BAB V

PENUTUP

Dari perencanaan yang telah dilakukan terhadap kapal General Cargo 6000 DWT dengan ukuran kapal :

- Panjang antara garis tegak (Lpp) : 98,68 m
- Lebar kapal (B) : 16,33 m
- Tinggi kapal (H) : 8,81 m
- Sarat air (T) : 7,15 m
- Koefisien blok (Cb) : 0,71 m
- Kecepatan (Vs) : 15 knot
- Klasifikasi : BKI
- Jarak pelayaran : 1801 mil

Maka diambil kesimpulan sebagai berikut :

- Besarnya daya continous rating (MCR) yang diperlukan agar kapal dapat mencapai kecepatan 15 knot adalah 6450 kw.
- Motor penggerak utama dipilih mesin diesel empat langkah dengan spesifikasi sebagai berikut :

  Merk : Wartsila

  Type : 16V32

Daya : 6450 kw / 8810 HP

Jumlah silinder : 16 V

Putaran : 720 rpm

SFOC : 177 gr / kW.h

Bore x Stroke : 320 mm x 350 mm

Ukuran : Panjang x Lebar x Tinggi

: 6883 mm x 2765 mm x 2360 mm

Jumlah : 1 Buah dan mesin dipasang diburitan kapal

- Untuk mendukung kegiatan kapal selama berlayar dibutuhkan Genset sebesar 478,30 kW dengan spesifikasi sebagai berikut:

Merk : Yanmar

Type : 6NY16L-DN

Gen. Capacity : 3 × 280 kw

Putaran mesin : 1200 Rpm

No. of Cylinder : 6

Jumlah : 3 Buah

D = 4200 mm

a. Panjang Blade Elemen Dari Centre Line ke Leading Edge (h_D)

r/R (1)	h_D / D (2)	h_D (3)
0,2	0.116	487.200
0,3	0.129	541.800
0,4	0.136	571.200
0,5	0.137	575.400
0,6	0.132	554.400
0,7	0.118	495.600
0,8	0.092	386.400
0,9	0.051	214.200
0,95	0.020	84.000
1,00	-0.053	-222.600

b. Panjang Total Blade Elemen (C)

r/R (1)	C / D (2)	C (3)
0,2	0.208	873.600
0,3	0.241	1012.200
0,4	0.263	1104.600
0,5	0.276	1159.200
0,6	0.279	1171.800
0,7	0.269	1129.800
0,8	0.241	1012.200
0,9	0.184	772.800
0,95	0.135	567.000
1,00	0.000	0.000

c. Panjang Blade Elemen Dari Centre Line ke Trailing Edge (h_{TE})

r/R (1)	h_{TE} = C - h_D (2)
0,2	386.400
0,3	470.400
0,4	533.400
0,5	583.800
0,6	617.400
0,7	634.200
0,8	625.800
0,9	558.600
0,95	483.000
1,00	222.600

d. Jarak Ordinate Maksimum dari Leading Edge (h_T)

r/R (1)	h_T / C (2)	h_T (3)
0,2	0.350	305.760
0,3	0.387	391.721
0,4	0.420	463.932
0,5	0.450	521.640
0,6	0.475	556.605
0,7	0.493	556.991
0,8	0.500	506.100
0,9	0.500	386.400
0,95	0.500	283.500
1,00	0.500	0.000

e. Ketebalan Blade Maksimum Pada Ordinate (t)

r/R (1)	t / D (2)	t (3)
0,2	0.0366	153.720
0,3	0.0324	136.080
0,4	0.0282	118.440
0,5	0.0240	100.800
0,6	0.0198	83.160
0,7	0.0156	65.520
0,8	0.0114	47.880
0,9	0.0072	30.240
0,95	0.0051	21.420
1,00	0.0030	12.600

Jarak Ordinat Belakang & Muka Dari Ordinate Maksimum
Ordinate Belakang

Trailing Edge

r / R	T.E	mm	80	mm	60	mm	40	mm	20	mm	y_1	mm	y_2	mm
0,2	0.386	59.336	0.63	96.844	0.805	123.745	0.919	141.269	0.981	150.799	0.380	58.414	0.375	57.645
0,3	0.338	45.995	0.598	81.376	0.787	107.095	0.911	123.969	0.979	133.222	0.343	46.675	0.325	44.226
0,4	0.289	34.229	0.565	66.919	0.769	91.080	0.903	106.951	0.977	115.716	0.307	36.361	0.274	32.453
0,5	0.233	23.486	0.521	52.517	0.742	74.794	0.892	89.914	0.975	98.280	0.270	27.216	0.218	21.974
0,6	0.171	14.220	0.477	39.667	0.712	59.210	0.875	72.765	0.97	80.665	0.000	0.000	0.151	12.557
0,7	0.102	6.683	0.436	28.567	0.687	45.012	0.859	56.282	0.965	63.227	0.000	0.000	0.076	4.980
0,8	0.073	3.495	0.407	19.487	0.669	32.032	0.852	40.794	0.963	46.108	0.000	0.000	0.037	1.772
0,9	0.116	3.508	0.434	13.124	0.682	20.624	0.859	25.976	0.965	29.182	0.000	0.000	0.058	1.754
0,95	0.163	3.491	0.464	9.939	0.699	14.973	0.866	18.550	0.967	20.713	0.000	0.000	0.082	1.756

Leading Edge

r / R	20	mm	40	mm	60	mm	70	mm	80	mm	85	mm	90	mm	95	mm	LE	mm
0,2	0.984	151.260	0.932	143.267	0.844	129.740	0.783	120.363	0.708	108.834	0.662	101.763	0.608	93.462	0.538	82.701	0.000	0.000
0,3	0.981	133.494	0.924	125.738	0.826	112.402	0.759	103.285	0.676	91.990	0.626	85.186	0.569	77.430	0.497	67.632	0.000	0.000
0,4	0.979	115.953	0.915	108.373	0.804	95.226	0.732	86.698	0.637	75.446	0.582	68.932	0.523	61.944	0.444	52.587	0.000	0.000
0,5	0.978	98.582	0.900	90.720	0.774	78.019	0.692	69.754	0.591	59.573	0.531	53.525	0.463	46.670	0.377	38.002	0.000	0.000
0,6	0.975	81.081	0.881	73.264	0.737	61.289	0.647	53.805	0.530	44.075	0.465	38.669	0.386	32.100	0.298	24.782	0.171	14.220
0,7	0.968	63.423	0.866	56.740	0.698	45.733	0.590	38.657	0.465	30.467	0.390	25.553	0.305	19.984	0.210	13.759	0.102	6.683
0,8	0.963	46.108	0.852	40.794	0.669	32.032	0.546	26.142	0.407	19.487	0.330	15.800	0.249	11.922	0.163	7.804	0.073	3.495
0,9	0.965	29.182	0.859	25.976	0.682	20.624	0.567	17.146	0.434	13.124	0.361	10.917	0.284	8.588	0.202	6.108	0.116	3.508
0,95	0.967	20.713	0.866	18.550	0.699	14.973	0.590	12.638	0.464	9.939	0.395	8.461	0.322	6.897	0.245	5.248	0.163	3.491

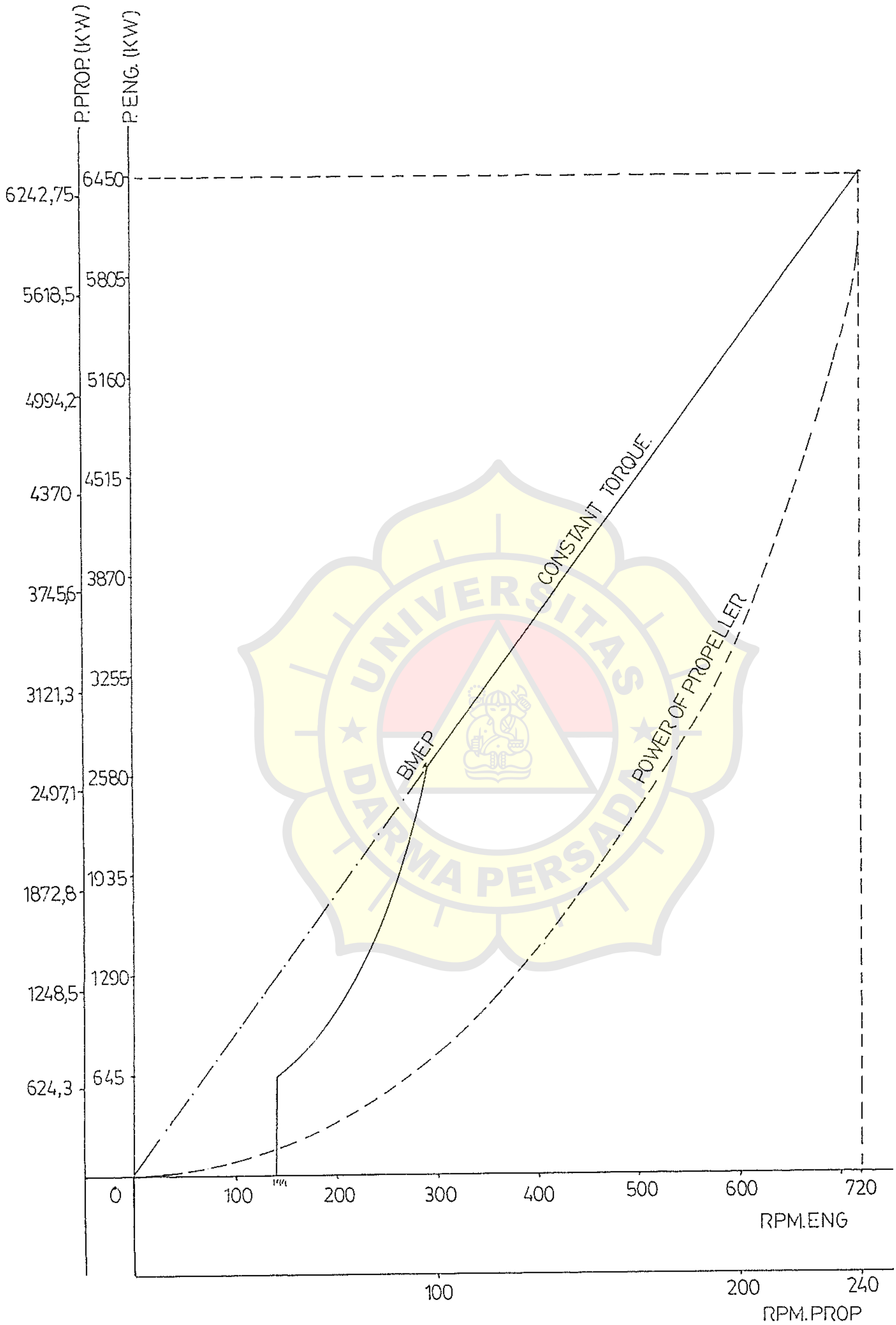
2. Ordinat Muka

Trailing Edge

r / R	T.E	mm	80	mm	60	mm	40	mm	20	mm
0,2	0.364	55.954	0.210	32.281	0.105	16.141	0.041	6.303	0.009	1.383
0,3	0.312	42.457	0.178	24.222	0.087	11.839	0.033	4.491	0.007	0.953
0,4	0.259	30.676	0.139	16.463	0.068	8.054	0.025	2.961	0.005	0.592
0,5	0.199	20.059	0.100	10.080	0.045	4.536	0.013	1.310	0.003	0.302
0,6	0.129	10.728	0.050	4.158	0.016	1.331	0.000	0.000	0.000	0.000
0,7	0.048	3.145	0.013	0.852	0.000	0.000	0.000	0.000	0.000	0.000
0,8	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
0,9	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
0,95	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000

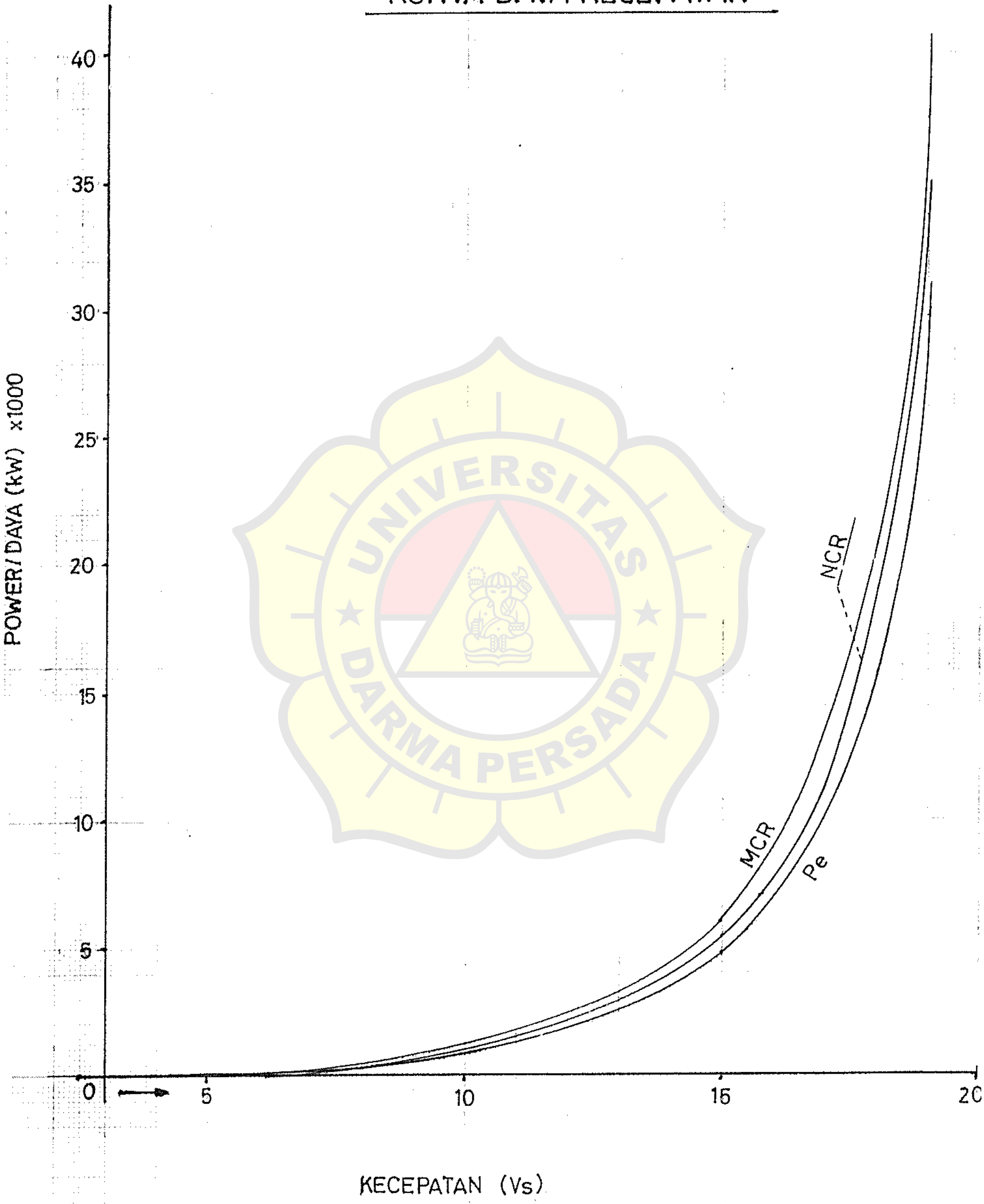
Leading Edge

r / R	20	mm	40	mm	60	mm	70	mm	80	mm	85	mm	90	mm	95	mm	LE	mm
0,2	0.007	1.076	0.029	4.458	0.066	10.146	0.094	14.450	0.132	20.291	0.157	24.134	0.190	29.207	0.237	36.432	0.000	0.000
0,3	0.005	0.680	0.020	2.722	0.048	6.532	0.070	9.526	0.100	13.608	0.121	16.466	0.149	20.276	0.192	26.127	0.000	0.000
0,4	0.003	0.355	0.012	1.421	0.030	3.553	0.045	5.330	0.067	7.935	0.084	9.949	0.108	12.792	0.146	17.292	0.000	0.000
0,5	0.000	0.000	0.004	0.403	0.012	1.210	0.021	2.117	0.037	3.730	0.045	4.536	0.068	6.854	0.101	10.181	0.000	0.000
0,6	0.000	0.000	0.000	0.000	0.000	0.000	0.003	0.249	0.012	0.998	0.020	1.663	0.034	2.827	0.058	4.823	0.129	10.728
0,7	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.002	0.131	0.004	0.262	0.012	0.786	0.048	3.145
0,8	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
0,9	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000
0,95	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000	0.000



MATCHING OF ENG.-PROP. CURVE

KURVA DAYA KECEPATAN



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Bab

Lima

PENENTUAN TAHANAN KAPAL

5.1. PENDAHULUAN

Dalam membuat usulan awal untuk kapal baru atau melakukan studi transportasi, pertanyaan vital yang sering dihadapi pemilik kapal, arsitek kapal, politikus, ahli ekonomi, atau mahasiswa adalah besarnya daya yang diperlukan. Jawabannya dapat dicari dengan berbagai cara. Seperti halnya dalam perancangan awal kapal, ada tiga kelompok yang dapat dipilih :

Metode kapal pembanding
Metode statistik
Metode satu per satu

Jika memakai metode yang pertama maka harus dipilih suatu kapal pembanding. Kapal pembanding ini harus merupakan jenis yang sama dengan jenis kapal yang disyaratkan dalam usulan. Selain itu, ukuran utama dan kecepatan kapal pembanding tersebut harus tidak jauh berbeda dengan yang diharapkan untuk kapal yang akan diusulkan. Koefisien admiralty A_c untuk kapal pembanding dihitung dengan memakai rumus

$$A_c = \frac{\Delta^{2/3} V^3}{P} \tag{5.1.1}$$

P adalah daya yang diperlukan untuk menggerakkan kapal pada displasemen Δ dan kecepatan V. Kemudian daya P_p untuk kapal yang diusulkan dapat dihitung dengan

$$P_p = \frac{\Delta_p^{2/3} V_p^3}{A_c} \tag{5.1.2}$$

Δ_p dan V_p masing-masing adalah displasemen dan kecepatan kapal yang diusulkan. Di sini daya yang diperlukan dianggap berbanding lurus dengan tahanan total kapal.

Bila memakai metode yang kedua maka data propulsi dari seperangkat kapal dikumpulkan dan dipelajari statistiknya. Hasilnya dapat diberikan berupa program untuk perhitungan atau seperangkat diagram yang menyatakan daya sebagai fungsi dari, mungkin, koefisien blok, displasemen, dan rasio panjang–displasemen. Seperangkat diagram semacam itu dapat dilihat di Bab 9.

Berbeda dengan kedua metode tadi, dalam metode yang ketiga tahanan kapal yang diusulkan itu sendirilah yang harus diketahui. Tahanan ini dapat diperkirakan dengan berbagai cara. Gagasan melakukan percobaan model di air untuk memperkirakan tahanan kapal berukuran penuh sebagaimana disebutkan di Bab 3 merupakan gagasan yang sudah timbul sejak lama, yaitu mulai dari sekitar tahun 1500 (Turnisi, 1953), namun demikian hingga tahun 1868 tidak ada metode yang dapat dipakai untuk mentransformasi data model ke kapal yang sebenarnya (Stoot, 1959). William Froude kemudian mengusulkan hukum perbandingannya dan menunjukkan cara pemakaiannya dalam praktek untuk memprakirakan tahanan kapal dari hasil model. Dalam bab ini akan diuraikan dan dibahas metode Froude dan metode yang paling akhir untuk menentukan tahanan kapal.

Tambahan kelonggaran ini sangat tergantung pada jalur pelayaran. Kelonggaran rata-rata untuk pelayaran dinas (kadang-kadang disebut sea margin atau service margin) untuk tahanan atau daya efektif diusulkan sebagai berikut :

Jalur pelayaran Atlantik Utara, ke Timur, untuk musim panas 15% dan musim dingin 20%

Jalur pelayaran Atlantik Utara, ke Barat, untuk musim panas 20% dan musim dingin 30% (5.5.28)

Jalur pelayaran Pasifik, 15 – 30%

Jalur pelayaran Atlantik Selatan dan Autralia, 12 – 18%

Jalur pelayaran Asia Timur, 15 – 20%

Tahanan total harus dihitung dengan memakai rumus

$$R_T = C_T(\tfrac{1}{2}\rho V^2 S) \tag{5.5.29}$$

S adalah luas permukaan basah badan kapal.

Banyak sekali metode untuk memperkirakan S. Dianjurkan untuk memakai salah satu dari dua metode berikut ini :

1. Publikasi FORMDATA I – V (Guldhammer, 1962, 1963, 1967, 1969, 1973) memuat data hidrostatik dari suatu seri yang sangat baik yang terdiri dari sejumlah bentuk kapal yang divariasikan secara sistematik. Permukaan basah semua bentuk memakai koefisien

$$\boxed{S} = \frac{S}{L(B + 2{,}5T)} \tag{5.5.30}$$

Jika dari perancangan awal kapal bentuk badan kapal yang sebenarnya hampir sama dengan salah satu bentuk yang diberikan dalam FORMDATA tersebut maka akan diperoleh S dengan kesalahan kurang dari 1%.

2. Permukaan basah untuk kapal niaga yang normal dapat dihitung dengan memakai rumus berikut ini (versi rumus Mumford) :

$$S = 1{,}025 L_{pp}(\delta_{pp} B + 1{,}7T) \tag{5.5.31}$$

Semua diagram $\boxed{S}$ dan rumus yang disertakan dalam FORMDATA berlaku untuk bentuk kapal yang buritan dan haluannya masing-masing terletak pada garis tegak belakang dan garis tegak depan. Hampir semua kapal mempunyai luas permukaan basah yang sesuai dengan asumsi tersebut, karena luas yang kurang dan luas yang lebih akan saling berimbang. Untuk kapal yang mempunyai juntaian (= bagian yang menggantung = overhang), atau lekukan (= bagian yang masuk = cutout), di dalam air yang besar maka hal tersebut harus diperhitungkan (diberikan kelonggaran).

Perhitungan tahanan dan daya efektif dapat dilakukan dengan prosedur seperti yang ditunjukkan dalam Contoh Formulir untuk Menghitung Daya Efektif (lihat halaman 132). Perhitungan dapat dilakukan dengan memakai komputer mini. Kini banyak arsitek kapal yang mempunyai program untuk perhitungan demikian itu.

Dalam tahap perancangan pertanyaan utama yang harus dituntaskan adalah jenis dan ukuran mesin (misalnya banyaknya dan ukuran silinder, jika memakai mesin disel). Tahanan harus ditentukan dengan tingkat kepastian yang memadai sehingga, atas dasar daya efektif P_E, daya poros akan dapat ditentukan dengan tingkat ketepatan yang cukup untuk dapat menjawab dengan aman pertanyaan vital tadi.

Di lain pihak, upaya untuk mencapai ketepatan yang melebihi dari yang diperlukan untuk menyelesaikan masalah tahanan tidak mempunyai arti yang besar. Tingkat ketidakpastian dalam faktor yang terlibat cukup tinggi, dan pembaca diingatkan untuk tidak membuang waktu untuk memburu ketepatan yang tersisa dengan perhitungan yang sifatnya hanya pendekatan.

Untuk kapal yang bertenaga mesin disel, merubah jumlah silinder, katakanlah dari 6 menjadi 7, atau dari 11 menjadi 12, akan berarti merubah daya masing-masing sebesar sekitar 17% atau 8%. Dengan memodifikasi tekanan efektif rata-rata dan jumlah kisaran maka akan dapat merubah luaran menerus (continuous output) sebesar sekitar 10%.

Mesin turbin mempunyai tingkatan menurut jenisnya

Atas dasar pertimbangan tersebut barangkali tingkat ketepatan yang diperlukan dalam penentuan P_E untuk perancangan awal kapal dapat ditentukan sebesar 1 hingga 5%. Ketepatan ini dapat dengan mudah diperoleh dengan memakai diagram dan formulir Perhitungan yang diberikan di sini

Diagram dan rumus tersebut dapat pula dipakai dengan cara sebagai berikut. Setiap hasil yang diperoleh dari percobaan yang dilakukan sendiri oleh si arsitek kapal di tangki percobaan dicocokkan dengan diagram. Data ini kemudian dikoreksi dengan memakai rumus dan diagram tadi dan selanjutnya dipakai sebagai dasar materi untuk menentukan tahanan kapal baru yang akan diajukan dalam usulan. Sering bahwa dengan prosedur demikian ini dapat diperoleh hasil yang sangat baik.

ANGGOTA BADAN KAPAL

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

TAHANAN TAMBAHAN

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

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

(5.5.23)

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

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

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

ANGGOTA BADAN KAPAL

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

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

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

TAHANAN UDARA DAN TAHANAN KEMUDI

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

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

Koreksi untuk tahanan kemudi mungkin sekitar

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

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

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

KONDISI PELAYARAN DINAS

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

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

BENTUK BADAN KAPAL (BENTUK PENAMPANG MELINTANG DAN HALUAN)

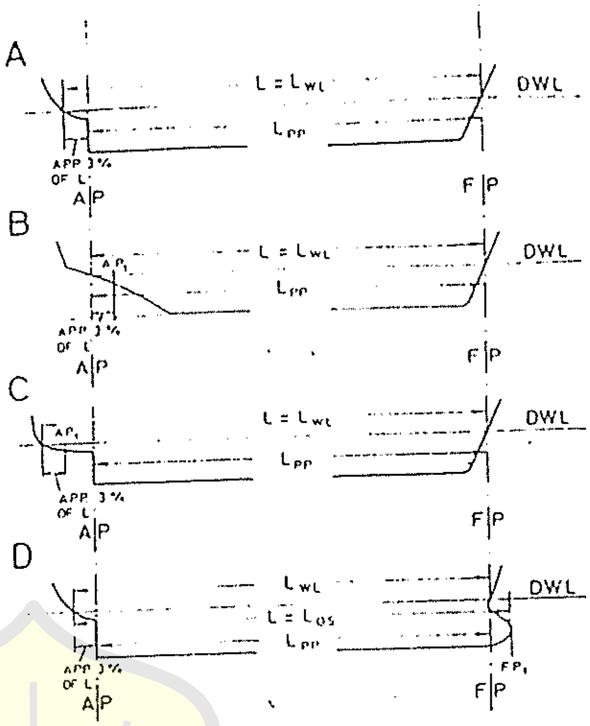
Sebagaimana disebutkan sebelumnya, kurva tahanan (yang diperoleh berdasarkan Gb. 5.5.5 – 5.5.13) dianggap berlaku untuk yang mempunyai bentuk "standar", yaitu penampangnya bukan yang benar-benar berbentuk U ataupun V. Karena itu, dalam menghitung daya efektif untuk perancangan awal umumnya tidak diperlukan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

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

(5.5.20)

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

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



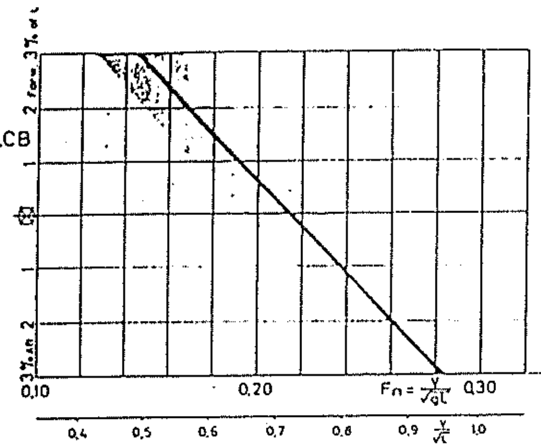
Gambar 5.5.17. Definisi L dan LCB. (a) Bentuk normal. Panjang buritan pada garis air umumnya 3% L. (b) Badan kapal tanpa linggi buritan (sternpost). AP umumnya diletakkan di ujung belakang DWL. Untuk koreksi LCB dipakai AP_1 3% L di depan ujung belakang garis air. (c) Badan kapal dengan panjang buritan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

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

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

F_n = 0,15	0,18	0,21 0,253	0,24 0,253	0,27	0,30	0,33	0,36	φ
		+ 0,2	0	– 0,2	– 0,4	– 0,4	– 0,4	0,50
		+ 0,2	0	– 0,2	– 0,3	– 0,3	[illegible]	0,60
	+ 0,2	0	– 0,2	– 0,3	– 0,3	[illegible]	[illegible]	0,70
+ 0,1	0	– 0,2				[illegible]	[illegible]	0,80

(5.5.21)



Gambar 5.5.15. LCB standar. Letak longitudinal titik benam yang dipandang terbaik.

Dalam hal ini, LCB standar tersebut didefinisikan sebagai fungsi linier angka Froude F_n. Karena tidak adanya ketergantungan yang pasti pada parameter lainnya yang tercatat maka LCB standar tersebut disajikan sebagai garis tunggal. Daerah yang diberi warna gelap di sekitar garis ini menunjukkan lingkup materi yang dikaji.

Sebagaimana disebutkan sebelumnya, karena letak LCB standar dianggap merupakan letak yang memberikan tahanan yang paling kecil maka letak yang lain pada prinsipnya akan memberikan tahanan yang lebih besar. Penambahan tahanan tersebut harus dicari dengan jalan mengalikan penyimpangan LCB dari standar, yaitu

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

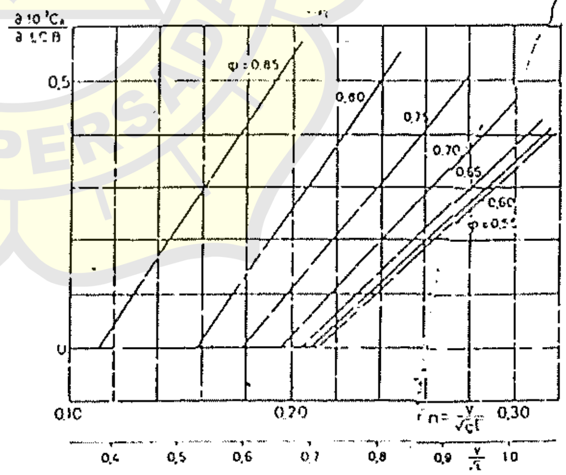
dengan faktor $\partial 10^3 C_R / \partial LCB$. Harga faktor ini dapat diperoleh dari Gb. 5.5.16, dan ini hanya berlaku untuk LCB yang berada di depan $LCB_{standar}$. Mengenai LCB yang berada di belakang $LCB_{standar}$, semua sumber yang ada mempunyai pendapat yang saling bertentangan. Namun demikian, karena kecenderungan terjadinya letak demikian itu sangat kecil maka pengabaian koreksi dalam hal itu tidak akan memberikan kesalahan yang berarti.

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

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

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

LCB didefinisikan sebagai letak longitudinal titik benam, yaitu jarak antara titik ini dengan penampang tengah kapal, dan positif di belakang penampang tersebut. Midship section (penampang melintang tengah kapal, atau penampang tengah kapal, atau bidang tengah kapal, atau bidang tengah kapal) didefinisikan sebagai penampang melintang yang terletak sejauh 48.5%L dari batas depan displasemen. L adalah



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

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

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

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

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

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

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

Dalam hal ini koefisien tahanan totalnya adalah

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

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

C_F = koefisien tahanan gesek dan dapat dihitung dengan memakai

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

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

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

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

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

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

B/T

Karena diagram tersebut dibuat berdasarkan rasio lebar – sarat

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

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

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

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

Koreksi ini dapat mempunyai harga yang negatif atau positif.

LCB

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

1. Semua data diacukan pada daerah (lingkup) model, dan tahanan model (R_{Tm}) ditentukan sebagai fungsi kecepatan.
2. Koefisien tahanan total spesifik model (C_{Tm}) ditentukan :

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

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

3. Koefisien tahanan sisa spesifik ditentukan dari

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

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

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

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

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

4. C_R dinyatakan sebagai fungsi angka Froude

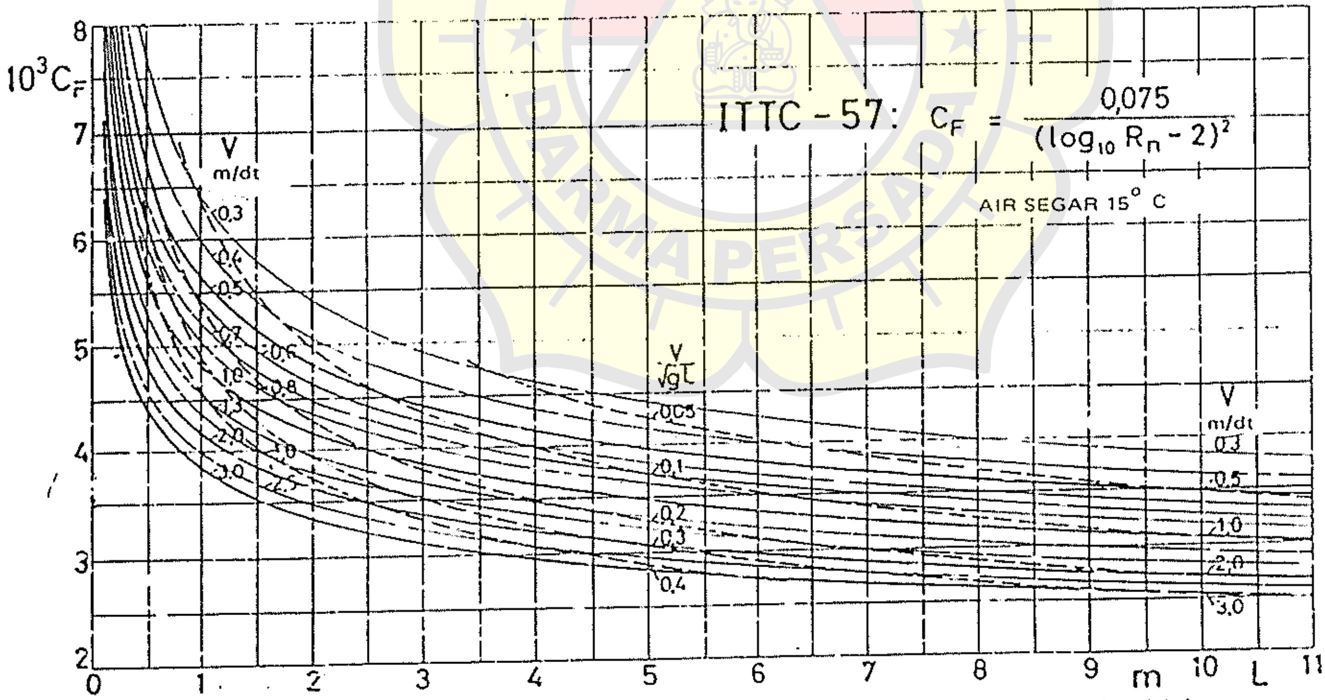
$$F_n = \frac{V}{\sqrt{gL}} \qquad (5.5.9)$$

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

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

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

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



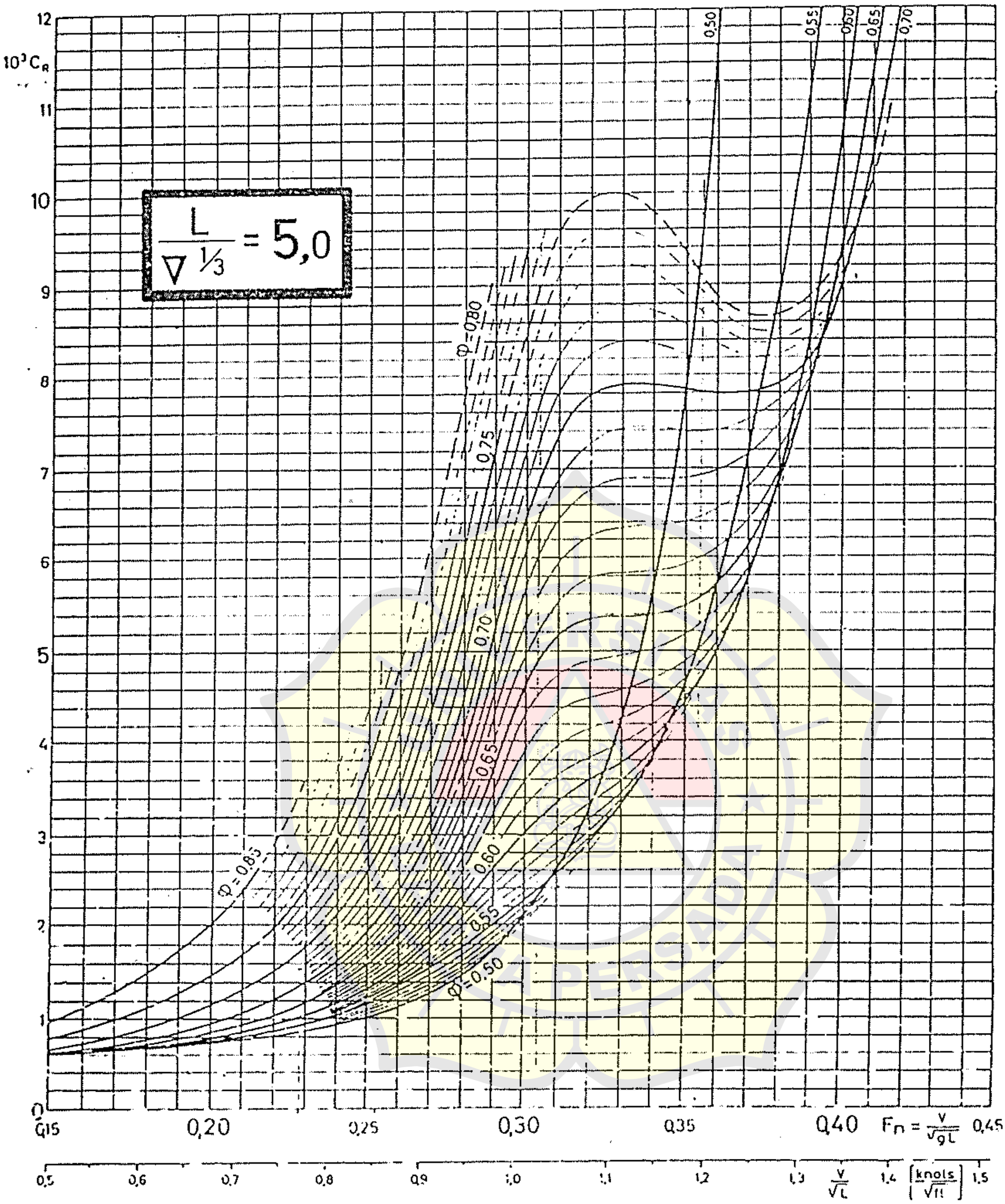
Gambar 5.5.4. Koefisien tahanan gesek C_F (menurut ITTC 1957) sebagai fungsi dari panjang model kapal L dan kecepatan V.

Table 18.2 Anchor, Chain Cables and Ropes

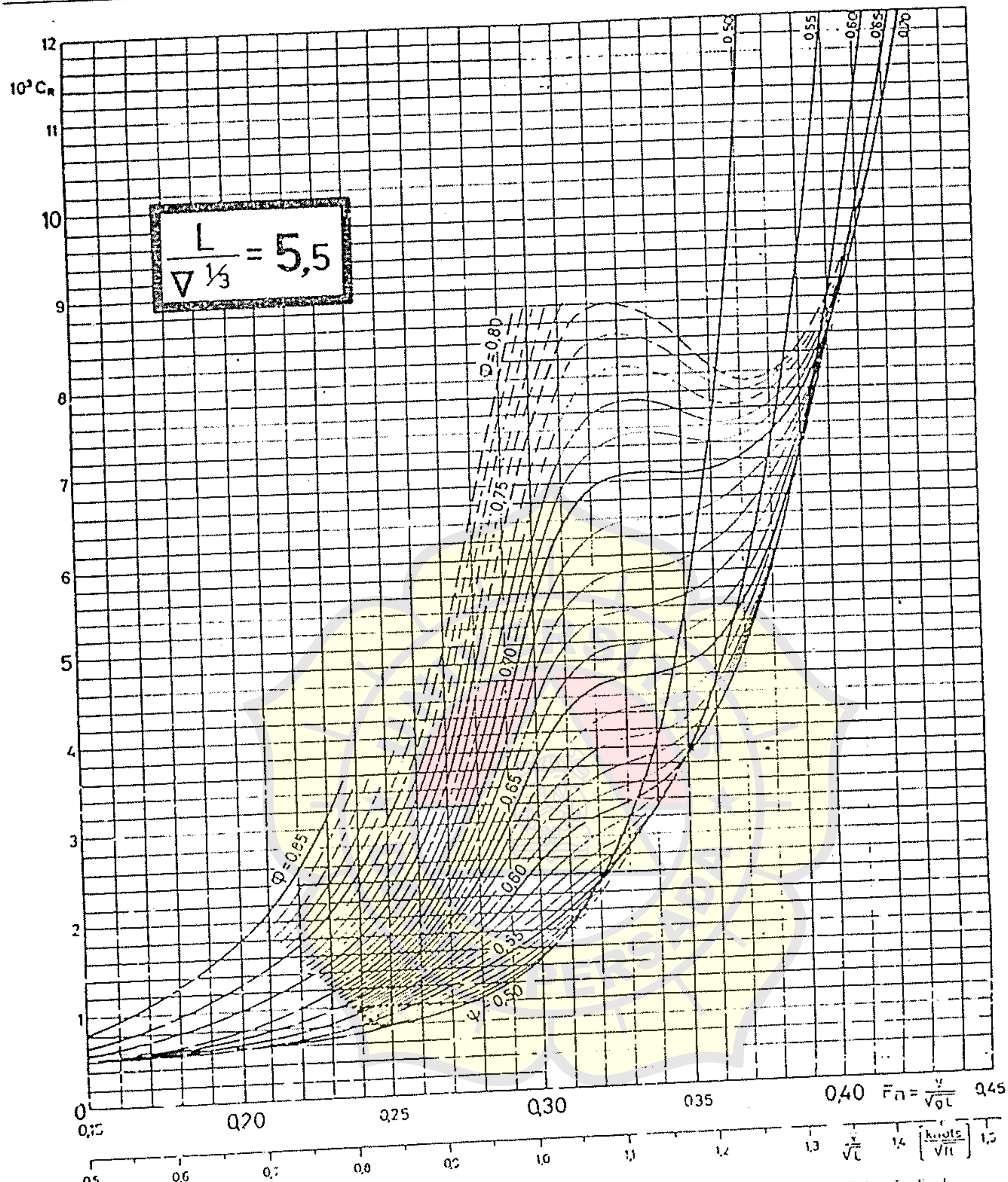
No. for Reg.	Equipment numeral Z	Stockless anchor			Stud link chain cables						Recommended ropes				
		Bower anchor		Stream anchor	Bower anchors				Stream wire or chain for stream anchor		Towline		Mooring ropes		
		Number [1]	Mass per anchor		Total length	Diameter d1	Diameter d2	Diameter d3	Length	Br. load [2]	Length	Br. load [2]	Number	Length	Br. load [2]
			[kg]		[m]	[mm]	[mm]	[mm]	[m]	[kN]	[m]	[kN]		[m]	[kN]
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
101	up to - 50	2	120	40	165	12,5	12,5	12,5	80	65	180	100	3	80	35
102	50 - 70	2	180	60	220	14	12,5	12,5	80	65	180	100	3	80	35
103	70 - 90	2	240	80	220	16	14	14	85	75	180	100	3	100	40
104	90 - 110	2	300	100	247,5	17,5	16	16	85	80	180	100	3	110	40
105	110 - 130	2	360	120	247,5	19	17,5	17,5	90	90	180	100	3	110	45
106	130 - 150	2	420	140	275	20,5	17,5	17,5	90	100	180	100	3	120	50
107	150 - 175	2	480	165	275	22	19	19	90	110	180	100	3	120	55
108	175 - 205	2	570	190	302,5	24	20,5	20,5	90	120	180	110	3	120	60
109	205 - 240	3	660		302,5	26	22	20,5			180	130	4	120	65
110	240 - 280	3	780		330	28	24	22			180	150	4	120	70
111	280 - 320	3	900		357,5	30	26	24			180	175	4	140	80
112	320 - 360	3	1020		357,5	32	28	24			180	200	4	140	85
113	360 - 400	3	1140		385	34	30	26			180	225	4	140	95
114	400 - 450	3	1290		385	36	32	28			180	250	4	140	100
115	450 - 500	3	1440		412,5	38	34	30			180	275	4	140	110
116	500 - 550	3	1590		412,5	40	34	30			190	305	4	160	120
117	550 - 600	3	1740		440	42	36	32			190	340	4	160	130
118	600 - 660	3	1920		440	44	38	34			190	370	4	160	145
119	660 - 720	3	2100		440	46	40	36			190	405	4	160	160
120	720 - 780	3	2280		467,5	48	42	36			190	440	4	170	170
121	780 - 840	3	2460		467,5	50	44	38			190	480	4	170	185
122	840 - 910	3	2640		467,5	52	46	40			190	520	4	170	200
123	910 - 980	3	2850		495	54	48	42			190	560	4	170	215
124	980 - 1060	3	3060		495	56	50	44			200	600	4	180	230
125	1060 - 1140	3	3300		495	58	50	46			200	645	4	180	250
126	1140 - 1220	3	3540		522,5	60	52	46			200	690	4	180	270
127	1220 - 1300	3	3780		522,5	62	54	48			200	740	4	180	285
128	1300 - 1390	3	4050		522,5	64	56	50			200	785	4	180	305
129	1390 - 1480	3	4320		550	66	58	50			200	835	4	180	325
130	1480 - 1570	3	4590		550	68	60	52			220	890	5	190	325
131	1570 - 1670	3	4890		550	70	62	54			220	940	5	190	335
132	1670 - 1790	3	5250		577,5	73	64	56			220	1025	5	190	350
133	1790 - 1930	3	5610		577,5	76	66	58			220	1110	5	190	375
134	1930 - 2080	3	6000		577,5	78	68	60			220	1170	5	190	400
135	2080 - 2230	3	6450		605	81	70	62			240	1260	5	200	425
136	2230 - 2380	3	6900		605	84	73	64			240	1355	5	200	450
137	2380 - 2530	3	7350		605	87	76	66			240	1455	5	200	480
138	2530 - 2700	3	7800		632,5	90	78	68			260	1470	6	200	480
139	2700 - 2870	3	8300		632,5	92	81	70			260	1470	6	200	490
140	2870 - 3040	3	8700		632,5	95	84	73			260	1470	6	200	500
141	3040 - 3210	3	9300		660	97	84	76			280	1470	6	200	520
142	3210 - 3400	3	9900		660	100	87	78			280	1470	6	200	555
143	3400 - 3600	3	10500		660	102	90	78			280	1470	6	200	590
144	3600 - 3800	3	11100		687,5	105	92	81			300	1470	6	200	620
145	3800 - 4000	3	11700		687,5	107	95	84			300	1470	6	200	650
146	4000 - 4200	3	12300		687,5	111	97	87			300	1470	7	200	650
147	4200 - 4400	3	12900		715	114	100	87			300	1470	7	200	660
148	4400 - 4600	3	13500		715	117	102	90			300	1470	7	200	670
149	4600 - 4800	3	14100		715	120	105	92			300	1470	7	200	680
150	4800 - 5000	3	14700		742,5	122	107	95			300	1470	7	200	685
151	5000 - 5200	3	15400		742,5	124	111	97			300	1470	8	200	685
152	5200 - 5500	3	16100		742,5	127	111	97			300	1470	8	200	695
153	5500 - 5800	3	16900		742,5	130	114	100			300	1470	8	200	705
154	5800 - 6100	3	17800		742,5	132	117	102			300	1470	9	200	705
155	6100 - 6500	3	18800		742,5		120	107			300	1470	9	200	715
156	6500 - 6900	3	20000		770		124	111			300	1470	9	200	725
157	6900 - 7400	3	21500		770		127	114			300	1470	10	200	725
158	7400 - 7900	3	23000		770		132	117			300	1470	11	200	725
159	7900 - 8400	3	24500		770		137	122			300	1470	11	200	735
160	8400 - 8900	3	26000		770		142	127			300	1470	12	200	735
161	8900 - 9400	3	27500		770		147	132			300	1470	13	200	735
162	9400 - 10000	3	29000		770		152	132			300	1470	14	200	735
163	10000 - 10700	3	31000		770			137			300	1470	15	200	735
164	10700 - 11500	3	33000		770			142			300	1470	16	200	735
165	11500 - 12400	3	35500		770			147			300	1470	17	200	735
166	12400 - 13400	3	38500		770			152			300	1470	18	200	735
167	13400 - 14600	3	42000		770			157			300	1470	19	200	735
168	14600 - 16000	3	46000		770			162			300	1470	21	200	735

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

1 see C.1.
2 see F.1.2



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



Gambar 5.5.8. Koefisien tahanan sisa terhadap rasio kecepatan–panjang untuk harga koefisien prismatik longitudinal yang berbeda-beda. $L/\nabla^{1/3} = 5,5$.

- external thread diameter:

$d_g = 0{,}65 \cdot d_0$

See Fig. 14.6.

4.1.6 It is to be proved that 50% of the design yield moment will be solely transmitted by friction in the cone couplings. This can be done by calculating the required push-up pressure and push-up length according to 4.2.3 for a torsional moment $Q'_F = 0{,}5 \cdot Q_F$

4.2 Cone couplings with special arrangements for mounting and dismounting the couplings

4.2.1 Where the stock diameter exceeds 200 mm the press fit is recommended to be effected by a hydraulic pressure connection. In such cases the cone should be more slender (c ≈ 1 : 12 to ≈ 1 : 20).

4.2.2 In case of hydraulic pressure connections the nut is to be effectively secured against the rudder stock or the pintle. A securing plate for securing the nut against the rudder body is not to be provided, see Fig. 14.7.

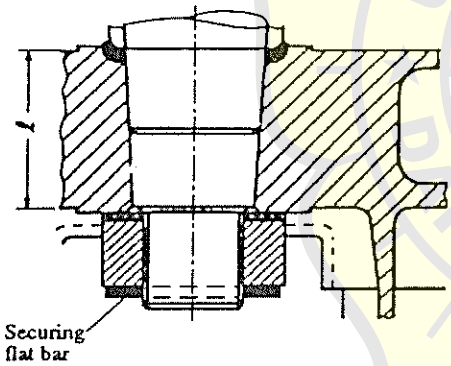


Fig. 14.7

4.2.3 For the safe transmission of the torsional moment by the coupling between rudder stock and rudder body the required push-up length and the push-up pressure are to be determined by the following formulae:

.1 required push-up pressure

$$p_{req1} = \frac{2 \cdot Q_F \cdot 10^3}{d_m^{\,2} \cdot \ell \cdot \pi \cdot \mu_0} \quad [\mathrm{N/mm^2}]$$

or

$$p_{req2} = \frac{6 \cdot M_b \cdot 10^3}{\ell^2 \cdot d_m} \quad [\mathrm{N/mm^2}]$$

Q_F = design yield moment of rudder stock according to F. in [Nm]

d_m = mean cone diameter in [mm]

ℓ = cone length in [mm]

μ_0 = 0,15 (frictional coefficient)

M_b = bending moment in the cone coupling (e.g. case of spade rudders) in [Nm].

The greater of the values p_{req1} or p_{req2} is to be taken.

It has to be proved that the required push-up pressure does not exceed the permissible surface pressure in the cone. The permissible surface pressure is to be determined by the following formula:

$$p_{perm} = \frac{0{,}8 \cdot R_{eH}(1 - \alpha^2)}{\sqrt{3} \cdot \alpha^4} \qquad [\mathrm{N/mm^2}]$$

R_{eH} = yield point in [N/mm²] of the material of the gudgeon

α = d_m/d_a (see Fig 14.6)

The outer diameter of the gudgeon should not be less than:

$d_a = 1{,}5 \cdot d_m$ [mm]

.2 required push-up length

$$\Delta\ell = \frac{p_{req} \cdot d_m}{E\left[\frac{1 - \alpha^2}{2}\right]c} + 0{,}8\,\frac{R_{tm}}{c} \quad [\mathrm{mm}]$$

R_{tm} = mean roughness in [mm]

R_{tm} ≈ 0,01 mm

c = taper on diameter according to 4.2.1

E = Young's modulus ($2{,}06 \cdot 10^5$ N/mm²)

A guidance figure for the minimum push-up length is:

$\Delta\ell_{min} = d_m/150$ [mm].

This value is not to be greater than:

$$\Delta\ell = \frac{1{,}62 \cdot R_{eH} \cdot d_m}{\sqrt{3 + \alpha^4}\; E \cdot c} + 0{,}8\,\frac{R_{tm}}{c} \quad [\mathrm{mm}]$$

Guidance

In case of hydraulic pressure connections the required push-up force P_e for the cone may be determined by the following formula:

$P_e = P_{req} \cdot d_m \cdot \pi \cdot \ell \, (c/2 + 0{,}02)$ [N]

Where due to the fitting procedure a partial push-up effect caused by the rudder weight is given, this may be taken into account when fixing the required push-up length, subject to approval by BKI.

4.2.4 The required push-up pressure for pintle bearings is to be determined by the following formula:

$$p_{req} = 0{,}4 \frac{B_1 \cdot d_0}{d_m^{\,2} \cdot \ell} \quad [N/mm^2]$$

B_1 = supporting force in the pintle bearing in [N], see also Fig. 14.4

d_m, ℓ see 4.2.3

d_0 = pintle diameter in [mm] according to Fig. 14.6.

E. Rudder Body, Rudder Bearings

1. Strength of rudder body

1.1 The rudder body is to be stiffened by horizontal and vertical webs in such a manner that the udder body will be effective as a beam. The rudder hould be additionally stiffened at the aft edge.

.2 The strength of the rudder body is to be roved by direct calculation according to C.3.

.3 For rudder bodies without cut-outs the permissible stress are limited to:

ending stress due to M_R:

$_b$ = 110 N/mm²

ıear stress due to Q_1:

= 50 N/mm²

ıuivalent stress due to bending and shear:

$$= \sqrt{\sigma_b^{\,2} + 3\tau^2} = 120 \quad N/mm^2$$

$_R$, Q_1 see C.3.3.

case of openings in the rudder plating for access to ne coupling or pintle nut the permissible stresses cording to 1.4 apply. Smaller permissible stress lues may be required if the corner radii are less than 15 · h, where h = height of opening.

| In rudder bodies with cut-outs (semi-spade lders) the following stress values are not to be exceeded:

bending stress due to M_R

σ_b = 90 N/mm²

shear stress due to Q_1:

τ = 50 N/mm²

torsional stress due to M_t

τ_t = 50 N/mm²

equivalent stress due to bending and shear and equivalent stress due to bending and torsion:

$$\sigma_{v1} = \sqrt{\sigma_b^{\,2} + 3\tau^2} = 120 \quad N/mm^2$$

$$\sigma_{v2} = \sqrt{\sigma_b^{\,2} + 3\tau^2} = 100 \quad N/mm^2$$

$$M_R = C_{R2} \cdot f_1 + B_1 \frac{f_2}{2} \quad [Nm]$$

$$Q_1 = C_{R2} \quad [N]$$

f_1, f_2 see Fig. 14.8.

The torsional stress may be calculated in a simplified manner as follows:

$$\tau_t = \frac{M_t}{2 \cdot \ell \cdot h \cdot t} \quad [N/mm^2]$$

$M_t = C_{R2} \cdot e$ [Nm]

C_{R2} = partial rudder force in [N] of the partial rudder area A_2 below the cross section under consideration

e = lever for torsional moment in [m] (horizontal distance between the centroid of area A_2 and the centre line a-a of the effective cross sectional area under consideration, see Fig. 14.8. The centroid is to be assumed at $0{,}33 \cdot c_2$ aft of the forward edge of area A_2, where c = mean breadth of area A_2)

h, ℓ, t in [cm], see Fig. 14.8.

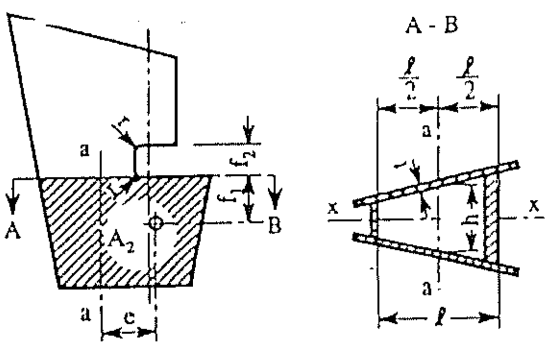


Fig. 14.8

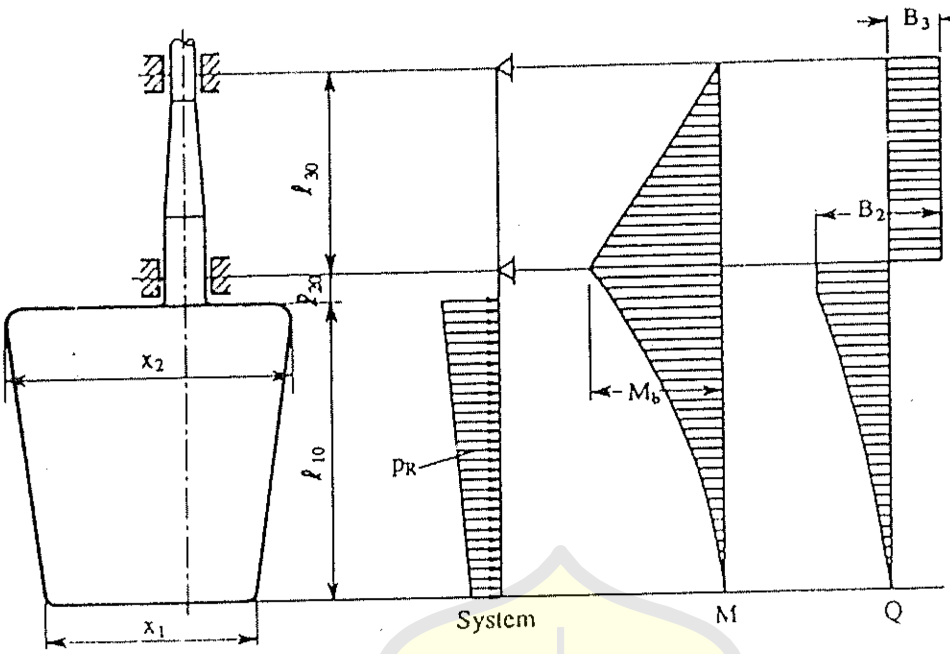


Fig. 14.5 Spade rudder

D. Rudder Couplings

1. General

1.1 The couplings are to be designed in such a way as to enable them to transmit the full torque of the rudder stock.

1.2 The distance of bolt axis from the edges of the flange is not to be less than 1,2 the diameter of the bolt. In horizontal couplings, at least 2 bolts are to be arranged forward of the stock axis.

1.3 The coupling bolts are to be fitted bolts. Their nuts are to be effectively locked, e.g., according to recognized standards.

1.4 For spade rudders horizontal couplings according to 2. are permissible only where the required thickness of the coupling flanges t_f is less than 50 mm, other wise cone couplings according to 4. are to be applied. For spade rudders of the high lift type, only cone couplings according to 4. are permitted.

2. Horizontal couplings

2.1 The diameter of coupling bolts is not to be less than:

$$d_b = 0{,}62 \sqrt{\frac{D^3 \cdot k_b}{k_r \cdot n \cdot e}} \quad [\text{mm}]$$

D = rudder stock diameter according to C. in [mm]

n = total number of bolts, which is not to be less than 6

e = mean distance of the bolt axes from the centre of bolt system in [mm]

k_r = material factor for the rudder stock as given in A.4.2

k_b = material factor for the bolts analogue to A.4.2.

2.2 The thickness of the coupling flanges is not to be less than determined by the following formulae:

$$t_f = 0{,}62 \sqrt{\frac{D^3 \cdot k_f}{k_r \cdot n \cdot e}} \quad [\text{mm}]$$

$$t_{fmin} = 0{,}9 \cdot d_b$$

k_f = material factor for the coupling flanges analogue to A.4.2.

The thickness of the coupling flanges clear of the bolt holes is not to be less than $0{,}65 \cdot t_f$.

The width of material outside the bolt holes is not to be less than $0{,}67 \cdot d_b$.

2.3 The coupling flanges are to be equipped with a fitted key according to recognized standards for relieving the bolts.

The fitted key may be dispensed with if the diameter of the bolts is increased by 10%.

2.4 Horizontal coupling flanges should be forged together with the rudder stock. If the flanges are welded to the rudder stock, the stock should have a weld flange[1]) with a diameter of 1,1 D (but not less than D + 20 mm) and with a thickness equal to the flange thickness (max. flange thickness + 5 mm).

2.5 For the connection of the coupling flanges with the rudder body see also Section 19, B.4.4.

3. Vertical couplings

3.1 The diameter of the coupling bolts is not to be less than:

$$d_b = \frac{0{,}81 \cdot D}{\sqrt{n}} \sqrt{\frac{k_b}{k_r}} \quad [mm]$$

D, k_b, k_r, n see 2.1, where n is not to be less than 8

.2 The first moment of area of the bolts about the centre of the coupling is not to be less than:

$= 0{,}00043\ D^3\ \ [cm^3]$.

.3 The thickness of the coupling flanges is not to be less than

$= d_b\ \ [mm]$

The width of material outside the bolt holes is not to be less than $0{,}67 \cdot d_b$.

Cone couplings

1 Cone couplings with key

1.1 Cone couplings should have a taper c on diameter of 1: 8 - 1:12.

$c = (d_0 - d_u)/\ell$ according Fig. 14.6.).

The cone shape should be very exact. The nut is to be carefully secured, e.g. as shown in Fig. 14.6.

[1] In special cases (e.g., for small stock diameter) weld flange may be dispensed with

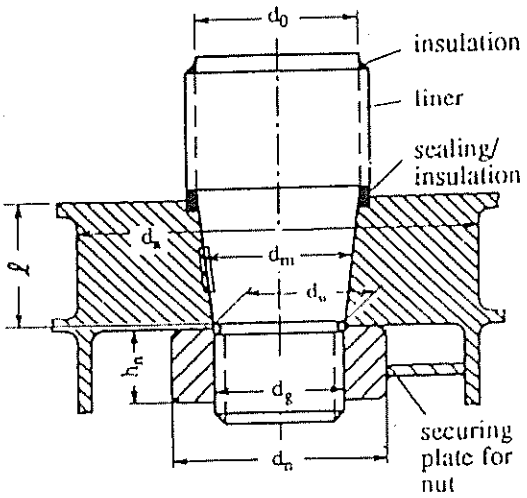


Fig. 14.6

4.1.2 The coupling length ℓ should, in general not be less than $1{,}5 \cdot d_0$.

4.1.3 For couplings between stock and rudder a key is to be provided, the shear area of which is not to be less than:

$$a_s = \frac{16 \cdot Q_F}{d_k \cdot R_{eH1}} \quad [cm^2]$$

Q_F = design yield moment of rudder stock in [Nm] according to F.

d_k = diameter of the conical part of the rudder stock in [mm] at the key

R_{eH1} = minimum nominal upper yield point of the key material in [N/mm²]

4.1.4 The effective surface area of the key (without rounded edges) between key and rudder stock or cone coupling, is not to be less than:

$$a_k = \frac{5 \cdot Q_F}{d_k \cdot R_{eH2}} \quad [cm^2]$$

R_{eH2} = minimum nominal upper yield point of the key, stock or coupling material in [N/mm²], whichever is less.

4.1.5 The dimensions of the slugging nut are to be as follows:

- height:

  $h_n = 0{,}6 \cdot d_g$

- outer diameter (the greater value to be taken)

  $d_n = 1{,}2 \cdot d_u$ or $d_n = 1{,}5 \cdot d_g$

3. Analysis

3.1 General

The evaluation of bending moments, shear forces and support forces for the system rudder - rudder stock may be carried out for some basic rudder types as shown in Figs. 14.3 - 14.5 as outlined in 3.2. - 3.3.

3.2 Data for the analysis

ℓ_{10} - ℓ_{50} = lengths of the individual girders of the system in [m]

I_{10} - I_{50} = moments of inertia of these girders in [cm^4]

For rudders supported by a sole piece the length ℓ_{20} is the distance between lower edge of rudder body and centre of sole piece, and I_{20} is the moment of inertia of the pintle in the sole piece

load on rudder body (general):

$$P_R = \frac{C_R}{\ell_{10} \cdot 10^3} \quad [kN/m]$$

load on semi-spade rudders:

$$P_{R10} = \frac{C_{R2}}{\ell_{10} \cdot 10^3} \quad [kN/m]$$

$$P_{R20} = \frac{C_{R1}}{\ell_{20} \cdot 10^3} \quad [kN/m]$$

C_R, C_{R1}, C_{R2} see B.1. and B.2.

Z = spring constant of support in the sole piece or rudder horn respectively

for the support in the sole piece (Fig. 14.3)

$$Z = \frac{6{,}18 \cdot I_{50}}{\ell_{50}^{\,3}} \quad [kN/m]$$

for the support in the rudder horn (Fig. 14.4)

$$Z = \frac{1}{f_b + f_t} \quad [kN/m]$$

f_b = unit displacement of rudder horn in [m] due to a unit force of 1 kN acting in the centre of support

$$f_b = 0{,}21 \frac{d^3}{I_n} \quad [m/kN] \text{ (guidance value)}$$

I_n = moment of inertia of rudder horn around the x-axis at d/2 in [cm^4] (see also Fig. 14.4)

f_t = unit displacement due to torsion

$$f_t = \frac{d \cdot e^2 \cdot \sum u_i/t_i}{3{,}14 \cdot 10^8 \cdot F_T^{\,2}} \quad [m/kN]$$

F_T = mean sectional area of rudder horn in [m^2]

u_i = breadth in [mm] of the individual plates forming the mean horn sectional area

t_i = plate thickness within the individual breadth u_i in [mm]

e, d = distances in [m] according to Fig. 14.4.

3.3 Moments and forces to be evaluated

3.3.1 The bending moment M_R and the shear force Q_1 in the rudder body, the bending moment M_b in the neck bearing and the support forces B_1, B_2, B_3 are to be evaluated.

The so evaluated moments and forces are to be used for the stress analyses required by 2. and E.1. of this Section and by Section 13, C.4. and C.5.

3.3.2 For spade rudders the moments and forces may be determined by the following formulae:

$$M_b = C_R \left[\ell_{20} + \frac{\ell_{10}(2x_1 + x_2)}{3(x_1 + x_2)} \right] \quad [Nm]$$

$$B_3 = \frac{M_b}{\ell_{30}} \quad [N]$$

$$B_2 = C_R + B_3 \quad [N]$$

4. Rudder trunk

Where the rudder stock is arranged in a trunk in such a way that the trunk is stressed by forces due to rudder action, the scantlings of the trunk are to be as such that the equivalent stress due to bending and shear does not exceed $0{,}35 \cdot R_{eH}$ of the material used.

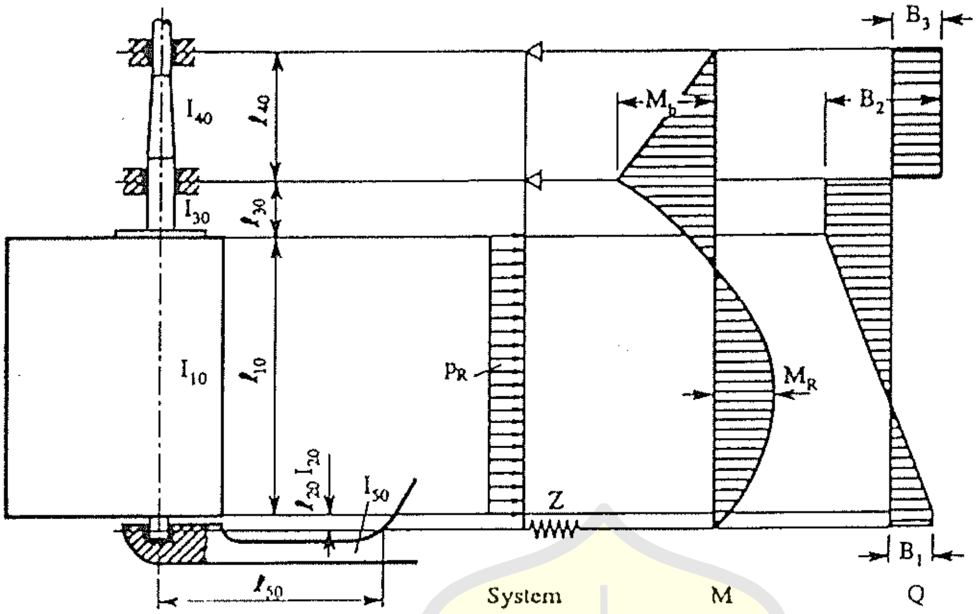


Fig. 14.3 Rudder supported by sole piece

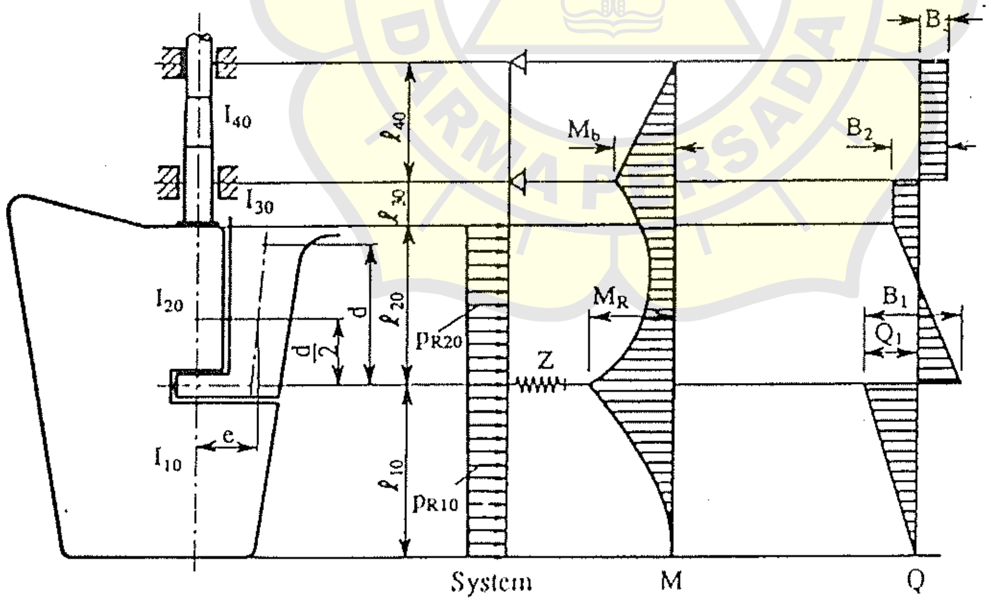


Fig. 14.4 Semi-spade rudder

Harvald disajikan untuk kapal-kapal yang besar dimana coefficient penampang tengah kapal $C_{\otimes}$ harganya mendekati 1,0.

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

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

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

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

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

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

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

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

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

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

Untuk keperluan praktis dapatlah dipakai rumus Taylor seperti dimuka yaitu;

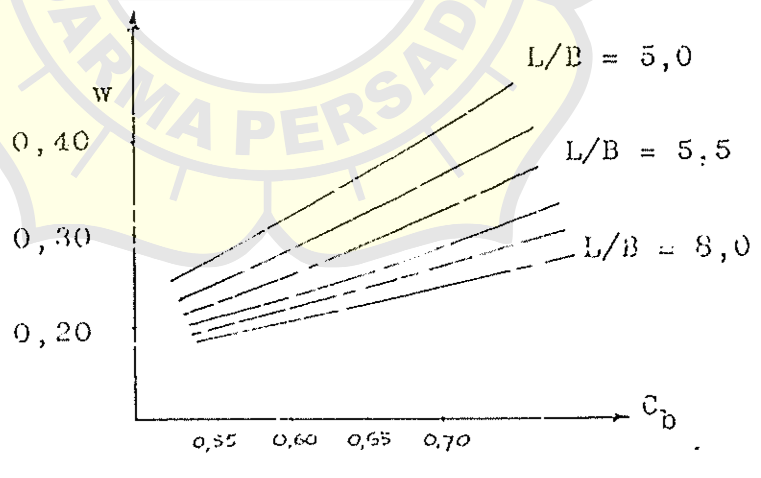
$t = 0,6\ w$ untuk kapal2 single screw.

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

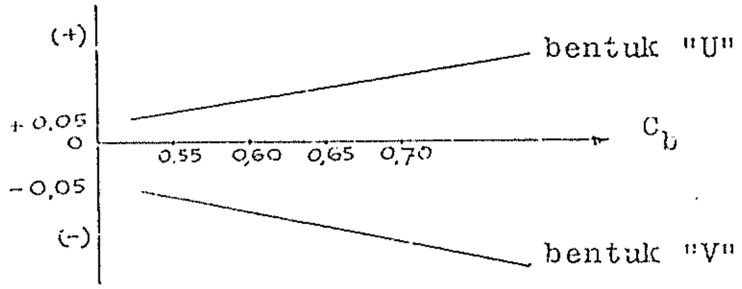
Setelah didapatkannya harga w maka kalau kecepatan kapal diketahui Vs dapatlah harga kecepatan air yang melewati piringan baling-baling (intake velocity = advance speed) dihitung yaitu :

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

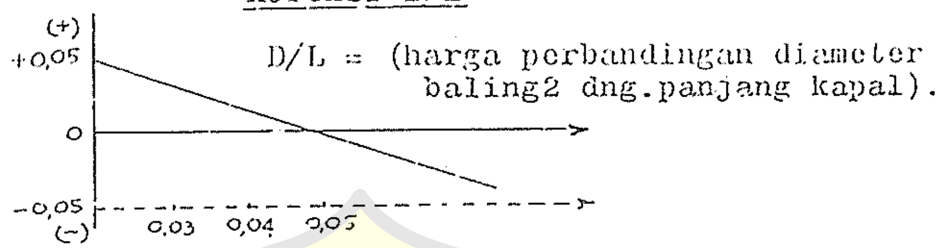
Sketsa diagram Harvald untuk mencari w :



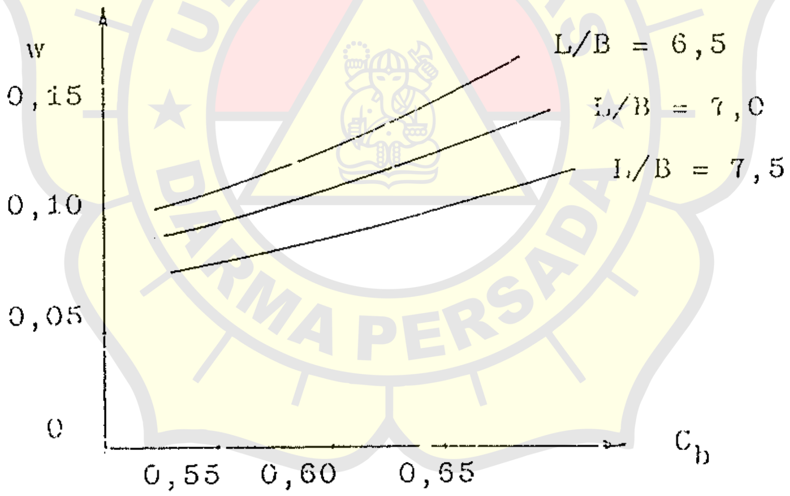
Koreksi bentuk badan kapal,



Koreksi D/L



Untuk kapal-kapal twin screw;



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

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

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

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

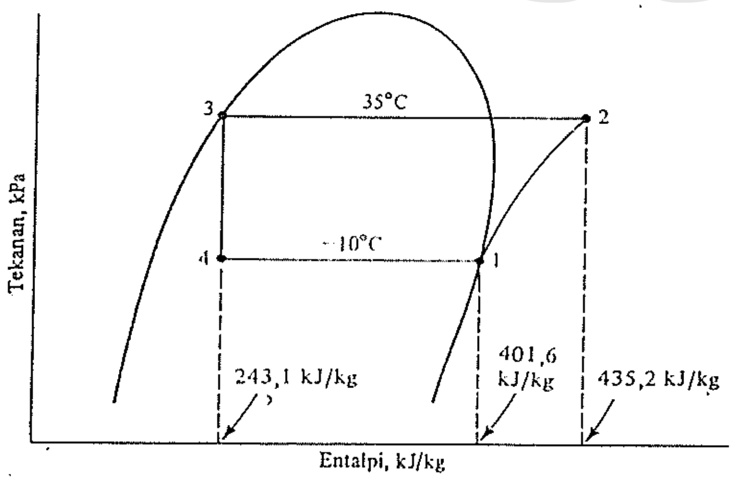
Untuk menemukan h_2 melalui garis entropi tetap geser titik 1 hingga mencapai tekanan jenuh yang sesuai dengan suhu 35°C. Tekanan pengembunan ini adalah 1354 kPa, dan nilai h_2 = 435,2 kJ/kg.

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

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

(*a*) Dampak refrigerasi:

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



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

t, °C	P, kPa	Entalpi, kJ/kg h_f	h_g	Entropi, kJ/kg · K s_f	s_g	Volume spesifik, L/kg v_f	v_g
−60	37,48	134,763	379,114	0,73254	1,87886	0,68208	537,152
−55	49,47	139,830	381,529	0,75599	1,86389	0,68856	414,827
−50	64,39	144,959	383,921	0,77919	1,85000	0,69526	324,557
−45	82,71	150,153	386,282	0,80216	1,83708	0,70219	256,990
−40	104,95	155,414	388,609	0,82490	1,82504	0,70936	205,745
−35	131,68	160,742	390,896	0,84743	1,81380	0,71680	166,400
−30	163,48	166,140	393,138	0,86976	1,80329	0,72452	135,844
−28	177,76	168,318	394,021	0,87864	1,79927	0,72769	125,563
−26	192,99	170,507	394,896	0,88748	1,79535	0,73092	116,214
−24	209,22	172,708	395,762	0,89630	1,79152	0,73420	107,701
−22	226,48	174,919	396,619	0,90509	1,78779	0,73753	99,9362
−20	244,83	177,142	397,467	0,91386	1,78415	0,74091	92,8432
−18	264,29	179,376	398,305	0,92259	1,78059	0,74436	86,3546
−16	284,93	181,622	399,133	0,93129	1,77711	0,74786	80,4103
−14	306,78	183,878	399,951	0,93997	1,77371	0,75143	74,9572
−12	329,89	186,147	400,759	0,94862	1,77039	0,75506	69,9478
−10	354,30	188,426	401,555	0,95725	1,76713	0,75876	65,3399
−9	367,01	189,571	401,949	0,96155	1,76553	0,76063	63,1746
−8	380,06	190,718	402,341	0,06585	1,76394	0,76253	61,0958
−7	393,47	191,868	402,729	0,97014	1,76237	0,76444	59,0996
−6	407,23	193,021	403,114	0,97442	1,76082	0,76636	57,1820
−5	421,35	194,176	403,496	0,97870	1,75928	0,76831	55,3394
−4	435,84	195,335	403,876	0,98297	1,75775	0,77028	53,5682
−3	450,70	196,497	404,252	0,98724	1,75624	0,77226	51,8653
−2	465,94	197,662	404,626	0,99150	1,75475	0,77427	50,2274
−1	481,57	198,828	404,994	0,99575	1,75326	0,77629	48,6517
0	497,59	200,000	405,361	1,00000	1,75279	0,77834	47,1354
1	514,01	201,174	405,724	1,00424	1,75034	0,78041	45,6757
2	530,83	202,351	406,084	1,00848	1,74889	0,78249	44,2702
3	548,06	203,530	406,440	1,01271	1,74746	0,78460	42,9166
4	565,71	204,713	406,793	1,01694	1,74604	0,78673	41,6124
5	583,78	205,899	407,143	1,02116	1,74463	0,78889	40,3556
6	602,28	207,089	407,489	1,02537	1,74324	0,79107	39,1441
7	621,22	208,281	407,831	1,02958	1,74185	0,79327	37,9759
8	640,59	209,477	408,169	1,03379	1,74047	0,79549	36,8493
9	660,42	210,675	408,504	1,03799	1,73911	0,79775	35,7624
10	680,70	211,877	408,835	1,04218	1,73775	0,80002	34,7136
11	701,44	213,083	409,162	1,04637	1,73640	0,80232	33,7013
12	722,65	214,291	409,485	1,05056	1,73506	0,80465	32,7239
13	744,33	215,503	409,804	1,05474	1,73373	0,80701	31,7801
14	766,50	216,719	410,119	1,05892	1,73241	0,80939	30,8683
15	789,15	217,937	410,430	1,06309	1,73109	0,81180	29,9874
16	812,29	219,160	410,736	1,06726	1,72978	0,81424	29,1361
17	835,93	220,386	411,038	1,07142	1,72848	0,81671	28,3131
18	860,08	221,615	411,336	1,07559	1,72719	0,81922	27,5173
19	884,75	222,848	411,629	1,07974	1,72590	0,82175	26,7477
20	909,93	224,084	411,918	1,08390	1,72462	0,82431	26,0032
21	935,64	225,324	412,202	1,08805	1,72334	0,82691	25,2829
22	961,89	226,568	412,481	1,09220	1,72206	0,82954	24,5857
23	988,67	227,816	412,755	1,09634	1,72080	0,83221	23,9107
24	1016,0	229,068	413,025	1,10048	1,71953	0,83491	23,2572
25	1043,9	230,324	413,289	1,10462	1,71827	0,83765	22,6242
26	1072,3	231,583	413,548	1,10876	1,71701	0,84043	22,0111
27	1101,4	232,847	413,802	1,11290	1,71576	[illegible]	[illegible]

t, °C	P, kPa	Entalpi, kJ/kg h_f	h_g	Entropi, kJ/kg · K s_f	s_g	Volume spesifik, L/kg v_f	v_g
29	1161,1	235,387	414,293	1,12116	1,71325	0,84899	20,2829
30	1191,9	236,664	414,530	1,12530	1,71200	0,85193	19,7417
31	1223,2	237,944	414,762	1,12943	1,71075	0,85491	19,2168
32	1255,2	239,230	414,987	1,13355	1,70950	0,85793	18,7076
33	1287,8	240,520	415,207	1,13768	1,70826	0,86101	18,2135
34	1321,0	241,814	415,420	1,14181	1,70701	0,86412	17,7341
35	1354,8	243,114	415,627	1,14594	1,70576	0,86729	17,2686
36	1389,2	244,418	415,828	1,15007	1,70450	0,87051	16,8168
37	1424,3	245,727	416,021	1,15420	1,70325	0,87378	16,3779
38	1460,1	247,041	416,208	1,15833	1,70199	0,87710	15,9517
39	1496,5	248,361	416,388	1,16246	1,70073	0,88048	15,5375
40	1533,5	249,685	416,561	1,16659	1,69946	0,88392	15,1351
41	1571,2	251,015	416,726	1,17073	1,69819	0,88741	14,7439
42	1609,6	252,352	416,883	1,17486	1,69692	0,89097	14,3636
43	1648,7	253,694	417,033	1,17900	1,69564	0,89459	13,9938
44	1688,5	255,042	417,174	1,18315	1,69435	0,89828	13,6341
45	1729,0	256,396	417,308	1,18730	1,69305	0,90203	13,2841
46	1770,2	257,756	417,432	1,19145	1,69174	0,90586	12,9436
47	1812,1	259,123	417,548	1,19560	1,69043	0,90976	12,6122
48	1854,8	260,497	417,655	1,19977	1,68911	0,91374	12,2895
49	1898,2	261,877	417,752	1,20393	1,68777	0,91779	11,9753
50	1942,3	263,264	417,838	1,20811	1,68643	0,92193	11,6693
52	2032,8	266,062	417,983	1,21648	1,68370	0,93047	11,0806
54	2126,5	268,891	418,083	1,22489	1,68091	0,93939	10,5214
56	2223,2	271,754	418,137	1,23333	1,67805	0,94872	9,98952
58	2323,2	274,654	418,141	1,24183	1,67511	0,95850	9,48319
60	2426,6	277,594	418,089	1,25038	1,67208	0,96878	9,00062
62	2533,3	280,577	417,978	1,25899	1,66895	0,97960	8,54016
64	2643,5	283,607	417,802	1,26768	1,66570	0,99104	8,10023
66	2757,3	286,690	417,553	1,27647	1,66231	1,00317	7,67934
68	2874,7	289,832	417,226	1,28535	1,65876	1,01608	7,27605
70	2995,9	293,038	416,809	1,29436	1,65504	1,02987	6,88899
75	3316,1	301,399	415,299	1,31758	1,64472	1,06916	5,98334
80	3662,3	310,424	412,898	1,34223	1,63239	1,11810	5,14862
85	4036,8	320,505	409,101	1,36936	1,61673	1,18328	4,35815
90	4442,5	332,616	402,653	1,40155	1,59440	1,28230	3,56440
95	4883,5	351,767	386,708	1,45222	1,54712	1,52064	2,55133

Tabel A-7 Refrigeran 22: sifat-sifat uap gas panas lanjut[5]

t, °C	v, L/kg	h, kJ/kg	s, kJ/kg · K	v, L/kg	h, kJ/kg	s, kJ/kg · K	v, L/kg	h, kJ/kg	s, kJ/kg · K
	Suhu jenuh, −20°C			Suhu jenuh, −10°C			Suhu jenuh, 0°C		
-20	92,8432	397,467	1,7841						
-15	95,1474	400,737	1,7969						
-10	97,4256	404,017	1,8095	65,3399	401,555	1,7671			
-5	99,6808	407,307	1,8219	67,0081	404,983	1,7800			
0	101,915	410,610	1,8341	68,6524	408,412	1,7927	47,1354	405,361	1,7518
5	104,130	413,926	1,8461	70,2751	411,845	1,8052	48,3899	408,959	1,7649
10	106,328	417,258	1,8580	71,8785	415,283	1,8174	49,6215	412,567	1,7777
15	108,510	420,606	1,8697	73,4644	418,730	1,8295	50,8328	416,159	1,7903
20	110,678	423,970	1,8813	75,0346	422,186	1,8414	52,0259	419,649	1,8026
25	112,832	426,353	1,8928	76,5904	425,653	1,8531	53,2028	423,339	1,8148
	Suhu jenuh, 5°C			Suhu jenuh, 10°C			Suhu jenuh, 15°C		
5	40,3556	407,143	1,7446						
10	41,4580	410,851	1,7578	34,7136	408,835	1,7377			
15	42,5379	414,542	1,7708	35,6907	412,651	1,7511	29,9874	410,430	1,7311
20	43,5979	418,222	1,7834	36,6454	416,442	1,7642	30,8606	414,362	1,7556
25	44,6401	421,894	1,7958	37,5804	420,215	1,7769	31,7114	418,260	1,7578
30	45,6665	425,562	1,8080	38,4981	423,974	1,7394	32,5427	422,133	1,7707
35	46,6786	429,229	1,8200	39,4002	427,724	1,8017	33,3568	425,985	1,7833
40	47,6779	432,897	1,8319	40,2884	431,469	1,8137	34,1556	429,823	1,7956
45	48,6656	436,569	1,8435	41,1642	435,211	1,8256	34,9409	433,650	1,8078
50	49,6427	440,247	1,8550	42,0286	438,954	1,8373	35,7139	437,470	1,8197

Tabel A-7 (lanjutan)

	Suhu jenuh, 20°C			Suhu jenuh, 25°C			Suhu jenuh, 30°C		
20	26,0032	411,918	1,7246						
25	26,7900	415,977	1,7383	22,6242	413,289	1,7183			
30	27,5542	419,991	1,7517	23,3389	417,487	1,7322	19,7417	414,530	1,7120
35	28,2989	423,970	1,7646	24,0306	421,627	1,7458	20,3962	418,881	1,7262
40	29,0264	427,922	1,7774	24,7027	425,721	1,7590	21,0272	423,159	1,7400
45	29,7389	431,852	1,7899	25,3575	429,779	1,7718	21,6381	427,378	1,7534
50	30,4379	435,766	1,8021	25,9974	433,807	1,7844	22,2316	431,549	1,7664
55	31,1250	439,668	1,8141	26,6239	437,813	1,7967	22,8101	435,683	1,7791
60	31,8012	443,561	1,8258	27,2386	441,801	1,8087	23,3733	439,787	1,7915
65	32,4678	447,450	1,8374	27,8427	445,777	1,8206	23,9288	443,867	1,8036
	Suhu jenuh, 32°C			Suhu jenuh, 34°C			Suhu jenuh, 36°C		
35	19,0907	417,648	1,7182	17,8590	416,325	1,7099			
40	19,7093	422,014	1,7322	18,4675	420,792	1,7243	17,2953	419,483	1,7162
45	20,3062	426,310	1,7458	19,0526	425,174	1,7382	17,8708	423,961	1,7304
50	20,8847	430,549	1,7591	19,6178	429,487	1,7517	18,4247	428,358	1,7442
55	21,4471	434,743	1,7719	20,1660	433,747	1,7647	18,9603	432,690	1,7575
60	21,9956	438,900	1,7845	20,6994	437,963	1,7775	19,4802	436,970	1,7704
65	22,5318	443,028	1,7968	21,2199	442,143	1,7899	19,9865	441,207	1,7830
70	23,0571	447,133	1,8089	21,7289	446,294	1,8021	20,4807	445,410	1,7954
75	23,5726	451,219	1,8207	22,2278	450,424	1,8141	20,9643	449,586	1,8074
80	24,0794	455,292	1,8323	22,7176	454,535	1,8258	21,4385	453,739	1,8193

(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

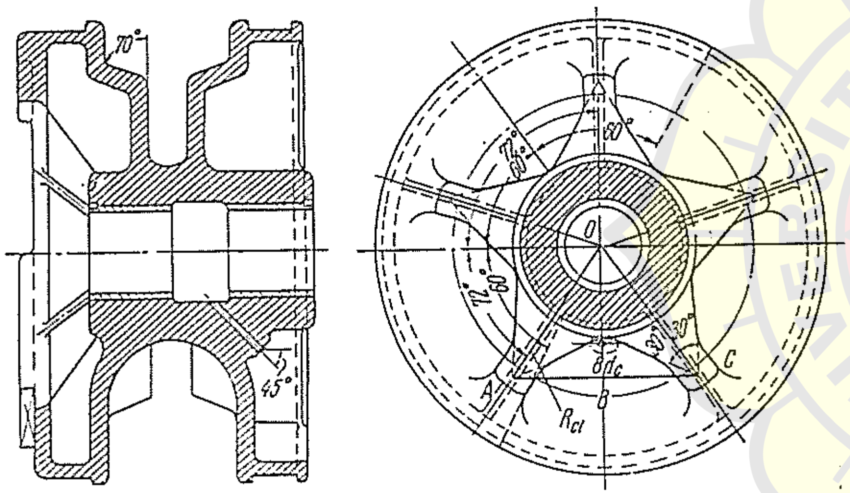


Fig. 170.

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

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

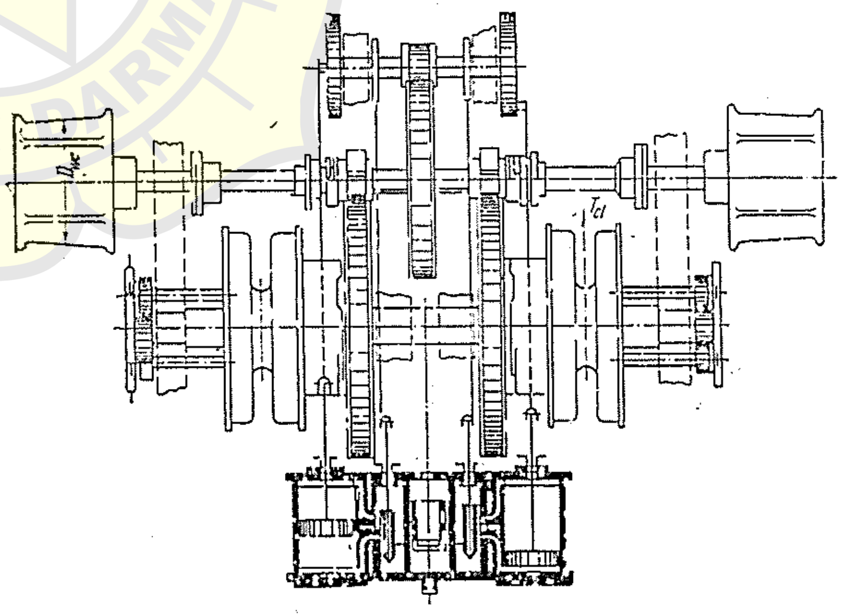


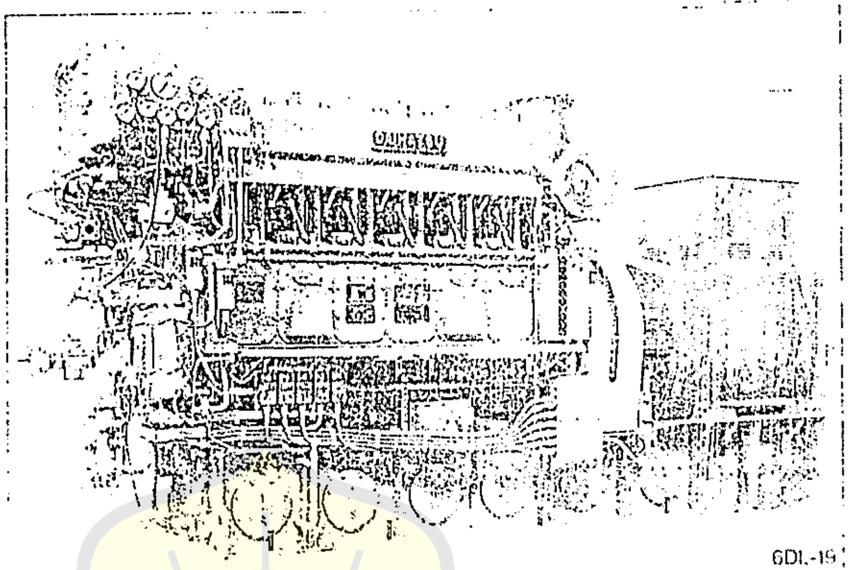
Fig. 171.

MARINE AUXILIARY DIESEL ENGINE

DL-19

BORE:190mm
STROKE:230mm
No. of cylinder:6
Pme:15.33～19.17kg/cm²
Piston speed:5.52m/sec
(at 720rpm)

...ke. Direct injection type
2 inlet & 2 exhaust valves.
...charger and inter cooler



6DL-19

PRINCIPAL PARTICULARS

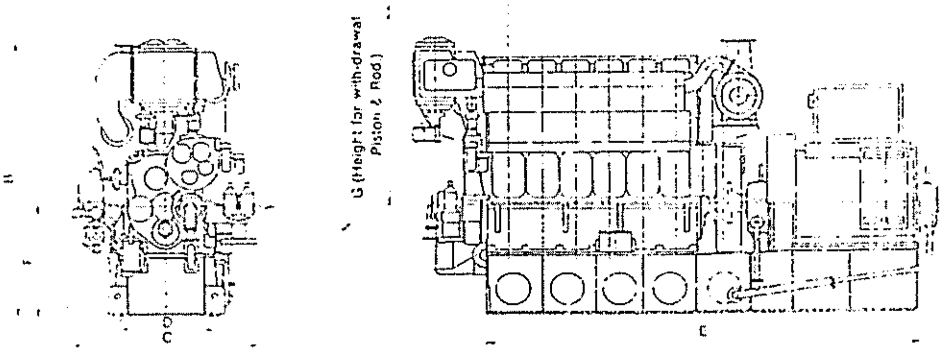
Model	Fuel oil / Rev (rpm) / Engine Output	[illegible] 720	[illegible] 750	[illegible] 900	[illegible] 1000	Diesel oil 720	Diesel oil 750	Diesel oil 900	Diesel oil 1000
6DLB-19	PS	480	480	600	600	480	480	600	600
	kW	350	350	440	440	350	350	440	440
6DL-19	PS	540	540	660	660	600	600	750	750
	kW	395	395	485	485	440	440	550	550

ISO 1[illegible] 1,359[illegible]

DIMENSIONS & WEIGHTS (at 720rpm)

(mm) (ton)

Model / Mark	A	B	C	D	E	F	G	H	J	Dry Weight (appr.) Eng.	Dry Weight (appr.) Gen.	Dry Weight (appr.) Common Bed
6DLB-19 / 6DL-19	4530	2310	1430	800	3520	900	1470	790	1790	6.0	2.8	1.5



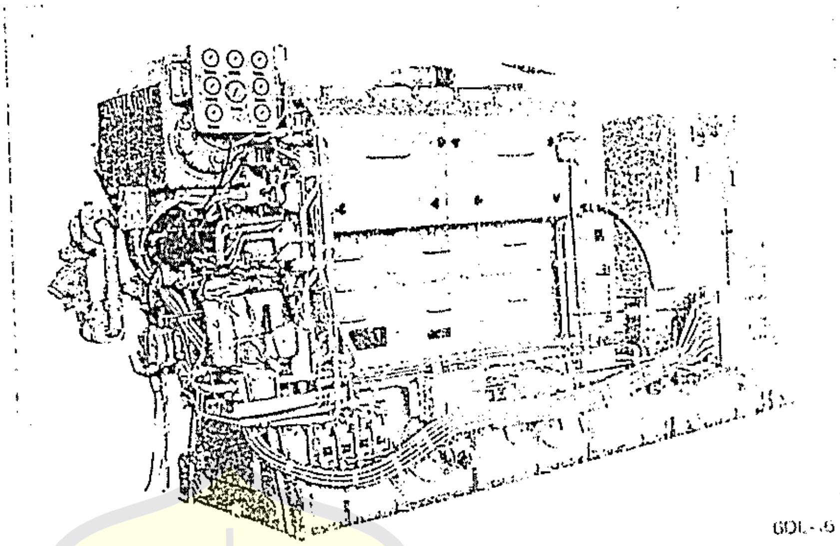
DL Series

MARINE AUXILIARY DIESEL ENGINE

DL-16

BORE:165mm
STROKE:210mm
No. of cylinder:6
Pme:16.70kg/cm²
Piston speed:8.4m/sec
(at 1200rpm)

4 stroke. Direct injection type
with 2 inlet & 2 exhaust valves.
turbo-charger and inter cooler



6DL-16

PRINCIPAL PARTICULARS

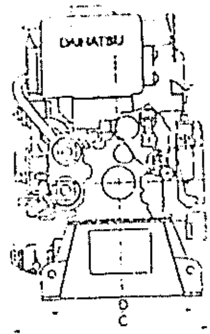
Model		Fuel oil	[illegible]			Diesel Oil		
		Rev. (rpm)	900	1000	1200	900	1000	1200
6DL-16	Engine Output	PS	420	450	540	450	500	600
		kW	310	330	395	330	365	44[illegible]

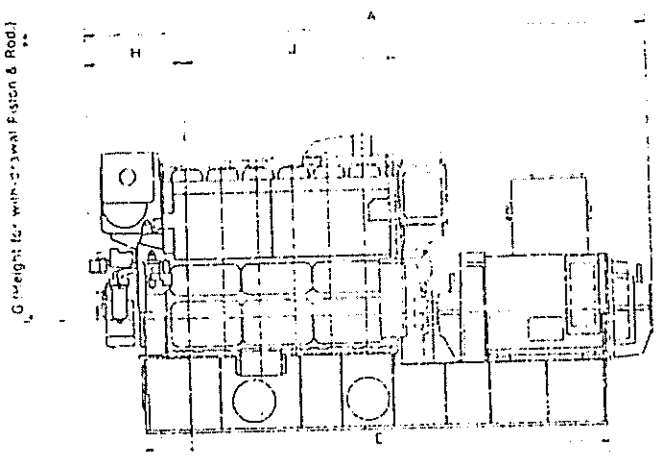
ISO 1 kW = 1.359 PS

DIMENSIONS & WEIGHTS (at 900rpm)

(mm) (t)

Model \ Mark	A	B	C	D	E	F	G	H	J	[illegible]		
6DL-16	3700	1800	[illegible]	950	3260	750	1195	645	1417.5	3.0	2.0	0.9





di mana Re: Bilangan Reynolds (tak berdimensi)
v: Kecepatan rata-rata aliran di dalam pipa (m/s)
D: Diameter dalam pipa (m)
ν: Viskositas kinematik zat cair (m^2/s)

Pada $Re < 2300$, aliran bersifat laminer.
Pada $Re > 4000$, aliran bersifat turbulen.
Pada $Re = 2300 - 4000$ terdapat daerah transisi, di mana aliran dapat bersifat laminer atau turbulen tergantung pada kondisi pipa dan aliran.

(I) Aliran laminer

Dalam hal aliran laminer, koefisien kerugian gesek untuk pipa (λ) dalam pers. (2.12) dapat dinyatakan dengan

$$\lambda = \frac{64}{Re} \tag{2.14}$$

(II) Aliran turbulen

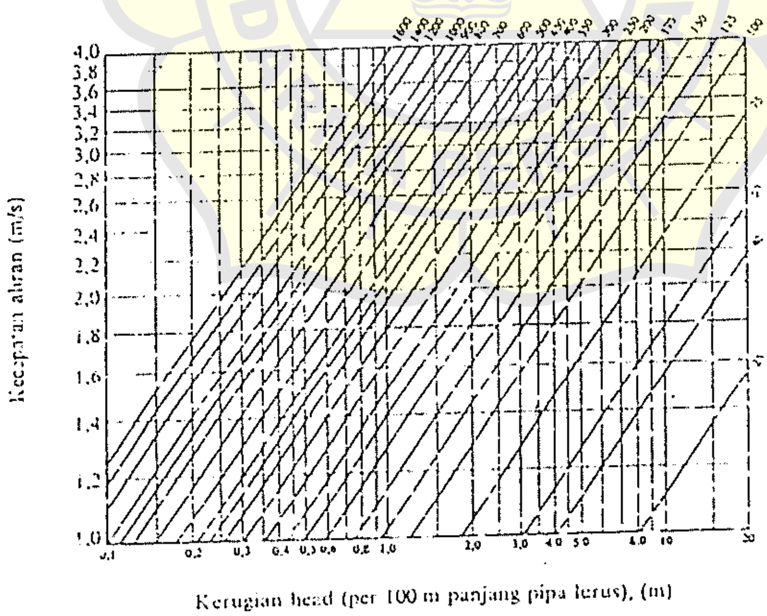
Untuk menghitung kerugian gesek dalam pipa pada aliran turbulen terdapat berbagai rumus empiris. Di bawah ini akan diberikan cara perhitungan dengan rumus Darcy dan Hazen-Williams.

1) *Formula Darcy*

Dengan cara Darcy, koefisien kerugian gesek λ dari Pers. (2.12) dihitung menurut rumus

$$\lambda = 0,020 + \frac{0,0005}{D} \tag{2.15}$$

di mana D adalah diameter dalam pipa (m). Rumus ini berlaku untuk pipa baru dari besi cor. Jika pipa telah dipakai selama bertahun-tahun, harga λ akan menjadi 1,5



Gb. 2.4 Kerugian gesek pada pipa lurus (rumus Darcy).

di mana h_p: Head tekanan (m)
p: Tekanan (kgf/cm^2)
γ: Berat per satuan volume zat cair yang dipompa (kgf/l)

Apabila tekanan diberikan dalam kPa, dapat dipakai rumus berikut:

$$h_p = \frac{1}{9{,}8}\frac{p'}{\rho} \tag{2.9}$$

di mana p': Tekanan (Pa)
ρ: Rapat masa (kg/l)

Menurut ISO, energi spesifik Y (J/kg) kadang-kadang dipakai sebagai pengganti head H (m). Adapun hubungannya adalah sebagai berikut:

$$Y = gH \tag{2.10}$$

Sebagaimana diutarakan di atas, untuk menentukan head total yang harus disediakan pompa, perlu dihitung lebih dahulu head kerugian h_l. Di bawah ini akan diuraikan cara menghitung kerugian head tersebut.

2.4.2 Head Kerugian

Head kerugian (yaitu head untuk mengatasi kerugian-kerugian) terdiri atas head kerugian gesek di dalam pipa-pipa, dan head kerugian di dalam belokan-belokan, reduser, katup-katup, dsb. Di bawah ini akan diberikan cara menghitungnya, satu per satu.

(1) Head kerugian gesek dalam pipa

Untuk menghitung kerugian gesek di dalam pipa dapat dipakai salah satu dari dua rumus berikut ini:

$$v = CR^pS^q \tag{2.11}$$

$$\boxed{h_f = \lambda\frac{L}{D}\frac{v^2}{2g}} \tag{2.12}$$

di mana v: Kecepatan rata-rata aliran di dalam pipa (m/s)
C, p, q: Koefisien-koefisien
R: Jari-jari hidrolik (m)

$$R = \frac{\text{Luas penampang pipa, tegak lurus aliran } (m^2)}{\text{Keliling pipa atau saluran yang dibasahi (m)}}$$

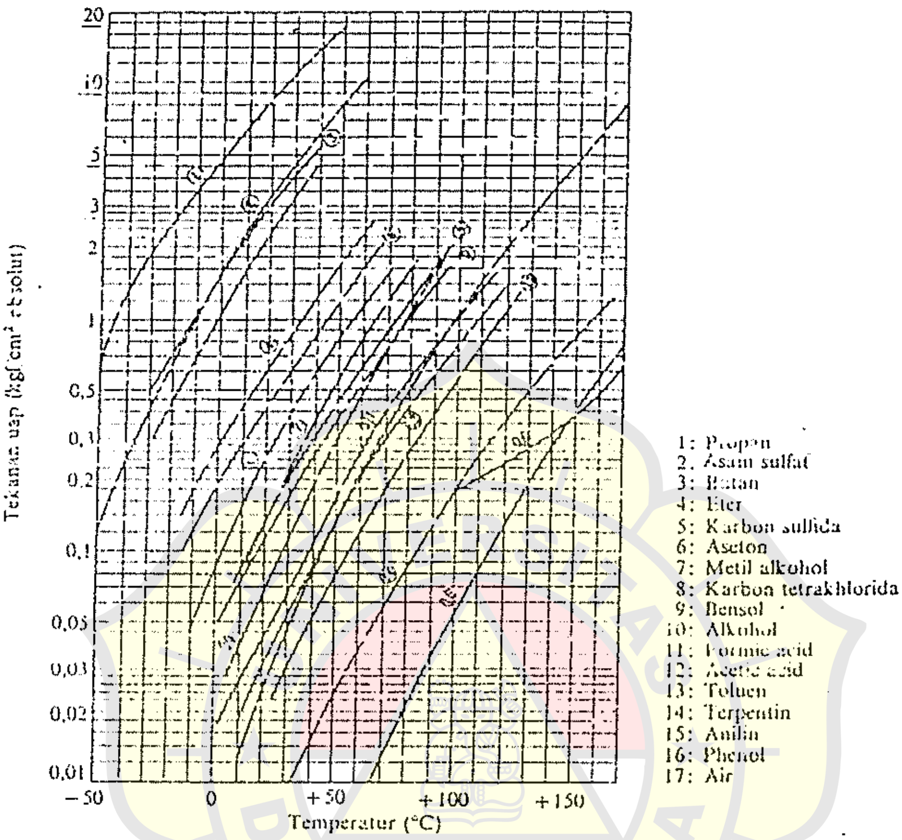
S: Gradien hidrolik

$$S = \frac{h_f}{L}$$

h_f: Head kerugian gesek dalam pipa (m)
λ: Koefisien kerugian gesek
g: Percepatan gravitasi (9,8 m/s^2)
L: Panjang pipa (m)
D: Diameter dalam pipa (m)

Selanjutnya, untuk aliran yang laminer dan yang turbulen, terdapat rumus yang berbeda. Sebagai patokan apakah suatu aliran itu laminer atau turbulen, dipakai bilangan Reynolds:

$$Re = \frac{vD}{\nu} \tag{2.13}$$



(b) Tekanan uap berbagai zat cair
(*Catatan*: 1 kgf/cm² = 0,1 M Pa)

Gb. 2.1 Sifat-sifat fisik berbagai zat cair.

2.4 Head

2.4.1 Head Total Pompa

Head total pompa yang harus disediakan untuk mengalirkan jumlah air seperti direncanakan, dapat ditentukan dari kondisi instalasi yang akan dilayani oleh pompa. Seperti diperlihatkan dalam Gb. 2.2, head total pompa dapat ditulis sebagai berikut:

$$\boxed{H = h_a + \Delta h_p + h_l + \frac{v_d^2}{2g}} \qquad (2.6)$$

di mana H: Head total pompa (m)

h_a: Head statis total (m)
Head ini adalah perbedaan tinggi antara muka air di sisi keluar dan di sisi isap; tanda positip (+) dipakai apabila muka air di sisi ke luar lebih tinggi dari pada sisi isap.

Δh_p: Perbedaan head tekanan yang bekerja pada kedua permukaan air (m),

$$\Delta h_p = h_{p_2} - h_{p_1},$$

h_l: Berbagai kerugian head di pipa, katup, belokan, sambungan, dll (m), Mula-mula perlu ditentukan jumlah limpasan keseluruhan dari air hujan di tanah pertanian dengan rumus

$$Q = 10fRA10 \qquad (2.2.a)$$

di mana Q: Limpasan keseluruhan (m^3)
R: Curah hujan standar (mm)
f: Koefisien limpas
A: Luas wilayah drainase (ha)

Dari jumlah limpasan yang dihitung dengan cara di atas kemudian dapat diperkirakan kapasitas pompa drainase yang diperlukan dengan rumus

$$Q_p = \frac{Q}{24 \times 3600 \times D} \qquad (2.2.b)$$

di mana Q_p: Kapasitas pompa drainase (m^3/s)
D: Lamanya genangan yang diperbolehkan (hari)

Koefisien limpas yang dipakai untuk menentukan limpasan total dipengaruhi oleh curah hujan total seperti diberikan di dalam Tabel 2.7.

Jumlah hari limpas harus dihitung secara coba-coba dengan memperhatikan bahwa limpasan total akan terdistribusikan seperti dalam Tabel 2.8.

Tabel 2.7 Curah hujan total dan koefisien limpas total.

Curah hujan total (mm)	Kurang dari 10	10 - 30	30 - 50	50 - 100	100 - 200	200 - 300	Lebih dari 300
Koefisien limpas total	[illegible]	0,10	0,30	0,50	0,80	0,90	0,95

Tabel 2.8 Faktor distribusi limpasan dari curah hujan tunggal.

Curah hujan (mm) \ Hari	Hari ke-1	Hari ke-2	Hari ke-3	Hari ke-4	Jumlah
Kurang dari 30	100%				100%
30 - 50	70%	30%		–	100%
50 - 100	60%	30%	10%	–	100%
Lebih dari 100	50%	30%	15%	5%	100%

Untuk penentuan akhir dari spesifikasi perencanaan, kondisi limpasan air hujan dan kondisi fluktuasi muka air harus diperhitungkan. Dalam hal ini perlu dipelajari buku-buku profesional dalam bidang tersebut.

(5) Pengairan tanah pertanian

Ditinjau dari cara pengairan, tanah pertanian dapat dibedakan antara sawah dan ladang.

(a) Pengairan sawah

1) *Keperluan air*

Sawah untuk tanaman padi harus digenangi air dengan kedalaman tertentu. Untuk memelihara kedalaman tersebut diperlukan tambahan air terus menerus guna mengganti penyusutan karena transpirasi tanaman, penguapan sawah, dan perkolasi*. Jadi:

* Transpirasi : penguapan melalui permukaan tanaman
Penguapan : penguapan langsung dari air ke udara
Perkolasi : peresapan air ke dalam tanah.

h) 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.

- +5% lower fore peak tank
- +3% upper fore peak tank
- +2% double bottom tank.

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

Sounding/ullage tables delivered by yard.

i) Provisions/persons/luggage

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

Type and Location of Main Engine

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

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

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

additions see fuel oil!

Lubrication oil

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

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

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

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

Fresh water

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

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

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

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

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

In general

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

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

16. Consumables and tanks

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

- consumables
- provisions
- ballast.

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

- fuel oil

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

last brackets for reserve:

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

Rumus TAYLOR

Untuk Wake fraction : Kapal berbaling2 tunggal;

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

Kapal berbaling2 ganda;

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

Untuk Thrust deduction factor :

Kapal berbaling2 tunggal: $t \approx w$

Kapal berbaling2 ganda; $t \quad w$

dimana harga k adalah sebagai berikut :

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

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

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

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

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

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

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

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

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

$h = f_b + \sum h'$

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

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

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

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

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

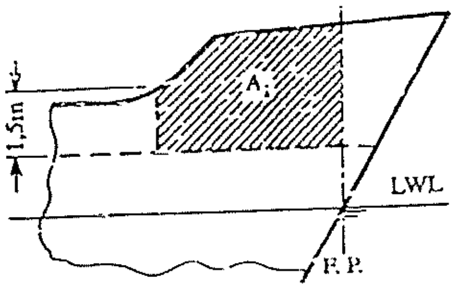


Fig. 18.1

C. Anchors

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

Guidance

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

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

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

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

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

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

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

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

Section 18

Equipment

A. General

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

Guidance

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

B. Equipment numeral

The equipment numeral is to be calculated as follows:

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

1.6 Pipe connections

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

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

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

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

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

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

Calculation of pipe diameters

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

2 Dry cargo and passenger ships

main bilge pipes

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

branch bilge pipes

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

where

d_H	[mm]	calculated inside diameter of main bilge pipe
d_Z	[mm]	calculated inside diameter of branch bilge pipe
L	[m]	length of ship between perpendiculars
B	[m]	moulded breadth of ship
H	[m]	depth of ship to the bulkhead deck
l	[m]	length of the watertight compartment

2.3 Tankers

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

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

where:

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

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

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

2.4 Minimum diameter

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

2.5 Maximum diameter

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

2.6 Deviations

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

3. Bilge pumps

3.1 Capacity of bilge pumps

Each bilge pump must be capable of delivering:

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

where:

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

n [Rpm] rated shaft speed

F [-] factor for the type of propulsion installation

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

= 100
for all other propulsion installations

b) Propeller shafts
= 100
for all types of installations

C_w [-] material factor

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

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

k [-] Factor for the type of shaft

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

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

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

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

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

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

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

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

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

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

D. Design

1. General

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

2. Shaft tapers and propeller nut threads

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

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

Section 4

Main Shafting

A. General

1. Scope

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

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

2. Documents for approval

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

B. Materials

1. Approved materials

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

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

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

. Testing of materials

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

C. Shaft Dimensions

1. General

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

2. Minimum diameter

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

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

d [mm] required outside diameter of shaft

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

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

d_a [mm] actual shaft diameter

P_W [kW] rated power transmitted by shaft

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

The distance between the vertical webs should not exceed 1,2 · h.

The radii in the rudder plating are not to be less than 4 - 5 times the plate thickness, but in no case less than 50 mm.

1.5 It is recommended to keep the natural frequency of the fully immersed rudder at least 10% above the exciting frequency of the propeller (number of revolutions x number of blades).

2. Rudder plating

2.1 The thickness of the rudder plating is to be determined according to the following formula:

$$t = 1{,}74 \cdot a\sqrt{p_R \cdot k} + t_K \quad [mm]$$

$$p_R = 10 \cdot T + \frac{C_R}{10^3 \cdot A} \quad [kN/m2]$$

a = the smaller unsupported width of a plate panel in [m].

t_K = corrosion allowance, see Section 3, K.

The influence of the aspect ratio of the plate panels may be taken into account as given in Section 3, A.3.

The thickness shall, however, not be less than the thickness t_2 of the shell plating at the ends according to Section 6, B.3.

2.2 For connecting the side plating of the rudder to the webs tenon welding is not to be used. Where application of fillet welding is not practicable, the side plating is to be connected by means of slot welding to flat bars which are welded to the webs.

2.3 The thickness of the webs is not to be less than 70% of the thickness of the rudder plating according to 2.1, but not less than:

$$t_{min} = 8\sqrt{k} \quad [mm]$$

3. Transmitting of the rudder torque

3.1 For transmitting the rudder torque, the rudder plating according to 2.1 is to be increased by 25% in way of the coupling. A sufficient number of vertical webs is to be fitted in way of the coupling.

3.2 If the torque is transmitted by a prolonged shaft extended into the rudder, the latter must have the diameter D_t or D_1, whichever is greater, at the upper 10% of the intersection length. Downwards it may be tapered to 0,6 D_t, in spade rudders to 0,4 times the strengthened diameter, if sufficient support is provided for.

4. Rudder bearings

4.1 In way of bearings liners and bushes are to be fitted. Where in case of small ships bushes are not fitted, the rudder stock is to be suitably increased in diameter in way of bearings enabling the stock to be re-machined later.

4.2 An adequate lubrication is to be provided.

4.3 The bearing forces result from the direct calculation mentioned in C.3. As a first approximation the bearing force may be determined without taking account of the elastic supports. This can be done as follows:

.1 normal rudder with two supports:

The rudder force C_R is to be distributed to the supports according to their vertical distances from the centre of gravity of the rudder area.

.2 semi-spade rudders:

- support force in the rudder horn:

$$B_1 = C_R \cdot b/c \quad [N]$$

- support force in the neck bearing:

$$B_2 = C_R - B_1 \quad [N]$$

For b and c see Fig. 13.6 in Section 13.

4.4 The projected bearing surface A_b (bearing height x external diameter of liner) is not to be less than

$$A_b = \frac{B}{q} \quad [mm^2]$$

B = support force in [N]

q = permissible surface pressure according to Table 14.2

4.5 Stainless and wear resistant steels, bronze and hot-pressed bronze-graphite materials have a considerable difference in potential to non-alloyed steel. Respective preventive measures are required.

Table 14.2

Bearing material	q [N/mm²]
lignum vitae or synthetic material of little hardness [1]	2,5
white metal, oil lubricated	4,5
synthetic material with adequate hardness [1,2]	5,5
steel[3], bronze and hot-pressed bronze-graphite materials	7,0

1 Synthetic materials to be of approved type
2 Indentation hardness test to be carried out at 23 °C and with 50% moisture according to recognized standards, e.g. DIN 53456.
3 Stainless and wear resistant steel in an approved combination with stock liner

Higher values may be taken for q if they are verified by test.

4.6 The bearing height shall be equal to the bearing diameter, however, is not to exceed 1,2 times the bearing diameter. Where the bearing depth is less than the bearing diameter, higher specific surface pressures may be allowed.

4.7 The wall thickness of pintle bearings in sole piece and rudder horn shall be approximately ¼ of the pintle diameter.

5. **Pintles**

5.1 Pintles are to have scantlings complying with the conditions given in 4.4 and 4.6. The pintle diameter is not to be less than:

$$d = 0{,}35\sqrt{B_1 \cdot k_r} \quad [\text{mm}]$$

B_1 = support force in [N]

k_r see A.4.2.

5.2 The thickness of any liner or bush shall not be less than:

$$t = 0{,}01\sqrt{B_1} \quad [\text{mm}]$$

t_{min} = 8 mm for metallic materials and synthetic material

t_{min} = 22 mm for lignum vitae

5.3 Where pintles are of conical shape, they are to comply with the following

taper on diameter 1 : 8 to 1 : 12 if keyed by slugging nut.

taper on diameter 1 : 12 to 1 : 20 if mounted with oil injection and hydraulic nut.

5.4 The pintles are to be arranged in such a manner as to prevent unintentional loosening and falling out.

For nuts and threads the requirements of D.4.1.5 and 4.2.2 apply accordingly.

6. **Guidance values for bearing clearances**

For metallic bearing material the bearing clearance should generally not be less than:

$$\frac{d_b}{1000} + 1{,}0 \quad [\text{mm}] \text{ or } 1{,}5 \text{ mm as a minimum}$$

d_b = inner diameter of bush.

If non-metallic bearing material is applied, the bearing clearance is to be specially determined considering the material's swelling and thermal expansion properties.

F. Design Yield Moment of Rudder Stock

The design yield moment of the rudder stock is to be determined by the following formula:

$$Q_F = 0{,}02664 \frac{D_t^3}{k_r} \quad [\text{Nm}]$$

D_t = stock diameter in [mm] according to C.1.

Where the actual diameter D_{ta} is greater than the calculated diameter D_t, the diameter D_{ta} is to be used. However, D_{ta} need not be taken greater than 1,145 D_t.

G. Stopper, Locking Device

1. **Stopper**

The motions of quadrants or tillers are to be limited on either side by stoppers. The stoppers and their foundations connected to the ship's hull are to be of strong construction so that the yield point of the applied materials is not exceeded at the design yield moment of the rudder stock.

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, $\alpha = 0{,}4$ may be used for the ahead condition

k_b = balance factor as follows

k_b = A_f/A

k_b = 0,08 for unbalanced rudders

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

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

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

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

The resulting force of each part may be taken as:

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

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

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

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

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

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

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

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

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

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

c_1 = A_1/b_1

$C_2 = A_2/b_2$

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

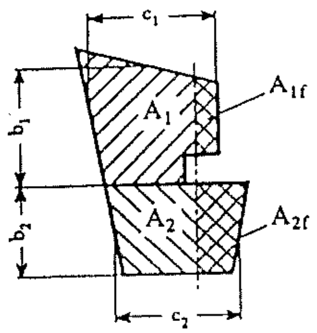


Fig. 14.2

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

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

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

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

for ahead condition

The greater value is to be taken.

C. Scantlings of the Rudder Stock

1. Rudder stock diameter

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

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

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

The related torsional stress is:

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

k_r see A.4.2.

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

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

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

2. Strengthening of rudder stock

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

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

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

Bending stress:

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

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

Torsional stress:

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

D_1 = increased rudder stock diameter in [cm]

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

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

Q_R see B.1.2 and B.2.2 - 2.3

D_t see 1.1.

Guidance

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

Section 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.

Materials

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

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

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

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

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

Before significant reductions in rudder stock
eter due to the application of steels with R_{eH}
ding 235 N/mm^2 are granted, the Society may
re the evaluation of the elastic rudder stock
tions. Large deflections should be avoided in
to avoid excessive edge pressures in way of
gs.

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

Definitions

rudder force in [N]

rudder torque in [Nm]

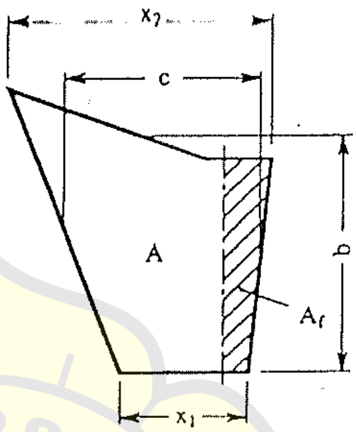
total movable area of the rudder in [m^2]
For nozzle rudders, A is not to be taken less than 1,35 times the projected area of the nozzle;

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

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

b = mean height of rudder area in [m]

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



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

Fig. 14.1

Λ = aspect ratio of rudder area A_t

Λ = b^2/A_t

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

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

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

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

B. Rudder Force and Torque

1. Rudder force and torque for normal rudders

1.1 The rudder force is to be determined ac-

Table 60

Mooring and Warping Ropes

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

Continued

Characteristic	Towing rope: Length, m	Towing rope: Circumference of hemp rope, mm	Towing rope: Diameter of steel rope, mm	Warping hawsers: Total length, m	Warping hawsers: Number of ropes	Warping hawsers: Circumference of hemp rope, mm	Warping hawsers: Diameter of steel rope, mm	Cable warps: Total length, m	Cable warps: Number of ropes	Cable warps: Circumference of hemp rope, mm	Cable warps: Diameter of steel rope, mm
2700	220	350	34.5	640	4	225	24	200	2	200	21.5
3000	220	350	34.5	640	4	225	24	200	2	200	21.5
3300	240	375	39	640	4	250	26	200	2	200	21.5
3600	240	375	39	640	4	250	26	200	2	200	21.5
3900	240	400	43.5	640	4	250	26	200	2	200	21.5
4200	240	400	43.5	640	4	250	26	200	2	225	24
4500	240	425	48.5	720	4	250	26	200	2	225	24
4800	240	425	48.5	720	4	250	26	200	2	225	24
5100	240	—	53	720	4	275	28	240	2	225	24
5400	240	—	53	800	4	275	28	240	2	250	26
5800	240	—	53	880	4	275	28	240	2	250	26
6200	240	—	57	960	6	300	30	240	2	250	26
6600	240	—	57	960	6	300	30	240	2	250	26
7000	240	—	57	960	6	300	30	240	2	250	26
7400	240	—	57	960	6	300	30	480	4	250	26
7800	240	—	57	960	6	300	30	480	4	250	26
8200	240	—	61.5	960	6	300	30	480	4	250	26
8600	240	—	61.5	960	6	325	32	480	4	250	26
9000	240	—	61.5	960	6	325	32	480	4	250	26
9600	240	—	61.5	960	6	325	32	480	4	250	26

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

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

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	—	8.8
2	75	2	200	25	125	13	50	—	8.8
3	100	2	250	50	125	15	50	—	11
4	150	2	300	50	150	16	50	—	11
5	200	2	350	50	175	17	75	—	11
6	250	2	450	75	200	18	75	11	13
7	300	2	500	75	225	19	75	13	13
8	350	2	600	100	250	20	75	14	15.5
9	400	2	700	100	275	21	75	14	15.5
10	450	2	750	125	300	22	100	15	17.5
11	500	2	800	150	300	24	100	16	17.5
12	550	2	900	175	325	25	100	16	17.5
13	600	3	1500	200	350	27	100	17	17.5
14	650	3	1700	225	350	28	100	18	19.5
15	700	3	1800	250	375	29	100	18	20.5
16	750	3	2100	250	375	30	100	19	20.5
17	800	3	2250	250	375	31	125	19	20.5
18	850	3	2400	275	375	32	125	20	22
19	900	3	2700	300	375	33	125	21	24
20	950	3	3000	300	400	34	125	21	24
21	1000	3	3200	350	400	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	28	30
29	1850	3	5500	600	450	46	150	28	30
30	2000	3	5750	700	450	46	150	29	31.5
31	2150	3	6000	700	475	48	175	29	31.5
32	2300	3	6500	800	500	49	175	29	32.5
33	2500	3	6750	800	500	50	175	29	32.5
34	2700	3	7500	900	500	52	175	30	33.5
35	3000	3	8250	1000	500	53	200	31	33.5
36	3300	3	9000	1000	500	55	200	31	33.5
37	3600	3	9750	1250	525	57	200	33	34.5
38	3900	3	10500	1250	550	59	225	33	34.5
39	4200	3	11000	1400	550	61	225	34	37
40	4500	3	11500	1500	550	62	225	35	37
41	4800	3	12900	1650	550	65	225	36	—
42	5100	3	13500	1750	550	67	250	37	—
43	5400	3	14500	1750	575	68	250	37	—
44	5800	3	15000	2000	600	70	250	40	—
45	6200	3	15800	2000	600	72	250	40	—
46	6600	3	16300	2250	600	74	275	43	—
47	7000	3	17600	2250	600	76	275	43	—
48	7400	3	18000	2250	600	77	275	44	—
49	7800	3	19500	2500	600	80	275	46	—
50	8200	3	20300	2700	600	82	275	48	—
51	8600	3	21000	2800	600	83	275	49	—
52	9000	3	22000	3000	600	85	275	50	—
53	9500	3	23000	3000	600	87	275	50	—

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

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

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

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

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

where d_c = chain bar size, mm

Table 42

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

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

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

whence

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

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

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

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

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

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

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

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

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

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

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

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

Table 43

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

Backward curved vanes are rarely employed and then only for low-pressure fans. The number of vanes is usually assigned so as to facilitate laying out and m...

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

(2) cargo hold fans, designed for ventila ting the holds of dry-store bulk carriers, tankers and refrigerated carg o 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 pollute d by mechanical impurities.

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

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

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

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

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

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

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

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

V_r = volume of the room, cu m

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

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

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

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

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

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

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

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

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

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

The amount of carbon dioxide, heat and vapour produced by persons in a room can be calculated f...

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

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

$$N_u = \frac{GH}{60\times60\times75}\ \text{hp} = \frac{GH}{60\times60\times102}\ \text{kW}$$

$$N_u = \frac{Q\gamma H}{60\times60\times75}\ \text{hp} = \frac{Q\gamma H}{60\times60\times102}\ \text{kW}$$

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

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

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

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

(A) PUMP CLASSIFICATION ACCORDING TO PURPOSE

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

1. General service pumps whose function is to ensure the seaworthiness of the ship and to provide for the domestic needs of the crew and passengers, and also to maintain the necessary sanitary conditions on board.
2. Pumps of the shipboard systems, designed to serve the main and auxiliary systems, and to facilitate the maintenance of normal conditions for their operation.
3. Special-purpose pumps in tankers, trawlers, ice-breakers, life-saving ships and dredgers.

General service pumps include:
(1) bilge pumps,
(2) sanitary pumps,
(3) fire pumps,
(4) emergency pumps.

Bilge Pumps

Bilge pumps include ballast and drainage pumps.

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

them of ballast and to transfer the ballast from certain tanks and peaks to others.

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

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

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

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

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

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

Table 3

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

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

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

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

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

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

2. Starting with compressed air

2.1 Main engines which are started with compressed air are to be equipped with at least two starting air compressors. At least one of the air compressors must be driven independently of the main engine and must supply at least 50 % of the total capacity required.

2.2 The total capacity of the starting air compressors is to be such that the starting air receivers designed in accordance with 2.4 or 2.5, as applicable, can be charged from atmospheric pressure to their final pressure within one hour.

Normally, compressors of equal capacity are to be installed.

This does not apply to an emergency air compressor which may be provided to meet the requirement stated in H.1.

2.3 If the main engine is started with compressed air, the available starting air is to be divided between at least two starting air receivers of approximately equal size which can be used independently of each other.

2.4 The total capacity of air receivers is to be sufficient to provide, without their being replenished, not less than 12 consecutive starts alternating between Ahead and Astern of each main engine of the reversible type, and not less than six starts of each main non-reversible type engine connected to a controllable pitch propeller or other device enabling the start without opposite torque. The number of starts refers to an engine in cold and ready-to-start condition.

2.5 With multi-engine installations the number of start-up operations per engine may, with the Society's agreement, be reduced according to the type of installation and the way in which the power is transmitted to the propeller.

2.6 If starting air systems for auxiliaries or for supplying pneumatically operated regulating and manoeuvering equipment or tyfon units are to be fed from the main starting air receivers, due attention is to be paid to the air consumption of this equipment when calculating the capacity of the main starting air receivers.

2.7 Other consumers with a high air consumption apart from those mentioned in 2.6 may not be connected to the main starting air system. Separate air supplies are to be provided for these units. Deviations to this require the agreement of the Society.

2.8 For the approximate calculation of the starting air storage capacity, use may be made of the formulae given in Part C of the appendix to this section.

3. Electrical starting equipment

3.1 Where main engines are started electrically, two mutually independent starter batteries are to be installed. The batteries are to be so arranged that they cannot be connected in parallel with each other. Each battery must enable the main engine to be started from cold.

The total capacity of the starter batteries must be sufficient for the execution within 30 minutes, without recharging the batteries, of the same number of start-up operations as is prescribed in H.2.4. or H.2.5, as appropriable, for starting with compressed air.

3.2 If two or more auxiliary engines are started electrically, at least two mutually independent batteries are to be provided. Where starter batteries for the main engine are fitted, the use of these batteries is acceptable.

The capacity of the batteries must be sufficient for at least three start-up operations per engine.
If only one of the auxiliary engines is started electrically, one battery is sufficient.

3.3 The starter batteries may only be used for starting (and preheating where applicable) and for monitoring equipment belonging to the engine.

3.4 Steps are to be taken to ensure that the batteries are kept charged and the charge level is monitored.

4. Start-up of emergency generating sets

4.1 Emergency generating sets are to be so designed that they can be started up readily even at a temperature of 0 °C.

If the set can be started only at higher temperatures, or where there is a possibility that lower ambient temperatures may occur, heating equipment is to be fitted to ensure ready reliable starting.

The operational readiness of the set must be guaranteed under all weather and seaway conditions. Fire flaps required in air inlet and outlet openings must only be closed in case of fire and are to be kept open at all other times. Warning signs to this effect are to be applied. If the flaps close, an alarm must be activated. No alarm is required in the case of automatic fire flap actuation dependent on the operation of the set. Air inlet and outlet openings must not-be fitted with weatherproof covers.

4.2 Each emergency generating set required to be capable of automatic starting is to be equipped with an automatic starting system approved by the Society, the capacity of which is sufficient for at least three successive starts (see Volume IV, Rules for Electrical Installation, Section 3, C).

In addition, a second energy source is to be installed

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

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

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

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

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

where ΔH = loss of head in the fan.

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

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

where N_{fr} = power lost in overcoming fluid friction
$\beta = (5 \text{ to } 15)\left(1 + 5\frac{b_2}{D_2}\right)$ = coefficient obtained from data compiled by the Central Institute of Aero- and Hydrodynamics
b_2 = width of the impeller at air outlet
D_2 = impeller diameter at air outlet

For backward-curved vanes—$\eta_{fr} \cong 0.6$ to 0.75
For forward-curved vanes—$\eta_r \cong 0.75$ to 0.9.

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

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

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

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

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

2-2. Design and Selection of Fans

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

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

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

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

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

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

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

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

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

$H = \overline{H}$ [illegible]

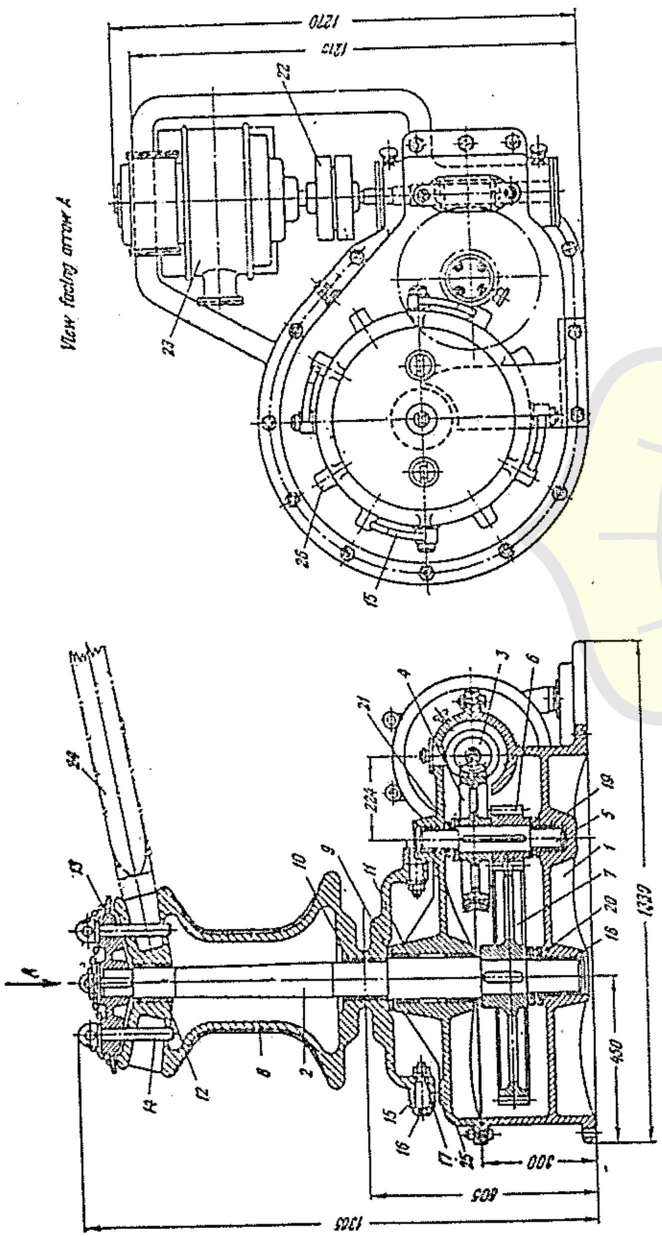


Fig. 169.

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

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

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

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

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

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

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

G_a = weight of the anchor, kg

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

L_a = length of the suspended cable, m

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

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

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

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

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

In hoisting one anchor

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

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

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

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

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

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* In breaking away one anchor from the bottom

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

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

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

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

The value of $(\alpha_i k_i p_{is} - p_{ss})$ 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

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

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

The steam consumption of the engine driving the anchoring arrangement is

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

where g_{ia} = specific steam consumption, kg per ihp-hr (the same values are taken as for a steam steering engine).

If need arises to determine the pull on the cable lifter from data measured on the anchoring mechanism, formula (390) can be used.

Solving Posdyunin's formula (389) for the torque developed on the shaft of the steam engine we can write

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

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

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

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

The diameter of the warp ends is taken equal to

(a) $D_{we} = (5 \text{ to } 8) d_w$ for hemp ropes (391)

(b) $D_{we} = (15 \text{ to } 20) d_w$ for steel ropes (392)

where d_w = diameter of the warping hawser.

Warp end diameters determined from the diameter of hemp ropes will be suitable for steel ropes as well.

Denoting the hawser heaving-in speed as v_w m per sec we can find the speed of the warping shaft from the length of hawser heaved in per minute. Thus

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

where v_w = hawser heaving-in speed, m per sec, is to be assigned according to the pull of the warp end (Table 58).

The gearing ratio between the warping shaft and the shaft of the motive unit is

$$i_w = \frac{n_m}{n_w}$$

The pulling force developed on the warp end is

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

where M_{we} = torque developed on the warp end
η_w = efficiency of the transmission between the warping and motive unit shafts.

If n_m rpm is the speed of the motive unit shaft, the speed at which the hawser is heaved in will be

$$v_w = \frac{\pi (D_{we} + d_w) n_m}{\text{[illegible]}} \quad \text{m per sec} \tag{395}$$

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 cable lifter (Fig. 170) [illegible] difference in pressures in the chambers will cause the vanes to turn clockwise.

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

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

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

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

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

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

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

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

Table 47

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

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

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

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

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

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

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

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

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

It follows from formula (314) that

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

Combining equations (315) and (316) we obtain

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

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

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

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

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

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

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

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

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

The shaft horse power can also be determined from the shaft

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

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

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

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

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

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

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

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

where c = minimum number of blocks.

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

Table 63

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

The winch head diameter is

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

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

The required winch head speed is found from the equation

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

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

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

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

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

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

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

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

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

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

and

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

The cylinder diameter and indicated power of steam boat winches are determined from the same maximum pressure, p_f kg per sq m, then the amount of liquid pumped is

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

This equation can be solved for V_e and V_f:

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

and

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

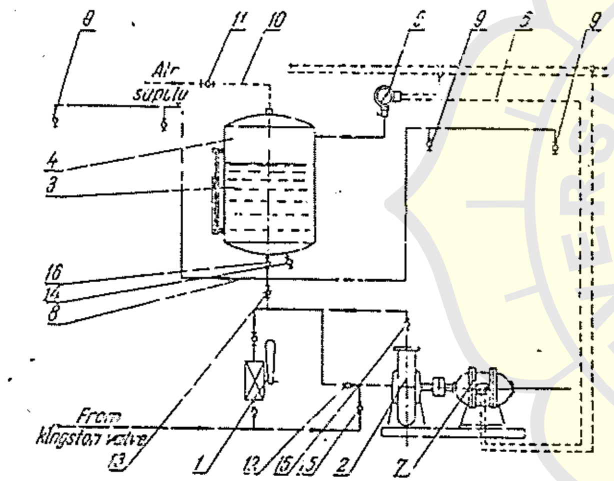


Fig. 189.

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

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

Therefore the minimum and maximum volumes of the air are

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

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

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

Such tanks may also be

(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

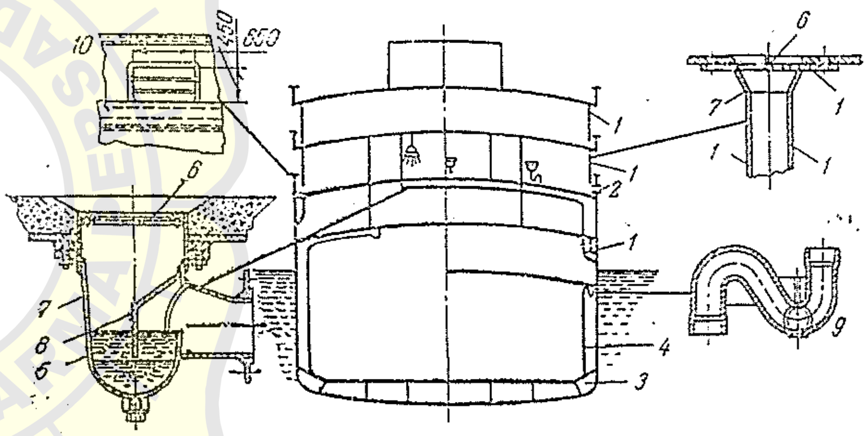


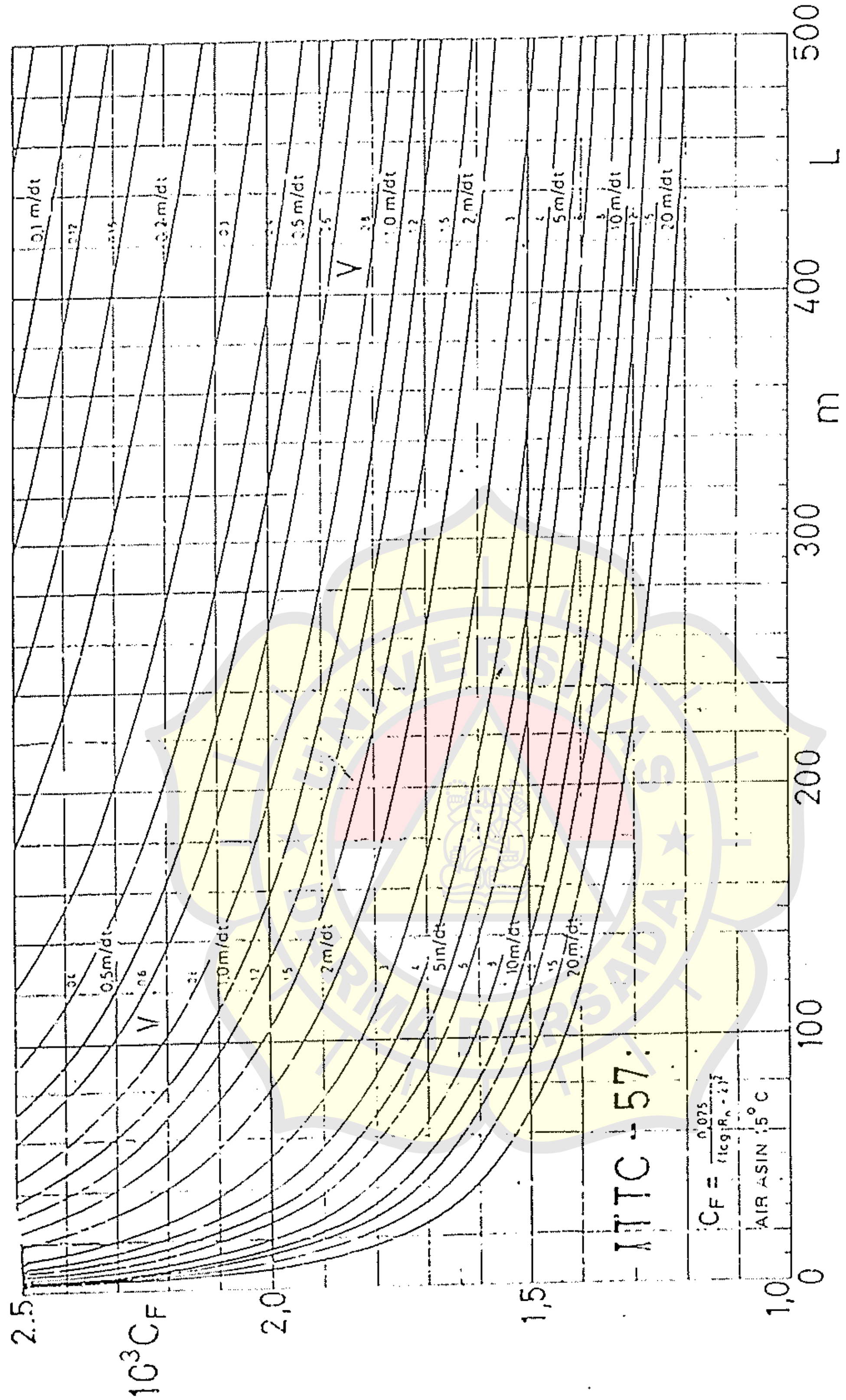
Fig. 190.

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.



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

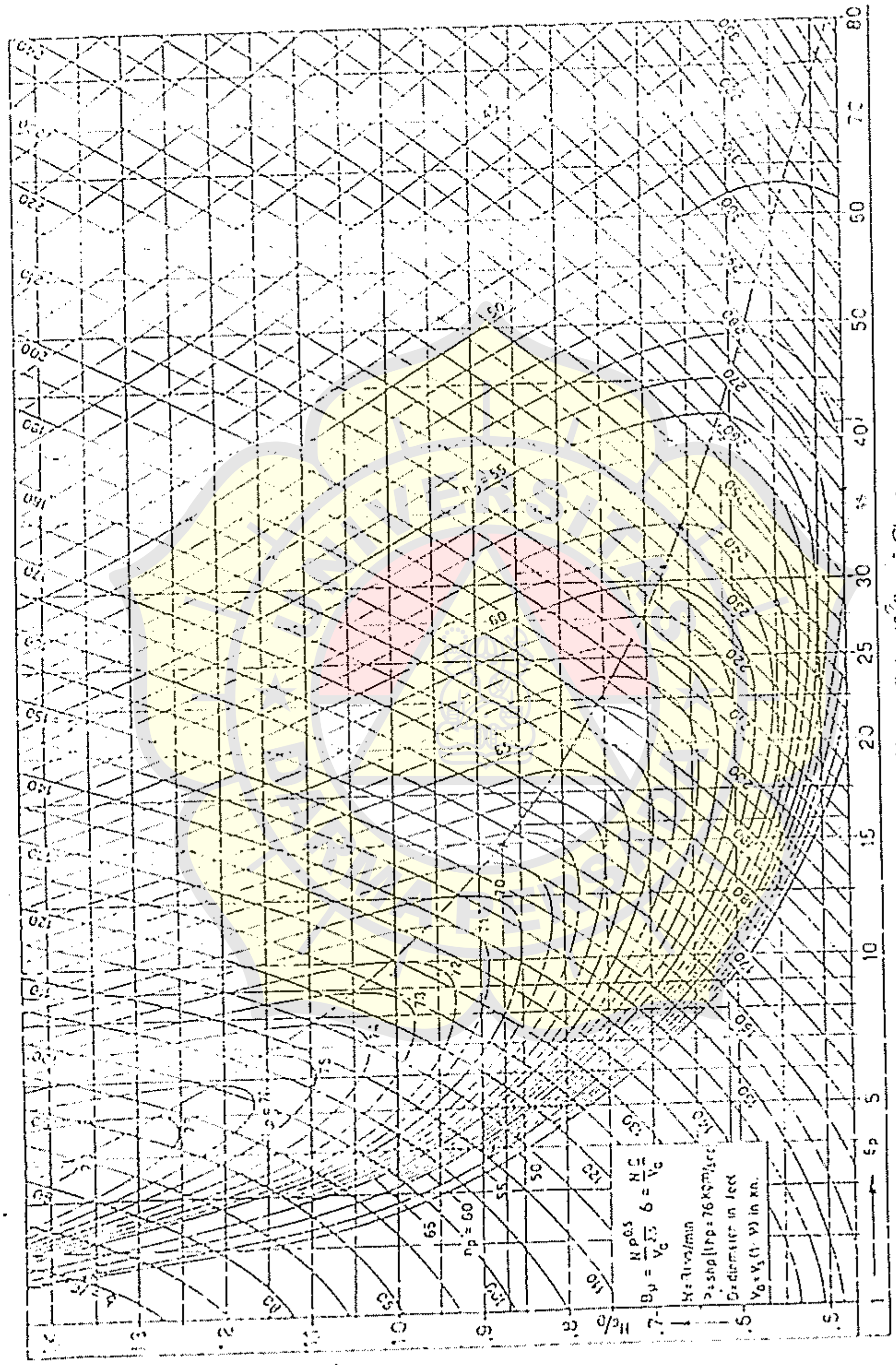
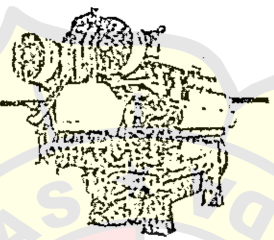


Fig. 3.15 Troost B.4 – 40 B_p – δ Chart

WÄRTSILÄ

NOHAB 25



Wärtsilä Nohab 25

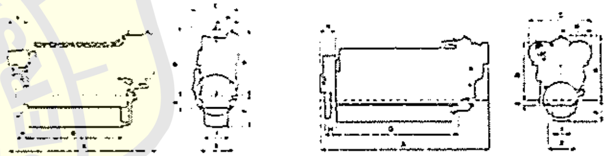
The Wärtsilä Nohab 25 medium-speed engine represents a further development of the popular F series. Incorporating experience from more than 1.800 engines, the Wärtsilä Nohab 25 is a reliable and compact high performance engine. It provides both low fuel consumption and running costs for cost-effective power. The high number of repeat orders is proof of customer confidence

MAIN DATA

Cylinder bore	250 mm
Piston stroke	300 mm
Speed	720–1000 rpm
Mean effective pressure	20.8–18.7 bar
Piston speed	7.2–10.0 m/s
FUEL SPECIFICATION:	
Fuel oil	380 cSt/50°C 3500 sRI/100°F

Rated power: Propulsion engines

Output	Output in kW/BHP at									
	720 rpm		750 rpm		825 rpm		900 rpm		1000 rpm	
Engine type	kW	BHP	kW	BHP	kW	BHP	kW	BHP	kW	BHP
6R25	1110	1500	1150	1560	1270	1730	1380	1880	1545	2100
8V25	1470	2000	1530	2080	1690	2300	1840	2500	[illegible]	[illegible]



Principal engine dimensions (mm) and weights (tons)

Engine type	A	B	C	D	E	F Max.	G
6R25	4245	1950	1355	2070	350	555*	2700
8V25	3615	1950	1855	1960	180	710*	2380
12V25	4665	2205	1960	1960	180	710*	3220
16V25	5650	2205	2110	1960	180	710*	4050

Engine type	H Max.	I	K	M	N	Weight
6R25	375**	660	920	650	560	9.9
8V25	375**	660	920	885	525	11.2
12V25	375**	660	920	955	525	16.6
16V25	375**	660	920	1055	525	21.1

* Max. with wet sump. ** Max. Depending on flywheel size.

Rated power: Generating sets

Output	Output at							
	720 rpm/ 60 Hz		750 rpm/ 50 Hz		900 rpm/ 60 Hz		1000 rpm/ 50 Hz	
Engine type	Eng. kW	Gen. kW	Eng. kW	Gen. kW	Eng. kW	Gen. kW	Eng. kW	Gen. kW
6R25	1110	1050	1150	1100	1380	1320	1545	1470
8V25	1470	1400	1530	1450	1840	1770	1840	1770
12V25	2210	2100	2300	2190	2760	2650	2760	2550
16V25	2940	2800	3070	2920	3680	3530	3680	3530

Principal genset dimensions (mm) and weights (tons)

Engine type	Length	Breadth	Height	Engine weight	Weight of gen. set
6R25	6600	1730	2950	10.9	20

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

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

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

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

Besarnya daya adiabatik teoritis dapat dihitung dengan rumus

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

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

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

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

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

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

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

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

Capacity table at varying r. p. m.

TYPE	Capacity m³/h	Capacity l/min	Branches Suc.	Branches Del.	1450 r.p.m. H	1450 r.p.m. N	1750 r.p.m. H	1750 r.p.m. N	2000 r.p.m. H	2000 r.p.m. N	2500 r.p.m. H	2500 r.p.m. N	2900 r.p.m. H	2900 r.p.m. N	3500 r.p.m. H	3500 r.p.m. N
SA-20-90/9	0	0	3/4"	3/4"					4,7	0,06	7,3	0,12	9,9	0,18	14,4	[illegible]
Max.	1,5	25	–	–					4,4	0,08	7,0	0,14	9,7	0,21	14,2	0,38
6000 r.p.m.	3,0	50	–	–					3,4	0,10	6,1	0,18	8,9	0,27	13,5	0,44
	4,5	75	–	–							4,2	0,21	7,1	0,30	11,7	0,50
SA-25-122/12	0	0	1"	1"	5,0	0,14	7,3	0,23	9,5	0,35	14,6	0,72	18,4	0,9	27	1,7
Max.	2	33	–	–	4,0	0,18	6,9	0,29	9,0	0,42	13,7	0,78	17,0	1,0	26	1,8
5000 r.p.m.	4	66	–	–	3,0	0,20	5,5	0,33	7,7	0,46	11,5	0,79	14,6	1,1	23	1,9
	6	100	–	–			3,0	0,36	5,4	0,52	8,0	0,80	10,7	1,1	18	2,0
SA-35-135/12	0	0	1½"	1½"	6,25	0,16	9,1	0,26	11,7	0,4	18,6	0,9	25	1,4	36	2,2
Max.	5	83	–	–	5,0	0,21	7,7	0,35	10,4	0,5	17,0	1,0	23	1,5	34	2,5
4500 r.p.m.	10	166	–	–	2,7	0,26	5,3	0,43	7,9	0,6	14,4	1,1	20	1,7	31	2,9
	15	250	–	–					4,3	0,7	10,3	1,3	16	1,9	27	3,3
[illegible]	0	0	2"	2"					18,0	0,9	28	1,4	38	2,7	55	[illegible]
Max.	6	100	–	–					16,5	1,1	26	2,1	36	3,2	52	[illegible]
3500 r.p.m.	12	200	–	–					12,5	1,4	22	2,5	32	3,6	48	[illegible]
	18	300	–	–							14	2,7	25	4,3	42	[illegible]
SA-50-180/8	0	0	2"	2"	11,5	0,7	16,4	1,2	21	1,7	33	3,3				
Max.	7,5	125	–	–	9,5	0,9	14,3	1,5	19	2,2	30	4,0				
2500 r.p.m.	15,0	250	–	–	6,6	1,0	11,2	[illegible]	16	[illegible]	27	4,5				
	22,5	375	–	–			5,3	[illegible]	11	[illegible]	22	5,0				
SA-65-250/8	0	0	3"	2½"					39	[illegible]	61	10	82	16	100	22
Max.	20	333	–	–					35	[illegible]	57	[illegible]	76	[illegible]	91	[illegible]
3500 r.p.m.	40	666	–	–					31	[illegible]	51	[illegible]	70	[illegible]	87	[illegible]
	60	1000	–	–							36	[illegible]	63	23	79	[illegible]
SA-65-250/17	0	0	3"	2½"	23	[illegible]	33	[illegible]	42	[illegible]						
Max.	20	333	–	–	21	[illegible]	32	[illegible]	40	[illegible]						
2200 r.p.m.	40	666	–	–	19	4,6	28	[illegible]	37	[illegible]						
	60	1000	–	–	16	5,4	25	[illegible]	33	[illegible]						
SA-80-160/17	0	0	3"	3"					14,0	1,2	22	[illegible]	30	4,0	44	7,0
Max.	20	333	–	–					13,0	2,1	21	4,0	28	5,6	42	9,4
4000 r.p.m.	40	666	–	–					9,5	[illegible]	18	4,6	26	7,2	39	11,4
	60	1000	–	–							9	[illegible]	18	8,2	33	13,0
SA-80-220/17	0	0	3"	3"	16,3	[illegible]	23	[illegible]	31	5,9	48	6,0	65	12	95	[illegible]
Max.	20	333	–	–	14,5	[illegible]	21	[illegible]	26	[illegible]	47	9,5	63	[illegible]	93	[illegible]
3500 r.p.m.	40	666	–	–	10,5	[illegible]	16	[illegible]	24	[illegible]	44	[illegible]	[illegible]	17	90	[illegible]
	60	1000	4"	–	3,7	[illegible]	11	[illegible]	[illegible]	[illegible]	33	[illegible]	55	19	84	[illegible]
SA-100-235/28	0	0	4"	4"	16	2,2	23	[illegible]	30	8,0	47	[illegible]				
Max.	40	666	–	–	15	4,5	22	7,7	29	11,3	46	21				
3000 r.p.m.	80	1333	–	–	12	[illegible]	19	[illegible]	27	13,5	44	[illegible]				
	120	2000	–	–	7	[illegible]	[illegible]	[illegible]	22	[illegible]	[illegible]	[illegible]				
SA-150-260/33	0	0	6"	6"	17	7,6	26	12	33	19						
Max.	80	1333	–	–	16	11,0	24	15	32	26						
2500 r.p.m.	160	2666	–	–	14	14,0	22	23	29	33						
	240	4000	–	–	9	17,0	17	27	25	38						
SA-200-320/50	0	0	8"	8"	23	25	40	45	53	67						
Max.	150	2500	–	–	27	34	39	57	51	84						
2000 r.p.m.	300	5000	–	–	24	43	36	69	48	100						
	450	7500	–	–	19	49	32	79	44	114						

Motor HK = N + 20 ÷ 10 %

Pipe line dimension

m³/h	Suction mm	Suction inch	Delivery mm	Delivery inch
2	20	3/4"	20	3/4"
3,5	25	1"	25	1"
5	32	1¼"	25	1"
9	40	1½"	32	1¼"
14	50	2"	40	1½"
24	65	2½"	50	2"
36	80	3"	65	2½"
57	100	4"	80	3"
34	125	5"	100	4"
125	150	6"	125	5"
225	200	8"	150	6"
350	250	10"	200	8"
500	300	12"	250	10"

Conversion factors:
1 m = 3.28 ft

As the operation figures of any pump may be changed by varying the diameter of the impeller and/or the blade width, or by changing the number of revolutions, DESMI pumps may in most cases be adapted to specific operation conditions – for instance requirements as to max. power consumption – in a manner to ensure that the high rate of efficiency is maintained. We should be pleased to forward data sheets stating the capacity area of each pump type.
The capacities apply at 0 m suction lift, and all numbers of revolutions apply to direct coupling. In case of belt drive please apply to us for details.