BAB V
KELISTRIKAN

V.1. MOTOR BANTU

V.1.1. Perhitungan Daya Kebutuhan Listrik Kapal

Motor bantu atau generator berfungsi sebagai sumber energi untuk berbagai kebutuhan listrik di kapal, dimana kapasitas dan jumlah yang diperlukan disesuaikan dengan kebutuhan. Kebutuhan listrik tersebut antara lain :

- pompa – pompa,
- penerangan serta keperluan – keperluan lain
  sehingga kapal dapat beroperasi dengan baik.

Sumber tenaga listrik di atas kapal dapat dibagi menjadi :

- Generator
- Battery Marine Use
-

Pemakaian beban listrik pada beberapa kondisi pelayaran :

- Kondisi saat olah gerak
- Kondisi saat berlayar
- Kondisi saat sandar

Untuk memudahkan perhitungan, maka pemakaian listrik untuk pemerangan di kapal di buat pada perhitungan tersendiri di bawah ini sedangkan untuk kebutuhan listrik seluruhnya dapat di lihat pada tabel perhitngan kebutuhan listrik di bawah ini

Kategori Pemakaian Daya Listrik

	Jml	Beban		Berlayar		Manuver		Sandar	
		Unit	KW	LF	kW	LF	kW	LF	kW
MACHINERY SPACE AUX.									
Fuel Oil Transfer Pump	2	0.82	1.64	1	1.64	1	1.64	0.1	0.164
Fuel Oil Service Pump	2	0.64	1.28	1	1.28	1	1.28	0.1	0.128
Lubrication Oil Pump	2	0.40	0.80	1	0.80	1	0.80	0.2	0.16
Fresh Cooling Water Pump	2	8.45	16.9	1	16.9	1	16.9	0.2	3.2
Sea cooling Water Pump	2	8.48	16.96	1	16.96	1	16.96	0.2	3.392
Main Air Compresor	2	5.153	10.30	0	0	0	0	0.1	1.030
Bilga Pump	2	4.025	8.05	0.5	4.025	1	8.05	0.5	4.025
Ballast Pump	2	2.083	4.166	0.5	2.083	1	4.166	0.5	2.083
Fire Fighting & GS Pump	1	8.352	8.352	0	0	0	0	0	0
SANITARY									
Fresh Water Pump	2	0.70	1.4	0.3	0.42	0.2	0.28	1	1.4
Sea Water Pump	2	1.298	2.6	0.3	0.78	0.2	0.52	1	2.6
MESIN GLADAK									
Steering Gear	2	8.37	16.74	1	16.74	1	16.74	0	0
Capstan	2	7.44	14.88	0	0	0	0	1	14.88
Windlass	2	11.58	23.16	0	0	0	0	1	23.16
Boat Winches	2	4.94	9.88	0	0	0	0	0	0
UDARA PENDINGIN									
Engine Room Vent	2	3.94	7.88	1	7.88	1	7.88	1	7.88
Fan Kamar Mesin	2	6.774	13.55	1	13.55	1	13.55	1	13.55
Refrigerator	1	0.09	0.09	1	0.09	1	0.09	1	0.09
PENERANGAN									
Ruang Kemudi	3	0.06	0.18	1	0.18	1	0.18	1	0.18
Kamar Kapten	2	0.05	0.10	0.7	0.07	1	0.10	0.9	0.09

Kamar Chief Enginer	2	0.05	0.10	0.7	0.07	1	0.10	0.9	0.09
Kamar ABK I	2	0.05	0.10	0.5	0.05	0.5	0.05	0.9	0.09
Kamar ABK II	2	0.05	0.10	0.5	0.05	0.5	0.05	0.9	0.09
Kamar ABK III	2	0.05	0.10	0.5	0.05	0.5	0.05	0.9	0.09
Kamar Mesin	6	0.06	0.36	1	0.36	1	0.36	1	0.36
Ruang	2	0.04	0.08	0.5	0.04	0.5	0.04	0.9	0.072
Kamar Mandi	2	0.04	0.08	1	0.08	1	0.08	1	0.08
Kamar WC	2	0.04	0.08	1	0.08	1	0.08	1	0.08
Lampu Sisi Kanan & Kiri	4	0.06	0.24	0.5	0.12	0.5	0.12	0	0
Lampu Buritan	2	0.06	0.12	0.5	0.06	0.5	0.06	0	0
Lampu Jangkar	1	0.05	0.05	0	0	0	0	0	0
Lampu Tiang	4	0.06	0.24	0.5	0.12	1	0.24	0	0
Lampu Morse	2	0.06	0.12	0.5	0.06	1	0.12	0	0
Galley	2	0.03	0.06	1	0.06	1	0.06	0.5	0.03
Gang Way	10	0.03	0.30	1	0.30	0.7	0.21	1	0.30
Alat Navigasi	1	0.75	0.75	1	0.75	1	0.75	1	0.75
Electric Mech Washing	1	0.10	0.10	0	0	0	0	1	0.10
Rice Cooker	1	0.30	0.30	1	0.30	0.2	0.06	0.5	0.15
TOTAL					85.948		94.146		80.295

Jumlah dari daya yang dipergunakan adalah :

- Kondisi saat berlayar = 85.948 kW
- Kondisi saat Manuver = 94.146 kW
- Kondisi saat sandar = 80.295 kW

V.1.2. Perencanaan Perhitungan Generator

Untuk memenuhi kebutuhan listrik tersebut maka akan direncanakan generator yang mampu memenuhi kebutuhan listrik terbesar yaitu pada waktu kapal melakukan olah gerak serta pertimbangan pada besarnya efisiensi dari generator :

Dimana :

Pmaks. = 94,146 kW (Daya maksimum dari pemakaian beban)

P_{gen} = Efisiensi generator = 80% = 0,8

$$P_{gen} = \frac{\text{Out put}}{\text{In put}} \times 0{,}8$$

$$= \frac{94.146}{\text{In put}} \times 0.8$$

$$= \frac{94{,}146}{0{,}8} = 117.68 \text{ kW}$$

Direncanakan pemakaian generator sebanyak 2 buah.

Masing – masing generator direncanakan mempunyai kapasitas daya yang sama, daya harus di tambah dengan daya yang di perlukan, sebanyak 10 % dari daya yang di dapat dari hasil perhitungan, di karenakan untuk mensupply daya yang tidak terhitung di dalam perhitungan perancangan ini. Antara lain adalah :

$$P_{gen} = 117{,}68 \text{ kW.}$$

$$= 117{,}68 + 11{,}768$$

$$= 129{,}45 \text{ kW}$$

Dari katalog yang ada dipilih generator dengan spesifikasi sebagai berikut :

Merk	:	YANMAR
Type	:	4 HAL 2 – TN
Cylinder	:	4
P generator Minimal	:	129,45 kW.
	:	135 kW.
Putaran Mesin	:	1800 Rpm
Cylinder (Bore x Stroke)	:	130 x 165
Berat	:	1030 Kg
Jumlah Generator	:	2 set

V.1.3. Battery Darurat

Battery darurat digunakan hanya untuk keperluan-keperluan tertentu / darurat.

Keperluan-keperluan tersebut adalah :

-	Peralatan Nautika	:	0,75	kW
-	Peralatan Radio dan Komunikasi	:	0,30	kW
-	Penerangan Darurat	:	0,25	kW
-	Gang way	:	0,30	kW
	Jumlah	:	1,60	kW

Dengan demikian lamanya penggunaan Battery dalam keadaan darurat adalah :

H = 2 x 200 AH / Total Pemakaian

= 2 x 200 / 1,60

= 2,5 jam

Daftar Pustaka

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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_2O.

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,

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

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

where H = the actual head created by the pump, mH_2O.

The mechanical efficiency, η_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

$$Q_b = 0.2825 d_b^2 v_b \text{ cu m per hr} \qquad (1-1)$$

where d_b = 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	70	360 to 480	140
40 to 75	80	480 to 620	150
75 to 120	90	620 to 800	160
120 to 190	100	800 to 1000	175
190 to 265	110	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

$$Q_b = 0.565 d_b^2 \text{ cu m per hr.} \qquad (1-2)$$

Because of water leakages this calculated capacity must be increased by 5 or 10 per cent. Ballast pump capacities range from 10 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_2O;

(2) medium-pressure fans developing a head up to 300 mmH_2O;

(3) high-pressure fans developing a head up to 1,500 mmH_2O.

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_2O, at normal atmospheric conditions and the fan wheel speed, n rpm, at the highest efficiency:

$$n_s = \frac{n\sqrt{Q}}{\sqrt[4]{H^3}}$$

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

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

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

V_r = volume of the room, cu m

$V_{mr} \approx 1$ = 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

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

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

t_r = given temperature of the room, °C

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

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

γ_{fa} = 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

$$Q_{hu} = \frac{100\, D_{hu}}{\varphi_r d_r - \varphi_{fa} d_{fa}} \text{ cu m per hour} \tag{275}$$

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

d_r and d_{fa} = absolute humidity of saturated air at the room temperature, t_r, and at the temperature, t_{fa}, of the entering air, g per cu m (see Table 38)

φ_r and φ_{fa} = 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	
	Plenum 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	15	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 $\varphi_{st}=50$ 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

$$Q_{st}=\frac{(1+\alpha t_{st})\,p_a Q_a}{p_{st}(1+\alpha t_a)}=Q_a\frac{\left(1+\frac{1}{273}20\right)}{\left(1+\frac{1}{273}t_a\right)}\frac{p_a}{760}=$$

$$=Q_a\frac{293}{273+t_a}\frac{p_a}{760}\text{ cu m per hour} \qquad (276)$$

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

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

where γ_{air} = density of air, kg per cu m
$\gamma_{wat}=1{,}000$ = density of water, kg per cu m
ρ = mass density of air, kg-sec² per m⁴.

Upon radial entry of the air onto the fan impeller vanes

$$H_{t\infty}=\rho c_{2u}u_2\text{ mmH}_2\text{O}$$

Taking into account the effect of having a finite number of impeller vanes on the developed head by the factor σ and for the losses of head in the fan by the hydraulic efficiency η_h we obtain the actual head

$$H=H_{t\infty}\sigma\eta_h=\sigma\rho c_{2u}u_2\eta_h=\sigma\rho\frac{c_{2u}}{u_2}u_2u_2\eta_h=$$
$$=\sigma\rho\varphi_h u_2^2\eta_h=\rho\psi_h u_2^2\text{ mmH}_2\text{O} \qquad (278)$$

where $\varphi_h=\frac{c_{2u}}{u_2}$ = eddy current factor

$\psi_h=\sigma\varphi_h\eta_h$ = 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

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

where ΔH = loss of head in the fan.

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

$$\eta_{fr} = \frac{N_{fr}}{N_a} = \frac{\beta 10^{-6} \rho D_2^2 u_2^3}{N_a}$$

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

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

b_2 = width of the impeller at air outlet

D_2 = impeller diameter at air outlet

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

For forward-curved vanes—$\eta_r \cong 0.75$ to 0.9.

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

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

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

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

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

2-2. Design and Selection of Fans

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

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

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

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

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

where v = mean velocity in the discharge connection of the fan.

On the basis of the discharge per second, Q_s, head, H, and speed, n, we next determine the specific velocity of the fan.

The specific velocity of a fan is a value that relates the air discharge, Q_{st} cu m per sec, the total head, H mmH$_2$O, and the impeller speed, n, at maximum efficiency:

$$n_s = \frac{n\sqrt{Q_s}}{\sqrt[4]{H^3}} \qquad (281)$$

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

$$\overline{Q}_k = \frac{Q_s}{Fu_2}$$

and

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

where F = area of the impeller, sq m

D_2 = outside diameter of the impeller, m.

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

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

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

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

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

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

$$H = \overline{H}_k u_2^2 \rho \text{ mmH}_2\text{O}$$
$$H_{st} = \overline{H}_{kst} u_2^2 \rho \text{ mmH}_2\text{O} \qquad (282)$$

difference in pressures in the chambers will cause the vanes to turn clockwise.

As soon as the helmsman stops turning the wheel the pressure in the system drops, valve *41* is returned to its central position by spring *44* and the rudder comes to rest.

In cases when the rudder is operated by emergency steering facilities (quadrants, rudder tackle, etc.), compression of the liquid in the chambers is prevented by opening the relief-bypass valve *33* by its spindle *45*.

The interaction of the parts of this steering gear for counter-clockwise rotation of the rudder can be followed out in Fig. 158.

4-4. Determining the Principal Data Required in the Design of Steam and Electric Steering Gears

The main initial data required to determine the principal dimensions of steering gears are the rudder characteristic, χ_r, the torque, M_{rs}, in kg-m developed on the rudder head and the time, τ, required to put over the rudder.

The time required to put the rudder from hard-over to hard-over, depending upon the purpose of the ship and used in steering gear design, is listed in Table 47. It should not exceed the standards established by the U. S. S. R. Shipping Register.

The time that elapses before the steering engine reaches its rated speed, which we shall call the starting time, must be taken into consideration by reducing the time τ for putting the rudder from hard-over to hard-over by 1.5 to 2 seconds.

If we denote the gearing ratio between the rudder stock and steering engine shaft as i_{sg}, the overall efficiency of the steering gear as η_{sg} and the speed at which the rudder stock turns,

Table 47

Type of ship	Time required to put rudder from hard-over to hard-over, sec	Speed of rudder movement, deg/sec, for rudder angle of	
		$2\alpha°=70°$	$2\alpha°=64°$
Ice breakers	15	4.66	4.25
Sea-going craft and transport ships . . .	25 to 30	2.8 to 2.34	2.56 to 2.13
Towboats	20 to 25	3.5 to 2.8	3.2 to 2.56
River craft	40 to 45	1.75 to 1.55	1.6 to 1.44

expressed in rpm, as n_{rs}, then the torque developed on the steering engine shaft and its speed, n_m rpm, will be

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

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

where n_m = 100 to 350 rpm for steam engines

n_m = 300 to 1,800 rpm for electric motors.

The angular velocity of rotation ω_{rs} of the rudder stock can be calculated from the following formulas:

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

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

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

It follows from formula (314) that

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

Combining equations (315) and (316) we obtain

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

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

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

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

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

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

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

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

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

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

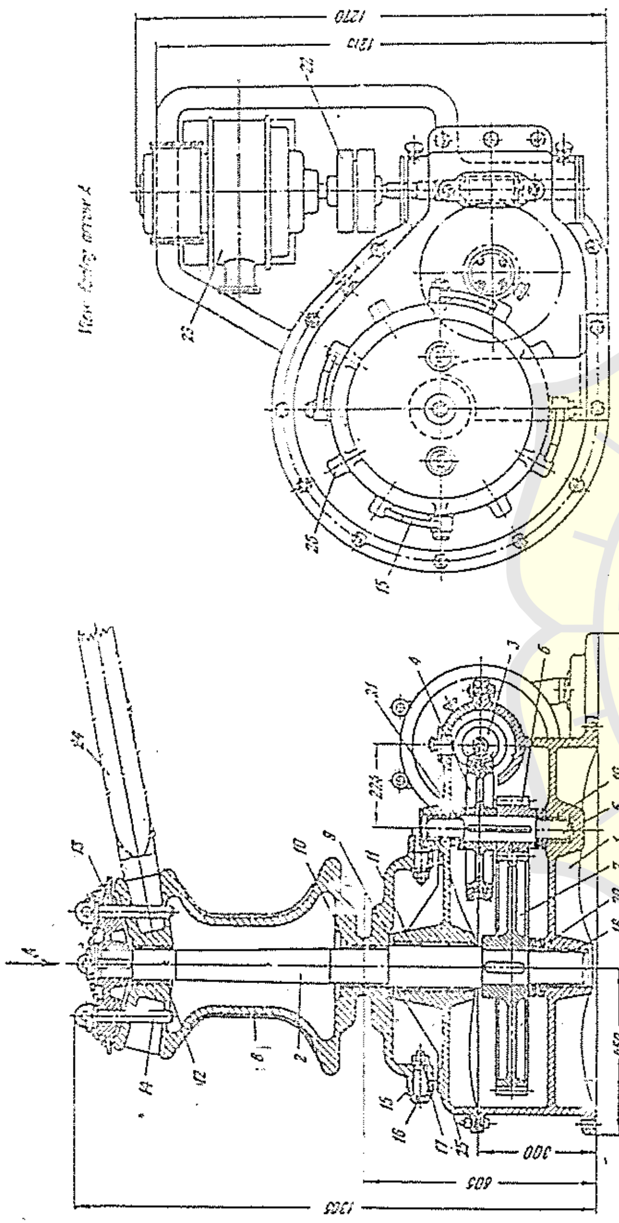


Fig. 169.

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
γ_a = 7,750 = density of the material of the anchor, kg per cu m
γ_w = 1,025 = density of sea water, kg per cu m
f_h = 1.28 to 1.35 = a factor taking into account the friction losses in the hawse hole and stopper.

The required pull of the cable lifter to hoist two anchors is

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

In hoisting one anchor

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

The following empirical formulas can be derived from a comparison of the weights of anchors and the size of their chains as stipulated by the U.S.S.R. Shipping Register, as well as the U.S.S.R. Standard on anchor chain:

The chain bar size $d_c \approx \sqrt{G_a}$ mm. The weight per running metre of anchor chain is

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

According to the U.S.S.R. Shipping Register the aft anchoring arrangement, usually consisting of a capstan, must break away the anchor and heave it in at a speed of at least 9 m per min.*

* In breaking away one anchor from the bottom

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

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

$$T_w = \frac{R_{br}}{6} \tag{385}$$

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:

$$X = L(B + H) + \Sigma\chi_i \tag{386}$$

where L = length of the ship at the summer load line, m

B = maximum breadth between the outer edges of the ship's hull, m

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

$\Sigma\chi_i$ = correction factor taking into account the sail effect of the superstructures.

Correction factors for the sail effect of the superstructures having a height h_i and length l_i consist of:

(a) correction factors for the superstructures of the forecastle, poop and midships, each having a length l_{sp} and height h_{sp}:

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

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

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

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

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

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

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

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

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

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

$$\chi_q = l_qh_q$$

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

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

Table 59

Self-Propelled Transport Ships with an Unlimited Region of Navigation

No.	Characteristic X	Anchors: Bower: Quantity	Anchors: Bower: Total weight, kg	Anchors: Stream anchor, kg	Chain cable for bower anchors: Total length of two cables, m	Chain cable for bower anchors: Anchor chain size, mm	Chain or steel rope for the stream anchor: Length, m	Chain or steel rope for the stream anchor: Anchor chain size, mm	Chain or steel rope for the stream anchor: Diameter of steel rope, mm
1	50	2	150	25	100	12	50	—	8.8
2	75	2	200	25	125	13	50	—	8.8
3	100	2	250	50	125	15	50	—	11
4	150	2	300	50	150	16	50	—	11
5	200	2	350	50	175	17	75	—	11
6	250	2	450	75	200	18	75	11	13
7	300	2	500	75	225	19	75	13	13
8	350	2	600	100	250	20	75	14	15.5
9	400	2	700	100	275	21	75	14	15.5
10	450	2	750	125	300	22	100	15	17.5
11	500	2	800	150	300	24	100	16	17.5
12	550	2	900	175	325	25	100	16	17.5
13	600	3	1500	200	350	27	100	17	17.5
14	650	3	1700	225	350	28	100	18	19.5
15	700	3	1800	250	375	29	100	18	20.5
16	750	3	2100	250	375	30	100	19	20.5
17	800	3	2250	250	375	31	125	19	20.5
18	850	3	2400	275	375	32	125	20	22
19	900	3	2700	300	375	33	125	21	24
20	950	3	3000	300	400	34	125	21	24
21	1000	3	3200	350	400	36	125	22	24
22	1100	3	3500	400	400	37	125	23	26
23	1200	3	3750	400	420	38	150	25	26
24	1300	3	4100	450	450	40	150	25	28
25	1400	3	4250	450	450	41	150	25	28
26	1500	3	4500	500	450	42	150	26	28
27	1600	3	4750	500	450	43	150	26	28
28	1700	3	5250	600	450	45	150	28	30
29	1850	3	5500	600	450	46	150	28	30
30	2000	3	5750	700	450	46	150	29	31.5
31	2150	3	6000	700	475	48	175	29	31.5
32	2300	3	6500	800	500	49	175	29	32.5
33	2500	3	6750	800	500	50	175	29	32.5
34	2700	3	7500	900	500	52	175	30	33.5

Continued

No.	Characteristic X	Anchors: Bower: Quantity	Anchors: Bower: Total weight, kg	Anchors: Stream anchor, kg	Chain cable for bower anchors: Total length of two cables, m	Chain cable for bower anchors: Anchor chain size, mm	Chain or steel rope for the stream anchor: Length, m	Chain or steel rope for the stream anchor: Anchor chain size, mm	Chain or steel rope for the stream anchor: Diameter of steel rope, mm
35	3000	3	825[illegible]	1000	500	53	200	31	33.5
36	3300	3	90[illegible]	1000	500	55	200	31	33.5
37	3600	3	975[illegible]	1250	525	57	210	33	34.5
38	3900	3	105[illegible]	1250	550	59	225	33	34.5
39	4200	3	11[illegible]	1400	550	61	225	34	37
40	4500	3	115[illegible]	1500	550	62	225	35	37
41	4800	3	12[illegible]	1650	550	65	225	36	—
42	5100	3	135[illegible]	1750	550	67	250	37	—
43	5400	3	145[illegible]	1750	575	66	250	37	—
44	5800	3	150[illegible]	2000	600	70	250	40	—
45	6200	3	155[illegible]	2000	600	72	250	40	—
46	6600	3	16[illegible]	2250	600	74	275	43	—
47	7000	3	17[illegible]	2250	600	76	275	43	—
48	7400	3	18[illegible]	2250	600	77	275	44	—
49	7800	3	195[illegible]	2500	600	80	275	46	—
50	8200	3	20[illegible]	2700	600	82	275	48	—
51	8600	3	21[illegible]	2800	600	83	275	49	—
52	9000	3	22[illegible]	3000	600	85	275	50	—
53	9500	3	23[illegible]	3000	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 600 or larger.

pentagon. If the bar size of the anchor chain cable is denoted as d_c mm, then the chain pitch equal to $8\,d_c$ 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

$$D_{cl} = 2R_{cl} = 2\frac{4\,d_c}{\sin\alpha'} = \frac{8d_c}{\sin\frac{360°}{10}} = 12.6\,d_c \text{ mm} = 0.013 d_c \text{ m} \quad (357)$$

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

$$l_c = 5t_c = 5\times 8d_c = 40d_c \text{ mm} = 0.04 d_c \text{ m} \quad (358)$$

where d_c = chain bar size, mm.

Mooring and Warping Ropes

Table 60

Characteristic	Towing rope: Length, m	Towing rope: Circumference of hemp rope, mm	Towing rope: Diameter of steel rope, mm	Warping hawsers: Total length, m	Warping hawsers: Number of ropes	Warping hawsers: Circumference of hemp rope, mm	Warping hawsers: Diameter of steel rope, mm	Cable warps: Total length, m	Cable warps: Number of ropes	Cable warps: Circumference of hemp rope, mm	Cable warps: 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	9.5	—	—	—	—
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	80	1	100	12
450	135	150	17.5	180	2	125	15	80	1	100	12
500	135	150	17.5	200	2	125	15	85	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: Length, m	Towing rope: Circumference of hemp rope, mm	Towing rope: Diameter of steel rope, mm	Warping hawsers: Total length, m	Warping hawsers: Number of ropes	Warping hawsers: Circumference of hemp rope, mm	Warping hawsers: Diameter of steel rope, mm	Cable warps: Total length, m	Cable warps: Number of ropes	Cable warps: Circumference of hemp rope, mm	Cable warps: 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	39	640	4	250	26	200	2	200	21.5
3600	240	375	39	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
4500	240	425	43.5	720	4	250	26	200	2	225	24
4800	240	425	43.5	720	4	250	26	200	2	225	24
5100	240	—	52	720	4	275	28	240	2	225	24
5400	240	—	52	800	4	275	28	240	2	250	26
5800	240	—	52	800	4	275	28	240	2	250	26
6200	240	—	57	960	6	300	30	240	2	250	26
6600	240	—	57	960	6	300	30	240	2	250	26
7000	240	—	57	960	6	300	30	240	2	250	26
7400	240	—	57	960	6	300	30	480	4	250	26
7800	240	—	57	960	6	300	30	480	4	250	26
8200	240	—	61.5	960	6	300	30	480	4	250	26
8600	240	—	61.5	960	6	325	32	480	4	250	26
9000	240	—	61.5	960	6	325	32	480	4	250	26
9600	240	—	61.5	960	6	325	32	480	4	250	26

Notes: 1. If the actual characteristic is 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 for nonpropelling vessels 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 (tugs) 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 fivefold 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_{cl}, in rpm, of the cable lifter from the equation

$$l_c n_{cl} = 60\, v_a$$

(a) for windlasses and capstans of bower anchors:

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

(b) for the stern anchoring capstan:

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

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

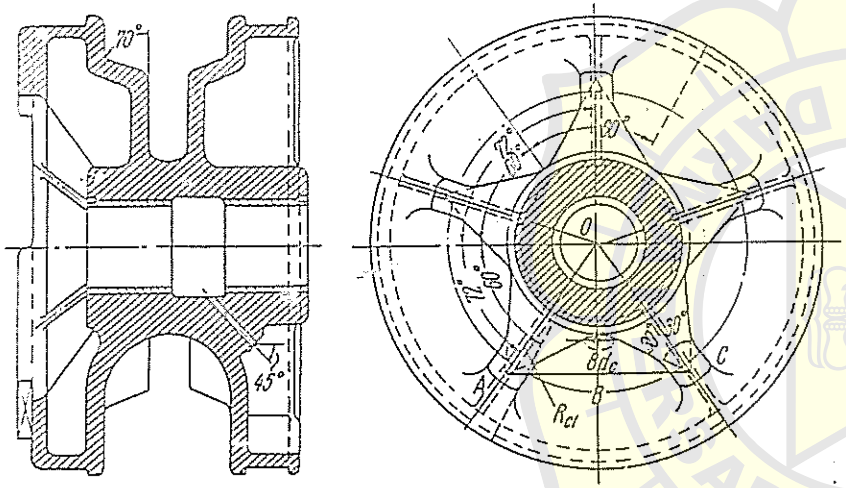


Fig. 170.

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

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

where η_{cl}, η_{sh}, η_{pg}, η_{wg} = efficiencies of the cable lifter, shaft bearings, pairs of spur gears and worm gearing

a and c = number of shaft bearings and pairs of spur gears.

The torque on the cable lifter is

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

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

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

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

Table 61

Anchor-handling gear	Motive unit shaft speed n_m, rpm	Gearing ratio of mechanism, i_a
Hand-operated capstans		4 to 40
Steam capstans	180 to 320	18 to 60
Electric capstans	800 to 1450	110 to 200
Hand-powered windlasses		9 to 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

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

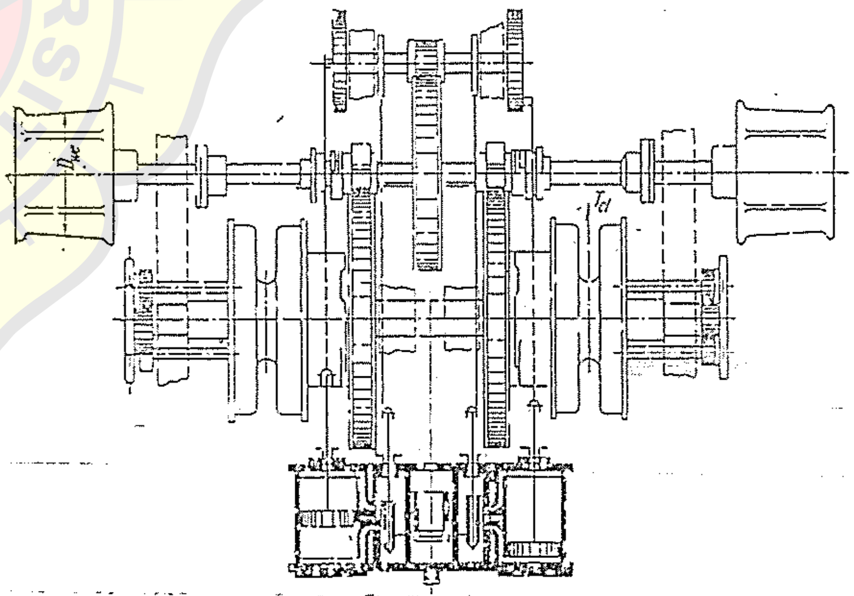


Fig. 171.

The mean shaft power of the motive unit should be

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

The mean indicated power is

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

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

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

where M_m = torque developed on the shaft of the engine, kg-cm
ψ_a = 0.85 to 1.7 = cylinder ratio, i.e., $S : D_{ca}$.

The value of $(\alpha_i k_i p_{is} - p_{ss})$ 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

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

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

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_a is the total gearing ratio of the transmission in the anchoring mechanism, then

$$M_m = \frac{M_{cl}}{\eta_a i_a} = \frac{T_{cl} D_{cl}}{2 \eta_a i_a} \quad \text{kg-cm}$$

Combining the last two equations we obtain

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

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

The diameter of the warp ends is taken equal to

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

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

where d_w = diameter of the warping hawser.

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

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

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

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

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

The pulling force developed on the warp end is

$$T_{we} = \frac{M_{we}}{\frac{1}{2}(D_{we} + d_w)} = \frac{2 M_m \eta_w i_w}{D_{we} + d_w} \leqslant \frac{R_{br}}{6} \qquad (394)$$

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

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

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

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_n = 0.9 to 1.1 = coefficient of nonequal distribution of the movable load due to the weight of the persons in the boat.

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

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

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

$$\eta_f = \frac{1}{m\varepsilon^m}\,\frac{\varepsilon^m - 1}{\varepsilon - 1} = \text{efficiency of the boat's falls}$$

ε = coefficient depending upon the ratio of the block diameter to the tackle fall diameter ($\varepsilon = 1.1$ for a hemp fall and $\varepsilon = 1.04$ to 1.06 for a steel wire rope)
η_r = 0.9 to 0.97 = efficiency of the davit guide roller
η_s = 0.9 to 0.97 = efficiency of the snatch-block
a = maximum number of blocks between the davit guide roller and the winch head.

The tension at the end of a rope that has run over the minimum number of blocks is

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

where c = minimum number of blocks.

The diameter, d_f, of a hemp fall is selected according to the breaking strength $(T_{max} + T_{min})\,6 \leqslant R_{br}$ as a function of the boat length from Table 63 (U.S.S.R. Shipping Register).

Table 63

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

The winch head diameter is

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

The speed, v_b, with which the boats are hoisted and lowered is assigned so that if the ship is rolling one of these operations can be carried out during the rolling period. This can be accomplished if the heaving-in speed is greater than the vertical component of the crest speed of waves running along the length of the ship. It has been established that the boat hoisting speed must be at least $v_b = 0.15$ m per sec under these conditions. The heaving-in speed of the tackle fall when single-sheave blocks are used must in this case be $v_f = 0.3$ m per sec.

The required winch head speed is found from the equation

$$\pi\,(D_h + d_f)\,n_h = 60v_f$$

$$n_h = \frac{60v_f}{\pi\,(D_h + d_f)} = 19.1\,\frac{v_f}{D_h + d_f} \quad \text{rpm}$$

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

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

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

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

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

$$M_h = \frac{T\,(D_h + d_f)}{2}$$

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

$$M_{mt} = \frac{M_h}{\eta_{bw} i_{bw}} = \frac{T\,(D_h + d_f)}{2\eta_{bw} i_{bw}}$$

and

$$N_e = \frac{M_{mt} n_m}{716.20} \quad \text{metric hp}$$

The cylinder diameter and indicated power of steam boat winches are determined from the same Posdyunin formulas used in maximum pressure, p_f kg per sq m, then the amount of liquid pumped is

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

This equation can be solved for V_e and V_f:

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

and

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

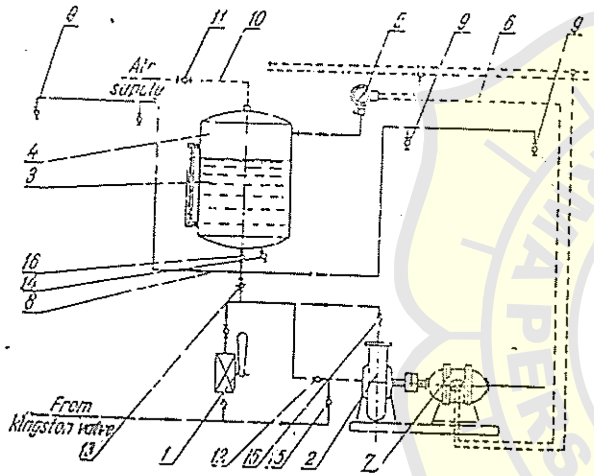


Fig. 189.

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

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

Therefore the minimum and maximum volumes of the air are

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

Denoting by V_0 the volume of liquid remaining in the tank at the lowest level, we find that the volume of the pneumatic tank is

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

Such tanks may also be used in the drinking and washing water systems.

(D) SANITARY AND SCUPPER SYSTEMS

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

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

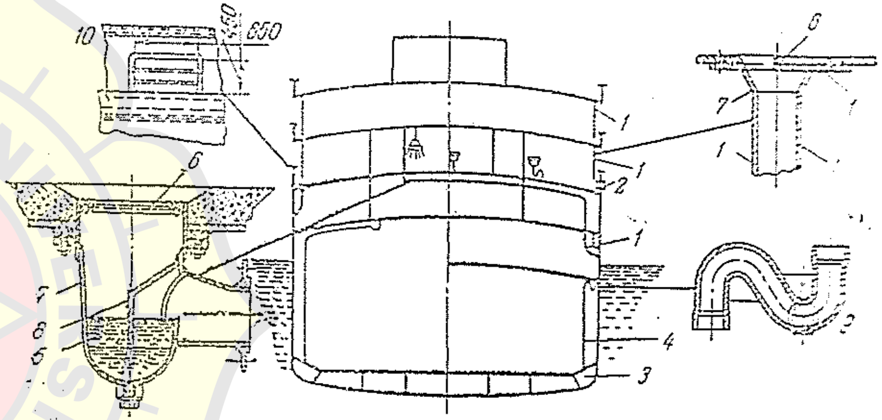


Fig. 190.

pipes until it reaches the last open deck above the load waterline from where it is discharged overboard through deck scuppers *2*. Large amounts of water drain from open decks through freeing ports *10* installed in the bulwarks.

Water is drained from decks located lower than the load waterline through scupper pipes *4* into bilge courses *3* or into dirty water tanks arranged in the double-bottom or side spaces from where it is discharged overboard by pumps.

Scuppers *7* with grates *6*, cowls *8* and sumps *5* avoid clogging of the scupper pipes. S-traps *9* are provided in scupper pipes which drain water from closed compartments to prevent the odour of the sewage spaces from getting into the compartments.

Shipside outlets of scupper pipes serving closed compartments are fitted with swing-check valves to exclude sea water in rough weather.

Sanitary pipelines made of galvanized pipe must be laid with a grade of at least 0.05 to ensure reliable water flow.

Section 4

Main Shafting

A. General

1. Scope

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

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

2. Documents for approval

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

B. Materials

1. Approved materials

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

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

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

2. Testing of materials

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

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).

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

d	[mm]	required outside diameter of shaft
d_i	[mm]	diameter of shaft bore, where present. If the bore in the shaft is ≤ 0,4 . d, the expression $1 - \left(\frac{d_i}{d_a}\right)^4 = 1,0$ may be applied
d_a	[mm]	actual shaft diameter
P_W	[kW]	rated power transmitted by shaft

[1]) For ships flying Indonesian flag in quadruplicate, one of which intended for the Indonesian Government.

n [Rpm] rated shaft speed

F [-] factor for the type of propulsion installation:

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

= 100
for all other propulsion installations

b) Propeller shafts
= 100
for all types of installations

C_w [-] material factor

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

R_m [N/mm²] Tensile strength of the shaft material (see also B.1)

k [-] Factor for the type of shaft

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

=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 of 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 shrunk 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 or 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

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 manoeuvering 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 appropriable, 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:

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

where

J	[dm^3]	total capacity of the starting air receivers
D	[mm]	cylinder bore
H	[mm]	stroke
V_h	[dm^3]	swept volume of one cylinder (in the case of double-acting engines, the swept volume of the upper portion of the cylinder)
$p_{e,perm}$	[bar]	maximum permissible working pressure of the starting air receiver
z	[-]	number of cylinders
p_{ee}	[bar]	mean effective working pressure in cylinder at rated power

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

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

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

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

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

c = 1, where $P_{e,perm}$ = 30 bar

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

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

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

If a pressure-reducing valve is fitted, which reduces the pressure $p_{e,perm}$ to the starting pressure P_A, then the value of "c" shown in Fig. 2.14 is to be used.

The following values of n_A are to be applied:

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

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

n_o	[Rpm]	= rated speed

2. Starting air for installations with non reversible engines

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

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

1.6 Pipe connections

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

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

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

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

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

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

2. Calculation of pipe diameters

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

2.2 Dry cargo and passenger ships

a) main bilge pipes

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

b) branch bilge pipes

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

where

d_H [mm] calculated inside diameter of main bilge pipe

d_z [mm] calculated inside diameter of branch bilge pipe

L [m] length of ship between perpendiculars

B [m] moulded breadth of ship

H [m] depth of ship to the bulkhead deck

l [m] length of the watertight compartment

2.3 Tankers

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

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

where:

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

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

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

2.4 Minimum diameter

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

2.5 Maximum diameter

The diameter of the main bilge line calculated according to 2.2 a) need not exceed ND 200.

2.6 Deviations

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

3. Bilge pumps

3.1 Capacity of bilge pumps

Each bilge pump must be capable of delivering:

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

where:

Q [m^3/h] minimum capacity

d_H [mm] calculated inside diameter of main bilge pipe

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

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

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

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

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

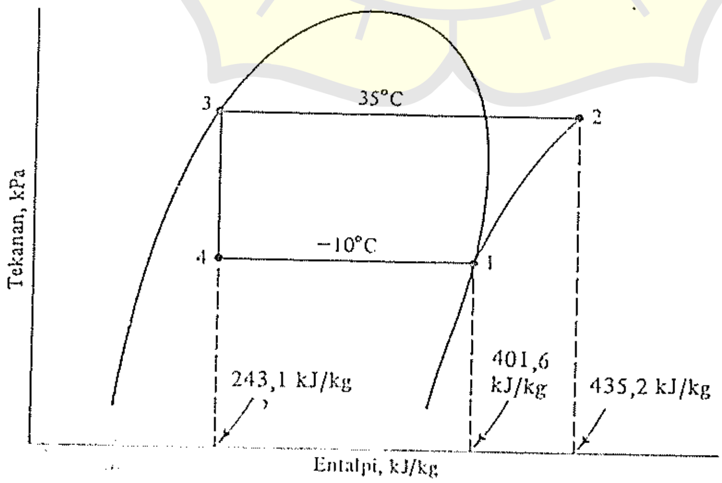
Nilai h_3 dan h_4 identik, dan sama dengan entalpi cairan jenuh pada 35°C, yaitu 243,1 kJ/kg. Sehingga

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

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

(*a*) Dampak refrigerasi:

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



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

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

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

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

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

(*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 m^3/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°C.

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

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

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

Dibandingkan dengan daur kompresi uap standar, sistem yang menggunakan penukar kalor nampaknya lebih memiliki keuntungan yang jelas karena naiknya dampak refrigerasi. Kapasitas dan koefisien prestasi tampaknya 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

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^2 and a minimum tensile strength of less than 400 N/mm^2 or more than 900 N/mm^2 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^2. If material is used having a R_{eH} differing from 235 N/mm^2, the material factor k_r is to be determined as follows:

$$k_r = \left[\frac{235}{R_{eH}}\right]^{0,75} \qquad \text{for } R_{eH} > 235\ N/mm^2$$

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

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

4.3 Before significant reductions in rudder stock diameter due to the application of steels with R_{eH} exceeding 235 N/mm^2 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 bearings.

4.4 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.

5. Definitions

C_R = rudder force in [N]

Q_R = rudder torque in [Nm]

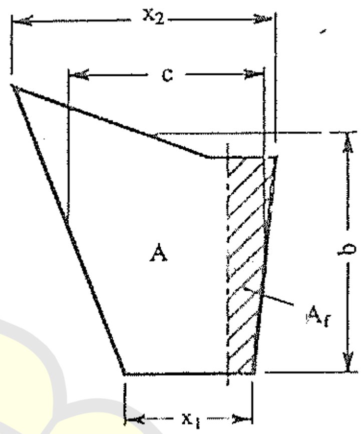
A = total movable area of the rudder in $[m^2]$
For nozzle rudders, A is not to be taken less than 1,35 times the projected area of the nozzle;

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

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

b = mean height of rudder area in [m]

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



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

Fig. 14.1

Λ = aspect ratio of rudder area A_t

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

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

B. Rudder Force and Torque

1. Rudder force and torque for normal rudders

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

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

v = v_0 for ahead condition

v = v_a for astern condition

κ_1 = coefficient, depending on the aspect ratio Λ

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

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

κ_3 = coefficient, depending on the location of the rudder

κ_3 = 0,8 for rudders outside the propeller jet

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

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

κ_t = coefficient depending on the thrust coefficient c_t

κ_t = 1,0 normally

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

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

Table 14.1

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

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

Q_R = $C_R \cdot r$ [Nm]

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

α = 0,33 for ahead condition

α = 0,66 for astern condition (general)

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

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

α = 0,25 for ahead condition

α = 0,55 for astern condition.

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

k_b = balance factor as follows:

k_b = A_f/A

k_b = 0,08 for unbalanced rudders

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

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

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

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

The resulting force of each part may be taken as:

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

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

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

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

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

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

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

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

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

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

C_1 = A_1/b_1

$C_2 = A_2/b_2$

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

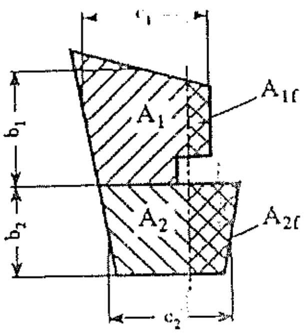


Fig. 14.2

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

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

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

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

for ahead condition

The greater value is to be taken.

C. Scantlings of the Rudder Stock

1. Rudder stock diameter

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

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

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

The related torsional stress is:

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

k_r see A.4.2.

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

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

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

2. Strengthening of rudder stock

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

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

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

Bending stress:

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

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

Torsional stress:

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

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.

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

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

It is assumed that under normal circum- nces a ship will use only one bow anchor and in cable at a time.

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

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

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

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

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

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

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

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

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

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

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

B. Equipment numeral

The equipment numeral is to be calculated as follows:

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

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

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

h = $f_b + \sum h'$

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

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

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

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

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

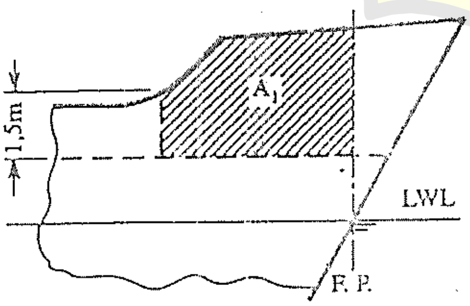


Fig. 18.1

C. Anchors

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

Guidance

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

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

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

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

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

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

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

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

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

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

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

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

D. Chain Cables

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

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

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

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

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

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

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

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

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

E. Chain Locker

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

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

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

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

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

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

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

F. Mooring Equipment

1. Ropes

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

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

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

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

1.4 Wire ropes shall be of the following type:

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

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

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

- 6 x 19 with 1 steel core
  type: Seale

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

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

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

Table 18.1

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

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

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

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

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

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

mooring line, however, should have a breaking load of less than 500 kN.

.2 The number of mooring lines may be reduced with corresponding increase of the breaking load of the individual mooring lines, provided that the total breaking load of all lines aboard ship is not less than the rule value specified in Table 18.2, however, the number of lines should not be less than 6.

2 Mooring winches, bollards, hawses

2.1 Mooring winches are to be designed taking into account the actual mooring lines and their nominal breaking loads.

2.2 Hawses, bollards and cleats shall be so designed as to protect the ropes against excessive wear. They are to be of proved construction and shall comply with relevant standards. Attention is drawn to relevant national standards.

3. Equipment for mooring at single point moorings

3.1 Upon request from the owner, BKI is prepared to certify that the vessel is specially fitted for compliance with Sections 2.1, 4.2 and 6. of the "Standards for equipment employed in the mooring of ships at single point moorings" published by the Oil Companies International Marine Forum (OCIMF).

3.2 The certificate may be issued if

.1 plans showing the equipment and the arrangement as well as necessary substructures are submitted for approval;

.2 the chain stopper, Smith bracket, or other device for securing the chafing chain to the ship and the structure to which it is attached are capable of withstanding a load not less than the breaking strength of the chain corresponding to the size of the ship as given in Section 6 of the standards stipulated in 3.1 above and calculations to demonstrate this capability are submitted;

.3 the chain bearing surface of the bow fairleads described in 6.1 of the standard stipulated in 3.1 above have a diameter at least seven times that of the associated chain;

.4 the installation on board the ship is surveyed by BKI - surveyor.

Table 18.2 Anchor, Chain Cables and Ropes

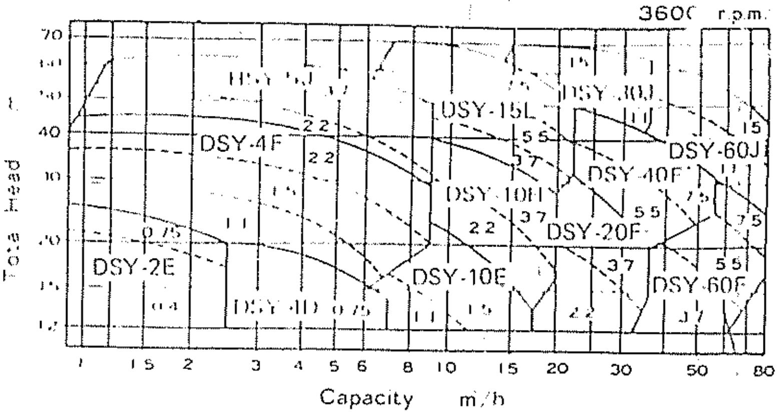
No. for Reg.	Equipment numeral Z	Stockless anchor			Stud link chain cables						Recommended ropes				
		Bower anchor		Stream anchor	Bower anchors				Stream wire or chain for stream anchor		Towline		Mooring ropes		
		Number [1]	Mass per anchor		Total length	Diameter d_1	Diameter d_2	Diameter d_3	Length	Br. load[2]	Length	Br. load[2]	Number	Length	Br. load[2]
			[kg]		[m]	[mm]	[mm]	[mm]	[m]	[kN]	[m]	[kN]		[m]	[kN]
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
101	up to - 50	2	120	40	165	12,5	12,5	12,5	80	65	180	100	3	80	35
102	50 - 70	2	180	60	220	14	12,5	12,5	80	65	180	100	3	80	35
103	70 - 90	2	240	80	220	16	14	14	85	75	180	100	3	100	40
104	90 - 110	2	300	100	247,5	17,5	16	16	85	80	180	100	3	110	40
105	110 - 130	2	360	120	247,5	19	17,5	17,5	90	90	180	100	3	110	45
106	130 - 150	2	420	140	275	20,5	17,5	17,5	90	100	180	100	3	120	50
107	150 - 175	2	480	165	275	22	19	19	90	110	180	100	3	120	55
108	175 - 205	2	570	190	302,5	24	20,5	20,5	90	120	180	110	3	120	60
109	205 - 240	3	660		302,5	26	22	20,5			180	130	4	120	65
110	240 - 280	3	780		330	28	24	22			180	150	4	120	70
111	280 - 320	3	900		357,5	30	26	24			180	175	4	140	80
112	320 - 360	3	1020		357,5	32	28	24			180	200	4	140	85
113	360 - 400	3	1140		385	34	30	26			180	225	4	140	95
114	400 - 450	3	1290		385	36	32	28			180	250	4	140	100
115	450 - 500	3	1440		412,5	38	34	30			180	275	4	140	110
116	500 - 550	3	1590		412,5	40	34	30			190	305	4	160	120
117	550 - 600	3	1740		440	42	36	32			190	340	4	160	130
118	600 - 660	3	1920		440	44	38	34			190	370	4	160	145
119	660 - 720	3	2100		440	46	40	36			190	405	4	160	160
120	720 - 780	3	2280		467,5	48	42	36			190	440	4	170	170
121	780 - 840	3	2460		467,5	50	44	38			190	480	4	170	185
122	840 - 910	3	2640		467,5	52	46	40			190	520	4	170	200
123	910 - 980	3	2850		495	54	48	42			190	560	4	170	215
124	980 - 1060	3	3060		495	56	50	44			200	600	4	180	230
125	1060 - 1140	3	3300		495	58	50	46			200	645	4	180	250
126	1140 - 1220	3	3540		522,5	60	52	46			200	690	4	180	270
127	1220 - 1300	3	3780		522,5	62	54	48			200	740	4	180	285
128	1300 - 1390	3	4050		522,5	64	56	50			200	785	4	180	305
129	1390 - 1480	3	4320		550	66	58	50			200	835	4	180	325
130	1480 - 1570	3	4590		550	68	60	52			220	890	5	190	325
131	1570 - 1670	3	4890		550	70	62	54			220	940	5	190	335
132	1670 - 1790	3	5250		577,5	73	64	56			220	1025	5	190	350
133	1790 - 1930	3	5610		577,5	76	66	58			220	1110	5	190	375
134	1930 - 2080	3	6000		577,5	78	68	60			220	1170	5	190	400
135	2080 - 2230	3	6450		605	81	70	62			240	1260	5	200	425
136	2230 - 2380	3	6900		605	84	73	64			240	1355	5	200	450
137	2380 - 2530	3	7350		605	87	76	66			240	1455	5	200	480
138	2530 - 2700	3	7800		632,5	90	78	68			260	1470	6	200	480
139	2700 - 2870	3	8300		632,5	92	81	70			260	1470	6	200	490
140	2870 - 3040	3	8700		632,5	95	84	73			260	1470	6	200	500
141	3040 - 3210	3	9300		660	97	84	76			280	1470	6	200	520
142	3210 - 3400	3	9900		660	100	87	78			280	1470	6	200	555
143	3400 - 3600	3	10500		660	102	90	78			280	1470	6	200	590
144	3600 - 3800	3	11100		687,5	105	92	81			300	1470	6	200	620
145	3800 - 4000	3	11700		687,5	107	95	84			300	1470	6	200	650
146	4000 - 4200	3	12300		687,5	111	97	87			300	1470	7	200	650
147	4200 - 4400	3	12900		715	114	100	87			300	1470	7	200	660
148	4400 - 4600	3	13500		715	117	102	90			300	1470	7	200	670
149	4600 - 4800	3	14100		715	120	105	92			300	1470	7	200	680
150	4800 - 5000	3	14700		742,5	122	107	95			300	1470	7	200	685
151	5000 - 5200	3	15400		742,5	124	111	97			300	1470	8	200	685
152	5200 - 5500	3	16100		742,5	127	111	97			300	1470	8	200	695
153	5500 - 5800	3	16900		742,5	130	114	100			300	1470	8	200	705
154	5800 - 6100	3	17800		742,5	132	117	102			300	1470	9	200	705
155	6100 - 6500	3	18800		742,5		120	107			300	1470	9	200	715
156	6500 - 6900	3	20000		770		124	111			300	1470	9	200	725
157	6900 - 7400	3	21500		770		127	114			300	1470	10	200	725
158	7400 - 7900	3	23000		770		132	117			300	1470	11	200	725
159	7900 - 8400	3	24500		770		137	122			300	1470	11	200	735
160	8400 - 8900	3	26000		770		142	127			300	1470	12	200	735
161	8900 - 9400	3	27500		770		147	132			300	1470	13	200	735
162	9400 - 10000	3	29000		770		152	132			300	1470	14	200	735
163	10000 - 10700	3	31000		770			137			300	1470	15	200	735
164	10700 - 11500	3	33000		770			142			300	1470	16	200	735
165	11500 - 12400	3	35500		770			147			300	1470	17	200	735
166	12400 - 13400	3	38500		770			152			300	1470	18	200	735
167	13400 - 14600	3	42000		770			157			300	1470	19	200	735
168	14600 - 16000	3	46000		770			162			300	1470	21	200	735

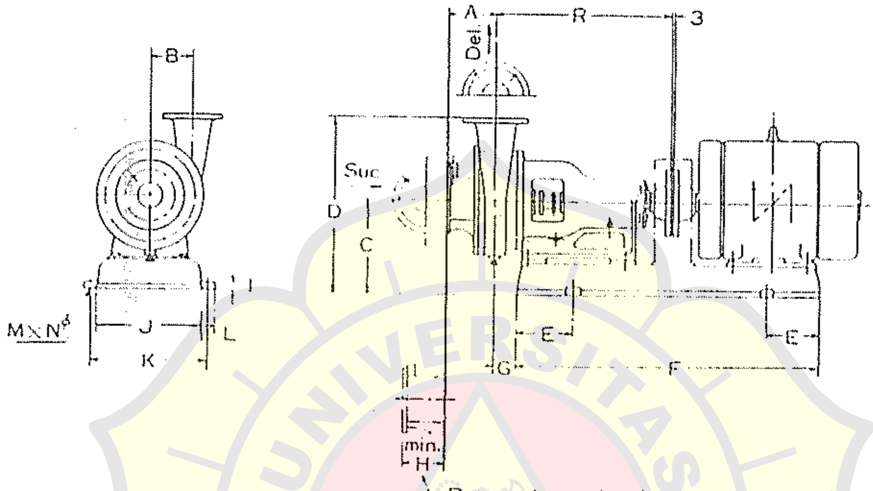
d_1 = Chain diameter Grade K 1 (Ordinary quality)
d_2 = Chain diameter Grade K 2 (Special quality)
d_3 = Chain diameter Grade K 3 (Extra special quality)
} See also D

1 see C.1.
2 see F.1.2

HORIZONTAL SINGLE STAGE SINGLE SUCTION

DSY Type





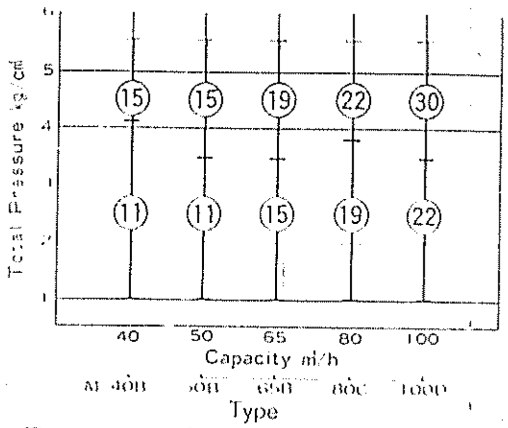
Put a short pipe here to remove the casing cover.

Dimensions—mm

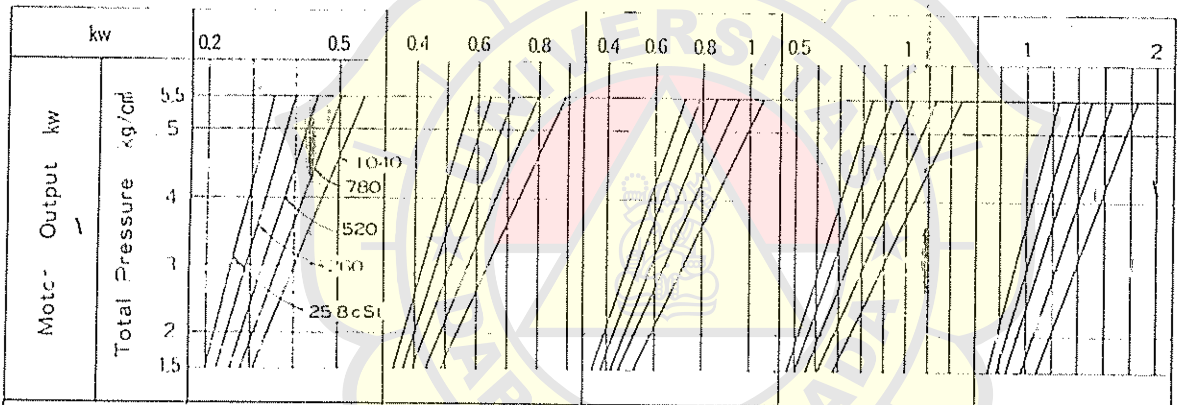
Type	Motor (kw)	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	R	Pump Weight (kg)
SY-2E	0.75	32	32	88	73	165	315	100	550	35	100	25	200	240	23	4	15	350	45
SY-4D	0.75	32	32	90	65	172	290	100	550	35	100	25	200	240	23	4	15	350	45
SY-4F	1.5	32	32	96	92	165	315	100	600	30	100	25	200	240	23	4	15	350	50
	2.2			96	92	175	325	100	620	35	100	25	240	280	23	4	15	350	50
SY-5J	2.2	32	32	94	95	182	325	100	620	35	100	25	240	280	23	4	15	350	60
	3.7			94	95	197	340	100	650	30	100	25	260	300	23	4	15	350	60
Y-10E	1.5	50	50	96	75	165	295	100	600	30	100	25	200	240	23	4	15	350	60
Y-10H	2.2	50	50	96	96	175	325	100	620	35	100	25	240	280	23	4	15	350	55
	3.7			96	96	190	340	100	650	30	100	25	260	300	23	4	15	350	55
Y-15L	3.7	50	50	110	120	190	390	125	700	28	100	25	300	340	23	4	15	370	80
	5.5			110	120	210	410	125	700	28	100	25	300	340	23	4	15	370	80
	7.5			110	120	210	410	150	750	28	100	25	300	340	23	4	15	370	80
Y-20F	2.2	65	65	105	95	175	315	100	620	35	100	25	240	280	23	4	15	350	75
	3.7			105	95	190	330	100	650	30	100	25	260	300	23	4	15	350	75
Y-40F	5.5	100	100	118	105	202	420	125	700	28	100	25	300	340	23	4	15	370	125
	7.5			118	105	202	420	150	750	28	100	25	300	340	23	4	15	370	125
-30J	11	65	65	120	120	249	445	175	860	35	100	25	360	400	23	4	15	425	110
	15			120	120	249	445	175	900	35	100	25	360	400	23	4	15	425	110
-60F	3.7	100	100	122	115	210	390	120	700	35	100	25	300	340	23	4	15	425	160
	5.5			122	115	210	390	150	750	30	100	25	300	340	23	4	15	425	160
	7.5			122	115	210	390	150	780	35	100	25	300	340	23	4	15	425	160
-60J	11	100	100	120	120	235	445	175	860	35	100	25	360	400	23	4	15	425	110
	15			120	120	235	445	175	900	35	100	25	360	400	23	4	15	425	110

HORISONTAL GEAR PUMP

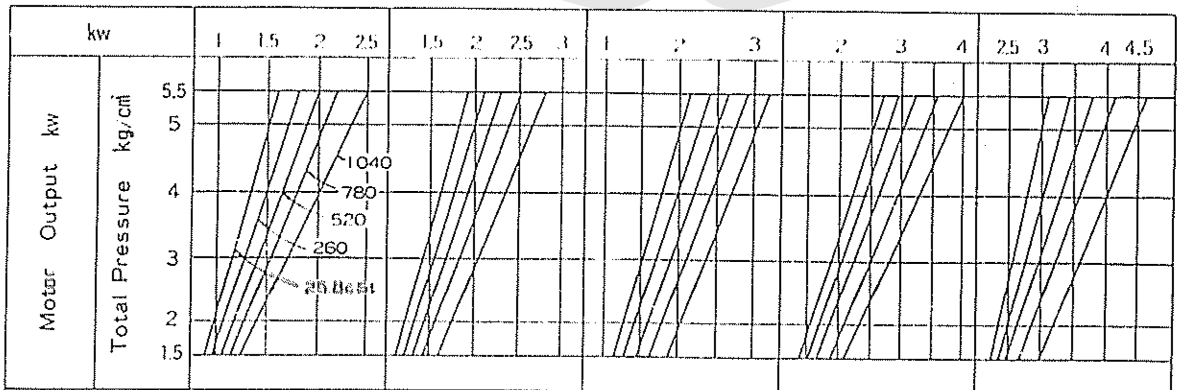
M Type PERFORMANCE CHART



The number in ○ mark indicates the output(kw) of the motor when 260cSt oil is used.



Nor. Capacity	0.5	1	1.5	2	3
Type	M-05B	M-1B	M-2B	M-2B	M-3B



Nor. Capacity	4	5	6	7.5	10
Type	M-4B	M-5B	M-6B	M-7.5B	M-10B

NAUTISERVO B.V.

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Tel. 01891-3955

P.O. Box 19
2650 AA Berkel-Rodenrijs
Telex 26774 Nauti-NL

Ankerwinde
Windlass

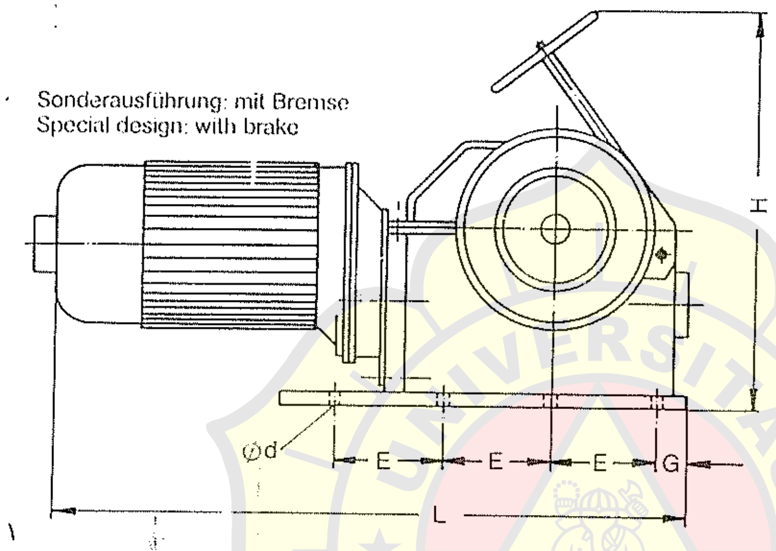
Blatt-Nr.

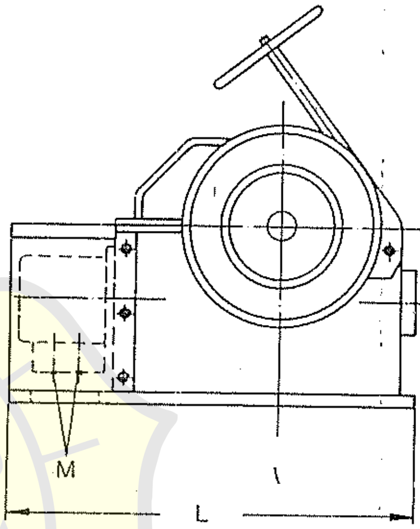
HP–0048

Type . . . . E
Antrieb durch Elektromotor
Drive by electric motor

Type . . . . H
Antrieb durch Hydraulikmotor
Drive by hydraulic motor

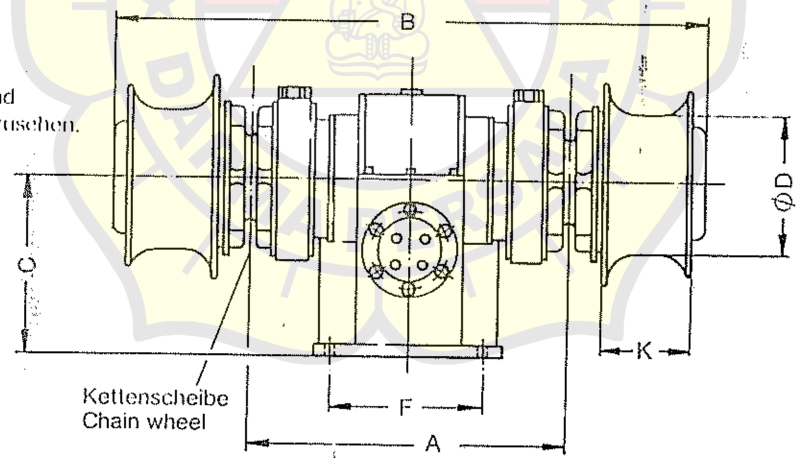
Sonderausführung: mit Bremse
Special design: with brake





Achtung!
Kettenstopper sind vom Kunden vorzusehen.

Chain stopper to be supplied by yard or customer.



Kettenscheibe
Chain wheel

Type	A	B	C	D	d	E	F	G	H	K	L		Z [kp] ø Kette chain	V [m/min] ø Kette chain	Antriebsleistung Driving power
720.03E	450	850	250	200	22	150	215	40	550	130	900		1500 ø 16	10 ø 16	4,4 kW/380 V DS
720.04H	450	850	250	200	22	150	215	40	550	130	580	R ½"	1500 ø 16	10 ø 16	Δp 70 bar/55 l/min
721.16E	535	1100	320	290	22	190	250	82	680	180	1121		3400 ø 18	10 ø 22	7,5 kW/380 V DS
721.15H	535	1100	320	290	22	190	250	82	680	180	785	R ¾"	3400 ø 18	10 ø 22	Δp 105 baf/65 l/min
722.01E	660	1270	395	290	26	240	350	85	690	180	1315		5600 ø 26	10 ø 28	13,5 kW/380 V DS
722.02H	660	1270	395	290	26	240	350	85	690	180	960	R ¾"	5600 ø 26	10 ø 28	Δp 135 bar/65 l/min

Konstruktionsänderungen vorbehalten/Subject to changes of design

Maßangaben und Leistungswerte sind nur für die aufgeführten Ketten gültig.
Dimensions and performance data only apply to the chains listed.

V = Hubgeschwindigkeit
Lifting speed

Z = max. Zugkraft
max. lifting power

Maße/Dimensions = [mm]

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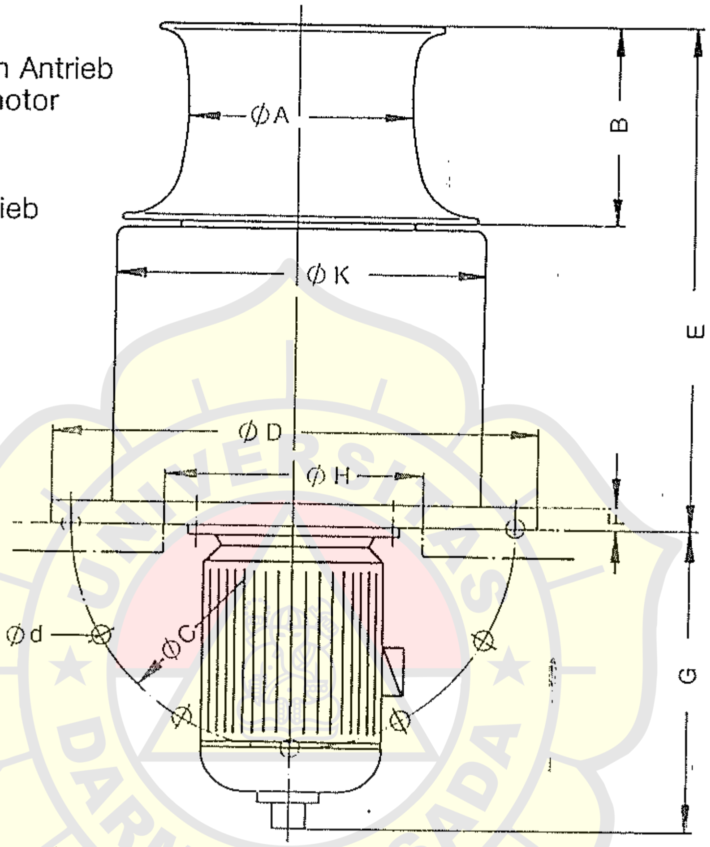
Tel. 01891-3955
P.O. Box 19
2650 AA Berkel-Rodenrijs
Telex 26774 Nauti-NL

Verholspill
Capstan

Blatt-Nr.
HP-0049

Type . . . . E
mit elektromotorischem Antrieb
with drive by electric motor

Type . . . . H
mit hydraulischem Antrieb
with hydraulic drive



Sonderausführung: mit Bremse am E-motor
Special design: electric motor with brake

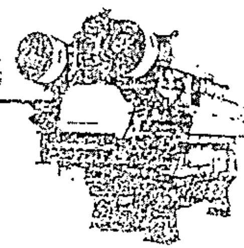
Type	A	B	C	D	E	F	G	H	K	d	V [m/min]	Z [kp]	Antriebsleistung Driving power	
2000 H	290	180	460	500	500	25	100	320	394	12x22	10	2000	38 l/min Δp 85 bar	R ½"
2000 E	290	180	460	500	500	25	310	320	394	12x22	10	2000	3 kW 380 V~	
4000 H	320	270	620	680	693	30	200	420	520	12x25	10	4000	50 l/min Δp 140 bar	R¾"
4000 E	320	270	620	680	693	30	370	420	520	12x25	10	4000	5,5 kW 380 V~	
6000 H	400	340	740	800	800	40	250	520	622	12x25	8	6000	55 l/min Δp 140 bar	R¾"
6000 E	400	340	740	800	800	40	410	520	622	12x25	8	6000	7,5 kW 380 V~	

Konstruktionsänderungen vorbehalten/Subject to changes of design

Maße/Dimensions = [mm]
V = Laufgeschwindigkeit max.
Speed max.
Z = max. Zugkraft in kp
max. tractive power in kp

5-'81

WÄRTSILÄ
NOHAB 25



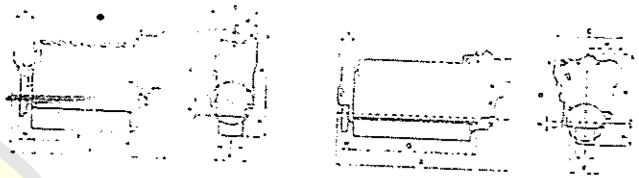
Wärtsilä Nohab 25

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 a cost-effective power. The high number of repeat orders is proof of customer confidence.

Main data

Cylinder bore	250 mm
Piston stroke	300 mm
Speed	720–1000 rpm
Mean effective pressure	20.8–16.7 bar
Piston speed	7.2–10.0 m/s
FUEL SPECIFICATION:	
Fuel oil	380 cSt/50°C
	3500 sRI/100°F

Output — Engine type	Output in kW/BHP at 720 rpm kW	720 rpm BHP	750 rpm kW	750 rpm BHP	825 rpm kW	825 rpm BHP	900 rpm kW	900 rpm BHP	1000 rpm kW	1000 rpm BHP
6R25	1110	1500	1150	1560	1270	1730	1380	1880	1545	2100
8V25	1470	2000	1530	2060	1590	2300	1840	2500	1840	2500
12V25	2210	3000	2300	3130	2530	3440	2760	3750	2760	3750
16V25	2940	4000	3070	4175	3360	4600	3680	5000	3680	5000



Principal dimensions (mm) and weights (tonnes)

Engine type	A	B	C	D	E	F Max.	G
6R25	4245	[illegible]	1355	2070	350	[illegible]	2700
8V25	3515	[illegible]	1855	1960	180	[illegible]	2360
12V25	4665	[illegible]	1960	1960	180	[illegible]	3220
16V25	5650	[illegible]	2110	1960	180	[illegible]	4050

Engine type	H Max.	I	K	M	N	Weight
6R25	375**	[illegible]	920	650	560	9.9
8V25	375**	[illegible]	920	665	525	11.2
12V25	375**	[illegible]	920	955	525	15.6
16V25	375**	[illegible]	920	1055	525	21.1

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

Generating sets

Output — Engine type	Output at 720 rpm/60 Hz Eng. kW	720 rpm/60 Hz Gen. kW	750 rpm/50 Hz Eng. kW	750 rpm/50 Hz Gen. kW	900 rpm/60 Hz Eng. kW	900 rpm/60 Hz Gen. kW	1000 rpm/50 Hz Eng. kW	1000 rpm/50 Hz Gen. kW
6R25	1110	1050	1150	1100	1380	1320	1545	1470
8V25	1470	1400	1530	1450	1840	1770	1840	1770
12V25	2210	2100	2300	2190	2760	2650	2760	2650
16V25	2940	2800	3070	2920	3680	3530	3680	3530

Generating set dimensions (mm) and weights (tonnes)

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	7500	2050	3205	17.6	33
16V25	8000	2050	3205	22.1	43

6LAAL

Engine output
243–669 kW (330–910 PS)

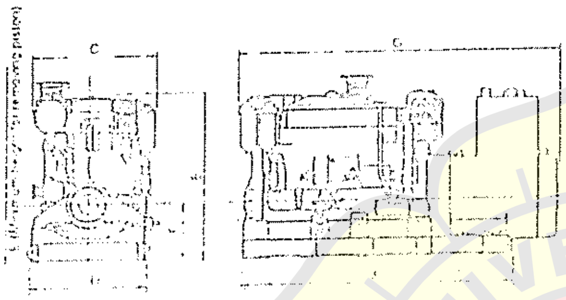
Specifications

Engine model		6LAAL-DTN	6LAAL-UTN			12LAAL-DTN		12LAAL-UTN	
No. of cylinders		6				V12			
Cylinder bore × stroke	mm	148×165				148×165			
Continuous rated output	kW (PS)	243 (330)	265 (360)	309 (420)	353 (480)	530 (720)	618 (840)	574 (780)	669 (910)
Engine speed	rpm	1200	1200	1500	1800	1500	1800	1500	1800
Generator capacity	kW	220	240	280	320	480	560	520	600
Starting system		Electric starting (Air-motor starting is available.)				Electric starting (Air-motor starting is available.)			
Dry weight	kg	1990	2050			3660		3680	

The engine dry weight may differ depending upon the specifications and attached accessories

Dimensions (Units: mm)

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



Engine model	6LAAL-DTN	6LAAL-UTN		12LAAL-DTN	12LAAL-UTN
Engine speed (rpm)	1200	1200	1500 1800	1500 1800	1500 1800
A	2340	2530	2340	2900	2900
B	1469	1469		1610	1610
C	1061	1061		1452	1452
D	1000	1000		1060	1060
E	1414	1414		1315	1315
F	439	439		640	640
G	2721	2785	2828	3544	3544
Dry weight of generating equipment (kg)	3380	3710	3620	6400	6400

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

4HAL2 6HAL2

Engine output
72–305 kW (98–414 PS)

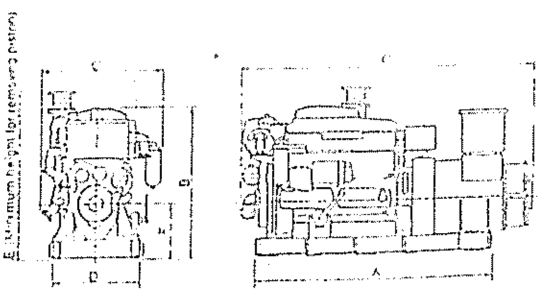
Specifications

Engine model		4HAL2-TN1			4HAL2-TN			6HAL2-N			6HAL2-TN			6HAL2-HTN			6HAL2-DTN		
No. of cylinders		4						6											
Cylinder bore × stroke	mm	130×165						130×165											
Continuous rated output	kW (PS)	72 (98)	89 (121)	116 (157)	90 (122)	115 (156)	135 (183)	90 (122)	115 (156)	135 (183)	120 (163)	150 (204)	180 (244)	160 (217)	220 (299)	265 (360)	200 (271)	255 (346)	305 (414)
Engine speed	rpm	1200	1500	1800	1200	1500	1800	1200	1500	1800	1200	1500	1800	1200	1500	1800	1200	1500	1800
Generator capacity	kW	64	80	104	80	100	120	80	100	120	104	136	160	144	200	240	180	232	280
Starting system		Electric starting (Air-motor starting is available.)																	
Dry weight	kg	1030			1030			1380			1395			1410			1420		

The engine dry weight may differ depending upon the specifications and attached accessories

Dimensions (Units: mm)

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



Engine model	4HAL2-TN1		4HAL2-TN		6HAL2-N		6HAL2-TN		6HAL2-HTN		6HAL2-DTN	
Engine speed (rpm)	1200	1500 1800	1200	1500 1800	1200	1500 1500	1200	1500 1800	1200	1500 1800	1200	1500 1800
A	1600		1600		1970		1970	2050	2150		2180	
B	1206		1206		1285		1551		1351		1351	
C	1013		1013		1115		1115		1115		1115	
D	800		800		800		800		800		800	
E	1233		1233		1268		1268		1268		1268	
F	450		450		485		485		485		485	
G	2021	2047	2021	2047	2369	2415	2484	2524	2625	2639	2682	2692
Dry weight of generating equipment (kg)	1750	1820	1810	1820	2180	2190	2290	2340	2630	2680	2750	2790

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

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

$$Q = 10fRA10 \tag{2.2.a}$$

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

$$Q_p = \frac{Q}{24 \times 3600 \times D} \tag{2.2.b}$$

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.

Hari / Curah hujan (mm)	Hari-ke-1	Hari ke-2	Hari ke-3	Hari ke-4	Jumlah
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;

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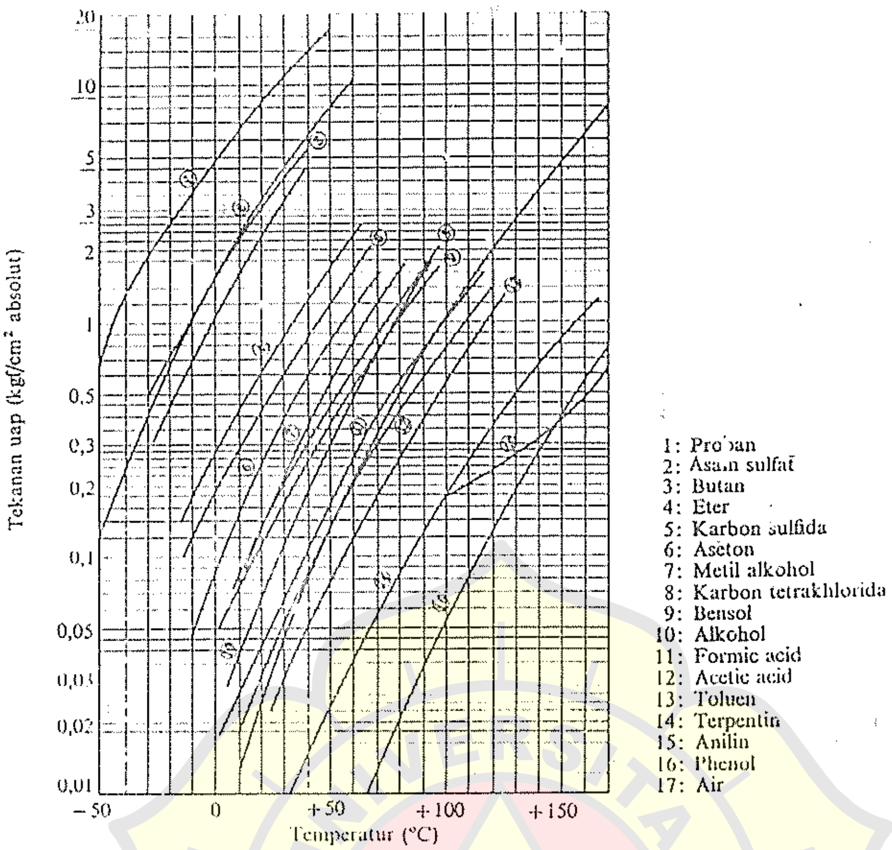
* Transpirasi = penguapan melalui pernapasan tanaman
Penguapan = penguapan langsung dari air ke udara
Perkolasi = peresapan air ke dalam tanah.



(b) Tekanan uap berbagai zat cair
(*Catatan*: 1 kgf/cm² = 0,1 M Pa)

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:

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

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 positip (+) dipakai apabila muka air di sisi ke luar lebih tinggi dari pada sisi isap.

Δh_p: Perbedaan head tekanan yang bekerja pada kedua permukaan air (m),

$$\Delta h_p = h_{p_2} - h_{p_1},$$

h_l: Berbagai kerugian head di pipa, katup, belokan, sambungan, dll (m), di mana h_p: Head tekanan (m)

p: Tekanan (kgf/cm^2)

γ: Berat per satuan volume zat cair yang dipompa (kgf/l)

Apabila tekanan diberikan dalam kPa, dapat dipakai rumus berikut:

$$h_p = \frac{1}{9{,}8}\frac{p'}{\rho} \tag{2.9}$$

di mana p': Tekanan (Pa)

ρ: Rapat masa (kg/l)

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

$$Y = gH \tag{2.10}$$

Sebagaimana diutarakan di atas, untuk menentukan head total yang harus disediakan pompa, perlu dihitung lebih dahulu head kerugian h_l. 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:

$$v = CR^pS^q \tag{2.11}$$

$$h_f = \lambda\frac{L}{D}\frac{v^2}{2g} \tag{2.12}$$

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

C, p, q: Koefisien-koefisien

R: Jari-jari hidrolik (m)

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

S: Gradien hidrolik

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

h_f: Head kerugian gesek dalam pipa (m)

λ: Koefisien kerugian gesek

g: Percepatan gravitasi (9,8 m/s^2)

L: Panjang pipa (m)

D: Diameter dalam pipa (m)

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

$$Re = \frac{vD}{\nu} \tag{2.13}$$

di mana Re: Bilangan Reynolds (tak berdimensi)

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

D: Diameter dalam pipa (m)

ν: Viskositas kinematik zat cair (m^2/s)

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

(I) Aliran laminer

Dalam hal aliran laminer, koefisien kerugian gesek untuk pipa (λ) dalam pers. (2.12) dapat dinyatakan dengan

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

(II) Aliran turbulen

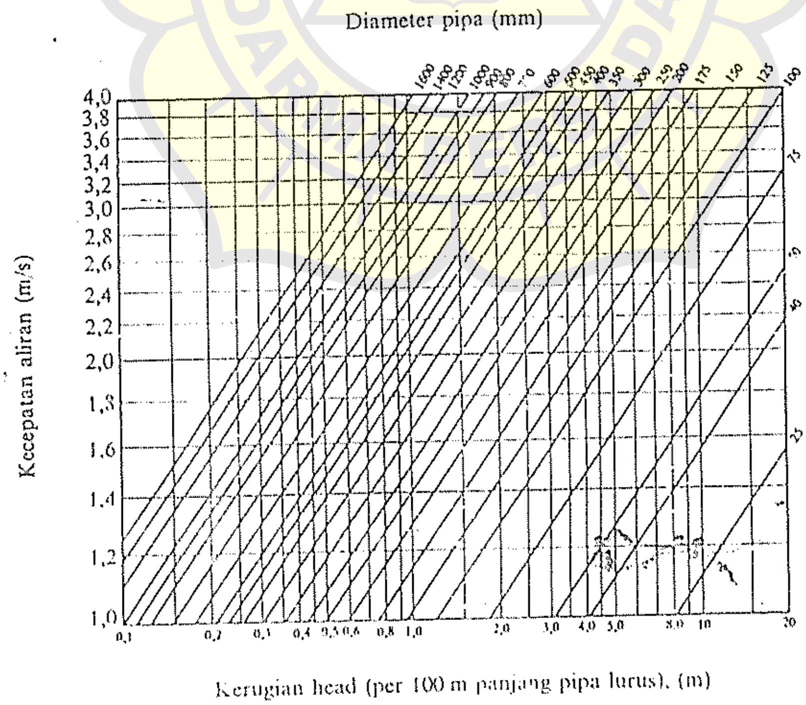
Untuk menghitung kerugian gesek dalam pipa pada aliran turbulen terdapat berbagai rumus empiris. Di bawah ini akan diberikan cara perhitungan dengan rumus Darcy dan Hazen-Williams.

1) Formula Darcy

Dengan cara Darcy, koefisien kerugian gesek λ 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 λ 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:

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

di mana η_{ad}: Efisiensi adiabatik kesekuruhan (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

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

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

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

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

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

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

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

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

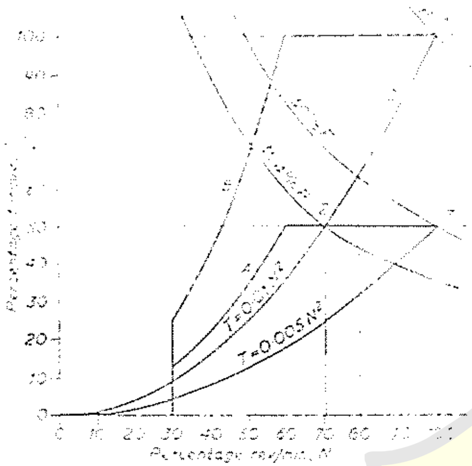


FIG. 11-5—*Torque—speed curves for two engines driving one propeller.*

propeller matched to absorb the full power (service rating) of the two engines together at full rev/min corresponding to point 1. The vessel may be propelled by one engine alone developing its full torque (equal to half the total torque) as at point 2. The revolutions will be reduced to 0·707 of full rev/min and the power available to 0·707 × 0·5 = 0·353 of full power giving the ship 0·707 of its full speed.

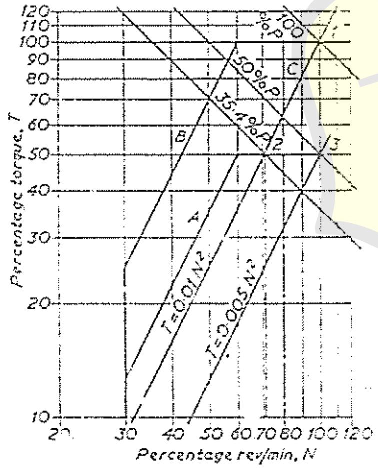


FIG. 11-6—*Torque—speed curves for two engines driving one propeller. log–log scales.*

If a controllable pitch propeller is fitted the pitch can be reduced when running on one engine permitting it to develop its full rev/min as well as its full torque as at point 3. The power is 0·5 of total full power and the vessel's speed is 0·8 of full speed.

... this nature the curves can be plotted easily and quickly on "log-log paper" as shown in Figure 11–6 which has been drawn using percentage scales. Curves of constant power and power proportional to (rev/min)² become groups of parallel straight lines which are easily located. The curves and points in Figure 11–6 correspond to those in Figure 11–5 and are lettered and numbered accordingly.

Vessels engaged on routes where reduced speed is required for a significant proportion of the running time can benefit economically from the installation of twin geared engines. As can be seen from Figure 11–7

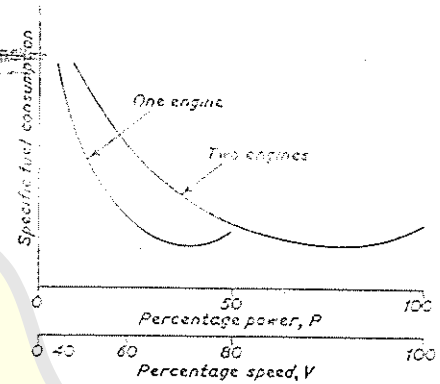


FIG. 11–7—*Fuel consumption, power and speed for two engines driving one controllable pitch propeller.*

which is drawn for twin engines and a controllable pitch propeller, the rise in specific fuel consumption at reduced power and speed can be countered by changing to single engine propulsion at speeds below 80% of full speed.

Installations may be designed with two engines of unequal power geared together or with three or four engines. The reduction in the speed of the vessel as a result of part of the total power not being used is easily calculated, as follows.

Let full power, speed, torque and rev/min be represented by P_f, V_f, T_f and N_f respectively,

and let the available power, speed, torque and rev/min be represented by Pa, Va, Ta and Na respectively.

The propeller will absorb the available torque and for a fixed pitch propeller the rev/min will be reduced.

Thus $$\frac{Ta}{T_f} = \left(\frac{Na}{N_f}\right)^2$$

$$\therefore \quad \frac{Pa}{P_f} = \frac{Ta\,Na}{T_f\,N_f} = \left(\frac{Ta}{T_f}\right)^{3/2} \longrightarrow = \frac{T_a}{T_f}\cdot\left(\frac{T_a}{T_f}\right)^{\frac{1}{2}}$$

and $$\frac{Va}{V_f} = \left(\frac{Pa}{P_f}\right)^{1/3} = \left(\frac{Ta}{T_f}\right)^{1/2}$$