	720	4	275	28	240	2	250	26
5600	240	425	48.5	830	4	275	28	240	2	250	26
6200	240	425	48.5	830	4	300	30	240	2	250	26
6600	240	425	48.5	830	4	300	30	240	2	250	26
7000	240	425	48.5	830	4	300	30	240	2	250	26
7400	240	425	48.5	830	4	300	30	240	2	250	26
7800	240	425	48.5	830	4	300	30	240	2	250	26
8200	240	425	48.5	830	4	300	30	240	2	250	26
8600	240	425	48.5	830	4	300	30	240	2	250	26
9000	240	425	48.5	830	4	300	30	240	2	250	26
9600	240	425	48.5	830	4	300	30	240	2	250	26

Notes: 1. If the actual characteristic's between two tabular values, data should be taken for the next larger tabular characteristic.  
 2. The diameter and circumference of ropes selected from the table for ships with square rigging are to be increased by one size.  
 3. The towing rope (or nonpropelling vessel) is taken one size larger than the tabular value (in diameter and circumference). In addition to the towing rope indicated in the table, towing vessels (lugs) must have a towing rope for towing other vessels. This latter is to be selected in accordance with the pulling capacity of the hook which is taken with a level of margin of safety.  
 4. If Manila or sisal hemp ropes are to be used instead of ordinary hemp, they can be taken one size less than the tabular value.

Denoting the heaving-in speed of the anchor cable as  $v_a$  m per sec, we can find the speed,  $n_{vt}$ , in rpm, of the cable lifter from the equation

$$L n_{vt} = 60 v_a$$

(a) for windlasses and capstans of bower anchors:

$$n_{cl} = \frac{60 v_a}{0.04 d_c} = \frac{60 \times 0.2}{0.04 d_c} = \frac{300}{d_c} \text{ rpm}$$

(b) for the stern anchoring capstan:

$$n_{cl} = \frac{9}{0.04 d_c} = \frac{225}{d_c} \text{ rpm}$$

The efficiency of the anchoring arrangement is  $\eta_a = 0.7$  to  $0.85$  for mechanisms with spur gearing and  $\eta_a = 0.65$  to  $0.75$  for mechanisms

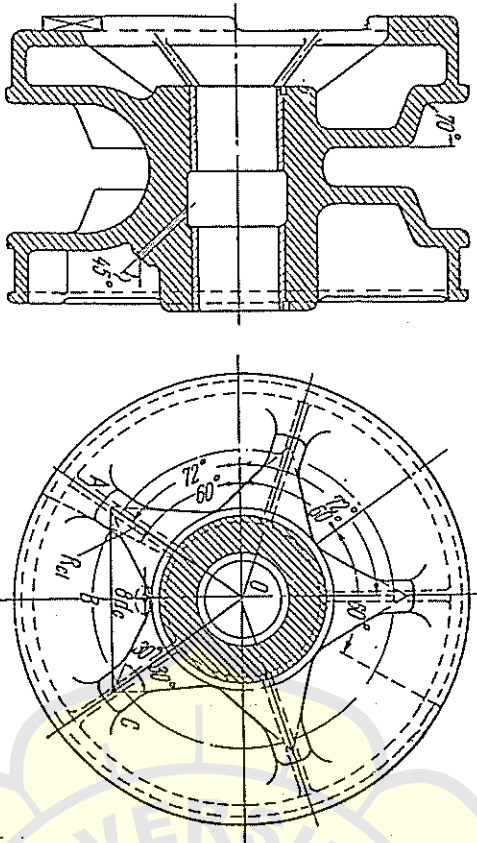


Fig. 170.

with worm gearing. It is the product of the efficiencies of the transmissions and shaft bearings in the gear train (Fig. 171):

$$\eta_a = \eta_{cl} \eta_{sh}^a \eta_{pg}^c \eta_{wg}^c \eta_{br}^c$$

where  $\eta_{cl}$ ,  $\eta_{sh}$ ,  $\eta_{pg}$ ,  $\eta_{wg}$  = efficiencies of the cable lifter, shaft bearings, pairs of spur gears and worm gearing  
 $a$  and  $c$  = number of shaft bearings and pairs of spur gears.

The torque on the cable lifter is

$$M_{cl} = \frac{T_{cl} D_{cl}}{2 \eta_{cl}} \text{ kg-m}$$

where  $\eta_{cl} = 0.9$  to  $0.92$  = efficiency of the cable lifter.

Denoting the engine shaft speed as  $n_m$ , the gearing ratio of the mechanism (Table 61) is

$$i_a = \frac{n_m}{n_{cl}}$$

Table 61

Ancher handling gear	Motive unit shaft speed $n_m$ , rpm	Gearing ratio of mechanism, $i_a$
Hand-operated capstans . . . . .	180 to 320	4 to 40
Steam capstans . . . . .	800 to 1450	18 to 60
Electric capstans . . . . .	800 to 1450	110 to 200
Hand-powered windlasses . . . . .	90 to 270	9 to 18
Steam windlasses . . . . .	90 to 270	6 to 30
Electric windlasses . . . . .	720 to 1550	105 to 250

The torque developed on the shaft of the motive unit is

$$M_m = \frac{M_{cl}}{i_a \eta_a} \text{ kg-m}$$

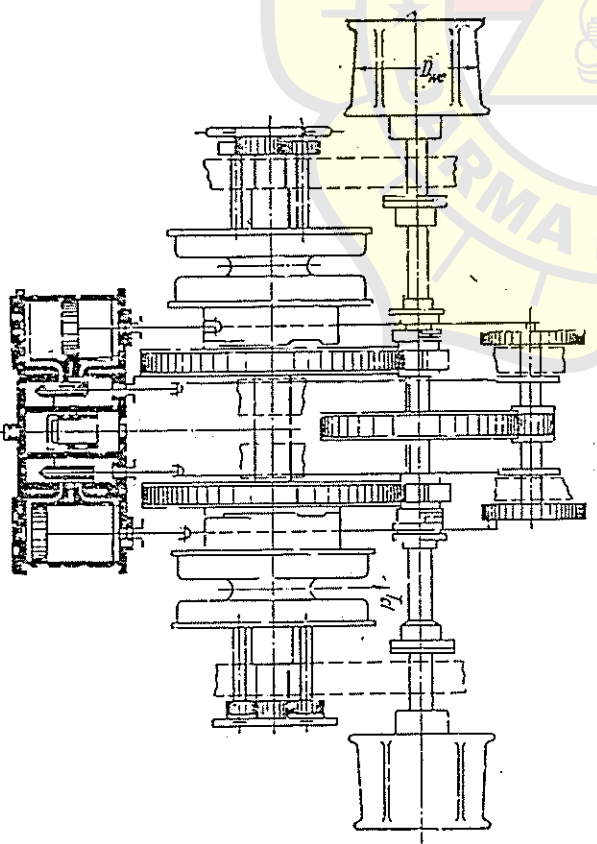


Fig. 171.

The mean shaft power of the motive unit should be

$$N_e = \frac{M_m \eta_m}{716.20} \quad \text{metric hp}$$

The mean indicated power is

$$N_{im} = \frac{N_e}{\eta_m}$$

The cylinder diameter of the steam engine, according to Posdyunin's formula which is based on the conditions for starting from a dead stop, is

$$D_{ca} = 1.37 \sqrt[3]{\frac{M_m}{\psi_a \eta_m (\alpha_i k_i p_{is} - p_{ss})}} \quad \text{cm} \quad (389)$$

where  $M_m$  = torque developed on the shaft of the engine, kg-cm

$\psi_a = 0.85$  to  $1.7$  = cylinder ratio, i.e.,  $S : D_{ca}$ .  
The value of  $(\alpha_i k_i p_{is} - p_{ss})$  is approximately from 10 to 15 per cent lower than that taken for a steering engine, due to longer distance from the anchoring mechanism to the steam supply, resulting in higher condensation losses in the pipelines. The other values in the formula are to be within the same limits as for steam steering engines.

The indicated power  $N_{ia}$  required to start the engine from rest and the coefficient of reserve power are

$$N_{ia} = \frac{\psi_a D_{ca}^3 (\alpha_i k_i p_{is} - p_{ss}) \eta_m}{143,300} \quad \text{metric hp} \quad (390)$$

$$\psi_{res} = \frac{N_{ia}}{N_{im}}$$

The steam consumption of the engine driving the anchoring arrangement is

$$G_{ia} = g_{ia} N_{ia} \quad \text{kg per hour}$$

where  $g_{ia}$  = specific steam consumption, kg per ihp-hr (the same values are taken as for a steam steering engine).

If need arises to determine the pull on the cable lifter from data measured on the anchoring mechanism, formula (390) can be used. Solving Posdyunin's formula (389) for the torque developed on the shaft of the steam engine we can write

$$M_m = \left( \frac{D_{ca}}{1.37} \right)^3 \eta_m \psi_a (\alpha_i k_i p_{is} - p_{ss}) \quad \text{kg-cm}$$

On the other hand, if  $i_g$  is the total gearing ratio of the transmission in the anchoring mechanism, then

$$M_m = \frac{M_e i_g}{\eta_m i_g} = \frac{T_e D_e i_g}{2 \eta_m i_g} \quad \text{kg-cm}$$

Combining the last two equations we obtain

$$T_{e1} = \frac{2 M_m \eta_m i_g}{D_{e1}} = \frac{2}{D_{e1}} \left( \frac{D_{ca}}{1.37} \right)^3 \eta_m \psi_a (\alpha_i k_i p_{is} - p_{ss}) \eta_m i_g$$

$$= 0.78 \frac{D_{ca}^3}{D_{e1}} \eta_m \psi_a (\alpha_i k_i p_{is} - p_{ss}) \eta_m i_g \quad \text{kg}$$

The diameter of the warp ends is taken equal to

$$(a) D_{we} = (5 \text{ to } 8) d_w \text{ for hemp ropes} \quad (391)$$

$$(b) D_{we} = (15 \text{ to } 20) d_w \text{ for steel ropes} \quad (392)$$

where  $d_w$  = diameter of the warping hawser.

Warp end diameters determined from the diameter of hemp ropes will be suitable for steel ropes as well.

Denoting the hawser heaving-in speed as  $v_w$ , m per sec we can find the speed  $\omega$ , the warping shaft from the length of hawser heaved in per minute. Thus

$$\eta_w = \frac{60 v_w}{\pi (D_{we} + d_w)} = 19.1 \frac{v_w}{D_{we} + d_w} \quad \text{rpm} \quad (393)$$

where  $v_w$  = hawser heaving-in speed, m per sec, is to be assigned according to the pull of the warp end (Table 58).

The gearing ratio between the warping shaft and the shaft of the motive unit is

$$i_g = \frac{n_m}{n_w}$$

The pulling force developed on the warp end is

$$T_{we} = \frac{1}{2} \frac{M_{we}}{(D_{we} + d_w)} = \frac{2 M_m \eta_m i_g}{D_{we} + d_w} \leq \frac{P_{br}}{\delta} \quad (394)$$

where  $M_{we}$  = torque developed on the warp end  
 $\eta_w$  = efficiency of the transmission between the warping and motive unit shafts.

If  $n_m$  rpm is the speed of the motive unit shaft, the speed at which the hawser is heaved in will be

$$v_w = \frac{\pi (D_{we} + d_w) n_m}{60 i_g} \quad \text{m per sec} \quad (395)$$

where  $Q_b = 570$  to  $2,175$  = weight of the fully rigged boat, kg  
 $Q_p$  = total weight of all persons allowed to embark (the weight of one person is approximately 75 kg; the number of persons in a boat may reach 78), kg  
 $Q_f = 0.05(Q_b + Q_p)$  = weight of the boat's falls, kg  
 $k_a = 0.9$  to  $1.1$  = coefficient of nonequal distribution of the movable load due to the weight of the persons in the boat.

The maximum tension of the fall at the winch head, after running over the maximum number of guide devices, is

$$T_{max} = \frac{0.5(Q_b + 1.1Q_p) + Q_f}{m\eta_f\eta_r\eta_s^a}$$

where  $m$  = total number of blocks in the boat's falls;

$\eta_f = \frac{1}{me^m} \frac{e^m - 1}{e - 1}$  = efficiency of the boat's falls

$e$  = coefficient depending upon the ratio of the block diameter to the tackle fall diameter ( $e=1.1$  for a hemp fall and  $e=1.04$  to  $1.06$  for a steel wire rope)  
 $\eta_r = 0.9$  to  $0.97$  = efficiency of the davit guide roller  
 $\eta_s = 0.9$  to  $0.97$  = efficiency of the snatch-block  
 $a$  = maximum number of blocks between the davit guide roller and the winch head.  
 The tension at the end of a rope that has run over the minimum number of blocks is

$$T_{min} = \frac{0.5(Q_b + 0.9Q_p) + Q_f}{m\eta_f\eta_r\eta_s^a}$$

where  $c$  = minimum number of blocks.  
 The diameter,  $d_f$ , of a hemp fall is selected according to the breaking strength ( $T_{max} + T_{min}$ )  $\leq R_{br}$  as a function of the boat length from Table 63 (U.S.S.R. Shipping Register).

Table 63

Boat length, m	Circumference of tackle fall, mm	Breaking strength, kg
8.25 to 9.14	95	6,100
7.62 to 8.25	89	5,400
7.35 to 7.62	83	4,600
6.72 to 7.35	73	3,900

The winch head diameter is

$$D_n = (5 \text{ to } 8) d_f$$

The speed,  $v_b$ , with which the boats are hoisted and lowered is assigned so that if the ship is rolling one of these operations can be carried out during the rolling period. This can be accomplished if the heaving-in speed is greater than the vertical component of the crest speed of waves running along the length of the ship. It has been established that the boat hoisting speed must be at least  $v_b = 0.15$  m per sec under these conditions. The heaving-in speed of the tackle fall when single-sheave blocks are used must in this case be  $v_f = 0.5$  m per sec.

The required winch head speed is found from the equation

$$\pi(D_n + d_f) n_n = 60 v_f$$

$$n_n = \frac{60 v_f}{\pi(D_n + d_f)} = 19.1 \frac{v_f}{D_n + d_f} \text{ rpm}$$

Assigning a motive unit speed ( $n_m = 500$  to  $1,600$  rpm for electric motors and  $n_m = 200$  to  $380$  rpm for steam engines), we can find the gearing ratio of the boat winch. Thus

$$i_{bw} = \frac{n_m}{n_n}$$

In designing nonreversible worm gearing the number of teeth on the worm wheel is taken in the range from 24 to 44. The pulling force on the winch head is equal to the sum of the pulling forces on the tackle falls:

$$T = T_{max} + T_{min}$$

Disregarding friction losses, the torque developed on the winch head shaft will be

$$M_n = \frac{T(D_n + d_f)}{2}$$

If the winch has an efficiency of  $\eta_{bw}$ , the torque and power on the motive unit shaft will be

$$M_{mt} = \frac{M_n}{\eta_{bw}} = \frac{T(D_n + d_f)}{2\eta_{bw}}$$

$$N_e = \frac{M_{mt} \omega}{7.16 \cdot 23} \text{ metric hp}$$

The cylinder diameter and indicated power of steam boat winches are determined from the same Posdyunin formulas used in

maximum pressure,  $p$ , kg per sq m, then the amount of liquid pumped is

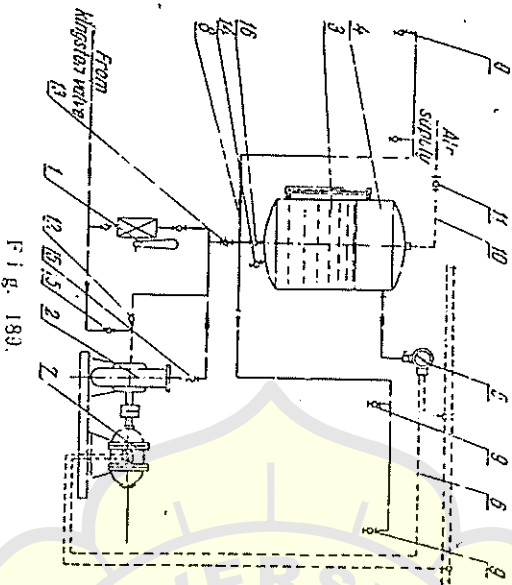
$$V_p = V_e - V_f = D_1 \quad \text{cu m}$$

This equation can be solved for  $V_e$  and  $V_f$ :

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

and

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



The equation of state for the air in the air cushion can be written as

$$V_e p_e = V_f p_f = \left( V_f + \frac{L'}{6} \right) p_e = \left( V_e - \frac{D}{6} \right) p_f$$

Therefore the minimum and maximum volumes of the air are

$$V_f = \frac{L p_e}{6(p_f - p_e)} \quad \text{and} \quad V_e = \frac{D p_f}{6(p_f - p_e)}$$

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

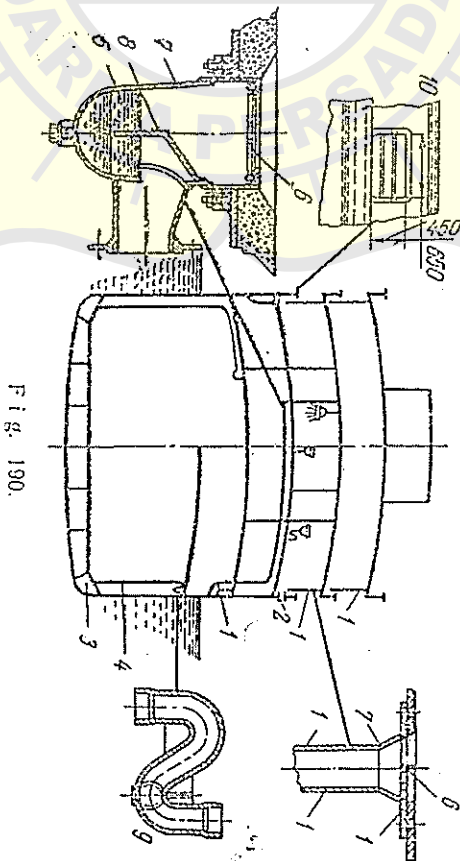
$$V_t = V_0 + V_e = V_0 + \frac{D p_f}{6(p_f - p_e)}$$

Such tanks may also be used in the drinking and ...

(D) SANITARY AND SCUPPER SYSTEMS

The sanitary and scupper systems serve to remove water from the deck and also to dispose of used water from baths, laundries, refreshment bars, galleys, storerooms, etc. Water is drained from the decks through scuppers and their pipes which range from 50 to 100 mm in diameter.

The diagram in Fig. 190 shows how water is removed through scupper pipes 1 from the upper decks and compartment decks. From each deck water runs down to the next lower deck through scupper



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 8 and sunps 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.

## 2. Starting with compressed air

2.1 Main engines which are started with compressed air are to be equipped with at least two starting air compressors. At least one of the air compressors must be driven independently of the main engine and must supply at least 50 % of the total capacity required.

2.2 The total capacity of the starting air compressors is to be such that the starting air receivers designed in accordance with 2.4 or 2.5, as applicable, can be charged from atmospheric pressure to their final pressure within one hour.

Normally, compressors of equal capacity are to be installed.

This does not apply to an emergency air compressor which may be provided to meet the requirement stated in H.1.

2.3 If the main engine is started with compressed air, the available starting air is to be divided between at least two starting air receivers of approximately equal size which can be used independently of each other.

2.4 The total capacity of air receivers is to be sufficient to provide, without their being replenished, not less than 12 consecutive starts alternating between Ahead and Astern of each main engine of the reversible type, and not less than six starts of each main non-reversible type engine connected to a controllable pitch propeller or other device enabling the start without opposite torque. The number of starts refers to an engine in cold and ready-to-start condition.

2.5 With multi-engine installations the number of start up operations per engine may, with the Society's agreement, be reduced according to the type of installation and the way in which the power is transmitted to the propeller.

2.6 If starting air systems for auxiliaries or for supplying pneumatically operated regulating and manoeuvring equipment or tyfon units are to be fed from the main starting air receivers, due attention is to be paid to the air consumption of this equipment when calculating the capacity of the main starting air receivers.

2.7 Other consumers with a high air consumption apart from those mentioned in 2.6 may not be connected to the main starting air system. Separate air supplies are to be provided for these units. Deviations to this require the agreement of the Society.

2.8 For the approximate calculation of the starting air storage capacity, use may be made of the formulae given in Part C of the appendix to this section.

## 3. Electrical starting equipment

3.1 Where main engines are started electrically, two mutually independent starter batteries are to be installed. The batteries are to be so arranged that they cannot be connected in parallel with each other. Each battery must enable the main engine to be started from cold.

The total capacity of the starter batteries must be sufficient for the execution within 30 minutes, without recharging the batteries, of the same number of start-up operations as is prescribed in H.2.4. or H.2.5, as appropriate, for starting with compressed air.

3.2 If two or more auxiliary engines are started electrically, at least two mutually independent batteries are to be provided. Where starter batteries for the main engine are fitted, the use of these batteries is acceptable.

The capacity of the batteries must be sufficient for at least three start-up operations per engine. If only one of the auxiliary engines is started electrically, one battery is sufficient.

3.3 The starter batteries may only be used for starting (and preheating where applicable) and for monitoring equipment belonging to the engine.

3.4 Steps are to be taken to ensure that the batteries are kept charged and the charge level is monitored.

## 4. Start-up of emergency generating sets

4.1 Emergency generating sets are to be so designed that they can be started up readily even at a temperature of 0 °C.

If the set can be started only at higher temperatures, or where there is a possibility that lower ambient temperatures may occur, heating equipment is to be fitted to ensure ready reliable starting.

The operational readiness of the set must be guaranteed under all weather and seaway conditions. Fire flaps required in air inlet and outlet openings must only be closed in case of fire and are to be kept open at all other times. Warning signs to this effect are to be applied. If the flaps close, an alarm must be activated. No alarm is required in the case of automatic fire flap actuation dependent on the operation of the set. Air inlet and outlet openings must not be fitted with weatherproof covers.

4.2 Each emergency generating set required to be capable of automatic starting is to be equipped with an automatic starting system approved by the Society, the capacity of which is sufficient for at least three successive starts (see Volume IV, Rules for Electrical Installation, Section 3, C).

In addition, a second energy source is to be installed

## 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:

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

where

J	[dm <sup>3</sup> ]	total capacity of the starting air receivers
D	[mm]	cylinder bore
H	[mm]	stroke
V <sub>h</sub>	[dm <sup>3</sup> ]	swept volume of one cylinder (in the case of double-acting engines, the swept volume of the upper portion of the cylinder)
P <sub>c,perm</sub>	[bar]	maximum permissible working pressure of the starting air receiver
z	[-]	number of cylinders
P <sub>ce</sub>	[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<sub>c,perm</sub> = 30 bar

$$c = \frac{0,0584}{1 - e^{(0,11 - 0,05 \cdot I_A \cdot P_{c,perm})}}$$

where P<sub>c,perm</sub> ≠ 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<sub>c,perm</sub> to the starting pressure P<sub>A</sub>, then the value of "c" shown in Fig. 2.14 is to be used.

The following values of n<sub>A</sub> are to be applied:

$$n_A = 0,06 \cdot n_o + 14 \quad \text{where } n_o \leq 1000$$

$$n_A = 0,25 \cdot n_o - 176 \quad \text{where } n_o > 1000$$

n<sub>o</sub> [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.

## Section 4

## Main Shafting

## A. General

## 1. Scope

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

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

## 2. Documents for approval

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

## B. Materials

## 1. Approved materials

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

In general, the tensile strength of steels used for shafting shall be between 400 N/mm<sup>2</sup> and 800 N/mm<sup>2</sup>. However, the value of R<sub>m</sub> used for calculation the material factor C<sub>w</sub> in accordance with formula (2) for propeller shafts shall not be greater than 600 N/mm<sup>2</sup>.

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

## 2. Testing of materials

All component parts of the shafting which assist in transmitting the torque from the ship's propulsion

plant are subject to the Society's Rules for Materials and must be tested. This requirement also covers metal propeller shaft liners. Where propeller shafts running in seawater are protected against seawater penetration not by a metal liner but by plastic coatings, the coating technique used must be approved by the Society.

## C. Shaft Dimensions

## 1. General

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

## 2. Minimum diameter

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

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

d [mm] required outside diameter of shaft

d<sub>i</sub> [mm] diameter of shaft bore, where present. If the bore in the shaft is ≤ 0,4 · d<sub>s</sub>, the expression

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

d<sub>s</sub> [mm] actual shaft diameter

P<sub>w</sub> [kW] rated power transmitted by shaft

<sup>1)</sup> For ships flying Indonesian flag in quadruplicate, one of which intended for the Indonesian Government.



[Rpm] rated shaft speed

[-] factor for the type of propulsion installation

- a) Intermediate and thrust shafts  
= 95  
for turbine installations, engine installations with slip couplings and electric propulsion installations  
= 100  
for all other propulsion installations

- b) Propeller shafts  
= 100  
for all types of installations

[-] material factor

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

[N/mm<sup>2</sup>] Tensile strength of the shaft material (see also B.1)

[-] Factor for the type of shaft

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

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

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

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

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

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

= 1,22 for propeller shafts from the area of the aft stern tube or shaft bracket bearing to the forward load-bearing face of the propeller boss subject to a minimum of 2,5 · d, if the

propeller is shrink fitted, without key, on to the tapered end of the propeller shaft using a method approved by the Society, or if the propeller is bolted to a flange forged on the propeller shaft, the propeller shaft runs in oil.

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

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

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

## D. Design

### 1. General

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

### 2. Shaft tapers and propeller nut threads

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

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

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.

### Pipe connections

To prevent the penetration of ballast water into the ship through the bilge system, two means of reverse-flow protection are to be fitted in bilge connections, one of which is to be a screw-down non-return valve.

If such means of protection is to be fitted in bilge suction line.

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.

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

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

The discharge lines of oily water separators are to be fitted with a non-return valve at the ship's side.

### Calculation of pipe diameters

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

#### Dry cargo and passenger ships

##### main bilge pipes

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

##### branch bilge pipes

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

[mm] calculated inside diameter of main bilge pipe

[mm] calculated inside diameter of branch bilge pipe

[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:

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

where:

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

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

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

### 2.4 Minimum diameter

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

### 2.5 Maximum diameter

The diameter of the main bilge line calculated according to 2.2 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:

$$Q = 5,75 \cdot 10 \quad (7)$$

where:

Q [m<sup>3</sup>/h] minimum capacity

$d_{it}$  [mm] calculated inside diameter of main bilge pipe

## 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 a displacement of more than 50.000 ton

= 1,7 for tugs and trawlers

$c_2$  = factor for the rudder type:

= 1,0 in general

= 0,9 for semi-spade rudders

= 0,8 for double rudders (per rudder)

= 0,7 for high lift rudders

$c_3$  = factor for the rudder profile:

= 1,0 for NACA-profiles and plate rudder

= 0,8 for hollow profiles

$c_4$  = factor for the rudder arrangement:

= 1,0 for rudders in the propeller jet

= 1,5 for rudders outside the propeller jet

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

4. Materials

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

4.2 In general materials having a minimum nominal upper yield point  $R_{eH}$  of less than 200 N/mm<sup>2</sup> and a minimum tensile strength of less than 400 N/mm<sup>2</sup> or more than 900 N/mm<sup>2</sup> shall not be used for rudder stocks, pintles, keys and bolts. The requirements of this Section are based on a material's minimum nominal upper yield point  $R_{eH}$  of 235 N/mm<sup>2</sup>. If material is used having a  $R_{eH}$  differing from 235 N/mm<sup>2</sup>, the material factor  $k_r$  is to be determined as follows:

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

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

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

3 Before significant reductions in rudder stock diameter due to the application of steels with  $R_{eH}$  exceeding 235 N/mm<sup>2</sup> are granted, the Society may require the evaluation of the elastic rudder stock deflections. Large deflections should be avoided in order to avoid excessive edge pressures in way of rings.

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

Definitions

- = rudder force in [N]
- = rudder torque in [Nm]
- = total movable area of the rudder in [m<sup>2</sup>]  
For nozzle Rudders, A is not to be taken less than 1,35 times the projected area of the nozzle;

- $A_t$  = A + area of a rudder horn, if any, in [m<sup>2</sup>]
- $A_f$  = portion of rudder area located ahead of the rudder stock axis in [m<sup>2</sup>]
- b = mean height of rudder area in [m]
- c = mean breadth of rudder area in [m] (see Fig. 14.1)

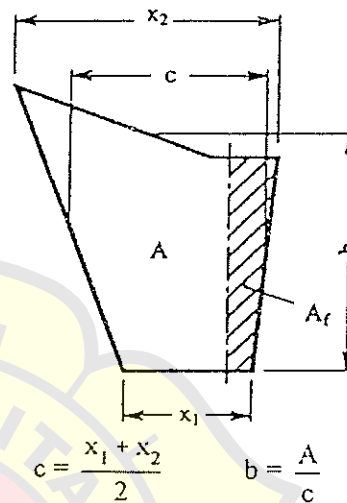


Fig. 14.1

- $\Lambda$  = 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-

according to the following formula:

$$C_R = 132 \cdot A \cdot v^2 \cdot \kappa_1 \cdot \kappa_2 \cdot \kappa_3 \cdot \kappa_4 \quad [N]$$

$v = v_0$  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

$\kappa_4 =$  coefficient depending on the thrust coefficient  $c_t$

$\kappa_4 = 1,0$  normally

In special cases for thrust coefficients  $c_t > 1,0$  determination of  $\kappa_4$  according to the following formula may be required:

$$\kappa_4 = \frac{C_R(c_t)}{C_R(c_t = 1,0)}$$

Table 14.1

Profile/ type of rudder	$\kappa_2$	
	ahead	astern
NACA-00 series Göttinger profiles	1,1	1,4
flat side profiles	1,1	1,4
hollow profiles	1,35	1,4
high lift rudders	1,7	to be specially considered; if not known: 1,7

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

$$Q_R = C_R \cdot r \quad [Nm]$$

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

$\alpha = 0,33$  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:

$$k_b = A_r/A$$

$k_b = 0,08$  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:

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

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

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

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

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

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

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

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

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

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

$$c_1 = A_1/b_1$$

$$C_2 = A_2/b_2$$

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

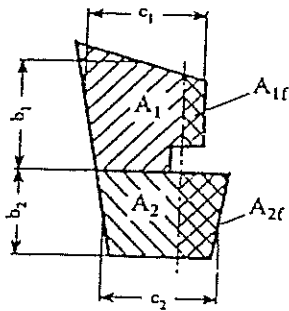


Fig. 14.2

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

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

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

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

for ahead condition

The greater value is to be taken.

### C. 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:

$$D_1 = 4,2 \sqrt[3]{Q_R \cdot k_r} \quad [\text{mm}]$$

see B. 1.2 and B. 2.2 - 2.3.

The related torsional stress is:

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

see A.4.2.

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

In case of mechanical steering gear the diameter of the rudder stock in its upper part which is intended for transmission of the torsional mo-

ment 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:

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

Bending stress:

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

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

Torsional stress:

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

$D_1$  = increased rudder stock diameter in [cm]

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

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

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

$D_t$  see 1.1.

### Guidance

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

## 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 is awaiting berth, tide, etc.*

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

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

4. *The equipment numeral formula for anchoring equipment required under this Section is based on an assumed current speed of 2.5 m/sec, wind speed of 25 m/sec and a scope of chain cable between 6 and 10, the scope being the ratio between length of chain paid out and water depth.*

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

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

Windlass and chain stopper, if fitted, are to comply with 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 snips 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:

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

$D$  = moulded displacement in [ton] (in sea water having a density of  $1,025 \text{ t/m}^3$ ) 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 [ $\text{m}^2$ ], in profile view of the hull, superstructures and houses, having a breadth greater than  $B/4$ , 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  $A$ .

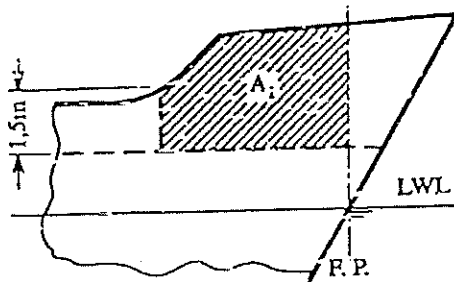


Fig. 18.1

### 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.



Table 18.2 Anchor, Chain Cables and Ropes

No. or Reg.	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 <sup>1</sup>	Mass per anchor	Total length	Diameter			Length	Br. load <sup>2</sup>	Length	Br. load <sup>2</sup>	Number	Length	Br. load <sup>2</sup>		
			[kg]	[m]	d <sub>1</sub> [mm]	d <sub>2</sub> [mm]	d <sub>3</sub> [mm]	[m]	[kN]	[m]	[kN]		[m]	[kN]		
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	
101	up to - 50	2	120	40	165	12.5	12.5	12.5	80	65	180	100	3	80	35	
102	50 - 70	2	180	60	220	14	12.5	12.5	80	65	180	100	3	80	35	
103	70 - 90	2	240	80	220	16	14	14	85	75	180	100	3	100	40	
104	90 - 110	2	300	100	247.5	17.5	16	16	85	80	180	100	3	110	40	
105	110 - 130	2	360	120	247.5	19	17.5	17.5	90	90	180	100	3	110	45	
106	130 - 150	2	420	140	275	20.5	17.5	17.5	90	100	180	100	3	120	50	
107	150 - 175	2	480	165	275	22	19	19	90	110	180	100	3	120	55	
108	175 - 205	2	570	190	302.5	24	20.5	20.5	90	120	180	110	3	120	60	
109	205 - 240	3	660		302.5	26	22	20.5			180	130	4	120	65	
110	240 - 280	3	780		330	28	24	22			180	150	4	120	70	
111	280 - 320	3	900		357.5	30	26	24			180	175	4	140	80	
112	320 - 360	3	1020		357.5	32	28	24			180	200	4	140	85	
113	360 - 400	3	1140		385	34	30	26			180	225	4	140	95	
114	400 - 450	3	1290		385	36	32	28			180	250	4	140	100	
115	450 - 500	3	1440		412.5	38	34	30			180	275	4	140	110	
116	500 - 550	3	1590		412.5	40	34	30			190	305	4	160	120	
117	550 - 600	3	1740		440	42	36	32			190	340	4	160	130	
118	600 - 660	3	1920		440	44	38	34			190	370	4	160	145	
119	660 - 720	3	2100		440	46	40	36			190	405	4	160	160	
120	720 - 780	3	2280		467.5	48	42	36			190	440	4	170	170	
121	780 - 840	3	2460		467.5	50	44	38			190	480	4	170	185	
122	840 - 910	3	2640		467.5	52	46	40			190	520	4	170	200	
123	910 - 980	3	2850		495	54	48	42			190	560	4	170	215	
124	980 - 1060	3	3060		495	56	50	44			200	600	4	180	230	
125	1060 - 1140	3	3300		495	58	50	46			200	645	4	180	250	
126	1140 - 1220	3	3540		522.5	60	52	46			200	690	4	180	270	
127	1220 - 1300	3	3780		522.5	62	54	48			200	740	4	180	285	
128	1300 - 1390	3	4050		522.5	64	56	50			200	785	4	180	305	
129	1390 - 1480	3	4320		550	66	58	50			200	835	4	180	325	
130	1480 - 1570	3	4590		550	68	60	52			220	890	5	190	335	
131	1570 - 1670	3	4890		550	70	62	54			220	940	5	190	350	
132	1670 - 1790	3	5250		577.5	73	64	56			220	1025	5	190	375	
133	1790 - 1930	3	5610		577.5	76	66	58			220	1110	5	190	400	
134	1930 - 2080	3	6000		577.5	78	68	60			220	1170	5	190	425	
135	2080 - 2230	3	6450		605	81	70	62			240	1250	5	200	450	
136	2230 - 2380	3	6900		605	84	73	64			240	1355	5	200	480	
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	490	
139	2700 - 2870	3	8300		632.5	92	81	70			260	1470	6	200	500	
140	2870 - 3040	3	8700		632.5	95	84	73			260	1470	6	200	520	
141	3040 - 3210	3	9200		660	97	84	76			280	1470	6	200	555	
142	3210 - 3400	3	9900		660	100	87	78			280	1470	6	200	590	
143	3400 - 3600	3	10500		660	102	90	78			300	1470	6	200	620	
144	3600 - 3800	3	11100		687.5	105	92	81			300	1470	6	200	650	
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	660	
147	4200 - 4400	3	12900		715	114	100	90			300	1470	7	200	670	
148	4400 - 4600	3	13500		715	117	102	90			300	1470	7	200	680	
149	4600 - 4800	3	14100		715	120	105	92			300	1470	7	200	685	
150	4800 - 5000	3	14700		742.5	122	107	95			300	1470	7	200	685	
151	5000 - 5200	3	15300		742.5	124	111	97			300	1470	8	200	695	
152	5200 - 5300	3	16190		742.5	127	111	97			300	1470	8	200	705	
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	715	
155	6100 - 6500	3	18800		770	124	120	107			300	1470	9	200	725	
156	6500 - 6900	3	20000		770	127	124	111			300	1470	10	200	725	
157	6900 - 7100	3	21500		770	132	127	117			300	1470	11	200	735	
158	7400 - 7900	3	23000		770	137	132	122			300	1470	11	200	735	
159	7900 - 8400	3	24500		770	142	137	127			300	1470	12	200	735	
160	8400 - 8900	3	26000		770	147	142	132			300	1470	13	200	735	
161	8900 - 9400	3	27500		770	152	147	137			300	1470	14	200	735	
162	9400 - 10000	3	29000		770	157	152	142			300	1470	15	200	735	
163	10000 - 10700	3	31000		770	162	157	147			300	1470	16	200	735	
164	10700 - 11500	3	33000		770						300	1470	17	200	735	
165	11500 - 12400	3	35500		770						300	1470	18	200	735	
166	12400 - 13400	3	38500		770						300	1470	19	200	735	
167	13400 - 14600	3	42600		770						300	1470	20	200	735	
168	14600 - 16000	3	46000		770						300	1470	21	200	735	

d<sub>1</sub> = Chain diameter Grade K 1 (Ordinary quality)  
 d<sub>2</sub> = Chain diameter Grade K 2 (Special quality)  
 d<sub>3</sub> = Chain diameter Grade K 3 (Extra special quality)

See also D

1 - see C.1.  
 2 - see F.1.2

Rumus yang sederhana dan praktis untuk perencanaan baling-baling adalah :

Rumus TAYLOR

Untuk Wake fraction : Kapal berbaling2 tunggal;

$$w = -0,05 + 0,5 C_b$$

Kapal berbaling2 ganda;

$$w = -0,20 + 0,55 C_b$$

Untuk Thrust deduction factor :

Kapal berbaling2 tunggal:  $t \approx w$

Kapal berbaling2 ganda;  $t \approx w$

dimana harga k adalah sebagai berikut :

Streamline rudder  $k = 0,55 - 0,70$

Rudder tipis  $k = 0,50$

Rudder tebal  $k = 0,70$

Untuk menghitung harga wake yang lebih teliti adalah memakai diagram yang dibuat oleh Harvald. Untuk dapat membuat diagram tersebut Harvald telah menggunakan 200 model kapal untuk percobaan2-nya di tangki percobaan dinegeri Belanda. Adapun parameter yang ia pilih untuk menentukan besarnya aliran wake adalah :

- Block coefficient  $C_b$ .
- Bentuk dari penampang-penampang melintang kapal bagian belakang.
- Diameter baling-baling  $D$ .
- Panjang kapal  $L$  dan harga perbandingan  $D/L$ .
- Rake dari daun baling-baling dan celah antara baling-baling dengan stern frame.

Block coeff.  $C_b$  mempunyai pengaruh kepada aliran wake. Percobaan Harvald memakai model kapal yg. mempunyai beban indentik tetapi diadakan beberapa perubahan-perubahan pada bagian muka badan kapal.

Ternyata bagian muka badan kapal juga mempunyai pengaruh terhadap besarnya aliran wake. Dari percobaan ini dapatlah diketahui bahwa harga  $w$  tidak hanya dipengaruhi oleh  $C_b$  badan kapal bagian belakang, tetapi oleh  $C_b$  dari keseluruhan badan kapal.

Adapun coefficient prismatic  $C_p$  tidak dipakai sebagai salah satu parameter berhubung percobaan

Harvald disajikan untuk kapal-kapal yang besar dimana coefficient penampang tengah kapal  $C_M$  harganya mendekati 1,0.

Lebar kapal B juga berpengaruh terhadap aliran potential yang menyelubungi badan kapal. Oleh karenanya harga perbandingan L/B merupakan salah satu parameter lain dalam pembuatan diagram aliran wake.

Untuk kapal-kapal samodera dimana harga perbandingan B/d disekitar 2,5 dimana diagram Harvald di tujukan untuk kapal-kapal tersebut pengaruh B/d kepada aliran wake tidak besar. Maka parameter B/d diabaikan.

Bentuk dari penampang melintang badan kapal dibagian belakang merupakan salah satu faktor yang tidak dapat diabaikan dalam pengaruhnya terhadap aliran wake. Suatu perbandingan antara aliran dua dan tiga dimensi menunjukkan bahwa aliran wake akan makin sedikit bilamana aliran tersebut condong kearah aliran tiga dimensi. Dengan kesimpulan itu maka dapatlah dimengerti bahwasanya aliran wake akan lebih besar pada kapal-kapal yang mempunyai penampang bentuk "U" bilamana dibanding terhadap kapal yang mempunyai penampang bentuk "V".

Diameter baling D juga mempunyai pengaruh besar kepada harga wake fraction. Makin besar diameter baling maka akan makin besar pula bagian baling yang bekerja diluar "boundary layer" kapal (baling kapal single screw). Akibatnya adalah besarnya harga rata-rata aliran wake dipiringan baling (propeller disc) akan lebih kecil.

Panjang kapal L menentukan tebal dari boundary layer kapal. Jadi seberapa bagian dari diameter baling D dimana baling bekerja didaerah boundary layer akan tergantung dari panjang kapal L yang menentukan tebalnya boundary layer yang terseret kapal.

Dengan demikian harga perbandingan D/L merupakan parameter lain yang juga harus diperhitungkan. Rake dari baling dan juga celah antara daun baling dengan stern frame merupakan faktor-faktor yang mempunyai pengaruh terhadap aliran wake.

Diagram Harvald untuk mencari harga w adalah sama halnya dengan w dari rumus Taylor yaitu merupakan harga wake rata-rata. Untuk kapal-kapal single screw, harga-harga w diplotkan terhadap harga  $C_b$  untuk beberapa harga L/B yang mana kemudian harus di adakan koreksi untuk bentuk penampang apakah bentuk "U" ataukah "V" dan koreksi untuk harga perbandingan D/L. Bagi kapal-kapal twin screw karena lokasi baling baling berlainan dengan bilamana baling berada ditengah kapal, maka diagram Harvald untuk harga w kapal twin screw tersebut tidak memakai koreksi baik untuk

bentuk penampang badan kapal maupun harga perbandingan  $D/L$ . Menurut hasil yang didapat, ternyata harga-harga  $w$  yang dihitung dengan rumus Taylor perbedaannya tidak seberapa terhadap harga  $w$  dari diagram Harvald terutama untuk kapal-kapal samodera.

Dengan begitu untuk keperluan praktis dalam perencanaan dapatlah dipergunakan rumus Taylor untuk menghitung besarnya harga  $w$ .

Harga thrust deduction factor  $t$  tidak dapat dibuat diagram seperti halnya harga  $w$ . Hal ini disebabkan harga  $t$  sangat terpengaruh sekali besarnya terhadap :

- Ukuran-ukuran stern frame.
- Bentuk kelangsingan (fineness) dari garis air (waterlines) badan kapal.
- Harga perbandingan tebal dan panjang serta bentuk dari daun kemudi, dll.

Untuk keperluan praktis dapatlah dipakai rumus Taylor seperti dimuka yaitu;

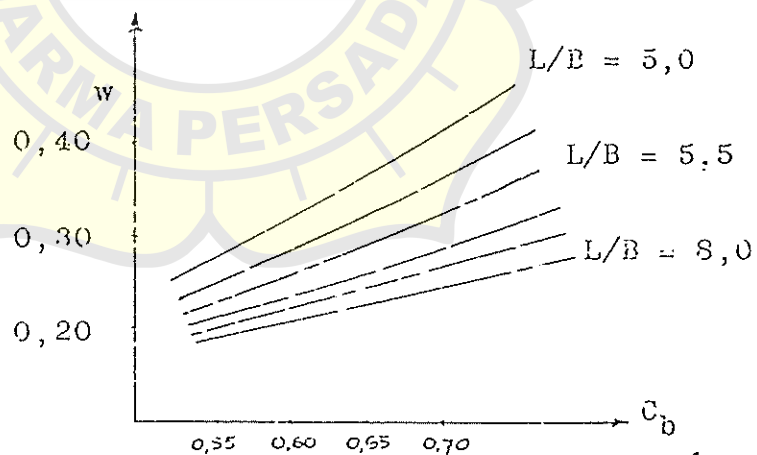
$$t = 0,6 w \text{ untuk kapal 2 single screw.}$$

$$t \approx w \text{ untuk kapal-kapal twin screw.}$$

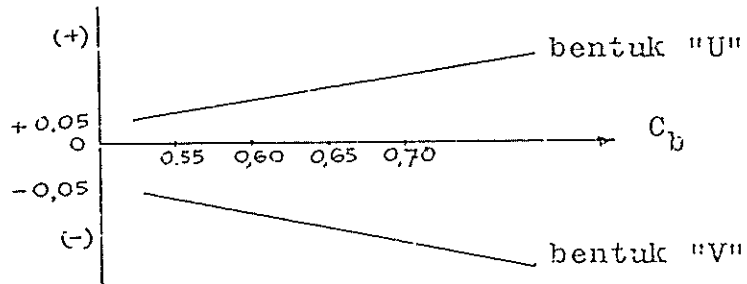
Setelah didapatkannya harga  $w$  maka kalau kecepatan kapal diketahui  $V_s$  dapatlah harga kecepatan air yang melewati piringan baling-baling (intake velocity = advance speed) dihitung yaitu :

$$V_a = V_s (1 - w)$$

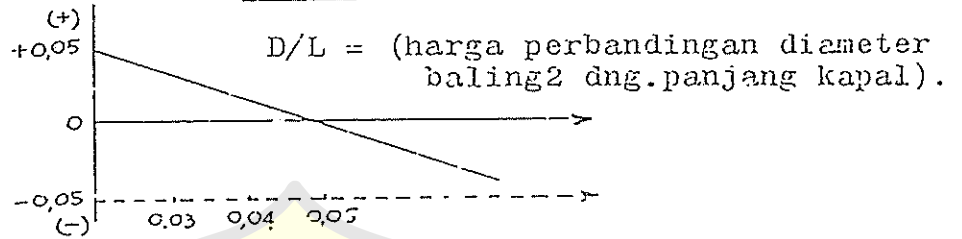
Sketsa diagram Harvald untuk mencari  $w$  :



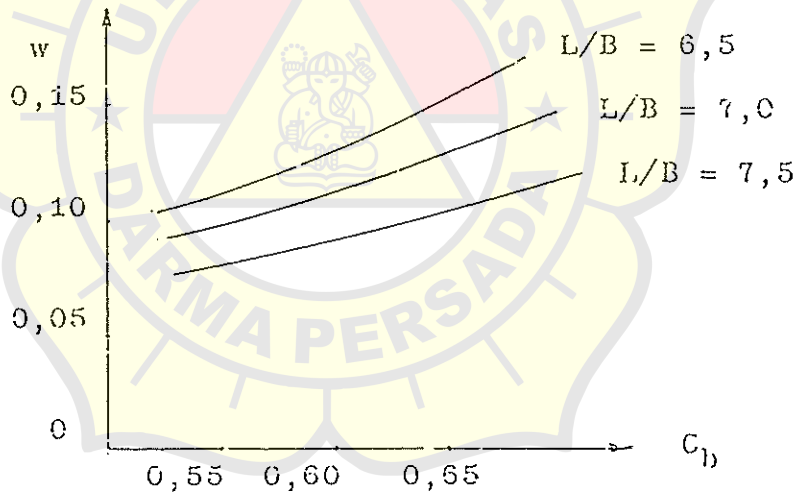
Koreksi bentuk badan kapal,



Koreksi D/L



Untuk kapal-kapal twin screw;



## 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

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

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

$$P_{p1} = \frac{\Delta_p^{2/3} V_p^3}{A_c} \quad (5.1.2)$$

$\Delta_p$  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 sendiri-lah 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.

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 :

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

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

3. Koefisien tahanan sisa spesifik ditentukan dari

$$C_R = C_{Tm} - C_{Fm} \quad (5.5.6)$$

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

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

$R_n$  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  $\nu = 1,139 \times 10^{-6} \text{ m s}^{-1}$ ,  $\rho = 1000 \text{ t/m}^3$ , dan  $T = 15^\circ\text{C}$ . 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

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

4.  $C_R$  dinyatakan sebagai fungsi angka Froude

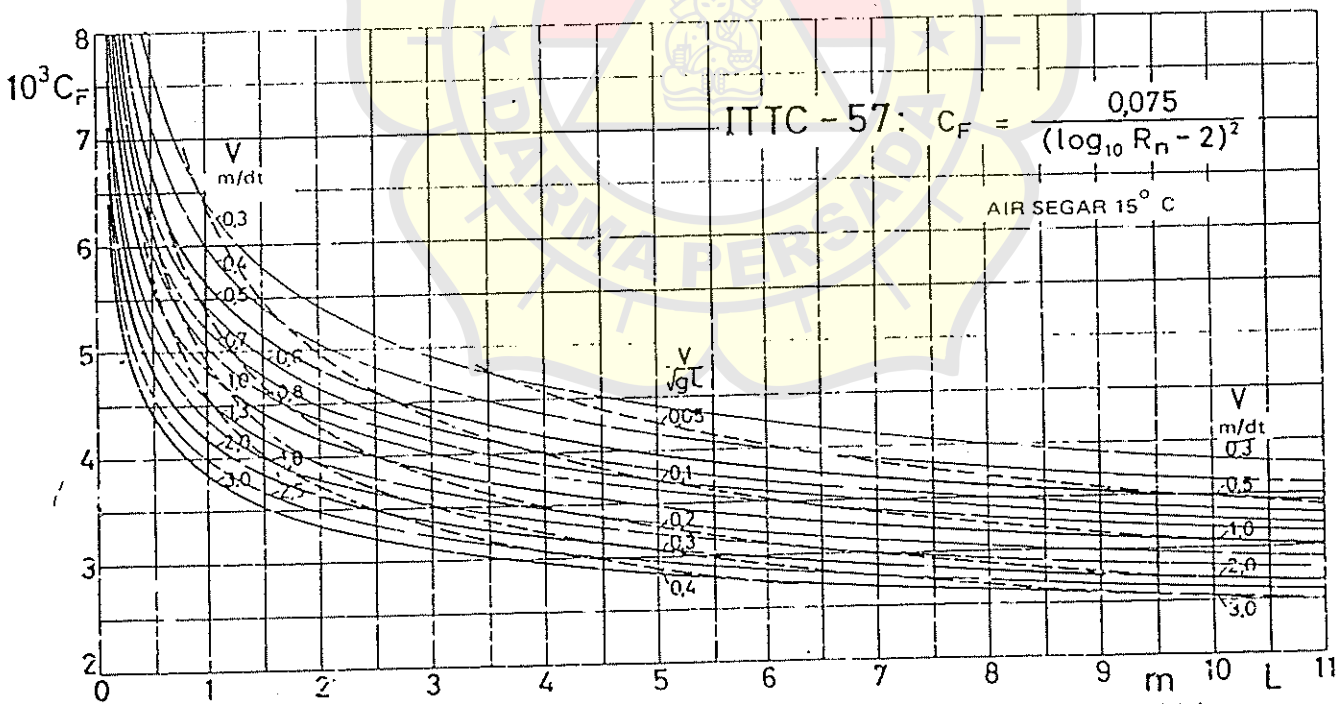
$$F_n = \frac{V}{\sqrt{gL}} \quad (5.5.9)$$

(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  $\phi$  model.  $V$  adalah volume displasemen dan

$$\phi = \frac{\nabla}{LBT\beta} \quad (5.5.10)$$

$B$  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, buritan-nya 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

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

$$P_E = RV \quad (kW) \quad (5.5.12)$$

Dalam hal ini koefisien tahanan totalnya adalah

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

$C_R$  = 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

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

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  $\nu = 1,188 \times 10^{-6} \text{ m}^2/\text{s}$ ,  $\rho = 1,025 \text{ t/m}^3$ , dan  $t = 15^\circ\text{C}$ . Untuk kondisi yang lain, yaitu massa jenis dan suhu yang lain, sebelum memakai diagram tersebut panjang kapal harus diubah dulu sebagai berikut :

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

$C_A$  = 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

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

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 :

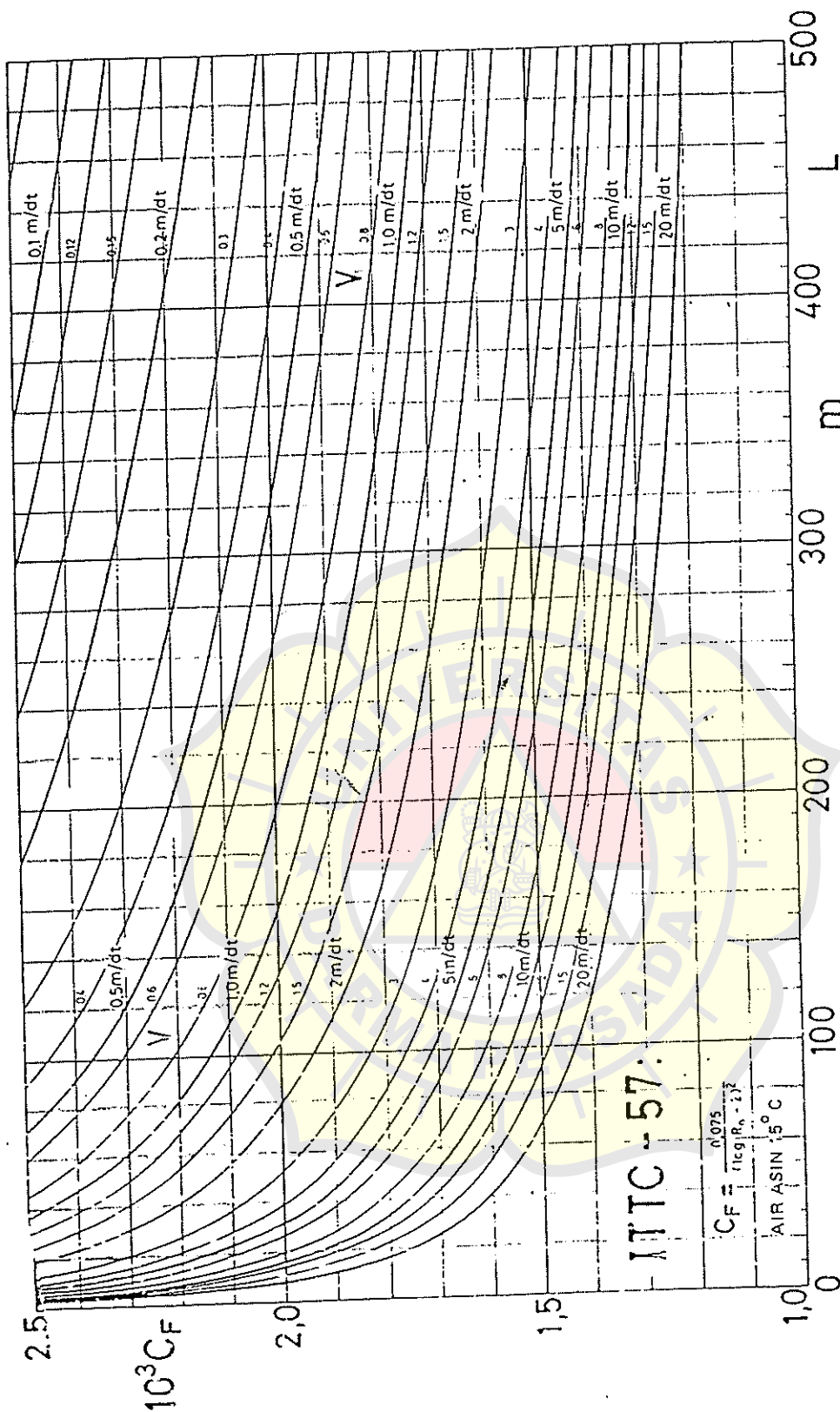
$$10^3 C_R = 10^3 C_{R(B/T=2,5)} + 0,16(B/T - 2,5) \quad (5.5.17)$$

Koreksi ini dapat mempunyai harga yang negatif atau positif.

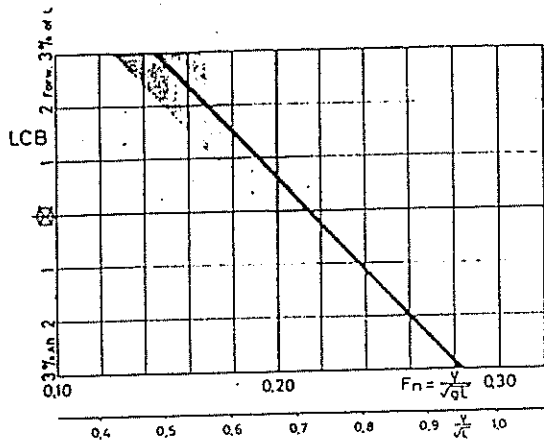
LCB

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

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

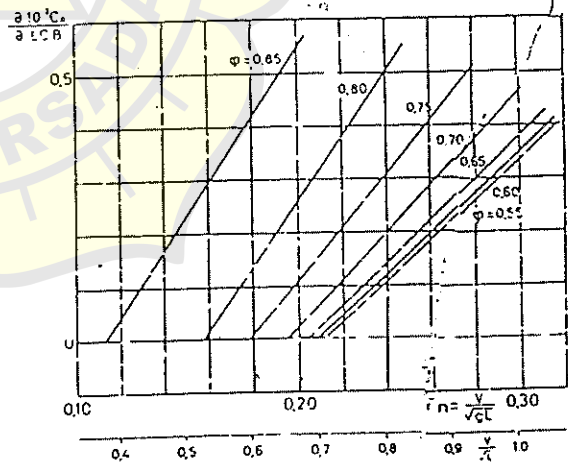
dengan faktor  $\partial 10^3 C_R / \partial LCB$ . Harga faktor ini dapat diperoleh dari Gb. 5.5.16, dan ini hanya berlaku untuk LCB yang berada di depan  $LCB_{\text{standar}}$ . Mengenai LCB yang berada di belakang  $LCB_{\text{standar}}$ , semua sumber yang ada mempunyai pendapat yang saling bertentangan. 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 :

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

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  $LO_s$  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  $(\partial 10^3 C_R / \partial LCB) |\Delta LCB|$ .  $\Delta LCB$  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 (auxiliary 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

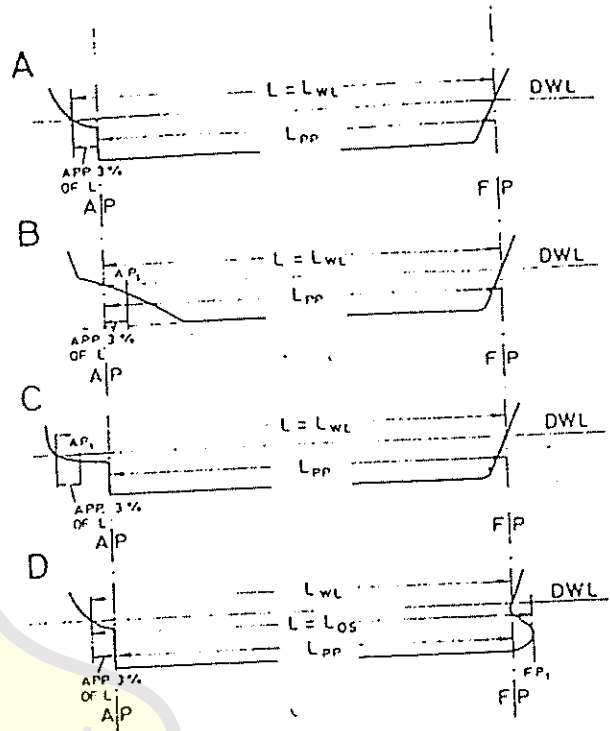
(5.5.20)

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  $A_{BT}/A_X \cong 0,10$  ( $A_{BT}$  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 :

$F_n = 0,15$	0,18	0,21	0,255	0,24	0,253	0,27	0,30	0,33	0,36	$\varphi$
										0,50
		+0,2	0			-0,2	-0,4	-0,4	-0,4	0,60
		+0,2	0			-0,2	-0,3	-0,3	-0,4	0,70
		0	-0,2			-0,3	-0,3	-0,3	-0,4	0,80
+0,1	0	-0,2								

(5.5.21)



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  $A_{RT}/A_X = 0,10$  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.

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 dinaikkan sebesar 3 – 5%	(5.5.22)
Braket dan poros baling-baling	Untuk kapal ramping dinaikkan sebesar 5 – 8%	

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,

$$C_{F'} = C_F \frac{S_1}{S} \quad (5.5.25)$$

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

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 $L \leq 100$ m,	$10^3 C_A = 0,4$
$= 150$ m	$= 0,2$
$= 200$ m	$= 0$
$= 250$ m	$= -0,2$
$\geq 300$ m	$= -0,3$

(5.5.23)

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

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

(5.5.24)

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

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  $10^3 C_{AA}$  sebagai berikut :

$$10^3 C_{AA} = 0,07 \quad (5.5.26)$$

Koreksi untuk tahanan kemudi mungkin sekitar

$$10^3 C_{AS} = 0,04 \quad (5.5.27)$$

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 Australia, 12 – 18%

Jalur pelayaran Asia Timur, 15 – 20%

Tahanan total harus dihitung dengan memakai rumus

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

$S$  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 hidrostatis dari suatu seri yang sangat baik yang terdiri dari sejumlah bentuk kapal yang divariasikan secara sistematis. Permukaan basah semua bentuk memakai koefisien

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

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) :

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

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 jantainya (= 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.

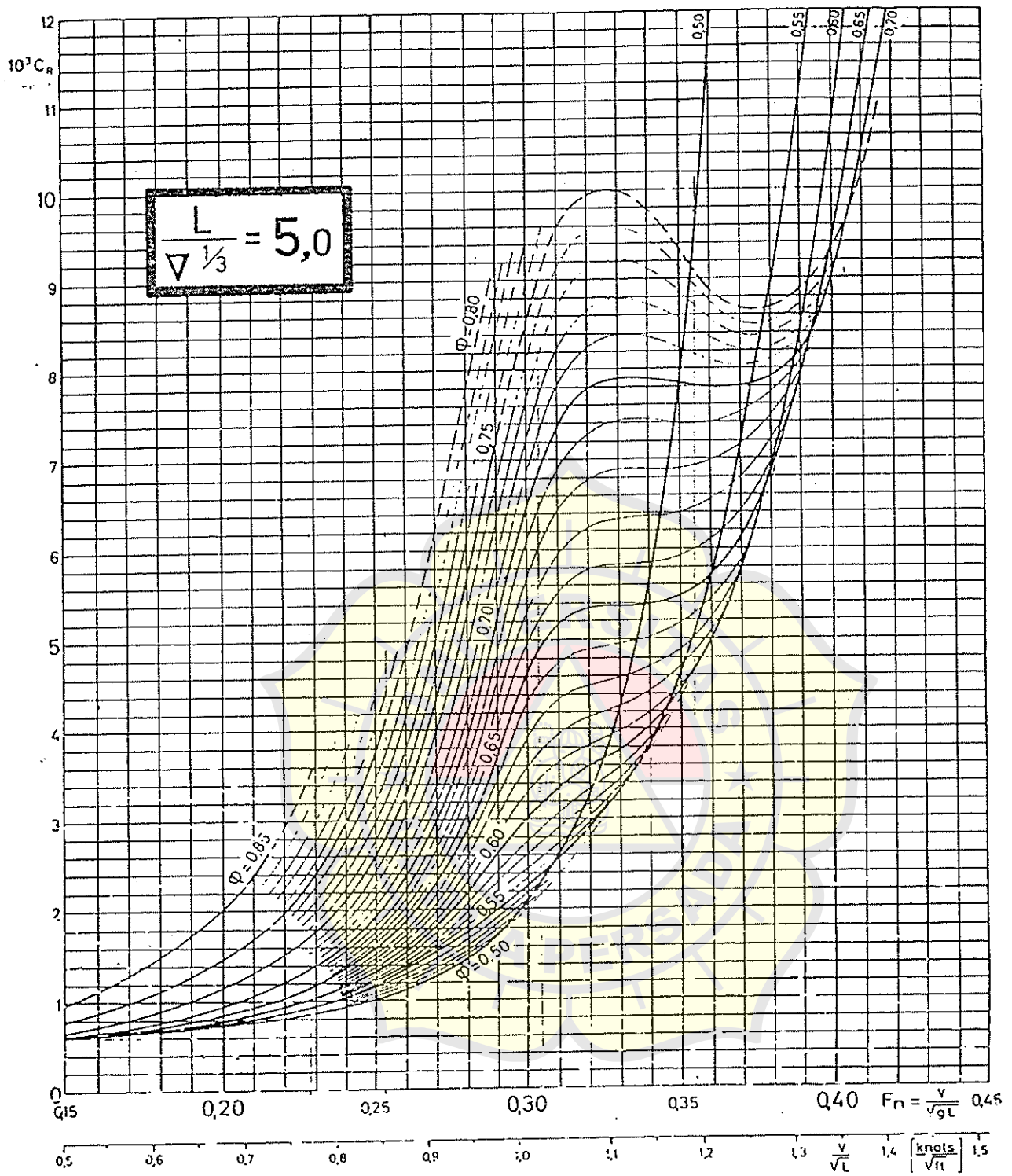
Di lain pihak, upaya untuk mencapai ketepatan yang melebihi dari yang diperlukan untuk menyelesaikan masalah tahanan tidak mempunyai arti yang besar. Tingkat ketidakpastian dalam faktor yang terlibat cukup tinggi, dan pembaca diingatkan untuk tidak membuang waktu untuk memburu ketepatan yang tersisa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

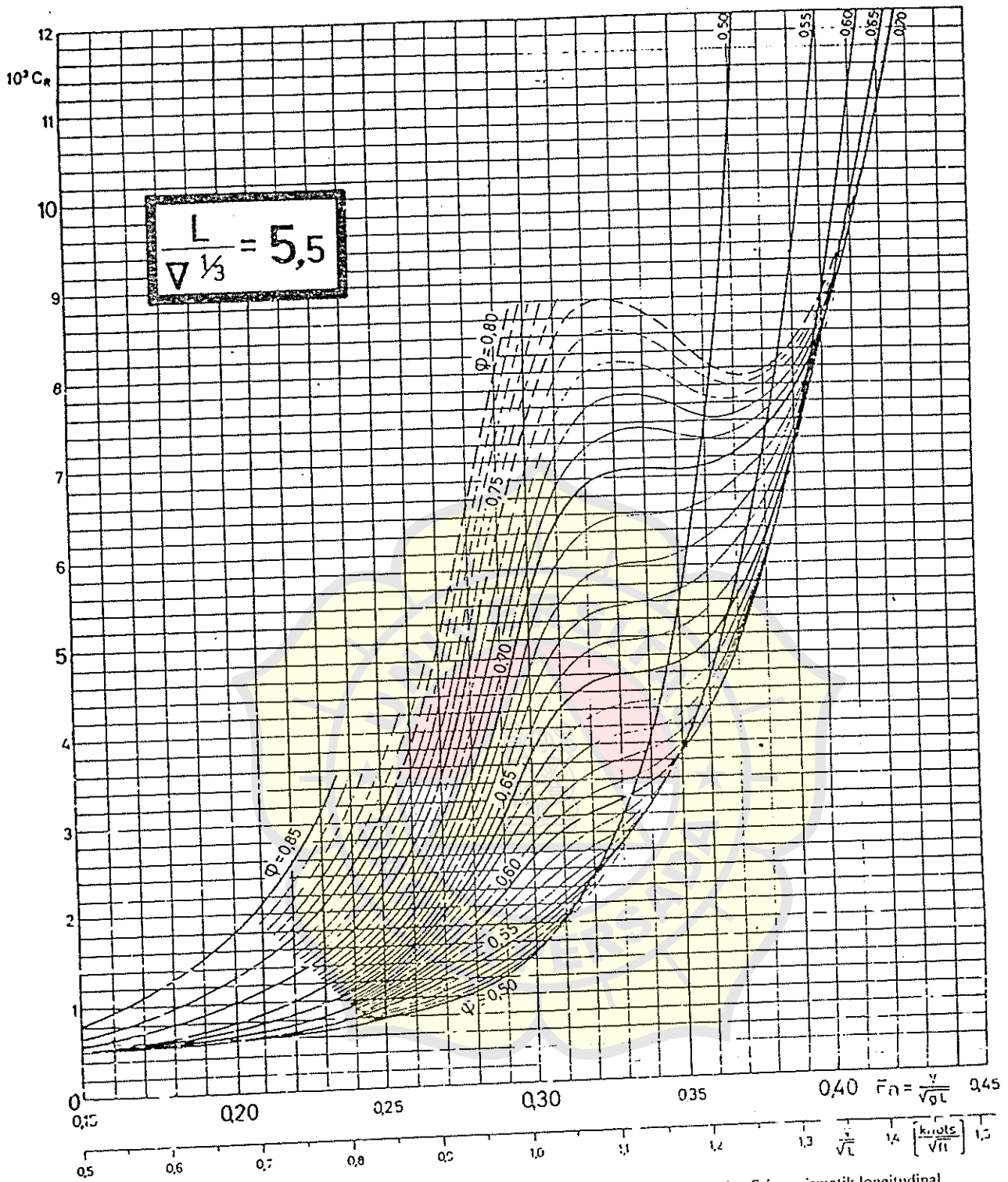
Mesin turbin mempunyai tingkatan menurut jenisnya

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.



Gambar 5.5.7. Koefisien tahanan sisa terhadap rasio kecepatan-panjang untuk harga koefisien prismatic longitudinal yang berbeda-beda.  $L/\Delta^{1/3} = 5,0$ .



Gambar 5.5.8. Koefisien tahanan sisa terhadap rasio kecepatan-panjang untuk harga koefisien prismatik longitudinal yang berbeda-beda.  $L/\Delta^{1/3} = 5.5$ .





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  $35^{\circ}\text{C}$  dan suhu penguapan  $-10^{\circ}\text{C}$ . 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) kju ali: 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-i2) 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  $-10^{\circ}\text{C}$ , yaitu 401,6 kJ/kg.

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

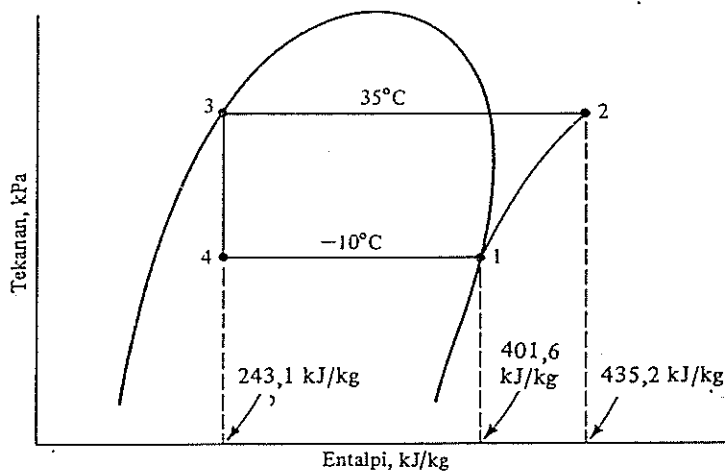
Nilai  $h_3$  dan  $h_4$  identik, dan sama dengan entalpi cairan jenuh pada  $35^{\circ}\text{C}$ , yaitu 243,1 kJ/kg. Sehingga

$$h_1 = 401,6 \text{ kJ/kg} \quad h_2 = 435,2 \text{ kJ/kg}$$

$$h_3 = h_4 = 243,1 \text{ kJ/kg}$$

(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 pendaaran 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

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

(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  $0,0654 \text{ m}^3/\text{kg}$ , sehingga

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

(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  $57^\circ\text{C}$ .

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,<sup>1</sup> 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 (sub-cools) cairan dari kondensor dengan uap isap (suction vapor) yang datang dari evaporator. Susunannya diperlihatkan dalam Gambar 10-13a, dan diagram tekanan-entalpi yang bersangkutan dalam Gambar 10-13b.

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 dapat ditingkatkan. 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

$t, ^\circ\text{C}$	$P, \text{kPa}$	$h_f$	$h_x$	$s_f$	$s_x$	$v_f$	$v_x$
29	1161,1	235,387	414,293	1,12116	1,71325	0,84899	26,2829
30	1191,9	236,664	414,520	1,12530	1,71200	0,85193	19,7417
31	1223,2	237,944	414,762	1,12943	1,71075	0,85491	19,2168
32	1255,2	239,230	414,887	1,13355	1,70956	0,85793	18,7076
33	1287,8	240,520	415,207	1,13768	1,70826	0,86101	18,2135
34	1321,0	241,814	415,420	1,14181	1,70701	0,86412	17,7341
35	1354,8	243,114	415,627	1,14594	1,70576	0,86729	17,2685
36	1389,2	244,418	415,828	1,15007	1,70450	0,87051	16,8168
37	1424,3	245,727	416,021	1,15420	1,70325	0,87378	16,3779
38	1460,1	247,041	416,208	1,15833	1,70199	0,87710	15,9517
39	1496,5	248,361	416,488	1,16246	1,70073	0,88048	15,5375
40	1533,5	249,685	416,661	1,16659	1,69946	0,88392	15,1351
41	1571,2	251,016	416,825	1,17073	1,69819	0,88741	14,7439
42	1609,6	252,352	416,983	1,17486	1,69692	0,89097	14,3636
43	1648,7	253,694	417,133	1,17900	1,69564	0,89459	13,9938
44	1688,5	255,042	417,274	1,18315	1,69435	0,89828	13,6341
45	1729,0	256,396	417,408	1,18730	1,69305	0,90200	13,2841
46	1770,2	257,756	417,532	1,19145	1,69174	0,90586	12,9436
47	1812,1	259,123	417,648	1,19560	1,69043	0,90976	12,6122
48	1854,8	260,497	417,758	1,19977	1,68911	0,91374	12,2895
49	1898,2	261,877	417,852	1,20393	1,68777	0,91779	11,9753
50	1942,3	263,264	417,938	1,20811	1,68643	0,92193	11,6693
52	2032,8	266,062	417,983	1,21648	1,68370	0,93047	11,0806
54	2126,5	268,891	418,083	1,22489	1,68091	0,93939	10,5214
56	2223,2	271,754	418,137	1,23333	1,67805	0,94872	9,98952
58	2323,2	274,654	418,141	1,24183	1,67511	0,95850	9,48319
60	2426,6	277,591	418,089	1,25038	1,67208	0,96878	9,00062
62	2533,3	280,577	417,978	1,25899	1,66895	0,97960	8,54016
64	2643,5	283,607	417,802	1,26767	1,66570	0,99104	8,10023
66	2757,3	286,680	417,553	1,27647	1,66231	1,00317	7,67934
68	2874,7	289,802	417,236	1,28531	1,65876	1,01608	7,27605
70	2995,9	293,008	416,809	1,29436	1,65504	1,02987	6,88999
75	3316,1	301,399	415,299	1,31758	1,64472	1,06916	5,98334
80	3662,3	310,454	412,898	1,34223	1,62239	1,11810	5,14862
85	4036,8	320,505	409,101	1,36936	1,61673	1,18328	4,35815
90	4442,5	332,616	402,653	1,40155	1,59440	1,28250	3,56440
95	4883,5	351,767	386,708	1,45222	1,54712	1,52064	2,55133

$t, ^\circ\text{C}$	$P, \text{kPa}$	$h_f$	$h_x$	$s_f$	$s_x$	$v_f$	$v_x$
-60	37,48	134,763	379,114	0,73254	1,87886	0,68308	537,152
-55	49,47	139,830	381,529	0,75599	1,86389	0,68856	414,827
-50	64,39	144,959	383,921	0,77919	1,85000	0,69526	324,557
-45	82,71	150,153	386,282	0,80216	1,83708	0,70219	256,990
-40	104,95	155,414	388,609	0,82490	1,82504	0,70936	205,745
-35	131,68	160,742	390,896	0,84743	1,81380	0,71680	165,400
-30	163,48	166,140	393,138	0,86976	1,80329	0,72452	135,844
-28	177,76	168,318	394,021	0,87864	1,79927	0,72769	125,563
-26	192,99	170,507	394,896	0,88748	1,79535	0,73092	107,701
-24	209,22	172,708	395,762	0,89630	1,79152	0,73450	99,9362
-22	226,48	174,919	396,619	0,90509	1,78779	0,73853	92,8432
-20	244,83	177,142	397,467	0,91386	1,78415	0,74291	86,3546
-18	264,29	179,376	398,305	0,92259	1,78059	0,74766	80,4103
-16	284,93	181,622	399,133	0,93129	1,77711	0,75286	75,0248
-14	306,78	183,878	399,951	0,93997	1,77371	0,75843	70,1952
-12	329,89	186,147	400,759	0,94862	1,77039	0,76444	65,9478
-10	354,30	188,426	401,555	0,95725	1,76713	0,77082	62,3399
-9	367,01	189,571	401,949	0,96155	1,76553	0,77663	63,1746
-8	380,06	190,718	402,341	0,96585	1,76394	0,78283	61,0958
-7	393,47	191,868	402,729	0,97014	1,76237	0,78944	59,0996
-6	407,23	193,021	403,114	0,97442	1,76082	0,79636	57,1820
-5	421,35	194,176	403,496	0,97870	1,75928	0,78311	55,3394
-4	435,84	195,335	403,876	0,98297	1,75775	0,77028	53,5682
-3	450,70	196,497	404,252	0,98724	1,75624	0,77226	51,8653
-2	465,94	197,662	404,626	0,99150	1,75475	0,77429	50,2274
-1	481,57	198,828	404,994	0,99575	1,75326	0,77629	48,6517
0	497,59	200,000	405,361	1,00000	1,75179	0,77834	47,1354
1	514,01	201,174	405,724	1,00424	1,75034	0,78041	45,6757
2	530,83	202,351	406,084	1,00848	1,74889	0,78249	44,2702
3	548,06	203,530	406,440	1,01271	1,74746	0,78460	42,9166
4	565,71	204,713	406,793	1,01694	1,74604	0,78673	41,6124
5	583,78	205,899	407,143	1,02116	1,74463	0,78889	40,3556
6	603,28	207,089	407,489	1,02537	1,74324	0,79107	39,1441
7	624,22	208,281	407,831	1,02958	1,74185	0,79327	37,9759
8	646,59	209,477	408,169	1,03379	1,74047	0,79549	36,8493
9	660,42	210,675	408,504	1,03799	1,73911	0,79775	35,7624
10	680,70	211,877	408,835	1,04218	1,73775	0,80002	34,7136
11	701,44	213,083	409,162	1,04637	1,73640	0,80232	33,7013
12	722,65	214,291	409,485	1,05056	1,73506	0,80465	32,7239
13	744,33	215,503	409,804	1,05474	1,73373	0,80701	31,7801
14	766,50	216,719	410,119	1,05892	1,73241	0,80939	30,8683
15	789,15	217,937	410,430	1,06309	1,73109	0,81180	29,9874
16	812,29	219,160	410,736	1,06726	1,72978	0,81424	29,1361
17	835,93	220,386	411,038	1,07142	1,72848	0,81671	28,3131
18	860,08	221,615	411,336	1,07559	1,72719	0,81922	27,5173
19	884,75	222,848	411,629	1,07974	1,72590	0,82175	26,7477
20	909,93	224,084	411,918	1,08390	1,72462	0,82431	26,0032
21	935,64	225,324	412,202	1,08805	1,72334	0,82691	25,2829
22	961,89	226,568	412,481	1,09220	1,72206	0,82954	24,5857
23	988,67	227,816	412,755	1,09634	1,72080	0,83221	23,9107
24	1016,0	229,068	413,025	1,10048	1,71953	0,83491	23,2572
25	1043,9	230,324	413,289	1,10462	1,71827	0,83765	22,6242
26	1072,3	231,583	413,548	1,10876	1,71701	0,84043	22,0111
27	1101,4	232,847	413,802	1,11290	1,71576	0,84324	21,4169
28	1130,9	234,115	414,050	1,11703	1,71450	0,84610	20,8411

Tabel A-7 Refrigeran 22: sifat-sifat uap gas panas lanjut<sup>5</sup>

$t, ^\circ\text{C}$	$v, \text{L/kg}$	$h, \text{kJ/kg}$	$s, \text{kJ/kg} \cdot \text{K}$	$v, \text{L/kg}$	$h, \text{kJ/kg}$	$s, \text{kJ/kg} \cdot \text{K}$	$v, \text{L/kg}$	$h, \text{kJ/kg}$	$s, \text{kJ/kg} \cdot \text{K}$
Suhu jenuh, $-20^\circ\text{C}$			Suhu jenuh, $-10^\circ\text{C}$			Suhu jenuh, $0^\circ\text{C}$			
-20	92,8432	397,467	1,7841						
-15	95,1474	400,737	1,7969						
-10	97,4256	404,017	1,8095	65,3399	401,555	1,7671			
-5	99,6808	407,307	1,8219	67,0081	404,983	1,7800			
0	101,915	410,610	1,8341	68,6524	408,412	1,7927	47,1354	405,361	1,7518
5	104,130	413,926	1,8461	70,2751	411,845	1,8052	48,3899	408,959	1,7649
10	106,328	417,258	1,8580	71,8785	415,283	1,8174	49,6215	412,567	1,7777
15	108,510	420,606	1,8697	73,4644	418,730	1,8295	50,8328	416,159	1,7903
20	110,678	423,970	1,8813	75,0346	422,186	1,8414	52,0259	419,649	1,8026
25	112,832	426,353	1,8928	76,5904	425,653	1,8531	53,2028	423,339	1,8148
Suhu jenuh, $5^\circ\text{C}$			Suhu jenuh, $10^\circ\text{C}$			Suhu jenuh, $15^\circ\text{C}$			
5	40,3556	407,143	1,7445						
10	41,4580	410,851	1,7578	34,7136	408,835	1,7377			
15	42,5379	414,542	1,7708	35,6907	412,651	1,7511	29,9874	410,430	1,7311
20	43,5979	418,222	1,7834	36,6454	416,442	1,7642	30,8606	414,362	1,7556
25	44,6401	421,894	1,7958	37,5804	420,215	1,7769	31,7114	418,260	1,7578
30	45,6665	425,562	1,8080	38,4981	423,974	1,7894	32,5427	422,133	1,7707
35	46,6786	429,229	1,8200	39,4002	427,724	1,8017	33,3568	425,985	1,7833
40	47,6779	432,897	1,8319	40,2884	431,469	1,8137	34,1556	429,823	1,7956
45	48,6656	436,569	1,8435	41,1642	435,211	1,8256	34,9409	433,650	1,8078
50	49,6427	440,247	1,8550	42,0286	438,954	1,8373	35,7139	437,470	1,8197

Refrigeran 22: sifat-sifat uap gas panas lanjut (lanjutan)

Tabel A-7 (lanjutan)

Suhu jenuh, $20^\circ\text{C}$				Suhu jenuh, $25^\circ\text{C}$			Suhu jenuh, $30^\circ\text{C}$		
20	26,0032	411,918	1,7246						
25	26,7900	415,977	1,7383	22,6242	413,289	1,7183			
30	27,5542	419,991	1,7517	23,3389	417,487	1,7322	19,7417	414,530	1,7120
35	28,2989	423,970	1,7646	24,0306	421,627	1,7458	20,3962	418,881	1,7262
40	29,0264	427,922	1,7774	24,7027	425,721	1,7590	21,0272	423,159	1,7400
45	29,7389	431,852	1,7899	25,3575	429,779	1,7718	21,6381	427,378	1,7534
50	30,4379	435,766	1,8021	25,9974	433,807	1,7844	22,2316	431,549	1,7664
55	31,1250	439,668	1,8141	26,6239	437,813	1,7967	22,8101	435,683	1,7791
60	31,8012	443,561	1,8258	27,2386	441,801	1,8087	23,3733	439,787	1,7915
65	32,4678	447,450	1,8374	27,8427	445,777	1,8206	23,9288	443,867	1,8036
Suhu jenuh, $32^\circ\text{C}$				Suhu jenuh, $34^\circ\text{C}$			Suhu jenuh, $36^\circ\text{C}$		
35	19,0907	417,648	1,7182	17,8590	416,325	1,7099			
40	19,7093	422,014	1,7322	18,4675	420,792	1,7243	17,2953	419,453	1,7162
45	20,3062	426,310	1,7458	19,0526	425,174	1,7382	17,8708	423,961	1,7304
50	20,8847	430,549	1,7591	19,6178	429,487	1,7517	18,4247	428,358	1,7442
55	21,4471	434,743	1,7719	20,1660	433,747	1,7647	18,9603	432,690	1,7575
60	21,9956	438,900	1,7845	20,6994	437,963	1,7775	19,4802	436,970	1,7704
65	22,5318	443,028	1,7968	21,2199	442,143	1,7899	19,9865	441,207	1,7830
70	23,0571	447,133	1,8089	21,7289	446,294	1,8021	20,4807	445,410	1,7954
75	23,5726	451,219	1,8207	22,2278	450,424	1,8141	20,9643	449,586	1,8078
80	24,0794	455,292	1,8323	22,7176	454,535	1,8258	21,4385	453,739	1,8193

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

$$V_{\text{trial}} \approx 1.06 \cdot V_{\text{service}} \quad (\text{this corresponds to a power margin of about } 20 - 25\%).$$

The propeller is designed for 85% ... 90% of the driving power, at 100% of revolutions.

#### 26. 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

$$W_{\text{fuel oil}} \text{ [t]} = P_{\text{Bme}} \cdot b_{\text{me}} + P_{\text{ae}} \cdot b_{\text{ae}} \cdot \frac{S}{V_{\text{serv}}} \cdot 10^{-6} \cdot [1.3 \dots 1.5]$$

last brackets for reserve:

- fuel rests in tanks
- seaway
- wind
- waiting time
- (- according to owner's desire!)

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 auxillary engines and for the main engine during estuary trading.

$$w_{\text{diesel}} = (0.1 \dots 0.2) \cdot w_{\text{heavy fuel oil}}$$

$$\text{specific weight } \gamma_{\text{diesel}} = 0.85 \text{ t/m}^3$$

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

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!).

$$w_{\text{lubr.}} = \rho_{\text{time}} \cdot b_{\text{mc}} \cdot \frac{S}{V_{\text{serv}}} + \text{addition}$$

$$b = 0.8 \dots 1.2 \quad [\text{g/KW}\cdot\text{h}] \quad \text{diesel engine two stroke}$$

$$b = 1.2 \dots 1.6 \quad [\text{g/KW}\cdot\text{h}] \quad \text{diesel engine four stroke}$$

$$b = 0.14 \quad [\text{g/KW}\cdot\text{h}] \quad \text{turbines and gearboxes}$$

$$\text{specific weight } \gamma_{\text{lubr}} = 0.90 \text{ t/m}^3 ; \quad v = \frac{w}{\gamma} \quad (\text{m}^3)$$

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

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.

- 15% lower fore peak tank
- 13% upper fore peak tank
- 12% 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.

*nyin, tempat*  
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).

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

Mula-mula perlu ditentukan jumlah limpasan keseluruhan dari air hujan di tanah pertanian dengan rumus

$$Q = 10fRA10 \quad (2.2.a)$$

di mana  $Q$ : Limpasan keseluruhan ( $m^3$ )

$R$ : Curah hujan standar (mm)

$f$ : Koefisien limpas

$A$ : Luas wilayah drainase (ha)

Dari jumlah limpasan yang dihitung dengan cara di atas kemudian dapat diperkirakan kapasitas pompa drainase yang diperlukan dengan rumus

$$Q_p = \frac{Q}{24 \times 3600 \times D} \quad (2.2.b)$$

di mana  $Q_p$ : Kapasitas pompa drainase ( $m^3/s$ )

$D$ : Lamanya genangan yang diperbolehkan (hari)

Koefisien limpas yang dipakai untuk menentukan limpasan total dipengaruhi oleh curah hujan total seperti diberikan di dalam Tabel 2.7.

Jumlah hari limpas harus dihitung secara coba-coba dengan memperhatikan bahwa limpasan total akan terdistribusikan seperti dalam Tabel 2.8.

Tabel 2.7 Curah hujan total dan koefisien limpas total.

Curah hujan total (mm)	Kurang dari 10	10-30	30-50	50-100	100-200	200-300	Lebih dari 300
Koefisien limpas total	0	0,10	0,30	0,50	0,80	0,90	0,95

Tabel 2.8 Faktor distribusi limpasan dari curah hujan tunggal.

Curah hujan (mm)	Hari				Jumlah
	Hari ke-1	Hari ke-2	Hari ke-3	Hari ke-4	
Kurang dari 30	100%	-	-	-	100%
30-50	70%	30%	-	-	100%
50-100	60%	30%	10%	-	100%
Lebih dari 100	50%	30%	15%	5%	100%

Untuk penentuan akhir dari spesifikasi perencanaan, kondisi limpasan air hujan dan kondisi fluktuasi muka air harus diperhitungkan. Dalam hal ini perlu dipelajari buku-buku profesional dalam bidang tersebut.

#### (5) Pengairan tanah pertanian

Ditinjau dari cara pengairan, tanah pertanian dapat dibedakan antara sawah dan ladang.

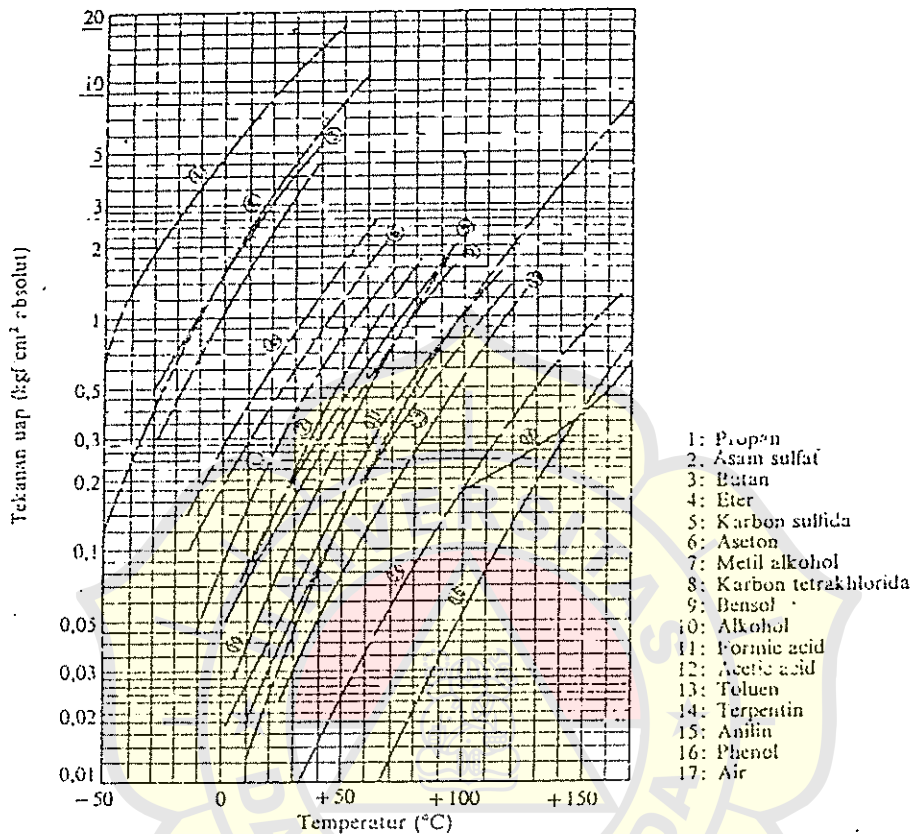
##### (a) Pengairan sawah

###### 1) Keperluan air

Sawah untuk tanaman padi harus digenangi air dengan kedalaman tertentu. Untuk memelihara kedalaman tersebut diperlukan tambahan air terus menerus guna mengganti penyusutan karena transpirasi tanaman, penguapan sawah, dan perkolasi\*. Jadi:

- \* Transpirasi = penguapan melalui pernapasan tanaman
- Penguapan = penguapan langsung dari air ke udara
- Perkolasi = peresapan air ke dalam tanah.





(b) Tekanan uap berbagai zat cair  
 (Catatan: 1 kgf/cm<sup>2</sup> = 0,1 MPa)

Gb. 2.1 Sifat-sifat fisik berbagai zat cair.

## 2.4 Head

### 2.4.1 Head Total Pompa

Head total pompa yang harus disediakan untuk mengalirkan jumlah air seperti direncanakan, dapat ditentukan dari kondisi instalasi yang akan dilayani oleh pompa. Seperti diperlihatkan dalam Gb. 2.2, head total pompa dapat ditulis sebagai berikut:

$$H = h_a + \Delta h_p + h_f + \frac{v_d^2}{2g} \quad (2.6)$$

di mana  $H$ : Head total pompa (m)

$h_a$ : Head statis total (m)

Head ini adalah perbedaan tinggi antara muka air di sisi keluar dan di sisi isap; tanda positif (+) dipakai apabila muka air di sisi ke luar lebih tinggi dari pada sisi isap.

$\Delta h_p$ : Perbedaan head tekanan yang bekerja pada kedua permukaan air (m),

$$\Delta h_p = h_{p2} - h_{p1}$$

$h_f$ : Berbagai kerugian head di pipa, katup, belokan, sambungan, dll (m),

di mana  $h_p$ : Head tekanan (m)

$p$ : Tekanan (kgf/cm<sup>2</sup>)

$\gamma$ : Berat per satuan volume zat cair yang dipompa (kgf/l)

Apabila tekanan diberikan dalam kPa, dapat dipakai rumus berikut:

$$h_p = \frac{1}{9,8} \frac{p'}{\rho} \quad (2.9)$$

di mana  $p'$ : Tekanan (Pa)

$\rho$ : Rapat masa (kg/l)

Menurut ISO, energi spesifik  $Y$  (J/kg) kadang-kadang dipakai sebagai pengganti head  $H$  (m). Adapun hubungannya adalah sebagai berikut:

$$Y = gH \quad (2.10)$$

Sebagaimana diutarakan di atas, untuk menentukan head total yang harus disediakan pompa, perlu dihitung lebih dahulu head kerugian  $h_f$ . Di bawah ini akan diuraikan cara menghitung kerugian head tersebut.

#### 2.4.2 Head Kerugian

Head kerugian (yaitu head untuk mengatasi kerugian-kerugian) terdiri atas head kerugian gesek di dalam pipa-pipa, dan head kerugian di dalam belokan-belokan, reduser, katup-katup, dsb. Di bawah ini akan diberikan cara menghitungnya, satu per satu.

##### (1) Head kerugian gesek dalam pipa

Untuk menghitung kerugian gesek di dalam pipa dapat dipakai salah satu dari dua rumus berikut ini:

$$s = CR^q S^q \quad (2.11)$$

$$h_f = \lambda \frac{L}{D} \frac{v^2}{2g} \quad (2.12)$$

di mana  $v$ : Kecepatan rata-rata aliran di dalam pipa (m/s)

$C, p, q$ : Koefisien-koefisien

$R$ : Jari-jari hidrolis (m)

$$R = \frac{\text{Luas penampang pipa, tegak lurus aliran (m}^2\text{)}}{\text{Keliling pipa atau saluran yang dibasahi (m)}}$$

$S$ : Gradien hidrolis

$$S = \frac{h_f}{L}$$

$h_f$ : Head kerugian gesek dalam pipa (m)

$\lambda$ : Koefisien kerugian gesek

$g$ : Percepatan gravitasi (9,8 m/s<sup>2</sup>)

$L$ : Panjang pipa (m)

$D$ : Diameter dalam pipa (m)

Selanjutnya, untuk aliran yang laminar dan yang turbulen, terdapat rumus yang berbeda. Sebagai patokan apakah suatu aliran itu laminar atau turbulen, dipakai bilangan Reynolds:

$$Re = \frac{vD}{\nu} \quad (2.13)$$

di mana  $Re$ : Bilangan Reynolds (tak berdimensi)  
 $v$ : Kecepatan rata-rata aliran di dalam pipa (m/s)  
 $D$ : Diameter dalam pipa (m)  
 $\nu$ : Viskositas kinematik zat cair (m<sup>2</sup>/s)

Pada  $Re < 2300$ , aliran bersifat laminar.

Pada  $Re > 4000$ , aliran bersifat turbulen.

Pada  $Re = 2300 - 4000$  terdapat daerah transisi, di mana aliran dapat bersifat laminar atau turbulen tergantung pada kondisi pipa dan aliran.

(I) Aliran laminar

Dalam hal aliran laminar, koefisien kerugian gesek untuk pipa ( $\lambda$ ) dalam pers. (2.12) dapat dinyatakan dengan

$$\lambda = \frac{64}{Re} \tag{2.14}$$

(II) Aliran turbulen

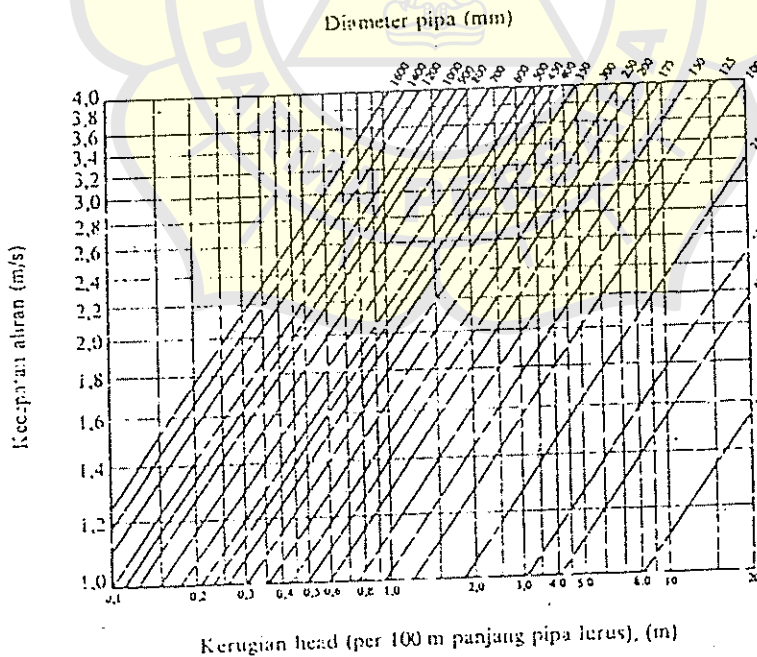
Untuk menghitung kerugian gesek dalam pipa pada aliran turbulen terdapat berbagai rumus empiris. Di bawah ini akan diberikan cara perhitungan dengan rumus Darcy dan Hazen-Williams.

1) *Formula Darcy*

Dengan cara Darcy, koefisien kerugian gesek  $\lambda$  dari Pers. (2.12) dihitung menurut rumus

$$\lambda = 0,020 + \frac{0,0005}{D} \tag{2.15}$$

di mana  $D$  adalah diameter dalam pipa (m). Rumus ini berlaku untuk pipa baru dari besi cor. Jika pipa telah dipakai selama bertahun-tahun, harga  $\lambda$  akan menjadi 1,5



Gb. 2.4 Kerugian gesek pada pipa lurus (rumus Darcy).

dinginan, 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:

$$\eta_{ad} = \frac{L_{ad}}{L_s} \quad (2.20)$$

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

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

$P_s$ : Tekanan isap tingkat pertama (kgf/m<sup>2</sup> abs)

$P_d$ : Tekanan keluar dari tingkat terakhir (kgf/m<sup>2</sup> abs)

$Q_s$ : Jumlah volume gas yang keluar dari tingkat terakhir (m<sup>3</sup>/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

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

Dalam Tabel 2.7 diberikan harga-harga daya adiabatik teoritis yang diperlukan untuk mengkompresikan 1 m<sup>3</sup>/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/cm<sup>2</sup> (g) atau 8,033 kgf/cm<sup>2</sup> 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  $m = 2$ .

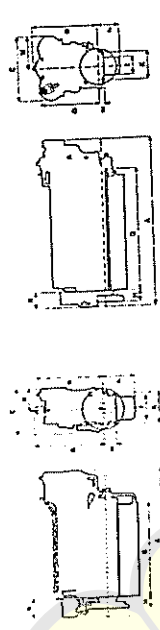
Selanjutnya efisiensi adiabatik keseluruhan dapat dihitung menurut contoh sebagai berikut. Seandainya untuk sebuah kompresor 2 tingkat yang memampatkan udara menjadi 7 kgf/cm<sup>2</sup> (g) diperlukan daya poros sebesar 5,4 kW, maka dengan daya adiabatik teoritis sebesar 4,022 kW, kompresi ini mempunyai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

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



### Wärtsilä Nohab 25

The Wärtsilä Nohab 25 medium-speed engine represents a further development of the popular F series. Incorporating experience from more than 1,800 engines, the Wärtsilä Nohab 25 is a reliable and compact high performance engine. It provides both low fuel consumption and running costs for cost-effective power. The high number of repeat orders is proof of customer confidence.



#### Principal engine dimensions (mm) and weights (tons)

Engine type	A	B	C	D	E	F Max.	G
6R25	4245	1950	1255	2070	350	555*	2700
8V25	3915	1950	1955	1950	180	710*	2380
12V25	4665	2205	1960	1960	180	710*	3220
16V25	5650	2205	2110	1960	180	710*	4060

Engine type	H. Max.	I	K	M	N	Weight
6R25	375**	650	920	650	550	9.9
8V25	375**	650	920	885	525	11.2
12V25	375**	650	920	955	525	16.6
16V25	375**	650	920	1055	525	21.1

\* Max. with wet sump. \*\* Max. Depending on flywheel size.

#### Rated power/generating sets

Engine type	Output at			
	720 rpm/60 Hz	750 rpm/50 Hz	900 rpm/60 Hz	1000 rpm/50 Hz
6R25	1110 kW	1150 kW	1380 kW	1545 kW
8V25	1470 kW	1530 kW	1770 kW	1840 kW
12V25	2210 kW	2300 kW	2650 kW	2760 kW
16V25	2940 kW	3070 kW	3530 kW	3530 kW

#### Principal generator dimensions (mm) and weights (tons)

Engine type	Length	Breadth	Height	Engine weight	Weight of gen. set
6R25	6600	1730	2950	10.9	20
8V25	6400	1920	3130	12.2	24
12V25	7600	2050	3205	17.4	33
16V25	8800	2050	3205	22.1	43

#### Principal engine specifications

Cylinder bore	250 mm
Piston stroke	300 mm
Speed	720-1000 rpm
Mean effective pressure	20.8-18.7 bar
Piston speed	7.2-10.0 m/s
FUEL SPECIFICATION:	380 cSt/50°C
Fuel oil	3500 6R/100°F

#### Output in kW/BHP at

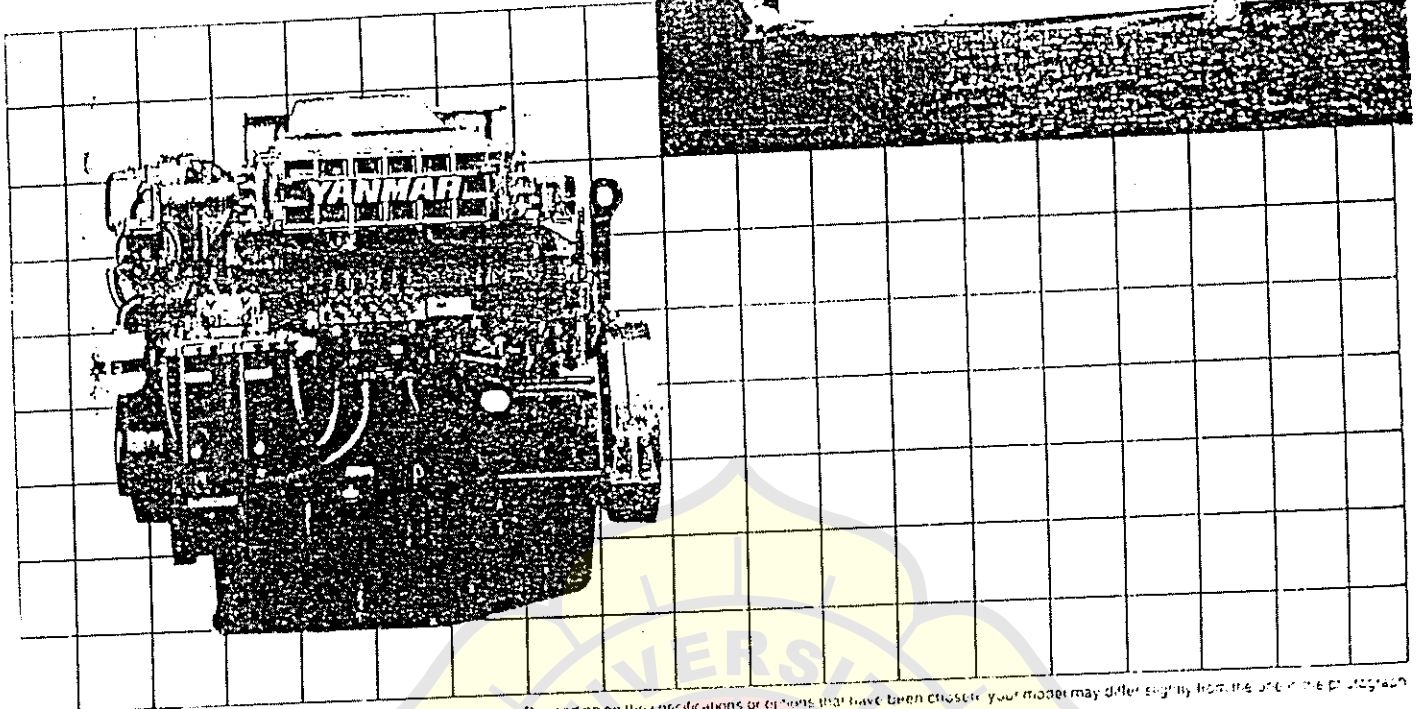
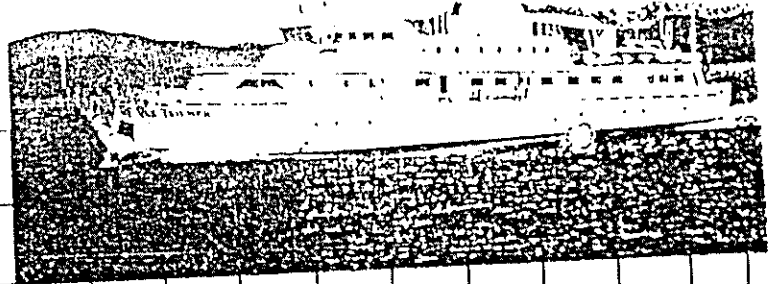
Engine type	Output in kW/BHP at					
	720 rpm	750 rpm	900 rpm	1000 rpm	900 rpm	1000 rpm
6R25	1110 kW	1150 kW	1380 kW	1545 kW	1380 BHP	1545 BHP
8V25	1470 kW	1530 kW	1770 kW	1840 kW	1840 BHP	1840 BHP
12V25	2210 kW	2300 kW	2650 kW	2760 kW	2500 BHP	2500 BHP
16V25	2940 kW	3070 kW	3530 kW	3530 kW	3680 BHP	3680 BHP

# Capacity table at varying r. p. m.

TYPE	H = Total pump head in m. N = Power consumption in hp at duty point.																																																								
	Capacity		Branches		1450 r.p.m.		1750 r.p.m.		2000 r.p.m.		2500 r.p.m.		2900 r.p.m.		3500 r.p.m.																																										
	m <sup>3</sup> /h	l/min	Suc.	Del.	H	N	H	N	H	N	H	N	H	N	H	N																																									
SA-20-90/9 Max. 6000 r.p.m.	0	0	3/4"	3/4"					4,7	0,06	7,3	0,12	9,9	0,18	14,4	0,32																																									
	1,5	25							4,4	0,08	7,0	0,14	9,7	0,21	14,2	0,38																																									
	3,0	50							3,4	0,10	6,1	0,18	8,9	0,27	13,5	0,44																																									
	4,5	75								4,2	0,21	7,1	0,30	11,7	0,50																																										
SA-25-122/12 Max. 5000 r.p.m.	0	0	1"	1"	5,0	0,14	7,3	0,23	9,5	0,35	14,6	0,72	18,4	0,9	27	1,7																																									
	2	33			4,0	0,18	6,9	0,29	9,0	0,42	13,7	0,78	17,0	1,0	26	1,8																																									
	4	66			3,0	0,20	5,5	0,33	7,7	0,48	11,5	0,79	14,6	1,1	23	1,9																																									
	6	100					3,0	0,38	5,4	0,52	8,0	0,90	10,7	1,1	18	2,0																																									
SA-35-135/12 Max. 4500 r.p.m.	0	0	1 1/2"	1 1/2"	6,25	0,16	9,1	0,28	11,7	0,4	18,6	0,9	25	1,4	36	2,2																																									
	5	83			5,0	0,21	7,7	0,35	10,4	0,5	17,0	1,0	23	1,5	34	2,5																																									
	10	166			2,7	0,28	5,3	0,48	7,9	0,6	14,4	1,1	20	1,7	31	2,0																																									
	15	250							4,3	0,7	10,3	1,3	16	1,9	27	3,3																																									
SA-50-160/8 Max. 3500 r.p.m.	0	0	2"	2"					18,0	0,8	28	1,4	38	2,7	55	4,8																																									
	6	100							16,5	1,1	26	2,1	36	3,2	52	6,3																																									
	12	200							12,5	1,4	22	2,5	32	3,6	48	8,5																																									
	18	300								14	2,7	25	4,3	42	7,5																																										
SA-50-180/8 Max. 2500 r.p.m.	0	0	2"	2"	11,5	0,7	16,4	1,2	21	1,7	33	3,3																																													
	7,5	125			9,5	0,9	14,3	1,5	19	2,2	30	4,0																																													
	15,0	250			6,6	1,0	11,2	1,7	16	2,5	27	4,5																																													
	22,5	375					5,3	1,8	11	2,8	22	5,0																																													
SA-65-250/8 Max. 3500 r.p.m.	0	0	3"	2 1/2"					39	5,7	61	10	82	15	100	22																																									
	20	333							35	7,5	57	13	76	21	94	33																																									
	40	666							31	9,3	51	16	70	25	87	45																																									
	60	1000								35	13	63	23	72	40																																										
SA-65-250/17 Max. 2200 r.p.m.	0	0	3"	2 1/2"	23	1,9	33	2,5	42	3,5																																															
	20	333			21	2,7	32	3,2	40	4,0																																															
	40	666			13	4,5	28	5,2	37	5,9																																															
	50	1000			15	5,4	25	6,1	33	7,0																																															
SA-80-160/17 Max. 4000 r.p.m.	0	0	3"	3"					14,0	1,2	22	2,5	30	4,0	44	7,0																																									
	20	333							13,0	2,1	21	4,0	28	5,6	42	9,4																																									
	40	666							9,5	2,7	18	4,6	26	7,2	39	11,4																																									
	60	1000								9	5,3	18	8,2	33	13,0																																										
SA-80-220/17 Max. 3500 r.p.m.	0	0	3"	3"	16,3	1,3	23	1,7	31	2,0	48	3,0	65	12	95	11																																									
	20	333			14,5	1,8	21	2,4	28	2,8	47	3,5	53	14	83	15																																									
	40	666			10,5	2,4	16	3,1	24	3,5	44	4,0	60	17	80	23																																									
	60	1000			3,7	3,1	11	4,7	15	6,2	33	6,5	55	19	64	31																																									
SA-100-235/23 Max. 3000 r.p.m.	0	0	4"	4"	15	1,2	23	1,6	30	2,0	47	3																																													
	40	666			15	4,5	22	7,7	29	11,3	46	21																																													
	90	1333			12	6,2	19	10,3	27	13,5	44	23																																													
	120	2000			7	8,5	14	13,8	22	18,0	39	23																																													
SA-150-260/33 Max. 2500 r.p.m.	0	0	6"	6"	17	7,6	26	12	33	19																																															
	80	1333			16	11,0	24	18	32	26																																															
	160	2666			14	14,0	22	23	29	33																																															
	240	4000			9	17,0	17	27	25	38																																															
SA-200-320/50 Max. 2000 r.p.m.	0	0	8"	8"	28	2,9	40	4,5	53	6,7																																															
	150	2500			27	3,4	39	5,7	51	8,4																																															
	300	5000			24	4,3	36	6,9	48	10,9																																															
	450	7500			19	5,9	32	9,4	44	14,4																																															
<p>As the operation figures of any pump may be changed by varying the diameter of the impeller and/or the blade width, or by changing the number of revolutions, DESMI pumps may in most cases be adapted to specific operation conditions — for instance requirements as to max. power consumption — in a manner to ensure that the high rate of efficiency is maintained. We should be pleased to forward data sheets stating the capacity area of each pump type.</p> <p>The capacities apply at 0 m suction lift, and all numbers of revolutions apply to direct coupling. In case of belt drive please apply to us for details.</p>																																																									
<p>Motor FIK # N = 2 x 10<sup>4</sup> W</p>																																																									
<p>Pipe line dimension</p> <table border="1"> <thead> <tr> <th>m<sup>3</sup>/h</th> <th>Section</th> <th>Delivery</th> </tr> </thead> <tbody> <tr> <td>2</td> <td>20 1/2"</td> <td>20 1/2"</td> </tr> <tr> <td>3,5</td> <td>25 1"</td> <td>25 1"</td> </tr> <tr> <td>5</td> <td>32 1 1/4"</td> <td>25 1"</td> </tr> <tr> <td>9</td> <td>40 1 1/2"</td> <td>32 1 1/4"</td> </tr> <tr> <td>14</td> <td>50 2"</td> <td>40 1 1/2"</td> </tr> <tr> <td>24</td> <td>65 2 1/2"</td> <td>50 2"</td> </tr> <tr> <td>38</td> <td>80 3"</td> <td>55 2 1/2"</td> </tr> <tr> <td>57</td> <td>100 4"</td> <td>80 3"</td> </tr> <tr> <td>34</td> <td>125 5"</td> <td>100 4"</td> </tr> <tr> <td>125</td> <td>150 6"</td> <td>125 5"</td> </tr> <tr> <td>225</td> <td>200 8"</td> <td>150 6"</td> </tr> <tr> <td>350</td> <td>250 10"</td> <td>200 8"</td> </tr> <tr> <td>500</td> <td>300 12"</td> <td>250 10"</td> </tr> </tbody> </table>																m <sup>3</sup> /h	Section	Delivery	2	20 1/2"	20 1/2"	3,5	25 1"	25 1"	5	32 1 1/4"	25 1"	9	40 1 1/2"	32 1 1/4"	14	50 2"	40 1 1/2"	24	65 2 1/2"	50 2"	38	80 3"	55 2 1/2"	57	100 4"	80 3"	34	125 5"	100 4"	125	150 6"	125 5"	225	200 8"	150 6"	350	250 10"	200 8"	500	300 12"	250 10"
m <sup>3</sup> /h	Section	Delivery																																																							
2	20 1/2"	20 1/2"																																																							
3,5	25 1"	25 1"																																																							
5	32 1 1/4"	25 1"																																																							
9	40 1 1/2"	32 1 1/4"																																																							
14	50 2"	40 1 1/2"																																																							
24	65 2 1/2"	50 2"																																																							
38	80 3"	55 2 1/2"																																																							
57	100 4"	80 3"																																																							
34	125 5"	100 4"																																																							
125	150 6"	125 5"																																																							
225	200 8"	150 6"																																																							
350	250 10"	200 8"																																																							
500	300 12"	250 10"																																																							
<p>Conversion Factors: 1 m = 3,28 ft 1 m<sup>3</sup> = 35,23 imp. gal.</p>																																																									

# 6KHHL

Engine output  
184-309 kW (250-420 PS)



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

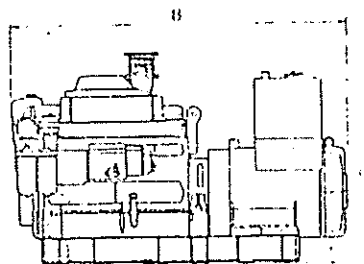
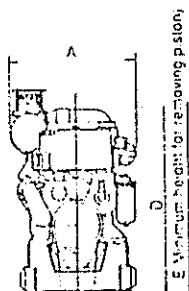
## 1 Specifications

Engine model		6KHHL-STN					
Type		Vertical water-cooled 4-cycle diesel engine					
No. of cylinders		6					
Cylinder bore × stroke		135 × 160					
Total displacement		13.74					
Continuous rated output		184 (250)	235 (320)	279 (380)	199 (270)	265 (360)	309 (420)
Engine speed		1200	1500	1800	1200	1500	1800
Net mean effective pressure		1.339 (13.65)	1.370 (13.97)	1.356 (13.83)	1.446 (14.74)	1.542 (15.72)	1.498 (15.28)
Generator capacity		160	200	240	180	240	280
Intake system		Direct injection					
Starting system		Electric starting (Air-motor starting is available)					
External dimensions	Overall length	1026					
	Overall width	991					
	Overall height	1434					
Dry weight		1320					

## Dimensions

(Units: mm)

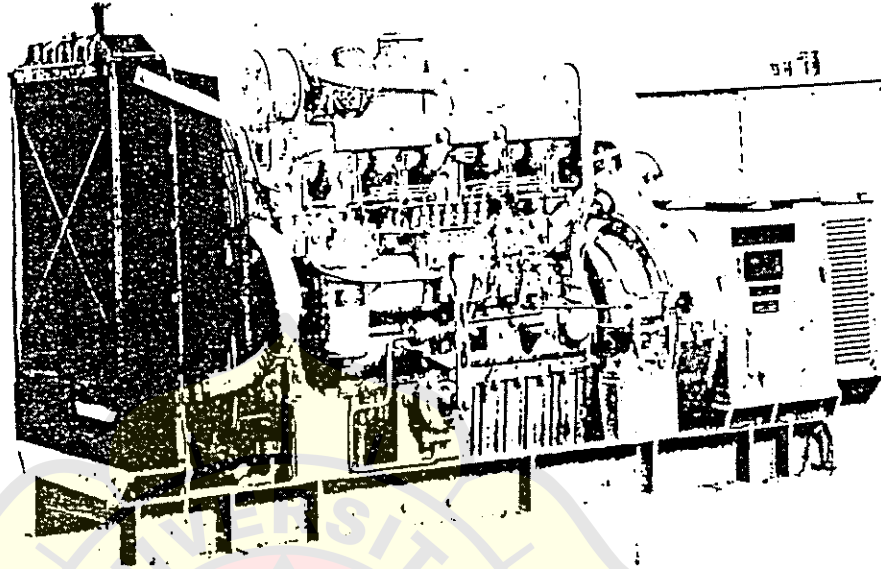
The dimensions and weights for the diesel engine generator sets are simply reference values. The values may differ for different generator manufacturers.



Engine model	6KHHL-STN
A	1026
B	2524
C	1401
D	1454
E	1394
Dry weight of generator set (reference)	2600

# Emergency-Use Marine Power Generating Sets

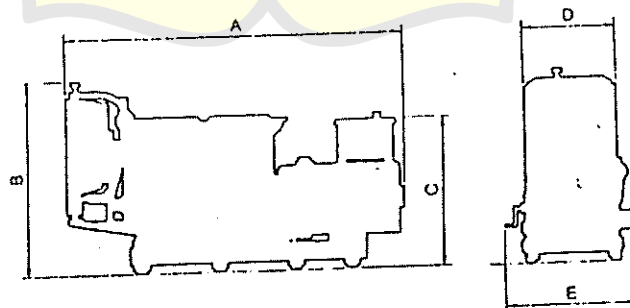
## YMGH Series



### Specifications

Model		YMGH30	YMGH40	YMGH55	YMGH80	YMGH100	YMGH120	YMGH150	YMGH200	YMGH250	
Generator	Capacity	kVA	30	40	55	80	100	120	150	200	250
	Frequency	Hz	60								
	Voltage	V	450								
	Power factor	%	80								
Diesel 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)	89 (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.



### YMGH-A Series Dimensions

(Unit: mm)

Model	A	B	C	D	E	Weight (kg)
YMGH-30A	1640	950	830	645	815	735
YMGH-40A	1680	965	830	645	815	770
YMGH-55A	1920	1050	925	690	945	940
YMGH-80A	1900	1050	960	690	945	1120
YMGH-100A	2320	1310	1050	720	845	1530
YMGH-120A	2300	1310	1155	720	845	1640
YMGH-150A	2600	1460	1210	720	1015	2110
YMGH-200A	2670	1460	1210	720	1015	2270
YMGH-250A	2725	1480	1210	720	1015	2440

### YMGH-B Series Dimensions

(Unit: mm)

Model	A	B	C	D	E	Weight (kg)
YMGH-30B	1590	965	995	645	815	765
YMGH-40B	1630	965	995	645	815	780
YMGH-55B	1800	1050	925	690	945	880
YMGH-80B	1920	1050	925	690	945	1000
YMGH-100B	2290	1310	1070	720	845	1400
YMGH-120B	2345	1310	1070	720	815	1440
YMGH-150B	2615	1460	1125	720	1015	1910
YMGH-200B	2705	1460	1205	720	1015	2190
YMGH-250B	2830	1480	1340	720	1015	2400



PROPELLER  
KW ENGINE

