BAB VI
KESIMPULAN

6. 1. Kesimpulan

Dengan selesainya perencanaan dan perhitungan tugas merancang ini, maka penulis dapat mengambil kesimpulan yang berhubungan dengan kapal perencanaan kapal Tanker **6. 500 DWT** sebagai berikut :

- **Panjang kapal seluruhnya** (Loa) = 105 m
- **Panjang kapal antara garis tegak** (Lpp) = 99 m
- **Lebar kapal** (B) = 18,8 m
- **Tinggi kapal** (H) = 9,5 m
- **Sarat air kapal** (T) = 6,0 m
- **Koeffisien blok kapal** (Cb) = 0,747
- **Koeffisien prismatik kapal** (Cp) = 0,76
- **Koeffisien garis air kapal** (Cw) = 0,823
- **Koeffisien tengah kapal** (Cm) = 0,986
- **Displacement kapal** (Δ) = 8.550,446 ton
- **Volume kapal** (∇) = 8.509,41 m^3
- **Jumlah anak buah kapal** (ABK) = 26 orang
- **Mesin induk yang digunakan :**

Jumlah mesin	: 1 (satu) buah
Merk mesin	: MAN B&W
Type	: S26MC
Daya	: 2.180 HP/ 1.600 kW
Putaran mesin	: 250 rpm
Bore x Stroke	: 260 mm x 980 mm
Jumlah silinder	: 4

Cycle	**: 2 langkah**
SFOC	**: 132 g/ BHP. h (179 g/ kW.h)**
SLOC	**: 1,5 kg/ cyl. 24 h**
Kecepatan dinas	**: 11 knot.**

❋ **Mesin bantu yang digunakan :**

Jumlah mesin	**: 3 (tiga) buah**
Merk mesin	**: YANMAR**
Type	**: 6 HAL 2 – DTN Vertical 4 cycle**
Daya	**: 180 kW**
Putaran mesin	**: 1.200 rpm**
Bore x Stroke	**: 130 mm x 165 mm**
Jumlah silinder	**: 6**
Power factor	**: 0,8**
Cycle	**: 4 langkah**

Berdasarkan perhitungan dan hasil pengamatan untuk kapal yang hampir sama kesimpulan sebagai berikut :

1. Kapal memerlukan tenaga penggerak minimum 2.180 HP. Pada perencanaan ini dipilih motor induk dengan daya sebesar 2.180 HP pada putaran 250 rpm.
2. Dengan jumlah ABK 26 orang dan route pelayaran yang ditempuh lebih kurang 9.500 Nautical mile, kapasitas maksimum kebutuhan listrik untuk mensuplai peralatan yang ada sebesar 316,197 kW. Dalam perencanaan ini digunakan 3 unit generator yang masing-masing sebesar 180 kW.

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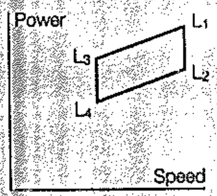
8. Tahara H. Sularso, *Pompa dan Kompressor*, cetakan ketiga, Jakarta, PT. Pranacya Paramita, cetakan ke-6,1996.

Power, Speed and SFOC



L35MC

Stroke: 1 050 Bore: 350

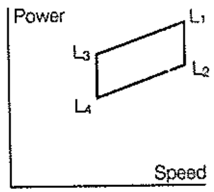
		L_1	L_2	L_3	L_4
Speed	r/min	210	210	178	178
mep	bar	18.4	14.7	18.4	14.7
Cylinder	**Power**				
4	kW	**2 600**	**2 080**	**2 200**	**1 760**
	BHP	3 520	2 820	3 000	2 400
5	kW	**3 250**	**2 600**	**2 750**	**2 200**
	BHP	4 400	3 525	3 750	3 000
6	kW	**3 900**	**3 120**	**3 300**	**2 640**
	BHP	5 280	4 230	4 500	3 600
7	kW	**4 550**	**3 640**	**3 850**	**3 080**
	BHP	6 160	4 935	5 250	4 200
8	kW	**5 200**	**4 160**	**4 400**	**3 520**
	BHP	7 040	5 640	6 000	4 800
9	kW	**5 850**	**4 680**	**4 950**	**3 960**
	BHP	7 920	6 345	6 750	5 400
10	kW	**6 500**	**5 200**	**5 500**	**4 400**
	BHP	8 800	7 050	7 500	6 000
11	kW	**7 150**	**5 720**	**6 050**	**4 840**
	BHP	9 680	7 755	8 250	6 600
12	kW	**7 800**	**6 240**	**6 600**	**5 280**
	BHP	10 560	8 460	9 000	7 200

Specific Fuel Oil Consumption (SFOC)

Conventional turbocharger				
g/kWh	177	171	177	171
g/BHPh	130	126	130	126

Lubricating oil consumption:	approximately 2 kg/cyl. 24h
Cylinder oil consumption:	1.0-1.4 g/kWh ≈ 0.7-1.0 g/BHPh

Power, Speed and SFOC



S26MC

Stroke: 980 Bore: 260

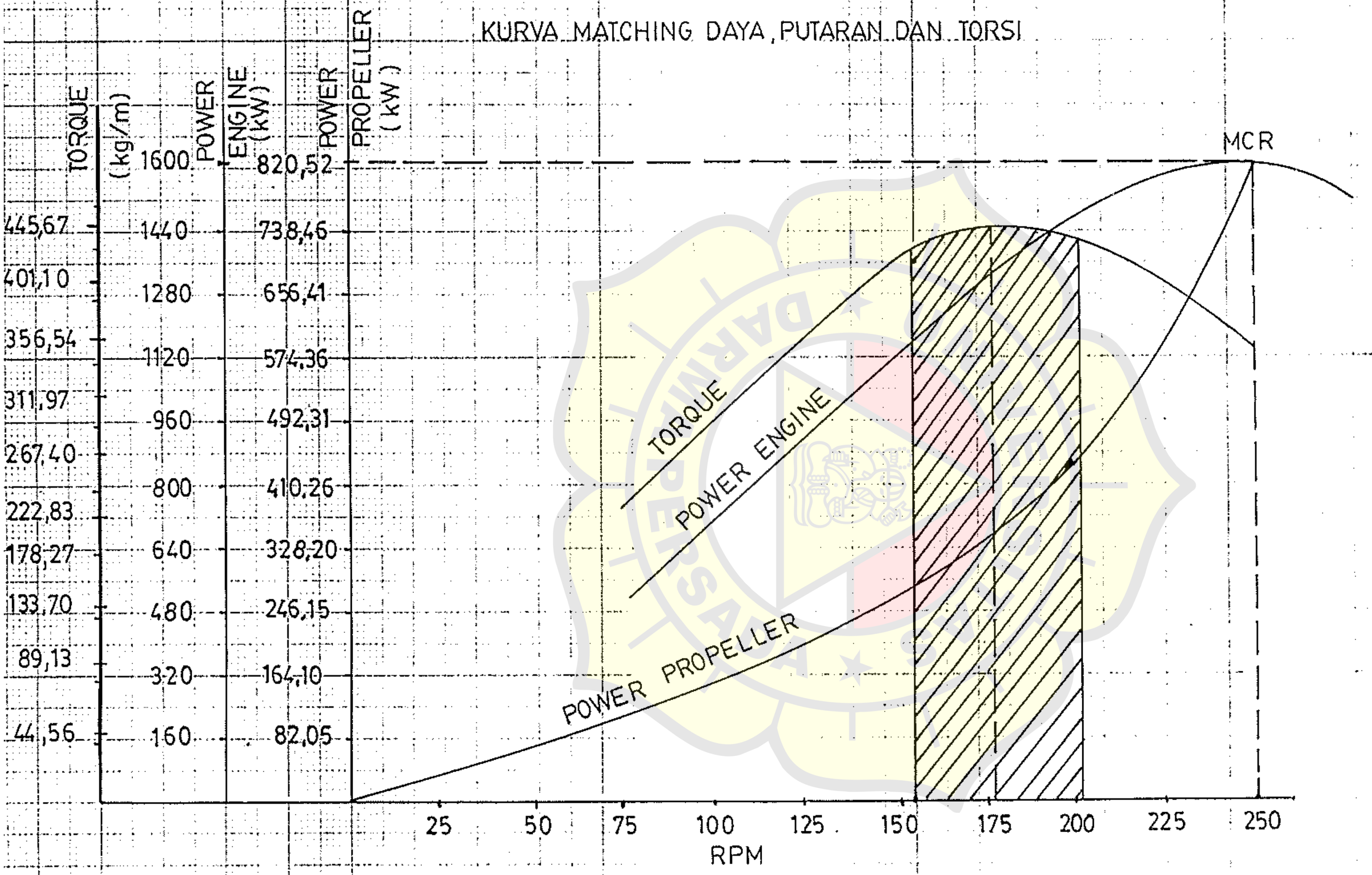
		L_1	L_2	L_3	L_4
Speed	r/min	250	250	212	212
mep	bar	18.4	14.8	18.4	14.8
Cylinder	**Power**				
4	kW	**1 600**	**1 280**	**1 360**	**1 100**
	BHP	2 180	1 740	1 860	1 480
5	kW	**2 000**	**1 600**	**1 700**	**1 375**
	BHP	2 725	2 175	2 325	1 850
6	kW	**2 400**	**1 920**	**2 040**	**1 650**
	BHP	3 270	2 610	2 790	2 220
7	kW	**2 800**	**2 240**	**2 380**	**1 925**
	BHP	3 815	3 045	3 255	2 590
8	kW	**3 200**	**2 560**	**2 720**	**2 200**
	BHP	4 360	3 480	3 720	2 960
9	kW	**3 600**	**2 880**	**3 060**	**2 475**
	BHP	4 905	3 915	4 185	3 330
10	kW	**4 000**	**3 200**	**3 400**	**2 750**
	BHP	5 450	4 350	4 650	3 700
11	kW	**4 400**	**3 520**	**3 740**	**3 025**
	BHP	5 995	4 785	5 115	4 070
12	kW	**4 800**	**3 840**	**4 080**	**3 300**
	BHP	6 540	5 220	5 580	4 440

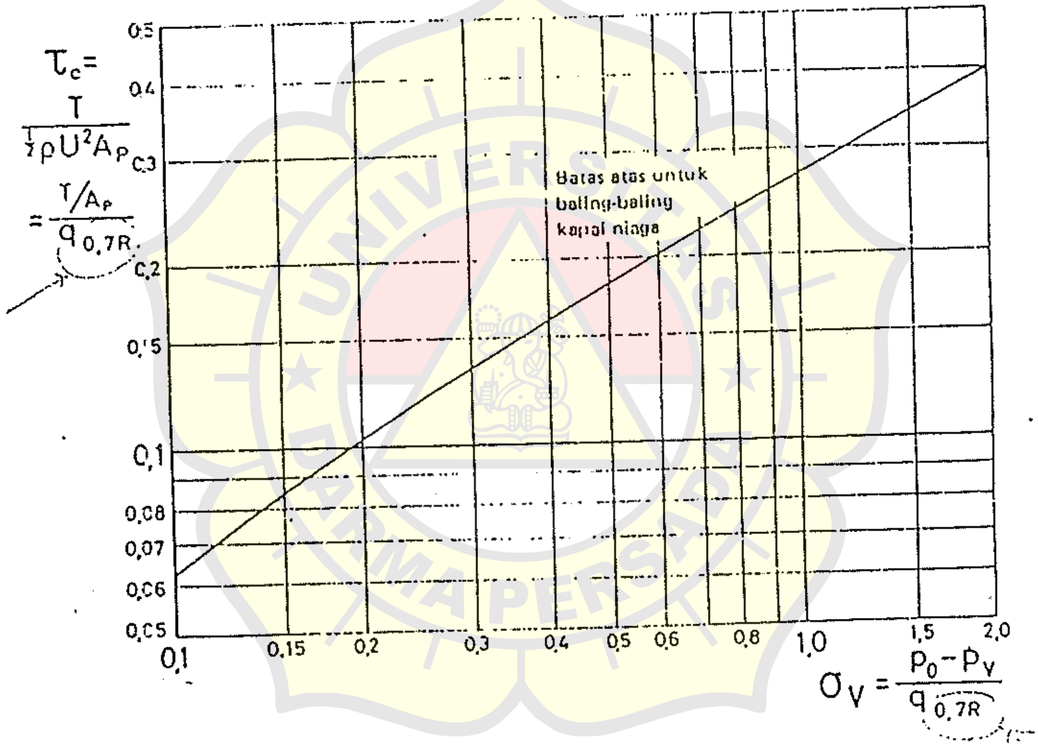
Specific Fuel Oil Consumption (SFOC)

Conventional turbocharger				
g/kWh	179	174	179	174
g/BHPh	132	128	132	128

Lubricating oil consumption:	approximately 1.5 kg/cyl. 24h
Cylinder oil consumption:	1.0-1.4 g/kWh ≈ 0.7-1.0 g/BHPh

KURVA MATCHING DAYA, PUTARAN DAN TORSI





Gbr.No. 2 Diagram Burril

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

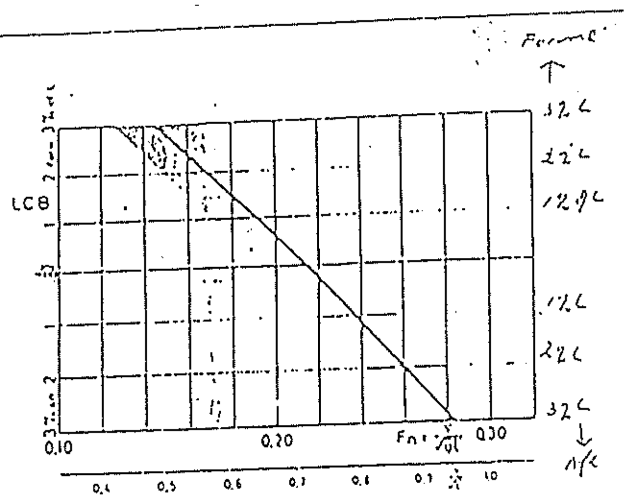
Gbr No.6 Koreksi Bentuk Anggota Badan

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

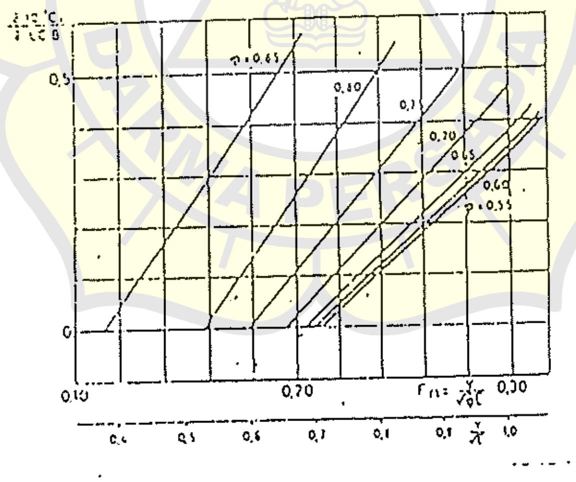
Gbr.No. 6 Koreksi Tahanan Tambahan

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

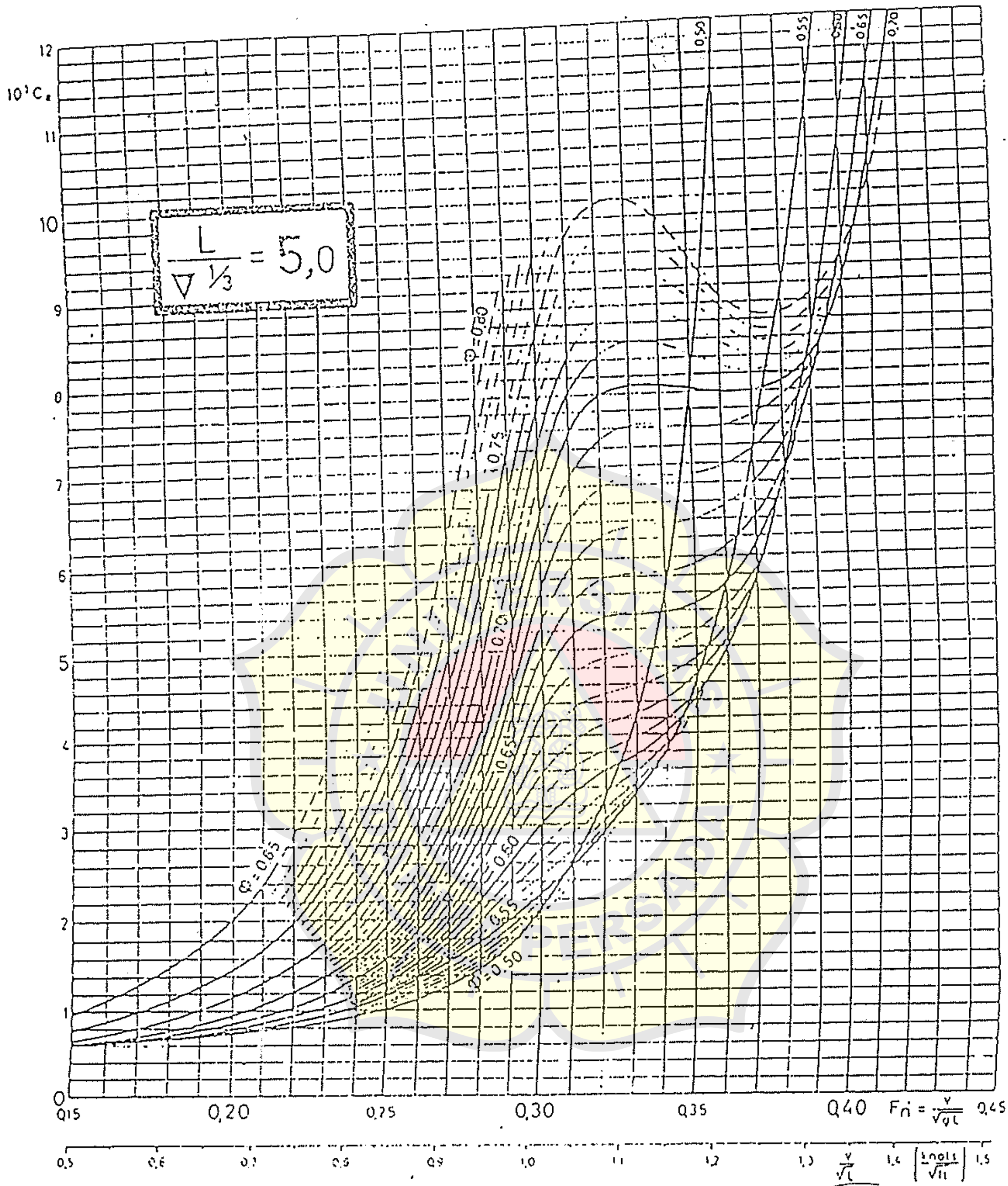
Gbr.No.4 Koreksi Bentuk Haluan



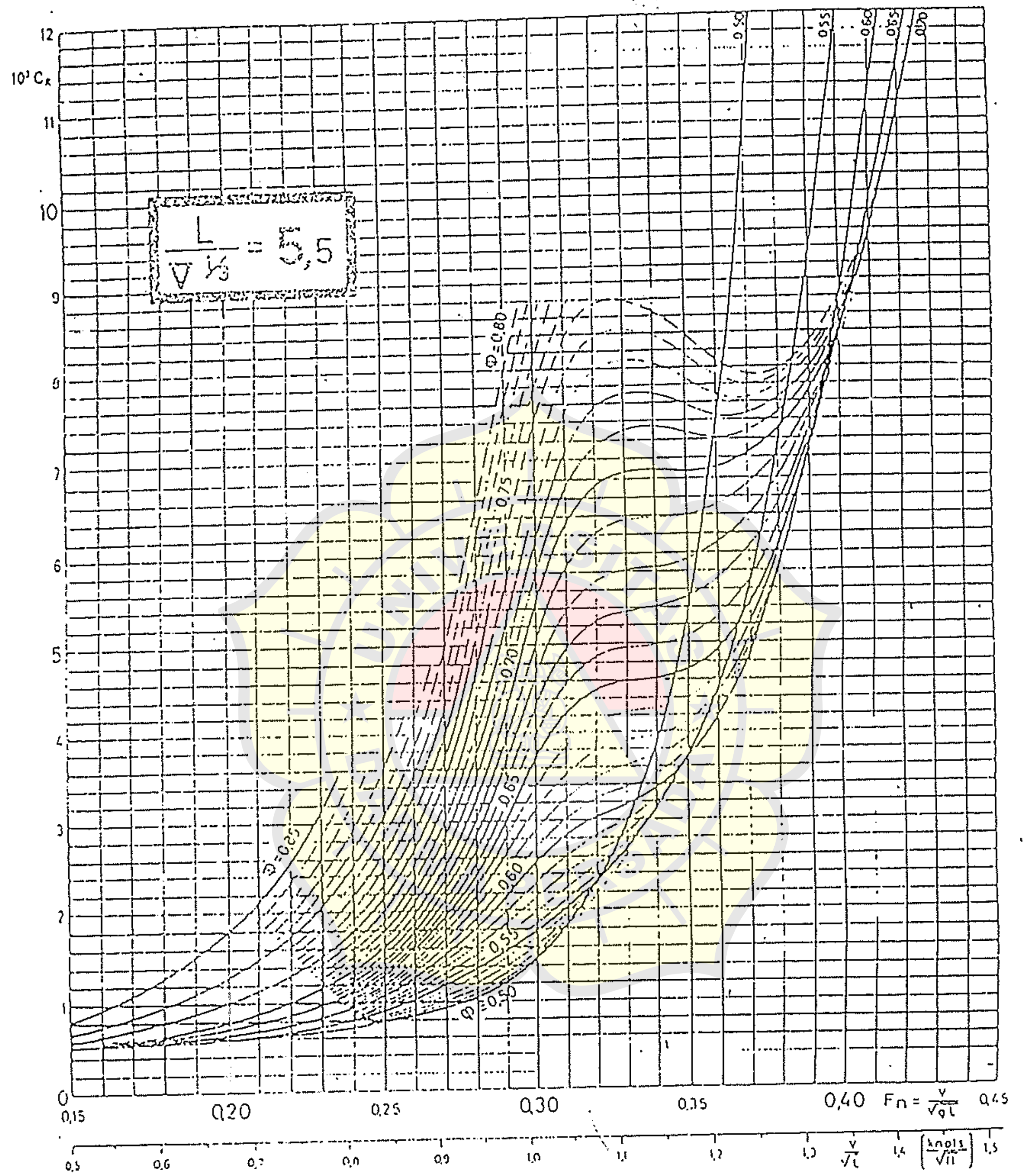
Gbr.No.3 LCB Standar



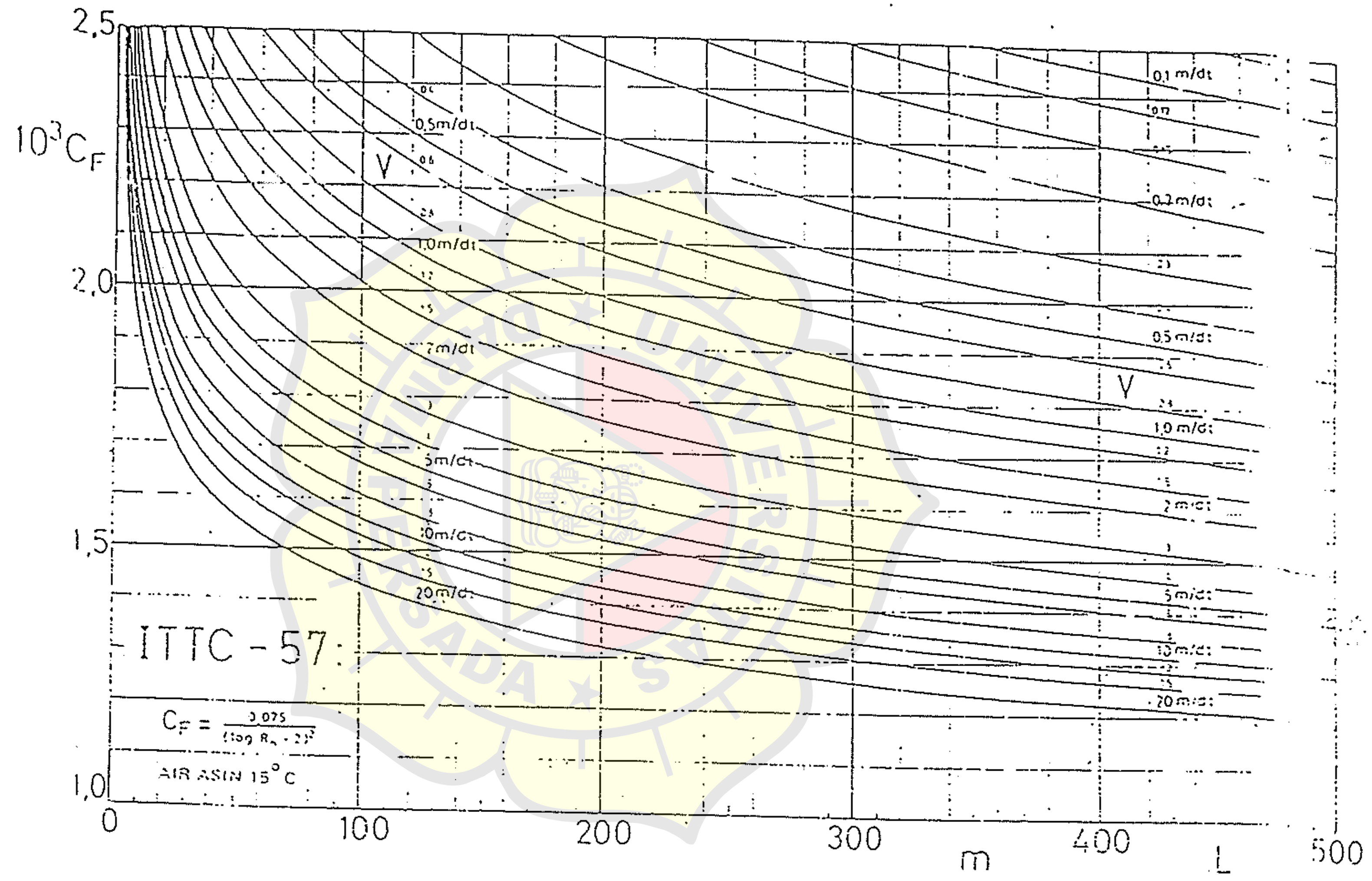
Gbr.No.3 Koreksi LCB



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



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



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

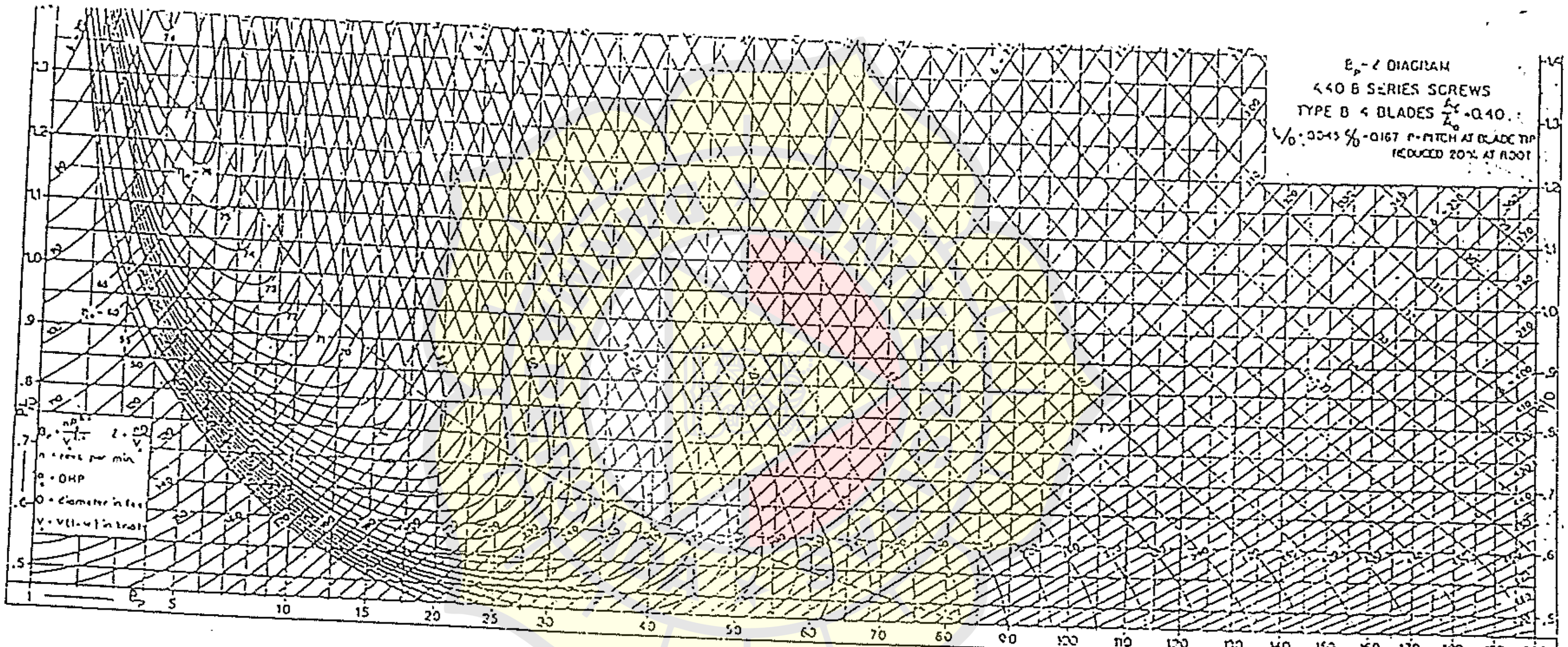
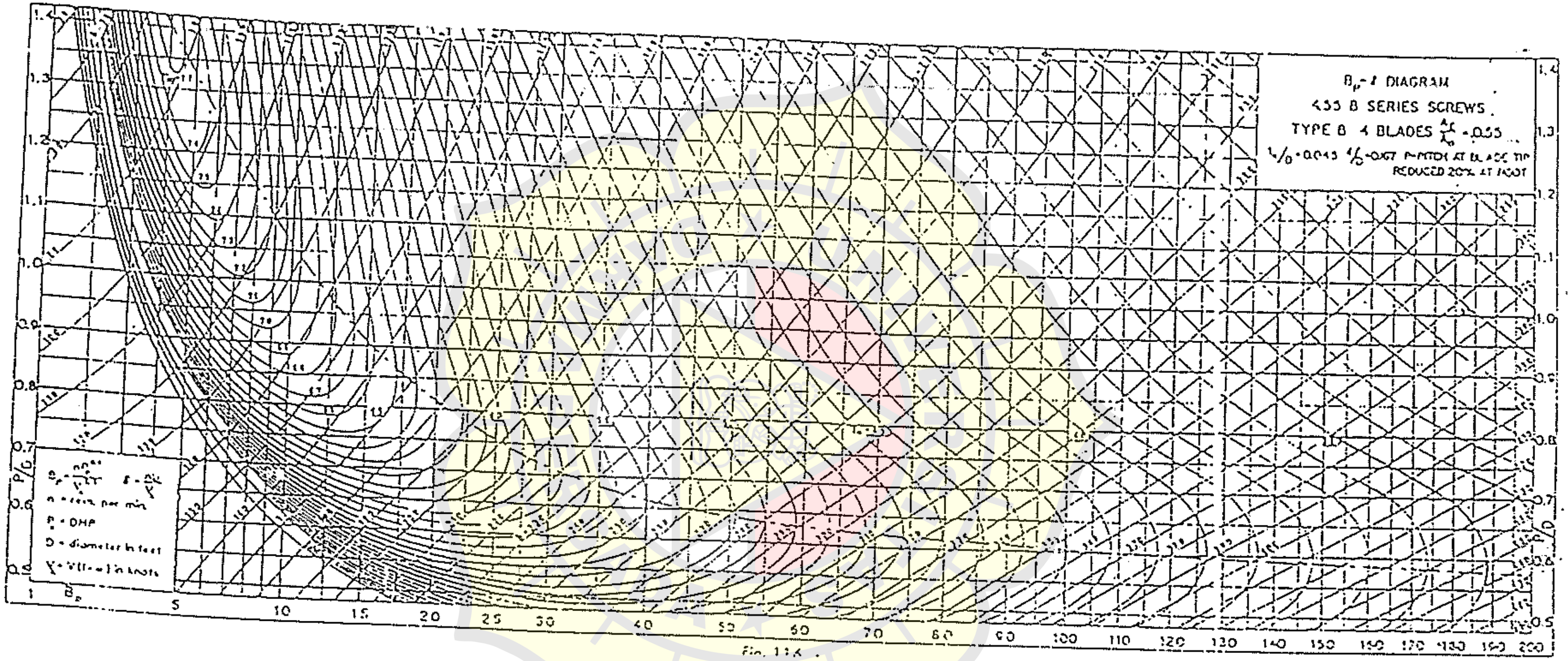


Fig. 115

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



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

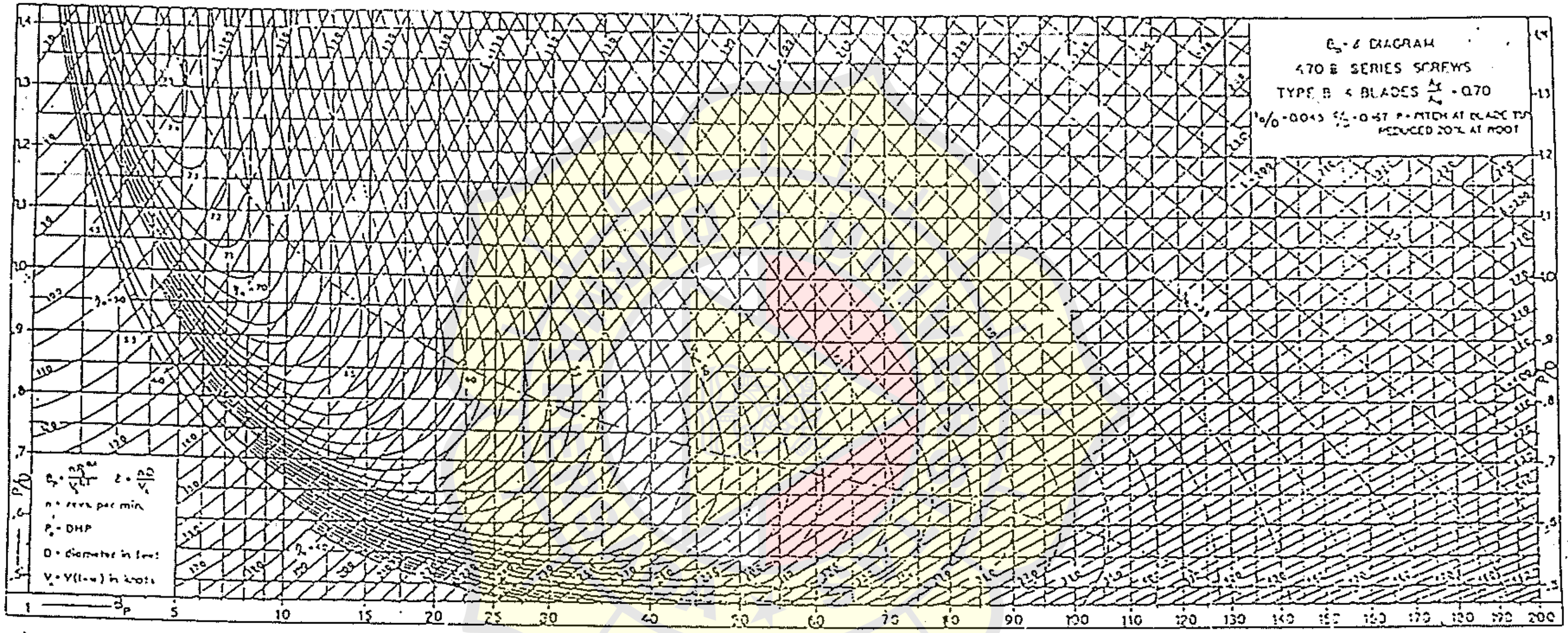
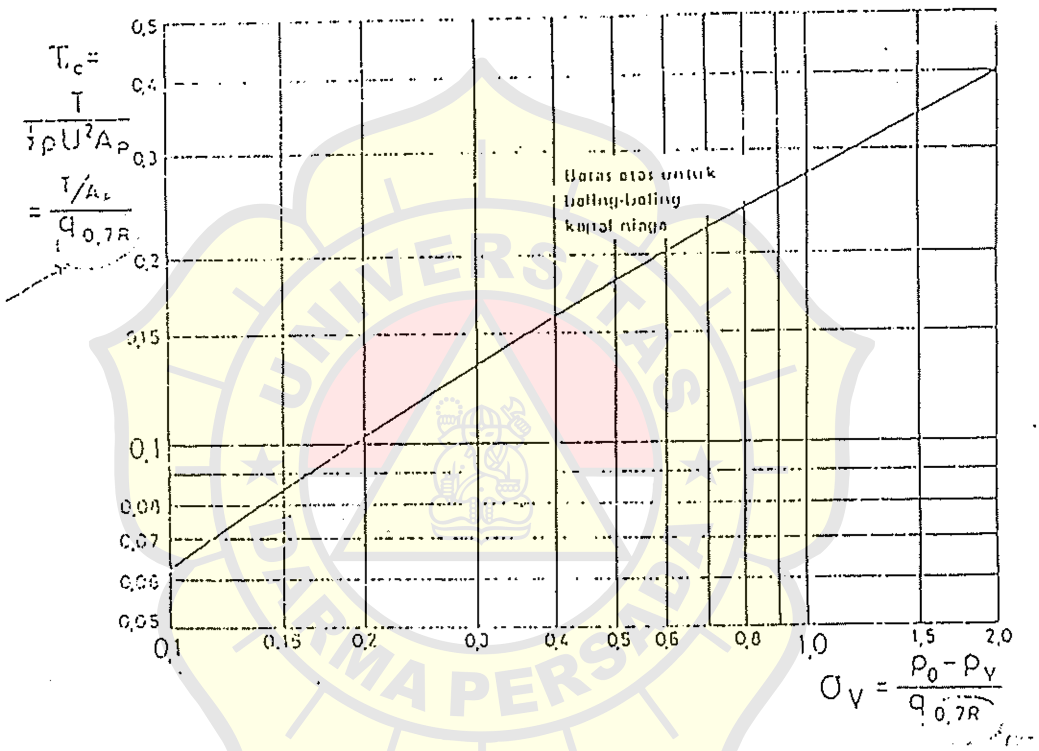


Fig. 117

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



Gbr.No. 2 Diagram Burril

Marine Pumps

Centrifugal Pumps

Naniwa centrifugal pumps are available as single- or double suctions, single- or double volutes, vertical- or horizontal installations, single- or multi stages, etc. to ideally suit any specification requirement.

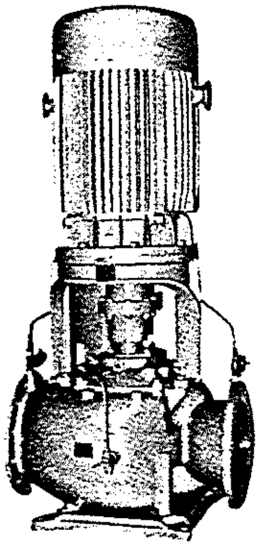
The materials of the principal parts of Naniwa centrifugal pumps are standardized as follows:

Casing: Bronze for sea water
Cast iron for fresh water
Stainless steel for chemicals

Impeller: Phosphor bronze for sea water and fresh water
Stainless steel for sea water and chemicals

Shaft: Stainless steel for sea water and fresh water

Vertical type



FEV
FEWV

Applications:
Cooling water
Water service
Ballast

Specifications:
Vertical single stage
double volute
Centrifugal
Capacity 30–3,000 m³/h
Head 13–50 m

FEWV-300

FBV
FBSV

Applications:
Cooling water
Water service
Fire & general service

Specifications:
Vertical single stage
single suction
Centrifugal
Capacity 30–500 m³/h
Head 10–75 m

FBWV
FCDV

Applications:
Cooling water
Condenser circulating
Ballast
Water service

Specifications:
Vertical single stage
double suction
Centrifugal
Capacity 200–15,000 m³/h
Head 5–50 m

EVPM

Applications:
Product cargo
Chemical cargo

Specifications:
Vertical multi-stage
single suction
Centrifugal
Capacity 50–1,500 m³/h
Head 50–150 m

TOM

Applications:
Lublicating oil
Oil service

Specifications:
Vertical multistage
single suction
Centrifugal
Capacity 30–1,000 m³/h
Head 2–15 kg/cm²

TOM-200

FDDV
2FDDV

Applications:
Condensate
Drain

Specifications:
Vertical 1–2 stage
single (double) suction
Centrifugal
Capacity 2–110 m³/h
Head 50–110 m

FB2V
FBCV
FE2V

Applications:
Fire & general service
Bilge & ballast
Water service

Specifications:
Vertical two stage
single suction
Centrifugal
Capacity 40–800 m³/h
Head 25–170 m

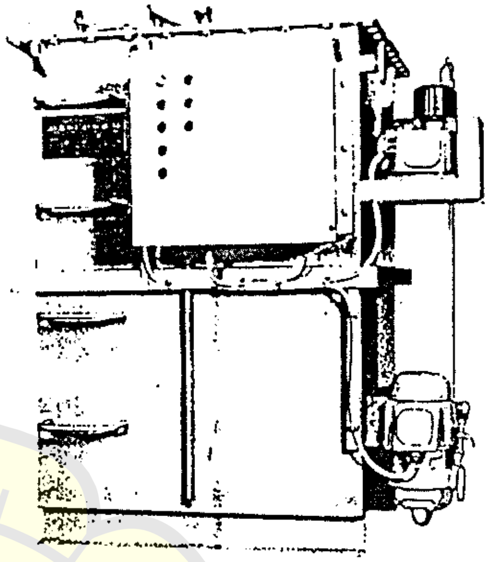
Taiko Kikai Industries Co., Ltd.

Sewage Treatment Unit

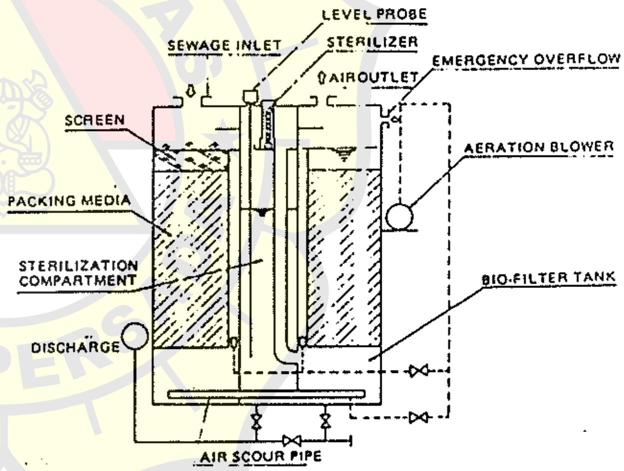
TAIKO SHIP CLEAN "SBT Series" are compact sized sewage treatment system with superior performance capabilities, designed exclusively for marine installations, and which were developed by TAIKO with high technology of many years experiences for Marine Public Nuisance. These device thereby more enables to be installed in all vessels of 200 or more gross tons, or accommodating 10 or more peoples as described in the Annex IV of MARPOL TREATY 73/78.

Features:

1. High Capabilities
   The use of a "Submerged Bio-Filter System" and the transposition of the sterilization compartment to the center of the device enable it to be more compact. These state of art devices are thereby more stable under condition of pitch and roll.
2. Comply with MARPOL TREATY 73/78
   In accordance with the IMO recommended MEPC2 (V1) test standard, the certified authorization for USCG and/or UK/DOT has been obtained.
3. Fully automatic integrated type
   A pump and a blower are mounted on the device. Therefore, piping and wiring works are simplified. These device are fully automatic except few maintenance and control, such as removal of sludge and/or filling disinfectant, etc.
4. The installation can be made at the offshore works.
   It is ultra small, accordingly the installation is feasible without requiring the works in docks.
5. Others
   The stability perioc is shorten. (3 days)



SBT-Series



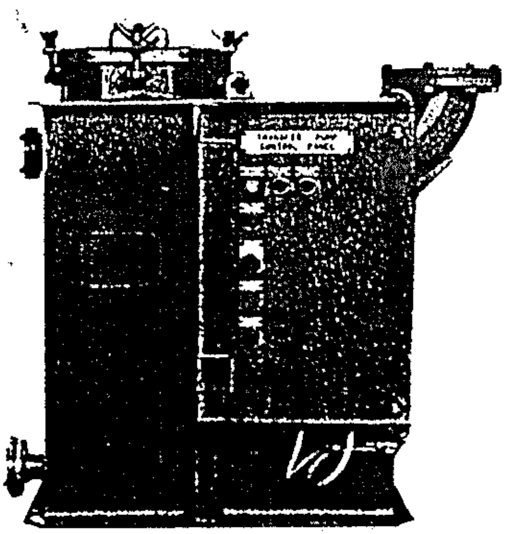
SBT-Series Sectional Drawing

Specification

Item \ Model		SBT-15	SBT-25	SBT-40	SBT-65
Number of persons persons/days		15	25	40	65
Average of sewage processed ℓ/days		900	1500	2400	3900
Processed amount of sewage ℓ/hours × frequency/days		94 × 3	156 × 3	250 × 3	406 × 3
BOD5 amount g/days		202.5	337.5	540	877.5
Blower	Air flow m^3/minutes	0.22	0.37	0.60	0.98
	Pressure kgf/cm^2	0.2			
	Motor power input kW	0.4	0.4	0.75	0.75
Discharge pump	Capacity m^3/hours	7 (60 Hz)		6 (50 Hz)	
	Head m	20 (60 Hz)		12 (50 Hz)	
	Motor power input kW	1.5			

Collecting Tank Model: SCT-200

Toilets on vessels are scattered. When pipings cannot be made to collect sewage from every toilet to a sewage treatment device (BFT-40) please install this sewage collecting tank. The sewage collected in this tank is transferred automatically to the sewage treatment device by an attached sewage transfer pump.



Collection Tank SCT-200

Marine Pumps

Centrifugal Pumps

Naniwa centrifugal pumps are available as single- or double suctions, single- or double volutes, vertical- or horizontal installations, single- or multi stages, etc. to ideally suit any specification requirement.

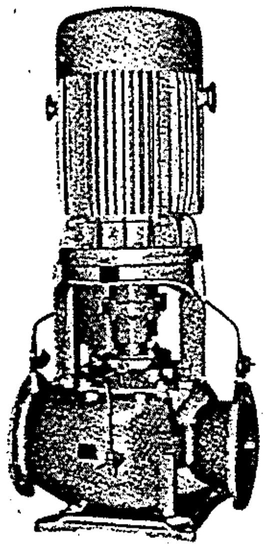
The materials of the principal parts of Naniwa centrifugal pumps are standardized as follows:

Casing: Bronze for sea water
Cast iron for fresh water
Stainless steel for chemicals

Impeller: Phosphor bronze for sea water and fresh water
Stainless steel for sea water and chemicals

Shaft: Stainless steel for sea water and fresh water

Vertical type



FEWV-300

FEV
FEWV

Applications:
Cooling water
Water service
Ballast

Specifications:
Vertical single stage
double volute
Centrifugal
Capacity 30—3,000 m^3/h
Head 13—50 m

TOM-200

TOM

Applications:
Lublicating oil
Oil service

Specifications:
Vertical multistage
single suction
Centrifugal
Capacity 30—1,000 m^3/h
Head 2—15 kg/cm^2

FBV
FBSV

Applications:
Cooling water
Water service
Fire & general service

Specifications:
Vertical single stage
single suction
Centrifugal
Capacity 30—500 m^3/h
Head 10—75 m

FBWV
FCDV

Applications:
Cooling water
Condenser circulating
Ballast
Water service

Specifications:
Vertical single stage
double suction
Centrifugal
Capacity 200—15,000 m^3/h
Head 5—50 m

EVPM

Applications:
Product cargo
Chemical cargo

Specifications:
Vertical multi-stage
single suction
Centrifugal
Capacity 50—1,500 m^3/h
Head 50—150 m

FDDV
2FDDV

Applications:
Condensate
Drain

Specifications:
Vertical 1—2 stage
single (double) suction
Centrifugal
Capacity 2—110 m^3/h
Head 50—110 m

FB2V
FBCV
FE2V

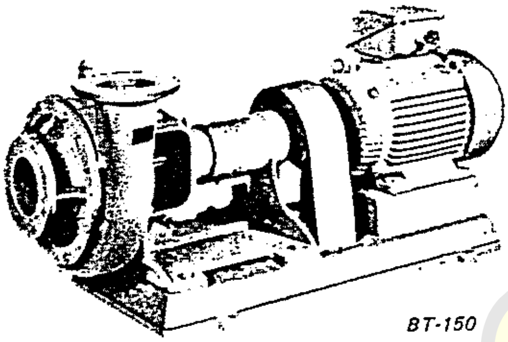
Applications:
Fire & general service
Bilge & ballast
Water service

Specifications:
Vertical two stage
single suction
Centrifugal
Capacity 40—800 m^3/h
Head 25—170 m

Naniwa Pump Mfg. Co., Ltd.

Centrifugal Pumps

Horizontal type

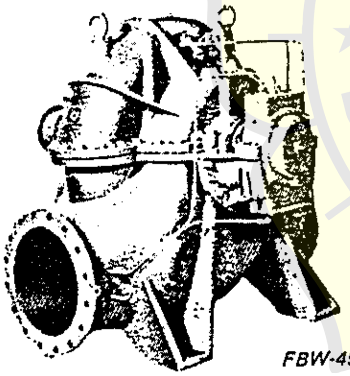


BT-150

BT
BH

Applications:
Cooling water
Water service
Fresh water
Sanitary

Specifications:
Horizontal single stage
single suction
Centrifugal
Capacity 2–700 m^3/h
Head 9–130 m



FBW-450

FBW

Applications:
Cooling water
Condenser circulating
Ballast
Water service

Specifications:
Horizontal single stage
double suction
Centrifugal
Capacity 200–3,500 m^3/h
Head 13–50 m

SGH
SH

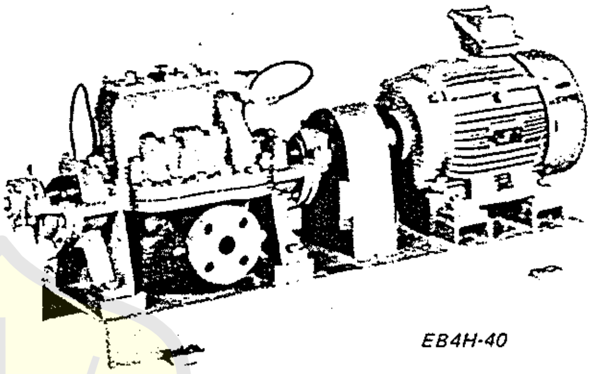
Applications:
Ballast
Water service
Fire & general service

Specifications:
Horizontal single stage
self priming
Centrifugal
Capacity 2–700 m^3/h
Head 10–65 m

FB2H

Applications:
Fire
Emergency fire

Specifications:
Horizontal 2 stage
Single suction
Centrifugal
Capacity 100–500 m^3/h
Head 50–150 m

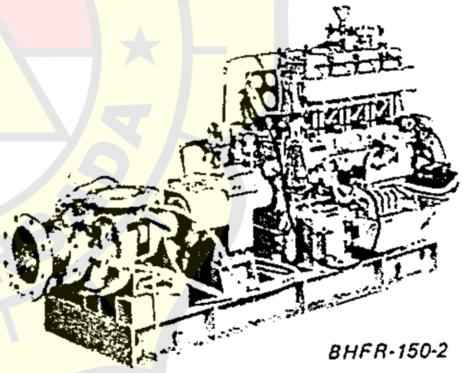


EB4H-40

EBH
EB2H
EB4H

Applications:
Boiler feed
Water service

Specifications:
Horizontal multi stage
single suction
Centrifugal
Capacity 1–80 m^3/h
Head 50–310 m



BHFR-150-2

BHF

Applications:
Fire
Emergency fire

Specifications:
Horizontal single stage
self priming
Centrifugal
Capacity 25–200 m^3/h
Head 40–90 m

BBH-L
BBH-S

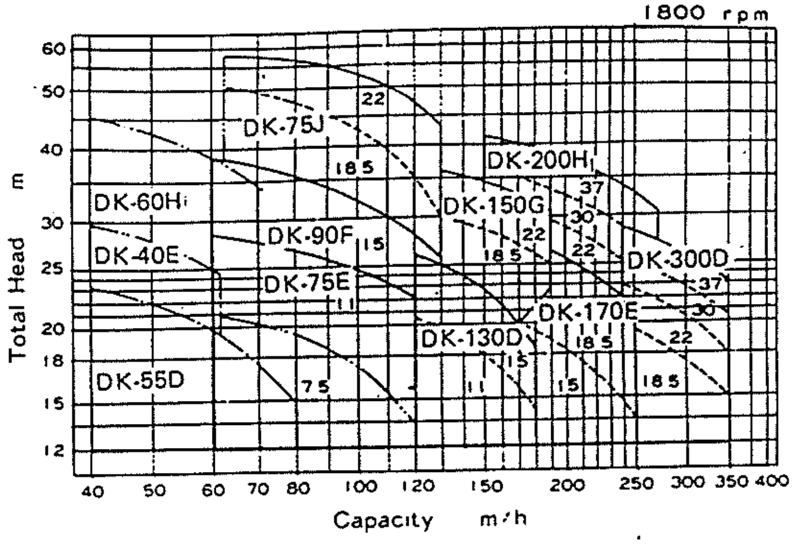
Applications:
Boiler water circulating
Head transfer
Liquids circulating

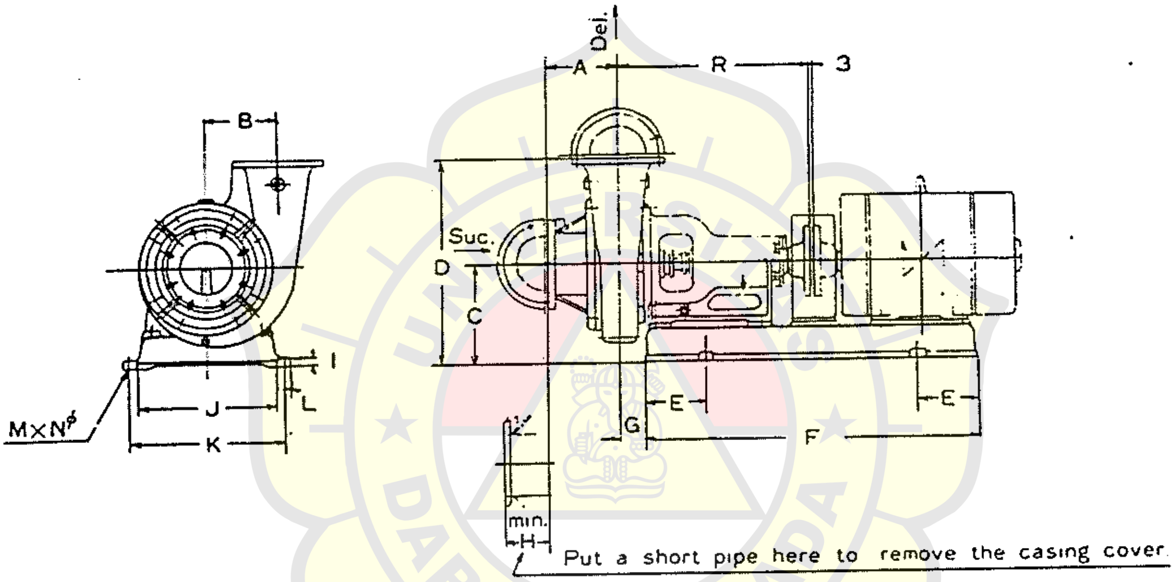
Specifications:
Horizontal single stage
single suction
Centrifugal
Capacity 2–80 m^3/h
Head 15–65 m

EBHU

Applications:
Fire
Salvage

Specifications
Horizontal 2–3 stage
single suction
Centrifugal
Capacity 2–80 m^3/h
Head 80–210 m





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

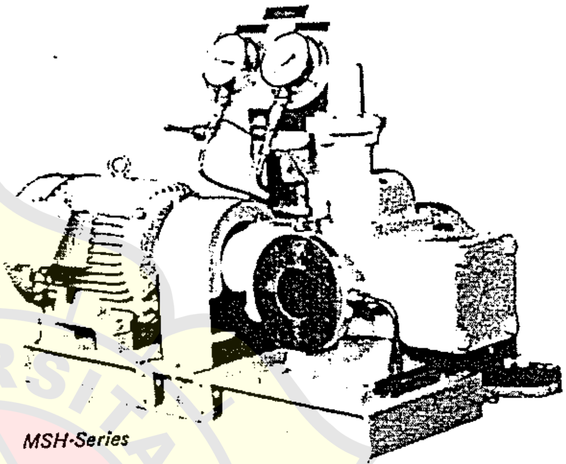
Three Rotor Screw Pump

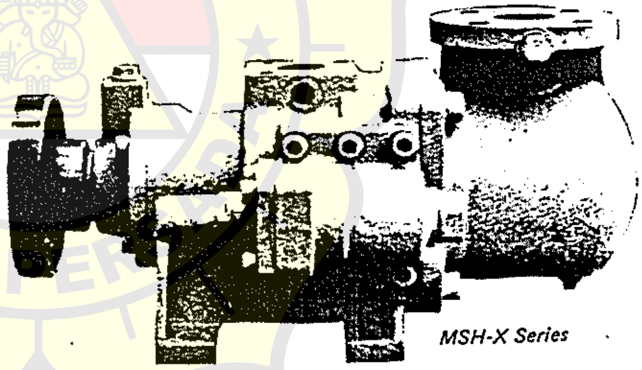

Three Rotor Screw

Features:

1. Simplified construction
   Radically new design using one drive screw and two driven screws.
2. Ease of handling
   Easy to handle as all mechanisms are rationally designed and engineered.
3. Simple and straight forward operation
   The rotating section is at a minimum and each screw rotates by liquid pressure to ensure freedom from vibration and noise.
4. Pulsating and agitation free
   Pulsating and agitation are totally absent to offer continuous pumping and supply liquid.
5. Long service life
   Liquid pressure exerted on the rotating section is properly balanced to reduce minimum. Also driven screw pressure is minimal for minimized wear.
6. High pumping efficiency
   Such screw pump characteristics as force transmission loss, friction loss, and leakage loss are minimized to achieve unusually high pumping efficiency.
7. Small starting torque
   Small inertia on the rotating section means small starting torque.
8. Compact design but large capacity
   High-speed operation and relatively compact design.

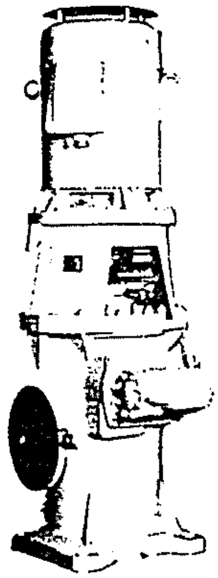
Application: F.O. trans pump, main L.O. pump, hydraulic oil pump, cross head L.O. pump, burning pump, F.O. booster pump.

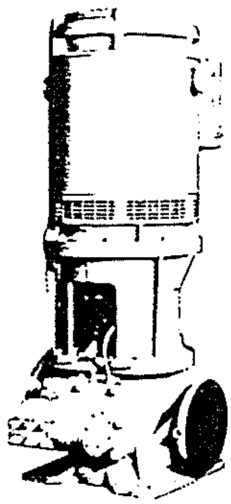

MSH-Series


MSH-X Series

(1) MSH-/MSH-X Series
(Horizontal, internal bearing type)
Capacity : 0.5–150 m^3/hr
Head : Max. 60 kgf/cm^2
H. Power : 0.4–500 kW

(2) MSV/MST Series
(Vertical, internal bearing type)
Capacity : 30–600 m^3/hr
Head : Max. 16 kgf/cm^2
H. Power : 11–150 kW


MSV-Series


MST-Series

Taiko Kikai Industries Co., Ltd.

UST-Series

Certificate No.

Model	CAPA m³/h	H.K. Cert. No.	USCG Cert. No.
UST-01	0.15	820S-p No. 1016	162.050/1056/0
UST-03	0.25	80TK-h No. 108554	162.050/1056/0
UST-05	0.5	80TK-h No. 108555	162.050/1058/0
UST-10	1.0	80TK-h No. 108556	162.050/1059/0
UST-20N	2.0	80TK-h No. 108557	162.050/1060/0
UST-30N	3.0	80TK-h No. 108558	162.050/1061/0
UST-50N	5.0	80TK-h No. 108559	162.050/1062/0

Other Approvals
The UK DOT, The NASASN (Sweden), The Greek Government, The Norwegian Maritime Directorate, The Republic of Panama, Republic of Korea.

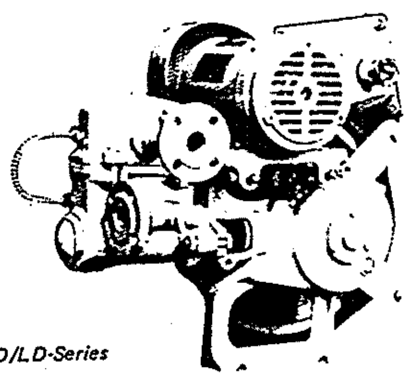
Piston Pump

Features:
1. Strong "Self-Priming", no need for initial feed of water or liquid.
2. Minimum number of parts to facilitate maintenance and for reduced parts stock.

Application: Bilge pump, oily water separator pump F.O. trans pump

(A) Horizontal Type

1) PD-03
   Capacity : 0.25 m^3/hr
   Head : 2 kgf/cm^2
   H. Power : 0.4 kW
2) LD-Series
   Capacity : 0.5–5 m^3/hr
   Head : 2–3 kgf/cm^2
   H. Power : 0.4–1.5 kW

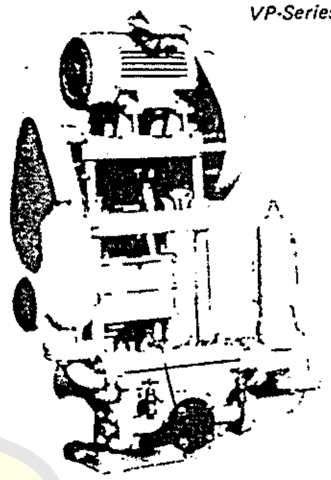


PD/LD-Series

(B) Vertical Type

VP-Series
Capacity : 2–60 m^3/hr
Head : 2–4 kgf/cm^2
H. Power : 0.75–15 kW

VP-Series



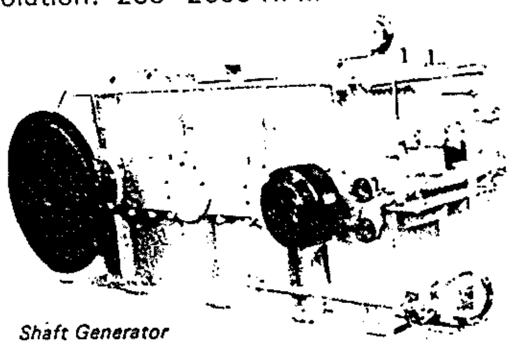
Transmission

Shaft Generator

Due to the increase in oil prices in recent years, the problem of "SAVING ENERGY" has to be solved. Especially, the "FUEL OIL CONSUMPTION RATIO" for vessels and such equipment as the engine, propellers, generator, etc., required improvement.
As a result, we have succeeded in developing new, compact, high-speed models of our SHAFT GENERATOR.

Features:
1) Conventional auxiliary engines use type A-Heavy Fuel Oil. The main engine uses type C-Heavy F.O. except when entering or leaving port. Therefore, since the fuel oil consumption ratio of the main engine, SHAFT GENERATOR reduces both the amount of fuel consumed and the unit price of that fuel. Thus the system allows a dual economy regarding fuel.
2) Maintenance requirements for the aux. engine are eliminated, and engine changes are reduced.
3) Generator, cargo pumps, hydraulic pumps, etc., are driven by a compact transmission system, thus reducing the engine room space required by aux. engines.

Capacity: Up to 3000 HP
Revolution: 200–2000 RPM



Shaft Generator

ʒtarting Compressor (Water-cooled)

Model No.	Speed (r.p.m.)	25kgf/cm²			30kgf/cm²		
		Capacity m^3/hr (FA)	Power required PS	Motor (kW)	Capacity m^3/hr (FA)	Power required PS	Motor (kW)
H-63	1,200	65	16.5	15	60	17	15
	1,500	80	20	18.5	75	21	18.5
	1,800	95	25	22	90	26.5	22
H-64	1,200	90	23	18.5	85	24	18.5
	1,500	110	29	25	105	30	25
	1,800	135	36.5	30	130	37.5	30
H-264	1,200	175	45.5	37	170	48	37
	1,500	215	57	45	210	60	50
	1,800	260	70.5	55	255	73.5	60
H-73	1,200	110	28	25	105	29	25
	1,500	135	35	30	130	36	30
	1,800	160	46	37	155	47	37
H-74	1,200	140	35	30	135	36	30
	1,500	175	45	37	170	46	37
	1,800	205	58	45	200	60	50
H-273	1,200	220	55	45	215	58	45
	1,500	275	69	55	270	73	60
	1,800	325	86	65	320	88	70
H-274	1,200	275	70	55	270	73	55
	1,500	340	88	70	335	91	75
	1,800	405	108	85	400	111	90
H-373	1,200	330	83	65	320	87	70
	1,500	410	104	85	400	109	90
	1,800	485	128	100	475	132	110
H-374	1,200	415	105	85	405	110	90
	1,500	515	130	110	505	135	110
	1,800	610	162	125	600	168	132

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

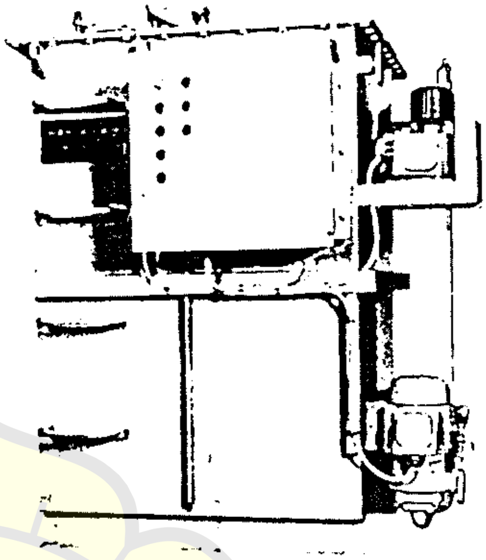
ıergency Compressor (Vertical 2-stage Air-cooled)

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

Taiko Kikai Industries Co., Ltd.

Sewage Treatment Unit

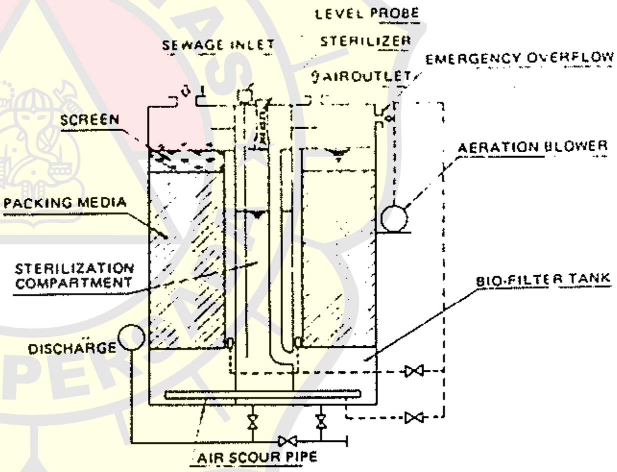
TAIKO SHIP CLEAN "SBT Series" are compact sized sewage treatment system with superior performance capabilities, designed exclusively for marine installations, and which were developed by TAIKO with high technology of many years experiences for Marine Public Nuisance. These device thereby more enables to be installed in all vessels of 200 or more gross tons, or accommodating 10 or more peoples as described in the Annex IV of MARPOL TREATY 73/78.



SBT-Series

Features:

1. High Capabilities
   The use of a "Submerged Bio-Filter System" and the transposition of the sterilization compartment to the center of the device enable it to be more compact. These state of art devices are thereby more stable under condition of pitch and roll.
2. Comply with MARPOL TREATY 73/78
   In accordance with the IMO recommended MEPC2 (V1) test standard, the certified authorization for USCG and/or UK/DOT has been obtained.
3. Fully automatic integrated type
   A pump and a blower are mounted on the device. Therefore, piping and wiring works are simplified. These device are fully automatic except few maintenance and control, such as removal of sludge and/or filling disinfectant, etc.
4. The installation can be made at the offshore works.
   It is ultra small, accordingly the installation is feasible without requiring the works in docks.
5. Others
   The stability period is shorten. (3 days)



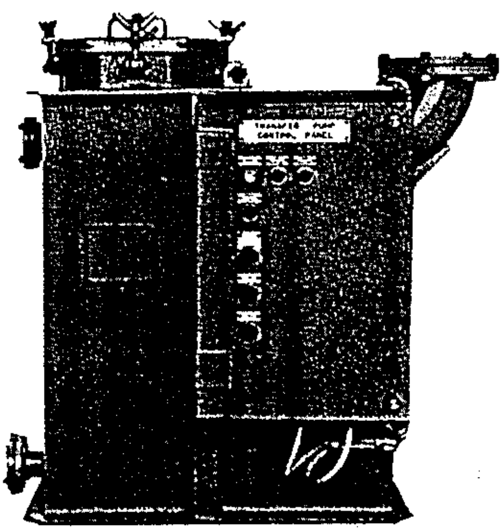
SBT-Series Sectional Drawing

Specification

Item \ Model		SBT-15	SBT-25	SBT-40	SBT-65
Number of persons persons/days		15	25	40	65
Average of sewage processed ℓ/days		900	1500	2400	3900
Processed amount of sewage ℓ/hours × frequency/days		94 x 3	156 x 3	250 x 3	406 x 3
BOD₅ amount g/days		202.5	337.5	540	877.5
Blower	Air flow m³/minutes	0.22	0.37	0.60	0.98
	Pressure kgf/cm²	0.2			
	Motor power input kW	0.4	0.4	0.75	0.75
Discharge pump	Capacity m³/hours	7 (60 Hz)		6 (50 Hz)	
	Head m	20 (60 Hz)		12 (50 Hz)	
	Motor power input kW	1.5			

Collecting Tank Model: SCT-200

Toilets on vessels are scattered. When pipings cannot be made to collect sewage from every toilet to a sewage treatment device (BFT-40) please install this sewage collecting tank. The sewage collected in this tank is transferred automatically to the sewage treatment device by an attached sewage transfer pump.



Collection Tank SCT-200

LAMPIRAN 2

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

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

† 1 kg/(m·s) = 0,0209 slug/(ft·s); 1 m²/s = 10,76 ft²/s.

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

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

BERAT JENIS BEBERAPA FLUIDA YANG LAZIM

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

LAMPIRAN 3

Pumps

Hose diameter d_h, mm	Hose length l_h, m	Nozzle orifice diameter d_n, mm: 10	13	16	19
		Characteristic B			
50	0	0.121	0.346	0.793	1.577
	10	0.119	0.331	0.722	1.320
	20	0.118	0.318	0.622	1.130
	40	0.114	0.304	0.568	0.882
	60	0.111	0.274	0.498	0.723
	80	0.108	0.257	0.442	0.612
	100	0.105	0.241	0.398	0.531
65	0	0.121	0.346	0.793	1.577
	10	0.1205	0.342	0.776	1.51
	20	0.120	0.339	0.758	1.44
	40	0.1195	0.332	0.726	1.33
	60	0.1185	0.326	0.696	1.23
	80	0.118	0.320	0.669	1.15
	100	0.117	0.314	0.644	1.03

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

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

LAMPIRAN 4

Jangkar, rantai dan tali

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

LAMPIRAN 5

Mooring and Warping Ropes

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

LAMPIRAN 6

Self-Propelled Transport Ships with an Unlimited Region of Navigation

No.	Characteristic X	Anchors: Bower: Quantity	Anchors: Bower: Total weight, kg	Anchors: Stream anchor, kg	Chain cable for bower anchors: Total length of two cables, m	Chain cable for bower anchors: Anchor chain size, mm	Chain or steel rope for the stream anchor: Length, m	Chain or steel rope for the stream anchor: Anchor chain size, mm	Chain or steel rope for the stream anchor: Diameter of steel rope, mm
1	50	2	150	25	100	12	50	—	8.8
2	75	2	200	25	125	13	50	—	8.8
3	100	2	250	50	125	15	50	—	11
4	150	2	300	50	150	16	50	—	11
5	200	2	350	50	175	17	75	—	11
6	250	2	450	75	200	18	75	11	13
7	300	2	500	75	225	19	75	13	13
8	350	2	600	100	250	20	75	14	15.5
9	400	2	700	100	275	21	75	14	15.5
10	450	2	750	125	300	22	100	15	17.5
11	500	2	800	150	300	24	100	16	17.5
12	550	2	900	175	325	25	100	16	17.5
13	600	3	1500	200	350	27	100	17	17.5
14	650	3	1700	225	350	28	100	18	19.5
15	700	3	1800	250	375	29	100	18	20.5
16	750	3	2100	250	375	30	100	19	20.5
17	800	3	2250	250	375	31	125	19	20.5
18	850	3	2400	275	375	32	125	20	22
19	900	3	2700	300	375	33	125	21	24
20	950	3	3000	300	400	34	125	21	24
21	1000	3	3200	350	400	35	125	22	24
22	1100	3	3500	400	400	37	125	23	26
23	1200	3	3750	400	420	38	150	25	26
24	1300	3	4100	450	450	40	150	25	28
25	1400	3	4250	450	450	41	150	25	28
26	1500	3	4500	500	450	42	150	25	28
27	1600	3	4750	500	450	43	150	26	28
28	1700	3	5250	600	450	45	150	28	30
29	1850	3	5500	600	450	46	150	28	30
30	2000	3	5750	700	450	46	150	29	31.5
31	2150	3	6000	700	475	48	175	29	31.5
32	2300	3	6500	800	500	49	175	29	32.5
33	2500	3	6750	800	500	50	175	29	32.5
34	2700	3	7500	900	500	52	175	30	33.5
35	3000	3	8250	1000	500	53	200	31	33.5
36	3300	3	9000	1000	500	55	200	31	33.5
37	3600	3	9750	1250	525	57	200	33	34.5
38	3900	3	10500	1250	550	59	225	33	34.5
39	4200	3	11000	1400	550	61	225	34	37
40	4500	3	11500	1500	550	62	225	35	37
41	4800	3	12900	1650	550	65	225	36	—
42	5100	3	13500	1750	550	67	250	37	—
43	5400	3	14500	1750	575	68	250	37	—
44	5800	3	15000	2000	600	70	250	40	—
45	6200	3	15800	2000	600	72	250	40	—
46	6600	3	16300	2250	600	74	275	43	—
47	7000	3	17600	2250	600	76	275	43	—
48	7400	3	18000	2250	600	77	275	44	—
49	7800	3	19500	2500	600	80	275	46	—
50	8200	3	20300	2700	600	82	275	48	—
51	8600	3	21000	2800	600	83	275	49	—
52	9000	3	22000	3000	600	85	275	50	—
53	9500	3	23000	3000	600	87	275	50	—

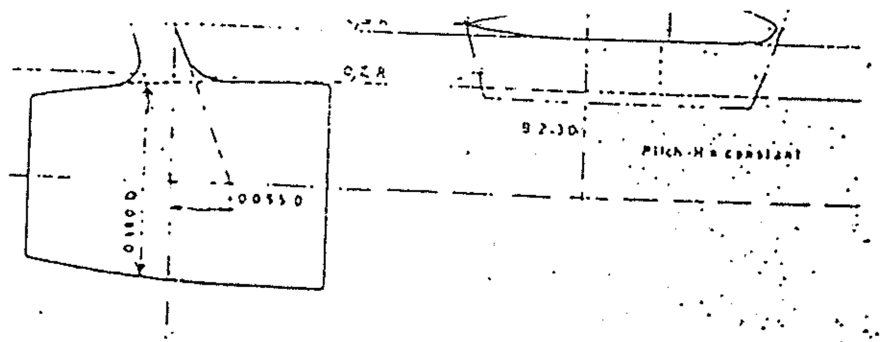


Fig. 33. *General plan of the two-bladed propeller of type B 2-30*

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

Distance of the ordinates from the maximum thickness

R	From maximum thickness to trailing edge 100%	80%	60%	40%	20%	From maximum thickness to leading edge 20%	40%	60%	80%	90%	95%	100%
						Ordinates for the back						
	—	53.35	72.65	86.90	96.45	98.60	94.50	87.00	74.40	64.35	56.95	—
	—	50.95	71.60	86.80	96.80	98.40	94.00	85.30	72.50	62.65	54.90	—
	—	47.70	70.25	86.55	97.00	98.20	93.25	84.30	70.40	60.15	52.20	—
	—	43.40	68.40	86.10	96.95	98.10	92.40	82.30	67.70	56.80	48.60	—
	—	40.20	67.15	85.40	96.80	98.10	91.25	79.35	63.60	52.20	43.35	—
	—	39.40	66.90	84.90	96.65	97.60	88.80	74.90	57.00	44.20	35.00	—
	—	40.95	67.80	85.30	96.70	97.00	35.30	68.70	48.25	34.55	25.45	—
	—	45.15	70.00	87.00	97.00	97.00	87.00	70.00	45.15	30.10	22.00	—
	—	44.80	72.00	88.00	97.20	97.20	88.80	72.00	44.80	29.50	21.60	—
						Ordinates for the face						
	30.00	18.20	10.90	5.45	1.55	0.45	2.30	5.90	13.45	20.30	26.20	40.00
	25.35	12.20	5.80	1.70	—	0.05	1.30	4.60	10.85	16.55	22.20	37.55
	17.85	6.20	1.50	—	—	—	0.30	2.65	7.80	12.50	17.90	34.50
	9.70	1.75	—	—	—	—	—	0.70	4.30	8.45	13.30	30.40
	5.1	—	—	—	—	—	—	—	0.80	4.45	8.40	24.50
	—	—	—	—	—	—	—	—	—	0.40	2.45	16.05
	—	—	—	—	—	—	—	—	—	—	—	7.40

he percentages of the ordinates relate to the maximum thickness of the corresponding sections, the curve of thickness ...
ssumed to be rectilinear. The connecting lines of ...

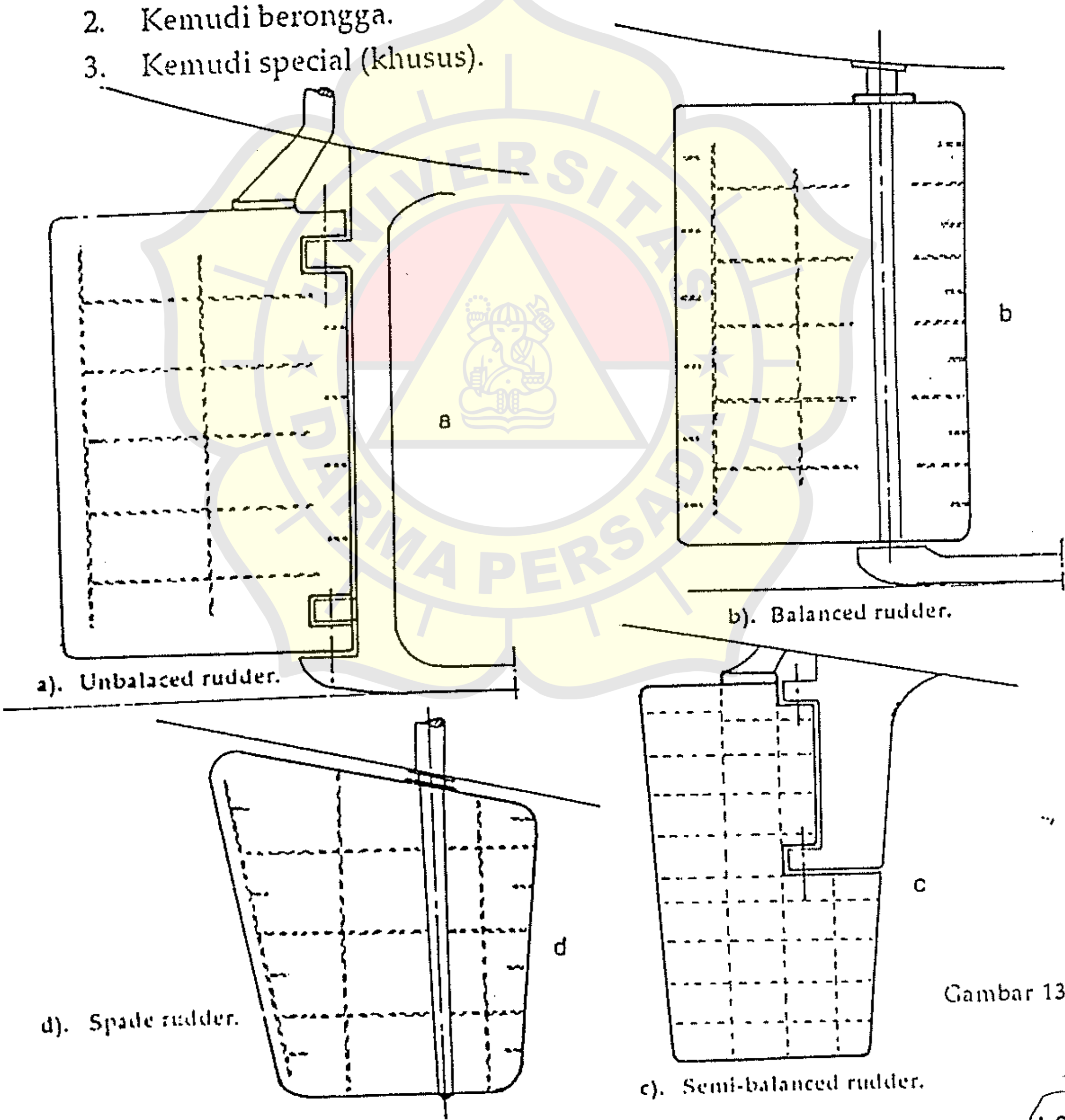
2. Kemudi balansir, dimana luas sayap kemudi terbagi dua, bagian dimuka dan dibelakang sumbu putar kemudi (gambar b).
3. Kemudi setengah balansir, dimana bagian atas sayap kemudi termasuk kemudi biasa, sedang bagian bawah merupakan kemudi balansir sedangkan bagian atas dan bawah tetap merupakan satu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

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

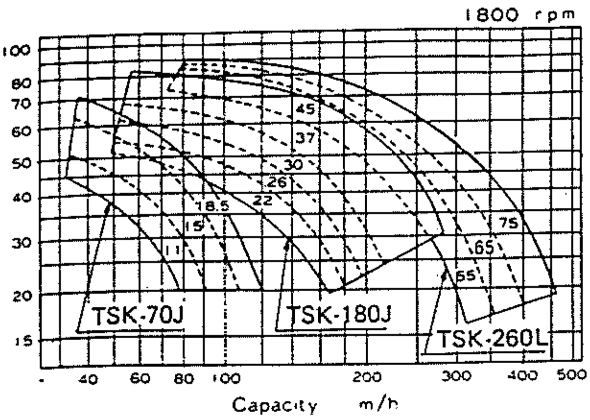
1. Kemudi meletak (gambar a dan b)
2. Kemudi menggantung (gambar d)
3. Kemudi setengah menggantung (gambar c)

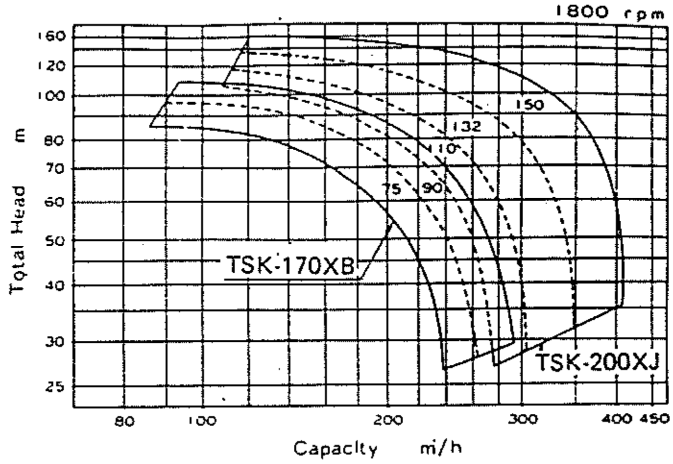
C). Dipandang dari konstruksinya dibagi :

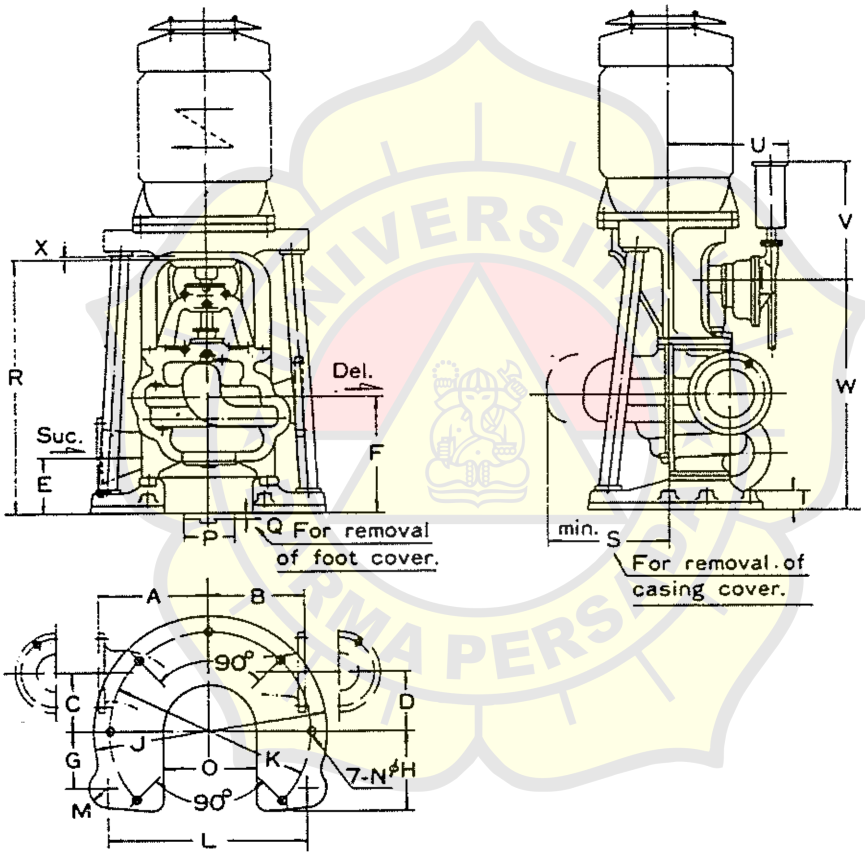
1. Kemudi plat (satu lapis plat).
2. Kemudi berongga.
3. Kemudi special (khusus).



a). Unbalaced rudder.
b). Balanced rudder.
d). Spade rudder.
c). Semi-balanced rudder.
Gambar 130.





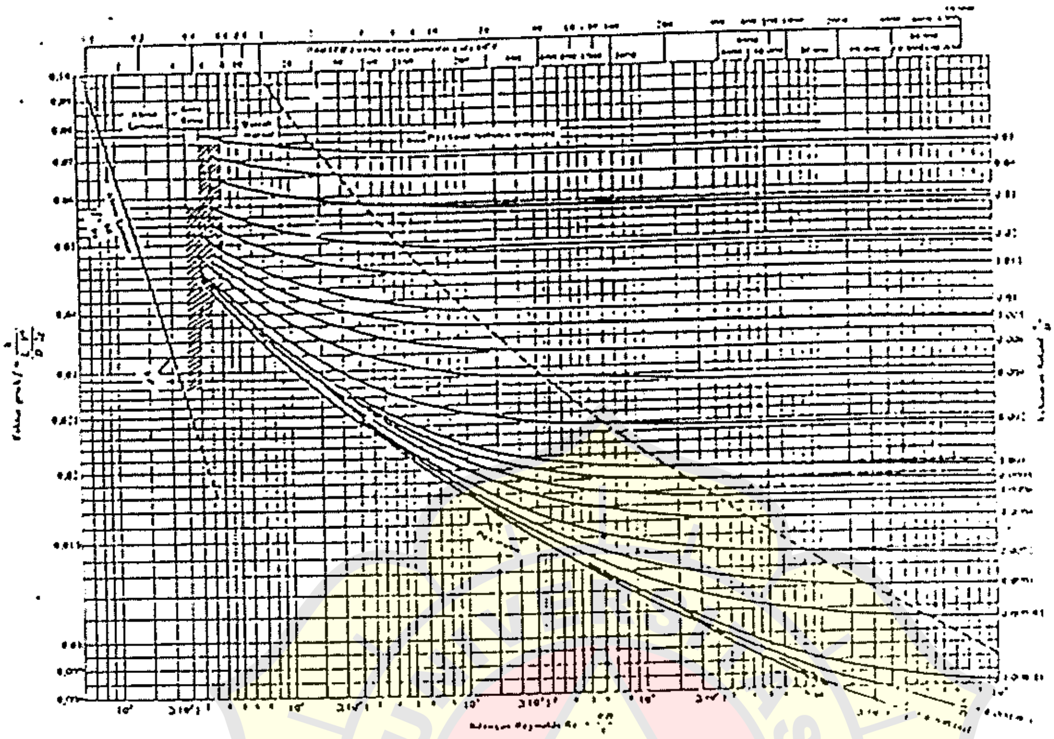


Type	Bore Suc.	Bore Del.	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Vacuum Pump	Pump Weight (kg)
TSK-50H	100	100	250	250	155	155	155	295	140	230	600	530	540	63	23	270	158	40	767	340	50	408	465	677	3	V-18	300
TSK-40N	100	100	260	260	190	190	165	295	140	230	600	530	540	63	23	270	158	5	767	310	50	408	465	677	3	V-18	370
TSK-70J	125	125	275	275	175	175	180	344	190	260	700	630	630	65	27	280	220	25	800	390	65	420	465	703	3	V-50	400
TSK-95J	125 / 150	125 / 150	350	300	185	185	203 / 190	381	190	260	700	630	630	65	27	280	188	5	840	425	65	420	465	743	3	V-50	430
TSK-100P	125	125	300	300	210	210	225	410	210	300	800	730	680	65	27	300	220	25	923	450	80	416	465	826	3	V-50	370
TSK-180J	200	200	350	350	200	200	250	495	210	300	800	730	680	65	27	300	188	30	997	456	80	416	465	900	3	V-50	470
TSK-260L	250	250	370	400	240	240	350	650	230	340	900	810	800	75	33	380	230	–	1312	520	110	430	465	1200	4	V-50	1010
TSK-200XJ	250	250	450	450	250	280	380	680	320	450	1200	1070	1100	87	39	500	250	–	1360	600	120	490	465	1220	4	V-50	1120
TSK-170XB	200	200	390	390	230	230	340	585	270	380	1000	900	960	87	33	470	188	30	1230	510	110	490	465	1090	4	V-50	920

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

L. B. H (m)	L. B. H. (ft^3)	Kapasitas (ft^3)	Jumlah orang	berat sekoci (kg)	Berat Orang (kg)	berat perlengkapan(kg)	Total berat (kg).
9,4 x 2,74 x 1 x1,114	30 x 9 x 3,75	607	60	2205	4500	356	7061
8,84 x 2,74 x 1,10	29 x 8,75 x 3,60	545	54	1976	4050	356	6382
8,53 x 2,59 x 1,07	28 x 8,50 x 3,50	500	50	1824	3750	330	5894
8,23 x 2,51 x 1,04	27 x 8,25 x 3,40	454	45	1646	3376	330	5351
7,92 x 2,44 x 0,99	26 x 8,00 x 3,25	405	40	473	3000	305	4778
7,62 x 2,36 x 0,96	25 x 7,75 x 3,15	366	36	1326	2700	305	4331
7,31 x 2,29 x 0,91	24 x 7,50 x 3,0	324	32	1180	2400	254	3843
7,01 x 2,29 x 0,88	23 x 7,50 x 2,90	300	30	1087	2250	254	3591 ✓
6,71 x 2,21 x 0,84	22 x 7,25 x 2,75	236	26	955	1950	229	3134
6,40 x 2,13 x 0,82	21 x 7,0 x 2,70	238	23	264	1725	229	2818
6,10 x 2,06 x 0,79	20 x 6,75 x 2,60	210	21	762	1575	203	2540
5,79 x 1,98 x 0,76	19 x 6,50 x 2,50	182	18	650	1350	178	2178
5,49 x 1,90 x 0,75	18 x 6,25 x 2,40	162	16	590	1200	152	1942
5,18 x 1,83 x 0,715	17 x 6,0 x 2,30	145	14	508	1050	152	1710
4,88 x 1,75 x 0,70	16 x 5,75 x 2,30	127	12	452	900	127	1494

LAMPIRAN 1



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

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

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

† Bersentuhan dengan udara.

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

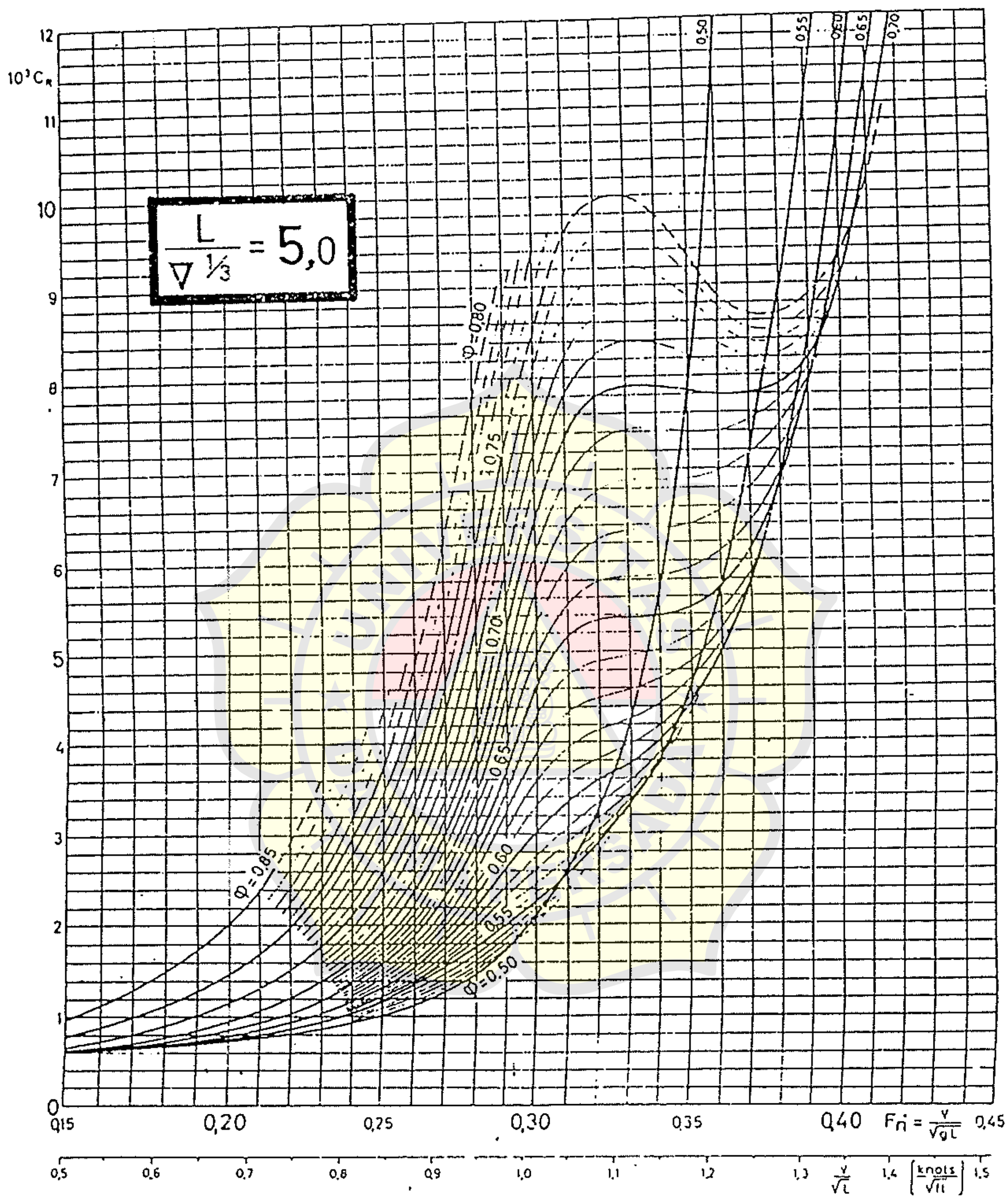
Gbr No.6 Koreksi Bentuk Anggota Badan

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

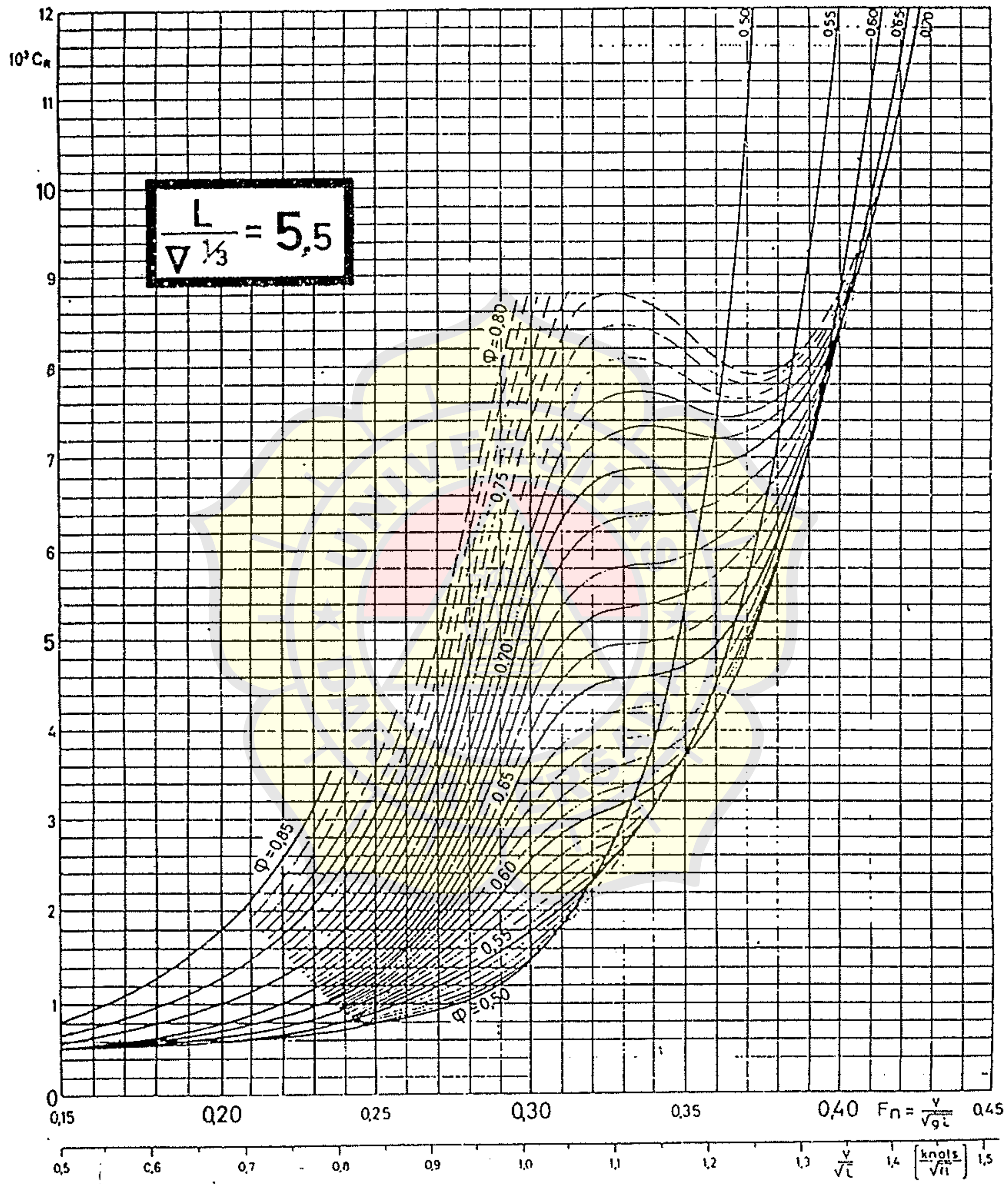
Gbr.No. 6 Koreksi Tahanan Tambahan

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

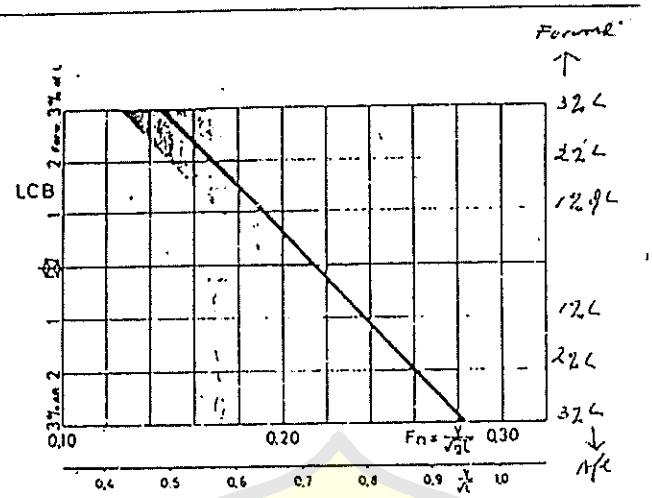
Gbr.No.4 Koreksi Bentuk Haluan



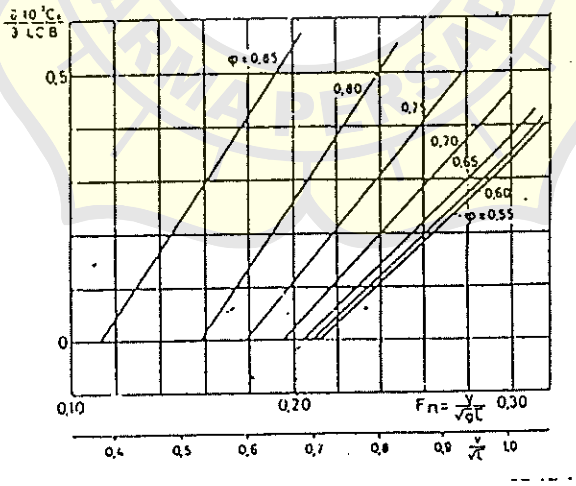
Gbr. No. 2 Koefisien Hambatan Sisa ($L / \nabla^{1/3}$)



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



Gbr.No.3 LCB Standar



Gbr.No.3 Koreksi LCB

ANGGOTA BADAN KAPAL

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

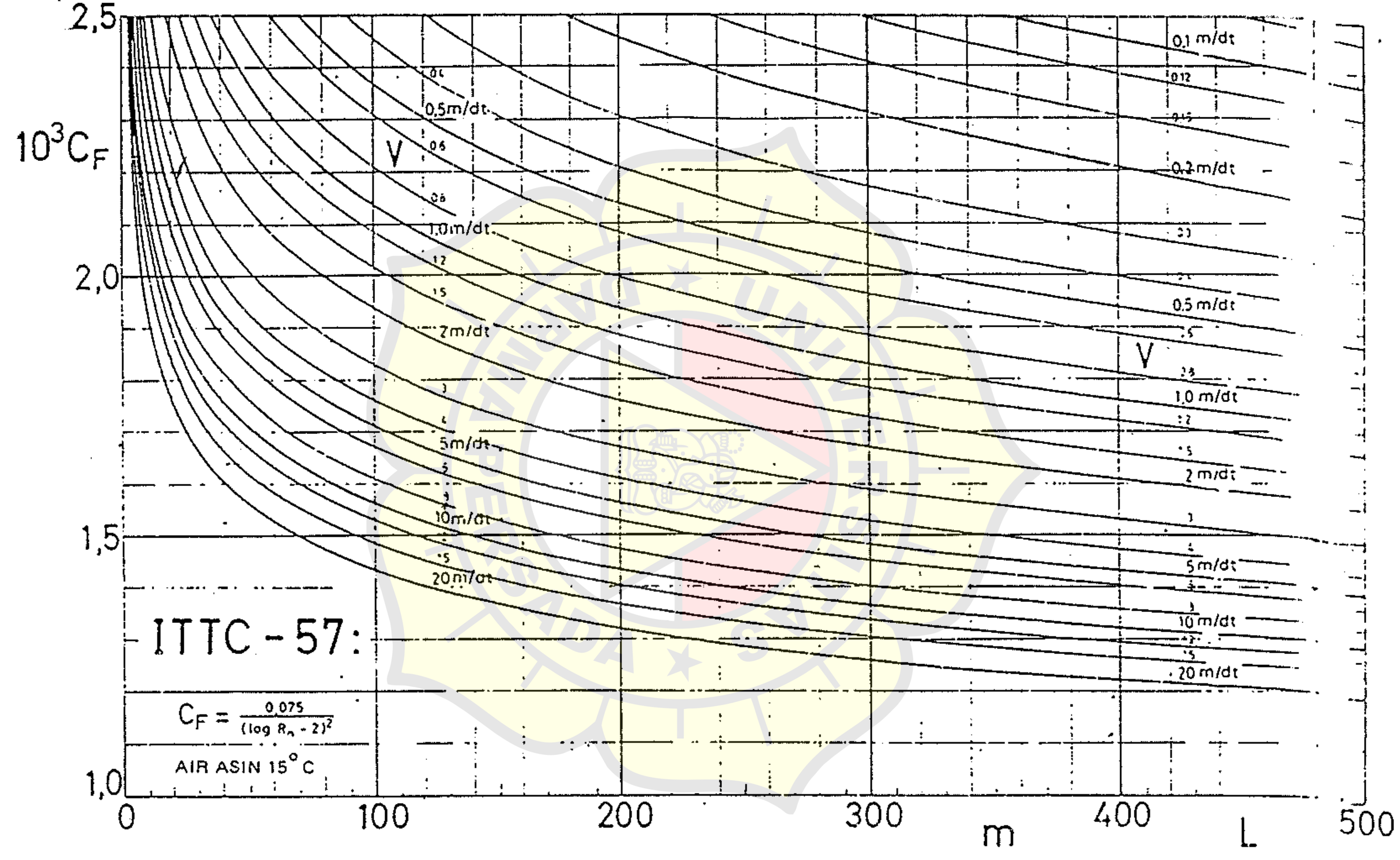
Gbr No.6 Koreksi Bentuk Anggota Badan

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

Gbr.No. 6 Koreksi Tahanan Tambahan

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

Gbr.No.4 Koreksi Bentuk Haluan



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

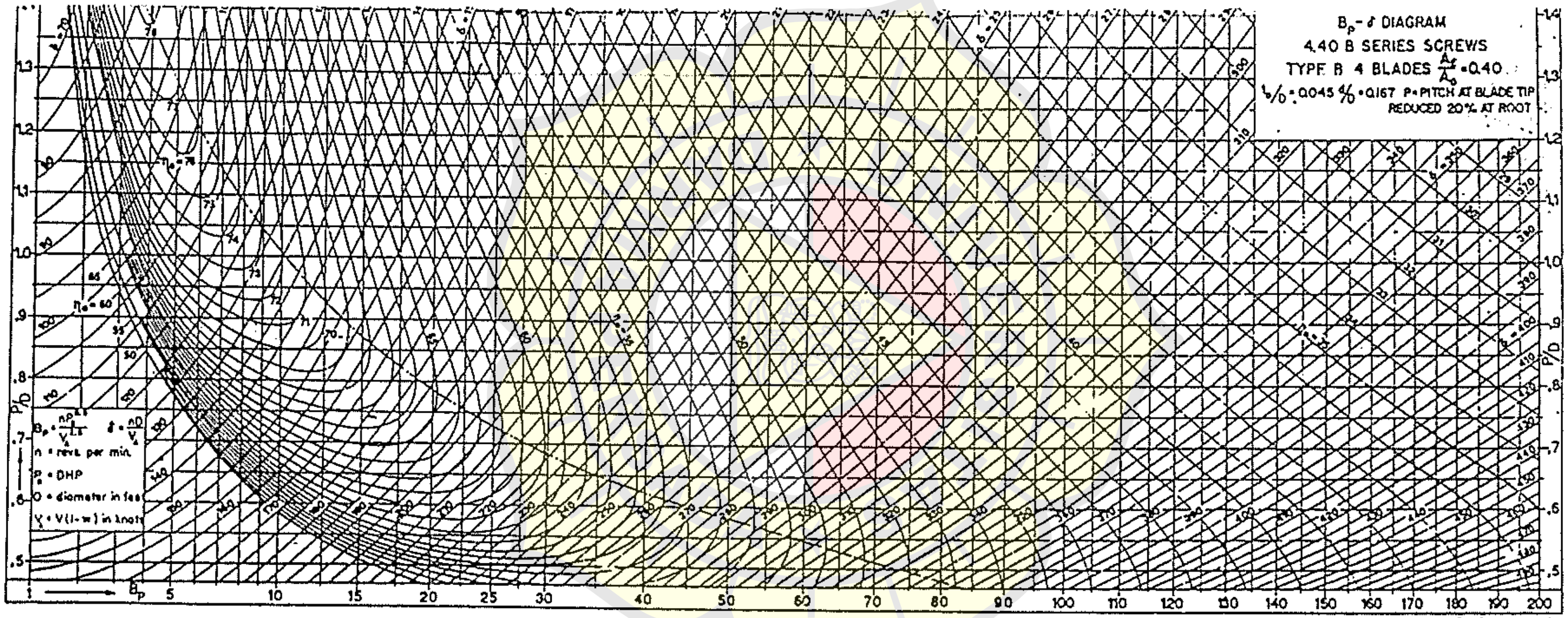
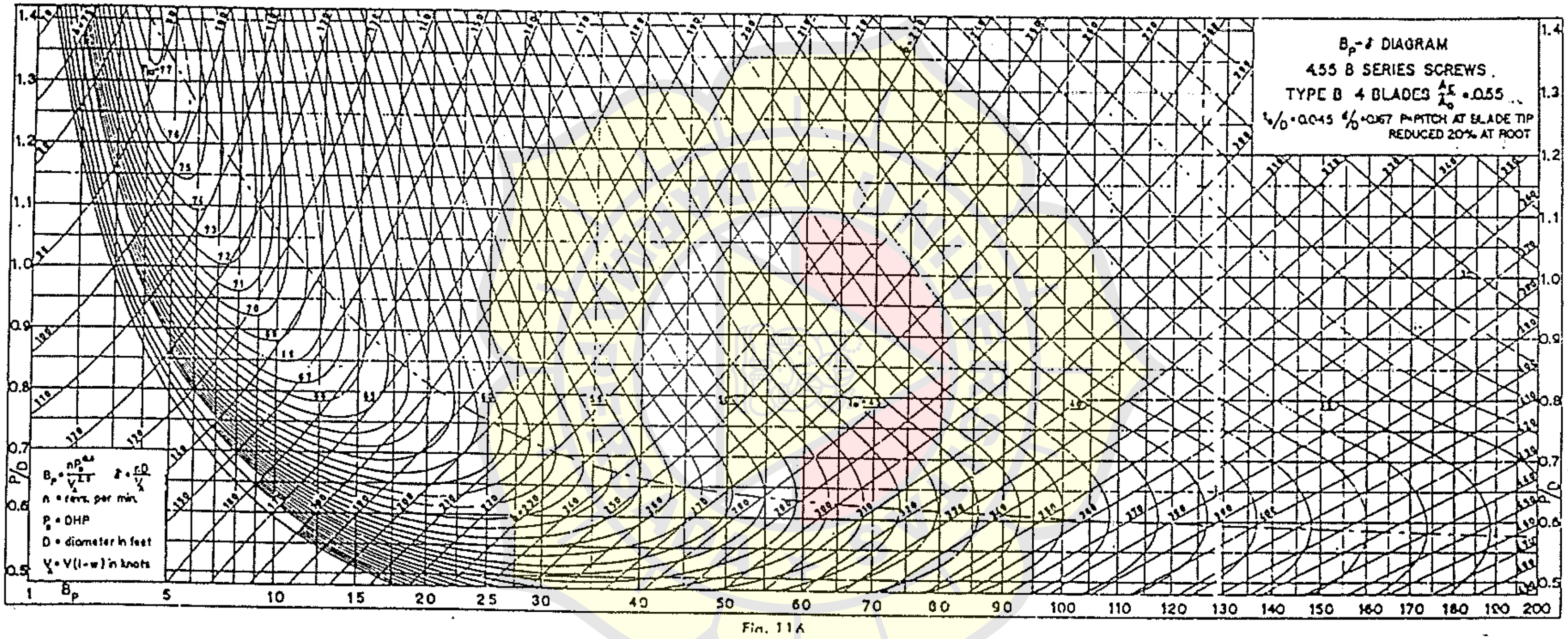


Fig. 115

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



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

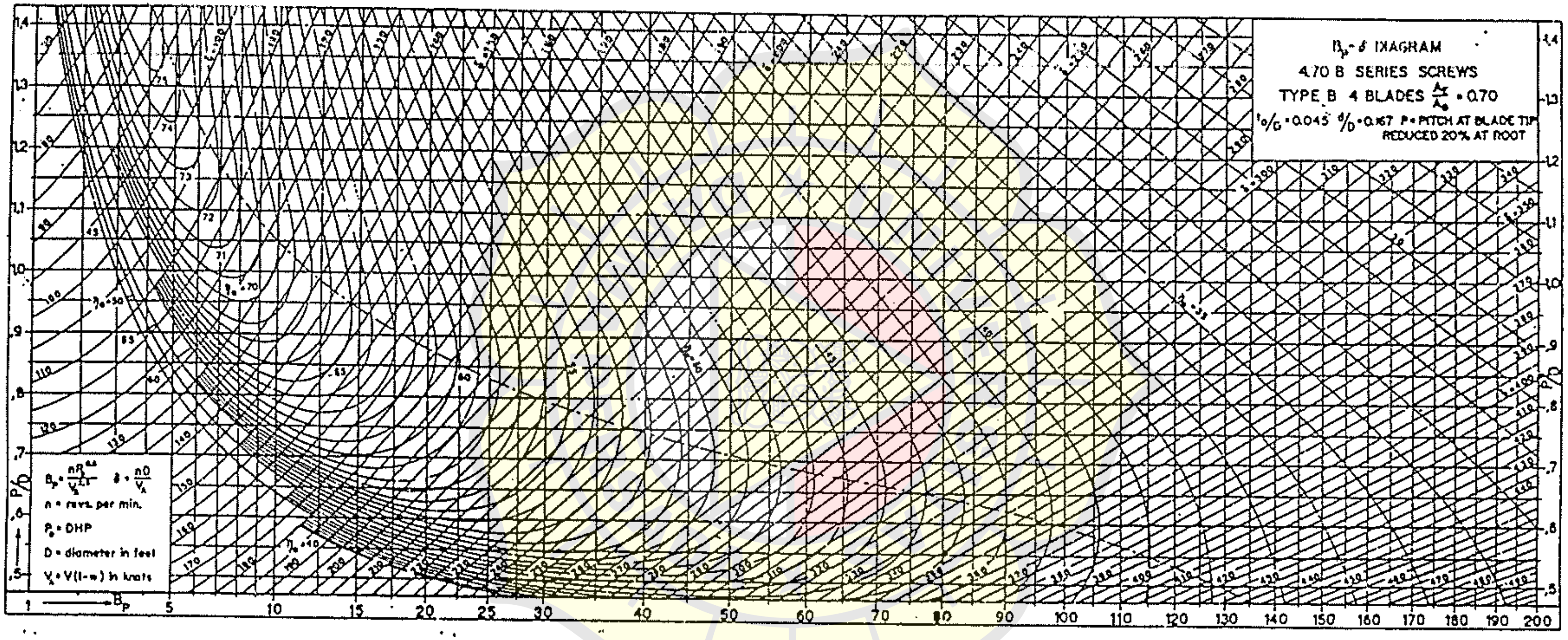
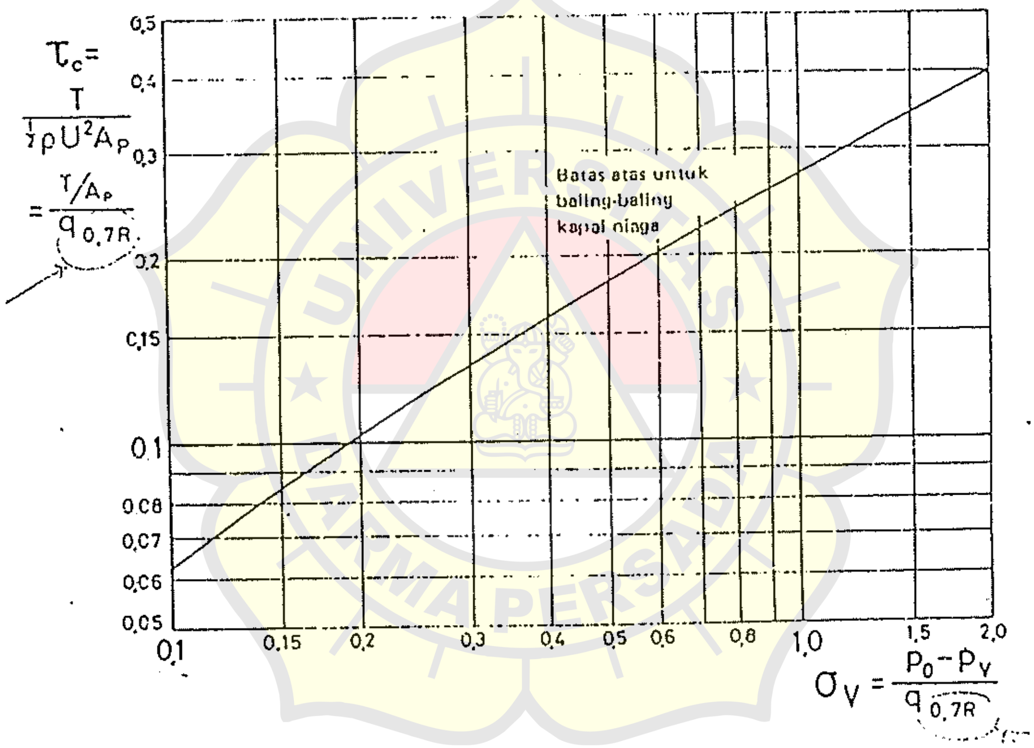


Fig. 117

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



Gbr.No. 2 Diagram Burril

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

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

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

3. Koefisien tahanan sisa spesifik ditentukan dari

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

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

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

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

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

4. C_R dinyatakan sebagai fungsi angka Froude

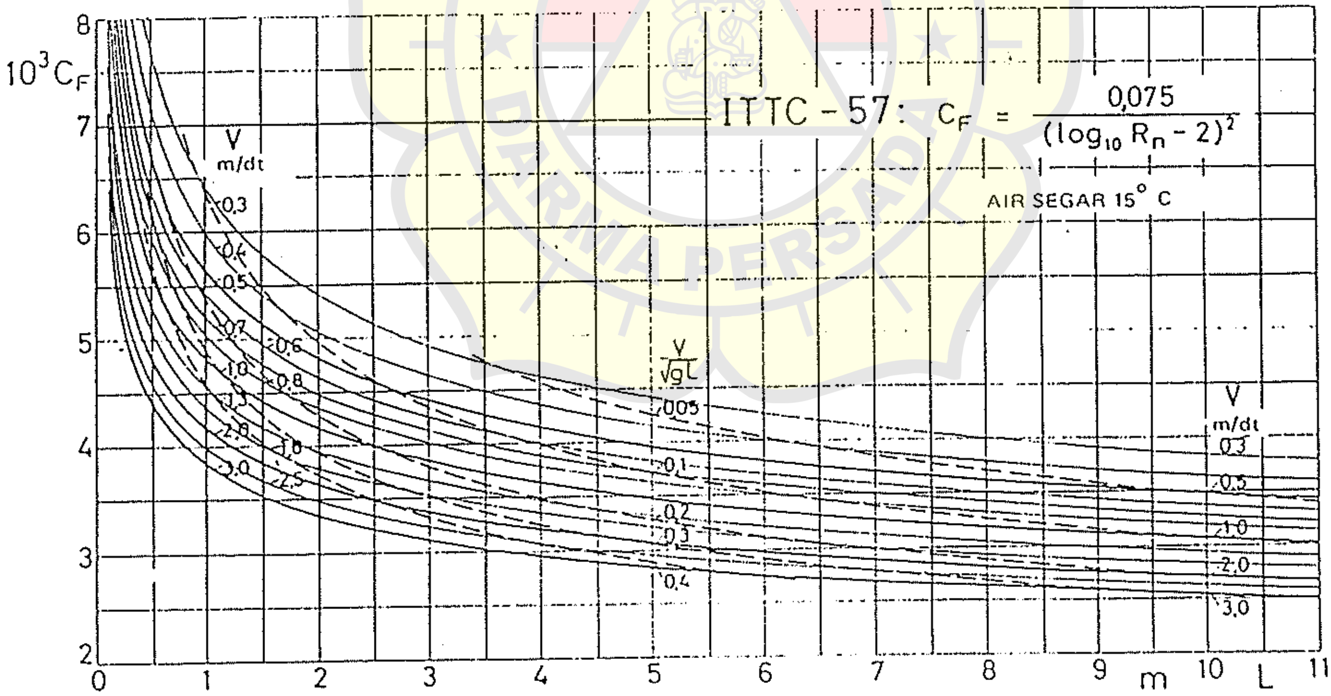
$$F_n = \frac{V}{\sqrt{gL}} \qquad (5.5.9)$$

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

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

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

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



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

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

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

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

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

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

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

Dalam hal ini koefisien tahanan totalnya adalah

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

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

C_F = koefisien tahanan gesek dan dapat dihitung dengan memakai

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

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

$$L_1 = \frac{1{,}188}{10^6 \nu} L \qquad (5.5$$

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

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

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

B/T

Karena diagram tersebut dibuat berdasarkan ra lebar – sarat

$$B/T = 2{,}5 \qquad (5.5.$$

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

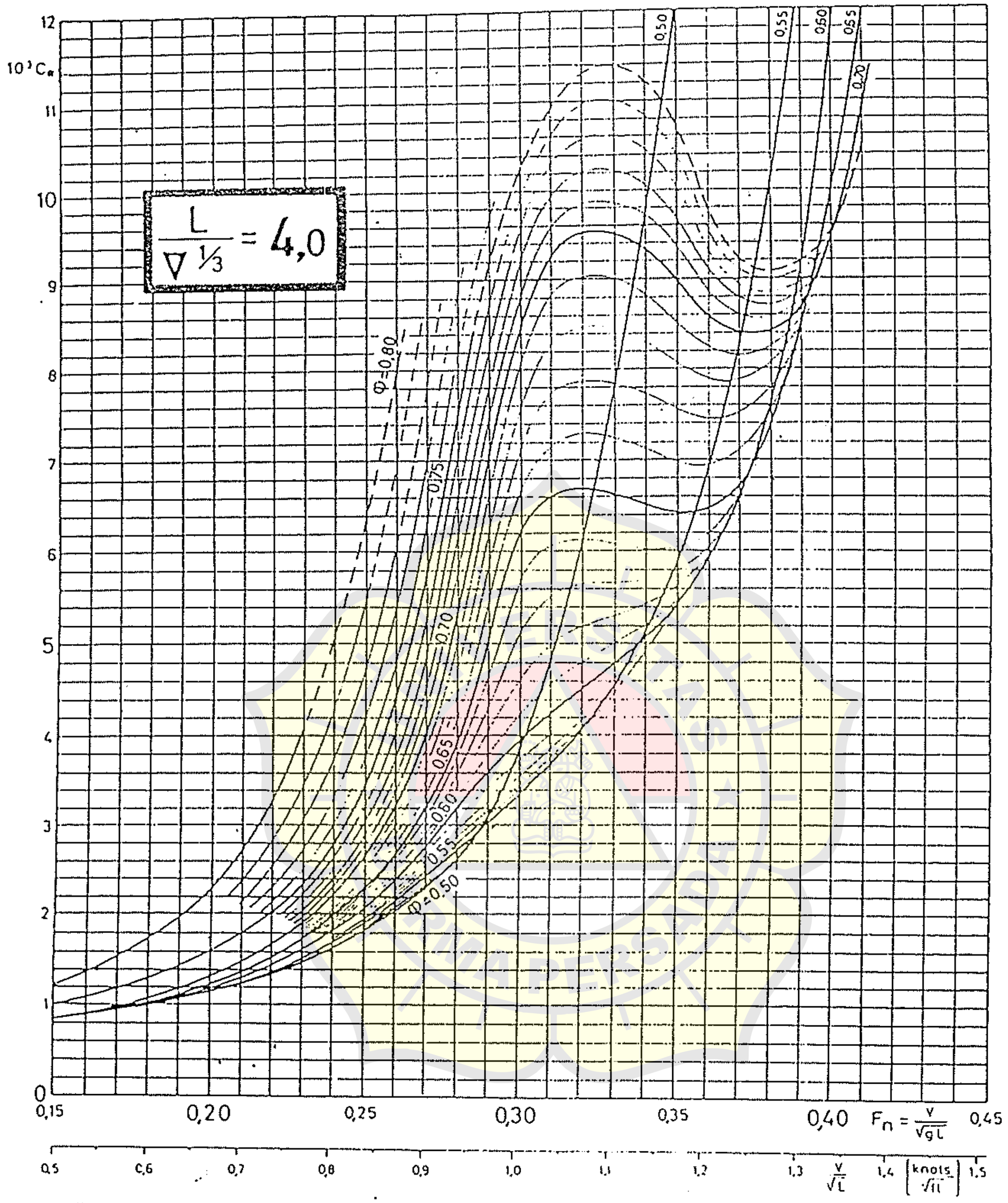
Berdasarkan hasil pemeriksaan materi pengui yang ada saat ini maka disarankan untuk mema rumus koreksi berikut ini :

$$10^3 C_R = 10^3 C_{R(B/T=2,5)} + 0{,}16(B/T - 2{,}5) \qquad (5.5.$$

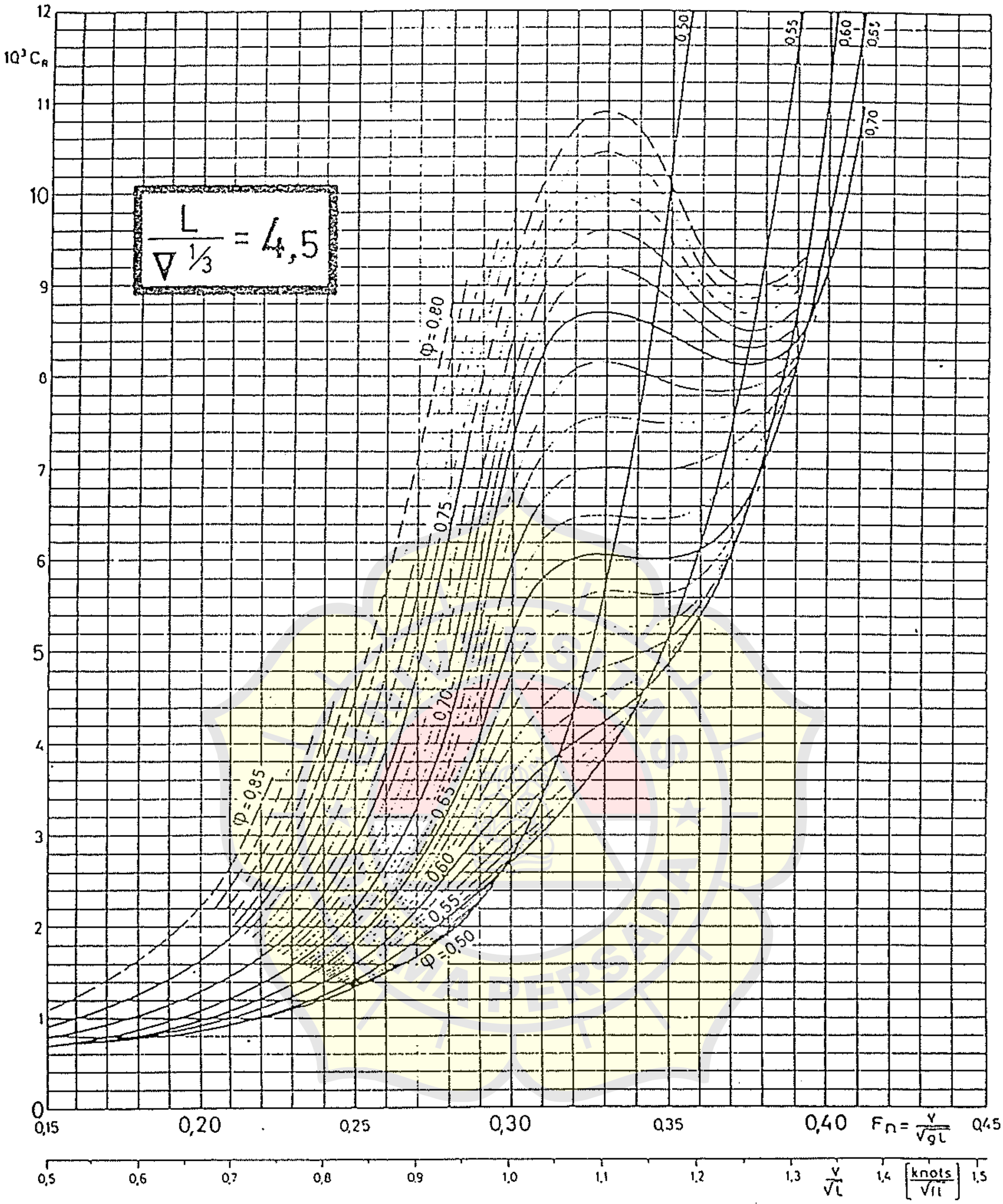
Koreksi ini dapat mempunyai harga yang negatif a positif.

LCB

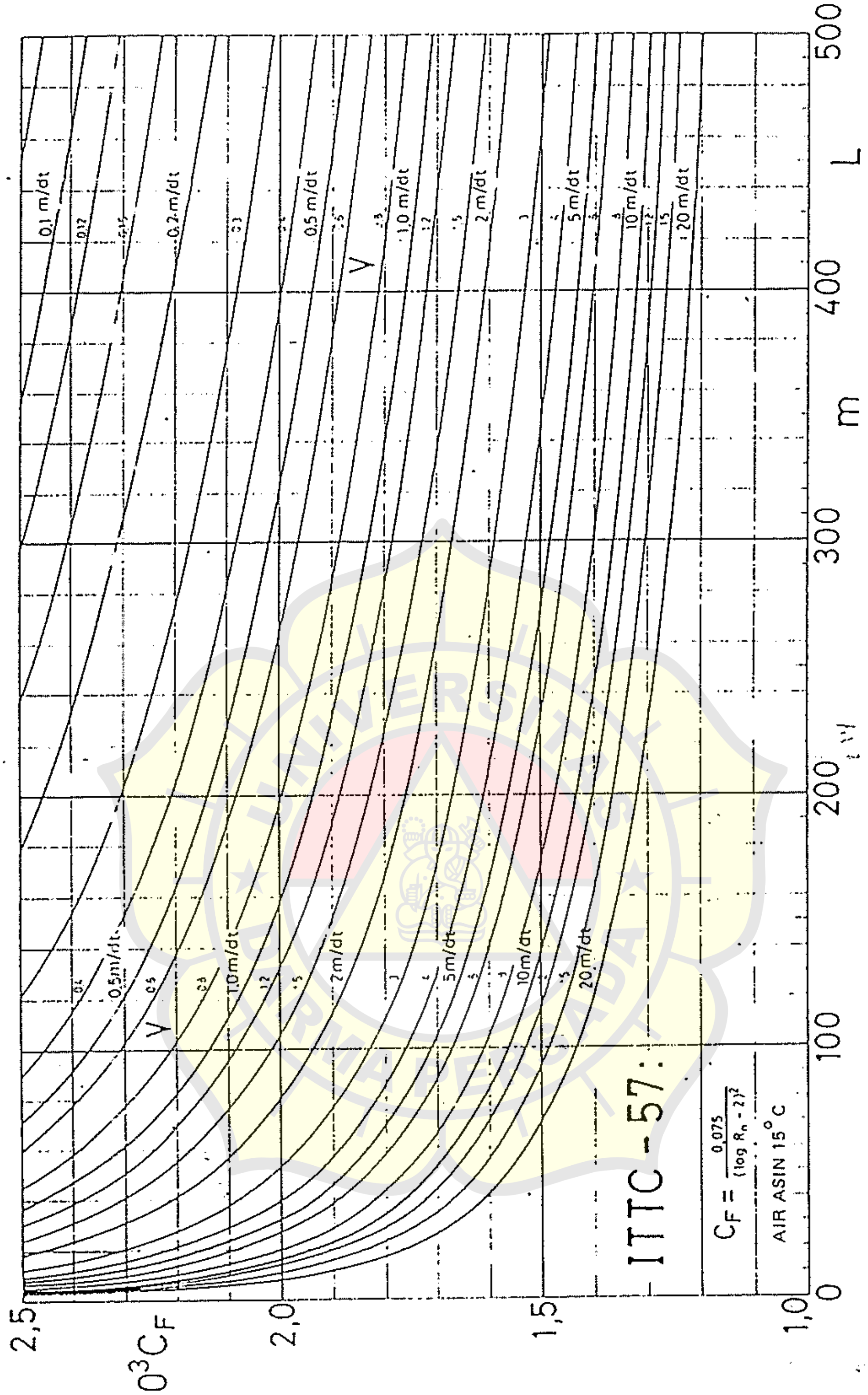
Semua kurva C_R tersebut dimaksudkan untuk ka yang letak titik benam longitudinalnya dekat den letak yang dewasa ini dipandang sebagai letak y terbaik yang memungkinkan. Letak LCB yang optim merupakan kuantitas yang masih agak meragukan, semua kepustakaan yang ada menunjukkan penda yang berbeda-beda sehingga memberikan gamba yang agak membingungkan. Namun demiki ketergantungan tahanan kapal pada LCB nampak j pada kecepatan yang tinggi. Sebagai upaya un mengatasi kerancuan tersebut maka semua inforn yang ada dikumpulkan dan diringkas pada Gb. 5.5 Namun ini harus dipandang sebagai LCB standar un metode itu saja.



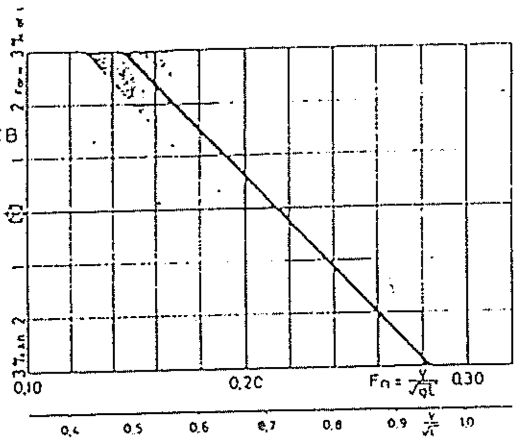
Gambar 5.5.5. Koefisien tahanan sisa terhadap rasio kecepatan - panjang untuk harga koefisien prismatik longitudinal yang berbeda-beda. $L/\nabla^{1/3} = 4,0$.



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



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

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

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

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

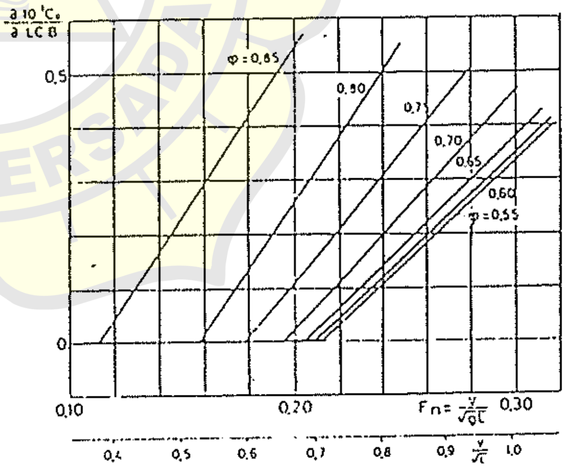
dengan faktor $\partial 10^3 C_R / \partial LCB$. Harga faktor ini dapat diperoleh dari Gb. 5.5.16, dan ini hanya berlaku untuk LCB yang berada di depan $LCB_{standar}$. Mengenai LCB yang berada di belakang $LCB_{standar}$, semua sumber yang ada mempunyai pendapat yang saling bertentangan. Namun demikian,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

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

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

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.

panjang perhitungan menurut definisi tadi. Dengan demikian maka penampang tengah kapal menurut definisi ini adalah pertengahan antara kedua garis tegak bantu (auxuliary perpendiculars), $AP_1 - FP_1$; banding-kan 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)

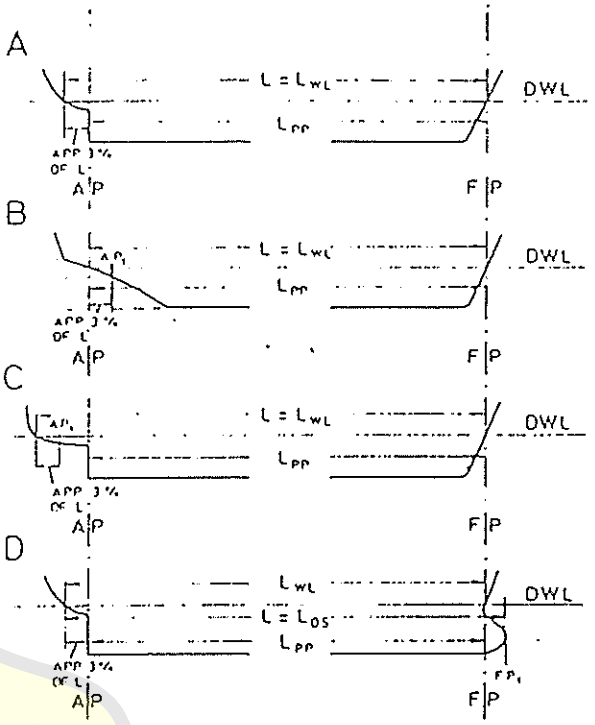
Sebagaimana disebutkan sebelumnya, kurva tahanan (yang diperoleh berdasarkan Gb. 5.5.5 – 5.5.13) dianggap berlaku untuk yang mempunyai bentuk "standar", yaitu penampangnya bukan yang benar-benar berbentuk U ataupun V. Karena itu, dalam menghitung daya efektif untuk perancangan awal umumnya tidak diperlukan 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-bènar baru dan catnyapun masih segar. Koefisien penambahan tahanan untuk korelasi model – kapal umumnya ditentukan sebesar $C_A = 0{,}0004$. Namun demikian, pengalaman lebih lanjut menunjukkan bahwa cara demikian itu tidak selalu benar. Karena itu, diusulkan koreksi untuk pengaruh kekasaran dan pengaruh sebagai berikut untuk kondisi pelayaran percobaan :

Untuk kapal dengan $L \leqq 100$ m, $10^3 C_A = 0{,}4$

$L \leqq 100$ m	$10^3 C_A =$	0,4
$= 150$ m	$=$	0,2
$= 200$ m	$=$	0
$= 250$ m	$=$	$-0{,}2$
$\geqq 300$ m	$=$	$-0{,}3$

(5.5.23)

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

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

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

ANGGOTA BADAN KAPAL

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

$$C_{F'} = C_F \frac{S_1}{S} \tag{5.5.25}$$

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

TAHANAN UDARA DAN TAHANAN KEMUDI

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

$$10^3 C_{AA} = 0{,}07 \tag{5.5.26}$$

Koreksi untuk tahanan kemudi mungkin sekitar

$$10^3 C_{AS} = 0{,}04 \tag{5.5.27}$$

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

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

KONDISI PELAYARAN DINAS

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

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

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

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.

2.3 Sifat-sifat Zat Cair

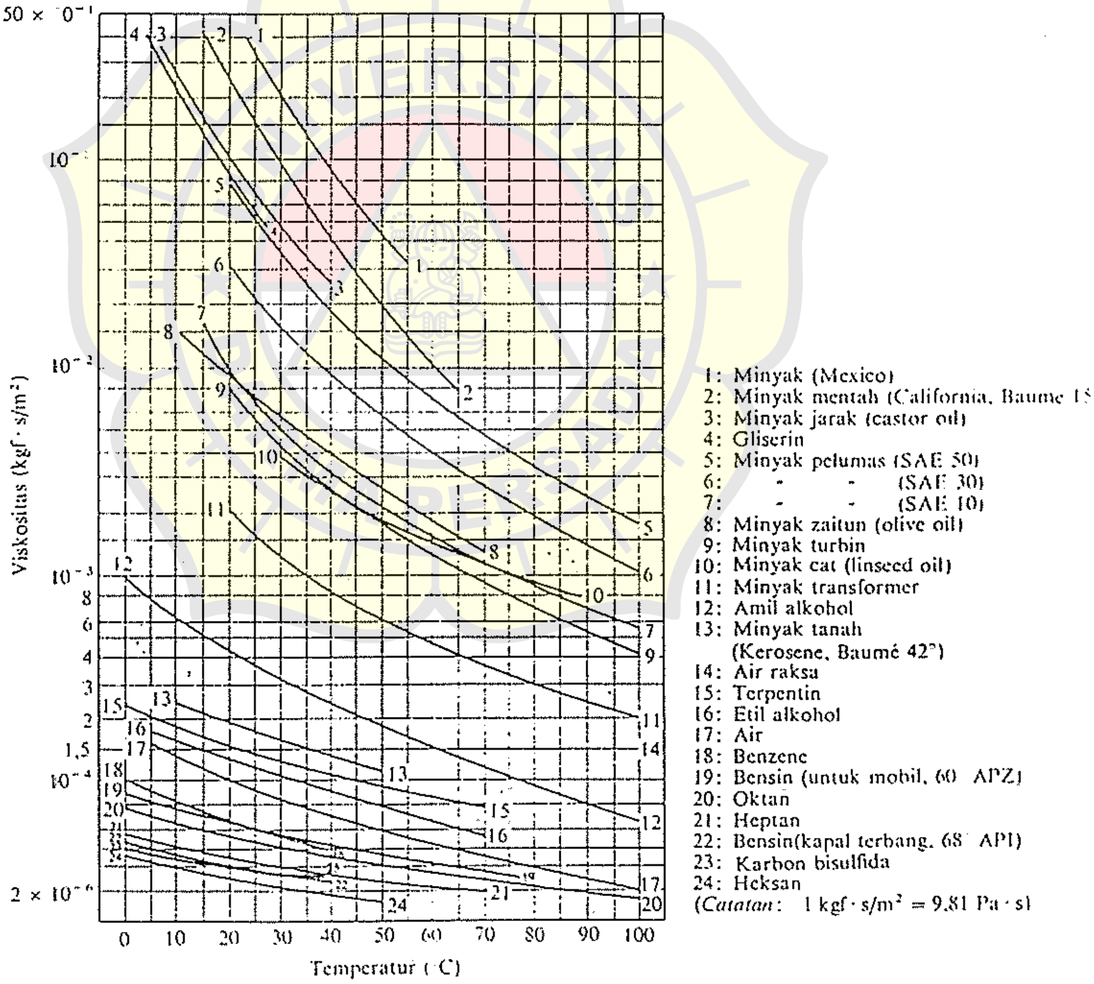
Performansi sebuah pompa dapat berubah-ubah tergantung pada karakteristik zat cair yang dialirkan. Jadi, dalam menentukan spesifikasi pompa, karakteristik ini harus diperhatikan. Sifat-sifat air dan beberapa fluida penting diberikan di bawah ini.

2.3.1 Sifat-sifat Air

Berat per satuan volume, viskositas kinematik, dan tekanan uap air untuk berbagai temperatur diberikan di dalam Tabel 2.12.

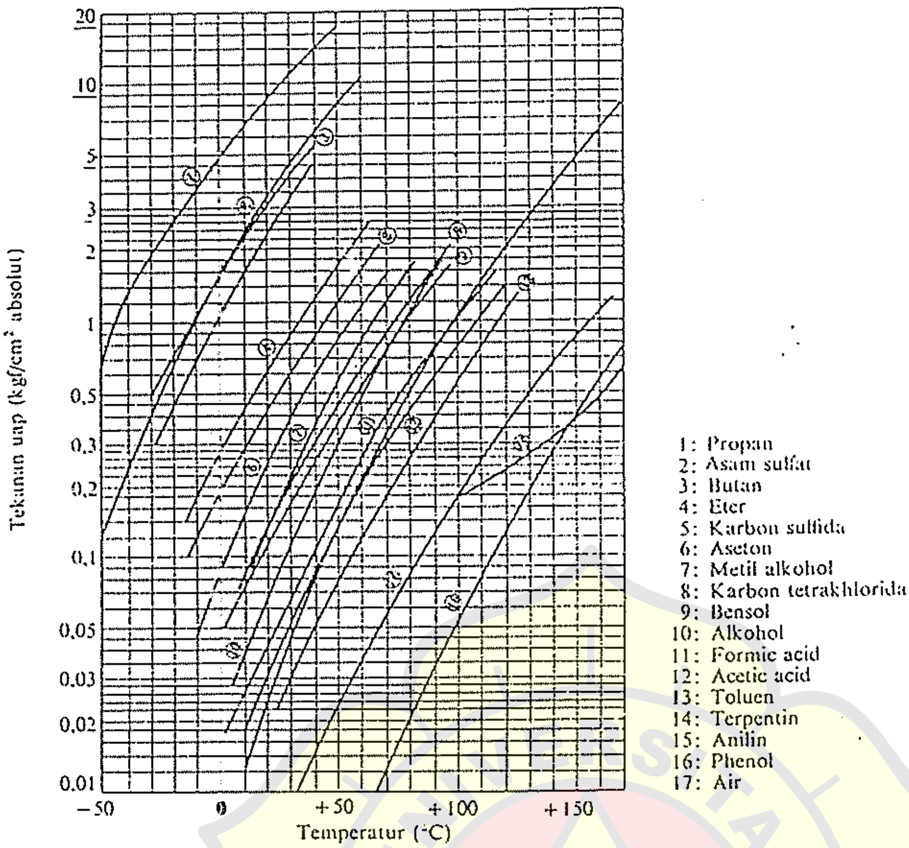
2.3.2 Sifat-sifat Zat Selain Air

Sifat-sifat fisik zat cair yang banyak dijumpai dalam teknik diberikan dalam Tabel 2.13 dan Gb. 2.1.



(a) Viskositas μ

Gb. 2.1 Sifat-sifat fisik berbagai zat cair.



(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:

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

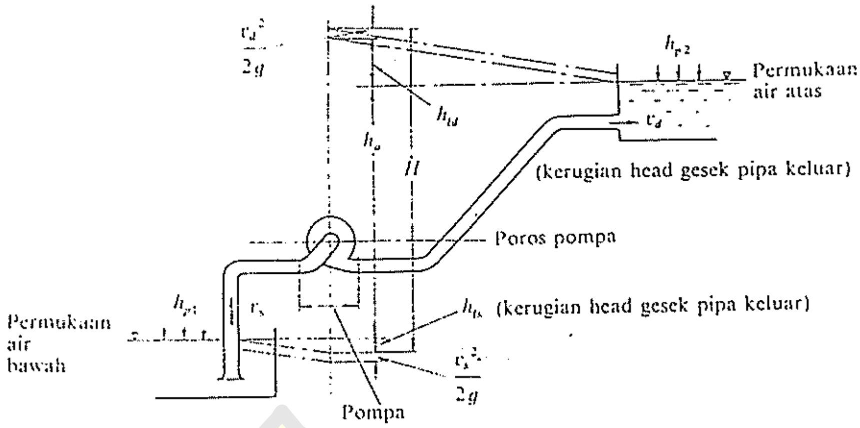
di mana H: Head total pompa (m)

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

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

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

h_l: Berbagai kerugian head di pipa, katup, belokan, sambungan, dll (m),



Gb. 2.2 Head pompa (1).

$$h_l = h_{ld} + h_{ls}$$

$v^2/2g$: Head kecepatan keluar (m)

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

Dalam hal pompa menerima energi dari aliran yang masuk ke sisi isapnya, seperti pada pompa penguat (pompa booster), maka head total pompa dapat dihitung dengan rumus berikut:

$$H = h_a + \Delta h_p + h_l + \frac{1}{2g}(v_d^2 - v_s^2) \tag{2.7}$$

di mana h_a: Perbedaan tinggi antara titik sebarang Ⓐ di pipa keluar, dan sembarang titik Ⓑ di pipa isap (m) (Lihat Gb. 2.3).

Δh_p: Perbedaan tekanan statis antara titik Ⓐ dan titik Ⓑ (m)

h_l: Berbagai kerugian head di pipa, katup, belokan dll, antara titik Ⓐ dan titik Ⓑ (m)

v_d: Kecepatan aliran rata-rata di titik Ⓐ (m/s)

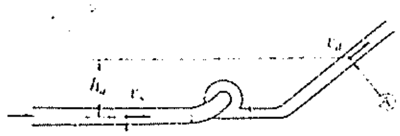
v_s: Kecepatan aliran rata-rata di titik Ⓑ (m/s)

Untuk pompa tegak yang tidak mempunyai pipa isap, $h_l = h_{ld}$.

Apabila permukaan air berubah-ubah dengan perbedaan besar, head statis total harus ditentukan dengan mempertimbangkan karakteristik pompa, besarnya selisih perubahan permukaan air, dan dasar yang dipakai untuk menentukan jumlah air yang harus dipompa.

Adapun hubungan antara tekanan dan head tekanan dapat diperoleh dari rumus berikut:

$$h_p = 10 \times \frac{p}{\gamma} \tag{2.8}$$



Gb. 2.3 Head pompa (2).

di mana h_p: Head tekanan (m)

p: Tekanan (kgf/cm^2)

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

Apabila tekanan diberikan dalam kPa, dapat dipakai rumus berikut:

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

di mana p': Tekanan (Pa)

ρ: Rapat masa (kg/l)

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

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

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

2.4.2 Head Kerugian

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

(1) Head kerugian gesek dalam pipa

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

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

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

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

C, p, q: Koefisien-koefisien

R: Jari-jari hidrolik (m)

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

S: Gradien hidrolik

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

h_f: Head kerugian gesek dalam pipa (m)

λ: Koefisien kerugian gesek

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

L: Panjang pipa (m)

D: Diameter dalam pipa (m)

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

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

di mana Re: Bilangan Reynolds (tak berdimensi)

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

D: Diameter dalam pipa (m)

ν: Viskositas kinematik zat cair (m^2/s)

Pada $Re < 2300$, aliran bersifat laminer.

Pada $Re > 4000$, aliran bersifat turbulen.

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

(I) Aliran laminer

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

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

(II) Aliran turbulen

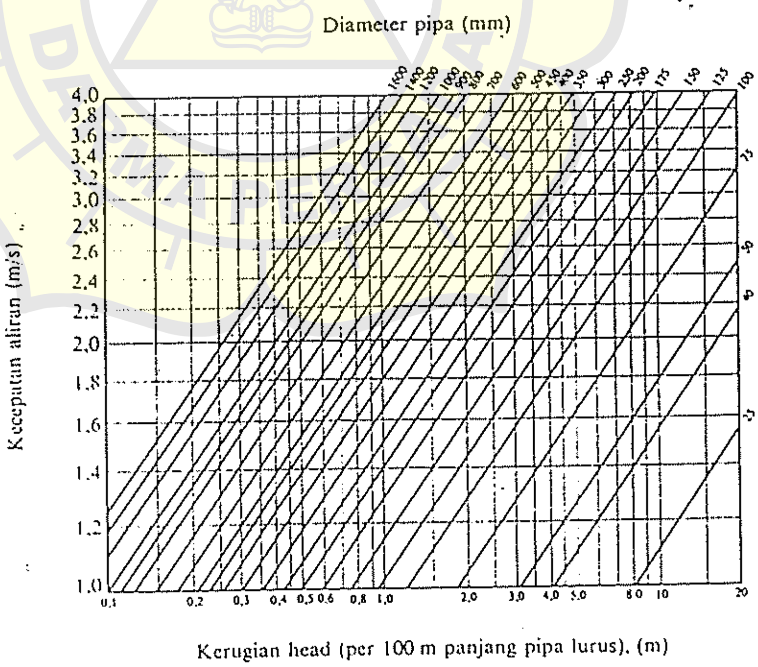
Untuk menghitung kerugian gesek dalam pipa pada aliran turbulen terdapat berbagai rumus empiris. Di bawah ini akan diberikan cara perhitungan dengan rumus Darcy dan Hazen-Williams.

1) Formula Darcy

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

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

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



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

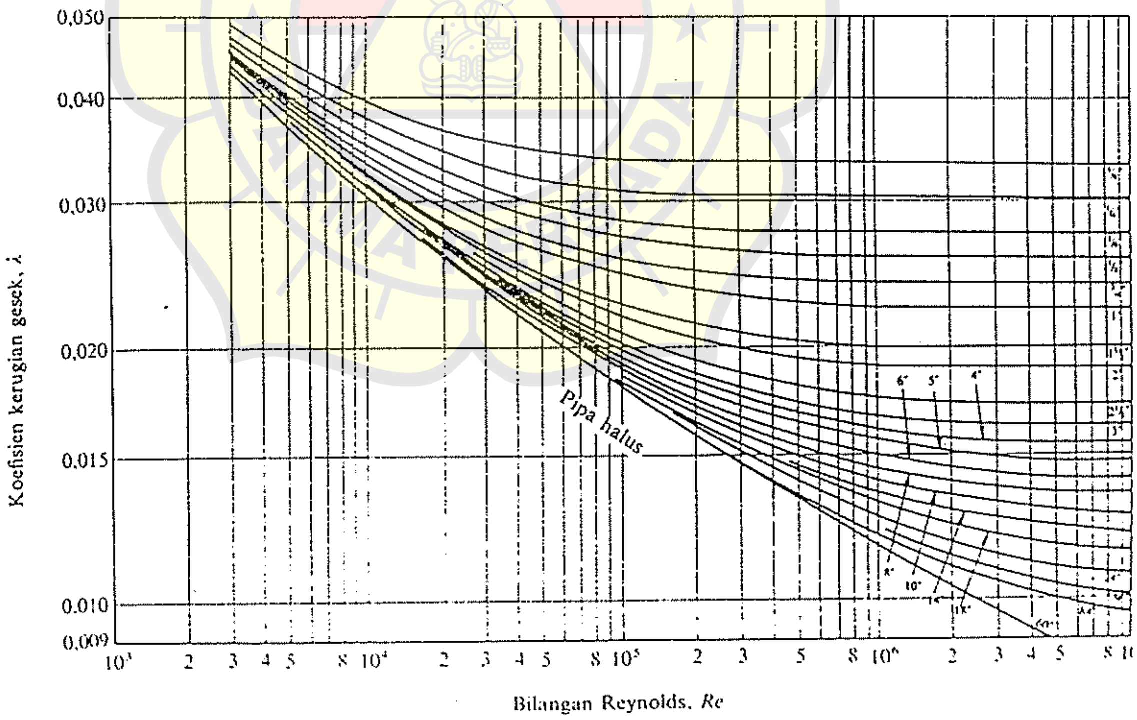
Tabel 2.21 Panjang pipa lurus ekivalen, L_f.

Nama peralatan pipa	Panjang pipa lurus ekivalen L_f	Nama peralatan pipa	Panjang pipa lurus ekivalen L_f
Belokan 45 (1"-3")	15-20 d	Meteran air: Jenis cakram jenis turbin Katup sorong: terbuka penuh " 3/4 " 1/2 " 1/4	135-400 D 200-300 D 0-7 D 10-40 D 100-200 D 800 D
Belokan 90 (jari-jari lengkung standar)	32 D		
Belokan (R/D = 3) 90 (R/D = 4)	24 D 10 D	Katup bola 1" – 2½" 3" – 6" 7" – 10"	 45 D 60 D 75 D
Belokan 180	75 D		
Sambungan silang Sambungan -T	50 D 40-80 D		
Meteran air jenis torak	600D D		

pipa lurus. Harga-harga L_f untuk berbagai peralatan pipa yang umum, diberikan dalam Tabel 2.21.

(5) Head kerugian gesek untuk zat cair istimewa

Untuk menghitung kerugian head pada pipa yang mengalirkan zat cair bukan air, dapat dilakukan sebagai berikut. Mula-mula harus dihitung bilangan Reynolds Re dari aliran. Kemudian kerugian head ditentukan dengan cara seperti pada air di mana koefisien kerugian gesek diambil untuk bilangan Reynolds yang bersangkutan.



Gb. 2.17 Bilangan Reynolds dan koefisien gesek.
(angka-angka menunjukkan diameter dalam pipa).

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:

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

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

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

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

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

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

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

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

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

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

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

If a pressure-reducing valve is fitted, which red[uces] the pressure $p_{e,perm}$ to the starting pressure P_A, [t]he value of "c" shown in Fig. 2.14 is to be use[d].

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 reversible engines

For each non-reversible main engine driving a [con]trollable pitch propeller or where starting wit[hout] torque resistance is possible the calculated sta[rting] air supply may be reduced to 0,5 . J though no[t less] than that needed for six start-up operations.

M. Exhaust Gas Lines

1. Pipe layout

1.1 Engine exhaust gas pipes are to be installed separately from each other, taking the structural fire protection into account. Other designs are to be submitted for approval. The same applies to boiler exhaust gas pipes.

1.2 Account is to be taken of thermal expansion when laying out and suspending the lines.

1.3 Where exhaust gas lines discharge near water level, provisions are to be taken to prevent water from entering the engines.

2. Silencers

2.1 Engine exhaust pipes are to be fitted with effective silencers.

2.2 Silencers are to be provided with an inspection opening.

3. Water drains

Exhaust lines and silencers are to be provided with suitable drains of adequate size.

4. Insulation

4.1 Exhaust gas lines, silencers and exhaust gas boilers are to be effectively insulated to prevent the ignition of combustible materials on them.

4.2 Insulating materials must be incombustible.

4.3 Exhaust gas lines inside engine rooms are to be provided with a metal sheathing or other approved type of hard sheathing.

5. For special Rules for tankers refer to Section 15, B.9.3.

N. Bilge Systems

1. Bilge lines

1.1 Layout of bilge lines

1.1.1 Bilge lines and bilge suction are to be so arranged that the bilges can be completely pumped even under disadvantageous trim conditions.

1.1.2 Bilge suction are normally to be located on both sides of the ship. For compartments located fore and aft in the ship, one bilge suction may be considered sufficient provided that it is capable of completely draining the relevant compartment.

1.1.3 Spaces located forward of the collision bulkhead and aft of the stern tube bulkhead and not connected to the general bilge system are to be drained by other suitable means of adequate capacity.

1.1.4 The required pipe thicknesses of bilge lines are to be in accordance with Table 11.4.

1.2 Pipes laid through tanks

1.2.1 Bilge pipes may not be led through tanks for lubricating oil, thermal oil, drinking water or feedwater.

1.2.2 Where bilge pipes are led through fuel tanks located above the double bottom and terminate in spaces which are not accessible during the voyage, an additional non-return valve is to be fitted in the bilge pipe where the pipe from the suction enters the fuel tank.

1.3 Bilge suction and strums

1.3.1 Bilge suction are to be so arranged as not to impede the cleaning of bilges and bilge wells. They are to be fitted with easily detachable, corrosion-resistant strums.

1.3.2 Emergency bilge suction are to be arranged in such a manner that they are accessible, with free flow and at a suitable distance from the tank top or the ship's bottom.

1.3.3 For the size and design of bilge wells see Rules for Hull Construction, Volume II, Section 8. B.6.2.

1.4 Bilge valves

1.4.1 Valves in connecting pipes between the bilge and the seawater and ballast water system, as well as between the bilge connections of different compartments, are to be so arranged that even in the event of faulty operation or intermediate positions of the valves, penetration of seawater through the bilge system will be safely prevented.

1.4.2 Bilge discharge pipes are to be fitted with shutoff valves at the ship's side.

1.4.3 Bilge valves are to be arranged so as to be always accessible irrespective of the ballast and loading condition of the ship.

1.5 Reverse-flow protection

1.5.1 A screw-down non-return valve is recognized as reverse-flow protection.

Table 28.1

Anchors, Chain Cables and Ropes of Fishing Vessels

No for Reg	Equipment Numeral Z	Stockless Bower Anchors		Stud Link Chain Cables 1) for Bower Anchors				Recommended Mooring Ropes		
		Number	Weight p. Anchor	Total Length	diameter d1	diameter d2	diameter d3	Number	Length	Br. Load
			kg	m	mm	mm	mm		m	kN
1	2	3	4	6	7	8	9	14	15	16
101	up to 30	2	70	137.5	11	11	11	2	40	25
102	30 - 40	2	80	165	11	11	11	2	50	30
103	40 - 50	2	100	192.5	11	11	11	2	60	30
104	50 - 60	2	120	192.5	12.5	12.5	12.5	2	60	30
105	60 - 70	2	140	192.5	12.5	12.5	12.5	2	80	30
106	70 - 80	2	160	220	14	12.5	12.5	2	100	35
107	80 - 90	2	180	220	14	12.5	12.5	2	100	35
108	90 - 100	2	210	220	16	14	14	2	110	35
109	100 - 110	2	240	220	16	14	14	2	110	40
110	110 - 120	2	270	247.5	17.5	16	16	2	110	40
111	120 - 130	2	300	247.5	17.5	16	16	2	110	45
112	130 - 140	2	340	275	19	17.5	17.5	2	120	45
113	140 - 150	2	390	275	19	17.5	17.5	2	120	50
114	150 - 175	2	480	275	22	19	19	2	120	55
115	175 - 205	2	570	302.5	24	20.5	20.5	2	120	60
116	205 - 240	2	660	302.5	26	22	20.5	2	120	65
117	240 - 280	2	780	330	28	24	22	3	120	70
118	280 - 320	2	900	357.5	30	26	24	3	140	80
119	320 - 360	2	1020	357.5	32	28	24	3	140	85
120	360 - 400	2	1140	385	34	30	26	3	140	95
121	400 - 450	2	1290	385	36	32	28	3	140	100
122	450 - 500	2	1440	412.5	38	34	30	3	140	110
123	500 - 550	2	1590	412.5	40	34	30	4	160	120
124	550 - 600	2	1740	440	42	36	32	4	160	130
125	600 - 660	2	1920	440	44	38	34	4	160	145
126	660 - 720	2	2100	440	46	40	36	4	160	160

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 o the rudder horn may be included into the rudder are: 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:

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

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

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

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

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

5. Definitions

C_R = rudder force in [N]

Q_R = rudder torque in [Nm]

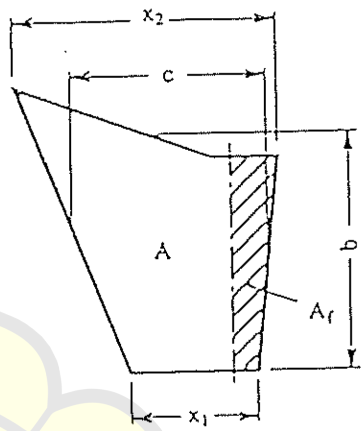
A = total movable area of the rudder in [m^2]
For nozzle rudders, A is not to be taken less than 1,35 times the projected area of the nozzle;

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

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

b = mean height of rudder area in [m]

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



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

Fig. 14.1

Λ = aspect ratio of rudder area A_t

Λ = b^2/A_t

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

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

v_a = astern speed of ship in [kn]; if the astern speed $v_a \leq 0,4 \cdot v_0$ 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, α = 0,4 may be used for the ahead condition

k_b = balance factor as follows:

k_b = A_f/A

k_b = 0,08 for unbalanced rudders

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

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

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

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

The resulting force of each part may be taken as:

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

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

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

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

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

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

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

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

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

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

$C_1 = A_1/b_1$

$C_2 = A_2/b_2$

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

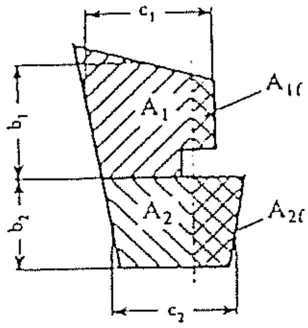


Fig. 14.2

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

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

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

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

for ahead condition

The greater value is to be taken.

C. Scantlings of the Rudder Stock

1. Rudder stock diameter

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

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

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

The related torsional stress is:

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

k_r see A.4.2.

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

1.3 In case of mechanical steering gear the diameter of the rudder stock in its upper part which is only intended for transmission of the torsional moment from the auxiliary steering gear may be 0.9 D_t. The length of the edge of the quadrangle for the auxiliary tiller must not be less than 0,77 D_t and the height not less than 0,8 D_t.

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

2. Strengthening of rudder stock

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

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

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

Bending stress:

$$\sigma_b = \frac{10.2 \cdot M_b}{D_1^3} \quad [N/mm^2]$$

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

Torsional stress:

$$\tau = \frac{5.1 \cdot Q_R}{D_1^3} \quad [N/mm^2]$$

D_1 = increased rudder stock diameter in [cm]

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

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

Q_R see B.1.2 and B.2.2 - 2.3

D_t see 1.1.

Guidance

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

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

Rumus TAYLOR

Untuk Wake fraction : Kapal berbaling2 tunggal;

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

Kapal berbaling2 ganda;

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

Untuk Thrust deduction factor :

Kapal berbaling2 tunggal; $t \approx w$

Kapal berbaling2 ganda; $t \quad w$

dimana harga k adalah sebagai berikut :

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

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

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

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

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

Adapun coefficient prismatic C_p tidak dipakai sebagai salah satu parameter berhu- bung percobaan

Additions to the volume

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

Diesel oil

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

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

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

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

additions see fuel oil!

Lubrication oil

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

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

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

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

Fresh water

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

additions to the tank volume: 3 ... 4% for special coatings in case of fresh wate[r]

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

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

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

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

In general

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

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

<u>Consumables and tanks</u>

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

- consumables
- provisions
- ballast.

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

- fuel oil

$$w_{fuel\ oil}\ [t] = P_{Bmc} \cdot b_{mc} + P_{ae} \cdot b_{ae} \cdot \frac{S}{V_{serv}} \cdot 10$$

$$\cdot [1.3 \ldots 1.$$

last brackets for reserve:
- fuel rests in tanks
- seaway
- wind
- waiting time
(- according to owner's desire).

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:

$$\eta_{ad} = \frac{L_{ad}}{L_s} \qquad (2.20)$$

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

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

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 tekanan dan temperatur isap

k: c_p/c_v

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

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

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

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

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

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

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

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

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

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

V_r = volume of the room, cu m

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

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

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

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

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

t_r = given temperature of the room, °C

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

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

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

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

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

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

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

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

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

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

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

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

Table 42

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

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

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

whence

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

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

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

$$H_{t\infty} = \frac{1}{g}(c_{2u}u_2 - c_{1u}u_1) = \frac{1{,}000}{g}\frac{\gamma_{air}}{\gamma_{wat}}(c_{2u}u_2 - c_{1u}u_1) =$$
$$= \rho(c_{2u}u_2 - c_{1u}u_1)\ \mathrm{mmH_2O} \qquad (277)$$

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

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

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

Upon radial entry of the air onto the fan impeller vanes

$$H_{t\infty} = \rho c_{2u}u_2\ \mathrm{mmH_2O}$$

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

$$H = H_{t\infty}\sigma\eta_h = \sigma\rho c_{2u}u_2\eta_h = \sigma\rho\frac{c_{2u}}{u_2}u_2u_2\eta_h =$$
$$= \sigma\rho\varphi_h u_2^2\eta_h = \rho\psi_h u_2^2\ \mathrm{mmH_2O} \qquad (278)$$

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

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

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

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

Table 43

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

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

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

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

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

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

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

where ΔH = loss of head in the fan.

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

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

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

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

b_2 = width of the impeller at air outlet

D_2 = impeller diameter at air outlet

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

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

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

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

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

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

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

2-2. Design and Selection of Fans

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

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

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

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

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

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

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

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

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

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

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

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

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

G_a = weight of the anchor, kg

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

L_a = length of the suspended cable, m

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

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

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

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

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

In hoisting one anchor

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

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

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

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

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

---

* In breaking away one anchor from the bottom

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

maximum pressure, p_f kg per sq m, then the amount of liquid pumped is

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

This equation can be solved for V_e and V_f:

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

and

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

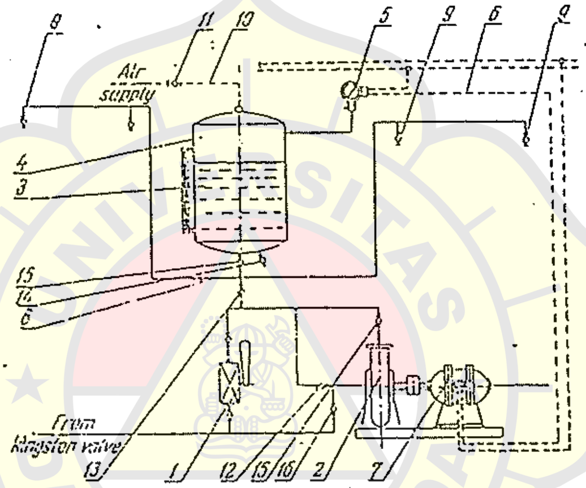


Fig. 169.

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

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

Therefore the minimum and maximum volumes of the air are

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

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

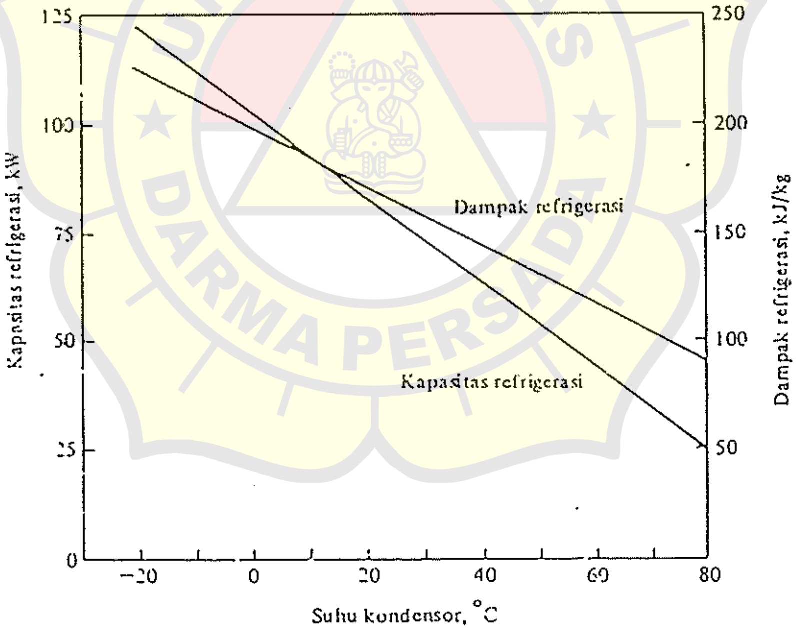
$$V_t = V_0 + V_e = V_0 + \frac{D p_f}{6 (p_f - p_e)}$$

Such tanks may also be used in the drinking and washing water systems.

si volumetrik yang mempengaruhi laju alir massa, yang menunjukkan suatu penurunan akibat naiknya suhu kondensor. Gambar 11-10 menunjukkan penurunan tersebut yang progresif. Kapasitas refrigerasi adalah hasil kali antara dampak refrigerasi dan laju aliran massa, yang keduanya akan turun bila suhu kondensor naik. Jadi kapasitas refrigerasi turun agak lebih cepat karena naiknya suhu kondensor.

Karakteristik yang penting lagi adalah daya – yang diperlihatkan dalam Gambar 11-11. Daya kompresor adalah hasil perkalian antara kerja kompresi yang bersatuan kilojoule per-kilogram dan laju alir massa. Bila suhu kondensor naik, maka kerja kompresi dan laju alir massa menurun, sehingga daya naik mencapai puncak dan kemudian mulai turun. Sifat yang sama dengan daya ini, yaitu sebagai fungsi dari suhu evaporator, ditunjukkan dalam Gambar 11-6.

Beberapa penjelasan tentang arti dan sifat-sifat yang terdapat di dalam Gambar 11-9 hingga 11-11 adalah sebagai berikut: pencapaian puncak-puncak daya dapat terjadi dalam kompresor-kompresor nyata seperti juga pada kompresor ideal, tetapi hanya terjadi bila dilakukan pemompaan dari suhu-suhu rendah evaporator. Kompresi satu tingkat dari suhu penguapan –20°C hingga suhu pengembunan 60°C yang menghasilkan puncak seperti pada Gambar 11-11, tidaklah umum. Dengan perbedaan suhu yang lebih sedikit antara kondensor dan evaporator, diperkirakan bila suhu kondensor naik, akan ada kenaikan daya pada kompresor, walaupun kenaikan tersebut mungkin hanya sedikit. Kapasitas refrigerasi selalu turun bila suhu kondensor naik. Karakteristik lain yang penting, tidak digambarkan dalam grafik, adalah *koefisien prestasi* (coefficient of performance), yang turun secara monoton bila suhu kondensor naik.



Gambar 11-10 Dampak refrigerasi dan kapasitas refrigerasi untuk kompresor ideal dengan refrigeran 22, volume sisa 4,5 persen, laju volume langkah 50 L/det, dan suhu evaporator –20°C.

Bertitik tolak dari daya dan efisiensi, diinginkan suhu kondensor yang rendah, jadi kondensor tersebut harus menggunakan udara atau air yang terdingin yang tersedia, mengalir secara maksimum dan ekonomis, serta permukaannya harus dijaga tetap bersih. Udara atau gas-gas yang tak dapat mengembun di dalam kondensor juga mengakibatkan tingginya tekanan kondensor tersebut.

CHAPTER ELEVEN

Propulsion Machinery

11.1. *Relation between engine and propeller power*

When a propeller is rotated behind a vessel it absorbs power in the form of torque and rotational speed, and converts it, more or less efficiently, into thrust and linear speed. The relationship between the power absorbed and the revolutions per minute approximates to a law of the form $P \propto N^3$, where P is the power and N is the rotational speed. Some authorities prefer to use $P \propto N^{2.8}$ as being a little more accurate.

The main engine provides this power, again in the form of torque and rotational speed, but for most applications of medium and high speed engines the torque is lower and the speed higher than that which is suitable for the propeller and a speed reduction gear must be included between engine output and propeller shaft. Some means of reversing the thrust of the propeller is needed which may be provided by direct reversing of the engine, or by a reversing gear train in the gearbox or by a variable pitch propeller. Each of these means is in present day use.

11.2. *Limits of engine power*

In Chapter 2 the limits to engine output were considered and it was seen that each design of engine has limits of b.m.e.p. and rev/min which cannot be safely exceeded. It is customary for engine manufacturers to ensure that these limits are not overstepped by fitting limit stops to the fuel pump and governor. A stop restricting the movement of the fuel pump rack (or corresponding component) in the direction of increasing fuel will determine the maximum amount of fuel injected into the cylinder per cycle and so restrict the b.m.e.p. to a maximum value. For a given engine design this corresponds to a limiting maximum torque. The setting of this limit stop will be carried out on the test berth at the full rated rev/min and usually corresponds to the maximum continuous rating of the engine. Changes in the volumetric efficiency of fuel injection equipment with rev/min may result in a variation in the limiting torque produced down the speed range but for the majority of engines it may be assumed that the maximum output characteristic is one of constant torque at all revolutions, see Figure 11–1. At the high end of the speed range a limit is set by restricting the governor speed setting mechanism to give the maximum rated rev/min when carrying the maximum continuous rated b.m.e.p. If

value at which the running becomes irregular and unstable. The idling speed is set just a little above this point. In the case of turbo-charged engines insufficient airflow may prevent high loads being carried at low rev/min. These conditions result in limits at the low speed end of the range which are again shown in Figure 11–1. These same limits can also be displayed on power and rev/min co-ordinates as shown in Figure 11–2.

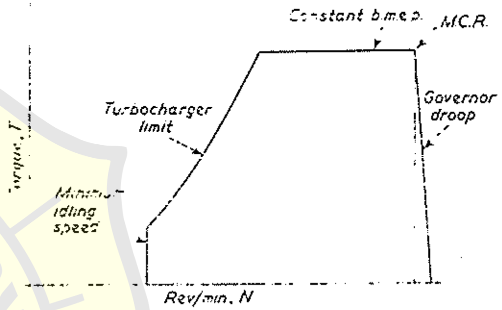


FIG. 11–1—*Engine output limits.*

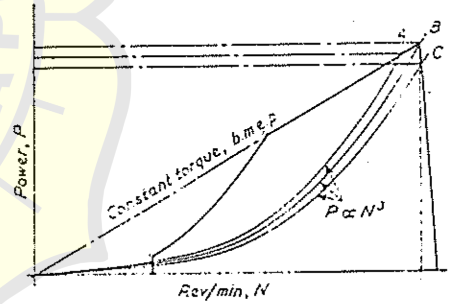


FIG. 11–2—*Engine output and propeller demand.*

11.3. *Power demand of a propeller*

As the ratio of engine rev/min to propeller rev/min will have a fixed value the propeller law curve may be plotted on the same co-ordinates. Making allowance for the power losses in the gearbox this will give a curve of $P \propto N^3$ form as shown at A. B and C in Figure 11–2. If the engine, propeller and gearbox are correctly matched the propeller curve will pass through the maximum continuous rating point of the engine, as shown by B. A propeller that is too small will reach full rev/min at less than full torque as at C, and a propeller that is too large will absorb the full torque of the engine before full rev/min is attained as at A. In either of the cases A and C the full rated power will not be available and the intended speed of the vessel may not be achieved in consequence. This illustrates the possible or not depends on the criteria which are considered to limit the rating for that particular design of engine.

If the resistance to forward motion of a vessel having a fixed pitch propeller is increased, for example by using it to tow another vessel, the propeller law curve will become steeper although still of $P \propto N^3$ form as shown in Figure 11–3. A heavier tow will steepen it further, corresponding

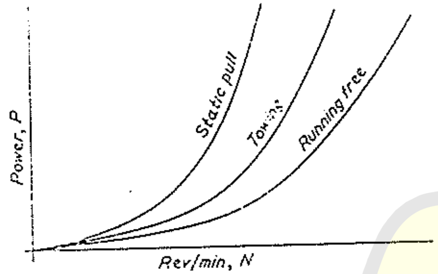


FIG. 11–3—*Effect of increasing the resistance of the vessel.*

to a lower forward speed of the vessel until the most sharply rising curve is produced by the static pull condition. Vessels designed to tow must take these circumstances into account. The machinery and propellers of docking tugs are usually designed so that the static pull curve passes through the maximum continuous rating point; this results in the highest possible value for the static pull but precludes the use of the maximum installed horse-power once the tow is under way. A tug intended for continuous towing will have machinery designed to match the propeller curve at the intended towing speed and a similar design procedure will be followed for a trawler which tows gear at a relatively high speed.

11.4. *Variable pitch propellers*

A variable pitch propeller has blades which can be moved to take up various pitch angles. Any one setting results in a particular propeller law curve for a vessel of given resistance and a whole family of propeller law curves can be produced for such a vessel by altering the blades to different settings. A similar set can be formed for any given increase in resistance of the vessel. It is therefore possible, within the design limits of the propeller, to choose settings which would match the maximum con-tinuous rating of the engine under any condition from static pull to running free, and so utilize the maximum continuous rated power of the engine to obtain the highest tow rope pull and the highest speed attainable in the circumstances. Compared with a fixed pitch propeller the variable pitch propeller makes better use of the power available from the engine over a wide range of diverse conditions. On the other hand it has a disadvantage in that the efficiency is not so high as that of the fixed pitch propeller

A variable pitch propeller can also be used to improve the economy of operation of a vessel of constant resistance but which is called upon to operate at different forward speed during its service. If lines of constant fuel consumption of the kind shown in Figure 2–8, are added to the limits of Figure 11–2 their appearance is as in Figure 11–4. The family of curves

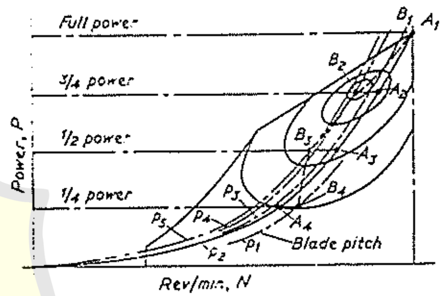


FIG. 11–4—*Optimizing fuel consumption by controllable pitch propeller.*

for a variable pitch propeller and vessel of constant resistance lies across these lines of constant fuel consumption and by choosing combinations of pitch angle and rev/min, the operating line for various speeds can be made to pass through the regions of lowest fuel consumption as is shown. It must, of course, be remembered that alteration of propeller pitch slightly affects propeller efficiency and therefore the relative location of load points and minimum fuel consumption points must be kept in mind in establishing the best relation between optimum engine thermal efficiency and optimum propeller efficiency. In Figure 11–4, the fuel consumption points at full load, three quarter load, half load and quarter load positions with a fixed pitch propeller are indicated at A1, A2, A3 and A4, whilst the improved fuel consumption points obtainable with variable pitch propeller are indicated at B1, B2, B3 and B4, on the dotted line passing through the isofuel consumption loops. Bearing in mind the change in propeller efficiency it may be that in practice optimum performance will be attained between these two settings. Such settings of pitch angle corresponding to rev/min may be selected automatically by appropriate control gear once they have been determined.

11.5. *Two engines geared to one propeller*

By gearing two or more medium speed engines together to drive a single propeller a high powered installation can be designed to occupy a small space and to have a low weight. Benefits in economy due to the use of slower turning higher efficiency propellers and the ability to use one engine only for low speed operation can be justifiably claimed for the arrangement as also can increased reliability and availability of the ship.

When two engines of equal power are geared to one propeller the

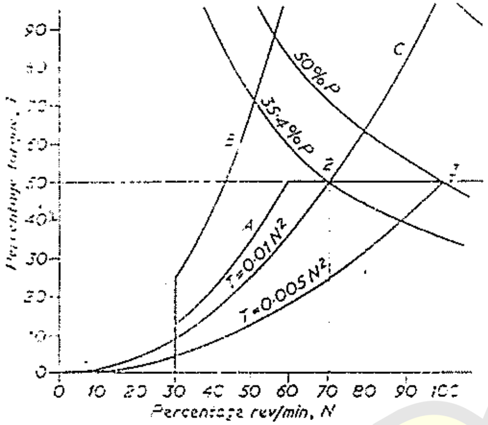


FIG. 11–5—*Torque—speed curves for two engines driving one propeller.*

propeller matched to absorb the full power (service rating) of the two engines together at full rev/min corresponding to point 1. The vessel may be propelled by one engine alone developing its full torque (equal to half the total torque) as at point 2. The revolutions will be reduced to 0·707 of full rev/min and the power available to 0·707 × 0·5 = 0·353 of full power giving the ship 0·707 of its full speed.

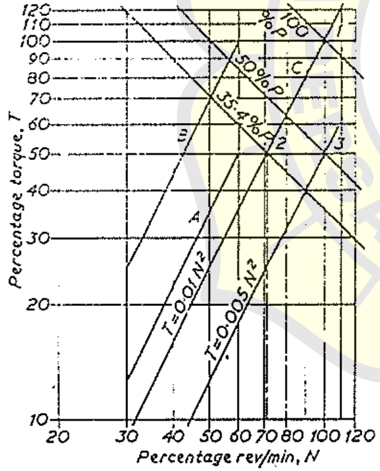


FIG. 11–6—*Torque—speed curves for two engines driving one propeller. log–log scales.*

If a controllable pitch propeller is fitted the pitch can be reduced when running on one engine permitting it to develop its full rev/min as well as its full torque as at point 3. The power is 0·5 of total full power and the vessel's speed is 0·8 of full speed.

... using percentage scales. Curves of constant power and power proportional to (rev/min)^n become groups of parallel straight lines which are easily located. The curves and points in Figure 11–6 correspond to those in Figure 11–5 and are lettered and numbered accordingly.

Vessels engaged on routes where reduced speed is required for a significant proportion of the running time can benefit economically from the installation of twin geared engines. As can be seen from Figure 11–7

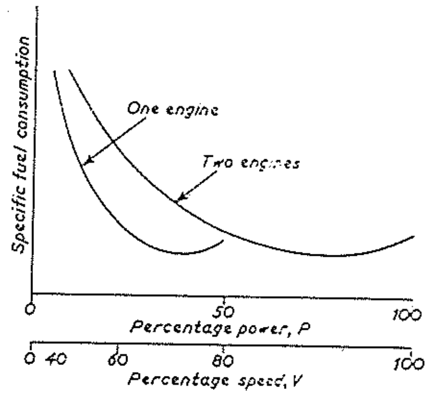


FIG. 11–7—*Fuel consumption, power and speed for two engines driving one controllable pitch propeller.*

which is drawn for twin engines and a controllable pitch propeller, the rise in specific fuel consumption at reduced power and speed can be countered by changing to single engine propulsion at speeds below 80% of full speed.

Installations may be designed with two engines of unequal power geared together or with three or four engines. The reduction in the speed of the vessel as a result of part of the total power not being used is easily calculated, as follows.

Let full power, speed, torque and rev/min be represented by P_f, V_f, T_f and N_f respectively,

and let the available power, speed, torque and rev/min be represented by Pa, Va, Ta and Na respectively.

The propeller will absorb the available torque and for a fixed pitch propeller the rev/min will be reduced.

Thus $$\frac{Ta}{T_f} = \left(\frac{Na}{N_f}\right)^2$$

$$\therefore \quad \frac{Pa}{P_f} = \frac{Ta\,Na}{T_f\,N_f} = \left(\frac{Ta}{T_f}\right)^{3/2} \longrightarrow = \frac{Ta}{T_f}\cdot\left(\frac{Ta}{T_f}\right)^{\frac{1}{2}} = ($$

and $$\frac{Va}{V_f} = \left(\frac{Pa}{P_f}\right)^{1/3} = \left(\frac{Ta}{T_f}\right)^{1/2}$$

$$\therefore \quad \frac{\cdots}{P_f} = \frac{\cdots}{T_f N_f} = \frac{\cdots}{T_f}$$

and $$\frac{Ta}{Tf} = \left(\frac{Pa}{P_f}\right)^{1/3} = \left(\frac{Ta}{T_f}\right)^{1/3}$$

11.6. *Speed reduction gearboxes*

Medium and high speed engines, because of their crankshaft revolutions being high compared with propeller revolutions, are almost always installed with speed reduction gearboxes. A few high speed craft, naval applications and cross channel ferries, are the exceptions in which the engines are direct coupled to the propeller. In most cases the engines are derated, running at rev/min below their maximum rated speed with a corresponding reduction in horsepower, although often not a proportionate reduction as there may be an increase in b.m.e.p. above that which can be carried at the maximum continuous rating, as will be appreciated from a study of Figure 2–8.

Once a speed reduction gear is to be included in the propulsion machinery installation the revolutions of the propeller may be chosen from propulsion considerations alone unfettered by engine requirements, its size being restricted only by the aperture and hull form. The efficiency of such a propeller may be considerably higher than is usually the case with propellers for direct coupled low speed engines, the improvement outweighing the losses in the gearing.

The speed ratio of reduction gears is generally of the order of 3:1 to 4:1 for single engines of about 1,000 hp and below. For installations of higher power, particularly for twin engines (or multi-engines) driving a single propeller and ships of large displacement, ratios of 4:1 to 6:1 are used.

11.7. *Gears for two or more engines and single propeller*

Twin engines of equal power are the most popular form of multi-engined geared installations. The basic arrangement is seen in Figure 11–8(a), using either fluid couplings or mechanical clutch-couplings between the engines and the gears. If mechanical clutch-couplings are used in conjunction with a fixed pitch propeller and direct reversing engine the clutch surfaces will have to be of adequate size to withstand the demands of crash reversal conditions and also clutches must be situated so that the heat generated during these manoeuvres can be dissipated as described in article 11.11. By using quill shafts for the input pinions the clutches can be separated from the couplings and placed aft of the gears, as in Figure 11–8(b). This arrangement often meets the requirements more easily and uses less space.

The quill shaft drive also permits the accommodation aft of the gearbox of an electric generator which can be driven by one of the engines permit the use of smaller higher speed electrical generators.

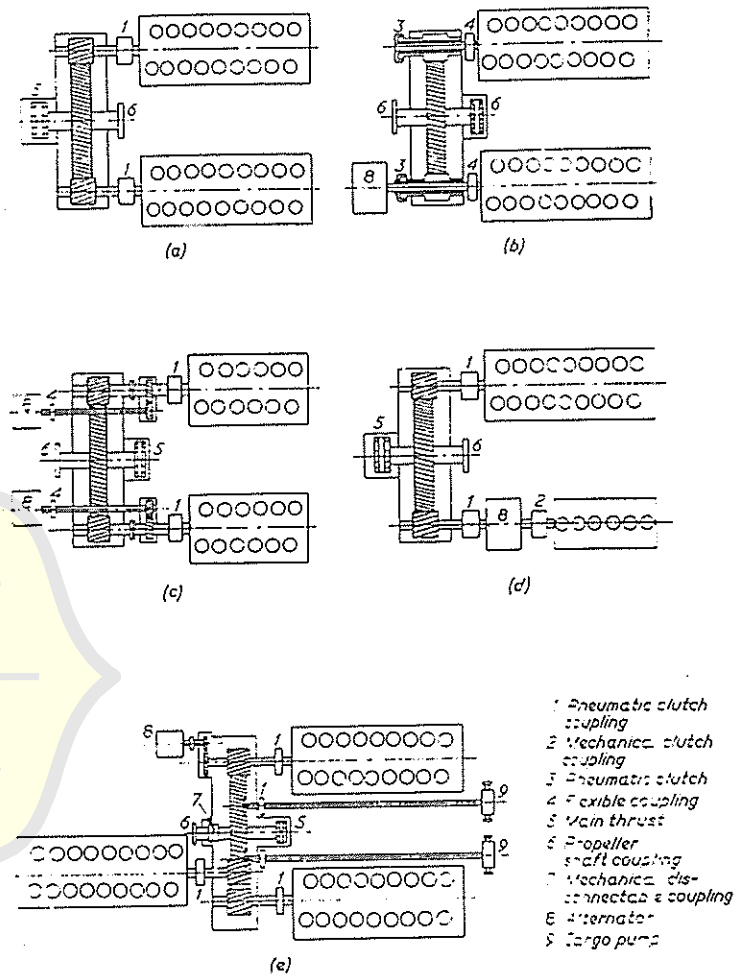


FIG. 11–8—*Arrangements of geared engines.*

The use of quill shafts for the input pinions with clutches or clutch couplings aft gives space to accommodate the thrust block forward of the main gear wheel; a position which many designers consider to be preferable from the point of view of the strength of the seating.

Figure 11–8(d) shows a design in which the two engines geared together are of widely differing power, sometimes termed a "father and son" arrangement. The smaller engine drives a generator (in port or under controllable pitch propeller with this combination ...

Three or four engines may be geared to one propeller and the variety of combinations of auxiliary drives is endless, Figure 11–8(e) shows one example. The use of controllable pitch propellers simplifies many of the arrangements, overcoming problems arising from reversing requirements. They are, of course essential if ac electrical machinery is to be driven from the propelling engines.

11.8. *Reversing gears*

It is in the method of reversing that the widest variation in practice between high powered and low powered installations is seen. High speed engines developing comparatively low powers, in the main, rely on reversing gear boxes for astern thrust. A typical form is the epicyclic train shown in Figure 17–6. For ahead operation the clutch plates are engaged and the whole gear revolves as one unit, for astern running the band brake prevents the rotation of the carrier in which the bevel pinions are mounted and as the clutch plates are released the drive is taken through the bevel gears. Neutral is obtained by releasing both clutch plates and band brake. A speed reduction gear is mounted abaft the reverse gear and embodies the thrust bearing, the whole assembly being built integral with the engine.

This simple reverse gear train and construction of gearboxes is quite inadequate for larger powers and medium speed engines are associated with separately mounted gearboxes of heavier scantlings. When reversing is carried out by gearing it takes the form of a separate train within the box, the ahead or the astern train of gears being engaged by the operation of appropriate clutches. These clutches are mostly of multi-drive plate design compactly arranged within the gearwheels and operated hydraulically.

For high powered medium speed engines the cost of incorporating reversing gear trains in the speed reduction boxes becomes higher than the cost of providing direct reversing of the engine and the latter method is the more popular in installations of large power, particularly for twin and multi-engine arrangements where flexibility for manoeuvring can be claimed in addition.

Controllable pitch propellers offer a means of obtaining astern thrust that is being increasingly adopted by all sizes of high speed and medium speed engines.

11.9. *Flexible couplings*

The separate mounting of engine and gearbox, the higher torques and the larger sizes of components make it essential to have some form of flexible coupling between the engine and the gearbox. This coupling must not only protect the gears from misalignment and impact loading but ... transmitting the torque from the driving member to the driven member through a flexible element which may be in the form of steel springs or of rubber blocks. In both these cases it is possible to manufacture transmission elements covering a wide variation of flexibility and so obtain a range of couplings of different stiffnesses. Selection of the appropriate stiffness value enables torsional vibration to be controlled by "tuning" the shaft system.

Some flexible couplings are designed so that they have non-linear torque deflexion characteristics. This gives them a "detuning" property which is of assistance in dealing with torsional vibration. There are other designs of coupling which go further than this and are arranged to damp torsional vibrations to which they are subjected.

11.10. *Fluid couplings and electro-magnetic couplings*

An alternative to directly mechanically coupling the engine and gearbox is to be found in the use of either fluid couplings or electro-magnetic couplings. In fluid couplings the torque is transmitted between the driving and driven members by means of an hydraulic fluid which circulates between the two. In the case of electro-magnetic couplings the torque is transmitted by magnetic flux. Both these couplings have the great advantage that they isolate the engine and its vibrations completely from the gearbox and the stern gear. With both of them, however, there is some loss of power due to the fact that there must be a slip between the driving and driven members in order to transmit the torque. In fact the more torque being transmitted the greater is the slip. Figure 11–9 shows the torque slip

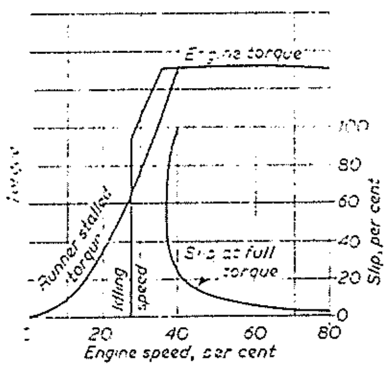


FIG. 11–9—*Torque slip characteristics of fluid coupling.*

characteristics of a fluid coupling. The size of the coupling is determined so that the amount of slip and the power loss is not excessive; about 3% of the engine power is usually considered tolerable. This power loss appears as heat in the hydraulic fluid or as heat in the electrical circuit of the are large in size, heavy and ... been used quite frequently.

Both fluid couplings and electro-magnetic couplings provide a ready means by which the engine may be disconnected from the propeller. This can be a useful feature if the engine is sometimes used for purposes other than propulsion on occasion and is practically essential when two or more engines are geared together to drive one output shaft.

11.11. *Clutches*

Disconnexion may be desired when mechanical flexible couplings are used and if this is the case then some form of mechanical clutch must be incorporated. Mechanical clutches can take the form of disks, drums or cones and they may be operated manually, pneumatically or hydraulic-

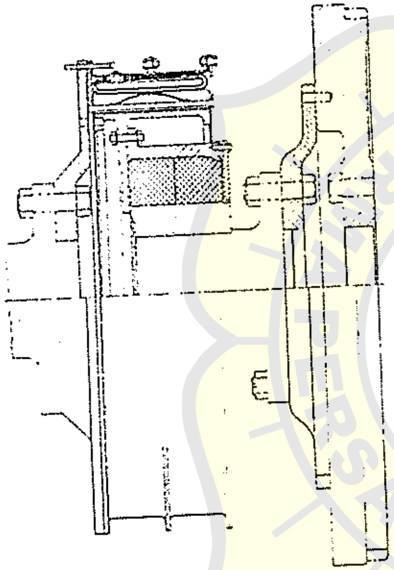


FIG. 11-10.

ally. On engines of over one hundred horsepower manual operation is usually impossible and pneumatic or oil operation is the rule. In order to save space clutches and couplings are frequently combined, the clutch being built around the flexible resilient centre forming the coupling. With this form of construction drum type or cone type couplings are the most easily accommodated. Examples of these are shown in Figures 11-10 and 11-11.

In determining the size of a clutch the first requirement is that it should be capable of carrying the maximum steady torque in the shaft system. However, in many cases the clutch is required to perform a far

With direct reversing engines and fixed pitch propellers the sequence of carrying out the manoeuvre is as follows. Fuel is cut off from the engines by closing the fuel pump racks and the engine and propeller rev/min fall together, the ship continuing with virtually no change in speed until there is zero torque in the shaft, which is achieved in some seven to ten seconds.

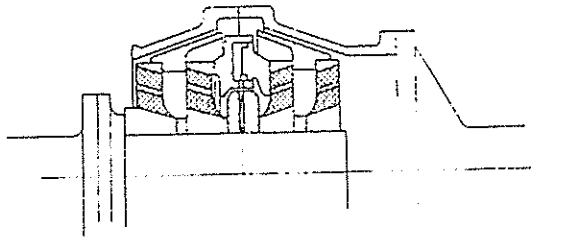


FIG. 11-11—*Cone type clutch-coupling.*

The clutches are then disengaged and the engines reversed. The engines must now be run at a speed sufficiently high above idling to ensure that they will not stall during subsequent re-engagement of the clutches. In each clutch one half is now turning astern driven by the engine and the other half is turning ahead driven by the trailing propeller, and, as re-engagement takes place energy is dissipated at the clutch friction surfaces in the form of heat. The clutches must be large enough and so designed that this operation does not cause them to burn out. If this re-engagement is performed properly the vessel will be moving ahead at a speed practically equal to its full speed when the manoeuvre commences and the propeller trailing rev/min and torque will be high and hence the rate of energy dissipation at the clutches will also be high. If time is allowed for the ship to slow down the trailing rev/min, torque and rate of energy dissipation will all be reduced but an unacceptably long stopping time or distance may result. A curve showing the variation of torque with propeller rev/min during the manoeuvre will have an appearance as shown in Figure 11-12. The vertical and horizontal axes of torque and rev/min respectively have been extended into the negative regions. Negative torque and negative rev/min represent torques and rotational speed in the astern direction. Starting at the condition of full torque and full ahead revolutions, point A, engine power is cut off and the rev/min falls until an equilibrium position is reached at which the propeller trails, the only torque in the shaft being that imparted by the trailing propeller to overcome friction in bearings and gears, point B. Between B and C the propeller continues to turn ahead but losing rev/min as torque in the astern direction of rotation is provided either by brakes or by clutches in the act of being engaged to the engine which, by now, has been reversed and is running astern.

of the ahead shaft.
Figure 11–1. If the torque–rev/min curve of the propeller passes into this shaded area then the clutches can be engaged without waiting for the ship to slow down and the period of engagement is as indicated on the graph. If

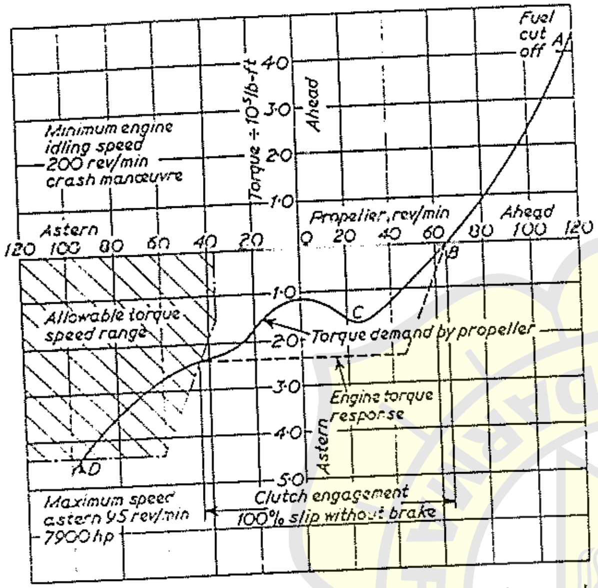


FIG. 11–12—*F.P.P. torque—rev/min characteristics for crash stop.*

the torque–rev/min curve does not pass through the shaded area then it will be necessary to allow the ship to slow down until a condition is reached in which this characteristic has lower torque values and enters the shaded area. In any event at the final astern rev/min the torque must not exceed that which the engine is capable of giving. (Refs. 1, 2 and 3)

Clutches therefore must be selected not only to be capable of transmitting the maximum torque of the engine but also of dissipating the energy which is converted into heat at their friction surfaces when undergoing a manoeuvre of this kind. Clutches that are combined with couplings and placed separately between the engine and gearbox are located in a position where they can be made amply large to cater for this condition.

When a reversing gear is used precisely similar actions take place as far as the clutches are concerned. Because of the demands of space within the gearbox, clutches operating astern gears are often restricted in size and are frequently not capable of coping with crash reversals at the full speed of the vessel. Some of the energy to be dissipated can be absorbed by a brake fitted on the propeller shaft. This is used to slow down and hold the propeller shaft stationary. The propeller can be held in this condition

clutches are sufficiently large to dissipate the energy then the addition and use of a shaft brake will not significantly improve the time for a crash

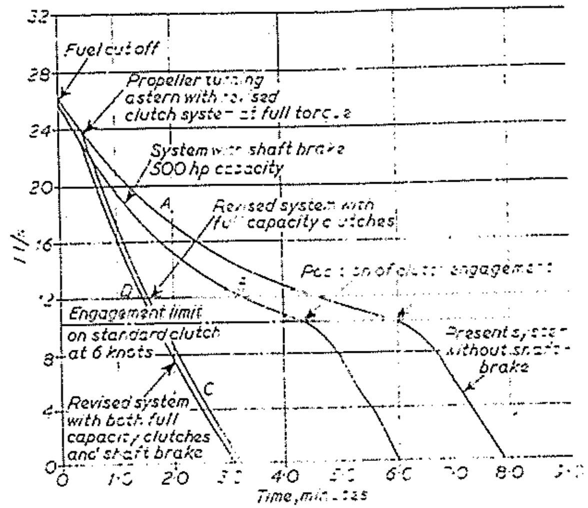


FIG. 11–13—*F.P.P. time taken to stop.*

reversal. Figure 11–13 illustrates four cases for a vessel having the following details:

Length B.P.	102 m (335 ft)	Engine speed—8·77 rev/sec (525 rev/min)	
Breadth (MLD)	16·8 m (55 ft)	rating—1·400 MN/m² (203 p.s.i., b.m.e.p.)	
Draught (MLD) loaded	7·8 m (25 ft 6 in)	No. engines—2	
Displacement loaded	8380 tonnes (8260 tons)	No. cylinders—6	
Block coefficient	0·64	Propeller dia.—4·19 m	(13·75 ft)
Loaded speed	7·96 m/s (15·5 knots)	P/D—0·865	
Power	3500 kW (4700 S.H.P.)	B.A.R.—0·55	
Estimated loaded prop. rev/min	2·58 rev/sec (155 rev/min)		

At A is seen the speed time relationship for a vessel with clutches designed to take the full torque from the engine but not large enough to dissipate the energy necessary for early engagement. At B is shown the condition when a shaft brake is added to the system capable of being applied at the start of the manoeuvre. The time to bring the vessel to rest is reduced by these means from 7·9 minutes to 6 minutes but by using a clutch system of adequate size the time can be reduced to 3·2 minutes as shown by curve C. The use of a brake in conjunction with the large clutches rapidly to a peak followed by an equally rapid reduction to a steady value remaining virtually constant until the ship is dead in the water. The sharp peak of thrust corresponds with peak torque reaction from the propeller. This cannot be allowed to reach a value much higher than the full torque of the engine, as if it did the engine would stall. In fact any excess above the full torque causes a momentary fall in engine speed. All these features are exhibited in Figure 11–14. Control of the rate at which the pitch of the blades is altered may be necessary to avoid such excessive torque. In this case, therefore, the clutches may be selected on the basis of the maximum engine torque plus a small margin.

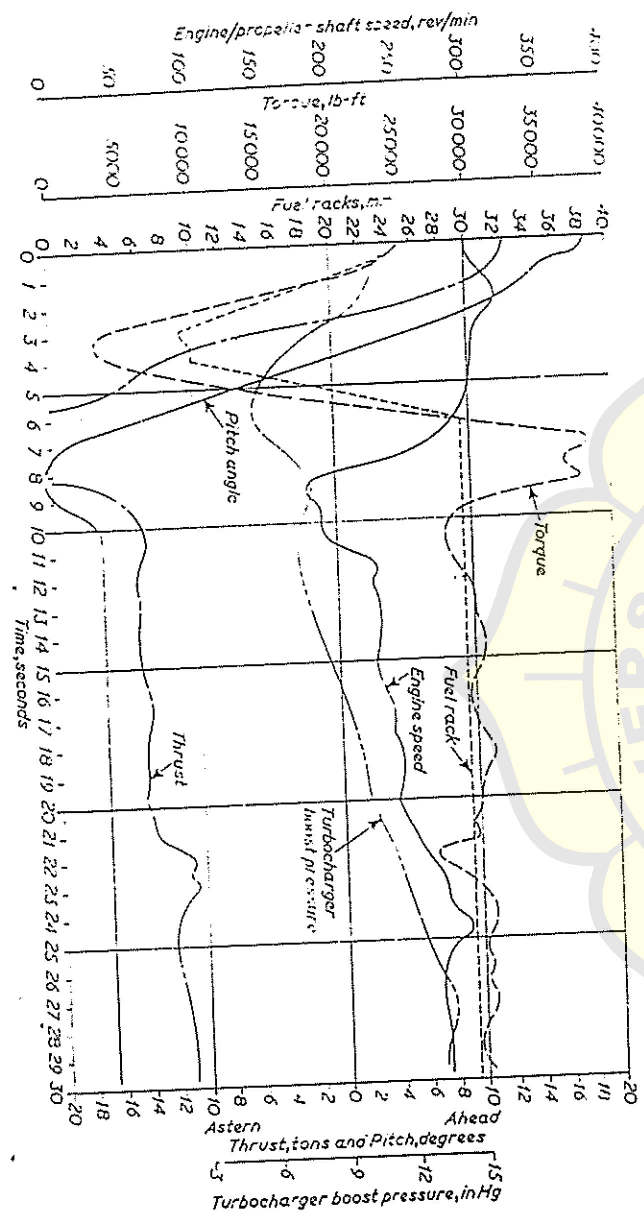


FIG. 11–14—C.P.P. emergency stop by full reversal of pitch.

REFERENCES

1. HENSHALL, S. H., *Development of a standardized power pack*, Trans. I. Mar. E. IMAS '69.
2. ADLEY, A. A. and LEA, K. E., *Selections, applications and installation of medium speed machinery systems*. Paper to branches of I. Mar. E.
3. GOODWIN, A. J. H., IRVINE, J. H. and FORREST, J., *The practical application of computers in marine engineering*, Trans. I. Mar. E., Vol. 80, 1968.

The seaman making his first voyage may not at first realise that the essentials of certain arrangements do not vary greatly from ship to ship. After some experience he will find that he can familiarise himself in a strange ship quite rapidly with, for example, bilge, ballast, oil fuel ~~transfer,~~ fresh water, sanitary water and other important systems. It is a good idea to be painstaking and to trace the piping carefully, even laboriously, until the arrangements are fully understood. The knowledge so gained may be very useful on one wild night in the darkness.

Remote control

In the last ten years, developments have been numerous and rapid; the powers of single engines have increased dramatically, remote and automated controls (with provision for local manual operation) operated from air-conditioned, sound-proof compartments have come into common use. As these devices are now reliable, engine rooms are often unmanned for long periods. These changes have naturally been accompanied by some simplification of main systems, but have brought about a great increase in auxiliary equipment. Low temperature (and therefore very low pressure) cascade evaporators are usual, often utilising waste heat. Relatively elaborate treatment of oil fuel, lubricating oil, boiler feed water and drinking water have become normal practice.

1–4
105 –
125

A large amount of control equipment demands clean, dry, oil-free air and the compressors to provide it. The oil-lubricated stern-tube is general, as is underwater discharge of sewage, often treated and the refrigeration load for air-conditioning commonly exceeds that for cargo. Tables 1.1 and 1.2 list the more important auxiliaries in four modern vessels; ships A and B are single-screw motor-ships, ship C single and ship D twin-screw turbine steamers. The proliferation of tanks, small pumps and heat exchangers is apparent. It will be noticed that there is a tendency to provide pumps in duplicate for essential services and to keep systems simple and separate, though it is not unusual to instal three pumps,

It is usual to have only two sea inlet openings, one port, one starboard, with a pipe connection from a valve at a higher level, for use in shallow water. Each opening on which all sea suction valves are mounted, is fitted with a strainer and weed-clearing valve (also sometimes a heating connection). The freshwater engine-cooling systems in motorships work under positive head, i.e. a header tank placed at a greater height than the highest engine cooling water outlet is connected in the system; all air vents are led to this tank and all make-up water is introduced.

Table 1.1 Auxiliaries found in single-screw motor ships

	Ship A	Ship B
Main L.O. pumps	2	2
Main S.W. pumps	2	2
Auxiliary pumps	2	2
Main jacket cooling pumps	2	2
Main piston cooling pumps	2	2
O.F. transfer (heavy) pumps	1	1
O.F. transfer (diesel) pumps	1	—
Heavy O.F. separators	2	2
L.O. separators	1	1
Sludge pumps	2	3
Boiler feed pumps	2	2
Fire wash-deck pumps	1	2
Gen. service pumps	1	1
Ballast pumps	2	1
E.R. Bilge pumps	1	1
Refrigerating circulating pumps	2	1
Fresh water pumps	2	3
Sanitary pumps	1	1
Starting air compressors	2	2
Starting air reservoirs	2	2
Main L.O. coolers	1	1
Main jacket water coolers	1	1
Main piston water coolers	1	1
Generating engines	4	4
Storage, drain, sludge, etc. tanks	22	40
Heat exchangers	9	20
Small pumps	10	23

	Ship C	Ship D
Main circulating pumps	2	4
Main feed pumps	2	4
Auxiliary pumps	1	2
Main condensate pumps	2	4
Main L.O. pumps	2	4
Oil fuel transfer pumps	2	2
Diesel pumps	—	2
Auxiliary bilge pumps	1	1
Bilge and ballast pumps	2	2
Fire and washdeck pumps	2	2
Auxiliary F.W. pumps	—	2
Auxiliary S.W. pumps	—	2
Sanitary pumps	1	1
Fresh water pumps	2	2
Turbo-alternator/generator	1	1
Diesel alternator/generator	3	3
Main L.O. coolers	1	2
Storage, drain, sludge, etc. tanks	35	45
Heat exchangers	23	25
Small pumps	17	20

CIRCULATING WATER SYSTEMS

In motorships there are two main circuits, one salt, one fresh. The salt circuit is: sea inlet box-pump(s) — oil cooler(s) — f.w.cooler(s) in series — turbo-blower aftercooler(s) in parallel — overboard. Branches may be taken to blower oil cooler(s), fuel valve cooling oil cooler(s) and from the outlet side, to evaporator sea inlets or domestic warm water systems; baths, etc. There may be a blanked connection to the fresh water circuit.

The fresh water circuit being under positive head, is closed, i.e. pump — fresh water coolers — cylinder jackets — cylinder heads — exhaust valves (if any) — turbo-blower(s) (if any), by branches — pump. There may be a closed circuit branch to evaporator primary-stage heating inlet(s). If the engine being: pump — cooler — inlet manifold — telescopic tubes — outlet manifold — pump. The f.w.temperature should be kept as high as practicable by the use of the salt water bypass valves on oil and f.w. coolers. These may be butterfly valves controlled by thermo-pneumatic devices. It is usual to provide for warming the fresh circulating water before the main engines are started, either by steam or more usually, by bleeding from the auxiliary fresh cooling circuit.

The auxiliary salt cooling circuit is: sea inlet — pump — oil cooler(s) — fresh water cooler(s) — overboard. The air compressor inter- and after-coolers are likely to be supplied in parallel; alternatively, their fresh water coolers. It is unusual for supercharger blowers to have aftercoolers; if they have, they will be circulated similarly. The auxiliary fresh water system is similar to the main and may use the same header tank if the resulting head is not too great; if it is, a separate header tank will be provided at a lower level.

If the pumps for each service are not duplicated, they will be dual-connected. If one pump fails, the survivor becomes the fresh water pump and a clean service e.g. ballast pump circulates the sea water.

In steamers the main circulating system is: sea inlet-pump(s) — main condenser — overboard, with branches to the main oil coolers and possibly, turbo-generator condensers. Pumps are usually centrifugal, engine, turbine or motor driven. In some fast, high-powered ships, axial-flow pumps are used in conjunction with scoops. At speed, the scoop gives an adequate flow of water, the pump impeller idling; the pump is used in other circumstances. If the largest clean service pump (e.g. ballast pump) has adequate capacity, it will act as stand-by circulator, if not, two main pumps are usual.

The use of auxiliary condensers is limited to small services and port use and they will have separate circulating pumps, which may serve also the oil coolers; wherever possible feed water is used for cooling the auxiliary exhausts. Many steamers have diesel-engined generators, the circulating systems being as for motorships.

KVÆRNER EUREKA	DATA SHEET for CENTRIFUGAL PUMP		
Sign KR / Dato 29/11/94		Page Side 1	of av 1

CUSTOMER	:	P.T.DOK SURABAYA
CUSTOMER REF.	:	1X6.500 DWT TANKER
KVÆRNER TENDER NO.	:	94-178
KVÆRNER ORDER NO.	:	620033

Rev.: 01
Sign : ZW
Date : 22.11.96

ITEM NO. 01 CARGO OIL PUMPS

MAIN DATA

No. off	:	3 - Three
Type	:	Double suction, double volute, one stage, radially split centrifugal pump
Model	:	C04BX 6-10 AAN H91
Rotation	:	Anti clockwise (Seen towards pump shaft driven end).
Shop no.	:	96-20413 to 96-20415

PUMP SELECTION CRITERIA / PERFORMANCE DATA

Fluid	:	Cargo oil		Density	:	900	kg/m³
Capacity	:	300	m³/h	Viscosity	:	33	cSt at °C
Diff. pressure	:	90	mLC	Power consumption	:	107	kW
Speed	:	3568	rpm	Min. driver rating	:	125	kW

DESIGN DATA

Shaft seal	:	Mechanical seal	Flanges:	DN mm :	Pressure class :
Coupl. bearing	:	Grease lubricated ball bearing	Suction	250	DIN 2501-PN16
Aft bearing	:	Cargo lubricated journal bearing			
Weight	:	255 kg approx.	Discharge	150	DIN 2501-PN16
Materials	:	Casing : Ni-Al-Bronze	Evac.	3/4" BSP	
		Impeller : Ni-Al-Bronze			
		Shaft : Stainless Steel			
Coating	:	Alkyd finish, blue - RAL 5015			

DOCUMENTATION

Description	:	SU 04623-000	Outline Drawing	:	TU 17038-004
Performance curve	:	TU 16353-688	Sectional drawing	:	TU 17329-001
Schematic drawing	:	TU 17275-001	Material list	:	TU 17329-002

TEST

Performance test in accordance with ISO 2548 Engineering grade 2. Hydrostatic pressure test to 1.5 times pump delivery pressure against closed discharge valve.

CERTIFICATE

Official certificate from ABS

ACCESSORIES

Pump shaft coupling mounted on.
Blind counter flanges will be provided.

REMARKS

Mechanical seal leakage alarm (high and low level alarm for barrier fluid) will be provided.