BAB VII
PENUTUP

VII.1 Kesimpulan

Dari hasil perhitungan yang telah dilakukan pada kapal rancangan yaitu Kapal Curah 13.000 DWT dengan dimensi sebagai berikut :

Panjang keseluruhan (Loa)	: 132,59 m
Panjang antara garis tegak (Lbp)	: 120,54 m
Lebar kapal (B)	: 20,09 m
Sarat kapal (T)	: 8,4 m
Kecepatan (Vs)	: 14,52 Knot
Dead Weight (DWT)	: 13000 Ton
Jarak Pelayaran (S)	: 525 mil
Klasifikasi	: BKI

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

⇨ Di dalam perancangan kamar mesin, tidak terlepas dari adanya asumsi asumsi yang diberikan untuk mempermudah dalam perhitungan dengan tidak mengabaikan tanggung secara teknis, ekonomis dan peraturan peraturan yang ada, sehingga hasil perhitungan dapat mendekati keadaan yang sebenarnya.

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

⇨ Peletakan permesinan berpengaruh pada stabilitas kapal.

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

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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

$$A_c = \frac{\Delta^{2/3} V^3}{P} \qquad (5.1.1)$$

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

$$P_p = \frac{\Delta_p^{2/3} V_p^3}{A_c} \qquad (5.1.2)$$

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.

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 terse[...] untuk $\nu = 1{,}139 \times 10^{-6}$ m s^{-1}, $\rho = 1000$ t/[...] dan $T = 15°$C. Karena itu untuk memakai diagr[...] tersebut dengan kondisi yang lain, yaitu massa j[...] dan suhu yang lain, panjang kapal harus diubah [...] sebelum memakai diagram tersebut sebagai ber[...]

$$L_1 = \frac{1{,}139}{10^6 \nu} L \qquad (5.5[...]$$

4. C_R dinyatakan sebagai fungsi angka Fro[...]

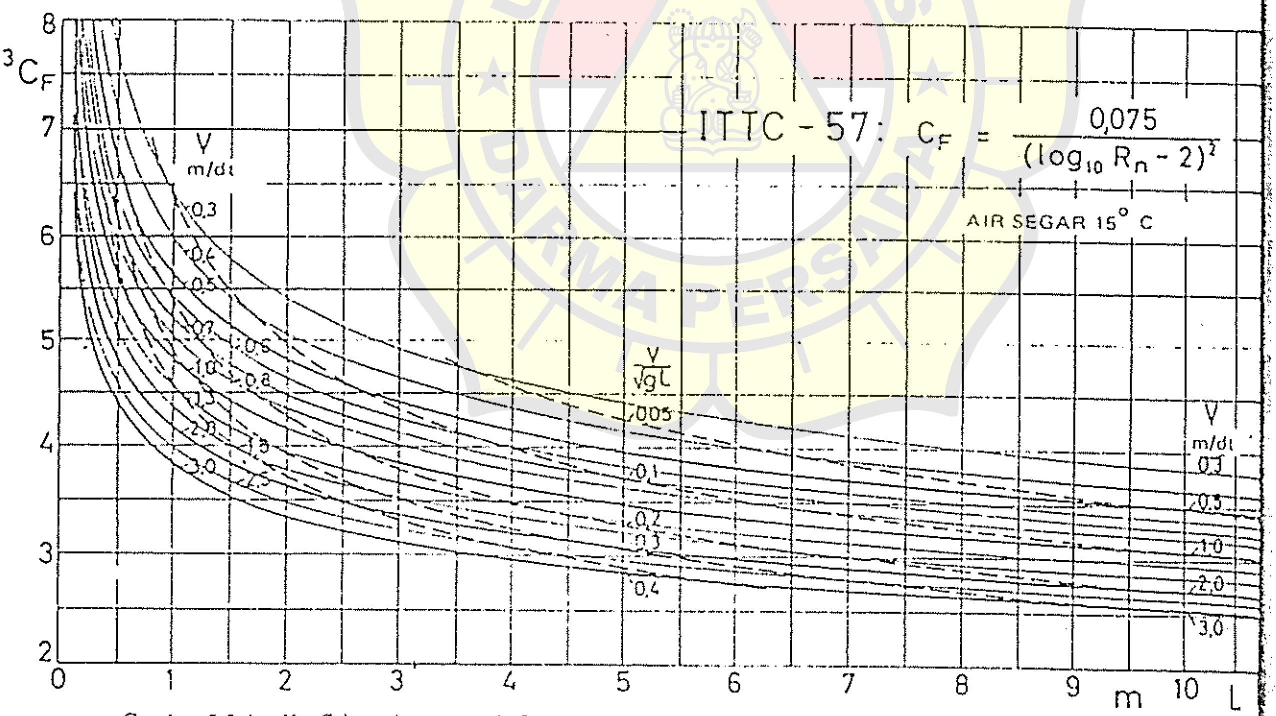
$$F_n = \frac{V}{\sqrt{gL}} \qquad (5.5[...]$$

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

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

$$\varphi = \frac{\nabla}{LBT\beta} \qquad (5.5.[...]$$

B adalah lebar, T sarat, dan β koefisien penampa[...] melintang tengah kapal.



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

6. Diagram [...] kurva ra[...] 2,5. Dia[...] 5.5.13.

Dalam diag[...] garis terpu[...] tersebut did[...] jumlahnya [...] itu kergua[...] Selain itu, [...] dekat daera[...] yang menyc[...] negatif, ti[...] Perubahan [...] dalam daera[...] berarti pad[...] Perlu pu[...] tersebut ber[...] standar, yai[...] nya standar [...] nya merupa[...] moderat, da[...] condong (r[...] Tahanan [...] dapat dihitu[...]

R

P_L

Dalam hal

C_R = k[...] yang "st[...] 5.5.5 – 5[...]

C_F = k[...] dengan [...]

atau dap[...] ini kont[...] yang be[...] panjang [...] 1,188 × [...] 15°C. [...] jenis dar[...] tersebut [...] berikut [...]

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/m³, 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.

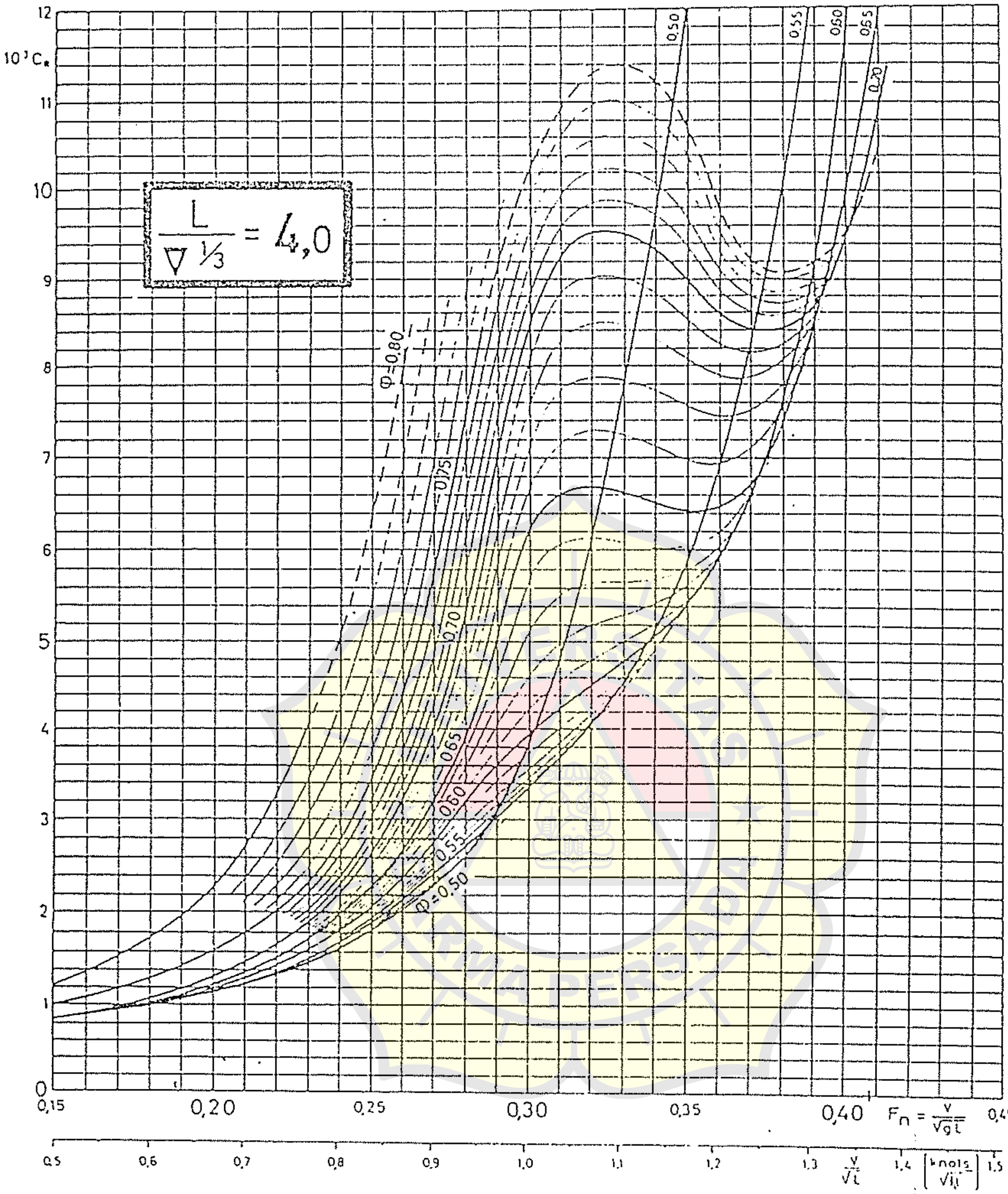
Berdasarkan hasil pemeriksaan materi pengujian yang ada saat ini maka disarankan untuk memakai rumus koreksi berikut ini :

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

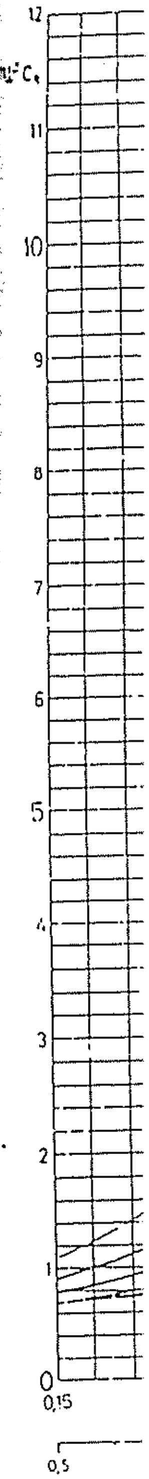
Koreksi ini dapat mempunyai harga yang negatif atau positif.

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.



Gambar 5.5.5. Koefisien tahanan sisa terhadap rasio kecepatan – panjang untuk harga koefisien prismatik longitudinal yang berbeda-beda. $L/\nabla^{1/3} = 4,0$.



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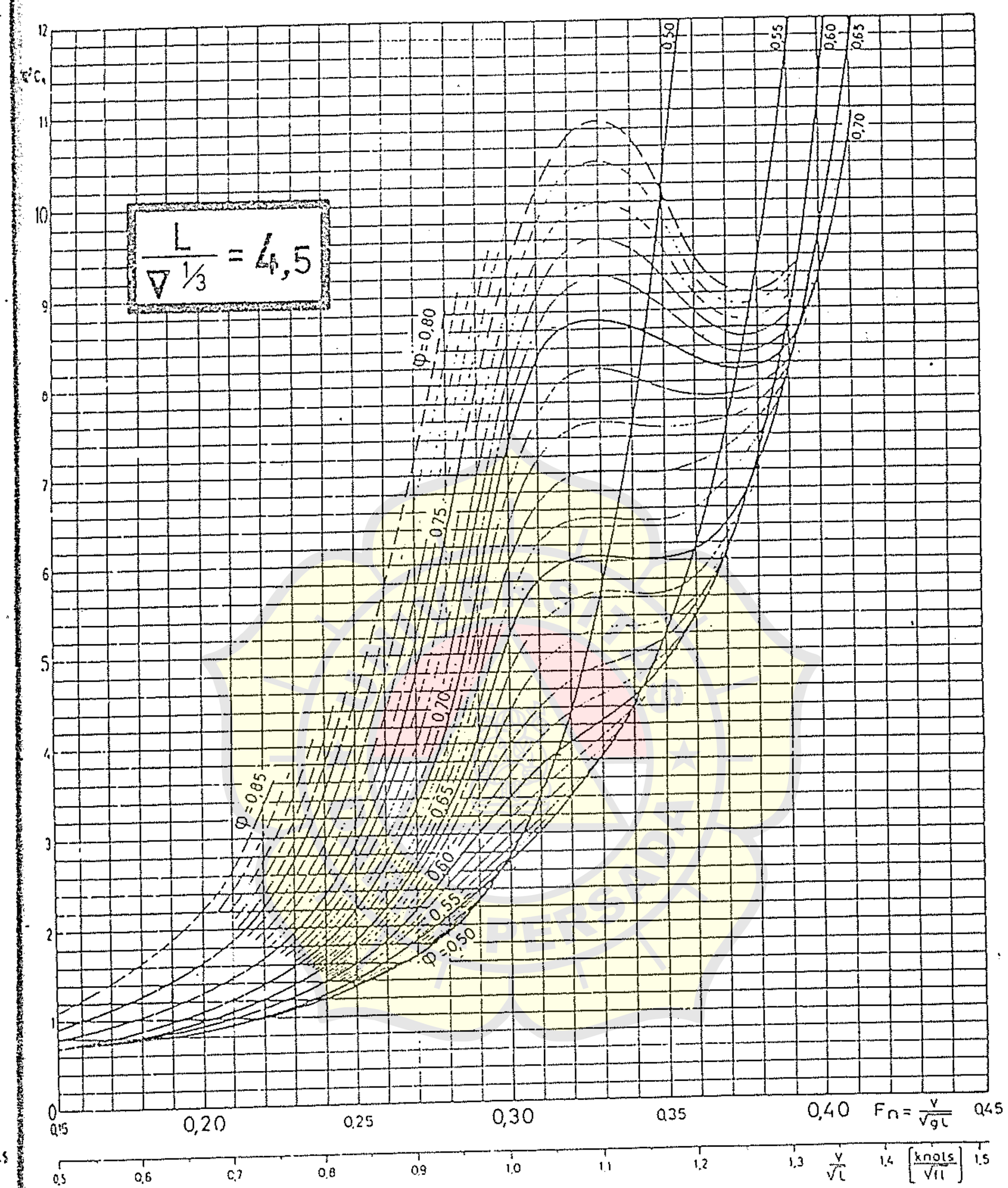
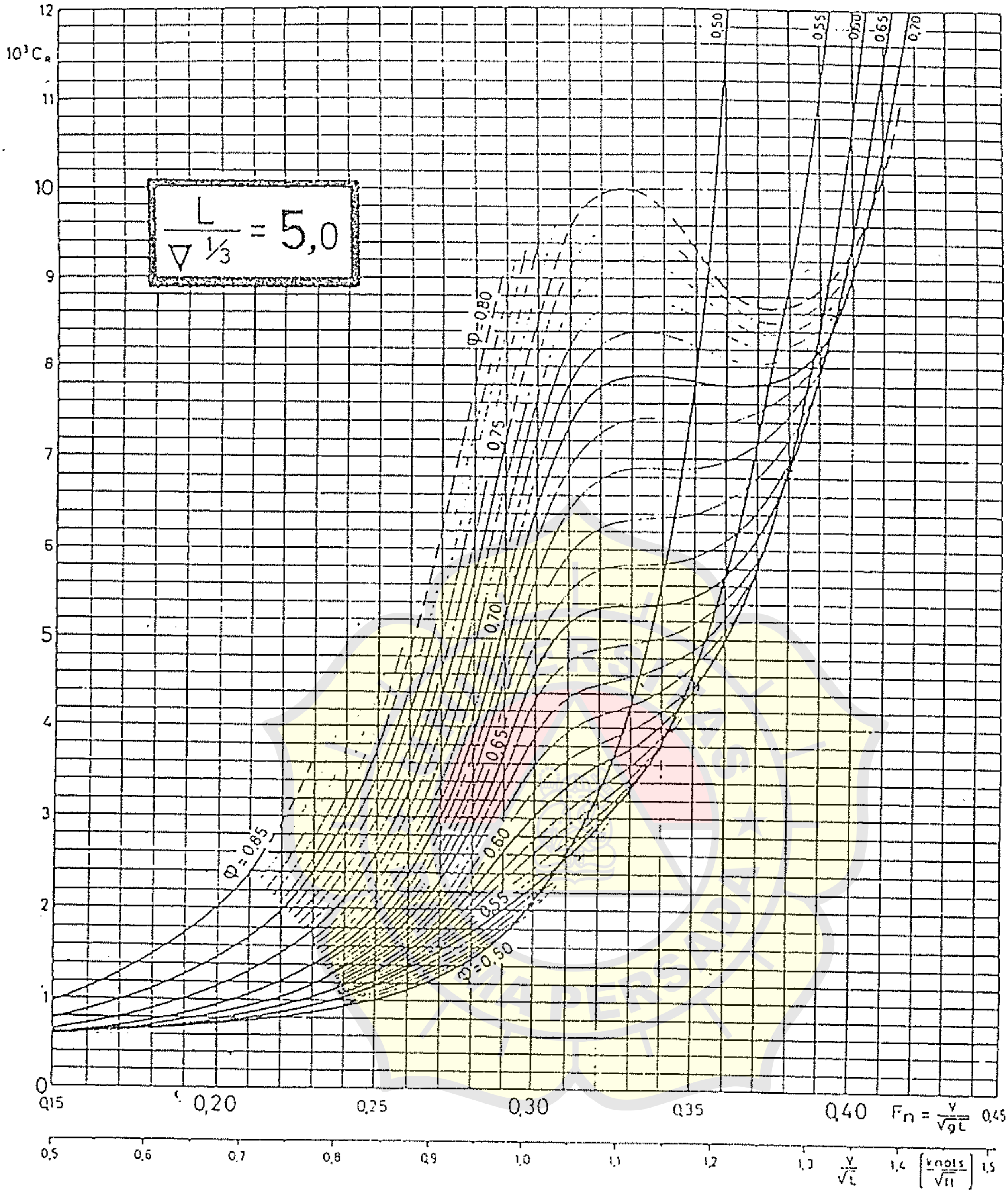
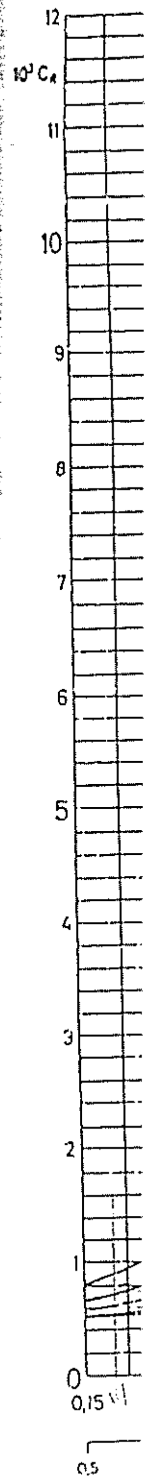
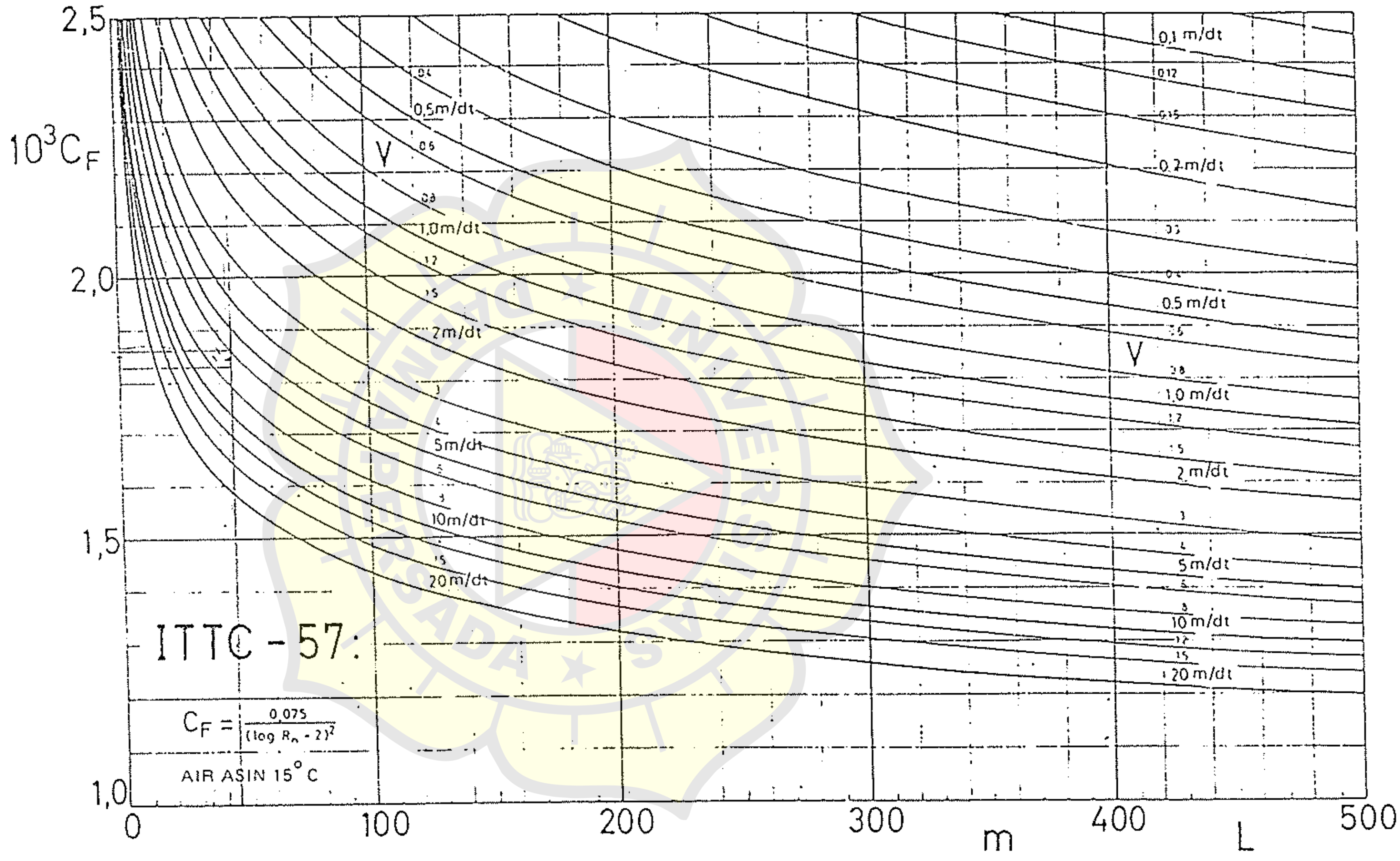


Gambar 5.5.6. Koefisien tahanan sisa terhadap rasio kecepatan-panjang untuk harga koefisien prismatik longitudinal yang berbeda-beda. $L/\nabla^{1/3} = 4,5$.

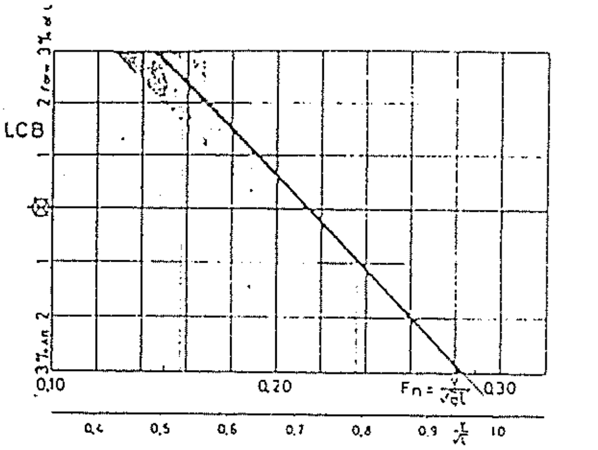


Gambar 5.5.7. Koefisien tahanan sisa terhadap rasio kecepatan-panjang untuk harga koefisien prismatik longitudinal yang berbeda-beda, $L/\nabla^{1/3} = 5{,}0$





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



ɔar 5.5.15. LCB standar. Letak longitudinal titik benam yang dang terbaik.

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

bagaimana disebutkan sebelumnya, karena letak standar dianggap merupakan letak yang berikan tahanan yang paling kecil maka letak yang pada prinsipnya akan memberikan tahanan yang besar. Penambahan tahanan tersebut harus dicari ın jalan mengalikan penyimpangan LCB dari ar, yaitu

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

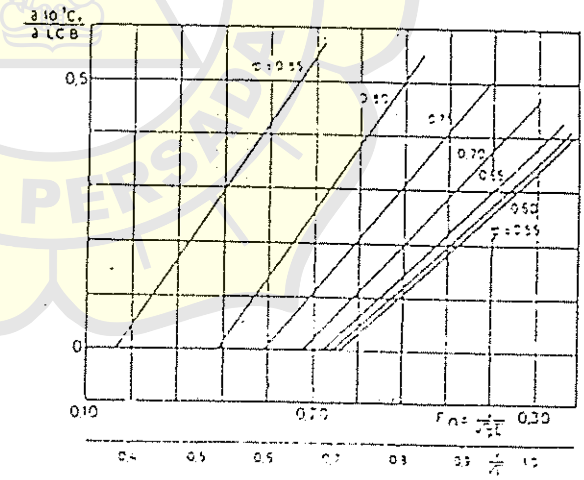
n faktor $\partial 10^3 C_R / \partial LCB$. Harga faktor ini dapat oleh dari Gb. 5.5.16, dan ini hanya berlaku untuk yang berada di depan $LCB_{standar}$. Mengenai LCB berada di belakang $LCB_{standar}$, semua sumber ada mempunyai pendapat yang saling tangan. Namun demikian, karena kecenderungan nya letak demikian itu sangat kecil maka bagian koreksi dalam hal itu tidak akan memberi-salahan yang berarti.

gan demikian maka koefisien tahanan sisa koreksi tersebut untuk kapal yang mempunyai i depan LCB standar adalah :

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

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.

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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)

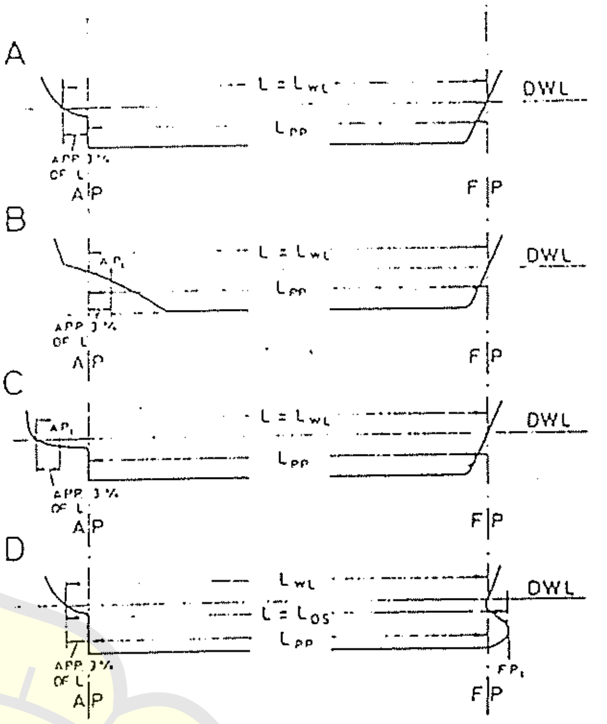
Sebagaimana disebutkan sebelumnya, kurva tahanan (yang diperoleh berdasarkan Gb. 5.5.5 – 5.5.13) dianggap berlaku untuk yang mempunyai bentuk "standar", yaitu penampangnya bukan yang benar-benar berbentuk U ataupun V. Karena itu, dalam menghitung daya efektif untuk perancangan awal umumnya tidak diperlukan 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	0,24	0,27	0,30	0,33	0,36	φ
		+ 0,2	0	– 0,2	– 0,4	– 0,4	– 0,4	0,50
		+ 0,2	0	– 0,2	– 0,3	– 0,3		0,60
	+ 0,2	0	– 0,2	– 0,3	– 0,3			0,70
+ 0,1	0	– 0,2						0,80

(5.5.21)

ANGGOTA BADAN KAPAL

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

TAHANAN TAMBAHAN

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

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

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

$$
\begin{array}{rl}
\text{Displasemen} & \\
\hline
1.000 \text{ t} & C_A = 0{,}6 \times 10^{-3} \\
10.000 \text{ t} & \quad\; = 0{,}4 \times 10^{-3} \\
100.000 \text{ t} & \quad\; = 0 \\
1.000.000 \text{ t} & \quad\; = -0{,}6 \times 10^{-3}
\end{array}
\qquad (5.5.24)
$$

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

ANGGOTA BADAN KAPAL

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

$$C_{F'} = C_F \frac{S_1}{S} \qquad (5...$$

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

TAHANAN UDARA DAN TAHANAN KEMUDI

Tahanan udara dapat ditentukan dengan memaka... mengenai struktur yang berada di atas air da... udara. Namun demikian, besarnya tahanan ... umumnya tidak terlalu penting, dan upaya yang ... dilakukan untuk mendapatkan hasil perhitunga... tepat mungkin tidak memadai dengan penti... pengaruh udara tersebut. Karena itu, jika data me... angin dalam perancangan kapal tidak diketahui ... disarankan untuk mengoreksi $10^3 C_R$ sebagai beri...

$$10^3 C_{AA} = 0{,}07 \qquad (5...$$

Koreksi untuk tahanan kemudi mungkin s...

$$10^3 C_{AS} = 0{,}04 \qquad (5...$$

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

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

KONDISI PELAYARAN DINAS

Tahanan dan daya efektif yang dihitung de... memakai diagram yang diberikan di sini berlaku ... kapal dalam kondisi pelayaran percobaan, yaitu, ... kondisi ideal dari segi angin, gelombang, kedalama... dan kemulusan badan kapal. Untuk kondisi ra... pelayaran dinas harus diberikan kelonggaran tam... pada tahanan dan daya efektif yang disebabka... angin, laut, erosi, dan fouling pada badan k...

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

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

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

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

   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) :

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

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.

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	
		$2\alpha^\circ = 70^\circ$	$2\alpha^\circ = 64^\circ$
Ice breakers	15	4.66	4.25
Sea-going craft and transport ships . . .	25 to 30	2.8 to 2.34	2.50 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:

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

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

where α° = 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

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

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

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

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

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

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

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

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

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

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

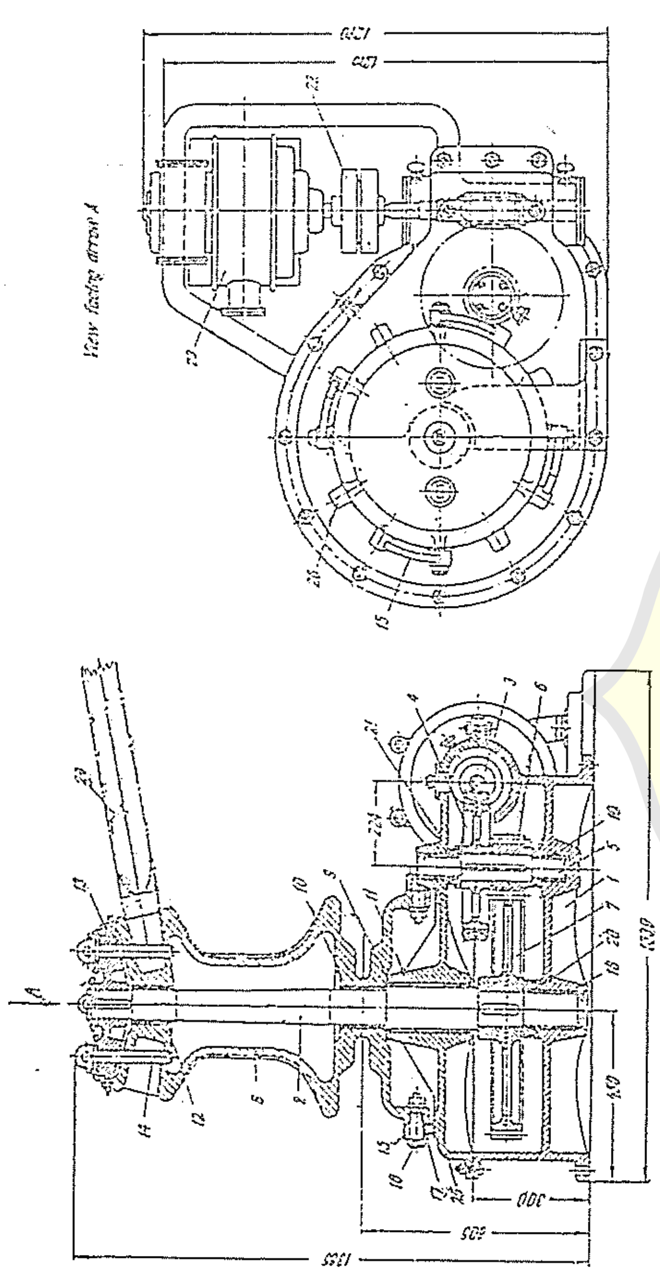


Fig. 169.

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

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

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

80 m if each anchor weighs 1,000 kg or less
90 m if the anchor weighs from 1,500 to 3,000 kg
100 m if the anchor weighs from 3,000 to 6,000 kg.

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

G_a = weight of the anchor, kg
p_a = weight per running metre of the chain cable, kg
L_a = length of the suspended cable, m
γ_a = 7,750 = density of the material of the anchor, kg per cu m
γ_w = 1,025 = density of sea water, kg per cu m
f_h = 1.28 to 1.35 = a factor taking into account the friction losses in the hawse hole and stopper.

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

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

In hoisting one anchor

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

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

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

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

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

* In breaking away one anchor from the bottom

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

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

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

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

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

Table 58

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

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

$$\chi_q = l_q h_q$$

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

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

(a) for windlasses and capstans of bower anchors:

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

(b) for the stern anchoring capstan:

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

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

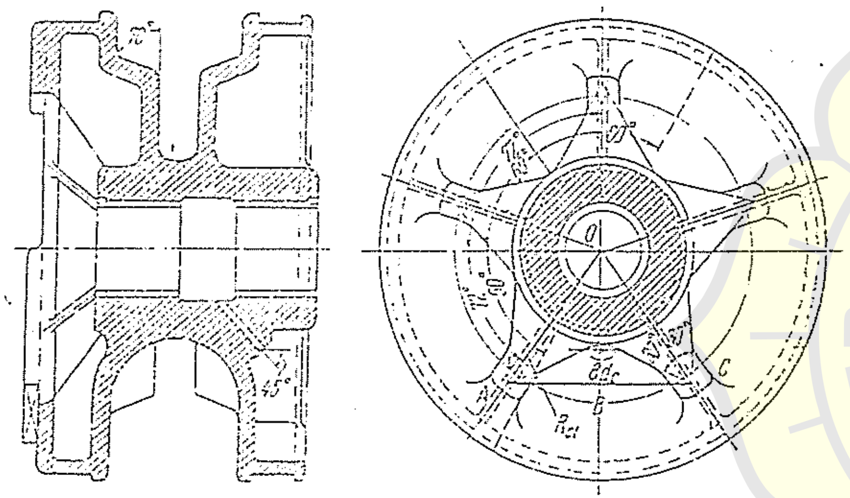


Fig. 170.

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

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

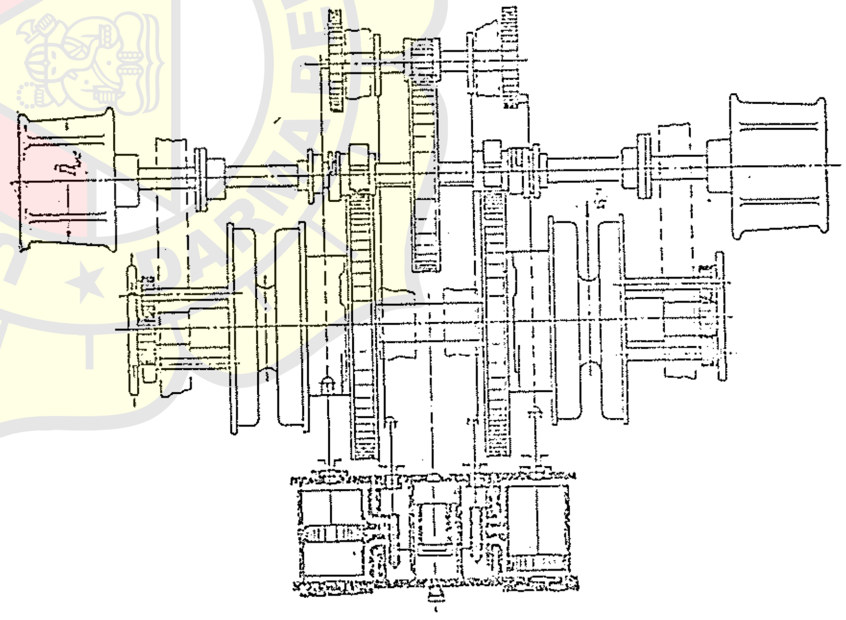


Fig. 171.

The mean shaft power of the motive unit should be

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

The mean indicated power is

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

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

$$D_{ca} = 1.57 \sqrt[3]{\frac{M_m}{\psi_a \eta_m (\alpha_i k_i p_{ls} - p_{ss})}} \quad \text{cm} \qquad (389)$$

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

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

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

$$N_{ia} = \frac{\psi_a l D_{ca}^2 (\alpha_i k_i p_{ls} - p_{ss}) n_m}{143{,}300} \quad \text{metric hp} \qquad (390)$$

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

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

$$M_m = \left(\frac{D_{ca}}{1.57}\right)^3 \eta_m \psi_a (\alpha_i k_i p_{ls} - p_{ss}) \quad \text{kg-cm}$$

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

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

Combining the last two equations we obtain

$$T_{cl} = \frac{2M_m \eta_a i_a}{D_{cl}} = 2\left(\frac{D_{ca}}{1.57}\right)^3 \frac{\eta_m \psi_a (\alpha_i k_i p_{ls} - p_{ss}) \eta_a i_a}{D_{cl}} =$$

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

The diameter of the warp ends is taken equal to

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

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

where d_w = diameter of the warping hawser.

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

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

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

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

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

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

The pulling force developed on the warp end is

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

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

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

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

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 Posdyunin formulas used in

determining these values for other steam-powered auxiliary machinery.

The required motor power rating for an electric winch can be calculated from the formula

$$N_e = \frac{(T_{max} + T_{min})\, v_b}{75 \times 60 \eta_{bw}} \quad \text{metric hp}$$

6-5. Winch Operation

Before starting a winch it is necessary to make sure by inspecti[on] that it is in order and that no foreign objects hinder the movi[ng] parts of the winch. Then, warm up the steam supply line and the winch engine, apply lubricant to all friction surfaces, check the starting gear by turning it to the "hoist" and "lower" positions, ch[eck] whether the speed-changing clutches engage and disengage proper[ly] and whether the braking gear is in order. After this, test the f[or]ward and reverse operation of the winch; if no knocking is hear[d] and reversal is rapid and smooth, the winch is ready for regu[lar] operation. The winch is started and stopped and the shaft speed changed in operation by opening and closing a stop valve or the starting valve. The winch is reversed either by shifting the li[nk] of the reversing gear or by operating the starting valve.

During winch operation it is necessary to see that lubricant [is] being properly fed, to check the temperature of the parts subject to friction and to listen for knocking. As soon as abnormal no[ise] is heard, stop the winch, find and eliminate the cause of the nois[e].

The load lowering speed should be regulated by applying the brake. Backsteam should be resorted to only when the load dro[ps] too fast even after applying the brake.

If hoisting operations are interrupted for short periods in winte[r], the winch should be run idle at low speed with open blow-off valve[s].

If the winch is not to be operated for a prolonged period in win[ter], it is necessary to drain the condensate from the cylinders and the live and exhaust steam lines.

Winch operation is prohibited if cracks are found in critic[al] parts, if the motive mechanism, steam distribution or braking gear is out of order or if the gears are excessively worn or some of the teeth are missing.

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.

4. Materials

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

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

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

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

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

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

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

5. Definitions

C_R = rudder force in [N]

Q_R = rudder torque in [Nm]

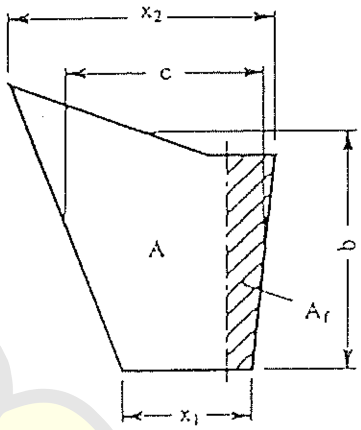
A = total movable area of the rudder in $[m^2]$
For nozzle rudders, A is not to be taken less than 1,35 times the projected area of the nozzle;

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

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

b = mean height of rudder area in [m]

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



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

Fig. 14.1

Λ = aspect ratio of rudder area A_t

Λ = b^2/A_t

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

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

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

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

B. Rudder Force and Torque

1. Rudder force and torque for normal rudders

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

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

v = v_0 for ahead condition

v = v_a for astern condition

κ_1 = coefficient, depending on the aspect ratio Λ

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

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

κ_3 = coefficient, depending on the location of the rudder

κ_3 = 0,8 for rudders outside the propeller jet

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

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

κ_t = coefficient depending on the thrust coefficient c_t

κ_t = 1,0 normally

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

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

Table 14.1

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

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

Q_R = $C_R \cdot r$ [Nm]

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

α = 0,33 for ahead condition

α = 0,66 for astern condition (general)

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

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

α = 0,25 for ahead condition

α = 0,55 for astern condition.

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

k_b = balance factor as follows:

k_b = A_f/A

k_b = 0,08 for unbalanced rudders

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

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

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

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

The resulting force of each part may be taken as:

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

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

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

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

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

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

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

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

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

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

C_1 = A_1/b_1

$C_2 \;=\; A_2/b_2$

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

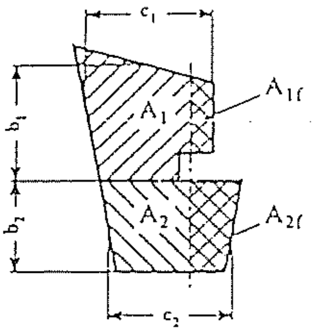


Fig. 14.2

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

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

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

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

for ahead condition

The greater value is to be taken.

C. Scantlings of the Rudder Stock

1. Rudder stock diameter

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

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

Q_R see B. 1.2 and B. 2.2 - 2.3

The related torsional stress is

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

k_r see A.4.2.

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

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

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

2. Strengthening of rudder stock

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

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

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

Bending stress:

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

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

Torsional stress:

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

D_1 = increased rudder stock diameter in [cm]

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

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

Q_R see B.1.2 and B.2.2 - 2.3

D_t see 1.1.

Guidance

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

Section 4

Main Shafting

A. General

1. Scope

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

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

2. Documents for approval

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

B. Materials

1. Approved materials

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

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

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

2. Testing of materials

All component parts of the shafting which assist in transmitting the torque from the ship's propulsion plant are [illegible] and must be tested [illegible] covers metal propeller [illegible] running in [illegible] penetration not by [illegible] 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 [illegible] with the requirements relating to torsional vibrations set out in Section 16. The dimensions of the shafting shall be based [illegible] Where the [illegible] be dimensioned [illegible] 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 [illegible] the shaft is [illegible]
		$1 - \left(\frac{d_i}{d_a}\right)^4 = 1.0$ 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

[Rpm] rated shaft speed

[-] 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

[-] material factor

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

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

[-] 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 [illegible] of the propeller shaft [illegible] method approved by the Society or if the propeller is [illegible] flange forged on the propeller shaft, the propeller shaft [illegible]

k =1,26 for propeller shafts [illegible] area specified [illegible] the propeller is [illegible] 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 [illegible].

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 [illegible] shaft [illegible] tube [illegible] 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 [illegible] at the surface of the shaft taper [illegible] 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 of 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 18

Equipment

A. General

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

Guidance

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

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

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

4. The equipment numeral formula for anchoring equipment required under this Section is based on an assumed current speed of 2,5 m/sec, wind speed of 25 m/sec and a scope of chain cable between 6 and 10, the scope being the ratio between length of chain paid out and water depth.

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

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

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

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

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

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

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

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

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

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

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

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

B. Equipment numeral

The equipment numeral is to be calculated as follows:

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

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

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

h = $f_b + \sum h'$

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

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

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

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

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

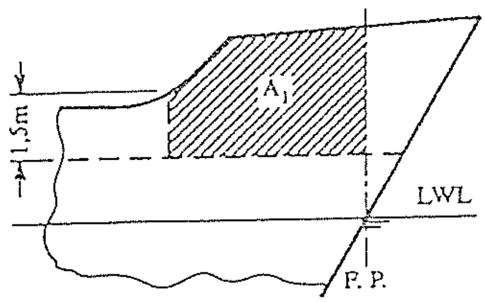


Fig. 18.1

C. Anchors

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

Guidance

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

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

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

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

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

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

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

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

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 d_1	Diameter d_2	Diameter d_3	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	87	76			280	1470	6	200	520
142	3210 - 3400	3	9900		660	100	90	78			280	1470	6	200	555
143	3400 - 3600	3	10500		660	102	92	78			280	1470	6	200	590
144	3600 - 3800	3	11100		687,5	105	95	81			300	1470	6	200	620
145	3800 - 4000	3	11700		687,5	107	97	84			300	1470	6	200	650
146	4000 - 4200	3	12300		687,5	111	100	87			300	1470	7	200	650
147	4200 - 4400	3	12900		715	114	102	87			300	1470	7	200	660
148	4400 - 4600	3	13500		715	117	105	90			300	1470	7	200	670
149	4600 - 4800	3	14100		715	120	107	92			300	1470	7	200	680
150	4800 - 5000	3	14700		742,5	122	111	95			300	1470	7	200	685
151	5000 - 5200	3	15400		742,5	124	111	97			300	1470	8	200	[illegible]
152	5200 - 5500	3	16100		742,5	127	114	97			300	1470	8	200	[illegible]
153	5500 - 5800	3	16900		742,5	130	117	100			300	1470	8	200	705
154	5800 - 6100	3	17800		742,5	132	120	102			300	1470	9	200	705
155	6100 - 6500	3	18800		742,5		124	107			300	1470	9	200	715
156	6500 - 6900	3	20000		770		127	111			300	1470	9	200	725
157	6900 - 7400	3	21500		770		132	114			300	1470	10	200	725
158	7400 - 7900	3	23000		770		137	117			300	1470	11	200	[illegible]
159	7900 - 8400	3	24500		770		142	122			300	1470	11	200	[illegible]
160	8400 - 8900	3	26000		770		147	127			300	1470	12	200	[illegible]
161	8900 - 9400	3	27500		770		152	132			300	1470	13	200	735
162	9400 - 10000	3	29000		770			132			300	1470	14	200	735
163	10000 - 10700	3	31000		770			137			300	1470	15	200	735
164	10700 - 11500	3	33000		770			142			300	1470	16	200	735
165	11500 - 12400	3	35500		770			147			300	1470	17	200	735
166	12400 - 13400	3	38500		770			152			300	1470	18	200	735
167	13400 - 14600	3	42000		770			157			300	1470	19	200	735
168	14600 - 16000	3	46000		770			162			300	1470	21	200	735

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

1 see C.1
2 see E.1.2

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

Appendix to Section 2

Part C :

Approximate Calculation of the Starting Air Supply

1. Starting air for installations with reversible engines

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

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

where

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

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

- for two-stroke engines: a = [illegible]714
- [illegible]

The following [illegible] to be used:

- [illegible]
- for four-stroke engines: b = 0,056

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

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

$$c = \frac{[illegible]}{1 - e^{(0{,}11 - 0{,}05 \cdot 1_e \cdot p_{c,perm})}}$$

where $p_{c,perm}$ ≠ 30 bar, if no pressure-reducing valve is fitted.

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

If a pressure-reducing valve is fitted, which reduces the pressure [illegible] to the starting air receiver [illegible] then the [illegible] is to be [illegible].

The following values of n_A are to be applied:

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

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

n_o [Rpm] = rated speed

2. Starting air for installations with non reversible engines

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

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

1.6 Pipe connections

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

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

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

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

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

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

2. Calculation of pipe diameters

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

2.2 Dry cargo and passenger ships

a) main bilge pipes

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

b) branch bilge pipes

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

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 [illegible]

H [m] depth [illegible] 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 tank[illegible] calculated using the for[illegible]

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

where:

l_1 [m] total length of spaces between [illegible] bulkhead and stern tube bulkhead

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

Branch bilge pipes are to be dimensioned in accordance with [illegible] spaces in the cargo area of tankers and bulk cargo/oil carriers see Section 15.

2.4 Minimum [illegible]

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 of gravity are not yet exactly known in the early project stage. If the model does not accomplish the req... ...ed the designer has to alter the hull. This al... ever, is possible in the early project stage o... the trial speed in ballast condition ... 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 co... margin ...)

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

$$W_{fuel\ oil}\ [t] = P_{Bmc} \cdot b_{mc} + P_{ac} \cdot b_{ac} \cdot \frac{s}{V_{serv}} \cdot 10^{...} \cdot [1.3 \ldots 1.5]$$

last brackets for reserve:

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

Additions to the volume

2%   for double bottom tanks

1 ... 2% for top tanks and deep tanks

2%   for thermal expansion, i.e. 98% filled only.

Diesel oil

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

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

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

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

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!).

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

e) Ballast capacity used for

- trim (immersion of propeller; ...)
- 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 ... are larger at the ends of the ship.

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

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

Sounding/ullage tables delivered by yard.

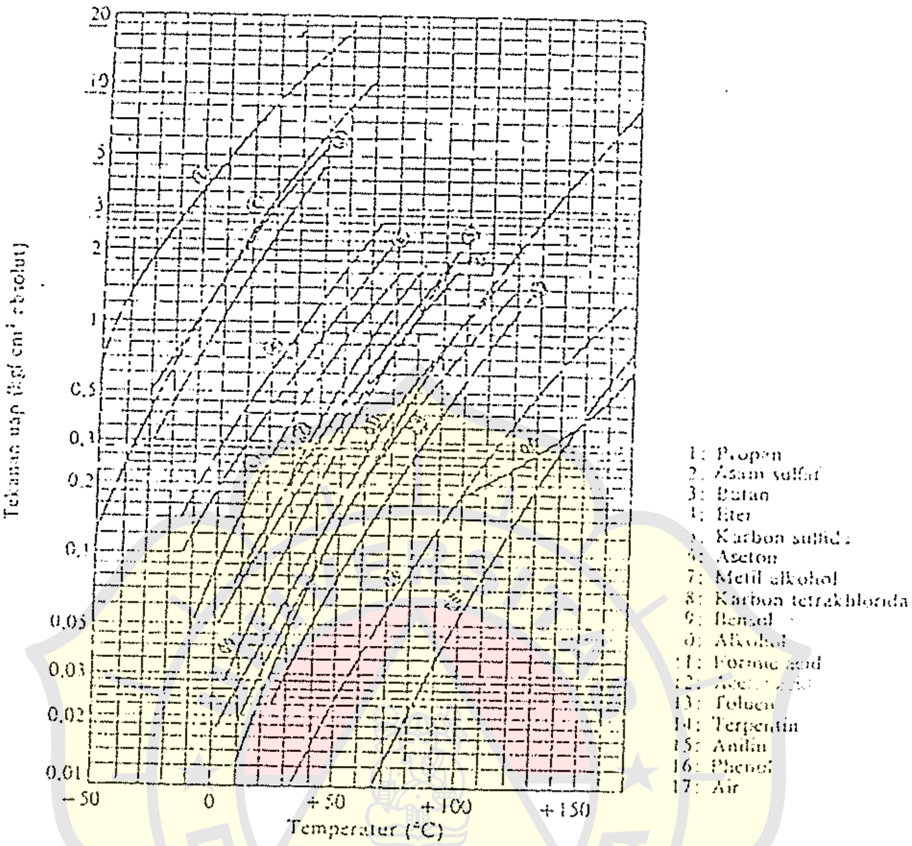
f) Provisions/persons/luggage

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 ...)

Type and Location of Main Engine

is another part of the contract influencing ship design. (Ship weight, volume, fuel consumption).

... is determined by the choice of the main engine type, als



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

Gb. 2.1 Sifat-sifat fisik berbagai zat cair.

2.4 Head

2.4.1 Head Total Pompa

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

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

di mana H: Head total pompa (m)

h_a: Head statis total (m)
Head ini adalah perbedaan tinggi antara muka air di sisi keluar dan 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),

$$\Delta h_p = h_{p2} - h_{p1}.$$

h_l: Berbagai kerugian head di pipa, katup, belokan, sambungan, dll (m), di mana h_p: Head tekanan (m)

p: Tekanan (kgf/cm^2)

γ: Berat per satuan volume zat cair yang dipompa (kgf/l)

Apabila tekanan diberikan dalam kPa, dapat dipakai rumus berikut:

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

di mana p': Tekanan (Pa)

ρ: Rapat masa (kg/l)

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

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

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

2.4.2 Head Kerugian

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

(1) Head kerugian gesek dalam pipa

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

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

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

di mana v: Kecepatan rata-rata aliran di dalam pipa (m/s)

C, p, q: Koefisien-koefisien

R: Jari-jari hidrolik (m)

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

S: Gradien hidrolik

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

h_f: Head kerugian gesek dalam pipa (m)

λ: Koefisien kerugian gesek

g: Percepatan gravitasi (9,8 m/s^2)

L: Panjang pipa (m)

D: Diameter dalam pipa (m)

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

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

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

Pada $Re < 2300$, aliran bersifat laminer.
Pada $Re > 4000$, aliran bersifat turbulen.
Pada $Re \doteq 2300 - 4000$ terdapat daerah transisi, di mana aliran dapat bersifat laminer atau

(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.

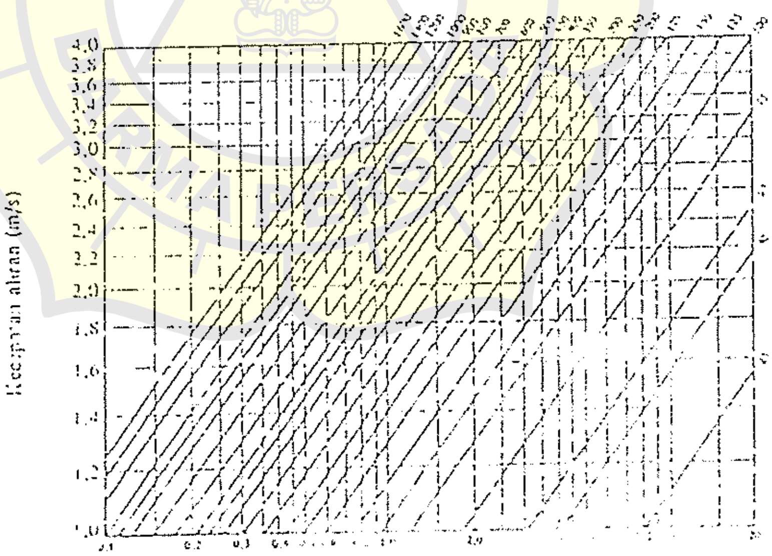
1) *Formula Darcy*

Dengan cara Darcy, koefisien kerugian gesek

rumus

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

di mana D adalah diameter dalam pipa 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).

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

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

P_s: Tekanan isap tingkat pertama (kgf/m^2 abs)

P_d: Tekanan keluar dari tingkat terakhir (kgf/m^2 abs)

Q_s: Jumlah volume gas yang keluar dari tingkat terakhir (m^3/min) dinyatakan pada kondisi tekan dan temperatur isap

k: c_p/c_v

m: Jumlah tingkat kompresi; lihat keterangan pada Pers. (2.16).

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

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

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

Sebagai contoh, dari Tabel 2.7 terbaca bahwa untuk kompresi 1 tingkat sampai 7 kgf/cm^2 (g) atau 8,033 kgf/cm^2 abs, diperlukan daya sebesar 4,7074 kW. Ini diperoleh dari Pers. (2.21) dengan mengambil harga $k = 1{,}4$ dan $m = 1$. Daya sebesar 4,7074 kW tersebut juga akan diperlukan untuk kompresi 2 tingkat tanpa pendingin antara. Namun jika digunakan pendingin antara maka daya yang diperlukan menjadi sebesar 4,022 kW. Harga ini dapat diperoleh dari Pers. (2.21a) jika diambil $k = 1{,}4$ dan $m = 2$.

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

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

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 [illegible] 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

Power, Speed and SFOC

L42MC

Stroke: 1 360 mm Bore: 420 mm

Speed	r/min	[illegible]	176	132	132
mep	bar	[illegible]	11.5	18.0	11.5
Cylinder	Power				
4	kW	3 980	2 540	2 980	1 920
	BHP	[illegible]	3 480	4 000	2 600
5	kW	4 975	3 475	3 725	2 400
	BHP	[illegible]	4 345	5 075	3 250
6	kW	5 970	3 810	4 470	2 880
	BHP	[illegible]	5 190	6 090	3 900
7	kW	6 965	4 445	5 215	3 360
	BHP	[illegible]	6 055	7 105	4 550
8	kW	7 960	5 080	5 960	3 840
	BHP	[illegible]	6 920	7 105	5 200
9	kW	8 955	5 715	6 705	4 320
	BHP	[illegible]	7 805	9 135	5 850
10	kW	9 950	6 350	7 450	4 800
	BHP	[illegible]	8 650	10 150	6 500
11	kW	10 945	6 985	8 195	5 280
	BHP	[illegible]	9 535	11 165	7 150
12	kW	11 940	7 620	8 940	5 760
	BHP	[illegible]	10 390	12 180	7 800

	Specific Fuel Oil Consumption (SFOC)			
g/kWh	177	165	177	165
g/BHPh	[illegible]	121	130	121
Lubricating oil consumption	approximately 3 kg/cyl. 24h			
Cylinder oil consumption	[illegible] 1.4 g/kWh - 0.65-1.0 g/BHPh			

Power, Speed and SFOC

S35MC

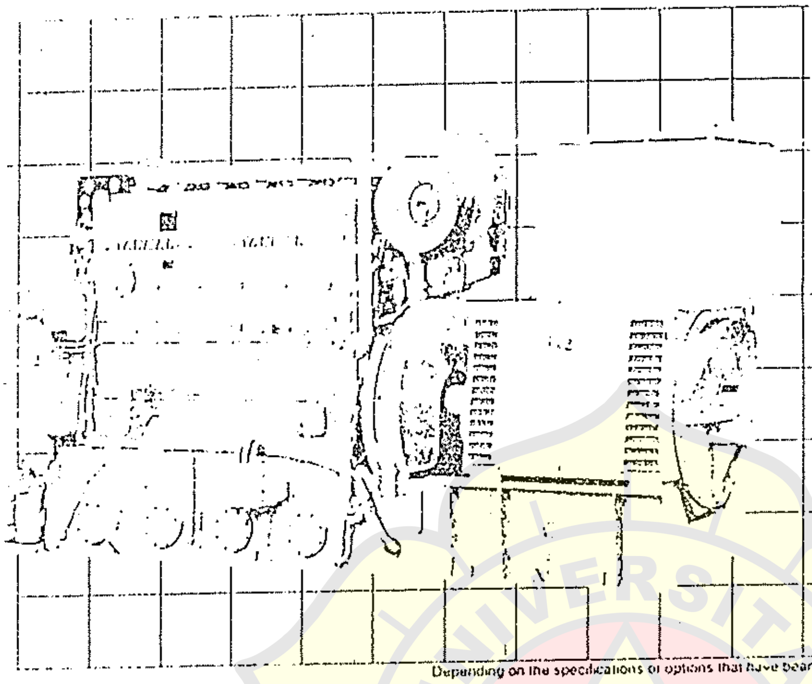
Stroke: 1 400 mm Bore: 350 mm

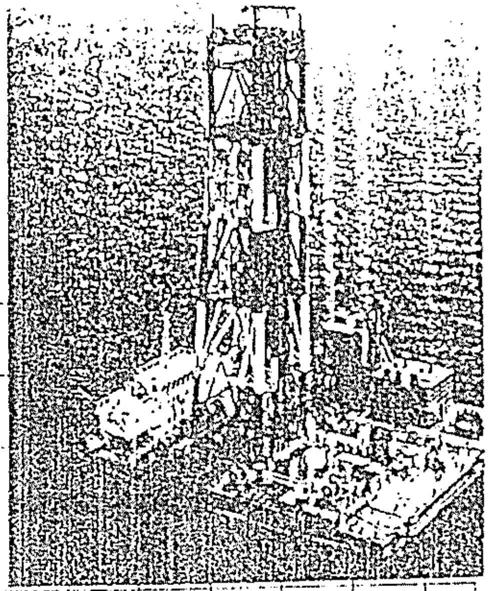
Speed	r/min	170	170	145	145
mep	bar	[illegible]	14.7	18.4	14.7
Cylinder	Power				
4	kW	2 800	2 240	2 300	1 900
	BHP	[illegible]	3 040	3 220	2 580
5	kW	3 500	2 800	2 975	2 375
	BHP	4 750	3 800	4 025	3 225
6	kW	4 200	3 360	3 570	2 850
	BHP	[illegible]	4 560	4 830	3 870
7	kW	4 900	3 920	4 165	3 325
	BHP	[illegible]	5 320	5 635	4 515
8	kW	5 600	4 480	4 760	3 800
	BHP	7 600	6 080	6 440	5 160
9	kW	6 300	5 040	5 335	4 275
	BHP	8 550	6 840	7 245	5 805
10	kW	7 000	5 600	5 950	4 750
	BHP	[illegible]	7 600	8 050	6 450
11	kW	7 700	6 160	6 545	5 225
	BHP	[illegible]	8 360	8 855	7 095
12	kW	8 400	6 720	7 140	5 700
	BHP	11 400	9 120	9 660	7 740

	Specific Fuel Oil Consumption (SFOC)			
g/kWh	175	170	175	170
g/BHPh	[illegible]	125	129	125
Lubricating oil consumption	approximately 2 kg/cyl. 24h			
Cylinder oil consumption	1.1-1.6 g/kWh - 0.8-1.2 g/BHPh			

S185L

Engine output
309 485 kW (420 660 PS)





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

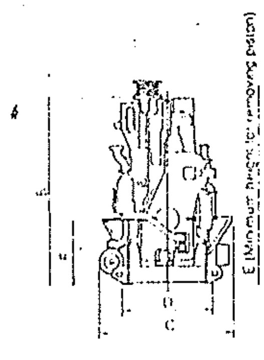
Specifications

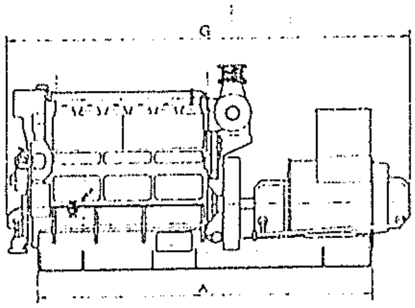
Engine model		S185DL-UT		S185DL-ST		S185DL-ET		S185L-UT		S185L-ST		S185L-ET		S185AL-UT	S185AL-ST
Type		Vertical water-cooled 4-cycle diesel engine													
No. of cylinders		6													
Cylinder bore × stroke	mm	185 × 230													
Total displacement	ℓ	37.09													
Continuous rated output	kW (PS)	309 (420)		353 (480)		397 (540)		397 (540)		441 (600)		485 (660)		441 (600)	485 (660)
Engine speed	rpm	720	750	720	750	720	750	900	1000	900	1000	900	1000	1200	1200
Net mean effective pressure	MPa (kgf/cm²)	1.388 (14.15)	1.333 (13.59)	1.586 (16.17)	1.523 (15.53)	1.785 (18.20)	1.713 (17.47)	1.428 (14.56)	1.294 (13.10)	1.587 (16.18)	1.428 (14.56)	1.745 (17.79)	1.570 (16.01)	1.190 (12.13)	1.309 (13.35)
Generator capacity	KW	280		320		360		360		400		450		400	450
Combustion system		Direct injection													
Starting system		Compressed air													
External dimensions: Overall length	mm	2687													
External dimensions: Overall width	mm	1134													
External dimensions: Overall height	mm	1749													
Dry weight	kg	5400						5000						5000	

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

Dimensions (Units: mm)

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





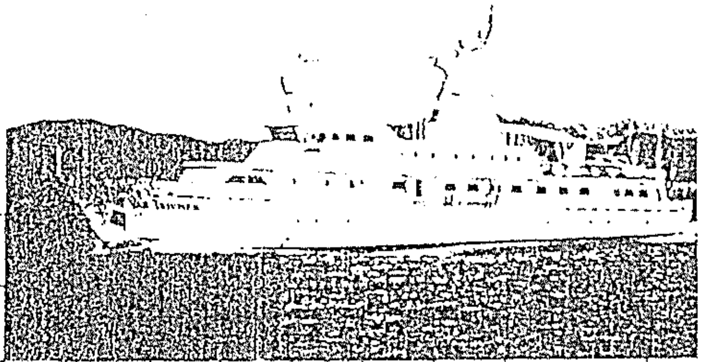
Engine model	S185DL-UT S185DL-ST	S185DL-ET	S185L-UT S185L-ST	S185L-ET	S185AL-UT S185AL-ST
A	3650	3800	3590	3590	3590
B	2124	2179	2124	2179	2124
C	1455	1455	1455	1455	1455
D	950	980	960	980	930
E	2245	2245	2245	2245	2245
F	800	800	800	800	800
G	4291	4400	4231	4231	4240
Dry weight of generating equipment (kg)	9400	9400	9200	9200	9100

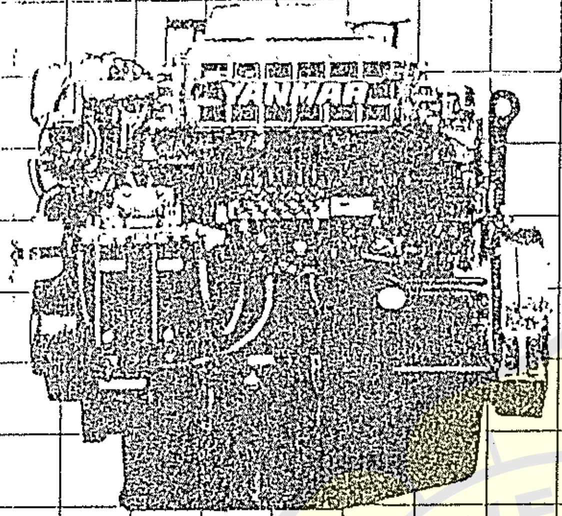
Note: Above data shows the case of common bed and built-in L.O. sump tank.

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

6KHL

Engine output
184 309 kW (250 420 PS)





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

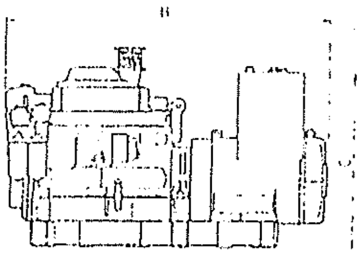
Specifications

Engine model			6KHL-STN					
Type			Vertical water-cooled 4-cycle diesel engine					
No. of cylinders			6					
Cylinder bore × stroke		mm	135 × 160					
Total displacement		ℓ	13.74					
Continuous rated output		kW (PS)	184 (250)	235 (320)	279 (380)	199 (270)	265 (360)	309 (420)
Engine speed		rpm	1200	1500	1800	1200	1500	1800
Net mean effective pressure		MPa (kgf/cm²)	1.339 (13.65)	1.370 (13.97)	1.356 (13.83)	1.446 (14.74)	1.542 (15.72)	1.498 (15.28)
Generator capacity		kW	160	200	240	180	240	280
Combustion system			Direct injection					
Starting system			Electric starting (Air-motor starting is available.)					
External dimensions	Overall length	mm	1626					
	Overall width	mm	991					
	Overall height	mm	1434					
Dry weight		kg	1320					

Dimensions

(Units: mm)

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



Engine model	6KHL-STN
A	1028
B	2524
C	1401
D	1454
E	1394
Dry weight of generating equipment (kg)	2500

Additional specifications, etc. on the separate delivery specifications sheet.

H = Total pump head in m. N = Power consumption in hp at duty point.

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
[illegible]	0	0	¾"	¾"					4,7	[illegible]	7,3	0,12	9,9	0,18	14,4	0,32
	1,5	25	—	—					4,4	0,08	7,0	0,14	10,7	0,21	14,2	0,38
Max.	3,0	50	—	—					3,4	0,19	6,1	0,18	8,9	0,27	13,5	0,44
6000 r.p.m.	4,5	75	—	—							4,2	0,21	7,1	0,30	11,7	0,50
[illegible]	0	0	1"	1"	5,0	0,14	7,3	0,23	9,5	0,36	14,6	0,72	18,4	0,9	27	1,7
	2	33	—	—	4,0	0,18	6,9	0,29	9,0	[illegible]	13,7	0,73	17,0	1,0	26	1,8
Max.	4	66	—	—	3,0	[illegible]	5,5	0,31	7,7	0,45	11,5	0,79	14,6	1,1	23	1,9
5000 r.p.m.	6	100	—	—			3,0	0,33	5,4	[illegible]	8,0	0,80	10,7	1,1	18	2,0
[illegible]	0	0	1½"	1½"	6,35	0,56	9,1	0,26	11,7	0,4	18,6	0,9	25	1,4	36	2,2
	5	85	—	—	5,0	[illegible]	7,7	[illegible]	10,4	[illegible]	17,0	1,0	23	1,5	34	2,5
Max.	10	166	—	—	2,7	[illegible]	5,3	0,43	7,9	[illegible]	14,4	1,1	20	1,7	31	2,9
4500 r.p.m.	15	250	—	—					4,3	[illegible]	10,3	1,3	16	1,9	27	3,3
[illegible]	0	0	2"	2"					18,0	[illegible]	23	1,4	38	2,7	55	4,6
	6	100	—	—					16,5	1,1	26	2,1	35	3,2	52	5,3
Max.	12	200	—	—					12,5	1,4	22	[illegible]	32	3,6	48	6,5
3500 r.p.m.	18	300	—	—							14	2,7	25	4,3	42	7,5
SA-[illegible]	0	0	2"	2"	11,5	[illegible]	15,4	[illegible]	21	[illegible]	33	[illegible]				
	7,5	125	—	—	9,5	[illegible]	14,3	1,5	19	[illegible]	30	4,0				
Max.	15,0	250	—	—	6,6	[illegible]	11,2	[illegible]	16	[illegible]	27	4,5				
2500 r.p.m.	22,5	375	—	—			5,3	[illegible]	11	[illegible]	22	5,0				
[illegible]	0	0	3"	2½"					39	[illegible]	61	10	82	16	100	22
	20	333	—	—					35	[illegible]	57	13	76	21	94	28
Max.	40	666	—	—					31	[illegible]	51	[illegible]	70	25	87	35
3500 r.p.m.	60	1000	—	—							39	[illegible]	63	28	79	40
[illegible]	0	0	3"	2½"	23	[illegible]	33	[illegible]	42	[illegible]						
	20	333	—	—	21	3,7	32	9,2	40	[illegible]						
Max.	40	666	—	—	19	4,6	28	[illegible]	37	11,0						
2200 r.p.m.	60	1000	—	—	15	[illegible]	25	[illegible]	33	[illegible]						
[illegible]	0	0	3"	3"					14,0	[illegible]	22	[illegible]	30	4,0	44	7,0
	20	333	—	—					13,0	[illegible]	21	[illegible]	29	5,6	42	9,4
Max.	40	666	—	—					9,5	[illegible]	16	4,6	26	7,2	39	11,4
4000 r.p.m.	60	1000	—	—							9	5,3	18	8,2	33	13,0
SA-50-220/17	0	0	3"	3"	16,3	[illegible]	23	[illegible]	31	[illegible]	43	[illegible]	65	12	95	21
	20	333	—	—	14,5	[illegible]	21	[illegible]	28	[illegible]	47	[illegible]	63	14	93	25
Max.	40	666	—	—	10,5	[illegible]	18	[illegible]	24	[illegible]	43	[illegible]	60	17	90	28
3500 r.p.m.	60	1000	4"	—	3,7	[illegible]	11	[illegible]	18	[illegible]	38	[illegible]	55	19	84	31
SA-100-235/23	0	0	4"	4"	15	[illegible]	23	[illegible]	30	[illegible]	47	[illegible]				
	40	666	—	—	15	4,5	22	7,7	29	11,3	46	21				
Max.	80	1333	—	—	12	[illegible]	19	[illegible]	27	[illegible]	44	[illegible]				
3000 r.p.m.	120	2000	—	—	7	[illegible]	14	11,4	22	15,0	39	28				
SA-150-200/33	0	0	6"	6"	17	7,8	26	12	33	19						
	80	1333	—	—	16	11,0	24	[illegible]	32	[illegible]						
Max.	160	2666	—	—	14	14,0	22	23	29	[illegible]						
2500 r.p.m.	240	4000	—	—	9	16,0	17	27	25	[illegible]						
[illegible]	0	0	8"	8"	28	[illegible]	40	[illegible]	53	67	Motor HK = N + 20 - 10 %					
	150	2500	—	—	27	[illegible]	39	[illegible]	51	[illegible]						
Max.	300	5000	—	—	24	[illegible]	36	[illegible]	48	[illegible]						
2000 r.p.m.	450	7500	[illegible]	—	19	[illegible]	32	[illegible]	44	111						

Pipe line dimension

m³/h	Suction mm	Suction "	Delivery mm	Delivery "
2	20	¾"	20	¾"
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"
30	80	3"	65	2½"
57	100	4"	80	3"
84	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.
1 m³/h = 3.67 imp. g.p.m.

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.