BAB VI

KESIMPULAN

6. 1. KESIMPULAN

Dari perencanaan dan perhitungan berdasarkan data kapal berikut :

- Type kapal : TANKER
- Lwl : 85,68 m
- Lpp : 84 m
- B : 15 m
- T : 5 m
- H : 7 m
- Cb : 0,661
- Vs : 13 knots

Maka dapat disimpulkan :

1. Kapal memerlukan tenaga penggerak minimum 2.179,633 HP. Pada perencanaan ini dipilih motor induk dengan daya sebesar 2.180 HP pada putaran 250 rpm.
2. Dengan jumlah crew 30 orang dan route pelayaran yang ditempuh lebih kurang 11.000 mil, kapasitas maksimum kebutuhan listrik untuk mensuplai peralatan yang ada sebesar 199,229 kW. Dalam perencanaan ini digunakan 3 unit generator yang sama besar dimana 2 unit sebagai generator utama dan 1 unit dipakai sebagai generator cadangan ataupun standby generator. Untuk itu dipilih generator dengan kapasitas masing-masing sebesar 104 kW.

Frekwensi	60 Hz
Power factor	0,8
Combustion	DI (Direct Injection)

5.3. Analisa Beban Dan Pemilihan Generator

5.3.1. Analisa Beban Generator

Didalam menentukan besarnya daya listrik yang harus disediakan oleh generator, sebelumnya harus dilakukan suatu analisa penggunaan daya listrik sehingga didapat nilai yang effisien dalam pemakaian generator.

Untuk memenuhi tujuan tersebut pengguaan daya listrik khususnya diatas kapal dapat dikategorikan pada tiga kondisi, yaitu : penggunaan daya listrik pada saat kapal sedang melakukan bongkar muat baik pada waktu siang maupun malam hari.

Adapun analisa beban generator ini dapat dilihat pada tabel beban generator.

5.3.2. Pemilihan Generator

Berdasarkan tabel analisa beban generator dapat dilihat pemakaian listrik yang terbesar adalah pada saat kapal sedang manuver pada malam hari yaitu sebesar **199,229 kW**. Untuk itu direncanakan dipakai 3 unit generator yang sama dengan masing-masing **104 kW**, dimana salah satu dari ketiga generator tersebut disiapkan sebagai generator cadangan atau sebagai standby generator.

Dari hasil perkiraan perhitungan diatas, maka dipilih GEN-SET dengan data-data sebagai berikut :

Tabel.5.3.2. Spesifikasi Gen-Set Tanker 3. 500 DWT :

Merk generator	YANMAR
Model	6HAL2-TN
Type	Vertical 4-cycle
Daya	104 kW
Putaran	1.200 rpm
Bore	130 mm
Stroke	165 mm

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Starting Compressor (Water-cooled)

Model No.	Speed (r.p.m.)	25kgf/cm² Capacity m³/hr (FA)	25kgf/cm² Power required PS	25kgf/cm² Motor (kW)	30kgf/cm² Capacity m³/hr (FA)	30kgf/cm² Power required PS	30kgf/cm² Motor (kW)
H-63	1,200	65	16.5	15	60	17	15
	1,500	80	20	18.5	75	21	18 5
	1,800	95	25	22	90	26.5	22
H-64	1,200	90	23	18.5	85	24	18 5
	1,500	110	29	25	105	30	25
	1,800	135	36.5	30	130	37.5	30
H-264	1,200	175	45.5	37	170	48	37
	1,500	215	57	45	210	60	50
	1,800	260	70.5	55	255	73 5	60
H-73	1,200	110	28	25	105	29	25
	1,500	135	35	30	130	36	30
	1,800	160	46	37	155	47	37
H-74	1,200	140	35	30	135	36	30
	1,500	175	45	37	170	46	37
	1,800	205	58	45	200	60	50
H-273	1,200	220	55	45	215	58	45
	1,500	275	69	55	270	73	60
	1,800	325	86	65	320	88	70
H-274	1,200	275	70	55	270	73	55
	1,500	340	88	70	335	91	75
	1,800	405	108	85	400	111	90
H-373	1,200	330	83	65	320	87	70
	1,500	410	104	85	400	109	90
	1,800	485	128	100	475	132	110
H-374	1,200	415	105	85	405	110	90
	1,500	515	130	110	505	135	110
	1,800	610	162	125	600	168	132

LEGEND: Capacity (free air) referred to inlet condition, measured according to vessel charging test method.

Emergency Compressor (Vertical 2-stage Air-cooled)

Model No.	Speed (r.p.m)	15kg/cm² m³/hr FA	15kg/cm² PS	25 ~ 30kg/cm² m³/hr FA	25 ~ 30kg/cm² PS
LSHC-20B	900	4.7	1.4	4.3	1.6
	1000	5.2	1.5	5.0	1.7
LSHC-30A	900	13.5	4.8	12.8	5.3
	1000	14.8	5.3	13.8	5.8
LSHC-40A	900	20.4	7.2	19.4	8.0
	1000	22.3	7.9	21.2	8.9

" STANDAR UKURAN SEKOCI OLEH BOT (BOARD OF TRADE) ENGLAND "

L. B. H (m)	L. B. H. (ft³)	Kapasitas (ft^3)	Jumlah orang	berat sekoci (kg)	Berat Orang (kg)	berat perlengkapan (kg)	Total berat (kg).
9,4 x 2,74 x 1 x1,114	30 x 9 x 3,75	607	60	2205	4500	356	7061
8,84 x 2,74 x 1,10	29 x 8,75 x 3,60	545	54	1976	4050	356	6382
8,53 x 2,59 x 1,07	28 x 8,50 x 3,50	500	50	1824	3750	330	5894
8,23 x 2,51 x 1,04	27 x 8,25 x 3,40	454	45	1646	3376	330	5351
7,92 x 2,44 x 0,99	26 x 8,00 x 3,25	405	40	473	3000	305	4778
7,62 x 2,36 x 0,96	25 x 7,75 x 3,15	366	36	1326	2700	305	4331
7,31 x 2,29 x 0,91	24 x 7,50 x 3,0	324	32	1180	2400	254	3843
7,01 x 2,29 x 0,88	23 x 7,50 x 2,90	300	30	1087	2250	254	3591 ✓
6,71 x 2,21 x 0,84	22 x 7,25 x 2,75	236	26	955	1950	229	3134
6,40 x 2,13 x 0,82	21 x 7,0 x 2,70	238	23	864	1725	229	2818
6,10 x 2,06 x 0,79	20 x 6,75 x 2,60	210	21	762	1575	203	2540
5,79 x 1,98 x 0,76	19 x 6,50 x 2,50	182	18	650	1350	178	2178
5,49 x 1,90 x 0,73	18 x 6,25 x 2,40	162	16	590	1200	152	1942
5,18 x 1,83 x 0,715	17 x 6,0 x 2,30	145	14	508	1050	152	1710
4,83 x 1,75 x 0,70	16 x 5,75 x 2,30	127	12	452	900	127	1494

LAMPIRAN 6

Self-Propelled Transport Ships with an Unlimited Region of Navigation

No.	Characteristic X	Anchors: Bower: Quantity	Anchors: Bower: Total weight, kg	Anchors: Stream anchor, kg	Chain cable for bower anchors: Total length of two cables, m	Chain cable for bower anchors: Anchor chain size, mm	Chain or steel rope for the stream anchor: Length, m	Chain or steel rope for the stream anchor: Anchor chain size, mm	Chain or steel rope for the stream anchor: Diameter of steel rope, mm
1	50	2	150	25	100	12	50	—	8.8
2	75	2	200	25	125	13	50	—	8.8
3	100	2	250	50	125	15	50	—	11
4	150	2	300	50	150	16	50	—	11
5	200	2	350	50	175	17	75	—	11
6	250	2	450	75	200	18	75	11	13
7	300	2	500	75	225	19	75	13	13
8	350	2	600	100	250	20	75	14	15.5
9	400	2	700	100	275	21	75	14	15.5
10	450	2	750	125	300	22	100	15	17.5
11	500	2	800	150	300	24	100	16	17.5
12	550	2	900	175	325	25	100	16	17.5
13	600	3	1500	200	350	27	100	17	17.5
14	650	3	1700	225	350	28	100	18	19.5
15	700	3	1800	250	375	29	100	18	20.5
16	750	3	2100	250	375	30	100	19	20.5
17	800	3	2250	250	375	31	125	19	20.5
18	850	3	2400	275	375	32	125	20	22
19	900	3	2700	300	375	33	125	21	24
20	950	3	3000	300	400	34	125	21	24
21	1000	3	3200	350	400	35	125	22	24
22	1100	3	3500	400	400	37	125	23	26
23	1200	3	3750	400	420	38	150	25	26
24	1300	3	4100	450	450	40	150	25	28
25	1400	3	4250	450	450	41	150	25	28
26	1500	3	4500	500	450	42	150	25	28
27	1600	3	4750	500	450	43	150	26	28
28	1700	3	5250	600	450	45	150	28	30
29	1850	3	5500	600	450	46	150	28	30
30	2000	3	5750	700	450	46	150	29	31.5
31	2150	3	6000	700	475	48	175	29	31.5
32	2300	3	6500	800	500	49	175	29	32.5
33	2500	3	6750	800	500	50	175	29	32.5
34	2700	3	7500	900	500	52	175	30	33.5
35	3000	3	8250	1000	500	53	200	31	33.5
36	3300	3	9000	1000	500	55	200	31	33.5
37	3600	3	9750	1250	525	57	200	33	34.5
38	3900	3	10500	1250	550	59	225	33	34.5
39	4200	3	11000	1400	550	61	225	34	37
40	4500	3	11500	1500	550	62	225	35	37
41	4800	3	12900	1650	550	65	225	36	—
42	5100	3	13500	1750	550	67	250	37	—
43	5400	3	14500	1750	575	68	250	37	—
44	5800	3	15000	2000	600	70	250	40	—
45	6200	3	15800	2000	600	72	250	40	—
46	6600	3	16300	2250	600	74	275	43	—
47	7000	3	17600	2250	600	76	275	43	—
48	7400	3	18000	2250	600	77	275	44	—
49	7800	3	19500	2500	600	80	275	46	—
50	8200	3	20300	2700	600	82	275	48	—
51	8600	3	21000	2800	600	83	275	49	—
52	9000	3	22000	3000	600	85	275	50	—
53	9500	3	23000	3000	600	87	275	50	—

Mooring and Warping Ropes

Characteristic	Towing rope: Length, m	Towing rope: Circumference of hemp rope, mm	Towing rope: Diameter of steel rope, mm	Warping hawsers: Total length, m	Warping hawsers: Number of ropes	Warping hawsers: Circumference of hemp rope, mm	Warping hawsers: Diameter of steel rope, mm	Cable warps: Total length, m	Cable warps: Number of ropes	Cable warps: Circumference of hemp rope, mm	Cable warps: Diameter of steel rope, mm
50	50	75	—	50	1	65	—				
75	50	90	11	50	1	65	—	—	—	—	—
100	75	90	11	75	1	65	8.5	—	—	—	—
150	75	100	12	75	1	75	9.5	—	—	—	—
200	100	100	12	100	2	75	9.5	—	—	—	—
250	100	125	15	140	2	100	12	—	—	—	—
300	110	125	15	160	2	100	12	—	—	—	—
350	110	150	17.5	160	2	100	12	—	—	—	—
400	135	150	17.5	180	2	125	15	—	—	—	—
450	135	150	17.5	180	2	125	15	80	1	100	12
500	135	150	17.5	200	2	125	15	80	1	100	12
550	135	175	19.5	200	2	125	15	85	1	100	12
600	135	175	19.5	220	2	150	17.5	85	1	100	12
650	135	175	19.5	240	2	150	17.5	90	1	100	12
700	150	200	21.5	240	2	150	17.5	90	1	100	12
750	150	200	21.5	360	4	150	17.5	90	1	100	12
800	150	200	21.5	360	4	150	17.5	90	1	125	15
850	175	200	21.5	360	4	150	17.5	90	1	125	15
900	175	225	24	360	4	175	19.5	90	1	125	15
950	175	225	24	360	4	175	19.5	120	2	125	15
1000	175	225	24	360	4	175	19.5	120	2	125	15
1100	175	225	24	360	4	175	19.5	120	2	150	17.5
1200	190	250	26	360	4	175	19.5	140	2	150	17.5
1300	190	250	26	400	4	200	21.5	140	2	150	17.5
1400	190	275	28	400	4	200	21.5	150	2	150	17.5
1500	190	275	28	450	4	200	21.5	150	2	150	17.5
1600	200	300	30	480	4	200	21.5	150	2	150	17.5
1700	200	300	30	480	4	200	21.5	150	2	150	17.5
1850	200	325	32.5	540	4	200	21.5	180	2	150	17.5
2000	200	350	34.5	540	4	200	21.5	180	2	175	19.5
2150	200	350	34.5	540	4	200	21.5	150	2	175	19.5
2300	220	350	34.5	540	4	200	21.5	180	2	175	19.5
2500	220	350	34.5	640	4	225	24	180	2	175	19.5
2700	220	350	34.5	640	4	225	24	200	2	175	19.5
3000	220	350	34.5	640	4	225	24	200	2	200	21.5
3300	240	375	39	640	4	250	26	200	2	200	21.5
3600	240	375	39	640	4	250	26	200	2	200	21.5
3900	240	400	43.5	640	4	250	26	200	2	200	21.5
4200	240	400	43.5	640	4	250	26	200	2	200	21.5
4500	240	425	48.5	720	4	250	26	200	2	225	21
4800	240	425	45.5	720	4	250	26	200	2	225	24
5100	240	—	53	750	4	275	25	200	2	225	24
5400	240	—	53	800	4	275	25	240	2	225	24
5900	240	—	53	880	4	275	28	240	2	250	26
6200	240	—	57	860	6	300	30	240	2	250	26
6600	240	—	57	860	6	300	30	240	2	250	26
7000	240	—	57	560	6	300	30	240	2	250	26
7400	240	—	57	860	6	300	30	240	2	250	26
7800	240	—	57	860	6	300	30	480	4	250	26
8200	240	—	61.5	860	6	300	30	450	4	250	26
8600	240	—	61.5	960	6	325	32	410	4	250	26
9000	240	—	61.5	860	6	325	32	480	4	250	26
9600	240	—	61.5	960	6	325	32	410	4	250	26
								480	4	250	26

LAMPIRAN 3

Pumps

Hose diameter d_h, mm	Hose length l_h, m	Nozzle orifice diameter d_n, mm: 10	13	16	19	Hose diameter d_h, mm	Hose length l_h, m	Nozzle orifice diameter d_n, mm: 10	13	16	19
		Characteristic D						Characteristic D			
50	0	0.121	0.346	0.793	1.577	65	0	0.121	0.346	0.793	1.577
	10	0.119	0.331	0.722	1.320		10	0.1205	0.342	0.776	1.51
	20	0.118	0.318	0.622	1.130		20	0.120	0.339	0.758	1.44
	40	0.114	0.304	0.568	0.882		40	0.1195	0.332	0.726	1.33
	60	0.111	0.274	0.498	0.723		60	0.1185	0.326	0.696	1.23
	80	0.108	0.257	0.442	0.612		80	0.118	0.320	0.669	1.15
	100	0.105	0.241	0.398	0.531		100	0.117	0.314	0.644	1.05

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

Inside diameter of the drainage main, mm	Capacity of each drainage pump, cu m per h	Inside diameter of the drainage main, mm	Capacity of each drainage pump, cu m per h
50	15	133	103
57	19	140	113
64	23	146	124
70	28	152	135
76	34	158	146
82	40	165	158
89	46	171	171
95	53	178	183
103	60	184	197
108	68	190	210
114	76	197	224
120	84	205	240
127	93		

LAMPIRAN 2

KEKENTALAN DAN KEKENTALAN KINEMATIK DELAPAN FLUIDA PADA 1 ATM DAN 20°C

Fluida	μ, kg/(m·s)†	Nisbah $\mu/\mu(H_2)$	ρ, kg/m³	ν, m²/s†	Nisbah $\nu/\nu(Hg)$
Hidrogen	8,8 E–6	1,0	0,084	1,05 E–4	920
Udara	1,8 E–5	2,1	1,20	1,51 E–5	130
Bensin	2,9 E–4	33	680	4,22 E–7	3,7
Air	1,0 E–3	114	998	1,01 E–6	8,7
Ethanol	1,2 E–3	135	789	1,52 E–6	13
Air-raksa	1,5 E–3	170	13.580	1,16 E–7	1,0
Minyak pelumas SAE	0,29	33.000	891	3,25 E–4	2.850
Gliserin	1,5	170.000	1.264	1,18 E–3	10.300

† 1 kg/(m·s) = 0,0209 slug/(ft·s); 1 m²/s = 10,76 ft²/s.

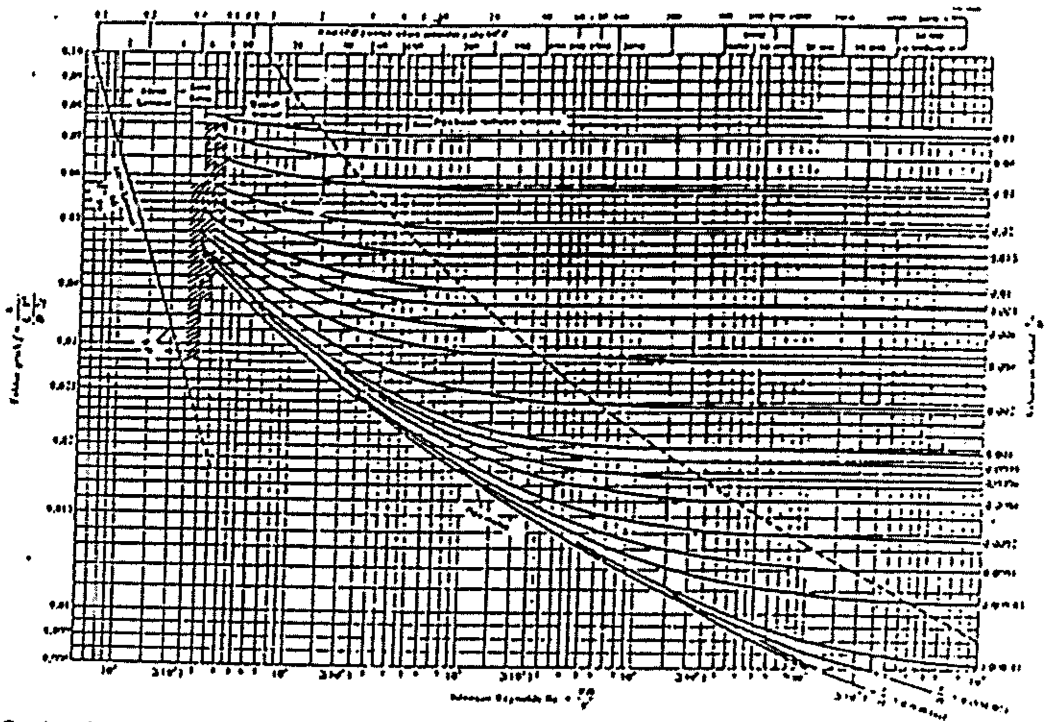
CONTOH GRAVITASI JENIS BEBERAPA ZAT CAIR PADA 20°C (68°F)

Zat Cair	Gravitasi Jenis
Bensin	0.66 – 0.69
Alkohol denaturasi	0.80
Minyak tanah	0.80 – 0.84
Minyak mentah	0.80 – 0.92
Minyak kastor	0.97
Air laut	1.025
Karbon tetraklorida	1.594
Aseulena tetrabromida	2.962
Air raksa (Hg)	13.546

BERAT JENIS BEBERAPA FLUIDA YANG LAZIM

Fluida	Berat Jenis ρg pada 68°F = 20°C	
	lbf/ft³	N/m³
Udara (pada 1 atm)	0,0752	11,8
Etanol	49,2	7.733
Minyak pelumas SAE 30	57,3	8.996
Air	62,4	9.790
Air laut	64,0	10.050
Gliserin	78,7	12.360
Karbon tetraklorida	99,1	15.570
Air-raksa	846	133.100

LAMPIRAN 1



Gambar 6.13 Bagan Moody untuk gesekan pipa berdinding halus/kasar.

SIFAT-SIFAT ZAT CAIR YANG LAZIM PADA 1 atm DAN 20°C (68°F)

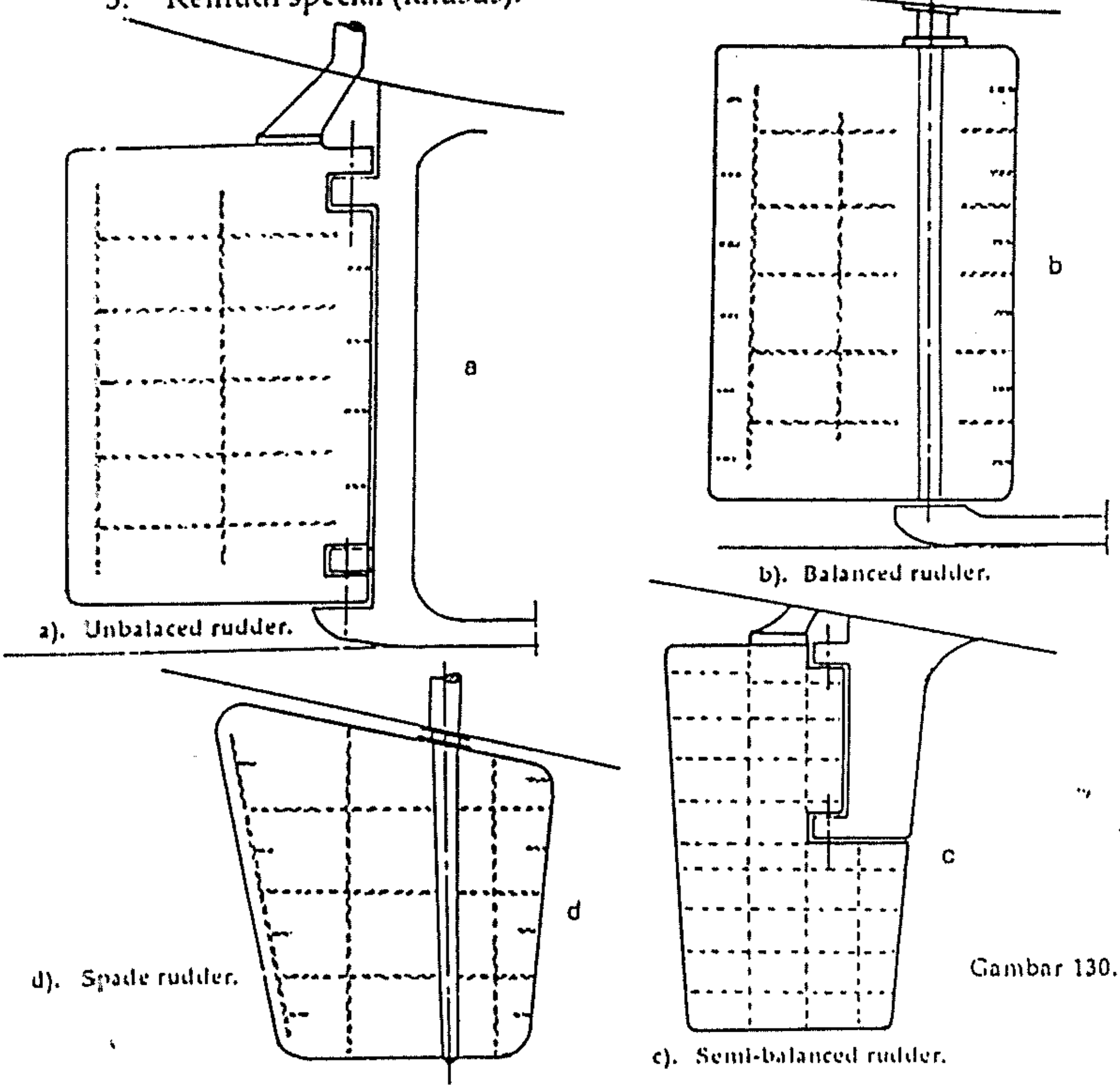
Zat cair	ρ, kg/m³	μ, (N·s)/m²	Υ, N/m†	p_v, N/m²	Modulus limbak, N/m²
Amonia	608	2.20 E−4	2.13 E−2	9.10 E+5	
Bensin	881	6.51 E−4	2.88 E−2	1.01 E+4	1.05 E+9
Karbon tetrakhlorida	1,590	9.67 E−4	2.70 E−2	1.20 E+4	[illegible]
Etanol	789	1.20 E−3	2.28 E−2	5.7 E+3	8.96 E+8
Gasolin	680	2.92 E−4	2.16 E−2	5.51 E+4	9.58 E+8
Gliserin	1,260	1.49	6.33 E−2	1.4 E−2	4.34 E+9
Minyak tanah	804	1.92 E−3	2.8 E−2	3.11 E+3	1.43 E+9
Air-raksa	13,550	1.56 E−3	4.84 E−1	1.1 E−3	2.55 E+10
Metanol	791	5.98 E−4	2.25 E−2	1.34 E+4	8.27 E+8
Pelumas SAE 10	917	1.04 E−1	3.6 E−2	...	1.31 E+9
Pelumas SAE 30	917	2.90 E−1	3.5 E−2	...	1.38 E+9
Air	998	1.00 E−3	7.28 E−2	2.34 E+3	2.19 E+9
Air laut	1,025	1.07 E−3	7.28 E−2	2.34 E+3	2.28 E+9

† Bersentuhan dengan udara.

2. Kemudi balansir, dimana luas sayap kemudi terbagi dua, bagian dimuka dan dibelakang sumbu putar kemudi (gambar b).
3. Kemudi setengah balansir, dimana bagian atas sayap kemudi termasuk kemudi biasa, sedang bagian bawah merupakan kemudi balansir sedangkan bagian atas dan bawah tetap merupakan satu bagian (gambar c).

B). Dipandang dari sulfies (sepatu linggi) dibagi :
1. Kemudi meletak (gambar a dan b)
2. Kemudi menggantung (gambar d)
3. Kemudi setengah menggantung (gambar c)

C). Dipandang dari konstruksinya dibagi :
1. Kemudi plat (satu lapis plat).
2. Kemudi berongga.
3. Kemudi special (khusus).



a). Unbalaced rudder.
b). Balanced rudder.
c). Semi-balanced rudder.
d). Spade rudder.

Gambar 130.

TABLE 3. *Dimensions of the two-bladed screws, type B 2.30*

	r/R	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	
Length of the blade sections as percentages of the maximum length of the blade sections at 0.6 R.	from centre line to trailing edge	28.68	32.67	36.62	40.53	44.18	46.97	48.22	45.46	14.87	Length of blade section at 0.6 R = 0.3313 D if F_a/F = 0.30
	from centre line to leading edge	46.05	51.24	54.91	56.52	55.82	52.22	44.63	30.31	—	
	total length	74.73	83.91	91.53	97.05	100.00	99.19	92.85	75.77	—	
Blade-thickness ratio as percentages of the diameter		4.46	3.94	3.42	2.90	2.38	1.86	1.34	0.82	0.30	Maximum thickness at centre of shaft = = 0.055 D
Distance of maximum thickness from leading edge as percentages of the length of the sections		35.00	35.00	35.00	35.50	38.90	44.20	47.80	50.00	—	

TABLE 4. *Dimensions of the three-bladed screws, types B 3.35, B 3.50 and B 3.65*

	r/R	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	
Length of the blade sections as percentages of the maximum length of the blade sections at 0.6 R.	from centre line to trailing edge	28.68	32.67	36.62	40.53	44.18	46.97	48.22	45.46	14.87	Length of blade section at 0.6 R = 0.3698 D if F_a/F = 0.50
	from centre line to leading edge	46.05	51.25	54.91	56.52	55.82	52.22	44.63	30.31	—	
	total length	74.73	83.91	91.53	97.05	100.0	99.19	92.85	75.77	—	
Blade-thickness ratio as percentages of the diameter		4.06	3.59	3.12	2.65	2.18	1.71	1.24	0.77	0.30	Maximum thickness at centre of shaft = = 0.05 D
Distance of maximum thickness from leading edge as percentages of the length of the sections		35.0	35.0	35.0	35.5	38.9	44.2	47.8	50.0	—	

TABLE 5. *Dimensions of the four-bladed screws, types B 4.40, B 4.55 and B 4.70*

	r/R	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	
ength of the blade sections percentages of the maxim length of the blade tions at 0.6 R.	from centre line to trailing edge	29.18	33.32	37.30	40.78	43.92	46.68	48.35	47.00	20.14	Length of blade section at 0.6 R = 0.2187 D if F_a/F = 0.40
	from centre line to leading edge	46.9	52.64	56.32	57.60	56.08	51.40	41.65	25.35	—	
	total length	76.08	85.96	93.62	98.38	100.0	98.08	90.00	72.35	—	
de-thickness ratio as percentages of the diameter		3.66	3.24	2.82	2.40	1.98	1.56	1.14	0.72	0.30	Maximum thickness at centre of shaft = = 0.045 D
ance of maximum thickness from leading as percentages of the length of the sections		35.0	35.0	35.0	35.5	38.9	44.3	47.9	50.0	—	

TABLE 6. *Dimensions of the five-bladed screws, types B 5.45 and B 5.60*

	r/R	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	
h of the blade sections centages of the maxilength of the blade s at 0.6 R.	from centre line to trailing edge	29.18	33.32	37.30	40.78	43.92	46.68	48.35	47.00	20.14	Length of blade section at 0.6 R = 0.1968 D if F_a/F = 0.45
	from centre line to leading edge	46.90	52.64	56.32	57.60	56.08	51.40	41.65	25.35	—	
	total length	76.08	85.96	93.62	98.38	100.00	98.08	90.00	72.35	—	
hickness ratio as percentages of the diameter		3.26	2.89	2.52	2.15	1.78	1.41	1.04	0.67	0.30	Maximum thickness at centre of shaft = = 0.040 D
e of maximum thickness from leading percentages of the length of the sections		35.00	35.00	35.00	35.50	38.90	44.30	47.90	50.00	—	

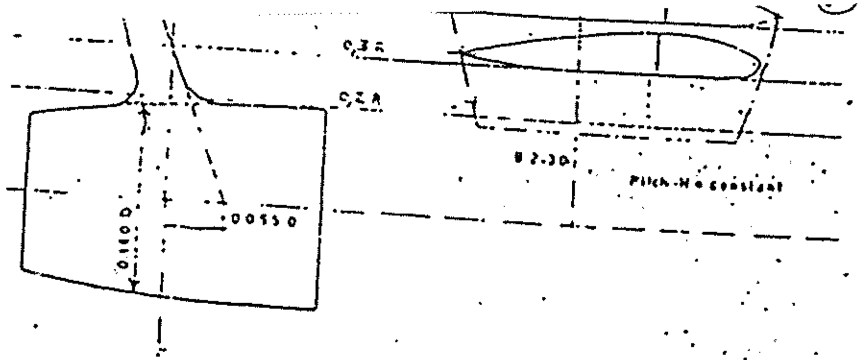


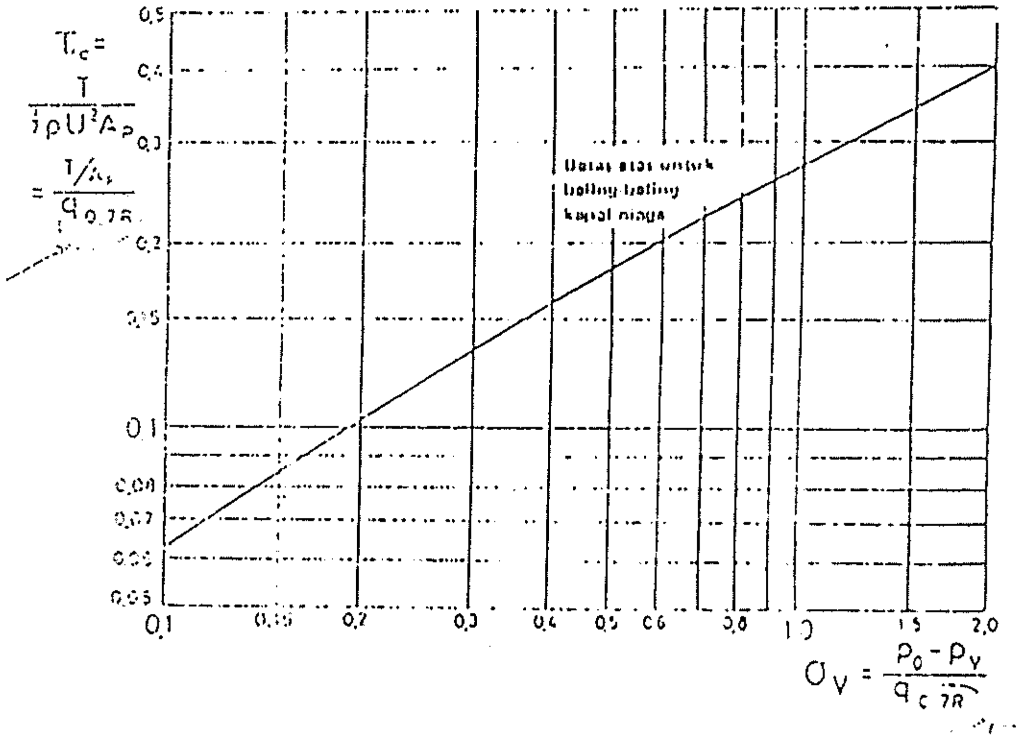
Fig. 33. *General plan of the two-bladed propeller of type B 2-30*

TABLE 2. *Table of ordinates of the B series*

Distance of the ordinates from the maximum thickness

r/R	From maximum thickness to trailing edge 100%	80%	60%	40%	20%	From maximum thickness to leading edge 20%	40%	60%	80%	90%	95%	100%
						Ordinates for the back						
0.2	—	53.35	72.65	86.90	96.45	98.60	94.50	87.00	74.40	64.35	56.95	—
0.3	—	50.95	71.60	86.80	96.80	98.40	94.00	85.30	72.50	62.65	54.90	—
0.4	—	47.70	70.25	86.55	97.00	98.20	93.25	84.30	70.40	60.15	52.20	—
0.5	—	43.40	68.40	86.10	96.95	98.10	92.40	82.30	67.70	56.80	48.60	—
0.6	—	40.20	67.15	85.40	96.80	98.10	91.25	79.35	63.60	52.20	43.35	—
0.7	—	39.40	66.90	84.90	96.65	97.60	88.80	74.90	57.00	44.20	35.00	—
0.8	—	40.95	67.80	85.30	96.70	97.00	85.30	68.70	48.25	34.55	25.45	—
0.9	—	45.15	70.00	87.00	97.00	97.00	87.00	70.00	45.15	30.10	22.00	—
0.95	—	44.80	72.00	88.00	97.20	97.20	88.80	72.00	44.80	29.50	21.60	—
						Ordinates for the face						
0.2	30.00	18.20	10.90	5.45	1.55	0.45	2.30	5.90	13.45	20.30	26.20	40.00
0.3	25.35	12.20	5.80	1.70	—	0.05	1.30	4.60	10.85	16.55	22.20	37.55
0.4	17.85	6.20	1.50	—	—	—	0.30	2.65	7.80	12.50	17.90	34.50
0.5	9.70	1.75	—	—	—	—	—	0.70	4.30	8.45	13.30	30.40
0.6	5.1	—	—	—	—	—	—	—	0.80	4.45	8.40	24.50
0.7	—	—	—	—	—	—	—	—	—	0.40	2.45	16.05
0.8	—	—	—	—	—	—	—	—	—	—	—	7.40

Note: The percentages of the ordinates relate to the maximum thickness of the corresponding sections ... assumed to be rectilinear. The connecting lines of the ...



Gbr.No. 2 Diagram Burril

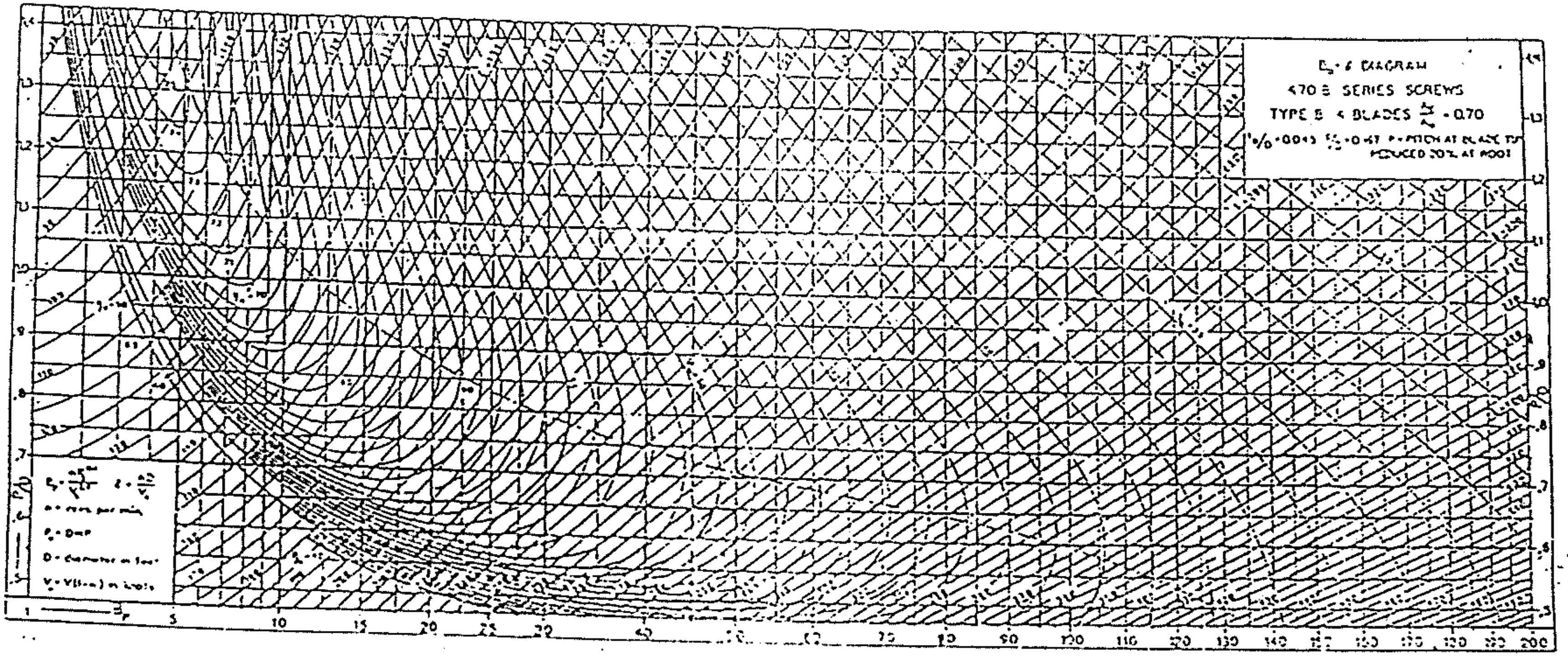


Fig 117

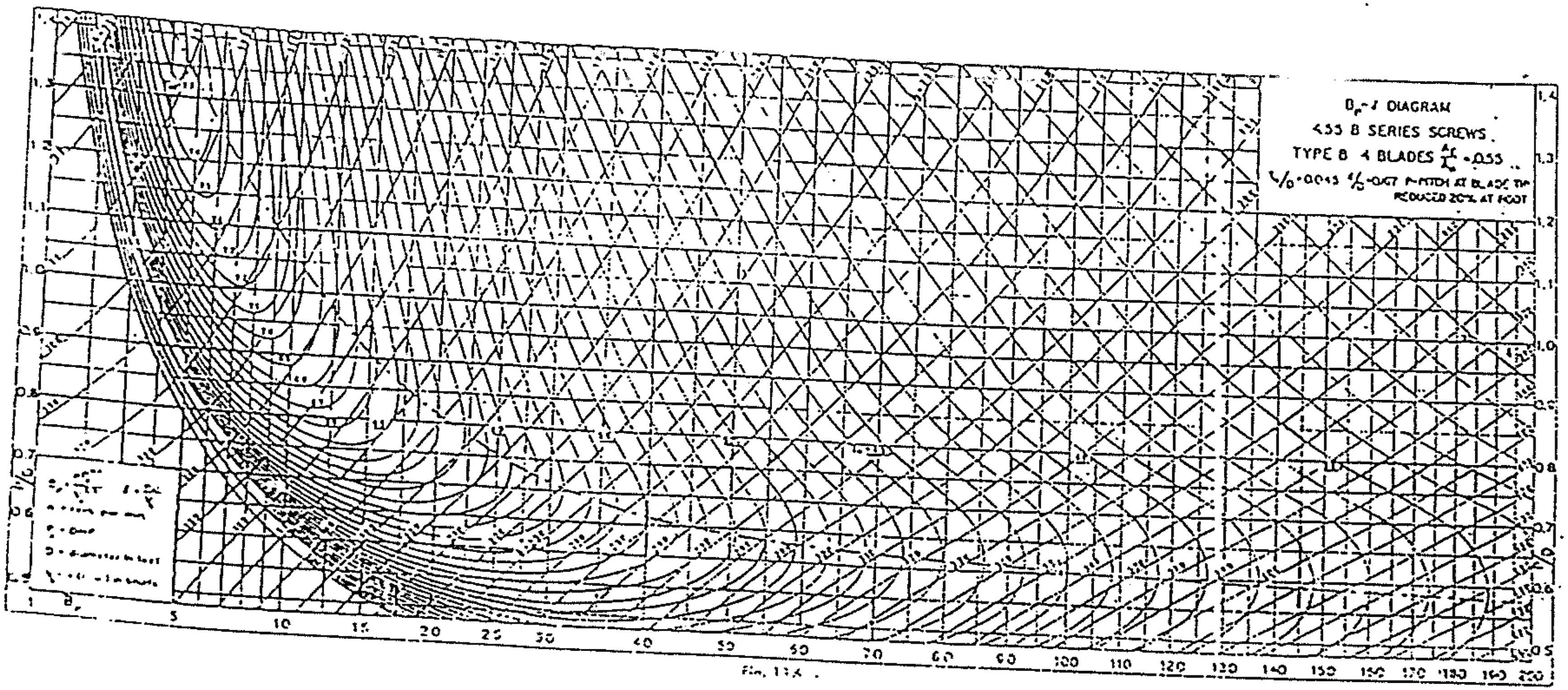
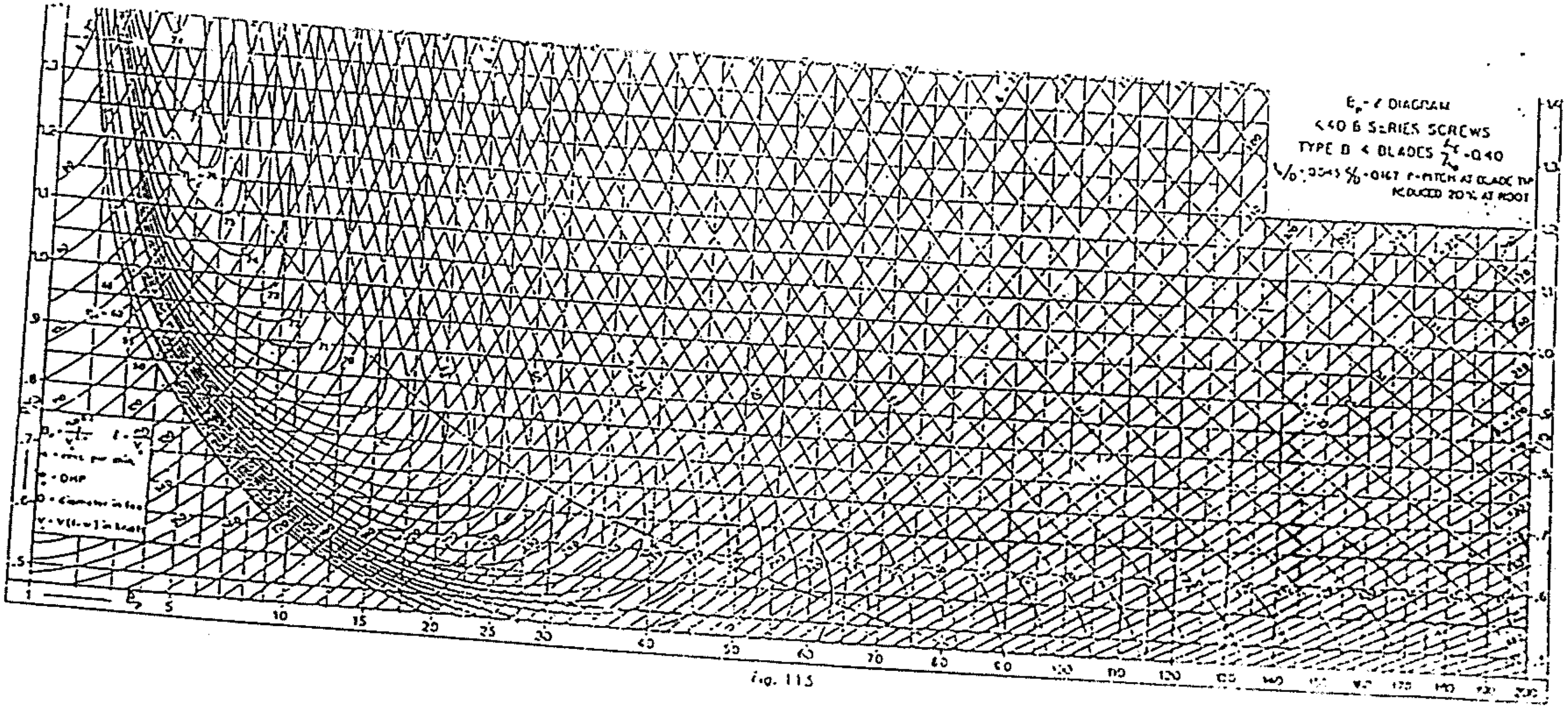
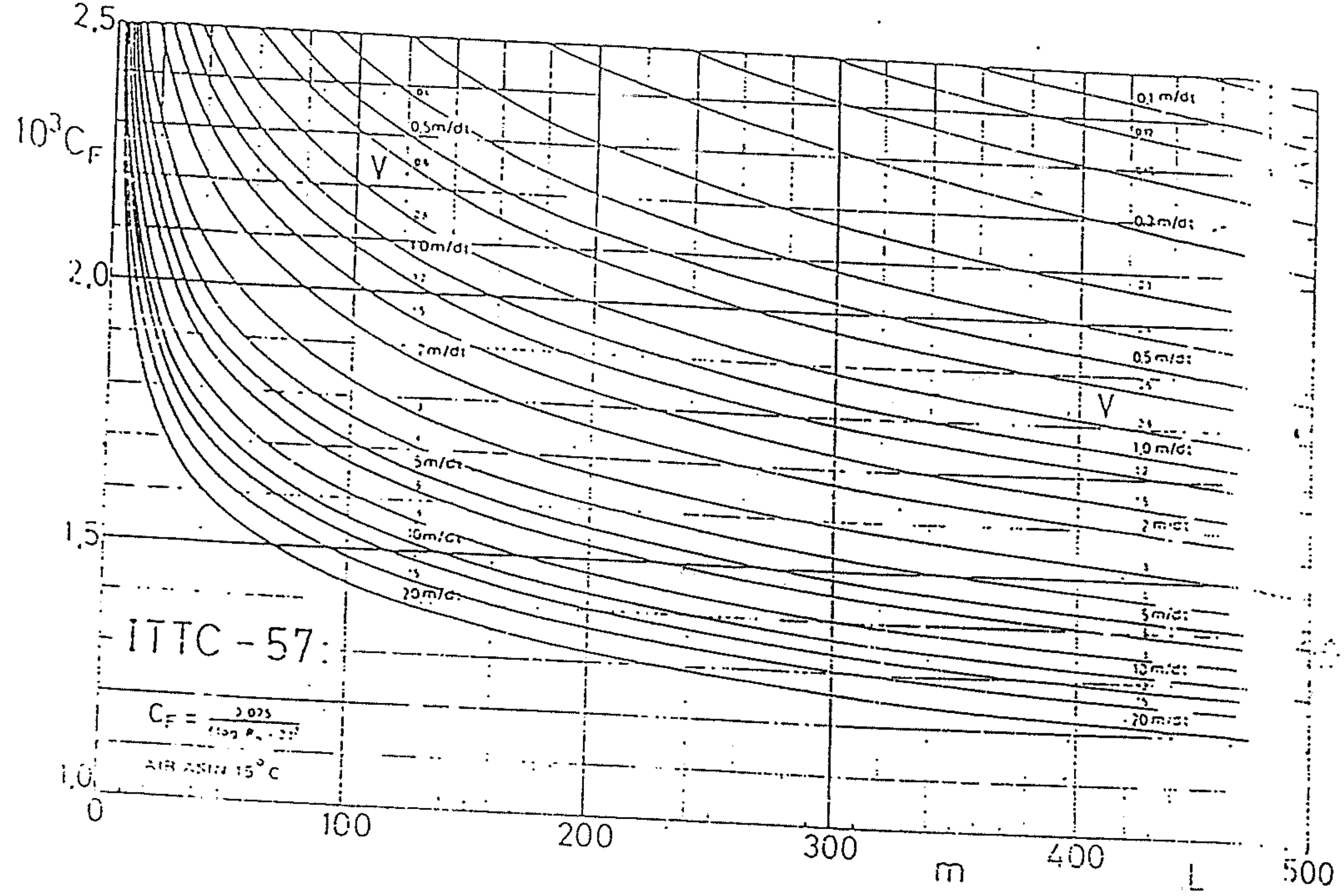


Fig. 11.X

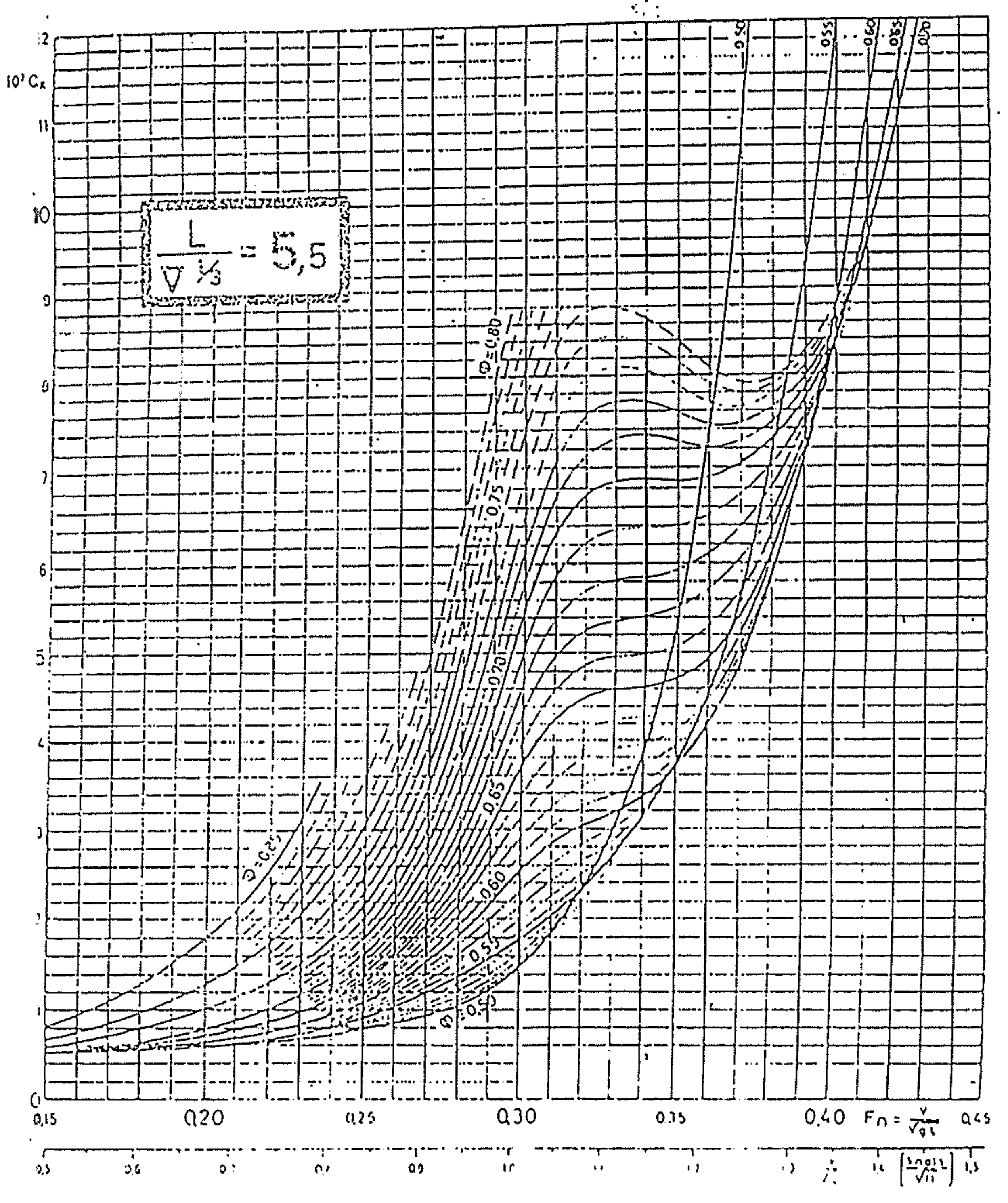
Gbr. No. 4 Diagram Bp - δ B Series Tipe B - 4



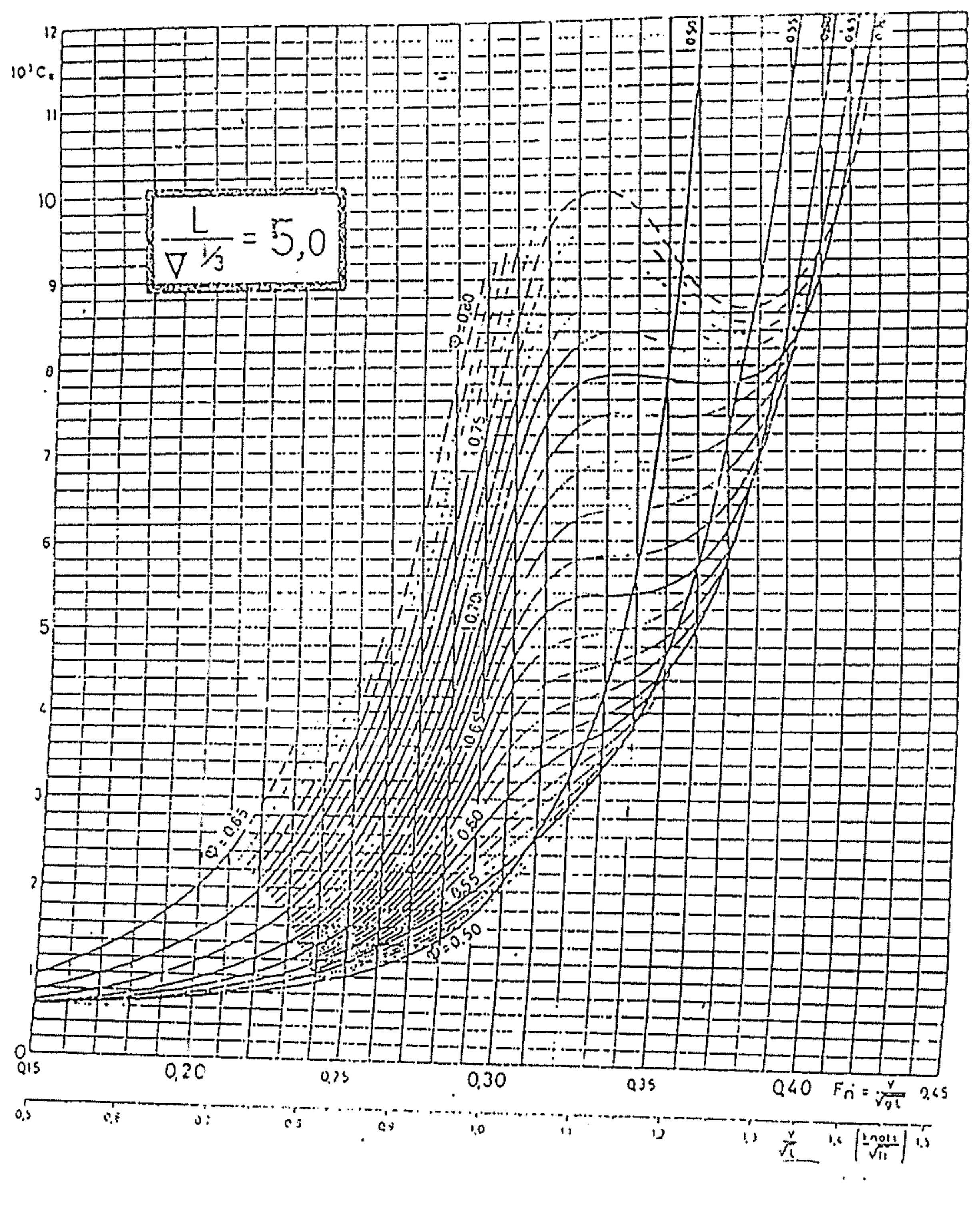
Gbr. No. 7 Diagram Bp - δ B Series Type B - 40



Gbr.No. 5 " Hambatan Gesekan " (ITTC - 57)



Gbr. No. . Koefisien Hambatan Sisa ($L/\nabla^{1/3}$)



Gbr. No. 2 Koefisien Hambatan Sisa ($L / \nabla^{1/3}$)

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%
Braket dan poros baling-baling	Untuk kapal ramping C_R dinaikkan sebesar 5 – 8%

Gbr No.6 Koreksi Bentuk Anggota Badan

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

Gbr.No. 6 Koreksi Tahanan Tambahan

F_n = 0,15	0,18	0,21	0,24	0,27	0,30	0,33	0,36	φ
		+ 0,2	0	– 0,2	– 0,4	– 0,4	– 0,4	0,50
		+ 0,2	0	– 0,2	– 0,3	– 0,3		0,60
	+ 0,2	0	– 0,2	– 0,3	– 0,3			0,70
+ 0,1	0	– 0,2						0,80

Gbr.No.4 Koreksi Bentuk Haluan

... used, for example by using it to tow another vessel, the ... fixed pitch propeller law curve will become steeper although still of $P \propto N^3$ form as shown in Figure 11–3. A heavier tow will steepen it further, corresponding

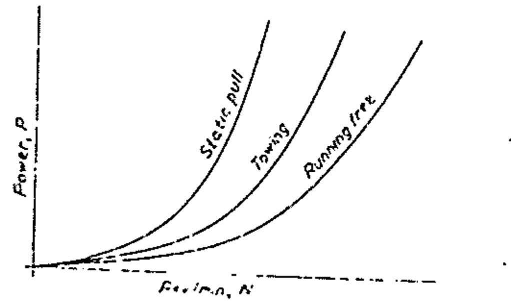


FIG. 11–3—Effect of increasing the resistance of the vessel.

to a lower forward speed of the vessel until the most sharply rising curve is produced by the static pull condition. Vessels designed to tow must take these circumstances into account. The machinery and propellers of docking tugs are usually designed so that the static pull curve passes through the maximum continuous rating point, this results in the highest possible value for the static pull but precludes the use of the maximum installed horsepower once the tow is under way. A tug intended for continuous towing will have machinery designed to match the propeller curve at the intended towing speed and a similar design procedure will be followed for a trawler which tows gear at a relatively high speed.

11.4. *Variable pitch propellers*

A variable pitch propeller has blades which can be moved to take up various pitch angles. Any one setting results in a particular propeller law curve for a vessel of given resistance and a whole family of propeller law curves can be produced for such a vessel by altering the blades to different settings. A similar set can be formed for any given increase in resistance of the vessel. It is therefore possible, within the design limits of the propeller, to choose settings which would match the maximum continuous rating of the engine under any condition from static pull to running free, and so utilize the maximum continuous rated power of the engine to obtain the highest tow rope pull and the highest speed attainable in the circumstances. Compared with a fixed pitch propeller the variable pitch propeller makes better use of the power available from the engine over a wide range of diverse conditions. On the other hand it has a disadvantage in that the efficiency is not so high as that of the fixed pitch propeller operating at optimum conditions but this is slight and obviously only of importance in a vessel that spends its time regularly at one speed and in one loaded condition.

... a vessel of constant resistance but which is called upon to operate at different forward speed during its service. If lines of constant fuel consumption of the kind shown in Figure 2–8, are added to the limits of Figure 11–2 their appearance is as in Figure 11–4. The family of curves

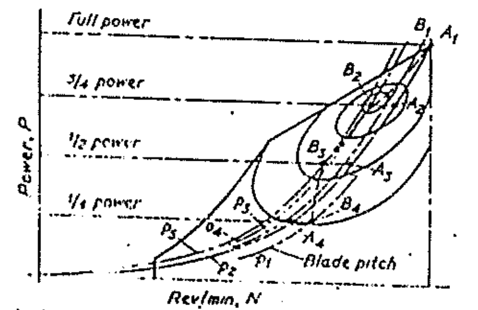


FIG. 11–4 Optimizing fuel consumption by controllable pitch propeller.

for a variable pitch propeller and vessel of constant resistance lies across these lines of constant fuel consumption and by choosing combinations of pitch angle and rev/min, the operating line for various speeds can be made to pass through the regions of lowest fuel consumption as is shown. It must, of course, be remembered that alteration of propeller pitch slightly affects propeller efficiency and therefore the relative location of load points and minimum fuel consumption points must be kept in mind in establishing the best relation between optimum engine thermal efficiency and optimum propeller efficiency. In Figure 11-4, the fuel consumption points at full load, three quarter load, half load and quarter load positions with a fixed pitch propeller are indicated at A1, A2, A3 and A4, whilst the improved fuel consumption points obtainable with variable pitch propeller are indicated at B1, B2, B3 and B4, on the dotted line passing through the isofuel consumption loops. Bearing in mind the change in propeller efficiency it may be that in practice optimum performance will be attained between these two settings. Such settings of pitch angle corresponding to rev/min may be selected automatically by appropriate control gear once they have been determined.

11.5. *Two engines geared to one propeller*

By gearing two or more medium speed engines together to drive a single propeller a high powered installation can be designed to occupy a small space and to have a low weight. Benefits in economy due to the use of slower turning higher efficiency propellers and the ability to use one engine only for low speed operation can be justifiably claimed for the arrangement as also can increased reliability and availability of the ship.

When two engines of equal power are geared to one propeller the relationship is as shown in Figure 11–5. Curve *A* represents the torque–rev/min characteristic of each single engine and curve *B* that of the two

... for example by using it to tow another vessel, the propeller law curve will become steeper although still of $P \propto N^3$ form as shown in Figure 11–3. A heavier tow will steepen it further, corresponding

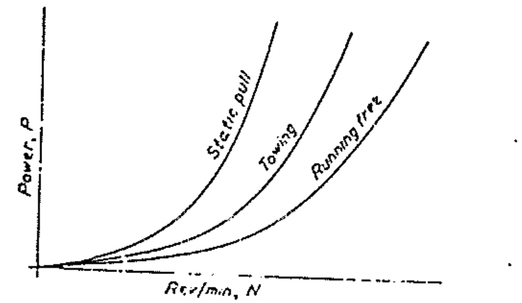


FIG. 11-3—*Effect of increasing the resistance of the vessel.*

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... a vessel of constant resistance but which is called upon to operate at different forward speed during its service. If lines of constant fuel consumption of the kind shown in Figure 2–8, are added to the limits of Figure 11–2 their appearance is as in Figure 11–4. The family of curves

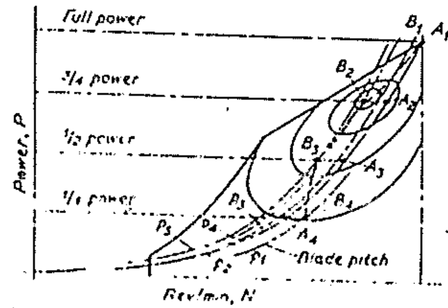


FIG. 11-4. *Optimizing fuel consumption by controllable pitch propeller.*

for a variable pitch propeller and vessel of constant resistance lies across these lines of constant fuel consumption and by choosing combinations of pitch angle and rev/min, the operating line for various speeds can be made to pass through the regions of lowest fuel consumption as is shown. It must, of course, be remembered that alteration of propeller pitch slightly affects propeller efficiency and therefore the relative location of load points and minimum fuel consumption points must be kept in mind in establishing the best relation between optimum engine thermal efficiency and optimum propeller efficiency. In Figure 11-4, the fuel consumption points at full load, three quarter load, half load and quarter load positions with a fixed pitch propeller are indicated at A1, A2, A3 and A4, whilst the improved fuel consumption points obtainable with variable pitch propeller are indicated at B1, B2, B3 and B4, on the dotted line passing through the isofuel consumption loops. Bearing in mind the change in propeller efficiency it may be that in practice optimum performance will be attained between these two settings. Such settings of pitch angle corresponding to rev/min may be selected automatically by appropriate control gear once they have been determined.

11.5. *Two engines geared to one propeller*

By gearing two or more medium speed engines together to drive a single propeller a high powered installation can be designed to occupy a small space and to have a low weight. Benefits in economy due to the use of slower turning higher efficiency propellers and the ability to use one engine only for low speed operation can be justifiably claimed for the arrangement as also can increased reliability and availability of the ship.

When two engines of equal power are geared to one propeller the relationship is as shown in Figure 11–5. Curve *A* represents the torque–rev/min characteristic of each single engine and curve *B* that of the two

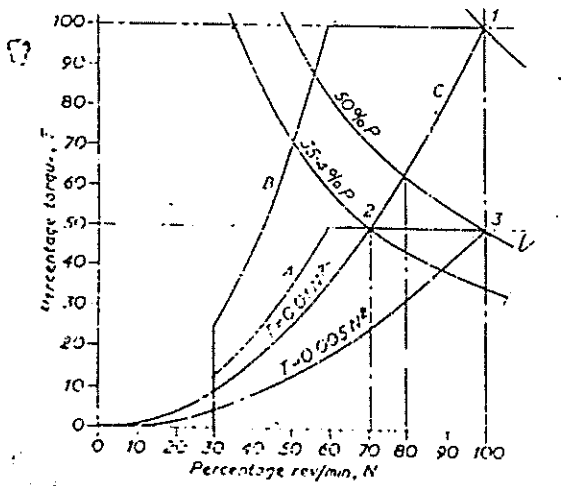


FIG. 11-5 Torque-speed curves for two engines driving one propeller.

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

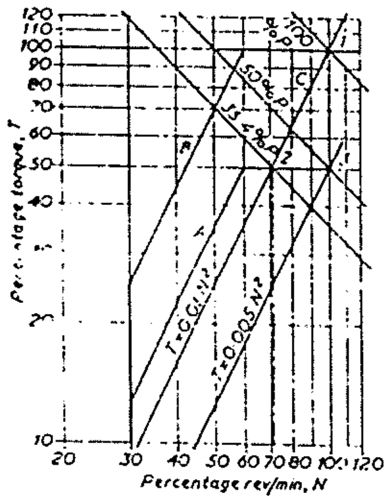


FIG. 11-6 Torque—speed curves for two engines driving one propeller. log-log scales.

If a controllable pitch propeller is fitted the pitch can be reduced when running on one engine permitting it to develop its full rev/min as well as its full torque as at point 3. The power is 0·5 of total full power and the curves can be plotted easily and quickly on "log-log paper" as shown in Figure 11–6 which has been drawn using percentage scales. Curves of constant power and power proportional to (rev/min)ⁿ become groups of parallel straight lines which are easily located. The curves and points in Figure 11–6 correspond to those in Figure 11–5 and are lettered and numbered accordingly.

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

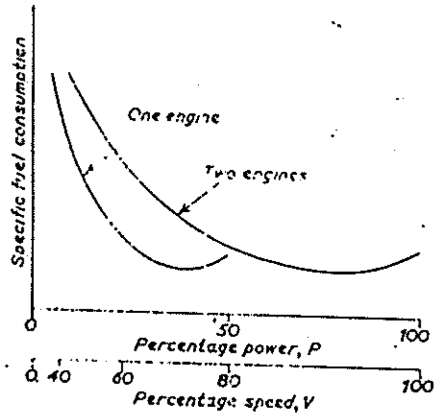


FIG. 11–7—Fuel consumption, power and speed for two engines driving one controllable pitch propeller.

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

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

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

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

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

Thus :

N = rotax
P - Power
V = speed
T - Tors.

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

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

$$Va \quad \left(\frac{Pa}{}\right)^{1/3} \quad \left(\frac{Ta}{}\right)^{1/2}$$

3.1 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.

- 6% lower fore peak tank
- 3% upper fore peak tank
- 2% double bottom tank.

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

Sounding/ullage tables delivered by yard.

.) Provisions/persons/luggage

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

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, a!

- 1

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. Charges are dependent on tonnage, for example in ports, canals, for pilots ... . Most of the shipbuilding statistics are based on tonnage.

Tonnage unit: gross ton

1 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

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

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

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

D = depth in $[m]$ "

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

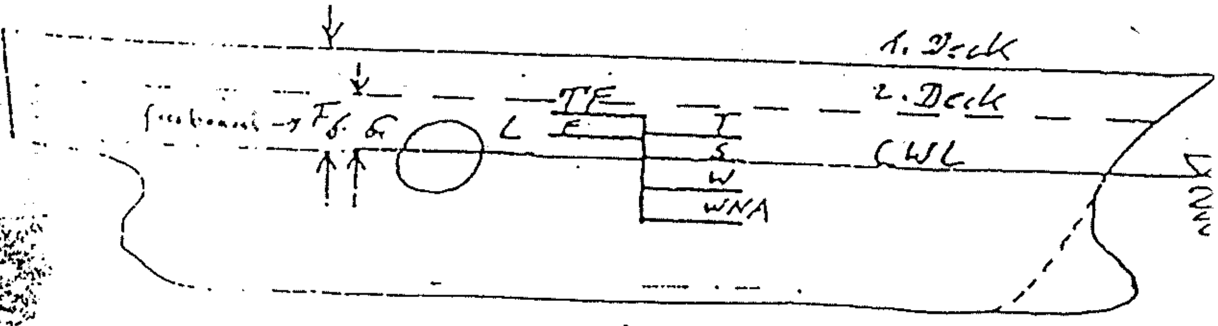
N_2 = number of other passengers.

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

Definition of freeboard:

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

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

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

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

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

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

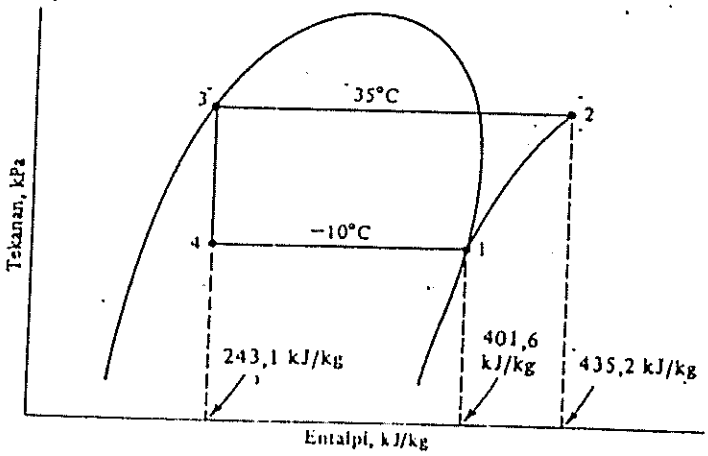
Untuk menemukan h_2 melalui garis entropi tetap geser titik 1 hingga mencapai tekanan jenuh yang sesuai dengan suhu 35°C. Tekanan pengembunan ini adalah 1354 kPa, dan nilai h_2 = 435,2 kJ/kg.

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

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

(*a*) Dampak refrigerasi:

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



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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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Tabel II-

L. B. H (m)	L. B. H (ft3)	Kapasitas (ft3)	Jumlah orang	berat sekoci (kg)	Berat Orang (kg)	berat perlengkapan (kg)	Total berat (kg)
9,4 x 2,74 x 1 x 1,114	30 x 9 x 3,75	607	60	2205	4500	356	7061
8,84 x 2,74 x 1,10	29 x 8,75 x 3,60	545	54	1976	4050	356	6382
8,53 x 2,59 x 1,07	28 x 8,50 x 3,50	500	50	1824	3750	330	5894
8,23 x 2,51 x 1,04	27 x 8,25 x 3,40	454	45	1646	3376	330	5351
7,92 x 2,44 x 0,99	26 x 8,00 x 3,25	405	40	473	3000	305	4778
7,62 x 2,36 x 0,96	25 x 7,75 x 3,15	366	36	1326	2700	305	4331
7,31 x 22,9 x 0,91	24 x 7,50 x 3,00	324	32	1180	2400	254	3843
7,01 x 2,29 x 0,88	23 x 7,50 x 2,90	300	30	1087	2250	254	3591
6,71 x 2,21 x 0,84	22 x 7,25 x 2,75	236	26	955	1950	229	3134
6,40 x 2,13 x 0,82	21 x 7,00 x 2,70	238	23	864	1725	229	2818
6,10 x 2,06 x 0,79	20 x 6,75 x 2,60	210	21	762	1575	203	2540
5,79 x 1,98 x 0,76	19 x 6,50 x 2,50	182	18	650	1350	178	2178
5,49 x 1,90 x 0,73	18 x 6,25 x 2,40	162	16	590	1200	152	1942
5,18 x 1,83 x 0,715	17 x 6,00 x 2,30	143	14	508	1050	152	1710
4,88 x 1,75 x 0,70	16 x 5,75 x 2,30	127	12	475	900	127	1484

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

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

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

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

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

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

P_d: Tekanan keluar dari silinder tingkat pertama (kgf/cm² abs),

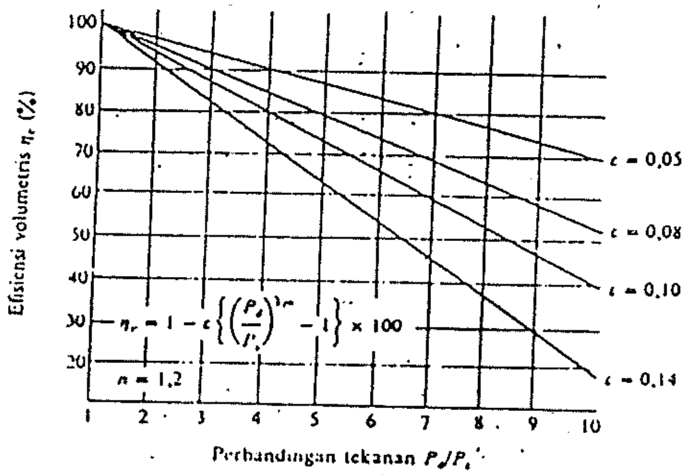
P_s: Tekanan isap dari silinder tingkat pertama (kgf/cm² abs).

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

Tanda ≈ 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 pendinginan, dll. Namun, menentukan secara tepat pengaruh masing-masing faktor tersebut adalah sangat sulit. Karena itu faktor-faktor ini digabungkan dalam efisiensi adiabatik keseluruhan.

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

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

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

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

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

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

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

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

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

Tabel 2.7 Daya yang diperlukan untuk kompresi adiabatis teoritis.

Tekanan (kgf/cm² (G))	Kompresi 1-tingkat (kW)	Kompresi 2-tingkat (kW)	Tekanan (kgf/cm² (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 m³/menit udara bebas. 1 kgf/cm² = 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%.

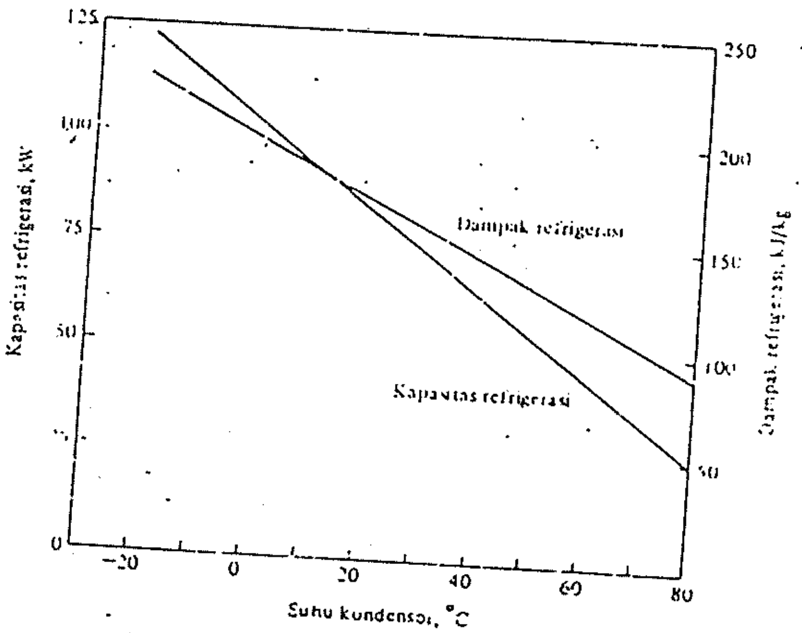
Selanjutnya, karena harga daya adiabatis teoritis untuk kompresor 1 tingkat berbeda dengan harga untuk kompresor 2 tingkat, maka memperbandingkan efisiensi kompresor harus dilakukan di antara yang sama jumlah tingkatnya.

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

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

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

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



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

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

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

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

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

where n_m = 100 to 350 rpm for steam engines
n_m = 300 to 1,800 rpm for electric motors.

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

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

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

where α° = maximum rudder angle from the middle-line plane.

It follows from formula (314) that

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

Combining equations (315) and (316) we obtain

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

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

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

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

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

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

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

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

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

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

Table 47

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

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

re, R_{br} = breaking strength of the warping hawser.
The speed at which a capstan barrel heaves in a warping hawser be taken from Table 58 which has been compiled from the manu-aring specifications for capstans worked out by the Central ine 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

he speed at which a warping hawser is heaved in by a windlass l limited by the values in Table 58, and usually is equal to l 0.4 m per sec.

has been stated previously that the same machinery is em-d both for anchoring and warping purposes. It follows that lasses and capstans must be designed so as to ensure normal tion of both the anchoring and warping arrangements.

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

ie number of anchors, their weight, the size of the anchor cables, the circumference of warping hawsers and towing ropes, heir length are determined from the tables of the pertinent ations of the Shipping Register. To find these values it is nec-to calculate the rigging characteristic of the anchoring and ng arrangement:

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

L = length of the ship at the summer load line, m
B = maximum breadth between the outer edges of the ship's hull, m

... 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 port cable.

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

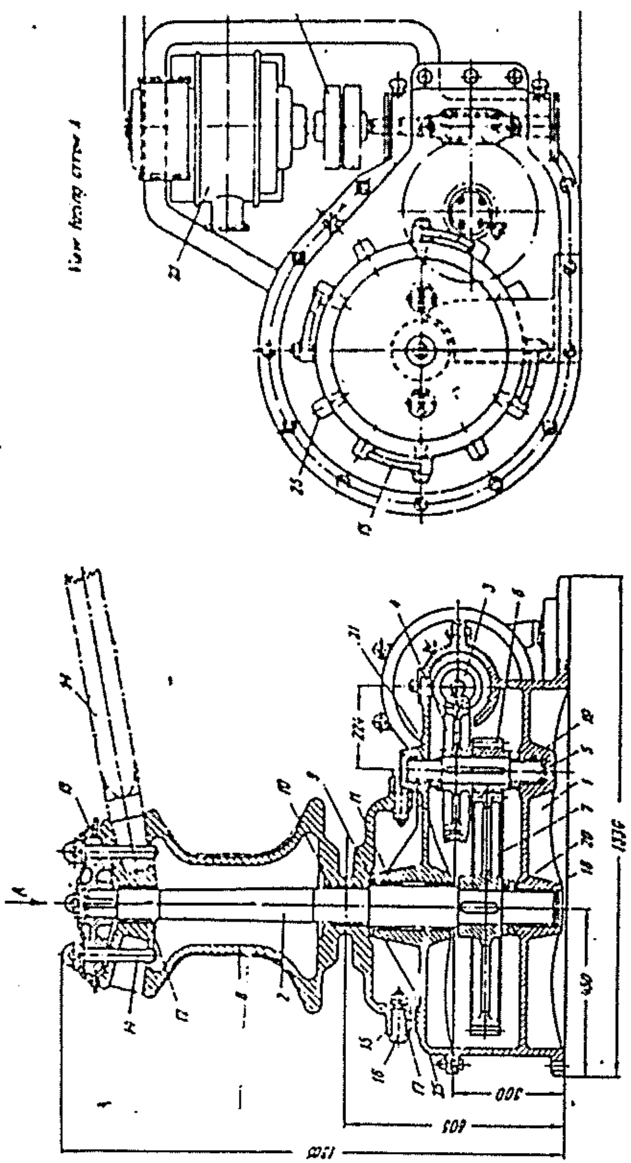


Fig. 169.

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

$$T_{cl} = 2f_h(G_a + p_aL_a)\left(1 - \frac{\gamma_w}{\gamma_a}\right) = 2\times 1.35\,(G_a + p_aL_a)\left(1 - \frac{1.025}{7.75}\right) = 2.35\,(G_a + p_aL_a)\ \ \text{kg} \qquad (383)$$

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

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

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

* In breaking away one anchor from the bottom

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

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

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

This equation can be solved for V_e and V_f:

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

and

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

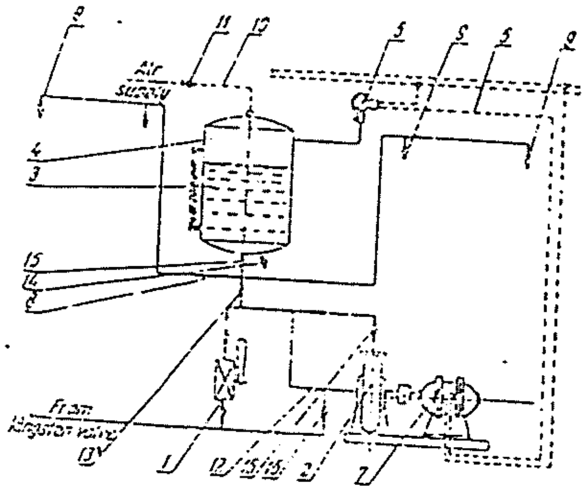


Fig. 189.

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

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

Therefore the minimum and maximum volumes of the air are

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

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

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

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

(b) SANITARY AND SCUPPER SYSTEMS

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

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

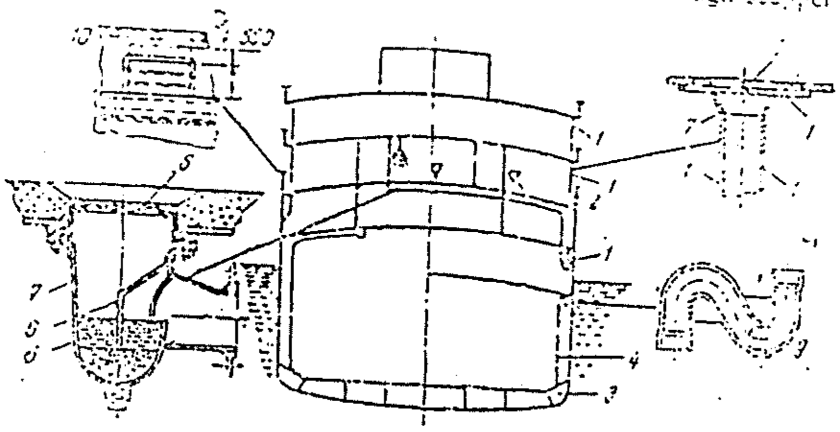


Fig. 190.

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

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

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

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

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

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.

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).

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: $\boxed{\gamma = 0.95\ t/m^3}$

Required volume of storage tanks

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

Additions to the volume:

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

Diesel oil

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

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

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

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

additions see fuel oil!

Lubrication oil

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

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

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

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

Fresh water

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

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

Fresh water tanks have to be separated from all other tanks by cofferdams.

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

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

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

:

[mm] Inside diameter of main bilge line

[mm] Inside diameter of branch bilge line

[m] Breadth of ship

[m] Moulded depth of ship

[m] Total length of cargo area

[m] Length of watertight compartment

[m] Maximum breadth of cargo tanks

[m] Maximum depth of cargo tanks

[m] Total length of all cargo tanks

[m] Length of tanks in the watertight compartment.

›city of each bilge pump is to be calculated g to Section 11, N.3.1. At least two bilge ›e to be provided.

When separate bilge pumps, e.g. ejectors ›ided for compartments with watertight › the pump capacity is to be evaluated as ›n 4.1.3 and is to be divided according the the individual compartments. For each ›ent two bilge pumps are to be fitted of a ›f not less than 5 m³/h each.

›aces for independent tanks are to be pro- sounding arrangements.

›ast or cooling water lines are fitted in ›ndependent tanks bilge level alarms are ›led.

›e pumping of cargo pump rooms cofferdams in the cargo area

›e pumping equipment is to be located in ›ea to serve the cargo pump rooms and A cargo pump may also be used as a On oil tankers used exclusively for the flammable liquids with flash points ›, cargo pump rooms and cofferdams ›cted to the engine room bilge system.

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

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

4.3 Ballast systems in the cargo area

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

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

a) Minimum wall thicknesses

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

b) Only completely welded pipes or equivalent are permitted

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

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

5. Ventilation and gas-freeing

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

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

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

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

Appendix to Section 2

Part C :

Approximate Calculation of the Starting Air Supply

1. Starting air for installations with reversible engines

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

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

where

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

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

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

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

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

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

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

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

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

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

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

The following values of n_A are to be applied:

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

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

n_o [Rpm] = rated speed

2. Starting air for installations with non reversible engines

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

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

1.6 Pipe connections

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

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

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

1.6.3 The direct bilge suction and the emergency 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.

Calculation of pipe diameters

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

2 Dry cargo and passenger ships

main bilge pipes

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

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	[m^3/h]	minimum capacity
d_H	[mm]	calculated inside diameter of main bilge pipe

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

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

3.4 Use of other pumps for bilge pumping

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

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

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

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

3.5 Number of bilge pumps for cargo ships

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

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

3.6 **Number of bilge pumps for passenger ships**

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

4. **Bilge pumping for various spaces**

4.1 **Machinery spaces**

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

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

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

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

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

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

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

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

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

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

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

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

4.2 **Shaft tunnel**

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

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

6.3.2 Bilge lines

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

6.3.3 Ballast pipes

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

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

6.3.4 Fuel pipes

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

6.3.5 Cargo pipes

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

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

6.4 Control stands

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

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

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

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

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

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

6.5 Power units

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

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

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

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

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

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

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

7. Pumps

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

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

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

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

4.10 Cofferdams, pipe tunnels and void spaces

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

4.11 Chain lockers

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

5. Additional Rules for passenger vessels

5.1.1 The arrangement of bilge pipes

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

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

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

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

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

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

5.2 Bilge suctions

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

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

5.3 Arrangement of bilge pumps

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

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

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

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

c) bilge pumps are installed above the bulkhead deck

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

5.4 Passenger vessels for limited range of service

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

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

Additional Rules for tankers

Section 15, B.4.

Bilge testing

ilge arrangements are to be tested under the ty's supervision.

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

Oily water separating equipment

Ships of 400 tons gross and above shall be ith an oily water separator or a filter plant for aration of oil/water mixtures.

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

A sampling device is to be arranged in the ge line of oily water separating equiptering systems.

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

Discharge of fuel and oil residues

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

self-priming pump is to be provided for charge to reception facilities. The capacity ip shall be such that the sludge tank can be n a reasonable time.

separate discharge line is to be provided ge of fuel and oil residues to reception fa-

n regard to the installation on ships of oily water arators, filter plants, oil collecting tanks, oil harge lines and a monitoring and control system or ppm alarm device in the water outlet of oily water rators, compliance is required with the provisions e International Convention for the Prevention of ution from Ships, 1973, (MARPOL) and the ocol of 1978.

F.136 is to submitted for approval.

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.

Section 14

Rudder and Manoeuvring Arrangement

A. General

1. Manoeuvring arrangement

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

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

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

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

Guidance

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

1.5 For ice-strengthening see Section 15

2. Structural details

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

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

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

3. Size of rudder area

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

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

c_1 = factor for the ship type

- = 1,0 in general
- = 0,9 for bulk carriers and tankers having a displacement of more than 50.000 t
- = 1,7 for tugs and trawlers

c_2 = factor for the rudder type

- = 1,0 in general
- = 0,9 for semi-spade rudders
- = 0,8 for double rudders (per rudder)
- = 0,7 for high lift rudders

c_3 = factor for the rudder profile

- = 1,0 for NACA-profiles and plate rudder
- = 0,8 for hollow profiles

c_4 = factor for the rudder arrangement.

- = 1,0 for rudders in the propeller jet
- = 1,5 for rudders outside the propeller jet

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

Materials

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

In general materials having a minimum nominal upper yield point R_{eH} of less than 200 N/mm² and a minimum tensile strength of less than 400 N/mm² or more than 900 N/mm² shall not be used for rudder stocks, pintles, keys and bolts. The requirements of this Section are based on a material's minimum nominal upper yield point R_{eH} of 235 N/mm². If material is used having a R_{eH} differing from 235 N/mm², the material factor k_r is to be determined as follows:

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

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

R_{eH} = minimum nominal upper yield point of material used in [N/mm²]. R_{eH} is not to be taken greater than $0{,}7 \cdot R_m$ or 450 N/mm², whichever is less. R_m = tensile strength of the material used.

Before significant reductions in rudder stock diameter due to the application of steels with R_{eH} exceeding 235 N/mm² are granted, the Society may require the evaluation of the elastic rudder stock deflections. Large deflections should be avoided in order to avoid excessive edge pressures in way of bearings.

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

Definitions

C_R = rudder force in [N]

Q_R = rudder torque in [Nm]

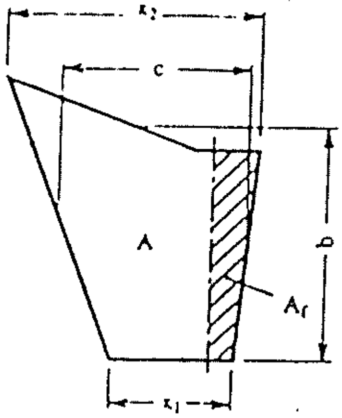
A = total movable area of the rudder in [m²]
For nozzle rudders, A is not to be taken less than 1,35 times the projected area of the nozzle;

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

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

b = mean height of rudder area in [m]

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



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

Fig. 14.1

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

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

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

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

B. Rudder Force and Torque

1. Rudder force and torque for normal rudders

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

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

v = v_0 for ahead condition

v = v_a for astern condition

κ_1 = coefficient, depending on the aspect ratio Λ

κ_1 = $(\Lambda + 2)/3$, 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

$c_1 = A_1/b_1$

= A_2/b_2

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

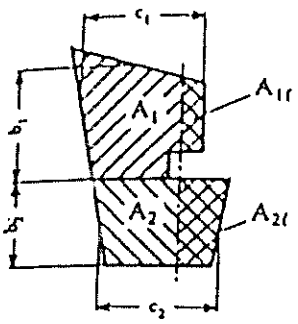


Fig. 14.2

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

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

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

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

for ahead condition

The greater value is to be taken.

Scantlings of the Rudder Stock

Rudder stock diameter

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

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

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

The related torsional stress is:

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

k_r see A.4.2.

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

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

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

2. Strengthening of rudder stock

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

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

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

Bending stress:

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

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

Torsional stress:

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

D_1 = increased rudder stock diameter in [cm]

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

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

Q_R see B.1.2 and B.2.2 - 2.3

D_t see 1.1.

Guidance

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

Section 18

Equipment

A. General

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

Guidance

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

The equipment numeral formula for anhoring equipment required under this Section is ased on an assumed current speed of 2,5 m/sec, ind speed of 25 m/sec and a scope of chain cable tween 6 and 10, the scope being the ratio between ngth of chain paid out and water depth.

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

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

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

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

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

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

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

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

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

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

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

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

B. Equipment numeral

The equipment numeral is to be calculated as follows:

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

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

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

h = $f_b + \sum h'$

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

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

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

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

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

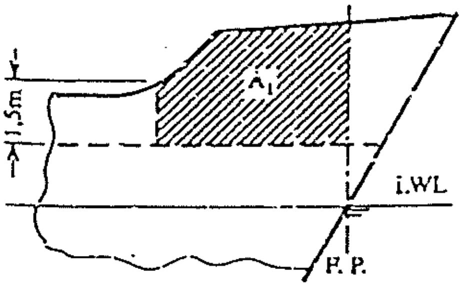


Fig. 18.1

C. Anchors

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

Guidance

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

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

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

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

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

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

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