BAB VII
PENUTUP

VII. 1 Kesimpulan

Dari hasil perhitungan yang telah dilakukan pada kapal rancangan yaitu kapal Ferry Ro-Ro 820 GT dengan dimensi utama kapal sebagai berikut :

Panjang keseluruhan kapal	L_{OA}	: 88,47 m
Panjang antara garis tegak kapal	L_{PP}	: 81,00 m
Panjang antara garis air	L_{WL}	: 85,05 m
Lebar kapal	B	: 15,60 m
Tinggi kapal	H	: 5,00 m
Sarat air kapal	T	: 3,75 m
Kecepatan	Vs	: 16,00 knot
Gross Tonage (GT)		: 820 Ton
Radius Pelayaran		: 1200 Miles
Klasifikasi		: BKI

⇨ Untuk dapat menentukan besarnya daya motor induk sebagai penggerak utama kapal, maka faktor kecepatan daerah pelayaran serta dimensi dari dari kapal mempunyai pengaruh sangat besar.

⇨ Didalam perancangan 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.

⇨ Tata letak mesin induk, mesin bantu, maupun peralatan peralatan lain hendaknya diatur seefisien mungkin, hal ini untuk mempermudah dalam perawatan dan perbaikan peralatan yang ada di kamar mesin itu sendiri.

⇨ Peletakan permesinan berpengaruh pada stabilitas kapal.

⇨ Pemilihan mesin bantu tergantung dari jumlah daya yang harus disuplai pada kondisi operasi kapal yang berbeda beda.

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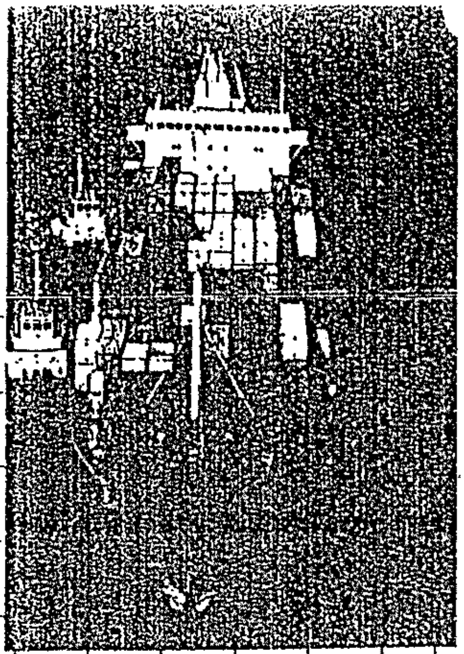
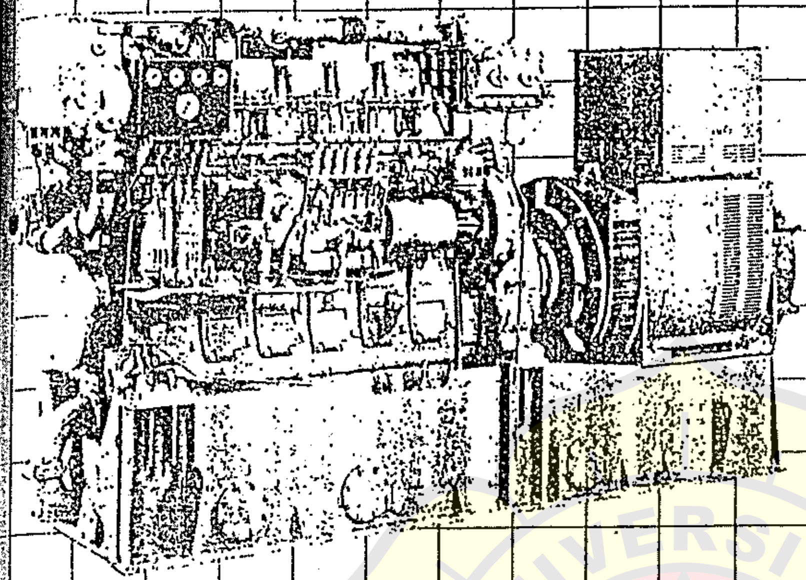
REFERENSI

6NY16L

Engine output
200 441 kW (272–600 PS)





Depending on the specifications or options that have been chosen, your model may differ slightly from the one in the photograph

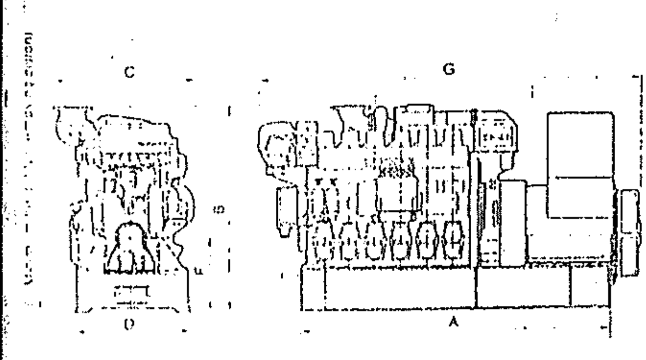
Specifications

Engine model		6NY16L-HN		6NY16L-DN		6NY16L-UN		6NY16L-SN		6NY16L-EN	
		Vertical water-cooled 4-cycle diesel engine									
of cylinders		6									
nder bore × stroke	mm	160 × 200									
displacement	ℓ	24.13									
nuous rated output	kW (PS)	200 (272)	265 (360)	235 (320)	310 (421)	270 (367)	355 (483)	310 (421)	400 (544)	353 (480)	441 (600)
e speed	rpm	1000	1200	1000	1200	1000	1200	1000	1200	1000	1200
mean effective pressure	MPa (kgf/cm²)	0.995 (10.15)	1.097 (11.19)	1.171 (11.94)	1.283 (13.09)	1.343 (13.69)	1.472 (15.01)	1.540 (15.70)	1.658 (16.91)	1.756 (17.90)	1.829 (18.65)
rator capacity	kW	180	240	200	280	240	320	280	360	320	400
bustion system		Direct Injection									
ting system		Compressed air									
nal ensions Overall length	mm	1996									
Overall width	mm	1085									
Overall height	mm	1532									
weight	kg	2880									

The engine dry weight may differ depending upon the specifications and attached accessories

Dimensions (Units: mm)

The dimensions and weights for the diesel engine generator sets are simply reference values. The values may differ for different generator manufacturers.



Engine model	6NY16L-HN	6NY16L-DN	6NY16L-UN	6NY16L-SN	6NY16L-EN
A	2530	2530	2530	2530	2530
B	1613	1613	1613	1613	1613
C	1136	1136	1136	1136	1136
D	940	940	940	940	940
E	1725	1725	1725	1725	1725
F	600	600	600	600	600
G	2991	2991	2991	2991	2991
Dry weight of generating equipment (kg)	5500	5500	5500	5500	5500

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

ok.

Pengertian - pengertian power (keterangan gambar):

- **EHP = Effective Horse Power (PE),** adalah daya yang diperlukan untuk menggerakkan kapal di air atau untuk menarik kapal, tidak termasuk tenaga yang memutar propeller dari engine.

- **IHP = Indicate Horse Power (PI),** adalah daya motor bakar dalam engine tersebut yang keluar dan pembakaran secara langsung atau internal combution engine. Biasanya EHP/IHP adalah 50 %, artinya IHP besarnya 2 kali EHP, tetapi hal tersebut bukan patokan utama.

- **BHP = Breake Horse Power (PB),** adalah daya yang dipakai untuk indicated power, atau daya yang dikeluarkan oleh mesin dengan dipengaruhi beban engine itu sendiri (losses dalam engine al; gesekan-gesekan, panas engine dll). Maksimum BHP adalah tenaga maksimal engine yang diteruskan oleh engine, biasanya dan selalu pada maksimum Rpm yang diijinkan.

- **SHP = Shaft Horse Power (PS),** adalah daya yang disalurkan oleh mesin penggerak ke shaft propeller (daya poros). SHP adalah BHP dikurangi tenaga yang hilang (losses) sebelum tersalur ke shaft (Losses pada gear box sebesar ± 3%, di shaft bearing ± 1,5%. Hal tersebut penting untuk diingat bahwa SHP adalah ukuran yang seharusnya benar-benar digunakan dalam membuat perhitungan propeller.

- **DHP = Delivered Horse Power (PD),** adalah daya yang disalurkan ke baling-baling, dengan kerugian mekanis bracket dan bearingnya.

- **THP = Thrust Horse Power (PT),** adalah daya yang disalurkan oleh baling-baling kapal tanpa pengaruh cuaca atau keadaan lingkungan.

- Vs = Speed of service, adalah kecepatan kapal
T = Total Resistance, jumlah tahanan total kapal Sangat menarik untuk melihat secara detail dimana energi dan bahan bakar yang tersalurkan, ± 35 % hilang karena panas di udara terbuka, ± 25 % hilang karena panas dan vibrasi pada air, ± 2% hilang pada shaft propeller, dan kesemuanya tersebut hanya tinggal ± 38% secara kasar digunakan untuk mengatasi tahanan-tahanan kapal.

Gambar 1 - Engine performence curve salah satu mesin diesel

Engine Performance Curve (Envelope Curve)
Tenaga yang dapat mendorong sebuah kapal dimana tenaga tersebut dapat dikonversikan ke gaya berputar yang memutar propeller, gaya berputar tersebut dinamakan torsi. Tenaga dan torsi didapatkan dari engine benar-benar dapat dilihat dikurva engine performance. Kurva tersebut tersedia pada engine performance curve yang didistribusikan oleh hampir semua engine maker, dimana kurva tersebut meng-eplotkan BHP, torsi dan jumlah bahan bakar yang dikonsumsi dibandingkan dengan Rpm.

...da kemudi balansir, untuk mengurangi kemungkinan getaran, bagian ...ansir dianjurkan ≤ 23 ... dari seluruh luas kemudi dan lebar bagian pada potongan-potongan horisontal < 0,35 lebar sayap kemudi.

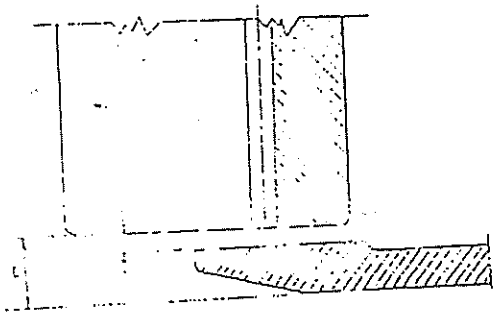
...ada kapal-kapal yang mempunyai batas sarat air yang cukup ...mempunyai ukuran yang tinggi (λ = hp/bp cukup tinggi). ...etapi tinggi kemudi harus diperlihatkan pula menurut bentuk ...kapal.

...rapa batasan untuk harga λ :

...al barang dan kapal penumpang : λ = 1.?
...al coaster : λ = 1,05 - 1,15
...al tunda, pandu : λ = 1,8
...al ikan ukuran sedang : λ = 1,55 - 2,0

Dianjurkan tinggi tiap-tiap kemudi harus menutupi diameter ...aling. Bagian bawah kemudi untuk menjaga kerusakan-kerusakan ...etan dengan dasar laut harus lebih tinggi dari garis dasar kapal.

Batas-batasnya sebagai berikut :



- Untuk kemudi menggantung atau setengah menggantung
  t = (4 - 10%) h
- Untuk kemudi bertumpu;
  t = (6 - 12)% h

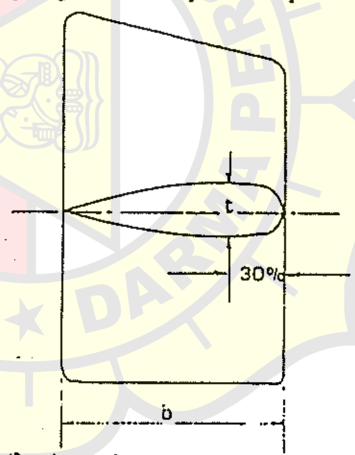
Dimana :
h = tinggi kemudi.

Contoh : Umumnya untuk semua bentuk diambil ketentuan :
t : 50 m/m.

Van Lammeren ditetapkan batasan-batasan λ = h/b sebagai berikut :

Type kapal dan kemudi	h/d
Kapal barang 1 baling-baling dan kapal penumpang semuanya dengan kemudi balansir.	1,8
Kapal pantai 1 baling-baling dengan kemudi balansir.	1,15
4. Untuk semua kapal dengan 2 baling-baling dengan kemudi biasa.	1,5
5. Untuk kapal-kapal 2 baling-baling dengan kemudi setengah balansir.	1,1
6. Untuk kapal-kapal dengan 2 baling-baling dengan dua kemudi.	2,2

Bentuk kemudi harus dibuat sedemikian supaya dengan perubahan letak kemudi dalam sudut attack yang tidak begitu besar, kapal dapat membuat belokan besar, dengan catatan pada saat yang sama dengan perubahan letak kemudi tersebut diperhitungkan supaya tidak mempengaruhi kecepatan kapal.



Gambar 142.

Berdasarkan praktek yang dilakukan, koefisien tebal plat profil kemudi :

Ct = t/b terletak dalam batas-batas : 0,18 - 0,22.

Tetapi untuk kemudi setengah menggantung pada kapal besar hanya Ct mencapai 0,5.

Untuk kemudi biasa (tak balansir) untuk twin screw diambil batas-batas.

Ct = 0,15 - 0,18

Untuk setengah balansir :

Ct = 0,18 - 0,22

Kemudi kembar menggantung biasanya lebih tebal dari kemudi yang bertumpu, tetapi untuk menjaga kekuatan, kemudi tersebut mempunyai harga : Ct = 0,2

Untuk menghindari getaran dianjurkan supaya jarak maximum penampang kemudi yaitu 30% lebar profil, dihitung dari permukaan depan.

Koefisien kompensasi dihitung dengan rumus pendekatan yang menghasilkan perhitungan moment putar yang sangat kecil di poros, sehingga memperkecil kekuatan motor penggerak kemudi ...

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	25	360	4	175	19.5	140	2	150	17.5
1300	190	250	25	400	4	200	21.5	150	2	150	17.5
1400	[illegible]	[illegible]	25	400	4	200	21.5	150	2	150	17.5
1500	[illegible]	[illegible]	[illegible]	[illegible]	4	200	21.5	180	2	150	17.5
1600	200	300	30	480	4	200	21.5	180	2	150	17.5
1700	200	300	30	480	4	200	21.5	180	2	175	19.5
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
[illegible]	[illegible]	350	34.5	540	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	400	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

B SERIES ... ①

Tabel 2 ORDINATES

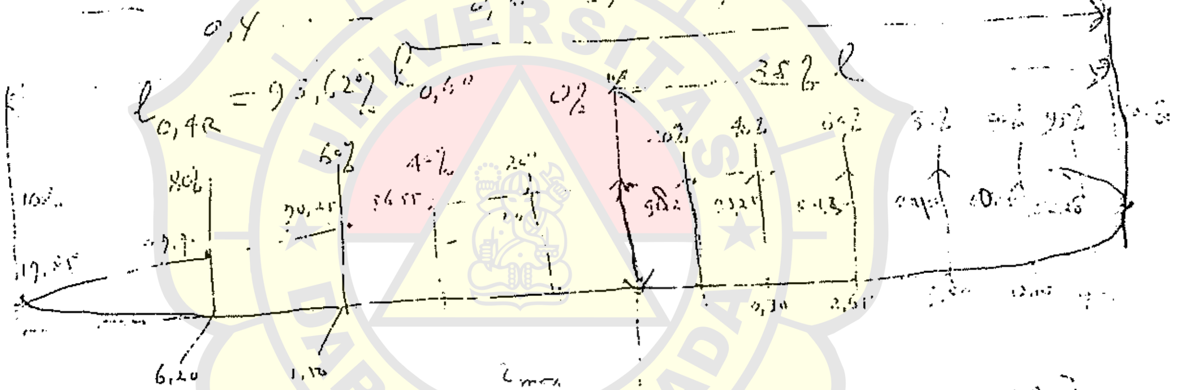
Ordinates from maximum thickness

r/R from max thickness TRAILING edge

		100%	80%	60%	40%	20%
Back	0,4	-	41,90	70,25	86,15	[illegible]
Face	0,4	17,85	[illegible]	1,50	-	-

% from max thickness to LEADING edge

		20%	40%	60%	80%	90%	95%	100%
Back	0,4	95,20	[illegible]	84,50	70,40	60,[illegible]	52,[illegible]	-
Face	0,4	-	0,[illegible]	2,65	7,80	12,10	17,[illegible]	28,[illegible]



$l_{0,6R} = 0,2187 D$

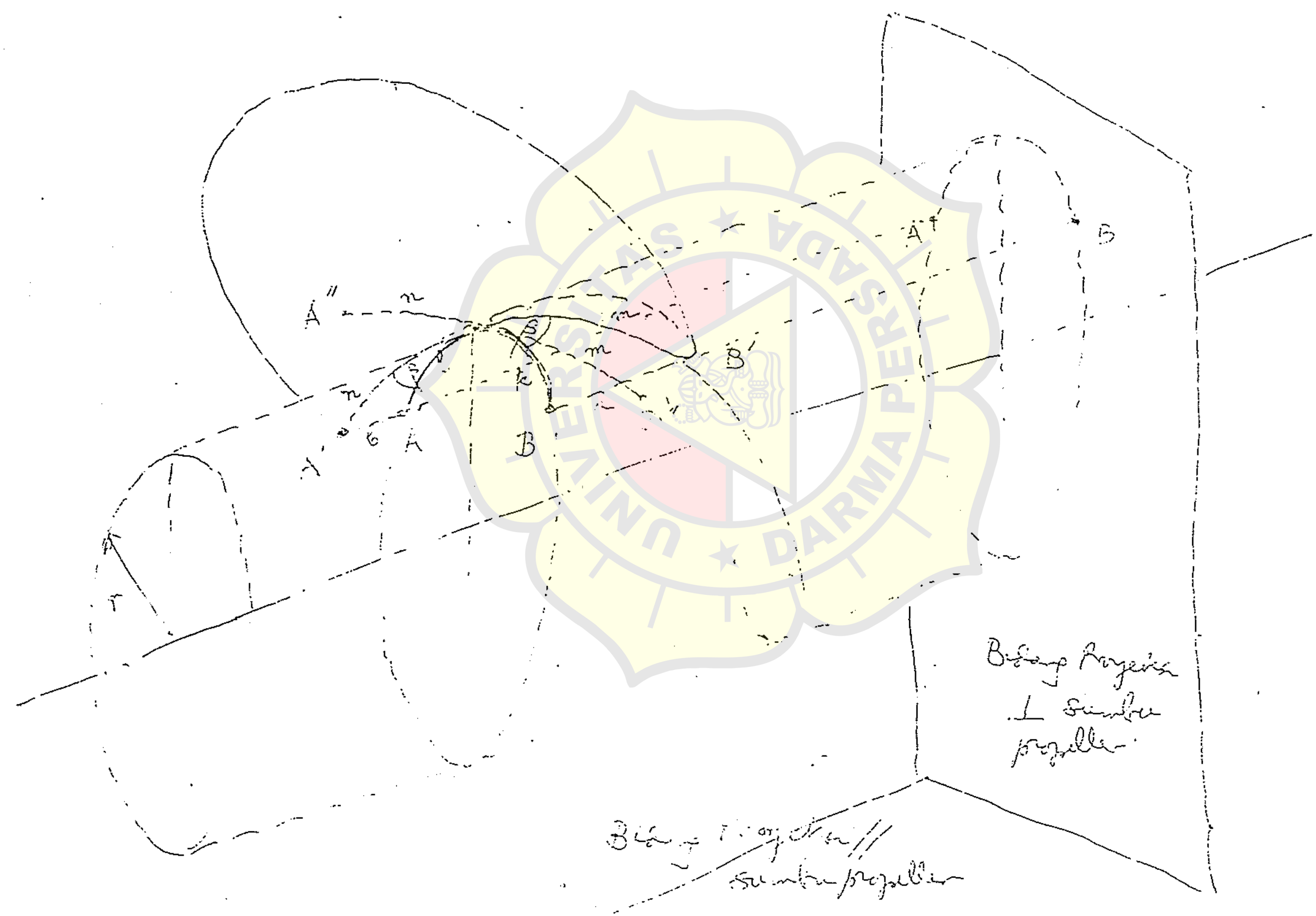
Tabel Dimension B4-40

r/R		0,20	0,30	0,40	0,50	0,60	0,70	0,80	0,90	1,0
l : length of the blade section % max. length of the blade section at 0,6R ; $l_{0,6R} = 0,2187 D$	from Center Line to TRAILING edge			37,30		43,92				
	from Center Line to LEADING edge			56,32		56,08				
	l = Total length			93,62		1,00				
				2,82		1,98				
max. thickness ratio % D				3,50		3,9				
Distance of maximum thickness from leading edge as % of the length of the section				56,32% ($2l$)						

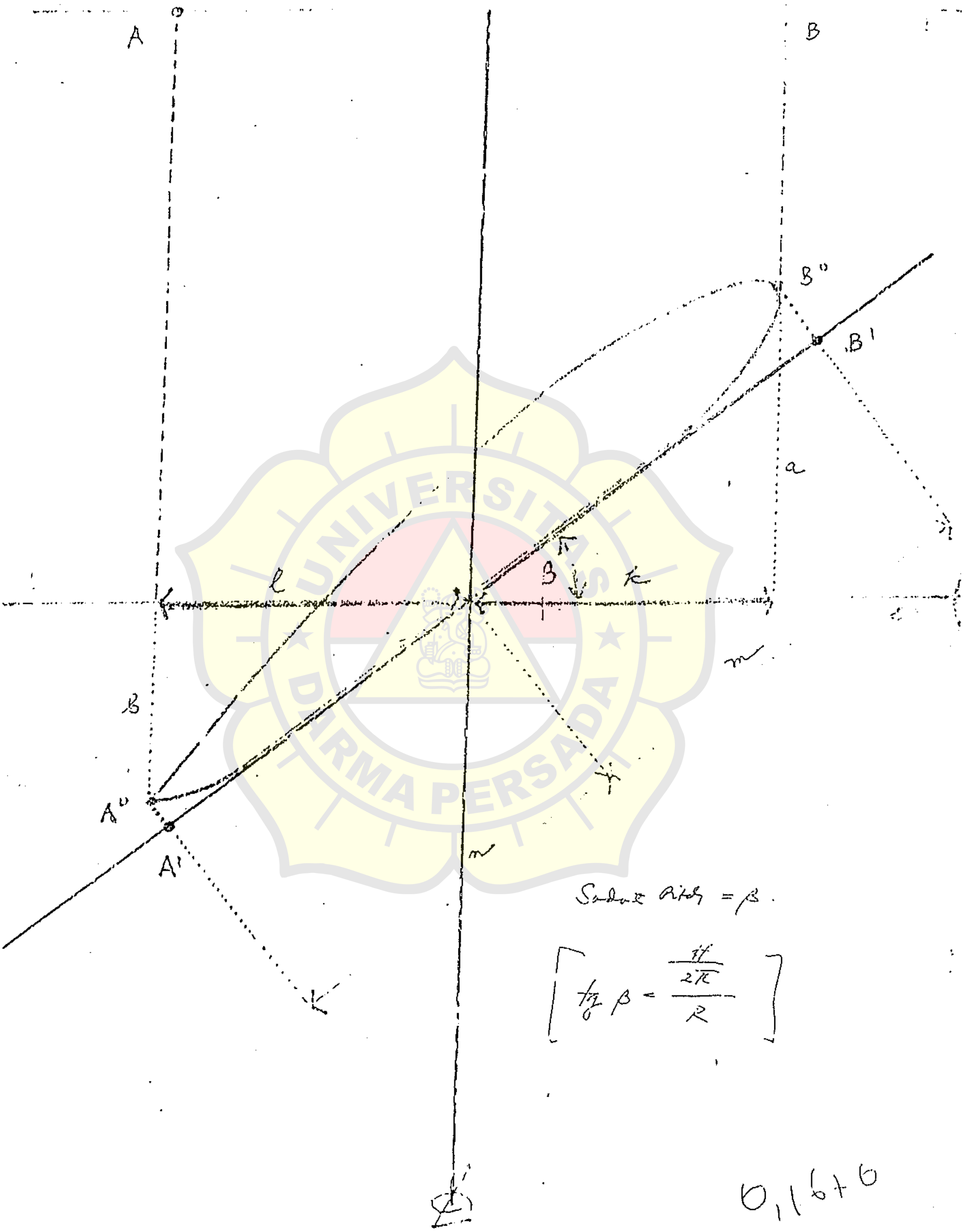
37,30% $l_{0,5R}$ — Trailing Edge

56,32% $l_{0,5R}$ — Center line

Trailing Edge



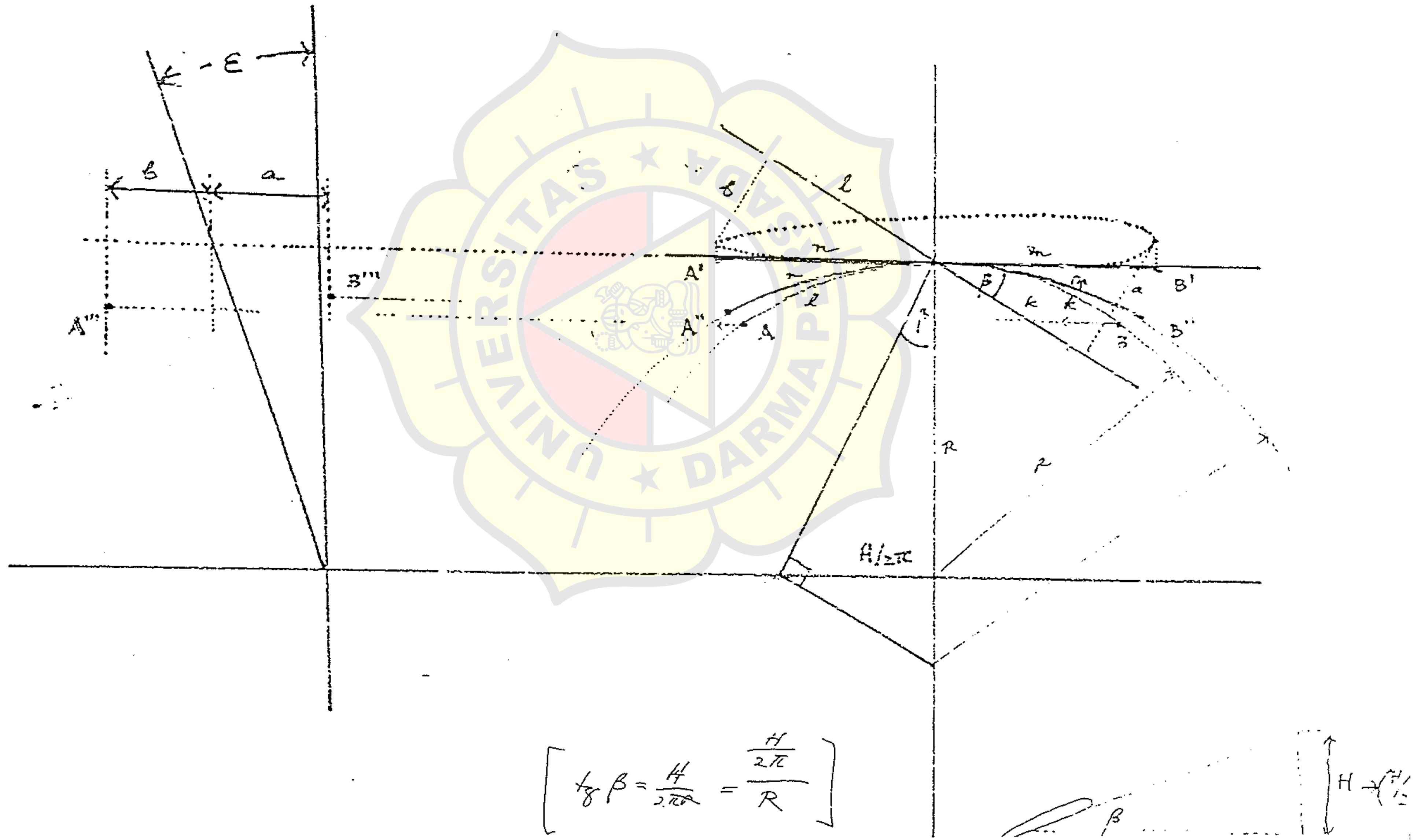
A''
n
A'
B
B'
m
k
A
B
r
Bidang Proyeksi ⊥ sumbu propeller
Bidang Proyeksi // sumbu propeller



Sudut Ritch $= \beta$

$$\left[\ tg\ \beta = \frac{\frac{H}{2\pi}}{R}\ \right]$$

0,1676



$$\left[\ tg\,\beta = \frac{H}{2\pi R} = \frac{\frac{H}{2\pi}}{R}\ \right]$$

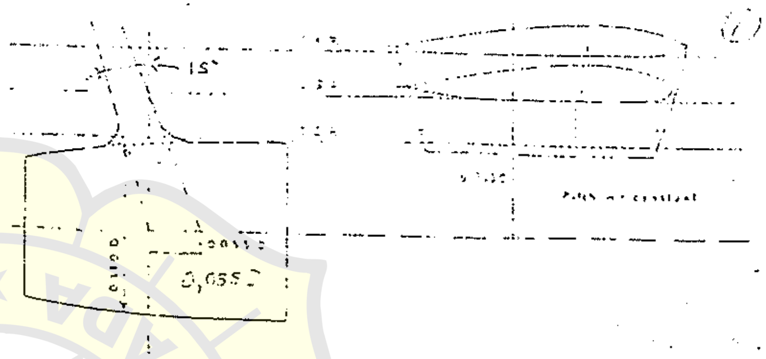


Fig. 31. General plan of the four-bladed propeller of type B 4-50

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

Distance of the ordinates from the maximum thickness

r/R	From maximum thickness to trailing edge 100%	80%	60%	40%	20%	From maximum thickness to leading edge 20%	40%	60%	80%	90%	95%	100%
						Ordinates for the back						
0.2	—	53.35	72.65	86.90	96.45	98.60	94.50	87.00	74.40	64.35	56.95	—
0.3	—	50.95	71.60	86.50	96.80	98.40	94.00	85.50	72.50	62.65	54.90	—
0.4	—	47.70	70.25	86.55	97.00	98.20	93.25	84.50	70.40	60.15	52.20	—
0.5	—	43.40	68.40	86.10	96.95	98.10	92.40	82.30	67.70	56.80	45.60	—
0.6	—	40.20	67.15	85.40	96.80	98.10	91.75	79.35	63.60	52.20	43.35	—
0.7	—	39.40	66.90	84.90	95.65	97.60	88.90	74.90	57.00	44.20	35.00	—
0.8	—	40.95	67.80	85.30	96.70	97.00	85.30	68.70	48.25	34.55	25.45	—
0.9	—	45.15	70.00	87.00	97.00	97.00	87.00	70.00	45.15	30.10	22.00	—
0.95	—	44.80	72.00	88.00	97.20	97.20	88.80	72.00	44.80	29.50	21.50	—
						Ordinates for the face						
0.2	30.00	18.20	10.90	5.45	1.55	0.45	2.30	5.90	13.45	20.30	26.20	40.00
0.3	25.35	12.20	5.80	1.70	—	0.05	1.30	4.60	10.85	16.55	22.20	37.55
0.4	17.85	6.20	1.50	—	—	—	0.30	2.60	7.80	12.50	17.90	34.50
0.5	9.70	1.75	—	—	—	—	—	0.70	4.30	8.45	13.30	30.40
0.6	5.1	—	—	—	—	—	—	—	0.80	4.45	8.40	24.50
0.7	—	—	—	—	—	—	—	—	—	0.40	2.45	16.05
0.8	—	—	—	—	—	—	—					

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

	r/R	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	
Length of the blade sections as percentages of the maximum length of the blade sections at 0.6 R.	from centre line to trailing edge	21.61	31.67	36.62	40.3[illegible]	44.35	46.97	48.27	43.46	[illegible]	Length of blade section at 0.6 R = 0.3[illegible] D if F_a/F = 0.30
	from centre line to leading edge	47.05	51.24	54.91	56.12	55.82	52.22	44.63	30.31	—	
	total length	74.73	83.91	[illegible]	97.05	100.00	99.19	92.85	73.77	—	
Blade thickness ratio as percentages of the diameter		4.46	3.94	[illegible].42	2.90	2.38	1.86	1.34	0.82	0.30	Maximum thickness at centre of shaft = 0.055 D
Distance of maximum thickness from leading edge as percentages of the length of the sections		35.00	35.00	35.00	35.10	38.90	44.20	47.80	50.00	—	

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

	r/R	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	
Length of the blade sections as percentages of the maximum length of the blade sections at 0.6 R.	from centre line to trailing edge	28.65	[illegible].67	[illegible]	40.13	[illegible]13	46.97	48.22	43.46	14.87	Length of blade section at 0.6 R = 0.[illegible]698 D if F_a/F = 0.50
	from centre line to leading edge	46.01	[illegible]21	53.91	56.52	55.82	52.22	44.63	30.31	—	
	total length	74.73	[illegible].91	[illegible]	97.05	100.0	99.19	92.85	[illegible]7	—	
Blade thickness ratio as percentages of the diameter		4.06	[illegible]59	3.12	2.65	2.18	1.71	1.24	0.77	0.30	Maximum thickness at centre of shaft 0.05 D
Distance of maximum thickness from leading edge as percentages of the length of the sections		35.0	35.0	35.0	35.5	38.9	44.2	47.8	50.0	—	

TABLE 5. *Dimensions of the four-bladed screws, types B 4.40, B 4.55 and B 4.70* → 70 | 40 70, ue

Max is c

	r/R	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0	
Length of the blade sections as percentages of the maximum length of the blade sections at 0.6 R.	from centre line to trailing edge	29.18	3[illegible].32	37.30	40.78	43.92	46.68	45.35	47.00	20.14	Length of blade section at 0.6 R = 0.2187 D if F_a/F = 0.40
	from centre line to leading edge	46.9	52.64	56.32	57.60	55.08	51.40	41.61	25.35	—	
	total length	76.08	85.96	93.62	98.35	100.0	98.08	90.00	72.35	—	
Blade thickness ratio as percentages of the diameter		3.66	3.24	2.82	2.40	[illegible].98	1.56	1.14	0.72	0.30	Maximum thickness at centre of shaf 0.045 D
Distance of maximum thickness from leading edge as percentages of the length of the sections		35.0	35.0	35.0	35.5	38.9	44.3	47.9	50.0	—	

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

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

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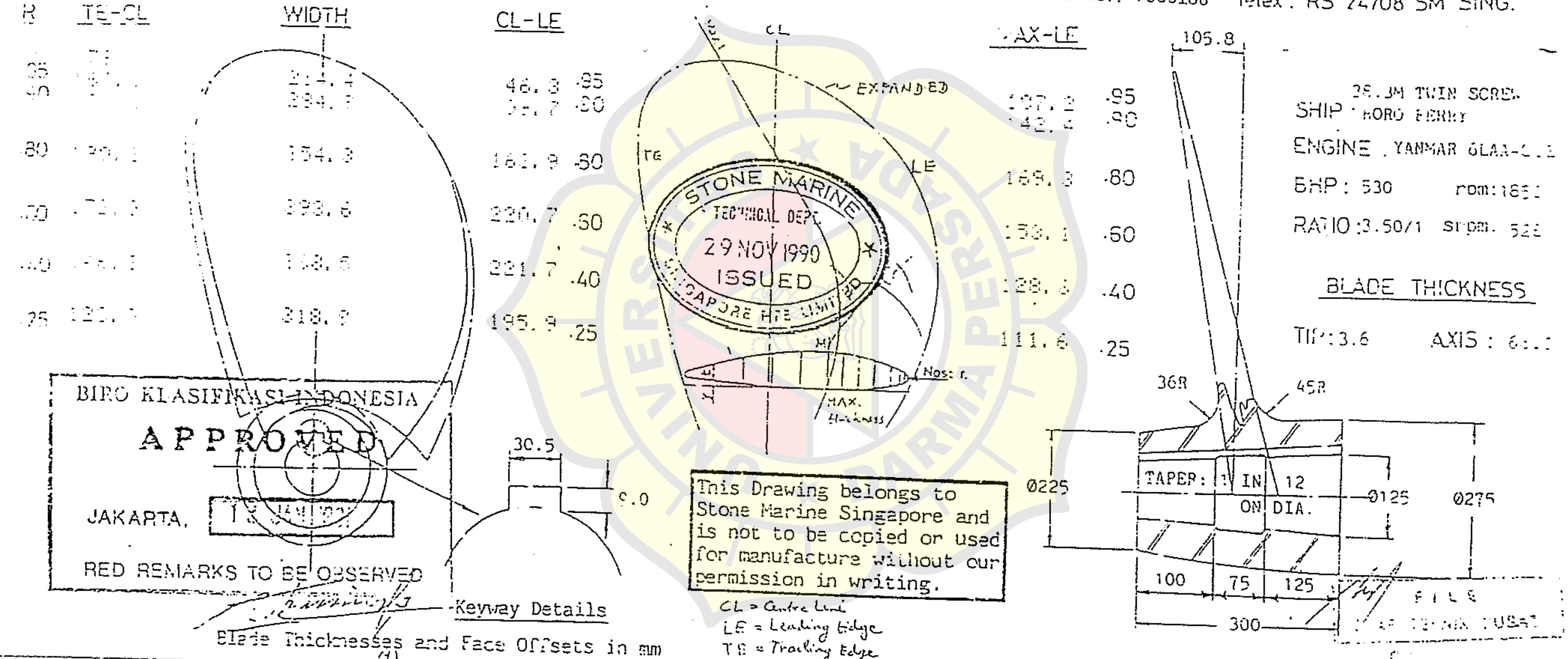

Blade Thicknesses and Face Offsets in mm

R	100	80	60	40	20	0	20	40	60	80	90	95	100	Nose r
.95	0.0	3.0	4.8	6.0	6.5	6.7	6.5	6.0	4.8	3.0	2.0	1.5	0.0	
.90	0.0	4.4	6.9	8.6	9.5	9.8	9.5	8.6	6.9	4.4	3.0	2.2	0.0	
.80	0.0	6.6	10.9	13.7	15.5	16.1	15.6	13.9	11.2	[illegible]	5.1	3.6	0.0	
.60	0.0	11.5	19.2	24.4	27.6	28.6	28.0	25.9	21.5	14.8	9.8	6.7	0.0	
y	1.5	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.1	0.5	2.9	1.3
.40	0.0	17.0	25.2	35.5	39.8	41.0	40.0	37.0	31.7	21.7	15.9	11.2	0.0	
y	7.3	2.5	0.6	0.0	0.0	0.0	0.0	0.1	0.7	2.4	[illegible]	[illegible]	[illegible]	3.3
.25	0.0	18.6	32.1	42	48	50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	

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

DIAMETER	1200 mm
PITCH	1090 mm
B.A.R.	0.60
BLADES	4RH & 4LH
MATERIAL	NI BRONZE

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
c	[-]	Coefficient for shrunk joints = 1,0 for engine and turbine gear transmissions = 12 for direct drives
C_G	[-]	Size factor in accordance wit formula (2)
C_{Dyn}	[-]	Dynamic factor in accordanc with formula (3)
C_w	[-]	Characteristic value fo propeller material as shown i Table 6.1 (corresponds to th minimum tensile strength of the propeller material whe

[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 $= R \cdot \tan \varepsilon$
Sudut.		
E_F	[-]	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.

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 spe[illegible] 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

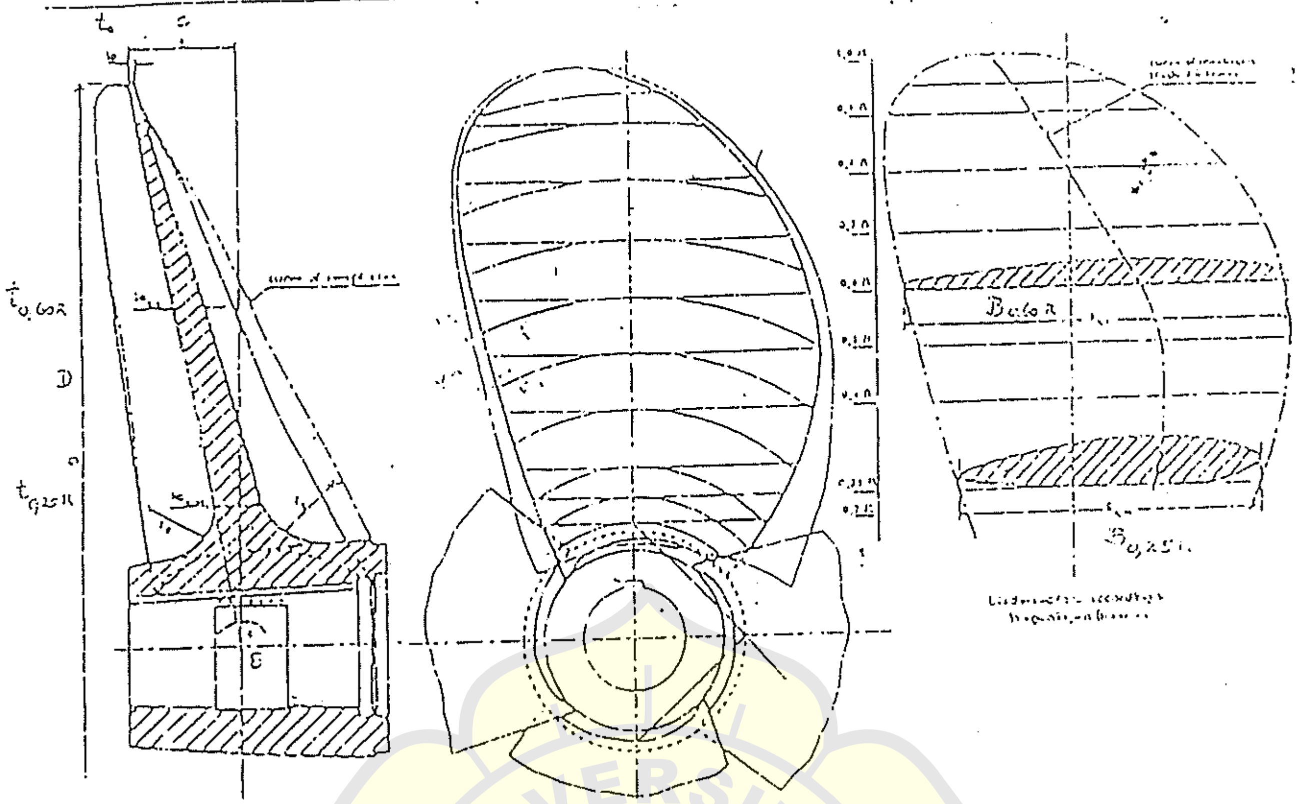

Fig. 6.1 Blade sections

T_K	[Nm]	Impact moment
V_s	[kn]	Speed of ship
w	[-]	Wake factor
W_x	[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

$$\alpha_{0,25} = \text{arc tan}\ \frac{1,27 \cdot H}{D}$$

$$\alpha_{0,35} = \text{arc tan}\ \frac{0,91 \cdot H}{D}$$

$$\alpha_{0,60} = \text{arc tan}\ \frac{0,53 \cdot H}{D}$$

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 generatrix 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).

$$\boxed{t = K_0 \cdot k \cdot K_1 \cdot C_G \cdot C_{Dyn}} \qquad (1)$$

$$K_0 = 1 + \frac{e \cdot \cos\alpha}{H} + \frac{n}{15000}$$

k as in Table 6.2 PITCH (m)

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

C_G [-] Size factor

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

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

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

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

σ_{max}/σ_m can be roughly calculated from the thrust-stimulating factor E_r 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".)

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

$$E_r = \frac{\hat{\delta}_{KT}}{\delta_J} \cdot \frac{J}{K_T} \qquad (5)$$

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

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:

$$k' = k \cdot \sqrt{\frac{\beta_1}{\beta'_1}} \text{ with } \beta'_1 = \frac{W_1}{t^2 \cdot B} \qquad (6)$$

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 submitted 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, $\beta_1 = 0,12$	73	62	44
Segmental profiles with parabolic back, $\beta_1 = 0,11$	77	66	47
Blade profiles as for Wageningen B Series propellers where $\beta_{1,0,25} = 0,10$ $\beta_{1,0,35} = 0,11$ $\beta_{1,0,60} = 0,12$	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 $\leq 4 \cdot t$.

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.

$$T_M = \frac{0,65 \cdot 10^6 \cdot R_{p0,2} \cdot P_w \cdot L_M \cdot C_G^2}{n \cdot z \cdot C_w \cdot D} \quad (7)$$

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):

$$d_k = 1,58 \cdot \sqrt{\frac{\alpha_A \cdot F_M}{R_{eH}}} \quad (8)$$

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

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

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 1-1, Yaesu 2-chome, Chuo-ku, Tokyo 104, Japan Telex: (222-4733 Yanmar [illegible]										
S165 series	4	6L	[illegible]65	210	8.40-9.45	1200-1350		117-441	5.57-14.65	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	197
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	188
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

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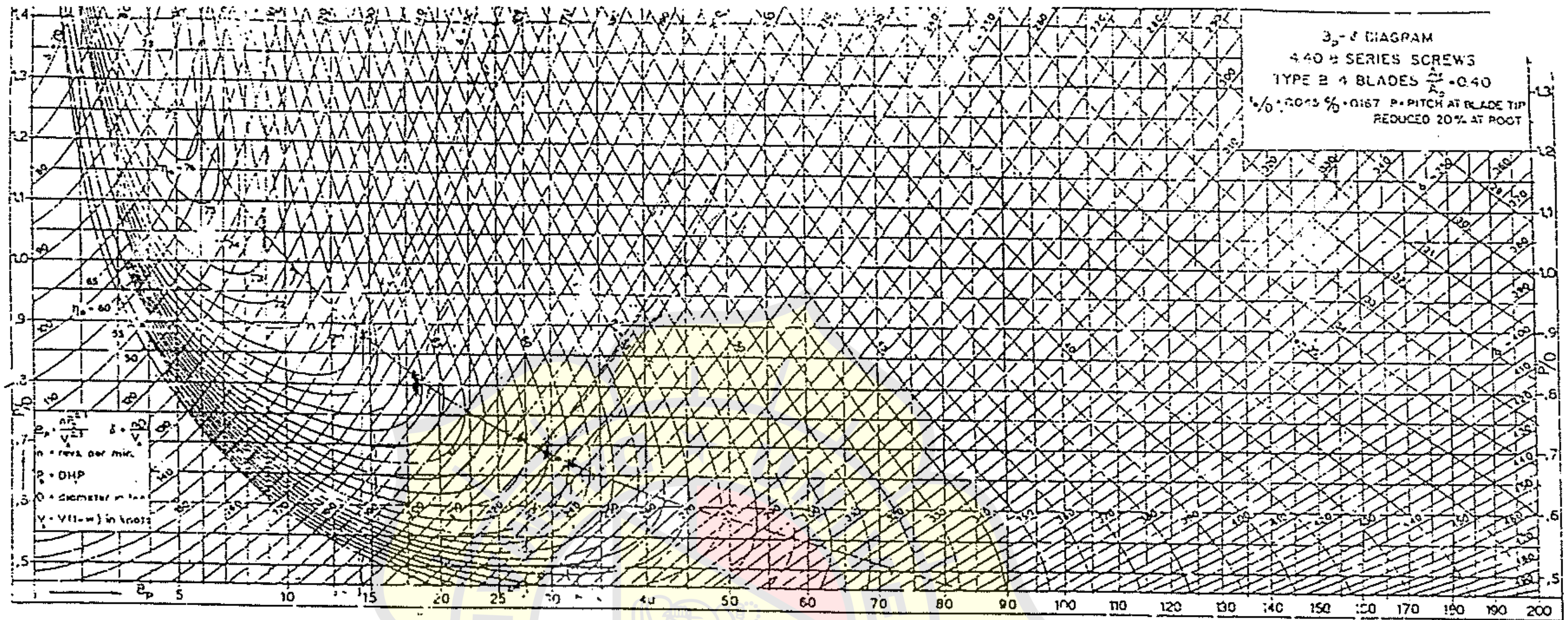
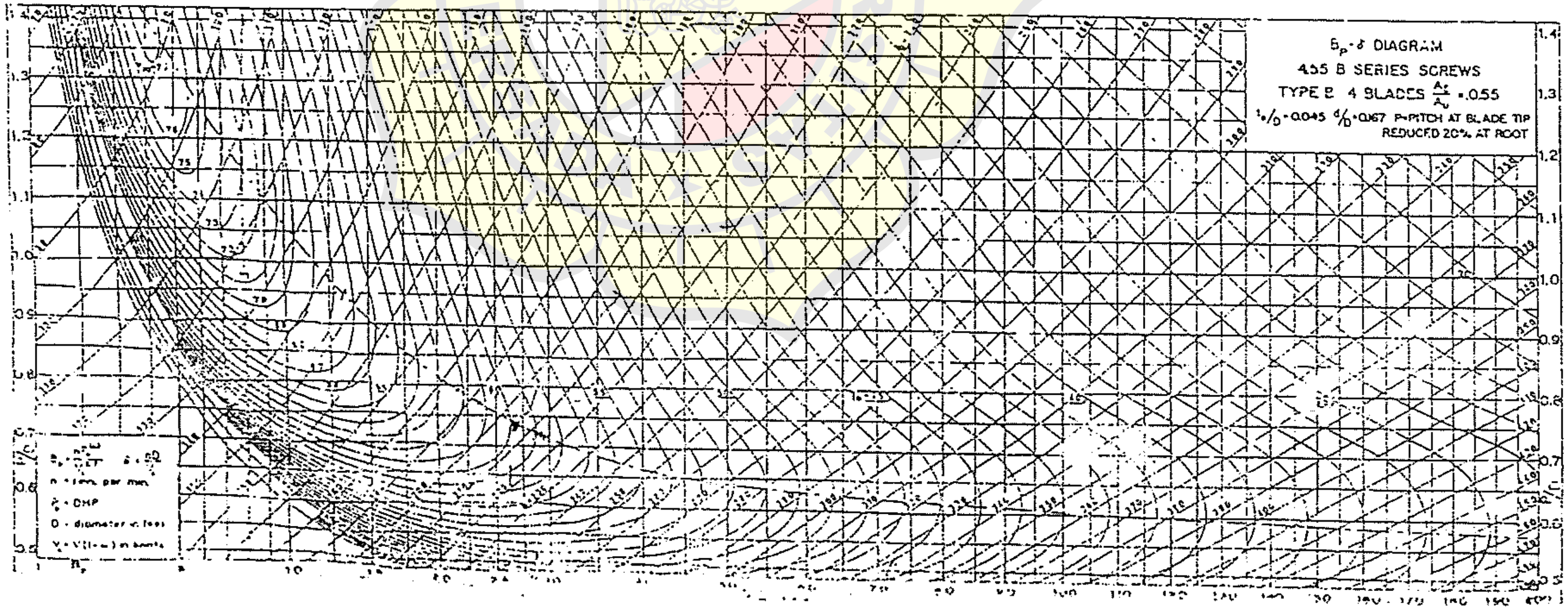


B_p-δ DIAGRAM
4.40 B SERIES SCREWS
TYPE B 4 BLADES $\frac{A_E}{A_O}$ = 0.40
t/D = 0.045 d/D = 0.167 P = PITCH AT BLADE TIP
REDUCED 20% AT ROOT
$B_p = \frac{nP^{0.5}}{V_a^{2.5}}$ $\delta = \frac{nD}{V_a}$
n = revs. per min.
P = DHP
D = diameter in feet
$V_a = V(1-w)$ in knots
B_p
P/D

Fig. 115



B_p-δ DIAGRAM
4.55 B SERIES SCREWS
TYPE B 4 BLADES $\frac{A_E}{A_O}$ = 0.55
t/D = 0.045 d/D = 0.167 P = PITCH AT BLADE TIP
REDUCED 20% AT ROOT
D = diameter in feet
P/D

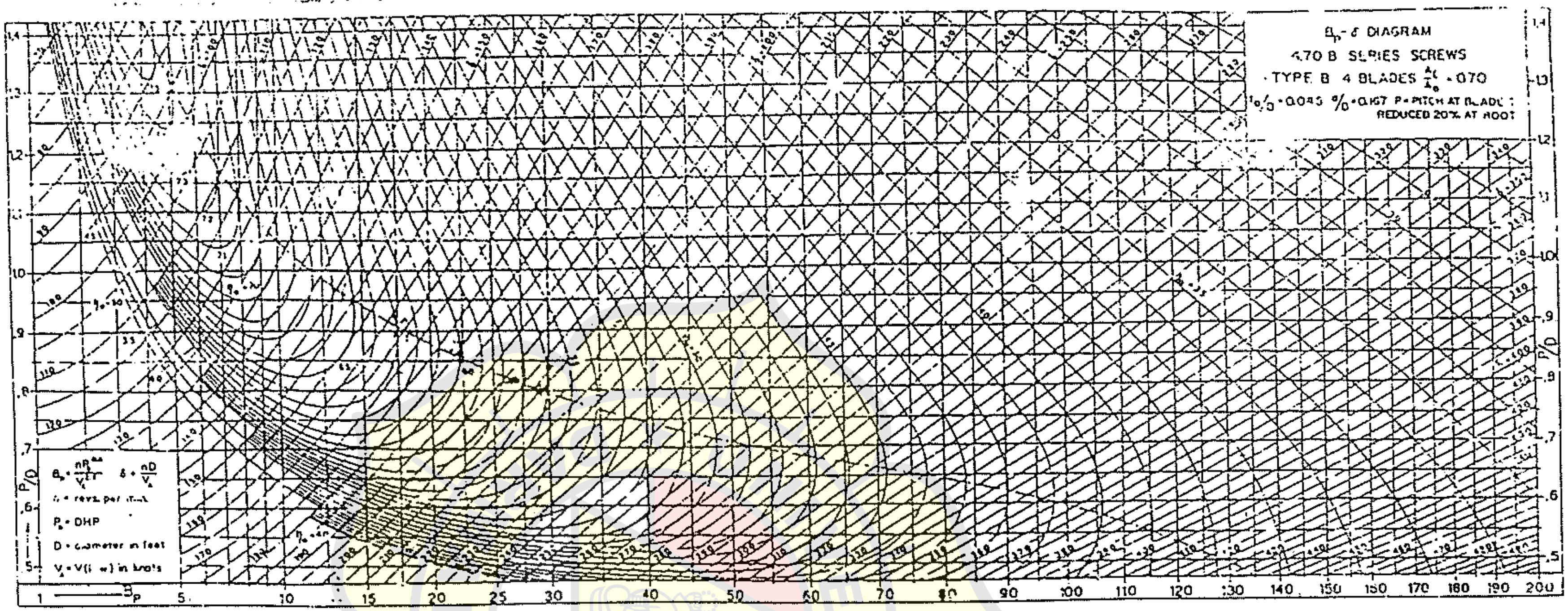


Fig. 117

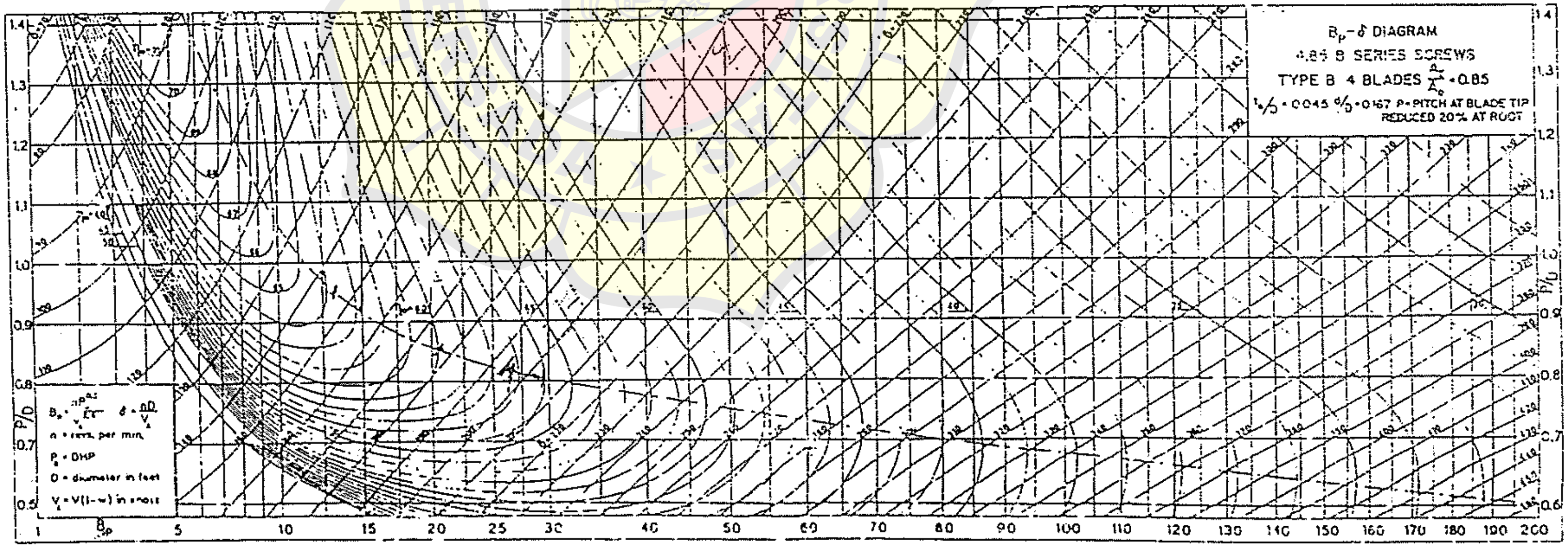


Fig. 118

... power 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).

1 HP (horsepower) = 550 foot pounds per second = 0.746 kilowatt = 1.014 PS (or Cheval Vapeur)

1 PS (Pferdestärke) or CV (Cheval Vapeur) = 542.5 foot pounds per second = 0.986 hp = 0.736 kw

1 kw (kilowatt) = 1000 watts = 1.340 hp = 1.359 PS or CV = 737 foot-pounds/s

1 long ton = 1.016 t, 1 short ton = 0.907 t

Motors:

Specific fuel oil consumption:

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

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

b = 196 ... 209 [g/KW·h]

for full power: addition 5%

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

For steam turbines:

Standard circulation without furnace gas reheat

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

b = 270 ... 286 [g/KW·h]

with furnace gas reheat

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

b = 252 ... 265 [g/KW·h]

For gas turbines:

Gasoline and light crude oils

b = 299 ... 312 [g/KW·h]

Specific weight of heavy fuel oil: γ = 0.95 t/m³

Required volume of storage tanks

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

1 kw = 0,736 PS

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

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.

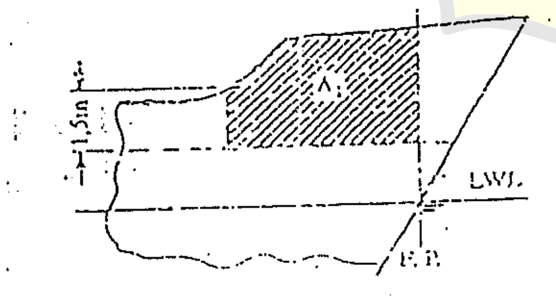


Fig. 18.1

Anchors

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. three bower anchors are required the third anchor 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

... ision of spare anchor may need ...

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 value 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 approximately the same mass.

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

Table 38

Density, kg/m³	Absolute humidity, g/m³	Vapour pressure, mm Hg	Temperature, °C	Density, kg/m³	Absolute humidity, g/m³	Vapour pressure, mm Hg
1,424	0.64	0,540	+13	1,235	11.32	11.162
1,418	0.71	0.600	+14	1,230	12.03	11.908
1,406	0.86	0,745	+16	1,222	13.59	13.536
1,401	0.95	0,825	+17	1,217	14.43	14.421
1,395	1.05	0,910	+18	1,213	15.31	15.357
1,390	1.15	1,000	+19	1,209	16.25	16.346
1,384	1.25	1,095	+20	1,205	17.22	17.391
1,379	1.35	1,190	+21	1,201	18.25	18.495
1,374	1.46	1,290	+22	1,197	19.33	19.659
1,368	1.58	1,400	+23	1,193	20.48	20.888
1,363	1.70	1,520	+24	1,189	21.68	22.184
1,358	1.83	1,635	+25	1,185	22.93	23.550
1,353	1.98	1,760	+26	1,181	24.24	24.988
1,347	2.14	1,930	+27	1,177	25.64	26.505
1,342	2.31	2,093	+28	1,173	27.09	28.101
1,337	2.49	2,267	+29	1,169	28.62	29.782
1,332	2.69	2,455	+30	1,165	30.21	31.548
1,327	2.90	2,658	+31	1,161	31.89	33.406
1,322	3.13	2,876	+32	1,157	33.64	35.350
1,317	3.37	3,113	+33	1,154	35.48	37.411
1,312	3.64	3,368	+34	1,149	37.40	39.565
1,308	3.92	3,644	+35	1,146	39.41	41.827
1,303	4.22	3,941	+36	1,142	41.51	44.201
1,298	4.55	4,263	+37	1,139	43.71	46.691
1,293	4.89	4,600	+38	1,135	46.00	49.302
1,288	5.23	4,940	+39	1,131	48.40	52.039
1,284	5.60	5,302	+40	1,128	50.91	54.906
1,279	5.98	5,687	+41	1,124	53.52	57.910
1,275	6.39	6,097	+42	1,121	56.25	61.055
1,270	6.82	6,534	+43	1,117	59.09	64.346
1,265	7.28	6,998	+44	1,114	62.05	67.790
1,261	7.76	7,492	+45	1,110	65.14	71.391
1,256	8.26	8,017	+46	1,107	68.36	75.158
1,252	8.82	8,574	+47	1,103	71.73	79.093
1,247	9.39	9,165	+48	1,100	75.22	83.204
1,243	10.01	9,792	+49	1,096	78.86	83.499
1,239	10.64	10,457	+50	1,093	82.63	91.932

Table 39

Locality	Data on fresh air and sea water: Warmest period of navigation — Temperature of outside air t_{fn}, °C	Water temperature t_{w}, °C	Relative humidity of outside air φ_{fn}, %	Coldest period of navigation — Temperature of outside air t_{fn}, °C	Water temperature t_{w}, °C	Relative humidity of outside air φ_{fn}, %
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 −15 to −20	0 to −2 4	80 to 85 75 to 85
Warm seas	25 to 30	20 to 25	55 to 55	20	27	70
Tropical seas	30	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	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 60
Passageways of living and service accommodations	15	$t_{fa}+5$°C	75 to 85
Bath- and shower-rooms	25	$t_{fa}+10$°C	70
Cloak-rooms	20	$t_{fa}+5$°C	70 to 80
Wash-rooms and laundries	15	$t_{fa}+5$°C	50
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

= sudut antara letak bidang sayap kemudi dengan bidang yang sejajar dengan bidang simetri didalam bujur kapal. Pada saat kemudi berada ditengah-tengah.

... kemudi tunggal yang dipasang pada bidang bujur kapal ... vertikal dan horisontal sama dengan ...

... kemudi.

... ketentuan "... Veritas" 1974 luas kemudi dirumuskan ...

$$F = \frac{T.L}{100} \left(1 + 25 \left(\frac{B}{L} \right) \right) \; m^2$$

dimana : T = sarat air (m)

L = panjang kapal antara garis tegak atau 0,96 LWL jika angka ini lebih besar (m).

B = lebar kapal (m).

... catatan :

Kemudi yang tak bekerja langsung dibelakang baling-baling ditambah dengan 30% dari ketentuan di atas. Untuk kapal-kapal kemudi kembar dianjurkan, jumlah luas ... jadi 3% LT.

Untuk pengontrolan dapat dipakai pedoman batas-batas : menurut G.W. Sabolev

$$\frac{0{,}025}{\sqrt[3]{\frac{L_1}{\delta B} - 6{,}2}} < \frac{F}{L_1 T} < \frac{0{,}03}{\sqrt[3]{\frac{L_1}{\delta B} - 72}}$$

dimana : B = lebar kapal

δ = koefisien blok

L_1 = panjang kapal

= 0,95 LWL.

Luas daun kemudi dapat pula dinyatakan dalam % LT sebagai berikut :

Type Kapal	% LT
1. Kapal barang, single screw dengan kecepatan sedang.	1,5 - 2,5
2. Kapal barang, single screw dengan kecepatan tinggi.	1,6 - 2,0
3. Kapal barang kecil, single screw.	2,0 - 2,5
4. Kapal barang, twin screw, single rudder.	1,5 - 2,1
5. Kapal barang, twin screw, twin rudder.	2,1 - 3,0
6. Kapal tangker ukuran sedang.	1,3 - 1,9
7. Super tangker.	1,7 - 2,1
8. Kapal penumpang, kecepatan tinggi (L 60 m).	1,2 - 1,7
9. Kapal penumpang & barang besar kecepatan sedang.	1,6 - 2,0
10. Kapal penumpang ukuran sedang, kecepatan tinggi.	1,7 - 2,0
11. Kapal penumpang ukuran kecil kecepatan lambat.	1,7 - 2,3
12. Kapal pelayaran pantai (*coaster*).	2,0 - 3,3
13. Kapal ikan.	2,5 - 5,5
14. Kapal tunda.	3,0 - 6,0
15. Kapal layar besar.	2,0 - 2,5
16. Kapal layar sedang.	2,0 - 3,0
17. Kapal pandu.	2,3 - 4,0
18. Kapal kecil.	4,0 - 4,5
19. Kapal tak bermotor.	4,0 - 5,0

Bentuk sayap kemudi diperhitungkan menurut bentuk bagian belakang kapal (cruiser stern, biasa dan lain-lain dan ukuran bentuk sepatu linggi).

Umumnya pada teknologi pembuatan kapal dipilih bentuk sayap yang sederhana, empat persegi, tetapi untuk mendapat gaya tekan air yang maximum pada sayap kemudi, kadang-kadang dibagian atas dibuat miring membesar.

Untuk kapal-kapal yang mempunyai satu baling-baling dimana bentuk bagian belakang yang agak runcing, biasanya memakai kemudi yang setengah menggantung dengan bentuk trapesium termasuk rongga porosnya, dengan lebar bagian bawahnya kecil dengan demikian juga tebal profilnya makin ke bawah makin berkurang.

(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-lore 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

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

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

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

$V_{cc} \approx 0.3$ = 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

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

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

t_r = given temperature of the room, °C

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

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

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

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

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

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

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

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 20

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.

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

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).

Table 41

Source of heat	Heat emitted into surroundings, Q_t, kcal/h	Notation
Steam boilers	(0.03 to 0.05) $G_f Q_l$	Q_f = total fuel consumption in boiler, k[illegible] 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 valu[e] of the fuel, kc[al] per kg
Auxiliary machinery	(0.02 to 0.03) $G_s \Delta i$	
Steam lines	0.01 $G_s \Delta i$	G_s = steam consumptio[n] kg per hour
Internal combustion engines	0.02 $N_e g_e Q_l$	Δi = useful heat dro[p] 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_c = mean current den[]sity per sq mm c[]conductor cross sec[]tion, A per sq m[m]
Lighting fixtures	864 N	
Wires, bus bars, cables and fittings	2,160 i_c^2	η = efficiency of elec[]trical 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 hea[t] transmission c[] room walls, kcal pe[r] hour per sq m pe[r] deg C; F = area of the roo[m] walls, sq m; Δt = difference in tem[]peratures of roo[m] wall surfaces an[d] external surfaces o[f] 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 [illegible]

Table 42

Compartment	Number of air renewals per hour for: Plenum ventilation	Exhaust ventilation
[illegible]	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

$p_{st}=760$ mm Hg, 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

[illegible]

whence

$$Q_{st} = [illegible] \; Q_a \left(\frac{1+\frac{1}{273}\,20}{1+\frac{1}{273}\,t_a}\right)\frac{p_a}{760} = [illegible] \text{ per hour} \qquad (276)$$

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

$$H_{t\infty} = \frac{1}{g}(c_{2u}u_2 - c_{1u}u_1) = \frac{1{,}000}{g}\frac{\gamma_{air}}{\gamma_{wat}}(c_{2u}u_2 - c_{1u}u_1) = \rho(c_{2u}u_2 - c_{1u}u_1) \text{ mm } H_2O \qquad (277)$$

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

$\gamma_{wat}=1{,}000$ = density of water, kg per cu m

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

Upon radial entry of the air onto the fan impeller vanes

$$H_{t\infty} = \rho c_{2u}u_2 \text{ mm } H_2O$$

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

$$H = H_{t\infty}\sigma\eta_h = \sigma\rho c_{2u}u_2\eta_h = \sigma\rho\frac{c_{2u}}{u_2}u_2u_2\eta_h = \sigma\rho\varphi_h u_2^2\eta_h = \rho\psi_h u_2^2 \text{ mm } H_2O \qquad (278)$$

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 [illegible]

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

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

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

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

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

where ΔH = loss of head in the fan.

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

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

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—$\eta_{fr} \cong 0.75$ to 0.9.

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

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

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

$$\eta_t = \eta_h \eta_{fr} \eta_m = 0.4 \text{ to } 0.75 \tag{279}$$

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

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

$$H = H_{st} + H_{dyn} = H_{st} + \frac{v^2}{2g} \times 10^{-3}\ \text{mm H}_2\text{O} \tag{280}$$

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 mm H$_2$O, and the impeller speed, n, at maximum efficiency:

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

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

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

$$H_{st} = \overline{H}_{kst} u_2^2 \rho \text{ mm H}_2\text{O} \tag{282}$$

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 counterclockwise rotation of the rudder can be followed out in Fig. 158.

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

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

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

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

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

Table 47

	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
Passenger 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

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

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

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:

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

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

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

It follows from formula (314) that

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

Combining equations (315) and (316) we obtain

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

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

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

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

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

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

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

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

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

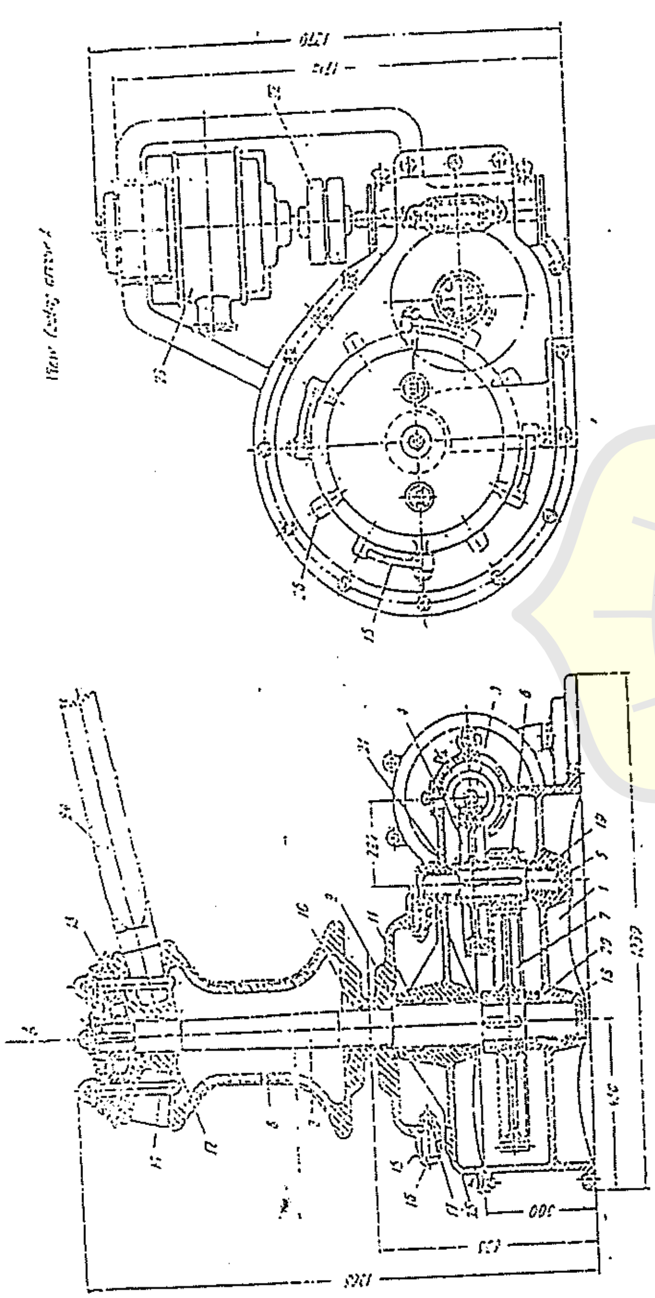


Fig. 169.

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

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

in hoisting one anchor

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

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

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

According to the U.S.S.R. Shipping Register the all 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

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

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

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

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:

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

$$\chi_q = l_q h_q$$

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

A section taken through the central plane of the usual five-snug

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	—	6.3
2	75	2	200	25	125	13	50	—	8.8
3	100	2	250	50	125	15	50	—	11
4	150	2	300	50	150	16	50	—	11
5	200	2	350	50	175	17	75	—	11
6	250	2	450	75	200	18	75	11	13
7	300	2	500	75	225	19	75	13	13
8	350	2	600	100	250	20	75	14	15.5
9	400	2	700	100	275	21	75	14	15.5
10	450	2	750	125	300	22	100	15	17.5
11	500	2	800	150	300	24	100	16	17.5
12	550	2	900	175	325	25	100	16	17.5
13	600	3	1500	200	350	27	100	17	17.5
14	650	3	1700	225	350	28	100	18	19.5
15	700	3	1800	250	375	29	100	18	20.5
16	750	3	2100	250	375	30	100	19	20.5
17	800	3	2250	250	375	31	125	19	20.5
18	850	3	2400	275	375	32	125	20	22
19	900	3	2700	300	375	33	125	21	24
20	950	3	3000	300	400	34	125	21	24
21	1000	3	3200	350	400	36	125	22	24
22	1100	3	3500	400	400	37	125	23	26
23	1200	3	3750	400	420	38	150	25	26
24	1300	3	4100	450	450	40	150	25	28
25	1400	3	4250	450	450	41	150	25	28
26	1500	3	4500	500	450	42	150	26	28
27	1600	3	4750	500	450	43	150	26	28
28	1700	3	5250	600	450	45	150	26	30
29	1850	3	5500	600	450	46	150	28	30
30	2000	3	5750	700	450	46	150	29	31.5
31	2150	3	6000	700	475	48	175	29	31.5
32	2300	3	6500	800	500	49	175	29	32.5
33	2500	3	6750	800	500	50	175	29	32.5
34	2700	3	7500	900	500	52	175	30	33.5

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	34.5
38	3900	3	10500	1250	550	59	225	33	34.5
39	4200	3	11000	1400	550	61	225	34	37
40	4500	3	11500	1500	550	62	225	35	37
41	4800	3	12900	1650	550	65	225	36	—
42	5100	3	13500	1750	550	67	250	37	—
43	5400	3	14500	1750	575	68	250	37	—
44	5800	3	15000	2000	600	70	250	40	—
45	6200	3	15800	2000	600	72	250	40	—
46	6600	3	16300	2250	600	74	275	43	—
47	7000	3	17600	2250	600	76	275	43	—
48	7400	3	18000	2250	600	77	275	44	—
49	7800	3	19500	2500	600	80	275	46	—
50	8200	3	20300	2700	600	82	275	48	—
51	8600	3	21000	2800	600	83	275	49	—
52	9000	3	22000	3000	600	85	275	50	—
53	9500	3	23000	3000	600	87	275	50	—

Note: Two bower anchors with a total weight of at least 2/3 of the tabular values 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 $8d_c$ 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

$$D_{cl} = 2R_{cl} = 2\frac{4d_c}{\sin\alpha} = \frac{8d_c}{\sin\frac{360°}{10}} = 13.6\,d_c \text{ mm} = 0.0136 d_c \text{ m} \quad (287)$$

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

$$l_c = 5l_c = 5 \times 8d_c = 40d_c \text{ mm} = 0.04d_c \text{ m} \quad (288)$$

where d_c = chain bar size, mm.

(a) for windlasses and capstans of bower anchors:

$$n_{cl}=\frac{60\,v_a}{0.04\,d_c}=\frac{60\times0.2}{0.04\,d_c}=\frac{300}{d_c}\quad \text{rpm}$$

(b) for the stern anchoring capstan:

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

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

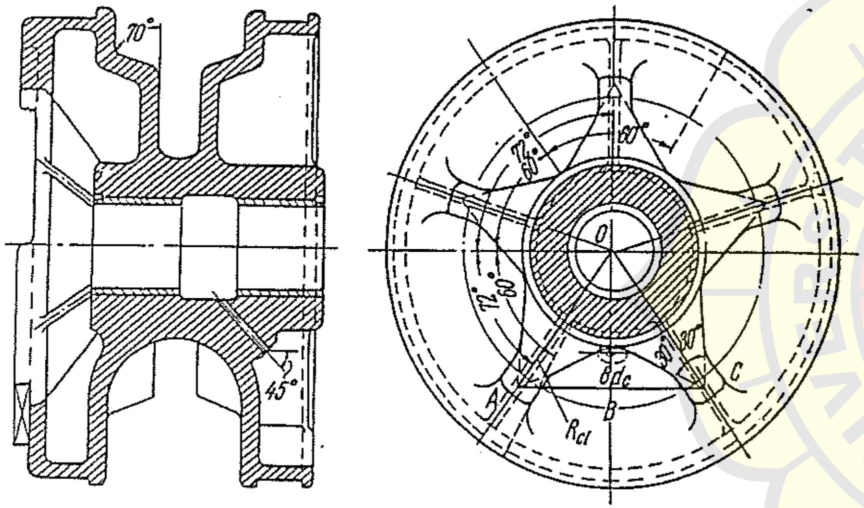


Fig. 170.

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

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

where η_{cl}, 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

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

where $\eta_{cl}=0.9$ 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

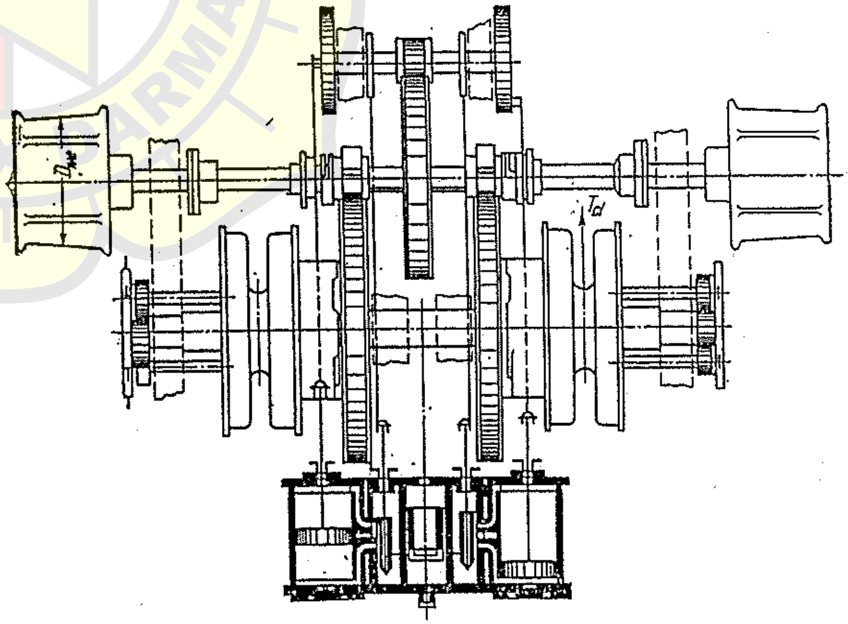
$$i_a=\frac{n_m}{n_{cl}}$$

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	180 to 320	18 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

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



The mean shaft power of the motive unit should be

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

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

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

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 $(\alpha_i k_i p_{ls} - p_{ss})$ 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

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

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

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

$$M_m = \left(\frac{D_{ca}}{1.37}\right)^3 \eta_m \psi_a (\alpha_i k_i p_{ls} - p_{ss}) \quad \text{kg-cm}$$

On the other hand, if i_a is the total gearing ratio of the transmission in the anchoring mechanism, then

$$M_m = \frac{M_{cl}}{\eta_a i_a} = \frac{T_{cl} D_{cl}}{2 \eta_a i_a} \quad \text{kg-cm}$$

Combining the last two equations we obtain

$$T_{cl} = \frac{2 M_m \eta_a i_a}{D_{cl}} = 2 \left(\frac{D_{ca}}{1.37}\right)^3 \frac{\eta_m \psi_a (\alpha_i k_i p_{ls} - p_{ss}) \eta_a i_a}{D_{cl}} =$$

$$= 0.78 \frac{D_{ca}^3}{D_{cl}} \eta_m \psi_a (\alpha_i k_i p_{ls} - p_{ss}) \eta_a i_a \quad \text{kg}$$

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

$$n_w = \frac{60 v_w}{\pi (D_{we} + d_w)} = 19.1 \frac{v_w}{D_{we} + d_w} \quad \text{rpm} \qquad (393)$$

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

$$T_{we} = \frac{M_{we}}{\frac{1}{2}(D_{we} + d_w)} = \frac{2 M_m \eta_w i_w}{D_{we} + d_w} \leqslant \frac{R_{br}}{6} \qquad (394)$$

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

$$v_w = \frac{\pi (D_{we} + d_w) n_m}{60 i_w} \quad \text{m per sec} \qquad (395)$$

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:

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

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

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

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_{pg}), barrel (η_b) and worm gearing (η_{wg}). Thus

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

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:*

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

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

where $k_l = \frac{1 + \ln\Delta}{\Delta}$ = coefficient of mean theoretical indicated pressure for a ratio of steam expansion Δ
M_{md} = torque developed on the engine shaft, kg-cm.

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

$$N_i = \frac{D_{cw}^3 (\alpha_i k_l p_{ls} - p_{ss}) n_m \Psi_r}{143{,}300} \quad \text{hp} \tag{427}$$

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

Table 62

δ	0.5	0.6	0.7	0.8	0.9	1
k_l	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

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

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

$$n_{.} = \frac{n_m}{\ } \quad \text{rpm} \tag{431}$$

consideration all parasitic resistances in the boat's falls and in the various guide blocks through which the tackle fall runs to the winch heads.

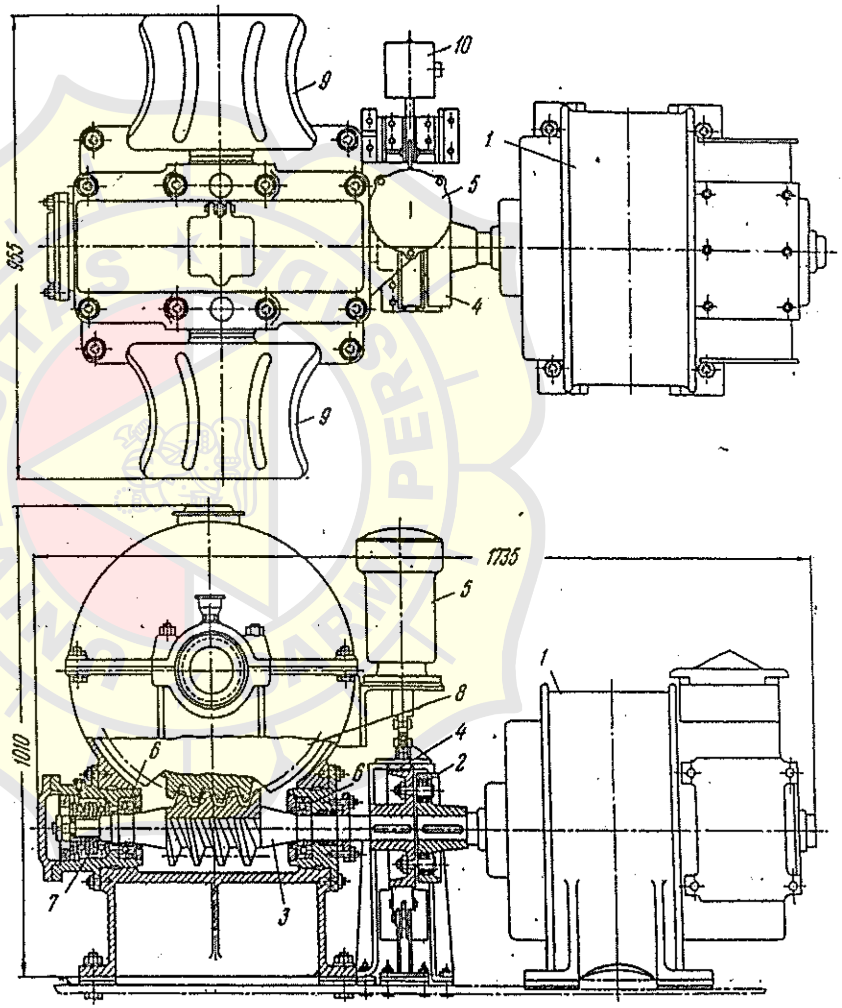


Fig. 184.

The design load Q acting on the falls of one davit is determined from the formula

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 (ε=1.1 for a hemp fall and ε=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 $(T_{max}+T_{min})\,6\leqslant R_{br}$ 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

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

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

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

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

and

$$N=\frac{M_{mb}n_m}{716.20}\ \text{metric hp}$$

The cylinder diameter and indicated power of steam boat 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:

$$V_e = V_f + D_1 = V_1 + \frac{D}{6}$$

and

$$V_f = V_1 - D_1 = V_1 - \frac{D}{6}$$

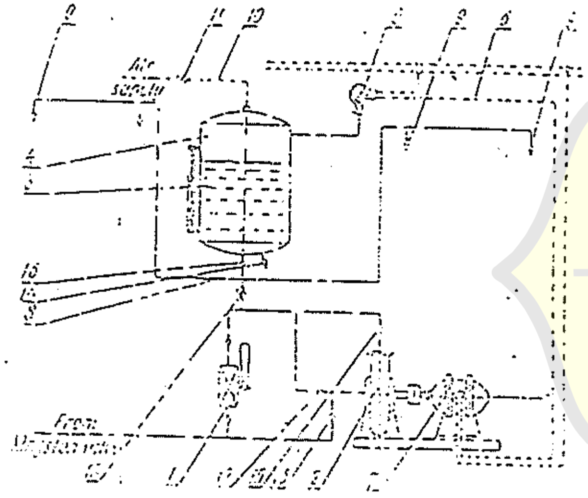


Fig. 189.

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

$$V_e p_e = V_f p_f = \left(V_1 + \frac{D}{6}\right) p_e = \left(V_1 - \frac{D}{6}\right) p_f$$

and [illegible] volumes of [illegible] are

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

Denoting [illegible] volume of liquid remaining in the tank at the lowest [illegible] that the volume [illegible] the pneumatic tank is

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

Such tanks [illegible] also be used in the drinking and washing water systems.

(D) SANITARY AND SCUPPER SYSTEMS

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

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

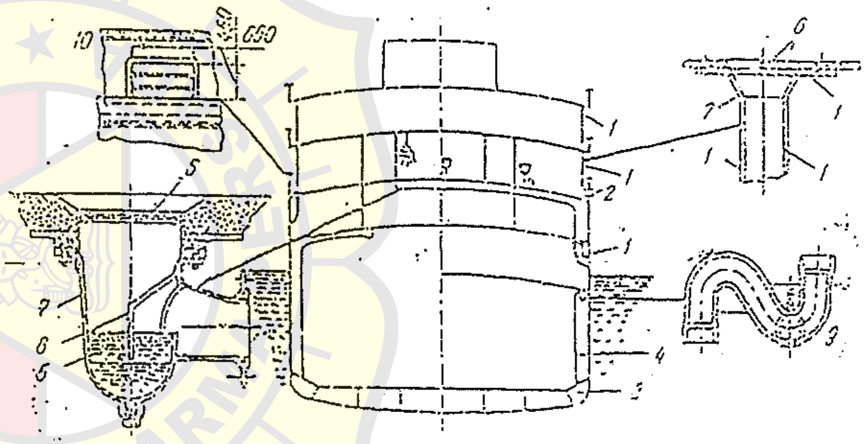


Fig. 190.

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

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

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

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

Sanitary pipelines made of [illegible]

Notes to Table 43.

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

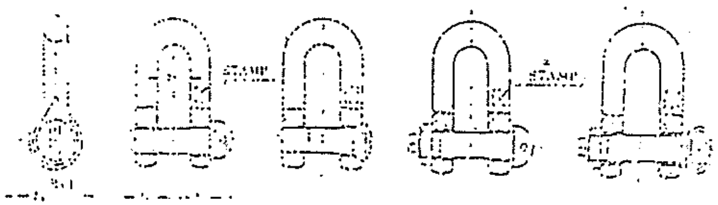
according to DIN 82101, Feb. 76

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 Ø	Pin Thread
	t	mm	mm	mm	mm	
1	1	21	13	32	16	M 16
1,6	1,6	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 20	1 to 20	25 to 100



Pemasangan sekat tubrukan pada suatu kapal sangat dibutuhkan karena sekat ini untuk menghindari mengalirnya air keruangan yang ada dibelakangnya apabila terjadi kebocoran di ceruk haluan akibat menubruk sesuatu dan dengan rusaknya ceruk haluan kapal masih selamat, tidak tenggelam.

Pemasangan sekat tubrukan menurut BKI 2004 adalah sebagai berikut:

- Kapal kargo dengan $L_c \leq 200$ m harus mempunyai sekat tubrukan yang jaraknya tidak kurang dari 0,05 L_c dari arah garis tegak haluan. Kapal kargo dengan $L_c > 200$ m dipasang sekat tubrukan sejarak > 10 m dari arah garis tegak haluan.

- Semua kapal kargo mempunyai sekat tubrukan yang ditempatkan tidak lebih dari pada 0,08 L_c dari garis tegak haluan. Jarak yang lebih besar disetujui dalam hal-hal khusus.

- Untuk kapal yang mempunyai beberapa bagian bawah air yang melewati garis tegak haluan, seperti haluan bola, jarak yang diisyaratkan seperti hal-hal diatas boleh diukur dari suatu titik referensi yang ditempatkan pada jarak x didepan garis tegak haluan dengan harga terkecil.
  Dimana : a) $x = a/2$
  b) $x = 0{,}015\ L_c$ dengan harga terbesar
  $x = 3$ m.

- Sekat tubrukan harus kedap air sampai geladak lambung timbul.

- Jika kapal mempunyai bangunan atas yang menerus atau bagunan atas yang panjang, sekat tubrukan harus diteruskan sampai kegeladak bangunan atas. Penerusan ini tidak perlu diletakkan langsung diatas sekat bawah. Bukaan-bukaan dengan alat penutup yang kedap cuaca dapat diizinkan sebelah atas geladak lambung timbul pada sekat tubrukan dan pada tingkat-tingkat relung yang disebut terdahulu. Jumlah lubang harus sedikit mungkin, sesuai dengan kebutuhan dan fungsi kapal.

5. Pembujur alas (Bottom Longitudinal)
6. Penumpu samping terputus (Intercostal side girder)
7. Penegar (Flat bar stiffener)
8. Pelat tepi miring (Margin plate)
9. Pelat lutut (Bracket)

G. Konstruksi Lambung

Sistem konstruksi lambung sebagai kerangka lambung kapal pada pokoknya terdiri atas dua sistem yaitu sistem kerangka gading melintang dan sistem kerangka gading memanjang.

1. Gading

Konstruksi kerangka gading-gading melintang merupakan penegar-penegar tegak yang dipasang pada pelat lambung dan berfungsi untuk memperkuat pelat lambung dari tekanan air di luar kapal. Pada kapal dengan geladak jamak (lebih dari satu) gading-gading ini diberi nama sesuai dengan letaknya. Gading-gading yang terletak di bawah geladak terakhir atau geladak utama disebut gading utama, yang terletak di antara dua geladak disebut gading antara, sedangkan yang disebut gading bangunan atas adalah gading yang terletak di bangunan atas.

Gading-gading melintang pada umumnya dipasang pada kapal-kapal yang lebih kecil dari 100 m karena masih belum memerlukan kekuatan memanjang yang lebih besar pada daerah lambung.

Gading-gading pada geladak dihubungkan dengan balok geladak melintang dan lutut sedangkan di bagian dasar dengan pelat lutut bilga.

Jarak gading melintang di lambung bervariasi dan sangat bergantung pada ukuran panjang kapal. BKI menentukan jarak gading standar a_0 dari sekat ceruk buritan hingga 0,2 L dari garis tegak haluan dihitung dengan rumus sebagai berikut :

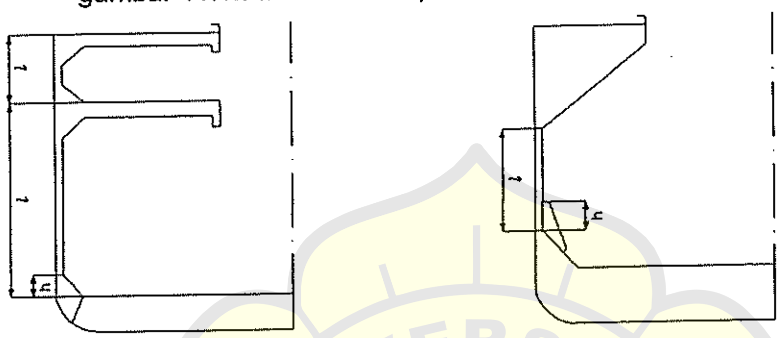
$a_0 = L/500 + 0{,}48$ (m),
a_0 maksimum = 1 m.

Untuk penampang profil gading-gading utama, BKI menentukan berdasarkan hasil perhitungan modulus penampang gading tersebut. Modulus penampang gading-gading tidak boleh kurang dari :

$W = k\ n\ a\ l^2\ P_s\ f$ (cm^3)

di mana :

a = Jarak gading
K = Faktor bahan. Berharga = 1 untuk kapal dengan baja normal.
l = Jarak bentang (m), termasuk pengikatan bagian-bagian ujung, biasanya tidak kurang dari
P_s = Besar beban tekan untuk gading (kN/m^2)
n = 0,63 – L/400, untuk L <_ 100 m..
= 0,38, untuk L \> 100 m.
f = 1,4 – h (fmin = 0,9)
h = Tinggi lutut di sisi atas wrang atau pelat alas dalam (lihat gambar 11.12 a dan11.12 b)



Gambar 11.12 Penentuan Jarak Bentang dari Rumus BKI 2004

Modulus penampang gading dalam tangki harus ditambah 10% dari hasil rumusan tersebut di atas dan tidak boleh kurang dari modulus penegar dalam tangki. Jika tangki muatan juga digunakan sebagai tangki balas, modulus penampang gading-gading tidak boleh kurang dari :

$$W = k\ 0{,}55\ a\ l2\ P_1\ (cm2)$$

dimana :
P_1 = Besar beban tekan pada tangki (kN/m2)
Untuk modulus penampang gading-gading geladak antara dan gading-gadng bangunan atas tidak boleh kurang dari :

$$W = k.0{,}8.a.l^2.P_s\ (cm^3)$$

dimana :
P_s = Tidak boleh kurang dari $P_{min} = 0{,}4.P_L.\ (b\ /\ l)^2\ (kN/m^3)$.
B = Panjang balok geladak dibawah gading-gading geladakantara (m).
P_L= Beban pada geladak antara, untuk puncak tangki adalah setengah jarak antara puncak tangki dan ujung atas pipa limpah akan tetapi tidak kurang dari 12,3 (kN/m^2)

Additions to the volume

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

Diesel oil

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

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

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

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

additions see fuel oil!

Lubrication oil

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

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

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

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

$b = 0.14$ [g/KW·h] turbines and gearboxes

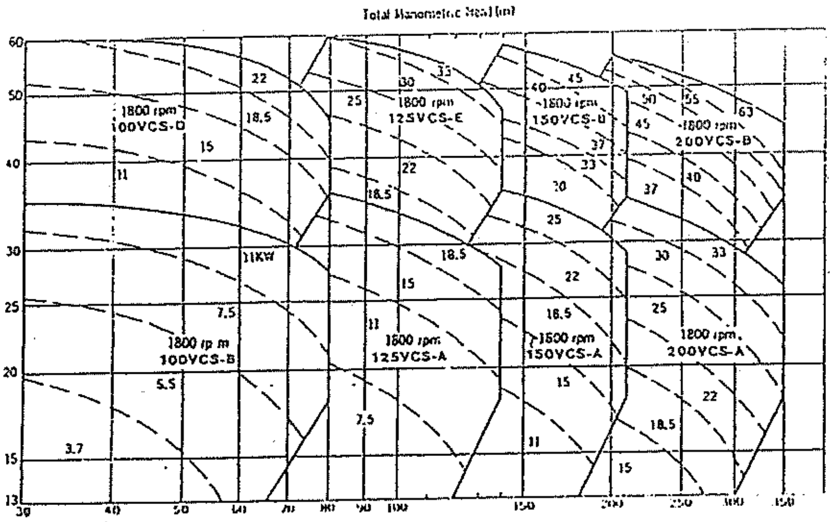
specific weight $\gamma_{lubr} = 0.90\ t/m^3$; $V = \dfrac{W}{\gamma}\ (m^3)$

Fresh water

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

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

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



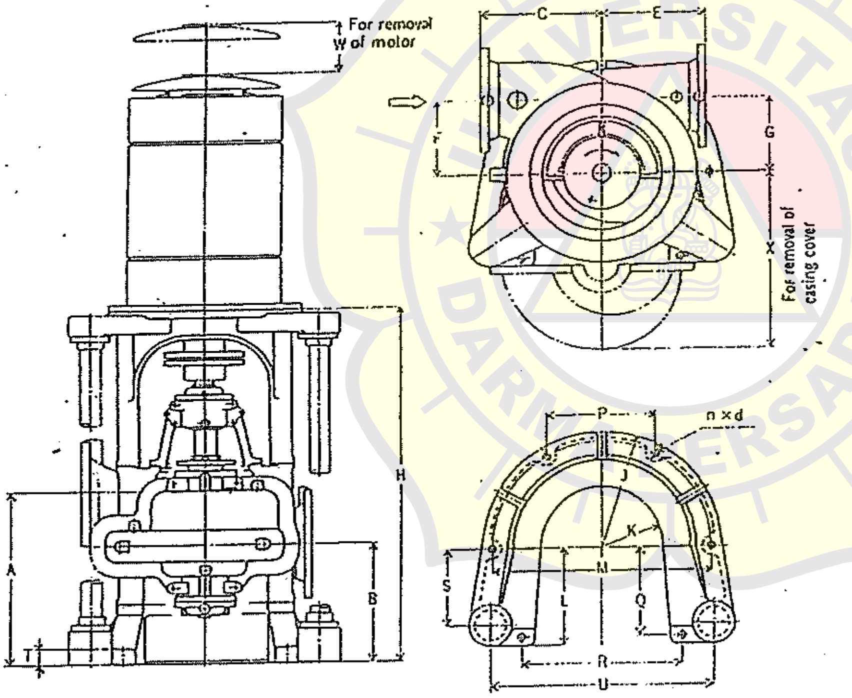
EXPLANATION ON PERFORMANCE CHART

In selecting the size of a pump pattern, if the required specified point of Q-H falls just on the boundary line in the performance chart, please select the smaller size of the nominal bore of the patterns from the adjoining ones.

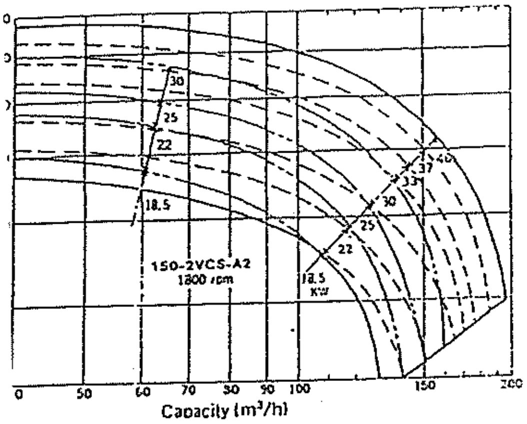
The numerals entered between diagonal dotted lines, show the required capacity of the driver in Kw.

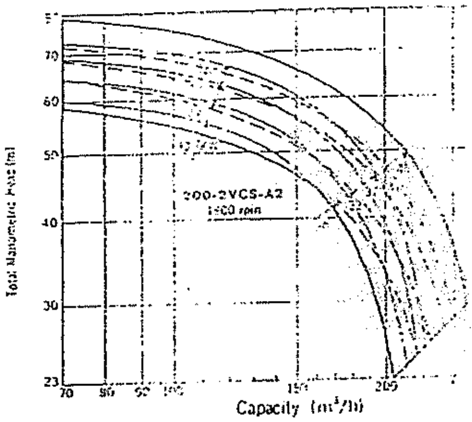
The driver with this capacity will never be overloaded at any point on the Q-H curve developed by the pump at the rated speed.

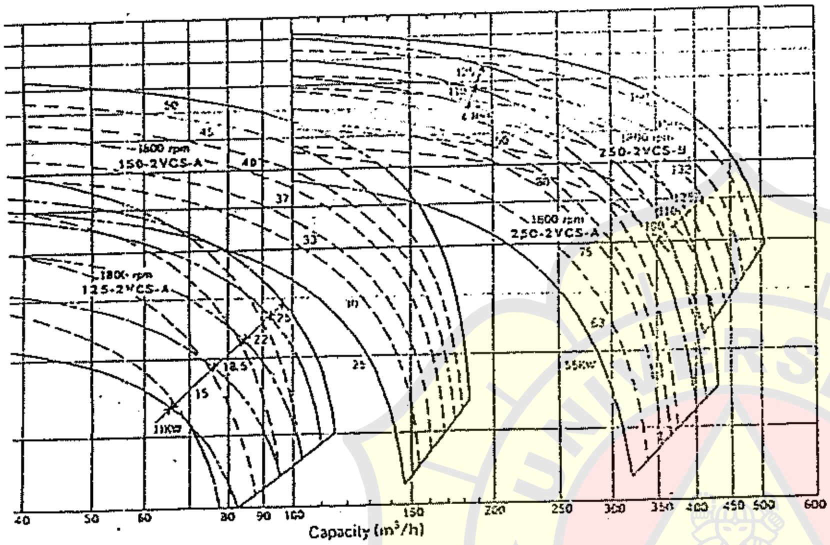
Ex. In case, the specified capacity, total head and speed are 125m³/h, 30m and 1,750 rpm respectively,
Select 125 VCS-A from the adjoining patterns of 125 VCS-A, 125 VCS-E and 150 VCS-A.
Capacity of the driver, 18.5 KW.

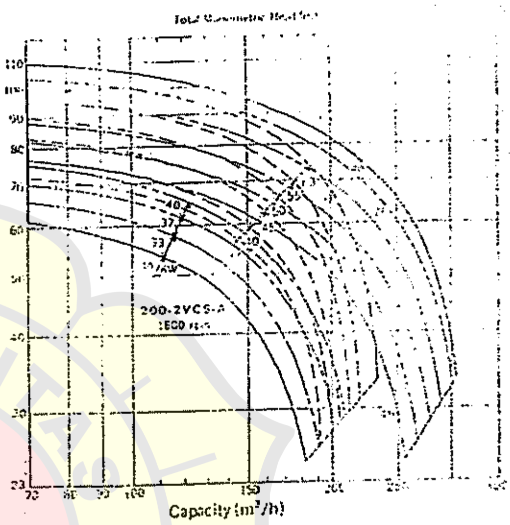


E	MOTOR		NOMINAL BORE		DIMENSIONS (mm)																					WEIGHT (kg) PUMP	
	KW	rpm	SUC	DEL	A	B	C	E	F	G	H	J	K	L	M	P	Q	R	S	T	U	nxd	W	X	FC CASING	BC CASING
S-B	3.7~11	1800	100	100	357	270	250	220	180	180	790	260	150	260	510	255	235	420	-	30	-	6x23	110	330	218	218
S-D	11~22				377	290	280	250	210	210	850	"	"	"	"	"	"	"	-	"	-	"	"	400	228	[illegible]
S-A	7.5~18.5		125	125	374	270	250	220	200	200	797	"	"	"	"	"	"	"	-	"	-	"	"	340	[illegible]	[illegible]
S-E	18.5~33				394	290	290	260	230	230	880	"	"	"	"	"	"	"	-	"	-	"	140	420	[illegible]	[illegible]
S-A	11~25		150	150	456	326	310	270	210	210	901	340	100	280	630	315	255	470	-	"	-	"	110	360	[illegible]	[illegible]
S-B	30~45				493	373	340	300	220	220	1061	"	"	"	"	"	"	"	-	"	-	"	140	500	[illegible]	[illegible]
S-A	15~33		200	200	486	326	320	280	230	230	953	"	"	"	"	"	"	"	220	"	650	"	"	490	[illegible]	427
S-B	37~63				485	325	350	300	"	"	1022	"	"	"	"	"	"	"	"	"	"	"	"	520	462	[illegible]





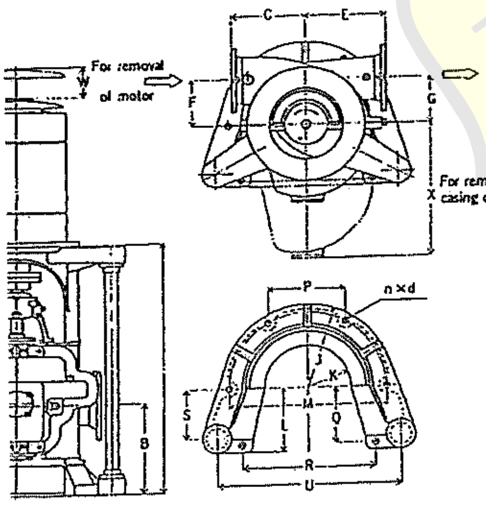




In selecting the size of pump pattern, if the required specified point of Q-H falls just on the boundary lines in the performance chart, please select the smaller size of the nominal bore of the pattern from the adjoining ones.

Dotted and chain lines show the limit of the required motor output, and additionally the tendency of the characteristic Q-H curves of the pump. If the specified point of Q-H falls on one of these lines, the numeral entered (in kw) just below that line shall be taken as the rated motor output.

Further, the applicable impellers will be different depending upon the variation in combination of required Q and H, such as 2 or 3 points are specified for instance. Accordingly, the characteristic curves will become different as shown in dotted or chain lines in the figures.

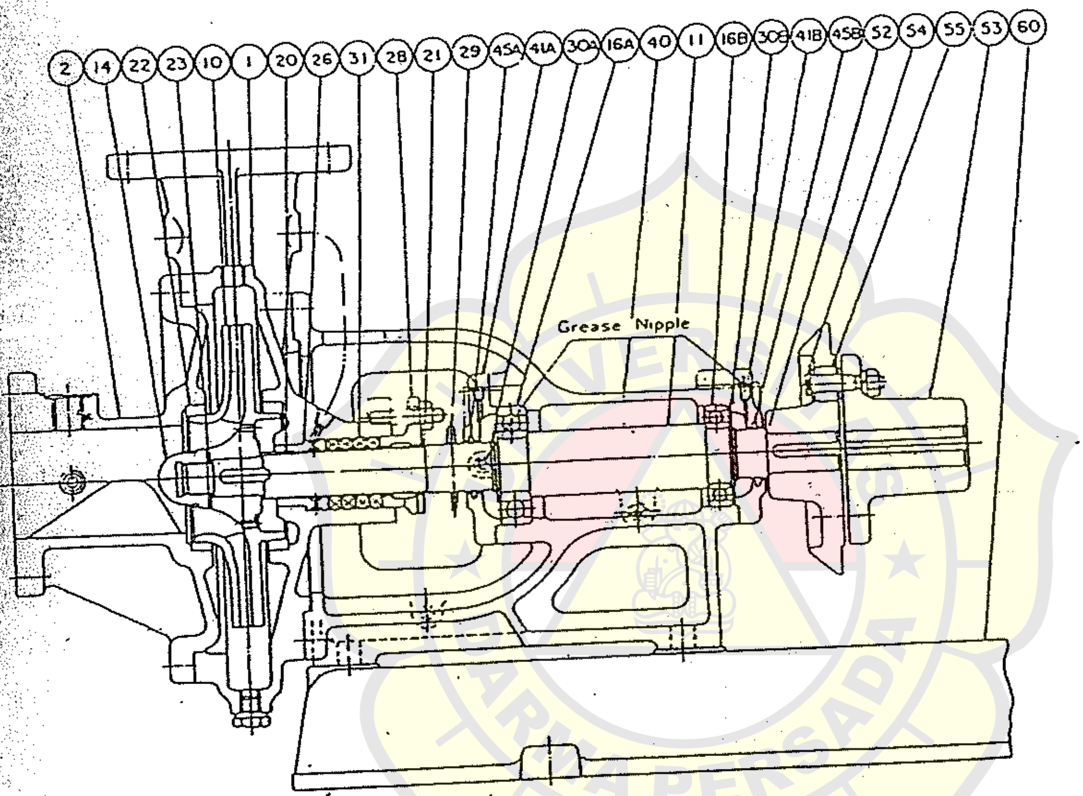


	MOTOR		NOMINAL BORE		DIMENSIONS (mm)																				WEIGHT (kg) PUMP	
	KW	rpm	SUC.	DEL.	A	B	C	E	F	G	H	J	K	L	M	P	Q	R	S	T	U	n×d	W	X	TC CASING	BC CASING
S-A	11 ~ 25	1800	125	125	538	370	320	320	180	180	1117	340	190	280	630	315	255	470	220	30	650	6x23	140	570	602	612
S-A	18.5~50		150	150	556	366	350	300	220	230	1109	"	"	"	"	"	"	"	"	"	"	"	"	"	705	720
S-A	30 ~ 80		200	200	638	417	"	370	"	220	1191	400	205	340	740	370	310	560	270	34	780	6x27	"	620	762	772
S-A	55~125		250	250	730	465	400	430	245	245	1298	"	"	"	"	"	"	"	"	"	"	"	"	720	902	907
												450	250	440	840	420	410	630	355	"	930	8x27	176	850	965	975

ONTAL SINGLE STAGE SINGLE SUCTION

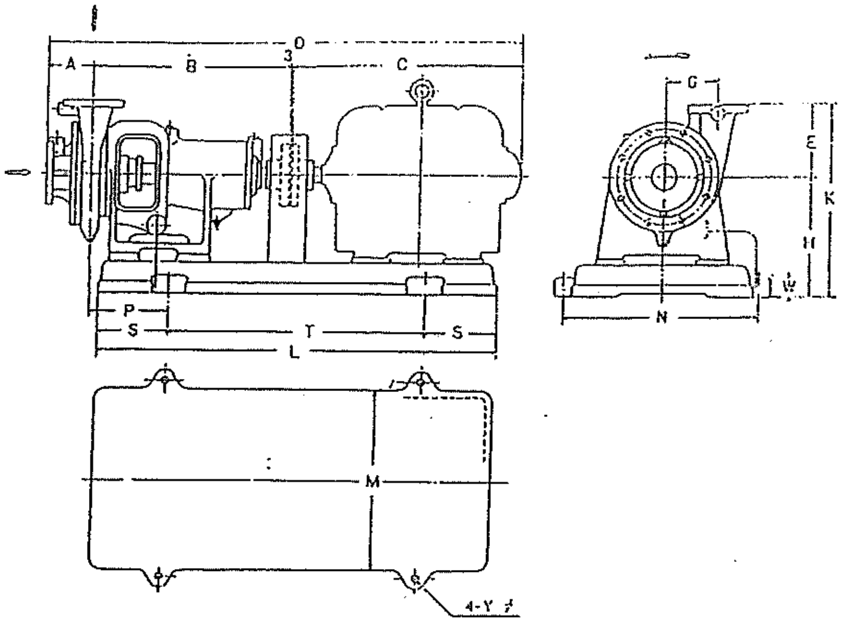
Type FIRE & G.S PUMP

...es are used mainly for Fire/G.S pumps, and ...pumps, and are fitted with a vacuum pump.
...T for materials of the main parts etc.



NAME OF PART	MATERIAL NOMINATION	QUANTITY
...ASING	BRONZE	1
...ASING COVER	BRONZE	1
...IPELLER	PHOS. BRONZE	1
...HAFT WITH KEY	STAINLESS STEEL	1 SET
...IPELLER NUT	HIGH- TEN. BRASS	1
...ALL BEARING		1
...ALL BEARING		1
...LEEVE	STAINLESS STEEL	1
..."O" RING	SYNTH. RUBBER	1
...OUTH RING	LEADED BRONZE	1 – 2
...ASHER	BRASS	1
...EAL BUSH	LEADED BRONZE	1
...ACKING GLAND	BRONZE	1
...ATER SHELTER	SYNTH. RUBBER	1

PART NO.	NAME OF PART	MATERIAL NOMINATION	QUANTITY
30A	RETAINING RING	CARBON T. STEEL	1
30B	RETAINING RING	CARBON T. STEEL	1
31	GLAND PACKING	PILLAR NO. 6501L	1 SET
40	BEARING CASE	CAST IRON/BRONZE	1
41A	BEARING COVER	CAST IRON/BRONZE	1
41B	BEARING COVER	CAST IRON/BRONZE	1
45A	FELT RING	FELT	1
45B	FELT RING	FELT	1
52	FLEXIBLE COUPLING	CAST IRON	1
53	FLEXIBLE COUPLING	CAST IRON	1
54	COUPLING BOLT	MILD STEEL	6 – 8
55	BUFFER RING	SYNTH. RUBBER	6 – 8
60	COMMON BED	CAST IRON	1



TYPE	MOTOR		NOMINAL BORE		DIMENSIONS (mm)																WEIGHT (kg) PUMP WITH BED	
	KW	rpm	SUC.	DEL.	A	B	C*	D*	E	G	H	K	L	M	N	P	S	T	W	Y	FC CASING	SC CASING
50MS-A	1.5	3600	50	50	95	400	297	795	150	105	230	380	760	290	320	150	125	510	25	15	102	105
	2.2	"	"	"	"	"	325	823	"	"	"	"	"		"	"	"	"	"	"	"	"
50MS-B	0.4	1800	"	"	"	"	207	705	160	120	230	390	760	290	320	150	125	510	25	15	107	110
	0.75	"	"	"	"	"	248	746	"	"	"	"	"	"	"	"	"	"	"	"	"	"
	1.5	"	"	"	"	"	297	795	"	"	"	"	"	"	"	"	"	"	"	"	"	"
	3.7	3600	"	"	"	"	355	853	"	"	"	"	"	"	"	"	"	"	"	"	"	"
	5.5	"	"	"	"	"	414	912	"	"	"	"	"	"	"	"	"	"	"	"	"	"
65MS-F	2.2	3600	65	65	105	400	325	833	160	105	230	390	760	290	320	150	125	510	25	15	132	[illegible]
	3.7	"	"	"	"	"	355	863	"	"	"	"	"	"	"	"	"	"	"	"	"	"
	5.5	"	"	"	"	"	414	922	"	"	"	"	"	"	"	"	"	"	"	"	"	"
65MS-G	1.5	1800	65	65	105	490	297	895	170	120	310	480	930	360	390	182	140	650	40	15	174	[illegible]
	2.2	"	"	"	"	"	325	923	"	"	"	"	"	"	"	"	"	"	"	"	"	"
	5.5	3600	"	"	"	"	414	1012	"	"	"	"	"	"	"	"	"	"	"	"	"	"
	7.5	"	"	"	"	"	452	1050	"	"	"	"	"	"	"	"	"	"	"	"	"	"
	11	"	"	"	"	"	556	1154	"	"	"	"	"	"	"	"	"	"	"	"	"	"
100MS-B	7.5	3600	100	100	136	490	452	1081	180	125	320	500	1060	360	390	167	160	740	40	19	210	216
	11	"	"	"	"	"	556	1185	"	"	"	"	"	"	"	"	"	"	"	"	"	"
	15—18.5	"	"	"	"	"	600	1229	"	"	"	"	"	"	"	"	"	"	"	"	"	"
100MS-D	3.7	1800	100	100	130	488	355	976	220	160	310	530	850	330	360	175	140	570	40	15	199	205
	5.5	"	"	"	"	"	414	1035	"	"	"	"	"	"	"	"	"	"	"	"	"	"
	7.5	"	"	"	"	"	452	1073	"	"	"	"	"	"	"	"	"	"	"	"	"	"
125MS-A	15—18.5	3600	125	125	150	493	600	1246	220	150	320	540	1080	400	430	170	160	760	40	19	240	247
	22	"	"	"	"	"	625	1271	"	"	"	"	"	"	"		"	"	"		"	"
	25—30	"	"	"	"	"	663	1309	"	"	"	"	"	"	"	"	"	"	"		"	"
125MS-B	7.5	1800	125	125	150	493	452	1098	250	195	320	570	1060	360	300	170	160	740	40	19	245	246
	11	"	"	"	"	"	556	1202	"	"	"	"	"	"	"	"	"	"	"	"	"	"
	15	"	"	"	"	"	600	1246	"	"	"	"	"	"	"	"	"	"	"	"	"	"

Note: Asterisked dimensions vary somewhat with driver.

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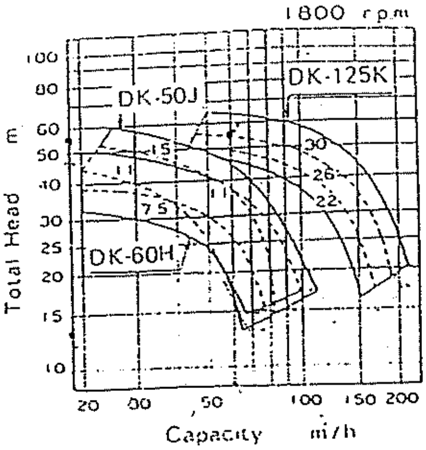
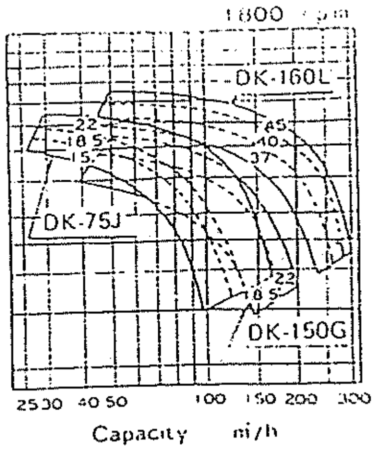
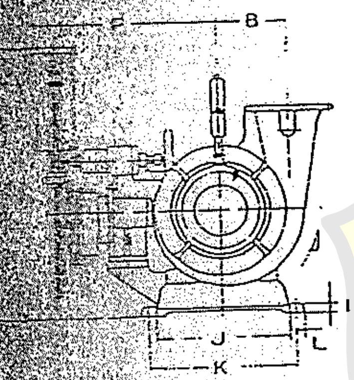
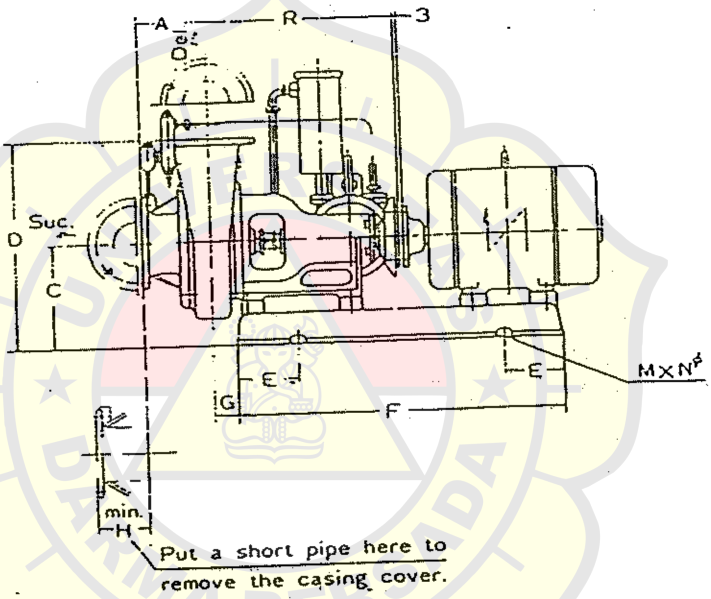
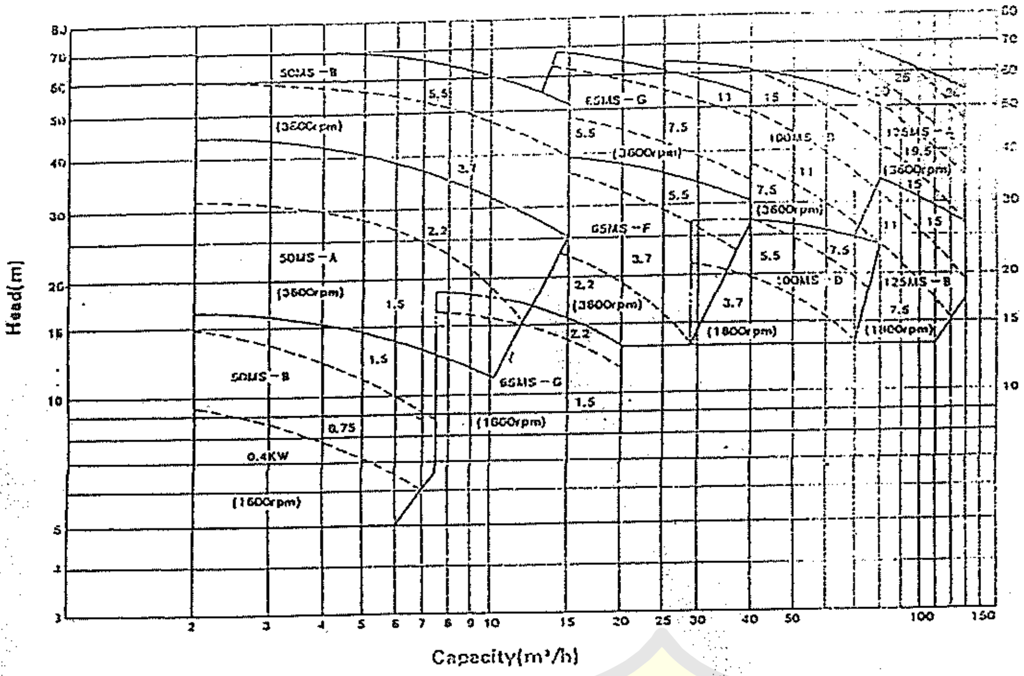








Motor (kW)	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	H	I	J	K	L	M	N	P	Q	R	Vacuum Pump	Pump Weight (kg)
[illegible]	100	100	140	220	240	520	150	900	30	100	25	360	400	23	4	15	405	705	420	V-18	250
15			140	220	240	520	150	900	30	100	25	360	400	23	4	15	405	705	420	V-18	250
[illegible]	100	100	117	177	240	470	150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



EXPLANATION ON PERFORMANCE CHART

selecting the size of a pump pattern, if the required specified point of Q-H falls just on the boundary line in the rmance chart, please select the small size of nominal bore of the pattern from the adjoining ones.

ıe numerals entered between diagonal dotted lines in the performance chart show the required capacity of the driver in . The driver with this capacity will never be overloaded at any point on the Q-H curve developed by the pump at the speed.

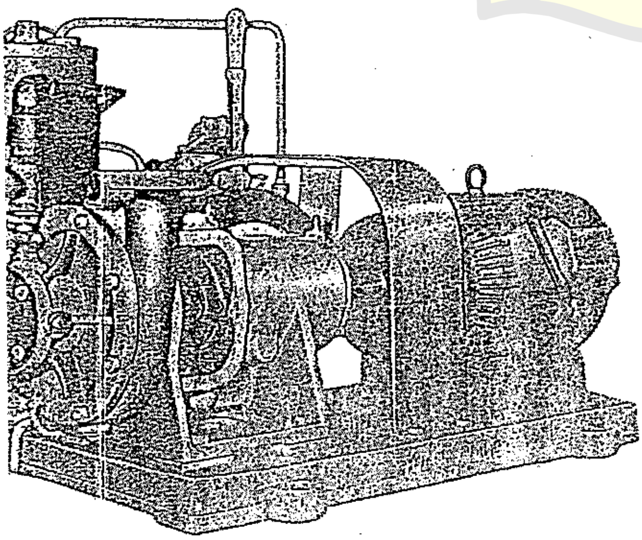
Ex.- In case, the specified capacity, total head and speed are 30 m³/h, 15 m and 3,450 rpm, respectively, Select 50 MS-B from between the adjoining patterns of 50 MS-B and 65MS-F, capacity of driver, 3.7 KW.

SELF PRIMING DEVICE

ə pump can be supplied, if required, with automatic priming equipment including its necessary accessories such as sealing tank, non-return valve, float valve and piping.

ə feature of this automatic priming system is as follws:-

ə primer is driven from the main pump shaft through combination coupling and friction pulley. The engagement and agement of the pulley are controlled automatically by means of a mechanism which is subjected to the discharge pressure developed by the main pump.

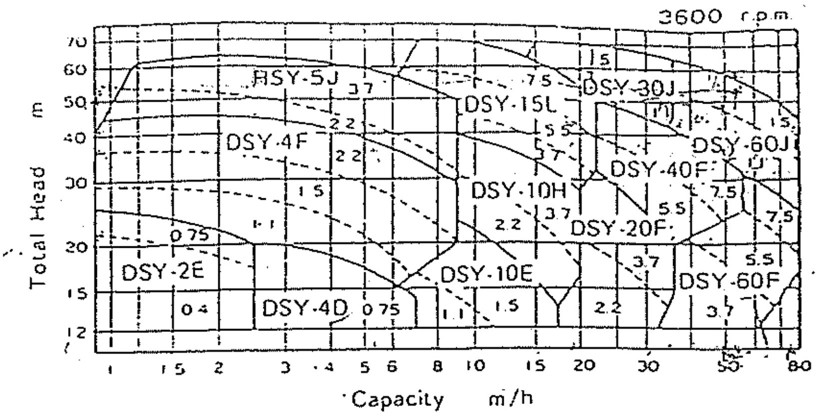


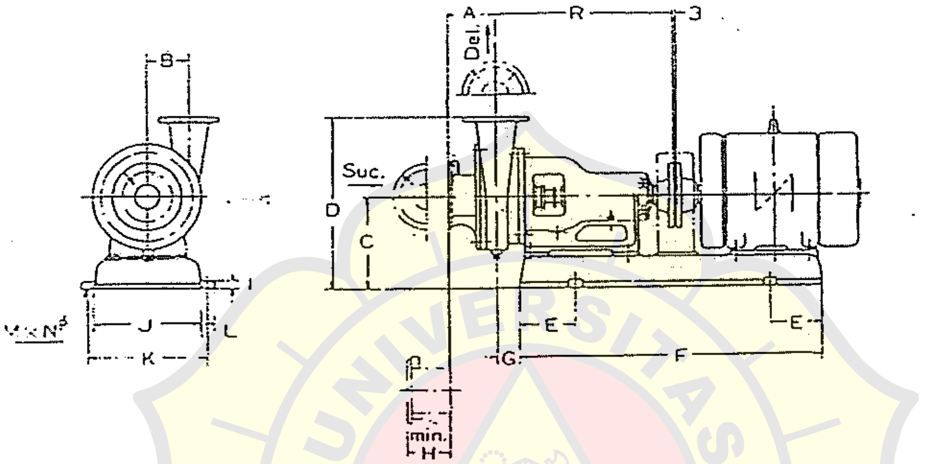
The primer ceases operation automatically on the accomplishment of the priming of the main pump and remains idle during the main pump is in service.

If the air breaks into the main pump for some reason, resulting in going down of the discharge pressure developed by the main pump, the primer begins to work automatically and the cycle recommences.

... SINGLE STAGE SINGLE SUCTION

SY Type





Put a short pipe here to remove the casing cover.

Dimensions—mm

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)
2E	[illegible]	32	32	88	73	165	315	100	550	35	100	25	200	240	23	4	15	350	45
4D	[illegible]	32	32	90	65	172	290	100	550	35	100	25	200	240	23	4	15	350	45
[illegible]	[illegible]	32	32	96	92	165	315	100	600	30	100	25	200	240	23	4	15	350	50
	2.2			96	92	175	325	100	620	35	100	25	240	280	23	4	15	350	50
[illegible]	2.2	32	32	94	95	182	325	100	620	35	100	25	240	280	23	4	15	350	60
	3.7			94	95	197	340	100	650	30	100	25	260	300	23	4	15	350	60
10E	1.5	50	50	96	75	165	295	100	600	30	100	25	200	240	23	4	15	350	60
10H	2.3	50	50	96	96	175	325	100	620	35	100	25	240	280	23	4	15	350	55
	3.7			96	96	190	340	100	650	30	100	25	260	300	23	4	15	350	55
15L	3.7	50	50	110	120	190	390	125	700	28	100	25	300	340	23	4	15	370	80
	5.5			110	120	210	410	125	700	28	100	25	300	340	23	4	15	370	80
	7.5			110	120	210	410	150	750	28	100	25	300	340	23	4	15	370	80
20F	2.2	65	65	105	95	175	315	100	620	35	100	25	240	280	23	4	15	350	75
	3.7			105	95	190	330	100	650	30	100	25	260	300	23	4	15	350	75
40F	5.5	100	100	118	105	202	420	125	700	28	100	25	300	340	23	4	15	370	125
	7.5			118	105	202	420	150	750	28	100	25	300	340	23	4	15	370	125
30J	11	65	65	120	120	249	445	175	860	35	100	25	360	400	23	4	15	425	110
	15			120	120	249	445	175	900	35	100	25	360	400	23	4	15	425	110
0F	3.7	100	100	122	115	210	390	120	700	35	100	25	300	340	23	4	15	425	160
	5.5			122	115	210	390	150	750	30	100	25	300	340	23	4	15	425	160
	7.5			122	115	210	390	150	780	35	100	25	300	340	23	4	15	425	160
0J	11	100	100	120	120	235	445	175	860	35	100	25	360	400	23	4	15	425	110
	15			120	120	235	445	175	900	35	100	25	360	400	23	4	15	425	110

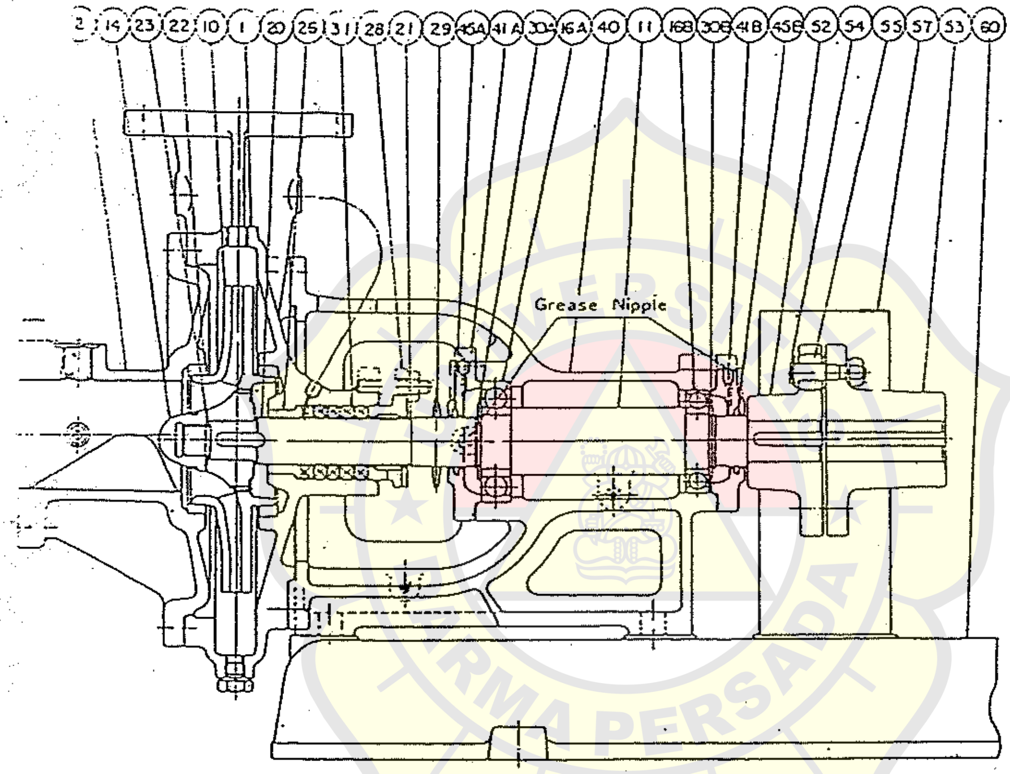
... type GENERAL PUMP

... speed of HSY type is 3600 r.p.m.
... r.p.m.

... materials of main parts.

... seen in every performance chart
... synchronous revolution of electric
... performance in the performance
... the revolution at which the motors
... with pumps.

Service
- Fresh Water Pump
- Drinking Water Pump
- Sanitary Pump
- Cooling Water Pump



NAME OF PART	MATERIAL NOMINATION	QUANTITY	PART NO.	NAME OF PART	MATERIAL NOMINATION	QUANTITY
CASING		1	30A	RETAINING RING	CARBON T. STEEL	1
CASING COVER		1	30B	RETAINING RING	CARBON T. STEEL	1
IMPELLER		1	31	GLAND PACKING	PILLAR NO. 6501L	1 SET
SHAFT WITH KEY	STAINLESS STEEL	1 SET	40	BEARING CASE	BRONZE/CAST IRON	1
IMPELLER NUT	HIGH-TEN. BRASS	1	41A	BEARING COVER	BRONZE/CAST IRON	1
BALL BEARING		1	41B	BEARING COVER	BRONZE/CAST IRON	1
BALL BEARING		1	45A	FELT RING	FELT	1
SLEEVE	STAINLESS STEEL	1	45B	FELT RING	FELT	1
O-RING	SYNTH. RUBBER	1	52	FLEXIBLE COUPLING	CAST IRON	1
MOUTH RING	LEADED BRONZE	1 – 2	53	FLEXIBLE COUPLING	CAST IRON	1
WASHER	BRASS	1	54	COUPLING BOLT	MILD STEEL	6 – 8
SEAL BUSH	LEADED BRONZE	1	55	BUFFER RING	SYNTH. RUBBER	6 – 8
PACKING GLAND	BRONZE	1	57	COUPLING COVER	MILD STEEL	1
WATER SHELTER	SYNTH. RUBBER	1	60	COMMON BED	CAST IRON	1

PERHITUNGAN CONSUMABLES

1. Berat Bahan Bakar Mesin Induk

$W_{fo} = BHP_{me} . b_{me} . S/Vs . 10^{-6} . C$ (ton)

Dimana:
- BHP_{me} = Bhp mesin induk (katalog mesin) kW
- b_{me} = spesifik konsumsi bahan bakar mesin induk (171 g/kWh)
- S = jarak pelayaran (mil)
- Vs = kecepatan dinas (knot)
- C = koreksi cadangan (1,3 – 1,5)

Menentukan volume bahan bakar mesin induk:

$V (W_{fo}) = W_{fo}/\rho$ (m^3)   dimana: ρ = 0,95 ton/m^3

Volume bahan bakar mesin induk ada penambahan karena:
- Double bottom (2 %)
- Ekspansi karena panas (2 %)

2. Berat Bahan Bakar Mesin Bantu (Wfb)

W_{fb} = (0,1 – 0,2) W_{fo} (ton)

Menentukan bahan bakar mesin bantu (Vfb):

$V_{fb} = W_{fb} / \rho$ diesel (m^3) dimana: : ρ = 0,95 ton/m^3

Volume tangki bahan bakar mesin bantu ada penambahan sebesar 4 % V_{fb}.

3. Berat Minyak Pelumas (W_{lo})

W_{lo} = BHPme . b_{lo} . S/Vs . 10^{-6} . (1,3 – 1,5) (ton)

Dimana: b_{lo} = 1,2 – 1,6

Menentukan volume minyak pelumas (lubricating oil):

$V_{lo} = W_{lo} / \rho$ (m^3) dimana: ρ = 0,90 ton/m^3

Volume tangki ada penambahan sebesar 4 % V_{lo}.

Ballast capacity used for

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

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

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

- +15% lower fore peak tank
- +3% upper fore peak tank
- +2% double bottom tank.

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

Sounding/ullage tables delivered by yard.

Provisions/persons/luggage

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

...e and Location of Main Engine

...another part of the contract influencing ship design.
...ip weight, volume, fuel consumption).
...nomy is determined by the choice of the main engine type, also

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

$c_1 = A_1/b_1$

14-4

$c_2 = A_2/b_2$

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

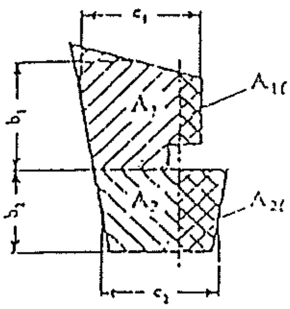


Fig. 14.2

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

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

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

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

for ahead condition

The greater value is to be taken.

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:

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

$D_t = 4.2 \times \sqrt[3]{Q_r \cdot k_r}$

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

The related torsional stress is:

$\tau_t = \frac{68}{k_r}$ [N/mm²]

k_r see A.4.2.

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

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

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

2. Strengthening of rudder stock

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

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

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

Bending stress:

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

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

Torsional stress:

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

D_1 = increased rudder stock diameter in [cm]

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

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

Q_R see B.1.2 and B.2.2 - 2.3

D_t see 1.1.

Guidance

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

$$y = 1 - \frac{F_1(1 - Y_1)^2 + F_2(1 - \ldots)^2}{F}$$

dimana $y_1 = (0{,}68 - 0{,}43 - \Delta y + 0{,}18 \; \frac{2h_1 + l_1}{H})w$

$$y_2 = (0{,}68 - 0{,}43 - \Delta y + 0{,}18 \; \frac{2h_2 + l_2}{H})w$$

$$F = F_1 + F_2$$

...

... luas kemudi yang setengah menggantung.

... *permukaan air dan ... yang pada kerja Kemudi.*

Terbentuk putaran air pada sayap kemudi menyebabkan pengurangan tekanan pada kemudi juga menurunkan gaya angkat, kalau letak kemudi lebih dalam dari batas permukaan air.

Jika permukaan air laut tenang, kecepatan kapal dan sudut letak kemudi kecil maka bagian kemudi yang ada di atas permukaan air akan bekerja lebih baik dari pada letak kemudi yang lebih dalam dari permukaan air.

Untuk kapal-kapal yang mempunyai kecepatan besar dan dengan adanya gelombang yang tak teratur, efek baiknya (yaitu dengan sebagian kemudi di atas permukaan air) tidak akan terjadi.

Jadi untuk kapal barang atau kapal penumpang yang berlayar dilautan terbuka dianjurkan letak kemudi jika mungkin dibawah waterline konstruksi.

Dalam percobaan model kapal yang sebenarnya menerangkan bahwa arus gelombang dapat menambah besar moment pada kemudi dalam pelayaran lurus ataupun dalam sirkulasi.

Menurut percobaan *"Henson"* dan *"Hautteker"* (dari Swedia). Gelombang dengan ketinggian 4 meter dapat menambah besarnya moment yang maximum dalam sirkulasi dan besarnya 35% jika dibandingkan dengan moment maximum dalam sirkulasi dilautan yang tenang.

Percobaan lain yang dilakukan di Jepang, untuk laut yang bergelombang maka moment yang timbul 2,5 kali moment maximum diair tenang.

Percobaan di Holland menunjukkan moment yang timbul di air bergelombang, 3 kali moment maximum di air pada sudut kemudi 30.

... *... geometris Kemudi.*

A). Dipandang dari letak sayap kemudi terhadap porosnya maka kemudi dapat dibagi :

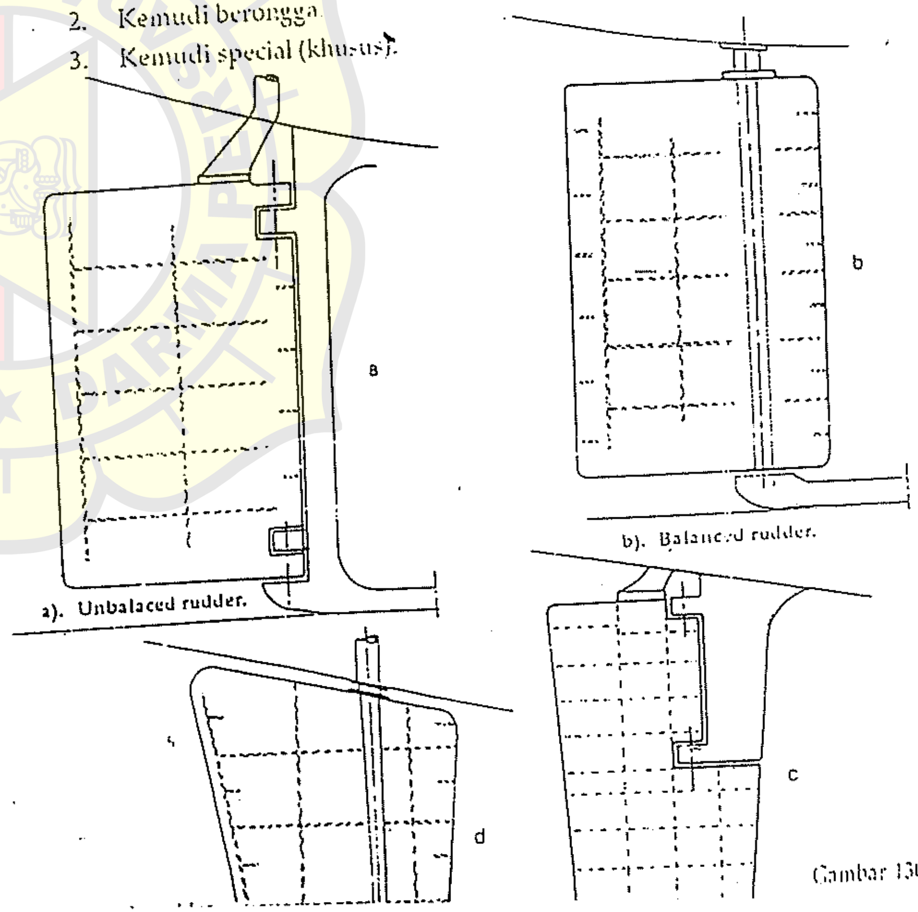
1. Kemudi biasa, dimana semua luas sayap kemudi terletak dibelakang
2. Kemudi balansir, dimana luas sayap kemudi terbagi dua, bagian dimuka dan dibelakang sumbu putar kemudi (gambar b).
3. Kemudi setengah balansir, dimana bagian atas sayap kemudi termasuk kemudi biasa, sedang bagian bawah merupakan kemudi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

B). Dipandang dari sulfies (sepatu linggi) dibagi :

1. Kemudi meletak (gambar a dan b)
2. Kemudi menggantung (gambar d)
3. Kemudi setengah menggantung (gambar c)

C). Dipandang dari konstruksinya dibagi :

1. Kemudi plat (satu lapis plat)
2. Kemudi berongga
3. Kemudi special (khusus).



Gambar 130.

Section 9

Framing System

Bukunya disebut!

A. Transverse Framing

1. General

1.1 Frame spacing

Forward of the collision bulkhead and aft of the afterpeak bulkhead, the frame spacing shall in general not exceed 600 mm.

1.2 Definitions

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

. = unsupported span in [m] according to Section 3, C., see also Fig. 9.1

$_{min}$ = 2,0 m

$_{Ku}$, $_{Ko}$ = length of lower/upper bracket connection of main frames within the length l in [m], see Fig. 9.1

m_a = $0{,}204\ \frac{a}{\cdot}\left[4\ \ \left(\frac{a}{\cdot}\right)^2\right]$, where $\frac{a}{\cdot} \le 1$

e = spacing of web frames [m]

p = p_s or p_e as the case may be

p_s = load on ship's sides [kN/m^2] according to Section 4, B.2.1

p_e = load on bow structures [kN/m^2] according to Section 4, B.2.2 or stern structures according to Section 4, B.2.3 as the case may be

p_L = 'tween deck load [kN/m^2] according to Section.4, C.1.

p_1, p_2 = pressure [kN/m^2] according to Section 4, D.1.

H_u = depth up to the lowest deck [m]

c_r = factor for curved frames

= $1{,}0\ \ 2\ \frac{s}{\cdot}$

c_{rmin} = 0,75

s = max. height of curve.

2. Main frames

2.1 Scantlings

2.1.1 The section modulus W_R and shear area A_R of the main frames including end attachments are not to be less than:

$$W_R = n \cdot c \cdot a \cdot\ ^2 \cdot p \cdot c_r \cdot k \ \ [cm^3]$$

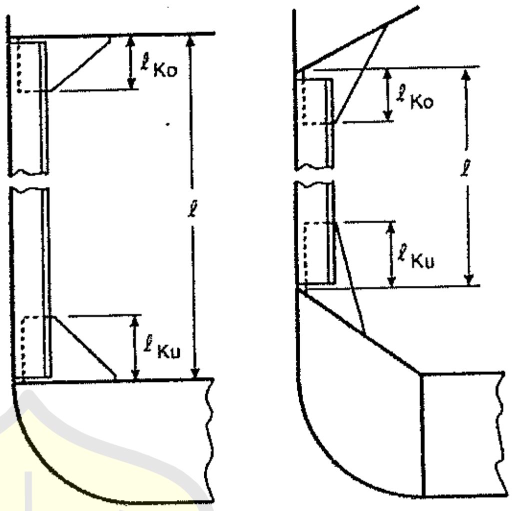


Fig. 9.1 Unsupported span of transverse frames

upper end shear area :

$$A_{RO} = (1\ \ 0{,}817 \cdot m_a)\ 0{,}04 \cdot a \cdot\ \cdot p \cdot k \ \ [cm^2]$$

lower end shear area :

$$A_{RU} = (1\ \ 0{,}817 \cdot m_a)\ 0{,}07 \cdot a \cdot\ \cdot p \cdot k \ \ [cm^2]$$

n = 0,9 0,0035 · L for L < 100 m

= 0,55 for L ≥ 100 m

$$c = 1{,}0\ \left(\frac{_{Ku}}{\cdot}\ \ 0{,}4\ \frac{_{Ko}}{\cdot}\right)$$

c_{min} = 0,6

Within the lower bracket connection the section modulus is not to be less than the value obtained for c = 1,0.

2.1.2 In ships with more than 3 decks the main frames are to extend at least to the deck above the lowest deck.

2.1.3 The scantlings of the main frames are not to be less than those of the 'tween deck frames above.

2.1.4 Where the scantlings of the main frames are determined by strength calculations, the following permissible stresses are to be observed:

bending stress: $\sigma_b = \frac{150}{k}$ [N/mm^2]

shear stress: $\tau = \frac{100}{k}$ [N/mm^2]

Section 11

Watertight Bulkheads

A. General

1. Watertight subdivision

1.1 All ships are to have a collision bulkhead, a stern tube bulkhead and one watertight bulkhead at each end of the engine room. In ships with machinery aft, the stern tube bulkhead may substitute the aft engine room bulkhead.

1.2 For ships without longitudinal bulkheads in the cargo hold area the number of watertight transverse bulkheads should, in general, not be less than given in Table 11.1.

Table 11.1 Number of watertight transverse bulkheads

L [m]	Arrangement of machinery space	
	aft	elsewhere
$L \leq 65$	3	4
$65 < L \leq 85$	4	4
$85 < L \leq 105$	4	5
$105 < L \leq 125$	5	6
$125 < L \leq 145$	6	7
$145 < L \leq 165$	7	8
$165 < L \leq 185$	8	9
$L > 185$	to be special considered	

1.3 One or more of the watertight bulkheads required by 1.2, may be dispensed with where the transverse strength of the ship is adequate. The number of watertight bulkheads will be entered into the Register Book.

1.4 Number and location of transverse bulkheads fitted in addition to those specified in 1.1 are to be so selected as to ensure sufficient transverse strength of the hull.

1.5 For ships which require proof of survival capability in damaged conditions, the watertight sub-division will be determined by damage stability calculations. For oil tankers see Section 24, A.2., for passenger vessels see Section 29-I, C., for special purpose ships see Section 29-II, C., for cargo ships of more than 100 m in length see Section 36 and for supply vessels see Section 34, A.2. For liquefied gas tankers see Rules for Ships Carrying Liquefied Gases in Bulk, Volume IX, Section 2, for chemical tankers see Rules for Ships Carrying Dangerous Chemicals in Bulk, Volume X, Section 2.

2. Arrangement of watertight bulkheads

2.1 Collision bulkhead

2.1.1 Cargo ships with $L_c \leq 200$ m shall have the collision bulkhead situated not less than 0,05 L_c from the forward perpendicular. Cargo ships with $L_c > 200$ m shall have the collision bulkhead fitted at least 10 m from the forward perpendicular.

2.1.2 All cargo ships shall have the collision bulkhead located not more than 0,08 L_c from the forward perpendicular. Upon application greater distances may be approved in special cases.

2.1.3 In the case of ships having any part of the underwater body extending forward of the forward perpendicular, e.g. a bulbous bow, the required distances specified in 2.1.1 and 2.1.2 are to be measured from a reference point located at a distance x forward of the forward perpendicular which shall be the lesser of:

$$x = \frac{a}{2}$$

$$= 0{,}015\ L_c$$

$$= 3{,}0\ \text{m.}$$

For passenger ships see Section 29-I, C.3.

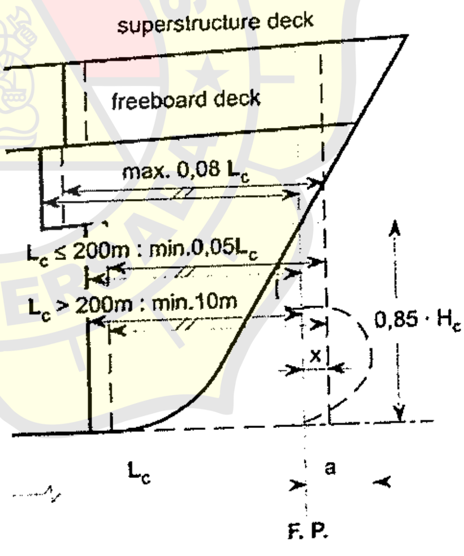
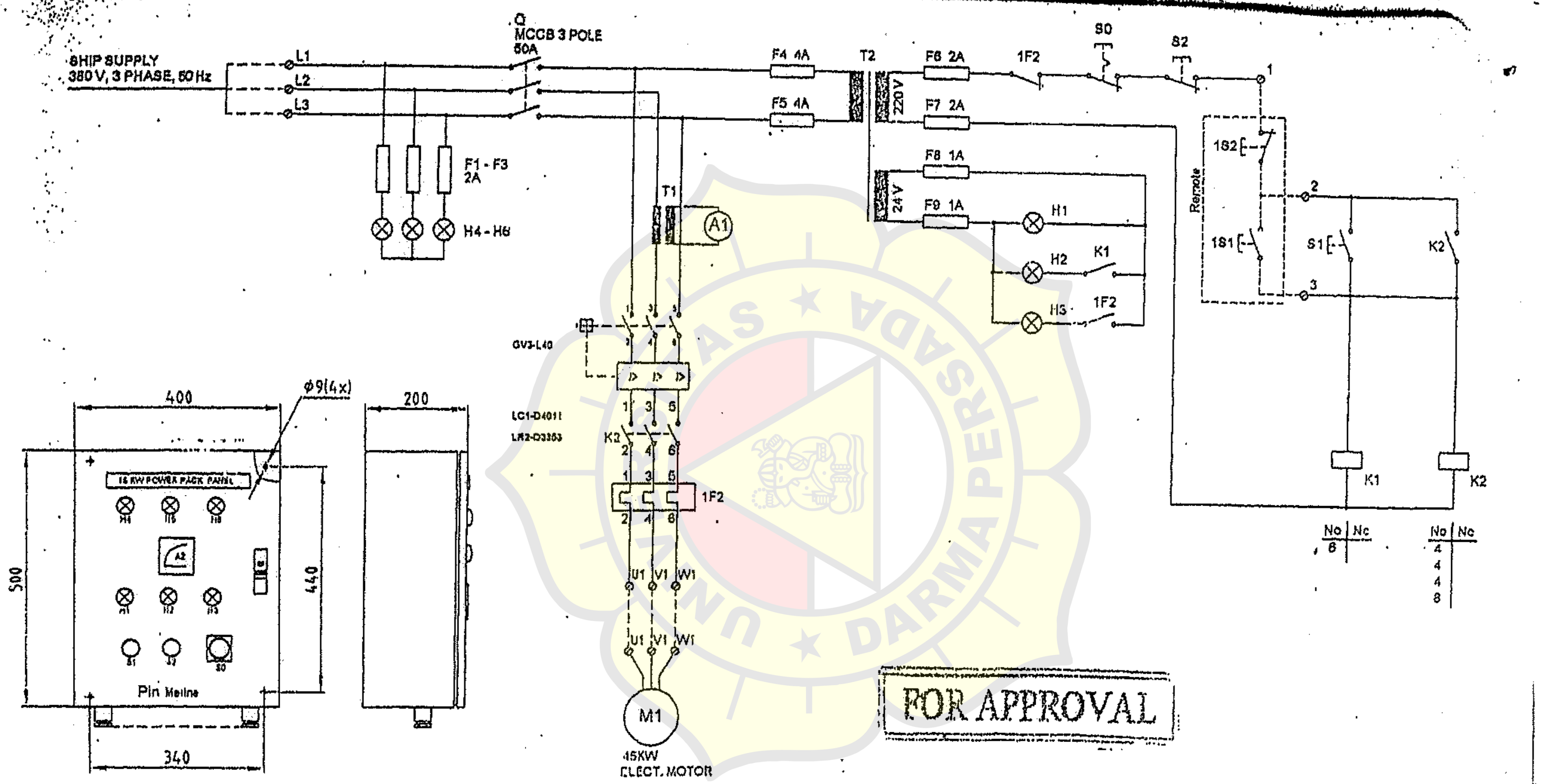


Fig. 11.1 Location of collision bulkhead

2.1.4 The collision bulkhead shall extend watertight up to the freeboard deck. Steps or recesses may be permitted provided 2.1.1, 2.1.2 and 2.1.3 are observed.

2.1.5 In ships having continuous or long superstructures, the collision bulkhead shall extend to the first deck above



S0	Emergency stop
S1	Hyd. pump on
S2	Hyd. pump off
1S1	Hyd. pump on (remote)
1S2	Hyd. pump off (remote)
H1	Source on indicator
H2	Hyd. pump on indicator
H3	Overload indicator
H4	Phase indicator
H5	Phase indicator
H6	Phase indicator

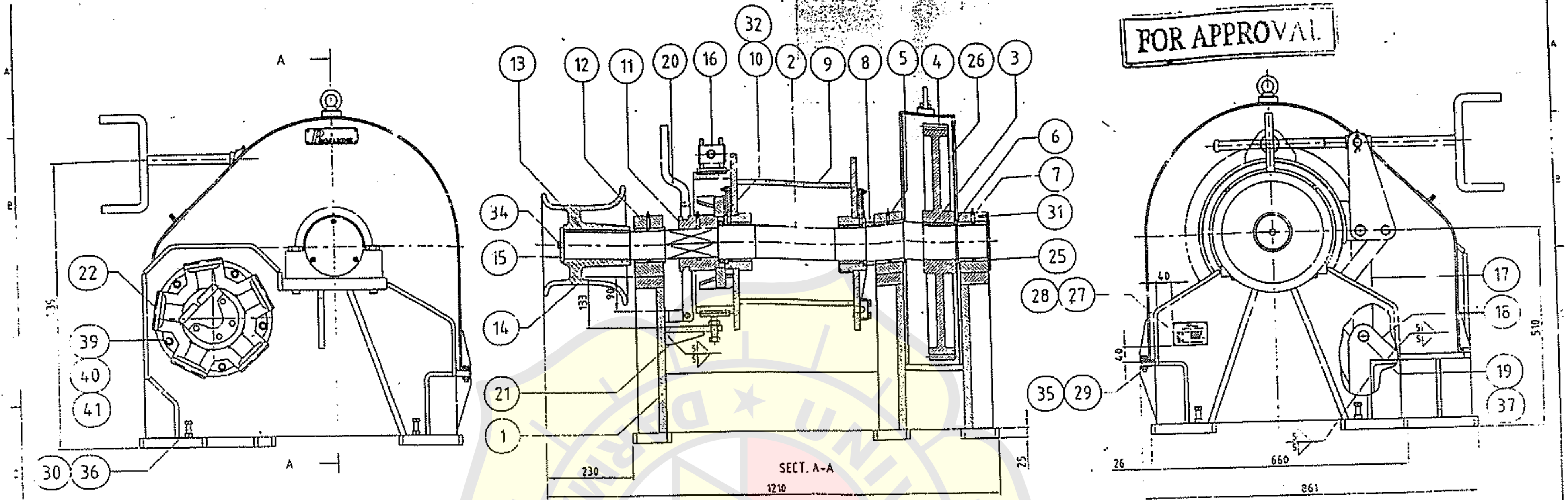
Nomor Bagian	Nama Bagian	Banyak nya	Bahan &. Spesifikasi	Keterangan :
Skala	Peru- a	c	e	
Ukuran : mm	bahan b	d	f	

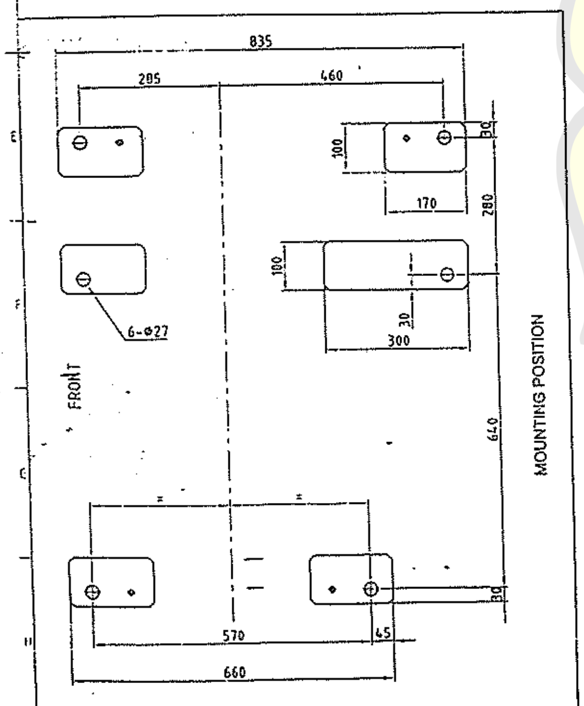
NAMA
ELECTRIC DIAGRAM
For 45 KW Power Pack (Roro 8/850 GRT)

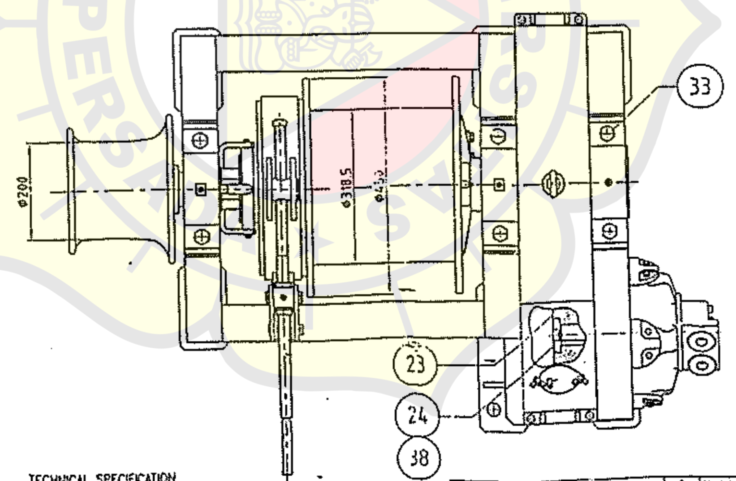
Digambar	Zie	Tgl. 21.01.'08
Diperiksa		Tgl.
Disetujui		Tgl.

PT. PINDAD (PERSERO)

No. Induk Gambar : Nomor Gambar :







TECHNICAL SPECIFICATION

MOORING WINCH

DRUM CAPACITY: 60 m ROPE #16 at 3 LAYERS

PULL/SPEED : 20 KN at 15 m/min

WARPING END

Pull/Speed : 31 KN at 9 m/min

WEIGHT : APPROX 600 kg

Nomor Bagian	Nama Bagian	Banyak nya	Bahan & Spesifikasi	Keterangan
41	HEX-BOLT	5	ISO 4014 – M12 x 80	
40	HEX NUT	5	ISO 4032 – M12	
39	SPRING WASHER	5	DIN 125 – A13	
38	CYLINDER HEAD CAP SCREW	4	DIN 912 – M14 x 26	
37	SPLIT PIN	1	ISO 1234 5x32	
36	HEX-BOLT	4	ISO 4016-M12x60	
35	HEX-BOLT	4	ISO 4017-M10x35	
34	HEX-BOLT	1	ISO 4016-M10x45	
33	HEX-BOLT	6	ISO 4016-M20x110	
32	HEX-SCREW	2	DIN 915-M8x16	
31	HEX-SCREW	3	ISO 10642-M6x20	
30	HEX NUT	4	ISO 4034 – M12	
29	NUT M10	4	DIN 934-M10	
28	GROOVED STUD	4	ISO 8746-3 x10	
27	SERIAL DATA PLATE	1	SM400A, t2x80x150	GP-SAW05-04.1271-0
26	GEAR COVER-D652 ASSY-T62	1	-	GP30R02-04.1175-0
25	BUSHING CAP-D150	1	SM400A, Ø150x65	GP4S03-04.1208-0
24	PINION GEAR CAP-D70	1	SM400A, Ø28x15	GP4S03-06.1775-0
23	PINION GEAR Z20,7045-D42	1	42CrMo4, Ø132x76	GP4P03-06.1256-0
22	HYDRAULIC MOTOR+PJCT	1	H46-4000 PARALLEL	-
21	BRAKE STOPPER-L130 ASSY	1	-	GP4BR02-06.1537-0
20	CLUTCH LEVER B130 ASSY-RD	1	-	GP4CT02-06.1212-0
19	BRAKE PIN-D28L50	1	SUS304, Ø28x45	GP4BR02-06.1536-0
18	BOTTOM LINK-B150	1	SM400A, t50x150x243	GP4BR02-06.1535-0
17	BRAKE LINK-L372	1	SM400A, t25x50x372	GP4BR02-06.1540-0
16	BRAKE ASSY B350L785	1	-	GP3BR02-06.1534-0
15	CAP-WARPING END D200	1	SM400A, Ø90x8	GP4WE04-03.1004-0
14	WARPING END D200	1	GC-25, Ø70x308	GP4WE04-03.1003-0
13	PARALLEL KEY-Y20X176	1	S45C, t12x20x176	GP4WE04-03.1005-0
12	JOURNAL BEARING ASSY 1	1	-	GP4JB03-04.1045-0
11	CLUTCH ASSY-ROPE DRUM-L474	1	-	GP4CT02-04.1005-0
10	SPACER 3	1	SM400A, Ø135x17	GP4SC02-03.1041-0
9	ROPE DRUM D315-L310R16 ASSY	1	-	GP3DC02-02.2722-0
8	SPACER 1	1	SM400A, Ø110x23	GP4SC02-03.1045-0
7	JOURNAL BEARING ASSY3	1	-	GP4JB03-05.1561-0
6	SPACER-D90L94	1	SM400A, Ø110x7.5	GP4SC02-06.1144-0
5	JOURNAL BEARING ASSY 2	1	-	GP4JB02-03.1031-0
4	GEAR Z102,7045-D90	1	S55C, Ø612x85	GP3GR02-03.1078-1
3	PARALLEL KEY-A25X14X85	1	S55C, t14x25x85	GP4KE03-05.1112-0
2	MAIN SHAFT-D70D95-L1130	1	S55C, Ø95x1134	GP3MS01-08.1145-0
1	FRAME ASSY	1	-	GP3FR02-08.2001-0

Skala 1:5	Perubahan	a	c	e
Ukuran: mm		b	d	f

Judul: 20 KN HYD. RAMP DOOR WINCH

Digambar

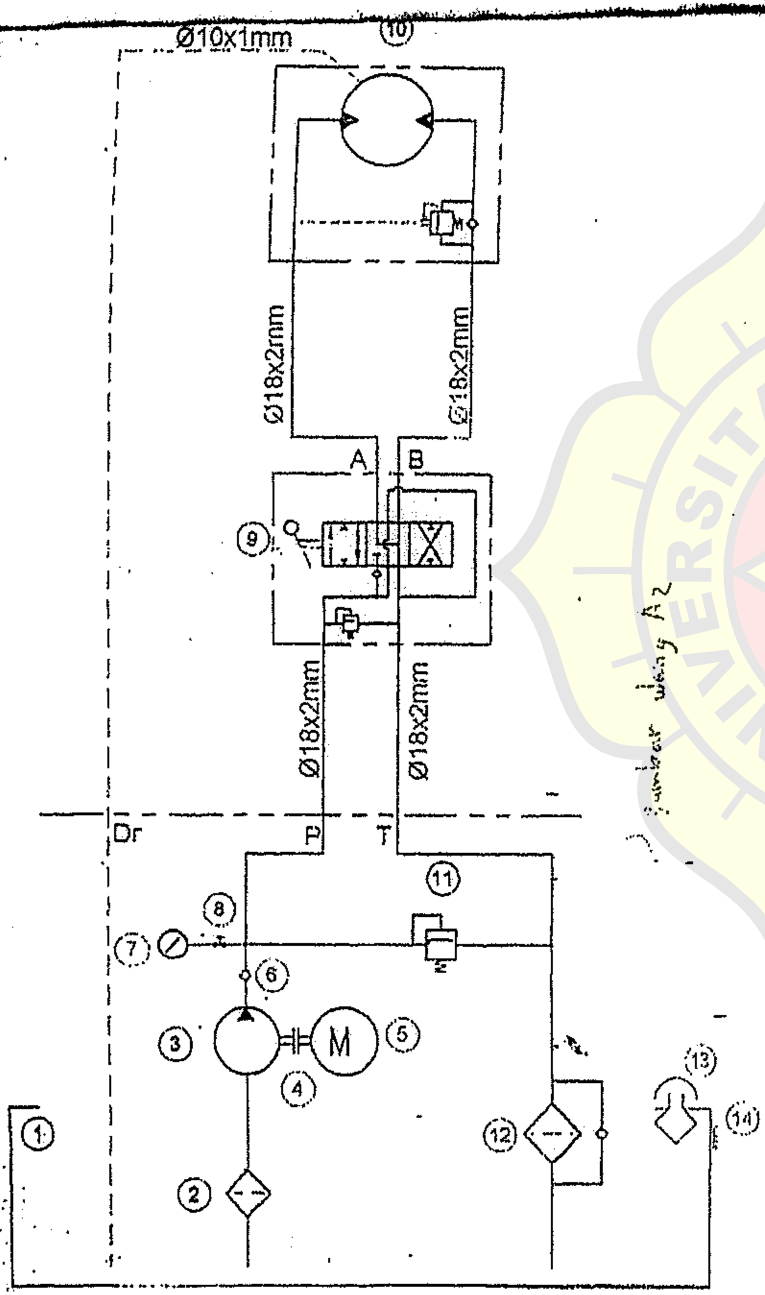
Diperiksa

Disetujui

PT. PINDAD (PERSERO)

Nomor Gambar: GP1WW02-08.1187-0

Lembar ke: 1/1



PIPE MATERIAL : Seamless St 35.4 High quality tube, free of scale (NBK) and phosphated and oiled

keterangan:

1. Work Pressure: ~140 bar
2. Power : Electric motor 15 KW

FOR APPROVAL

14	Oil level gauge	1	UC-FLT- 69221	
13	Air breather	1	UC-AB-1163	
12	Return filter	1	UC-MX-1591-101	
11	Pressure relief valve	1	CT-06-F .	
10	Hydraulic motor	1	IHM3-6-400B-D601	
9	Directional control valve	1	CM11-N02R30BL	
8	Shut off valve	1	DTNG-04 1/4	
7	Pressure gauge	1	UC-PG -4511120	(0-250bar
6	Check Valve	1	DT B PI -06	
5	Electric motor	1	11KW/380V/50Hz	IP 55
4	Drive Coupling	1	UC-DC-42P8	
3	Hydraulic pump	1	20VQ-5	
2	Suction Filter	1	UC-SE-1323	
1	Oil Tank 150L	1		GP3HD03-07.217

NAMA
HYDRAULIC DIAGRAM
FOR RAMPWINCH 800 GRT DECK MACHINERY

Digambar: zie — Tgl.20.
Diperiksa — Tgl.