BAB VI

KESIMPULAN

6. 1. Kesimpulan

Dengan selesainya perencanaan dan perhitungan tugas merancang ini, maka penulis dapat mengambil kesimpulan yang berhubungan dengan kapal perencanaan kapal Tanker 7. 000 **DWT** sebagai berikut :

- ❊ **Panjang kapal seluruhnya (Loa) = 105 m**
- ❊ **Panjang kapal antara garis tegak (Lpp) = 99 m**
- ❊ **Lebar kapal (B) = 18,8 m**
- ❊ **Tinggi kapal (H) = 9,5 m**
- ❊ **Sarat air kapal (T) = 6,0 m**
- ❊ **Koeffisien blok kapal (Cb) = 0,692**
- ❊ **Koeffisien prismatik kapal (Cp) = 0,704**
- ❊ **Koeffisien garis air kapal (Cw) = 0,784**
- ❊ **Koeffisien tengah kapal (Cm) = 0,983**
- ❊ **Displacement kapal (Δ) = 7.920,895 ton**
- ❊ **Volume kapal (∇) = 7.576,179 m^3**
- ❊ **Jumlah anak buah kapal (ABK) = 26 orang**
- ❊ **Mesin induk yang digunakan :**

Jumlah mesin	: 1 (satu) buah
Merk mesin	: MAN B&W
Type	: S26MC
Daya	: 3.270 HP/ 2.400 kW
Putaran mesin	: 250 rpm
Bore x Stroke	: 260 mm x 980 mm
Jumlah silinder	: 6
Cycle	: 2 langkah
SFOC	: 132 g/ BHP. jam

SLOC : 1,5 kg/ cyl. 24 h

Kecepatan dinas : 13 knot.

Berdasarkan perhitungan dan hasil pengamatan untuk kapal yang hampir sama kesimpulan sebagai berikut :

1. Kapal memerlukan tenaga penggerak minimum 3.270 HP. Pada perencanaan ini dipilih motor induk dengan daya sebesar 3.270 HP pada putaran 250 rpm.
2. Dengan jumlah ABK 26 orang dan route pelayaran yang ditempuh lebih kurang 11.000 mil, kapasitas maksimum kebutuhan listrik untuk mensuplai peralatan yang ada sebesar 361,962 kW. Dalam perencanaan ini digunakan 3 unit generator yang masing-masing sebesar 200 kW, dimana 2 unit generator berfungsi sebagai generator utama dan satu unit dipakai sebagai generator cadangan ataupun standby generator.
3. Berdasarkan hasil perhitungan rancangan propeller, maka spesifikasi propeller untuk penggerak kapal Tanker 7. 000 DWT ini adalah :
   1. Type propeller : B.4-40
   2. Diameter propeller : 3,247 m
   3. Pitch ratio propeller (Ho/ D) : 0,568
   4. Jumlah daun propeller (Z) : 4 buah
   5. Blade area ratio (Fa/ F) : 0,40.
   6. Effisiensi propeller (ηp) : 52 %.

TABEL ANALISA BEBAN GENERATOR

No	NAMA ALAT	DAYA			BERLABUH				BONGKAR MUAT				BERLAYAR				MANUVER			
					SIANG		MALAM		SIANG		MALAM		SIANG		MALAM		SIANG		MALAM	
		Watt	jml	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW
1	Steering Engine	28,467	2	56,934	-	-	-	-	-	-	-	-	0.8	45,547	0.8	45,547	1	56,934	1	56,934
2	Windlass	44,000	1	44	0.8	35.2	0.8	35.2	-	-	-	-	-	-	-	-	1	44	1	44
3	Capstan	8.581	1	8,581	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
4	Cooling S.W.Pump	5,400	2	10.8	-	-	-	-	0.8	8.64	0.8	8.64	0.8	8.64	0.8	8.64	1	10.8	1	10.8
5	S.W.Serv.Pp(for refre)	5,400	1	5,4	-	-	-	-	0.8	4,32	0.8	4,32	0.8	4,32	0.8	4,32	1	5,4	1	5,4
6	Fire & GS.Pump	0,340	1	9.34	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
7	Cool.S.W.Pp(for D/G)	1,000	2	2	-	-	-	-	0,8	1.6	0,8	1.6	0,8	1.6	0,8	1.6	1	2	1	2
8	D.O Purifier	1,500	1	1,5	-	-	-	-	0.4	0,6	-	-	0,2	0,3	0,2	0,3	1	1,5	1	1,5
9	Main Compressor	9,000	2	18	-	-	-	-	0,8	14,4	0,8	14,4	0,8	14,4	0,8	14,4	1	18	1	18
10	Emerg. Main air comp.	1.800	1	1,8	-	-	-	-	0.4	0,72	0.4	0,72	0,8	1,44	0,8	1,44	1	1,8	1	1,8
11	Electric welder	8.000	1	8,0	-	-	-	-	0,8	6,4	0,8	6,4	0,8	6,4	0,8	6,4	-	-	-	-
12	Incinerator	6,000	1	6	-	-	-	-	0,8	4,8	0,8	4,8	0,8	4,8	0,8	4,8	-	-	-	-
13	Grinder	1.150	1	1.15	-	-	-	-	0,4	0,46	0,8	0,92	0,8	0,92	0,8	0,92	-	-	-	-
14	Main L.O.Pump	7,100	2	14,2	-	-	-	-	0,6	8,52	0,6	8,52	0,8	11,36	0,8	11,36	1	14,2	1	14,2
15	L.O.Trans.Pump	1,150	1	1.15	-	-	-	-	0.8	0,92	0.8	0,92	0.8	0,92	0.8	0.92	1	1.15	1	1.15
16	D.O.Serv.Pump	0.102	2	0.204	-	-	-	-	0.8	0,163	0.8	0,163	0.8	0,163	0.8	0.163	1	0.204	1	0.204
17	L.O.Purifier	7,500	1	7,5	-	-	-	-	0.8	6	0.8	6	0.8	6	0.8	6	1	7,5	1	7,5
18	Cargo Oil Pump	28,500	2	57	-	-	-	-	0.8	45,6	0.8	45,6	0.8	45,6	0.8	45,6	1	57	1	57
19	Cargo Winches	13,600	2	27,2	-	-	-	-	0.8	21,76	0.8	21,76	-	-	-	-	-	-	-	-
20	Ballast Pump	1,460	1	1,46	0.4	0,584	0.4	0,584	0.6	0,876	0.6	0,876	0.8	1,168	0.8	1,168	1	1,46	1	1,46
21	DO Transfer Pump	1,280	1	2.56	-	-	-	-	0.8	1,024	0.8	1,024	0.8	1,024	0.8	1,024	1	2.56	1	1,28
22	Sanitary S.W pump	2,040	2	4.08	-	-	-	-	0.8	3,264	0.8	3,264	0.8	3,264	0.8	3,264	1	4.08	1	4.08
23	Oily Water Separator	2,500	1	2,5	-	-	-	-	0.8	2	0.8	2	0.8	2	0.8	2	1	4,3	1	4,3
24	Lathe Machine	3.800	1	3,8	-	-	-	-	0.9	3,42	0,9	3,42	0,9	3,42	0,9	3,42	1	3,8	1	3,8
25	Jacket Cool F.W.Pump	6,250	2	12.5	-	-	-	-	0,8	10	0,8	10	0,8	10	0,8	10	1	12.5	1	12.5
26	Stripping Pump	17,000	2	34	-	-	-	-	0.8	27,2	0.8	27,2	0.8	27,2	0.8	27,2	1	34	1	34
27	Tank Cleaning Pump	37,000	1	37	-	-	-	-	0.8	29,6	0.8	29,6	0.8	29,6	0.8	29,6	1	37	1	37
28	Hot W.Circ.Pump	102	1	0,102	-	-	-	-	0,8	0,082	0,8	0,082	0,8	0,082	0,8	0,082	1	1,3	1	1,3
29	Bilge Pump.	2,010	1	2,01	0.2	0,402	0.2	0,402	0.6	1,206	0.6	1,206	0,8	1,608	0,8	1,608	1	2,01	1	2,01
30	Drill	1.050	1	1,15	0,2	0,23	0,2	0,23	0,6	0,69	0,6	0,69	0,2	0,23	0,2	0,23	-	-	-	-
31	Cargo Pp RM Ex.Fan	1.675	1	1,6	0,2	0,32	0,2	0,32	0,4	0,64	0,4	0,64	0,2	0,32	0,2	0,32	-	-	-	-
32	Drink W.sterilizer	0,500	1	0,5	0,2	0,1	0,2	0,1	0,2	0,1	-	-	-	-	0,5	0.25	-	-	-	-
33	Boat winch	3.000	1	3	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
34	Emergency Fire Pump	11,000	1	11	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-

No	NAMA ALAT	DAYA			BERLABUH				BONGKAR MUAT				BERLAYAR				MANUVER			
					SIANG		MALAM		SIANG		MALAM		SIANG		MALAM		SIANG		MALAM	
		Watt	jml	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW
35	San & Dom.FW Pp	652	2	1.304	0.8	1,043	0.8	1,043	0.8	1,043	0.8	1,043	0.8	1,043	0.8	1,043	1	1,043	1	1,043
36	Sewage Pump	220	2	0.44	0.8	0,176	0.8	0,176	0.8	0,176	0.8	0,176	-	-	-	-	-	-	-	-
37	Exhaust Fan	450	3	1.35	0.4	0.54	0.4	0.54	0.8	1.08	0.8	1.08	0.8	1.08	0.8	1.08	1	1.35	1	1.35
38	Supply Fan	350	3	1.05	0.4	0.42	0.4	0.42	0.8	0.84	0.8	0.84	0.8	0.84	0.8	0.84	1	1.05	1	1.05
39	Cold Storage	11.500	1	11.5	0.6	6.9	0.6	6.9	0.6	6.9	0.6	6.9	0.6	6.9	0.6	6.9	0.6	6.9	0.6	6.9
40	Stern Light	80	1	0.08	-	-	-	-	-	-	0.8	0.064	-	-	0.8	0.064	0.8	0.064	0.8	0.064
41	N.U.C.L	120	1	0.12	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
42	Wheel House	120	2	0.24	-	-	0.8	0.192	-	-	0.4	0.096	-	-	0.8	0.192	0.8	0.192	0.8	0.192
43	Radio Room	40	2	0.08	-	-	0.4	0.032	-	-	0.8	0.064	-	-	0.6	0.048	0.6	0.048	0.6	0.048
44	Radio Officer	40	2	0.08	-	-	0.8	0.064	-	-	0.8	0.064	-	-	0.8	0.064	0.8	0.064	0.8	0.064
45	Morse Lamp	80	1	0.08	-	-	-	-	-	-	0.4	0.032	-	-	0.4	0.032	-	-	-	-
46	Chart Room	40	-	0.04	-	-	0.2	0.008	-	-	0.2	0.008	-	-	0.2	0.008	-	-	0.2	0.008
47	Pilot	40	-	0.04	0.4	0.016	0.6	0.024	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032
48	KM/ WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
49	Pilot	40	-	0.04	0.4	0.016	0.6	0.024	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032
50	KM/ WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
51	Owner Room	40	-	0.04	0.4	0.024	0.6	0.024	0.4	0.016	0.8	0.024	0.4	0.016	0.8	0.024	0.4	0.016	0.8	0.024
52	KM/ WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
53	Office	60	-	0.06	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048
54	KM/WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
55	Bed Room	40	-	0.04	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032
56	KM/WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
57	Office	60	-	0.06	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048
58	KM/WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
59	Hospital	60	-	0.06	-	-	0.6	0.036	-	-	0.6	0.036	-	-	0.6	0.036	-	-	0.6	0.036
60	KM/WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
61	Storage	40	-	0.04	-	-	0.4	0.016	-	-	0.4	0.016	-	-	0.4	0.016	-	-	0.4	0.016
62	Laundry & Dry Room	40	-	0.04	0.2	0.008	0.6	0.024	0.2	0.008	0.6	0.024	0.2	0.008	0.6	0.024	0.2	0.008	0.6	0.024
63	Corridor	60	-	0.06	0.6	0.036	0.6	0.036	0.6	0.036	0.6	0.036	0.6	0.036	0.6	0.036	0.6	0.036	0.6	0.036
64	Office	60	-	0.06	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048
65	Chief Officer	40	-	0.04	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032
66	Chief Engineer	40	-	0.04	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032
67	KM/WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
68	KM/WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008

No	NAMA ALAT	DAYA			BERLABUH				BONGKAR MUAT				BERLAYAR				MANUVER			
					SIANG		MALAM		SIANG		MALAM		SIANG		MALAM		SIANG		MALAM	
		Watt	jml	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW
69	Corridor	60	-	0.06	0.6	0.036	0.6	0.036	0.6	0.036	0.6	0.036	0.6	0.036	0.6	0.036	0.6	0.036	0.6	0.036
70	I.Officer	40	-	0.04	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032
71	II.Officer	40	-	0.04	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032
72	I.Engineer	40	-	0.04	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032
73	II.Engineer	40	-	0.04	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032
74	KM/WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
75	KM/WC	20		0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
76	Pantry	60	-	0.06	0.2	0.012	0.6	0.036	0.2	0.012	0.6	0.036	0.2	0.012	0.6	0.036	0.2	0.012	0.6	0.036
77	Galley	40	-	0.04	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032	0.2	0.008	0.8	0.032
78	Crew Mess	80	-	0.08	0.4	0.032	0.8	0.064	0.4	0.032	0.8	0.064	0.4	0.032	0.8	0.064	0.4	0.032	0.8	0.064
79	Kamar-kamar	480	-	0.48	0.4	0.192	0.6	0.288	0.4	0.192	0.6	0.288	0.4	0.192	0.6	0.288	0.4	0.192	0.6	0.288
80	Stew Officer	40	-	0.04	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032	0.4	0.016	0.8	0.032
81	Deck Store	40	-	0.04	-	-	0.4	0.016	-	-	0.4	0.016	-	-	0.4	0.016	-	-	0.4	0.016
82	Officer Mess	80	-	0.08	0.4	0.032	0.8	0.064	0.4	0.032	0.8	0.064	0.4	0.032	0.8	0.064	0.4	0.032	0.8	0.064
83	Corridor	40	-	0.04	0.6	0.024	0.6	0.024	0.6	0.024	0.6	0.024	0.6	0.024	0.6	0.024	0.6	0.024	0.6	0.024
84	Pantry	60	-	0.06	0.2	0.012	0.6	0.036	0.2	0.012	0.6	0.036	0.2	0.012	0.6	0.036	0.2	0.012	0.6	0.036
85	Galley	60	-	0.06	0.2	0.012	0.8	0.048	0.2	0.012	0.8	0.048	0.2	0.012	0.8	0.048	0.2	0.012	0.8	0.048
86	Suitcases Room	40	-	0.04	0.2	0.008	0.4	0.016	0.2	0.008	0.4	0.016	0.2	0.008	0.4	0.016	0.2	0.008	0.4	0.016
87	Electric Room	40	-	0.04	0.4	0.016	0.6	0.024	0.4	0.016	0.6	0.024	0.4	0.016	0.6	0.024	0.4	0.016	0.6	0.024
88	Bay Room	40	-	0.04	0.2	0.008	0.6	0.024	0.2	0.008	0.6	0.024	0.2	0.008	0.6	0.024	0.2	0.008	0.6	0.024
89	Officer	60	-	0.06	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048
90	KM/WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
91	KM/WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
92	Lampu2 di km mesin	500	-	0.5	0.8	0.4	0.8	0.4	0.8	0.4	0.8	0.4	0.8	0.4	0.8	0.4	0.8	0.4	0.8	0.4
93	Officer	60	-	0.06	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048	0.4	0.024	0.8	0.048
94	KM/WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
95	KM/WC	20	-	0.02	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008	0.2	0.004	0.4	0.008
96	Lampu2 di km mesin	500	-	0.5	0.8	0.4	0.8	0.4	0.8	0.4	0.8	0.4	0.8	0.4	0.8	0.4	0.8	0.4	0.8	0.4
97	Red Light	200	1	0.2	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
98	Boat Deck Light	200	2	0.4	-	-	0.2	0.08	-	-	0.6	0.24	-	-	0.6	0.24	-	-	0.6	0.24
99	Day Light Sign. Lamp	200	1	0.2	-	-	0.2	0.04	-	-	0.6	0.12	-	-	0.6	0.12	-	-	0.6	0.12
100	Cargo Light	400	4	1.6	-	-	0.8	1.28	-	-	-	-	-	-	0.4	0.64	-	-	0.4	0.64
101	Flood Light	200	1	0.2	-	-	0.8	0.16	-	-	-	-	-	-	0.4	0.08	-	-	0.4	0.08
102	Berth Light	400	4	1.6	-	-	0.8	1.28	-	-	-	-	-	-	0.8	1.28	-	-	0.8	1.28
103	Search Light	1000	1	1	-	-	-	-	-	-	0.6	0.6	-	-	0.8	0.8	-	-	0.8	0.8
104	Lampu Peta	60	1	0.06	-	-	-	-	-	-	0.8	0.48	-	-	0.6	0.36	-	-	0.6	0.36

	NAMA ALAT	DAYA			BERLABUH				BONGKAR MUAT				BERLAYAR				MANUVER			
					SIANG		MALAM		SIANG		MALAM		SIANG		MALAM		SIANG		MALAM	
		Watt	jml	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW	pf	kW
105	Compass	1000	1	1	-	-	-	-	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8
106	Flood Light	200	1	0.2	-	-	0.8	0.16	-	-	-	-	-	-	0.4	0.08	-	-	0.4	0.08
107	Berth Light	400	4	1.6	-	-	0.8	1.28	-	-	-	-	-	-	0.8	1.28	-	-	0.8	1.28
108	Search Light	1000	1	1	-	-	-	-	-	-	0.6	0.6	-	-	0.8	0.8	-	-	0.8	0.8
109	Lampu Peta	60	1	0.06	-	-	-	-	-	-	0.8	0.48	-	-	0.6	0.36	-	-	0.6	0.36
110	Compass	1000	1	1	-	-	-	-	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8
111	Radio	1000	1	1	0.6	0.6	0.6	0.6	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8
112	Radar	3000	1	4	0.4	1.6	0.4	1.6	0.8	3.2	0.8	3.2	0.8	3.2	0.8	3.2	0.8	3.2	0.8	3.2
113	Telephone System	1000	1	1	0.4	0.4	0.4	0.4	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8
114	Battery Charger	10000	1	10	0.4	4	0.4	4	0.4	4	0.4	4	0.4	4	0.4	4	0.4	4	0.4	4
115	Air Conditioner	12000	1	12	0.6	7.2	0.6	7.2	0.8	9.6	0.8	9.6	0.8	9.6	0.8	9.6	0.8	9.6	0.8	9.6
116	Peralatan Bengkel	3000	-	3	0.4	1.2	0.4	1.2	0.4	1.2	0.4	1.2	-	-	-	-	-	-	-	-
117	Peralatan Dapur	1000	-	1	0.6	0.6	0.6	0.6	0.8	0.8	0.8	0.8	0.4	0.4	0.4	0.4	0.4	0.4	0.4	0.4
	Jumlah					63,061		68,399		238,57		241,97		264,12		127,61		355,12		361,962

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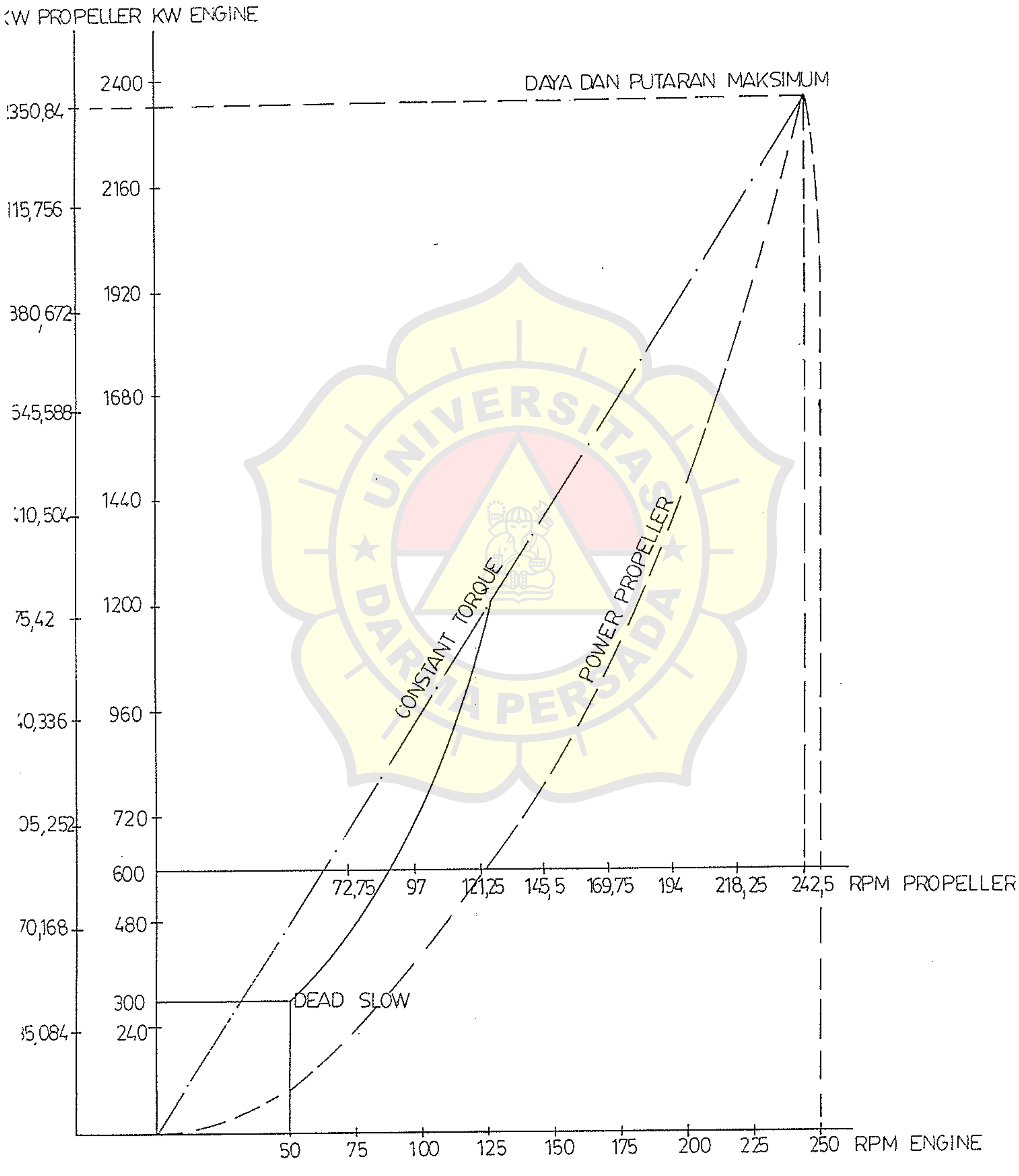


KW PROPELLER KW ENGINE
DAYA DAN PUTARAN MAKSIMUM
2400
2350,84
2160
2115,756
1920
1880,672
1680
1645,588
1440
1410,504
1200
1175,42
960
940,336
720
705,252
600
480
470,168
300
240
235,084
CONSTANT TORQUE
POWER PROPELLER
DEAD SLOW
72,75
97
121,25
145,5
169,75
194
218,25
242,5
RPM PROPELLER
50
75
100
125
150
175
200
225
250
RPM ENGINE

REFERENSI NO.1

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%

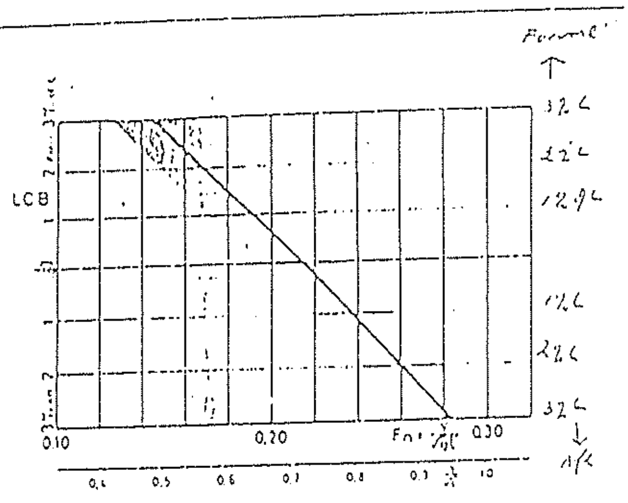
Gbr No.6 Koreksi Bentuk Anggota Badan

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

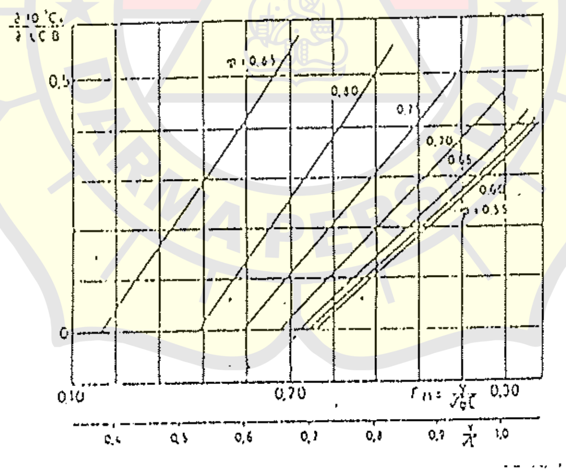
Gbr No. 6 Koreksi Tahanan Tambahan

= 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



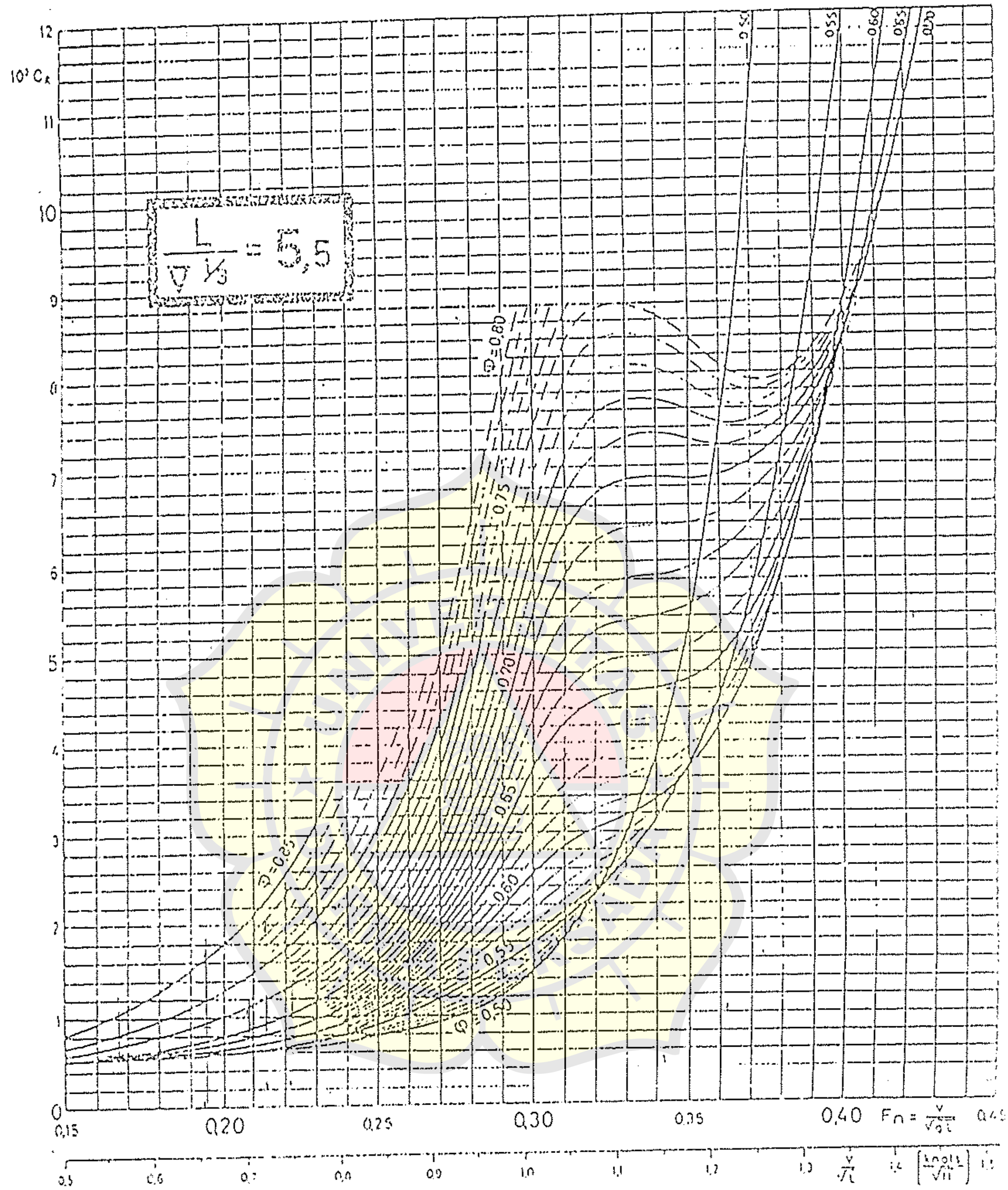
Gbr.No.3 LCB Standar



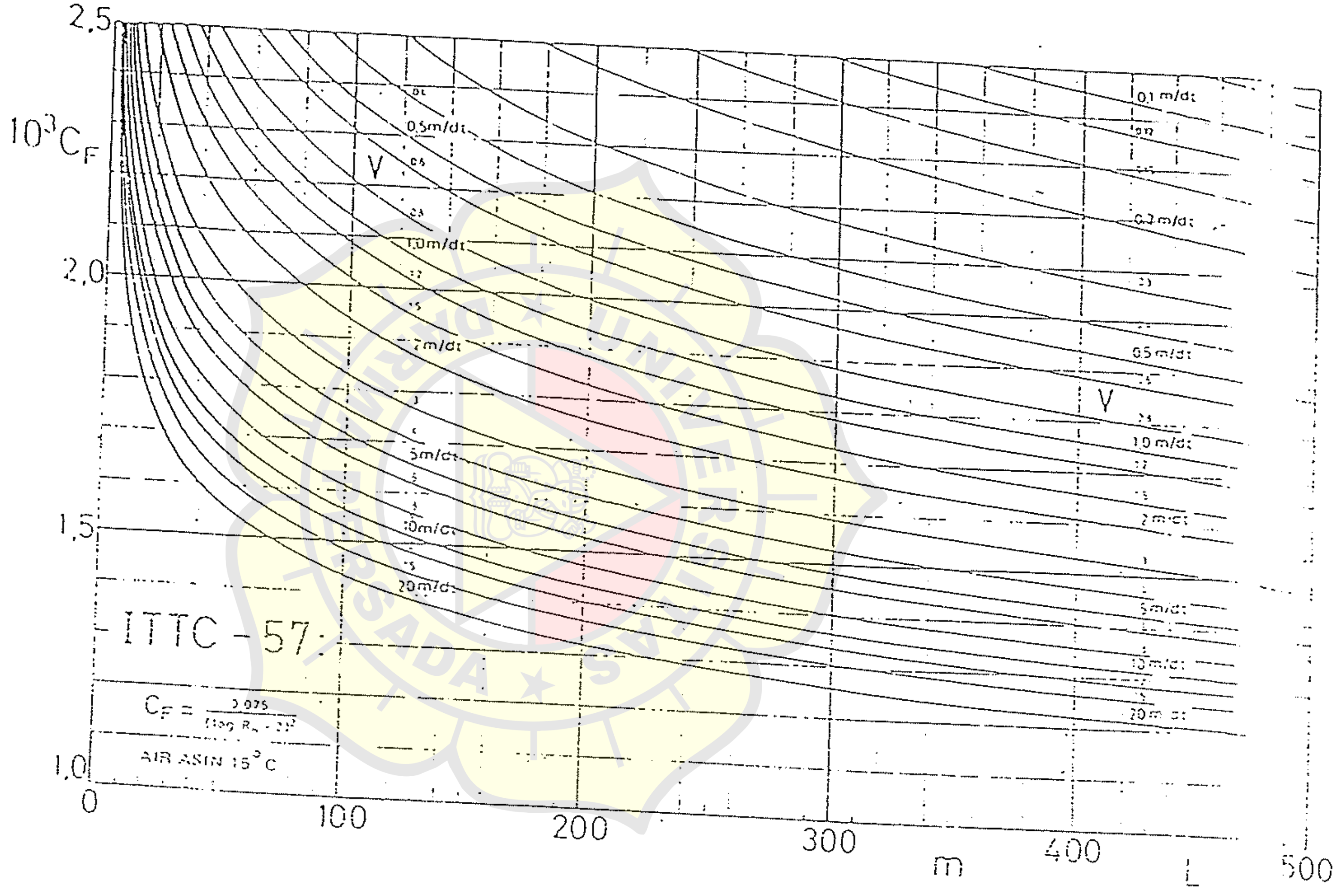
Gbr.No.3 Koreksi LCB

UNIVERSITAS DARMA PERSADA

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



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



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

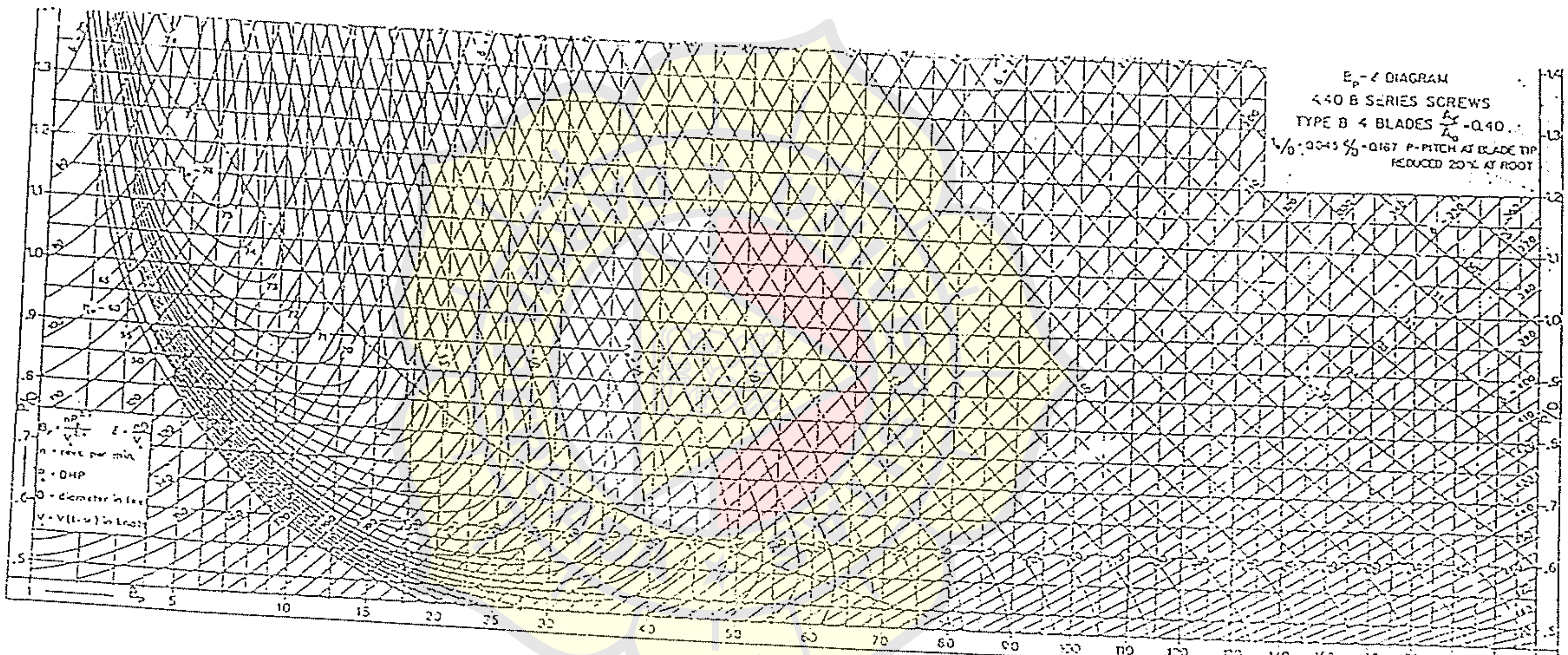
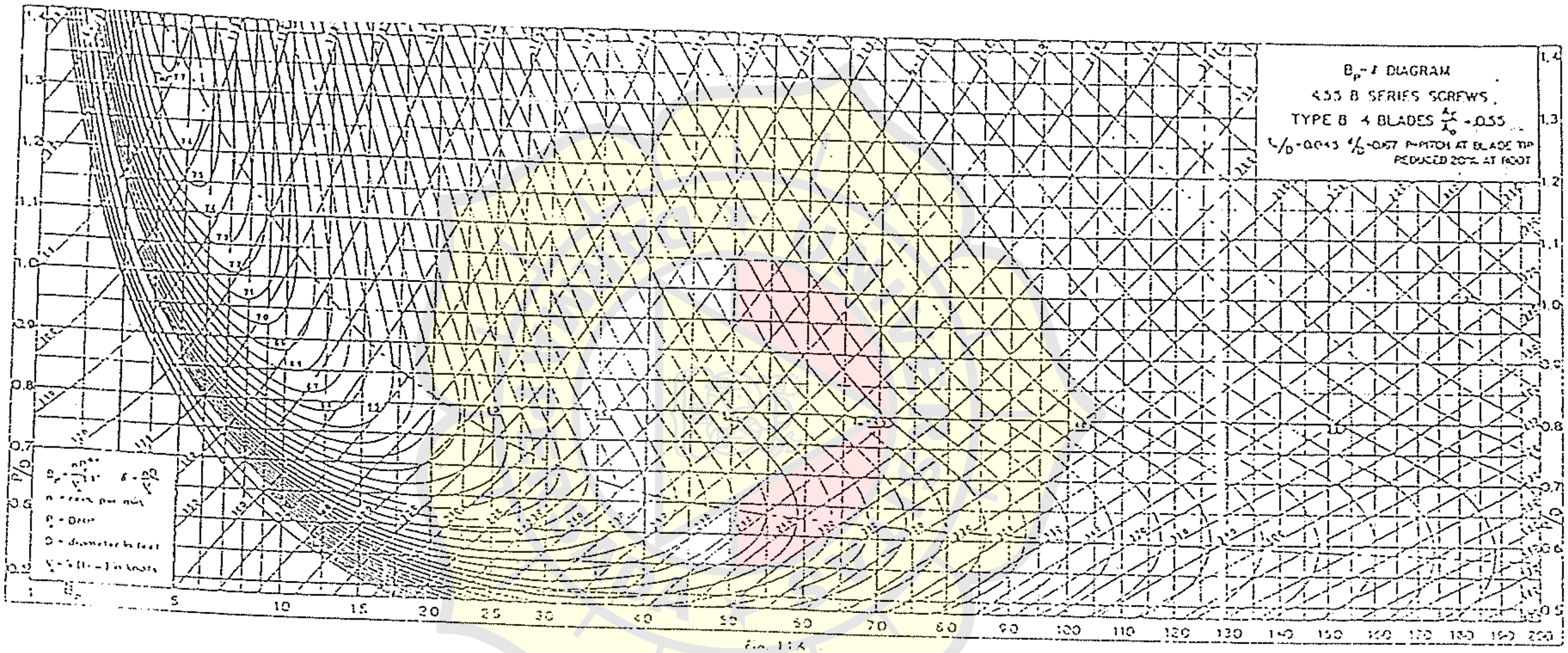


Fig. 115

Gbr. No. 7 Diagram Bp – δ B Series Type B4-40



Gbr. No. 4 Diagram Bp - δ B Series Tipe B - 55

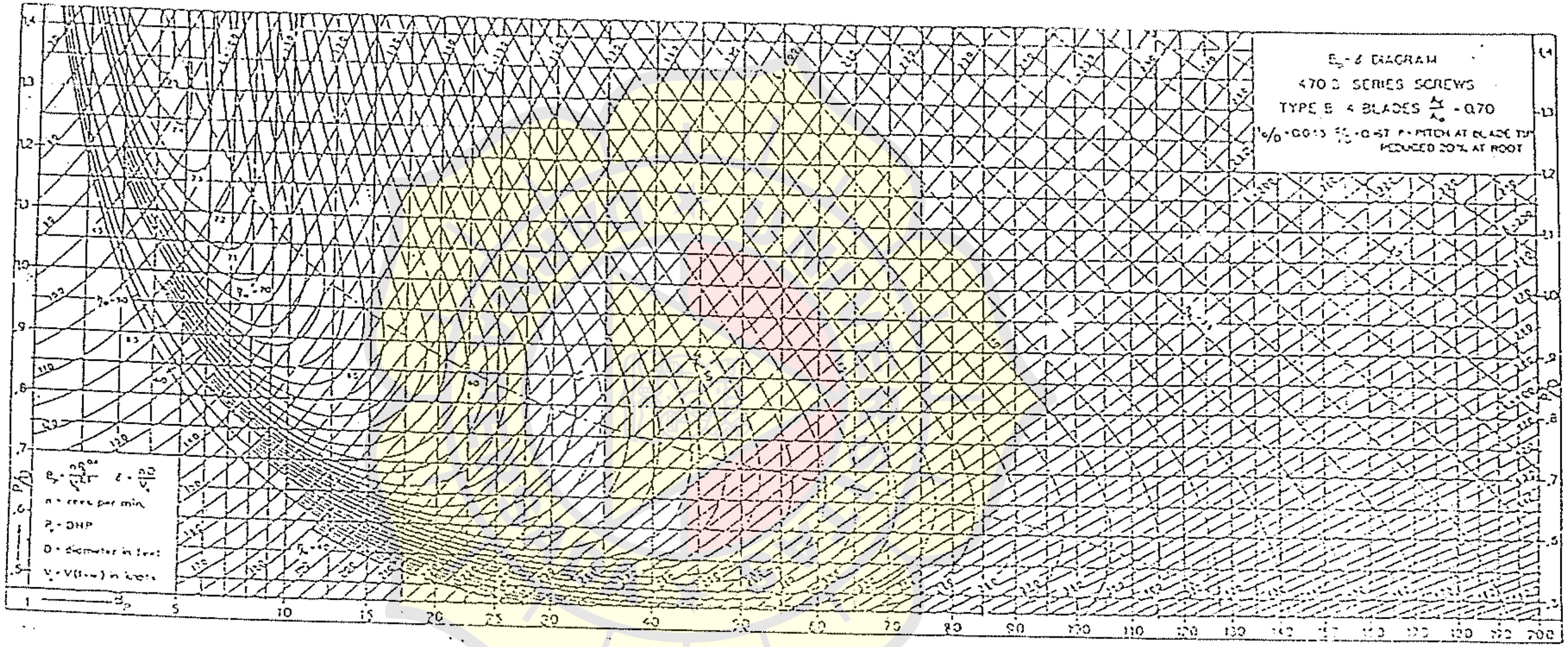
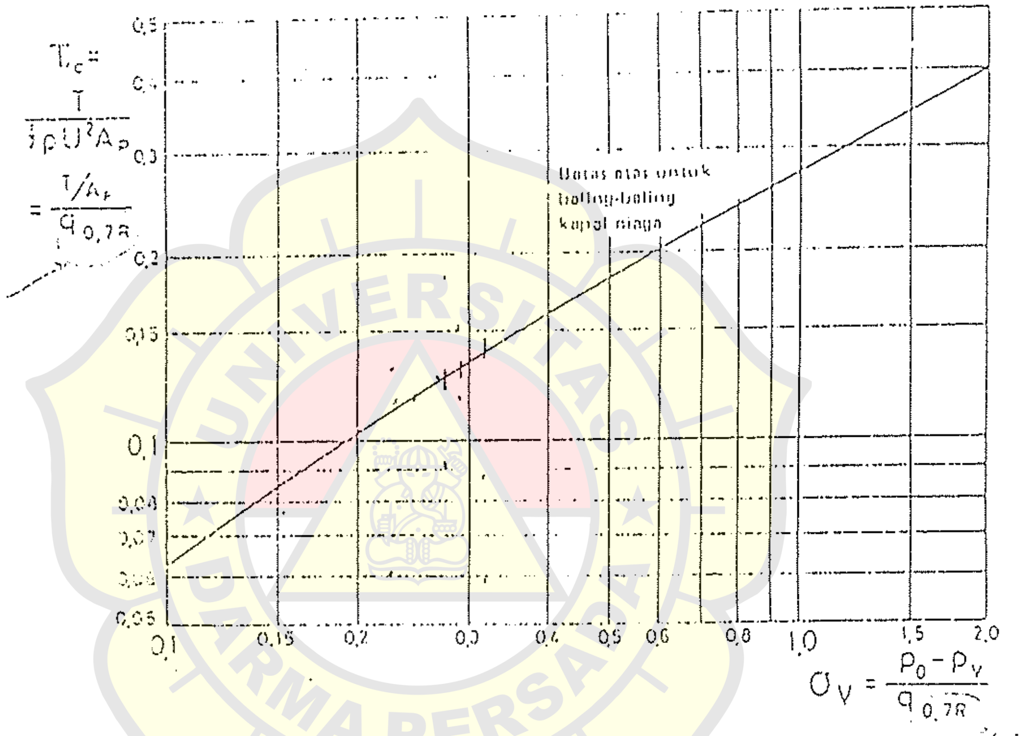


Fig. 117

Gbr. No. 7 Diagram Bp - δ B Series Tipe B - 70



Gbr.No. 2 Diagram Burril

[illegible]FICATION [illegible]

1. GENERAL SPECIFICATION

-1. RULE :ABS, MAPROL 73/78(EXCEPT MEPC33), JIS

-2. SOURCE

POWER CIRCUIT :AC440V, 60Hz, 3Φ

CONTROL CIRCUIT :AC220V, 60Hz, 1Φ

-3. PAINTING COLOR

INCINERATOR BODY :SILVER

CONTROL PANEL :MUNSELL NO. 2.5G7/2

-4. NAME PLATE :ENGLISH

2. INCINERATOR SPECIFICATION

- 1. TYPE :YI-30

- 2. QUANTITY :1SET

- 3. MAX. DISPOSAL CALORIFIC VALUE:32×10^4 kcal/h

- 4. DISPOSAL QUANTITY

EX. WATER CONTENT 40% THE OTHER ARE COMBUSTIBLE LIQUID

:ab. 50 L/h

- 5. SOLID WASTE :ab. 25 kg/h

- 6. COMBUSTION METHOD :WASTE OIL ATOMIZED BURNING

- 7. DRAFT SYSTEM :FORCED DRAFT SYSTEM

- 8. IGNITION SYSTEM :DIESEL OIL PRESSURE JET BURNER WITH HIGH TENSION DIRECT IGNITION

(IN CASE OF ASSISTED BURNING, PILOT BURNER IS TO BE USED CONTINUOUS.)

- 9. OPERATION SYSTEM :SEMI AUTOMATIC OPERATION SYSTEM

10. SOLID COMBUSTIBLES :SOLID COMBUSTIBLES CAN ALSO BE BURNED AND DISPOSED OFF.

-11. COMBUSTION AIR TEMPERATURE :NOR. 20°C , MAX. 45°C

-12. AIR AND STEAM SOURCE

AIR FOR ATOMIZING :4 ~ 10 kg/cm

-13. EXHAUST GAS COOLING SYSTEM :DILUTING AIR COOLING SYSTEM

SPECIFICATION OF SASAKURA OILY WATER SEPARATOR

Hull NO. : 045 JMI
Rule :
Mfd No. : 055 - 5983

Approved by :
Prepared by :
Date : APR. 1997

1. Particulars

Model	SMT-5A		Name plate	English
No.of set per ship	1 set			
Capacity	5 m3/h (Max.)		Unit	Metric, SI
Working press	Max.0.29MPa(3.0kgf/cm2)			
Power source	Control	AC220V 60Hz 1φ	Pointing color for separator	Munsell No.2.5G 7/2
	Pump	AC440V 60Hz 3φ		
Performance 性能	This equipment as a 15ppm Oily Water Separator has been certified by D.O.T.(U.K.) and Japanese Government in accordance with the Requirement of the Specification contained in the Annex of the Recommendation in IMO resolution MEPC.60(33).			
	IMO 決議 MEPC.60(33)に定める試験基準により、15ppm 用油水分離装置として日本政府及び英国D.O.T.に型式承認された装置です。			
Pump	Piston, Screw, or Plunger			

2. Materials

Part	Materials
Shell	Carbon steel (inside of shell is coated with anti-corrosive point.)
Coalescer	Stainless steel

3. Accessories

Article	Q'ty	Article	Q'ty
Oil level sensor	1	Bilge alarm (Type : FOCAS-1500C)	1 set
Motor valve	1	Safety valve	1
Pressure gauge	3	Steam heater	1
Test cock	1	Bilge feed pump	1 set
Oil drain valve	3	Starter	1

4. Remarks

Performance tests of shop and onboard are not to be carried out because the performance has been authorized by D.O.T.(U.K.) and Japanese Government as stated above.
本油水分離器は、日本政府及び英国 D.O.T.による型式承認品であるため、工場及び船内立会試験は施工いたしません。

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	
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	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	—	
Blade-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*
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	44.3	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	
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	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	—	
Blade-thickness 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*
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	44.30	47.90	50.00	—	

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	14.17	Length of blade section at 0.6 *R* = 0.3111 *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	44.30	47.90	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	14.17	Length of blade section at 0.6 *R* = 0.3498 *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	44.3	47.9	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	
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	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	—	
Blade-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*
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	44.3	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	
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	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* = 0.1961 *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	—	
Blade-thickness 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*
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	44.30	47.90	50.00	—	

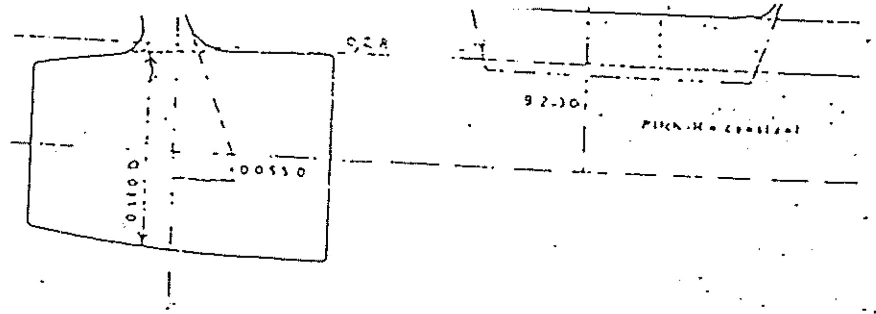


Fig. 33. *General plan of the two-bladed propeller of type B 2-30*

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	85.50	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	22.50	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, the curve of thicknesses being assumed to be rectilinear. The connecting lines of the points at which set-back and back intersect, cut each other at 0.15 *R*.

ting Compressor (Water-cooled)

Model No.	Speed (r.p.m.)	25kgf/cm²			30kgf/cm²		
		Capacity m³/hr (FA)	Power required PS	Motor (kW)	Capacity m³/hr (FA)	Power required PS	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

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

gency Compressor (Vertical 2-stage Air-cooled)

Model No.	Speed (r.p.m)	15kg/cm²		25 30kg/cm²	
		m³/hr FA	PS	m³/hr FA	PS
SHC-20B	900	4.7	1.4	4.3	1.6
	1000	5.2	1.5	5.6	1.7
SHC-30A	900	13.5	4.8	1[illegible].8	5.3
	1000	14.8	5.3	1[illegible].8	5.8
SHC-40A	900	20.4	7.2	19.4	8.0
	1000	22.3	7.9	2[illegible].2	8.9

Taiko Kikai Industries Co., Ltd.

Screw Pump

Single Rotor Screw Pump

Features:

1. Can handle both high and low viscosity liquids and semi-solid materials, thus providing a wide range of applications.
2. Has a strong vacuum capacity (8.5 mAq) so that it requires no priming.
3. Can handle slurry liquids.
4. Can achieve a fixed rate of transfer without pulsive fluctuation. It generates neither excess flow nor turbulence.
5. Can operate both forwards and backwards, so that the liquid flow direction can be changed easily.
6. Has a simple structure and a small number of parts, making its handling and maintenance easy.
7. Can achieve a large volume of outlet, about 1.5 to 1.6 times as large as the capacity achieved by competitor pumps of the same rotor diameter.

Application: Fuel oil trans. pump, bilge & sludge pump, waste oil pump, sewage pump, fish pump

(1) HNP-/HNP-CS, HNF-/HNF-B Series
(Horizontal external bearing type)
Capacity : 0.3–80 m^3/hr
Head : Max. 24 kgf/cm^2
H. Power : 0.4–55 kW

(2) HNV-Series
(Vertical submerged type)
Capacity : 0.3–80 m^3/hr
Head : Max. 24 kgf/cm^2
H. Power : 0.4–55 kW

Two Rotor Screw Pump

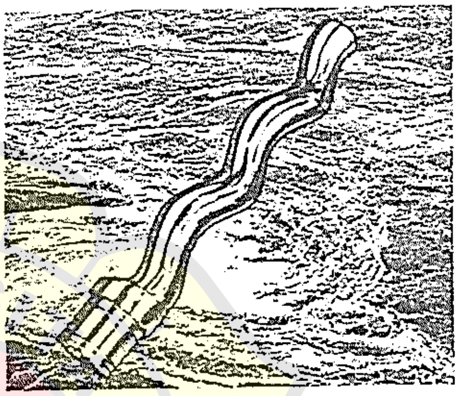
Two sets of screws, one left-handed and one right-handed are housed in the pump casing. The liquid entering from the intake branches into two directions is transferred to the center outlet from both sides of the casing. Therefore, complete shaft balance is maintained and there is no thrust load applied to the bearing. Since the intake is also the shaft accommodation hole, no leakage of liquid occurs. Thus, this pump is suitable for the handling of dangerous liquids.

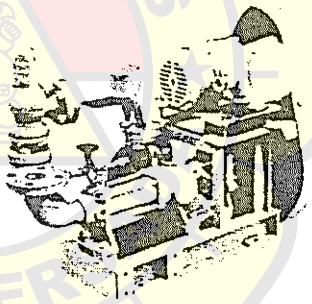
Application: F.O,Trans pump, L.O. pump, Cargo pump, stripping pump

(1) FSE-/FSH-Series
(Horizontal, external bearing type with timing gear)
Capacity : 3–1500 m^3/hr
Head : Max. 16 kgf/cm^2
H.Power : 1.5–1000 kW

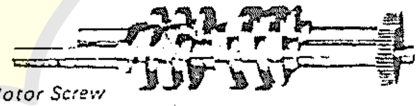
(2) FSU-/FSF-/FSG-Series
(Horizontal, internal bearing type with timing gear)
Capacity : 60–600 M^3/hr
Head : max. 10 kgf/cm^2
H. Power : 18.5–250 kW

(3) FST-/FSV-Series
(Vertical, external bearing type with timing gear)
Capacity : 30–1500 m^3/hr
Head : Max. 16 kgf/cm^2
H. Power : 11–1000 kW

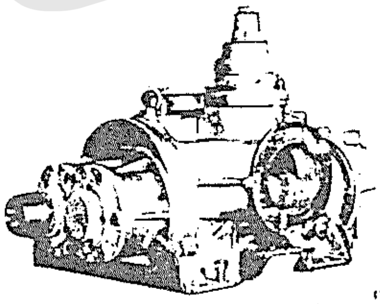




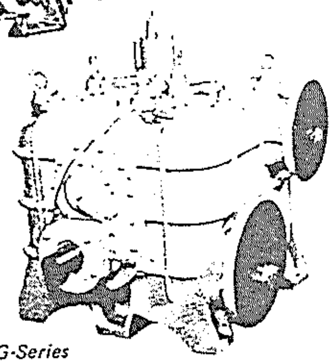
HNP-Series



Rotor Screw



FSE/FSH-Series



FSU-/FSF-/FSG-Series

Taiko Kikai Industries Co., Ltd.

Gear Pump

Merits of TAIKO-Type Gear Pump
(Segmental Gear)

1. The gears are one point contact type originally designed for gear pump application, therefore, neither cavitation nor noise will occur.
2. Smooth rotation and slip ratio of only 1/10 against the ordinary gear pump result in less wearing of gear teeth.
3. Compact size with large capacity.
4. Longer life of bearings.
5. Constant suction and delivery volume.
6. Suitable even for fluid of high viscosity.
7. Changing the delivery pressure and capacity is almost nil.
8. High efficiency and less power consumption.

nternal Bearing Type

\pplication: L.O. Pump, L.O. Trans. Pump, F.O. Trans. Pump, F.O. Booster Pump

1) NHG-Series
(Horizontal Low press. small capacity type)
Capacity : 0.1–30 m^3/hr
Head : Max. 6 kgf/cm^2
H. Power : 0.2–15 kW

2) NHGH-Series
(Horizontal High press. small capacity type)
Capacity : 0.5–20 m^3/hr
Head : Max. 16 kgf/cm^2
H. Power : 0.4–18.5 kW

3) HG-Series
(Horizontal Low press, Medium capacity type)
Capacity : 30–200 m^3/hr
Head : Max. 7 kgf/cm^2
H. Power : 5.5–75 kW

4) VG-Series
(Vertical Low press, Medium capacity type)
Capacity: : 20–200 m^3/hr
Head : Max. 7 kgf/cm^2
H. Power : 5.5–75 kW

xternal Bearing Type

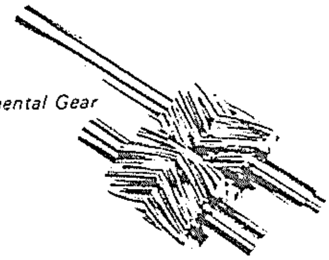
pplication: Low grade F.O. pump, bilge pump, cargo pump, stripping pump, waste oil pump

1) WL-, WA-, SL-, Series
(Horizontal, Low press, small capacity type)
Capacity : 1–100 m^3/hr
Head : Max. 16 kgf/cm^2
H. Power : 0.75–55 kW

2) CGL/CWL Series
(Horizontal, medium press, large capacity type)
Capacity : 60–1500 m^3/hr
Head : Max. 12 kgf/cm^2
H. Power : 15–500 kW

Other Used Pump

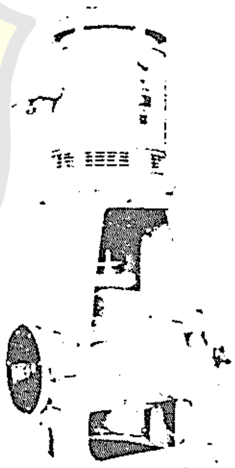
Engine Fixed Pump
Application: Main L.O. Pump, F.O. supply pump
OL/F Series (Engine driven bracket type)
Capacity : 0.3–300 m^3/hr
Head : - 8 kgf/cm^2



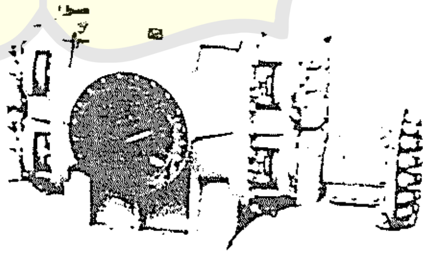
Segmental Gear

NHG, NHGH, HG-Series

WL-, WA-, Series



VG-Series



CGL, CWL-Series

OL/F Series

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REFERENSI NO.2

Appendix to Section 2

Part C :

Approximate Calculation of the Starting Air Supply

1. Starting air for installations with reversible engines

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

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

where

J	[dm³]	total capacity of the starting air receivers
D	[mm]	cylinder bore
H	[mm]	stroke
V_h	[dm³]	swept volume of one cylinder (in the case of double-acting engines, the swept volume of the upper portion of the cylinder)
$P_{c,perm}$	[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 $P_{c,perm}$ = 30 bar

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

where $p_{c,perm} \neq 30$ 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 $p_{c,perm}$ 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 $n_o \le 1000$

$n_A = 0{,}25 \cdot n_o - 176$ 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.

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

1.6 Pipe connections

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

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

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

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

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

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

2. Calculation of pipe diameters

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

2.2 Dry cargo and passenger ships

a) main bilge pipes

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

b) branch bilge pipes

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

where

d_H [mm] calculated inside diameter of main bilge pipe

d_z [mm] calculated inside diameter of branch bilge pipe

L [m] length of ship between perpendiculars

B [m] moulded breadth of ship

H [m] depth of ship to the bulkhead deck

l [m] length of the watertight compartment

2.3 Tankers

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

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

where:

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

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

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

2.4 Minimum diameter

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

2.5 Maximum diameter

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

2.6 Deviations

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

3. Bilge pumps

3.1 Capacity of bilge pumps

Each bilge pump must be capable of delivering:

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

where:

Q [m³/h] minimum capacity

d_H [mm] calculated inside diameter of main bilge pipe

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

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

3.4 Use of other pumps for bilge pumping

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

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

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

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

3.5 Number of bilge pumps for cargo ships

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

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

3.6 Number of bilge pumps for passenger ships

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

4. Bilge pumping for various spaces

4.1 Machinery spaces

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

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

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

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

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

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

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

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

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

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

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

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

4.2 Shaft tunnel

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

[1]) See SOLAS 1974, Chapter II-1, part-A, regulations 5 and 13

6.3.2 Bilge lines

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

6.3.3 Ballast pipes

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

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

6.3.4 Fuel pipes

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

6.3.5 Cargo pipes

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

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

6.4 Control stands

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

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

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

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

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

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

6.5 Power units

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

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

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

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

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

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

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

7. Pumps

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

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

Bilge pumps/bilge ejectors

Ballast pumps

Sea cooling water pumps

Fresh cooling water pumps

Fire extinguishing pumps

Emergency fire extinguishing pumps including drive units

Condensate pumps

Boiler feedwater pumps

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

4.10 Cofferdams, pipe tunnels and void spaces

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

4.11 Chain lockers

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

5. Additional Rules for passenger vessels

5.1.1 The arrangement of bilge pipes

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

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

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

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

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

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

5.2 Bilge suctions

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

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

5.3 Arrangement of bilge pumps

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

Ships with a length of 91.5 m or over or having a criterion numeral of 30[1]) or more are to have at least one bilge pump available in emergency cases. This requirement is satisfied if

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

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

c) bilge pumps are installed above the bulkhead deck.

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

5.4 Passenger vessels for limited range of service

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

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

6. Additional Rules for tankers

See Section 15, B.4.

- Bilge testing

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

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

1 Oily water separating equipment

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

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

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

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

2 Discharge of fuel and oil residues

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

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

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

[1] With regard to the installation on ships of oily water separators, filter plants, oil collecting tanks, oil discharge lines and a monitoring and control system or a 15 ppm alarm device in the water outlet of oily water separators, compliance is required with the provisions of the International Convention for the Prevention of Pollution from Ships, 1973, (MARPOL) and the Protocol of 1978.
Form F.336 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

REFERENSI NO.3

Section 14

Rudder and Manoeuvring Arrangement

A. General

1. Manoeuvring arrangement

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

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

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

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

Guidance

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

1.5 For ice-strengthening see Section 15.

2. Structural details

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

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

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

3. Size of rudder area

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

$$A = c_1 \cdot c_2 \cdot c_3 \cdot c_4 \cdot \frac{1.75 \cdot L \cdot T}{100} \quad [m^2]$$

c_1 = factor for the ship type

= 1,0 in general

= 0.9 for bulk carriers and tankers having displacement of more than 50.000 tdw

= 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]

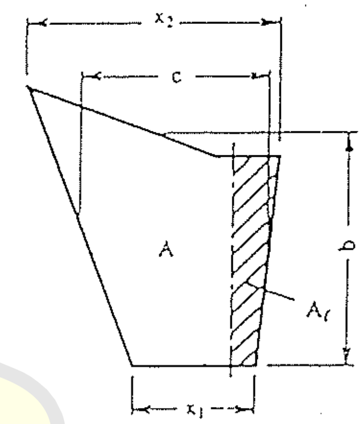
A = total movable area of the rudder in [m²]
For nozzle rudders, A is not to be taken less than 1,35 times the projected area of the nozzle;

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

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

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

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

B. Rudder Force and Torque

1. Rudder force and torque for normal rudders

1.1 The rudder force is to be determined 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 Gottinger profiles	1,1	1,4
flat side profiles	1,1	1,4
hollow profiles	1,35	1,4
high lift rudders	1,7	to be specially considered; if not known: 1,7

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

$Q_R = C_R \cdot r$ [Nm]

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

α = 0,33 for ahead condition

α = 0,66 for astern condition (general)

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

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

α = 0,25 for ahead condition

α = 0,55 for astern condition.

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

k_b = balance factor as follows:

$k_b = A_f/A$

k_b = 0,08 for unbalanced rudders

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

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

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

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

The resulting force of each part may be taken as:

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

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

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

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

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

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

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

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

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

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

$c_1 = A_1/b_1$

b_2 = A_2/b_2

h_1, h_2 = mean heights of the partial rudder areas A_1 and A_2 (see Fig. 14.2).

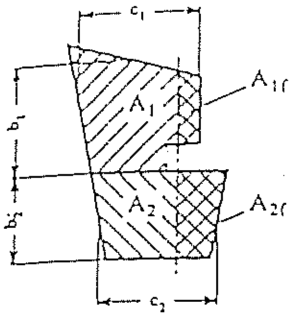


Fig. 14.2

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

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

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

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

for ahead condition

The greater value is to be taken.

Scantlings of the Rudder Stock

Rudder stock diameter

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

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

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

The related torsional stress is:

$\tau_t = \frac{68}{k_r}$ [N/mm²]

k_r see A.4.2.

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:

$\sigma_v = \sqrt{\sigma_b^2 + 3\tau^2} \le 118/k_r$ [N/mm²]

Bending stress:

$\sigma_b = \frac{10,2 \cdot M_b}{D_1^3}$ [N/mm²]

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

Torsional stress:

$\tau = \frac{5,1 \cdot Q_R}{D_1^3}$ [N/mm²]

D_1 = increased rudder stock diameter in [cm]

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

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

Q_R see B.1.2 and B.2.2 - 2.3

D_t see 1.1.

Guidance

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

Section 18

Equipment

A. General

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

Guidance

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

7. For 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/m³) 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 [m²], 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.

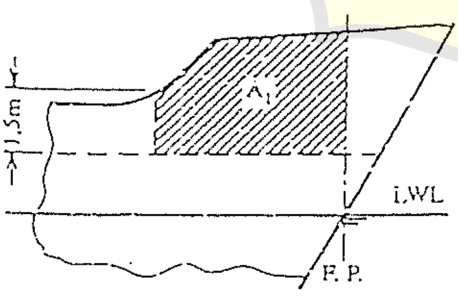


Fig. 18.1

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 pins 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.

Marine Pumps

Centrifugal Pumps

Naniwa centrifugal pumps are available as single- or double suctions, single- or double volutes, vertical- or horizontal installations, single- or multi stages, etc. to ideally suit any specification requirement.

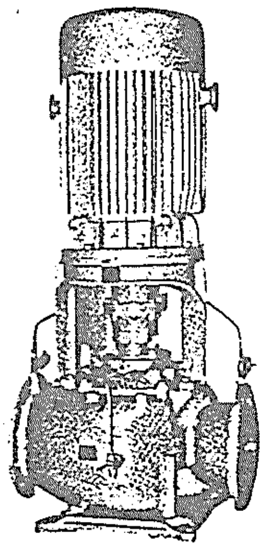
The materials of the principal parts of Naniwa centrifugal pumps are standardized as follows:

Casing: Bronze for sea water
Cast iron for fresh water
Stainless steel for chemicals

Impeller: Phosphor bronze for sea water and fresh water
Stainless steel for sea water and chemicals

Shaft: Stainless steel for sea water and fresh water

Vertical type



FEWV-300

FEV
FEWV

Applications:
Cooling water
Water service
Ballast

Specifications:
Vertical single stage
double volute
Centrifugal
Capacity 30–3,000 m³/h
Head 13–50 m

FBV
FBSV

Applications:
Cooling water
Water service
Fire & general service

Specifications:
Vertical single stage
single suction
Centrifugal
Capacity 30–500 m³/h
Head 10–75 m

FBWV
FCDV

Applications:
Cooling water
Condenser circulating
Ballast
Water service

Specifications:
Vertical single stage
double suction
Centrifugal
Capacity 200–15,000 m³/h
Head 5–50 m

EVPM

Applications:
Product cargo
Chemical cargo

Specifications:
Vertical multi-stage
single suction
Centrifugal
Capacity 50–1,500 m³/h
Head 50–150 m

TOM-200

TOM

Applications:
Lublicating oil
Oil service

Specifications:
Vertical multistage
single suction
Centrifugal
Capacity 30–1,000 m³/h
Head 2–15 kg/cm²

FDDV
2FDDV

Applications:
Condensate
Drain

Specifications:
Vertical 1–2 stage
single (double) suction
Centrifugal
Capacity 2–110 m³/h
Head 50–110 m

FB2V
FBCV
FE2V

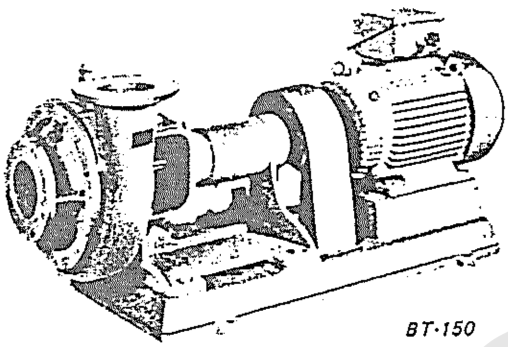
Applications:
Fire & general service
Bilge & ballast
Water service

Specifications:
Vertical two stage
single suction
Centrifugal
Capacity 40–800 m³/h
Head 25–170 m

Naniwa Pump Mfg. Co., Ltd.

Centrifugal Pumps

Horizontal type

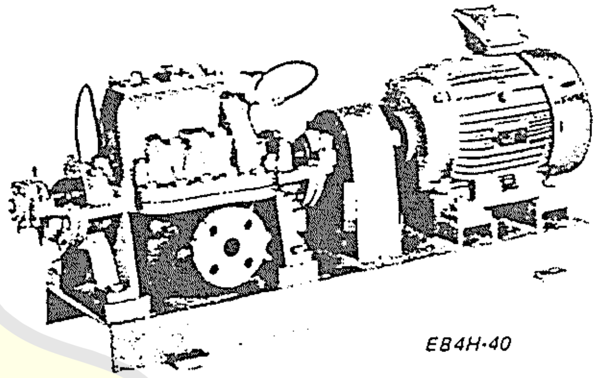


BT-150

BT
BH

Applications:
Cooling water
Water service
Fresh water
Sanitary

Specifications:
Horizontal single stage
single suction
Centrifugal
Capacity 2–700 m^3/h
Head 9–130 m

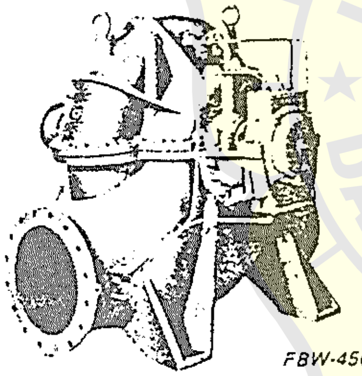


EB4H-40

EBH
EB2H
EB4H

Applications:
Boiler feed
Water service

Specifications:
Horizontal multi stage
single suction
Centrifugal
Capacity 1–80 m^3/h
Head 50–310 m

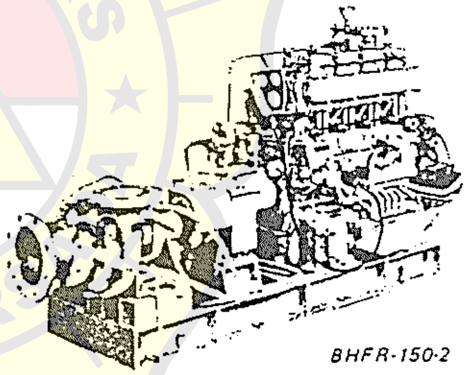


FBW-450

FBW

Applications:
Cooling water
Condenser circulating
Ballast
Water service

Specifications:
Horizontal single stage
double suction
Centrifugal
Capacity 200–3,500 m^3/h
Head 13–50 m



BHFR-150-2

BHF

Applications:
Fire
Emergency fire

Specifications:
Horizontal single stage
self priming
Centrifugal
Capacity 25–200 m^3/h
Head 40–90 m

SGH
SH

Applications:
Ballast
Water service
Fire & general service

Specifications:
Horizontal single stage
self priming
Centrifugal
Capacity 2–700 m^3/h
Head 10–65 m

BBH-L
BBH-S

Applications:
Boiler water circulating
Head transfer
Liquids circulating

Specifications:
Horizontal single stage
single suction
Centrifugal
Capacity 2–80 m^3/h
Head 15–65 m

FB2H

Applications:
Fire
Emergency fire

Specifications:
Horizontal 2 stage
Single suction
Centrifugal
Capacity 100–500 m^3/h
Head 50–150 m

EBHU

Applications:
Fire
Salvage

Specifications
Horizontal 2–3 stage
single suction
Centrifugal
Capacity 2–80 m^3/h
Head 80–210 m

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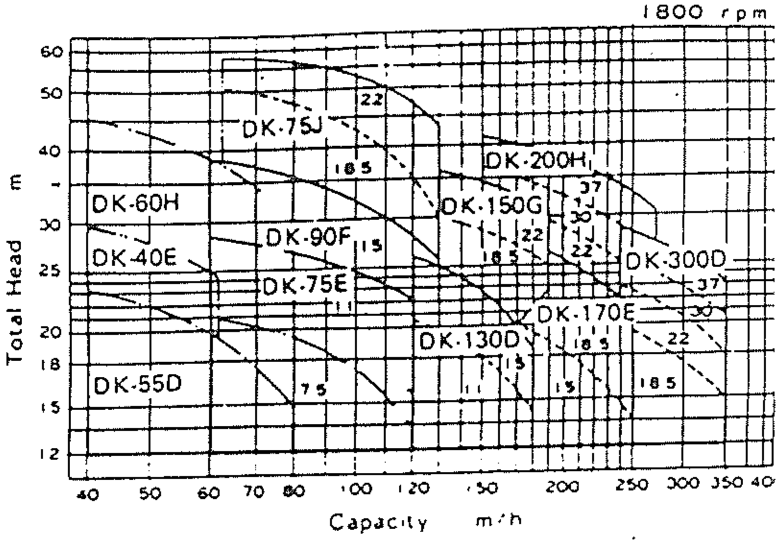
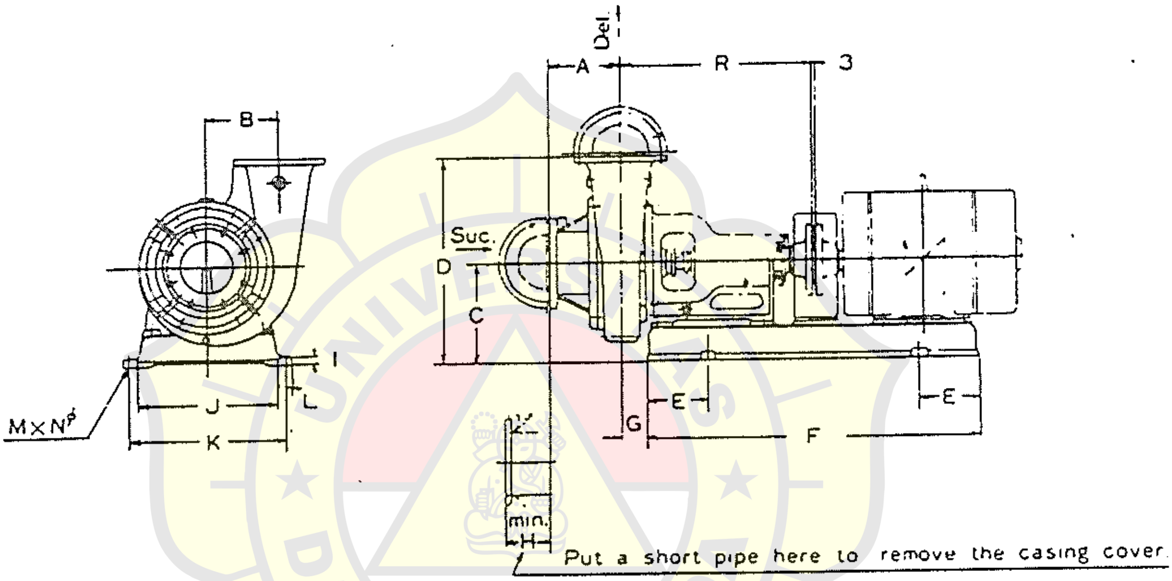




Type	Motor (kw)	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	R	Pump Weight (kg)
DK-75E	7.5	125	125	113	165	202	460	150	780	30	100	25	300	340	23	4	15	420	160
	11			113	165	227	485	175	860	30	100	25	360	400	23	4	15	420	160
DK-90F	15	125	125	150	180	231	505	175	900	33	100	25	360	400	23	4	15	423	210
DK-75J	18.5	125	125	156	205	290	580	150	960	35	120	30	390	440	25	4	19	480	240
	22			156	205	290	580	175	1000	35	120	30	370	410	25	4	19	480	240
DK-130D	11	150	150	160	160	225	485	175	860	38	100	25	360	400	23	4	15	428	160
	15			160	160	225	485	175	900	38	100	25	360	400	23	4	15	428	160
DK-150G	18.5	150	150	160	200	275	590	175	1000	58	120	30	400	450	25	4	19	548	250
	22			160	200	275	590	200	1050	53	120	30	400	450	25	4	19	548	250
DK-170E	15	200	200	178	190	246	590	150	950	40	120	30	350	400	25	4	19	485	250
	18.5			178	190	267	610	150	960	40	120	30	390	440	25	4	19	485	250
DK-200H	22	200	200	175	230	277	640	200	1050	65	120	30	400	450	25	4	19	560	320
	30			175	230	297	660	200	1100	65	120	30	450	500	25	4	19	560	320
	37			175	230	322	685	200	1150	65	120	30	490	540	25	4	19	560	320
DK-300D	18.5	250	250	185	235	255	640	175	1000	70	120	30	400	450	25	4	19	560	305
	22			185	235	255	640	200	1050	65	120	30	400	450	25	4	19	560	305
	30			185	235	275	660	200	1100	65	120	.30	450	500	25	4	19	560	305
	37			185	235	300	685	200	1150	65	120	30	490	540	25	4	19	560	305

Centrifugal Pump

TAIKO, a pioneer in the manufacture of centrifugal pumps, continues to offer the latest technological advances with its new family of centrifugal pumps. Various designs are offered, each having its advantages and preferred uses for which it is most suited. Unique in design and construction, TAIKO centrifugal pumps are packed with more exclusive money-saving features than any other models on the market today. They are manufactured under strict quality control, assuring long, trouble-free life under tough operating conditions.

TAIKO centrifugal pumps are recognized as the leader in centrifugal pumps. They are designed and manufactured according to ISO and other government and industrial standards.

Horizontal Pump

Suction	Stage	Model	Capacity m^3/h	Head m	Application
Single Suction	Single Stage	HC-Series EHC-Series EHS-Series TMC-Series SP-Series	2–2000	10–85	Cooling S.W. Pump Cooling F.W. Pump Bilge & Ballast Pump General Service Pump F.W. Pump Boiler Water Circ. Pump Sanitary Pump Fire & G.S. Pump Emergency Fire Pump Sewage Pump Sand Pump
	Double Stage	2MS-Series	10–300	15–300	Fire Pump Boiler Feed Water Pump Tank Cleaning Pump
	Multi Stage	nMT-Series nMS-Series			
Double Suction	Single Stage	HD-Series	200–4000	10–170	Cargo Pump Ballast Pump Cooling S.W. Pump Cooling F.W. Pump Sea Water Service Pump

Vertical Pump

Suction	Stage	Model	Capacity m^3/h	Head m	Application
Single Suction	Single Stage	VC-Series EVC-Series EMC-Series ESC-Series	30–500	10–65	Ballast Pump Sea Water Service Pump Cooling S.W. Pump Cooling F.W. Pump Hydrophor Unit
	Double Stage	VS-Series VSS-Series EMS-Series CVS-Series	30–600	20–100	Bilge & Ballast Pump Fire & G.S. Pump Cooling S.W. Pump Cooling F.W. Pump Condensate Pump
Double Suction	Single Stage	VD-Series ESD-Series	200–4000	10–170	Cargo Pump Ballast Pump Cooling S.W. Pump Cooling F.W. Pump Sea Water Service Pump

Tank Mounting Pump

Suction	Stage	Model	Capacity m^3/h	Head m	Application
Single Suction	Single Stage	C1T-Series DVC-Series SVC-Series	30–750	20–100	L.O. Pump Piston Cooling F.W. Pump F.V. Cooling F.W. Pump Cargo Pump Tank Cleaning Pump
	Multi Stage	C2T-Series nVC-Series	100–300	40–170	

Self Priming System: EHS-Series has its own function (built in type)
Other series of pumps should install a vacuum pump or an air ejector.

hree Rotor Screw Pump


Three Rotor Screw

eatures:

Simplified construction
Radically new design using one drive screw and two driven screws.

Ease of handling
Easy to handle as all mechanisms are rationally designed and engineered.

Simple and straight forward operation
The rotating section is at a minimum and each screw rotates by liquid pressure to ensure freedom from vibration and noise.

Pulsating and agitation free
Pulsating and agitation are totally absent to offer continuous pumping and supply liquid.

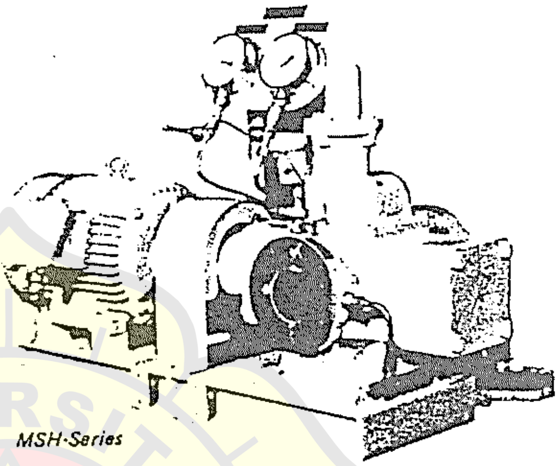
Long service life
Liquid pressure exerted on the rotating section is properly balanced to reduce minimum. Also driven screw pressure is minimal for minimized wear.

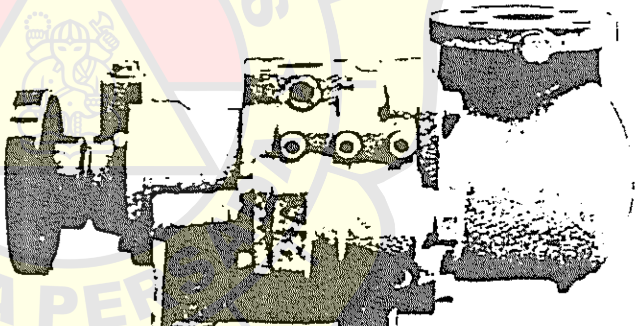
High pumping efficiency
Such screw pump characteristics as force transmission loss, friction loss, and leakage loss are minimized to achieve unusually high pumping efficiency.

Small starting torque
Small inertia on the rotating section means small starting torque.

Compact design but large capacity
High-speed operation and relatively compact design.

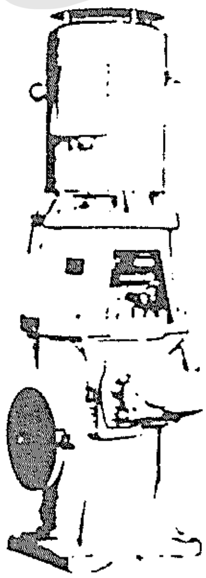
plication: F.O. trans pump, main L.O. pump, hydraulic oil pump, cross head L.O. pump, burning pump, F.O. booster pump.

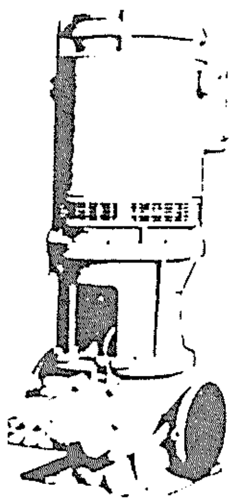

MSH-Series


MSH-X Series

MSH-/MSH-X Series
(Horizontal, internal bearing type)
Capacity : 0.5–150 m^3/hr
Head : Max. 60 kgf/cm^2
H. Power : 0.4–500 kW

MSV/MST Series
(Vertical, internal bearing type)
Capacity : 30–600 m^3/hr
Head : Max. 16 kgf/cm^2
H. Power : 11–150 kW


MSV-Series


MST-Series

iiko Kikai Industries Co., Ltd.

wage Treatment Unit

IKO SHIP CLEAN "SBT Series" are compact :d sewage treatment system with superior per- mance capabilities, designed exclusively for ·ine installations, and which were developed TAIKO with high technology of many years ·eriences for Marine Public Nuisance. These ice thereby more enables to be installed in all sels of 200 or more gross tons, or accommo- ing 10 or more peoples as described in the nex IV of MARPOL TREATY 73/78.

tures:

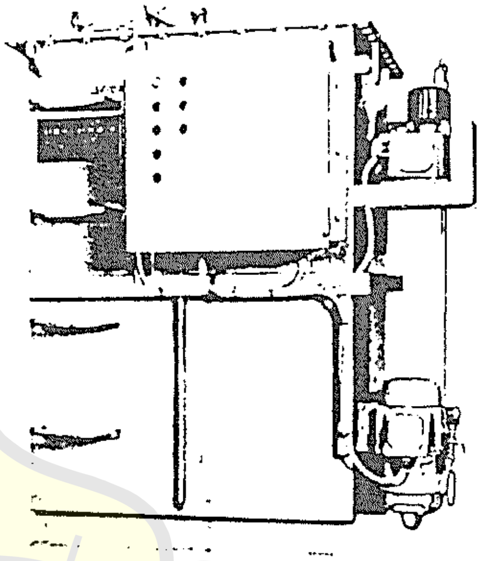
High Capabilities
The use of a "Submerged Bio-Filter System" and the transposition of the sterilization com- partment to the center of the device enable it to be more compact. These state of art de- vices are thereby more stable under condition of pitch and roll.

Comply with MARPOL TREATY 73/78
In accordance with the IMO recommended MEPC2 (V1) test standard, the certified authorization for USCG and/or UK/DOT has been obtained.

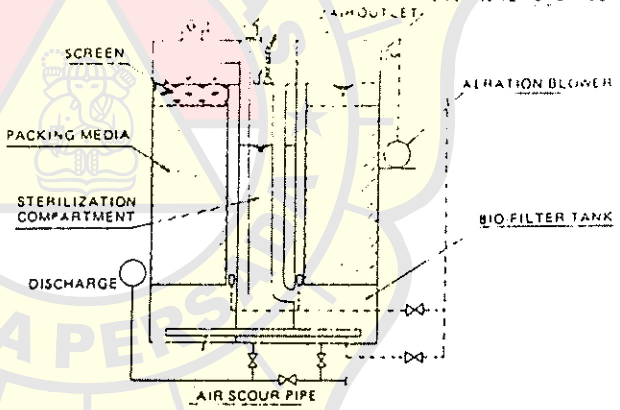
Fully automatic integrated type
A pump and a blower are mounted on the device. Therefore, piping and wiring works are implified. These device are fully automatic except few maintenance and control, such as emoval of sludge and/or filling disinfectant, tc.

The installation can be made at the offshore vorks.
t is ultra small, accordingly the installation is easible without requiring the works in docks.

Others
he stability period is shorten. (3 days)



SBT-Series



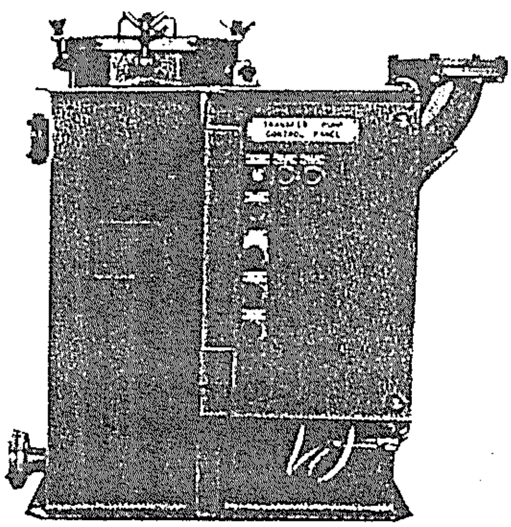
SBT-Series Sectional Drawing

ification

Model	SBT-15	SBT-25	SBT-40	SBT-65
of persons persons/days	15	25	40	65
of sewage d ℓ/days	900	1500	2400	3900
d amount of ℓ/ourt × frequency/days	94 × 3	156 × 3	250 × 3	406 × 3
ount g/days	202.5	337.5	540	877.5
Air flow m³/minutes	0.22	0.37	0.60	0.98
Pressure kgf/cm²	0.2			
Motor power input kW	0.4	0.4	0.75	0.75
Capacity m³/hours	7 (60 Hz)		6 (50 Hz)	
Head m	20 (60 Hz)		12 (50 Hz)	
Motor power input kW	1.5			

ting Tank Model: SCT-200

; on vessels are scattered. When pipings can- : made to collect sewage from every toilet sewage treatment device (BFT-40) please this sewage collecting tank. The sewage ed in this tank is transferred automatically sewage treatment device by an attached transfer pump.



Collection Tank SCT-200

Taiko Kikai Industries Co., Ltd.

JST-Series

Certificate No.

Model	CAPA m³/h	H.K. Cert. No.	USCG Cert. No.
UST-01	0.15	820S-p No. 1016	162.050/1056/0
UST-03	0.25	80TK-h No. 108554	162.050/1056/0
UST-05	0.5	80TK-h No. 108555	162.050/1058/0
UST-10	1.0	80TK-h No. 108556	162.050/1059/0
UST-20N	2.0	80TK-h No. 108557	162.050/1060/0
UST-30N	3.0	80TK-h No. 108558	162.050/1061/0
UST-50N	5.0	80TK-h No. 108559	162.050/1062/0

ther Approvals
ie UK DOT, The NASASN (Sweden), The eek Government, The Norwegian Maritime rectorate, The Republic of Panama, Republic Korea.

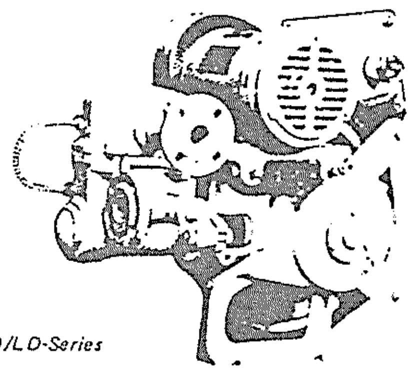
ston Pump

atures:
Strong "Self-Priming", no need for initial feed of water or liquid.
Minimum number of parts to facilitate maintenance and for reduced parts stock.
plication: Bilge pump, oily water separator pump F.O. trans pump

Horizontal Type

1) PD-03
Capacity : $0.25\ m^3/hr$
Head : $2\ kgf/cm^2$
H. Power : 0.4 kW

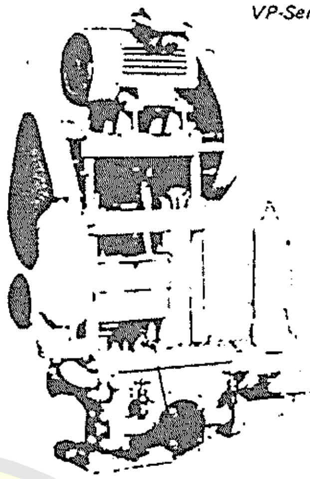
2) LD-Series
Capacity : $0.5–5\ m^3/hr$
Head : $2–3\ kgf/cm^2$
H. Power : 0.4–1.5 kW



PD/LD-Series

ertical Type

>-Series
pacity : $2–60\ m^3/hr$
ad : $2–4\ kgf/cm^2$
Power : 0.75–15 kW



VP-Series

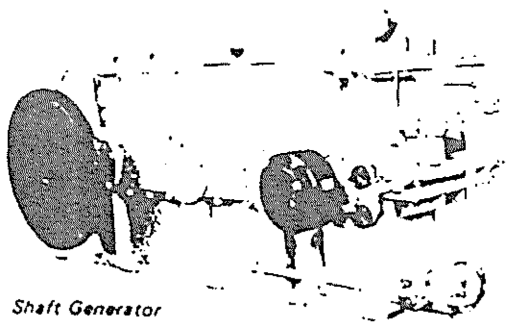
Transmission

Shaft Generator

Due to the increase in oil prices in recent years, the problem of "SAVING ENERGY" has to be solved. Especially, the "FUEL OIL CONSUMPTION RATIO" for vessels and such equipment as the engine, propellers, generator, etc., required improvement.
As a result, we have succeeded in developing new, compact, high-speed models of our SHAFT GENERATOR.

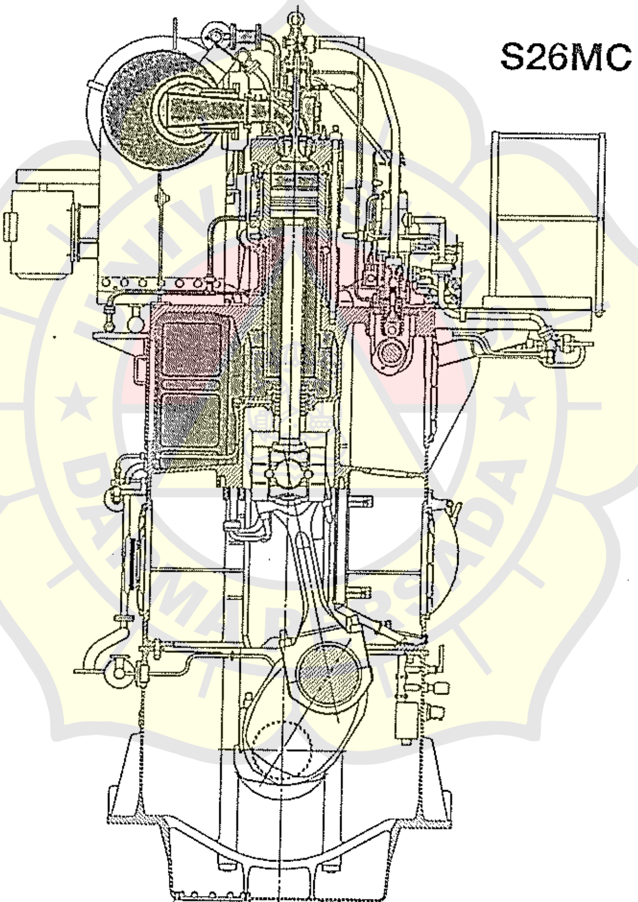
Features:
1) Conventional auxiliary engines use type A-Heavy Fuel Oil. The main engine uses type C-Heavy F.O. except when entering or leaving port. Therefore, since the fuel oil consumption ratio of the main engine, SHAFT GENERATOR reduces both the amount of fuel consumed and the unit price of that fuel. Thus the system allows a dual economy regarding fuel.
2) Maintenance requirements for the aux. engine are eliminated, and engine changes are reduced.
3) Generator, cargo pumps, hydraulic pumps, etc., are driven by a compact transmission system, thus reducing the engine room space required by aux. engines.

Capacity: Up to 3000 HP
Revolution: 200–2000 RPM

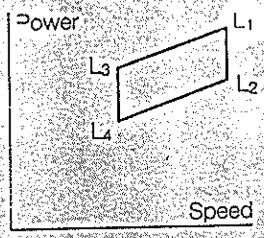


Shaft Generator

Engine cross section



Power, Speed and SFOC



S26MC

Stroke: 980 Bore: 260

		L1	L2	L3	L4
Speed	r/min	250	250	212	212
mep	bar	18.4	14.8	18.4	14.8
Cylinder	**Power**				
4	kW	1 600	1 280	1 360	1 100
	BHP	2 180	1 740	1 860	1 480
5	kW	2 000	1 600	1 700	1 375
	BHP	2 725	2 175	2 325	1 850
6	kW	2 400	1 920	2 040	1 650
	BHP	3 270	2 610	2 790	2 220
7	kW	2 800	2 240	2 380	1 925
	BHP	3 815	3 045	3 255	2 590
8	kW	3 200	2 560	2 720	2 200
	BHP	4 360	3 480	3 720	2 960
9	kW	3 600	2 880	3 060	2 475
	BHP	4 905	3 915	4 185	3 330
10	kW	4 000	3 200	3 400	2 750
	BHP	5 450	4 350	4 650	3 700
11	kW	4 400	3 520	3 740	3 025
	BHP	5 995	4 785	5 115	4 070
12	kW	4 800	3 840	4 080	3 300
	BHP	6 540	5 220	5 580	4 440

Specific Fuel Oil Consumption (SFOC)

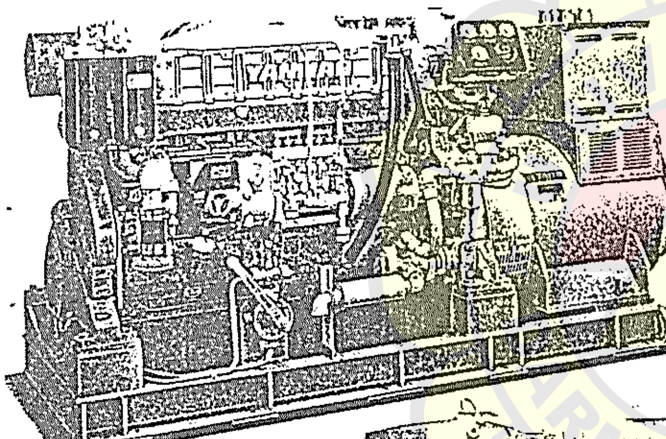
Conventional turbocharger	L1	L2	L3	L4
g/kWh	179	174	179	174
g/BHPh	132	128	132	128

Lubricating oil consumption:	approximately 1.5 kg/cyl. 24h
Cylinder oil consumption:	1.0-1.4 g/kWh ≈ 0.7-1.0 g/BHPh

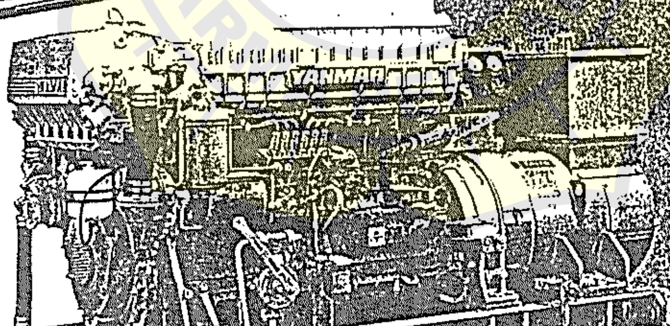
MARINE AUXILIARY DIESEL ENGINE

CHLN series 38~120hp
HAL2N series 122~414hp
LAALN series 330~840hp

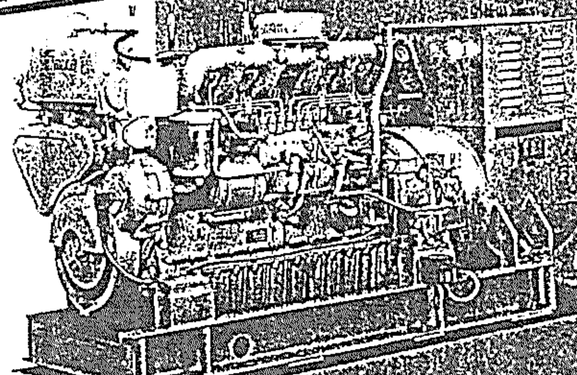
YANMAR



LAALN series



HAL2N



CHLN series

Certificate No. FM23056

Certificate No. JMI-0099

ISO 9001 Certified

THREE GREAT WAYS TO SAVE (

Yanmar's Marine Auxiliary Series with direc

;ENERAL FEATURE

lore and More Economy
major factor in their remarkable reduction of fuel consumption is a cooling stem which circulates fresh water at a constant high temperature, to eatly raise combustion efficiency.

ean and Light Design
e compact dimensions of the series offers space-saving features that are at designer's dream. They allow more effective use of engine room ace, more spacious cabin/locker areas, or larger fish holds.

Less Wear, More Work
Another benefit of the efficient constant t reduction in thermal load in the combusti wear on the piston liners increases engin frequency of major overhauls.

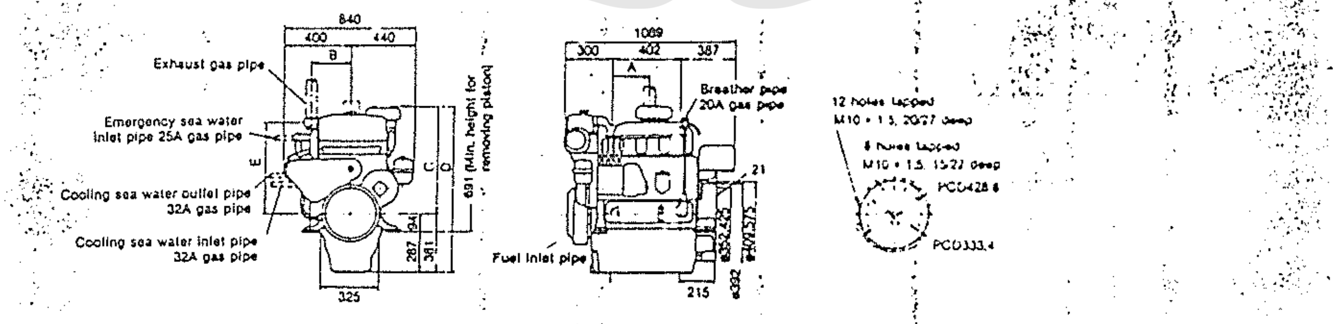
Speed Options
All three models are available with 1800r generators, and 1500rpm for 50Hz, 4-pol

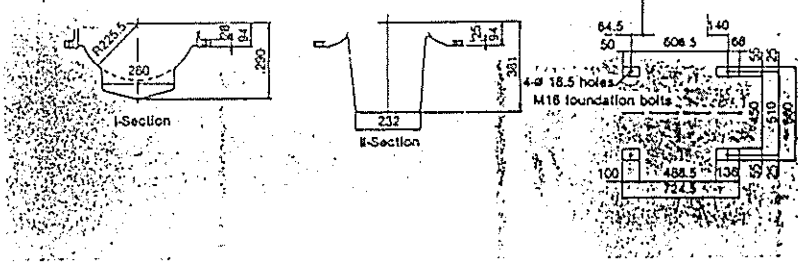
ECIFICATIONS

odel			4CHL-N		4CHL-TN		6CHL-N		6CHL-TN		6CHL-HT	
'pe												
ombustion type												
ımber of cylinders			In-line 4				In-line 6					
re×stroke		mm(in)	105×125 (4.13×4.92)									
splacement		lit. in³)	4.330 (264)				6.494 (396)					
ontinuous rated engine speed		rpm	1500	1800	1500	1800	1500	1800	1500	1800	1500	1ɛ
ontinuous rated output		hp	38	50	50	62	62	74	74	100	100	1
plicable generator capacity		kVA(kW)	30(24)	40(32)	40(32)	50(40)	50(40)	60(48)	60(48)	80(64)	80(64)	1(
equency		Hz	50	60	50	60	50	60	50	60	50	(
arting system			El									
oling system	Type		Constant high temperature fresh water forced circula									
	Fresh water capacity	lit(gal)	16.8 (4.4)				24 (6.3)					
rication system	Type		Forced lubrication with gear pump, water co									
	Oil filter											
	Oil capacity		16.5 (4.4)				32 (8.5)					
el system	Injection pump type		PE-AD									
	Governor type		Centrifugal speed control									
	Injection valve											
	Feed pump/Fuel filter		F									
bocharging system			Naturally aspirated		*1		Naturally aspirated		*1		*2	
ctrical system	Starter motor	24V-kW	4.0									
	Alternator	24V-W	600									
	Battery	V-AH×Q'ty	12V-120AH×2									
O.	Direction of rotation											
	Flywheel housing		Yanmar original									
cific fuel consumption 1500rpm/1800rpm		g/kW-hr	231		227		2.:0		218		208	
ine dry weight (Engine only)		kg(lbs)	500 (1100)		520 (1145)		625 (1380)		645 (1420)		675 (149	

ENSIONS (Unit:mm)

HL-N / 4CHL-TN





Model	A	B	C	D	E	Exhaust gas pipe
4CHL-N	220	232	641	1021	480	65A gas pipe
4CHL-TN	413	196	810	1191	700	80A gas pipe

N FUEL AND MAINTENANCE
fuel injection

ɪrature cooling system is a
ıamber area. This reduces
span, and reduces the

pecifications for 60Hz, 4-pole
ıerators.

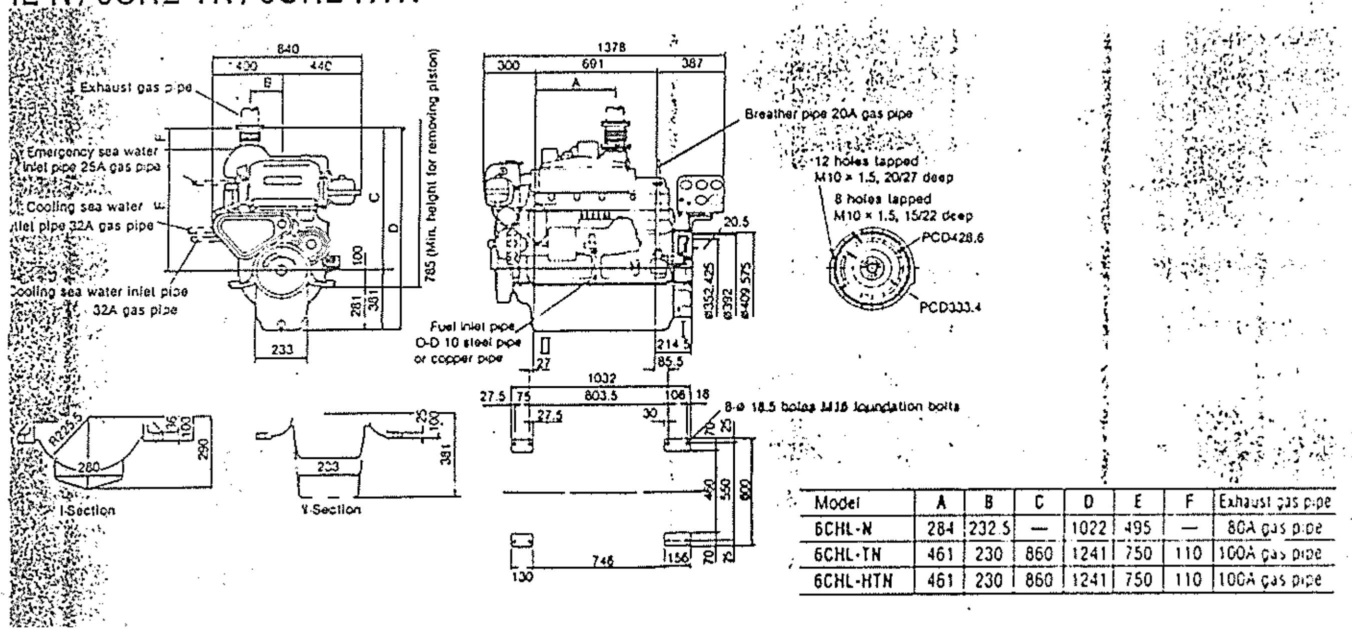
Versatile Technical Services

You can avail yourself of Yanmar's extensive experience in coupling technology, to create the most applicable system for your purposes. For generator coupling, multiple gen/compressor coupling, or for any other coupling combination, Yanmar can provide expert engineering services. These included, beside fabrication of a common bed, the use of special cooling systems, installation work, and special requirement such as a specified lubrication system, remote control devices, etc.

*1: Exhaust turbocharger *2: Exhaust turbocharger with intercooler

6HAL2-N			6HAL2-TN			6HAL2-HTN			6HAL2-DTN			6LAAL-DTN			6LAAL-UTN			12LAAL-DTN		
rtical 4-cycle, water cooled diesel engine																				
Direct injection																				
In-line 6												In-line 6						Vee 12		
130×165 (5.12×6.50)												148×165 (5.83×6.50)								
13.14 (802)												17.03 (1039)						34.06 (2078)		
1200	1500	1800	1200	1500	1800	1200	1500	1800	1200	1500	1800	1200	—	—	1200	1500	1800	1200	1500	1800
122	156	183	163	204	244	217	299	360	271	346	414	330	—	—	360	420	480	600	720	840
100(80)	125(100)	150(120)	130(104)	170(136)	200(160)	180(144)	250(200)	300(240)	225(180)	290(232)	350(280)	275(220)	—	—	300(240)	350(280)	400(320)	500(400)	600(480)	700(560)
60	50	60	60	50	60	60	50	60	60	50	60	60	—	—	60	50	60	60	50	60
(compressed air starting is available as option)																				
ıe with centrifugal sea and fresh water pumps (CHL series ; rubber impeller type sea water pump)																				
66 (17.4)												76 (20.1)						150 (39.6)		
ıltiple tube oil cooler, cylindrical oil pressure regulating valve with exterior adjusting screw																				
Paper filter																				
90 (23.8)												64 (16.9)						145 (38.3)		
P-S3S									P-S7C			YPE-1411								
Centrifugal speed control												Centrifugal speed control						Hydraulic governor		
Semi-long hole nozzle																				
type, attached on injection pump / paper filter																				
laturally aspirated			*1			*2						*2								
6.0												6.0×2						7.0×2		
600												600						840		
12V-120AH×2												12V-120AH×4						12V-150AH×4		
ınterclockwise viewed from flywheel side																				
SAE No.1												SAE No.0								
218	224	231	213	218	224	207	211	220	204	207	213	204	—	—	213	215	215	211	213	220
1380 (3042)			1395 (3075)			1410 (3109)			1420 (3131)			1990 (4390)			2050 (4525)			3500 (7718)		

ɪL-N / 6CHL-TN / 6CHL-HTN



Model	A	B	C	D	E	F	Exhaust gas pipe
6CHL-N	284	232.5	—	1022	495	—	80A gas pipe
6CHL-TN	461	230	860	1241	750	110	100A gas pipe
6CHL-HTN	461	230	860	1241	750	110	100A gas pipe

REFERENSI NO.4

UNIVERSITAS DARMA PERSADA

ngisapan) dari pompa; ini diukur dalam m H_2O. Tenaga yang dibutuhkan dapat dihitung dari rumus :

$$N = \frac{G.H}{60x60x75x\ \eta} \quad H.P = \frac{G.H}{60x60x102x\ \eta}\ K.W$$

$$N = \frac{Q.\ \gamma.\ H}{60x60x75x\ \eta} \quad H.P = \frac{Q.\ \gamma.\ H}{60x60x102x\ \eta}\ K.W$$

dimana :

Q = Output dalam m^3/jam

H = Head dalam m H_2O

γ = Berat jenis cairan dalam kg/m

γair tawar = 1,0 pada 4 C

γair laut = 1,025 - 1,03

γfuel oil = 0,825 - 0,95

η = Effisiensi total dari pompa, yang bervariasi antara 0,5 - 0,9 tergantung dari ukuran dan type pompa.

B. 2. Klasifikasi Pompa sesuai dng. Tujuan.

Sesuai dengan tujuannya, pompa diatas kapal dapat dibagi menjadi tiga golongan.

1. General service pumps yang funfsinya adalah untuk meyakinkan ke-layak lautan sebuah kapal dan untuk menyediakan kebutuhan sehari-hari dari awak kapal dan penumpang dan juga keperluan sanitary diatas kapal, Termasuk general service pumps :

   a. pompa bilga
   b. pompa saniter
   c. pompa pemadam kebakaran
   d. pompa darurat

2. Pompa-pompa yang direncanakan untuk melayani sistim dari mesin utama dan mesin-mesin bantu misalnya pompa pendingin, pompa bahan bakar, pompa pelumas dan sebagainya.

3. Pompa-pompa yang bertujuan khusus didalam tanker, trawlers, kapal pemecah es, kapal penyelamat dan kapal keruk.

B. 3. Pr

Ter an pompa

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

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

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

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

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

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

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

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

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

Table 47

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

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

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

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

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

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

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

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

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

It follows from formula (314) that

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

Combining equations (315) and (316) we obtain

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

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

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

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

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

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

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

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

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

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

4

... must not exceed

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

ere, 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 manufacturing specifications for capstans worked out by the Central rine 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
3,600	0.25	750
4,500	0.2	900
7,000	0.167	1,165
12,000	0.150	1,800

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

$$\chi_q = l_q h_q \, .$$

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

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

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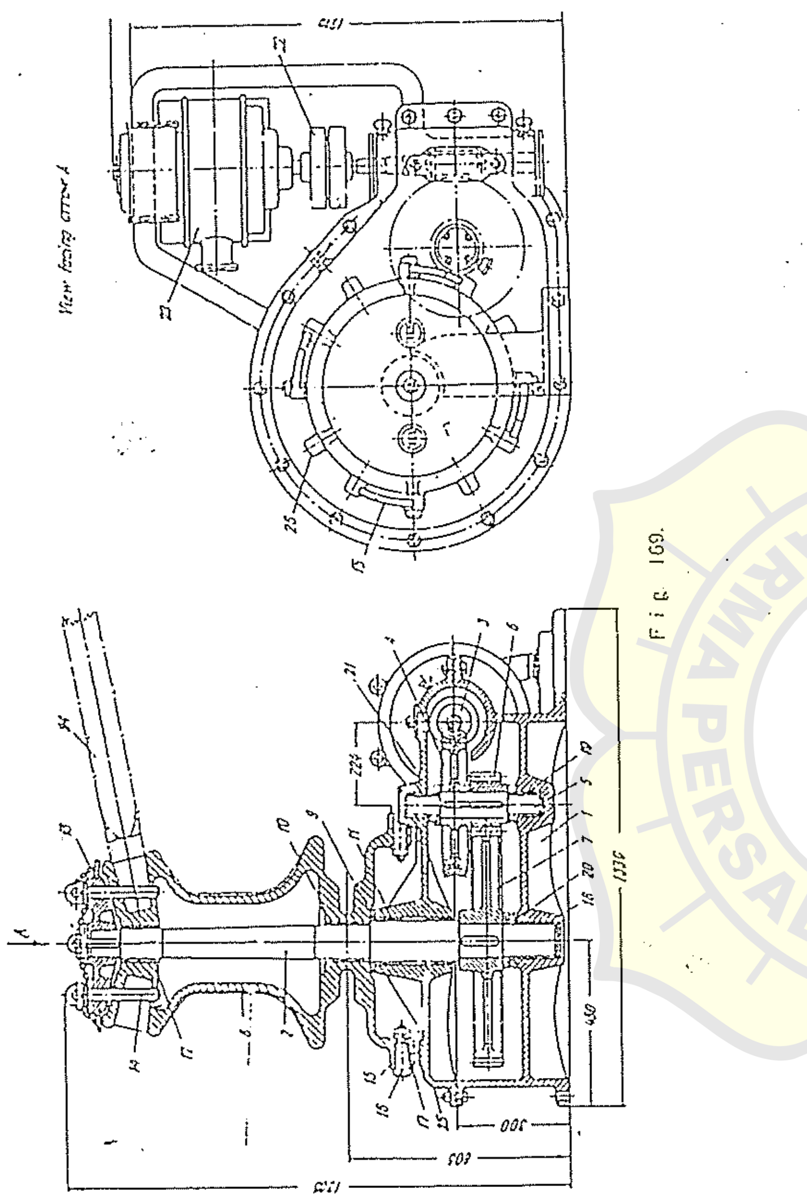

Fig. 169.

5-3. Determining the Principal Dimensions of Anchoring and Warping Machinery

The initial data used to determine the principal dimensions of anchoring machinery are the required pull of the cable lifter and the speed at which the anchor is weighed from the anchorage depth, which is equal to the distance from the hawse hole to the bottom.

It is advisable to determine the pull on the cable lifter so as to ensure that one anchor will be brought in at a speed of at least 12 m per min from the anchorage depth which is taken equal to:

80 m if each anchor weighs 1,000 kg or less

90 m if the anchor weighs from 1,500 to 3,000 kg

100 m if the anchor weighs from 3,000 to 6,000 kg.

The following notation will be used to derive the formulas for determining the pull on the cable lifter:

G_a = weight of the anchor, kg

p_a = weight per running metre of the chain cable, kg

L_a = length of the suspended cable, m

γ_a = 7,750 = density of the material of the anchor, kg per cu m

γ_w = 1,025 = density of sea water, kg per cu m

f_h = 1.28 to 1.35 = a factor taking into account the friction losses in the hawse hole and stopper.

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

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

In hoisting one anchor

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

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

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

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

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

* In breaking away one anchor from the bottom

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

... of liquid ... 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

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

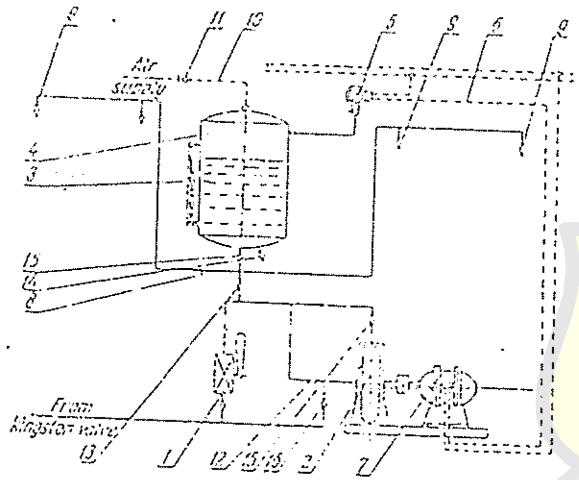

Fig. 189

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

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

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

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

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

(b) SANITARY AND SCUPPER SYSTEMS

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

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

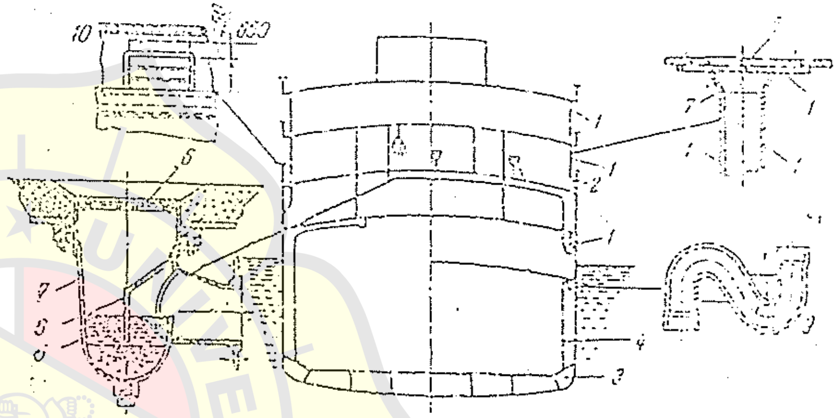

Fig. 190

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

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

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

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

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

REFERENSI NO.5

of gravity are not yet exactly known in the early project stage. If the model does not accomplish the required speed the designer has to alter the hull. This alteration, however, is possible in the early project stage only. If the trial speed in ballast condition corresponds to the model trial speed in ballast, it can be assumed that service speed in loaded condition is attained, too.

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

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

In general

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

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

76. Consumables and tanks

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

- consumables
- provisions
- ballast.

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

- fuel oil

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

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 auxillary engines [KW]

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

S = operating range [sm]

V_{serv} = speed [kn]

1 KW = 0.736 PS (BHP).

Motors:

Specific fuel oil consumption:

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

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

b = 196 ... 209 [g/KW·h]

for full power: addition 5%

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

For steam turbines:

Standard circulation without furnace gas reheat

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

b = 278 ... 286 [g/KW·h]

with furnace gas reheat

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

b = 252 ... 265 [g/KW·h]

For gas turbines:

Gasoline and light crude oils

b = 299 ... 312 [g/KW·h]

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

Required volume of storage tanks

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

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.

<u>Diesel oil</u>

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

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

additions see fuel oil!

<u>Lubrication oil</u>

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

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

$b = 0.8 \ldots 1.2$ [g/KW·h] diesel engine two stroke

$b = 1.2 \ldots 1.6$ [g/KW·h] diesel engine four stroke

$b = 0.11$ [g/KW·h] turbines and gearboxes

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

<u>Fresh water</u>

- 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.

[illegible]) 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.

- [illegible]% lower fore peak tank
- [illegible]3% upper fore peak tank
- [illegible]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.

[illegible]) 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, al:

- 1 -

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

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

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

where

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

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

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

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

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

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

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

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

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

4.3 Ballast systems in the cargo area

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

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

a) Minimum wall thicknesses

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

b) Only completely welded pipes or equivalent are permitted

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

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

5. Ventilation and gas-freeing

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

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

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

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

REFERENSI NO.6

Ref -6

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

7.8. Crew Members

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

7.9. Outfit and Equipment

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

7.10. Classification, Rules

have to be observed.

7.11. Restrictions of Dimensions

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

7.12. Tonnage of Ships

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

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

Tonnage unit: gross ton

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

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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

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

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

Net tonnage

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

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

T = draught in [m] (midships)

D = depth in [m] "

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

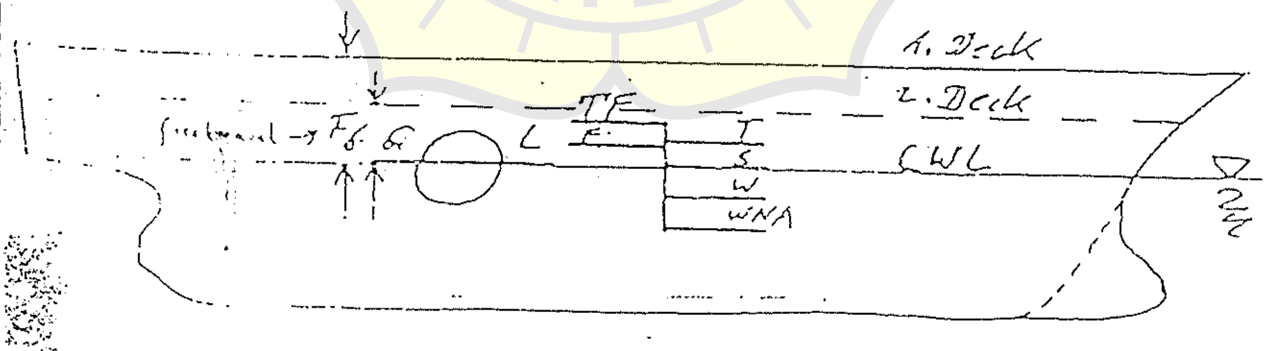
N_2 = number of other passengers.

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

Definition of freeboard:

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

ICLL International Freeboard Convention 1966.



Definition of superstructures:

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

REFERENSI NO.8

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.

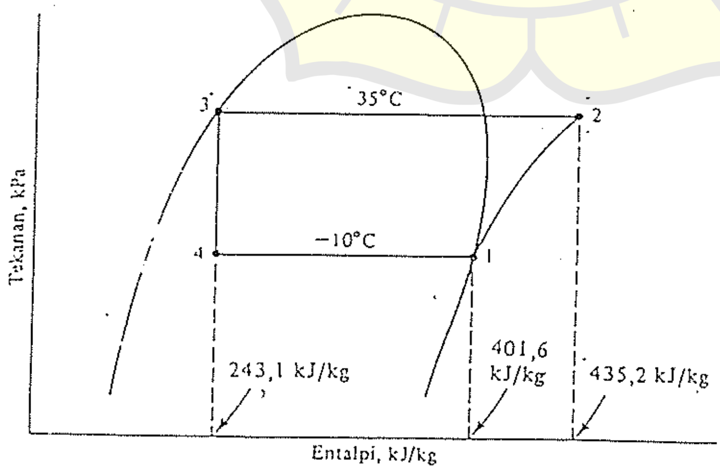
Untuk menemukan h_2 melalui garis entropi tetap geser titik 1 hingga mencapai tekanan jenuh yang sesuai dengan suhu 35°C. Tekanan pengembunan ini adalah 1355 kPa, dan nilai h_2 = 435,2 kJ/kg.

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.

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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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	1645	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	854	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/cm^2 abs),

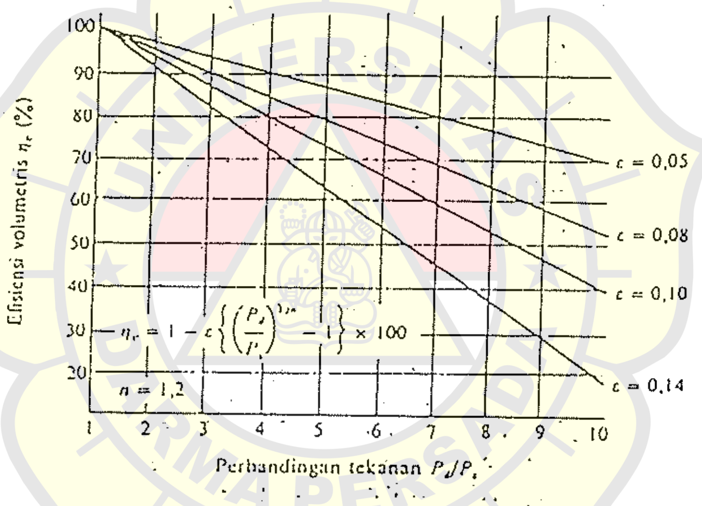
P_s: Tekanan isap dari silinder tingkat pertama (kgf/cm^2 abs).

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

Tanda ≈ 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:

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

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

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

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

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

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%.

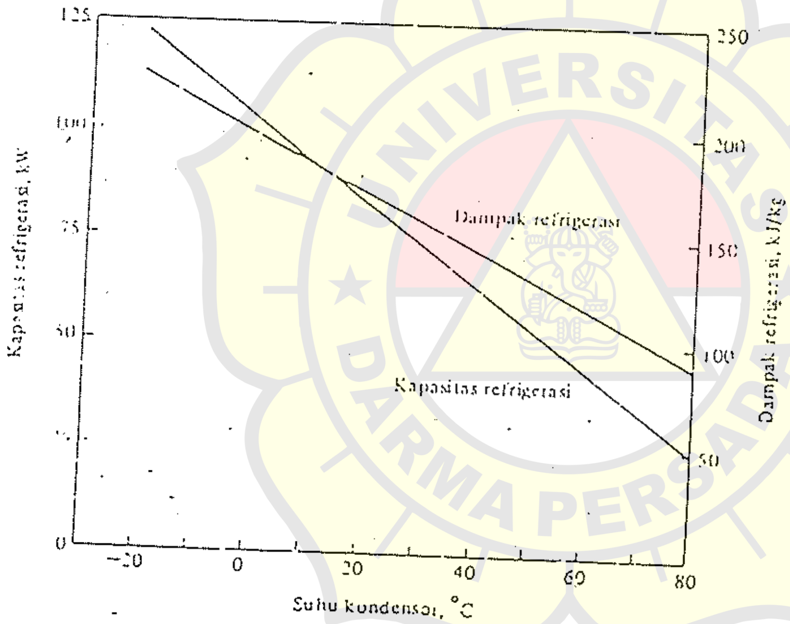
Selanjutnya, karena harga daya adiabatis teoritis untuk kompresor 1 tingkat berbeda dengan harga untuk kompresor 2 tingkat, maka memperbandingkan efisiensi kompresor harus dilakukan di antara yang sama jumlah tingkatnya.

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

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

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

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



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

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

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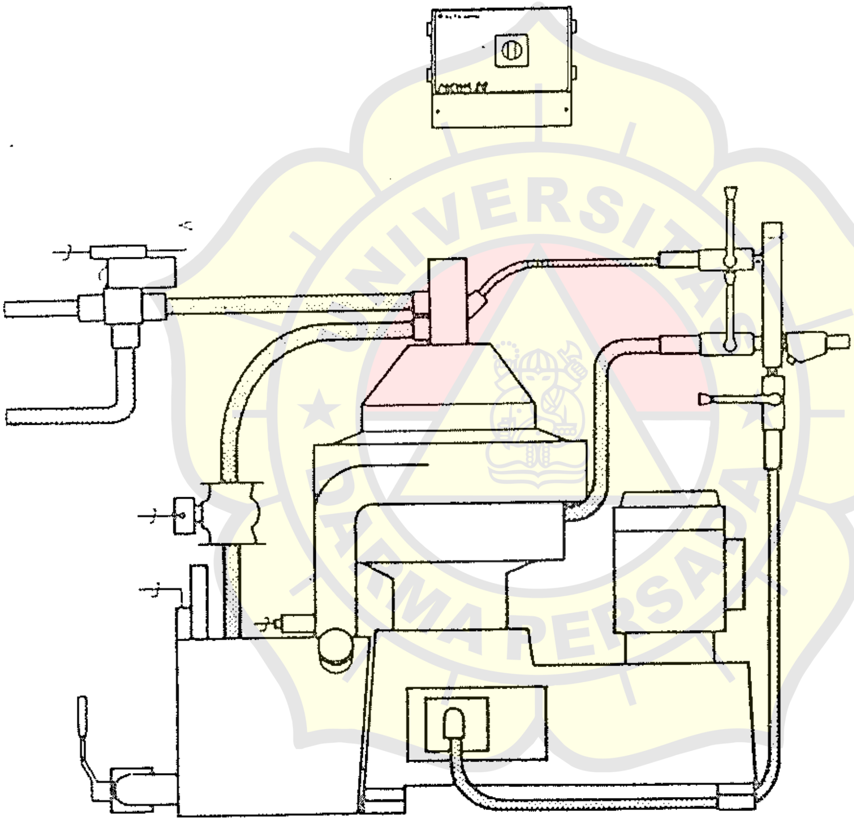

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工事番号：
Work No. R0−K501B