BAB VI
PENUTUP

Kesempurnaan dari suatu hasil penulisan adalah merupakan tujuan yang ingin dicapai penulis. Untuk itu penulis telah berusaha semaksimal mungkin dengan bantuan dan bimbingan dari dosen pembimbing.

Tetapi dalam hal ini penulis menyadari bahwa dalam penulisan masih banyak terdapat kekurangan dan kesalahan, maka dari itu penulis berharap adanya sumbangan pikiran untuk perbaikan dalam mengerjakan tugas mesin kapal ini.

VI.1. Kesimpulan

Dari hasil uraian diatas sebelumnya dimana mencakup dari pada pendahuluan, perencanaan perhitungan Mesin Induk dan Mesin Bantu. Maka penulis dapat menarik suatu kesimpulan yang berhubungan dengan perancanaan mesin kapal ferry 1350 GRT.

Adapun kesimpulan tersebut adalah sebagai berikut :

- Untuk menentukan besarnya daya motor induk kapal sebagai penggerak utama, maka faktor kecepatan, jarak pelayaran, serta dimensi dari kapal mempunyai pengaruh yang sangat besar.
- Didalam perencanaan kamar mesin, tidak terlepas dari adanya asumsi – asumsi yang diberikan untuk mempermudah dalam perhitungan dengan tidak mengabaikan tanggung jawab secara teknis, ekonomis dan peraturan – peraturan yang ada, sehingga hasil perhitungan dapat mendekati keadaan yang sebenarnya.
- Pemilihan mesin bantu tergantung dari jumlah daya yang harus disuplai pada kondisi operasi kapal yang berbeda – beda.
- Penempatan posisi mesin induk, mesin bantu serta peralatan – peralatan lainnya sangat berpengaruh pada stabilitas kapal.

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LAMPIRAN

SKOCI

STANDART UKURAN SEKOCI BERMOTOR :

L	B	H	Kapasitas	Jumlah orang	Berat sekoci dari kayu	Berat sekoci dari plat	Berat motor	Berat perlengkapan	Berat total
8,00	2,60	1,16	14,5	34	1700	1900	820	460	2550
8,50	2,60	1,16	15,4	33	1800	2100	820	480	2925
9,00	2,70	1,22	17,8	46	1900	2300	870	510	3450
9,50	2,80	1,22	19,4	50	2100	2500	1120	530	3750

STANDART UKURAN SEKOCI KERJA

L1	L	B	H	Kapasitas	Jumlah orang	Berat penumpang	Berat perlengkapan	Berat sekoci	Berat total
3,60	3,76	1,55	0,6	2,0	4	300	60	300	660
3,80	3,96	1,65	0,66	2,5	5	375	60	360	795
4,00	4,16	1,75	0,70	3,0	6	450	60	420	930
4,50	4,66	1,00	0,78	3,5	7	5,25	70	450	1045
5,00	5,18	1,85	0,72	4,0	8	600	70	500	1170
5,50	5,68	1,90	0,75	4,7	9	675	80	600	1355
6,00	6,18	2,00	0,80	5,8	11	825	80	700	1605

LAMPIRAN

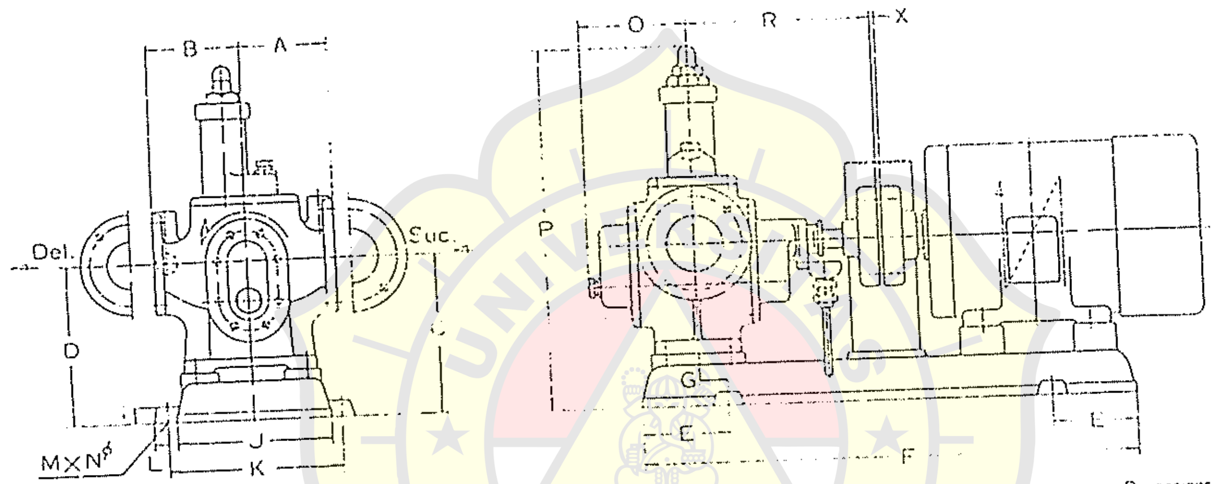
SKOCI

STANDART UKURAN SEKOCI OLEH BOT (BOARD OF TRADE) ENGLAND

Tabel II

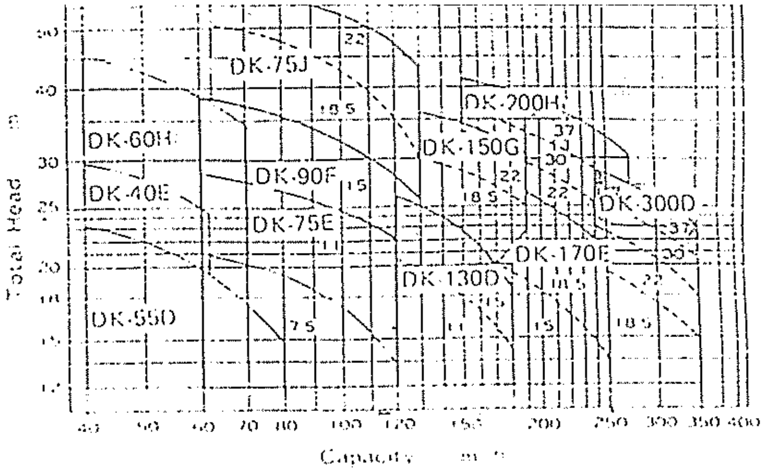
L. B. H (m)	L. B. H (ft3)	Kapasitas (ft3)	Jumlah orang	berat sekoci (kg)	Berat Orang (kg)	berat perlengkapan (kg)	Total berat (kg)
9,4 x 2,74 x 1 x 1,114	30 x 9 x 3,75	607	60	2205	4500	356	7061
8,84 x 2,74 x 1,10	29 x 8,75 x 3,60	545	54	1976	4050	356	6382
8,53 x 2,59 x 1,07	28 x 8,50 x 3,50	500	50	1824	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	15	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
4,90 x 1,75 x 0,70	16 x 5,75 x 2,30	127	12	475	900	127	1484

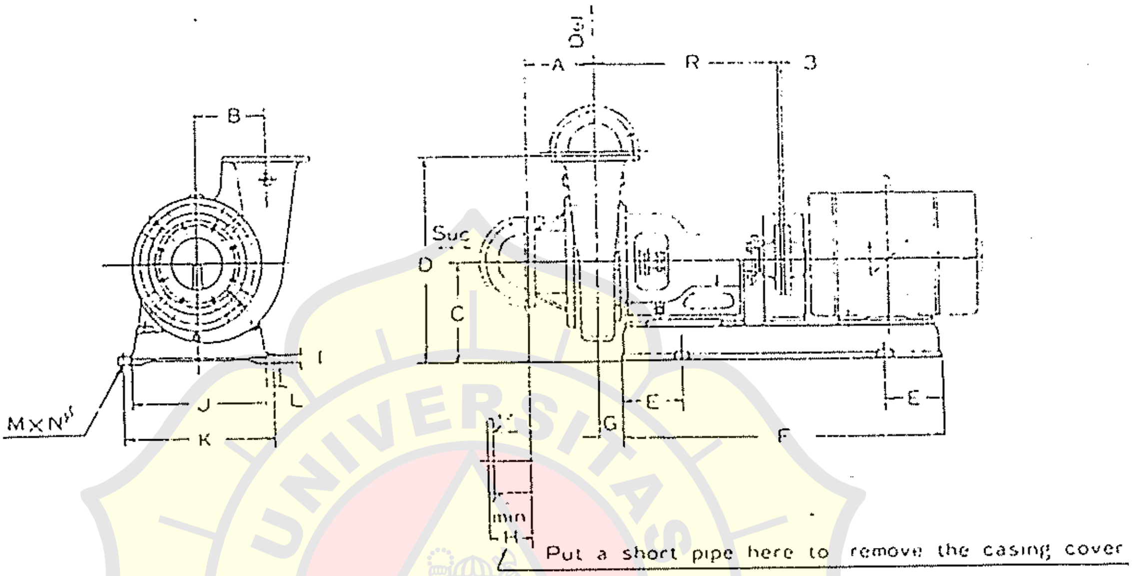
Type



Dimensions -- mm

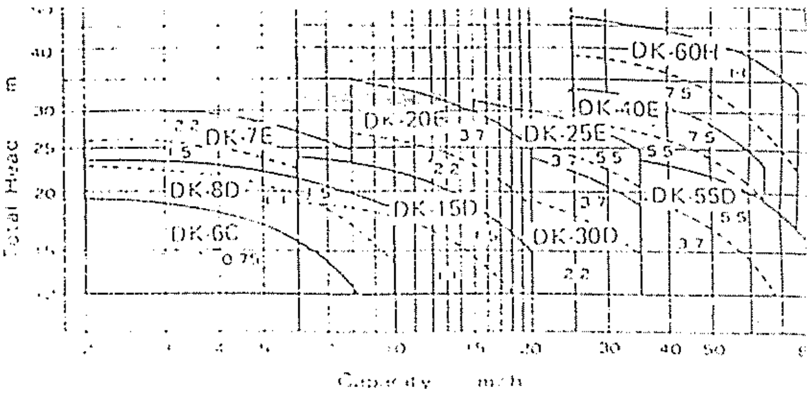
Type	No of Rev (r.p.m.)	Motor (kw)	Bore Suc	Bore Del	A	B	C	D	E	F	G	I	J	K	L	M	N	O	P	R	X	Pump Weight (kg)
05B	1200	0.4	25	20	110	110	200	200	50	365	21	20	170	140	15	4	15	55	330	122	3	30
1B	1200	0.75	32	25	95	95	195	195	100	500	60	25	260	300	23	4	15	89	380	175	3	47
2B	1200	0.75 1.5	40	32	95	85	195	195	100	500	60	25	260	300	23	4	15	96	385	175	3	50
3B	1200	0.75 1.5	50	40	100	100	210	210	100	550	35	25	260	300	23	4	15	98	412	185	3	55
4B	1200	1.5 2.2	65	50	105	105	225	225	100	600	20	25	280	320	23	4	15	144	470	200	3	70
5B	1200	1.5 2.2	65	50	105	105	225	225	100	600	20	25	280	320	23	4	15	111	470	200	3	72
6B	1200 / 1200	2.2 / 3.7	65	50	105 / 105	105 / 105	225 / 230	225 / 230	100 / 130	600 / 640	20 / 60	25	280 / 300	320 / 330	23 / 25	4	15	55 / 63	470 / 475	200	3	72 / 78
7.5B	1200	2.2 3.7	80	65	110	110	245	245	150	700	50	25	300	350	23	4	15	122	500	220	3	90
10B	1200	2.2 3.7	80	65	140	140	230	230	125	700	30	25	310	350	23	4	15	137	530	245	3	95
12B	1200 / 1200	3.7 / 5.5	80	65	140	140	230 / 230	230 / 230	125 / 125	700 / 750	30	25	310	350	23	4	15	152 / 152	530	250	3	103
15B	1200 / 1200	3.7 5.5 / 7.5	80	65	150	150	260 / 270	260 / 270	100 / 150	750 / 950	0. 20 / 65	25 / 30	310 / 350	350 / 390	23 / 22	4	15 / 19	165	565 / 575	285	3	140
20B	1200 / 1200	5.5 / 7.5	100	80	160	160	270	270	150 / 200	800 / 900	50 / 100	25	310 / 350	350 / 390	23	4	15	193	618	315	3	135
25B	1200 / 1200	5.5 / 7.5	100	80	160	160	270	270	150 / 200	800 / 900	50 / 100	25 / 25	310 / 350	350 / 390	23	4	15	193	618	315	3	135
30B	1200	7.5 11	125	100	175	175	320	320	200	1000	80	30	370	420	25	4	19	210	688	357	3	200
40B	1200	7.5 11	150	125	190	190	330	330	200	1050	40. 57	35	490	540	25	4	23	240	780	387	3	220
50B	1200	11 15	150	125	235	235	390	390	250	1100	90	30	450	500	25	4	19	225	835	385	3	365
65B	1200	15 18.5	150	125	235	235	390	390	200	1150	30	30	500	550	25	4	19	245	840	415	3	370
80C	900	18.5 22	150	125	250	250	460	460	200	1500	0	45	570	[illegible]	30	6	23	377	1005	577	3	490
100D	720	22 30	200	175	350	350	370	370	300	1730	-30	35	620	680	28	4	23	523	1395	730	4	550

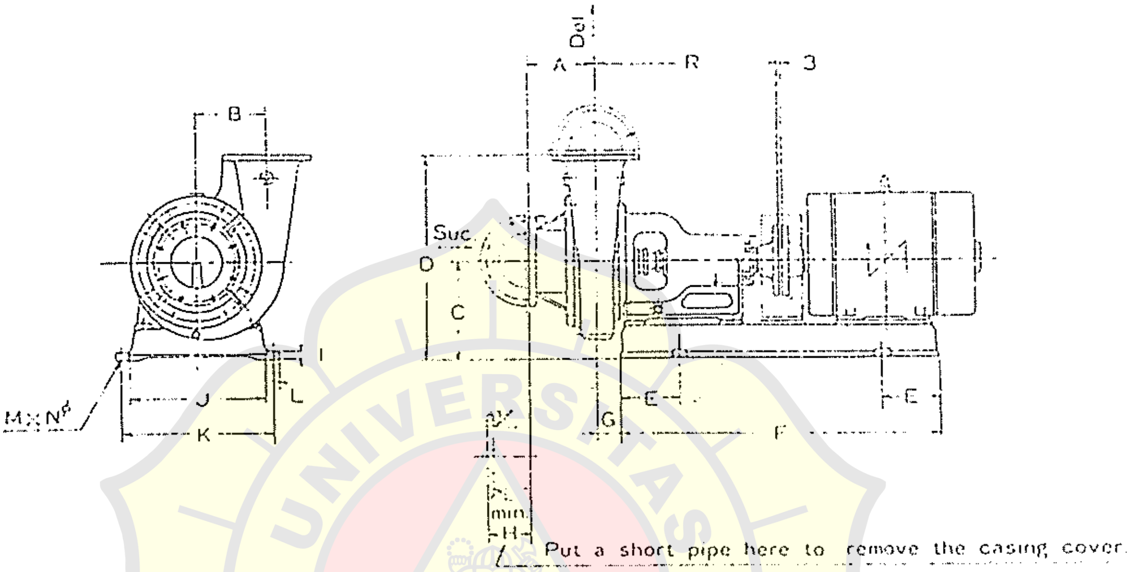




Type	Motor (kw)	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	R	Pump Weight (kg)
DK-75E	7.5	125	125	113	165	202	460	150	780	30	100	25	300	340	23	4	15	420	160
	11			113	165	227	485	175	860	30	100	25	360	400	23	4	15	420	160
DK-90F	15	125	125	150	180	231	505	175	900	33	100	25	360	400	23	4	15	423	210
DK-75J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

DK type





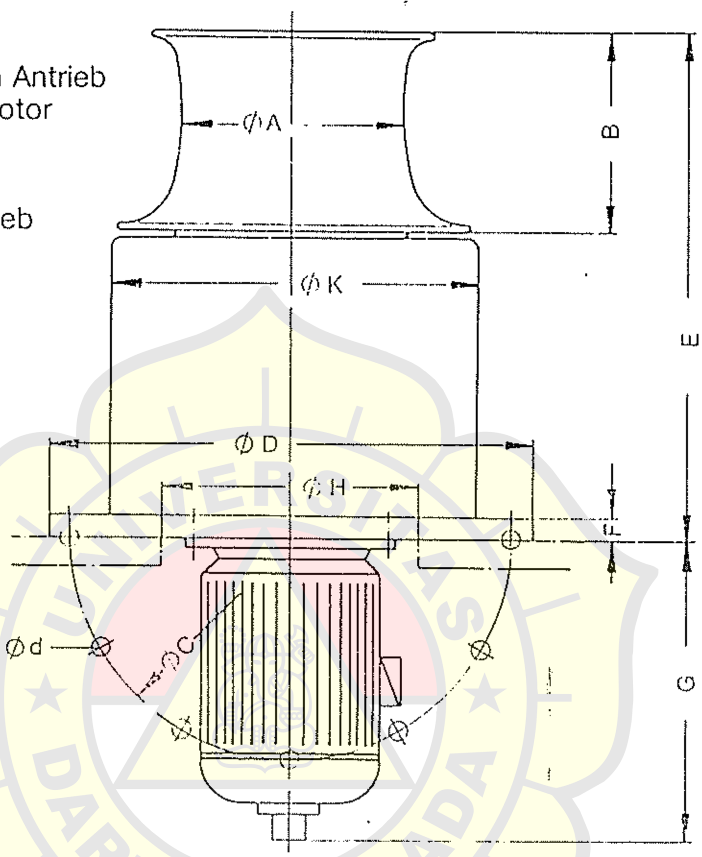
Type	Motor (kw)	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	R	Pump Weight (kg)
DK-6C	0.75	32	32	96	110	165	315	100	550	35	100	25	200	240	23	4	15	350	50
DK-8D	1.5			100	130	165	315	100	600	30	100	25	200	240	23	4	15	350	55
DK-7E	1.5	32	32	100	135	165	335	100	600	30	100	25	200	240	23	4	15	350	60
	2.2			100	135	175	345	100	620	35	100	25	240	280	23	4	15	350	60
DK-15D	1.5	50	50	108	125	165	325	100	600	30	100	25	200	240	23	4	15	350	95
DK-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
DK-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
DK-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
DK-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
DK-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
DK-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

NAUTISERVO B.V.
Industrieweg 64
2651 BD Berkel - Holland
Tel. 01891-39 55
P.O. Box 19
2650 AA Berkel-Rodenrijs
Telex 26774 Nauti-NL

		Blatt-Nr.
	Verholspill Capstan	HP-0049

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

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



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

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

Konstruktionsänderungen vorbehalten / Subject to changes of design

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

5-81

NAUTISERVO B.V.

Industrieweg 64
2651 BD Berkel - Holland
Tel. 01891-3955

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

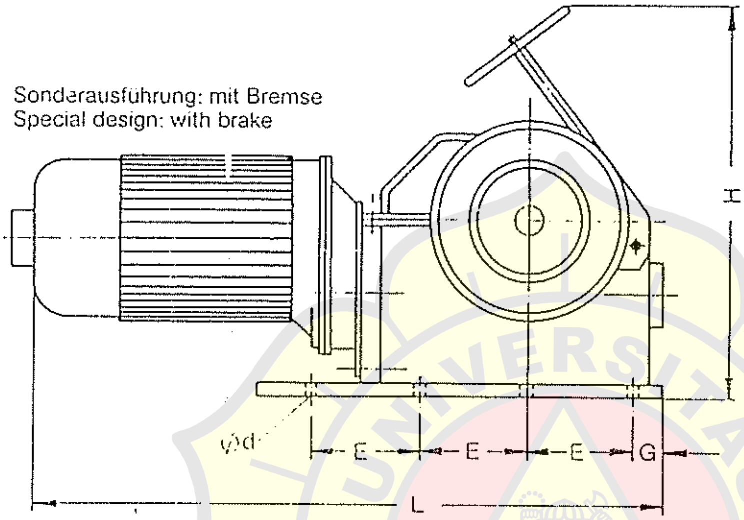
Ankerwinde
Windlass

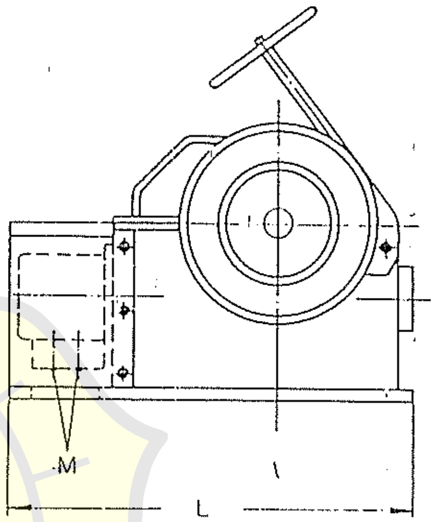
Blatt-Nr.
HP-0048

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

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

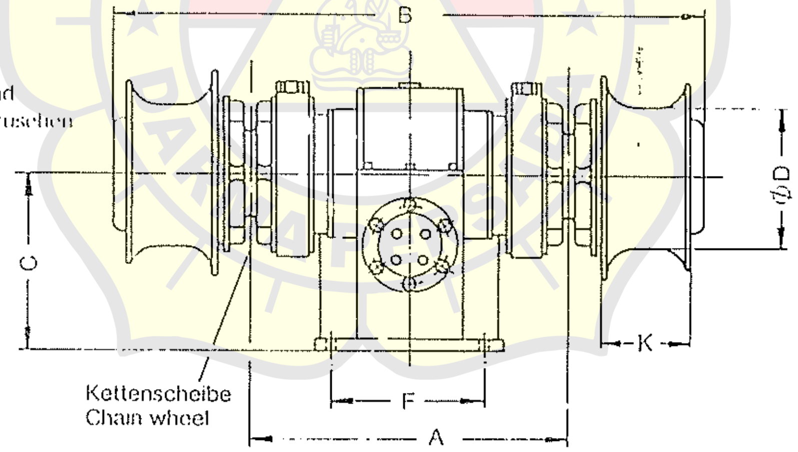
Sonderausführung: mit Bremse
Special design: with brake





Achtung!
Kettenstopper sind vom Kunden vorzusehen

Chain stopper to be supplied by yard or customer.



Kettenscheibe
Chain wheel

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

Konstruktionsänderungen vorbehalten / Subject to changes of design

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

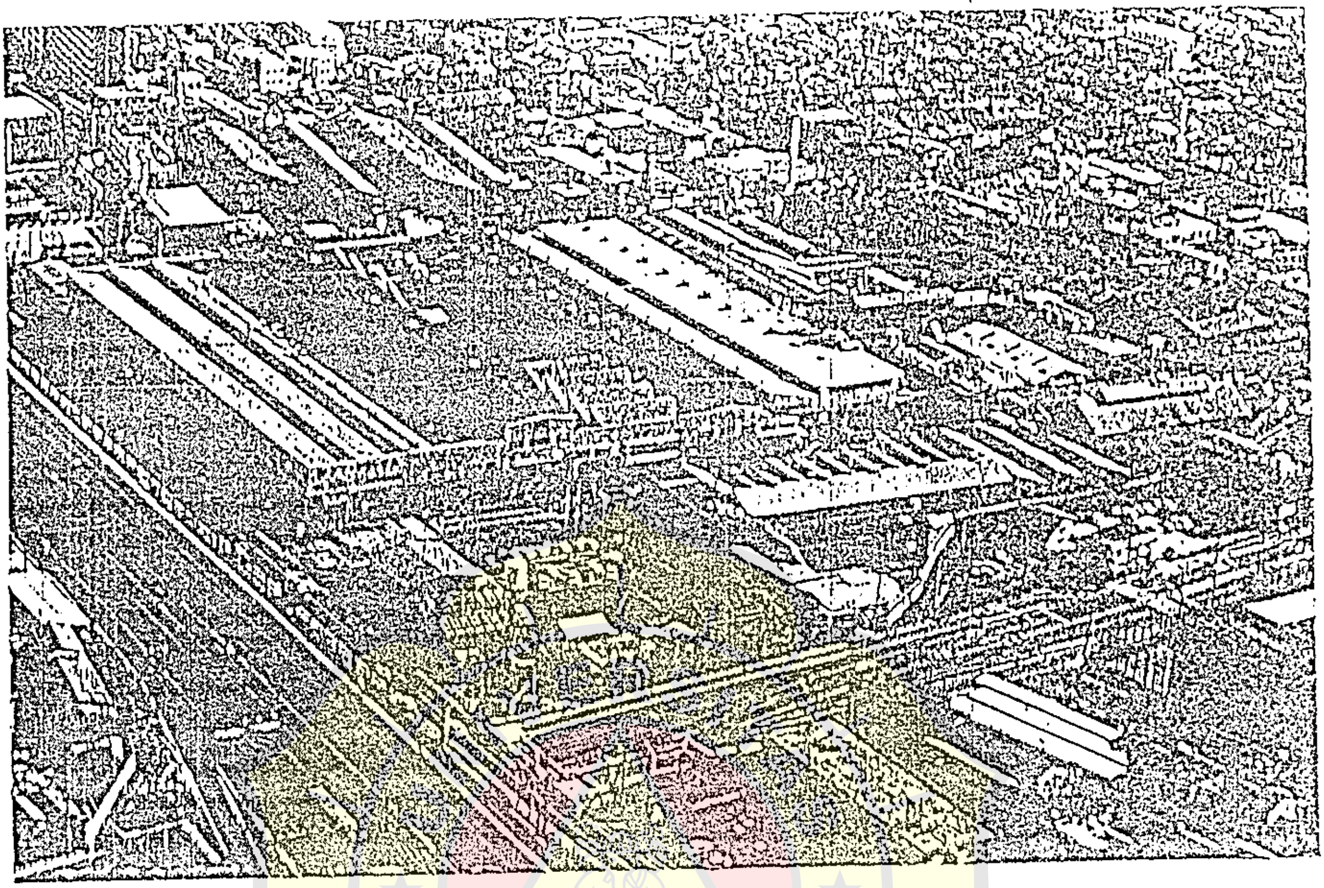
V = Hubgeschwindigkeit
Lifting speed

Z = max. Zugkraft
max. lifting power

Maße/Dimensions = [mm]

..'81

Yanmar Diesel Engine Co., Ltd.



President: Tadao Yamaoka
Capital: ¥1,200,000,000.–
Number of Employees: 4,568
Head Office:
Address: 1-32, Chayamachi, Kita-ku, Osaka
530, Japan
Tel: 06-372-1111
Telex: J63436
Cable Address: YANMAR OSAKA
Overseas Operations Division:
Address: 1-1, 2-chome, Yaesu, Chuo-ku,
Tokyo 104, Japan
Tel: 03-275-4932
Fax: 03-272-0687
Telex: 0222-4733 YANMAR J
Cable Address: YANMAR TOKYO
Main Products:
Main Propulsion and Auxiliary Diesel

Overseas Offices:
• Amsterdam Liaison Office
Rivierstaete, Amsteldijk 166, 1079 LH Amsterdam
Tel: 020-443035
Fax: 020-446951
• Singapore Liaison Office
600 House, 257 Jalan Ahmad Ibrahim, Singapore 2262
Tel 2642377
Telex: RS 35854 YANMAR
• Yanmar Diesel America Corp.
901 Corporate Grove Drive,
Buffalo Grove-Deerfield, IL 60015
Tel: 312-541-1900
Fax: 312-541-2161
Telex: 190179

Outline of Company

Yanmar is the world's largest manufacturer of non-vehicular use diesel engines for marine-use, agriculture and industry. The marine mains cover the range from 5 to 3600 hp, auxiliaries from 5.6 to 4800 hp. In addition, Yanmar produces a wide range of high quality marine gears for all kinds of vessel. Yanmar engines are renowned for their durability, reliability, compactness, lightness of weight, easy handling and fuel efficiency. They incorporate many of the latest technologies, including advanced fuel injection systems, ceramic parts and other new materials, laser-hardening, electronic controls and advanced sensor technologies. Yanmar's marine engine production facilities have been accredited by Lloyd's Register of Shipping, the American Bureau of Shipping and Nippon Kaiji Kyokai. Yanmar marine engines are sold in more than 100 countries, with service bases in 19 key oceanic ports around the world.

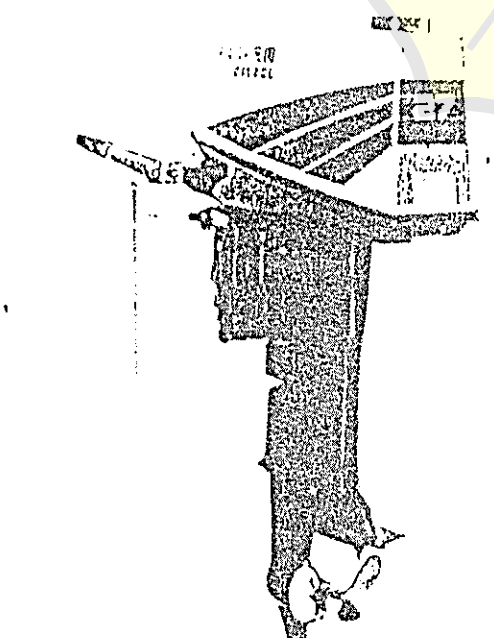
• Yanmar Diesel Engine (U.S.A.), Inc.
1424 North Hundley Street, Anaheim, CA 92806
Tel. 714-630-9415
Fax. 714-630-1567
Telex: 27-5330

The Yanmar Main Engine Line-up with True Marine Lineage

Yanmar's marine engines are designed to meet the realities of today's marine business, to get more nautical mileage out of every drop of fuel, and give more offshore service with less pause for maintenance. These abilities run through the great range of Yanmar marine engines up to 3,200 hp, serving abroad practically every type of vessel as mains, auxiliaries, and as inboard power for pleasure boats.

Fishing Boats

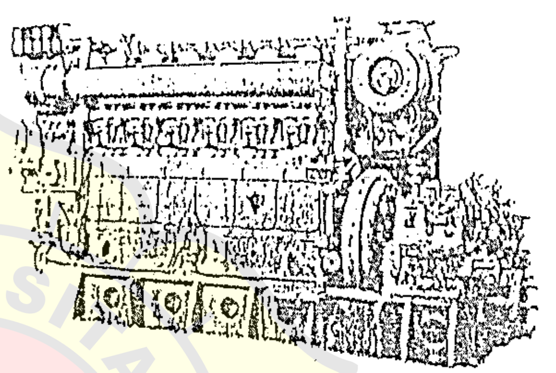
Yanmar is one of the world's most active companies in the supply of diesel engines to fishing boats. We export a major percentage of Japanese marine diesel under 300 hp for fishing boats. Yanmar's ruged and maintenance-free small horsepower diesels are specially suited for powering traditional boats including dhows, canoes and other wooden boats which have been long used in various areas of the world. Other advanced marine diesels precisely meet the demands of modern fishing with their stable power and all-speed performance. A recent technical breakthrough at Yanmar is the perfection of a lightweight diesel outboard which provide unheard of fuel savings and greatly boost productivity in coastal fishing. Together with our quality FRP fishing boats, produced in some 200 models, we are busy promoting modern coastal and offshore fishing in various part of the world.



Model D27

Work Boats

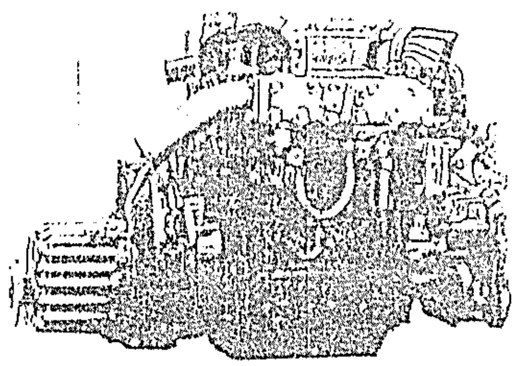
Yanmar's new high output, turbocharged marine diesel line-up is finding increasing and ever more diversified outlets for use in all kinds of cargo boats, tugs, dhows, passenger ferries and high-speed utility boats, thanks to powerful performance, lightness of weight, compactness and cost-cutting features. The new fuel pump delivers at high pressure with such incredibly precise timing that it can achieve almost 100% combustibility.



Model M220

Pleasure Boats

The name of Yanmar has a long and high reputation in major boating countries around Europe and U.S. for quality diesels. Yanmar's pleasure boat engine range covers a total of 28 models including V-drive, Angle-drive and Sail-drive models. Yanmar offers compact, tough, direct injection diesels in the max. output range from 9 hp to 140 hp to meet every type of marine pleasure requirement. Yanmar is also the fastest emerging force in the pleasure craft field, developing numerous sophisticated high speed power boats, especially for game fishing, all fitted with renowned Yanmar pleasure boat diesels.



Model 3GM30

Yanmar Diesel Engine Co., Ltd.

* Asterisk shows DIN6270B

Specifications (Mains)

Model	No. of cylinders	Bore x stroke: mm	Cont. rating output: hp/rpm	Dry weight: kg	Dimensions L x W x H: mm
D18	2	70 x 70	18/4500	74, 79	722 x 460 x 1286
D27	3	70 x 70	27/4500	82, 87	722 x 460 x 1368
D36	3	70 x 70	36/4500	114, 118	730 x 460 x 1433
1GM10	1	75 x 72	*9/3600	76	547 x 410 x 485
2GM20	2	75 x 72	*18/3600	106	638 x 455 x 495
2GM20F	2	75 x 72	*18/3600	114	643 x 482 x 545
3GM30	3	75 x 72	*27/3600	130	735 x 455 x 495
3GM30F	3	75 x 72	*27/3600	138	740 x 455 x 545
3HM35	3	80 x 85	*34/3400	158	786 x 485 x 617
3HM35F	3	80 x 85	*34/3400	167	791 x 475 x 638
2TD	2	100 x 115	26/2100	330	874 x 526 x 805
3TD	3	100 x 115	39/2100	400	1009.5 x 526 x 825
4TD	4	100 x 115	52/2100	510	1235.5 x 526 x 854.5
4JH2E	4	82 x 86	*50/3600	228	888.4 x 565 x 634.5
4JH2-TE	4	82 x 86	*62/3600	234	888.4 x 565 x 634.5
4JH2-HTE	4	82 x 86	*75/3600	244	888.4 x 565 x 643.5
4JH2-DTE	4	82 x 86	*88/3600	244	888.4 x 565 x 643.5
3ESDE	3	120 x 135	56/1800	680	1255 x 689 x 967
4ESDE	4	120 x 135	74/1800	800	1473 x 694 x 1015
4LH-TE	4	100 x 110	*110/3300	340	1058.2 x 649 x 726
4LH-HTE	4	100 x 110	*140/3300	350	1058.2 x 649 x 726
4CHE	4	105 x 125	70/2300	655	1372 x 688 x 1025
6CHE	6	105 x 125	105/2300	785	1661 x 690 x 1018
6CH-HTE	6	105 x 125	155/2300	830	1658 x 690 x 1056
6CH-DTE	6	105 x 125	190/2300	880	1658 x 690 x 1091
6CH-UTE	6	105 x 125	255/2550	915	1551.5 x 730 x 1111
4KDE	4	145 x 170	110/1450	1430	1701 x 731 x 1164
6KDE	6	145 x 170	165/1450	2263	2495 x 741 x 1202
6HA(M)E	6	130 x 150	165/2000	1145	1529 x 885 x 1097
6HA(M)-HTE	6	130 x 150	240/2000	1230	1529 x 939 x 1233
6HA(M)-DTE	6	130 x 150	300/2000	1250	1529 x 939 x 1233
6GH-UTE	6	117.9 x 140	350/2300	1335	1762 x 898.5 x 1247.5
6LAAE	6	148 x 165	240/1900	2120	1703 x 921 x 1275.5
6LA-DTE	6	148 x 165	400/1800	1890	1719 x 1012.5 x 1358
6LAA-UTE	6	148 x 165	530/1850	1890	1719 x 1012.5 x 1358
8LAA-DTE	Vee 8	148 x 165	530/1800	2420	1983 x 1430 x 1420
8LAA-UTE	Vee 8	142 x 165	650/1850	2420	1983 x 1430 x 1420
12LAA-DTE	Vee 12	148 x 165	800/1800	3300	2553 x 1430 x 1470
12LAA-UTE	Vee 12	148 x 165	1000/1850	3300	2553 x 1430 x 1470
S165	6	165 x 210	200/1200	3100	2574.5 x 1043 x 1586
S165-T	6	165 x 210	300/1300	3150	2574.5 x 1070 x 1586
S165-UT	6	165 x 210	450/1300	3600	2697 x 1070 x 1586
S165-ST	6	165 x 210	550/1300	3780	2697 x 1070 x 1586
S165-ET	6	165 x 210	600/1350	3780	2847 x 1070 x 1586

(Continued on next page)

Yanmar Diesel Engine Co., Ltd.

1 : 2,56

MAIN ENGINE

Model	No. of cylinders	Bore x stroke: mm	Cont. rating output: hp/rpm	Dry weight: kg	Dimensions L x W x H: mm
S185-UT	6	185 x 230	500/900	6000	3457 x 1170 x 1974
S185-ST	6	185 x 230	550/900	6040	3457 x 1170 x 1974
S185-ET	6	185 x 230	600/900	6090	3457 x 1170 x 2029
S185A-ET	6	185 x 230	650/950	6090	3457 x 1170 x 2029
M200D-UN	6	200 x 260	600/750	7350	3504 x 1120 x 1958
M200D-SN	6	200 x 260	660/750	7350	3508 x 1120 x 1958
M200-DN	6	200 x 260	600/900	6900	3411 x 1120 x 1958
M200-SN	6	200 x 260	800/900	7350	3508 x 1120 x 2013
M200-EN	6	200 x 260	900/900	7700	3650 x 1120 x 2013
M220-UN	6	220 x 300	1000/800	9100	3884 x 1162 x 2038
M220-SN	6	220 x 300	1100/800	9100	3910 x 1162 x 2143
M220-EN	6	220 x 300	1200/800	9100	3903 x 1162 x 2143
T240-UT	6	240 x 310	1000/750	10700	4131 x 1203 x 2244
T240-ST	6	240 x 310	1100/750	10700	4131 x 1203 x 2244
T240-ET	6	240 x 310	1200/750	10700	4131 x 1203 x 2244
T240A-ET	6	240 x 310	1400/800	11930	4303 x 1203 x 2244
T260-UT	6	260 x 330	1300/700	12930	4691 x 1443 x 2388
T260-ST	6	260 x 330	1400/700	12930	4691 x 1443 x 2388
T260-ET	6	260 x 330	1500/700	13080	4691 x 1443 x 2447
T260A-ET	6	260 x 330	1600/750	13300	4691 x 1443 x 2447
Z280-SN	6	280 x 360	1600/650	16550	4947.5 x 1540 x 2658
Z280-EN	6	280 x 360	1800/650	16550	4947.5 x 1540 x 2658
Z280A-EN	6	280 x 360	2000/720	17950	4947.5 x 1540 x 2658
Z280A-GN	6	280 x 360	2200/720	20900	5417 x 1481 x 2658
8Z280-SN	8	280 x 360	2100/650	22580	6288 x 1914 x 2651
8Z280-EN	8	280 x 360	2400/650	22580	6288 x 1914 x 2651
8Z280A-EN	8	280 x 360	2600/720	24330	6288 x 1914 x 2651
8Z280A-GN	8	280 x 360	2900/720	26600	6288 x 1914 x 2651
12T26-ST	Vee 12	260 x 330	2800/700	24800	5989 x 1857 x 2726
12T26-ET	Vee 12	260 x 330	3000/700	25200	6127 x 1982 x 2726
12T26A-ET	Vee 12	260 x 330	3200/750	25200	6127 x 1982 x 2850.5
MF24-HT	6	240 x 420	600/420	12450	4166 x 1363 x 2465
MF24-DT	6	240 x 420	700/420	12450	4166 x 1363 x 2465
MF24-UT	6	240 x 420	800/420	12450	4166 x 1363 x 2465
MF24-ST	6	240 x 420	950/420	12700	4237 x 1363 x 2465
MF26-HT	6	260 x 500	1000/350	16400	4607 x 1485 x 2840
MF26-ST	6	260 x 500	1200/380	17300	4897 x 1485 x 2840
MF28-HT	6	280 x 450	1000/380	18500	4803 x 1577 x 2880
MF28-DT	6	280 x 450	1100/380	19400	5093 x 1577 x 2880
MF28-UT	6	280 x 450	1200/380	19400	5093 x 1577 x 2880
MF28-ST	6	280 x 450	1300/380	19600	5093 x 1577 x 2925
MF33-DT	6	330 x 620	1600/300	26000	5297 x 1785 x 3440
MF33-UT	6	330 x 620	1800/300	26000	5297 x 1785 x 3440
MF33-ST	6	330 x 620	2000/300	26000	5297 x 1785 x 3440

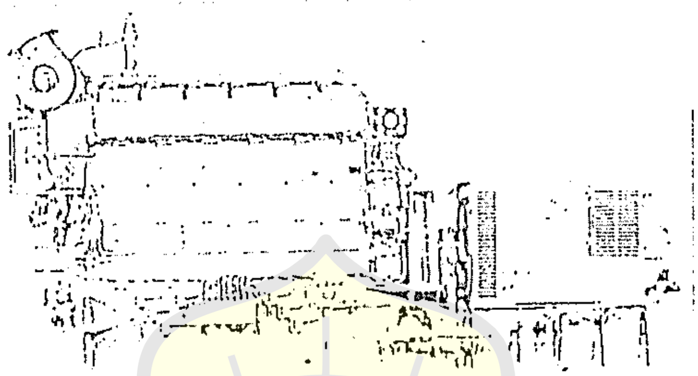
Yanmar Diesel Engine Co., Ltd.

Model	No. of cylinders	Bore x stroke: mm	Cont. rating output: hp/rpm	Dry weight: kg	Dimensions L x W x H: mm
S165L-DN	6	165 x 210	330/1000, 420/1200	2900*	2214 x 1070 x 1581
S165L-UN	6	165 x 210	360/1000, 480/1200	2900*	2214 x 1070 x 1581
S165L-SN	6	165 x 210	420/1000, 540/1200	2900*	2214 x 1070 x 1581
S165L-EN	6	165 x 210	480/1000, 600/1200	2900*	2214 x 1070 x 1581
S185DL-UT	6	185 x 230	420/720, 420/750	5400	2687 x 1134 x 1749
S185DL-ST	6	185 x 230	480/720, 480/750	5400	2687 x 1134 x 1749
S185DL-ET	6	185 x 230	540/720, 540/750	5400	2687 x 1134 x 1749
S185L-UT	6	185 x 230	540/900, 540/1000	5400	2687 x 1134 x 1749
S185L-ST	6	185 x 230	600/900, 600/1000	5000	2687 x 1134 x 1749
S185L-ET	6	185 x 230	660/900, 660/1000	5000	2687 x 1134 x 1749
S185AL-UT	6	185 x 230	600/1200	5000	2687 x 1134 x 1749
S185AL-ST	6	185 x 230	660/1200	5000	2687 x 1134 x 1749
M200L-UN	6	200 x 260	600/720, 600/750	5800	2919 x 1120.5 x 1844
M200L-SN	6	200 x 260	660/720, 660/750	5800	2923 x 1120.5 x 1880
M200L-EN	6	200 x 260	750/720, 750/750	5800	2977 x 1120.5 x 1883
M200L-UN	6	200 x 260	720/900, 720/1000	5800	2919 x 1120.5 x 1844
M200AL-SN	6	200 x 260	830/900, 830/1000	5800	2977 x 1120.5 x 1883
M200AL-EN	6	200 x 260	900/900, 900/1000	5800	2977 x 1120.5 x 1833
M220L-UN	6	220 x 300	830/720, 830/750	7200	3165 x 1162 x 2070
M220L-SN	6	220 x 300	900/720, 900/750	7200	3165 x 1162 x 2070
M220L-EN	6	220 x 300	1000/720, 1000/750	7200	3204 x 1162 x 2143
M220AL-UN	6	220 x 300	1000/900, 1000/1000	7200	3165 x 1162 x 2070
M220AL-SN	6	220 x 300	1100/900, 1100/1000	7200	3211 x 1162 x 2143
M220AL-EN	6	220 x 300	1200/900, 1200/1000	7200	3204 x 1162 x 2143
T240L-UT	6	240 x 310	1000/720, 1000/750	8400	3394 x 1203 x 2244
T240L-ST	6	240 x 310	1100/720, 1100/750	8400	3381 x 1203 x 2244
T240L-ET	6	240 x 310	1200/720, 1200/750	8400	3381 x 1203 x 2244
T240AL-ST	6	240 x 310	1200/900	8400	3381 x 1203 x 2244
T240AL-ET	6	240 x 310	1300/900	8400	3381 x 1203 x 2244
T260L-ST	6	260 x 330	1300/720, 1300/750	9600	3711 x 1313 x 2388
T260L-ST	6	260 x 330	1400/720, 1400/750	9600	3711 x 1313 x 2388
T260L-ET	6	260 x 330	1500/720, 1500/750	9750	3891 x 1343 x 2447
Z280L-UT	6	280 x 360	1600/720, 1600/750	12400	3895 x 1540 x 2658
Z280L-ST	6	280 x 360	1800/720, 1800/750	12600	3895 x 1540 x 2658
Z280L-ET	6	280 x 360	2000/720, 2000/750	12600	3895 x 1540 x 2658
8Z280L-UT	8	280 x 360	2200/720, 2200/750	16200	4888 x 1575 x 2651
8Z280L-ST	8	280 x 360	2400/720, 2400/750	16400	4888 x 1575 x 2651
8Z280L-ET	8	280 x 360	2600/720, 2600/750	16400	4888 x 1575 x 2651
12T26L-ST	Vee 12	260 x 330	2600/720, 2600/750	18600	4266 x 2360 x 2726
12T26L-ST	Vee 12	260 x 330	2800/720, 2800/750	18600	4266 x 2360 x 2726
12T26L-ET	Vee 12	260 x 330	3000/720, 3000/750	19000	4404 x 2360 x 2805
12ZL-UT	Vee 12	280 x 340	3200/720, 3200/750	26000	5108 x 2730 x 2937
12ZL-ST	Vee 12	280 x 340	3600/720, 3600/750	26500	5108 x 2730 x 3005
16ZL-ST	Vee 16	280 x 340	4800/720, 4800/750	34000	6216 x 2894 x 3286

Yanmar Diesel Engine Co., Ltd.

Marine Auxiliaries

Yanmar serves a great range of on-board power demands with its line-up of auxiliary engines. In the cost conscious world of marine trasporta tion, profitable fleet operation has been more important and to meet these needs Yanmar offers economical inboard co-generation systems, including MDO/HFO operation and various fuel pre-treatment systems. The lowest possible fuel consumption is a standard feature of all Yanmar marine auxiliaries, and all can stand extremely long hours continuous operation with major overhauls.



Model T220L

*The size excluding the fresh water cooler.

Specifications (Auxiliary)

Model	No. of cylinders	Bore x stroke: mm	Cont. rating output: hp/rpm	Dry weight: kg	Dimensions L x W x H: mm
1GM10L	1	75 x 72	6/3000, 7/3600	67	385 x 470 x 502
2GML	2	72 x 72	11/3000, 13/3600	97	481 x 470 x 512
2GMFL	2	72 x 72	11/3000, 13/3600	105	481 x 470 x 562
3GML	3	72 x 72	17/3000, 20/3600	122	566 x 470 x 512
3GMFL	3	72 x 72	17/3000, 20/3600	130	566 x 470 x 562
3HML	3	75 x 85	21.5/3000, 25.5/3600	147	606 x 470 x 617
3HMFL	3	75 x 85	21.5/3000, 25.5/3600	156	606 x 470 x 646
4CHLN	4	105 x 125	38/1500, 50/1800	500	1089 x 840 x 1021
6CHLN	6	105 x 125	62/1500, 74/1800	620	1378 x 840 x 1022
6CHL-TN	6	105 x 125	74/1500, 100/1800	640	1378 x 840 x 1242
6CHL-HTN	6	105 x 125	100/1500, 120/1800	670	1378 x 840 x 1241
3KDL	3	145 x 170	70/1200, 85/1500	940	1082 x 715 x 1182
4KDL	4	145 x 170	95/1200, 115/1500	1150	1351 x 715 x 1182
5KDL	5	145 x 170	120/1200, 140/1500	1345	1553 x 725 x 1182
6KFL	6	145 x 170	145/1200, 170/1500	1780	1798 x 837 x 1200
6KFL-T	6	145 x 170	185/1200, 225/1500	1890	2114 x 943 x 1448
6KFL-HT	6	145 x 170	220/1200, 270/1500	1930	2114 x 985 x 1448
6KFL-UT	6	145 x 170	270/1200, 300/1500	2050	2139 x 1106 x 1568
6HAL-N	6	130 x 150	125/1500, 150/1800	1150	2000 x 962 x 1278
★ 6HAL-TN	6	130 x 150	150/1500, 180/1800	1200	1910 x 856 x 1322
6HAL-HTN	6	130 x 150	200/1500, 240/1800	1250	1910 x 873.5 x 1322
6HAL-DTN	6	130 x 150	250/1500, 300/1800	1270	1910 x 873.5 x 1322
6LAAL-DTN	6	148 x 165	360/1500, 420/1800	1950	1766 x 1061 x 1529.5
8LAAL-DTN	Vee 8	148 x 165	480/1500, 560/1800	2400	1983 x 1316 x 1420
12LAAL-DTN	Vee 12	148 x 165	720/1500, 840/1800	3500	2553 x 1430 x 1470
S165L	6	165 x 210	200/1200	2700*	2181 x 1070 x 1581
S165SL-T	6	165 x 210	200/1200	2750*	2181 x 1070 x 1581
S165L-HN	6	165 x 210	270/1000, 360/1200	2900*	2214 x 1070 x 1581

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Model	Cycle	Cylinders	Bore (mm)	Stroke (mm)	Mean Piston Speed (m/s)	Speed (rev/min)	Output (kW/cyl)	Output range (kW)	Bmep (bar)	Sfoc (g/kWh)
Yanmar Diesel Engine Co Ltd 2-1-1 Yaesu 2-chome, Chuo-ku, Tokyo 105, Japan. Telex: 0222-4733 Yanmar J.										
S165 series	4	6L	165	210	8.40-9.45	1200-1350		147-441	5.57-14.85	210
6N165-EN	4	6L	165	232	10.83	1400		588	17.28	192
S185 series	4	6L	185	230	6.90-7.28	900-950		405-478	14.83-15.60	215
M200 series	4	6L	200	260	6.50-7.80	750-900		441-662	14.69-18.36	193
M220 series	4	6L	220	300	8.00	800		736-883	16.44-19.73	193
T240 series	4	6L	240	310	7.75-8.27	750-800		809-1030	15.69-18.72	192
T260 series	4	6L	260	330	7.70-8.47	700-770		1030-1177	17.12-18.26	201
6N260 series	4	6L	260	360	9.00	750		1177-1471	16.74-20.93	190
Z280 series	4	6L	280	360	7.80-8.64	650-720		1324-1471	18.74-18.80	197
8Z280 series	4	8L	280	360	7.80-8.64	650-720		1765-1912	18.74-18.80	[illegible]
6N280 series	4	6L	280	380	9.12	720		1471-1839	17.81-22.26	189
8N280 series	4	8L	280	380	9.12	720		1912-2354	17.36-21.37	189
6N330 series	4	6L	330	440	9.09	620		2207-2574	19.29-22.50	[illegible]
8N330 series	4	8L	330	440	9.09	620		2942-3310	19.29-21.70	188
MF24 series	4	6L	240	420	5.88	420		441-588	11.28-15.04	197
MF26 series	4	6L	260	500	5.83-6.67	350-400		588-956	11.90-18.36	194
MF29 series	4	6L	290	520	6.59	380		1030-1177	16.10-18.40	193
MF33 series	4	6L	330	620	6.20-6.82	300-330		1177-1618	15.09-18.86	190

XXX

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.

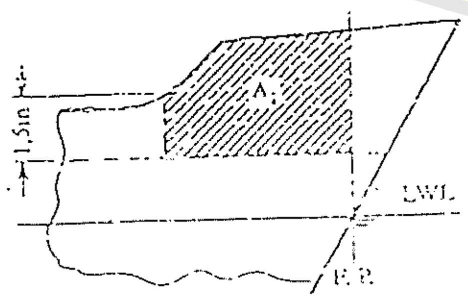


Fig. 18.1

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 pins and fittings, is not to be less than 60 percent of the total mass of the anchor.

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

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

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

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

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

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

Table 18.2 Anchor, Chain Cables and Ropes

No. for Reg.	Equipment numeral Z	Stockless anchor: Bower anchor: Number [1]	Stockless anchor: Bower anchor: Mass per anchor [kg]	Stockless anchor: Stream anchor: Mass per anchor [kg]	Stud link chain cables: Bower anchors: Total length [m]	Stud link chain cables: Bower anchors: Diameter d_1 [mm]	Stud link chain cables: Bower anchors: Diameter d_2 [mm]	Stud link chain cables: Bower anchors: Diameter d_3 [mm]	Stream wire or chain for stream anchor: Length [m]	Stream wire or chain for stream anchor: Br. load[2] [kN]	Recommended ropes: Towline: Length [m]	Recommended ropes: Towline: Br. load[2] [kN]	Recommended ropes: Mooring ropes: Number	Recommended ropes: Mooring ropes: Length [m]	Recommended ropes: Mooring ropes: Br. load[2] [kN]
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
101	up to - 50	2	120	40	165	12.5	12.5	12.5	80	65	180	100	3	80	35
102	50 - 70	2	180	60	220	14	12.5	12.5	80	65	180	100	3	80	35
103	70 - 90	2	240	80	220	16	14	14	85	75	180	100	3	100	40
104	90 - 110	2	300	100	247.5	17.5	16	16	85	80	180	100	3	110	40
105	110 - 130	2	360	120	247.5	19	17.5	17.5	90	90	180	100	3	110	45
106	130 - 150	2	420	140	275	20.5	17.5	17.5	90	100	180	100	3	120	50
107	150 - 175	2	480	165	275	22	19	19	90	110	180	100	3	120	55
108	175 - 205	3	570	190	302.5	24	20.5	20.5	90	120	180	110	3	120	60
109	205 - 240	3	660		302.5	26	22	20.5			180	130	4	120	65
110	240 - 280	3	780		330	28	24	22			180	150	4	140	70
111	280 - 320	3	900		357.5	30	26	24			180	175	4	140	80
112	320 - 360	3	1020		357.5	32	28	24			180	200	4	140	85
113	360 - 400	3	1140		385	34	30	26			180	225	4	140	95
114	400 - 450	3	1290		385	36	32	28			180	250	4	140	100
115	450 - 500	3	1440		412.5	38	34	30			180	275	4	160	110
116	500 - 550	3	1590		412.5	40	34	30			190	305	4	160	120
117	550 - 600	3	1740		440	42	36	32			190	340	4	160	130
118	600 - 660	3	1920		440	44	38	34			190	370	4	160	145
119	660 - 720	3	2100		440	46	40	36			190	405	4	160	160
120	720 - 780	3	2280		467.5	48	42	36			190	440	4	170	170
121	780 - 840	3	2460		467.5	50	44	38			190	480	4	170	185
122	840 - 910	3	2640		467.5	52	46	40			190	520	4	170	200
123	910 - 980	3	2850		495	54	48	42			190	560	4	170	215
124	980 - 1060	3	3060		495	56	50	44			200	600	4	180	230
125	1060 - 1140	3	3300		495	58	50	46			200	645	4	180	250
126	1140 - 1220	3	3540		522.5	60	52	46			200	690	4	180	270
127	1220 - 1300	3	3780		522.5	62	54	48			200	740	4	180	285
128	1300 - 1390	3	4050		522.5	64	56	50			200	785	4	180	305
129	1390 - 1480	3	4320		550	66	58	50			200	835	4	180	325
130	1480 - 1570	3	4590		550	68	60	52			220	890	5	190	325
131	1570 - 1670	3	4890		550	70	62	54			220	940	5	190	335
132	1670 - 1790	3	5250		577.5	73	64	56			220	1025	5	190	350
133	1790 - 1930	3	5610		577.5	76	66	58			220	1110	5	190	375
134	1930 - 2080	3	6000		577.5	78	68	60			220	1170	5	190	400
135	2080 - 2230	3	6450		605	81	70	62			240	1260	5	200	425
136	2230 - 2380	3	6900		605	84	73	64			240	1355	5	200	450
137	2380 - 2530	3	7350		605	87	76	66			240	1455	5	200	480
138	2530 - 2700	3	7800		632.5	90	78	68			260	1470	6	200	480
139	2700 - 2870	3	8300		632.5	92	81	70			260	1470	6	200	490
140	2870 - 3040	3	8700		632.5	95	84	73			260	1470	6	200	500
141	3040 - 3210	3	9300		660	97	87	76			280	1470	6	200	520
142	3210 - 3400	3	9900		660	100	90	78			280	1470	6	200	555
143	3400 - 3600	3	10500		660	102	92	78			280	1470	6	200	590
144	3600 - 3800	3	11100		687.5	105	95	81			300	1470	6	200	620
145	3800 - 4000	3	11700		687.5	107	97	84			300	1470	6	200	650
146	4000 - 4200	3	12300		687.5	111	100	87			300	1470	7	200	650
147	4200 - 4400	3	12900		715	114	102	[illegible]			300	1470	7	200	660
148	4400 - 4600	3	13500		715	117	105	90			300	1470	7	200	670
149	4600 - 4800	3	14100		715	120	107	92			300	1470	7	200	680
150	4800 - 5000	3	14700		742.5	122	111	95			300	1470	7	200	685
151	5000 - 5200	3	15400		742.5	124	111	97			300	1470	8	200	685
152	5200 - 5500	3	16100		742.5	127	114	97			300	1470	8	200	695
153	5500 - 5800	3	16900		742.5	130	117	100			300	1470	8	200	705
154	5800 - 6100	3	17800		742.5	132	120	102			300	1470	9	200	705
155	6100 - 6500	3	18800		742.5		124	107			300	1470	9	200	715
156	6500 - 6900	3	20000		770		127	111			300	1470	9	200	725
157	6900 - 7400	3	21500		770		132	114			300	1470	10	200	725
158	7400 - 7900	3	[illegible]		770		137	117			300	1470	11	200	725
159	7900 - 8400	3	[illegible]		770		142	122			300	1470	11	200	735
160	8400 - 8900	3	26000		770		147	127			300	1470	12	200	735
161	8900 - 9400	3	27500		770		152	132			300	1470	13	200	735
162	9400 - 10000	3	29000		770			132			300	1470	14	200	735
163	10000 - 10700	3	31000		770			137			300	1470	15	200	735
164	10700 - 11500	3	33000		770			142			300	1470	16	200	735
165	11500 - 12400	3	35500		770			147			300	1470	17	200	735
166	12400 - 13400	3	38500		770			152			300	1470	18	200	735
167	13400 - 14600	3	42000		770			157			300	1470	19	200	735
168	14600 - 16000	3	46000		770			162			300	1470	21	200	735

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

1 see C.1.
2 see F.1.2

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 rranged for attached bilge pumps to protect them gainst running dry, the bilge suction are also to be itted with two screw-down non-return valves.

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

. Calculation of pipe diameters

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

.2 Dry cargo and passenger ships

) main bilge pipes

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

) branch bilge pipes

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

here

	[mm]	calculated inside diameter of main bilge pipe
	[mm]	calculated inside diameter of branch bilge pipe
	[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:

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

where:

Q	[m³/h]	minimum capacity
d_H	[mm]	calculated inside diameter of main bilge pipe

Section 14

Rudder and Manoeuvring Arrangement

A. General

1. Manoeuvring arrangement

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

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

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

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

Guidance

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

1.5 For ice-strengthening see Section 15.

2. Structural details

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

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

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

3. Size of rudder area

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

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

c_1 = factor for the ship type:

= 1,0 in general

= 0,9 for bulk carriers and tankers having a displacement of more than 50.000 ton

= 1,7 for tugs and trawlers

c_2 = factor for the rudder type:

= 1,0 in general

= 0,9 for semi-spade rudders

= 0,8 for double rudders (per rudder)

= 0,7 for high lift rudders

c_3 = factor for the rudder profile:

= 1,0 for NACA-profiles and plate rudder

= 0,8 for hollow profiles

c_4 = factor for the rudder arrangement:

= 1,0 for rudders in the propeller jet

= 1,5 for rudders outside the propeller jet

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

cording to the following formula:

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

v = v_0 for ahead condition

v = v_a for astern condition

κ_1 = coefficient, depending on the aspect ratio Λ

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

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

κ_3 = coefficient, depending on the location of the rudder:

κ_3 = 0,8 for rudders outside the propeller jet

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

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

κ_t = coefficient depending on the thrust coefficient c_t

κ_t = 1,0 normally

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

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

Table 14.1

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

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

$Q_R = C_R \cdot r$ [Nm]

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

α = 0,33 for ahead condition

α = 0,66 for astern condition (general)

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

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

α = 0,25 for ahead condition

α = 0,55 for astern condition.

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

k_b = balance factor as follows:

$k_b = A_f/A$

k_b = 0,08 for unbalanced rudders

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

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

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

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

The resulting force of each part may be taken as:

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

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

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

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

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

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

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

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

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

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

$c_1 = A_1/b_1$

4. Materials

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

4.2 In general materials having a minimum nominal upper yield point R_{eH} of less than 200 N/mm^2 and a minimum tensile strength of less than 400 N/mm^2 or more than 900 N/mm^2 shall not be used for rudder stocks, pintles, keys and bolts. The requirements of this Section are based on a material's minimum nominal upper yield point R_{eH} of 235 N/mm^2. If material is used having a R_{eH} differing from 235 N/mm^2, the material factor k_r is to be determined as follows:

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

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

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

4.3 Before significant reductions in rudder stock diameter due to the application of steels with R_{eH} exceeding 235 N/mm^2 are granted, the Society may require the evaluation of the elastic rudder stock deflections. Large deflections should be avoided in order to avoid excessive edge pressures in way of bearings.

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

5. Definitions

C_R = rudder force in [N]

Q_R = rudder torque in [Nm]

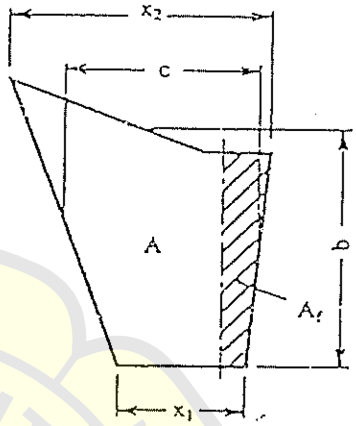
A = total movable area of the rudder in [m^2]
For nozzle rudders, A is not to be taken less than 1,35 times the projected area of the nozzle;

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

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

b = mean height of rudder area in [m]

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



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

Fig. 14.1

Λ = aspect ratio of rudder area A_t

Λ = b^2/A_t

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

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

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

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

B. Rudder Force and Torque

1. Rudder force and torque for normal rudders

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

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

$v = v_0$ for ahead condition

$v = v_a$ for astern condition

κ_1 = coefficient, depending on the aspect ratio Λ

$\kappa_1 = (\Lambda + 2)/3$, 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

$C_1 = A_1/b_1$

$C_2 = A_2/b_2$

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

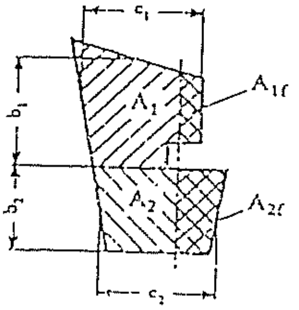


Fig. 14.2

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

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

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

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

for ahead condition

The greater value is to be taken.

C. Scantlings of the Rudder Stock

1. Rudder stock diameter

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

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

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

The related torsional stress is:

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

k_r see A.4.2

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

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

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

2. Strengthening of rudder stock

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

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

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

Bending stress:

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

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

Torsional stress:

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

D_1 = increased rudder stock diameter in [cm]

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

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

Q_R see B.1.2 and B.2.2 - 2.3

D_t see 1.1

Guidance

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

Section 6

Propellers

A. General

1. Scope

These Rules apply to screw-propellers and vane wheels. 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 and vane wheels as well as the position of the vane wheel on the ship are to be submitted to the Society in triplicate for examination The drawings are required to contain all the details necessary to verify compliance with the following Rules

B. Materials

1. Approved materials

Propellers and vane wheels are to be made of seawater-resistant cast copper alloys or cast steel alloys with a minimum tensile strength of 440 N/mm², cf. Rules for Materials. 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[1]). 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

2. Materials for blade retaining-bolts

Unless protected against contact with seawater the blade-retaining bolts 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, vane wheels 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

eff

c [-] Coefficient for shrunk joints

= 1,0

for engine and turbine gear transmissions

= 12

for direct drives

C_G [-] Size factor in accordance with formula (2)

C_{Dyn} [-] Dynamic factor in accordance with formula (3)

C_w [-] Characteristic value fo propeller material as shown i Table 6.1 (corresponds to th minimum tensile strength R of the propeller material whe

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1) Sufficient fatigue strength under alternating bending stresses must be proved by a method recognized by the Society. See also Section 1-2. of the Society's "Regulations for the Determination of Dynamic Stresses on Propellers", December 1971.

this has been shown to possess sufficient fatigue strength under alternating bending stresses in accordance with paragraph B.1.)

C	[-]	Conicity of shaft ends
	=	$\frac{\text{difference in taper diameter}}{\text{length of taper}}$
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 $= 2 \cdot R$
d_m	[mm]	Mean taper diameter
e	[mm]	Blade rake to aft $= R \cdot \tan \varepsilon$
E_T	[-]	Thrust stimulating factor in accordance with formula (5)
f, f_1, f_2, f_3	[-]	Factors in formulae (2) (3) (4) and (11)
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 $= \frac{\sum (R \cdot B \cdot H)}{\sum (R \cdot B)}$; in which R, B and H are to be substituted by values corresponding to the pitch at the various radii.
J	[-]	Degree of advance
k	[-]	Coefficient for various profile shapes in accordance with Table 6.2
k'	[-]	Coefficient calculated by applying formula (6) where use is made of profile shapes other than those given in Table 6.2
K_T	[-]	Thrust coefficient
L_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.

ℓ₃ sredut.

Table 6.1 Characteristic values C_w

Material	Description [1])	C_w
Cu 1	Cast manganese brass	440
Cu 2	Cast manganese nickel brass	440
Cu 3	Cast nickel aluminium bronze	590
Cu 4	Cast manganese aluminium bronze	630
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-austenitic 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.

L	[mm]	Pull-up length when mounting propeller on taper
L_{mech}	[mm]	Pull-up length at t = 35 °C
L_{temp}	[mm]	Temperature-related portion of pull-up length at t < 35 °C
n	[Rpm]	Propeller speed in rev/min.
P_w	[kW]	Shaft power
p	[N/mm²]	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

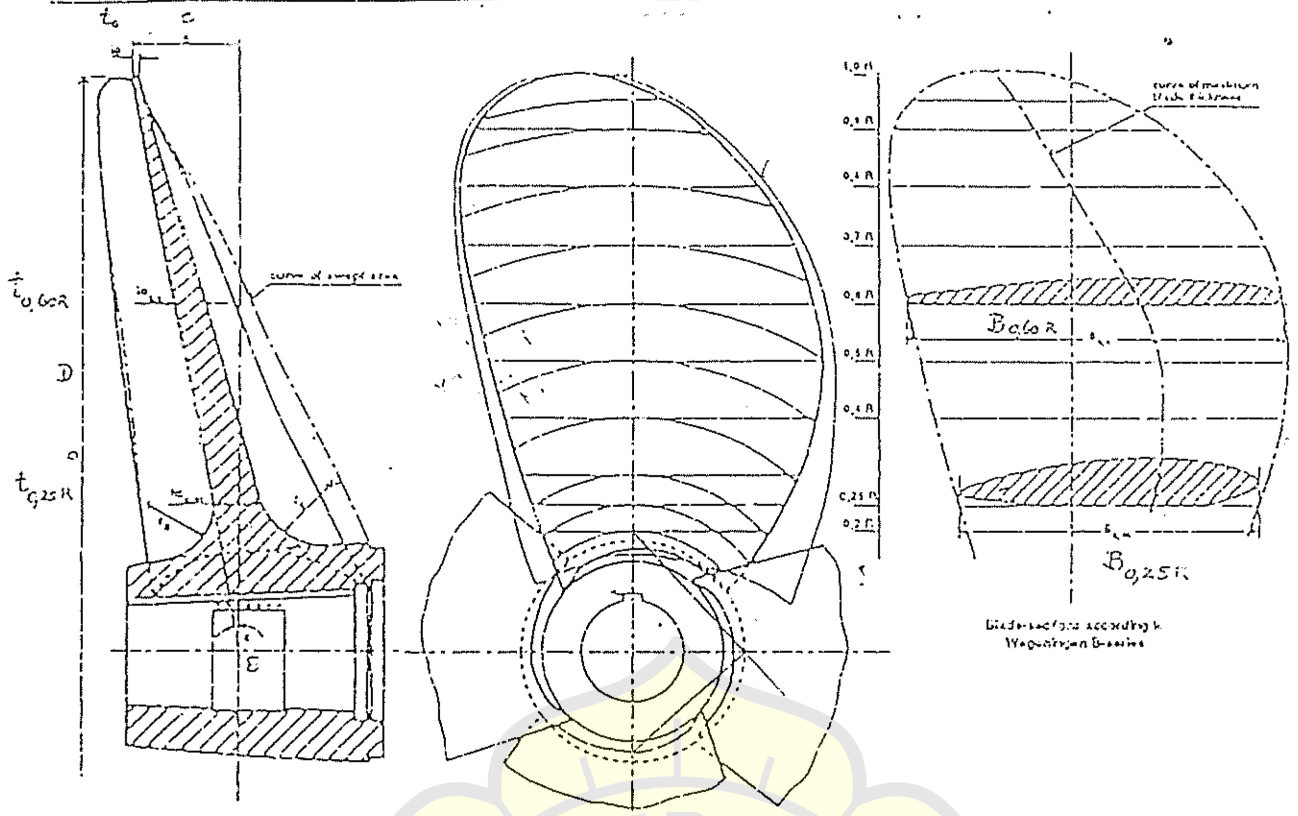


Fig. 6.1 Blade sections

T_{St}	[Nm]	Impact moment
V_s	[kn]	Speed of ship
w	[-]	Wake factor
W_v	[mm³]	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 = C / 2
μ_0	[-]	Coefficient of static friction = 0,13 for hydraulic oil shrunk joints = 0,18 for dry shrunk joints
$R_{p0,2}$	[N/mm²]	0,2 % proof stress of propeller material
R_{eH}	[N/mm²]	Yield strengths and
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).

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

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

k as in Table 6.2 PITCH (m)

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

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_{Dyn} [-] Dynamic factor

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

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

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".)

$$\frac{\sigma_{max}}{\sigma_m} = f_2 \cdot E_T + 1 \text{ with} \quad (4)$$

$$E_T = \frac{\delta_{KT}}{\delta_J} \cdot \frac{J}{K_T} \quad (5)$$

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

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 propellers are to be determined at radii 0,35 · R and 0,6 · R by applying formula (1).

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

For other ships the diameter/pitch ratio D/H_m applicable to open-water navigation can be used in formula (1).

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

2.4 The fillet radii at the transition from the face and the back of the blades to the propeller boss should correspond in the case of three and four bladed propellers, to about 3,5 % of the propeller diameter. For propellers with a larger number of blades the maximum fillet radii allowed by the propeller design should be aimed at, and the radii shall not in any case be made smaller than $0{,}4 \cdot t_{0,25}$

2.5 For blades of special shape, special mechanical strength calculations are to be submitted to the Society as evidence that the propeller blades are adequately dimensioned.

For profile shapes other than those given in Table 6.2 the following condition applies:

$$k' = k \cdot \sqrt{\frac{\beta_x}{\beta'_x}} \text{ with } \beta'_x = \frac{W_x}{t^2 \cdot B} \qquad (6)$$

D. Controllable Pitch Propellers

1. Documents for approval

In the case of controllable pitch propellers besides the design drawings of blade and propeller boss general and sectional drawings of the entire controllable pitch propeller installation are to be submited to the Society in triplicate. 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.

Table 6.2 Values of k for various profile shapes

Profile shape	Values of k		
	0,25 R	0,35 R	0,60 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 $\beta_{x\,0,25} = 0,10$ $\beta_{x\,0,35} = 0,11$ $\beta_{x\,0,60} = 0,12$	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 0,2 R is < 4 · t.

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 pressures 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 provided 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.

$$T_M = \frac{0,65 \cdot 10^6 \cdot R_{p02} \cdot P_w \cdot L_M \cdot C_G^2}{n \cdot z \cdot C_w \cdot D} \quad (7)$$

5. Blade retaining bolts

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

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

$$F_M = \frac{280 \cdot 10^6 \cdot R_{p0,2} \cdot P_w \cdot C_G^2}{n \cdot z \cdot Z \cdot C_w \cdot D} \quad (9)$$

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

The shank of blade retaining bolts 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 system 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 9).

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.

Section 4

Main Shafting

A. General

1. Scope

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

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

2. Documents for approval

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

B. Materials

1. Approved materials

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

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

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

2. Testing of materials

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

C. Shaft Dimensions

1. General

All parts of the shafting are to be dimensioned in accordance with the following formulae in compliance with the requirements relating to torsional vibrations set out in Section 16. The dimensions of the shafting shall be based on the total rated installed power. Where the geometry of a part is such that it cannot be dimensioned in accordance with these formulae, special evidence of the mechanical strength of the part or parts concerned is to be furnished to the Society.

2. Minimum diameter

The minimum shaft diameter is to be determined by applying formula (1).

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

d [mm] required outside diameter of shaft

d_i [mm] diameter of shaft bore, where present. If the bore in the shaft is ≤ 0,4 . d, the expression

$$1 - \left(\frac{d_i}{d_a}\right)^4 = 1,0 \text{ may be applied}$$

d_a [mm] actual shaft diameter

P_w [kW] rated power transmitted by shaft

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

n [Rpm] rated shaft speed

F [-] factor for the type of propulsion installation

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

= 100
for all other propulsion installations

b) Propeller shafts
= 100
for all types of installations

C_w [-] material factor

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

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

k [-] Factor for the type of shaft

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

k = 1,10 for intermediate shafts where the coupling flanges are mounted on the ends of the shaft with the aid of keys. At a distance of at least 0,2 · d from the end of the keyway, such shafts can be reduced to a diameter corresponding to k = 1,0.

k = 1,10 for intermediate shafts with radial holes whose diameter is not greater than 0,3 · d.

k = 1,10 for thrust shafts near the plain bearings on either side or the thrust collar, or near the axial bearings where an antifriction bearing design is used.

k = 1,15 for intermediate shafts designed as multi-splined shafts where d is the outside diameter of the splined shaft. Outside the splined section, the shafts can be reduced to a diameter corresponding to k = 1,0.

k = 1,20 for intermediate shafts with longitudinal slots where the length and width of the slot do not exceed 1,17 · d and 0,25 · d respectively.

k = 1,22 for propeller shafts from the area of the aft stern tube or shaft bracket bearing to the forward load-bearing face of the propeller boss subject to a minimum of 2,5 · d, if the propeller is shrink fitted, without key, on to the tapered end of the propeller shaft using a method approved by the Society, or if the propeller is bolted to a flange forged on the propeller shaft, the propeller shaft runs in oil.

k = 1,26 for propeller shafts in the area specified for k = 1,22, if the propeller is keyed to the tapered propeller shaft and the propeller shaft runs in oil, and also for water-lubricated propeller shafts which are protected against the penetration of seawater in accordance with D.3.2.

k = 1,40 for propeller shafts in the area specified for k = 1,22, if the shaft inside the stern tube is lubricated with grease.

k = 1,15 for propeller shafts forward portion of shafts to where they emerge from the stern tube. The portion of the propeller shaft located forward of the stern tube can be reduced to the size of the line shaft.

D. Design

1. General

Changes in diameter are to be effected by tapering or ample radiusing. For intermediate shafts, the radius at forged flanges is to be at least 0,08 ·d, that at the aft propeller shaft flange at least 0,125 ·d.

2. Shaft tapers and propeller nut threads

Keyways in the shaft taper for the propeller should be so designed that the forward end of the groove makes a gradual transition to the full shaft section. In addition, the forward end of the keyway should be spoon-shaped. The edges of the keyway at the surface of the shaft taper for the propeller may not be sharp. The forward end of the keyway must lie well within the seating of the propeller boss. Threaded holes to accommodate the securing screws for propeller keys should be located only in the aft half or the keyway (see Fig. 4.1).

In general, tapers for securing flange couplings should have a conicity of between 1: 10 and 1: 20. In the case of shaft tapers for propellers, the conicity must be between 1:10 and 1:15. Where the oil injection method is used to mount the propeller on its

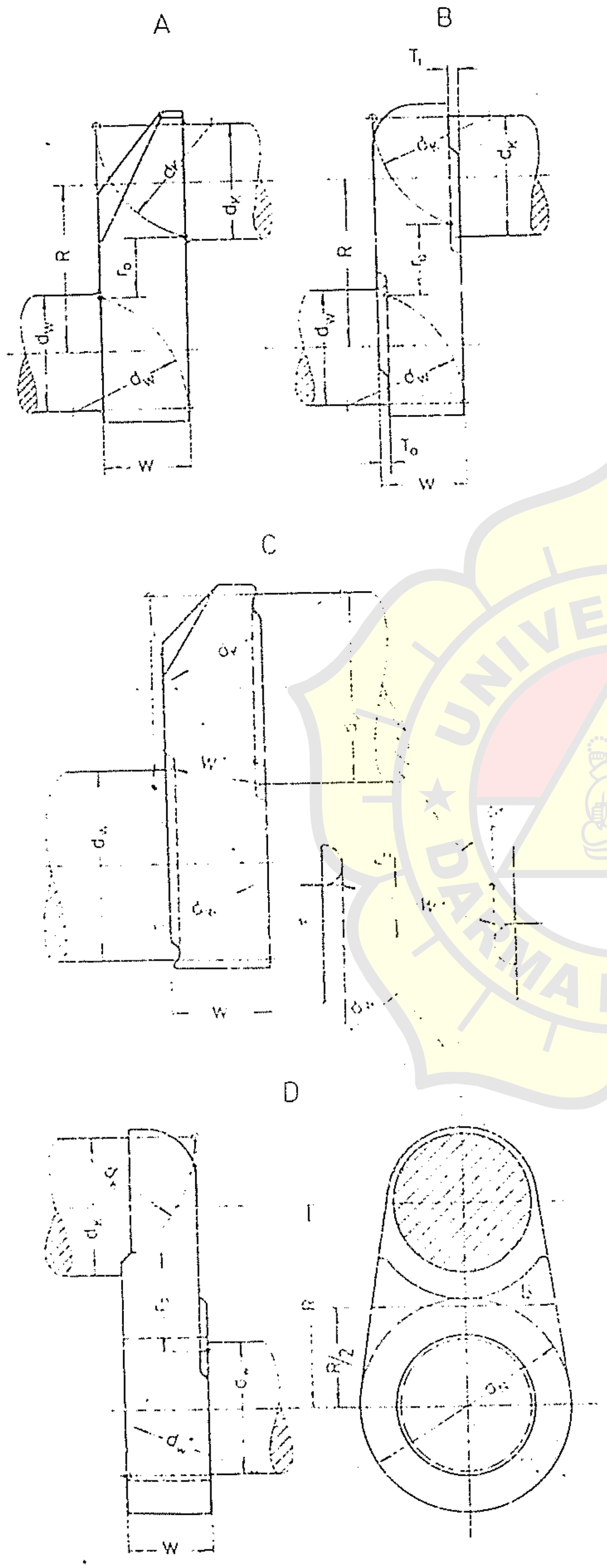


Fig. 2.11

$$r_o = 0{,}5\,(H + d_k + d_w) \cdot W\left(\sqrt{\frac{2d_k}{W} - 1} + \sqrt{\frac{2d_w}{W} - 1}\right) \tag{10}$$

In case of web undercut, W in formula (10) is to be replaced by:

$$W^* = 0{,}5\,(2 \cdot W - T_1 - T_a) \tag{11}$$

In the case of semi-built crankshafts in accordance with part D, the value d_W under the root sign only in formula (10) is to be replaced by:

$$d_w^* = 1/3\,(d_N - d_w) + d_w \tag{12}$$

In case of web undercut, W* is also to be substituted for W in accordance with formula (11)

Where there is a positive pin/journal overlap is ≥ 0) according to part C, the value W in formula (10) is to be replaced by:

$$W^* = \sqrt{(W - T_1 - T_a)^2 + \left[0{,}5\,(d_k + d_w - H)\right]^2} \tag{13}$$

For the conventional designs, where

B/d_w = 1,37 to 1,51 in the case of solid-forged crankshafts.

and

B/d_w = 1,51 to 1,63 in the case of semi-built crankshafts.

the influence of B in the normal calculation of r_o is already taken into account in the values of $\triangle_a$ in Fig. 2.9.

Where the values of B/d_W depart from the above (e.g. in the case of discs, oval webs etc.), the altered stiffening effect of B is to be allowed for by a fictitious web thickness W**, which is to be calculated by applying the following equations and is to be substituted for W in formula (10):

$W^{**} = W^* \cdot \sqrt[3]{\frac{B}{d_w} - 0{,}44}$	For solid-forged crankshafts	(14)
$W^{**} = W^* \cdot \sqrt[3]{\frac{B}{d_w} - 0{,}57}$	for semi-built crankshafts	(15)

Part C:

Approximate Calculation of the Starting Air Supply

1. 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 follows:

$$J = a \cdot \sqrt[3]{\frac{D}{H}} \cdot (z + b \cdot p_{e,e} \cdot n_A + 0{,}9) \cdot V_H \cdot c \cdot c' \tag{16}$$

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)
$P_{e,mp}$	[bar]	maximum permissible working pressure of the starting air receiver
z	[-]	number of cylinders
$P_{e,e}$		mean effective working pressure in cylinder at rated power

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

For two-stroke engines: a = 0,696
For four-stroke engines: a = 0,618

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 = 1,0 For single-shaft propulsion plants where one engine acts on the shaft directly or via gears.

c = 2,0 For single-shaft propulsion plants where two identical engines act on the shaft via a gear transmission and cannot be coupled and uncoupled in service.

c = 1,5 For single-shaft propulsion plants where two identical engines act on the shaft via a gear transmission and couplings which can be engaged and disengaged in service.

c = 1,5 For two-shaft propulsion plants where each engine acts on the corresponding shaft directly or via gears.

c = 3,0 For two-shaft propulsion plants where two identical engines in each case act on the corresponding shaft via a gear transmission and cannot be coupled and uncoupled in service.

c = 2,0 For two-shaft propulsion plants where two identical engines in each case act on the corresponding shaft via a gear transmission and couplings which can be engaged and disengaged in service.

c = 3,0 For four-shaft propulsion plants where each engine acts on the corresponding shaft directly or via gears.

Where the arrangement of the main propulsion plant differs from the above, the value of "c" is to be agreed with the Society in each individual case.

For installations with electrical propeller drive, "c" is to be given the value specified in 2.2.

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

$d = 1,$ where $P_{e,mp} = 30$ bar

$$d = \frac{0{,}0584}{1 - e^{(0{,}11 - 0{,}55 \cdot \ln P_{e,mp})}}$$

where $P_{e,mp} \neq 30$ 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 $P_{e,mp}$ to the starting pressure P_A, then the value of "d" shown in Fig. 2.12 is to be used.

The following values of n_x are to be applied:

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

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

n_o [min⁻¹] = rated speed

2. Starting air supply for plants with non-reversing engines

2.1 For each non-reversing main engine which drives a controllable pitch propeller or where starting is possible without resisting torque, the calculated supply of starting air may be reduced to 0,3 J, although it may not be less than that required for six starts.

2.2 Where diesel-electric propeller drive is installed, "c" in formula (16) is to be given the following values according to the number of generators n:

Table 2.14

n	1	2	3	4	5	6	7	8
c	0,30	0,60	0,84	1,08	1,26	1,38	1,44	1,50

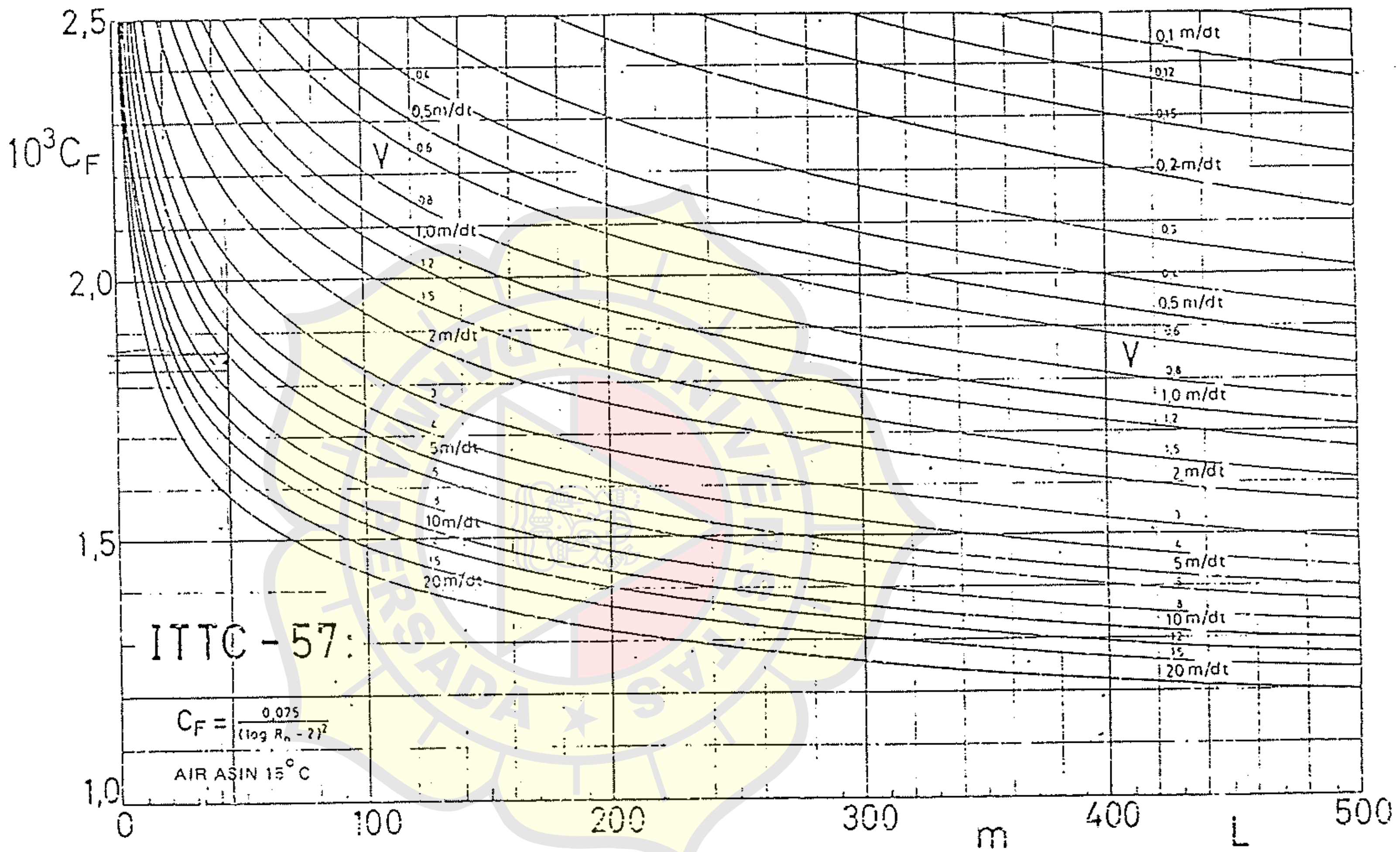
This assumes prime movers having the same dimensions and the same number of cylinders. Where the dimensions and numbers of cylinders differ, the values of "c" are to be interpolated accordingly.

3. Starting air supply for auxiliary engines on turbine ships

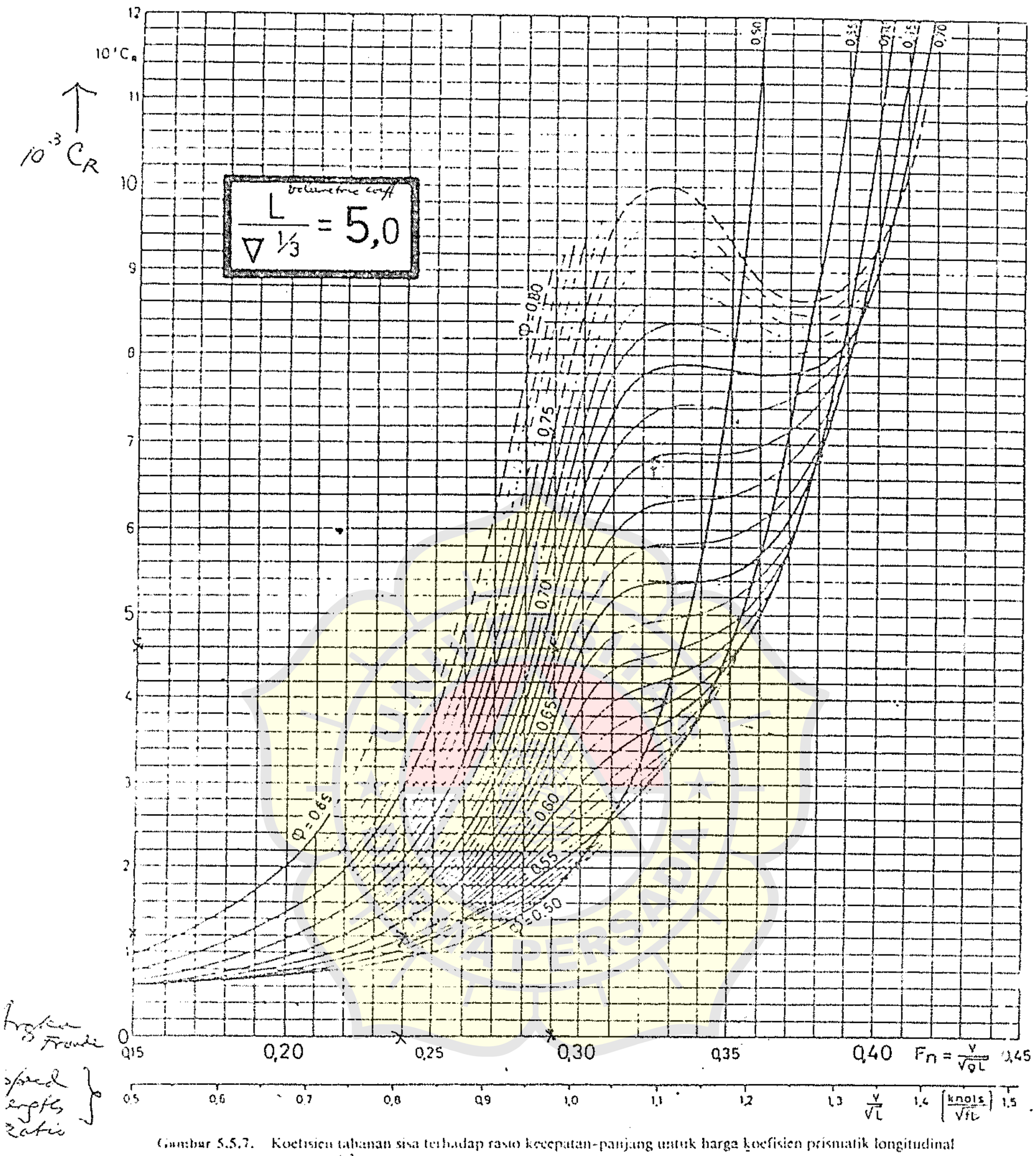
The supply of starting air is to be calculated according to formula (16). The value of "c" to be used depends on the number of auxiliary engines:

c = 0,30 for 1 auxiliary engine
c = 0,45 for 2 auxiliary engines
c = 0,60 for 3 auxiliary engines
c = 0,75 for 4 auxiliary engines or over

For engines with different numbers of cylinders and/or main dimensions the values of "c" are to be interpolated accordingly.

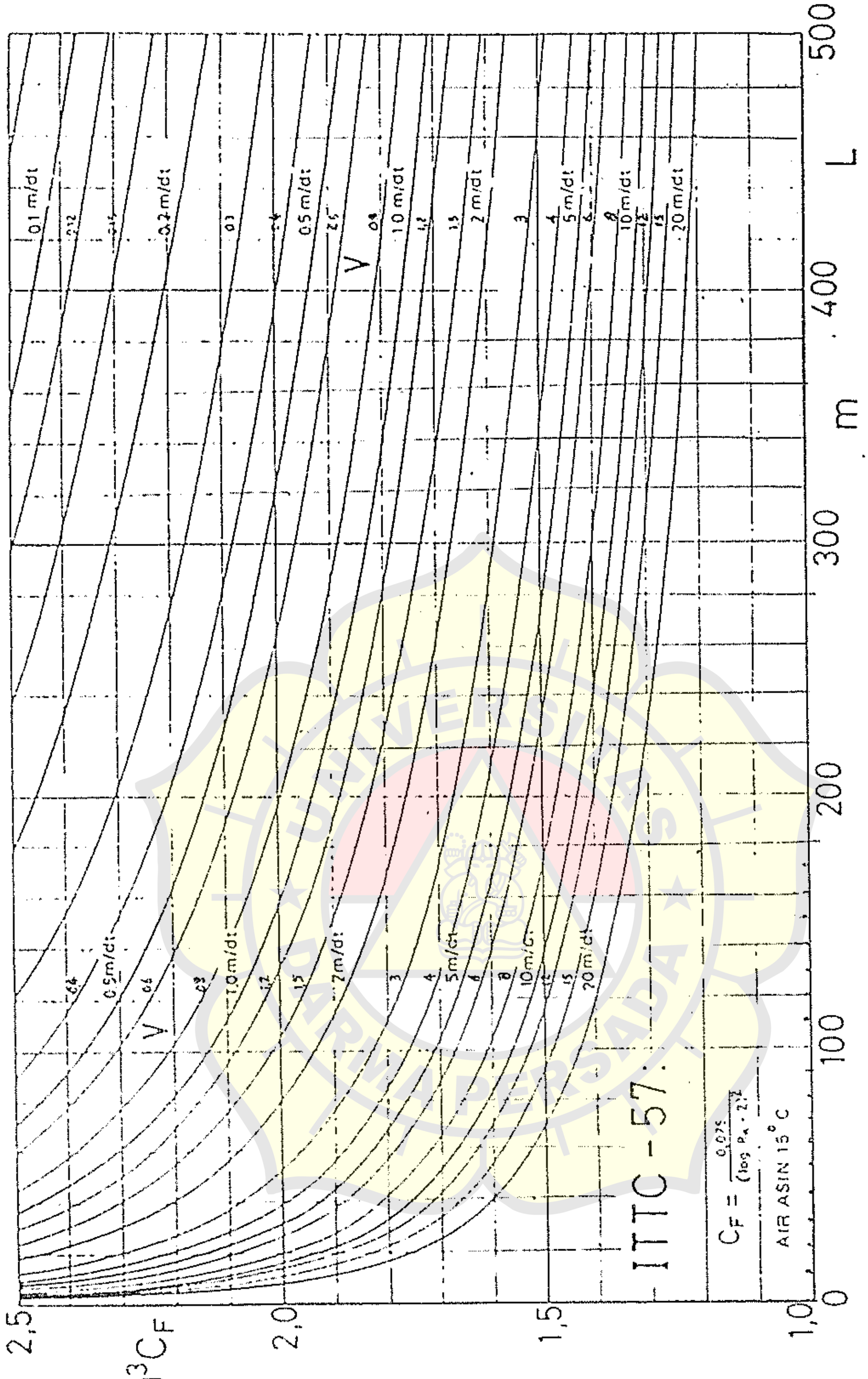


Gambar 5.5.14. Koefisien tahanan gesek C_F (menurut ITTC 1957) sebagai fungsi panjang kapal L dan kecepatan V.

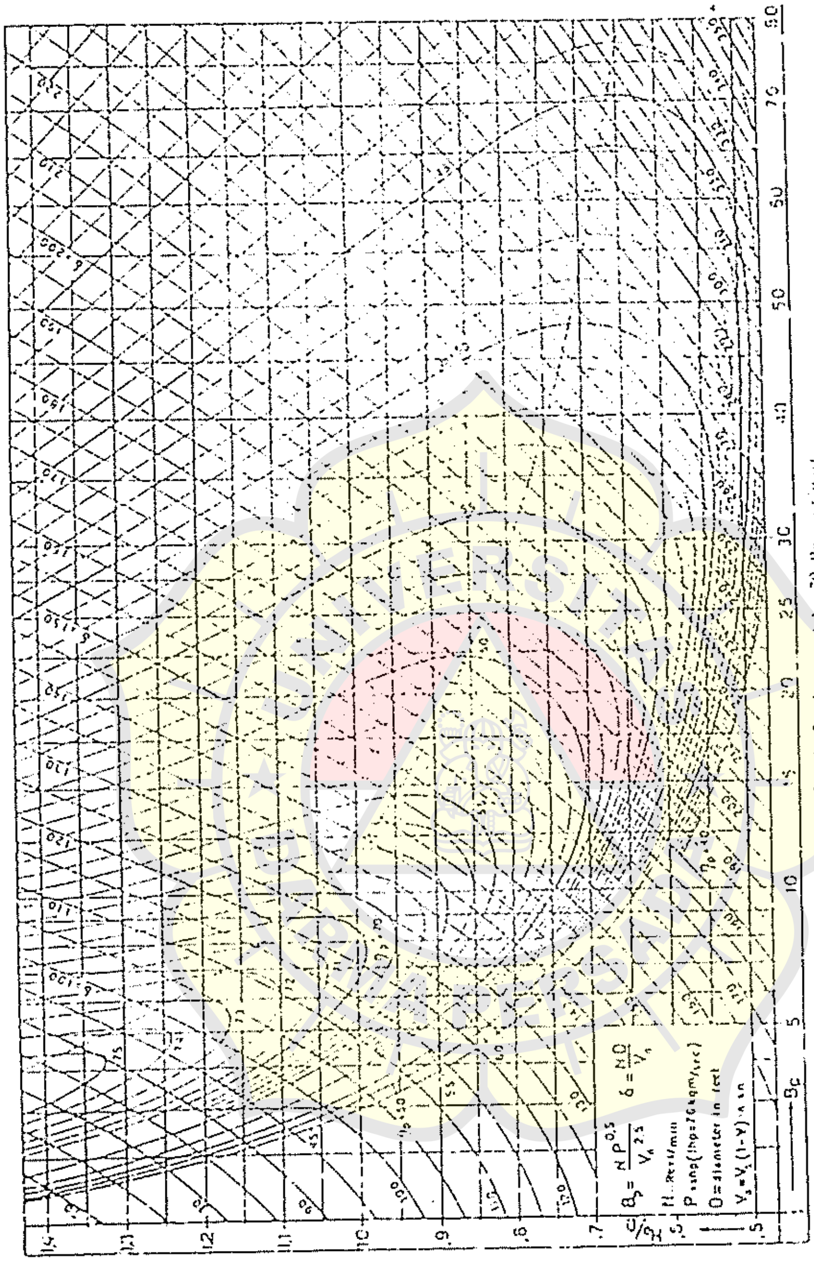


Gambar 5.5.7. Koefisien tahanan sisa terhadap rasio kecepatan-panjang untuk harga koefisien prismatik longitudinal yang berbeda-beda. $L/\nabla^{1/3} = 5,0$.

Ketr.: Prismatic Coefficient = φ (= C_p)



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



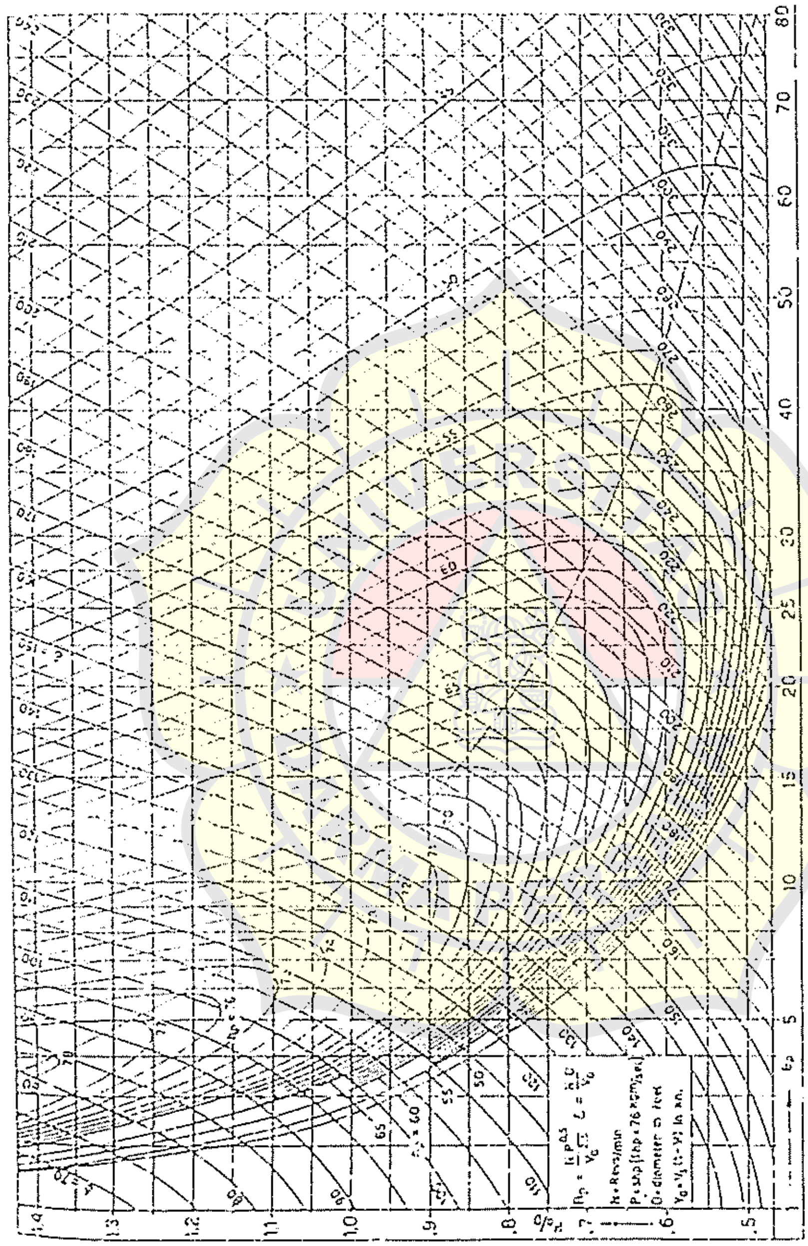
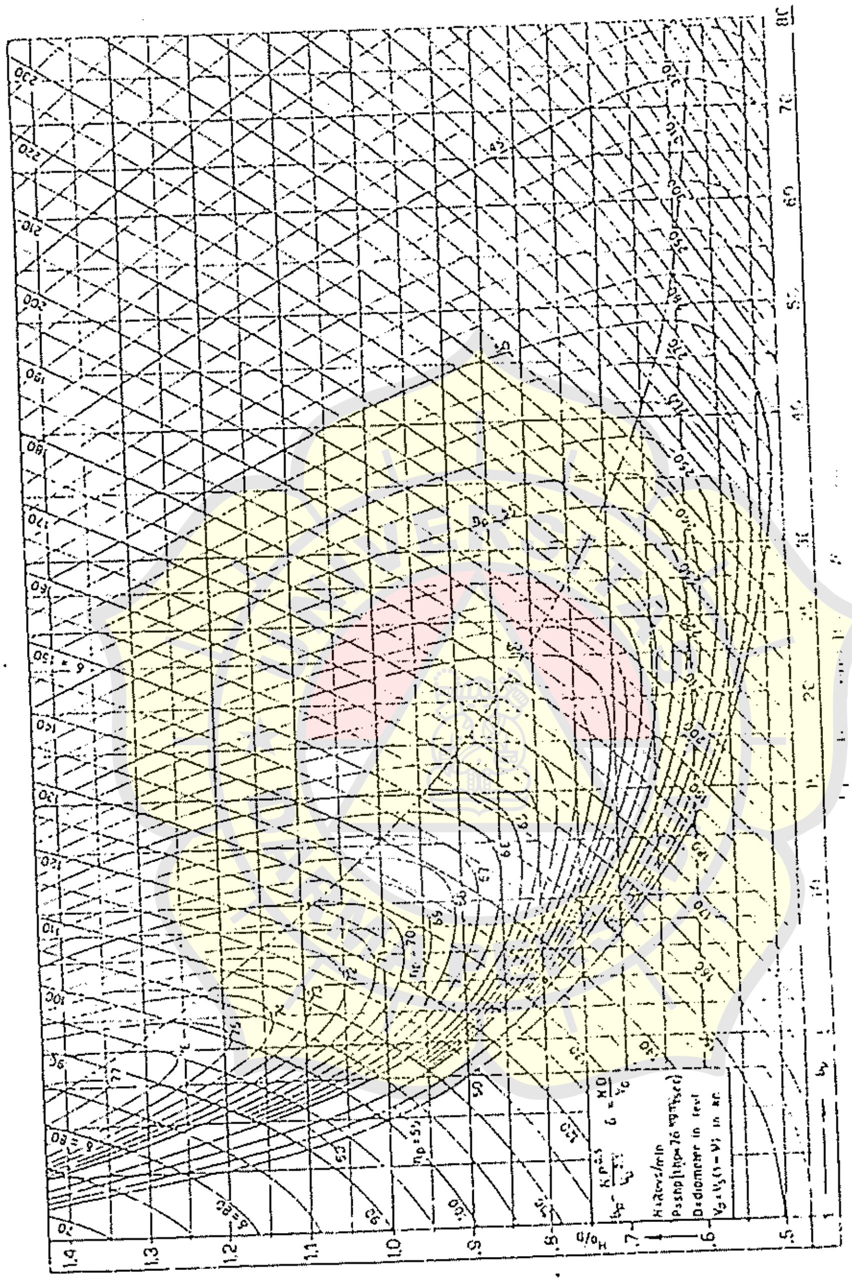


Fig. 3.15 Troost B.4–40 B_P–δ Chart



. Semua data diacukan pada daerah (lingkup) model, dan tahanan model (R_{Tm}) ditentukan sebagai fungsi kecepatan.

Koefisien tahanan total spesifik model (C_{Tm}) ditentukan :

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

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

Koefisien tahanan sisa spesifik ditentukan dari

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

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

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

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

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

4. C_R dinyatakan sebagai fungsi angka Froude

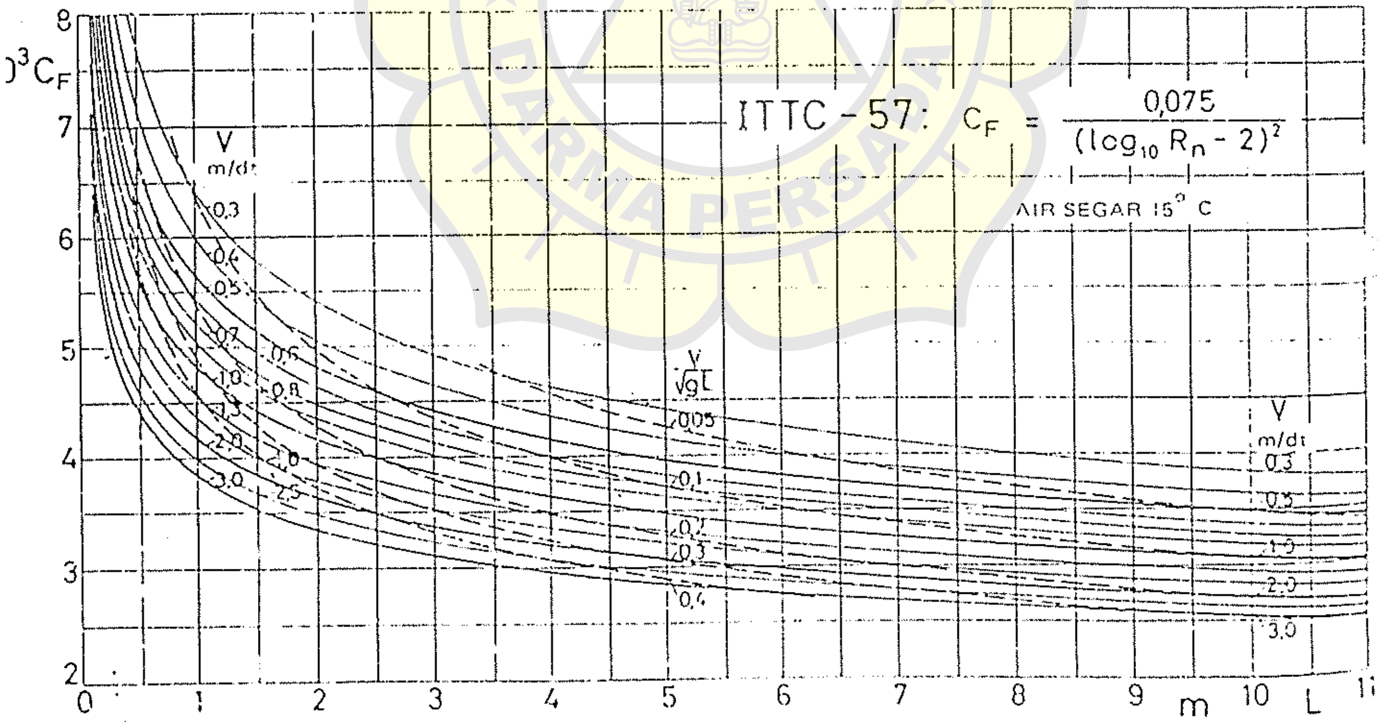
$$F_n = \frac{V}{\sqrt{gL}} \qquad (5.5.9)$$

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

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

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

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



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

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

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

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

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

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

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

Dalam hal ini koefisien tahanan totalnya adalah

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

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

C_F = koefisien tahanan gesek dan dapat dihitung dengan memakai

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

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

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

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

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

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

B/T

Karena diagram tersebut dibuat berdasarkan rasio lebar – sarat

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

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

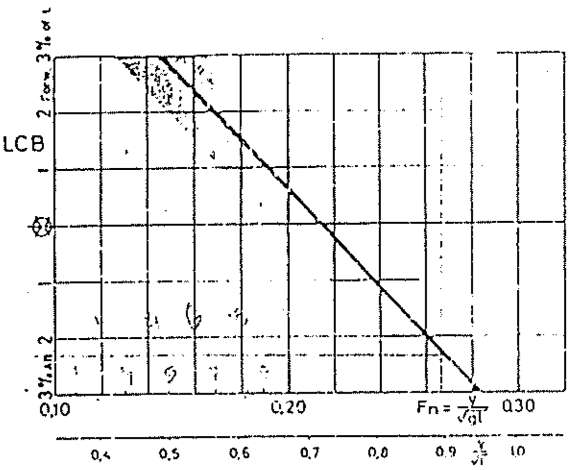
Berdasarkan hasil pemeriksaan materi pengujian yang ada saat ini maka disarankan untuk memakai rumus koreksi berikut ini :

$$10^3 C_R = 10^3 C_{R(B/T=2,5)} + 0{,}16(B/T - 2{,}5) \qquad (5.5.17)$$

Koreksi ini dapat mempunyai harga yang negatif atau positif.

LCB

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



;ambar 5.5.15. LCB standar. Letak longitudinal titik benam yang ipandang terbaik.

Dalam hal ini, LCB standar tersebut didefinisikan ebagai fungsi linier angka Froude F_n. Karena tidak danya ketergantungan yang pasti pada parameter lain- ya yang tercatat maka LCB standar tersebut disajikan ebagai garis tunggal. Daerah yang diberi warna gelap i sekitar garis ini menunjukkan lingkup materi yang ikaji.

Sebagaimana disebutkan sebelumnya, karena letak .CB standar dianggap merupakan letak yang nemberikan tahanan yang paling kecil maka letak yang iin pada prinsipnya akan memberikan tahanan yang ;bih besar. Penambahan tahanan tersebut harus dicari engan jalan mengalikan penyimpangan LCB dari :andar, yaitu

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

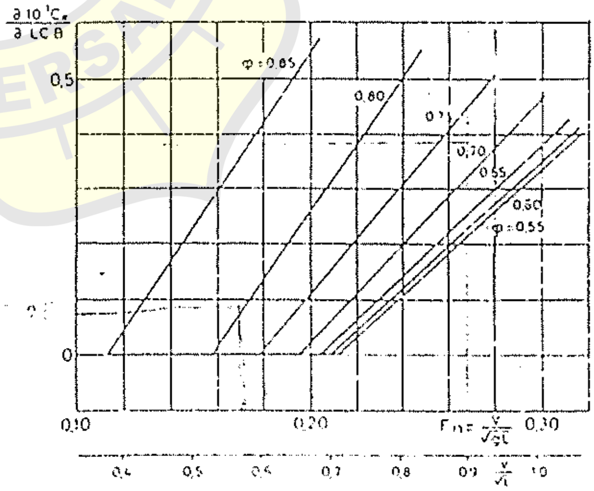
engan faktor $\partial 10^3 C_R / \partial \text{LCB}$. Harga faktor ini dapat iperoleh dari Gb. 5.5.16, dan ini hanya berlaku untuk .CB yang berada di depan $\text{LCB}_{\text{standar}}$. Mengenai LCB ang berada di belakang $\text{LCB}_{\text{standar}}$, semua sumber ang ada mempunyai pendapat yang saling ertentangan. Namun demikian, karena kecenderungan :rjadinya letak demikian itu sangat kecil maka engabaian koreksi dalam hal itu tidak akan memberi- an kesalahan yang berarti.

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

$$0^3 C_R = 10^3 C_{R(\text{standar})} + \frac{\partial 10^3 C_R}{\partial \text{LCB}} |\Delta \text{LCB}| \quad (5.5.19)$$

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

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



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

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

BENTUK BADAN KAPAL (BENTUK PENAMPANG MELINTANG DAN HALUAN)

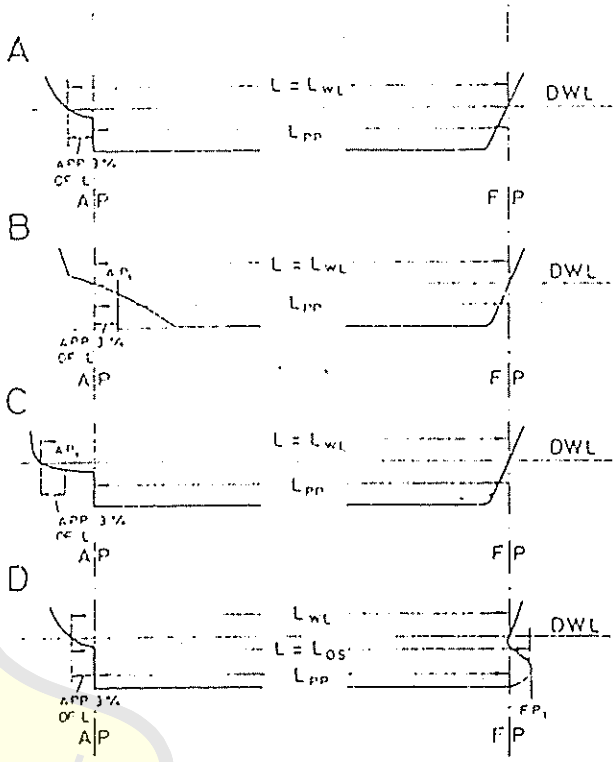
Sebagaimana disebutkan sebelumnya, kurva tahanan yang diperoleh berdasarkan Gb. 5.5.5 – 5.5.13) dianggap berlaku untuk yang mempunyai bentuk standar", yaitu penampangnya bukan yang benar-benar berbentuk U ataupun V. Karena itu, dalam menghitung daya efektif untuk perancangan awal umumnya tidak diperlukan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

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

(5.5.20)

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

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



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

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

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

$_n$ = 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

(5.5.21)

ANGGOTA BADAN KAPAL

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

TAHANAN TAMBAHAN

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

$$
\begin{aligned}
\text{Untuk kapal dengan } L &\leqq 100 \text{ m}, & 10^3 C_A &= 0{,}4\\
&= 150 \text{ m} & &= 0{,}2\\
&= 200 \text{ m} & &= 0\\
&= 250 \text{ m} & &= -0{,}2\\
&\geqq 300 \text{ m} & &= -0{,}3
\end{aligned}
\qquad (5.5.23)
$$

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

Dispiasemen		
1.000 t	$C_A = 0{,}6 \times 10^{-3}$	
10.000 t	$= 0{,}4 \times 10^{-3}$	(5.5.24)
100.000 t	$= 0$	
1.000.000 t	$= -0{,}6 \times 10^{-3}$	

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

ANGGOTA BADAN KAPAL

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

$$C_{F'} = C_F \frac{S_1}{S} \qquad (5.5.25)$$

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

TAHANAN UDARA DAN TAHANAN KEMUDI

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

$$10^3 C_{AA} = 0{,}07 \qquad (5.5.26)$$

Koreksi untuk tahanan kemudi mungkin sekitar

$$10^3 C_{AS} = 0{,}04 \qquad (5.5.27)$$

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

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

KONDISI PELAYARAN DINAS

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

6.5.5. Prakiraan Fraksi Deduksi Gaya Dorong

Rumus atau diagram untuk menentukan fraksi deduksi gaya dorong untuk model harus terdiri dari parameter yang telah dibahas di 6.5.4 berikut ini :

1. Koefisien blok δ
2. Rasio lebar-panjang B/L
3. Rasio diameter baling-baling dengan panjang kapal, D/L.
4. Koefisien bentuk penampang.

Umumnya keterangan mengenai t terkait dengan keterangan mengenai w. Karena itu kurva untuk menentukan fraksi deduksi gaya dorong digambarkan di Gb. 6.4.26 sebagai kurva untuk fraksi arus ikut. Kurva tersebut berlaku untuk buritan konvensional (lihat Gb. 6.5.5). Untuk buritan baling-baling bebas harga t akan berkurang sebesar

$$\Delta t = -0{,}5t \qquad (6.5.16)$$

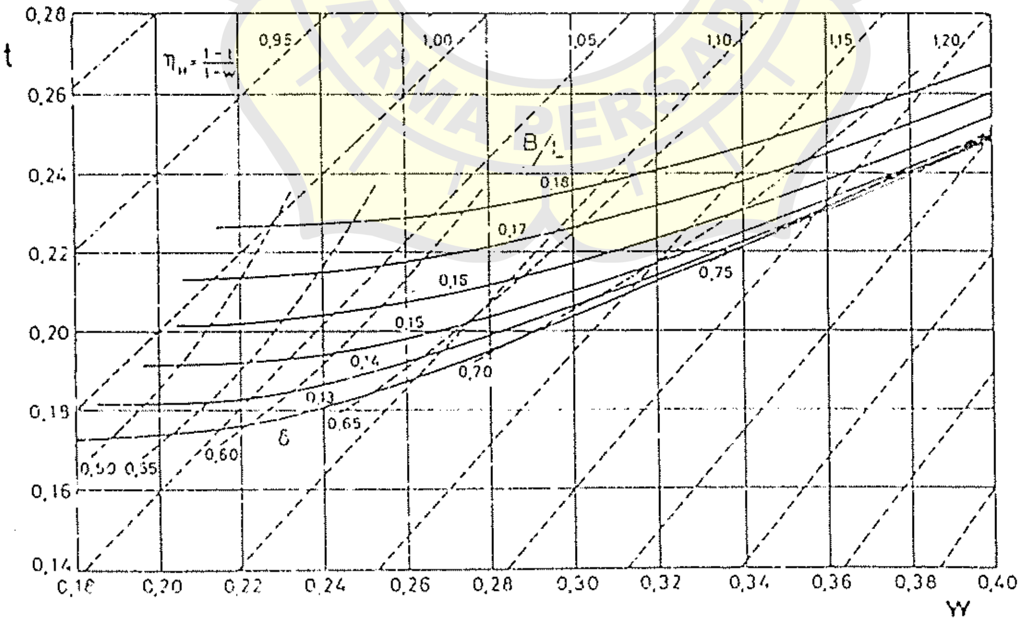
Buritan gembung memberikan pengertian bahwa t harus dikurangi sebesar

$$\Delta t = -0{,}25t \qquad (6.5.17)$$

Untuk kapal "standar" dengan bentuk penampang normal dan buritan konvensional, $D/L = 0{,}04$, serta $B/T = 2{,}5$, hubungan sederhana antara deduksi gaya dorong dengan arus ikut mudah dicari, dan hubungan ini ditunjukkan di Gb. 6.5.6. Dalam gambar ini koefisien arus ikut dipakai sebagai koordinat horizontal sedangkan ordinatnya adalah koefisien deduksi gaya dorong. Tiga perangkat kurva ditunjukkan dalam diagram tersebut. Perangkat yang pertama menunjukkan hubungan antara t dan w untuk harga koefisien blok yang tetap. Perangkat yang kedua menunjukkan hubungan yang sama tetapi untuk rasio lebar – panjang yang tetap, dan yang ketiga menunjukkan hubungan antara t dan w untuk efesiensi badan kapal yang tetap; $\eta_H = (1 - t)/(1 - w)$.

Sekalipun khusus hanya memandang kapal dengan bentuk yang normal dan mempunyai $D/L = 0{,}04$ akan terlihat bahwa antara t dan w tidak mempunyai hubungan yang proporsional. Lagi pula, t dan w bervariasi dengan cara sendiri-sendiri terhadap bentuk penampang kapal, garis tengah baling-baling, dan kecepatan. Karena itu Gb. 6.5.6 hanya dapat dipakai sebagai perkiraan yang sangat kasar untuk mendapatkan harga fraksi deduksi gaya dorong, fraksi arus ikut, dan efesiensi badan kapal dalam salah satu tahap perhitungan yang paling awal untuk menentukan daya yang diperlukan untuk propulsi kapal baru berbaling-baling tunggal.

Untuk memperkirakan fraksi deduksi gaya dorong kapal berbaling-baling ganda hanya pedoman dasarnya saja yang dapat diberikan. Yang jelas fraksi deduksi gaya dorong akan tergantung pada koefisien blok kapal.



Gambar 6.5.6. Hubungan antara fraksi deduksi gaya dorong, fraksi arus ikut, dan efesiensi badan kapal untuk kapal berbaling-baling tunggal dengan bentuk normal dan $D/L = 0{,}04$.

Koefisien blok yang besar akan memberikan harga fraksi deduksi gaya dorong yang tinggi seperti yang ditunjukkan di Gb. 6.5.7. Jika kapal yang bersangkutan tidak memakai bos tetapi memakai braket poros maka fraksi deduksi gaya dorongnya harus dikurangi dengan

$$\Delta t = -0{,}02 \qquad (6.5.18)$$

Jika harga rasio garis tengah-panjangnya berbeda dari D/L = 0,03 maka dapat dipakai koreksi berikut ini :

$$\Delta t = 4\left(\frac{D}{L} - 0{,}03\right) \qquad (6.5.19)$$

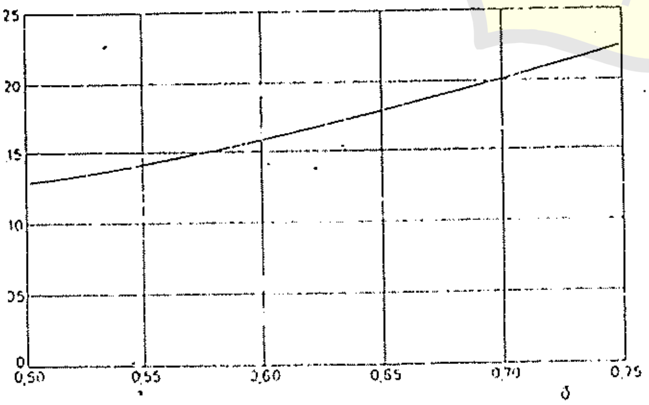
Selanjutnya, jika jarak kelonggaran ujung daun baling-baling (TC) tidak sebesar kira-kira 0,005L maka fraksi deduksi gaya dorongnya harus dikoreksi memakai :

$$\Delta t = -6\left(\frac{TC}{L} - 0{,}005\right) \qquad (6.5.20)$$

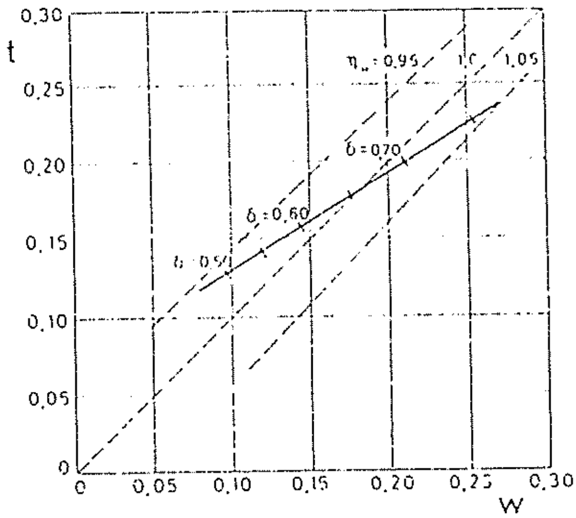
Dengan demikian maka harga t nya adalah

$$t = t_S + \Sigma\,\Delta t \qquad (6.5.21)$$

Gambar 6.5.8 menunjukkan hubungan antara fraksi deduksi gaya dorong, fraksi arus ikut, dan efesiensi badan kapal untuk kapal berbaling-baling ganda yang mempunyai bentuk yang normal dan D/L = 0,03 dan mungkin berguna untuk perkiraan awal.



Gambar 6.5.7. Fraksi deduksi gaya dorong untuk kapal berbaling-baling ganda, D/L = 0,03.



Gambar 6.5.8. Hubungan antara fraksi deduksi gaya dorong, fraksi arus ikut, dan efesiensi badan kapal untuk kapal berbaling-baling ganda yang mempunyai bentuk normal dan D/L = 0,03.

6.6. KAVITASI

6.6.1. Pendahuluan

Kavitasi merupakan fenomena yang dapat terjadi bila baling-baling bekerja dengan beban yang relatif tinggi. Kavitasi adalah proses dinamis. Dalam proses ini di dalam fluida yang tekanannya turun hingga pada tekanan uap fluida tersebut akan timbul sejumlah rongga (cavities) yang berisi uap. Jika pada baling-baling kapal timbul kavitasi maka, di atas kisaran kritis tertentu, akan terjadi pemecahan aliran yang terus meningkat, dan hal ini akan mengakibatkan berkurangnya gaya dorong. Kavitasi dapat menyebabkan kapal tidak dapat mencapai kecepatan yang diinginkan. Kavitasi juga dapat menimbulkan getaran, bunyi, dan erosi pada baling-baling. Jika pada seluruh permukaan suatu baling-baling kapal terdapat arus ikut yang berbeda-beda dan perbedaan itu besar maka pada permukaan itu akan cenderung terjadi kavitasi.

Dalam rekayasa umumnya kavitasi didefinisikan sebagai proses pembentukan fase uap dari suatu cairan ketika cairan tersebut mengalami pengurangan tekanan pada suhu sekeliling (ambient temperature) yang tetap. Secara umum suatu cairan dikatakan mengalami kavitasi jika di dalam cairan tersebut terlihat adanya gelembung yang terbentuk akibat turunnya tekanan. Untuk dapat memulai timbulnya kavitasi pada tekanan sebesar sekitar tekanan uap diperlukan sejumlah gelembung kecil, disebut inti (nuclei), sering cukup hanya dalam ukuran submikroskopis saja, yang mengandung gas permanen dan/atau uap cairan yang

p adalah perubahan tekanan dan merupakan karakteristik geometri aliran. σ_v disebut angka kavitasi uap. alam angka ini p_0 adalah tekanan statis, yaitu mlah dari tekanan hidrostatis dan tekanan atmosfir. ekanan uap p_v tidak tergantung pada suhu. Tekanan agnasi q tergantung pada massa jenis fluida dan pa- ι kecepatan aliran.

Agak terlalu optimistik kiranya menganggap bahwa vitasi mulai timbul ketika tekanan turun mencapai kanan uap air. Air laut mengandung banyak udara ng terikut (terbawa) dan larut didalamnya, dan engandung banyak sekali berbagai jenis inti yang pat mempengaruhi pembentukan awal rongga vitasi. Karena itu sebaiknya angka kavitasi didefinisin sebagai rasio antara selisih tekanan sekeliling yang solut p dan tekanan rongga kavitasi p_c dengan kanan dinamis aliran bebas (free stream dynamic essure)

$$\sigma = \frac{p - p_c}{q} \qquad (6.6.10)$$

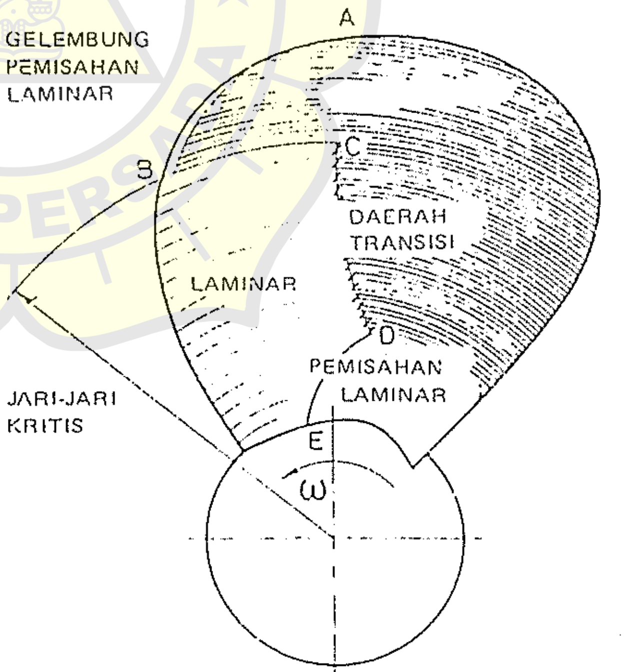
ngan demikian maka σ adalah karakteristik sistem ran-gas.

Tekanan rongga kavitasi adalah tekanan sebenarnya am kavitasi tunak atau kuasi tunak (quasisteady). kanan rongga kavitasi kira-kira sama dengan jumlah ua tekanan partial dari uap dan gas lainnya yang bawa dan tercampur (diffused) di dalam rongga. am sistem praktis definisi σ umumnya didasarkan la tekanan uap.

Harga angka kavitasi σ pada saat mulai terjadinya itasi di dalam suatu sistem aliran disebut angka itasi kritis σ. Kavitasi akan mulai timbul di suatu pat bila inti yang ada ditempat itu mencapai ukuran isnya akibat turunnya tekanan disekelilingnya. am fase awal riwayat kehidupan gelembung kavitasi di dalam tekanan yang turun itu gelembung tersebut n menjadi tidak stabil dan selanjutnya akan tumbuh gan cepat (kavitasi uap) atau tumbuh di dalam disi yang kuasi-setimbang (quasiequilibrium) karena si gas (kavitasi gas). Kandungan gas di dalam fluida at berupa kandungan gas larut atau tak larut. dungan gas seluruhnya sama dengan gas yang larut tak larut tersebut. Kandungan gas "bebas" (free) ι "terbawa" (entrained) merupakan istilah yang kai untuk kandungan gas yang tak larut. embung yang sedang mengembang permukaannya il.

etika suatu gelembung kavitasi transien (yang angsung sesaat) memasuki medan tekanan yang akin tinggi maka tibalah fase terakhir riwayat nbung tersebut. Permukaannya menjadi tidak stabil. Gelembung tersebut akan mengempis dan, kecuali jika mengandung gas asing dalam jumlah yang cukup, lenyap. Penggelembungan kembali (bubble rebound) adalah menggelembungnya kembali suatu kavitasi transien yang mengandung gas permanen dalam jumlah yang cukup setelah pertama kali mengempis. Ini karena adanya energi yang ditimbun di dalam gas yang mengalami pemampatan tersebut. Beberapa daur (cycles) pertumbuhan dan penggelembungan kembali kadang-kadang dapat diamati. Tekanan kempis gelembung (collapse pressure) adalah tekanan yang timbul di dalam medan gelembung kavitasi yang sedang dalam proses mengempis. Tekanan kempis ini dinyatakan dalam ribuan atmosfir dan diukur pada jari-jari minimum yang dicapai sebelum proses tersebut berhenti atau sebelum penggelembungan kembali terjadi.

Dalam uji model, aliran yang berada di sisi hisap daun baling-baling dapat berupa seperti yang ditunjukkan pada Gb. 6.6.2 [G. Kuiper (ITTC, 1978, bagian 2, halaman 148)]. Di daerah AB terdapat gelembung pemisahan laminar yang pendek yang kemudian diikuti dengan lapisan batas turbulen. Garis BC membedakan dengan jelas antara daerah turbulen setelah pemisahan dan daerah aliran laminar. Transisi alami (natural) berlangsung di daerah CD, sementara itu di dekat hub di suatu jarak dari tepi depan daun baling-baling dapat terjadi pemisahan laminar. Dalam hal ini semua penampang daun baling-baling dalam keadaan berhenti.



Gambar 6.6.2. Skema aliran lapisan batas pada sisi hisap daun baling-baling.

tak masing-masing titik $A-E$ pada daun baling-ling tentu saja tergantung pada geometri, beban, dan gka Reynolds baling-baling. Terutama titik B, titik bervariasi dari ujung daun hingga hub, tergantung ban baling-baling; sementara itu titik D dapat rvariasi dari C hingga E. Ditinjau menurut letak garis ord, daerah transisi CD sangat tergantung pada angka ynolds, dan akan bergeser menuju ke tepi depan daun ling-baling jika angka Reynoldsnya naik. Untuk gka Reynolds yang dipakai dalam praktek pelaksana-uji model (hingga sekitar 10^6) garis CD dan khusus-a titik C tidak akan pernah sampai dekat ke tepi pan daun baling-baling.

.3. Jenis kavitasi Baling-baling

boratorium uji kavitasi membuat sketsa atau motret pola kavitasi. Laboratorium demikian itu ing pula memberikan penjelasan mengenai hasil yang dapat berdasarkan penglihatan mata, yaitu mengenai vitasi uap (cloud), busa (foam), kabut (mist), nbaran (sheet), gelembung, buih (froth), bercak ot), dan garis (streak), dan sebagainya. Dari segi ika mengenai proses kavitasi, pembedaan kavitasi nurut jenisnya tidak perlu. Namun demikian mbedaan itu dalam praktek akan ada gunanya. Tidak a standar nyata yang dapat dipakai untuk menerang-n jenis kavitasi. Tetapi dapat dikatakan bahwa njelasan mengenai bentuk kavitasi harus mencakup terangan mengenai baik letak, ukuran, struktur, dan amika kavitasi, maupun dinamika aliran yang diacu ara benar.

Letak kavitasi dapat diterangkan sebagai berikut :

ung daun	Contoh : Kavitasi ujung (tip cavitation), yaitu kavitasi permukaan (surface cavitation) yang terjadi di dekat ujung daun baling-baling; kavitasi pusaran (vortex cavitation), yaitu kavitasi yang terjadi di dalam inti tekanan rendah pusaran ujung (tip vortex) baling-baling.
angkal daun (oot fillet)	Contoh : Kavitasi pangkal daun (root cavitation), yaitu kavitasi di dalam daerah tekanan rendah di pangkal daun baling-baling.

elah antara daun dan tabung baling-baling

ub atau konis one)	Contoh : Kavitasi hub atau kavitasi pusaran hub (hub vortex cavitation), yaitu kavitasi di dalam pusaran yang ditimbulkan oleh daun baling-baling pada hub. Jika baling-baling tersebut dianggap sebagai sayap maka akan diketahui bahwa di sebelah dalam atau di ujung hub pasti juga timbul pusaran. Tetapi karena rendahnya kecepatan penampang hub maka semakin dekat dengan pangkal daun sirkulasinya akan semakin berkurang dan pusarannya akan menjadi lebih lemah. Tetapi dalam kondisi beban yang tinggi pusaran demikian itu akan timbul pusaran hub yang menyusur ke belakang. Bentuknya seperti tali yang dipuntir dengan jumlah pilin yang sama dengan jumlah daun baling-baling.

Menurut letak penampang daun baling-baling tertentu, misalnya penampang di tengah (midchord).

Tepi depan	
Tepi ikut	Dalam kaitan ini, kavitasi pusaran ikut (trailing vortex cavitation) harus pula disebutkan. Kavitasi ini adalah kavitasi yang terus-menerus ada di dalam inti tekanan rendah pusaran ikut di dalam aliran yang meninggalkan baling-baling.

Alas

Sisi hisap (punggung)	Contoh : Kavitasi punggung (back side cavitation) adalah kavitasi yang terjadi pada punggung (sisi hisap) daun baling-baling.
Sisi tekanan (muka)	Contoh : Kavitasi muka (face cavitation) adalah kavitasi pada sisi tekanan (muka) daun baling-baling. Kavitasi ini umumnya ditimbulkan akibat kerja baling-baling yang demikian rupa hingga sudut pukul lokal daun baling-baling itu sangat negatif.
Antara baling-baling dan badan kapal	Kavitasi pusaran antara baling-baling dan badan kapal (propeller-hull vortex cavitation) diartikan sebagai kavitasi pusaran ujung daun baling-baling yang dalam interval tertentu merentang hingga mencapai permukaan badan kapal.

Jika ada kavitasi yang meluas (developed) maka ukuran kavitasi dapat dinyatakan dalam ukuran benda, misalnya, dengan menyatakannya menurut luas daun baling-baling yang diselimuti oleh suatu jenis kavitasi tertentu.

Struktur kavitasi dapat dinyatakan sebagai berikut :

Kavitasi lembaran (umumnya tipis, halus, tembus pandang, umumnya stabil, tidak stabil hanya di dalam medan arus ikut atau di dalam aliran yang miring)

Kavitasi bercak (bentuk khusus kavitasi lembaran; sempit, melekat pada permukaan, timbul pada bercak kekasaran yang terpencil atau pada bagian permukaan yang cacat)

Kavitasi garis (bentuk khusus kavitasi bercak; sempit, umumnya sejajar satu sama lain dan timbul pada bercak kekasaran yang terpencil atau pada bagian tepi depan daun yang cacat)

Kavitasi awan (di bagian belakang atau ujung patah kavitasi lembaran yang tak stabil di dalam medan arus ikut, massa dari rongga transien, umumnya terkait dengan erosi)

Kavitasi gelembung (terpencil, bergerak)

Kavitasi pusaran

Gambar yang menunjukkan contoh dari berbagai jenis kavitasi dapat dilihat di kepustakaan; lihat, misalnya, ITTC (1978, halaman 310).

Dinamika rongga kavitasi dapat dikategorikan sebagai :

Tunak (atau lebih baik, kuasi-tunak)

Tak tunak

Tidak menetap

Transien atau bergerak

Menempel (secara tetap atau berlangsung dalam interval waktu, dalam bentuk kavitasi yang mengembang sebagian atau sepenuhnya atau sebagai sejumlah pusaran)

Bergerak mengikut (misalnya, kavitasi pusaran)

Karakteristik dinamis aliran yang mengalami kavitasi dapat dinyatakan dengan memakai notasi berikut ini :

Lapisan batas laminar

Lapisan batas turbulen

Aliran tunak

Aliran tak tunak

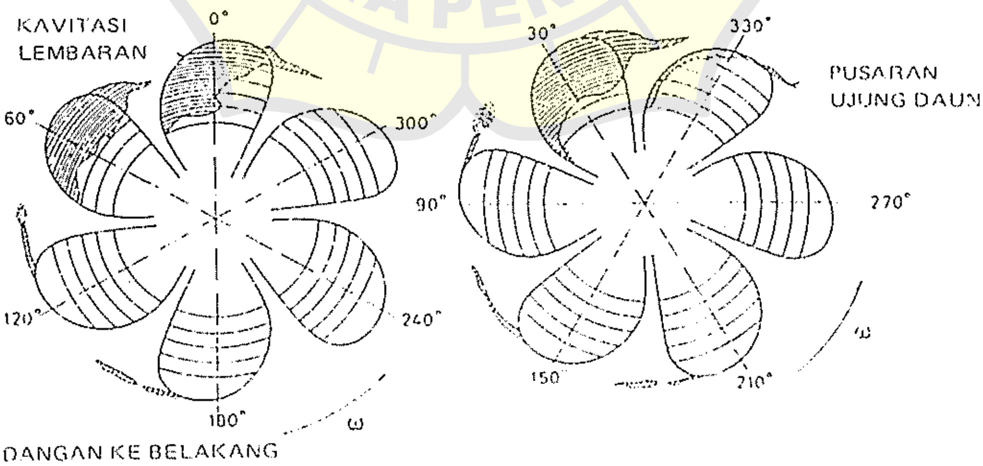
Aliran yang mengalami pemisahan

Pusaran bebas

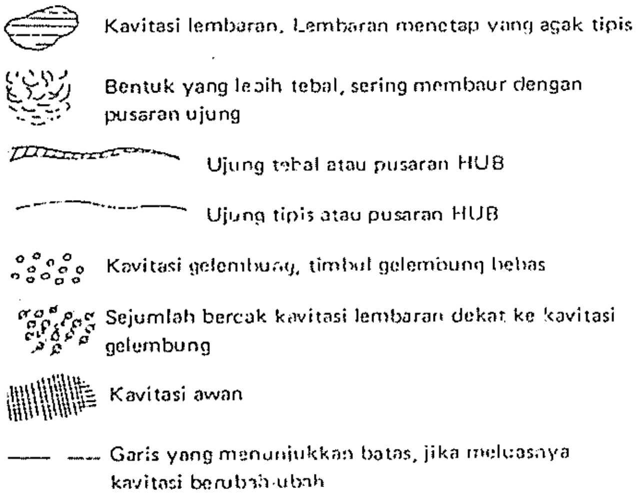
Lapisan geser (shear layers)

Aliran arus ikut (seragam, tak seragam)

Jika dipakai cara pengamatan yang tidak berdasarkan langsung dari penglihatan mata (misalnya, fotografi berkecepatan tinggi, holografi, penyebaran sinar, metode Schlieren, metode akustik) maka jenis kavitasi dapat dinyatakan memakai istilah khusus. Contoh penjelasan gambar kavitasi pada baling-baling berdaun enam untuk kapal pengangkut peti kemas berkecepatan tinggi diberikan di Gb. 6.6.3. Sering bahwa sketsa dalam bentuk demikian itu diberikan oleh pihak laboratorium kepada pihak pemilik kapal atau pihak galangan. Penyajian pola kavitasi secara skematis seperti itu masih belum distandarkan sepenuhnya, tetapi banyak galangan yang memakai notasi yang ditunjukkan di Gb. 6.6.4.



Gambar 6.6.3. Contoh hasil uji kavitasi dengan memakai model baling-baling kapal pengangkut peti kemas.



Gambar 6.6.4. Skema penyajian pola kavitasi.

.6.4. Pengaruh kavitasi yang merusak

Kavitasi pada baling-baling kapal mempunyai beberapa ›engaruh yang merusak. Pertama, efesiensi baling-›aling akan berkurang. Ini berarti bahwa dengan daya nesin penggerak yang sama baling-baling yang nengalami kavitasi akan memberikan kecepatan kapal ang lebih rendah daripada baling-baling yang bekerja anpa kavitasi. Dengan adanya kavitasi maka baling-aling akan tidak bekerja di dalam air yang homogen etapi di dalam cairan yang tercampur dengan uap dan as, dan ini menurunkan daya propulsi.

Kedua, kavitasi dapat menyebabkan erosi pada ahan. Seperti yang disebutkan di 6.6.2 pengempisan elembung kavitasi akan menghasilkan tekanan yang angat tinggi yang kadang-kadang dapat menyebabkan erusakan yang parah pada bahan. Cara yang nenyebabkan terjadinya kerusakan itu sendiri tidak apat dipahami sepenuhnya, tetapi barangkali karena danya hubungan fisik kimia–metalurgi yang timbal alik. Erosi baling-baling kapal dapat dibedakan ke alam dua kelas :

. Keausan umum atau pengasaran yang meliputi daerah yang cukup luas.
. Erosi cepat dan burik (pitting) pada luasan setempat.

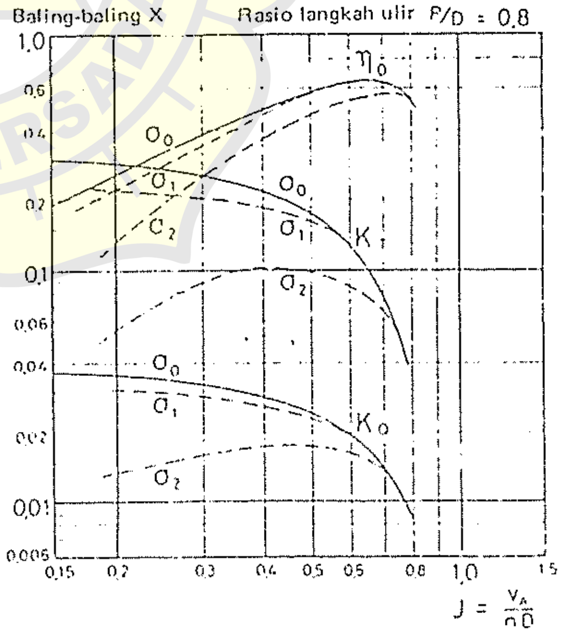
rosi pada daun baling-baling dapat menyebabkan ırunnya efesiensi baling-baling.

Ketiga, kavitasi dapat menyebabkan getaran dan unyi, dan ini sering merupakan sumber masalah yang esar, misalnya pada kapal tangki yang mempunyai aya mesin yang besar.

Masalah ini dapat dipelajari dengan melakukan ercobaan memakai sejumlah model yang sesuai di terowongan kavitasi (lihat Bab 3, 3.3), serta dapat dicarikan pula jalan keluar untuk mengurangi, bahkan barangkali menghindari sama sekali, pengaruh kavitasi yang merusak itu.

Untuk menentukan karakteristik unjuk kerja baling-baling pada berbagai angka kavitasi yang berbeda dapat dipakai terowongan kavitasi yang konvesional. Karakteristik tersebut digambarkan dengan cara yang sama seperti halnya hasil dari uji baling-baling terbuka, hanya saja untuk masing-masing harga angka kavitasi σ akan diperoleh perangkat kurva yang terpisah (lihat Gb. 6.6.5).

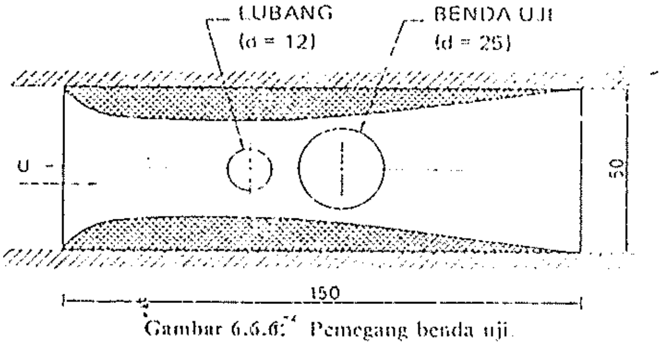
Kerusakan akibat kavitasi terjadi karena tumbukan (impact) ketika rongga kavitasi mengempis, dan gaya tumbuk (impact force) ini danggap berasal dari sejumlah gelombang kejut (shock waves) atau pancaran mikro (microjets). Alasan untuk gelombang kejut tersebut didukung oleh suatu laporan yang sistematik mengenai perhitungan tekanan untuk rongga kavitasi yang mengempis dan dengan percobaan yang dilakukan untuk mendapatkan perkiraan harga tekanan kempis yang terbesar. Tekanan kempis terbesar akan tidak kurang 10^9 N/m^2. Alasan untuk pancaran mikro tersebut didasarkan pada hasil pengamatan; yaitu bahwa gelembung itu tumbuh dan mengempis secara tidak simetris di dekat permukaan benda padat dan ketika pengempisan berlangsung timbul pancaran dengan kecepatan yang sangat tinggi yang menumbuk kuat-kuat permukaan benda padat tadi.



Gambar 6.6.5. Kurva karakteristik untuk baling-baling di dalam terowongan kavitasi. σ_0 adalah angka kavitasi pada tekanan atmosfir.

Banyak percobaan yang telah dilakukan untuk membandingkan tahanan relatif dari berbagai bahan dengan kerusakan akibat erosi. Diperkenalkan konsep kekuatan erosi (erosion strength). Konsep ini telah berhasil dipakai sekalipun cara penyaluran energi ke bahan sangat beraneka ragam. Cukup banyak pula upaya yang telah dilakukan untuk mencari hubungan antara beberapa sifat mekanis bahan baling-baling yang dapat langsung diukur dengan kemampuan bahan tersebut dalam menahan kerusakan akibat erosi, dengan percobaan kavitasi, tubrukan (impingement), atau lainnya. Dalam pelaksanaan pengujian, erosi pada benda uji di dalam fluida dapat ditimbulkan dengan jalan menggetarkan benda tersebut, misalnya, seperti yang diajukan dalam "Standard Method of Vibratory Cavitation Erosion Test". (Metode Standar untuk Pengujian Kavitasi dengan Gerakan) (ASTM, 1972).

Pengujian demikian itu dapat dilakukan di tempat yang mempunyai fasilitas untuk foil yang berputar, di tempat yang mempunyai aparatus untuk diskus yang berputar (**Dashnaw** dan kawan-kawan, 1980), atau di terusan aliran air dengan sirkulasi tertutup (**Hansson** dan **Mørch**, 1977). Bagian pengujian dari fasilitas tersebut mempunyai alat pemegang benda uji (specimen holder). Di alat ini benda akan diuji disisipkan demikian rupa sehingga merupakan bagian dari dinding induk (central wall) yang mulus. Gambar 6.6.6 menunjukkan sebuah alat pemegang benda uji. Aliran melewati ke dua sisi dinding tersebut secara simetris. Sebuah lubang di dalam dinding tersebut akan menimbulkan rongga kavitasi di dekat benda di dalam aliran yang menuju ke benda (upstream). Dengan mengatur tekanan dalam rentang tekanan kempis dan kecepatan aliran maka rongga tersebut akan mengempis di dekat permukaan benda uji. Salah satu cara untuk mengalibrasi berbagai keadaan kerusakan akibat kavitasi adalah dengan memakai aloi nikel yang kekuatan dan kekerasannya ditentukan lebih dulu sebagai bahan standar. Secara umum dapat diperhatikan bahwa semakin keras, kuat, dan kaku (modulusnya besar) material itu semakin tahan terhadap kerusakan akibat erosi.



Gambar 6.6.6. Pemegang benda uji.

Untuk dapat memperkirakan erosi baling-baling dengan cara yang dapat diandalkan maka telah dikembangkan suatu cara yang disebut teknik "permukaan lunak" (soft surface). Karena erosi kavitasi menjadi cepat ketika mencapai intensitas kritis maka pemakaian lapisan permukaan (coating) yang lunak yang intensitasnya pada model yang dipakai disesuaikan dengan skala model itu akan dapat memberikan kriteria yang memuaskan. Permukaan yang dipakai untuk model baling-baling dapat bervariasi dari aloi aluminium anoda dan timah lunak murni hingga tinta yang dipakai dalam rekayasa untuk membuat cetakan biru, tinta stensil, dan tinta bolpoin. Proses erosi pada permukaan yang dibuat dari bahan metal dapat memakan waktu beberapa hari, sedangkan pengujian dengan memakai lapisan dari tinta stensil akan dapat diselesaikan dalam waktu 5 menit saja. Metode permukaan lunak dengan waktu uji yang tepat terbukti memberikan petunjuk mengenai erosi pada skala penuh (benda yang sebenarnya) yang dapat dipercaya, dan memberikan perkiraan letak erosi yang lebih tepat daripada yang diperkirakan berdasarkan metode visual.

Badan kapal mendapatkan eksitasi dari baling-baling terutama dalam dua cara : (1) Beban daun baling-baling yang tidak tunak dapat disalurkan ke badan kapal melalui poros (gaya bantalan = bearing forces) dan (2) medan tekanan yang mengikuti kisaran daun baling-baling disalurkan melalui air ke badan kapal, menyebabkan timbulnya tekanan getar pada pelat badan kapal (gaya permukaan = surface forces). Hasil percobaan menunjukkan bahwa dalam kondisi tidak ada kavitasi kedua jenis gaya tersebut mempunyai besaran yang hampir sama. Karena adanya kavitasi tak tunak yang ekstensif pada baling-baling sebagian besar kapal niaga maka gaya permukaan umumnya beberapa kali lebih besar daripada gaya bantalan. Dengan demikian maka besarnya gaya permukaan sebagian besar ditentukan oleh perilaku kavitasi yang ada pada baling-baling yang bersangkutan. Jika akan menentukan gaya ini dengan percobaan model maka percobaan tersebut harus dilakukan di tempat yang mempunyai fasilitas demikian rupa sehingga model baling-baling tersebut akan bekerja dan mengalami kavitasi di dalam medan arus ikut dengan kondisi yang sedapat mungkin sama dengan kondisi yang sebenarnya. Jenis fasilitas berikut ini dapat dipakai untuk pengujian demikian itu :

1 Terowongan kavitasi konvensional (lihat Gb. 3.3.1*B*); medan arus ikut ditimbulkan dengan memakai beberapa model badan belakang (model tiruan = dummy models) yang dikombinasikan dengan jala.

2. Terowongan kavitasi yang tempat (ruang) ujinya mempunyai panjang dan luas yang dapat menampung model yang lengkap yang diperlukan untuk menimbulkan medan arus ikut (lihat Gb. 3.3.1*C*).
3. Fasilitas yang dapat dipakai untuk melakukan pengujian di permukaan air bebas (lihat Gb. 3.3.1*D* dan Gb. 3.3.1*G*).

Fluktuasi tekanan dapat diukur dengan transduser tekanan yang dipasang rata dengan permukaan badan kapal. Transduser tersebut dibuat dalam bentuk silinder dengan garis tengah sekitar 20 mm dan tinggi sekitar 15 mm. Perpindahan relatif antara inti ferit (ferrite core) yang dipasang pada membran dengan kumparan yang dipasang di dalam tempat transduser diukur dengan memakai jembatan elektris.

Jika bukan getaran tetapi bunyi akibat kavitasi yang merupakan obyek yang dikehendaki maka transduser tekanan tersebut diganti dengan hidropon (hydrophone). Dalam hal ini skala merupakan masalah yang sangat rumit, dan harus dipakai beberapa anggapan. Sebagai anggapan dasar adalah pola kavitasi pada model dan pola kavitasi dalam skala penuh keduanya memenuhi kesamaan geometris. Anggapan ini mempunyai pengertian bahwa jari-jari masing-masing gelembung berbanding lurus dengan faktor skala. Selain itu, lingkup daerah meluasnya gelembung kavitasi dan distribusi ukuran relatifnya yang timbul pada model dianggap sama dengan yang timbul pada skala penuh. Dari anggapan itu maka banyaknya gelembung yang timbul pada daun model baling-baling pada suatu posisi sudut dianggap sama dengan banyaknya gelembung yang timbul pada daun baling-baling yang sebenarnya pada posisi itu. Berikut ini akan dibahas lebih lanjut mengenai masalah itu.

6.6.5. Prosedur Uji Model di dalam Terowongan Kavitasi.

Beberapa fasilitas yang dapat dipakai untuk melakukan uji kavitasi dengan memakai model dibahas di Bab 3, 3.3. Pengujian kavitasi harus dilakukan demikian rupa sehingga semua gaya spesifik (seperti misalnya gaya dorong dan gaya torsi) yang bekerja pada model mirip dengan yang bekerja pada obyek dalam skala penuh. Karena itu syarat berikut ini harus dipenuhi :

1. Kesamaan geometris.
2. Kesamaan kinematis.
3. Kesamaan dinamis.

Menurut butir 1 maka model tersebut harus merupakan obyek yang sebenarnya yang diperkecil dalam suatu skala. Secara umum model baling-baling hampir merupakan jiplakan dari baling-baling yang sebenarnya. Begitu pula halnya dengan badan kapal, tetapi karena terbatasnya ukuran terowongan kavitasi atau fasilitas maka kondisi lingkungan di sekeliling model skala tidak dapat sama seperti kondisi lingkungan sebenarnya yang diperkecil dalam skala itu. Pasti akan ada masalah mengenai permukaan bebas dan akan ada pengaruh dinding terowongan. Contohnya, gelombang tekanan yang ditimbulkan oleh masing-masing rongga kavitasi akan dipantulkan dari dinding terowongan. Dengan demikian maka sinyal yang dicatat oleh transduser pada badan model adalah jumlah dari sinyal dari gelombang tekanan yang ditimbulkan langsung oleh rongga kavitasi dengan sinyal dari gelombang tekanan yang dipantulkan dari dinding terowongan. Agar sinyal dari gelombang tekanan yang dipantulkan dari dinding demikian itu dapat dikontrol maka kondisi pemantulan dari dinding terowongan harus diperhitungkan dalam prosedur kalibrasi.

Kesamaan kinematis (butir 2) akan terpenuhi jika kecepatan pada sisi model dan kecepatan pada sisi obyek yang sebenarnya semuanya mempunyai arah yang sama. Maka

$$\frac{V_{Am}}{n_m D_m} = \frac{V_{As}}{n_s D_s} \qquad (6.6.11)$$

$$J_m = J_s \qquad (6.6.12)$$

$$V_{Am} = \frac{n_n}{n_s} \cdot \frac{V_{As}}{\lambda} \qquad (6.6.13)$$

V_A adalah kecepatan maju baling-baling, n laju kisaran, D garis tengah baling-baling, J angka maju, dan rasio skala. Huruf m dan s yang ditulis di bawah masing-masing menunjukkan bahwa kuantitas tersebut berlaku untuk model dan untuk kapal. Ini juga berarti bahwa distribusi arus ikut pada model skala harus seperti distribusi arus ikut di belakang buritan baling-baling pada kapal yang sebenarnya. Medan arus ikut dapat ditimbulkan dengan memakai model kapal yang lengkap yang diletakkan di dalam tempat uji diterowongan kavitasi atau dengan memakai sejumlah model badan belakang yang dikombinasikan dengan memakai jala.

Untuk kesamaan dinamis (butir 3) hukum Froude dan hukum Reynolds harus dipenuhi :

$$V_{Am} = \frac{V_{As}}{\sqrt{\lambda}} \quad \text{(hukum Froude)} \qquad (6.6.14)$$

$$V_{Am} = V_{As}\lambda \quad \text{(hukum Reynolds)} \qquad (6.6.15)$$

ila dalam percobaan model terjadi kavitasi maka esamaan dinamis tersebut juga mensyaratkan agar (a) ukum kesamaan angka kavitasi, (b) hukum Weber, an (c) pengaruh kandungan udara di dalam air pada nomena kavitasi, harus pula diperhitungkan.

Untuk butir (a) diperlukan, antara model dan apal, fenomena kavitasi yang sama dan resiko kavitasi ang sama. Fenomena kavitasi yang sama berarti

$$\left(\frac{p - p_0}{\frac{1}{2}\rho U^2}\right)_m = \left(\frac{p - p_0}{\frac{1}{2}\rho U^2}\right)_s \qquad (6.6.16)$$

au

$$\frac{\Delta p_m}{q_m} = \frac{\Delta p_s}{q_s} \qquad (6.6.17)$$

n resiko kavitasi yang sama berarti

$$\left(\frac{p_0 - p_v}{q}\right)_m = \left(\frac{p_0 - p_v}{q}\right)_s \qquad (6.6.18)$$

au

$$\sigma_{vm} = \sigma_{vs} \qquad (6.6.19)$$

n ini menunjukkan bahwa angka kavitasi untuk odel harus sama dengan angka kavitasi untuk skala nuh. Simbol yang dipakai dalam Pers. (6.6.16) – 6.19) telah dijelaskan sebelumnya; juga lihat njelasan mengenai Pers. (6.6.1) – (6.6.9). Selanjutnya erlukan kesamaan dalam tegangan permukaan embung kavitasi. Ini memerlukan kesamaan dalam gka Weber W untuk rongga yang serupa :

$$W = \frac{\rho U^2 l}{T} \qquad (6.6.20)$$

adalah tegangan permukaan, ρ massa jenis fluida, U cepatan, l panjang karakteristik, dapat berupa garis ngah gelembung. Dengan memakai yang disebut pilaritas kinematis (kinematic capilarity)

$$\kappa = \frac{T}{\rho} \qquad (6.6.21)$$

maka berdasarkan hukum Weber

$$U_m = U_s \sqrt{\frac{\kappa_m}{\kappa_s}} \sqrt{\lambda} \qquad (6.6.22)$$

U_m adalah kecepatan air di dalam tempat uji di terowongan kavitasi.

Jelas bahwa kelima syarat yang disebutkan tadi :

(6.6.13) :	$U_m = c_1 U_s \lambda^{-1}$	$(J_m = J_s)$	(6.6.23)
(6.6.14) :	$U_m = c_2 U_s \lambda^{-1/2}$	(Froude)	
(6.6.15) :	$U_m = c_3 U_s \lambda$	(Reynolds)	
(6.6.19) :	$U_m = c_4 U_s$	$(\sigma_{vm} = \sigma_{vs})$	
(6.6.22) :	$U_m = c_5 U_s \lambda^{1/2}$	(Weber)	

dalam pelaksanaan pengujian di terowongan kavitasi, tidak dapat dipenuhi secara serentak. U adalah kecepatan aliran pada profil baling-baling, λ rasio skala, dan $c_1 - c_5$ merupakan koefisien yang berbeda. Persamaan (6.6.13), kesamaan angka maju, harus selalu dipenuhi. Persamaan (6.6.19), kesamaan angka kavitasi, harus juga dipenuhi untuk menjamin adanya kesamaan dalam fenomena kavitasi. Umumnya hukum Froude diabaikan seperti halnya dalam uji baling-baling terbuka yang biasa.

Harga angka Reynolds tidak boleh terlalu rendah. Jika harga angka Reynolds rendah maka akan ada resiko bahwa sebagian besar dari baling-baling model yang bersangkutan akan mempunyai aliran laminar, sedangkan yang skala penuh akan mempunyai aliran turbulen. Harga angka Reynolds terendah yang dapat dipakai tidak dapat digunakan untuk mendapatkan suatu kriteria. Harga angka Reynolds yang diperlukan sangat tergantung pada jenis dan ukuran profil baling-baling dan juga pada medan arus ikut. Secara kasar dapat dikatakan bahwa baling-baling yang mempunyai garis tengah 200 – 250 mm sebaiknya dioperasikan pada laju kisaran yang tidak kurang dari 25 – 30 kisaran per detik, dan ini berarti angka Reynolds sebesar sekitar 10^6. Dalam hal ini angka Reynolds didefinisikan sebagai

$$R_n = \frac{C_{0,75R}\sqrt{V_A^2 + (0,75\ \pi n D)^2}}{\nu} \qquad (6.6.24)$$

$c_{0,75R}$ adalah lebar daun baling-baling pada $0,75R$, R jari-jari baling-baling, D garis tengah, n laju kisaran, V_A kecepatan maju baling-baling, dan ν koefisien viskositas kinematis.

Angka Reynolds juga dapat didefinisikan sebagai

$$R_n = 5{,}3 \frac{A_E/A_0}{Z} \frac{nD^2}{\nu} \qquad (6.6.25)$$

ɛrsamaan ini memberikan harga angka Reynolds yang ɪmpir sama dengan yang diberikan oleh Pers. (6.6.24). $_E$ adalah luas bentang daun baling-baling, A_0 luas ɪkus, Z banyaknya daun baling-baling, dan n, D, serta seperti dalam Pers. (6.6.24).

Mengenai hukum Weber, sekalipun harga kritis gka Reynolds dilampaui kecepatan dalam pelaksanaan rcobaan umumnya tidak akan cukup untuk dapat ɛmenuhi hukum Weber tersebut. Selain itu, ndungan gas di dalam air yang berada di terowongan vitasi juga merupakan hal yang penting. Untuk ɛndapatkan hasil pengamatan yang tepat mengenai ɪomena kavitasi air tersebut harus mempunyai ndungan gas yang sesuai.

Pada bagian atas terowongan terdapat kubah (dome) ɪg berisi air yang mempunyai permukaan bebas (lihat). 3.3.2) dan udara di atas permukaan air di bawah sebut dapat dipompa keluar dengan memakai pompa cum hingga dicapai tekanan statis di tengah model uai dengan yang dikehendaki. Setelah beberapa saat nudian kandungan gas di dalam air tersebut juga ktis akan tetap. Sebagai ukuran kandungan gas akai rasio kandungan gas, yaitu rasio antara gas ·ut dan tak larut) di dalam cairan yang diuji dengan ɪdungan gas di dalam cairan jenuh (saturated) pada u dan tekanan standar

$$a_s = \frac{\alpha}{\alpha_s} \qquad (6.6.26)$$

ɪdungan gas di dalam cairan dapat dalam keadaan ɪt atau tak larut. Sebagaimana disebutkan di 6.6.1, ɪl terjadinya kavitasi diduga ada kaitannya dengan dalam keadaan tak larut yang dikandung di dalam . Agar di dalam air terdapat inti dalam jumlah yang up untuk dapat mengawali terjadinya kavitasi dan ɪyebabkan kavitasi dapat tumbuh, kandungan gas alam air tersebut harus melebihi harga batas tertentu salnya a_s = 0,3). Jika kandungan gas menjadi lebih lah daripada harga batas tersebut maka pertumbuh- dan tebal rongga kavitasi yang terjadi akan kurang dan fluktuasi tekanan pada badan model ɪngkali akan terlalu rendah.

ika percobaan dilakukan di terowongan kavitasi g tempat ujinya mempunyai panjang dan luas yang at menampung model yang lengkap maka dapat ɪrapkan bahwa harga fluktuasi tekanan yang dicatat hasil percobaan tersebut akan lebih tepat daripada hasil yang dicatat dari terowongan yang lebih kecil. Selain itu, jika medan arus ikut seluruhnya hanya ditimbulkan oleh badan model saja tanpa kontribusi dari jala maka dapat diharapkan bahwa interferensi antara baling-baling dan badan kapal yang penting yang dihasilkan dengan cara itu adalah benar.

Fasilitas yang mempunyai permukaan bebas seperti terowongan jenis D dan G (Gb. 3.3.1) dapat diharapkan memberikan keuntungan tambahan sebagai berikut :

1. Distribusi arus ikut yang dihasilkan agak lebih baik daripada yang dihasilkan di fasilitas tanpa permukaan bebas.
2. Percobaan dengan kondisi balas, yaitu baling-baling berada didekat permukaan air, dapat dilakukan.

Di lain pihak pemakaian fasilitas dengan permukaan bebas tersebut juga memberikan kerugian :

1. Karena adanya permukaan bebas maka kecepatan model harus sesuai dengan hukum Froude. Ini berarti bahwa kecepatan aliran akan agak rendah (1 – 3 m/detik). Agar dapat membuat angka kavitasi yang benar diperlukan tekanan statis yang sangat rendah di dalam terowongan kavitasi. Tekanan rendah ini dapat menyulitkan pengadaan inti dalam jumlah yang cukup atau spektrum inti yang sesuai untuk dapat menghasilkan bentuk kavitasi yang "benar." Untuk mengatasi kesulitan ini maka inti harus diadakan secara rekaan, misalnya dengan memasukkan udara ke dalam air atau dengan cara eletroklisa.
2. Keterbatasan kecepatan berarti rendahnya angka Reynolds. Ini akan menyebabkan tidak sesuainya pola kavitasi yang dihasilkan di terowongan dengan pola kavitasi dalam skala penuh. Masalah ini dapat diatasi sebagian dengan memakai model kapal yang lebih besar daripada yang umumnya dipakai ditangki percobaan (12 m dibandingkan dengan 6 – 8 m).

Dalam hal tertentu terowongan kavitasi harus dikalibrasi. Melalui the International Towing Tank Conference (ITTC) telah dilakukan pembandingan hasil percobaan dari berbagai terowongan. Dengan begitu maka masing-masing laboratorium dapat memeriksa ketepatan fasilitasnya. Beberapa laboratorium membandingkan foto yang diambil dari uji kavitasi dengan foto erosi baling-baling kapal yang diamati dalam pengedokan. Ini merupakan cara yang baik sekali untuk mengalibrasi terowongan kavitasi. Pemotretan kavitasi pada skala penuh dan pada model yang diambil dengan kecepatan tinggi juga dapat menghasilkan informasi yang berguna.

Sekalipun masih banyak masalah yang belum dapat dipecahkan sepenuhnya mengenai pelaksanaan uji model di terowongan kavitasi, percobaan demikian itu dapat memberikan banyak informasi dan petunjuk mengenai berbagai pengaruh yang merusak dari kavitasi.

6.6.6. Kriteria untuk Pencegahan Kavitasi

Dalam menyiapkan proposal awal untuk kapal baru hal yang ingin diketahui oleh pihak arsitek kapal dalam tahap dini adalah ukuran utama dan karakteristik baling-baling. Baling-baling harus demikian rupa hingga tidak terjadi kavitasi yang merusak; karena itu, perlu adanya kriteria sederhana untuk memprakirakan terjadinya kavitasi. Kriteria demikian itu dapat didasarkan pada gaya dorong baling-baling rata-rata tiap satuan luas proyeksi permukaan daun dalam hubungannya dengan angka kavitasi, kadang-kadang angka kavitasi setempat. **Burill** (1943) memakai koefisien yang τ_c yang didefinisikan sebagai

$$\tau_c = \frac{T/A_p}{\frac{1}{2}\rho(V_R)^2} = \frac{T/A_p}{q_{0,7R}} \qquad (6.6.27)$$

T = gaya dorong baling-baling
A_p = luas proyeksi daun
ρ = massa jenis
V_R = kecepatan relatif air pada 0,7 jari-jari ujung R
$q_{0,7R}$ = tekanan dinamis pada 0,7 jari-jari ujung

Dalam diagram yang diberikan oleh Burill τ_c digambar berdasarkan angka kavitasi setempat pada 0,7 jari-jari :

$$\sigma_{0,7R} = \frac{p_0 - p_v}{q_{0,7R}} \qquad (6.6.28)$$

$p_0 - p_v$ = tekanan pada garis pusat baling-baling
p_0 = tekanan sekeliling yang absolut (absolute ambient pressure)
p_v = tekanan uap air

Tekanan absolut sekitar (sekeliling) nya pada garis pusat baling-baling adalah tekanan atmosfir ditambah dengan tekanan dari kolom air di atas poros baling-baling; ini berarti

$$p_0 = \text{atm} + \rho g(T - E + \zeta_A) \qquad (6.6.29)$$

ρ adalah massa jenis, g percepatan gravitasi, T sarat kapal, E tinggi letak poros dari garis dasar, dan ζ_A adalah amplitudo gelombang. ζ_A dapat dianggap sekitar $0{,}0075L$ atau dapat perkirakan dengan memakai diagram di Gb. 6.4.12 atau 6.4.13. L adalah panjang kapal.

Jika tekanan atmosfir sama dengan 101,3 kN/m² (atau kPa) (tekanan atmosfir standar pada permukaan laut) maka $p_0 - p_v$ pada 15°C menjadi

$$p_0 - p_v \sim 99{,}6 - 10{,}05(T - E + \zeta_A) \quad \text{(kPa)} \qquad (6.6.30)$$

p_v pada 15°C adalah sekitar 1,7 kPa. Variasi p_v terhadap suhu ditunjukkan di Gb. 6.6.7. Kurva tersebut dianggap berlaku baik untuk air tawar maupun untuk air laut.

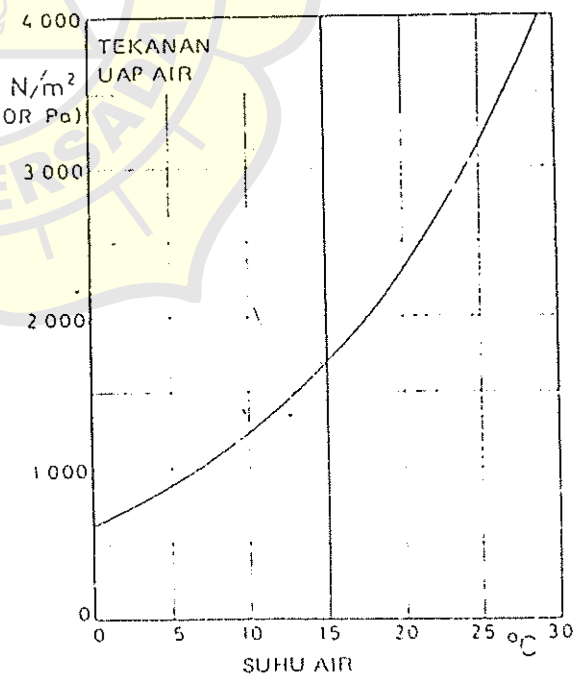
Kecepatan relatif air pada 0,7 jari-jari ujung adalah

$$V_R = \sqrt{V_A^2 + (0{,}7\,\pi D n)^2}$$

V_A = kecepatan maju baling-baling
D = garis tengah baling-baling
n = laju kisaran

Luas proyeksi daun baling-baling A_p hampir sama dengan

$$A_p \sim A_D(1{,}067 - 0{,}229\,P/D) \qquad (6.6.31)$$



Gambar 6.6.7. Kurva tekanan uap air terhadap suhu.

) adalah luas kembang daun baling-baling; dalam ·hitungan kasar luas ini dapat diganti dengan luas ıtang daun baling-baling A_E.

Gambar 6.6.8. menunjukkan salah satu kurva yang .jukan oleh **Burrill** (1943). Kurva tersebut merupa- ı kurva "batas atas yang disarankan untuk baling- .ing kapal niaga", yaitu berarti bahwa untuk nghindari kavitasi yang berlebihan dan erosi dalam ndisi pelayaran rata-rata di laut maka baling-baling ɔal yang bersangkutan harus bekerja di bawah kurva sebut.

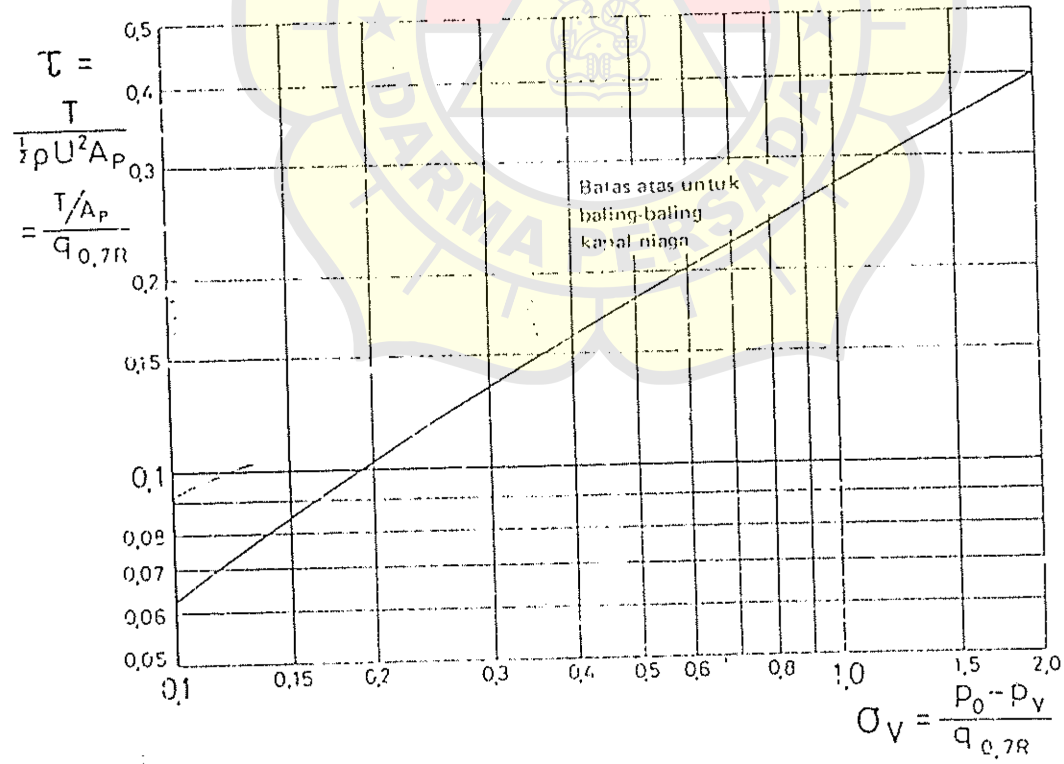
Kriteria tersebut dapat pula dinyatakan dalam syarat ıwa luas bentang yang diperlukan harus tidak kurang i

$$\left(\frac{E}{0}\right)_{nec}$$

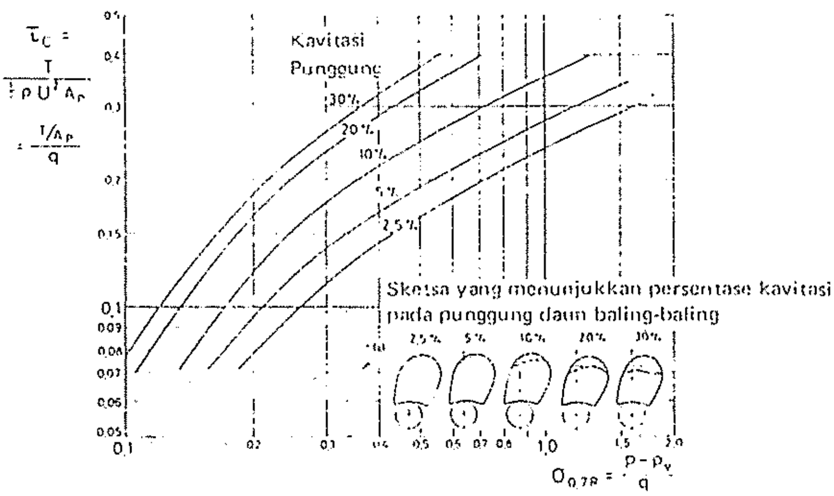
$$= \frac{T}{A_0(1{,}067 - 0{,}229P/D)(0{,}3\sigma_{0,7R}^{0,5} - 0{,}03)q_{0,7R}} \quad (6.6.32)$$

adalah luas diskus baling-baling $(=\pi D^2/4)$. teria ini sangat kasar. **Van Manen** memakai teori ;aran untuk menghitung seri baling-baling berdaun dua, tiga, empat, dan lima dengan berbagai rasio luas daun dan dengan berbagai rasio langkah ulir. Hasilnya digambar dalam diagram (Manen, 1957b, Gb. 66 dan 67), yaitu seperti Gb. 6.6.8. Hasil tersebut menunjukkan ketergantungan kriteria kravitasi tersebut pada parameter tadi, terutama pada rasio langkah ulir.

Hasil yang diberikan di Gb. 6.6.9 adalah hasil dari baling-baling berdaun empat dengan rasio luas bentang 0,60 dari seri baling-baling kapal niaga yang diuji di suatu terowongan kavitasi (Burrill dan Emerson, 1962–1963) terhadap koefisien maju dan angka kavitasi dalam rentang yang luas. Dalam gambar tersebut diberikan garis untuk 2,5%, 5%, 10%, 20%, dan 30% kavitasi punggung yang timbul. Dari gambar tersebut terlihat bahwa garis batas atas untuk baling-baling kapal niaga yang ditunjukkan di Gb. 6.6.8. terletak sangat dekat dengan garis untuk 5% kavitasi punggung. Hasil pengamatan baling-baling menunjukkan bahwa jika baling-baling tersebut bekerja pada kondisi perancangan atau pada kondisi kerja yang sesuai dengan garis 5% maka baling-baling itu akan didapatkan dalam keadaan yang cukup bersih dan bebas erosi, barangkali bukan karena mengkilapkan permukaan logam tersebut setelah beberapa tahun bekerja. Karena satu dan lain hal mengusahakan agar mendapatkan luas daun yang sekecil mungkin lebih disukai daripada mendapatkan kelebihan luas daun yang besar.



Gambar 6.6.8. Diagram kavitasi (Burrill).



Gambar 6.6.9. Diagram kavitasi untuk seri model baling-baling berdaun empat untuk kapal niaga.

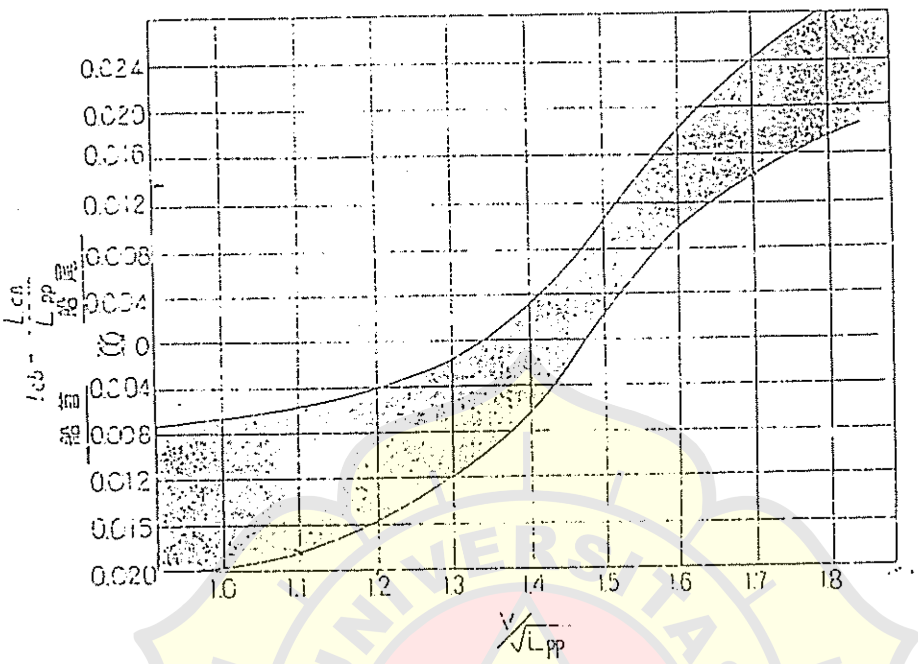
Untuk merancang baling-baling dengan memakai teori sirkulasi (lihat Bab 6, 6.7.5) perlu lebih dulu memilih garis tengah baling-baling, umumnya ditentukan dari diagram rancang (misalnya, Gb. 6.3.14). Karena itu untuk menghindari kavitasi diperlukan suatu kriteria yang agak umum dalam pemilihan luas daun. Diagram di Gb. 6.6.8 dapat dipakai sebagai pedoman demikian itu. Jika bentuk penampang daun telah diketahui maka distribusi tekanan di sekeliling penampang tersebut akan dapat dihitung (lihat Bab 2, 2.4 dan 2.6), atau mengukurnya di terowongan angin atau di terowongan air. Dengan memakai teori sirkulasi maka sudut insiden (angle of incidence) yang sebenarnya berikut pengurangan yang maksimum untuk tekanan pada punggung penampang dapat dicari. Tekanan yang dihitung tersebut kemudian dapat dibandingkan dengan tekanan statis $p_0 - p_v$ yang ada. Sudut insiden yang sebenarnya tergantung pada pola arus ikut di tempat bekerjanya baling-baling dan dalam satu kisaran baling-baling sudut tersebut akan berubah-ubah. Perhitungan tersebut harus dilakukan dengan memakai harga arus ikut mengeliling rata-rata pada setiap jari-jari tertentu. Dengan demikian maka kavitasi akan terjadi pada kisaran yang agak lebih rendah daripada yang dihitung, sehingga harus diberikan kelonggaran untuk itu. Sering bahwa setelah perhitungan selesai dilakukan kemudian dibuat model baling-balingnya dan dilakukan pengujian di terowongan kavitasi untuk memastikan tidak terjadinya pengaruh kavitasi yang merusak.

6.7. TEORI PERANCANGAN BALING-BALING

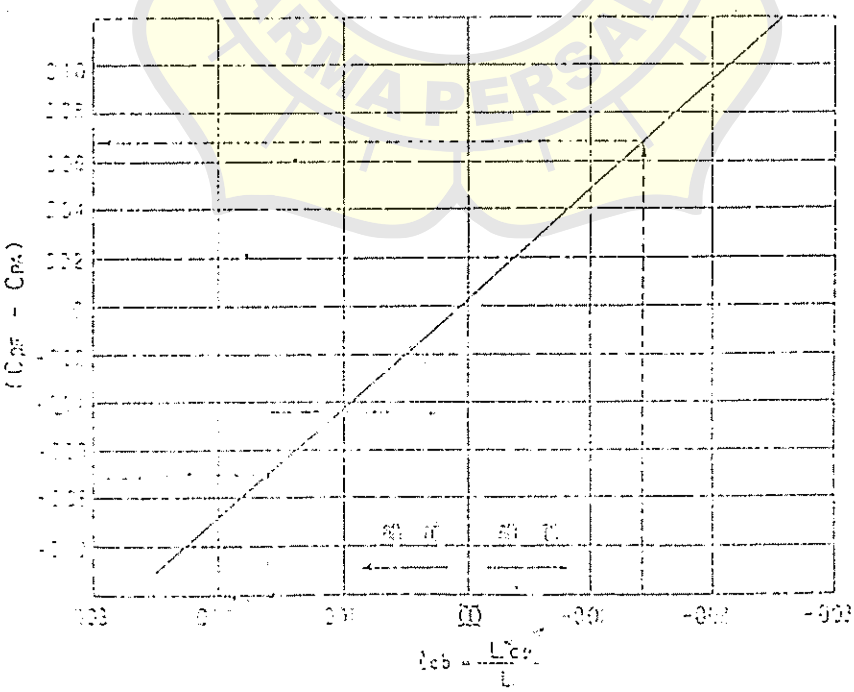
6.7.1. Pendahuluan

Telah banyak teori yang diajukan untuk menjelaskan cara sebuah baling-baling menghasilkan gaya dorong. Semua teori tersebut dikembangkan melalui pekerjaan yang sangat banyak, baik secara teoritis maupun memakai percobaan, yang dilakukan dalam cabang ilmu aerodinamika. Sekalipun demikian belum ada teori yang diajukan yang memperhitungkan semua faktor yang terlibat dalam aksi baling-baling. Selain itu, sekalipun konsep dari sebagian besar teori tersebut cukup sederhana matematiknya cukup rumit sehingga harus dipakai sejumlah anggapan tertentu untuk menyederhanakan masalahnya. Teori tersebut dapat diterapkan dalam praktek dengan memakai komputer, tetapi pemakaian teori yang akan diberikan berikut ini dan program komputer begitu saja tanpa memahaminya kadang-kadang dapat membuat malu yang besar. Karena itu, perancangan praktis baling-baling kapal yang cocok untuk kondisi yang diberikan masih sering tergantung pada hasil percobaan yang dilakukan secara sistematik dengan memakai model baling-baling; pemakaian pertimbangan yang baik merupakan hal yang hakiki. Di lain pihak, pengetahuan teoritis mengenai cara kerja baling-baling merupakan hal yang penting bagi pihak arsitek kapal untuk dapat menghasilkan rancang bangun baling-baling yang terbaik

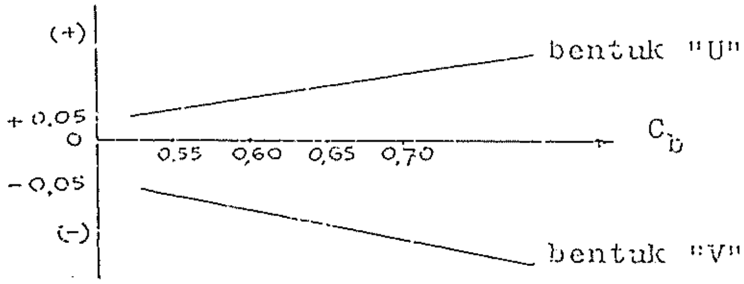
Lampiran 3. Diagram untuk menentukan letak LCB



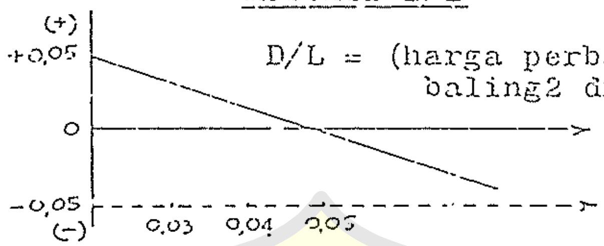
Lampiran 4. Diagram untuk menetukan koefisien depan dan belakang (Cpf - Cpa)



Koreksi bentuk badan kapal.

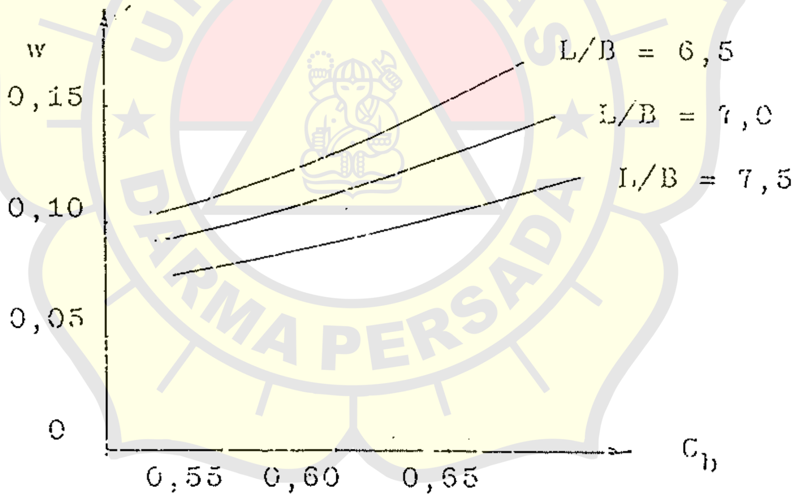


Koreksi D/L



D/L = (harga perbandingan diameter baling2 dng.panjang kapal).

Untuk kapal-kapal twin screw;



Harvald disajikan untuk kapal-kapal yang besar dimana coefficient penampang tengah kapal $C_{\not{g}}$ harganya mendekati 1,0.

Lebar kapal B juga berpengaruh terhadap aliran potential yang menyelubungi badan kapal. Oleh karenanya harga perbandingan L/B merupakan salah satu parameter lain dalam pembuatan diagram aliran wake.

Untuk kapal-kapal samodera dimana harga perbandingan B/d disekitar 2,5 dimana diagram Harvald di tujukan untuk kapal-kapal tersebut pengaruh B/d kepada aliran wake tidak besar. Maka parameter B/d diabaikan.

Bentuk dari penampang2 melintang badan kapal dibagian belakang merupakan salah satu faktor yang tidak dapat diabaikan dalam pengaruhnya terhadap aliran wake. Suatu perbandingan antara aliran dua dan tiga dimensi menunjukkan bahwa aliran wake akan makin sedikit bilamana aliran tersebut condong kearah aliran tiga dimensi. Dengan kesimpulan itu maka dapatlah dimengerti bahwasanya aliran wake akan lebih besar pada kapal-kapal yang mempunyai penampang bentuk "U" bilamana dibanding terhadap kapal yang mempunyai penampang bentuk "V".

Diameter baling2 D juga mempunyai pengaruh besar kepada harga wake fraction. Makin besar diameter baling2 maka akan makin besar pula bagian baling2 yang bekerja diluar "boundary layer" kapal (baling2 kapal single screw). Akibatnya adalah besarnya harga rata-rata aliran wake dipiringan baling2 (propeller disc) akan lebih kecil.

Panjang kapal L menentukan tebal dari boundary layer kapal. Jadi seberapa bagian dari diameter baling2 D dimana baling2 bekerja didaerah boundary layer akan tergantung dari panjang kapal L yang menentukan tebalnya boundary layer yang terseret kapal.

Dengan demikian harga perbandingan D/L merupakan parameter lain yang juga harus diperhitungkan. Rake dari baling2 dan juga celah antara daun baling2 dengan stern frame merupakan faktor-faktor yang mempunyai pengaruh terhadap aliran wake.

Diagram Harvald untuk memcari harga w adalah sama halnya dengan w dari rumus Taylor yaitu merupakanharga wake rata-rata. Untuk kapal-kapal single se screw, harga-harga w diplotkan terhadap harga C_b untuk beberapa harga L/B yang mana kemudian harus di adakan koreksi untuk bentuk penampang apakah bentuk "U" ataukah "V" dan koreksi untuk harga perbandingan D/L. Bagi kapal-kapal twin screw karena lokasi baling baling berlainan dengan bilamana baling2 berada ditengah kapal, maka diagram Harvald untuk harga w kapal2 twin screw tersebut tidak memakai koreksi2 baik untuk bentuk penampang badan kapal maupun harga perbandingan D/L. Menurut hasil yang didapat, ternyata harga-harga w yang dihitung dengan rumus Taylor perbedaannya tidak seberapa terhadap harga w dari diagram Harvald terutama untuk kapal-kapal samodera.

Dengan begitu untuk keperluan praktis dalam perencanaan dapatlah dipergunakan rumus Taylor untuk menghitung besarnya harga w.

Harga thrust deduction factor t tidak dapat dibuat diagram seperti halnya harga w. Hal ini disebabkan harga t sangat terpengaruh sekali besarnya terhadap :

- Ukuran-ukuran stern frame.
- Bentuk kelangsingan (fineness) dari garis air (waterlines) badan kapal.
- Harga perbandingan tebal dan panjang serta bentuk dari daun kemudi, dll.

Untuk keperluan praktis dapatlah dipakai rumus Taylor seperti dimuka yaitu;

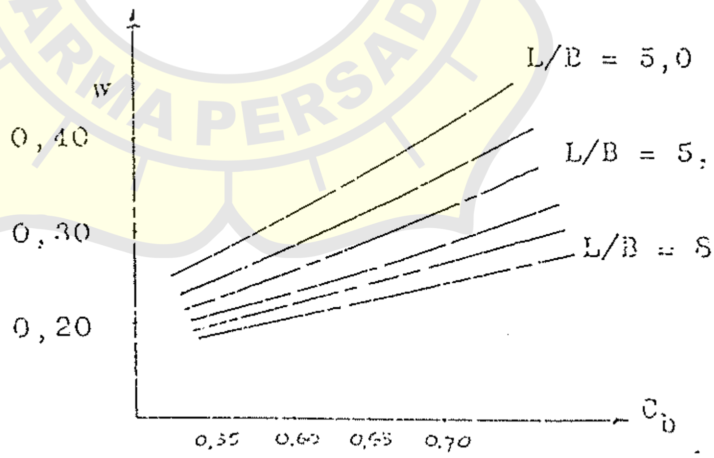
$t = 0,6\ w$ untuk kapal2 single screw.

$t \approx w$ untuk kapal-kapal twin screw.

Setelah didapatkannya harga w maka kalau kecepatan kapal diketahui Vs dapatlah harga kecepatan air yang melewati piringan baling-baling (intake velocity = advance speed) dihitung yaitu :

$$Va = Vs\ (1 - w)$$

Sketsa diagram Harvald untuk mencari w :



Rumus yang sederhana dan praktis untuk perencanaan baling-baling adalah :

Rumus TAYLOR

Untuk Wake fraction : Kapal berbaling2 tunggal;

$$w = -0,05 + 0,5\ C_b$$

Kapal berbaling2 ganda;

$$w = -0,20 + 0,55\ C_b$$

Untuk Thrust deduction factor :

Kapal barbaling2 tunggal: $t \approx w$

Kapal berbaling2 ganda; t w

dimana harga k adalah sebagai berikut :

Streamline rudder	$k = 0,55 - 0,70$
Rudder tipis	$k = 0,50$
Rudder tebal	$k = 0,70$

Untuk menghitung harga wake yang lebih teliti adalah memakai diagram yang dibuat oleh Harvald. Untuk dapat membuat diagram tersebut Harvald telah menggunakan 200 model kapal untuk percobaan2-nya di tangki percobaan dinegeri Belanda. Adapun parameter yang ia pilih untuk menentukan besarnya aliran wake adalah :

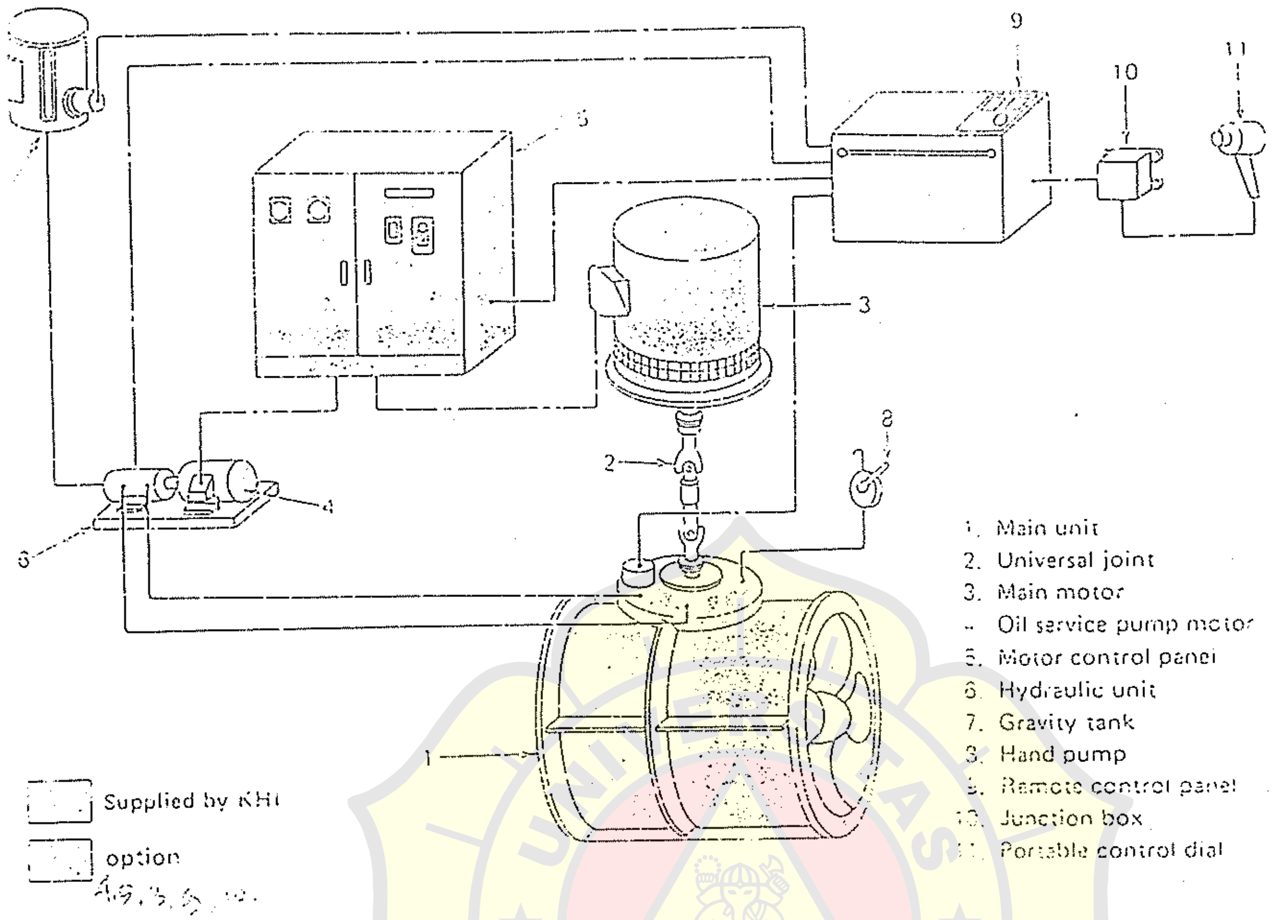
- Block coefficient C_b.
- Bentuk dari penampang-penampang melintang kapal bagian belakang.
- Diameter baling-baling D.
- Panjang kapal L dan harga perbandingan D/L.
- Rake dari daun baling-baling dan celah antara baling-baling dengan stern frame.

Block coeff. C_b mempunyai pengaruh kepada aliran wake. Percobaan Harvald memakai model kapal yg. mempunyai beban indentik tetapi diadakan beberapa perubahan-perubahan pada bagian muka badan kapal.

Ternyata bagian muka badan kapal juga mempunyai pengaruh terhadap besarnya aliran wake. Dari percobaan ini dapatlah diketahui bahwa harga w tidak hanya dipengaruhi oleh C_b badan kapal bagian belakang, tetapi oleh C_b dari keseluruhan badan kapal.

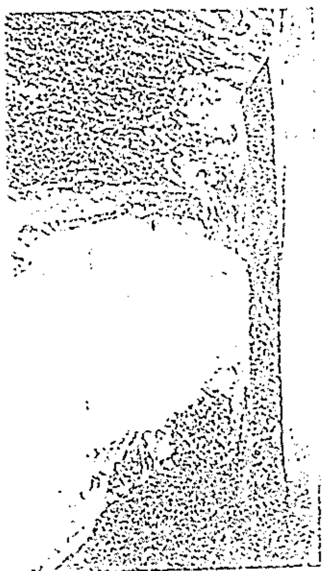
Adapun coefficient prismatic C_p tidak dipakai sebagai salah satu parameter berhubung percobaan

:NERAL ARRANGEMENT/一般配置

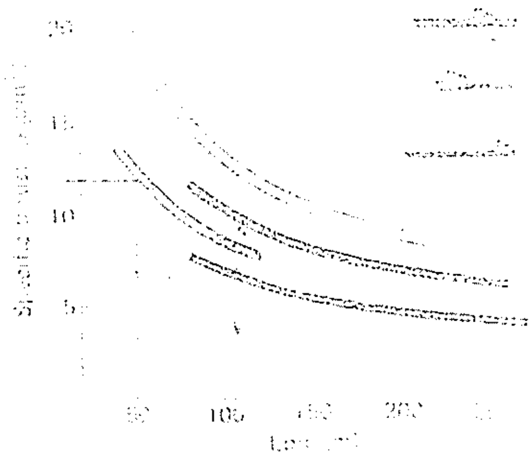


LECTING THE POWER/力量選定

following figure refers to the side thruster power and
d on our records. Specific thrust means thrust per unit
submerged hull's longitudinal projected area.
some vessels requiring fast turning or receiving large
orce, Specific thrust is to be increased as necessary.



Bollard test at sea



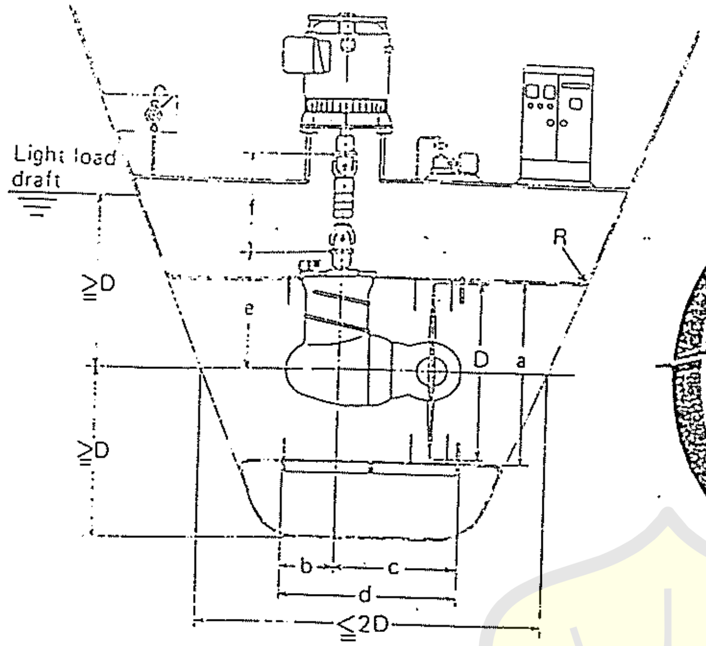
Ferries, Passenger ships

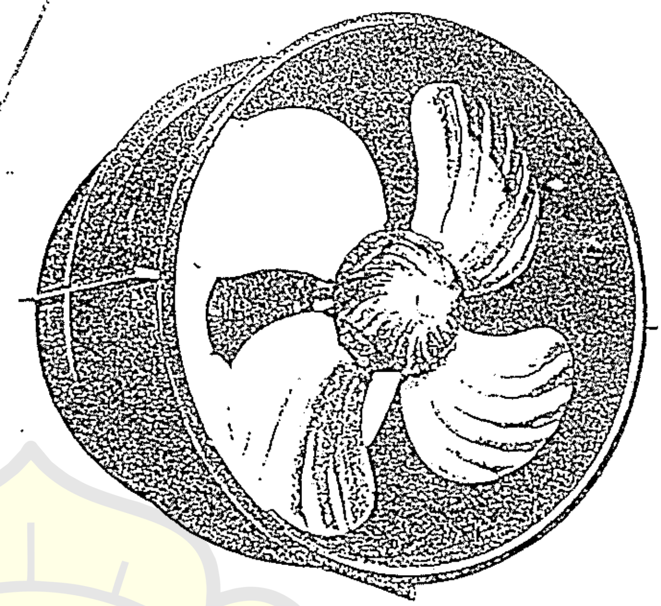
Cargo liners, container ships

Research ships, Tugs,
Fishing, ... vessels, etc.

Tankers, bulk carriers

DATA FOR THRUSTER/主要諸元





D: Propeller Dia.

DIMENSIONS/標準寸法

(mm)

Model 型式	KT-43B	KT-55B	KT-88B	KT-105B	KT-130B	KT-157B	KT-187B	KT-219B	KT-255B
D	1150	1300	1650	1800	2000	2200	2400	2600	2800
a	1219	1376	1749	1903	21.14	2324	2535	2744	2956
b	375	390	475	515	585	620	690	750	805
c	790	860	1075	1170	1295	1420	1560	1670	1795
d	1165	1250	1550	1685	1880	2040	2250	2420	2600
e	795	900	1130	1250	1375	1520	1620	1795	1935
f	690	670	670	870	870	880	880	1050	1190
R min.	60	65	85	90	100	110	120	130	140

WEIGHTS/標準重量

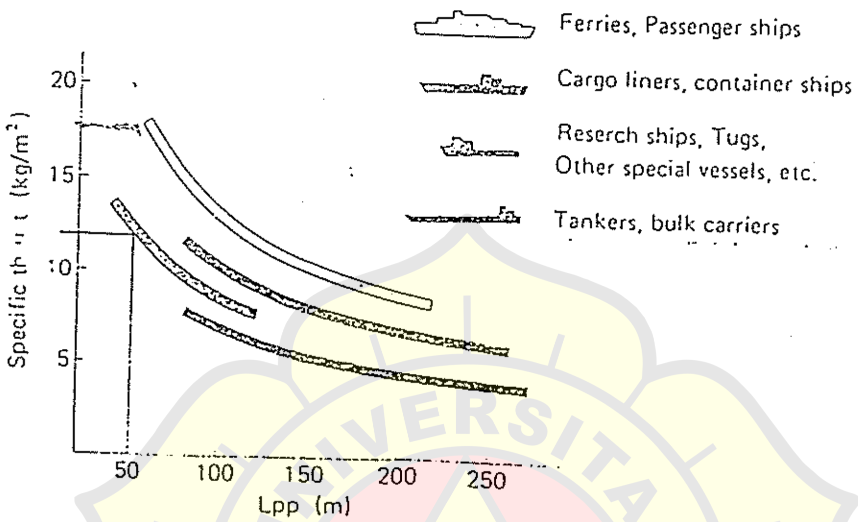
(kg)

Model 型式		KT-43B	KT-55B	KT-88B	KT-105B	KT-130B	KT-157B	KT-187B	KT-219B	KT-255B
ain unit	本体	1600	2300	4700	6100	8400	11200	14500	18500	23100
draulic unit	油圧ユニット	230	250	280	280	280	290	320	320	340
hers	その他重量	120	130	130	180	190	240	250	330	450

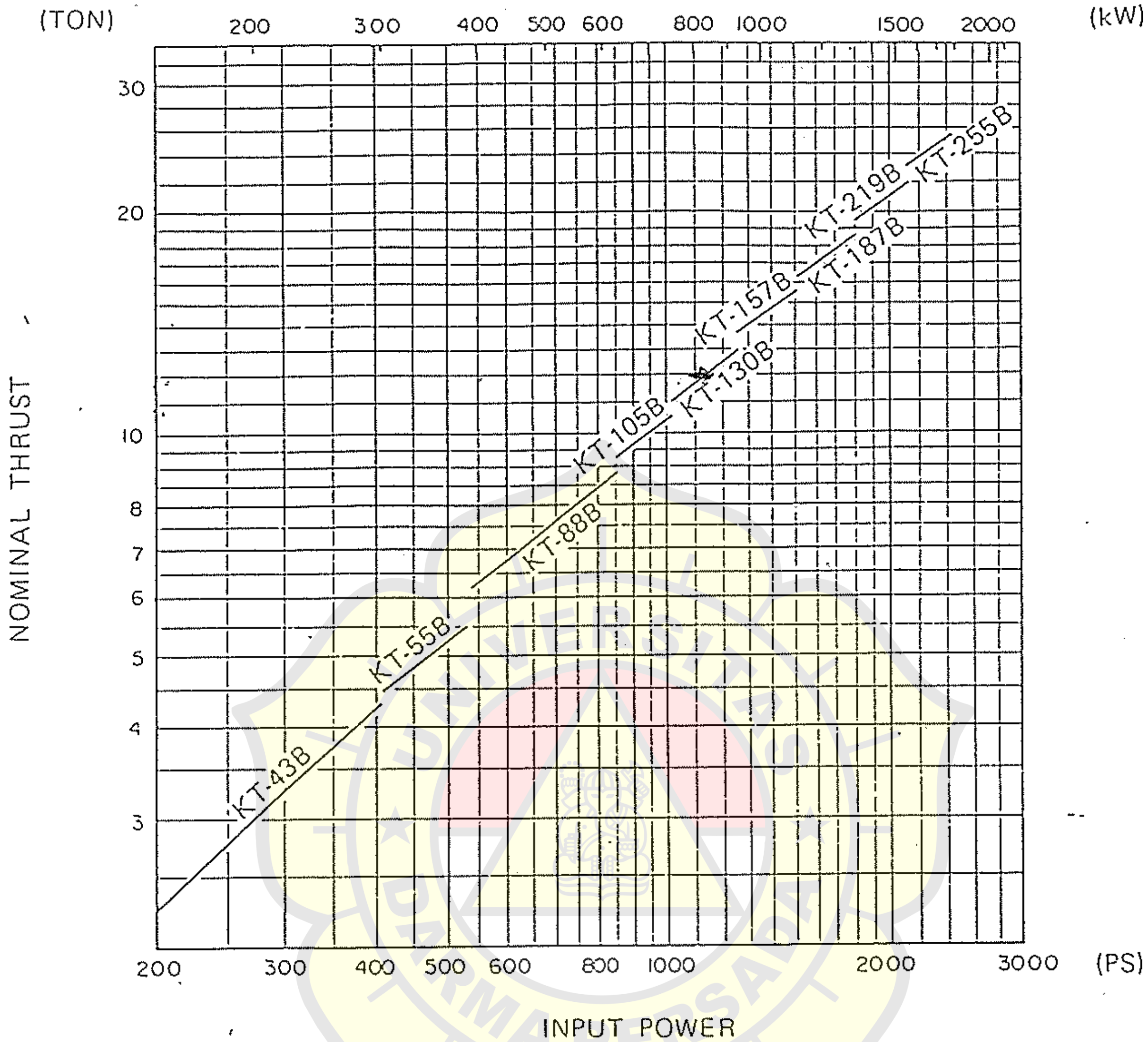
HYDRAULIC SYSTEM/補機要目

Model 型式			KT-43B	KT-55B	KT-88B	KT-105B	KT-130B	KT-157B	KT-187B	KT-219B	KT-255B
np capacity	ポンプ吐出量	ℓ/min	15	19	32	32	40	52	64	83	83
np pressure	ポンプ吐出圧力	kg/cm²	40	40	40	40	40	40	40	40	40
np R.P.M.	ポンプ回転数	rpm	1750	1750	1750	1750	1750	1750	1750	1750	1750
or power	ポンプモータ出力	kW	2.2	3.7	5.5	5.5	5.5	7.5	11	11	11
vity tank	重力タンク容量	ℓ	50	50	50	50	80	80	110	110	200
al oil	総油量	ℓ	140	180	320	400	560	710	930	1150	1500

Lampiran 17. Grafik Untuk Menentukan Bow Thruster



Lampiran 18. **Grafik Untuk Menentukan Tipe Bow Thruster**



Model	Propeller Dia. mmφ	Max. Input Power PS	Max. Input Power kW	Max. Thrust ton	Input RPM 60 Hz	Input RPM 50 Hz	Propeller rpm
KT- 43B	1150	410	300	4.3	1750	1450	509
KT- 55B	1300	530	390	5.5	1750	1450	450
KT- 88B	1650	850	625	8.8	1160	1450	355
KT-105B	1800	1010	745	10.5	1160	980	326
KT-130B	2000	1250	920	13.0	1160	980	293
KT-157B	2200	1510	1110	15.7	880	980	266
KT-187B	2400	1800	1325	18.7	880	980	244
KT-219B	2600	2110	1550	21.9	880	735	225
KT-255B	2800	2450	1800	25.5	880	735	209

The pulling force, T_b, on the winch barrel (Figs. 174 and 175) is

$$T_b = \frac{P+Q}{\eta_p^k} = P_g \qquad (412)$$

where P = weight of the useful load being hoisted, kg
Q = (0.0028 to 0.0022) P = weight of the cargo hook with the shackle, kg
η_p = 0.9 to 0.96 = efficiency of one pulley
k = number of intermediate pulleys between the boom iron and the winch barrel.

A two-speed winch is designed for a given pulling force, P_g, in double-gear operation. The gearing ratio for single-gear operation usually ranges from 4 to 6 and for double-gear operation, from 6 to 12

The diameter of the winch barrel is taken to be

$$D_b = (16.5 \text{ to } 18)\,d_r \qquad (413)$$

where the rope diameter, d_r, is selected according to its breaking strength, $R_{br} = 6P_g$.

The length of the winch barrel is

$$L_b = (1.1 \text{ to } 1.5)\,D_b \quad \text{cm}$$

The number of layers, z, of rope wound on the barrel depends upon the size of the latter and the length of the rope to be wound. This length, L_r, ranges from 40 to 75 m and the number of layers does not exceed five.

The diameter, d_r, and kind of rope is selected so that the actual tensile strength

$$R_{br} \geq k_m P_g \quad \text{kg} \qquad (414)$$

where $k_m \geq 6$ = margin of safety.

The number of turns that can be accommodated along the length of the barrel is

$$m = \frac{L_b}{d_r} \qquad (415)$$

The length of rope accommodated is:
in the first layer

$$l_{m1} = \pi\,(D_b + d_r)\,m$$

in the second layer

$$l_{m2} = \pi\,[D_b + (4d_r - d_r)]\,m = \pi\,(D_b + 3d_r)\,m$$

in the third layer

$$l_{m3} = \pi\,[D_b + (6d_r - d_r)]\,m = \pi\,(D_b + 5d_r)\,m$$

In the latter case, calculations are usually conducted using the design diameter of the barrel which is

$$D_{bd} = D_b + d_r(2z-1) \quad \text{m} \tag{420}$$

The torque developed on the barrel shaft is

$$M_{bd} = \frac{1}{2}[D_b + d_r(2z-1)]\frac{T_b}{\eta_b} \quad \text{kg-m} \tag{421}$$

where η_b = efficiency of the winch barrel.

The rotational speed, n_{bd}, of the barrel is found from the following equation for a load hoisting speed v_{ld} with the double gearing of the winch engaged:

$$n_{bd} = \frac{60v_{ld}}{\pi D_{bd}} = 19.1\frac{v_{ld}}{D_{bd}} \quad \text{rpm} \tag{422}$$

The overall gearing ratio of the winch with the double gearing engaged is

$$i_{wd} = \frac{n_m}{n_{bd}} = \frac{n_m}{\frac{60v_{ld}}{\pi D_{bd}}} = \frac{n_m \pi D_{bd}}{60v_{ld}} \tag{423}$$

where n_m = 80 to 250 = speed of the winch steam engine shaft, rpm
n_m = 500 to 3,000 = shaft speed of the electric motor, rpm.

The overall efficiency, η_{wd}, of the winch when the double gearing is engaged is the product of the efficiencies of the shafts (η_{sh}), pairs of spur gears (η_{sg}), barrel (η_b) and worm gearing (η_{wg}). Thus

$$\eta_{wd} = \eta_{sh}^{a}\eta_{sg}^{c}\eta_b\eta_{wg} \tag{424}$$

where a and c = number of shafts and pairs of gears, respectively
η_{wd} = 0.7 to 0.85 for winches with spur gearing
η_{wd} = 0.65 to 0.75 for winches with worm gearing.

The required shaft torque of the motive unit is

$$M_{md} = \frac{M_{bd}}{i_{wd}\eta_{wd}} \quad \text{kg-m} \tag{425}$$

The diameter of the steam engine cylinder and the required power to start from rest are determined from Posdyunin's formula:*

$$D_{cw} = 1.37\sqrt[3]{\frac{M_{md}}{\Psi_r\eta_m(\alpha_i k_i p_{ls} - p_{ss})}} \quad \text{cm} \tag{426}$$

* The symbols denote the same values as in the case of steering engines.

where $k_i = \frac{1 + \ln\Delta}{\Delta}$ = coefficient of mean theoretical indicated pressure for a ratio of steam expansion Δ

M_{ms} = torque developed on the engine shaft, kg-cm.

The indicated power of the engine required to start from rest under load is

$$N_i = \frac{D_{mp}^2(\sigma_i k_i p_{is} - p_{es})\, n_m \psi_e}{143{,}300} \quad \text{hp} \tag{427}$$

Values of k_i as a function of the admission ratio (reciprocal of the expansion ratio) $\delta = \frac{1}{\Delta}$ are listed in Table 62.

Table 62

δ	0.5	0.6	0.7	0.8	0.9	1
k_i	0.848	0.907	0.95	0.979	0.995	1

If T_{bs} is the given rated pulling force for the single gearing engagement of the winch, calculated from equation (412) for the given load hoisting capacity, then, according to equation (421), the torque developed on the winch barrel is

$$M_{bs} = \frac{1}{2}\left[D_b + d_r(2z-1)\right] T_{bs} \quad \text{kg-m} \tag{428}$$

Assuming that the motive unit shaft rotates at a constant speed n_m we can write

$$\frac{M_{bd}}{i_{wd}\eta_{wd}} = \frac{M_{bs}}{i_{ws}\eta_{ws}} \tag{429}$$

where η_{ws} = overall efficiency of the winch when the single gearing is engaged

i_{ws} = gearing ratio of the winch with the single gearing engaged.

It follows that the required gearing ratio is

$$i_{ws} = \frac{M_{bs}}{M_{bd}} \frac{\eta_{wd}}{\eta_{ws}} i_{wd} \tag{430}$$

The speed of the winch barrel for single gearing is

$$n_{bs} = \frac{n_m}{i_{ws}} \quad \text{rpm} \tag{431}$$

The suction lift, or simply lift, is the loss of head required to overcome resistance in the suction line of the pumping plant; it is measured in mH_2O.

The useful power of a pump is the energy increment in the flow of liquid passing through the pump in unit time and is expressed in horsepower or kilowatts. Thus,

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

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

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

The mechanical efficiency, η_m, of a pump determines the loss in energy in its operation and enables the required power input N to be calculated:

$$N = \frac{N_u}{\eta_m}\ \text{hp (kW)}.$$

1-3. Pump Classification According to Purpose and Principle of Operation

(A) PUMP CLASSIFICATION ACCORDING TO PURPOSE

In accordance with their purpose, shipboard pumps can be divided into three groups:

1. General service pumps whose function is to ensure the seaworthiness of the ship and to provide for the domestic needs of the crew and passengers, and also to maintain the necessary sanitary conditions on board.

2. Pumps of the shipboard systems, designed to serve the main and auxiliary systems, and to facilitate the maintenance of normal conditions for their operation.

3. Special-purpose pumps in tankers, trawlers, ice-breakers, life-saving ships and dredgers.

General service pumps include:

(1) bilge pumps,
(2) sanitary pumps,
(3) fire pumps,
(4) emergency pumps.

Bilge Pumps

Bilge pumps include ballast and drainage pumps.

Ballast Pumps. The purpose of these pumps is to take in liquid ballast from overboard, to fill the ballast tanks and peaks, to empty them of ballast and to transfer the ballast from certain tanks and peaks to others.

The capacity of ballast pumps must be sufficient to enable the tanks they serve to be emptied within 4 to 10 hours, depending upon the size of the ship.

Tanks that hold from 20 to 265 cu m of ballast are usually pumped dry within 1 to 3.5 hours; those holding from 360 to 1300 cu m are emptied within 4 to 5.5 hours.

The required capacity of a ballast pump can be determined from the formula

$$Q_b = 0.2825 d_p^2 v_b \text{ cu m per hr} \tag{11}$$

where d_p = diameter, in cm, of the inlet pipe of the largest ballast tank. According to the regulations of the U.S.S.R. Shipping Register this value is to be taken from Table 3, depending upon the tank capacity in tons

v_b = 2 to 2.5 = velocity of water flow in the input line of the pump, m per sec.

Table 3

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

At a water velocity $v_b = 2$ m per sec (recommended by the U.S.S.R. Shipping Register) the required ballast pump capacity will be

$$Q_b = 0.565 d_p^2 \text{ cu m per hr.} \tag{12}$$

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

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

Self-contained ballast pumps must be installed on oil tankers to serve the fore ballast tank.

Table 37

WIRE ROPES

Nominal strength	1570 N/mm²	1570 N/mm²	1570 N/mm²
Nominal dia of rope	accord. to DIN [illegible]  Round rope 6 × 7 Constr. 6(6 + 1) + FE	accord. to DIN 3060  Round rope Standard 6 × 19 Constr. 6(12 + 6 + 1) + FE	accord. to DIN 3063  Round rope Standard 6 × 24 Constr. 6(15 + 9 + FE) + FE
	Nominal breaking load¹)		
mm	kN	kN	kN
8	33,4	30,9	28,2
10	52,2	48,2	44,0
12	75,1	69,5	63,3
14	102	94,6	86,2
16	134	124	113
18	169	156	143
20	209	193	176
22	252	234	213
24	300	278	253
26	353	326	297
28	409	378	345
32	534	494	450
36	[illegible]	625	570
40	835	772	704
44	–	934	851
48	–	1110	1100
52	–	1300	1190
56	–	1510	1330
[illegible]	[illegible]	Shrouds, Stays, Guy pendants, Preventers, fastened by means of rope sockets to the hull, Ropes for single reeved span, not to be tapped under load	Cargo runners (hoisting ropes), Span wire ropes (luffing ropes), Guy pendant, Guy tackle, Preventer

¹) [illegible] for ropes with core of fibre material. The nominal breaking loads are [illegible] of same construction are used with core of steel wire rope.

Nominal strength	1570 N/mm²	1770 N/mm²	1570 N/mm²	1770 N/mm²
Nominal dia of rope	DIN 3066  Round rope Standard 6 × 37 Construction 6(18 + 12 + 6 + 1) + FE		DIN 3051  Round rope Warrington-Seale 6 × 36 Construction 6(14 + 7/7 + 7 + 1) + FE	
	Nominal breaking load¹)			
mm	kN	kN	kN	kN
8	29,6	33,4	–	–
10	46,3	52,2	51,3	58,4
12	66,6	75,1	74,5	84,1
14	90,7	102	102	114
16	113	134	133	149
18	150	169	168	189
20	185	209	207	234
22	224	253	251	283
24	267	301	298	336
26	313	353	350	395
28	363	409	406	458
32	474	534	530	598
36	600	676	671	757
40	741	835	829	934
44	896	1010	1006	1130
48	1070	1200	1190	1350
52	1250	1410	1400	1580
56	1450	1640	1620	1830
60	1670	1880	1860	2100
64	1900	2140	–	–
Approved for gears with a SWL up to 10 t; Approved for gears with a SWL up to 10 t	Nominal strength 1570 N/mm²: cargo runners (hoisting ropes), span ropes (luffing ropes), guy tackles, preventers, [illegible] are carried not permanently set. Nominal strength 1770 N/mm²: cargo runners (hoisting ropes), span ropes (luffing ropes), [illegible]			

Denomination of a rope made of round strands

according to nominal diameter of rope, DIN-standard, type of core, surface of wires, nominal strength of wires, kind and direction of impact

e.g.: rope 20 DIN 3066 – FE zn k 1570 sZ

According to DIN 3051, Sheet 1, Table 1

meaning of the following abbreviations is

FE = fibre core

zn k = wires drawn zinced

Note to Table 44.

Shackles

1. Shackles may be subjected only to tensile loads.

2. Wherever possible, shackles should be so connected that the bolt side is attached to a round eye and the strap side to an elongated eye or chain link.

3. Type C shackles are to be used for fastening cargo and span blocks, for attaching guy blocks, span runners and guy pendants to the head fitting and for beckets to the eyes of blocks.

4. Shackles for cargo hooks to Table 45, cargo chains and cargo hook swivels must have slotted bolts (type B).

5. Type A shackles may only be used for connecting the lower guy blocks and snatch blocks to the deck.

Table 44

SHACKLES

according to DIN 82101, Feb. 76

Member	Material
Bow	RSt 37-2, St 44-2 DIN 17100, C 22 DIN 17200 or C 15 DIN 17210
Pin	St 42 K DIN 1652, St 44-2 DIN 17100 or C 22 DIN 17200

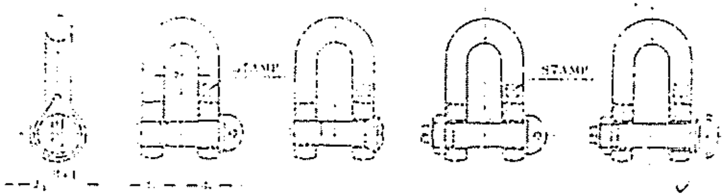
Nominal size	Working load limit "WLL"	b_1	d_1	d_2	Pin ∅	Pin Thread
	t	mm	mm	mm	mm	
1	1	21	13	32	16	M 16
1,6	1,5	27	17	40	20	M 20
2	2	30	19	44	22	M 22
2,5	2,5	33	21	48	24	M 24
3	3,2	38	24	54	27	M 27
4	4	42	27	60	30	M 30
5	5	47	30	72	36	M 36
6	6,3	53	34	78	39	M 39
8	8	60	38	90	45	M 45
10	10	66	42	96	48	M 48
12	12,5	73	47	104	52	M 52
16	16	81	52	120	60	M 60
20	20	90	58	136	68	M 68
25	25	100	63	144	72	M 72 × 6
32	32	110	70	160	80	M 80 × 6
40	40	125	79	180	90	M 90 × 6
50	50	140	88	200	100	M 100 × 6
63	63	155	96	220	110	M 110 × 6
80	80	175	110	250	125	M 125 × 6
100	100	200	125	280	140	M 140 × 6

Symbol

according to Form nominal size and No. of Table.

e.g.: Shackle A 16 - [11]

Nominal size	Form A	Form B	Form C	
	1 to 20	1 to 29	1 to 20	25 to 100



Temperature, °C	Density, kg/m³	Absolute humidity, g/m³	Vapour pressure, mmHg	Temperature, °C	Density, kg/m³	Absolute humidity, g/m³	Vapour pressure, mmHg
−25	1,424	0.61	0,540	+13	1,235	11.32	11.162
−24	1,418	0.71	0,600	+14	1,230	12.03	11.908
−22	1,406	0.86	0,745	+16	1,222	13.59	13.536
−21	1,401	0.95	0,825	+17	1,217	14.43	14.421
−20	1,395	1.05	0,910	+18	1,213	15.31	15.357
−19	1,390	1.15	1,000	+19	1,209	16.25	16.346
−18	1,384	1.25	1,095	+20	1,205	17.22	17.391
−17	1,379	1.35	1,190	+21	1,201	18.25	18.495
−16	1,374	1.46	1,290	+22	1,197	19.33	19.639
−15	1,368	1.58	1,400	+23	1,193	20.48	20.888
−14	1,363	1.70	1,520	+24	1,189	21.68	22.184
−13	1,358	1.83	1,635	+25	1,185	22.93	23.550
−12	1,353	1.98	1,780	+26	1,181	24.24	24.988
−11	1,347	2.14	1,930	+27	1,177	25.64	26.505
−10	1,342	2.31	2,093	+28	1,173	27.09	28.101
−9	1,337	2.49	2,267	+29	1,169	28.62	29.782
−8	1,332	2.69	2,455	+30	1,165	30.21	31.548
−7	1,327	2.90	2,658	+31	1,161	31.89	33.406
−6	1,322	3.13	2,876	+32	1,157	33.64	35.350
−5	1,317	3.37	3,113	+33	1,154	35.48	37.411
−4	1,312	3.64	3,368	+34	1,149	37.40	39.565
−3	1,308	3.92	3,644	+35	1,146	39.41	41.827
−2	1,303	4.22	3,941	+36	1,142	41.51	44.201
−1	1,298	4.55	4,263	+37	1,139	43.71	46.691
0	1,293	4.89	4,600	+38	1,135	46.00	49.302
+1	1,288	5.23	4.940	+39	1,131	48.40	52.039
+2	1,284	5.60	5,302	+40	1,128	50.91	54.906
+3	1,279	5.98	5,687	+41	1,124	53.52	57.910
+4	1,275	6.39	6,097	+42	1,121	56.25	61.055
+5	1,270	6.82	6,534	+43	1,117	59.09	64.346
+6	1,265	7.28	6.998	+44	1.114	62.05	67.790
+7	1,261	7.76	7,492	+45	1,110	65.14	71.391
+8	1,256	8.28	8,017	+46	1,107	68.36	75.158
+9	1,252	8.82	8,574	+47	1,103	71.73	79.093
+10	1,247	9.39	9,165	+48	1,100	75.22	83.204
+11	1,243	10.01	9,792	+49	1,096	78.86	88.499
+12	1,239	10.64	10,457	+50	1,093	82.63	91.982

Table 39

Data on fresh air and sea water

Locality	Warmest period of navigation: Temperature of outside air t_{fa}, °C	Warmest period: Water temperature t_w, °C	Warmest period: Relative humidity of outside air φ_{fa}, %	Coldest period of navigation: Temperature of outside air t_{fa}, °C	Coldest period: Water temperature t_w, °C	Coldest period: Relative humidity of outside air φ_{fa}, %
Rivers that freeze	20 to 30	16 to 25	55 to 65	−5	4	75 to 85
Seas in high or temperate latitudes	10 to 25	5 to 15	65 to 75	−25 to −35	0 to −2	80 to 85
Warm seas	25 to 30	20 to 25	55 to 65	−15 to −20	4	75 to 85
Tropical seas	30	27	70	20	27	70
Navigation in any localities	30	27	70	−25	0	80

Air requirements for shipboard accommodations

Accommodations	Air temperature: Coldest period of navigation, t_r, °C	Air temperature: Warmest period of navigation, t_r, °C	Relative humidity φ_r, %
Living and passenger accommodations, state-rooms and ward-rooms	18	$t_{fa}+5°C$	40 to 6
Passageways of living and service accommodations	15	$t_{fa}+5°C$	75 to 8
Bath- and shower-rooms	25	$t_{fa}+10°C$	70
Cloak-rooms	20	$t_{fa}+5°C$	70 to 8
Wash-rooms and laundries	15	$t_{fa}+5°C$	60
Toilets	12	$t_{fa}+5°C$	80
Galleys	8 to 22	$t_{fa}+10°C$	80
Pantries	8	$t_{fa}+5°C$	80
Wet provisions and vegetable storage rooms	2	$t_{fa}+5°C$	80

Table 42

Compartment	Number of air renewals per hour for	
	Plenum ventilation	Exhaust ventilation
Passengers', officers' and crew accommodations	10 to 15	—
Public rooms (staterooms, dining saloons, etc.)	15 to 20	10 to 15
Smoking rooms	—	15 to 20
Gymnasiums	15	20
Swimming pools	15	20
Russian baths	—	10 to 20
Galleys	5 to 10	40 to 60
Provision rooms without cooling facilities	5 to 10	10 to 15
Bathrooms, toilets and laundries	5	15 to 20
Sick bays	5 to 10	10 to 20
Baggage rooms	—	20
Deck refreshment bars	10 to 15	25 to 30
Upper deck passageways	—	6
Middle deck passageways	—	7
Lower deck passageways	—	8
Engine and boiler rooms	30	35

$p_{st}=760$ mmHg, relative humidity of $\varphi_{st}=50$ per cent and density $\gamma_{st}=1.2$ kg per cu m. The capacity of the fan determined for air in a given state, having a pressure p_a, volume Q_a and temperature t_a, can be converted to the standard air capacity by using formula (276) which is derived from the equation

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

whence

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

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

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

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

where γ_{air} = density of air, kg per cu m

γ_{wat} = 1,000 = density of water, kg per cu m

ρ = mass density of air, kg-sec² per m⁴.

Upon radial entry of the air onto the fan impeller vanes

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

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

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

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

$\psi_h = \sigma\varphi_h\eta_h$ = head factor taken equal to: 0.8 to 1.1 for forward-curved vanes; 0.6 to 0.8 for radial, or straight, vanes; 0.5 to 0.7 for backward-curved vanes.

The fan head required to accommodate a given ventilating system depends upon the resistance and characteristic curves of the latter.

The permissible maximum peripheral speeds (tip speeds) of an impeller, based upon fan design and strength considerations, are listed in Table 43. The table also lists the most widely used inlet and outlet angles of the vanes.

Table 43

Type of fan	Peripheral speed, m per sec	Inlet angle	Outlet angle
Low-pressure	30 to 40	95 to 105	15 to 25
Medium-pressure	40 to 50	125 to 130	30 to 35
High-pressure	50 to 90	140 to 145	40 to 45

Backward curved vanes are rarely employed and then only for low-pressure fans. The number of vanes is usually assigned so as to facilitate laying out and may be equal to 4, 6, 8, 12, 16, 24, 32 or 48.

(1) fans of service and living compartments, designed to provide induced ventilation in these spaces;

(2) cargo hold fans, designed for ventilating the holds of dry-store bulk carriers, tankers and refrigerated cargo vessels, as well as refrigerated provision chambers;

(3) boiler plant fans, designed to produce artificial draught for the steam boilers;

(4) coal bunker fans.

Depending upon the way they are installed fans are classified as:

(1) supply fans in which the fan discharge is connected with the spaces being served;

(2) exhaust fans in which the fan inlet is connected to the spaces being served;

(3) ceiling fans, designed to produce air movement in the spaces without providing exchange.

As regards the pressure they develop, fans are divided into:

(1) low-pressure fans developing a head up to 100 mmH_2O;

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

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

According to the mechanical composition of the gas they handle, there are:

(1) fans for delivering pure gases;

(2) dust fans designed for delivering gases polluted by mechanical impurities.

The specific velocity, n_s, of a fan is a value relating the air discharge, Q cu m per hour, full head, H mmH_2O, at normal atmospheric conditions and the fan wheel speed, n rpm, at the highest efficiency:

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

Gas passing through the fan is compressed to only the slightest degree and is therefore assumed to be an incompressible fluid. In this case all the main principles in the theory and operation of centrifugal pumps are valid for fans as well.

The capacity of a fan for ventilating definite service quarters must be sufficient to maintain the chemical composition, humidity and temperature of the air within the requirements of sanitary regulations.

The unwholesomeness, or contamination, of the air in a room or compartment due to the presence of people is usually estimated by the carbon dioxide content, which increases with an increase of harmful impurities in the air. The carbon dioxide content of the air must not exceed 0.1 to 0.15 per cent by volume.

The fan capacity required to maintain a stipulated chemical composition of the air in a compartment is

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

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

V_r = volume of the room, cu m

$V_{mr} \approx 1$ = the maximum carbon dioxide content per cu m of the given room, litres per cu m

$V_{ca} \approx 0.3$ = carbon dioxide content per cu m of sea air entering the room, litres per cu m.

The volume of air required to maintain the prescribed temperature in a room is

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

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

t_r = given temperature of the room, °C

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

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

γ_{fa} = density of the fresh air entering the room, kg per cu m

$\gamma_0 \approx 1.29$ = density of dry air at 0°C and a pressure of 760 mmHg, kg per cu m

$\alpha = \frac{1}{273}$ = coefficient of volumetric expansion of air.

The amount of external air required to maintain the relative humidity in a room is

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

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

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

φ_r and φ_{fa} = relative humidity of the air in the room and of the entering air, per cent.

Data on the relative humidity and temperature of the outside air depending upon the locality in which the ship is operating, and the permissible values for various accommodations are listed in Table 39.

The amount of carbon dioxide, heat and vapour produced by persons in a room can be calculated from the data of Table 40.

Each adult produces per hour	Carbon dioxide, litres/h	Heat, kcal/h	Vapour, g/h
At hard physical work	45	150	130
At quiet work	23	100	75
At rest	23	75	70
At sleep	23	75	40
Each child up to 12 years of age produces	12	50	23

It should be noted that the amount of vapour produced in a room by the operation of steam engines and steam lines is approximately equal to 1 or 2 per cent of the steam consumption of the engines or lines.

The heat generated by various sources and introduced into the premises they occupy can be determined from the formulas listed in Table 41.

In calculating the fan capacity required for engine and boiler rooms it is necessary to take into consideration the amount of air required for the operation of internal combustion engines and boilers.

The approximate volumes of air required for the operation of internal combustion engines, V_{eic}, and boilers, V_b, are found from the following formulas:

$$V_{eic} = 60\,\alpha_{ex} V_{cyl} n \text{ cu m per hour}$$

where V_{cyl} = total displacement of the cylinders, cu m
n = engine shaft speed, rpm
α_{ex} = 1.3 to 1.5 = excess air coefficient.

$$V_b = 1.15\,\alpha_b (1 + \alpha t_{fa})\, B \frac{Q_l}{1{,}000} \text{ cu m per hour}$$

where $\alpha_b \approx 1.2$ to 1.5 = excess air coefficient
α = coefficient of volumetric expansion of air
B = fuel consumption, kg per hour
Q_l = lower calorific value of the fuel, kcal per kg.

The required fan capacities calculated from formulas (273), (274) and (275) will not be the same and therefore the highest value should be taken for any given compartment.

Tentative values of the required capacity can be estimated on the basis of the number of air renewals per hour n_{rc} as established by experience for various accommodations (Table 42).

Source of heat	Heat emitted into surroundings, Q_l, kcal/h	Notation
Steam boilers	(0.03 to 0.05) $G_f Q_l$	Q_f = total fuel consumption in boiler, kg per hour
Steam turbines	0.005 $G_s \Delta i$	
Steam engines	(0.005 to 0.01) $G_s \Delta i$	Q_l = lower calorific value of the fuel, kcal per kg
Auxiliary machinery	(0.02 to 0.03) $G_s \Delta i$	
Steam lines	0.01 $G_s \Delta i$	G_s = steam consumption, kg per hour
Internal combustion engines	0.02 $N_e g_e Q_l$	Δi = useful heat drop, kcal per kg
Electrical machinery: (a) with recirculating cooling system	$64\,N \frac{1-\eta}{\eta}$	N_e = effective power; N = power, kW
(b) without cooling	$864\,N \frac{1-\eta}{\eta}$	i_e = mean current density per sq mm of conductor cross section, A per sq mm
Lighting fixtures	864 N	
Wires, bus bars, cables and fittings	2,160 i_e^2	η = efficiency of electrical machine
Heat introduced from outside by transmission through ship's hull	$\Sigma kF\Delta t$	g_e = fuel consumption, kg per hp-h; k = coefficient of heat transmission of room walls, kcal per hour per sq m per deg C; F = area of the room walls, sq m; Δt = difference in temperatures of room wall surfaces and external surfaces of ship's hull, deg C

In this case, if V_{com} is the volume of the compartment in cu m, the required hourly capacity of the fan will be

$$Q_a = n_{rc} V_{com} \text{ cu m per hour}$$

The fan capacity needed is selected on the basis of what is called standard air. This means air at a temperature $t_{st} = 20°C$, pressure

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

$$N_a = \frac{Q_a H}{75\eta_f 3{,}600} \text{ hp}$$

The overall efficiency of a fan is made up of the following efficiencies:

1. Hydraulic efficiency, which takes into consideration the loss of head in the fan

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

where ΔH = loss of head in the fan.

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

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

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

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

b_2 = width of the impeller at air outlet

D_2 = impeller diameter at air outlet

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

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

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

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

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

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

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

2-2. Design and Selection of Fans

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

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

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

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

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

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

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

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

$$n_s = \frac{n\sqrt{Q_{st}}}{\sqrt[4]{H^3}} \quad (281)$$

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

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

and

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

where F = area of the impeller, sq m

D_2 = outside diameter of the impeller, m.

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

Table 47

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

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

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

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

where n_m = 100 to 350 rpm for steam engines

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

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

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

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

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

It follows from formula (314) that

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

Combining equations (315) and (316) we obtain

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

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

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

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

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

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

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

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

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

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

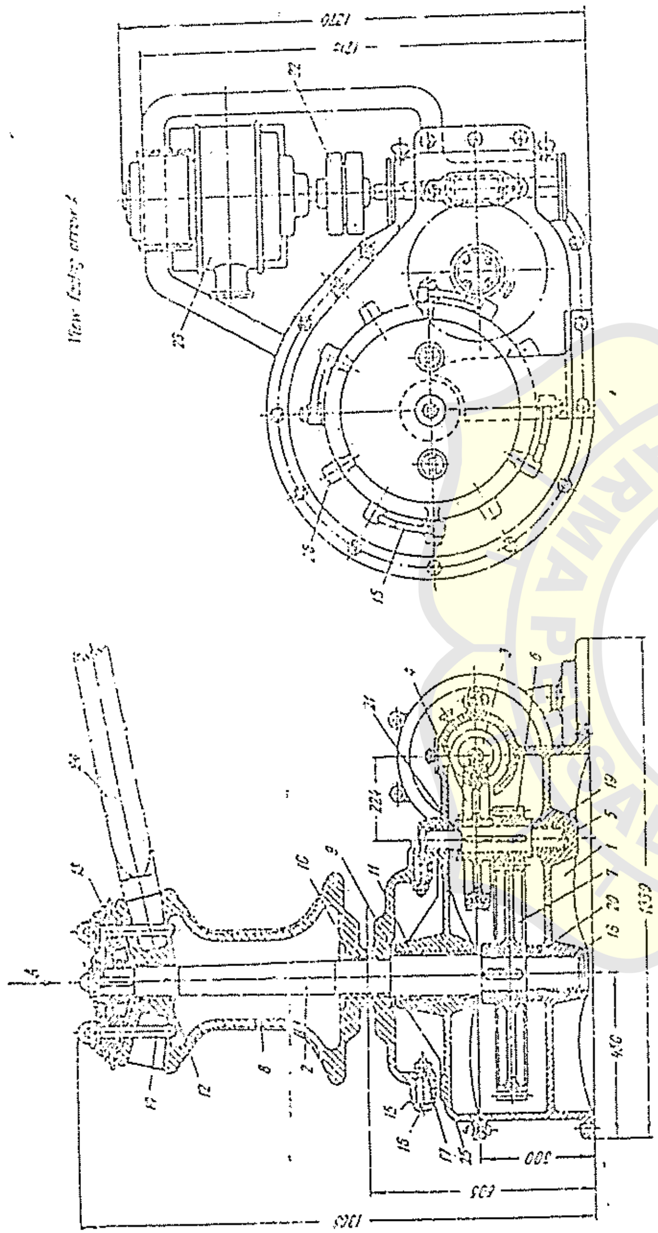


Fig. 169.

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

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

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

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

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

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

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

G_a = weight of the anchor, kg

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

L_a = length of the suspended cable, m

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

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

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

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

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

in hoisting one anchor

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

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

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

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

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

* In breaking away one anchor from the bottom

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

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

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

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

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

Table 58

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

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

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

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

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

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

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

H = height of the side amidships, measured from the upper edge of the keel to the lower edge of the strength deck stringer, m
$\Sigma\chi_i$ = correction factor taking into account the sail effect of the superstructures.

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

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

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

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

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

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

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

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

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

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

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

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

$$\chi_q = l_q h_q$$

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

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

Table 59

Self-Propelled Transport Ships with an Unlimited Region of Navigation

No.	Characteristic X	Anchors: Bower: Quantity	Anchors: Bower: Total weight, kg	Anchors: Stream anchor, kg	Chain cable for bower anchors: Total length of two cables, m	Chain cable for bower anchors: Anchor chain size, mm	Chain or steel rope for the stream anchor: Length, m	Chain or steel rope for the stream anchor: Anchor chain size, mm	Chain or steel rope for the stream anchor: Diameter of steel rope, mm
1	50	2	150	25	100	12	50	—	6.8
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	18	20.5
17	800	3	2250	250	375	31	125	19	20.5
18	850	3	2400	275	375	32	125	19	20.5
19	900	3	2700	300	375	33	125	20	22
20	950	3	3000	300	400	34	125	21	24
21	1000	3	3200	350	400	36	125	21	24
22	1100	3	3500	400	400	37	125	22	24
23	1200	3	3750	400	420	38	125	23	26
24	1300	3	4100	450	450	40	150	25	26
25	1400	3	4250	450	450	41	150	25	28
26	1500	3	4500	500	450	42	150	25	28
27	1600	3	4750	500	450	43	150	26	28
28	1700	3	5250	600	450	45	150	26	28
29	1850	3	5500	600	450	46	150	28	30
30	2000	3	5750	700	450	46	150	28	30
31	2150	3	6000	700	475	48	175	29	31.5
32	2300	3	6500	800	500	49	175	29	31.5
33	2500	3	6750	800	500	50	175	29	32.5
34	2700	3	7500	900	500	52	175	30	33.5

Continued

No.	Characteristic X	Anchors: Bower: Quantity	Anchors: Bower: Total weight, kg	Anchors: Stream anchor, kg	Chain cable for bower anchors: Total length of two cables, m	Chain cable for bower anchors: Anchor chain size, mm	Chain or steel rope for the stream anchor: Length, m	Chain or steel rope for the stream anchor: Anchor chain size, mm	Chain or steel rope for the stream anchor: Diameter of steel rope, mm
35	3000	3	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	31.5
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	17000	2250	600	76	275	43	—
48	7400	3	18000	2250	600	77	275	44	—
49	7800	3	19500	2500	600	80	275	46	—
50	8200	3	20300	2700	600	82	275	48	—
51	8600	3	21000	2800	600	83	275	49	—
52	9000	3	22000	3000	600	85	275	50	—
53	9500	3	23000	3000	600	87	275	50	—

Note. Two bower anchors with a total weight of at least 2/3 of the tabular value are sufficient for ships navigating in the Caspian Sea and having a characteristic of 600 or larger.

pentagon. If the bar size of the anchor chain cable is denoted as d_c mm, then the chain pitch equal to $8\,d_c$ is to be accommodated along one side AC of the pentagon. Thus, since $AB = BC = 4d_c$, it is evident from triangle OBC that the effective diameter of the cable lifter is

$$D_{cl} = 2R_{cl} = 2\frac{4\,d_c}{\sin\alpha'} \frac{8d_c}{\sin\frac{360^\circ}{10}} = 13.6\,d_c \text{ mm} = 0.013d_c \text{ m} \quad (387)$$

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

$$l_c = 5t_c = 5\times 8d_c = 40d_c \text{ mm} = 0.04d_c \text{ m} \quad (388)$$

where d_c=chain bar size, mm.

Table 60

Mooring and Warping Ropes

Characteristic	Towing rope: Length, m	Towing rope: Circumference of hemp rope, mm	Towing rope: Diameter of steel rope, mm	Warping hawsers: Total length, m	Warping hawsers: Number of ropes	Warping hawsers: Circumference of hemp rope, mm	Warping hawsers: Diameter of steel rope, mm	Cable warps: Total length, m	Cable warps: Number of ropes	Cable warps: Circumference of hemp rope, mm	Cable warps: Diameter of steel rope, mm
50	50	75	—	50	1	65	—	—	—	—	—
75	50	90	11	50	1	65	—	—	—	—	—
100	75	90	11	75	1	65	8.5	—	—	—	—
150	75	100	12	75	1	75	9.5	—	—	—	—
200	100	100	12	100	2	75	9.5	—	—	—	—
250	100	125	15	140	2	100	12	—	—	—	—
300	110	125	15	160	2	100	12	—	—	—	—
350	110	150	17.5	160	2	100	12	—	—	—	—
400	135	150	17.5	180	2	125	15	80	1	100	12
450	135	150	17.5	180	2	125	15	80	1	100	12
500	135	150	17.5	200	2	125	15	85	1	100	12
550	135	175	19.5	200	2	125	15	85	1	100	12
600	135	175	19.5	220	2	150	17.5	90	1	100	12
650	135	175	19.5	240	2	150	17.5	90	1	100	12
700	150	200	21.5	240	2	150	17.5	90	1	100	12
750	150	200	21.5	360	4	150	17.5	90	1	125	15
800	150	200	21.5	360	4	150	17.5	90	1	125	15
850	175	200	21.5	360	4	150	17.5	90	1	125	15
900	175	225	24	360	4	175	19.5	120	2	125	15
950	175	225	24	360	4	175	19.5	120	2	125	15
1000	175	225	24	360	4	175	19.5	120	2	150	17.5
1100	175	225	24	360	4	175	19.5	140	2	150	17.5
1200	190	250	26	360	4	175	19.5	140	2	150	17.5
1300	190	250	26	400	4	200	21.5	150	2	150	17.5
1400	190	275	28	400	4	200	21.5	150	2	150	17.5
1500	190	275	28	480	4	200	21.5	150	2	150	17.5
1600	200	300	30	480	4	200	21.5	180	2	150	17.5
1700	200	300	30	460	4	200	21.5	180	2	150	17.5
1850	200	325	32.5	540	4	200	21.5	180	2	175	19.5
2000	200	350	34.5	540	4	200	21.5	180	2	175	19.5
2150	200	350	34.5	540	4	200	21.5	180	2	175	19.5
2300	220	350	34.5	540	4	225	24	180	2	175	19.5
2500	220	350	34.5	640	4	225	24	200	2	175	19.5

Continued

Characteristic	Towing rope: Length, m	Towing rope: Circumference of hemp rope, mm	Towing rope: Diameter of steel rope, mm	Warping hawsers: Total length, m	Warping hawsers: Number of ropes	Warping hawsers: Circumference of hemp rope, mm	Warping hawsers: Diameter of steel rope, mm	Cable warps: Total length, m	Cable warps: Number of ropes	Cable warps: Circumference of hemp rope, mm	Cable warps: Diameter of steel rope, mm
2700	220	350	34.5	640	4	225	24	200	2	200	21.5
3000	220	350	34.5	640	4	225	24	200	2	200	21.5
3300	240	375	39	640	4	250	26	200	2	200	21.5
3600	240	375	39	640	4	250	26	200	2	200	21.5
3900	240	400	43.5	640	4	250	26	200	2	200	21.5
4200	240	400	43.5	640	4	250	26	200	2	225	24
4500	240	425	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	430	4	250	26
8600	240	—	61.5	960	6	325	32	480	4	250	26
9000	240	—	61.5	960	6	325	32	480	4	250	26
9600	240	—	61.5	960	6	325	32	480	4	250	26

Notes: 1. If the actual characteristic is between two tabular values, data should be taken for the next larger tabular characteristic.

2. The diameter and circumference of ropes selected from the table for ships with square rigging are to be increased by one size.

3. The towing rope for nonpropelling vessels is taken one size larger than the tabular value (in diameter and circumference). In addition to the towing rope indicated in the table, towing vessels (tugs) must have a towing rope for towing other vessels. This latter is to be selected in accordance with the pulling capacity of the hook which is taken with a fivefold margin of safety.

4. If Manilla or sisal hemp ropes are to be used instead of ordinary hemp, they can be taken one size less than the tabular value.

Denoting the heaving-in speed of the anchor cable as v_a m per sec, we can find the speed, n_{cl}, in rpm, of the cable lifter from the equation

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

(a) for windlasses and capstans of bower anchors:

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

(b) for the stern anchoring capstan:

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

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

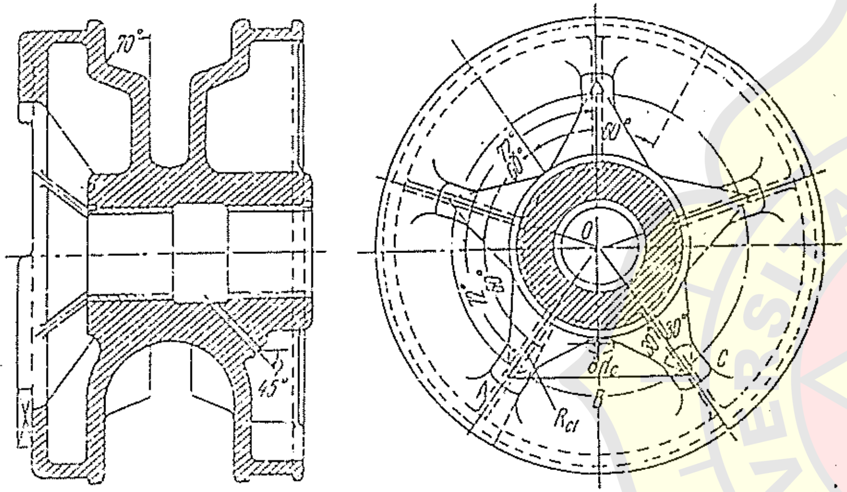


Fig. 170.

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

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

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

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

The torque on the cable lifter is

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

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

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

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

Table 61

Anchor-handling gear	Motive unit shaft speed n_m, rpm	Gearing ratio of mechanism, i_a
Hand-operated capstans		4 to 40
Steam capstans	160 to 320	12 to 60
Electric capstans	800 to 1450	110 to 200
Hand-powered windlasses		9 to 18
Steam windlasses	90 to 270	6 to 30
Electric windlasses	720 to 1550	105 to 250

The torque developed on the shaft of the motive unit is

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

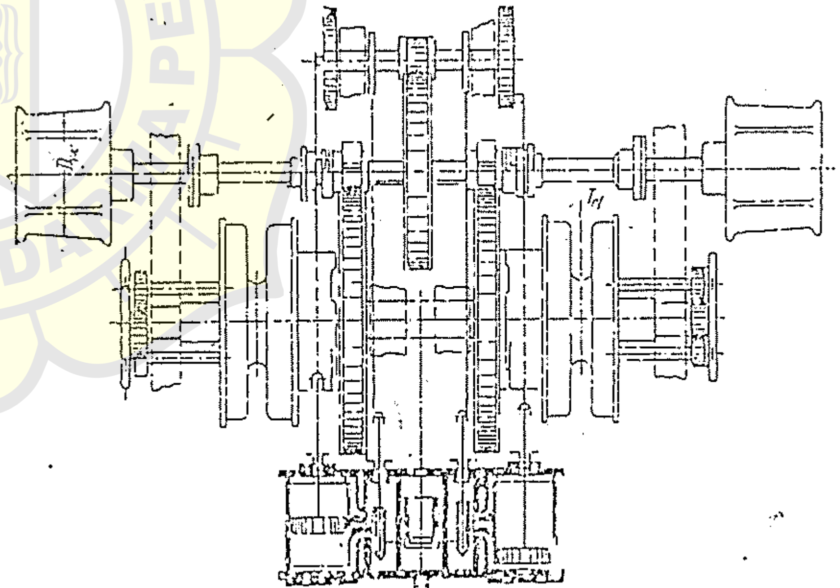


Fig. 171.

where Q_b = 570 to 2,175 = weight of the fully rigged boat, kg
Q_p = total weight of all persons allowed to embark (the weight of one person is approximately 75 kg; the number of persons in a boat may reach 78), kg
$Q_f = 0.05(Q_b + Q_p)$ = weight of the boat's falls, kg
k_n' = 0.9 to 1.1 = coefficient of nonequal distribution of the movable load due to the weight of the persons in the boat.

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

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

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

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

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

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

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

where c = minimum number of blocks.

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

Table 63

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

The winch head diameter is

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

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

The required winch head speed is found from the equation

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

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

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

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

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

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

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

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

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

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

and

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

The cylinder diameter and indicated power of steam boat winches are determined from the same Posdyunin formulas used in

The mean shaft power of the motive unit should be

$$N_e = \frac{M_m n_m}{71620} \quad \text{metric hp}$$

The mean indicated power is

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

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

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

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

The value of $(\alpha_i k_i p_{ts} - p_{ss})$ is approximately from 10 to 15 per cent lower than that taken for a steering engine, due to longer distance from the anchoring mechanism to the steam supply, resulting in higher condensation losses in the pipelines. The other values in the formula are to be within the same limits as for steam steering engines.

The indicated power N_{ia} required to start the engine from rest and the coefficient of reserve power are

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

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

The steam consumption of the engine driving the anchoring arrangement is

$$G_{ia} = g_{ia} N_{ia} \quad \text{kg per hour}$$

where g_{ia} = specific steam consumption, kg per ihp-hr (the same values are taken as for a steam steering engine).

If need arises to determine the pull on the cable lifter from data measured on the anchoring mechanism, formula (390) can be used.

Solving Posdyunin's formula (389) for the torque developed on the shaft of the steam engine we can write

$$M_m = \left(\frac{D_{ca}}{1.37}\right)^3 \eta_m \psi_a (\alpha_i k_i p_{ts} - p_{ss}) \quad \text{kg-cm}$$

On the other hand, if i_a is the total gearing ratio of the transmission in the anchoring mechanism, then

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

Combining the last two equations we obtain

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

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

The diameter of the warp ends is taken equal to

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

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

where d_w = diameter of the warping hawser.

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

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

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

where v_w = hawser heaving-in speed, m per sec, is to be assigned according to the pull of the warp end (Table 58).

The gearing ratio between the warping shaft and the shaft of the motive unit is

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

The pulling force developed on the warp end is

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

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

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

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

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

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

This equation can be solved for V_e and V_f:

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

and

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

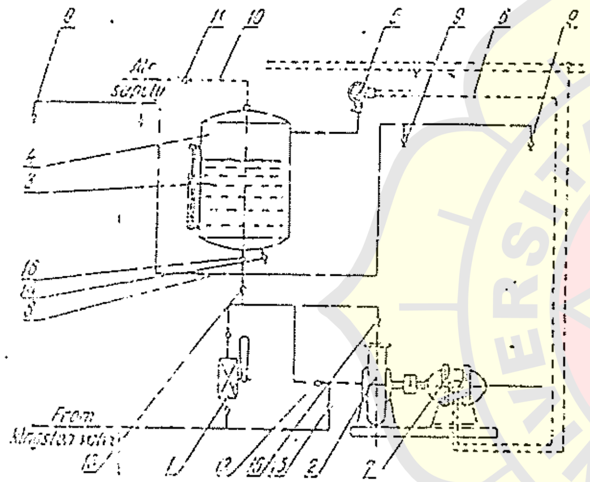


Fig. 189.

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

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

Therefore the minimum and maximum volumes of the air are

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

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

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

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

(D) SANITARY AND SCUPPER SYSTEMS

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

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

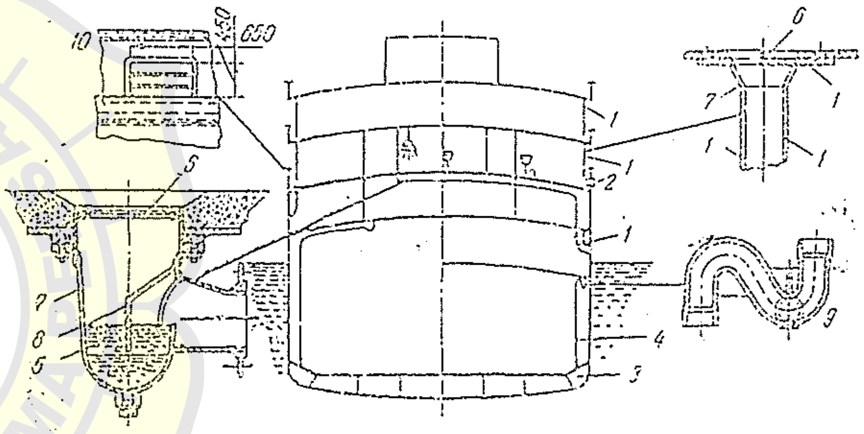


Fig. 190.

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

Water is drained from decks located lower than the load waterline through scupper pipes 1 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

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

last brackets for reserve:

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

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

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

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

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

s = operating range [sm]

V_{serv} = speed [kn]

1 KW = 0.736 PS (BHP).

Powercraft features
1 HP (Horsepower)
= 550 foot pounds per second
= 0.746 kilowatt
= 1.014 PS (or Cheval Vapeur)
1 PS (Pferdestärke) or CV (Cheval Vapeur)
= 542 foot pounds per second
= 0.986 hp
= 0.735 kw

Long ton = 1.01605 metric ton
Short ton = 0.907185 ton

Motors:

Specific fuel oil consumption:

1 kw (kilowatt)
= 1000 watts
= 1.340 hp
= 1.359 PS or CV
= 737 foot pounds per second

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

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

$b = 196 \ldots 209$ [g/KW·h]

for full power: addition 5%

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

For steam turbines:

Standard circulation without furnace gas reheat

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

$b = 278 \ldots 286$ [g/KW·h]

with furnace gas reheat

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

$b = 252 \ldots 265$ [g/KW·h]

For gas turbines:

Gasoline and light crude oils

$b = 299 \ldots 312$ [g/KW·h]

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

Required volume of storage tanks

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

1 kw = 0.7355

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Additions to the volume

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

<u>Diesel oil</u>

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

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

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

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

additions see fuel oil!

<u>Lubrication oil</u>

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

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

$b = 0.8 \ldots 1.2$ [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 $\gamma_{lubr} = 0.90\ t/m^3$; $V = \dfrac{w}{\gamma}\ (m^3)$

<u>Fresh water</u>

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

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

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

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.

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

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

Sounding/ullage tables delivered by yard.

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).

Type and Location of Main Engine

Is another part of the contract influencing ship design. (Ship weight, volume, fuel consumption).
Economy is determined by the choice of the main engine type, also

Tabel A-7 Refrigeran 22: sifat-sifat uap gas panas lanjut[5]

t, °C	v, L/kg	h, kJ/kg	s, kJ/kg · K	v, L/kg	h, kJ/kg	s, kJ/kg · K	v, L/kg	h, kJ/kg	s, kJ/kg · K
	Suhu jenuh, −20°C			Suhu jenuh, −10°C			Suhu jenuh, 0°C		
−20	92,8432	397,467	1,7841						
−15	95,1474	400,737	1,7969						
−10	97,4256	404,017	1,8095	65,2399	401,555	1,7671			
−5	99,6808	407,307	1,8219	67,0081	404,983	1,7800			
0	101,915	410,610	1,8341	68,6524	408,412	1,7927	47,1354	405,361	1,7518
5	104,130	413,926	1,8461	70,2751	411,845	1,8052	48,3899	408,959	1,7649
10	106,328	417,258	1,8580	71,8785	415,283	1,8174	49,6215	412,567	1,7777
15	108,510	420,606	1,8697	73,4644	418,730	1,8295	50,5328	416,159	1,7903
20	110,678	423,970	1,8813	75,0346	422,186	1,8414	52,0259	419,649	1,8026
25	112,832	426,353	1,8928	76,5904	425,653	1,8531	53,2028	423,339	1,8148
	Suhu jenuh, 5°C			Suhu jenuh, 10°C			Suhu jenuh, 15°C		
5	40,3556	407,143	1,7446						
10	41,4580	410,851	1,7578	34,7136	408,835	1,7377			
15	42,5379	414,542	1,7708	35,6907	412,651	1,7511	29,9574	410,430	1,7311
20	43,5979	418,222	1,7834	36,6454	416,442	1,7642	30,8606	414,362	1,7556
25	44,6401	421,894	1,7958	37,5804	420,215	1,7769	31,7114	418,260	1,7578
30	45,6665	425,562	1,8080	38,4981	423,974	1,7894	32,5427	422,133	1,7707
35	46,6786	429,229	1,8200	39,4002	427,724	1,8017	33,3568	425,955	1,7833
40	47,6779	432,897	1,8319	40,2884	431,469	1,8137	34,1556	429,823	1,7956
45	48,6656	436,569	1,8435	41,1642	435,211	1,8256	34,9409	433,650	1,8075
50	49,6427	440,247	1,8550	42,0286	438,954	1,8373	35,7139	437,470	1,8197

Tabel A-7 (lanjutan)

t, °C	v, L/kg	h, kJ/kg	s, kJ/kg · K	v, L/kg	h, kJ/kg	s, kJ/kg · K	v, L/kg	h, kJ/kg	s, kJ/kg · K
	Suhu jenuh, 20°C			Suhu jenuh, 25°C			Suhu jenuh, 30°C		
20	26,0032	411,918	1,7246						
25	26,7900	415,977	1,7383	22,6242	413,289	1,7183			
30	27,5542	419,991	1,7517	23,3389	417,487	1,7322	19,7417	414,530	1,7120
35	28,2989	423,970	1,7646	24,0306	421,627	1,7458	20,3962	418,881	1,7262
40	29,0264	427,922	1,7774	24,7027	425,721	1,7590	21,0272	423,159	1,7400
45	29,7389	431,852	1,7899	25,3575	429,779	1,7718	21,6381	427,378	1,7534
50	30,4379	435,766	1,8021	25,9974	433,807	1,7844	22,2316	431,549	1,7664
55	31,1250	439,668	1,8141	26,6239	437,813	1,7967	22,8101	435,683	1,7791
60	31,8012	443,561	1,8258	27,2386	441,801	1,8087	23,3733	439,787	1,7915
65	32,4678	447,450	1,8374	27,8427	445,777	1,8206	23,9288	443,867	1,8036
	Suhu jenuh, 32°C			Suhu jenuh, 34°C			Suhu jenuh, 36°C		
35	19,0907	417,648	1,7182	17,8590	416,325	1,7099			
40	19,7093	422,014	1,7322	18,4675	420,792	1,7243	17,2953	419,453	1,7162
45	20,3062	426,310	1,7458	19,0526	425,174	1,7382	17,8708	423,961	1,7304
50	20,8847	430,549	1,7591	19,6178	429,487	1,7517	18,4247	428,358	1,7442
55	21,4471	434,743	1,7719	20,1660	433,747	1,7647	18,9603	432,690	1,7575
60	21,9956	438,900	1,7845	20,6994	437,963	1,7775	19,4802	436,970	1,7704
65	22,5318	443,028	1,7968	21,2199	442,143	1,7899	19,9865	441,207	1,7830
70	23,0571	447,133	1,8089	21,7289	446,294	1,8021	20,4807	445,410	1,7954
75	23,5726	451,219	1,8207	22,2278	450,424	1,8141	20,9643	449,586	1,8074
80	24,0794	455,292	1,8323	22,7176	454,535	1,8258	21,4385	453,739	1,8193

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.

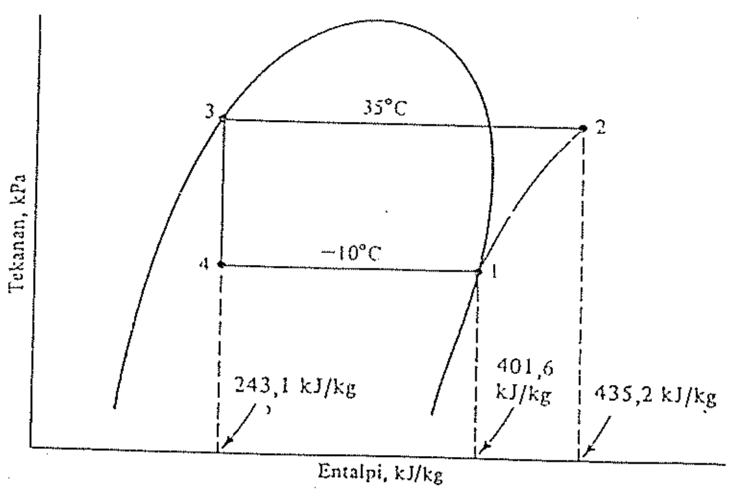
Untuk menemukan h_2 melalui garis entropi tetap geser titik 1 hingga mencapai tekanan jenuh yang sesuai dengan suhu 35°C. Tekanan pengembunan ini adalah 1354 kPa, dan nilai h_2 = 435,2 kJ/kg.

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.

Sifat-sifat zat yang melebur dan membeku

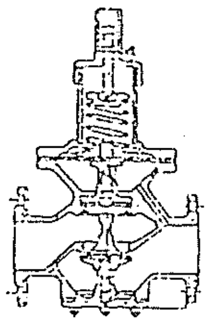
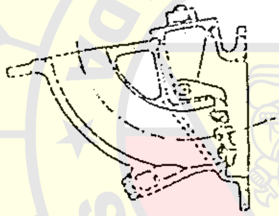
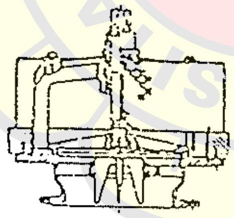
1. Waktu melebur atau membeku suhu zat tetap tidak berubah
2. Umumnya zat sebelum mencair/membeku didahului oleh kelumeran atau meleleh
3. Umumnya zat yang mencair, volumenya mengembang (kecuali es, besi, perak dan bismut)
4. Pada umumnya titik lebur/titik beku itu naik/turun apabila tekanannya bertambah tinggi/rendah
5. Titik lebur logam paduan, biasanya lebih rendah daripada titik lebur logam-logam asalnya. Misalnya timah solder (200°C) terdiri dari 50 % timbal (328°C) dan 50 % timah (232°C).
6. Tidak semua zat dapat mencair/membeku, ada yang tinggal tetap tidak berubah, ada pula yang dipisahkan secara kimia.
7. Dalam keadaan tertentu(tenang), beberapa macam zat cair dapat didinginkan sampai ke suhu di bawah titik bekunya. Apabila zat cair tersebut digerakkan, maka akan segera membeku.

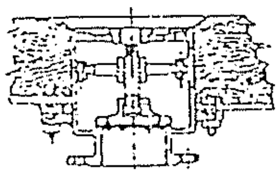
KALOR LEBUR & KALOR UAP

Pada tekanan satu atmosfir

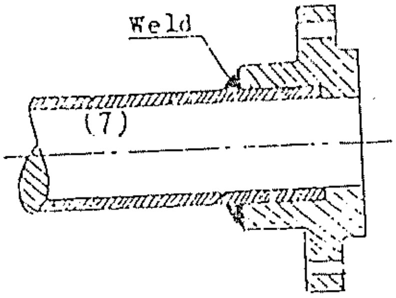
NAMA ZAT	MELEBUR/MEMBEKU			MENGUAP/MENGEMBUN		
	Titik lebur °C	Kalor lebur kkal/kg	Kalor lebur Btu/lb	Titik didih °C	Kalor uap kkal/kg	Kalor uap Btu/lb
Air (H_2O)	–	–	–	100	539	970,4
Es	0	80,4	144	–	–	–
(Air) Raksa(Hg)	–39	3	5,4	361	71	127,2
Amonia (NH_3)	–75	106	190,4	–33,3	327	565
R – 12(CCl_2F_2)	–158			–29,8	39,47	71,0
R – 22($CHClF_2$)	–160			–40,8	55,81	100,5
R – 502				–45,5	41,2	76,0
Timbal(Pb)	327	[illegible]	[illegible]	1740	175	313,4
Timah(Sn)	232	[illegible]	[illegible]	2270		
Aluminium(Al)	660	96	[illegible]	2100		
Tembaga(Cu)	1083	49	87,7	2360	1750	3134
Seng(Zn)	420	28,1	50,3	907		
Emas(Au)	1063	12,6	22,6	2960		
Perak(Ag)	960	21	37,6	2210		
Besi(Fe)	1530	28	50,1	2735		

- VALVES -

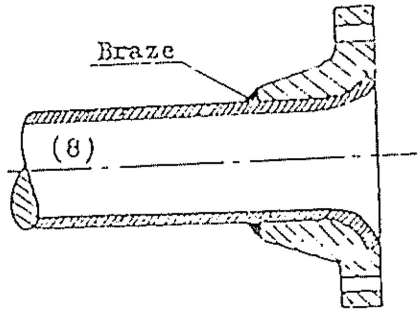
Type	Construction	Features
Reducing valve		Serves to reduce the pressure of the liquid flowing beyond the valve, and to maintain this pressure automatically, but it cannot be used as a shutoff device.
Relief valve		Serves to prevent the pressure in the object on which it is installed from increasing above the permissible value.
Swing-check valve		Used as an automatic shutoff device operated by the pressure of the liquid or for mounting at the end of pipelines discharging at the shipside.
Drain valve		Designed for draining liquid from a space which does not have its own drainage facilities.

Type	Construction	Features
Scupper valve		Serves to dispose of water from decks located below the waterline.

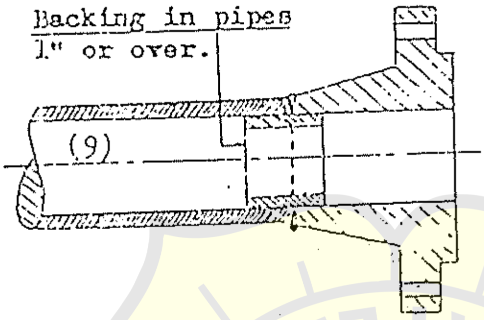
Gambar 5



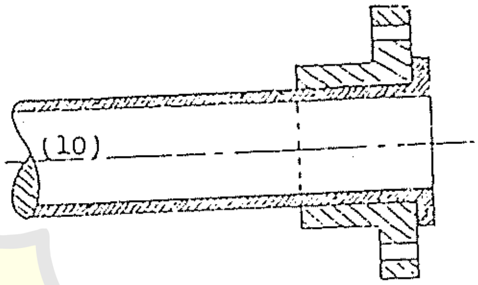
(7). Socket Welded.



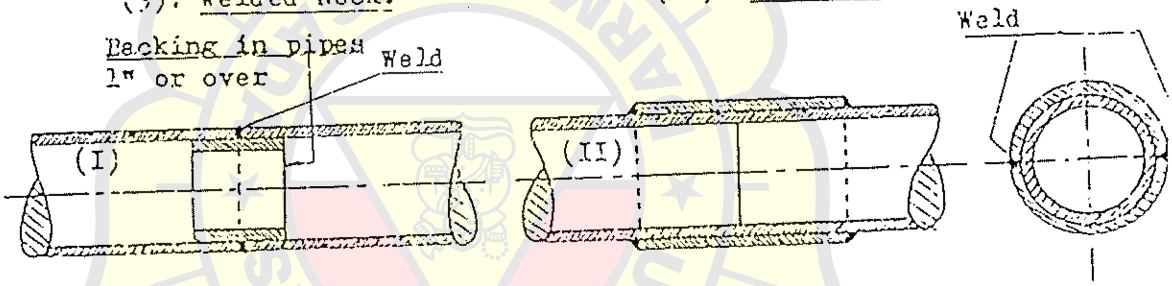
(8). Expanded and Brazed.



(9). Welded Neck.

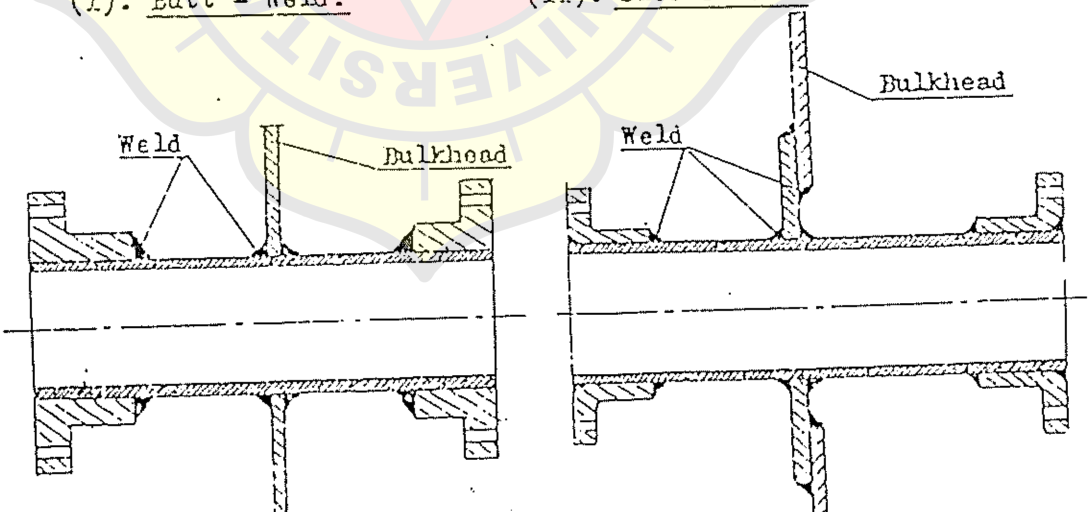


(10). Van Stone.



(I). Butt - Weld.

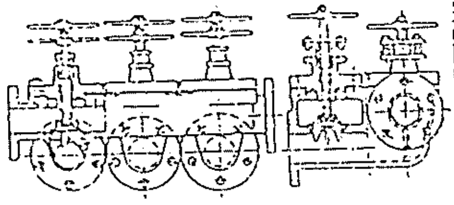
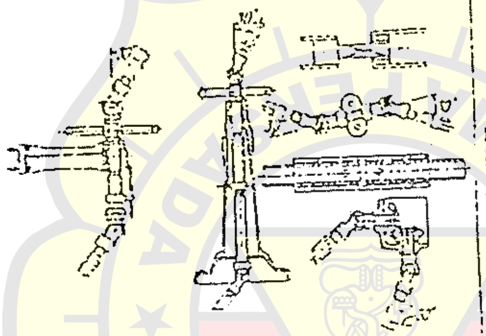
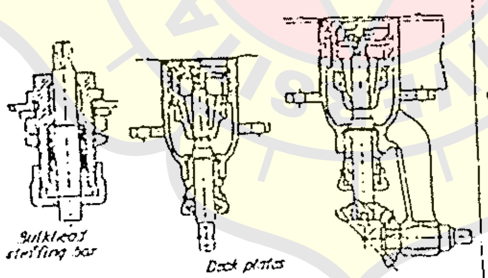
(II). Sleeve Weld.



Sambungan pipa
menembus Bulkhead.

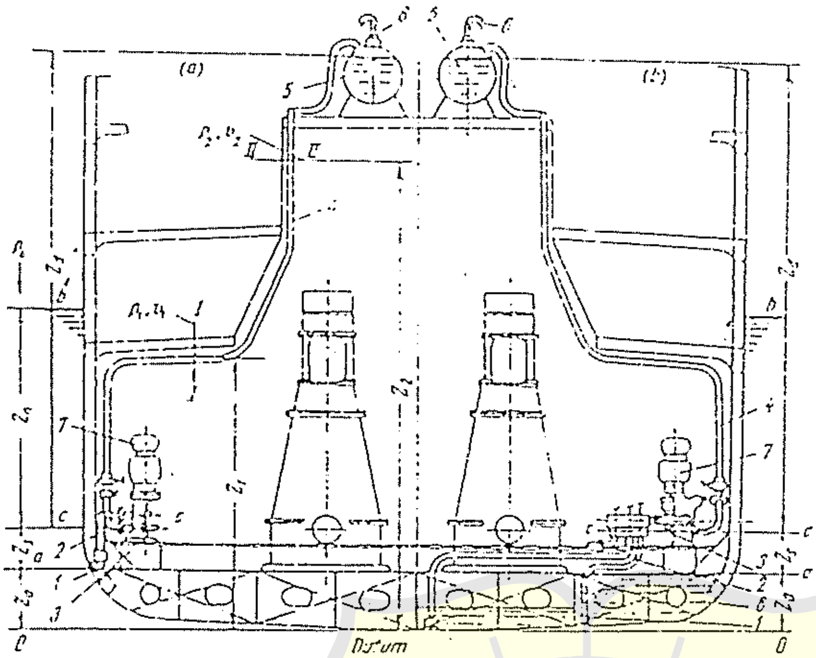
Gambar 6

- VALVES -

Construction	Type and features
	Valve chest with separate inlets and a common outlet. This chest with a separate inlet to each valve and a common outlet from all the valves is usually mounted in the suction line of a pump serving several objects and connects the object being served at the given time to the pump.
	Rod-type remote-control gear. Remote-control gear is designed for convenience in controlling fittings located in places difficult of access, and also in emergencies when access to the local gear is impossible.
	Bulkhead stuffing boxes and deck plates. These fittings are used to ensure the watertightness of bulkheads and the deck at places where shafts or rods pass through them.

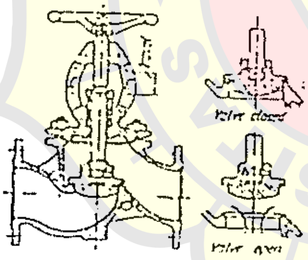
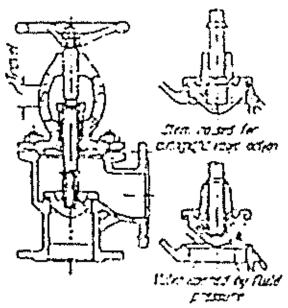
Gambar 7

NO 8



Gambar 1

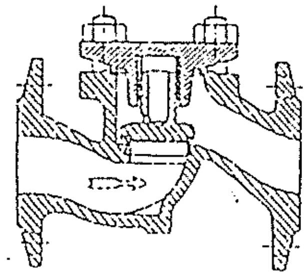
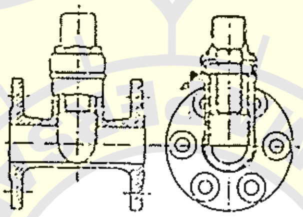
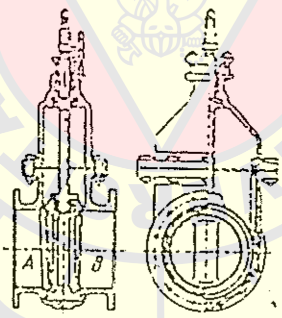
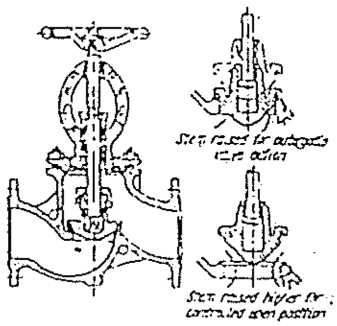
- Liquid Flow and Principles of Pump Action -

Type	Construction	Features
Stop valve		The valve disk is linked rigidly to the stem and can be actuated only from the handwheel. Serves as a shut-off device.
Screw-down nonreturn (check) valve		The valve disk is not fixed to the stem and can slide along it. When the stem is screwed down tight the valve acts as a shutoff device. When the stem is raised the valve operates as an adjustable-lift check, or nonreturn, valve. The stem does not raise the disk.

- VALVES -

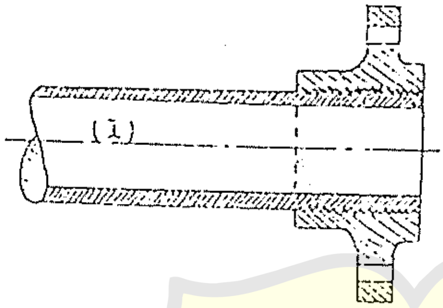
Gambar 2

- VALVES -

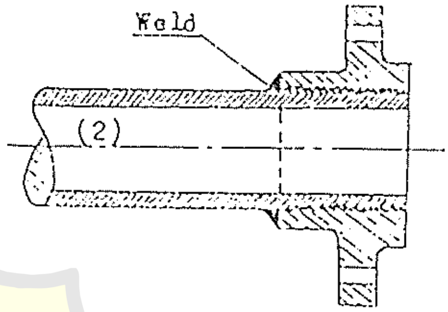
Type	Construction	Features
Nonreturn (lift-check) valve		Allows liquid to flow only in one direction. The valve disk is lifted from or returned to its seat by the pressure of the liquid acting on the corresponding surface of the disk.
Throttle valve		Serves to reduce the pressure of the liquid by increasing its velocity and reducing the passage area. It does not ensure constant pressure after the valve in cases when the pressure before it is variable.
Gate valve		Serves as a shutoff device for large size pipelines. It can be used as an overflow valve for passing liquid from one compartment to another.
Controlled nonreturn valve		The valve disk is freely mounted on the stem but its lift, without raising the stem, is limited. The free travel of the disk along the stem enables the valve to operate as an adjustable-lift check valve. The limited nature of the travel allows the stem to lift the disk whenever required.

- Flens dari besi tuang dapat digunakan dengan sistim sambungan yang disekrup, dan hanya boleh dipakai didalam sistim dimana penggunaannya tidak dilarang (kebocoran).
- Pipa-pipa non ferrous harus dipatri (solder brazed), tetapi untuk diameter lebih kecil atau sama dengan 2 inci dapat dengan disekrup.

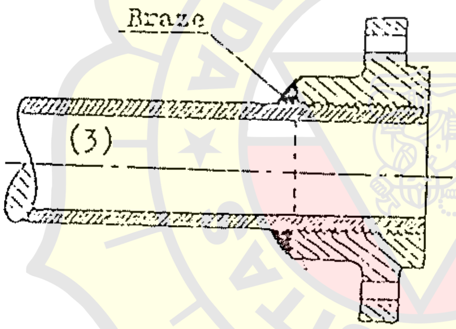
Beberapa cara untuk pengikatan flens yang telah disetujui, dapat dilihat pada gambar dibawah ini.



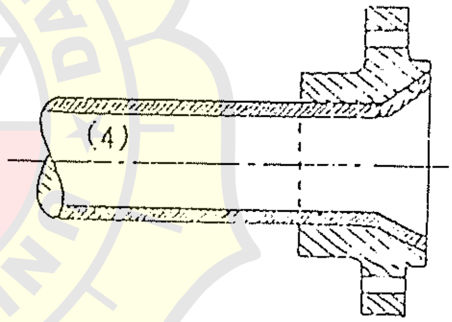
(1). Screwed.



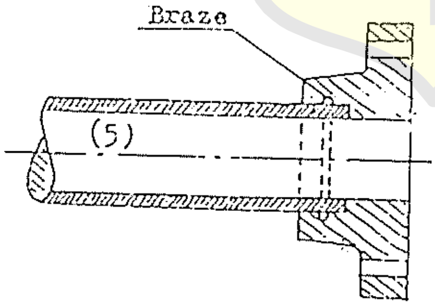
(2). Screwed and Welded.



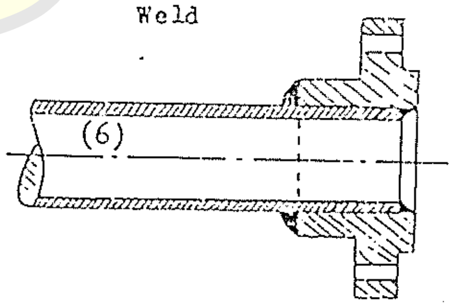
(3). Screw and Brazed.



(4). Expanded.



(5). Silver Brazed.



(6). Slip-on Weld.

Gambar 4

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

$$Q = 10 fRA10 \qquad (2.2.a)$$

di mana Q: Limpasan keseluruhan (m^3)
R: Curah hujan standar (mm)
f: Koefisien limpas
A: Luas wilayah drainase (ha)

Dari jumlah limpasan yang dihitung dengan cara di atas kemudian dapat diperkirakan kapasitas pompa drainase yang diperlukan dengan rumus

$$Q_p = \frac{Q}{24 \times 3600 \times D} \qquad (2.2.b)$$

di mana Q_p: Kapasitas pompa drainase (m^3/s)
D: Lamanya genangan yang diperbolehkan (hari)

Koefisien limpas yang dipakai untuk menentukan limpasan total dipengaruhi oleh curah hujan total seperti diberikan di dalam Tabel 2.7.

Jumlah hari limpas harus dihitung secara coba-coba dengan memperhatikan bahwa limpasan total akan terdistribusikan seperti dalam Tabel 2.8.

Tabel 2.7 Curah hujan total dan koefisien limpas total.

Curah hujan total (mm)	Kurang dari 10	10-30	30-50	50-100	100-200	200-300	Lebih dari 300
Koefisien limpas total	0	0,10	0,30	0,50	0,80	0,90	0,95

Tabel 2.8 Faktor distribusi limpasan dari curah hujan tunggal.

Curah hujan (mm) \ Hari	Hari ke-1	Hari ke-2	Hari ke-3	Hari ke-4	Jumlah
Kurang dari 30	100%			–	100%
30-50	70%	30%		–	100%
50-100	60%	30%	10%	–	100%
Lebih dari 100	50%	30%	15%	5%	100%

Untuk penentuan akhir dari spesifikasi perencanaan, kondisi limpasan air hujan dan kondisi fluktuasi muka air harus diperhitungkan. Dalam hal ini perlu dipelajari buku-buku profesional dalam bidang tersebut.

(5) Pengairan tanah pertanian

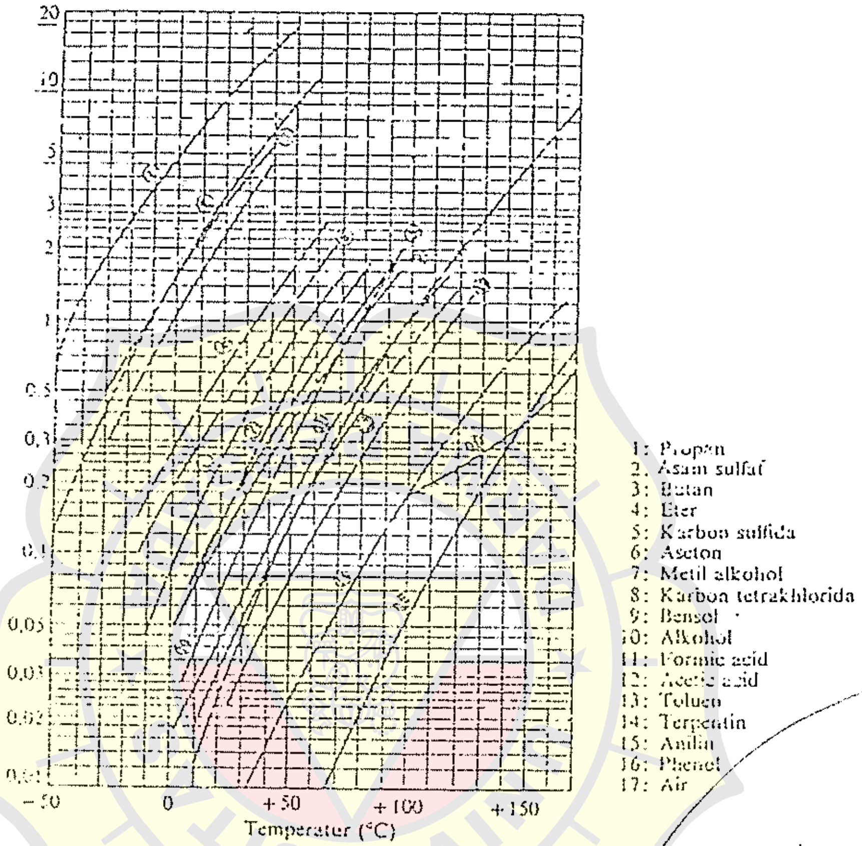
Ditinjau dari cara pengairan, tanah pertanian dapat dibedakan antara sawah dan ladang.

(a) Pengairan sawah

1) *Keperluan air*

Sawah untuk tanaman padi harus digenangi air dengan kedalaman tertentu. Untuk memelihara kedalaman tersebut diperlukan tambahan air terus menerus guna mengganti penyusutan karena transpirasi tanaman, penguapan sawah, dan perkolasi*. Jadi:

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



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

Gb. 2.1 Sifat-sifat fisik berbagai zat cair.

2.4 Head

2.4.1 Head Total Pompa

Head total pompa yang harus disediakan untuk mengalirkan jumlah air seperti direncanakan, dapat ditentukan dari kondisi instalasi yang akan dilayani oleh pompa. Seperti diperlihatkan dalam Gb. 2.2, head total pompa dapat ditulis sebagai berikut:

$$\boxed{H = h_a + \Delta h_p + h_l + \frac{v_d^2}{2g}} \tag{2.6}$$

di mana H: Head total pompa (m)

h_a: Head statis total (m)
Head ini adalah perbedaan tinggi antara muka air di sisi keluar dan di sisi isap; tanda positip (+) dipakai apabila muka air di sisi ke luar lebih tinggi dari pada sisi isap.

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

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

h_l: Berbagai kerugian head di pipa, katup, belokan, sambungan, dll (m),

di mana h_p: Head tekanan (m)
p: Tekanan (kgf/cm^2)
γ: Berat per satuan volume zat cair yang dipompa (kgf/l)

Apabila tekanan diberikan dalam kPa, dapat dipakai rumus berikut:

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

di mana p': Tekanan (Pa)
ρ: Rapat masa (kg/l)

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

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

Sebagaimana diutarakan di atas, untuk menentukan head total yang harus disediakan pompa, perlu dihitung lebih dahulu head kerugian h_l. Di bawah ini akan diuraikan cara menghitung kerugian head tersebut.

2.4.2 Head Kerugian

Head kerugian (yaitu head untuk mengatasi kerugian-kerugian) terdiri atas head kerugian gesek di dalam pipa-pipa, dan head kerugian di dalam belokan-belokan, reduser, katup-katup, dsb. Di bawah ini akan diberikan cara menghitungnya, satu per satu.

(1) Head kerugian gesek dalam pipa

Untuk menghitung kerugian gesek di dalam pipa dapat dipakai salah satu dari dua rumus berikut ini:

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

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

di mana v: Kecepatan rata-rata aliran di dalam pipa (m/s)
C, p, q: Koefisien-koefisien
R: Jari-jari hidrolik (m)

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

S: Gradien hidrolik

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

h_f: Head kerugian gesek dalam pipa (m)
λ: Koefisien kerugian gesek
g: Percepatan gravitasi (9,8 m/s^2)
L: Panjang pipa (m)
D: Diameter dalam pipa (m)

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

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

di mana Re: Bilangan Reynolds (tak berdimensi)

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

D: Diameter dalam pipa (m)

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

Pada $Re < 2300$, aliran bersifat laminer.

Pada $Re > 4000$, aliran bersifat turbulen.

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

(I) Aliran laminer

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

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

(II) Aliran turbulen

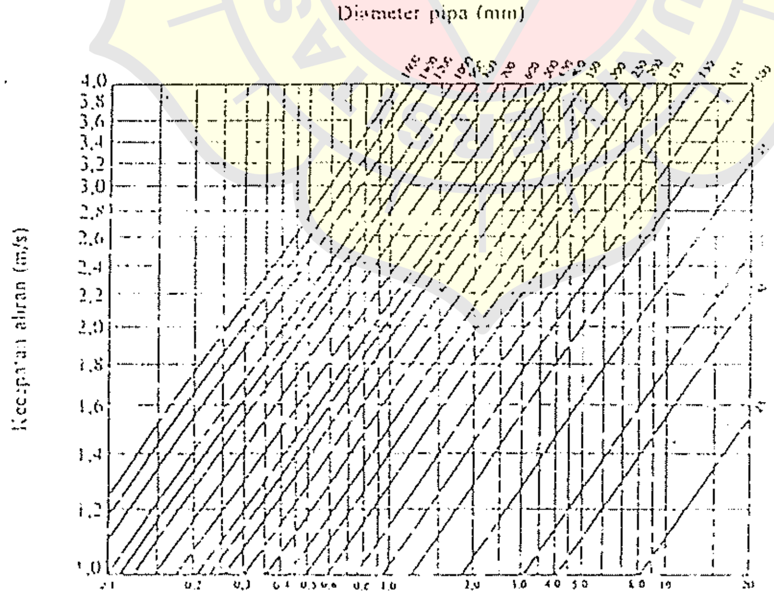
Untuk menghitung kerugian gesek dalam pipa pada aliran turbulen terdapat berbagai rumus empiris. Di bawah ini akan diberikan cara perhitungan dengan rumus Darcy dan Hazen-Williams.

1) *Formula Darcy*

Dengan cara Darcy, koefisien kerugian gesek λ dari Pers. (2.12) dihitung menurut rumus

$$\lambda = 0{,}020 + \frac{0{,}0005}{D} \tag{2.15}$$

di mana D adalah diameter dalam pipa (m). Rumus ini berlaku untuk pipa baru dari besi cor. Jika pipa telah dipakai selama bertahun-tahun, harga λ akan menjadi 1,5



Gb. 2.4 Kerugian gesek pada pipa lurus (rumus Darcy).