BAB.VII.
PENUTUP

Dengan selesainya penyusunan tugas merancang ini, maka penulis dapat mengambil kesimpulan yang berhubungan dengan perancanaan kapal Ikan 350 GT sebagai sarana penangkapan Ikan yang dapat menunjang perkembangan penangkapan Ikan di Indonesia.Adapun kesimpulan penulisan tersebut adalah sebagai berikut :

1. Ringkasan spesifikasi teknis dari kapal Ikan 350 GT :
   - Panjang seluruhnya (LOA : 47,30 m
   - Panjang antara garis tegak (LPP) : 43,00 m
   - Lebar (B) : 9,00 m
   - Tinggi (H) : 4,50 m
   - Koefisien blok (C_b) : 0,65
   - Koefisien prismatik (C_p) : 0,66
   - Koefisien garis air (C_w) : 0,78
   - Koefisien tengah kapal (C_m) : 0,97
   - Displasement : 902,436 ton
   - Volume :
   - Jumlah anak buah kapal (ABK) : 20 orang
   - Alat Penggerak yang digunakan

     Jumlah mesin : 1 (satu) buah

     Merk : MAN B & W

Tipe	: V 20 / 27
Daya	: 1400 kW / 1900 HP
Putaran mesin	: 1000 Rpm
Bore x Stroke	: 200 m x 270 m
SFOC	: 199 g / kWh
SLOC	: 1,2 g /kWh
Jumlah Silinder	: 14 V
Diameter propeller	: 2,00 m
Jumlah Daun	: 4 (Buah)

- Kecepatan Dinas : 14 Knots.

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

3. Dalam menentukan generator set didasarkan pada pembebanan penggunaan daya yang terbesar yaitu pada saat kapal melakukan menuver sebesar 112,99 kW, dengan menggunakan 2 buah generator masing - masing 150 kW daya yang dibutuhkan dapat terpenuhi.

4. Dalam perencanaan kamar mesin,tidak terlepas adanya asumsi - asumsi yang diberikan untuk mempermudah dalam perhitungan dengan tidak mengabaikan tanggungjawab secara teknis,ekonomis

serta peraturan - peraturan yang ada sehingga hasil perhitungann dapat mendekati keadaan yang sebenarnya.

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

DAFTAR PUSTAKA

1. Biro Klasifikasi Indonesia volume II, " ***Rules for Construction***" 1996
2. Biro Klasifikasi Indonesia volume III, " ***Rules For Installations***" 1996
3. Harvald,SV Aa " Tahanan dan Propulsi Kapal ". Airlangga University Press edisi I.1992.
4. Khetagurov M. ***"Marine Auxilary Machinery end System".*** Peace Publishers Moscow.
5. Poels, ***"Ship Propulsion"*** Edisi II.1992
6. Sastrodiwongso.T. ***"Ship Propulsion"*** Edisi II.1992
7. Ctoecker FW "Refrigerasi dan Pengkondisian udara".Erlangga,edisi II.1994
8. Taha.H.Sularso "Pompa dan Kompresor".PT.Pramadya Paramita Cetakan ke-6.1996
9. The Marine Engineering Society in Japan ***"Marine out fitting design manual",***volume I,Piping System for Diesel Ships.

Referensi.No.1.

cording to the following formula:

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

v = v_0 for ahead condition

v = v_a for astern condition

κ_1 = coefficient, depending on the aspect ratio Λ

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

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

κ_3 = coefficient, depending on the location of the rudder

κ_3 = 0,8 for rudders outside the propeller jet

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

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

κ_t = coefficient depending on the thrust coefficient c_t

κ_t = 1,0 normally

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

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

Table 14.1

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

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

$Q_R = C_R \cdot r$ [Nm]

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

α = 0,33 for ahead condition

α = 0,66 for astern condition (general)

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

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

α = 0,25 for ahead condition

α = 0,55 for astern condition.

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

k_b = balance factor as follows:

k_b = A_f/A

k_b = 0,08 for unbalanced rudders

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

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

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

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

The resulting force of each part may be taken as:

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

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

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

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

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

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

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

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

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

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

$C_1 = A_1/b_1$

Referensi.No.1.

b_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).

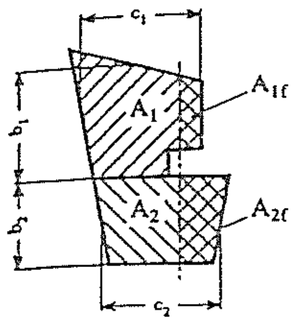


Fig. 14.2

.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.

Scantlings of the Rudder Stock

Rudder stock diameter

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

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

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

The related torsional stress is:

$\tau_t = \frac{68}{k_r}$ [N/mm²]

k_r see A.4.2.

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

.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:

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

Bending stress:

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

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

Torsional stress:

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

D_1 = increased rudder stock diameter in [cm]

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

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

Q_R see B.1.2 and B.2.2 - 2.3

D_t see 1.1.

Guidance

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

Referensi.No.1.

Section 18

Equipment

A. General

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

Guidance

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

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

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

4. The equipment numeral formula for anchoring equipment required under this Section is based on an assumed current speed of 2,5 m/sec, wind speed of 25 m/sec and a scope of chain cable between 6 and 10, the scope being the ratio between length of chain paid out and water depth.

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

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

Windlass and chain stopper, if fitted, are to comply

For the substructures of windiasses and chain s pers, see Section 10, B.5.

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

3. For ships having the navigation nota "L" (Small Coasting Service) affixed to their c acter of classification, the equipment may be termined as for one numeral range lower than requ in accordance with the equipment numeral Z.

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

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

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

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

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

7. For ships having three or more propelle reduction of the weight of the bower anchors an chain cables may be considered.

B. Equipment numeral

The equipment numeral is to be calculated as

Referensi.No.1.

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

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

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

h = $f_b + \sum h'$

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

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

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

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

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

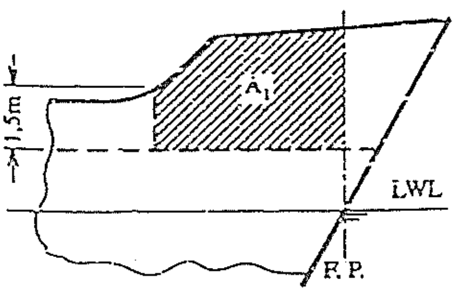


Fig. 18.1

C. Anchors

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

Guidance

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

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

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

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

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

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

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

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

Referensi.No.1.

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

Referensi.No.2.

Appendix to Section 2

Part C :

Approximate Calculation of the Starting Air Supply

1. Starting air for installations with reversible engines

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

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

where

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

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

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

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

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

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

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

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

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.

Referensi.No.2.

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

1.6 Pipe connections

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

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

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

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

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

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

2. Calculation of pipe diameters

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

2.2 Dry cargo and passenger ships

a) **main bilge pipes**

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

b) **branch bilge pipes**

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

where

d_H [mm] calculated inside diameter of main bilge pipe

d_z [mm] calculated inside diameter of branch bilge pipe

L [m] length of ship between perpendiculars

B [m] moulded breadth of ship

H [m] depth of ship to the bulkhead deck

l [m] length of the watertight compartment

2.3 Tankers

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

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

where:

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

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

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

2.4 Minimum diameter

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

2.5 Maximum diameter

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

2.6 Deviations

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

3. Bilge pumps

3.1 Capacity of bilge pumps

Each bilge pump must be capable of delivering:

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

where:

Q [m³/h] minimum capacity

d_H [mm] calculated inside diameter of main bilge pipe

Referensi.No.2.

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

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

3.4 Use of other pumps for bilge pumping

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

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

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

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

3.5 Number of bilge pumps for cargo ships

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

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

3.6 Number of bilge pumps for passenger ships

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

4. Bilge pumping for various spaces

4.1 Machinery spaces

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

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

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

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

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

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

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

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

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

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

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

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

4.2 Shaft tunnel

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

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

Referensi.No.2.

drained to the shaft tunnel or machinery space, provided that the drain line is fitted with a self-closing shutoff valve at a clearly visible and easily accessible position. The drain pipes shall have an inside diameter of at least 40 mm.

4.10 Cofferdams, pipe tunnels and void spaces

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

4.11 Chain lockers

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

5. Additional Rules for passenger vessels

5.1.1 The arrangement of bilge pipes

- within 0,2 B of the ship's side measured at the level of the subdivision load line,
- in the double bottom lower than 460 mm above the base line or
- below the horizontal level specified in Rules for Hull Construction, Volume II, Section 29.F.

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

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

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

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

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

5.2 Bilge suctions

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

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

5.3 Arrangement of bilge pumps

5.3.1 Bilge pumps must be installed in separate watertight compartments which are to be so arranged that they are unlikely to be simultaneously flooded in the event of damage to the ship.

Ships with a length of 91,5 m or over or having a criterion numeral of 30[1]) or more are to have at least one bilge pump available in emergency cases. This requirement is satisfied if

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

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

c) bilge pumps are installed above the bulkhead deck.

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

5.4 Passenger vessels for limited range of service

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

---

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

Referensi.No.2.

6. Additional Rules for tankers

See Section 15, B.4.

7. Bilge testing

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

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

1. Oily water separating equipment

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

1.2 Ships of 10.000 tons gross and above shall be fitted, in addition to the equipment required in Paragraph 1.1, with an oil discharge monitoring and control system or with a 15 ppm alarm system.

1.3 A sampling device is to be arranged in the discharge line of oily water separating equipment/filtering systems.

1.4 By-pass lines are not permitted for oilywater separatoring equipment/filtering systems.

2. Discharge of fuel and oil residues

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

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

2.3 A separate discharge line is to be provided for discharge of fuel and oil residues to reception facilities.

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

P. Ballast Systems

1. Ballast lines

1.1 Arrangement of piping - general

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

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

1.2 Pipes passing through tanks

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

1.3 Piping systems

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

1.3.2 Where ballast water tanks may be used exceptionally as dry cargo holds, such tanks are also to be connected to the bilge system. The requirements specified in N.4.5 are applicable.

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

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

Referensi.No.2.

6.3.2 Bilge lines

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

6.3.3 Ballast pipes

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

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

6.3.4 Fuel pipes

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

6.3.5 Cargo pipes

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

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

6.4 Control stands

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

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

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

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

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

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

6.5 Power units

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

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

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

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

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

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

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

7. Pumps

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

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

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

Referensi.No.2.

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

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

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

where

d_H	[mm]	Inside diameter of main bilge line
d_Z	[mm]	Inside diameter of branch bilge line
B	[m]	Breadth of ship
H	[m]	Moulded depth of ship
l_2	[m]	Total length of cargo area
l	[m]	Length of watertight compartment
b	[m]	Maximum breadth of cargo tanks
h	[m]	Maximum depth of cargo tanks
l_{T2}	[m]	Total length of all cargo tanks
l_T	[m]	Length of tanks in the watertight compartment.

The capacity of each bilge pump is to be calculated according to Section 11, N.3.1. At least two bilge pumps are to be provided.

4.1.4 When separate bilge pumps, e.g. ejectors are provided for compartments with watertight bulkheads the pump capacity is to be evaluated as specified in 4.1.3 and is to be divided according the length of the individual compartments. For each compartment two bilge pumps are to be fitted of a capacity of not less than 5 m^3/h each.

4.1.5 Spaces for independent tanks are to be provided with sounding arrangements.

When ballast or cooling water lines are fitted in spaces for independent tanks bilge level alarms are to be provided.

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

4.2.1 Bilge pumping equipment is to be located in the cargo area to serve the cargo pump rooms and cofferdams. A cargo pump may also be used as a bilge pump. On oil tankers used exclusively for the carriage of flammable liquids with flash points above 60 °C, cargo pump rooms and cofferdams may be connected to the engine room bilge system.

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

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

4.3 Ballast systems in the cargo area

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

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

a) Minimum wall thicknesses

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

b) Only completely welded pipes or equivalent are permitted

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

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

5. Ventilation and gas-freeing

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

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

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

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

Referensi.No.3.

Bab

Lima

PENENTUAN TAHANAN KAPAL

5.1. PENDAHULUAN

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

- Metode kapal pembanding
- Metode statistik
- Metode satu per satu

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

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

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

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

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

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

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

A : 0,419 (4 langkah)

B : 0,059 (2 langkah)

N_o : Putaran mesin : 1000 rpm

C : 1 untuk tekanan kerja 30 bar

n $\cancel{N_a}$: 0,06 x No + 14

: 0,06 x 1000 + 14

: 74 RPM.

Maka :

$$Q = 0{,}419x(270/200)x(14 + 0{,}056x14{,}15x74 + 0{,}9)x8{,}478x1{,}00$$

$$= 288{,}42 \text{ dm}^3$$

KOMPRESOR UDARA

a. Kapasitas kompresor dapat dihitung dengan rumus pendekatan (Ref. No. 2. Hl 38) adalah :

$$Q = 1{,}70 \text{ x } J \text{ x } (P - 9)$$

Dimana :

J : Kapasitas dari botol angin : 288,42 dm^3

P : Tekanan Discharge : 30 bar = 30 Kg/cm^2

Maka :

$$Q = 1{,}70 \text{ x } 288{,}42 \text{ x } (30 - 9)$$

$$= 10{,}296 \text{ m}^3/\text{jam}$$

$$= 0{,}1716 \text{ m}^3/\text{menit}$$

Jumlah kompresor yang digunakan adalah dua buah dengan penggerak motor listrik.

b. Daya yang dibutuhkan kompresor

Daya yang dibutuhkan kompresor dapat dihitung dengan rumus pendekatan (Ref. NO.2 HAL.[illegible]) 28)

$$N = \frac{mxk}{k-1} x \frac{PsxQ}{0,75x6120} \left[(P_d/P_s)\right]^{k-1/mxk} - 1$$ Ref. ... hal ....

Dimana :

m	: jumlah tingkat kompresi	:2
k	: Konstanta	: 1,4
P_s	: Tekanan hisap tingkat pertama	: 10332,6 Kg/m²
P_d	: Tekanan Discharge	: 300000 Kg/m²
Q	: Kapasitas kompresor	: 0,1716 m³/menit

Maka :

$$N = \frac{2x1,4}{1,4-1} x \frac{10332,6x0,1716}{0,75x6120} \left[(300000/10332,6)\right]^{1,4-1/2x1,4} - 1$$

$$= 1,672 \text{ kW}$$

Section 14

...d Manoeuvring Arrangement

...manoeu-
...sufficient

...udes all
...steering

...er bear-
...Section.
...ume III,

...shall be
...eparated
...I-I, Reg.

...al in the
...s, the re-
...on con-

5.

...d for sup
...hout ex-
...r carrier
...ock. The
...is to be

...ovided to

...rough the
...or glands

are to be fitted above the deepest load waterline, t[o] prevent water from entering the steering gear com-partment and the lubricant from being washed aw[ay] from the rudder carrier. If the top of the rudder tru[nk] is below the deepest waterline two separate stuffin[g] boxes are to be provided.

3. Size of rudder area

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

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

c_1 = factor for the ship type

= 1,0 in general

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

= 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 ar[ea] A.

Referensi.No.1.

Materials

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

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

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

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

= 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.

Before significant reductions in rudder stock meter due to the application of steels with R_{eH} eding 235 N/mm^2 are granted, the Society may uire the evaluation of the elastic rudder stock ections. Large deflections should be avoided in er to avoid excessive edge pressures in way of rings.

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

Definitions

C_R = rudder force in [N]

Q_R = rudder torque in [Nm]

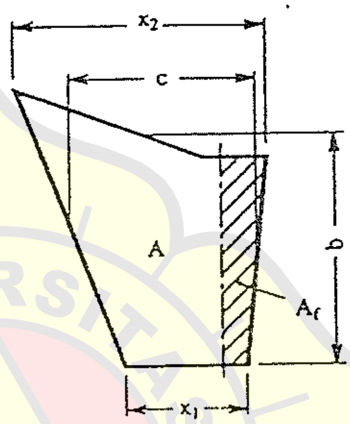
A = total movable area of the rudder in [m^2]
For nozzle rudders, A is not to be taken less than 1,35 times the projected area of the nozzle.

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

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

b = mean height of rudder area in [m]

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



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

Fig. 14.1

Λ = aspect ratio of rudder area A_t

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

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

B. Rudder Force and Torque

1. Rudder force and torque for normal rudders

1.1 The rudder force is to be determined ac-

Referensi.No.1.

cording to the following formula:

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

$v = v_0$ for ahead condition

$v = v_a$ for astern condition

κ_1 = coefficient, depending on the aspect ratio Λ

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

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

κ_3 = coefficient, depending on the location of the rudder

κ_3 = 0,8 for rudders outside the propeller jet

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

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

κ_t = coefficient depending on the thrust coefficient c_t

κ_t = 1,0 normally

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

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

Table 14.1

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

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

$Q_R = C_R \cdot r$ [Nm]

α = 0,33 for ahead condition

α = 0,66 for astern condition (general)

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

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

α = 0,25 for ahead condition

α = 0,55 for astern condition.

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

k_b = balance factor as follows:

$k_b = A_f/A$

k_b = 0,08 for unbalanced rudders

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

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

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

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

The resulting force of each part may be taken as:

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

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

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

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

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

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

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

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

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

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

Referensi.No.1.

A_2/b_2

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

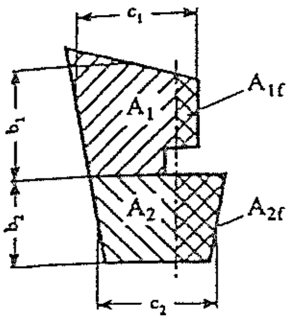


Fig. 14.2

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

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

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

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

for ahead condition

the greater value is to be taken.

Scantlings of the Rudder Stock

Rudder stock diameter

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

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

see B. 1.2 and B. 2.2 - 2.3.

The related torsional stress is:

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

see A.4.2.

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

In case of mechanical steering gear the diameter of the rudder stock in its upper part which is 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:

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

Bending stress:

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

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

Torsional stress:

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

D_1 = increased rudder stock diameter in [cm]

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

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

Q_R see B.1.2 and B.2.2 - 2.3

D_t see 1.1.

Guidance

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

Referensi.No.1.

Section 18

Equipment

A. General

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

Guidance

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

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

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

4. The equipment numeral formula for anchoring equipment required under this Section is based on an assumed current speed of 2,5 m/sec, wind speed of 25 m/sec and a scope of chain cable between 6 and 10, the scope being the ratio between length of chain paid out and water depth.

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

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

Windlass and chain stopper, if fitted, are to comply

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

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

3. For ships having the navigation nota "L" (Small Coasting Service) affixed to their c acter of classification, the equipment may be termined as for one numeral range lower than requ in accordance with the equipment numeral Z.

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

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

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

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

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

7. For ships having three or more propelle reduction of the weight of the bower anchors an chain cables may be considered.

B. Equipment numeral

The equipment numeral is to be calculated as

Referensi.No.1.

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

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.

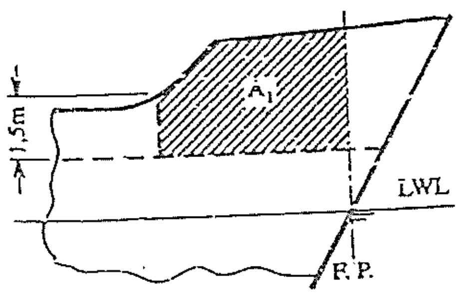


Fig. 18.1

C. Anchors

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

Guidance

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

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

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

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

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

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

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

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

Referensi.No.1.

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	d_2	d_3	Length	Br. load[2]	Length	Br. load[2]	Number	Length	Br. load[2]
			[kg]		[m]	[mm]	[mm]	[mm]	[m]	[kN]	[m]	[kN]		[m]	[kN]
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	489	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

Referensi.No.2.

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:

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

where

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

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

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

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

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

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

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

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

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

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

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

The following values of n_A are to be applied:

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

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

n_o [Rpm] = rated speed

2. Starting air for installations with non reversible engines

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

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

1.6 Pipe connections

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

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

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

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

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

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

2. Calculation of pipe diameters

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

2.2 Dry cargo and passenger ships

a) **main bilge pipes**

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

b) **branch bilge pipes**

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

where

d_H [mm] calculated inside diameter of main bilge pipe

d_z [mm] calculated inside diameter of branch bilge pipe

L [m] length of ship between perpendiculars

B [m] moulded breadth of ship

H [m] depth of ship to the bulkhead deck

l [m] length of the watertight compartment

2.3 Tankers

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

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

where:

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

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

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

2.4 Minimum diameter

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

2.5 Maximum diameter

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

2.6 Deviations

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

3. Bilge pumps

3.1 Capacity of bilge pumps

Each bilge pump must be capable of delivering:

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

where:

Q [m³/h] minimum capacity

d_H [mm] calculated inside diameter of main bilge pipe

Referensi.No.2.

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

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

3.4 Use of other pumps for bilge pumping

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

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

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

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

3.5 Number of bilge pumps for cargo ships

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

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

3.6 Number of bilge pumps for passenger ships

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

4. Bilge pumping for various spaces

4.1 Machinery spaces

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

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

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

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

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

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

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

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

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

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

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

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

4.2 Shaft tunnel

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

[1] See SOLAS 1974, Chapter II-1, part-A, regulations 5

Referensi.No.2.

drained to the shaft tunnel or machinery space, provided that the drain line is fitted with a self-closing shutoff valve at a clearly visible and easily accessible position. The drain pipes shall have an inside diameter of at least 40 mm.

4.10 Cofferdams, pipe tunnels and void spaces

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

4.11 Chain lockers

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

5. Additional Rules for passenger vessels

5.1.1 The arrangement of bilge pipes

- within 0,2 B of the ship's side measured at the level of the subdivision load line,
- in the double bottom lower than 460 mm above the base line or
- below the horizontal level specified in Rules for Hull Construction, Volume II, Section 29.F.

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

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

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

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

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

5.2 Bilge suctions

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

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

5.3 Arrangement of bilge pumps

5.3.1 Bilge pumps must be installed in separate watertight compartments which are to be so arranged that they are unlikely to be simultaneously flooded in the event of damage to the ship.

Ships with a length of 91,5 m or over or having a criterion numeral of 30[1]) or more are to have at least one bilge pump available in emergency cases. This requirement is satisfied if

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

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

c) bilge pumps are installed above the bulkhead deck.

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

5.4 Passenger vessels for limited range of service

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

---

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

Referensi.No.2.

6. Additional Rules for tankers

See Section 15, B.4.

7. Bilge testing

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

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

1. Oily water separating equipment

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

1.2 Ships of 10.000 tons gross and above shall be fitted, in addition to the equipment required in Paragraph 1.1, with an oil discharge monitoring and control system or with a 15 ppm alarm system.

1.3 A sampling device is to be arranged in the discharge line of oily water separating equipment/filtering systems.

1.4 By-pass lines are not permitted for oilywater separatoring equipment/filtering systems.

2. Discharge of fuel and oil residues

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

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

2.3 A separate discharge line is to be provided for discharge of fuel and oil residues to reception facilities.

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

P. Ballast Systems

1. Ballast lines

1.1 Arrangement of piping - general

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

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

1.2 Pipes passing through tanks

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

1.3 Piping systems

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

1.3.2 Where ballast water tanks may be used exceptionally as dry cargo holds, such tanks are also to be connected to the bilge system. The requirements specified in N.4.5 are applicable.

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

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

Referensi.No.2.

6.3.2 Bilge lines

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

6.3.3 Ballast pipes

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

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

6.3.4 Fuel pipes

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

6.3.5 Cargo pipes

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

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

6.4 Control stands

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

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

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

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

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

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

6.5 Power units

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

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

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

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

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

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

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

7. Pumps

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

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

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

Referensi.No.2.

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

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

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

where

d_H	[mm]	Inside diameter of main bilge line
d_Z	[mm]	Inside diameter of branch bilge line
B	[m]	Breadth of ship
H	[m]	Moulded depth of ship
l_2	[m]	Total length of cargo area
l	[m]	Length of watertight compartment
b	[m]	Maximum breadth of cargo tanks
h	[m]	Maximum depth of cargo tanks
l_{T2}	[m]	Total length of all cargo tanks
l_T	[m]	Length of tanks in the watertight compartment.

The capacity of each bilge pump is to be calculated according to Section 11, N.3.1. At least two bilge pumps are to be provided.

4.1.4 When separate bilge pumps, e.g. ejectors are provided for compartments with watertight bulkheads the pump capacity is to be evaluated as specified in 4.1.3 and is to be divided according the length of the individual compartments. For each compartment two bilge pumps are to be fitted of a capacity of not less than 5 m³/h each.

4.1.5 Spaces for independent tanks are to be provided with sounding arrangements.

When ballast or cooling water lines are fitted in spaces for independent tanks bilge level alarms are to be provided.

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

4.2.1 Bilge pumping equipment is to be located in the cargo area to serve the cargo pump rooms and cofferdams. A cargo pump may also be used as a bilge pump. On oil tankers used exclusively for the carriage of flammable liquids with flash points above 60 °C, cargo pump rooms and cofferdams may be connected to the engine room bilge system.

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

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

4.3 Ballast systems in the cargo area

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

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

a) Minimum wall thicknesses

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

b) Only completely welded pipes or equivalent are permitted

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

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

5. Ventilation and gas-freeing

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

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

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

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

Referensi.No.3.

Bab

Lima

PENENTUAN TAHANAN KAPAL

5.1. PENDAHULUAN

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

Metode kapal pembanding
Metode statistik
Metode satu per satu

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

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

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

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

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

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

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

Referensi.No.3

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

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

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

3. Koefisien tahanan sisa spesifik ditentukan dari

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

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

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

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

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

4. C_R dinyatakan sebagai fungsi angka Froude

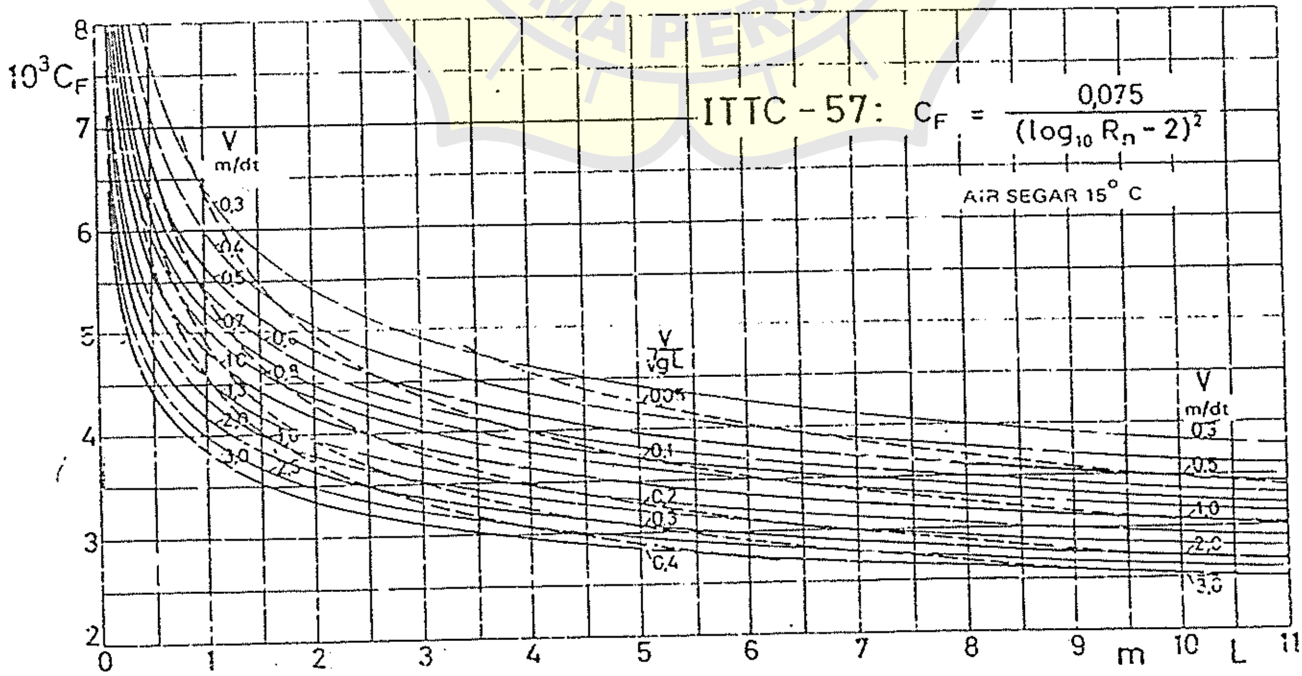
$$F_n = \frac{V}{\sqrt{gL}} \qquad (5.5.9)$$

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

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

$$\varphi = \frac{\nabla}{LBT\beta} \qquad (5.5.10)$$

B adalah lebar, T sarat, dan β koefisien penampang melintang tengah kapal.



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

Referensi.No.3

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

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

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

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

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

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

Dalam hal ini koefisien tahanan totalnya adalah

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

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

C_F = koefisien tahanan gesek dan dapat dihitung dengan memakai

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

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

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

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

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

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

B/T

Karena diagram tersebut dibuat berdasarkan rasio lebar – sarat

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

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

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

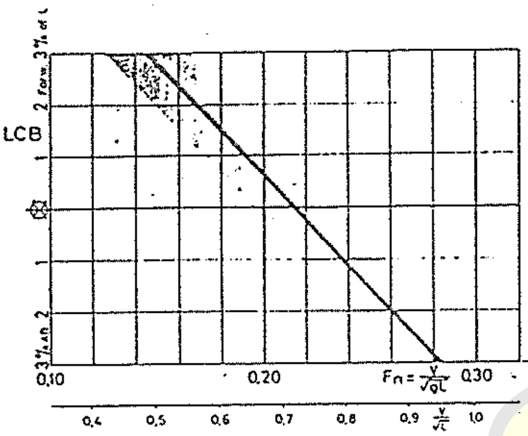
$$10^3 C_R = 10^3 C_{R(B/T=2,5)} + 0.16(B/T - 2{,}5) \qquad (5.5.17)$$

Koreksi ini dapat mempunyai harga yang negatif atau positif.

LCB

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

Referensi.No.3.



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

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

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

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

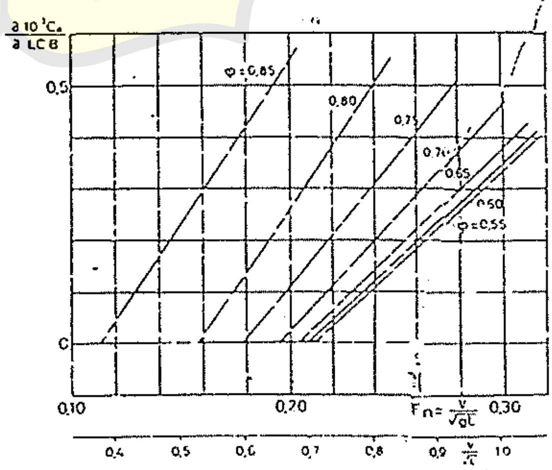
dengan faktor $\partial 10^3 C_R / \partial \mathrm{LCB}$. Harga faktor ini dapat diperoleh dari Gb. 5.5.16, dan ini hanya berlaku untuk LCB yang berada di depan $\mathrm{LCB}_{\mathrm{standar}}$. Mengenai LCB yang berada di belakang $\mathrm{LCB}_{\mathrm{standar}}$, semua sumber yang ada mempunyai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

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

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

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

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



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

Referensi.No.3.

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

BENTUK BADAN KAPAL (BENTUK PENAMPANG MELINTANG DAN HALUAN)

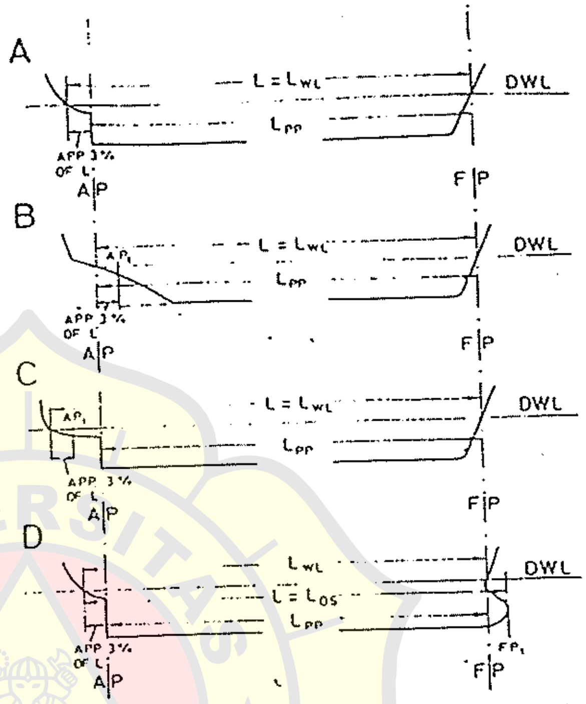
Sebagaimana disebutkan sebelumnya, kurva tahanan (yang diperoleh berdasarkan Gb. 5.5.5 – 5.5.13) dianggap berlaku untuk yang mempunyai bentuk "standar", yaitu penampangnya bukan yang benar-benar berbentuk U ataupun V. Karena itu, dalam menghitung daya efektif untuk perancangan awal umumnya tidak diperlukan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

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

(5.5.20)

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

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



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

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

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

F_n = 0,15	0,18	0,21 0,253	0,24 0,253	0,27	0,30	0,33	0,36	φ
		+0,2	0	−0,2	−0,4	−0,4	−0,4	0,50
		+0,2	0	−0,2	−0,3	−0,3	[illegible]	0,60
	+0,2	0	−0,2	−0,3	−0,3	[illegible]	[illegible]	0,70
+0,1	0	−0,2				[illegible]	[illegible]	0,80

(5.5.21)

Referensi.No.3.

ANGGOTA BADAN KAPAL

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

TAHANAN TAMBAHAN

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

$$
\begin{array}{llll}
\text{Untuk kapal dengan } L \leq & 100 \text{ m}, & 10^3 C_A = & 0{,}4 \\
= & 150 \text{ m} & = & 0{,}2 \\
= & 200 \text{ m} & = & 0 \\
= & 250 \text{ m} & = & -0{,}2 \\
\geq & 300 \text{ m} & = & -0{,}3
\end{array}
\tag{5.5.23}
$$

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

$$
\begin{array}{lll}
\text{Displasemen} & & \\
1.000 \text{ t} & C_A = & 0{,}6 \times 10^{-3} \\
10.000 \text{ t} & = & 0{,}4 \times 10^{-3} \\
100.000 \text{ t} & = & 0 \\
1.000.000 \text{ t} & = & -0{,}6 \times 10^{-3}
\end{array}
\tag{5.5.24}
$$

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

ANGGOTA BADAN KAPAL

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

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

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

TAHANAN UDARA DAN TAHANAN KEMUDI

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

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

Koreksi untuk tahanan kemudi mungkin sekitar

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

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

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

KONDISI PELAYARAN DINAS

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

Referensi.No.3.

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

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

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

Jalur pelayaran Pasifik, 15 – 30%

Jalur pelayaran Atlantik Selatan dan Autralia, 12 – 18%

Jalur pelayaran Asia Timur, 15 – 20%

Tahanan total harus dihitung dengan memakai rumus

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

S adalah luas permukaan basah badan kapal.

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

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

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

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

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

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

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

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

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

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

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

Mesin turbin mempunyai tingkatan menurut jenisnya.

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

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

Referensi.No.4.

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,

$$N_u = \frac{GH}{60\times60\times75}\ \text{hp} = \frac{GH}{60\times60\times102}\ \text{kW}$$

$$N_u = \frac{Q\gamma H}{60\times60\times75}\ \text{hp} = \frac{Q\gamma H}{60\times60\times102}\ \text{kW}$$

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

$$Q_b = 0.2825 d_b^2 v_b \text{ cu m per hr} \qquad (11)$$

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

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

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

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

Referensi.No.4.

(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

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

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

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

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

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

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.

Referensi.No.4.

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

$$\frac{p_{st}Q_{st}}{1+\alpha t_{st}}=\frac{p_aQ_a}{1+\alpha t_a}$$

whence

$$Q_{st}=\frac{(1+\alpha t_{st})\,p_aQ_a}{p_{st}(1+\alpha t_a)}=Q_a\frac{\left(1+\frac{1}{273}20\right)}{\left(1+\frac{1}{273}t_a\right)}\frac{p_a}{760}=$$
$$=Q_a\frac{293}{273+t_a}\frac{p_a}{760}\text{ cu m per hour} \tag{276}$$

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

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

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-sec^2 per m^4.

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

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

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.

Referensi.No.4.

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

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

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

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

b_2 = width of the impeller at air outlet

D_2 = impeller diameter at air outlet

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

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

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

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

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

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

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

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

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

and

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

where F = area of the impeller, sq m

D_2 = outside diameter of the impeller, m.

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

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

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

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

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

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

$$H = \overline{H}_k u_2^2 \rho \text{ mmH}_2\text{O}$$

Referensi.No.4.

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

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

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

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

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

The main initial data required to determine the principal dimensions of steering gears are the rudder characteristic, 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 $2\alpha° = 70°$	Speed of rudder movement, deg/sec, for rudder angle of $2\alpha° = 64°$
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

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

$$n_m = i_{sg} n_{rs} \text{ rpm} \tag{313}$$

where n_m = 100 to 350 rpm for steam engines

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

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

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

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

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

It follows from formula (314) that

$$n_{rs} = \frac{30\omega_{rs}}{\pi} \text{ rpm} \tag{316}$$

Combining equations (315) and (316) we obtain

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

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

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

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

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

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

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

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

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

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

Referensi.No.4.

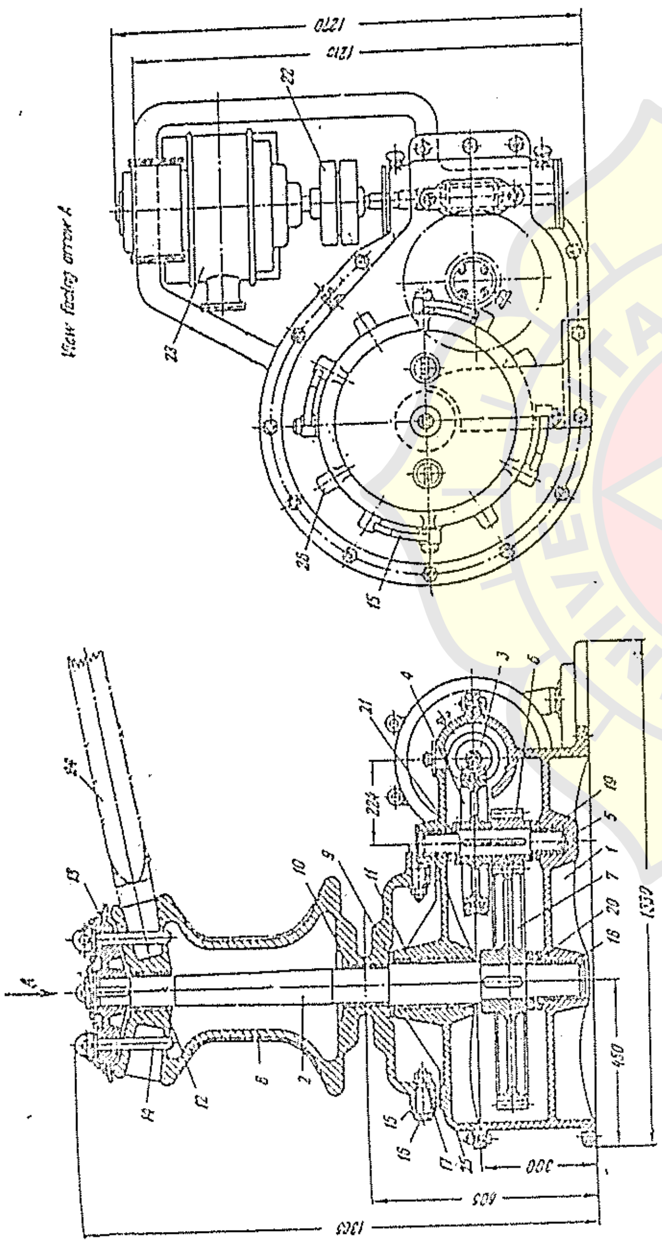


Fig. 169.

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

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

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

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

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

This equation can be solved for V_e and V_f:

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

and

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

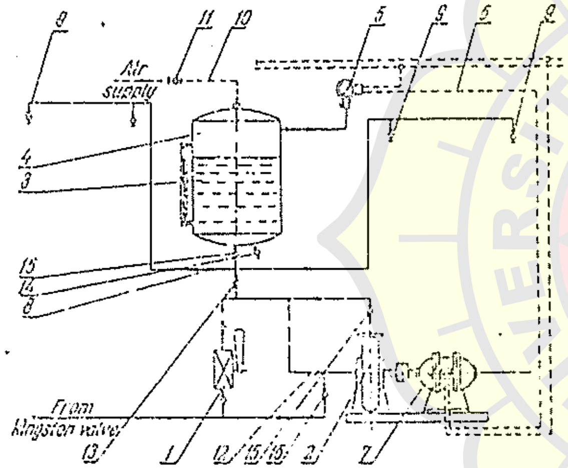


Fig. 189.

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

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

Therefore the minimum and maximum volumes of the air are

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

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

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

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

(b) SANITARY AND SCUPPER SYSTEMS

The sanitary and scupper systems serve to remove water from the deck and also to dispose of used water from baths, lavatories, 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

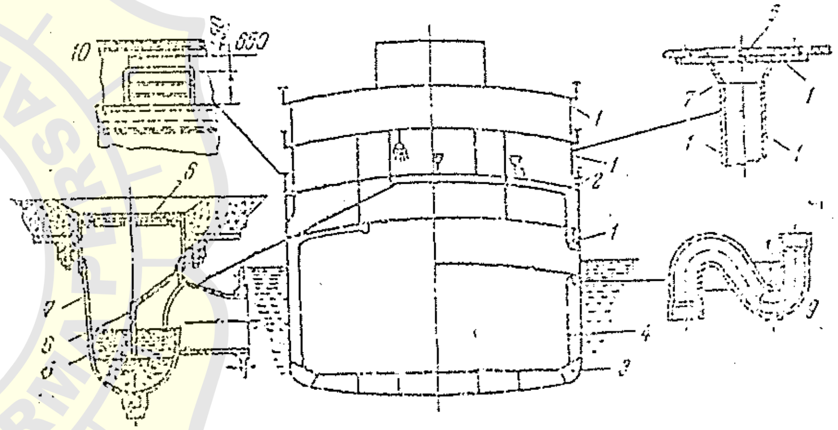


Fig. 190.

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.

Referensi.No.4

Referensi.No.4.

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

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

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

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

Table 58

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

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

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

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

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

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

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

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

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

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

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

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

$$\chi_{sp} = k_{1sp}\Sigma l_{sp}h_{sp}$$

where k_{1sp} = 0.75 if the total length of the superstructures is equal to or less than 0.5 L

$k_{1sp} = 1.5\frac{l_{sp}}{L}$ if the total length of the superstructures exceeds 0.5 L

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

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

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

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

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

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

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

$$\chi_q = l_q h_q$$

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

A section taken through the central plane of the usual five-snu cable lifter (Fig. 170) perpendicular to the shaft will be a regula of gravity are not yet exactly known in the early project stage. If the model does not accomplish the required speed the designer has to alter the hull. This alteration, however, is possible in the early project stage only. If the trial speed in ballast condition corresponds to the model trial speed in ballast, it can be assumed that service speed in loaded condition is attained, too.

Service speed of a ship is smaller than trial speed because of:

- increase of resistance by wind more than Beaufort 2
- increase of resistance by seaway
- increase of resistance by fouling on shell plating.

In general

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

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

7.6. Consumables and tanks

There are some more special requirements in ship design: Capacities of

- consumables
- provisions
- ballast.

a) consumables are (depending on type of engine plant, time for one round trip, number of crew members):

- fuel oil

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

last brackets for reserve:

- fuel rests in tanks
- seaway
- wind
- waiting time

(- according to owner's desire).

Referensi.No.6.

P_{Bme} = break horsepower of the main engine [KW]

b_{me} = specific fuel oil consumption main engine [g/KW·h]

P_{ae} total power of auxiliary engines [KW]

b_{ae} = specific fuel oil consumption auxiliary engines [g/KW·h]

S = operating range [sm]

V_{serv} = speed [kn]

1 KW = 0.736 PS (BHP).

Motors:

specific fuel oil consumption:

for two-stroke engines $b = 205 \ldots 211$ [g/KW·h]

for four-stroke engines with cylinder power more than 300 KW

$b = 196 \ldots 209$ [g/KW·h]

for full power: addition 5%

for diesel fuel: reduction 5% (dependent on heating value of diesel fuel)

For steam turbines:

Standard circulation without furnace gas reheat

livesteam: 64 ... 82 bar at 513 ... 538°C

$b = 278 \ldots 286$ [g/KW·h]

with furnace gas reheat

livesteam: 80 ... 110 bar at 513 ... 538°C

$b = 252 \ldots 265$ [g/KW·h]

For gas turbines:

Gasoline and light crude oils

$b = 299 \ldots 312$ [g/KW·h]

Specific weight of heavy fuel oil: $\gamma = 0.95\ t/m^3$

Required volume of storage tanks

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

Referensi.No.6.

Additions to the volume:

2% for double bottom tanks
1 ... 2% for top tanks and deep tanks
2% for thermal expansion, i.e. 98% filled only.

Diesel oil

used for auxiliary engines and for the main engine during estuary trading.

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

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

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

additions see fuel oil!

Lubrication oil

In general ships have about 30 ... 50 t lubrication oil, because otherwise the tanks will get too small. (According to owner's desire!).

$$w_{lubr.} = P_{Bme} \cdot b_{me} \cdot \frac{S}{V_{serv}} + \text{addition}$$

b = 0.8 ... 1.2 [g/KW·h] diesel engine two stroke
b = 1.2 ... 1.6 [g/KW·h] diesel engine four stroke
b = 0.14 [g/KW·h] turbines and gearboxes

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

Fresh water

- drinking water 10 ... 20 kg/pers · day
- washing water 60 kg/pers · day without bathing room
  up to 200 kg/pers · day with bathing room
- boiler feed water 0.14 kg/KW·h plus first filling

additions to the tank volume: 3 ... 4% for special coatings in case of fresh water

Fresh water tanks have to be separated from all other tanks

h) Ballast capacity used for

- trim (immersion of propeller; resistance)
- providing of sufficient stability (at the end of the voyage)
- heeling (heavy lift vessels; RoRo-vessels; container ships, because of container guides)
- longitudinal strength (bulker, tanker)
- immersion of ship (tanker, to avoid heavy motions in seaway; therefore light or heavy ballast).

Ballast capacity to be provided depending on ship type and on desires of the owner: between 10% and 50% of deadweight.

Additions to required ballast tank volumina are larger at the ends of the ship.

- +5% lower fore peak tank
- +3% upper fore peak tank
- +2% double bottom tank.

The new IMCO-rules recommend segregated ballast tanks to avoid pollution. Cargo oil tanks are separated from the ballast tank system. The economy decreases and more tank capacity is needed.

Sounding/ullage tables delivered by yard.

i) Provisions/persons/luggage

Weight of provisions	3 ... 5 kg/pers · day
weight of persons	75 kg (crew and passengers)
weight of luggage	20 kg/pers (short distance)
	60 kg/pers (long distance passenger and crew).

Type and Location of Main Engine

Is another part of the contract influencing ship design. (Ship weight, volume, fuel consumption). Economy is determined by the choice of the main engine type, also by the quality of the crew (maintenance). The degree of possible automation depends on the personal quality as well. Sometimes the choice of the engine depends on the route because of maintenance and engine maker.

8. Crew Members

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

9. Outfit and Equipment

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

10. Classification, Rules

have to be observed.

11. Restrictions of Dimensions

- Draught (because of port depth, estuary trading, canals)
- breadth (canals, locks)
- length (locks, length of berth)
- stability requirements.

12. Tonnage of Ships

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

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

Tonnage unit: gross ton

1 grt = 100 cbf = 2.83 m^3.

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

Gross tonnage

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

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

Net tonnage

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

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

T = draught in $[m]$ (midships)

D = depth in $[m]$ "

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

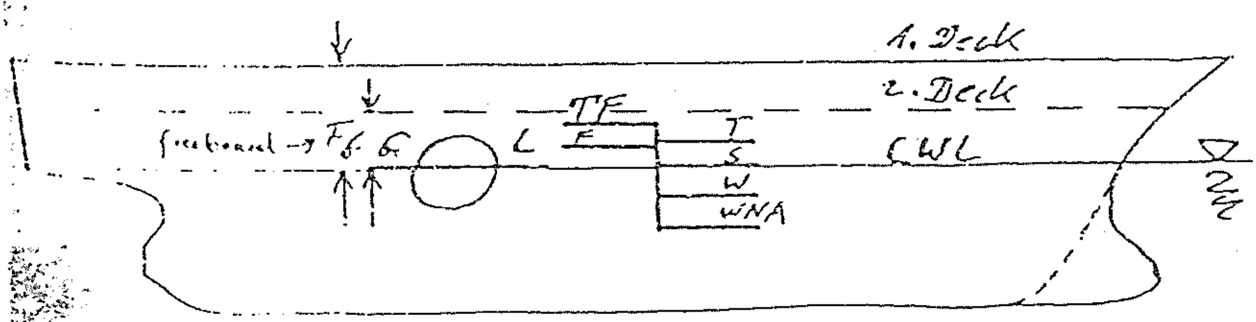
N_2 number of other passengers.

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

Definition of freeboard:

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

Lit.: International Freeboard Convention 1966.



Definition of superstructures:

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

Referensi.No.7.

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

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

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

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

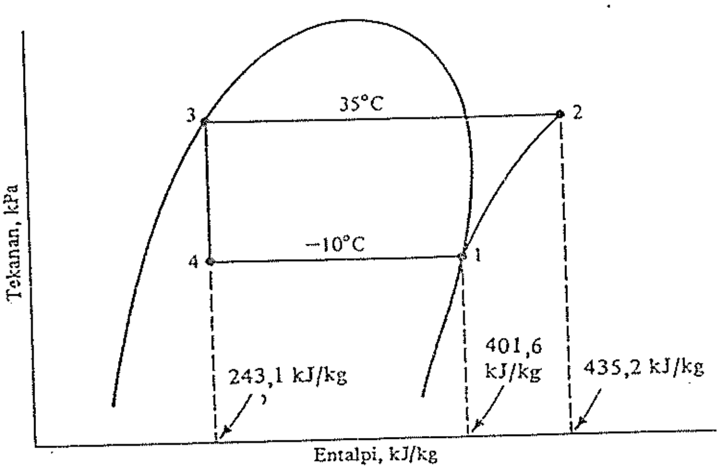
Untuk menemukan h_2 melalui garis entropi tetap geser titik 1 hingga mencapai tekanan jenuh yang sesuai dengan suhu 35°C. Tekanan pengembunan ini adalah 1354 kPa, dan nilai h_2 = 435,2 kJ/kg.

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

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

(*a*) Dampak refrigerasi:

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



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

Referensi.No.7.

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

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

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

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

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

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

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

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

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

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

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

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

10-15 Penukar kalor (heat exchangers) Beberapa sistem refrigerasi dilengkapi dengan penukar kalor jalur cair-ke-hisap (liquid-to-suction), yang menurunkan suhu (subcools) cairan dari 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

Referensi.No.8.

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 keseluruhan (biasanya dinyatakan dalam %).
L_{ad}: Daya adiabatik teoritis (kW)
L_s: Daya yang masuk pada poros kompresor (kW).

Besarnya daya adiabatik teoritis dapat dihitung dengan rumus

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

P_s: Tekanan isap tingkat pertama (kgf/m^2 abs)
P_d: Tekanan keluar dari tingkat terakhir (kgf/m^2 abs)
Q_s: Jumlah volume gas yang keluar dari tingkat terakhir (m^3/min) dinyatakan pada kondisi tekan dan temperatur isap
k: c_p/c_v
m: Jumlah tingkat kompresi; lihat keterangan pada Pers. (2.16).

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

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

Dalam Tabel 2.7 diberikan harga-harga daya adiabatik teoritis yang diperlukan untuk mengkompresikan 1 m^3/min udara dengan kondisi standar sebagai hasil perhitungan berdasarkan rumus di atas. Dari tabel terlihat bahwa daya yang diperlukan untuk kompresi 2 tingkat harganya lebih kecil dari pada kompresi 1 tingkat. Harga yang lebih rendah ini diperoleh pada kompresor 2 tingkat yang menggunakan pendingin antara (inter-cooler) di antara tingkat pertama dan tingkat ke dua. Penggunaan pendingin antara akan memperkecil kerja kompresi. Jika tidak digunakan pendingin antara, maka daya yang diperlukan untuk kompresi 2 tingkat adalah sama besarnya dengan daya untuk 1 tingkat, pada perbandingan tekanan yang sama.

Sebagai contoh, dari Tabel 2.7 terbaca bahwa untuk kompresi 1 tingkat sampai 7 kgf/cm^2 (g) atau 8,033 kgf/cm^2 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/cm^2 (g) diperlukan daya poros sebesar 5,4 kW, maka dengan daya adiabatik teoritis sebesar 4,022 kW, kompresi ini mempunyai efisiensi adiabatik keseluruhan sebesar

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

Referensi.No.8.

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

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

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

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

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

di mana ε: V_c/V_s, volume sisa (clearance) relatip,

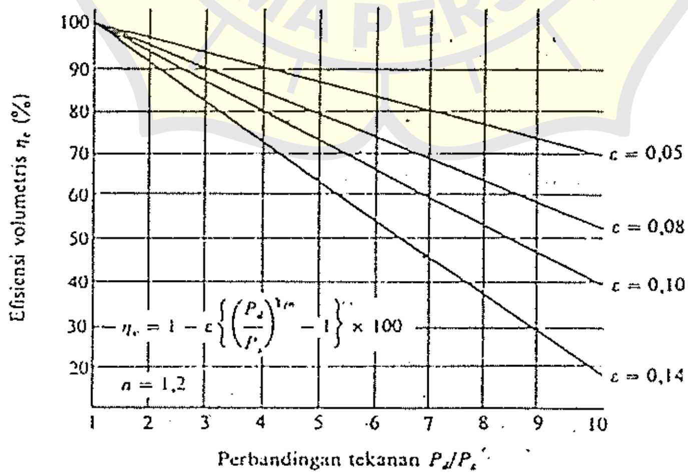
P_d: Tekanan keluar dari silinder tingkat pertama (kgf/cm^2 abs),

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

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

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

Dalam Gb. 2.11 diperlihatkan pengaruh ε dan P_d/P_s pada efisiensi volumetris η_v. Sehubungan dengan hal-hal di atas dapat dimengerti jika efisiensi volumetris juga tergantung pada faktor-faktor rancangan kompresor seperti bentuk dan ukuran silinder, serta bentuk, ukuran, dan susunan katup-katup.



Gb. 2.11 Efisiensi volumetris dan perbandingan tekanan.

2.4.2 Efisiensi adiabatik keseluruhan

Efisiensi kompresor ditentukan oleh berbagai faktor seperti tahanan aerodinamik di dalam katup-katup, saluran-saluran, pipa-pipa, kerugian mekanis, efektivitas pen-

Referensi.No.8.

Tabel 2.7 Daya yang diperlukan untuk kompresi adiabatis teoritis.

Tekanan (kgf/cm^2 (G))	Kompresi 1-tingkat (kW)	Kompresi 2-tingkat (kW)	Tekanan (kgf/cm^2 (G))	Kompresi 2-tingkat (kW)
0,5	0,7053		11	4,9639
1	1,2608		12	5,1563
1,5	1,7256		13	5,3365
2	2,1288		14	5,5060
2,5	2,4869		15	5,6661
3	2,8105		16	5,8178
3,5	3,1065		17	5,9621
4	3,3801	2,9994	18	6,0997
4,5	3,6348	3,2012	19	6,2313
5	3,8736	3,3879	20	6,3573
5,5	4,0987	3,5618	21	6,4783
6	4,3118	3,7247	22	6,5947
6,5	4,5143	3,8779	23	6,7068
7	4,7074	4,0227	24	6,8150
7,5	4,8922	4,1599	25	6,9195
8	5,0693	4,2904	26	7,0215
8,5	5,2396	4,4148	27	7,1195
9	5,4036	4,5338	28	7,1246
9,5	5,5619	4,6477	29	7,3069
10	5,7149	4,7572	30	7,3965

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

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

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

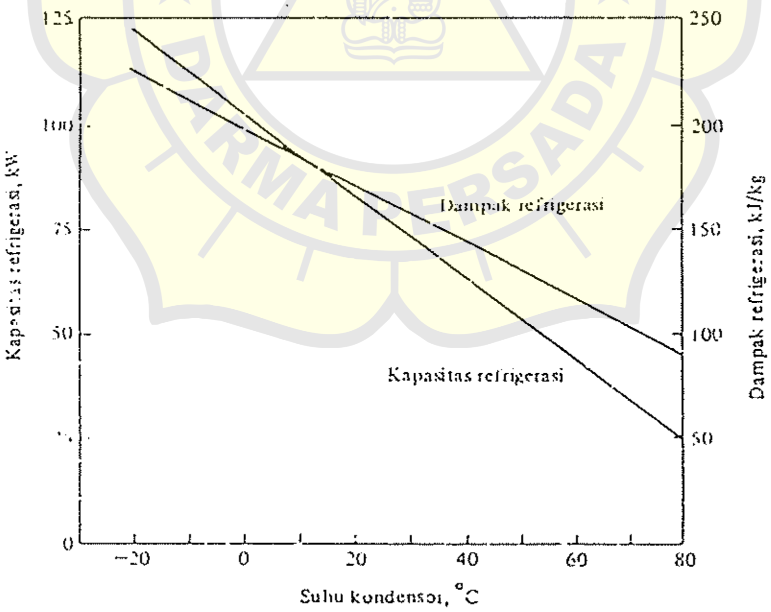
Sebagai kesimpulan dapat dikemukakan bahwa efisiensi diabatik keseluruhan merupakan petunjuk bagi baik buruknya performansi dan ekonomi sebuah kompresor. Adapun efisiensi volumetris hanya merupakan suatu koefisien yang diperlukan oleh perencana kompresor dan tidak penting artinya bagi pemakai.

Referensi.No.8.

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

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

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



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

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

Lampiran. I.

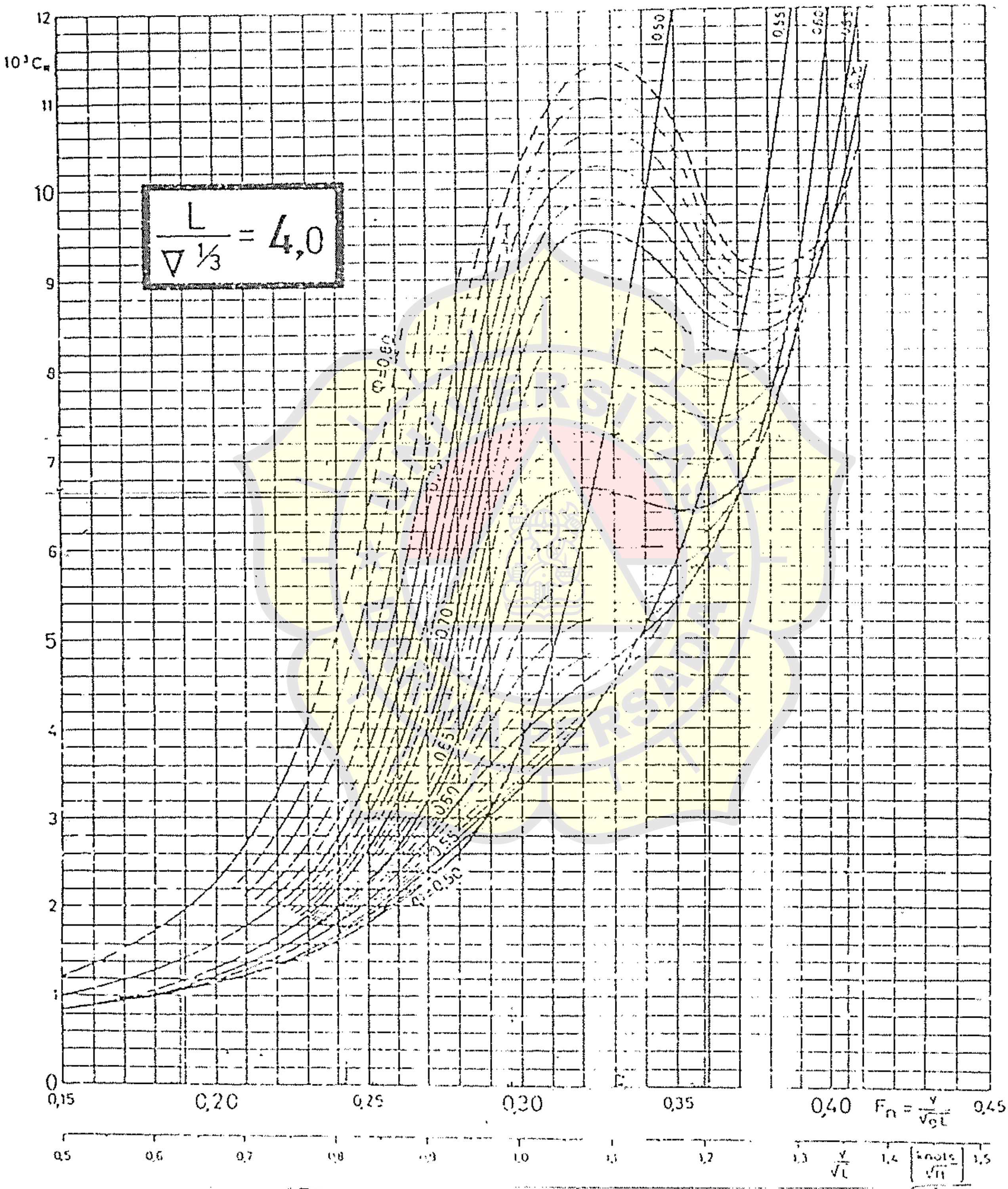


Diagram Koefisien tahanan sisa terhadap rasio kecepatan-panjang untuk harga koefisien prismatik longitudinal yang berbeda-beda.

Lampiran.2.

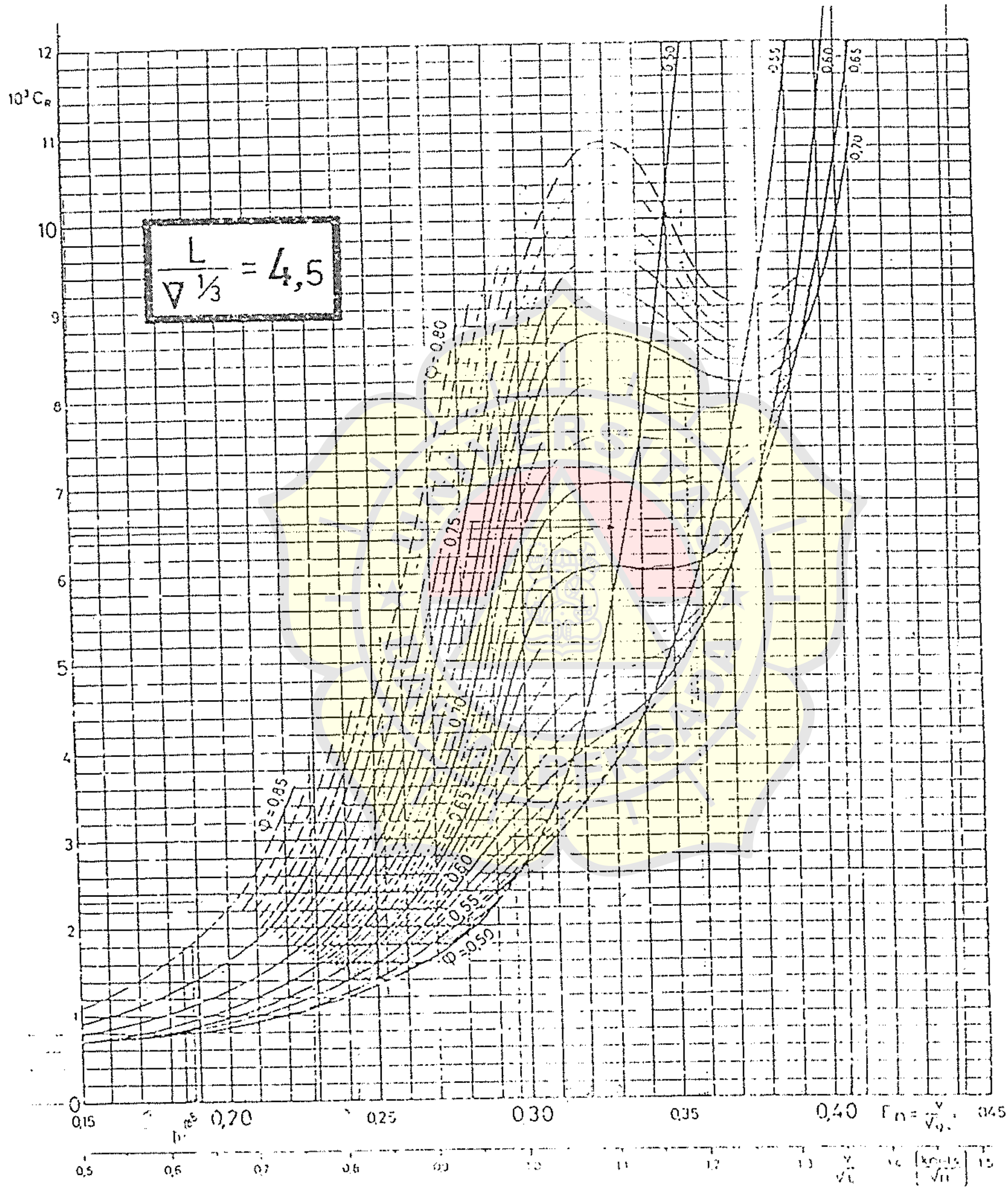
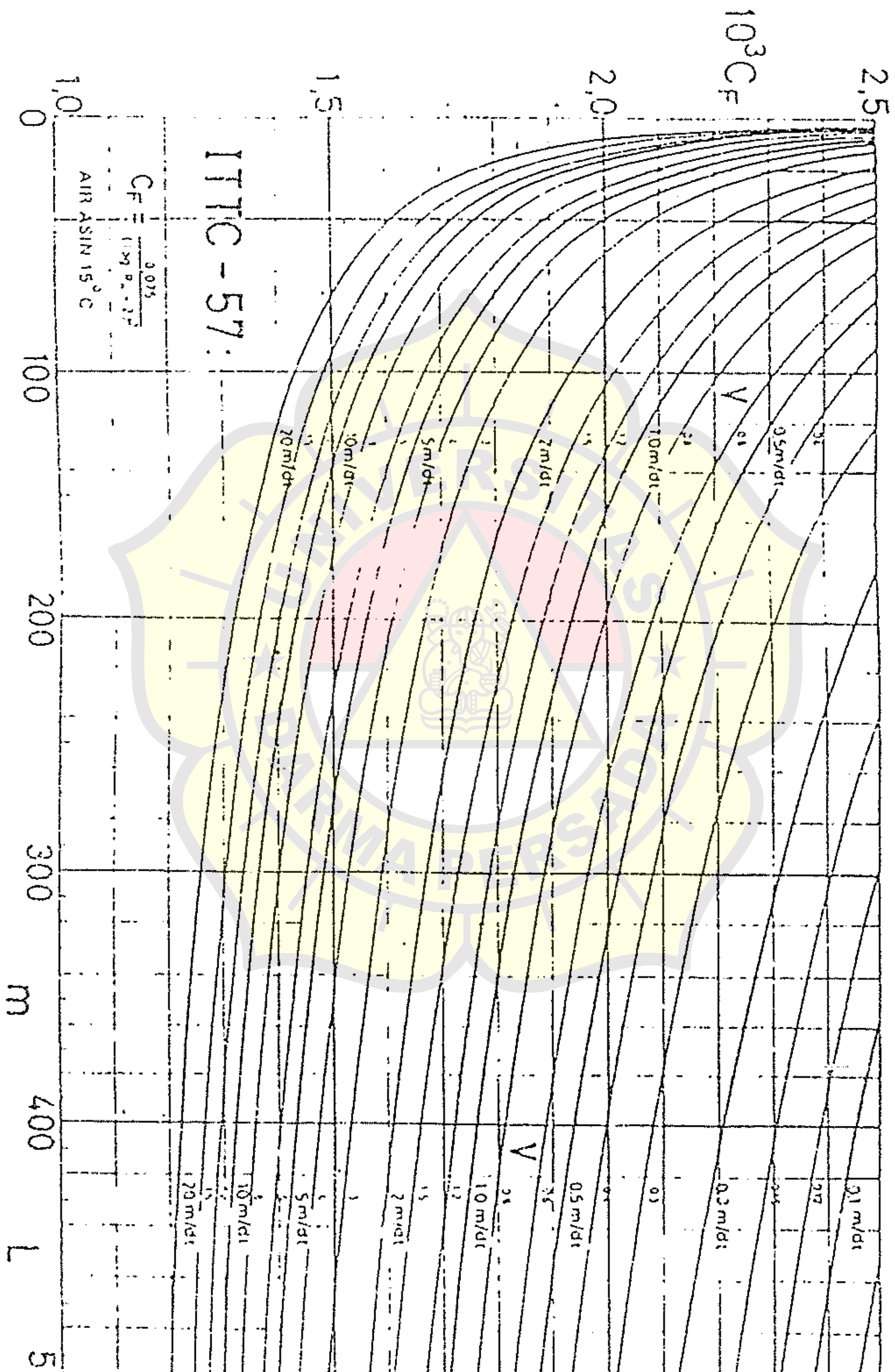


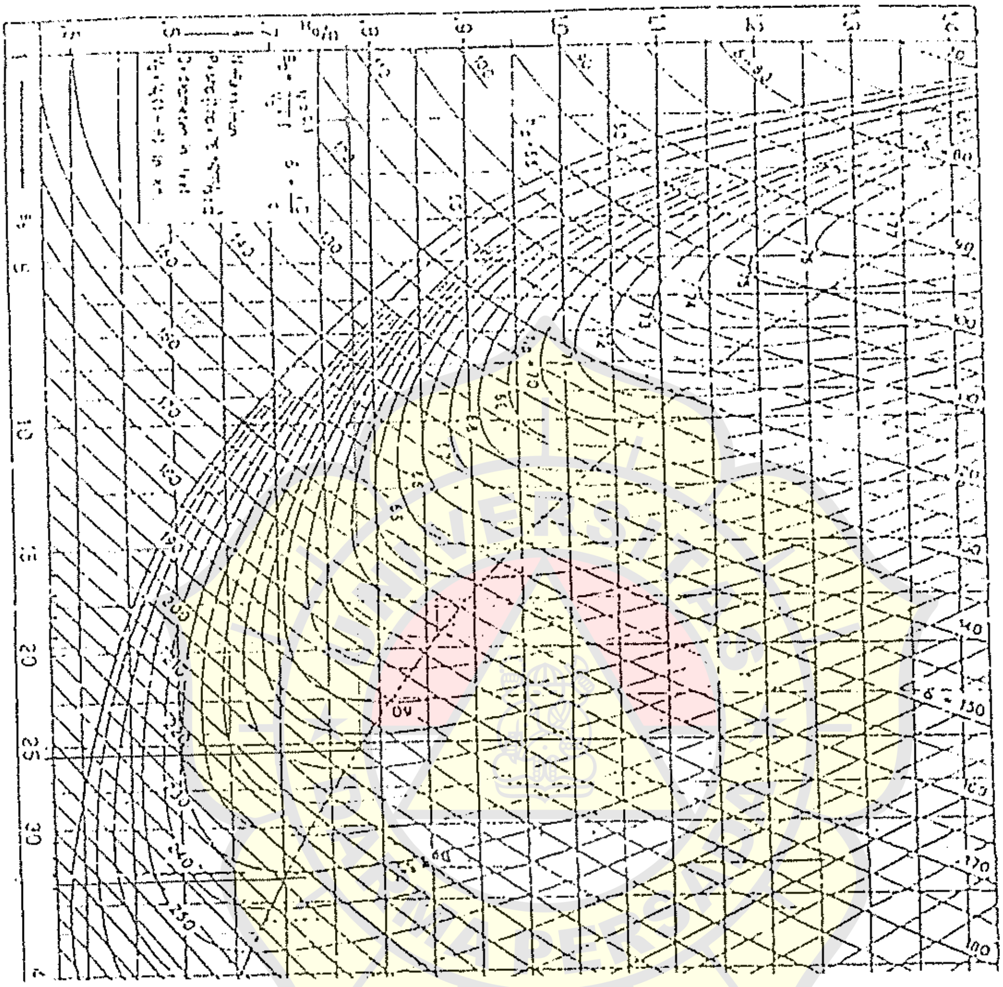
Diagram Koefisien tahanan sisa terhadap rasio kecepatan-panjang untuk harga koefisien prismatik longitudinal yang berbeda-beda.

Lampiran.3.



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

Lampiran.4.



Lampiran.5.

Tabel B.1. Harga Konduktivitas Thermal Beberapa Material

Material	Description	Thermal Conductivity (k) W/m K	Thermal Conductance (C) W/m² K
Masonry	Brick, common	0.72	
	Brick, face	1.30	
	Concrete, mortar or plaster	0.72	
	Concrete, sand aggregate	1.73	
	Concrete block		
	Sand aggregate 100 mm		7.95
	Sand aggregate 200 mm		5.11
	Sand aggregate 300 mm		4.43
	Cinder aggregate 100 mm		5.11
	Cinder aggregate 200 mm		3.29
	Cinder aggregate 300 mm		3.01
	Gypsum plaster 13 mm		17.72
	Tile, hollow clay 100 mm		5.11
	Tile, hollow clay 150 mm		3.75
	Tile, hollow clay 200 mm		3.07
Woods	Maple, oak, similar hardwoods	0.16	
	Fir, pine, similar softwoods	0.12	
	Plywood 13 mm		9.09
	Plywood 19 mm		6.08
Roofing	Asphalt roll roofing		36.91
	Built-up roofing 9 mm		11.03
Insulating materials	Blanket or batt, mineral or glass fiber	0.039	
	Board or slab		
	Cellular glass	0.058	
	Corkboard	0.043	
	Glass fiber	0.036	
	Expanded polystyrene (smooth)	0.029	
	Expanded polystyrene (cut cell)	0.036	
	Expanded polyurethane	0.025	
	Loose fill		
	Milled paper or wood pulp	0.039	
	Sawdust or shavings	0.065	
	Mineral wool (rock, glass, slag)	0.039	
	Redwood bark	0.037	
	Wood fiber (soft woods)	0.043	
Surface conductance (convection coefficient)	Still air		9.37
	Moving air (3.35 m/s or 12 km/h)		22.70
	Moving air (6.7 m/s or 24 km/h)		34.10
Glass	Single pane		6.42
	Two pane		2.61
	Three pane		1.65
	Four pane		[illegible]

Adapted from ASHRAE Data Book [illegible] 1972 Edition, by permission of [illegible] American Society of Heating, Refrigerating and Air-Conditioning Engineers

Lampiran.6.

Tabel C.1. Sifat Cairan dan Gas Jenuh Refrigeran 134a

TEMP °C	PRES bar	VOLUME m³/kg ×10⁻³		DENSITY kg/m³		ENTHALPY kJ/kg			ENTROPY kJ/kg K		TEMP °C
		LIQUID v_f	VAPOR v_g	LIQUID $1/v_f$	VAPOR $1/v_g$	LIQUID h_f	LATENT h_{fg}	VAPOR h_g	LIQUID s_f	VAPOR s_g	
0	4.9759	0.77834	47.1354	1.28479	0.02122	200.000	205.361	405.361	1.00000	1.75175	0
1	5.1401	0.78041	45.6757	1.28139	0.02189	201.174	204.550	405.724	1.00424	1.75034	1
2	5.3083	0.78249	44.2702	1.27797	0.02259	202.351	203.733	406.084	1.00848	1.74889	2
3	5.4806	0.78460	42.9166	1.27453	0.02330	203.530	202.910	406.440	1.01271	1.74746	3
4	5.6571	0.78673	41.6124	1.27108	0.02403	204.713	202.080	406.793	1.01694	1.74604	4
5	5.8378	0.78889	40.3556	1.26761	0.02478	205.899	201.243	407.142	1.02116	1.74463	5
6	6.0228	0.79107	39.1447	1.26412	0.02555	207.089	200.400	407.489	1.02537	1.74324	6
7	6.2122	0.79327	37.9759	1.26061	0.02633	208.281	199.550	407.831	1.02958	1.74185	7
8	6.4059	0.79549	36.8490	1.25709	0.02714	209.477	198.693	408.169	1.03379	1.74047	8
9	6.6042	0.79775	35.7624	1.25353	0.02796	210.675	197.829	408.504	1.03799	1.73911	9
10	6.8070	0.80002	34.7136	1.24997	0.02881	211.877	196.958	408.835	1.04218	1.73775	10
11	7.0144	0.80232	33.7013	1.24638	0.02967	213.083	196.079	409.162	1.04637	1.73640	11
12	7.2265	0.80465	32.7239	1.24277	0.03056	214.291	195.194	409.485	1.05056	1.73506	12
13	7.4433	0.80701	31.7801	1.23915	0.03147	215.503	194.301	409.804	1.05474	1.73373	13
14	7.6650	0.80939	30.8683	1.23550	0.03240	216.719	193.400	410.119	1.05892	1.73241	14
15	7.8915	0.81180	29.9874	1.23183	0.03335	217.937	192.492	410.430	1.06309	1.73109	15
16	8.1229	0.81424	29.1361	1.22813	0.03432	219.160	191.577	410.736	1.06726	1.72978	16
17	8.3593	0.81671	28.3131	1.22442	0.03532	220.385	190.653	411.038	1.07142	1.72848	17
18	8.6008	0.81922	27.5170	1.22068	0.03634	221.615	189.721	411.336	1.07559	1.72719	18
19	8.8475	0.82175	26.7477	1.21692	0.03739	222.848	188.782	411.629	1.07974	1.72590	19
20	9.0993	0.82431	26.0032	1.21313	0.03846	224.084	187.834	411.918	1.08390	1.72462	20
21	9.3564	0.82691	25.2829	1.20932	0.03955	225.324	186.877	412.202	1.08805	1.72334	21
22	9.6189	0.82954	24.5857	1.20548	0.04067	226.566	185.913	412.481	1.09220	1.72206	22
23	9.8867	0.83221	23.9107	1.20162	0.04182	227.816	184.939	412.755	1.09634	1.72080	23
24	10.160	0.83491	23.2572	1.19773	0.04300	229.068	183.957	413.025	1.10048	1.71953	24
25	10.439	0.83765	22.6242	1.19382	0.04420	230.324	182.965	413.289	1.10462	1.71827	25
26	10.723	0.84043	22.0111	1.18987	0.04543	231.583	181.965	413.548	1.10876	1.71701	26
27	11.014	0.84324	21.4169	1.18590	0.04669	232.847	180.955	413.802	1.11290	1.71576	27
28	11.309	0.84610	20.8411	1.18190	0.04798	234.115	179.935	414.050	1.11703	1.71450	28
29	11.611	0.84899	20.2829	1.17787	0.04930	235.387	178.906	414.293	1.12116	1.71325	29
30	11.919	0.85193	19.7417	1.17381	0.05065	236.664	177.867	414.530	1.12530	1.71200	30
31	12.232	0.85491	19.2168	1.16971	0.05204	237.944	176.817	414.762	1.12943	1.71075	31
32	12.552	0.85793	18.7076	1.16559	0.05345	239.230	175.758	414.987	1.13355	1.70950	32
33	12.878	0.86101	18.2135	1.16143	0.05490	240.520	174.687	415.207	1.13768	1.70826	33
34	13.210	0.86412	17.7341	1.15724	0.05639	241.814	173.606	415.420	1.14181	1.70701	34
35	13.548	0.86729	17.2686	1.15302	0.05791	243.114	172.514	415.627	1.14594	1.70576	35
36	13.892	0.87051	16.8168	1.14875	0.05946	244.418	171.410	415.828	1.15007	1.70450	36
37	14.243	0.87378	16.3779	1.14445	0.06106	245.727	170.294	416.021	1.15420	1.70325	37
38	14.601	0.87710	15.9517	1.14012	0.06269	247.041	169.167	416.208	1.15833	1.70199	38
39	14.965	0.88048	15.5375	1.13574	0.06436	248.361	168.027	416.388	1.16246	1.70073	39
40	15.335	0.88392	15.1351	1.13133	0.06607	249.686	166.875	416.561	1.16659	1.69946	40
41	15.712	0.88741	14.7439	1.12687	0.06782	251.016	165.710	416.726	1.17073	1.69819	41
42	16.096	0.89097	14.3636	1.12237	0.06962	252.352	164.531	416.883	1.17486	1.69692	42
43	16.487	0.89459	13.9979	1.11783	0.07144	253.694	163.339	417.033	1.17900	1.69564	43
44	16.885	0.89828	[illegible]	1.11324	0.07332	255.042	162.133	417.174	1.18315	1.69435	44
45	17.290	0.90203	13.2841	1.10861	0.07528	256.396	160.912	417.308	1.18730	1.69305	45
46	17.702	0.90586	12.9436	1.10392	0.07726	257.756	159.676	417.432	1.19145	1.69175	46
47	18.121	0.90976	12.6122	1.09919	0.07929	259.123	158.425	417.548	[illegible]	[illegible]	47
48	18.548	0.91374	12.2895	1.09441	0.08137	260.497	157.156	417.653	[illegible]	[illegible]	48
49	18.982	0.91779	11.9753	1.08958	0.08351	261.877	155.875	417.752	[illegible]	[illegible]	49

Courtesy E.I. du Pont de Nemours & Co., Inc.

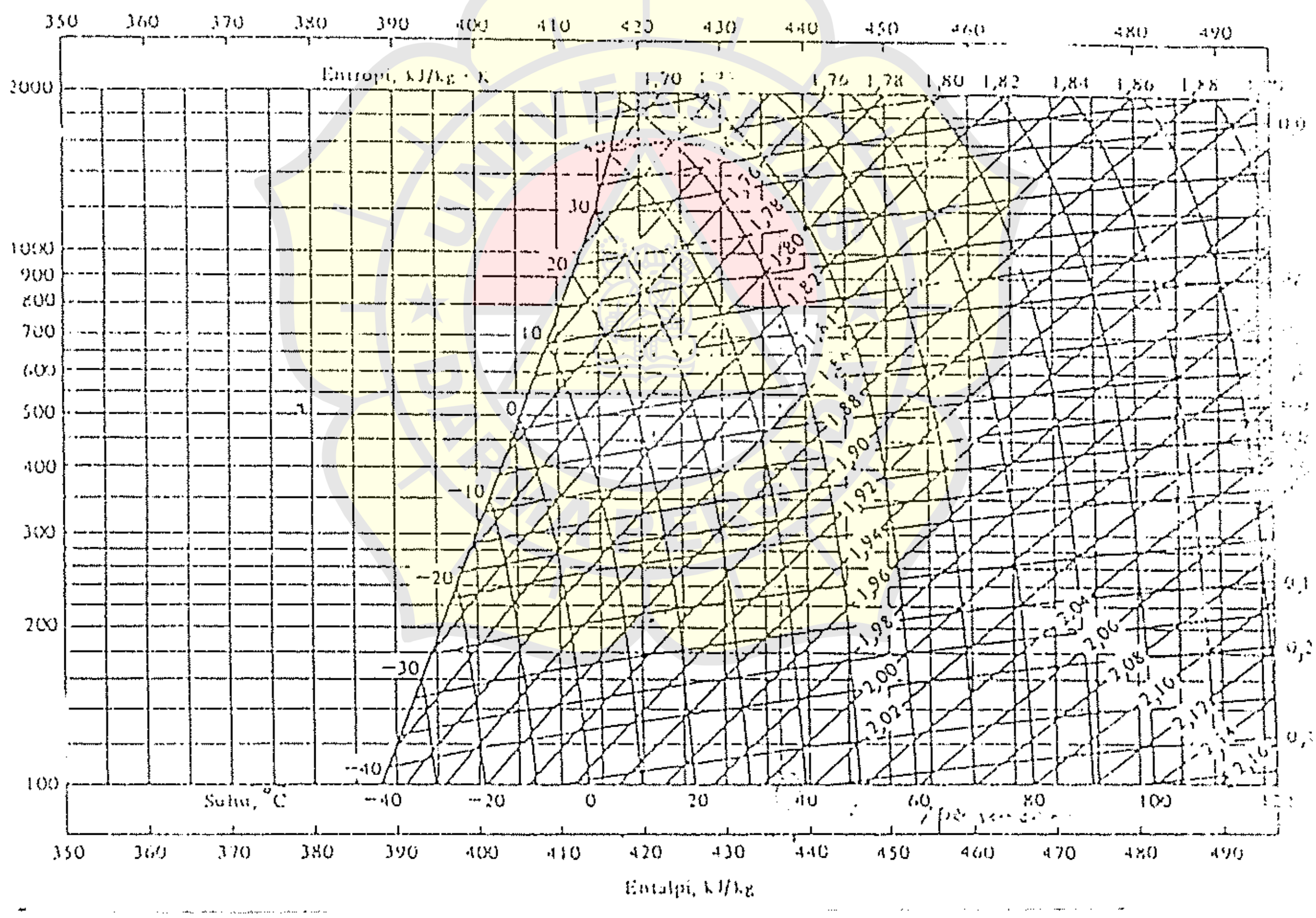
Lampiran.7.

Lanjutan Tabel C.1. Sifat Cairan dan Gas Jenuh Refrigeran 134a

TEMP °C	PRES bar	VOLUME m^3/kg 10^3		DENSITY kg/m^3 10^3		ENTHALPY kJ/kg			ENTROPY kJ/kg K		TEMP °C
		LIQUID v_f	VAPOR v_g	LIQUID $1/v_f$	VAPOR $1/v_g$	LIQUID h_f	LATENT h_{fg}	VAPOR h_g	LIQUID s_f	VAPOR s_g	
-50	0.6439	0.69526	324.557	1.43831	0.00308	144.959	238.962	383.921	0.77919	1.85000	-50
-49	0.6776	0.69663	309.466	1.43549	0.00323	145.992	238.403	384.395	0.78380	1.84734	-49
-48	0.7128	0.69800	295.217	1.43266	0.00339	147.029	237.840	384.869	0.78841	1.84472	-48
-47	0.7494	0.69939	281.753	1.42982	0.00355	148.067	237.274	385.341	0.79300	1.84214	-47
-46	0.7875	0.70078	269.027	1.42698	0.00372	149.109	236.704	385.813	0.79756	1.83959	-46
-45	0.8271	0.70219	256.9[illegible]	1.42412	0.00389	150.153	236.129	386.282	0.80216	1.83708	-45
-44	0.8682	0.70360	245.600	1.42126	0.00407	151.200	235.551	386.751	0.80672	1.83460	-44
-43	0.9110	0.70502	234.817	1.41839	0.00426	152.249	234.968	387.217	0.81128	1.83216	-43
-42	0.9555	0.70646	224.603	1.41551	0.00445	153.301	234.381	387.683	0.81583	1.82976	-42
-41	1.0016	0.70790	214.923	1.41262	0.00465	154.356	233.790	388.147	0.82036	1.82738	-41
-40	1.0495	0.70936	205.745	1.40972	0.00486	155.414	233.195	388.609	0.82490	1.82504	-40
-39	1.0992	0.71082	197.040	1.40682	0.00508	156.474	232.596	389.070	0.82942	1.82273	-39
-38	1.1507	0.71230	188.778	1.40390	0.00530	157.537	231.992	389.529	0.83383	1.82045	-38
-37	1.2041	0.71379	180.933	1.40097	0.00553	158.603	231.383	389.985	0.83844	1.81821	-37
-36	1.2594	0.71529	173.482	1.39804	0.00576	159.671	230.771	390.442	0.84294	1.81599	-36
-35	1.3168	0.71680	166.400	1.39510	0.00601	160.742	230.153	390.896	0.84743	1.81380	-35
-34	1.3761	0.71832	159.668	1.39214	0.00626	161.816	229.532	391.348	0.85191	1.81164	-34
-33	1.4375	0.71985	153.264	1.38918	0.00652	162.893	228.905	391.798	0.85638	1.80951	-33
-32	1.5011	0.72139	147.170	1.38620	0.00679	163.972	228.274	392.247	0.86085	1.80741	-32
-31	1.5668	0.72295	141.369	1.38322	0.00707	165.055	227.639	392.693	0.86531	1.80534	-31
-30	1.6348	0.72452	135.844	1.38022	0.00736	166.140	226.998	393.138	0.86976	1.80329	-30
-29	1.7050	0.72610	130.580	1.37722	0.00766	167.227	226.352	393.580	0.87420	1.80126	-29
-28	1.7776	0.72769	125.563	1.37420	0.00796	168.318	225.703	394.021	0.87863	1.79927	-28
-27	1.8525	0.72930	120.779	1.37116	0.00828	169.411	225.048	394.459	0.88306	1.79730	-27
-26	1.9299	0.73092	116.214	1.36812	0.00860	170.507	224.388	394.895	0.88748	1.79535	-26
-25	2.0098	0.73255	111.859	1.36509	0.00894	171.606	223.724	395.330	0.89190	1.79342	-25
-24	2.0922	0.73420	107.701	1.36203	0.00928	172.708	223.054	395.762	0.89630	1.79152	-24
-23	2.1772	0.73585	103.730	1.35896	0.00964	173.812	222.379	396.191	0.90070	1.78965	-23
-22	2.2648	0.73752	99.[illegible]	1.35588	0.01001	174.919	221.700	396.619	0.90509	1.78779	-22
-21	2.3552	0.73921	96.3101	1.35279	0.01038	176.029	221.015	397.044	0.90948	1.78596	-21
-20	2.4483	0.74091	92.8432	1.34968	0.01077	177.142	220.325	397.467	0.91386	1.78415	-20
-19	2.5442	0.74263	89.5273	1.34657	0.01117	178.258	219.629	397.887	0.91823	1.78236	-19
-18	2.6429	0.74436	86.3545	1.34344	0.01158	179.376	218.929	398.305	0.92259	1.78059	-18
-17	2.7446	0.74610	83.3179	1.34030	0.01200	180.498	218.223	398.721	[illegible]	1.77884	-17
-16	2.8493	0.74786	80.4103	1.33714	0.01244	181.622	217.512	399.134	[illegible]	1.77711	-16
-15	2.9570	0.74964	77.6254	1.33397	0.01288	182.749	216.795	399.544	0.93564	1.77540	-15
-14	3.0678	0.75143	74.9572	1.33079	0.01334	183.878	216.073	399.951	0.93997	1.77371	-14
-13	3.1817	0.75324	72.3997	1.32760	0.01381	185.011	215.345	400.356	0.94430	1.77204	-13
-12	3.2959	0.75506	69.9978	1.32439	0.01430	186.147	214.612	400.759	0.94862	1.77039	-12
-11	3.4193	0.75690	67.5951	1.32117	0.01479	187.285	213.873	401.158	0.95294	1.76875	-11
-10	3.5430	0.75876	65.3399	1.31794	0.01530	188.426	213.129	401.555	0.95725	1.76713	-10
-9	3.6701	0.76063	63.1746	1.31469	0.01583	189.571	212.379	401.949	0.96155	1.76553	-9
-8	3.8006	0.76253	61.1958	1.31143	0.01637	190.716	211.623	402.341	0.96585	1.76394	-8
-7	3.9347	0.76444	59.0956	1.30815	0.01692	191.868	210.861	402.729	0.97014	1.76237	-7
-6	4.0723	0.76637	57.1820	1.30486	0.01749	193.021	210.094	403.114	0.97442	1.76082	-6
-5	4.2135	0.76831	55.3394	1.30155	0.01807	194.176	209.320	403.496	0.97870	1.75928	-5
-4	4.3584	0.77028	53.5632	1.29823	0.01867	195.335	208.540	403.875	0.98297	1.75775	-4
-3	4.5070	0.77225	51.8653	1.29490	0.01929	196.497	207.755	404.252	0.98724	1.75624	-3
-2	4.6594	0.77427	50.2274	1.29154	0.01991	197.662	206.962	404.625	0.99150	1.75475	-2
-1	4.8157	0.77629	48.6517	1.28817	0.02055	198.829	206.165	404.994	0.99575	1.75326	-1

Lampiran.8.

Gambar C.1. Diagram Tekanan – Entalphi Panas Lanjut Refrigeran 134a



Lampiran.9.

Tabel C.2. Perbandingan Sifat Refrigeran

Refrigerant	ASA B9 Safety Code Group Classification	Nat'l Fire Underwriters Group Number	Toxicity Lethal or Serious Injury[3]: Refrigerant in Air: Duration of Exposure (hr)	% by Vol	lb/1000 cu ft	Products of Decomposition by Flame: Duration of Exposure (min)	% by Vol[4]	Flammable or Explosive Limits of Concentration in Air: % by Vol
Methane	3[1]	4-5						4.9-15.0
R-14	1[1]	6[1]						Nonflam.
Ethylene	3[1]	4-5						3.0-25.0
Nitrous oxide			8	0.0025				Nonflam.
R-13	1[1]	6[1]						Nonflam.
Ethane	3	5	2	37.4-51.7				3.3-10.6
Carbon dioxide	1	5	½ to 1	29-30	33.2-34.3			Nonflam.
Kulene-131	1[1]	6[1]						Nonflam.
Propane	3	5	2	37.5-51.7	42.4-58.5			2.3-7.3
R-22	1	5A				16	1.0	Nonflam.[2]
Ammonia	2	2	½	0.5-0.6	0.231-0.256			16.0-25.0
Carrene-7	1	5A	2	19.4-20.3	50.2-52.7	25	1.1	Nonflam.
R-12	1	6	2	28.5-30.4	89.6-95.7	20	1.0	Nonflam.
Methyl chloride	2	4	2	2-2.5	2.62-3.28	30	2.4	8.1-17.2
Isobutane	3	4-5						1.8-8.4
Sulfur dioxide	2	1	1/12	0.7	1.165			Nonflam.
Butane	3	5	2	37.5-51.7				1.6-6.5
R-114	1	6	2	20.1-21.5	90.5-96.8	15	1.0	Nonflam.
R-21	1		½	10.2	27.1			Nonflam.
Ethyl chloride	2	4	1	4.0	6.72	18	2.0	3.7-12.0
R-11	1	5	2	10	35.7	5	1.0	Nonflam.
Methyl formate	2	3	1	2-2.5	3.12-3.9			4.5-20.0
Methylene chloride	1	4A	½	5.1-5.3	17.25-11.[illegible]	20	1.0	Nonflam.
R-113	1	4	1	4.8-5.2	23.3-25.2	16	1.2	Nonflam.
Dichlorethylene	2	4	½	2-2.5	5.04-6.3	5	2.1	5.6-11.4

[1] Unofficial.
[2] Very slightly flammable, but for practical purposes considered nonflammable.
[3] To guinea pigs.
[4] Initial concentration.
From the *ASRE Data Book*, Design Volume, 1957-58 Edition, by permission of the American Society of Heating Refrigerating, and Air-Conditioning Engineers.

Lampiran.10.

Tabel D.1. Volume Udara Infiltrasi

Air Infiltration into Cold Storage Rooms

Average Air Changes Per 24 Hours for Storage Rooms Above 32°F Due to Door Opening and Infiltration*

Volume Cu Ft	Air Changes Per 24 Hr	Volume Cu Ft	Air Changes Per 24 Hr	Volume Cu Ft	Air Changes Per 24 Hr	Volume Cu Ft	Air Changes Per 24 Hr
200	44.0	800	20.0	5000	7.2	25,000	3.0
250	38.0	1000	17.5	6000	6.5	30,000	2.7
300	34.5	1500	14.0	8000	5.5	40,000	2.3
400	29.5	2000	12.0	10,000	4.9	50,000	2.0
500	26.0	3000	9.5	15,000	3.9	75,000	1.6
600	23.0	4000	8.2	20,000	3.5	100,000	1.4
						350,000	1.13†
						700,000	0.97†

Average Air Changes Per 24 Hours for Storage Rooms Below 32°F Due to Door Opening and Infiltration[b]

Volume Cu Ft	Air Changes Per 24 Hr	Volume Cu Ft	Air Changes Per 24 Hr	Volume Cu Ft	Air Changes Per 24 Hr	Volume Cu Ft	Air Changes Per 24 Hr
200	33.5	800	15.3	5000	5.6	25,000	2.3
250	29.0	1000	13.5	6000	5.0	30,000	2.1
300	26.2	1500	11.0	8000	4.3	40,000	1.8
400	22.5	2000	9.3	10,000	3.8	50,000	1.6
500	20.0	3000	7.4	15,000	3.0	75,000	1.3
600	18.0	4000	6.3	20,000	2.6	100,000	1.1
						150,000	[illegible]
						200,000	[illegible]

*For heavy usage multiply the above values by a service factor of 2. For long storage multiply the above values by 0.6

[b]For heavy usage multiply the above values by a service factor of 2. For long storage multiply the above values by 0.6. For 24 hr [illegible] two open doors on adjacent or opposite walls [illegible] not be tolerated

†Extrapolated

Courtesy Dunham-Bush, Inc.

Tabel D.2. Beban Panas Udara Infiltrasi

Heat Removed in Cooling Air to Storage Room Temperature (Btu/ft³)

Storage Room Temperature °F	85 °F, 50%	85 °F, 60%	90 °F, 50%	90 °F, 60%	95 °F, 50%	95 °F, 60%	100 °F, 50%	100 °F, 60%
65	0.65	0.85	0.93	1.17	1.24	1.54	1.58	1.95
60	0.85	1.03	1.13	1.37	1.44	1.74	1.78	2.15
55	1.12	1.34	1.41	1.66	1.72	2.01	2.06	2.44
50	1.32	1.54	1.62	1.87	1.93	2.22	2.28	2.65
45	1.50	1.73	1.80	2.06	2.12	2.42	2.47	2.85
40	1.69	1.92	2.00	2.26	2.31	2.62	2.67	3.06
35	1.86	2.00	2.17	2.43	2.49	2.79	2.85	3.24
30	2.00	2.24	2.26	2.50	2.64	2.94	2.95	3.35

Storage Room Temperature °F	40 °F, 70%	40 °F, 80%	50 °F, 70%	50 °F, 80%	90 °F, 50%	90 °F, 60%	100 °F, 50%	100 °F, 60%
30	0.24	0.29	0.58	0.66	2.26	2.53	2.9[illegible]	[illegible]
25	0.41	0.45	0.75	0.83	2.44	2.71	3.1[illegible]	[illegible]
20	0.56	0.61	0.91	0.99	2.62	2.90	3.3[illegible]	[illegible]
15	0.71	0.75	1.06	1.14	2.80	3.07	3.[illegible]	[illegible]
10	0.85	0.89	1.19	1.27	2.93	3.20	3.64	[illegible]
5	0.99	1.03	1.34	1.42	3.12	3.40	3.84	[illegible]
0	1.12	1.17	1.46	1.55	3.28	3.56	4.0[illegible]	[illegible]
−5	1.23	1.28	1.58	1.67	3.41	3.69	4.15	[illegible]
−10	1.35	1.41	1.70	1.81	3.56	3.85	4.31	4.74
−15	1.50	1.53	1.85	1.92	3.67	3.96	4.42	4.9[illegible]
−20	1.63	1.68	2.01	2.03	3.88	4.18	4.66	[illegible]
−25	1.77	1.80	2.12	2.21	4.00	4.30	4.78	5.21
−30	1.90	1.95	2.29	2.38	4.21	4.51	4.90	5.44

Courtesy Dunham-Bush, Inc.

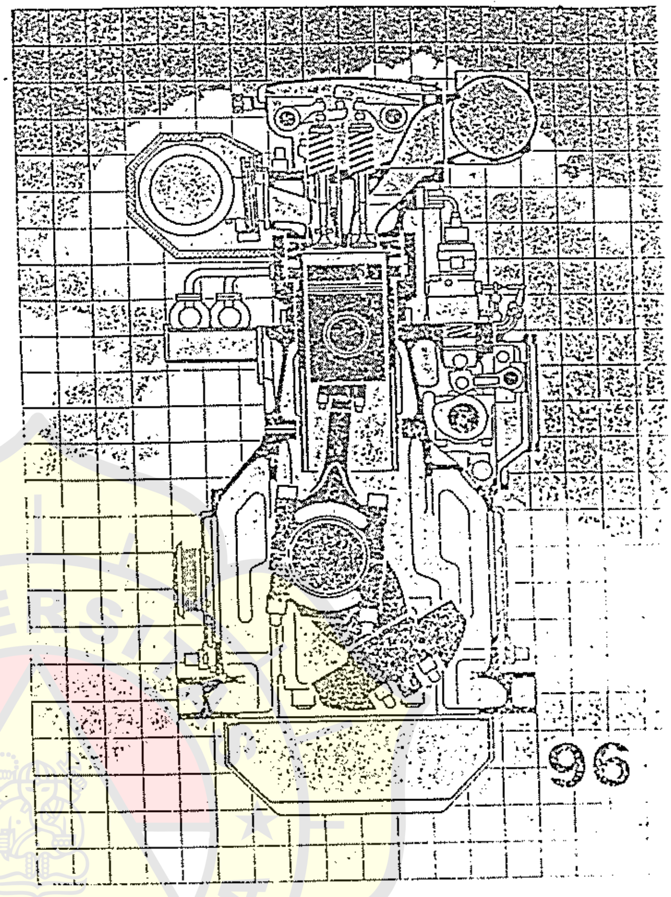
Lampiran.11.

AN B&W Diesel AG
stfach 10 00 80
86135 Augsburg
lephone (08 21) 32 20
lex 53 796-0 man d
lefax (08 21) 322 3382

tus: August 1996

Four-stroke Diesel engine programme

MAN B&W



bject to modifications
the interest of technical progress.

Table of contents

Type	Power range kW	Speed range 1/min	Page
Definitions			4/5
Diesel engines			
L-V 20/27	450– 1800	900–1000	6
L-V 32/40	2200– 7920	720– 750	7
L 40/54	4200– 6480	500– 550	8
L-V 48/60	6300–18900	500– 514	9
L 58/64	7800–12510	400– 428	10
Dual fuel engines			
L-V 20/27 DG	405– 1620	900–1000	11
L-V 32/40 DG	2400– 7200	720– 750	12
L-V 48/60 DG	5400–16200	500– 514	13
Holeby Genset			
L 16/24	450– 900	1000–1200	14
L 23/30 H	530– 1280	720– 900	15
L-V 28/32 H	880– 3960	720– 750	16
L 32/40	3200– 3960	720– 750	17
Spark-fired gas engines			
L-V 28/32 SI	1000– 3600	720– 750	18
Alpha Diesel propulsion systems			
L-V 23/30 A-KV/VO	800– 1920	800– 900	19
L-V 28/32 A-VO	1470– 3920	775	20
Standard rating values generating sets with gas- and dual-fuel engines			21
Standard rating values Diesel generating sets			22/23

Engine power ranges of marine propulsion engines

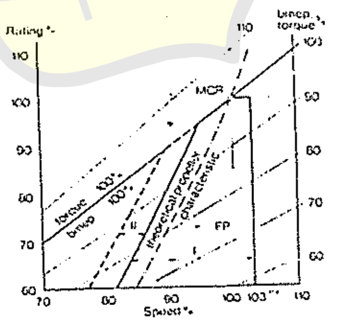


Fig. 1: Marine propulsion engine with fixed-pitch propeller

MCR = maximum continuous rating (fuel stop power)
I = operating range for continuous service
II = operating range temporarily admissible, e.g. during acceleration, manoeuvring (torque limit)
FP = design range for fixed-pitch propeller (Fig. 1)
VP = design range for controllable-pitch propeller with combinator (Fig. 2)

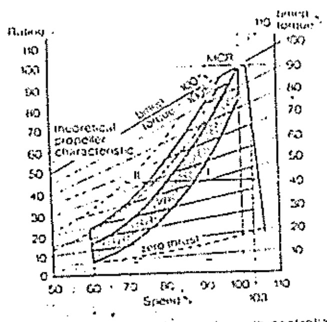


Fig. 2: Marine propulsion engine with controllable-pitch propeller

Lampiran.12.

L 20/27
V 20/27

No. of cyl.	Continuous rating P MCR kW/HP diesel and heavy fuel operation n (1/min) 1000	900	Dimensions mm L1/L2	W	H	Weight** tons
5 L	500 680	450 610	2700 1710	1350	1900	5.0
6 L	600 815	540 730	3000 2000	1350	1900	5.9
7 L	700 950	630 855	3300 2450	1350	1950	6.6
8 L	800 1090	720 980	3550 2730	1350	2150	7.4
9 L	900 1225	810 1100	3800 3010	1350	2150	8.0
12 V	1200 1630	1080 1470	3600 3400	1510	2600	10.6
14 V	1400 ~~1900~~	1260 1715	3950 3800	1510	2750	11.8
16 V	1600 2180	1440 1960	4300 4150	1510	2750	13.1
18 V	1800 2450	1620 2200	3650 4500	1510	2750	14.4

Bore (mm)	Stroke (mm)		n (1/min)	kW/cyl. (HP/cyl.)	p_e(bar)	c_m (m/s)
200	270	MCR	1000	100 (136)	14.15	9.0
			900	90 (122)		8.1

Fuel consumption (to ISO conditions):*					
		100% P		85% P	
L 20/27	MCR	200 g/kWh	147 g/HPh	200 g/kWh	147 g/HPh
V 20/27		199 g/kWh	146 g/HPh	199 g/kWh	146 g/HPh

Lube oil consumption: approx. 1.2 g/kWh 0.9 g/HPh

L 32/40
V 32/40

No. of cyl.	Continuous rating P MCR kW/HP diesel and heavy fuel operation n (1/min) 750	720	Dimensions mm L1/L2	W	H	Weight** tons
5 L	2200 3000	2200 3000	5095 3500	2100	3890	33
6 L	2640 3600	2640 3600	5625 4480	2100	3890	38
7 L	3080 4200	3080 4200	6155 5010	2570	4410	42
8 L	3520 4800	3520 4800	6685 5540	2570	4410	47
9 L	3960 5400	3960 5400	7215 6070	2570	4410	52
12 V	5280 7200	5280 7200	6250 5100	3100	3900	61
14 V	6160 8400	6160 8400	7100 5750	3100	4200	70
16 V	7040 9600	7040 9600	7750 6400	3100	4200	79
18 V	7920 10800	7920 10800	8400 7000	3100	4200	86

Bore (mm)	Stroke (mm)		n (1/min)	kW/cyl. (HP/cyl.)	p_e(bar)	c_m (m/s)
320	400	MCR	750	440 (600)	21.9	10.0
			720	440 (600)	22.8	9.6

Fuel consumption (to ISO conditions):*					
		100% P		85% P	
L 32/40	MCR	182 g/kWh	134 g/HPh	179 g/kWh	132 g/HPh
V 32/40		181 g/kWh	133 g/HPh	178 g/kWh	131 g/HPh

Lube oil consumption: approx. 1.0 g/kWh 0.7 g/HPh

Definitions

General definiton of Diesel engine ratings (acc. to ISO 3046/I)

P = Continuous rating

10% overload capacity for 1 hour within 12 operating hours

Reference conditions:
Air temperature: 298 K (25° C)
Air pressure: 1 bar
Cooling water temperature upstream of charge-air cooler: 298 K (25° C)

Marine auxiliary engines

P = Continuous Rating

10% overload capacity for 1 hour within 12 operating hours

Reference conditions:
Air temperature: 318 K (45° C)
Air pressure: 1 bar
Cooling water temperature upstream of charge-air cooler: 305 K (32° C)

Main marine engines

MCR = Maximum Continuous Rating (fuel stop power)

Reference conditions:
Air temperature: 318 K (45° C)
Air pressure: 1 bar
Cooling water temperature upstream of charge-air cooler: 305 K (32° C)

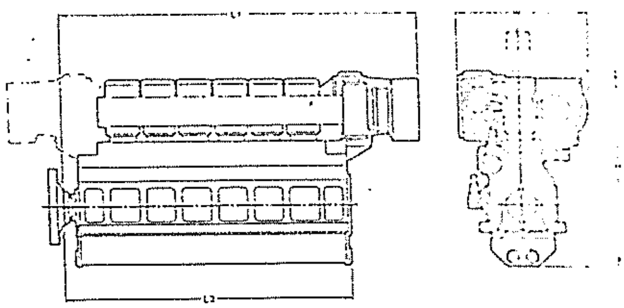
Definitions

- n — Rated engine speed
- Pe — Mean effective pressure
- Cm — Mean piston speed
- L1 — Overall engine length
- L2 — Length at crankshaft centreline
- W — Overall engine width
- H — Overall engine height
- Weight — Dry weight without fly wheel (tolerance 5%)

Key to engine type designations

e.g. L or V 20/27
- L = in-line engine
- V = V-engine
- 20 = bore in cm
- 27 = stroke in cm

e.g. L 28/32A-VO
- VO, KV = designation for complete propulsion system

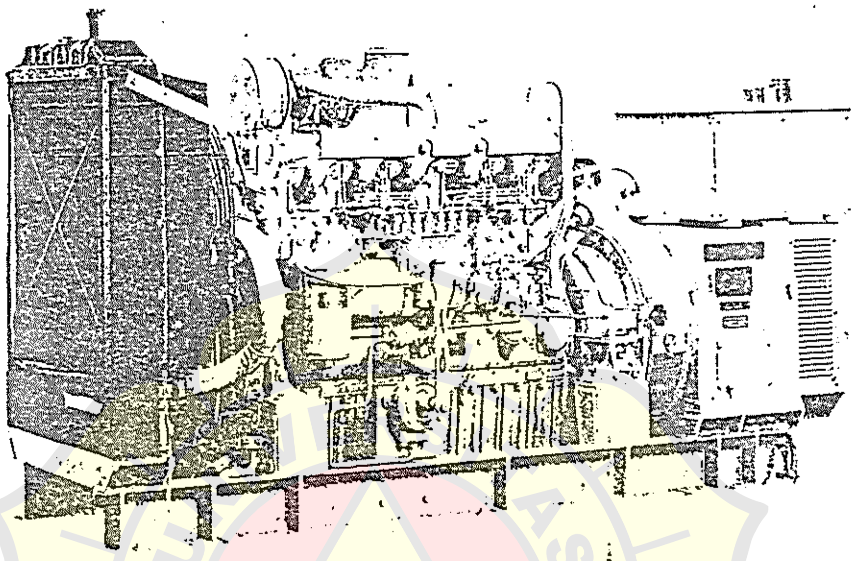


The fuel consumption rates are based on ISO conditions (25° C ambient temperature, 25° C cooling water temperature upstream of charge air cooler, 1 bar barometric pressure) and a net calorific value of the fuel of 42,700 kJ/kg. Tolerance + 3%

Lampiran.13.

:mergency-Use Marine Power ;enerating Sets

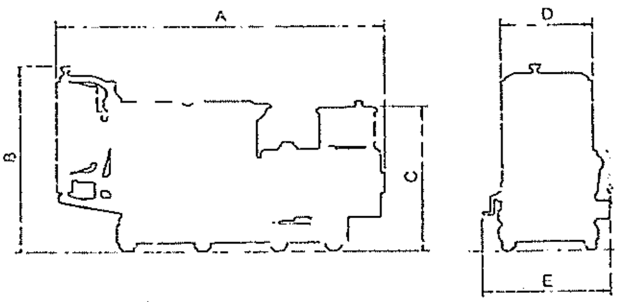
/MGH Series



ɔecifications

Model			YMGH30	YMGH40	YMGH55	YMGH80	YMGH100	YMGH120	YMGH150	YMGH200	YMGH250
ator	Capacity	kVA	30	40	55	80	100	120	150	200	25
	Frequency	Hz	60								
	Voltage	V	450								
	Power factor	%	80								
engine	Model	——	4TN100L-H	4TN100L-HI	4T112L-H	4TN112L-TH	6CHL-TH	6CHL-HTH	6HAL-H	6HAL-TH	6HAL-HTH
	Output	kW (PS)	31.3 (42.5)	40 (54)	51 (70)	72 (98)	89 (121)	107 (145)	132 (180)	177 (240)	221 (300)
	Engine speed	rpm	1800								
	Type	——	Vertical 4-cycle diesel engine (radiator cooling)								
	Combustion system	——	Direct injection								
	Starting system	——	Remote (automatic) starting by starting motor								

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



IGH-A Series Dimensions

(Unit: mm)

del	A	B	C	D	E	Weight (kg)
30A	1640	956	330	645	815	735
40A	1680	965	830	645	815	770
55A	1920	1050	925	690	945	940
80A	1900	1059	950	630	945	1120
100A	2320	1310	1050	720	845	1530
120A	2390	1310	1155	720	845	1640
150A	2609	1460	1210	720	1015	2110
200A	2670	1460	1210	720	1015	2270
250A	2725	1480	1219	720	1015	2440

YMGH-B Series Dimensions

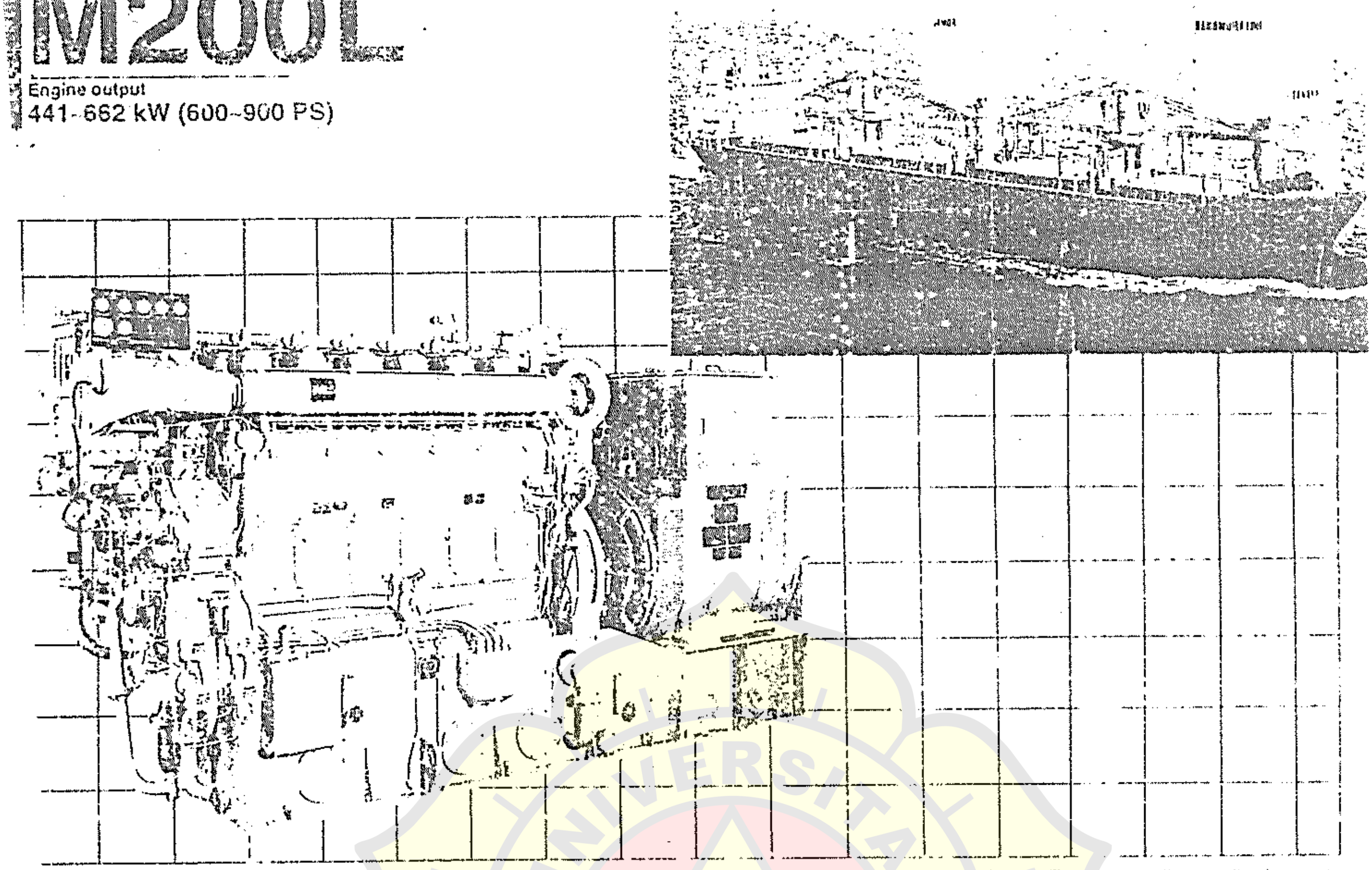
(Unit: mm)

Model	A	B	C	D	F	Weight (kg)
YMGH 30B	1590	965	905	645	815	765
YMGH- 40B	1630	965	905	645	815	780
YMGH- 55B	1800	1050	925	690	945	880
YMGH- 80B	1920	1059	925	690	945	1000
YMGH-100B	2290	1310	1070	720	845	1400
YMGH-120B	2345	1310	1070	720	845	1440
YMGH-150B	2615	1460	1125	720	1015	1940
YMGH-200B	2705	1460	1205	720	1015	2190
YMGH-250B	2630	1480	1340	720	1015	2400

Lampiran. 14.

M200L

Engine output
441~662 kW (600~900 PS)



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

pecifications

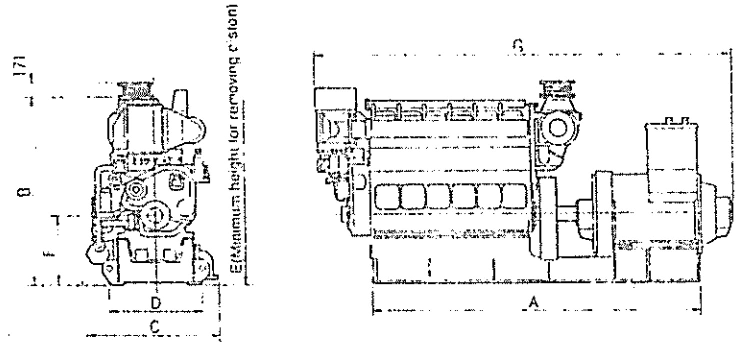
Engine model		M200L-UN (-UX)		M200L-SN (-SX)		M200L-EN (-EX)		M200AL-UN		M200AL-SN		M200AL-EN	
		Vertical water cooled 4-cycle diesel engine											
ı cylinders		6											
der bore × stroke	mm	200 × 260											
displacement	ℓ	49.01											
nuous rated output	kW (PS)	441 (600)		485 (660)		552 (750)		530 (720)		610 (830)		662 (900)	
ıe speed	rpm	720	750	720	750	720	750	900	1000	900	1000	900	1000
·ean effective pressure	MPa (kgf/cm²)	1.500 (15.30)	1.441 (14.69)	1.650 (16.83)	1.585 (16.16)	1.876 (19.13)	1.801 (18.36)	1.441 (14.69)	1.295 (13.22)	1.661 (16.94)	1.495 (15.24)	1.801 (18.36)	1.621 (16.53)
·rator capacity	kW	400		450		500		480		560		600	
)ustion system		Direct injection											
ng system		Compressed air											
ıal ·sions — Overall length	mm	2919		2923		2977		2919		2977		2977	
Overall width	mm	1120.5											
Overall height	mm	1844		1880		1883		1844		1883		1883	
·eight	kg	5800											

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

ımensions

(Units: mm)

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



Engine model	M200L-UN (-UX)	M200L-SN (-SX) M200AL-UN	M200L-EN (-EX) M200AL-SN M200AL-EN
A	3650	3650	4000
B	2271	2271	2326
C	1520	1520	1520
D	1040	1040	1040
E	2570	2570	2570
F	950	950	950
G	4550	4600	4600
Dry weight of generating set	11000	11000	11300

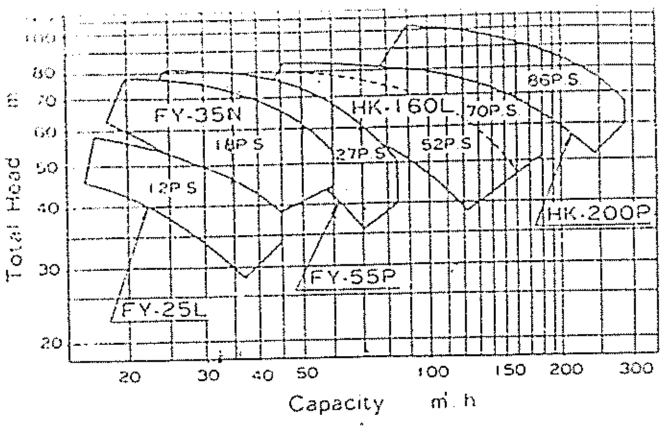
Note: Above data shows the case of [illegible]

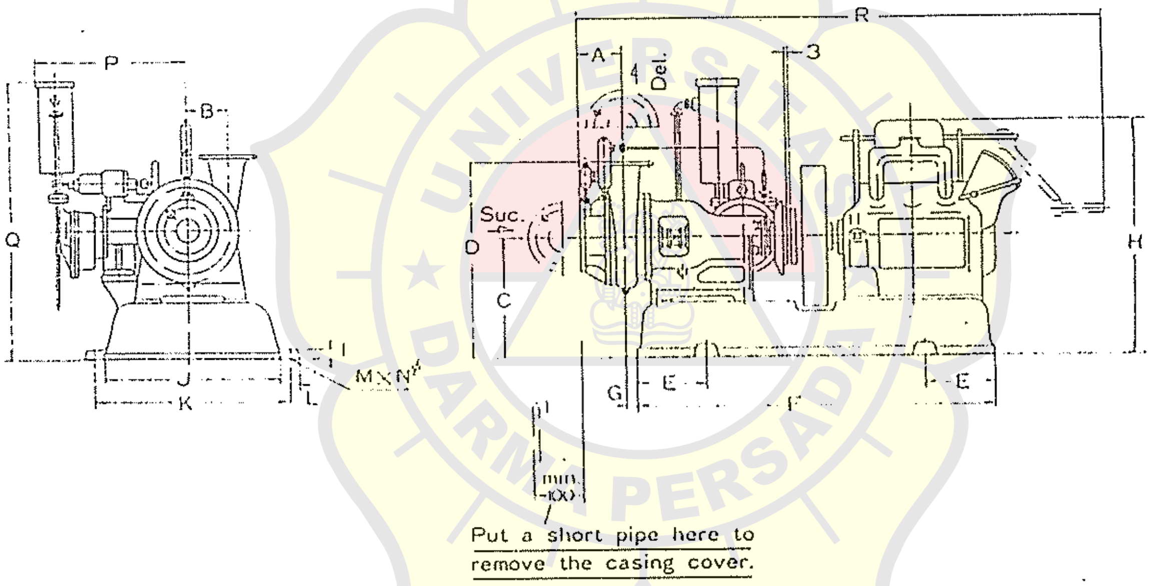
Lampiran.15.

CENTRIFUGAL PUMP
GEAR PUMP

OYAMA PUMP MANUFACTURING (INDONESIA)

Lampiran.10.

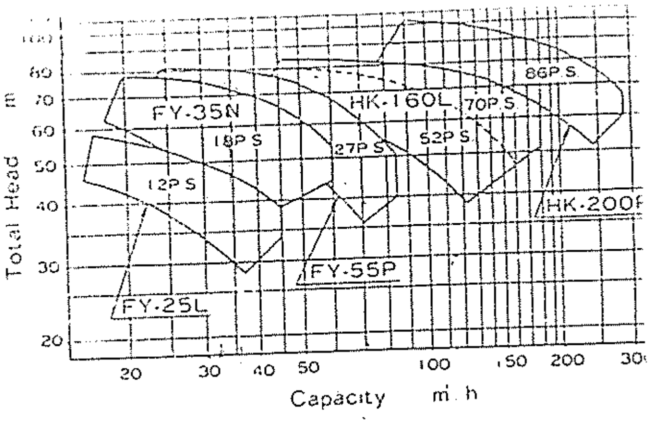


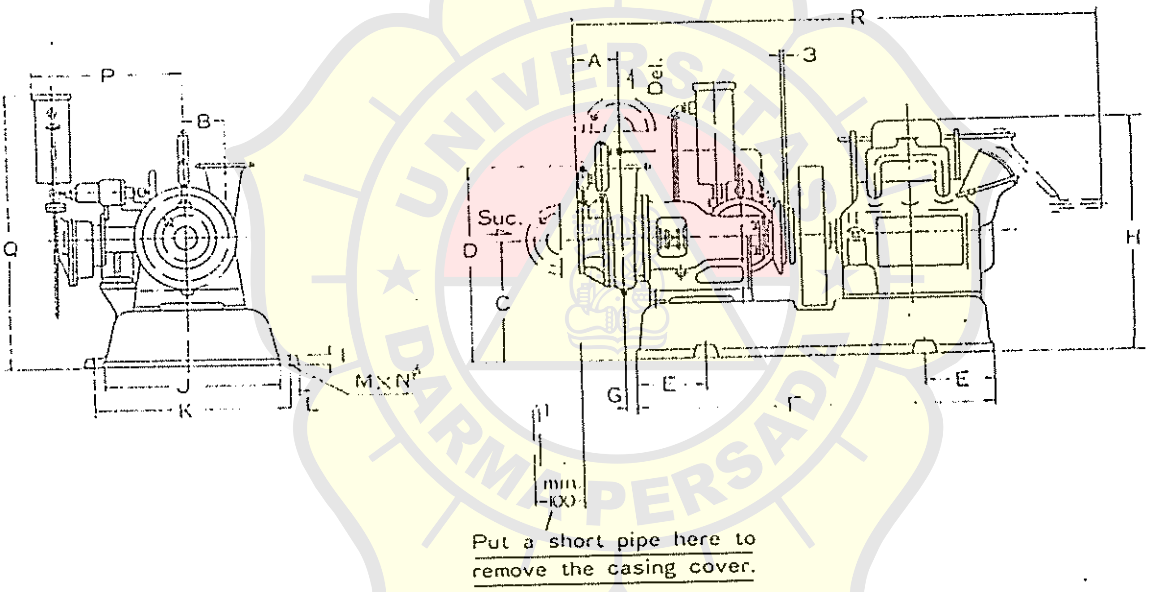


Dimensions—mm

'ype	Die. Engine (P.S)	Bore		A	B	C	D	E	F	G	H	I	J	K	L	M	N	P	Q	R	Vaccum Pump	Pump Weight (kg)
		Suc.	Del.																			
'5L	12	65	65	130	140	319	530	125	750	38	791	30	460	510	25	4	19	398	760	1264	V-18	290
'5N	18	80	80	140	140	330	520	125	950	38	813	30	480	530	25	4	19	398	795	1397	V-18	340
'5P	27	100	100	150	150	348	558	150	1140	48	831	30	500	550	25	4	19	398	813	1589	V-18	390
160L	52	125	125	152	235	403	710	120	1500	75	1040	35	550	610	28	6	23	420	855	1913	V-50	430
	70			152	235	433	740	200	1700	75	1170	35	550	610	28	6	23	420	885	2088	V-50	450
200P	86	200	200	200	270	488	945	200	1800	75	1457	50	570	490	40	6	27	442	960	2185	V-50	490

Lampiran.17.

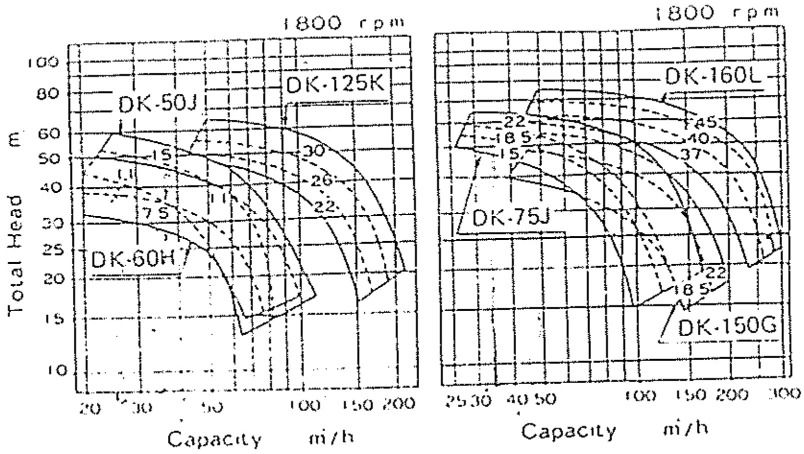


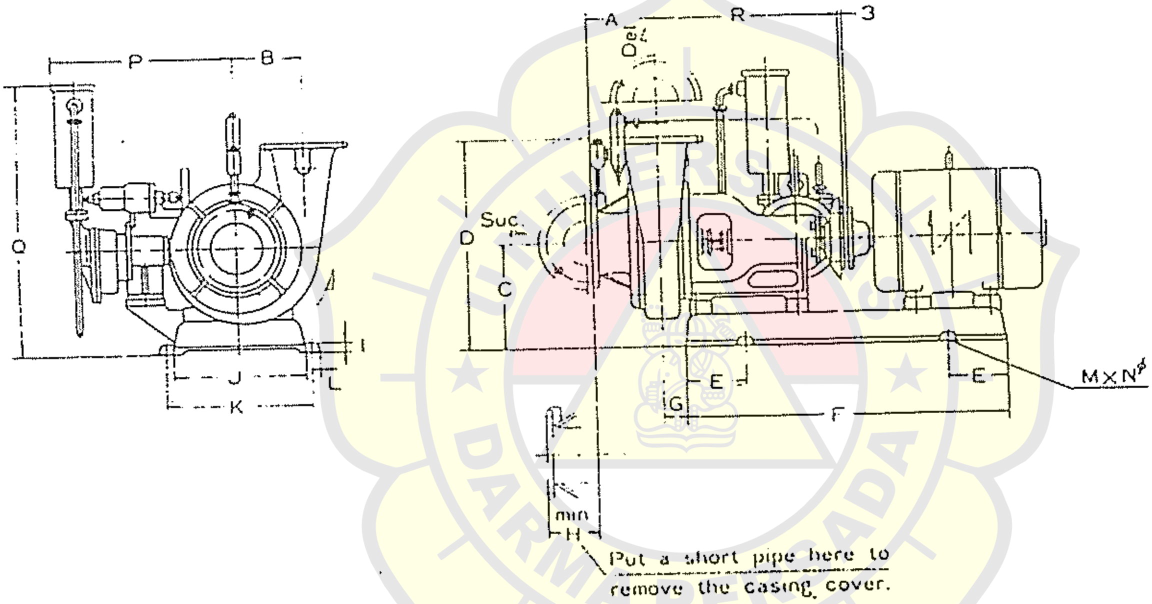


Dimensions—mm

Type	Die. Engine (P.S)	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	P	Q	R	Vaccum Pump	Pum. Weig. (kg)
Y-25L	(12)	65	65	130	140	319	530	125	750	38	791	30	460	510	25	4	19	398	760	1264	V-18	29
Y-35N	(18)	80	80	140	140	330	520	125	950	38	813	30	480	530	25	4	19	398	795	1397	V-18	34
Y-55P	27	100	100	150	150	348	558	150	1140	48	831	30	500	550	25	4	19	398	813	1589	V-18	39
HK-160L	52	125	125	152	235	403	710	120	1500	(75)	1040	35	550	610	28	6	23	420	855	1913	V-50	43
	70			152	235	433	740	200	1700	75	1170	35	550	610	28	6	23	420	885	2088	V-50	45
HK-200P	86	200	200	200	270	488	945	200	1800	75	1457	50	570	490	40	6	27	442	960	2185	V-50	49

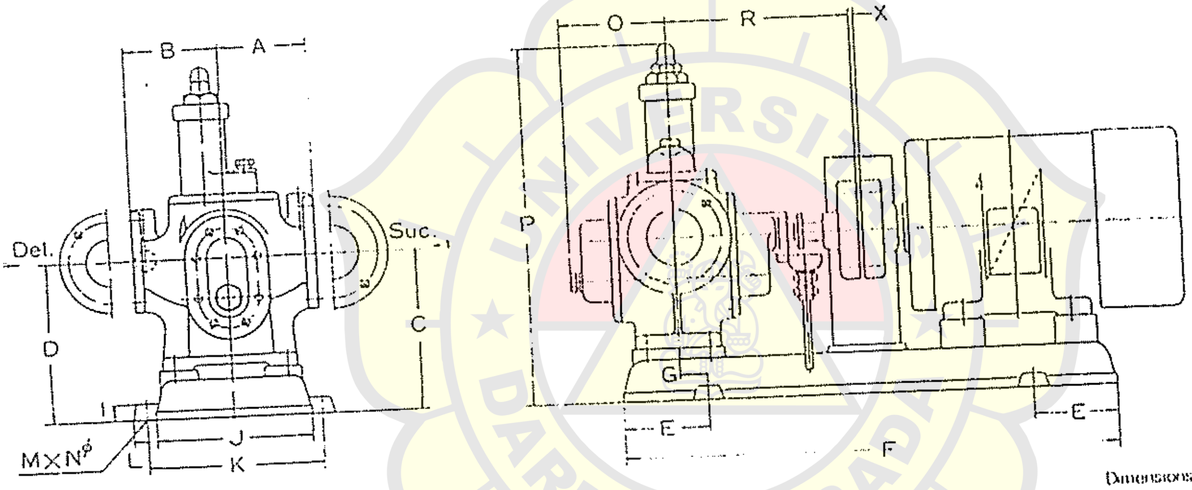
Lampiran. 18.





›e	Motor (kw)	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	P	Q	R	Vacuum Pump	Pump Weight (kg)
J	11	100	100	140	220	240	520	150	900	30	100	25	360	400	23	4	15	405	705	420	V-18	250
	15			140	220	240	520	150	900	30	100	25	360	400	23	4	15	405	705	420	V-18	250
H	7.5	100	100	117	177	240	470	150	900	32	100	25	360	400	23	4	15	405	705	422	V-18	240
	11			117	177	240	470	150	900	32	100	25	360	400	23	4	15	405	705	422	V-18	240
J	15	125	125	156	205	300	590	200	1020	35	120	30	390	430	25	4	23	405	765	480	V-50	270
	18.5			156	205	300	590	200	1020	35	120	30	390	430	25	4	23	405	765	480	V-50	270
	22			156	205	300	590	200	1020	35	120	30	390	430	25	4	23	405	765	480	V-50	270
0G	18.5	150	150	160	200	295	610	200	1050	53	120	30	450	500	25	4	23	420	775	548	V-50	280
	22			160	200	295	610	200	1050	53	120	30	450	500	25	4	23	420	775	548	V-50	280
5K	22	150	150	160	230	303	610	200	1050	60	120	30	450	500	25	4	23	420	775	555	V-50	280
	30			160	230	303	610	200	1100	60	120	30	450	500	25	4	23	420	775	555	V-50	280
0L	37	150	150	159	235	375	695	250	1340	68	100	30	490	550	28	4	23	420	840	760	V-50	430
	45			159	235	375	695	250	1340	68	100	30	490	550	28	4	23	420	840	760	V-50	430

Lampiran.19.



Dimensions - mm

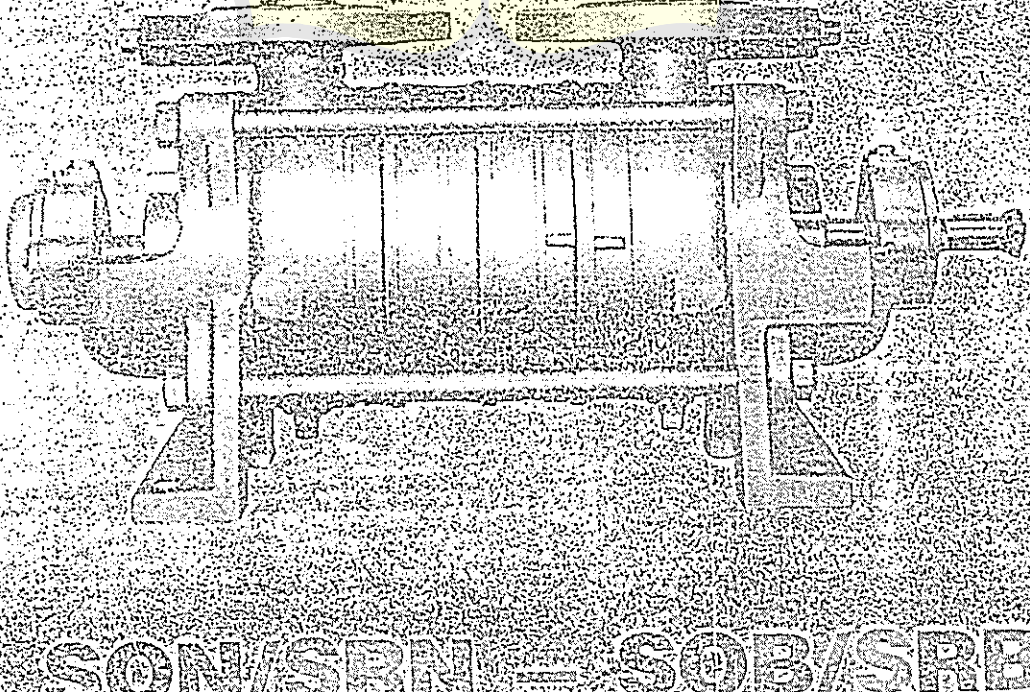
Model	No. of Rev (r.p.m.)	Motor (kw)	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	I	J	K	L	M	N	O	P	R	X	Pump Weight (kg)
	1200	0.4	25	20	110	110	200	200	50	365	21	20	170	140	15	4	15	55	330	122	3	30
	1200	0.75	32	25	95	95	195	195	100	500	60	25	260	300	23	4	15	89	380	175	3	47
—	1200	0.75 1.5	40	32	95	95	195	195	100	500	60	25	260	300	23	4	15	96	385	175	3	50
	1200	0.75 1.5	50	40	100	100	210	210	100	550	35	25	260	300	23	4	15	98	412	185	3	55
	1200	1.5 2.2	65	50	105	105	225	225	100	600	20	25	280	320	23	4	15	144	470	200	3	70
	1200	1.5 2.2	65	50	105	105	225	225	100	600	20	25	280	320	23	4	15	111	470	200	3	72
	1200	2.2	65	50	105	105	225	225	100	600	20	25	280	320	23	4	15	55	470	200	3	72
	1200	3.7			105	105	230	230	130	640	60		300	330	25			63	475			78
B	1200	2.2 3.7	80	65	110	110	245	245	150	700	50	25	300	350	23	4	15	122	500	220	3	90
3	1200	2.2 3.7	80	65	140	140	230	230	125	700	30	25	310	350	23	4	15	137	530	245	3	95
B	1200	3.7	80	65	140	140	230	230	125	700	30	25	310	350	23	4	15	152	530	250	3	103
	1200	5.5					230	230	125	750								152				
B	1200	3.7 5.5	80	65	150	150	260	260	100	750	0, 20	25	310	350	23	4	15	165	565	285	3	140
	1200	7.5					270	270	150	950	65	30	350	390	22		19		575			
)B	1200	5.5	100	80	160	160	270	270	150	800	50	25	310	350	23	4	15	193	618	315	3	135
	1200	7.5							200	900	100		350	390								
5B	1200	5.5	100	80	160	160	270	270	150	800	50	25	310	350	23	4	15	193	618	315	3	135
	1200	7.5							200	900	100	25	350	390								
0B	1200	7.5 11	125	100	175	175	320	320	200	1000	80	30	370	420	25	4	19	210	688	357	3	200
0B	1200	7.5 11	150	125	190	190	330	330	200	1050	40, 57	35	490	540	25	4	23	240	780	387	3	220
0B	1200	11 15	150	125	235	235	390	390	250	1100	90	30	450	500	25	4	19	225	835	385	3	365
5B	1200	15 18.5	150	125	235	235	390	390	200	1150	30	30	500	550	25	4	19	245	840	415	3	370
0C	900	18.5 22	150	125	250	250	460	460	200	1500	0	45	570	620	30	6	23	377	1005	577	3	490
00D	720	22 30	200	175	350	350	370	370	300	1730	-30	35	620	680	28	4	23	523	1395	730	4	550

mpiran.20.

SERO

Self-Priming Centrifugal Pumps

– side channel pumps –
(patented)



SON/SRN — SOB/SRB

Lampiran.21.

(—Material DIN No. 1.4581)

— oval flanges — SRN/SRB — round flanges

— 1450 RPM.

l/min	5	10	15	20	25	30	35	40	45	50	60	70	80	90	100	110	125	150	175	200	250	300	350	400	450	500	550	600
m³/h	0,3	0,6	0,9	1,2	1,5	1,8	2,1	2,4	2,7	3,0	3,6	4,2	4,8	5,4	6,0	6,6	7,5	9,0	10,5	12,0	15,0	18,0	21,0	24,0	27,0	30,0	33,0	36,0
Connections	Total dynamic head (metres) recommended motor size (kW)																											
R 1"	27	22	18	14	9	5																						
R 1"	50	42	33	25	17	9																						
R 1" NW 25	73	61	49	37	25	13																						
R 1"	32	29	26	23	19	15	11	7																				
R 1"	60	54	48	42	35	28	20	13																				
R 1" NW 25	88	79	70	61	51	41	30	19																				
NW 25	116	104	92	80	67	53	39	25																				
NW 25	144	129	114	100	83	66	49	31																				
NW 25	172	154	136	119	99	79	58	37																				
R 1"			38	36	33	30	26	22	17	13																		
R 1"			71	67	61	55	48	40	32	25																		
R 1"			104	98	89	80	70	58	47	37																		
NW 32			137	129	117	105	91	76	62	49																		
NW 32			170	159	145	130	113	95	77	61																		
NW 32			203	190	173	155	135	112	92	73																		
R 1¼"				42	40	38	35	33	30	28	23	17	10															
R 1¼"				80	76	72	67	63	56	54	44	33	20															
R 1¼"				118	112	106	99	93	86	80	65	49	39															
NW 32				156	148	140	131	123	114	106	86	65	40															
NW 32				194	184	174	163	153	143	132	107	81	50															
NW 32				232	220	208	195	183	170	158	120	97	60															
R 1½"										36	33	29	26	22	18	15	9											
R 1½"										68	62	54	48	41	34	28	17											
R 1½" NW 32										100	91	80	70	60	50	41	25											
NW 32										132	120	106	93	79	66	54	33											
NW 32										164	140	132	115	98	82	66	41											
NW 32										196	176	157	138	116	96	79	49											
NW 40															37	34	30	23	16	9								
NW 40															70	65	57	41	30	17								
NW 40															103	96	84	64	44	25								
NW 40															136	127	111	84	58	33								
NW 40															179	157	135	105	72	41								
NW 40															203	180	165	125	86	49								
NW 50																			38	33	25	18	10					
NW 50																			71	62	46	32	18					
NW 50																			104	91	67	47	26					
NW 50																			137	120	88	61	34					
NW 65																						34	31	26	21	16	11	6
NW 65																						64	58	49	39	30	20	[illegible]
NW 65																						94	86	72	58	44	30	[illegible]

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

Max. Casing Pressure:
SON, SOB 160 .... 330 : 16 bar
SRN, SRB 100 .... 550 : 25 bar
SRN 660 : 16 bar