BAB IX

KESIMPULAN

9. 1. KESIMPULAN

Dari perencanaan dan perhitungan berdasarkan data kapal berikut :

- Type kapal : **TANKER**
- Lwl : 85,68 m
- Lpp : 84 m
- B : 15 m
- T : 5 m
- H : 7 m
- Cb : 0,571
- Cm : 0,976
- Cp : 0,585
- Cw : 0,7
- Vs : 16 knots (8,2304 m/ dtk)

Maka dapat disimpulkan :

1. Pada perencanaan ini dipilih motor induk dengan daya sebesar 4.300 HP pada putaran 250 rpm.
2. Dengan jumlah crew 30 orang dan route pelayaran yang ditempuh 11.000 mil, kapasitas maksimum kebutuhan listrik untuk mensuplai peralatan yang ada sebesar **193,428 kW**. Dalam perencanaan ini digunakan 3 unit generator yang sama besar dimana 1 unit generator tadi dipakai sebagai generator cadangan ataupun standby generator. Untuk itu dipilih generator dengan kapasitas masing-masing sebesar **180 kW**.

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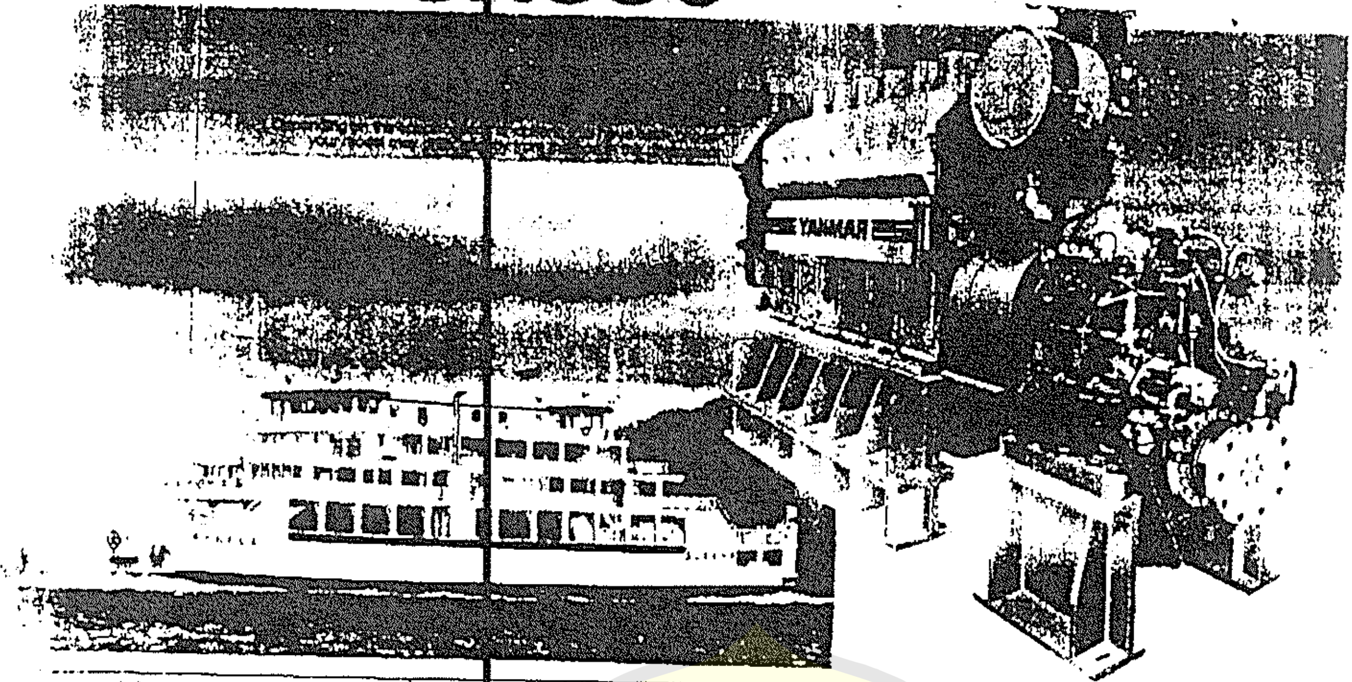
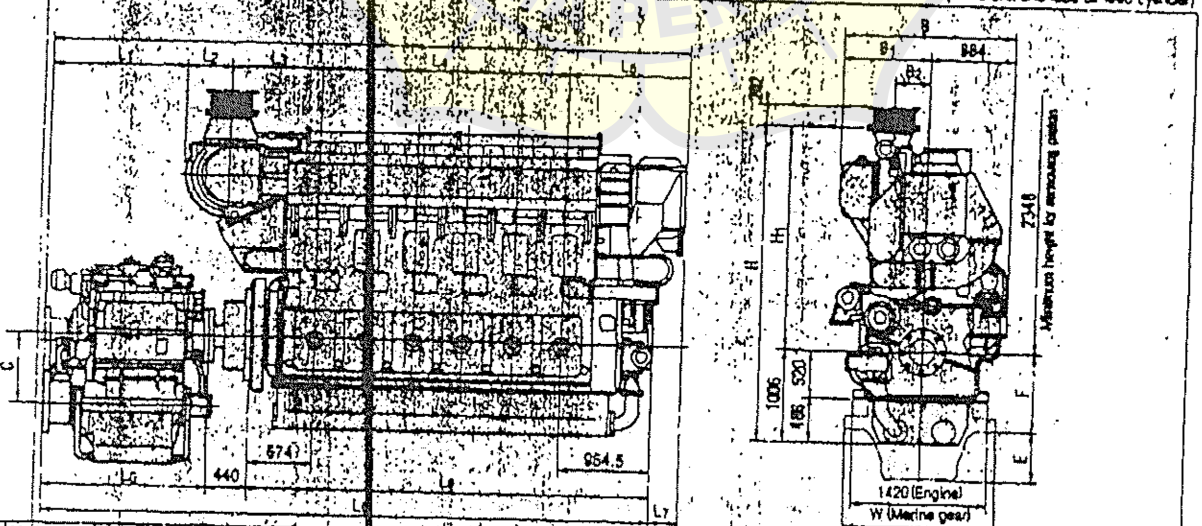
6N330/8N330



ngine model			6N330-UN	6N330-SN	6N330-EN	8N330-UN	8N330-SN	8N330-EN
umber of cylinder			6	6	6	8	8	8
ylinder bore × stroke		mm	330 × 440	330 × 440	330 × 440	330 × 440	330 × 440	330 × 440
ontinuous rated output		PS(kW)	3000(2207)	3300(2427)	3500(2574)	4000(2942)	4300(3163)	4500(3310)
ated engine speed		min^{-1} (rpm)	620	620	620	620	620	620
ngine dry mass		kg	30000	30000	30000	43000	43000	43000
ropeller type			for C.P.P.	for F.P.P.		for C.P.P.	for F.P.P.	
arine gear type			Reduction gear	Reverse-reduction gear		Reduction gear	Reverse-reduction gear	
andard marine gear	offset		YX-3500M	YX-3500		—	—	
	co-axial		YX-3500MC	YX-3500C		RCA-80F	DRA-80F	
duction gear ratios (ahead)	offset		2.55, 2.80, 3.09, 3.31	2.55, 2.80, 3.09, 3.31		—	—	
	co-axial		2.31, 2.54, 2.80, 3.00	2.31, 2.54, 2.80, 3.00		2.400, 2.687, 3.023	2.400, 2.687, 3.023	
arine gear mass	offset	kg	8700*	9400		—	—	
	co-axial		9300*	10000		11500	12000	

The engine dry weight may differ depending upon the specifications and attached accessories. The weight listed above does not include the high elastic coupling weight.
The asterisk indicates that the weight includes C.P.P. accessories (valve unit and lube oil feed cylinder)



Engine	Marine Gear	L	L1	L2	L3	L4	L5	L6	L7	L8	L9	B	B1	B2	H	H1	E	F	C	W	Flexible Coupling
6N330-EN	YX-3500M	6657	1409	460	950.5	2575	1265.5	4203.5	308	1795.5	6340	1789	905	373	3433	2427	850	878.5	776.5	1600	Geislinger BC72/17.5/85-12
	YX-3500	6568.5	1320.5	460	950.5	2575	1265.5	4203.5	308	1695	6278.6	1789	905	373	3433	2427	300	1126.6	776.5	1670	
	YX-3500C	6648.5	1400.5	460	950.5	2575	1265.5	4203.5	308	1697	6340.5	1789	905	373	3433	2427	300	430	80	1670	
8N330-EN	RCA-80F DRA-80F	8209	1816.5	547.5	1050	3505	1290	5233.5	495.5	2100	7773.5	1890	1006	349.5	3539	2533	560	250	0	1600	BC72/17.5/85-12

Please confirm all specifications, etc. on the separate delivery specifications sheet.
When installing the product, please request a separate outline drawing of the engine with detailed information regarding installation.

Starting Compressor (Water-cooled)

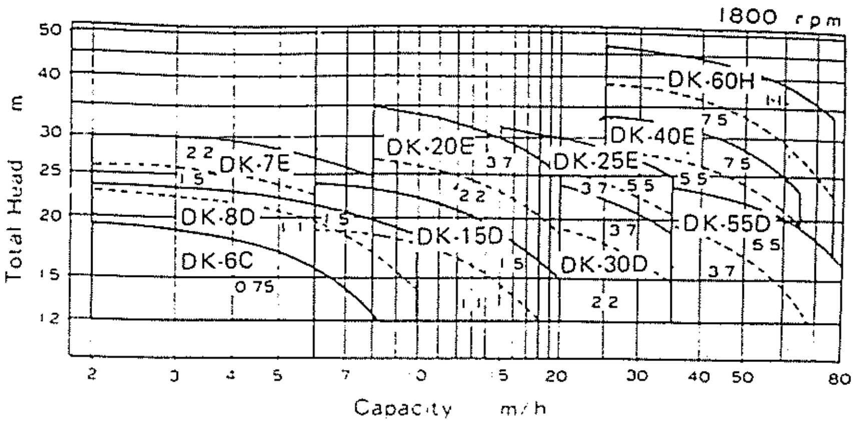
Model No.	Speed (r.p.m.)	25kgf/cm²			30kgf/cm²		
		Capacity m^3/hr (FA)	Power required PS	Motor (kW)	Capacity m^3/hr (FA)	Power required PS	Motor (kW)
H-63	1,200	65	16.5	15	60	17	15
	1,500	80	20	18.5	75	21	18.5
	1,800	95	25	22	90	26.5	22
H-64	1,200	90	23	18.5	85	24	18.5
	1,500	110	29	25	105	30	25
	1,800	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

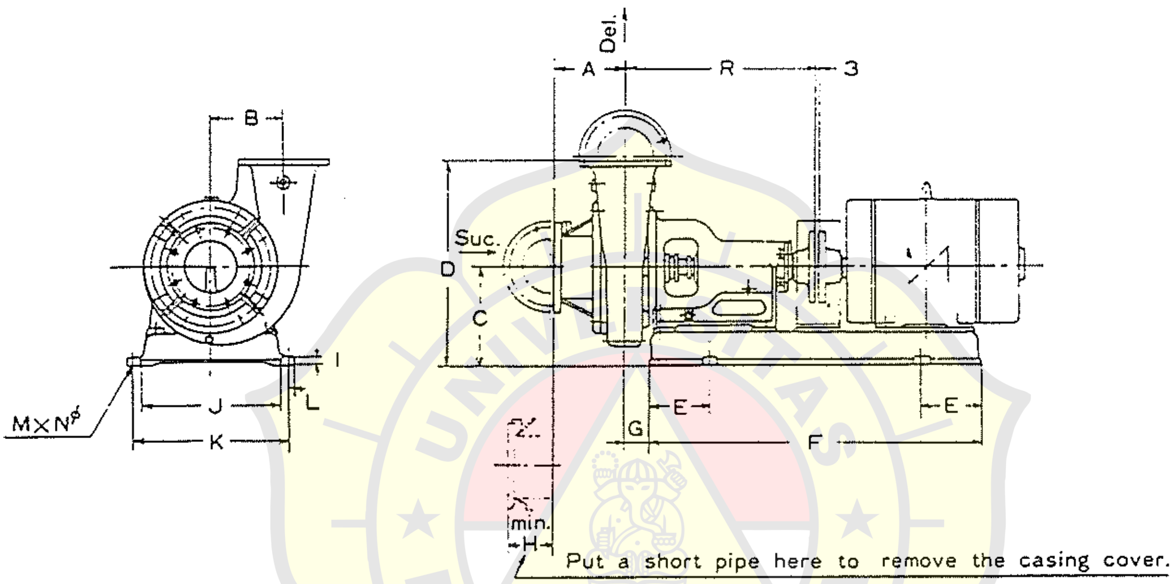
LEGEND: Capacity (free air) referred to inlet condition, measured according to vessel charging test method.

Emergency Compressor (Vertical 2-stage Air-cooled)

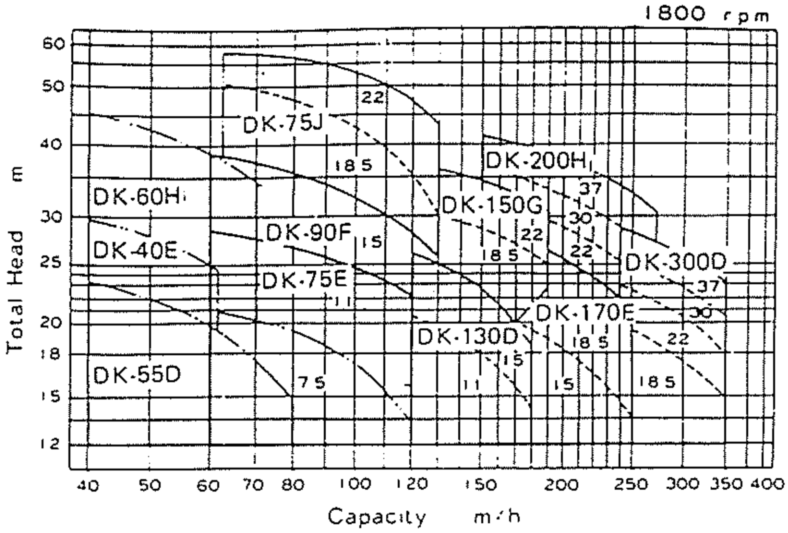
Model No.	Speed (r.p.m)	15kg/cm²		25 ~ 30kg/cm²	
		m^3/hr FA	PS	m^3/hr FA	PS
LSHC-20B	900	4.7	1.4	4.3	1.6
	1000	5.2	1.5	5.0	1.7
LSHC-30A	900	13.5	4.8	12.8	5.3
	1000	14.8	5.3	13.8	5.8
LSHC-40A	900	20.4	7.2	19.4	8.0
	1000	22.3	7.9	21.2	8.9

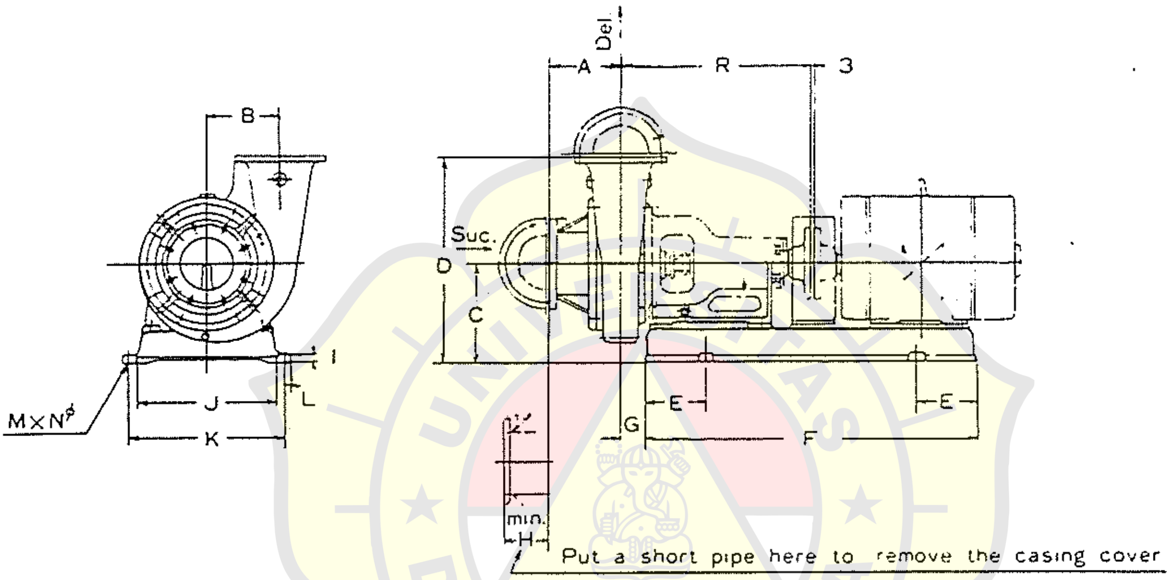
DK Type



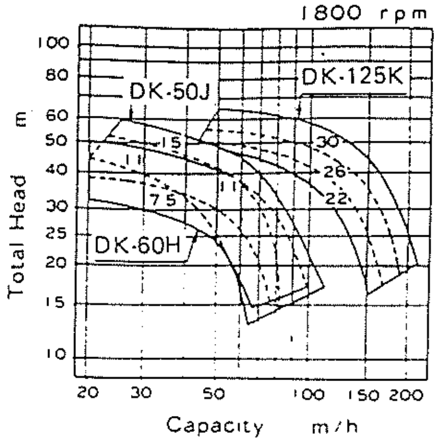


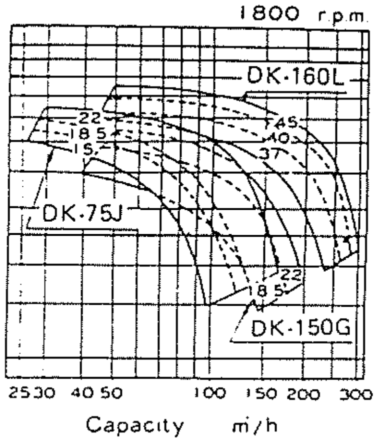
Type	Motor (kw)	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	R	Pump Weight (kg)
K-6C	0.75	32	32	96	110	165	315	100	550	35	100	25	200	240	23	4	15	350	50
K-8D	1.5			100	130	165	315	100	600	30	100	25	200	240	23	4	15	350	55
K-7E	1.5	32	32	100	135	165	335	100	600	30	100	25	200	240	23	4	15	350	60
	2.2			100	135	175	345	100	620	35	100	25	240	280	23	4	15	350	60
K-15D	1.5	50	50	108	125	165	325	100	600	30	100	25	200	240	23	4	15	350	95
K-20E	2.2	50	50	108	160	175	365	100	620	35	100	25	240	280	23	4	15	350	115
	3.7			108	160	190	380	100	650	30	100	25	260	300	23	4	15	350	115
K-25E	3.7	65	65	110	154	190	370	125	700	28	100	25	300	340	23	4	15	370	145
	5.5			110	154	210	390	125	700	28	100	25	300	340	23	4	15	370	145
K-30D	2.2	65	65	104	150	175	365	100	620	35	100	25	240	280	23	4	15	350	120
	3.7			104	150	190	380	100	650	30	100	25	260	300	23	4	15	350	120
K-55D	3.7	100	100	113	160	190	390	125	700	28	100	25	300	340	23	4	15	370	190
	5.5			113	160	210	410	125	700	28	100	25	300	340	23	4	15	370	190
K-40E	5.5	100	100	112	165	201	410	125	700	30	100	25	300	340	23	4	15	372	180
	7.5			112	165	201	410	150	750	30	100	25	300	340	23	4	15	372	180
K-60H	7.5	100	100	117	177	210	440	150	780	32	100	25	300	340	23	4	15	422	210
	11			117	177	235	465	175	860	32	100	25	360	400	23	4	15	422	210

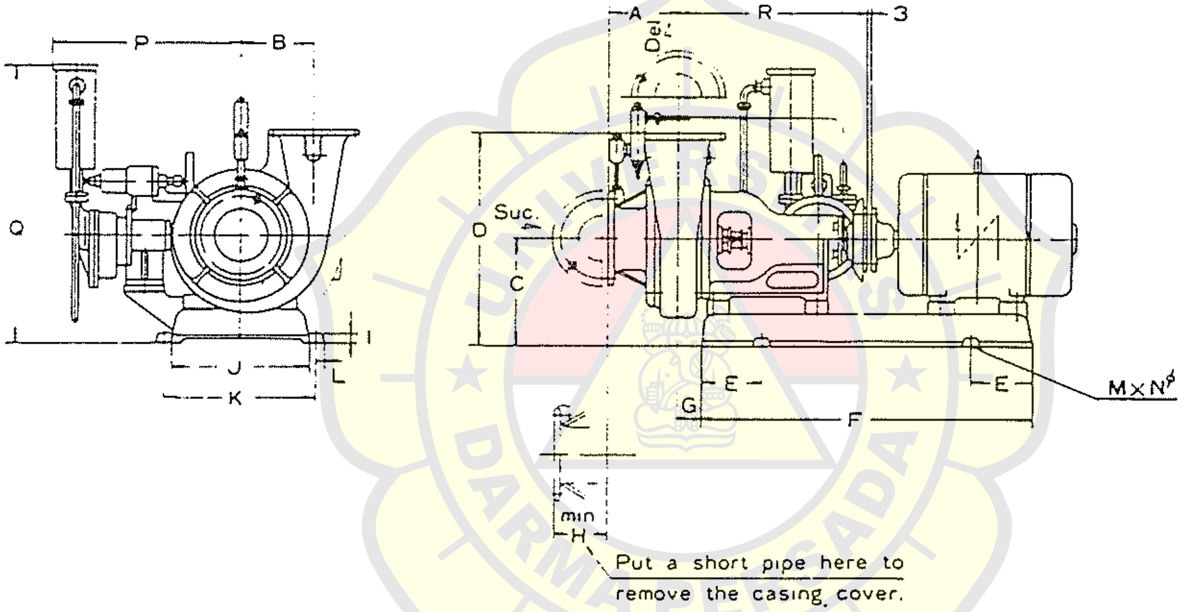




ype	Motor (kw)	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	R	Pump Weight (kg)
5E	7.5	125	125	113	165	202	460	150	780	30	100	25	300	340	23	4	15	420	160
	11			113	165	227	485	175	860	30	100	25	360	400	23	4	15	420	160
0F	15	125	125	150	180	231	505	175	900	33	100	25	360	400	23	4	15	423	210
5J	18.5	125	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
30D	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
50G	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
70E	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
00H	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
00D	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

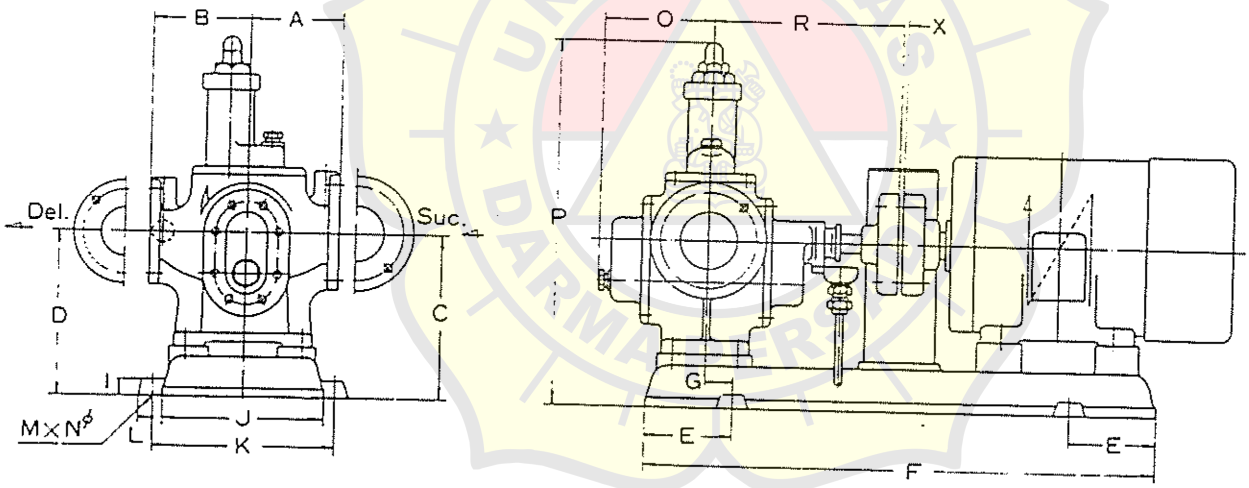






ype	Motor (kw)	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	P	Q	R	Vacuum Pump	Pump Weight (kg)
50J	11	100	100	140	220	240	520	150	900	30	100	25	360	400	23	4	15	405	705	420	V-18	250
	15			140	220	240	520	150	900	30	100	25	360	400	23	4	15	405	705	420	V-18	250
50H	7.5	100	100	117	177	240	470	150	900	32	100	25	360	400	23	4	15	405	705	422	V-18	240
	11			117	177	240	470	150	900	32	100	25	360	400	23	4	15	405	705	422	V-18	240
75J	15	125	125	156	205	300	590	200	1020	35	120	30	390	430	25	4	23	405	765	480	V-50	270
	18.5			156	205	300	590	200	1020	35	120	30	390	430	25	4	23	405	765	480	V-50	270
	22			156	205	300	590	200	1020	35	120	30	390	430	25	4	23	405	765	480	V-50	270
50G	18.5	150	150	160	200	295	610	200	1050	53	120	30	450	500	25	4	23	420	775	548	V-50	280
	22			160	200	295	610	200	1050	53	120	30	450	500	25	4	23	420	775	548	V-50	280
25K	22	150	150	160	230	303	610	200	1050	60	120	30	450	500	25	4	23	420	775	555	V-50	280
	30			160	230	303	610	200	1100	60	120	30	450	500	25	4	23	420	775	555	V-50	280
60L	37	150	150	159	235	375	695	250	1340	68	100	30	490	550	28	4	23	420	840	760	V-50	430
	45			159	235	375	695	250	1340	68	100	30	490	550	28	4	23	420	840	760	V-50	430

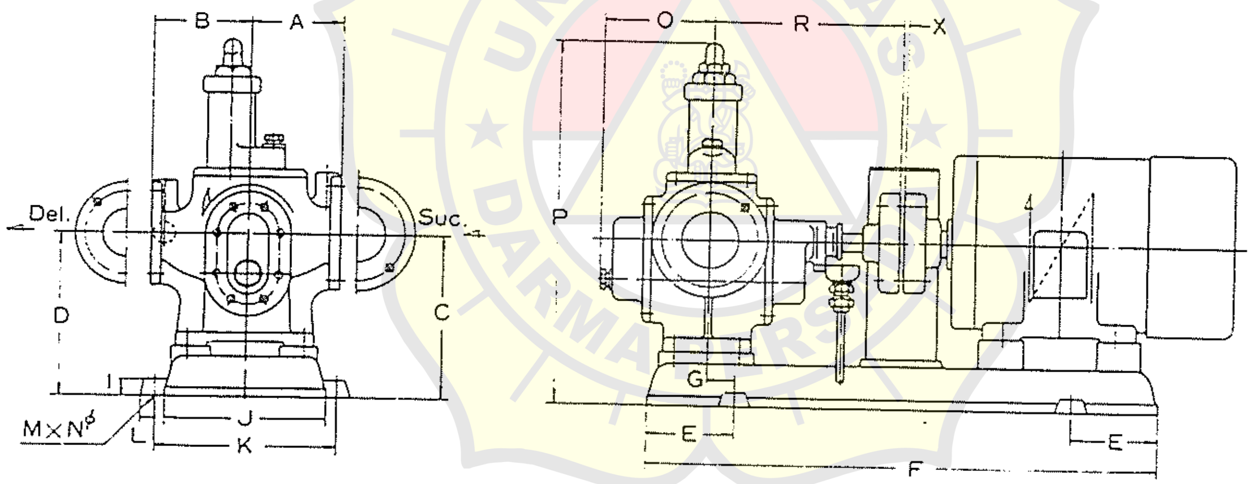
MA Type



Dimensions—mm

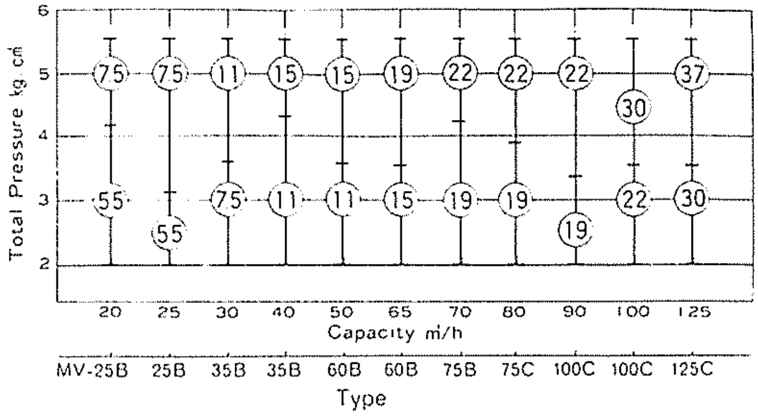
Type	Motor (kw)	Bore Suc.	Bore Del	A	B	C	D	E	F	G	I	J	K	L	M	N	O	P	R	X	Pump Weight (kg)
2B	1.5 / 2.2	40	32	105	105	210	210	100	550	35	25 / 22	260 / 280	300 / 320	23 / 25	4	15	98	412	185	3	65
3B	2.2	40	32	103	109	225	225	100	600	20	25	280	320	23	4	15	114	430	200	3	75
4B	2.2 / 3.7	50	40	105	105	225	225	100	600 / 630	20	25	280	320	23	4	15	119	470	200	3	77
[illegible]B	3.7	65	50	105	105	230	230	130	640	60	25	300	330	25	4	15	119	475	200	3	77
[illegible]B	3.7	65	50	105	105	225	225	110	640	45	25	310	350	23	4	15	230	470	220	3	77
[illegible]B	3.7 / 5.5	80	65	110	110	230	230 / 230	125 / 150	700 / 750	30 / 50	25	310	350	23	4	15	132	485	245	3	115
[illegible]0B	5.5 / 7.5	80	65	150	150	260 / 270	260 / 270	100 / 150	750 / 950	20 / 65	25 / 30	310 / 350	350 / 390	23 / 25	4	15 / 19	165	566 / 576	285	3	120
[illegible]5B	11 / 15	80	65	160	160	290	290	200	950 / 1000	100	30	350 / 400	400 / 450	25	4	19	193	648	315	3	135

MA Type

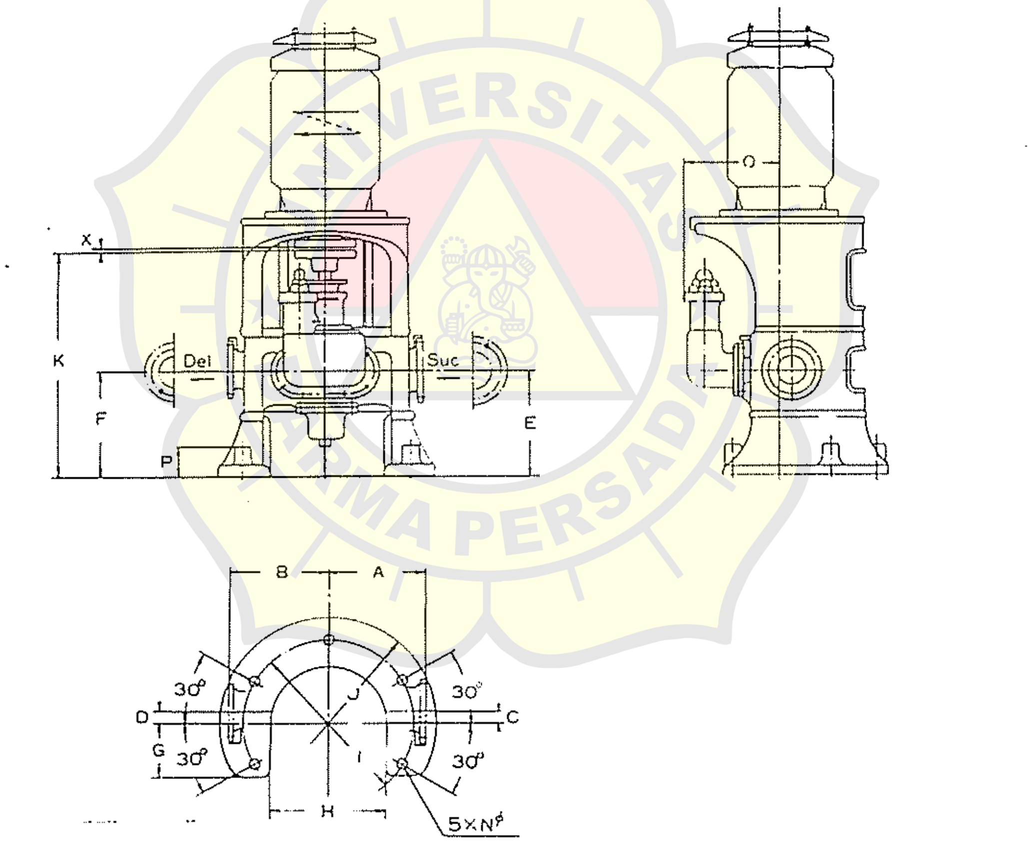


Dimensions—mm

pe	Motor (kw)	Bore Suc.	Bore Del	A	B	C	D	E	F	G	I	J	K	L	M	N	O	P	R	X	Pump Weight (kg)
2B	1.5/2.2	40	32	105	105	210	210	100	550	35	25/22	260/280	300/320	23/25	4	15	98	412	185	3	65
3B	2.2	40	32	103	109	225	225	100	600	20	25	280	320	23	4	15	114	430	200	3	75
4B	2.2/3.7	50	40	105	105	225	225	100	600/630	20	25	280	320	23	4	15	119	470	200	3	77
5B	3.7	65	50	105	105	230	230	130	640	60	25	300	330	25	4	15	119	475	200	3	77
6B	3.7	65	50	105	105	225	225	110	640	45	25	310	350	23	4	15	230	470	220	3	77
3B	3.7/5.5	80	65	110	110	230	230/230	125/150	700/750	30/50	25	310	350	23	4	15	132	485	245	3	115
0B	5.5/7.5	80	65	150	150	260/270	260/270	100/150	750/950	20/65	25/30	310/350	350/390	23/25	4	15/19	165	566/576	285	3	120
5B	11/15	80	65	160	160	290	290	200	950/1000	100	30	350/400	400/450	25	4	19	193	648	315	3	135

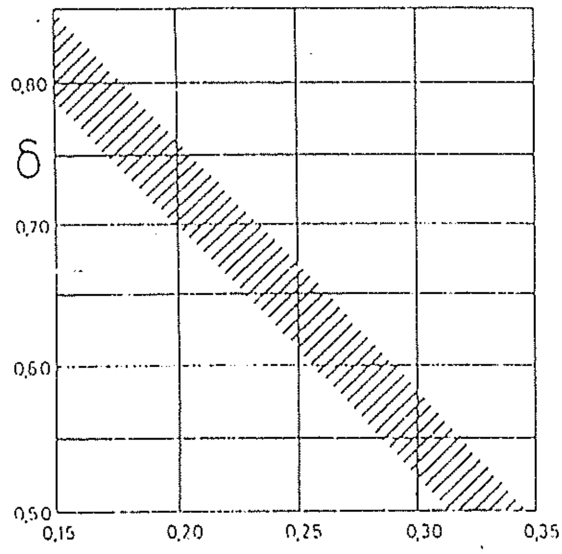


The number in ○ mark indicates the output (kw) of the motor when 260cSt oil is used.

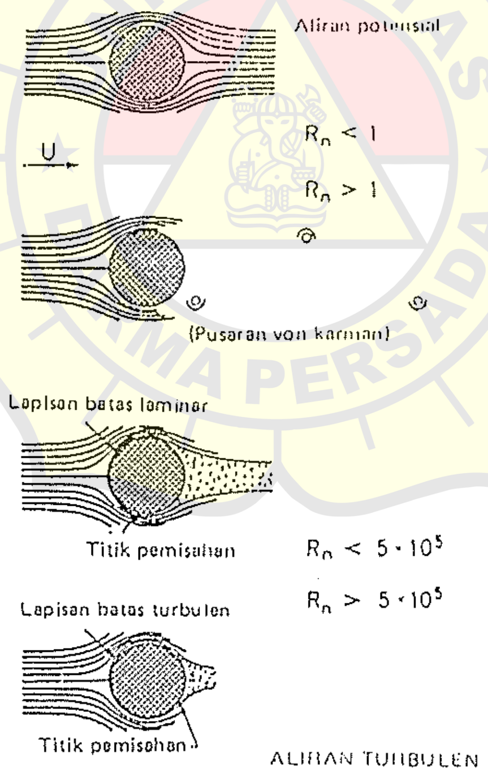


Dimensions — mm

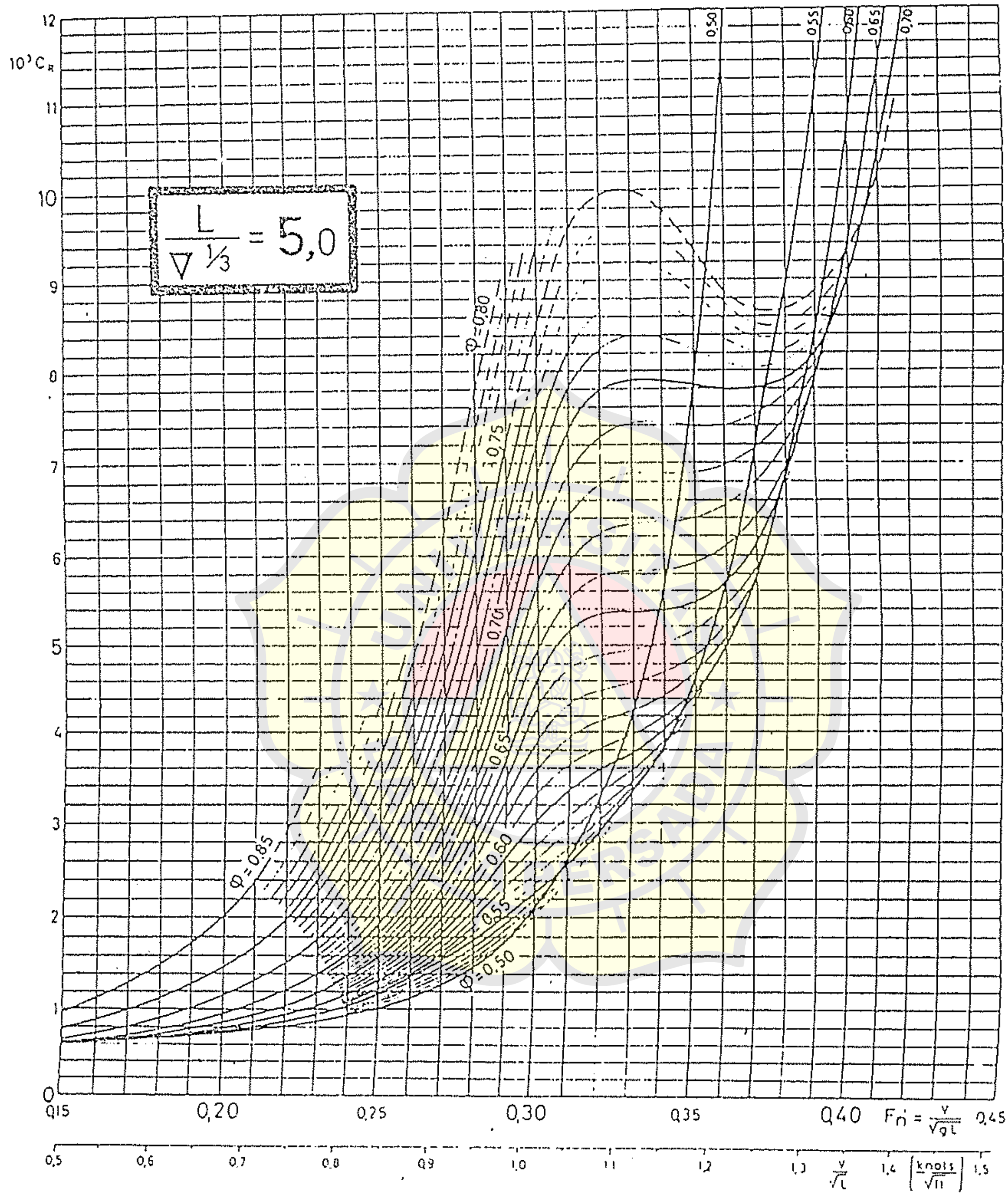
Type	No. of Rev. (r.p.m.)	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	H	I	J	K	N	O	P	X	Pump Weight (kg)
MV-35B	1200	125	100	345	345	35	35	350	350	206	456	685	760	715	27	355	115	3	420
MV-60B	1200	150	125	345	345	48	48	390	390	206	456	685	760	839	27	380	115	4	490
MV-75B	1200	150	125	405	405	48	48	405	405	206	456	685	760	870	25	395	115	4	510
MV-75C	900	150	125	430	430	48	48	560	560	240	490	780	860	1088	27	475	125	4	710
MV-100C	900	200	150	430	430	54	54	730	730	280	490	900	1000	1265	30	475	125	4	930
MV-125C	900	200	150	430	430	58	58	730	730	280	490	900	1000	1386	27	500	125	4	1000



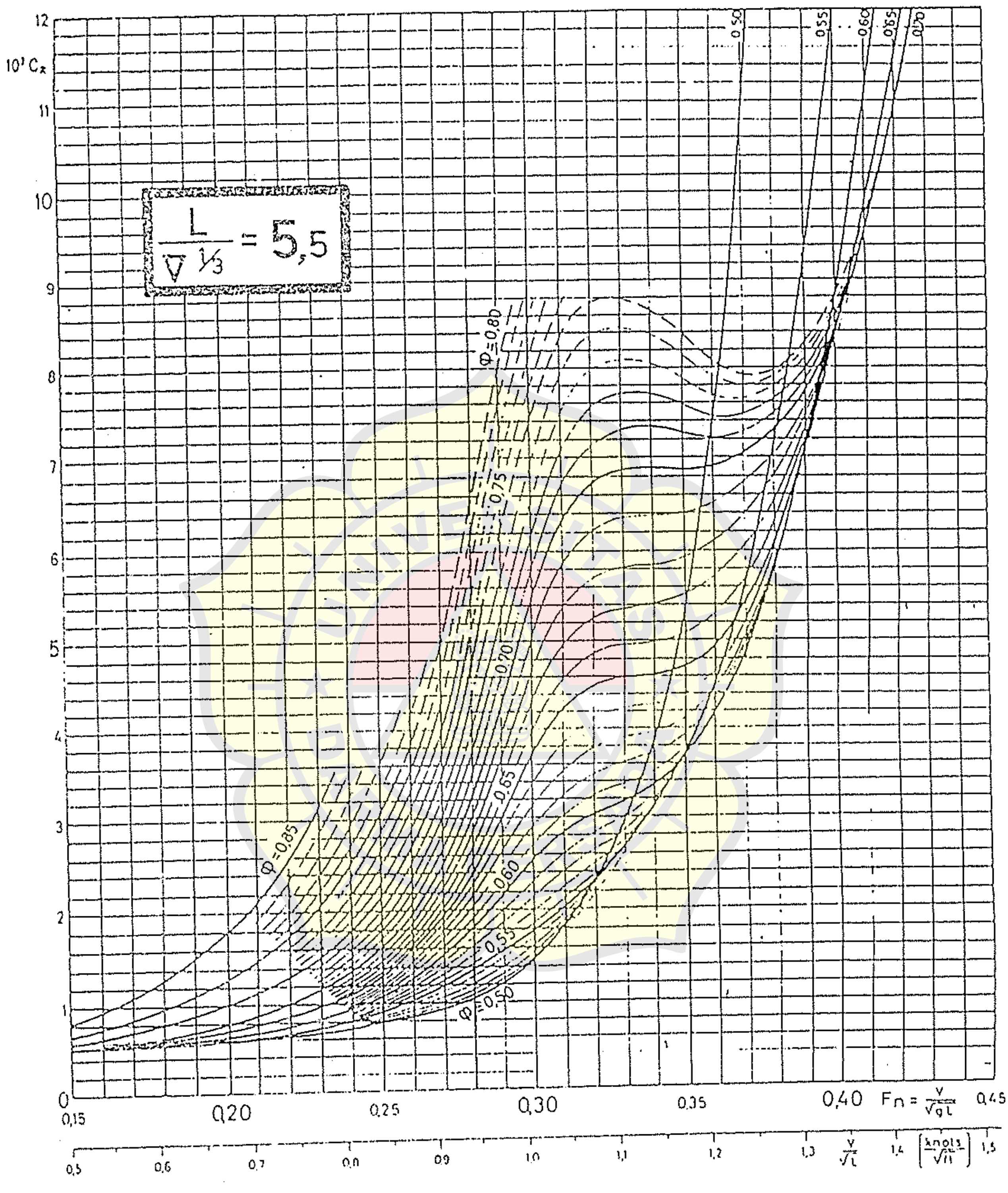
Gbr No.1 Koefisien Block (Cb) Standar



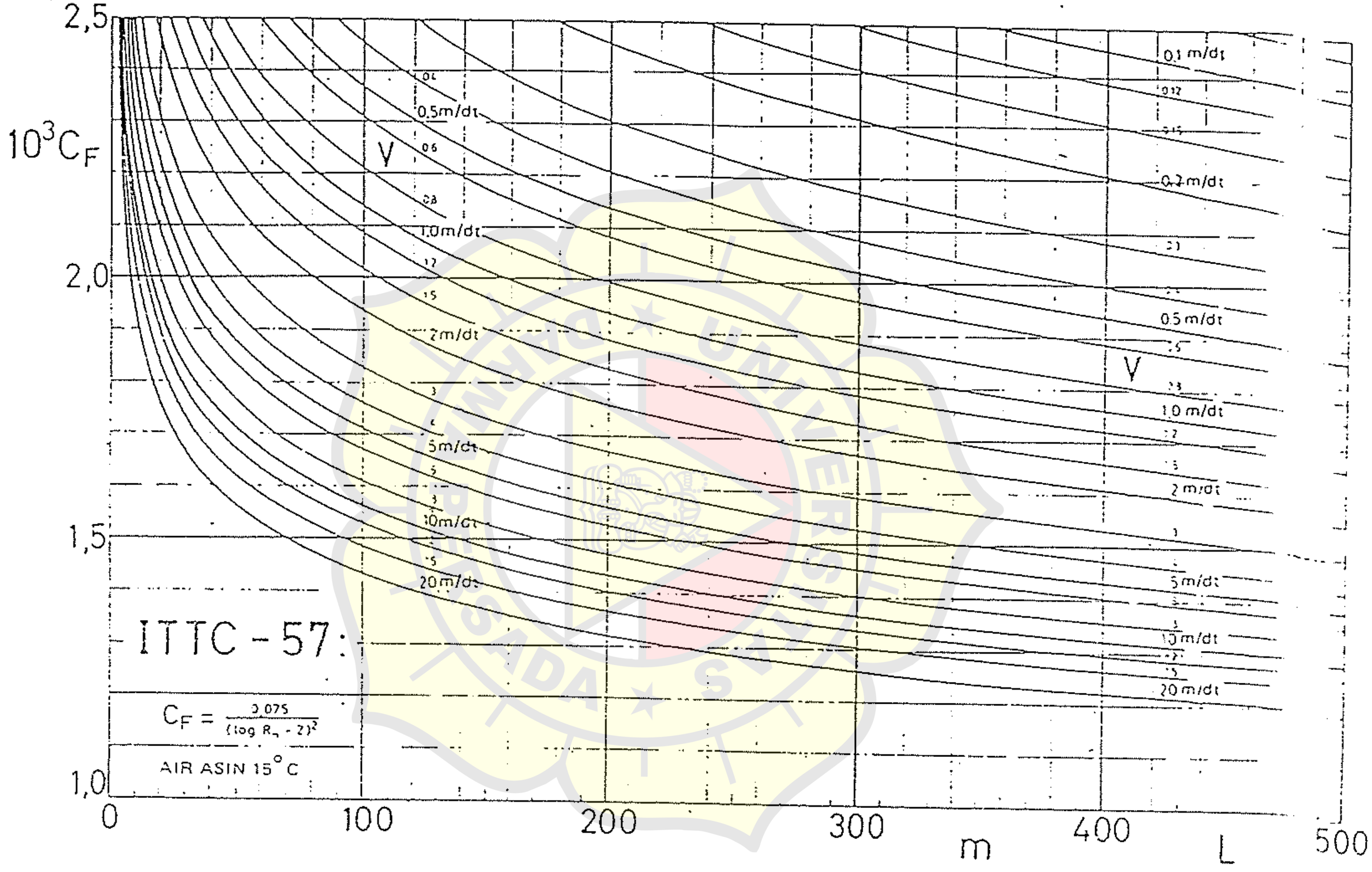
Tipe Aliran Sesuai Reynold Number (Rn)



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



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



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

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%

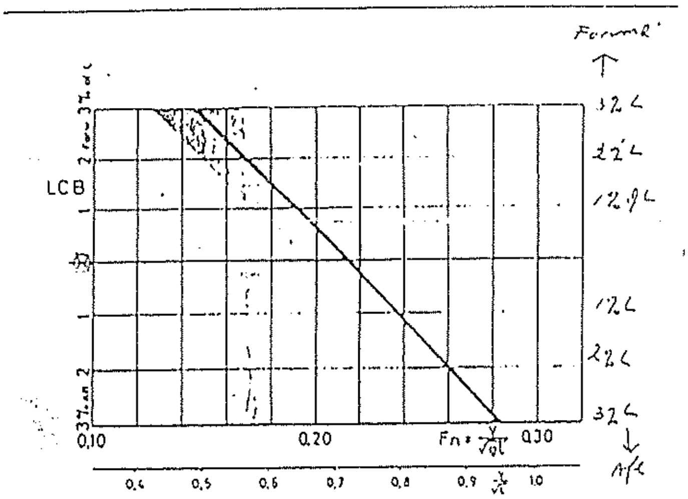
Gbr No.6 Koreksi Bentuk Anggota Badan

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

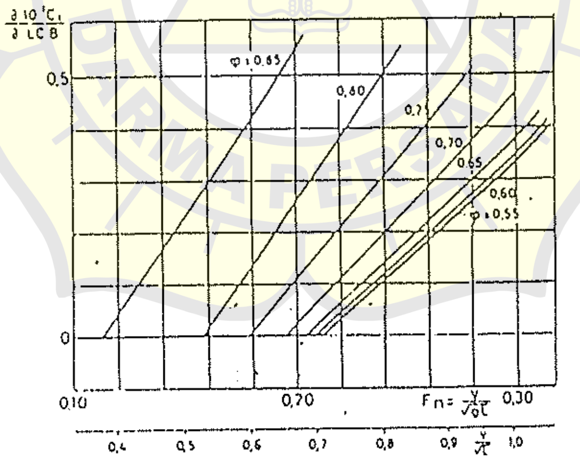
Gbr.No. 6 Koreksi Tahanan Tambahan

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

Gbr.No.4 Koreksi Bentuk Haluan



Gbr.No.3 LCB Standar



Gbr.No.3 Koreksi LCB

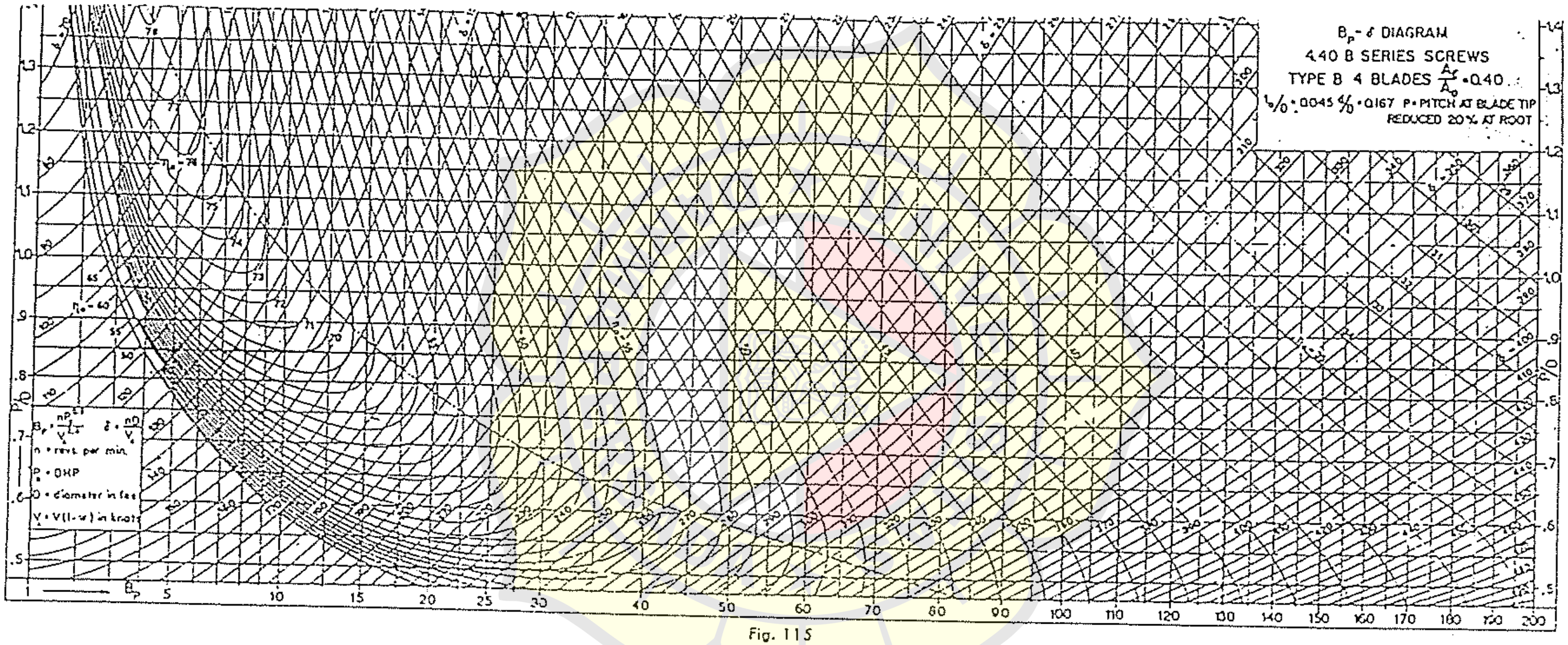
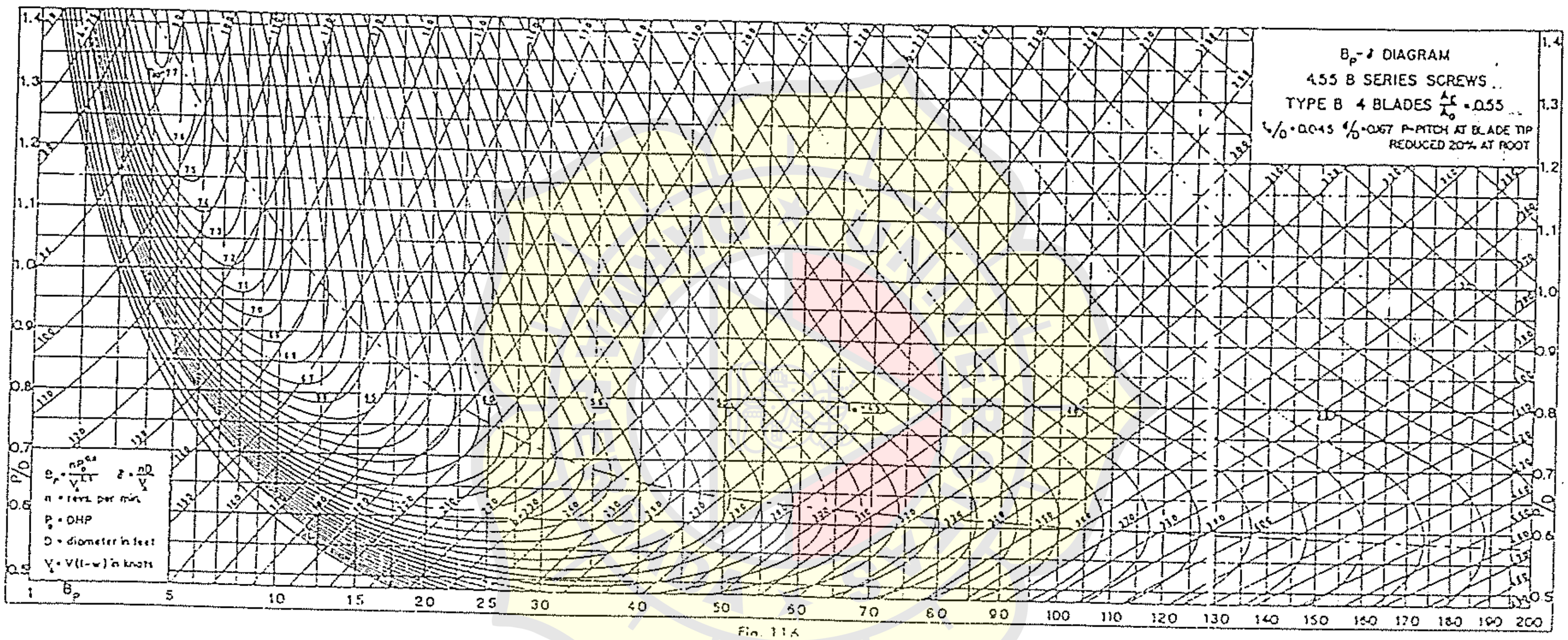


Fig. 115

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



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

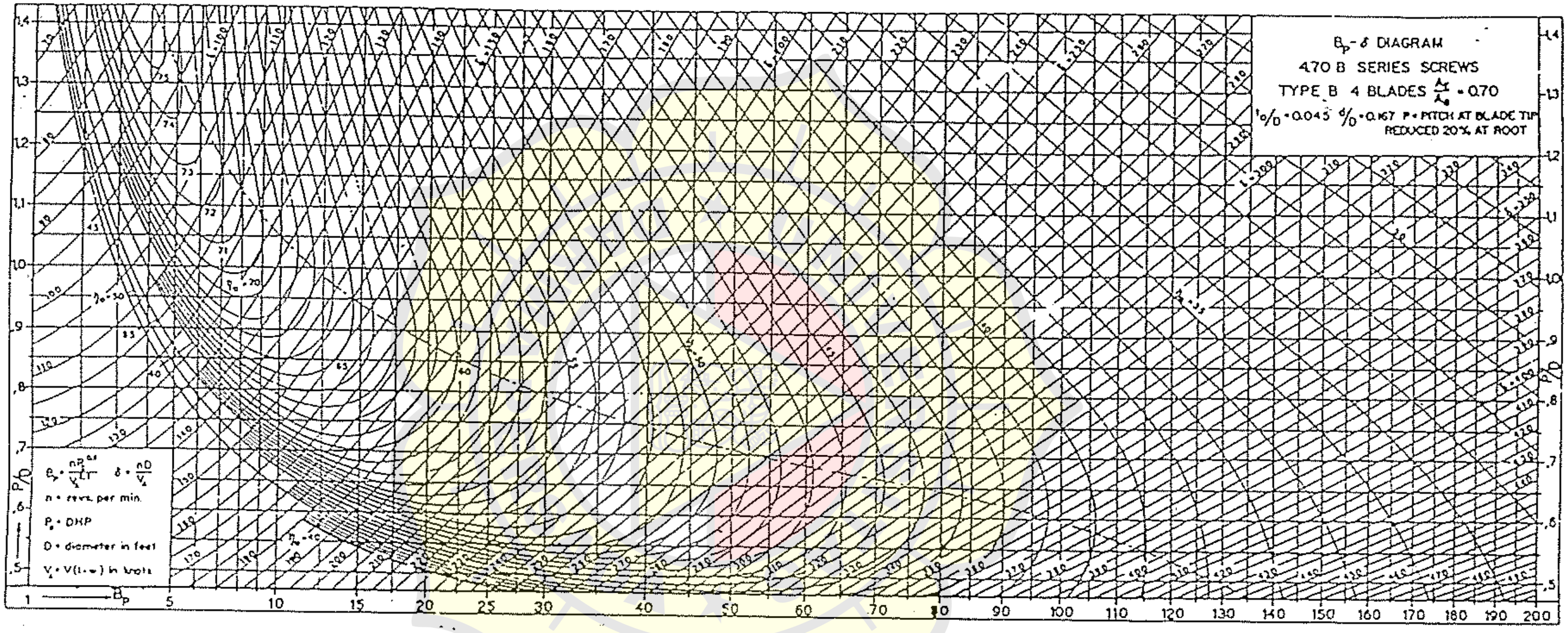
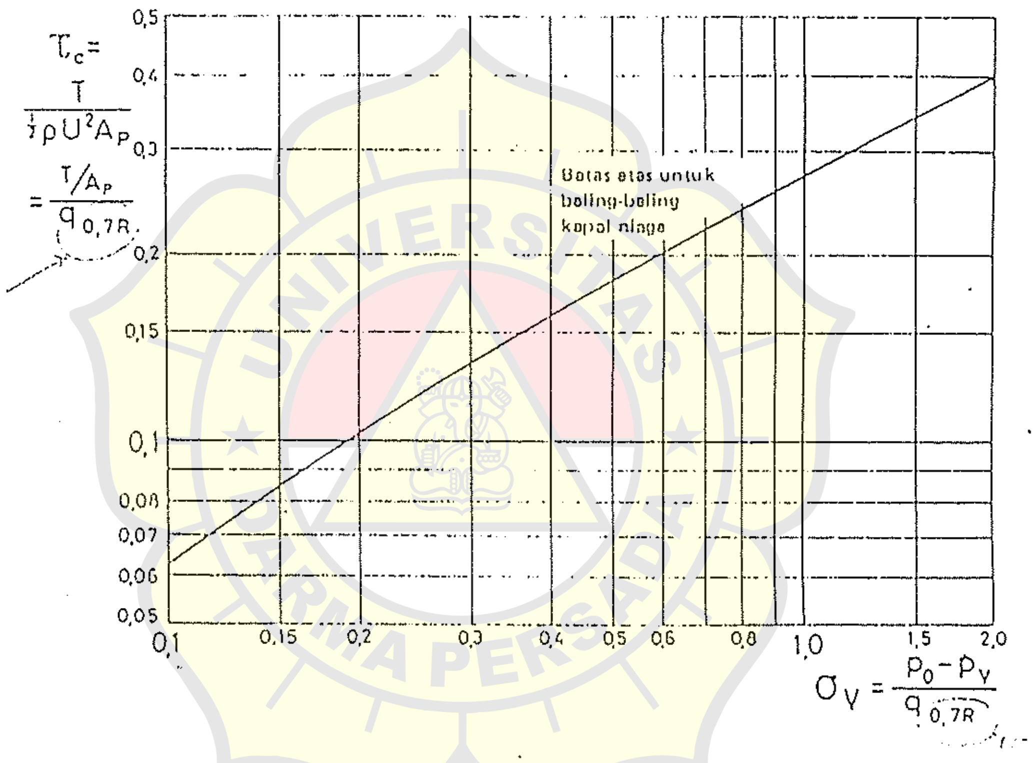


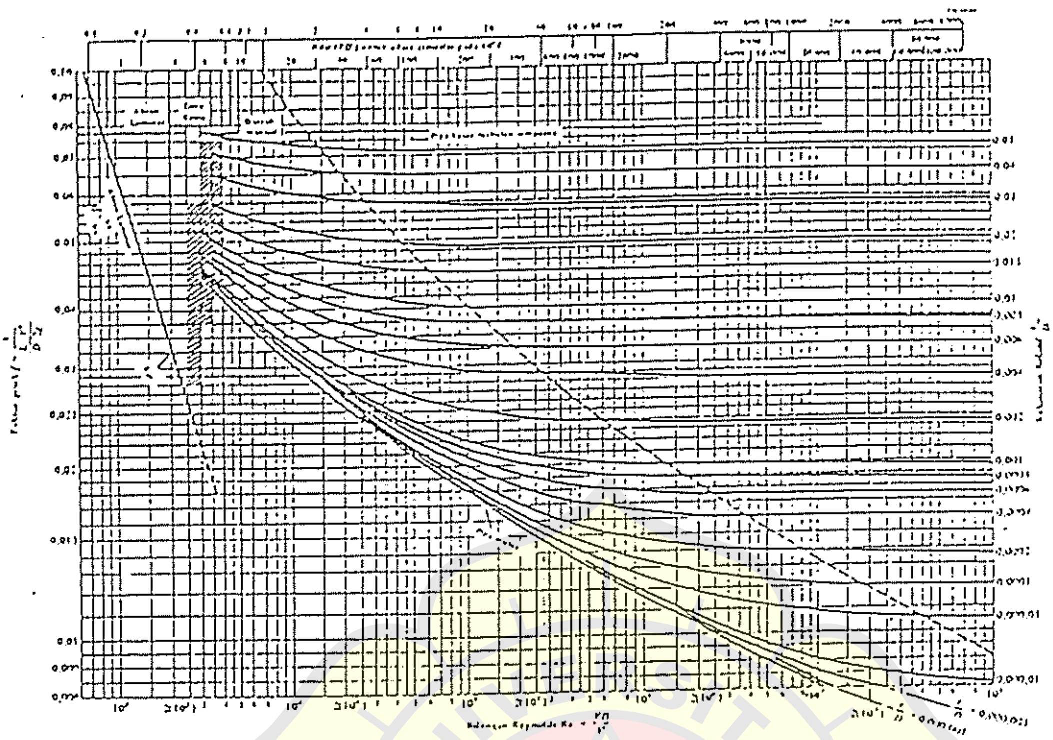
Fig. 117

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



Gbr.No. 2 Diagram Burril

LAMPIRAN 1



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

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

Zat cair	ρ, kg/m³	μ, (N·s)/m²	Υ, N/m†	p_v, N/m²	Modulus limbak, N/m²
Amonia	608	2.20 E−4	2.13 E−2	9.10 E+5	
Bensin	881	6.51 E−4	2.88 E−2	1.01 E+4	1.05 E+9
Karbon tetrakhlorida	1,590	9.67 E−4	2.70 E−2	1.20 E+4	9.65 E+8
Etanol	789	1.20 E−3	2.28 E−2	5.7 E+3	8.96 E+8
Gasolin	680	2.92 E−4	2.16 E−2	5.51 E+4	9.58 E+8
Gliserin	1,260	1.49	6.33 E−2	1.4 E−2	4.34 E+9
Minyak tanah	804	1.92 E−3	2.8 E−2	3.11 E+3	1.43 E+9
Air-raksa	13,550	1.56 E−3	4.84 E−1	1.1 E−3	2.55 E+10
Metanol	791	5.98 E−4	2.25 E−2	1.34 E+4	8.27 E+8
Pelumas SAE 10	917	1.04 E−1	3.6 E−2		1.31 E+9
Pelumas SAE 30	917	2.90 E−1	3.5 E−2		1.38 E+9
Air	998	1.00 E−3	7.28 E−2	2.34 E+3	2.19 E+9
Air laut	1,025	1.07 E−3	7.28 E−2	2.34 E+3	2.28 E+9

† Bersentuhan dengan udara.

LAMPIRAN 2

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

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

† 1 kg/(m·s) = 0,0209 slug/(ft·s); 1 m³/s = 10,76 ft²/s.

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

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

BERAT JENIS BEBERAPA FLUIDA YANG LAZIM

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

LAMPIRAN 3

Pumps

Hose diameter d_h, mm	Hose length l_h, m	Nozzle orifice diameter d_n, mm: 10	13	16	19	Hose diameter d_h, mm	Hose length l_h, m	Nozzle orifice diameter d_n, mm: 10	13	16	19
		Characteristic B						Characteristic B			
50	0	0.121	0.346	0.793	1.577	65	0	0.121	0.346	0.793	1.577
	10	0.119	0.331	0.722	1.320		10	0.1205	0.342	0.776	1.51
	20	0.118	0.318	0.622	1.130		20	0.120	0.339	0.758	1.44
	40	0.114	0.304	0.568	0.882		40	0.1195	0.332	0.726	1.33
	60	0.111	0.274	0.498	0.723		60	0.1185	0.326	0.696	1.23
	80	0.108	0.257	0.442	0.612		80	0.118	0.320	0.669	1.15
	100	0.105	0.241	0.398	0.531		100	0.117	0.314	0.644	1.08

Tank capacity, tons	Inside diameter of pipe and fittings, mm	Tank capacity, tons	Inside diameter of pipe and fittings, mm
Up to 20	60	265 to 360	125
20 to 40	70	360 to 480	140
40 to 75	80	480 to 620	150
75 to 120	90	620 to 800	160
120 to 190	100	800 to 1000	175
190 to 265	110	1000 to 1300	200

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

LAMPIRAN 4

Jangkar, rantai dan tali

No. urut Reg	Angka penunjuk Z	Jangkar tanpa tongkat: Jangkar jumlah	Jangkar tanpa tongkat: Berat satu jangkar (kg)	Jangkar tanpa tongkat: Jangkar arus (kg)	Rantai jangkar haluan: Panjang total (m)	Diameter d_1 (mm)	Diameter d_2 (mm)	Diameter d_3 (mm)	Kawat atau rantai arus: Panjang (m)	Kawat atau rantai arus: Beban putus (kg)	Tali tarik: Panjang (m)	Tali tarik: Beban putus (kg)	Tali tambat: Jumlah	Tali tambat: Panjang (m)	Tali tambat: Beban putus (kg)
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
101	50	2	120	40	165	12,5			80	6 000	180	10 000	2	100	3 500
102	51 – 70	2	180	60	220	14	12,5		85	6 600	180	10 000	2	100	3 500
103	71 – 90	2	240	80	220	16	14		85	7 500	180	10 000	2	100	3 750
104	91 – 110	2	300	100	247,5	17,5	16		90	8 300	180	10 000	2	110	4 000
105	111 – 130	2	360	120	247,5	19	17,5		90	9 100	180	10 000	2	110	4 500
106	131 – 150	2	420	140	275	20,5	17,5		90	10 000	180	10 000	2	120	5 000
107	150 – 175	2	480	165	275	22	19		90	11 000	180	10 000	2	120	5 500
108	175 – 205	2	570	190	302,5	24	20,5		90	12 000	180	11 400	2	120	6 000
109	205 – 240	3	660		302,5	26	22				180	13 200	2	120	6 000
110	240 – 280	3	780		330	28	24				180	15 300	3	120	7 250
111	280 – 320	3	900		357,5	30	26				180	17 700	3	140	8 000
112	320 – 360	3	1 020		357,5	32	28				180	21 100	3	140	8 750
113	360 – 400	3	1 140		385	34	30				180	22 800	3	140	9 500
114	400 – 450	3	1 290		385	36	32				180	25 500	3	140	10 250
115	450 – 500	3	1 440		412,5	38	34				180	28 200	3	140	11 000
116	500 – 550	3	1 590		412,5	40	34				190	31 200	3	160	11 500
117	550 – 600	3	1 740		440	42	36				190	34 500	4	160	12 000
118	600 – 660	3	1 920		440	44	38				190	37 800	4	160	12 500
119	660 – 720	3	2 100		440	46	40				190	41 400	4	160	13 000
120	720 – 780	3	2 280		467,5	48	42				190	45 000	4	170	13 500
121	780 – 840	3	2 460		467,5	50	44				190	48 900	4	170	14 000
122	840 – 910	3	2 640		467,5	52	46	40			190	52 800	4	170	14 500
123	910 – 980	3	2 850		495	54	48	42			190	57 000	4	170	15 000
124	980 – 1 060	3	3 060		495	56	50	44			200	61 500	4	180	16 000
125	1 060 – 1 140	3	3 300		495	58	50	46			200	66 000	4	180	17 000
126	1 140 – 1 220	3	3 540		522,5	60	52	46			200	70 500	4	180	18 000
127	1 220 – 1 300	3	3 780		522,5	62	54	48			200	75 300	4	180	19 000
128	1 300 – 1 390	3	4 050		522,5	64	56	50			200	80 100	4	180	20 000
129	1 390 – 1 480	3	4 320		550	66	58	50			200	85 200	4	180	21 000
130	1 480 – 1 570	3	4 590		550	68	60	52			220	90 000	5	190	22 000
131	1 570 – 1 670	3	4 890		550	70	62	54			220	96 000	5	190	23 000
132	1 670 – 1 790	3	5 250		577,5	73	64	56			220	104 400	5	190	24 000
133	1 790 – 1 930	3	5 610		577,5	76	66	58			220	113 100	5	190	25 000
134	1 930 – 2 080	3	6 000		577,5	78	68	60			220	119 100	5	190	26 000
135	2 080 – 2 230	3	6 450		605	81	70	62			240	128 400	5	200	27 000
136	2 230 – 2 380	3	6 900		605	84	73	64			240	138 300	5	200	28 000
137	2 380 – 2 530	3	7 350		605	87	76	66			240	148 200	5	200	29 000
138	2 530 – 2 700	3	7 800		632,5	90	78	68			260	150 000	6	200	30 000
139	2 700 – 2 870	3	8 300		632,5	92	81	70			260	150 000	6	200	31 000
140	2 870 – 3 040	3	8 700		632,5	95	84	73			260	150 000	6	200	32 000
141	3 040 – 3 210	3	9 300		660	97	84	76			280	150 000	6	200	33 000
142	3 210 – 3 400	3	9 900		660	100	87	78			280	150 000	6	200	34 000
143	3 400 – 3 600	3	10 500		660	102	90	78			280	150 000	6	200	35 000
144	3 600 – 3 800	3	11 100		687,5	105	92	81			300	150 000	6	200	36 000
145	3 800 – 4 000	3	11 700		687,5	107	95	84			300	150 000	6	200	37 000
146	4 000 – 4 200	3	12 300		687,5	111	97	87			300	150 000	7	200	38 000
147	4 200 – 4 400	3	12 900		715	114	100	87			300	150 000	7	200	39 000
148	4 400 – 4 600	3	13 500		715	117	102	90			300	150 000	7	200	40 000
149	4 600 – 4 800	3	14 100		715	120	105	92			300	150 000	7	200	41 000
150	4 800 – 5 000	3	14 700		742,5	122	107	95			300	150 000	7	200	42 000
151	5 000 – 5 200	3	15 400		742,5	124	111	97			300	150 000	8	200	44 000
152	5 200 – 5 500	3	16 100		742,5	127	111	97			300	150 000	8	200	46 000
153	5 500 – 5 800	3	16 900		742,5	130	114	100			300	150 000	8	200	48 000
154	5 800 – 6 100	3	17 800		742,5	132	117	102			300	150 000	9	200	50 000
155	6 100 – 6 500	3	18 800		742,5		120	107			300	150 000	10	200	50 000
156	6 500 – 6 900	3	20 000		770		124	111			300	150 000	11	200	50 000
157	6 900 – 7 400	3	21 500		770		127	114			300	150 000	12	200	50 000
158	7 400 – 7 900	3	23 000		770		132	117			300	150 000	13	200	50 000
159	7 900 – 8 400	3	24 500		770		137	122			300	150 000	14	200	50 000
160	8 400 – 8 900	3	26 000		770		142	127			300	150 000	15	200	50 000
161	8 900 – 9 400	3	27 500		770		147	132			300	150 000	16	200	50 000
162	9 400 – 10000	3	29 000		770		152	132			300	150 000	18	200	50 000

LAMPIRAN 5

Mooring and Warping Ropes

Characteristic	Towing rope			Warping hawsers							
								Cable warps			
	Length, m	Circumference of hemp rope, mm	Diameter of steel rope, mm	Total length, m	Number of ropes	Circumference of hemp rope, mm	Diameter of steel rope, mm	Total length, m	Number of ropes	Circumference of hemp rope, mm	Diameter of steel rope, mm
50	50	75	—	50	1	65	—	—	—	—	—
75	50	90	11	50	1	65	—	—	—	—	—
100	75	90	11	75	1	65	8.5	—	—	—	—
150	75	100	12	75	1	75	9.5	—	—	—	—
200	100	100	12	100	2	75	9.5	—	—	—	—
250	100	125	15	140	2	100	12	—	—	—	—
300	110	125	15	160	2	100	12	—	—	—	—
350	110	150	17.5	160	2	100	12	—	—	—	—
400	135	150	17.5	180	2	125	15	80	1	100	12
450	135	150	17.5	180	2	125	15	80	1	100	12
500	135	150	17.5	200	2	125	15	85	1	100	12
550	135	175	19.5	200	2	125	15	85	1	100	12
600	135	175	19.5	220	2	150	17.5	90	1	100	12
650	135	175	19.5	240	2	150	17.5	90	1	100	12
700	150	200	21.5	240	2	150	17.5	90	1	100	12
750	150	200	21.5	360	4	150	17.5	90	1	125	15
800	150	200	21.5	360	4	150	17.5	90	1	125	15
850	175	200	21.5	360	4	150	17.5	90	1	125	15
900	175	225	24	360	4	175	19.5	120	2	125	15
950	175	225	24	360	4	175	19.5	120	2	125	15
1000	175	225	24	360	4	175	19.5	120	2	150	17.5
1100	175	225	24	360	4	175	19.5	140	2	150	17.5
1200	190	250	26	360	4	175	19.5	140	2	150	17.5
1300	190	250	26	400	4	200	21.5	150	2	150	17.5
1400	190	275	28	400	4	200	21.5	150	2	150	17.5
1500	190	275	28	450	4	200	21.5	150	2	150	17.5
1600	200	300	30	480	4	200	21.5	180	2	150	17.5
1700	200	300	30	480	4	200	21.5	180	2	150	17.5
1850	200	325	32.5	540	4	200	21.5	180	2	175	19.5
2000	200	350	34.5	540	4	200	21.5	180	2	175	19.5
2150	200	350	34.5	540	4	200	21.5	180	2	175	19.5
2300	220	350	34.5	540	4	225	24	180	2	175	19.5
2500	220	350	34.5	640	4	225	24	200	2	175	19.5
2700	220	350	34.5	640	4	225	24	200	2	200	21.5
3000	220	350	34.5	640	4	225	24	200	2	200	21.5
3300	240	375	39	640	4	250	26	200	2	200	21.5
3600	240	375	39	640	4	250	26	200	2	200	21.5
3900	240	400	43.5	640	4	250	26	200	2	200	21.5
4200	240	400	43.5	640	4	250	26	200	2	225	24
4500	240	425	48.5	720	4	250	26	200	2	225	24
4800	240	425	48.5	720	4	250	26	200	2	225	24
5100	240	—	53	720	4	275	28	240	2	225	24
5400	240	—	53	800	4	275	28	240	2	250	26
5800	240	—	53	880	4	275	28	240	2	250	26
6200	240	—	57	960	6	300	30	240	2	250	26
6600	240	—	57	960	6	300	30	240	2	250	26
7000	240	—	57	960	6	300	30	240	2	250	26
7400	240	—	57	960	6	300	30	480	4	250	26
7800	240	—	57	960	6	300	30	480	4	250	26
8200	240	—	61.5	960	6	300	30	480	4	250	26
8600	240	—	61.5	960	6	325	32	480	4	250	26
9000	240	—	61.5	960	6	325	32	480	4	250	26
9600	240	—	61.5	960	6	325	32	480	4	250	26

LAMPIRAN 6

Self-Propelled Transport Ships with an Unlimited Region of Navigation

No.	Characteristic X	Anchors: Bower: Quantity	Anchors: Bower: Total weight, kg	Anchors: Stream anchor, kg	Chain cable for bower anchors: Total length of two cables, m	Chain cable for bower anchors: Anchor chain size, mm	Chain or steel rope for the stream anchor: Length, m	Chain or steel rope for the stream anchor: Anchor chain size, mm	Chain or steel rope for the stream anchor: Diameter of steel rope, mm
1	50	2	150	25	100	12	50	—	8.8
2	75	2	200	25	125	13	50	—	8.8
3	100	2	250	50	125	15	50	—	11
4	150	2	300	50	150	16	50	—	11
5	200	2	350	50	175	17	75	—	11
6	250	2	450	75	200	18	75	11	13
7	300	2	500	75	225	19	75	13	13
8	350	2	600	100	250	20	75	14	15.5
9	400	2	700	100	275	21	75	14	15.5
10	450	2	750	125	300	22	100	15	17.5
11	500	2	800	150	300	24	100	16	17.5
12	550	2	900	175	325	25	100	16	17.5
13	600	3	1500	200	350	27	100	17	17.5
14	650	3	1700	225	350	28	100	18	19.5
15	700	3	1800	250	375	29	100	18	20.5
16	750	3	2100	250	375	30	100	19	20.5
17	800	3	2250	250	375	31	125	19	20.5
18	850	3	2400	275	375	32	125	20	22
19	900	3	2700	300	375	33	125	21	24
20	950	3	3000	300	400	34	125	21	24
21	1000	3	3200	350	400	35	125	22	24
22	1100	3	3500	400	400	37	125	23	26
23	1200	3	3750	400	420	38	150	25	26
24	1300	3	4100	450	450	40	150	25	28
25	1400	3	4250	450	450	41	150	25	28
26	1500	3	4500	500	450	42	150	26	28
27	1600	3	4750	500	450	43	150	26	28
28	1700	3	5250	600	450	45	150	28	30
29	1850	3	5500	600	450	46	150	28	30
30	2000	3	5750	700	450	46	150	29	31.5
31	2150	3	6000	700	475	48	175	29	31.5
32	2300	3	6500	800	500	49	175	29	32.5
33	2500	3	6750	800	500	50	175	29	32.5
34	2700	3	7500	900	500	52	175	30	33.5
35	3000	3	8250	1000	500	53	200	31	33.5
36	3300	3	9000	1000	500	55	200	31	33.5
37	3600	3	9750	1250	525	57	200	33	34.5
38	3900	3	10500	1250	550	59	225	33	34.5
39	4200	3	11000	1400	550	61	225	34	37
40	4500	3	11500	1500	550	62	225	35	37
41	4800	3	12900	1650	550	65	225	36	—
42	5100	3	13500	1750	550	67	250	37	—
43	5400	3	14500	1750	575	68	250	37	—
44	5800	3	15000	2000	600	70	250	40	—
45	6200	3	15800	2000	600	72	250	40	—
46	6600	3	16300	2250	600	74	275	43	—
47	7000	3	17600	2250	600	76	275	43	—
48	7400	3	18000	2250	600	77	275	44	—
49	7800	3	19500	2500	600	80	275	46	—
50	8200	3	20300	2700	600	82	275	48	—
51	8600	3	21000	2900	600	83	275	49	—
52	9000	3	22000	3000	600	85	275	50	—
53	9500	3	23000	3000	600	87	275	50	—

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

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 x1,114	30 x 9 x 3,75	607	60	2205	4500	356	7061
8,84 x 2,74 x 1,10	29 x 8,75 x 3,60	545	54	1976	4050	356	6382
8,53 x 2,59 x 1,07	28 x 8,50 x 3,50	500	50	1824	3750	330	5894
8,23 x 2,51 x 1,04	27 x 8,25 x 3,40	454	45	1646	3376	330	5351
7,92 x 2,44 x 0,99	26 x 8,00 x 3,25	405	40	1473	3000	305	4778
7,62 x 2,36 x 0,96	25 x 7,75 x 3,15	366	36	1326	2700	305	4331
7,31 x 2,29 x 0,91	24 x 7,50 x 3,0	324	32	1180	2400	254	3845
7,01 x 2,29 x 0,88	23 x 7,50 x 2,90	300	30	1067	2250	254	3591 ✓
6,71 x 2,21 x 0,84	22 x 7,25 x 2,75	256	26	955	1950	229	3134
6,40 x 2,13 x 0,82	21 x 7,0 x 2,70	238	23	864	1725	229	2818
6,10 x 2,06 x 0,79	20 x 6,75 x 2,60	210	21	762	1575	203	2540
5,79 x 1,98 x 0,76	19 x 6,50 x 2,50	182	18	650	1350	178	2178
5,49 x 1,90 x 0,73	18 x 6,25 x 2,40	162	16	590	1200	152	1942
5,18 x 1,83 x 0,715	17 x 6,0 x 2,30	143	14	502	1050	152	1710
4,83 x 1,75 x 0,70	16 x 5,75 x 2,30	127	12	457	900	127	1494

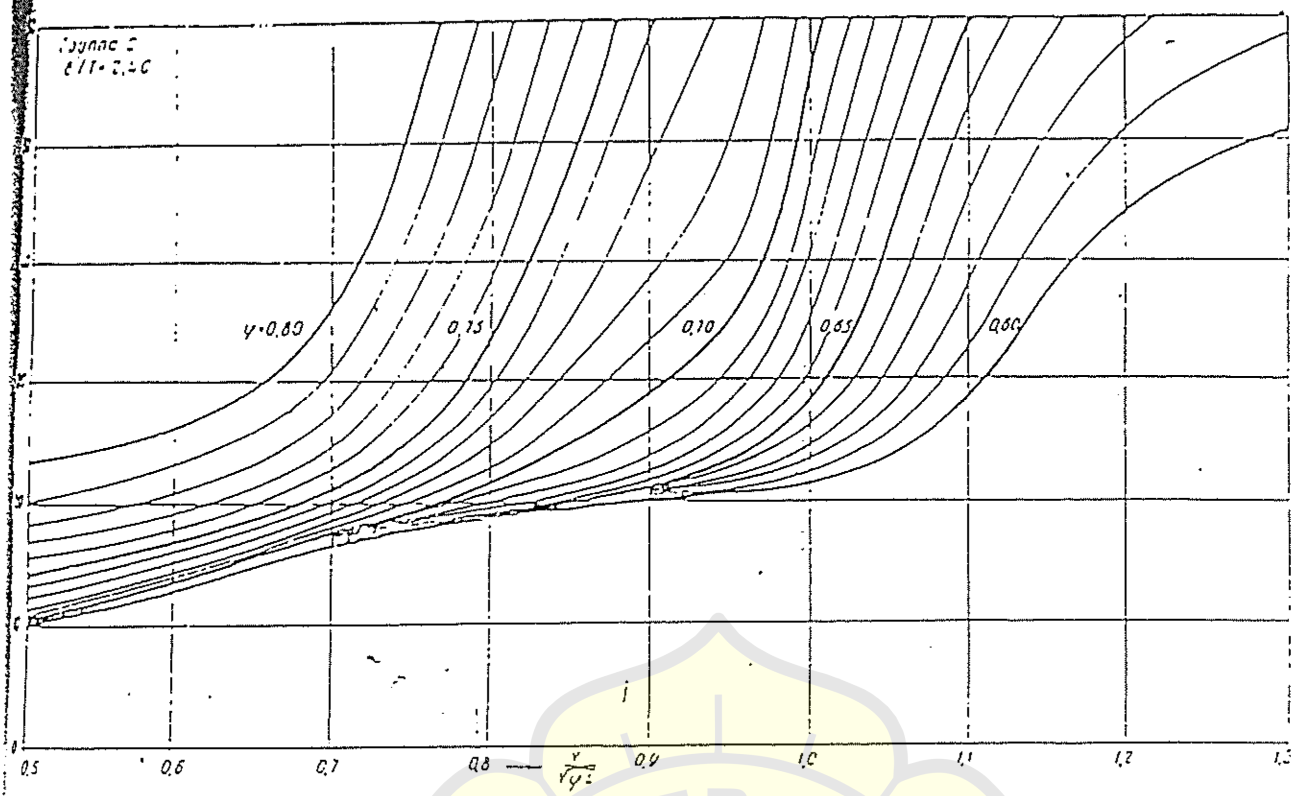


Рис. 6. Диаграмма А. И. Б. Лапа, группа C =96.39

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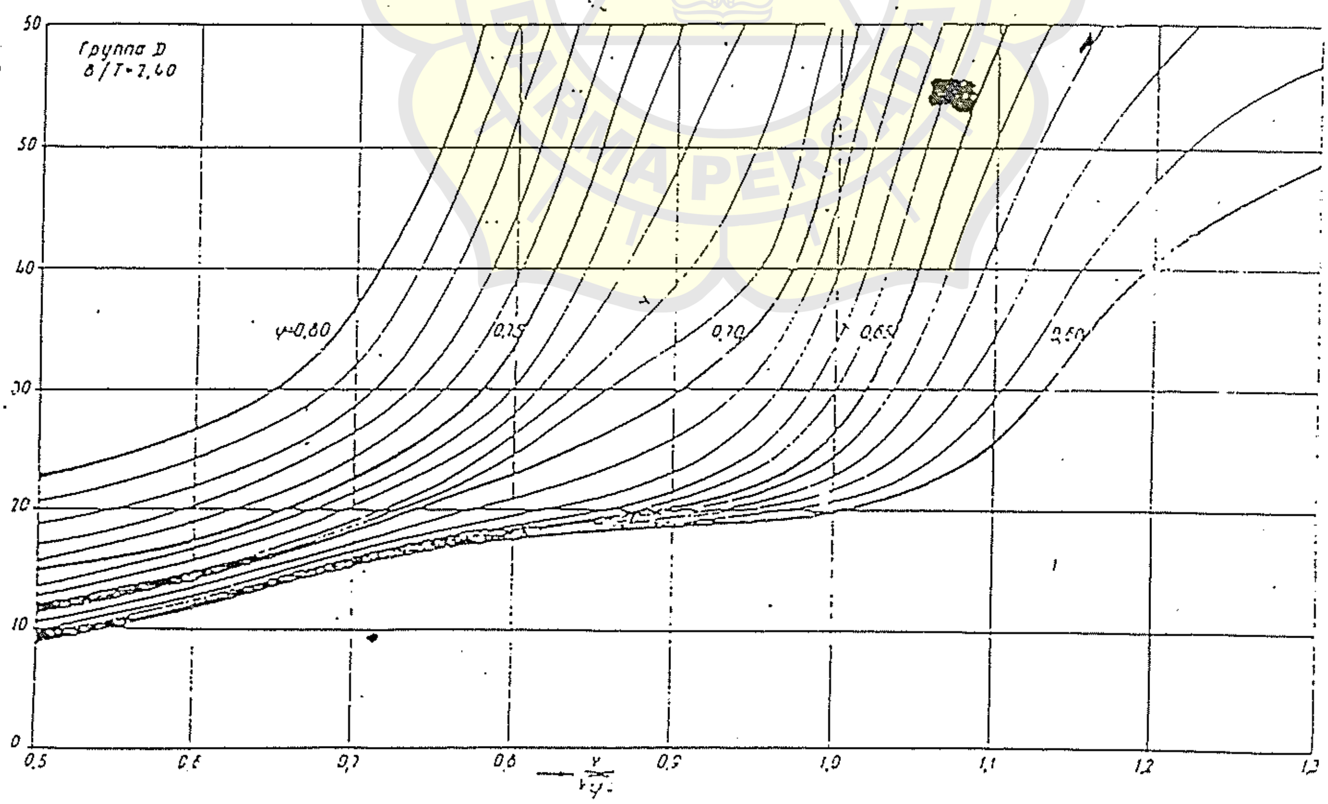

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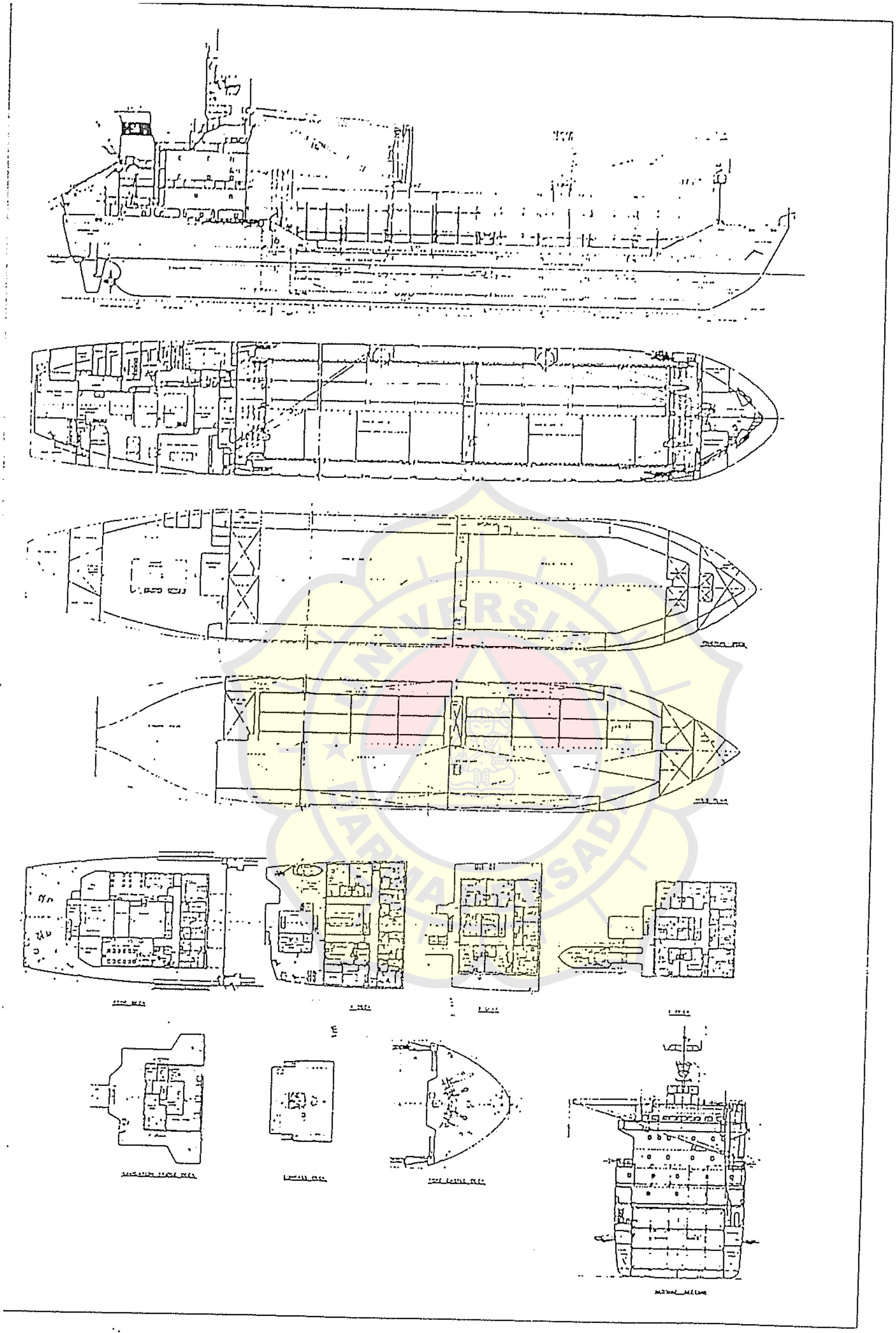

INDEX - MARINE ENGINE PERFORMANCE DATA

SALES MODEL	COMB SYS	ASP SYS TYPE	AFTERCOOLER TYPE	AFTERCOOLER WATER TEMP	EXH MFLD TYPE	COMP RATIO	GOV TYPE	APPLICATION TYPE	RATING TYPE	ENGINE POWER HP	ENGINE POWER KW	FLYWHEEL SPEED RPM	EFFECTIVE SERIAL NUMBER	PERF DATA REF NO.
3412C	DI	TA	JWAC	F	W/C	14.2:1		M PROP ENG	E-HIGH PERF	1250	932.0	2300	3JK	DM0545-03
3412C	DI	TA	JWAC	F	W/C	14.5:1		M PROP ENG	A-CONT	624	465.0	1800	60M05355	TM0008-04
3412C	DI	TA	JWAC	F	W/C	14.5:1		M PROP ENG	B-MED DUTY	671	500.0	1800	60M05355	TM1675-04
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	A-CONT	600	447.5	1200	69Z00501	TM0036-05
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	A-CONT	705	526.0	1200	69Z00498	TM0027-04
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	A-CONT	775	578.0	1600	96Y01050	TM0010-05
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	A-CONT	775	578.0	1800	96Y01128	TM0020-05
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	A-CONT	855	637.5	1600	96Y01088	TM0017-05
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	A-CONT	855	637.5	1800	96Y01139	TM0021-05
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	B-MED DUTY	805	600.0	1300	69Z00615	TM0028-05
3508	DI	TA	JWAC	130F	ASWC	13.0:1		M PROP ENG	B-MED DUTY	905	675.0	1600	96Y01050	TM0018-05
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	B-MED DUTY	960	716.0	1800	96Y01050	TM0022-05
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	C-INTER	820	611.5	1300	69Z00518	TM0034-05
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	C-INTER	850	634.0	1800	96Y01141	TM0048-05
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	C-INTER	940	701.0	1600	96Y01050	TM0019-06
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	C-INTER	1000	746.0	1800	96Y01050	TM0023-05
3508	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	D-PATROL CF	1150	857.5	1800	96Y01139	TM8297-05
3512	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	A-CONT	900	671.0	1200	66Z00615	TM0037-05
3512	DI	TA	JWAC	180F	ASWC	13.0:1		M PROP ENG	A-CONT	1060	790.5	1200	66Z00383	TM0038-05

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MARINE ENGINE PERFORMANCE DATA

TM0010-06 3509 DI TA JWAC M PROP ENG | 775 HP (578.0 KW) @ 1600 RPM A-CONT | ASWC MFLO | GOV

EFF SERIAL NO. 96T01050

ENG PERF DATA CURVE 1 - ZONE 1

ENGINE SPEED	RPM	1600	1500	1400	1300	1200	1100	1000	900	800
ENGINE POWER	HP	775	735	694	651	578	465	376	302	247
	KW	578.0	548.3	517.3	485.7	430.9	346.9	280.7	225.1	184.3
ENGINE TORQUE	FT-LB	2545	2575	2603	2632	2529	2221	1977	1762	1623
	N.M	3450	3491	3529	3568	3429	3011	2681	2389	2200
ENGINE BMEP	PSI	182	184	186	188	181	159	142	126	116
	KPA	1256	1271	1285	1299	1249	1097	976	870	801
SPECIFIC FUEL CONSUMPTIO	LB/HP-HR	.358	.355	.363	.372	.370	.373	.386	.404	.424
	G/KW-HR	218	216	221	226	225	227	235	246	258
FUEL RATE	GAL/HR	39.7	37.4	36.0	34.5	30.6	24.8	20.8	17.5	15.0
	LPH	150.1	141.4	136.1	130.8	115.7	93.9	78.6	66.1	56.8
INTAKE MANIFOLD TEMP ...	DEG F	184.5	182.5	181.6	180.5	178.2	176.7	175.6	174.9	174.7
	DEG C	84.7	83.6	83.1	82.5	81.2	80.4	79.8	79.4	79.3
INTAKE MANIFOLD PRESSURE	IN-HG	40.2	36.2	33.0	29.8	24.7	18.0	13.2	9.4	6.7
	KPA	135.6	122.3	111.5	100.8	83.4	60.9	44.7	31.6	22.5
INTAKE AIR FLOW	CFM	1755	1600	1451	1307	1116	929	791	675	579
	M3/MIN	50	45	41	37	32	26	22	19	16
EXHAUST MANIFOLD TEMP ..	DEG F	1023	1047	1075	1107	1131	1126	1119	1108	1099
	DEG C	550.7	563.8	579.4	597.3	610.4	608.0	604.0	598.0	592.7
EXHAUST MANIFOLD STACK TEMPERATURE...	DEG F	725	757	788	818	841	846	859	897	941
	DEG C	385	403	420	437	450	452	459	480	505
EXHAUST GAS FLOW	CFM	3888	3644	3401	3143	2733	2264	1963	1720	1526
	M3/MIN	110	103	96	89	77	64	56	49	43

MARINE ENGINE PERFORMANCE DATA

FEB 95 6202

TH0010-06 3508 DI TA JWAC M PROP ENG 775 HP (578.0 KW) @ 1600 RPM A-CONT ASHC MFLD GOV

EFF SERIAL NO. 96Y01050

PROPELLER DEMAND DATA FOR CURVE P

ENGINE SPEED	RPM	1600	1500	1400	1300	1200	1100	1000	900	800
ENGINE POWER	HP KW	775 578.0	639 476.3	519 387.2	416 310.0	327 243.8	252 187.8	189 141.1	138 102.9	97 72.3
ENGINE TORQUE	FT-LB N.M	2545 3450	2236 3032	1948 2641	1679 2277	1431 1940	1203 1631	994 1348	805 1092	636 862
ENGINE BMEP	PSI KPA	182 1256	160 1104	140 962	120 829	103 707	86 594	71 491	58 398	46 314
SPECIFIC FUEL CONSUMPTIO	LB/HP-HR G/KW-HR	.358 218	.358 218	.365 222	.368 224	.373 227	.390 237	.416 253	.441 268	.464 282
FUEL RATE	GAL/HR LPH	39.7 150.1	32.7 123.6	27.1 102.4	21.8 82.7	17.4 65.8	14.1 53.2	11.2 42.5	8.7 32.9	6.4 24.3
INTAKE MANIFOLD TEMP ...	DEG F DEG C	184.5 84.7	180.1 82.3	178.0 81.1	176.2 80.1	175.8 79.9	174.9 79.4	174.2 79.0	173.8 78.8	173.3 78.5
INTAKE MANIFOLD PRESSURE	IN-HG KPA	40.2 135.6	29.5 99.7	21.5 72.7	15.0 50.8	10.0 33.7	6.1 20.6	3.2 10.8	1.3 4.5	.3 .9
INTAKE AIR FLOW	CFM M3/MIN	1755 50	1437 41	1169 33	950 27	766 22	682 19	618 18	547 16	463 13
EXHAUST MANIFOLD TEMP ..	DEG F DEG C	1023 550.7	1009 543.0	993.7 534.3	961.9 516.6	918.5 492.5	850.3 454.6	764.2 406.8	667.4 353.0	560.1 293.4
EXHAUST MANIFOLD STACK TEMPERATURE...	DEG F DEG C	725 385	744 396	746 397	724 385	720 382	697 369	654 346	599 315	526 275
EXHAUST GAS FLOW	CFM M3/MIN	3888 110	3235 92	2642 75	2105 60	1692 48	1466 42	1282 36	1077 31	840 24

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MARINE ENGINE PERFORMANCE DATA

TH0010-06 3508 DI TA JWAC M PROP ENG

775 HP (578.0 KW) @ 1600 RPM A-CONT

ASWC MFLD GOV

EFF SERIAL NO. 96Y01050

ENG PERF DATA CURVE 1 - ZONE 1

ENGINE SPEED	RPM	1600	1500	1400	1300	1200	1100	1000	900	800
ENGINE POWER	HP	775	735	694	651	578	465	376	302	247
	KW	578.0	548.3	517.3	485.7	430.9	346.9	280.7	225.1	184.3
ENGINE TORQUE	FT-LB	2545	2575	2603	2632	2529	2221	1977	1762	1623
	N.M	3450	3491	3529	3568	3429	3011	2681	2389	2200
ENGINE BMEP	PSI	182	184	186	188	181	159	142	126	116
	KPA	1256	1271	1285	1299	1249	1097	976	870	801
SPECIFIC FUEL CONSUMPTIO	LB/HP-HR	.358	.355	.363	.372	.370	.373	.386	.404	.424
	G/KW-HR	218	216	221	226	225	227	235	246	258
FUEL RATE	GAL/HR	39.7	37.4	36.0	34.6	30.6	24.8	20.8	17.5	15.0
	LPH	150.1	141.4	136.1	130.8	115.7	93.9	78.6	66.1	56.8
INTAKE MANIFOLD TEMP ...	DEG F	184.5	182.5	181.6	180.5	178.2	176.7	175.6	174.9	174.7
	DEG C	84.7	83.6	83.1	82.5	81.2	80.4	79.8	79.4	79.3
INTAKE MANIFOLD PRESSURE	IN-HG	40.2	36.2	33.0	29.8	24.7	18.0	13.2	9.4	6.7
	KPA	135.6	122.3	111.5	100.8	83.4	60.9	44.7	31.6	22.5
INTAKE AIR FLOW	CFM	1755	1600	1451	1307	1116	929	791	675	579
	M3/MIN	50	45	41	37	32	26	22	19	16
EXHAUST MANIFOLD TEMP ..	DEG F	1023	1047	1075	1107	1131	1126	1119	1108	1099
	DEG C	550.7	563.8	579.4	597.3	610.4	608.0	604.0	598.0	592.7
EXHAUST MANIFOLD STACK TEMPERATURE...	DEG F	725	757	780	818	841	846	859	897	941
	DEG C	385	403	420	437	450	452	459	480	505
EXHAUST GAS FLOW	CFM	3888	3644	3401	3143	2733	2264	1963	1720	1526
	M3/MIN	110	103	96	89	77	64	56	49	43

MARINE ENGINE PERFORMANCE DATA

TM0010-06 3508 DI TA JWAC M PROP ENG — 775 HP (578.0 KW) a 1600 RPM A-CONT — ASWC MFLD — GOV — FEB 95 6202

EFF SERIAL NO. 96Y01050

HEAT REJECTION DATA FOR CURVE 4

ENGINE SPEED	RPM	1600	1500	1400	1300	1200	1100	1000	900	800
ENGINE POWER	HP	775	735	694	651	593	536	467	347	275
	KW	578.0	548.3	517.3	485.7	442.5	399.7	348.2	258.5	205.4
COOLANT TOT	BTU/MN	28321	25933	25705	25478	22975	20701	18141	13706	11203
	KW	498	456	452	448	404	364	319	241	197
ATMOSPHERE	BTU/MN	3298	3242	3128	3071	2957	2843	2730	2616	2559
	KW	58	57	55	54	52	50	48	46	45
EXHAUST TOT	BTU/MN	26103	24966	23828	22577	20644	18995	17288	14331	12739
	KW	459	439	419	397	363	334	304	252	224
EXH RECOVERY	BTU/MN	12227	12170	12000	11601	10748	10123	9270	8189	7450
	KW	215	214	211	204	189	178	163	144	131
FROM OIL CLR	BTU/MN	4550	4265	4095	3924	3583	3242	2900	2275	1934
	KW	80	75	72	69	63	57	51	40	34
FROM AFTRCLR	BTU/MN	3469	2730	2161	1649	1194	910	512	-114	-284
	KW	61	48	38	29	21	16	9	-2	-5
WORK ENERGY	BTU/MN	32871	31165	29402	27639	25136	22748	19791	14729	11658
	KW	578	548	517	486	442	400	348	259	205
LHV ENERGY .	BTU/MN	85760	80755	77798	74727	67960	61818	54993	43164	36340
	KW	1508	1420	1368	1314	1195	1087	967	759	639
HHV ENERGY .	BTU/MN	91333	86044	82859	79618	72395	65855	58576	45951	38728
	KW	1606	1513	1457	1400	1273	1150	1030	808	681

MARINE ENGINE PERFORMANCE DATA

FEB 95 6202

TM0010-06 3508 DI TA JWAC M PROP ENG

775 HP (578.0 KW) @ 1600 RPM A-CONT ASWC MFLD GOV

EFF SERIAL NO. 96Y01050

S O U N D - E X H A U S T - D A T A F O R C U R V E A

	ENG SPD RPM	ENGINE POWER HP	ENGINE POWER KW	OVERALL SOUND DB(A)	OBCF 63HZ DB	OBCF 125HZ DB	OBCF 250HZ DB	OBCF 500HZ DB	OBCF 1000HZ DB	OBCF 2000HZ DB	OBCF 4000HZ DB	OBCF 8000HZ DB
DATA @ 1.5M (4.9 FT)	1600	775	578.0	110	100	103	105	102	104	103	104	101
	1500	735	548.3	110	99	103	105	102	103	103	104	101
	1400	694	517.3	110	99	103	105	102	103	103	103	100
	1300	651	485.7	109	97	101	103	100	101	103	104	98
	1200	593	442.5	109	97	101	103	99	100	102	104	97
	1100	536	399.7	108	96	100	102	98	99	101	103	97
	1000	467	348.2	107	91	94	95	94	97	101	103	92
	900	347	258.5	106	90	93	93	93	96	100	102	90
	800	275	205.4	105	89	92	92	92	95	99	101	89
DATA @ 7.0M (22.9 FT)	1600	775	578.0	97	85	92	91	88	91	90	91	86
	1500	735	548.3	97	85	92	90	88	91	90	90	86
	1400	694	517.3	96	84	92	90	88	90	89	90	85
	1300	651	485.7	96	87	89	91	87	89	89	90	83
	1200	593	442.5	95	86	89	90	87	89	89	89	82
	1100	536	399.7	95	86	88	89	86	88	88	88	81
	1000	467	348.2	94	84	85	83	83	86	86	90	79
	900	347	258.5	92	83	83	81	81	85	85	88	77
	800	275	205.4	91	82	82	80	80	84	84	87	76
DATA @ 15 M (49.2 FT)	1600	775	578.0	90	78	86	84	82	84	83	84	79
	1500	735	548.3	90	78	85	84	81	84	83	84	79
	1400	694	517.3	90	78	85	83	81	84	83	83	79
	1300	651	485.7	89	80	83	84	81	83	83	83	76
	1200	593	442.5	89	80	82	83	80	82	82	82	75
	1100	536	399.7	88	79	81	83	79	82	81	82	75
	1000	467	348.2	87	77	78	76	76	80	80	83	72
	900	347	258.5	86	76	77	75	75	78	78	82	71
	800	275	205.4	85	75	75	74	74	77	77	81	70

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MARINE ENGINE PERFORMANCE DATA

TH0010-06 3508 DI TA JWAC M PROP ENG

775 HP (578.0 KW) a 1600 RPM A-CONT

ASHC MFLD GOV

EFF SERIAL NO. 96Y01050

SOUND - MECHANICAL - DATA FOR CURVE 4

	ENG SPD RPM	ENGINE POWER HP	ENGINE POWER KW	OVERALL SOUND DB(A)	OBCF 63HZ DB	OBCF 125HZ DB	OBCF 250HZ DB	OBCF 500HZ DB	OBCF 1000HZ DB	OBCF 2000HZ DB	OBCF 4000HZ DB	OBCF 8000HZ DB
DATA a 1.0M (3.2 FT)	1600	775	578.0	102	98	99	95	94	97	97	93	87
	1500	735	548.3	101	97	99	95	93	96	96	92	86
	1400	694	517.3	101	96	98	94	93	96	95	91	86
	1300	651	485.7	100	99	96	90	91	94	95	92	84
	1200	593	442.5	100	99	95	90	91	93	95	92	83
	1100	536	399.7	99	98	95	89	90	93	94	91	83
	1000	467	348.2	98	98	94	88	89	92	94	90	82
	900	347	258.5	98	97	93	88	89	91	93	90	81
	800	275	205.4	97	96	93	87	88	91	92	89	81
DATA a 7.0M (22.9 FT)	1600	775	578.0	90	86	87	83	82	85	85	81	75
	1500	735	548.3	89	85	87	83	81	84	84	80	74
	1400	694	517.3	89	84	86	82	81	84	83	79	74
	1300	651	485.7	88	87	84	78	79	82	83	80	72
	1200	593	442.5	88	87	83	78	79	81	83	80	71
	1100	536	399.7	87	86	83	77	78	81	82	79	71
	1000	467	348.2	86	86	82	76	77	80	82	78	70
	900	347	258.5	86	85	81	76	77	79	81	78	69
	800	275	205.4	85	84	81	75	76	79	80	77	69
DATA a 15 M (49.2 FT)	1600	775	578.0	84	80	81	77	76	79	79	75	69
	1500	735	548.3	83	79	81	77	75	78	78	74	68
	1400	694	517.3	83	78	80	76	75	78	77	73	68
	1300	651	485.7	82	81	78	72	73	76	77	74	66
	1200	593	442.5	82	81	77	72	73	75	77	74	65
	1100	536	399.7	81	80	77	71	72	75	76	73	65
	1000	467	348.2	80	80	76	70	71	74	76	72	64
	900	347	258.5	80	79	75	70	71	73	75	72	63
	800	275	205.4	79	78	75	69	70	73	74	71	63

MARINE ENGINE PERFORMANCE DATA

FEB 95 6202

TM0010-06 3508 DI TA JWAC M PROP ENG — 775 HP (578.0 KW) @ 1600 RPM A-CONT — ASWC MFLD — GOV

EFF SERIAL NO. 96Y01050

I D E N T I F I C A T I O N   R E F E R E N C E   A N D   N O T E S

ENGINE ARRANGEMENT	NUMBER	7N-8702
EFFECTIVE SERIAL	NUMBER	96Y01050
PRIMARY ENGINE TEST SPEC (OT)	NUMBER	0T-6901
PERFORMANCE PARM REF. ENGLISH	NUMBER	TE0015
METRIC	NUMBER	TM0015
PERFORMANCE DATA REF.	NUMBER	TM0010
AUX COOLANT PUMP PERF REF.	NUMBER	
COOLING SYSTEM PERF REF.	NUMBER	TD3094
CERTIFICATION	REF	
COMPRESSION RATIO	NUMBER	13.0
AFTERCOOLER TEMPERATURE	DEG F	180
	DEG C	82
CRANKCASE BLOWBY VOLUME	CU FT/H	388.46
	M3/H	11.0
FUEL RATE(RATED RPM) NO LOAD	GAL/HR	
	L/HR	

LUBE OIL PRESS @ LOW IDLE SPD	PSI	20
	KPA	138
LUBE OIL PRESS @ RATED SPEED	PSI	65
	KPA	445
PISTON SPD @ RATED END SPD	FT/MIN	1970.50
	M/SEC	10.01
MAX OPERATING ALTITUDE	FEET	10171
	METRE	3100
PEEC ELECT CONTROL MODULE REF	NUMBER	
PEEC PERSONALITY CONT MOD REF	NUMBER	
TURBOCHARGER MODEL REF.		TW8106-1.03
FUEL INJECTOR REF.	NUMBER	7E-3382
TIMING STATIC	DEG	
TIMING STATIC ADVANCE	DEG	
TIMING STATIC	MM	
UNIT INJECTOR TIMING	MM	86.40

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

8. Crew Members

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

9. Outfit and Equipment

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

10. Classification, Rules

have to be observed.

11. Restrictions of Dimensions

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

12. Tonnage of Ships

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

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

Tonnage unit: gross ton

1 grt = 100 cbf = 2.83 m^3.

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

Gross tonnage

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

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

Net tonnage

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

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

T draught in $[m]$ (midships)

D depth in $[m]$ "

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

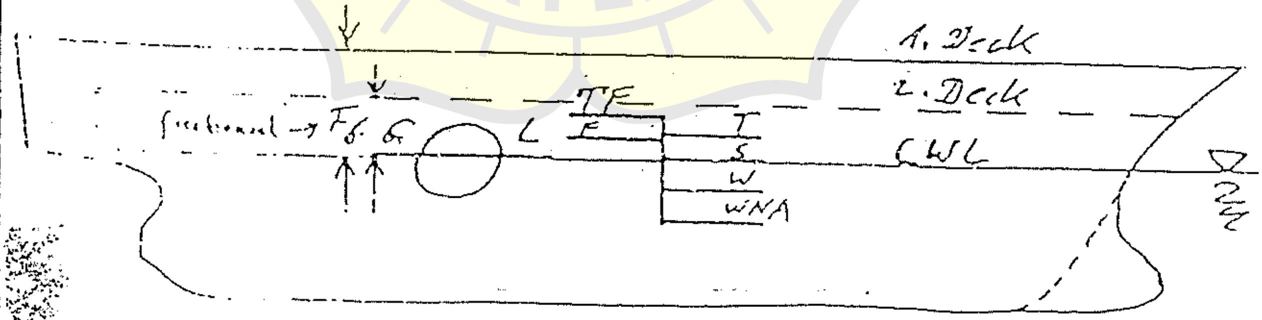
N_2 number of other passengers.

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

Definition of freeboard:

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

Lit.: International Freeboard Convention 1966.



Definition of superstructures:

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

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

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

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

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

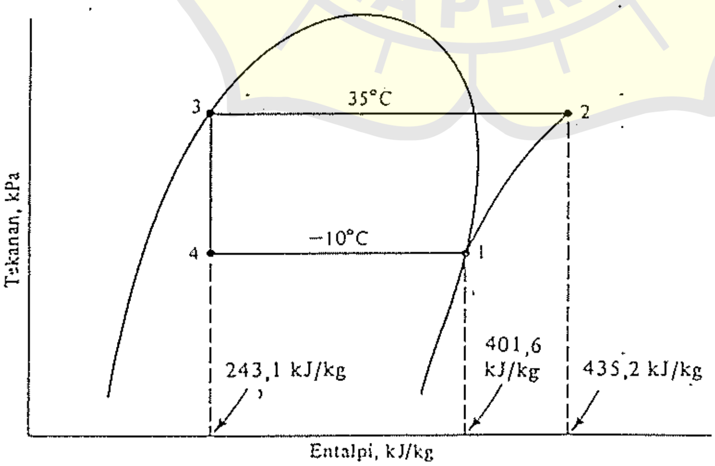
Untuk menemukan h_2 melalui garis entropi tetap geser titik 1 hingga mencapai tekanan jenuh yang sesuai dengan suhu 35°C. Tekanan pengembunan ini adalah 1354 kPa, dan nilai h_2 = 435,2 kJ/kg.

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

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

(*a*) Dampak refrigerasi:

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



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

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

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

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

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

(*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 kesetimbangan 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 dapatditingkatkan. 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

STANDART UKURAN SEKOCI OLEH BOT (BOARD OF TRADE) ENGLAND

Tabel II-

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

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

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

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

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

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

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

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

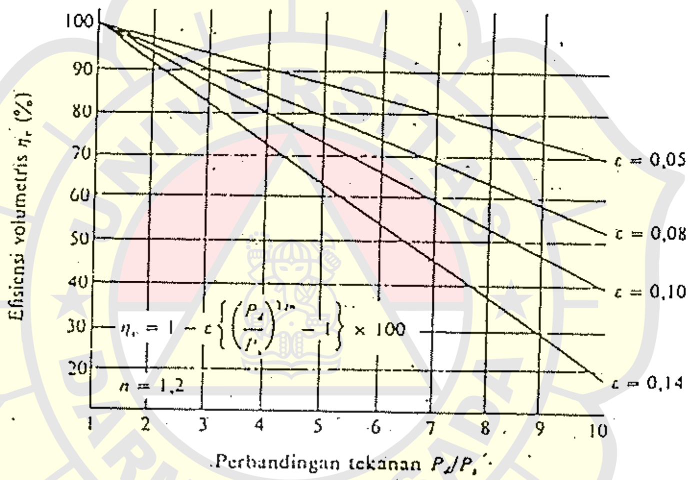
P_s: Tekanan isap dari silinder tingkat pertama (kgf/cm^2 abs).

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

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

Dalam Gb. 2.11 diperlihatkan pengaruh ε dan P_d/P_s pada efisiensi volumetris η_v.

Sehubungan dengan hal-hal di atas dapat dimengerti jika efisiensi volumetris juga tergantung pada faktor-faktor rancangan kompresor seperti bentuk dan ukuran silinder, serta bentuk, ukuran, dan susunan katup-katup.



Gb. 2.11 Efisiensi volumetris dan perbandingan tekanan.

2.4.2 Efisiensi adiabatik keseluruhan

Efisiensi kompresor ditentukan oleh berbagai faktor seperti tahanan aerodinamik di dalam katup-katup, saluran-saluran, pipa-pipa, kerugian mekanis, efektivitas pendinginan, dll. Namun, menentukan secara tepat pengaruh masing-masing faktor tersebut adalah sangat sulit. Karena itu faktor-faktor ini digabungkan dalam efisiensi adiabatik keseluruhan.

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

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

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

Besarnya daya adiabatik teoritis dapat dihitung dengan rumus

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

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

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

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

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

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

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

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

Tabel 2.7 Daya yang diperlukan untuk kompresi adiabatis teoritis.

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

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

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

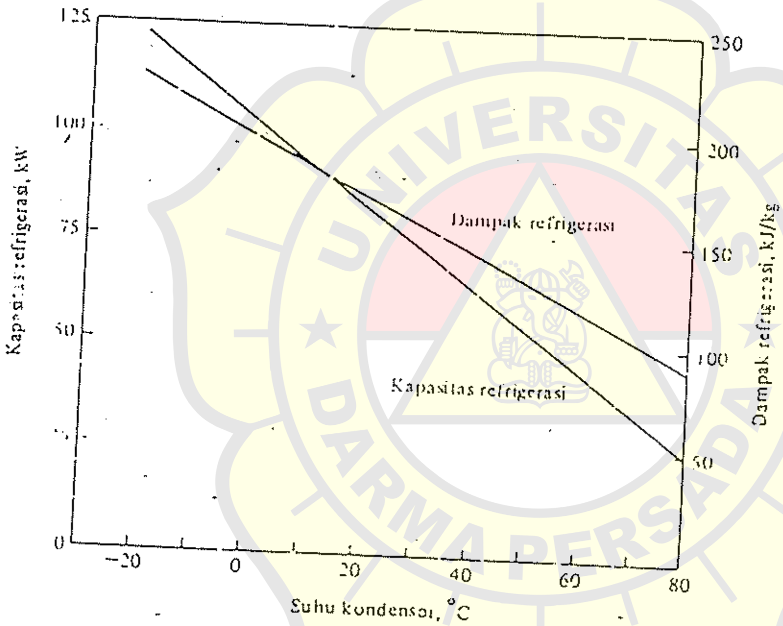
Selanjutnya, karena harga daya adiabatis teoritis untuk kompresor 1 tingkat berbeda dengan harga untuk kompresor 2 tingkat, maka memperbandingkan efisiensi kompresor harus dilakukan di antara yang sama jumlah tingkatnya.

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

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.

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

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

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

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

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

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

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

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

It follows from formula (314) that

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

Combining equations (315) and (316) we obtain

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

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

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

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

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

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

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

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

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

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

, the pull of the warp ends 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 manuturing 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
3,000	0.25	750
4,500	0.2	900
7,000	0.167	1,165
12,000	0.150	1,800

The speed at which a warping hawser is heaved in by a windlass 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 emyed 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 arrangel, 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 necery 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 cable lifter (Fig. 170) perpendicular to the shaft will be a regular

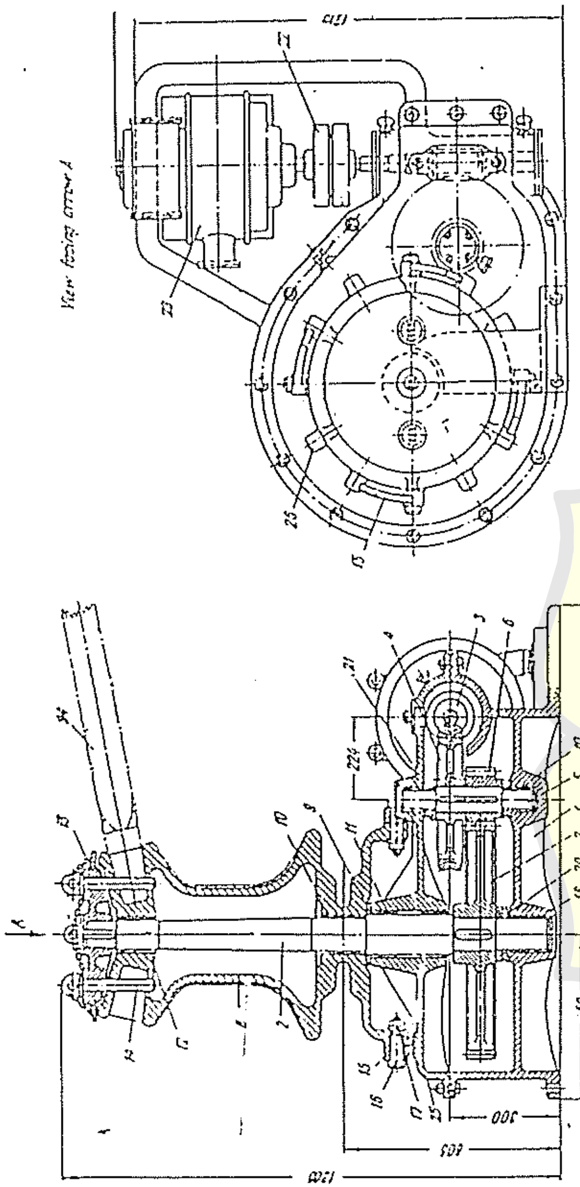


Fig. 169

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

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

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

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

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

* In breaking away one anchor from the bottom

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

...pumped is ... the amount of liquid

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

This equation can be solved for V_e and V_f:

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

and

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

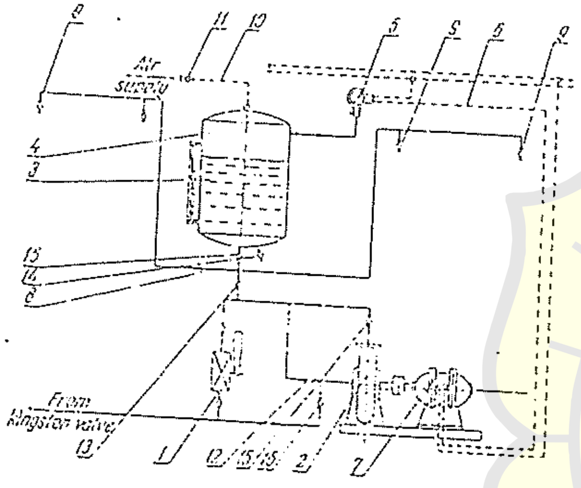


Fig. 189

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.

(D) SANITARY AND SCUPPER SYSTEMS

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

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

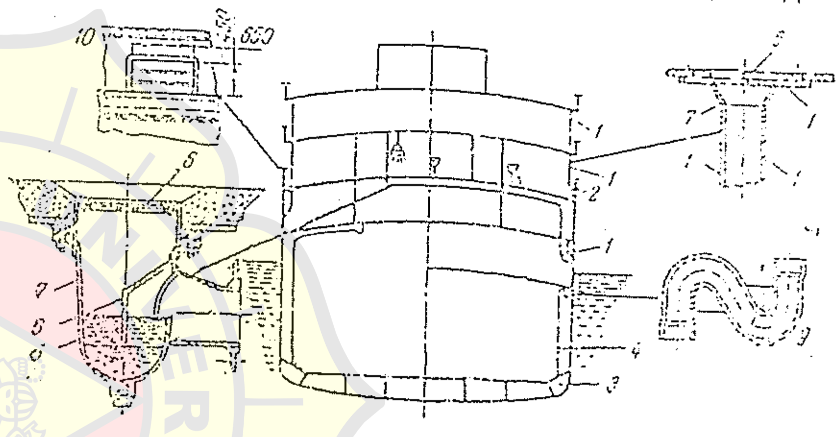


Fig. 190

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

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

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

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

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

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

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

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

In general

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

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

7.6. Consumables and tanks

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

- consumables
- provisions
- ballast.

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

- fuel oil

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

last brackets for reserve:

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

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

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

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

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

S = operating range [sm]

V_{serv} = speed [kn]

1 KW = 0.736 PS (BHP).

Motors:

Specific fuel oil consumption:

for two-stroke engines b = 205 ... 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: γ = 0.95 t/m³

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.

Diesel oil

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

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

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

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

additions see fuel oil!

Lubrication oil

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

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

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

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

Fresh water

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

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

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

b) Ballast capacity used for

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

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

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

- Sounding/ullage tables delivered by yard.

c) Provisions/persons/luggage

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

d) 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.

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

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

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

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

4.3 Ballast systems in the cargo area

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

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

a) Minimum wall thicknesses

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

b) Only completely welded pipes or equivalent are permitted

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

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

5. Ventilation and gas-freeing

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

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

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

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

Appendix to Section 2

Part C :

Approximate Calculation of the Starting Air Supply

1. Starting air for installations with reversible engines

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

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

where

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

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

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

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

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

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

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

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

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

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

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

The following values of n_A are to be applied:

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

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

n_o [Rpm] = rated speed

2. Starting air for installations with non reversible engines

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

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

1.6 Pipe connections

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

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

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

1.6.3 The direct bilge suction and the emergency 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 18

6.3.2 Bilge lines

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

6.3.3 Ballast pipes

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

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

6.3.4 Fuel pipes

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

6.3.5 Cargo pipes

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

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

6.4 Control stands

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

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

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

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

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

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

6.5 Power units

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

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

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

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

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

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

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

7. Pumps

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

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

Bilge pumps/bilge ejectors

Ballast pumps

Sea cooling water pumps

Fresh cooling water pumps

Fire extinguishing pumps

Emergency fire extinguishing pumps including drive units

Condensate pumps

Boiler feedwater pumps

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

4.10 Cofferdams, pipe tunnels and void spaces

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

4.11 Chain lockers

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

5. Additional Rules for passenger vessels

5.1.1 The arrangement of bilge pipes

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

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

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

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

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

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

5.2 Bilge suctions

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

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

5.3 Arrangement of bilge pumps

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

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

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

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

c) bilge pumps are installed above the bulkhead deck.

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

5.4 Passenger vessels for limited range of service

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

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

6. **Additional Rules for tankers**

See Section 15, B.4.

7. **Bilge testing**

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

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

1. **Oily water separating equipment**

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

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

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

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

2. **Discharge of fuel and oil residues**

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

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

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

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

P. Ballast Systems

1. **Ballast lines**

1.1 **Arrangement of piping - general**

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

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

1.2 **Pipes passing through tanks**

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

1.3 **Piping systems**

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

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

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

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[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 136 is to submitted for approval

Section 14

Rudder and Manoeuvring Arrangement

A. General

1. Manoeuvring arrangement

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

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

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

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

Guidance

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

1.5 For ice-strengthening see Section 15

2. Structural details

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

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

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

3. Size of rudder area

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

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

c_1 = factor for the ship type

= 1,0 in general

= 0,9 for bulk carriers and tankers having a displacement of more than 50 000 t

= 1,7 for tugs and trawlers

c_2 = factor for the rudder type

= 1,0 in general

= 0,9 for semi-spade rudders

= 0,8 for double rudders (per rudder)

= 0,7 for high lift rudders

c_3 = factor for the rudder profile

= 1,0 for NACA-profiles and plate rudder

= 0,8 for hollow profiles

c_4 = factor for the rudder arrangement

= 1,0 for rudders in the propeller jet

= 1,5 for rudders outside the propeller jet

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

Materials

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

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

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

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

R_{eH} = minimum nominal upper yield point of material used in [N/mm²]. R_{eH} is not to be taken greater than $0.7 \cdot R_m$ or 450 N/mm², whichever is less. R_m = tensile strength of the material used.

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

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

Definitions

C_R = rudder force in [N]

Q_R = rudder torque in [Nm]

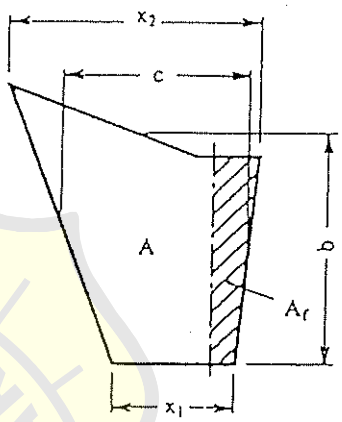
A = total movable area of the rudder in [m²]
For nozzle rudders, A is not to be taken less than 1,35 times the projected area of the nozzle;

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

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

b = mean height of rudder area in [m]

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



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

Fig. 14.1

Λ = aspect ratio of rudder area A_t

Λ = b^2/A_t

v_0 = ahead speed of ship in [kn] as defined in Section 1, H.5. if this speed is less than 10 kn, v_0 is to be taken as

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

v_a = astern speed of ship in [kn], if the astern speed $v_a \le 0{,}4 \cdot v_0$ or 6 kn, whichever is less, determination of rudder force and torque for astern condition is not required. For greater astern speeds special evaluation of rudder force and torque as a function of the rudder angle may be required. If no limitations for the rudder angle at astern condition is stipulated, the factor x_2 is not to be taken less than given in Table 14.1 for astern condition.

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

B. Rudder Force and Torque

1. Rudder force and torque for normal rudders

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

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

v = v_0 for ahead condition

v = v_a for astern condition

κ_1 = coefficient, depending on the aspect ratio Λ

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

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

κ_3 = coefficient, depending on the location of the rudder

κ_3 = 0,8 for rudders outside the propeller jet

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

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

κ_t = coefficient depending on the thrust coefficient c_t

κ_t = 1,0 normally

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

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

Table 14.1

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

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

$Q_R = C_R \cdot r$ [Nm]

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

α = 0,33 for ahead condition

α = 0,66 for astern condition (general)

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

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

α = 0,25 for ahead condition

α = 0,55 for astern condition.

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

k_b = balance factor as follows:

k_b = A_f/A

k_b = 0,08 for unbalanced rudders

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

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

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

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

The resulting force of each part may be taken as:

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

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

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

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

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

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

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

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

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

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

$c_1 = A_1/b_1$

$b_2 = A_2/b_2$

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

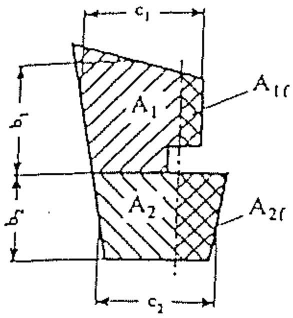


Fig. 14.2

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/m^3) to the summer load waterline

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

h = $f_b + \sum h'$

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

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

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

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

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

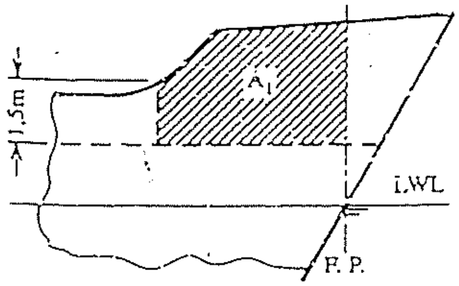


Fig. 18.1

C. Anchors

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

Guidance

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

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

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

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

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

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

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

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

KORT NOZZLE :

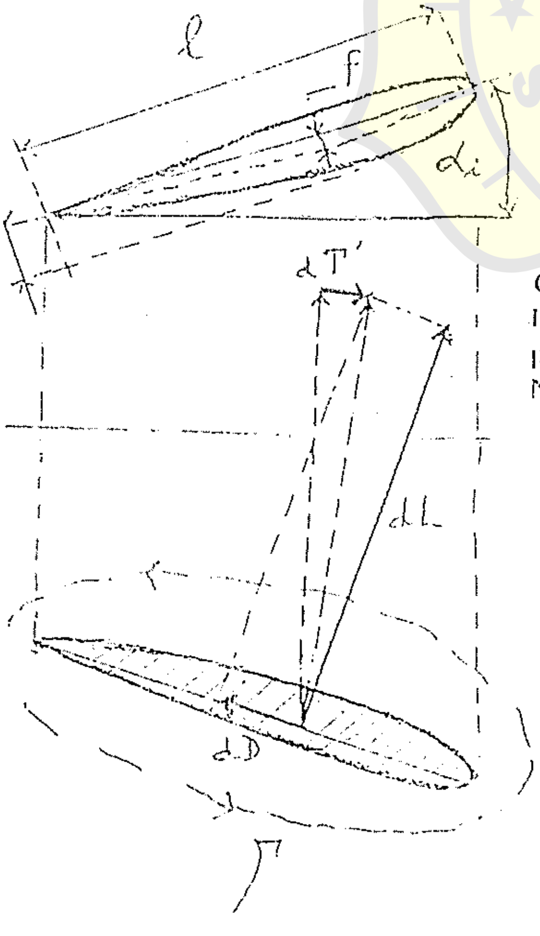
Tipe nozzle dengan menggunakan propeller tipe B4-55 yang telah diuji coba "open water" test oleh NSMB di Wageningen (sekarang MARIN) :

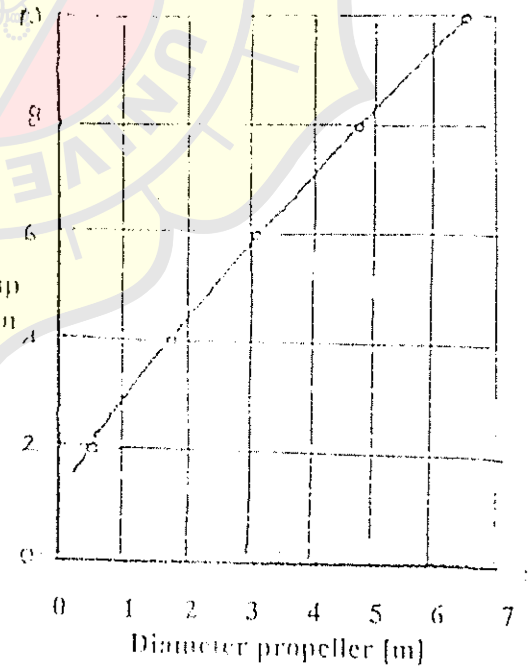
tebal nozzle — camber — sudut condong di nozzle

Nozzle :	l / D	s / l	f / l	α_i	Profile :
1	1,00	axial circular cylinder			
2	0,67	0,15	0,04	12,7°	NACA- 4415
3	0,50	0,15	0,04	12,7°	sda
4	0,83	0,15	0,04	12,7°	sda
5	0,50	0,15	0,04	15,2°	sda
6	0,50	0,15	0,04	15,2°	sda
7	0,50	0,16	0,05	12,7°	built up NACA-5415
8	0,50	0,15	0,03	12,7°	built up NACA-3415

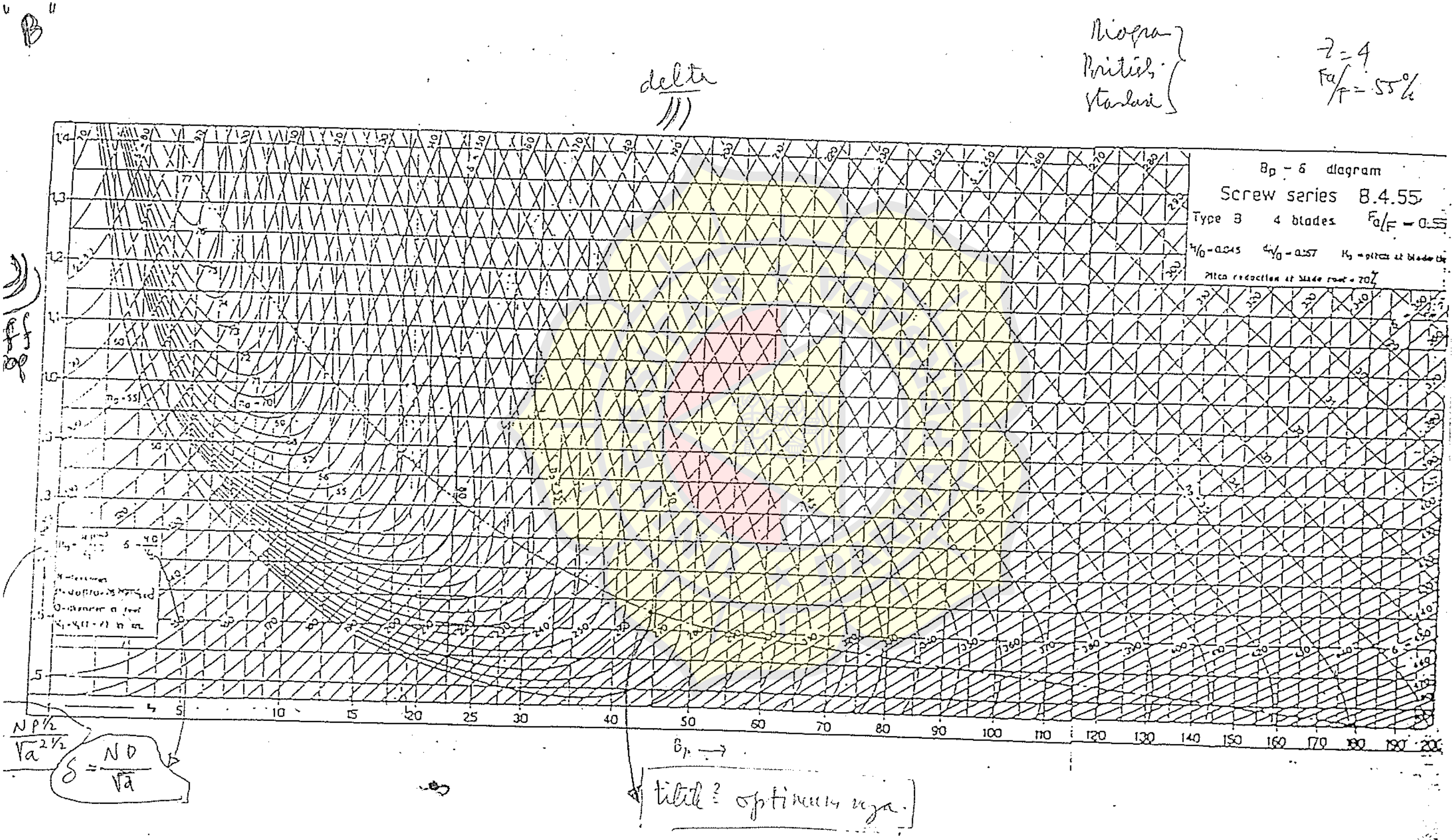
Model
s/d

bisa memperkecil adanya kavitasi (Tip Cortex)





"B"



delta
Diagram British standard
Z = 4
Fa/F = 55%
Bp - δ diagram
Screw series B.4.55
Type B 4 blades Fa/F = 0.55
Bp →
δ = ND/√a
NP½/Va^2½
titik ? optimum nya.

'Au' $\sqrt{Bp}$ – δ diagram

diagram troline }

4-BLADED PROPELLER. Type. AU
Constant Pitch
Exp A.R = 0.400 Boss Ratio = 0.180
B.T.R = 0.050 Rake Angle = 10°0'

$Bp = \frac{N P^{0.5}}{V_a^{2.5}}$ N = R.P.M
P = DHP (1HP = 75 kgm/sec)
$\delta = \frac{N D}{V_a}$ D = Diameter in m
V_a = Advance Speed in knot

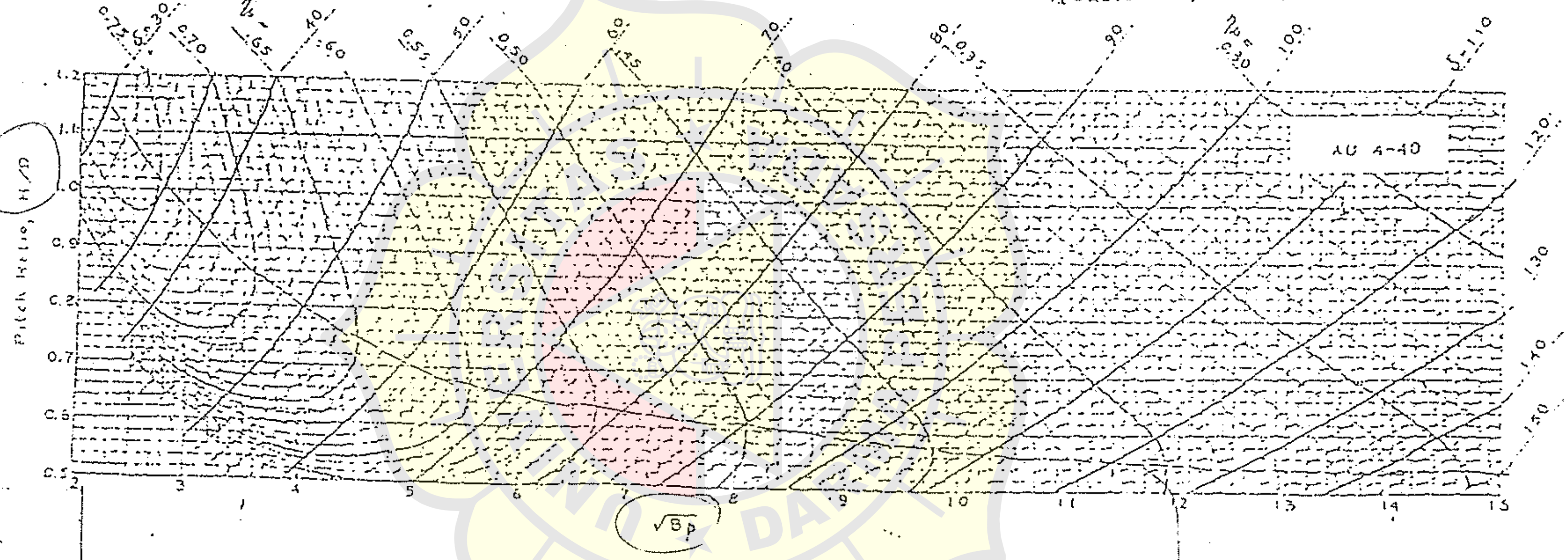


Fig. 10 - Design Diagram of AU4-40

optimum

$$Bp = \frac{N P^{1/2}}{V_{a\,(knot)}^{2\frac{1}{2}}}$$

$$\delta = \frac{N D^{(m)}}{V_{a\,(met)}}.$$

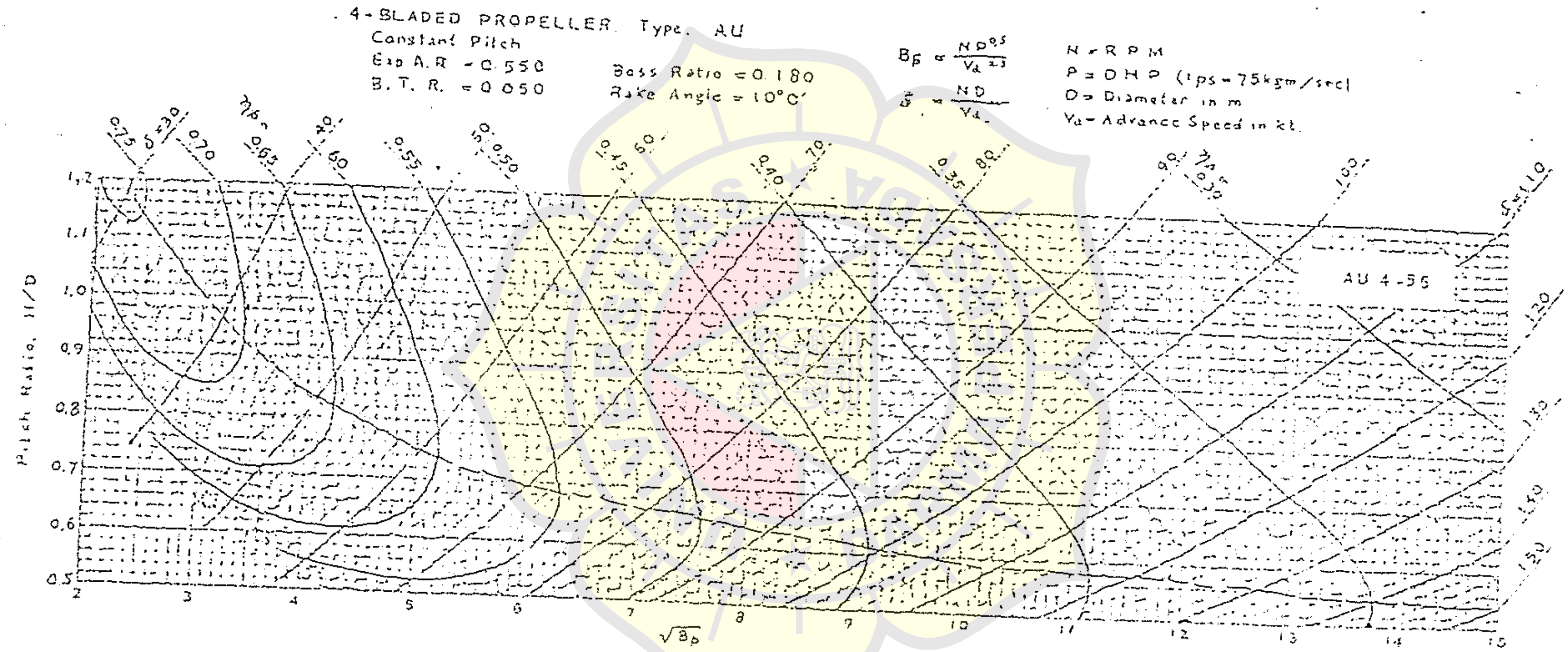


Fig. 11 - Design Diagram of AU4-55

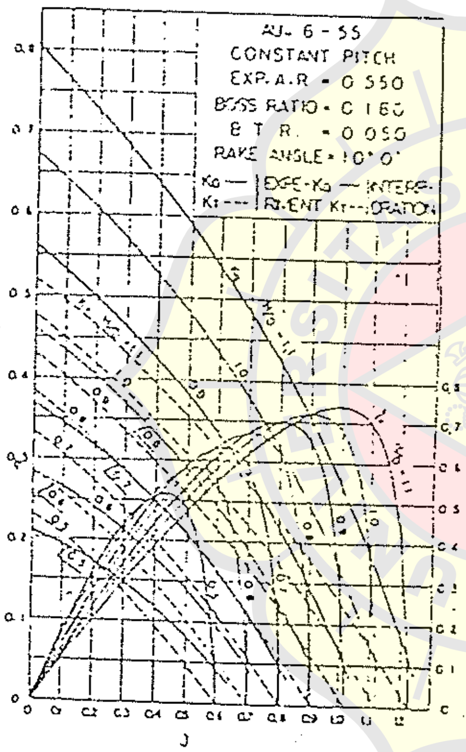


Fig. 6 - Open Test Result of AUw6-55

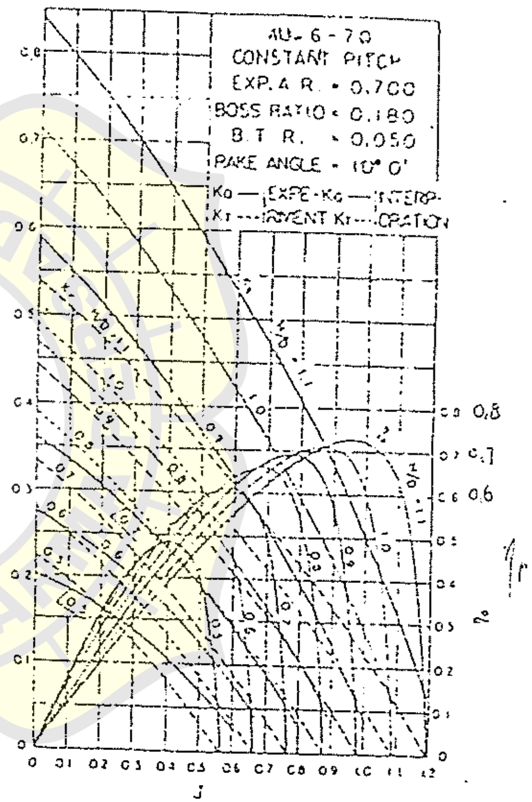


Fig. 7 - Open Test Result of AUw6-70

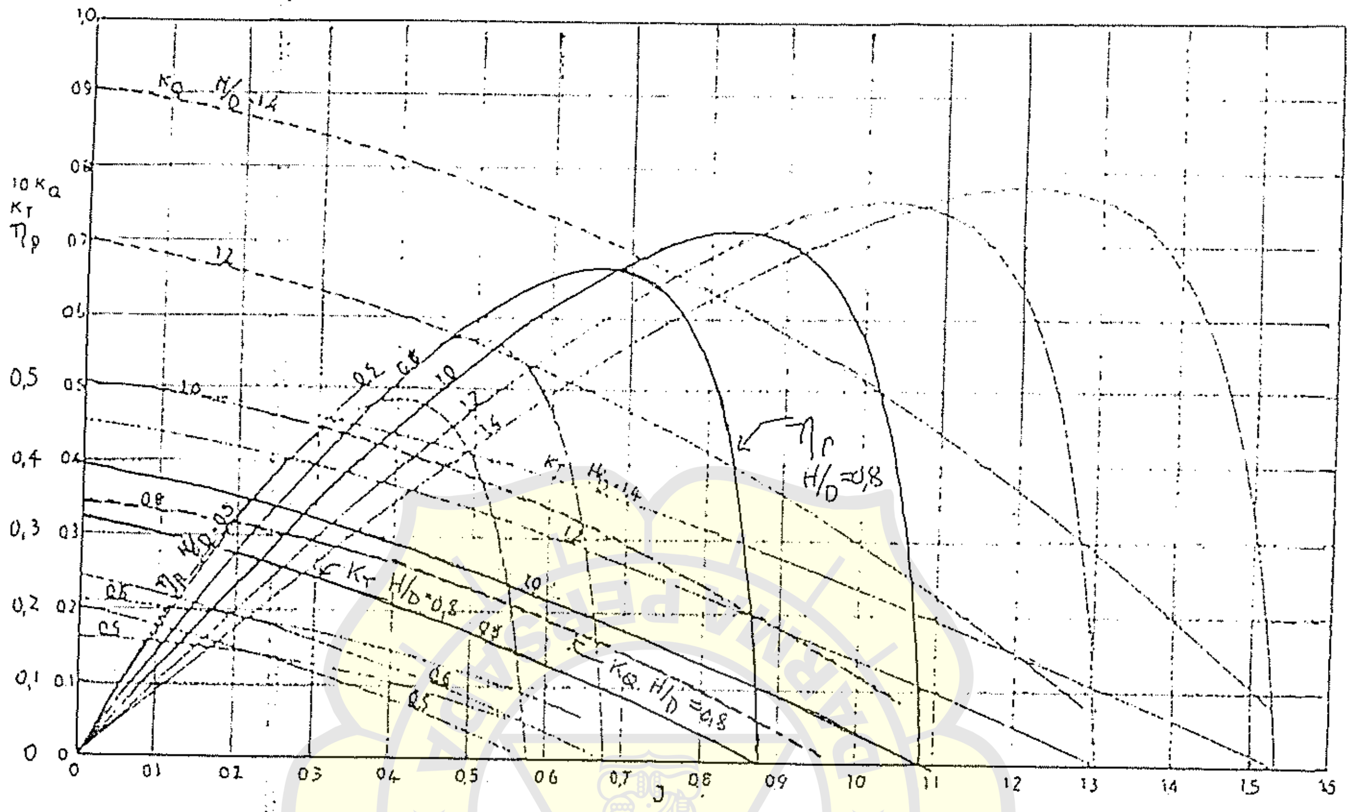


Fig. 5 Open-water test results of B 4-40 screw series

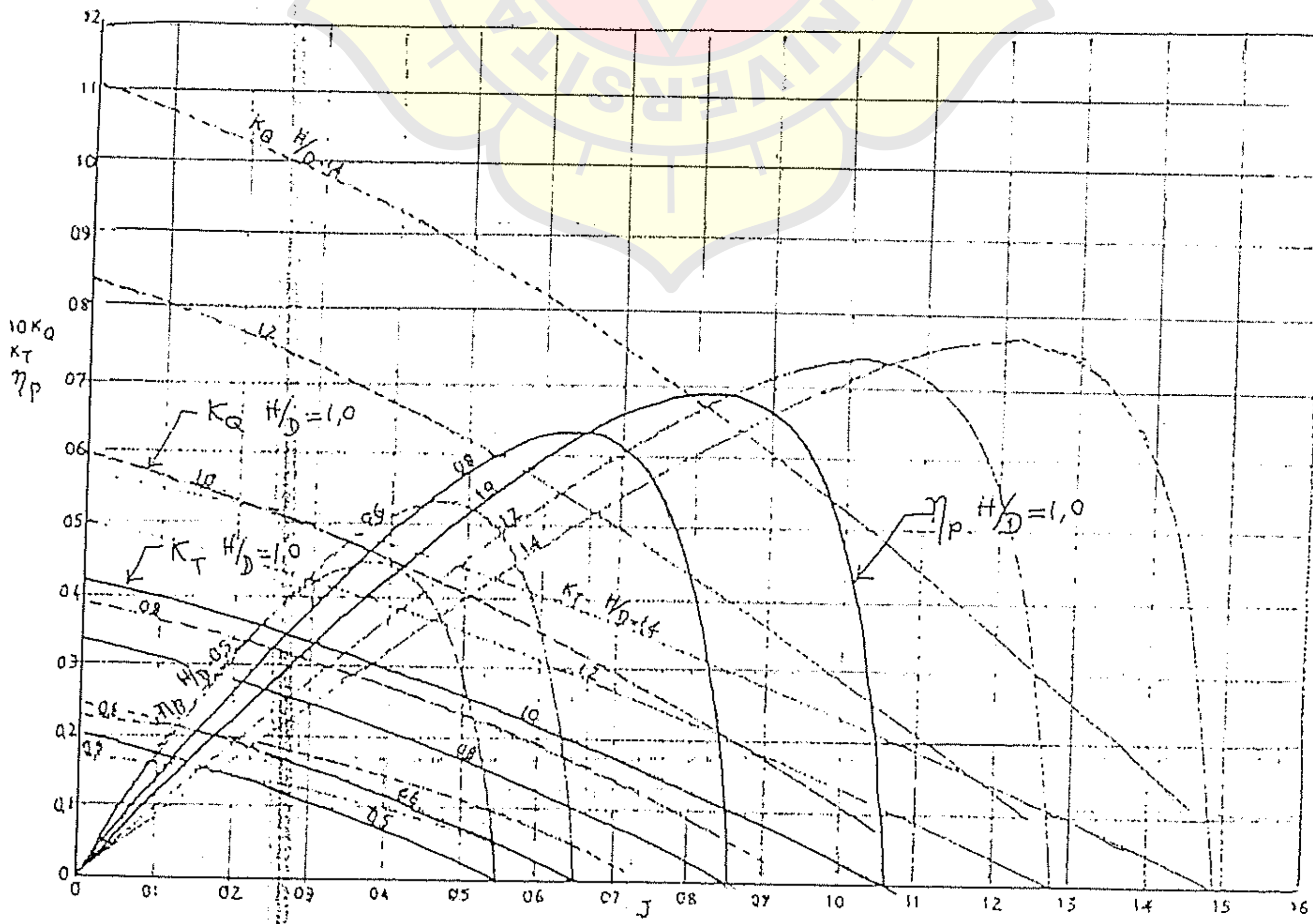
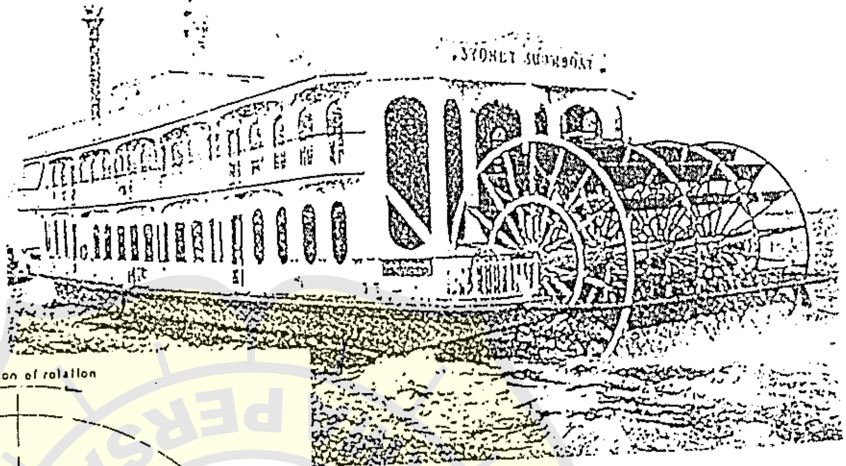
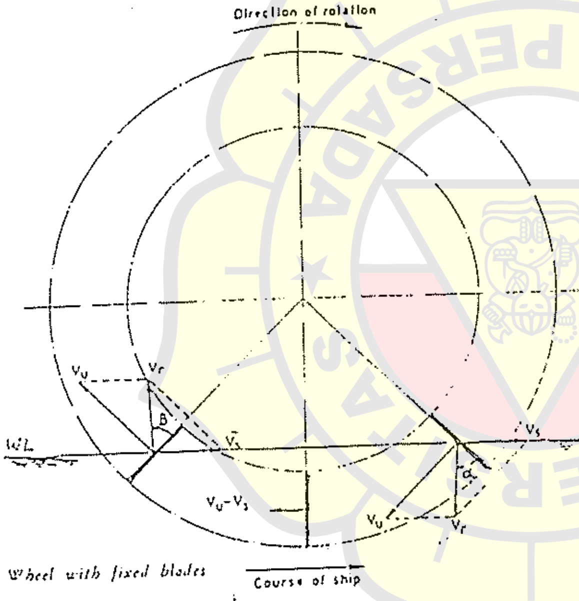


Fig. 6 Open-water test results of B 4-55 screw series

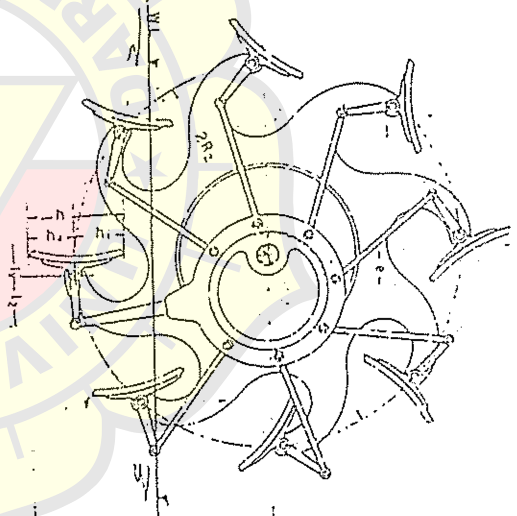
Jelasa

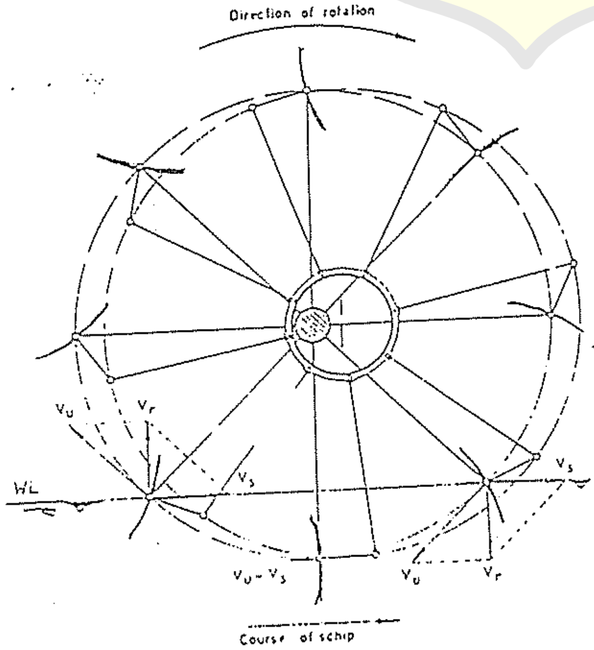
Alat Propulsi Mekanis : Paddle Wheel
(Roda Pedal/Roda Kayuh)



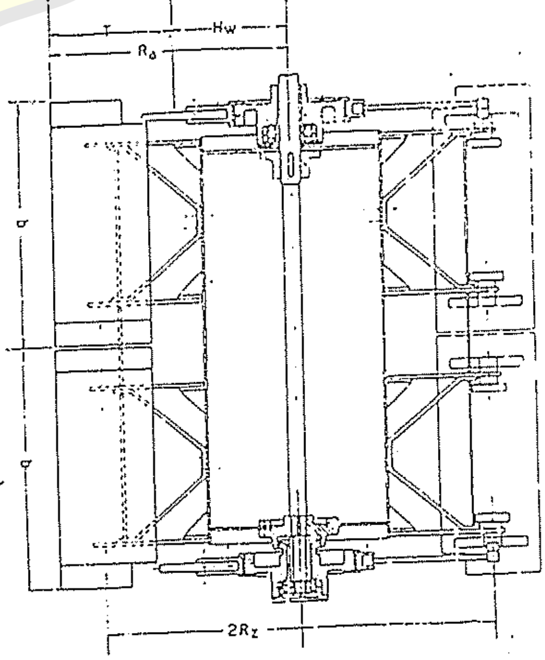


Wheel with fixed blades





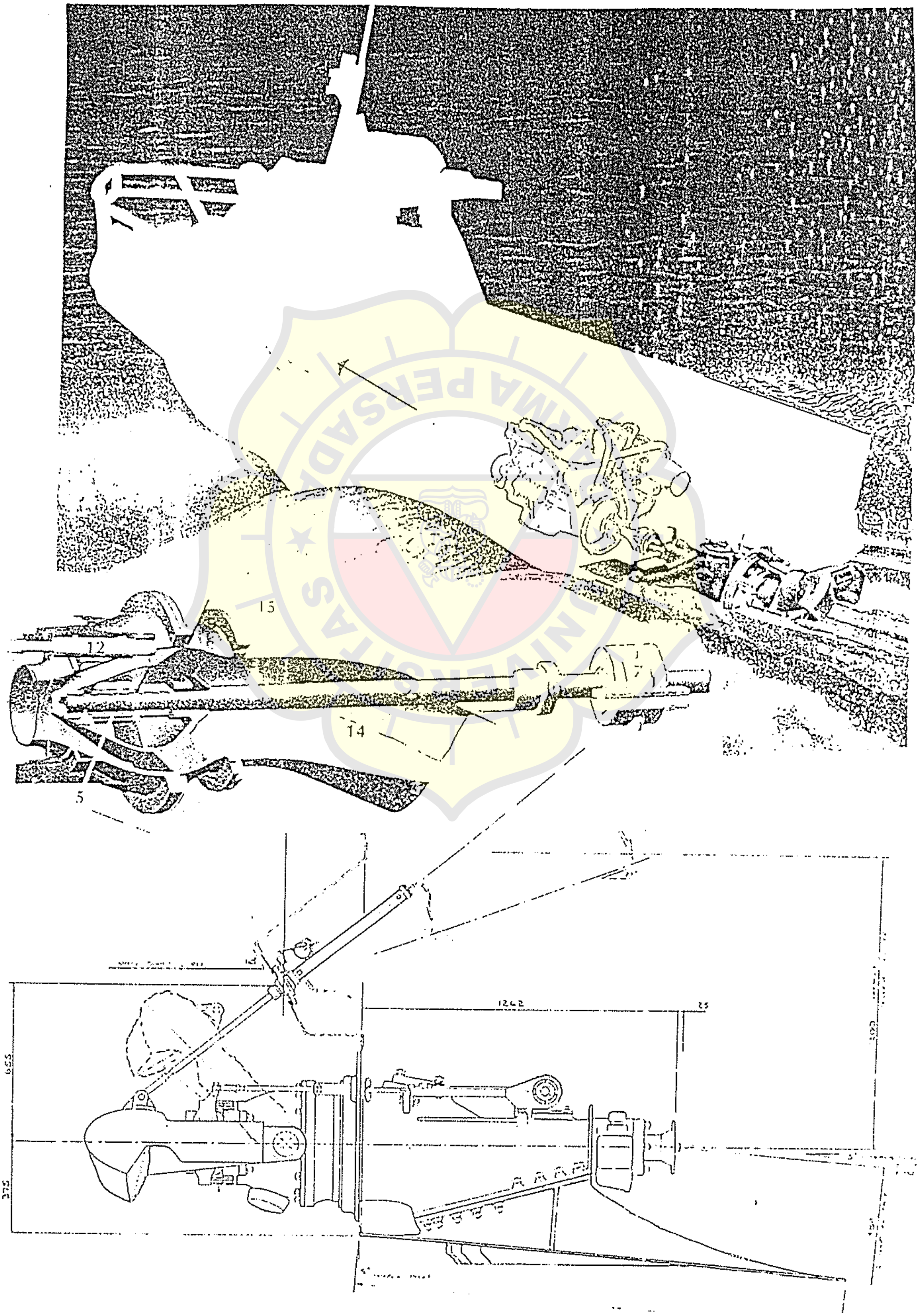
Wheel with adjustable blades
(ganyang, lalitat)
feathering



Model of a paddle wheel for a tug

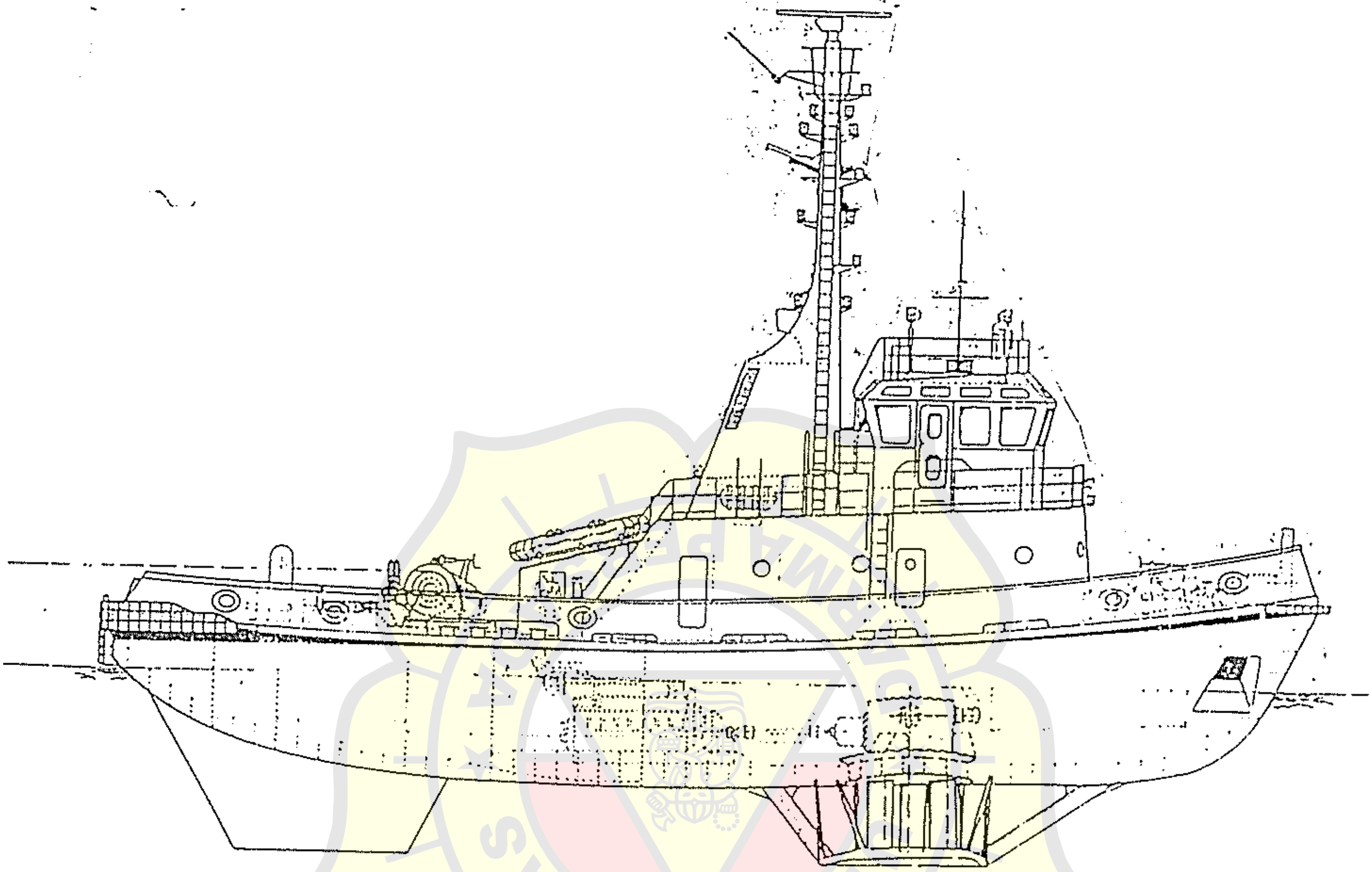
Propulsi Kapal

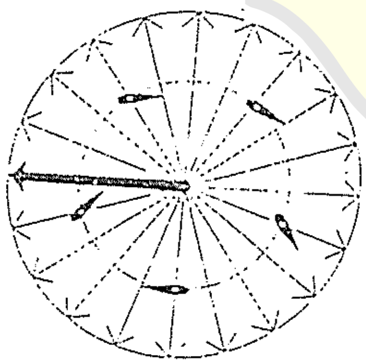
Alat Propulsi Mekanis : Water Jet Propulsion

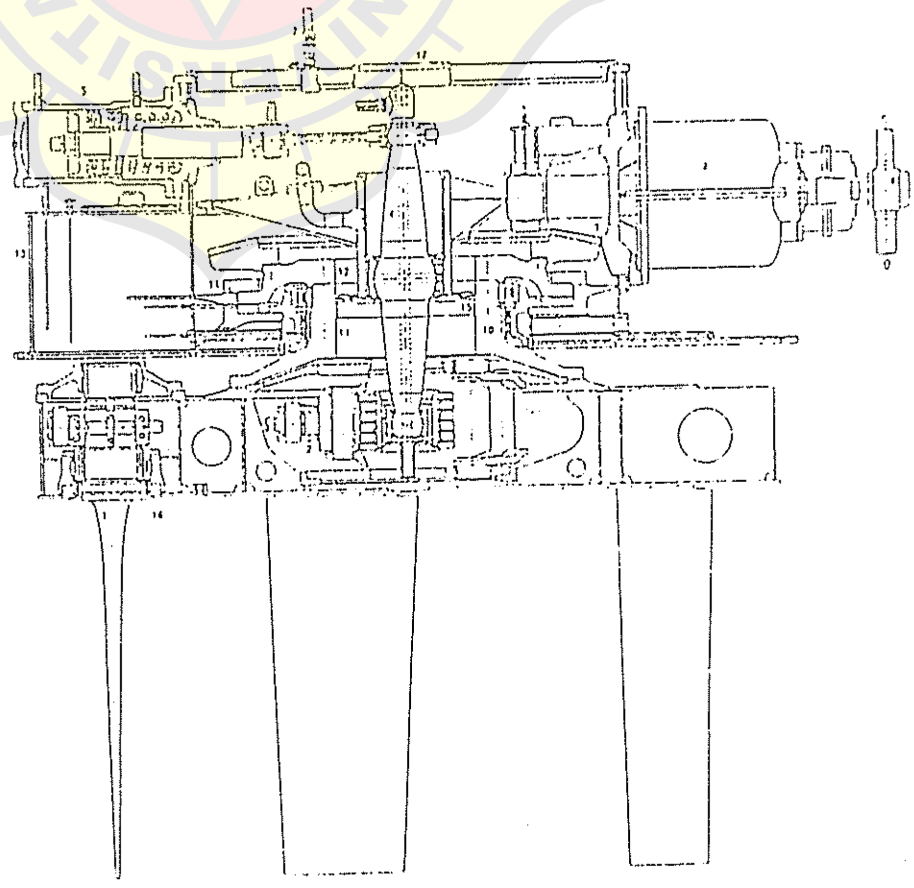


Propulsi Kapal

Alat Propulsi Mekanis : Voith Schneider Propeller

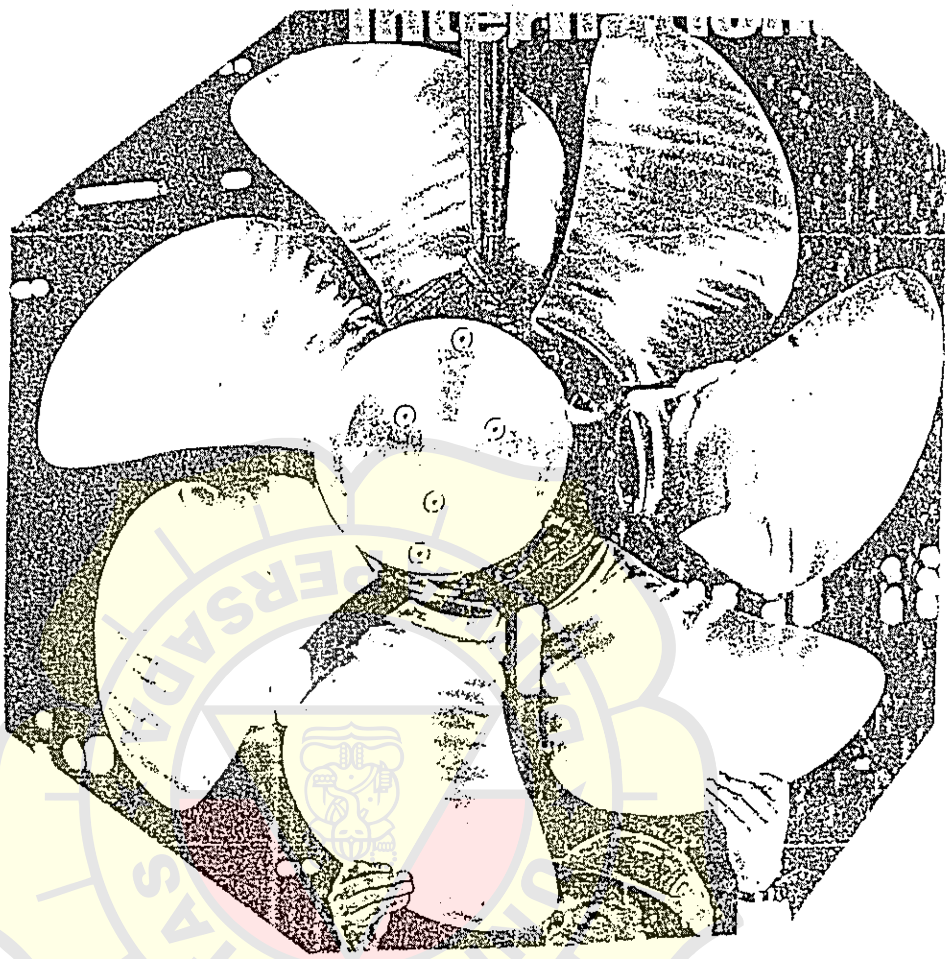


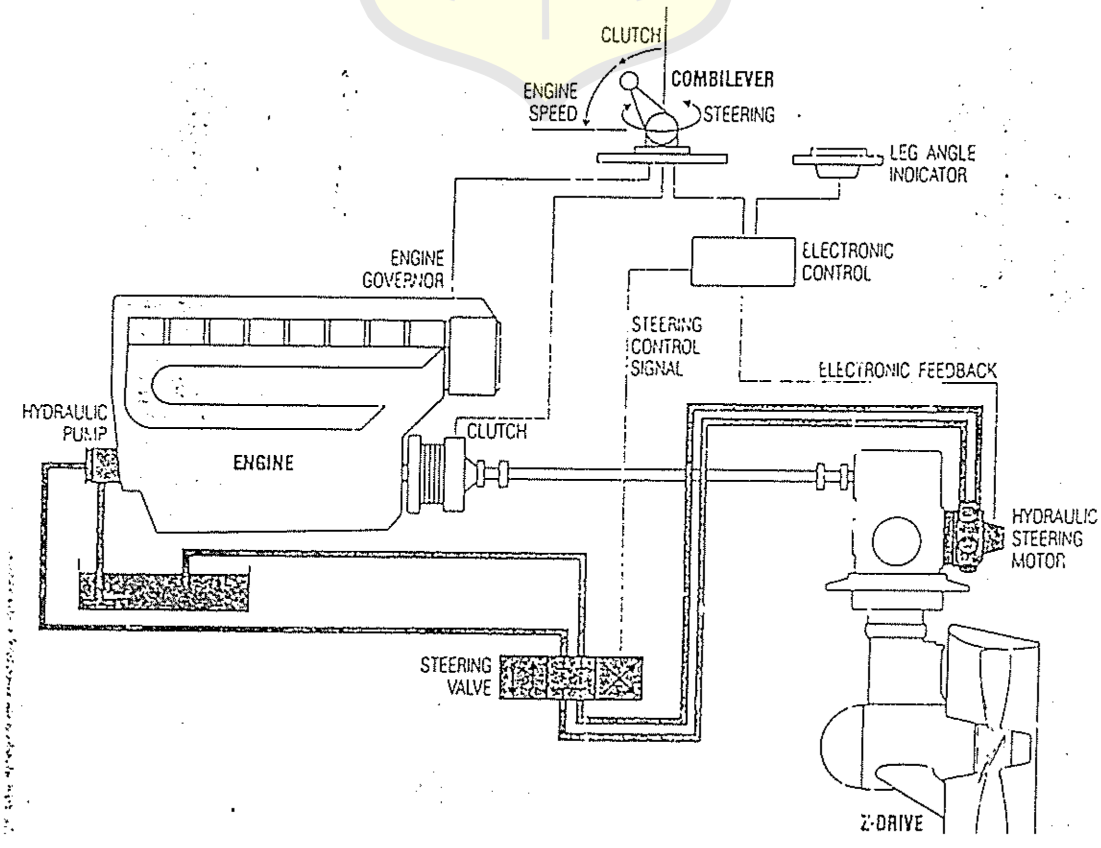




Propulsi Kapal

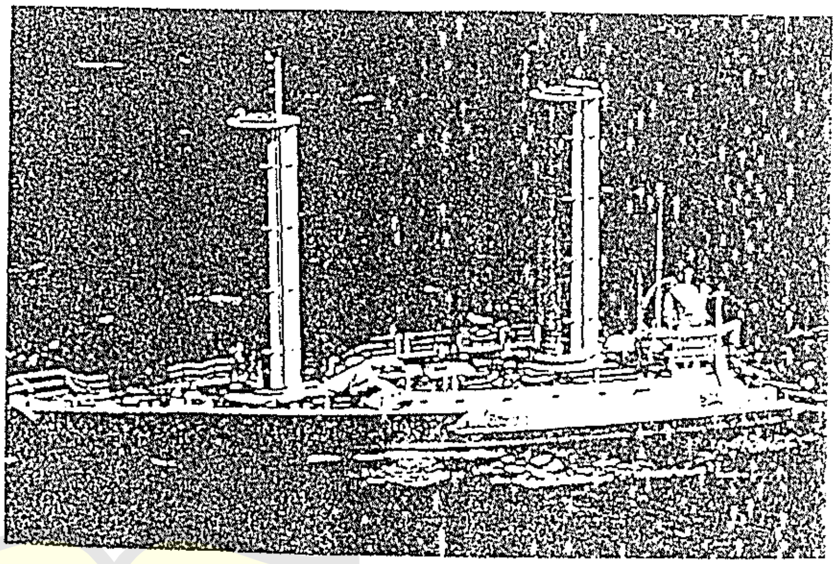
Alat Propulsi Mekanis : Screw Propeller
(baling-baling kapal)





Alat Propulsi Mekanis :

Fletner Rotor (masih uji coba)



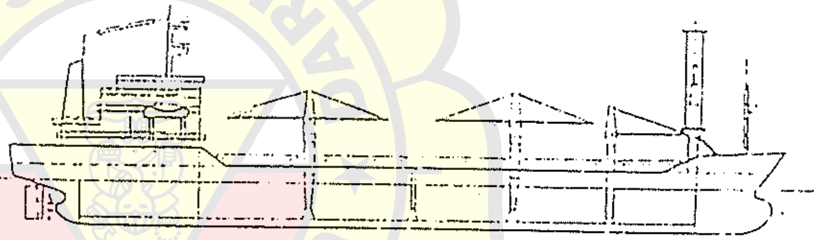
WIND ASSISTED PROPULSION

Rotor thruster dimensions
height 20 m, diameter 3.5 m
end plates diameter 5.4 m
Projected area ______ 70 m²
Max rotational speed ______ 300 rev/min
Max. power requirement ______ 25 kW
Average fuel cost savings ______ 6-8 %
Optimum condition savings ______ ~10 %
(8° B, beam wind)

BULK CARRIER
11800 DWT

MAIN DIMENSIONS

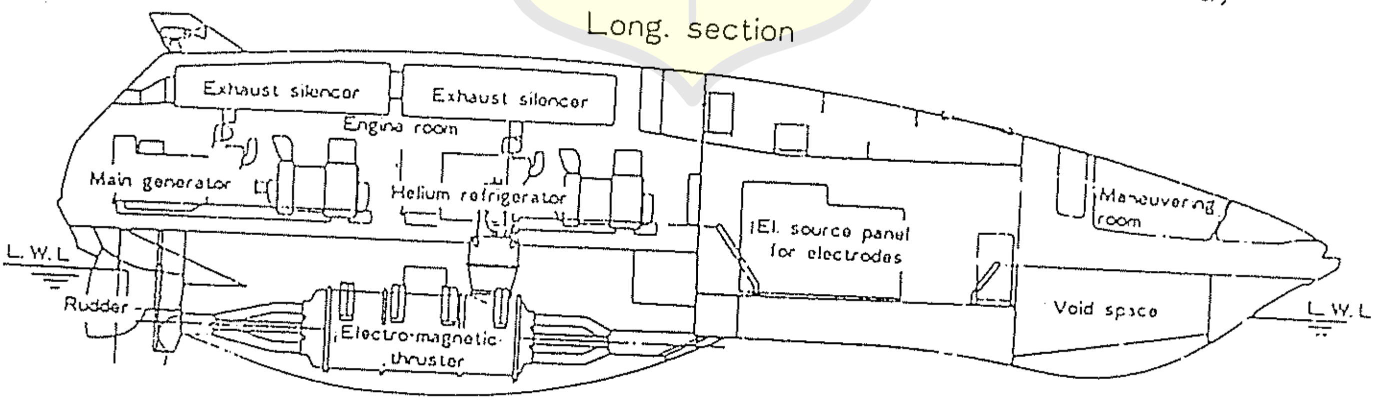
Lenght B.P. __ 132.4 m
Breadth ______ 19.4 m
Draugth ______ 8.2 m
Deadweight ______ 11800 t
Main engine ___ MAN/B&W H.CEGIELSKI
MCR = 3510 kW
Speed on trials ___ 13.9 kn
(85% MCR)



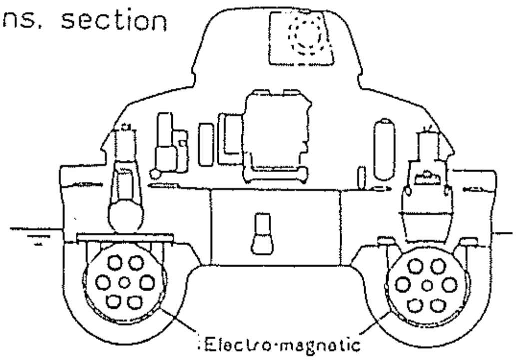
Fleming's left hand rule

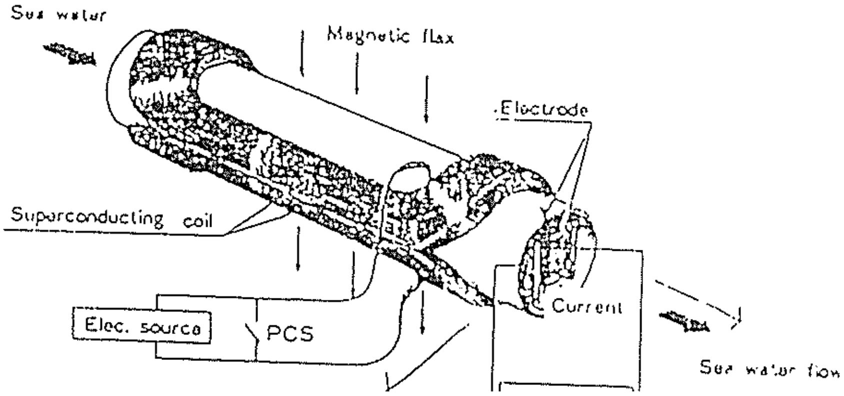
Alat Propulsi Mekanis : Superconductor Electro Magnetic Propulsion (masih uji coba)

Long. section



ans. section





Section 6

Propellers

A. General

1. Scope

These Rules apply to screw-propellers. Where a design is proposed to which the following Rules cannot be applied, special strength calculations are to be submitted to the Society. The tests to be carried out in each case are to be agreed with the Society. For the dimensions and materials of propellers for ships with ice classes, see Section 13

2. Documents for approval

Design drawings of propellers are to be submitted to the Society in triplicate[1] for examination. Drawings are to contain all the details necessary to verify compliance with the following Rules.

B. Materials

1. Approved materials

Propellers are to be made of seawater-resistant cast copper alloys or cast steel alloys with a minimum tensile strength of 440 N/mm², cf. Volume V Rules for Materials, Section 9, Part C, and Section 6, Part C and G. For the purpose of the following design Rules governing the thickness of the propeller blades, the requisite resistance to seawater of a cast copper alloy or cast steel alloy is considered to be achieved if the alloy used can be proved to withstand a fatigue test[2] under alternating bending stresses comprising 10^8 load cycles amounting to about 20% of the minimum tensile strength and carried out in a 3% NaCl solution, and if it can be proved that the fatigue strength under alternating bending stresses in natural seawater is not less than about 65% of the values established in 3% NaCl solution.

Grey cast iron with a minimum tensile strength of 250 N/mm² (GG 25 to DIN 1691) should be used only for spare propellers for short periods of service, and unalloyed or low alloy cast steels are primarily to be used only for navigating in icy waters, cf. Volume V.

2. Materials for blade retaining-bolts and studs

Unless protected against contact with seawater, the blade-retaining bolts or studs of assembled or controllable pitch propellers must likewise be made of materials which are resistant to seawater.

3. Novel materials.

Where it is proposed to use propeller materials whose serviceability is not attested by a sufficient period of practical experience, the Society must be provided with special proof of the suitability of such materials.

4. Material testing

The material of propellers and blade-retaining bolts or studs is to be tested in accordance with the Society's Rules for Materials.

C. Dimensions and Design of Propellers

1. Symbols and terms

A	[mm²]	Effective area of a shrink fit
B	[mm]	Developed blade width of cylindrical sections at radii 0,25 R, 0,35 R and 0,6 R
c	[-]	Coefficient for shrunk joints = 1,0 for engine and turbine gear transmissions = 1,2 for direct drives
C_{EP}, C_e	[-]	Ice class strengthening factors in accordance with Section 13 = 1,0 for machinery with no ice class
C_G	[-]	Size factor in accordance with formula (2)
C_{Dyn}	[-]	Dynamic factor in accordance with formula (3)
C_W	[-]	Characteristic value for propeller material as shown in Table 6.1 (corresponds to the minimum tensile strength R_m of the propeller material where this has been shown to possess sufficient fatigue strength under alternating bending stresses in accordance with paragraph B.1.)
C	[-]	Conicity of shaft ends $= \dfrac{\text{difference in taper diameter}}{\text{length of taper}}$
d	[mm]	Bolt-hole circle diameter of blade or propeller-fastening bolts
d_k	[mm]	Root diameter of blade or propeller-fastening bolts
D	[mm]	Diameter of propeller $= 2 \cdot R$

1) For Indonesian flag ships in quadruplicate, one for Government.

2) Sufficient fatigue strength under alternating bending stresses must be proved by a method recognized by the Society. See also Section F.2. of the Society's "Regulations for the Determination of Dynamic Stresses on Propellers", December, 1971.

Table 6.1 Characteristic values C_w for propeller materials

Material	Description[1]	C_w
Cu 1	Cast manganese brass	480
Cu 2	Cast manganese nickel brass	500
Cu 3	Cast nickel aluminium bronze	660
Cu 4	Cast manganese aluminium bronze	620
Fe 1	Unalloyed cast steel	380
Fe 2	Low-alloy cast steel	380
Fe 3	Martensitic cast chrome steel 13/1-6	600
Fe 4	Martensitic-autenitic cast steel 17/4	600
Fe 5	Ferritic-austenitic cast steel 24/8	600
Fe 6	Austenitic cast steel 18/8-11	500
Fe 7	Grey cast iron	200

1)For the chemical composition of the alloys, see the Society's Rules for Materials and Regulations for the Assessment and Repair of Defects on Propellers.

d_m	[mm]	Mean taper diameter
e	[mm]	Blade rake to aft $= R \cdot \tan \epsilon$
E_r	[-]	Thrust stimulating factor in accordance with formula (5)
f, f_1, f_2, f_3	[-]	Factors in formulae (11), (2), (4) and (3)
F_M	[N]	Bolt load
H	[mm]	Propeller blade face pitch at radii 0,25 R, 0,35 R and 0,6 R
H_m	[mm]	Mean effective propeller pitch on blade face for pitch varying with the radius

$$= \frac{\Sigma (R \cdot B \cdot H)}{\Sigma (R \cdot B)}$$

in which R, B and H are to be substituted by values corresponding to the pitch at the various radii.

J	[-]	Degree of advance
c	[-]	Coefficient for various profile shapes in accordance with Table 6.2
c'	[-]	Coefficient calculated by applying formula (6) where use is made of profile shapes other than those given in Table 6.2
c_T	[-]	Thrust coefficient
M	[mm]	2/3 of the leading-edge component of the blade width at 0,9 R, but at least 1/4 of the total blade width at 0,9 R for propellers with heavily raked blades.
	[mm]	Pull-up length when mounting propeller on taper
mech.	[mm]	Pull-up length at $t = 35°C$
$L_{temp.}$	[mm]	Temperature-related portion of pull-up length at $t < 35°C$
n_2	[min^{-1}]	Propeller speed in rev/min
P_w	[kW]	Shaft power
p	[N/mm^2]	Specific pressure in shrunk joint between propeller and shaft
Q	[N]	Peripheral force at mean taper diameter
S	[-]	Margin of safety against propeller slipping on taper = 2,8
t	[mm]	Maximum blade thickness of developed cylindrical section at radii 0,25 R, 0,35 R and 0,6 R
T	[N]	Propeller thrust
T_M	[Nm]	Impact moment
V_S	[kn]	Speed of ship
w	[-]	Wake factor
W_x	[mm^3]	Actual face modulus of developed cylindrical section referred to face blade pitch profiles about blade pitch line
Z	[-]	Total number of bolts used to retain one blade or propeller
z	[-]	Number of blades
α	[-]	Pitch angle of profile at radii 0,25 R, 0,35 R and 0,6 R

$$\alpha_{0,25} = \arctan \frac{1,27 \cdot H}{D}$$

$$\alpha_{0,35} = \arctan \frac{0,91 \cdot H}{D}$$

$$\alpha_{0,60} = \arctan \frac{0,53 \cdot H}{D}$$

α_A	[-]	Tightening factor for retaining bolts. and studs = 1,2 - 1,6 depending on the method of tightening used.
β_x	[-]	Factor for the section modulus of developed cylindrical section about blade pitch line for blade profiles in accordance with Table 6.2
β_x'	[-]	Factor for the section modulus of developed cylindrical section about blade pitch line for blade profiles other than those in Table 6.2
ϵ	[-]	Angle included by face generatix and normal
Θ	[-]	Half-conicity of shaft ends

$$= \frac{C}{2}$$

μ_o [-] Coefficient of static friction
= 0,13
for hydraulic oil shrunk joints
= 0,18
for dry shrunk joints

$R_{p\,0,2}$ [N/mm²] 0,2% proof stress of propeller material

R_{eH} [N/mm²] Yield strengths

R_m [N/mm²] Tensile strength of the material of fitted bolts or bolts

σ_{max}/σ_m [-] Ratio of maximum to mean stress at blade face

2. Calculation of blade thickness

2.1 At radii 0,25 R and 0,6 R, the blade thicknesses of solid propellers must, as a minimum requirement, comply with formula (1).

$$t = K_o \cdot k \cdot K_1 \cdot C_G \cdot C_{EP} \cdot C_{Dyn} \quad (1)$$

$$K_o = 1 + \frac{e \cdot \cos\alpha}{H} + \frac{n_2}{15000}$$

k as in Table 6.2

$$K_1 = \sqrt{\frac{P_w \cdot 10^5 \cdot \left(2.\dfrac{D}{H_m} \cdot \cos\alpha + \sin\alpha\right)}{n_2 \cdot B \cdot z \cdot C_w \cdot \cos^2\varepsilon}}$$

C_G [-] Size factor

$$1{,}1 \geq \sqrt{\frac{f_1 + D}{12{,}2}} \geq 0{,}85 \quad (2)$$

D to be inserted in [m]

f_1 = 7,2 for solid propellers

= 6,2 for separately cast blades of variable-pitch or built-up propellers

C_{EP} [-] Ice class strengthening factor in accordance with Section 13

= 1,0

for ships without ice class

C_{Dyn} [-] Dynamic factor

$$= \sqrt{\frac{(\sigma_{max}/\sigma_m - 1) + f_3}{0{,}5 + f_3}} \quad (3)$$

for $\dfrac{\sigma_{max}}{\sigma_m} > 1{,}5$

σ_{max} / σ_m can be roughly calculated from the thrust-stimulating factor E_T according to formula (5). (For a more accurate calculation of σ_{max} / σ_m see the "Regulations for the Determination of Dynamic Stresses on Propellers, 1971".)

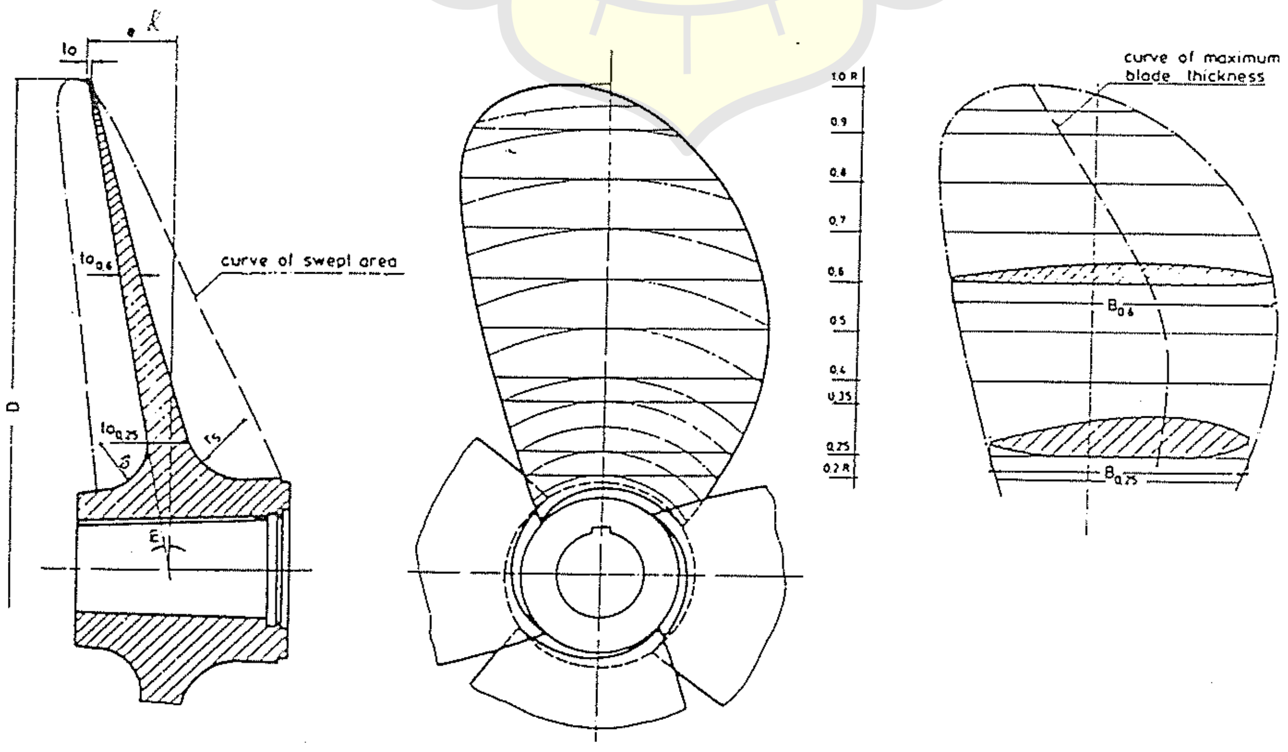


Fig. 6.1 Blade Sections

$$\frac{\sigma_{max}}{\sigma_m} = f_2 \cdot E_T + 1 \quad \text{where} \tag{4}$$

$$E_T = \frac{{}^{\sigma}K_T}{\sigma_J} \cdot \frac{J}{K_T} \tag{5}$$

$$= 4{,}3 \cdot 10^{-9} \; \frac{V_S \cdot n_2 \cdot (1 - w) \cdot D^3}{T}$$

f_2 = 0,4 - 0,6
for single-screw ships, the lower value applying to stern shapes with a wide propeller tip clearance and no rudder heel, and the larger value to sterns with little clearance and with rudder heel. Intermediate values are to be selected accordingly.

= 0,2
for twin-screw ships

f_3 = 0,2
for propeller materials which satisfy the requirements of B.1.

2.2 The blade thicknesses of controllable pitch propel-ers are to be determined at radii 0,35 R and 0,6 R by ıpplying formula (1).

ᐨor the controllable pitch propellers of tugs, trawlers ınd special-duty ships with similar operating conditions, he diameter/pitch ratio D/H_m for the maximum sta-ic bollard pull is to be used in formula (1).

'or other ships, the diameter/pitch ratio D/H_m appli-able to open-water navigation can be used in formula 1).

.3 The blade thicknesses calculated by applying for-ıula (1) are minima for the finish-machined propellers.

.4 The fillet radii at the transition from the face and ıe back of the blades to the propeller boss should cor-spond, in the case of three and four bladed propel-rs, to about 3,5% of the propeller diameter. For pro-:llers with a larger number of blades the maximum fil-: radii allowed by the propeller design should be ai-ed at, and the radii shall not in any case be made smal-· than $0{,}4 \cdot t_{0,25}$.

5 For blades of special shape, special mechanical engh calculations are to be submitted to the Society evidence that the propeller blades are adequately nensioned.

r profile shapes other than those given in Table 6.2, : following condition applies:

where

$$k^1 = K \cdot \sqrt{\frac{\beta_x}{\beta_x{}^1}} \quad \text{and} \quad \beta_x{}^1 = \frac{W_x}{t^2 \cdot B} \tag{6}$$

Table 6.2 Values of k for various profile shapes

Profile shape	Values of k		
	0,25 R	0,35 R	0,6 R
Segmental Profiles with circular arced back, $\beta_x = 0{,}12$	73	62	44
Segmental profiles with parabolic back, $\beta_x = 0{,}11$	77	66	47
Blade profiles as for Wageningen B Series propellers where $\beta_{x\,0.25} = 0{,}10$, $\beta_{x\,0.35} = 0{,}11$ $\beta_{x\,0.6} = 0{,}12$	80	66	44

Notes :
The Society reserves the right to specify an increase in the values of k in the case of special propellers where the blade width B at 0,25 R is $< 4 \cdot t$.

D. Controllable Pitch Propellers

1. Documents for approval

In the case of controllable pitch propellers, besides the design drawings of blades and propeller boss, general and sectional drawings of the entire controllable pitch propeller installation are to be submitted to the Society in triplicate.[1]. Diagrams of control systems and pipework are to be accompanied by a functional description. For new designs and controllable pitch propellers which are to be installed for the first time on ships with a BKI class, a description of the controllable pitch propeller system is to be submitted at the same time.

2. Testing of materials

In addition to the material tests specified in B.4, the Society reserves the right to require component parts of the pitch-adjusting mechanism including, in particular, those which are not accessible for shipboard repairs, to be tested in accordance with the Rules for Materials. Piping subject to pressure above 10 bar is to be tested in accordance with Section 11.

3. Hydraulic control equipment

Where the pitch-control mechanism is operated hydraulically, two mutually independent, power-driven pump sets are to be fitted. For propulsion plants up to 200 kW, one power-driven pump set is sufficient pro-

1) For Indonesian flag ships in quadruplicate, one for Government.

vided that, in addition, a hand-operated pump is fitted for controlling the blade pitch and that this enables the blades to be moved from the ahead to the astern position in a short enough time.

4. Pitch control mechanism

For the pitch-control mechanism, proof is required that, when subjected to impact moments T_M as defined by formula (7), the individual components still have a safety factor of 1,5 with respect to the yield strength of the materials used.

$$T_M = \frac{0{,}65 \cdot 10^6\, R_{p\,0,2} \cdot P_W \cdot I_M \cdot C_{EP}^2 \cdot C_G^2}{n_2 \cdot z \cdot C_W \cdot D} \qquad (7)$$

5. Blade retaining bolts and studs.

5.1 The root diameter of the bolts or studs used to attach blades is to be determined by applying formula (8):

$$d_k = 1{,}78 \cdot \sqrt{\frac{\alpha_A \cdot F_M}{R_{eH}}} \qquad (8)$$

$$F_M = \frac{0{,}28 \cdot 10^9 \cdot R_{p\,0,2} \cdot P_W \cdot C_{EP}^2 \cdot C_G^2}{n_2 \cdot z \cdot Z \cdot C_W \cdot d} \qquad (9)$$

5.2 The blade retaining bolts or studs are to be tightened in a controlled manner in such a way that the tension on the bolt or studs is about 60-70% of their yield strength.

The shank of blade retaining bolts or nuts may be designed with a minimum diameter equal to 0,9 times the root diameter of the thread. Blade retaining bolts must be secured against unintentional loosening.

6. Indicators

Controllable pitch propeller systems are to be provided with an engine room indicator showing the actual setting of the blades. Further blade position indicators are to be mounted on the bridge and in the engine room (see also Volume VII, and Volume IV, Section 8).

7. Failure of control system

Suitable devices are to be fitted to ensure that an alteration of the blade setting cannot overload the propulsion plant or cause it to stall.

Steps must be taken to ensure that, in the event of failure of the control system, the setting of the blades.

— does not change or
— assumes a final position slowly enough to allow the emergency control system to be put into operation.

8. Emergency control

Controllable pitch propeller systems must be equipped with means of emergency control enabling the controllable pitch propeller to remain in operation should the remote control system fail. It is recommended that a device be fitted which locks the propeller blades in the "ahead" setting.

E. Balancing and Testing

1. Balancing

The finished propeller and the blades of controllable pitch propellers are required to undergo static balancing.

2. Testing

The finished propeller is to be presented at the manufacturer's works to the BKI Surveyor for final inspection and verification of the dimensions. With regard to the assessment and the repair of defects on propellers, see the Society's Regulations for the Assessment and Repair of Defects on Propellers.

The Society reserves the right to require non-destructive tests to be conducted to detect surface cracks and casting defects.

In addition, controllabe pitch propeller systems are required to undergo pressure, tightness and operational tests.

F. Propeller Mounting

1. Tapered Mountings

1.1 Where the tapered joint between the shaft and the propeller is fitted with a key, the propeller is to be mounted on the tapered shaft in such a way that approximately the mean torque can be transmitted from the shaft to the propeller by the frictional bond. The "Reference data for the mounting length of keyed ships's propellers" issued by the Society can be used as a basis here. The propeller nut is to be secured in a suitable manner.

1.2 Where the tapered fit is effected by the hydraulic oil technique without the use of a key, the necessary pull-up distance on the tapered shaft is given by the expression.

$$L^{1)} = L_{mech.} + L_{temp.} \qquad (10)$$

where $L_{mech.}$ is determined according to the formulae of elasticity theory applied to shrunk joints for a specific pressure p [N/mm²] at the mean taper diameter found by applying formula (11) and for a water temperature of 35°C[2].

$$p = \frac{T}{A \cdot \Theta \cdot f}\left[\sqrt{1 + f\left(c^2 \cdot c_e^{\,6} \cdot \frac{Q^2}{T^2} + 1\right)} - 1\right] \qquad (11)$$

1) Where appropriate, allowance is also to be made for surface smoothing when calculating L.

2) The von Mises's equivalent stress based on the maximum specific pressure p and the tangential stress in the bore of the propeller hub may not exceed 75% of the 0,2% proof stress or yield strength of the propeller material.

$$\text{where } f = \left(\frac{\mu_O}{S \cdot \Theta}\right)^2 - 1 \geq 1{,}0 \qquad (12)$$

$$L_{temp.} = \frac{d_m}{C} \cdot 6 \cdot 10^{-6} \cdot (35 - t) \qquad (13)$$

t [°C] The temperature at which the propeller is mounted.

$L_{temp.}$ applies only to propellers made of bronze and austenitic steel.

The tapers of propellers which are mounted on the propeller shaft with the aid of the hydraulic oil technique should not be more than 1 : 15 or less than 1 : 20.

The propeller nut must be strongly secured to the propeller shaft.

2. Flange connections

For propellers attached by flanges, the root diameter d_k of the retaining bolts or studs is to be determined by applying formula (8). In this formula, the bolt force F_M shall not be less than that defined by formula (14).

$$F_M = \frac{0{,}28 \cdot 10^9 \cdot R_{p\,0,2} \cdot P_W \cdot C_{EP}^2 \cdot C_G^2 \cdot S}{n_2 \cdot z \cdot Z \cdot d \cdot C_W} \qquad (14)$$

S [-] Safety factor

$$= 2 - 0{,}3\left(\frac{d_f'}{d_f}\right)^2 \geq 1{,}5$$

$\left(\frac{d_f'}{d_f}\right)$ [-] Ratio of the actual and required diameters of any set pins to formula (5) in Section 4.

STONE MARINE SINGAPORE (PTE) LIMITED

128, HILLVIEW AVENUE SINGAPORE 2366
Telephone: 4684277 Cables: SHIPSPROP Telex: RS 24708 SM SING

OUR REF: NJD/9948

DATE: 28 NOV 90

ENGINE DETAILS: YANMAR 6 LAA-UTE
530 HP @ 1850 RPM
GEAR RATIO: 3.50/1

BLADE THICKNESS CALCULATION TO GERMANISCHER LLOYD RULES.

$D = 1200$

$C = 48$

$Ne = 537$ (METRIC)

$n = 528$

$z = 4$

$e = 105.8$

$\varepsilon = 10°$

$\eta_w = 0.97$

AT 0.25R	AT 0.60R
H = 931.4	H = 1090
B = 318.9	B = 393.6
c = 1.2	c = 1.0
k = 22	k = 12
α =	α = 25.7°

AT 0.60R

$$t = \left(1 + \frac{1.0 \times 105.8 \times 0.4338}{1090} + \frac{528}{15000}\right) 12 \sqrt{\frac{10^5 \left(2 \times \frac{1200}{1090} \times 0.9010 + 0.4338\right) 0.97 \times 537}{528 \times 393.6 \times 4 \times 48 \times 0.9698}}$$

$= 23.3$ mm

T 0.60 DESIGNED TO: 28.6 mm

NOTE
- MIN. TENSILE STRENGTH OF PROP. MATERIAL $G_T = 480\ N/mm^2$, AND TO BE COVERED WITH MATERIAL TESTING CERTIFICATE FROM BKI.
- GEOMETRY, SHAPE AND STATIC BALANCING TEST OF PROPELLER TO BE VERIFIED AND WITNESSED BY BKI SURVEYOR.

AT 0.25R

$$t = \left(1 + \frac{1.2 \times 105.8 \times 0.7020}{931.4} + \frac{528}{15000}\right) 22 \sqrt{\frac{10^5 \left(2 \times \frac{1200}{1090} \times 0.7122 + 0.7020\right) 0.97 \times 537}{528 \times 318.9 \times 4 \times 48 \times 0.9698}}$$

$= 48.3$ mm

T0.25 DESIGNED TO: 50.4 mm

CALCULATED BY: STEPHEN YEO.

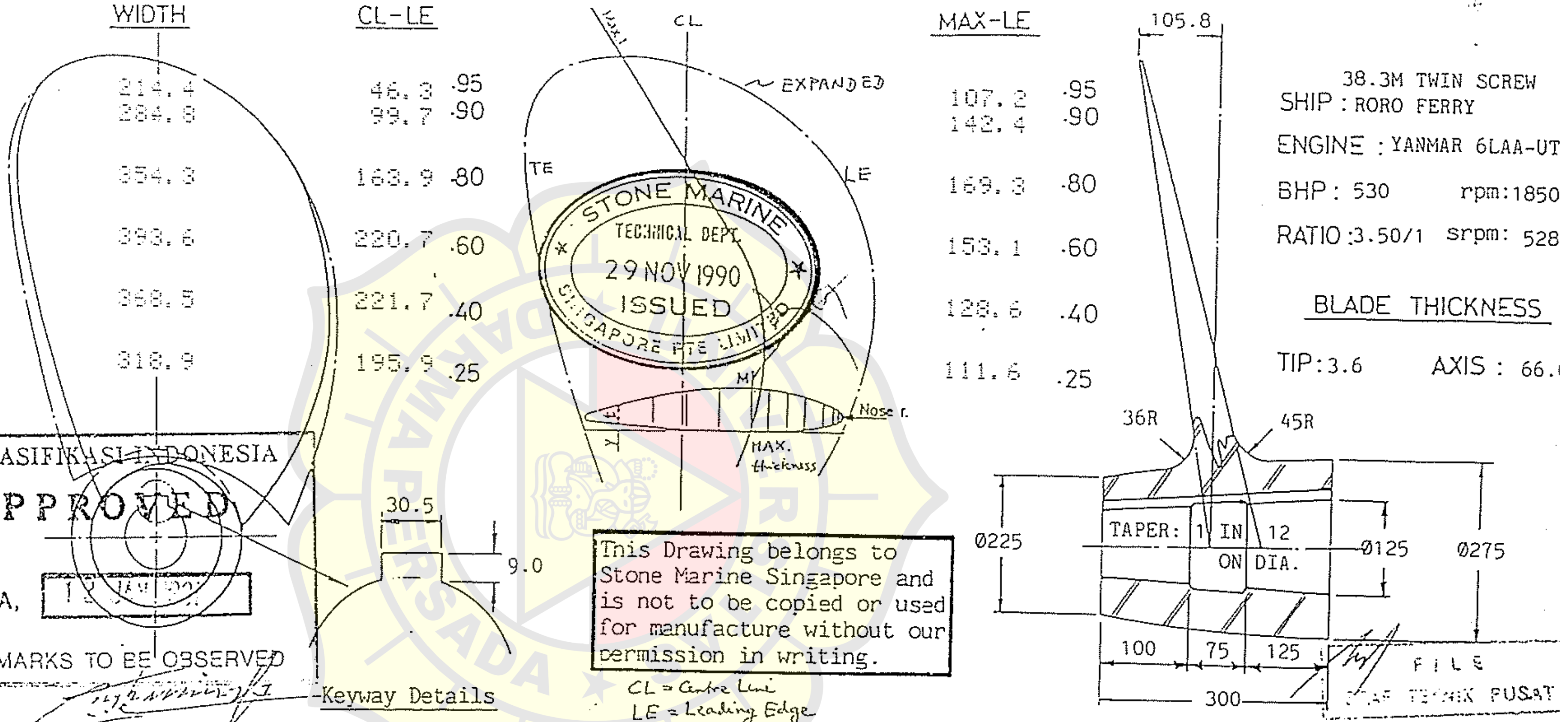
STONE MARINE SINGAPORE PTE. LTD. 128 HILLVIEW AVENUE SINGAPORE 23
FAX: 65-760038, Tel: 7600188 Telex: RS 24708 SM SING.

R	TE-CL	WIDTH	CL-LE	MAX-LE
1.0	79.3			
.95	168.1	214.4	46.3	107.2
.90	185.0	284.8	99.7	142.4
.80	190.3	354.3	163.9	169.3
.60	172.9	393.6	220.7	153.1
.40	146.8	368.5	221.7	128.6
.25	123.0	318.9	195.9	111.6

38.3M TWIN SCREW
SHIP : RORO FERRY
ENGINE : YANMAR 6LAA-UT
BHP : 530 rpm:1850
RATIO :3.50/1 srpm: 528

BLADE THICKNESS

TIP:3.6 AXIS : 66.



BIRO KLASIFIKASI INDONESIA
APPROVED
JAKARTA, 1?

RED REMARKS TO BE OBSERVED

-Keyway Details

This Drawing belongs to Stone Marine Singapore and is not to be copied or used for manufacture without our permission in writing.

CL = Centre Line
LE = Leading Edge
TE = Trailing Edge

STONE MARINE TECHNICAL DEPT. 29 NOV 1990 ISSUED SINGAPORE PTE LIMITED

FILE
... TEKNIK PUSAT

Blade Thicknesses (t) and Face Offsets in mm

R	100	80	60	40	20	0	20	40	60	80	90	95	100	Nose r.
.95	0.0	3.0	4.8	6.0	6.5	6.7	6.5	6.0	4.8	3.0	2.0	1.5	0.0	
.90	0.0	4.4	6.9	8.6	9.5	9.8	9.5	8.6	6.9	4.4	3.0	2.2	0.0	
.80	0.0	6.6	10.9	13.7	15.5	16.1	15.6	13.9	11.3	7.6	5.1	3.6	0.0	
.60	0.0	11.5	19.2	24.4	27.6	28.6	28.0	25.9	21.5	14.8	9.8	6.7	0.0	1.3
y	1.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.1	0.6	2.9	
.40	0.0	17.0	28.2	35.5	39.8	41.0	40.0	37.0	31.7	22.9	15.9	11.2	0.0	3.3
y	7.3	2.5	0.6	0.0	0.0	0.0	0.0	0.1	0.7	2.4	4.1	5.5	10.0	
.25	0.0	18.6	32.1	42.0	48.3	50.4	49.2	45.3	38.0	26.7	18.4	13.0	0.0	5.7
y	14.0	7.7	4.2	1.8	0.4	0.0	0.2	1.0	3.0	6.7	9.6	11.5	17.1	

R	PITCH
.95	1090.0
.90	1090.0
.80	1090.0
.60	1090.0
y	
.40	1035.5
y	
.25	931.4
y	

DIAMETER	1200 mm
PITCH	1090 mm
B.A.R.	0.60
BLADES	4RH & 4LH
MATERIAL	MN BRONZE
SURVEY	B.K.I.
DATE	29 NOV 90
DRAWING NO.	NJD/9948/T/A