BAB - IX
KESIMPULAN

Dengan selesainya penvusunan tugas merancang ini, maka penulis dapat mengambil kesimpulan yang berhubungan dengan perencanaan kapal Tanker 15.000 DWT sebagai sarana angkutan laut yang dapat menunjang perkembangan ekonomi di Indonesia. Adapun kesimpulan penulis adalah sebagai berikut

1. Data spesifikasi teknis dari kapal Tanker 15. 000 DWT:

Panjang seluruhnya	(Loa)	= 149,21 m
Panjang antara garis tegak	(Lpp)	= 140 m
Lebar	(B)	= 24,6 m
Tinggi	(H)	= 11,8 m
Sarat Air	(T)	= 7, 015 m
Koeffisien blok	(Cb)	= 0,809
Koeffisien prismatic	(Cp)	= 0,817
Koeffisien garis air	(Cw)	= 0,992
Koefisien tengah kapal	(Cm)	= 0,989
Dislacement	(Δ)	= 24.763,65
Volume	(∇)	=19.158,61
Jumlah anak buah kapal	(ABK)	= 32 orang

Alat penggerak yang digunakan

Jumlah mesin 1 (satu) buah

Merk	Warsila Vasa 32
Type	12V32
Daya	6.040 Hp/4.440 kW
Putaran mesin	720 rpm
Bore x Stroke	320 mm x 350 mm
Cycle	4 langkah
Jumlah silinder	12

Dimensi	: 5.696 mm (L);2.540 mm (W);3.653 mm (H)
SFOC	: **189 gr/ kW.h.**
SLOC	: 1,2 g/ kWh.
Diameter propeller	: 4,218 m
Jumnlah daun	: 3 (empat) buah
Kecepatan dinas	: 13 knot

2. Dalam rancangan, untuk dapat menentukan besarnya daya motor induk sebagai penggerak utama kapal, maka faktor kecepatan, daerah pelayaran serta dimensi dari kapal rancangan mempunyai pengaruh yang sangat besar.

3. Dalam menentukan generator set didasarkan pada pembebanan pengunaan daya yang terbesar yaitu pada saat kapal melakukan manuver sebesar 552,347 kW, dengan menggunakan 3 buah generator masing-masing berkapasitas 294 kW, dimana satu diantaranya berfungsi sebagai generator cadangan atau standby generator, daya yang dibutuhkan dapat terpenuhi.

4. Dalam perancangan kamar mesin, tidak lepas adanya asumsi-asumsi yang diberikan untuk mempermudah dalam perhitungan dengan tidak mengabaikan tanggung jawab secara teknis, ekonomis serta peraturan-peraturan yang ada
sehingga hasil perhitungan dapat mendekati keadaan yang sebenarnya.

5. Tata letak mesin induk, mesin bantu serta permesinan lainnya diatur seeffisien mungkin. hal ini untuk mempermudah dalam hal perawatan dan perbaikan peralatan yang ada di kamar mesin serta tata letaknya sangat berpengaruh pada stabilitas kapal.

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WÄRTSILÄ 26X

Main data

Cylinder bore	260 mm	Fuel specification:	
Piston stroke	320 mm	Marine diesel oil	ISO 8217, category ISO-F-DMA-DMB
Speed	1000 rpm		
Mean effective pressure	28.2 bar		
Piston speed	10.7 m/s		

The 26X is intended for naval applications and others such as fast ferries [illegible] higher performance ratings.

Rated power: Propulsion engines

Engine type	Output in kW/bhp at 1 000 rpm	
	kW	BHP
12V26X	4 800	6 525
16V26X	6 400	8 700
18V26X	7 200	9 785

WÄRTSILÄ 32

Main Data

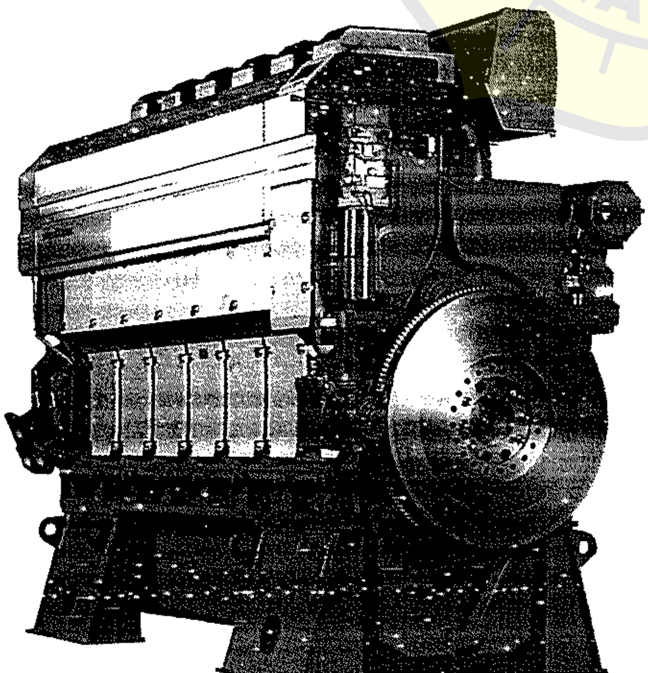
Cylinder bore	320mm		
Piston stroke	400 mm	Fuel specification:	
Speed	720 - 750 rpm	Fuel oil	730 cSt/50°C
Mean effective pressure	23.3 - 22.9 bar		7 200 sRI/100°F
Piston speed	9.6 - 10.0 m/s		ISO 8217, category ISO-F-RMK 55

Rated power: Propulsion engines

Engine type	Output 720 rpm		750 rpm	
	kW	BHP	kW	BHP
6L32	2 700	3 670	2 760	3 750
8L32	3 600	4 890	3 680	5 000
9L32	4 050	5 510	4 140	5 630
12V32	5 400	7 340	5 520	7 510
16V32	7 200	9 790	7 360	10 010
18V32	8 100	11 010	8 280	11 260

Rated power: Auxiliary engines

Engine type	Output 720 rpm/60 Hz		750 rpm/50 Hz	
	Engine kW	Gen. kW	Engine kW	Gen. kW
6L32	2 700	2 600	2 760	2 660
8L32	3 600	3 470	3 680	3 550
9L32	4 050	3 900	4 140	3 990
12V32	5 400	5 210	5 520	5 320
16V32	7 200	6 940	7 360	7 100
18V32	8 100	7 810	8 280	7 990



WÄRTSILÄ Vasa 32/32GD

Main Data

Cylinder bore	320 mm	Fuel specification:	
Piston stroke	350 mm	Fuel oil	730 cSt/50°C
Speed	720 - 750 rpm		7 200 sRI/100°F
Mean effective pressure	24.0 - 21.3 bar		ISO 8217, category ISO-F-RMK 55
Piston speed	8.4 - 8.75 m/s	Natural gas	

Rated power: Propulsion engines

Engine type	*D rating at 720 rpm		750 rpm		E rating at 720 rpm		750 rpm	
	kW	BHP	kW	BHP	kW	BHP	kW	BHP
4R32	1 480	2 010	1 500	2 040	1 620	2 200	1 640	2 230
6R32	2 220	3 020	2 250	3 060	2 430	3 300	2 460	3 350
8R32	2 960	4 030	3 000	4 080	3 240	4 410	3 280	4 460
9R32	3 330	4 530	3 375	4 590	3 645	4 960	3 690	5 020
12V32	4 440	6 040	4 500	6 120	4 860	6 610	4 920	6 690
16V32	5 920	8 050	6 000	8 160	6 480	8 810	6 560	8 920
18V32	6 660	9 060	6 750	9 180	7 290	9 910	7 380	10 040

*Available for Gas-Diesel

Rated power: Auxiliary engines

Engine type	E rating (max.) at 720 rpm/60 Hz		750 rpm/50 Hz	
	Eng. kW	Gen. kW	Eng. kW	Gen. kW
4R32	1 620	1 560	1 640	1 580
6R32	2 430	2 340	2 460	2 370
8R32	3 240	3 130	3 280	3 170
9R32	3 645	3 520	3 690	3 560
12V32	4 860	4 690	4 920	4 750
16V32	6 480	6 250	6 560	6 330
18V32	7 290	7 030	7 380	7 120

WÄRTSILÄ 38

Main data

Cylinder bore	380 mm	Fuel specification:	
Piston stroke	475 mm	Fuel oil	730 cSt/50°C
Speed	600 rpm		7 200 sRI/100°F
Mean effective pressure	24.5 bar		ISO 8217, category ISO-F-RMK 55
Piston speed	9.5 m/s		

Rated power: Propulsion engines

Engine type	A-output at 600 rpm		B-output at 600 rpm		B-output at 630 rpm	
	kW	BHP	kW	BHP	kW	BHP
6L38	3 960	5 385	4 350	5 915	4 350	5 915
8L38	5 280	7 180	5 800	7 890	5 800	7 890
9L38	5 940	8 080	6 525	8 875	6 525	8 875
12V38	7 920	10 770	8 700	11 830	8 700	11 830
16V38	–	–	11 600	15 775	11 600	15 775
18V38	11 880	16 155	13 050	17 750	13 050	17 750

Note: [illegible] available for [illegible] (subject to changes)

SULZER ZA40S

Main data

Cylinder bore	400 mm	Fuel specification:	
Piston stroke	560 mm	Fuel oil	730 cSt/50°C
Speed	510 rpm		7 200 sRI/100°F
Mean effective pressure	25.1 bar		ISO 8217, category ISO-F-RMK 55
Piston speed	9.5 m/s		

Rated power: Propulsion engines

Cyl.	Output in kW/bhp at 510 rpm * Engine MCR	
	kW	bhp
6L	4 500	6 120
8L	6 000	8 160
9L	6 750	9 180
12V	9 000	12 240
14V	10 500	14 280
16V	12 000	16 320
18V	13 500	18 360

* Speeds 500 [illegible] 14 rpm at same outputs also available for 50 and 60 Hz operation respectively

Yanmar Diesel Engine Co., Ltd.

Specifications (Mains)

* Asterisk shows DIN627

Model	No. of cylinders	Bore x stroke: mm	Cont. rating output: hp/rpm	Dry weight: kg	Dimensions L x W x H: mm
D18	2	70 x 70	18/4500	74, 79	722 x 460 x 1286
D27	3	70 x 70	27/4500	82, 87	722 x 460 x 1368
D36	3	70 x 70	36/4500	114, 118	730 x 460 x 1433
1GM10	1	75 x 72	*9/3600	76	547 x 410 x 485
2GM20	2	75 x 72	*18/3600	106	638 x 455 x 495
2GM20F	2	75 x 72	*18/3600	114	643 x 482 x 545
3GM30	3	75 x 72	*27/3600	130	735 x 455 x 495
3GM30F	3	75 x 72	*27/3600	138	740 x 455 x 545
3HM35	3	80 x 85	*34/3400	158	786 x 485 x 617
3HM35F	3	80 x 85	*34/3400	167	791 x 475 x 638
2TD	2	100 x 115	26/2100	330	874 x 526 x 805
3TD	3	100 x 115	39/2100	400	1009.5 x 526 x 825
4TD	4	100 x 115	52/2100	510	1235.5 x 526 x 854.5
4JH2E	4	82 x 86	*50/3600	228	888.4 x 565 x 634.5
4JH2-TE	4	82 x 86	*62/3600	234	888.4 x 565 x 634.5
4JH2-HTE	4	82 x 86	*75/3600	244	888.4 x 565 x 643.5
4JH2-DTE	4	82 x 86	*88/3600	244	888.4 x 565 x 643.5
3ESDE	3	120 x 135	56/1800	680	1255 x 689 x 967
4ESDE	4	120 x 135	74/1800	800	1473 x 694 x 1015
4LH-TE	4	100 x 110	*110/3300	340	1058.2 x 649 x 726
4LH-HTE	4	100 x 110	*140/3300	350	1058.2 x 649 x 726
4CHE	4	105 x 125	70/2300	655	1372 x 688 x 1025
6CHE	6	105 x 125	105/2300	785	1661 x 690 x 1018
6CH-HTE	6	105 x 125	155/2300	830	1658 x 690 x 1056
6CH-DTE	6	105 x 125	190/2300	880	1658 x 690 x 1091
6CH-UTE	6	105 x 125	255/2550	915	1551.5 x 730 x 1111
4KDE	4	145 x 170	110/1450	1430	1701 x 731 x 1164
6KDE	6	145 x 170	165/1450	2263	2495 x 741 x 1262
6HA(M)E	6	130 x 150	165/2000	1145	1529 x 885 x 1097
6HA(M)-HTE	6	130 x 150	240/2000	1230	1529 x 939 x 1233
6HA(M)-DTE	6	130 x 150	300/2000	1250	1529 x 939 x 1233
6GH-UTE	6	117.9 x 140	350/2300	1335	1762 x 898.5 x 1247
6LAAE	6	148 x 165	240/1900	2120	1703 x 921 x 1275.5
6LA-DTE	6	148 x 165	400/1800	1890	1719 x 1012.5 x 135
6LAA-UTE	6	148 x 165	530/1850	1890	1719 x 1012.5 x 135
8LAA-DTE	Vee 8	148 x 165	530/1800	2420	1983 x 1430 x 1420
8LAA-UTE	Vee 8	142 x 165	650/1850	2420	1983 x 1430 x 1420
12LAA-DTE	Vee 12	148 x 165	800/1800	3300	2553 x 1430 x 1470
12LAA-UTE	Vee 12	148 x 165	1000/1850	3300	2553 x 1430 x 1470
S165	6	165 x 210	200/1200	3100	2574.5 x 1043 x 15
S165-T	6	165 x 210	300/1300	3150	2574.5 x 1070 x 15
S165-UT	6	165 x 210	450/1300	3600	2697 x 1070 x 158
S165-ST	6	165 x 210	550/1300	3780	2697 x 1070 x 158
S165-ET	6	165 x 210	600/1350	3780	2847 x 1070 x 158

(Continued on nex

UNIVERSITAS DARMA PERSADA

Reduction gears

The core function of a reduction gearbox is to reduce the main engine speed to the optimum propeller speed. The Wärtsilä gears have been designed to meet the highest standards of operational efficiency, reliability and low noise and vibration.

Gear configurations

The gears can be supplied with built in multidisc clutches. Single input, single output gears are available with vertical or horizontal offsets of the shafts. Twin input single output gears can be delivered with up to 3.8 m horizontal offsets.

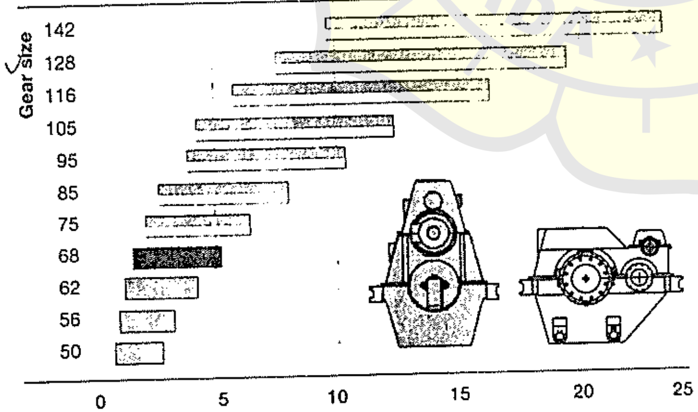
Power take-off arrangements

More than 90% of all gearbox deliveries include a built-in Power Take-Off (PTO) for shaft generators. The standardized solutions are primary driven, which means that the PTO is running also when the propeller has been disconnected. Customized solutions like secondary driven-, twin- and two-speed PTOs are also available.

Auxiliary propulsion drive for increased safety

The basic idea of the Auxiliary Propulsion Drive (APD) is to be able to utilise the power from the auxiliary engines for propulsion as back up for the main engine. To facilitate the APD option a standard gearbox with a multidisc clutch is supplied with an additional disconnecting coupling between the gear and the main engine. The APD may also be used for operation modes with low vessel speeds.

Wärtsilä reduction gears – Output range



Single stage gears – vertical offset SV, SCV
Single stage gears – horizontal offset SH, SCH

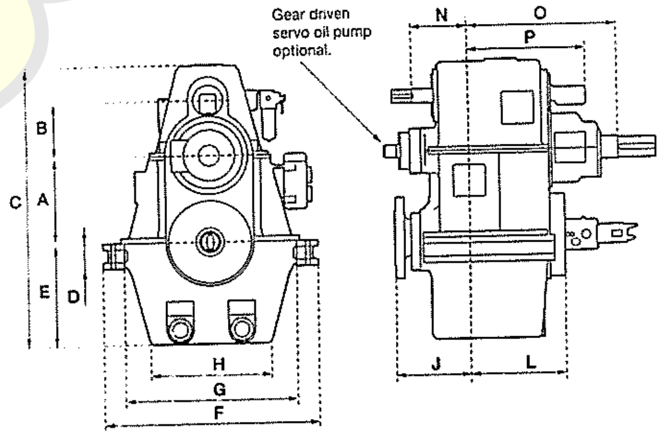
Integrated or separate hydraulic system for gear and CP propeller

Most of the Wärtsilä gears are purposely designed with an integrated hydraulic system for both the gear and the CP propeller. This will reduce installation cost for the yard and operational costs for the owner, as the complete hydraulic power unit for the CP propeller will be left out. For safety reasons the gear mechanically drives the main pump for the propeller. All gears can also be interfaced to a separate hydraulic power unit.

Single marine reduction gears

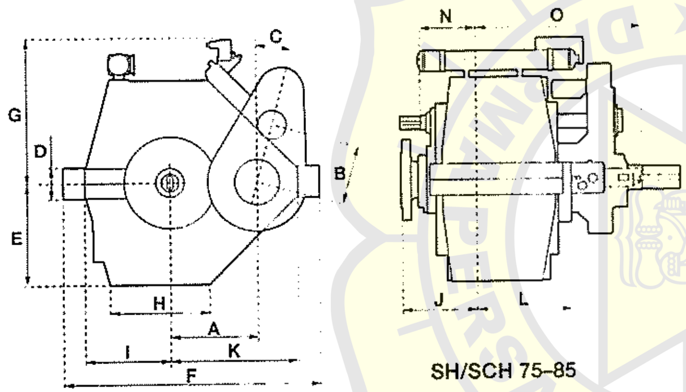
Vertical offset gears – Dimensions

SV,SCV Gear size	A	B Standard -Max	C	D	E	F	G	H	J	L	N	O SCV/SV	P
SCV 50	500	380	1724	150	590	1340	1024	720	470	592	420	1035	745
SCV 56	560	410	1848	160	645	1500	1110	800	530	650	450	1100	760
SCV 62	620	440-470	2210	180	740	1580	1240	880	570	662	350	1150	1000
SCV 68	680	460-510	2370	200	800	1720	1360	960	625	720	370	1250	1010
SCV 75	750	480-530	2460	220	880	1850	1480	1040	660	800	450	1300/1095	1035
SCV 85	850	510-560	2720	250	1000	2100	1680	1178	730	915	550	1470/1220	1170
SCV 95	950	580-630	3025	280	1145	2350	1880	1327	800	1025	450	1640/1350	1385
SCV 105	1050	630	3302	300	1265	2600	2100	1467	880	1125	500	1700/1400	1346
SCV 116	1160	650	3525	150	1400	2580	2300	1800	1535	765	885	1800/1025	1235
SCV 128	1280	800	3970	275	1536	3160	2645	1815	1700	840	900	2270/1120	1760
SCV 142	1420	1000	4520	305	1704	3505	2645	2012	1885	928	910	2270/1320	1950

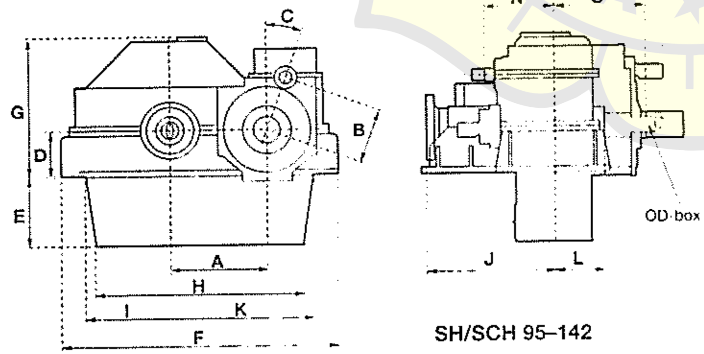


Horizontal offset gears – Dimensions

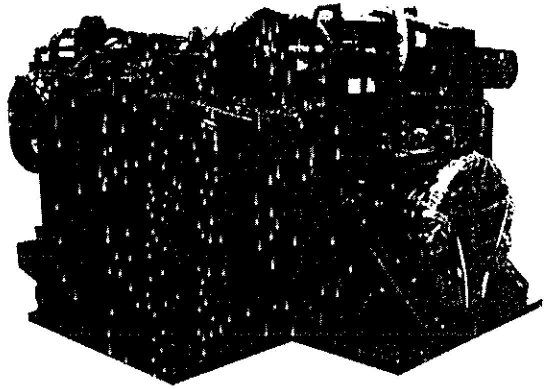
SH,SCH Gear size	A	B	C	D	E	F	G	H	I	J	K	L	N	O	P
SCH 75	750	530	15°	280	885	2230	1220	865	735	660	1115	800	515	1670	-
SCH 85	850	580	15°	320	1000	2495	1440	970	830	730	1245	915	550	1800	-
SCH 95	950	580	15°	450	750	2710	1520	2250	830	1215	1420	540	700	1640	1390
SCH 105	1050	630	20°	500	771	2995	1658	2195	910	1405	1545	560	750	1510/1700	1480
SCH 116	1160	670	20°	550	850	3300	2240	2500	1015	1535	1715	725	830	1800/1100	1150
SCH 128	1280	740	20°	590	1550	3640	1960	2675	1090	1600	1870	-	915	1915	-
SCH 142	1420	820	20°	620	1720	4040	2180	2970	1380	1700	2240	-	1015	2100	2000



SH/SCH 75–85



SH/SCH 95–142

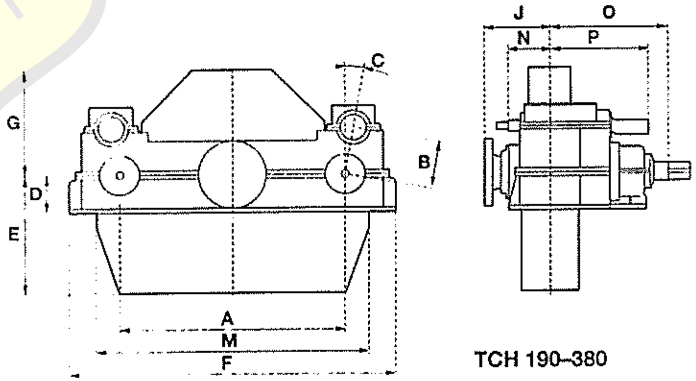


Wärtsilä Gear type TCH200V65/2. Twin input single output gear with two stage reduction, gear ratio 10:1, designed for diesel electric propulsion.

Twin input-single output reduction gears

Dimensions

Gear Size	A	B	C	D	E	F	G	J	M	N	O	P
TCH190	1900	400	10°	320	980	2750	890	555	2300	360	995	830
TCH250	2500	530	12.5°	450	1400	3700	1150	800	3230	570	1290	1170
TCH280	2800	650	10°	560	1450	4290	1305	990	3520	630	1700	-
TCH320	3200	760	10°	640	1660	4900	1490	1160	4020	720	1960	-
TCH350	3500	850	10°	700	1855	5370	1630	1270	4380	790	2140	1895
TCH380	3800	960	10°	760	2015	5800	1760	1380	4770	860	2300	-



TCH 190–380

Section 4

Main Shafting

A. General

1. Scope

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

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

2. Documents for approval

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

B. Materials

1. Approved materials

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

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

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

2. Testing of materials

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

C. Shaft Dimensions

1. General

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

2. Minimum diameter

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

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

d [mm] required outside diameter of shaft

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

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

d_a [mm] actual shaft diameter

P_W [kW] rated power transmitted by shaft

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

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.

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

1.6 Pipe connections

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

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

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

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

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

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

2. Calculation of pipe diameters

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

2.2 Dry cargo and passenger ships

a) main bilge pipes

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

b) branch bilge pipes

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

where

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

2.3 Tankers

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

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

where:

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

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

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

2.4 Minimum diameter

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

2.5 Maximum diameter

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

2.6 Deviations

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

3. Bilge pumps

3.1 Capacity of bilge pumps

Each bilge pump must be capable of delivering:

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

where:

Q	[m^3/h]	minimum capacity
d_H	[mm]	calculated inside diameter of main bilge pipe

Section 6

Propellers

A General

1. Scope

These Rules apply to screw-propellers and vane wheels. Where a design is proposed to which the following Rules cannot be applied, special strength calculations are to be submitted to the Society. The tests to be carried out in each case are to be agreed with the Society. For the dimensions and materials of propellers for ships with ice classes, see Section 13

2. Documents for approval

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

B. Materials

1. Approved materials

Propellers and vane wheels are to be made of seawater-resistant cast copper alloys or cast steel alloys with a minimum tensile strength of 440 N/mm², cf. Rules for Materials. For the purpose of the following design Rules governing the thickness of the propeller blades, the requisite resistance to seawater of a cast copper alloy or cast steel alloy is considered to be achieved if the alloy used can be proved to withstand a fatigue test[1]) under alternating bending stresses comprising 10^8 load cycles amounting to about 20 % of the minimum tensile strength and carried out in a 3 % NaCl solution, and if it can be proved that the fatigue strength under alternating bending stresses in natural seawater is not less than about 65 % of the values established in 3 % NaCl solution

2. Materials for blade retaining-bolts

Unless protected against contact with seawater the blade-retaining bolts of assembled or controllable pitch propellers must likewise be made of materials which are resistant to seawater.

3. Novel materials

Where it is proposed to use propeller materials whose serviceability is not attested by a sufficient period of practical experience the Society must be provided with special proof of the suitability of such materials.

4. Material testing

The material of propellers, vane wheels and blade-retaining bolts or studs is to be tested in accordance with the Society's Rules for Materials.

C. Dimensions and design of propellers

1. Symbols and terms

A	[mm²]	Effective area of a shrink fit
B	[mm]	Developed blade width of cylindrical sections at radii 0,25 R, 0,35 R and 0,6 R
c	[-]	Coefficient for shrunk joints = 1,0 for engine and turbine gear transmissions = 12 for direct drives
C_G	[-]	Size factor in accordance with formula (2)
C_{Dyn}	[-]	Dynamic factor in accordance with formula (3)
C_w	[-]	Characteristic value for propeller material as shown in Table 6.1 (corresponds to the minimum tensile strength R_m of the propeller material where

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

		this has been shown to possess sufficient fatigue strength under alternating bending stresses in accordance with paragraph B.1.)
C	[-]	Conicity of shaft ends
	=	$\frac{\text{difference in taper diameter}}{\text{length of taper}}$
d	[mm]	Bolt-hole circle diameter of blade or propeller-fastening bolts
d_k	[mm]	Root diameter of blade or propeller-fastening bolts
D	[mm]	Diameter of propeller
		$= 2 \cdot R$
d_m	[mm]	Mean taper diameter
e	[mm]	Blade rake to aft
		$= R \cdot \tan \varepsilon$
E_T	[-]	Thrust stimulating factor in accordance with formula (5)
f, f_1, f_2, f_3	[-]	Factors in formulae (2) (3) (4) and (11)
F_M	[N]	Bolt load
H	[mm]	Propeller blade face pitch at radii 0,25 R, 0,35 R and 0,6 R
H_m	[mm]	Mean effective propeller pitch on blade face for pitch varying with the radius
	=	$\frac{\sum (R \cdot B \cdot H)}{\sum (R \cdot B)}$
		in which R, B and H are to be substituted by values corresponding to the pitch at the various radii.
J	[-]	Degree of advance
k	[-]	Coefficient for various profile shapes in accordance with Table 6.2
k'	[-]	Coefficient calculated by applying formula (6) where use is made of profile shapes other than those given in Table 6.2
K_T	[-]	Thrust coefficient
L_M	[mm]	2/3 of the leading-edge component of the blade width at 0,9 R, but at least 1/4 of the total blade width at 0,9 R for propellers with heavily raked blades.

Table 6.1 Characteristic values C_w

Material	Description [1])	C_w
Cu 1	Cast manganese brass	440
Cu 2	Cast manganese nickel brass	440
Cu 3	Cast nickel aluminium bronze	590
Cu 4	Cast manganese aluminium bronze	630
Fe 1	Unalloyed cast steel	380
Fe 2	Low-alloy cast steel	380
Fe 3	Martensitic cast chrome steel 13/1-6	600
Fe 4	Martensitic-austenitic cast steel 17/4	600
Fe 5	Ferritic-austenitic cast steel 24/8	600
Fe 6	Austenitic cast steel 18/8-11	500
Fe 7	Grey cast iron	200

[1]) For the chemical compositon of the alloys, see the Society's Rules for Materials and Regulations for the Assessment and Repair of Defects on Propellers.

L	[mm]	Pull-up length when mounting propeller on taper
L_{mech}	[mm]	Pull-up length at t = 35 °C
L_{temp}	[mm]	Temperature-related portion of pull-up length at t < 35 °C
n	[Rpm]	Propeller speed in rev/min.
P_w	[kW]	Shaft power
p	[N/mm²]	Specific pressure in shrunk joint between propeller and shaft
Q	[N]	Peripheral force at mean taper diameter
S	[-]	Margin of safety against propeller slipping on taper = 2,8
t	[mm]	Maximum blade thickness of developed cylindrical section at radii 0,25 R, 0,35 R and 0,6 R
T	[N]	Propeller thrust

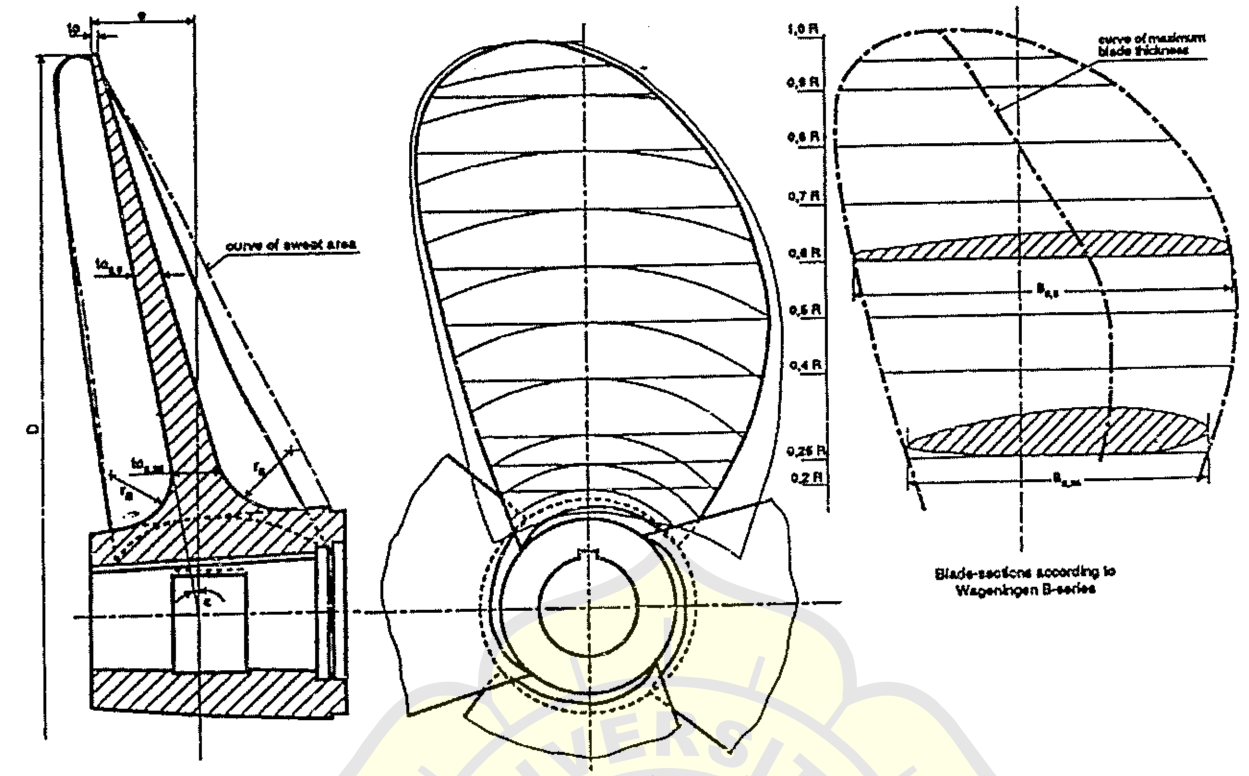

Fig. 6.1 Blade sections

T_M	[Nm]	Impact moment
V_s	[kn]	Speed of ship
w	[-]	Wake factor
W_x	[mm³]	Actual face modulus of developed cylindrical section referred to face blade pitch profiles about blade pitch line
Z	[-]	Total number of bolts used to retain one blade or propeller
z	[-]	Number of blades
α	[-]	Pitch angle of profile at radii 0,25 R, 0,35 R and 0,6 R

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

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

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

α_A	[-]	Tightening factor for retaining bolts and studs = 1,2 - 1,6 depending on the method of tightening used.

β_x	[-]	Factor for the section modulus of developed cylindrical section about blade pitch line for blade profiles in accordance with Table 6.2
β'_x	[-]	Factor for the section modulus of developed cylindrical section about blade pitch line for blade profiles other than those in Table 6.2
ε	[-]	Angle included by face generatix and normal
θ	[-]	Half-conicity of shaft ends = C / 2
μ_o	[-]	Coefficient of static friction = 0,13 for hydraulic oil shrunk joints = 0,18 for dry shrunk joints
$R_{p\,0,2}$	[N/mm²]	0,2 % proof stress of propeller material
R_{eH}	[N/mm²]	Yield strengths and
σ_{max}/σ_m	[-]	Ratio of maximum to mean stress at blade face

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Table 6.2 Values of k for various profile shapes

Profile shape	Values of k		
	0,25 R	0,35 R	0,60 R
Segmental profiles with circular arced back, $\beta_x = 0{,}12$	73	62	44
Segmental profiles with parabolic back, $\beta_x = 0{,}11$	77	66	47
Blade profiles as for Wageningen B Series propellers where $\beta_{x\,0,25} = 0{,}10$ $\beta_{x\,0,35} = 0{,}11$ $\beta_{x\,0,60} = 0{,}12$	80	66	44

Notes:
The Society reserves the right to specify an increase in the values of k in the case of special propellers where the blade width B at 0,2 R is < 4 • t.

2. Testing of materials

In addition to the material tests specified in B.4., the Society reserves the right to require component parts of the pitch-adjusting mechanism including in particular those which are not accessible for shipboard repairs to be tested in accordance with the Rules for Materials. Piping subject to pressures above 10 bar is to be tested in accordance with Section 11.

3. Hydraulic control equipment

Where the pitch-control mechanism is operated hydraulically two mutually independent, power-driven pump sets are to be fitted. For propulsion plants up to 200 kW one power-driven pump set is sufficient provided that in addition a hand-operated pump is fitted for controlling the blade pitch and that this enables the blades to be moved from the ahead to the astern position in a short enough time.

4. Pitch control mechanism

For the pitch-control mechanism proof is required that when subjected to impact moments T_M as defined by formula (7), the individual components still have a safely factor of 1,5 with respect to the yield strength of the materials used.

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

5. Blade retaining bolts

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

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

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

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

The shank of blade retaining bolts may be designed with a minimum diameter equal to 0,9 times the root diameter of the thread. Blade retaining bolts must be secured against unintentional loosening.

6. Indicators

Controllable pitch propeller system are to be provided with an engine room indicator showing the actual setting of the blades. Further blade position indicators are to be mounted on the bridge and in the engine room (see also Volume VII and Volume IV Section 9).

7. Failure of control system

Suitable devices are to be fitted to ensure that an alleration of the blade setting cannot overload the propulsion plant or cause it to stall.

Steps must be taken to ensure that, in the event of failure of the control system the setting of the blades

- does not change or
- assumes a final position slowly enough to allow the emergency control system to be put into operation.

8. Emergency control

Controllable pitch propeller systems must be equipped with means of emergency control enabling the controllable pitch propeller to remain in operation should the remote control syslem fail. It is recommended that a device be fitted which locks the propeller blades in the "**ahead**" setting.

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

1.6 Pipe connections

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

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

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

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

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

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

2. Calculation of pipe diameters

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

2.2 Dry cargo and passenger ships

a) main bilge pipes

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

b) branch bilge pipes

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

where

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l	[m]	length of the watertight compartment

2.3 Tankers

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

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

where:

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

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

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

2.4 Minimum diameter

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

2.5 Maximum diameter

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

2.6 Deviations

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

3. Bilge pumps

3.1 Capacity of bilge pumps

Each bilge pump must be capable of delivering:

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

where:

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

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

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

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

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

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

c L8

luasan balansir dianjurkan $\leq$ 23% dari seluruh luas kemudi dan lebar bagian balansir pada potongan-potongan horisontal < 0,35 lebar sayap kemudi.

Pada kapal-kapal yang mempunyai batas sarat air yang cukup tinggi, mempunyai ukuran yang tinggi (λ = hp/bp cukup tinggi).

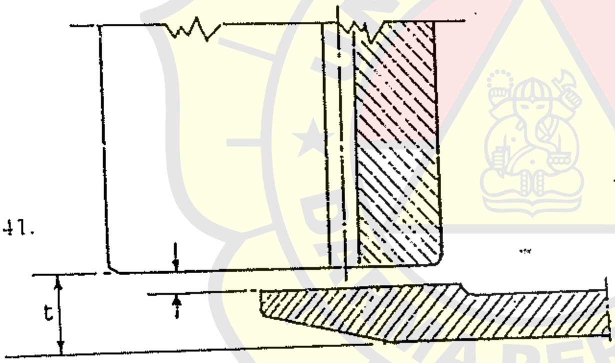
Tetapi tinggi kemudi harus diperlihatkan pula menurut bentuk buritan kapal.

Beberapa batasan untuk harga λ :

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

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

Batas-batasnya sebagai berikut :



Gambar 141.

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

Dimana :
h = tinggi kemudi.

Catatan : Umumnya untuk semua bentuk diambil ketentuan :

t 150 m/m.

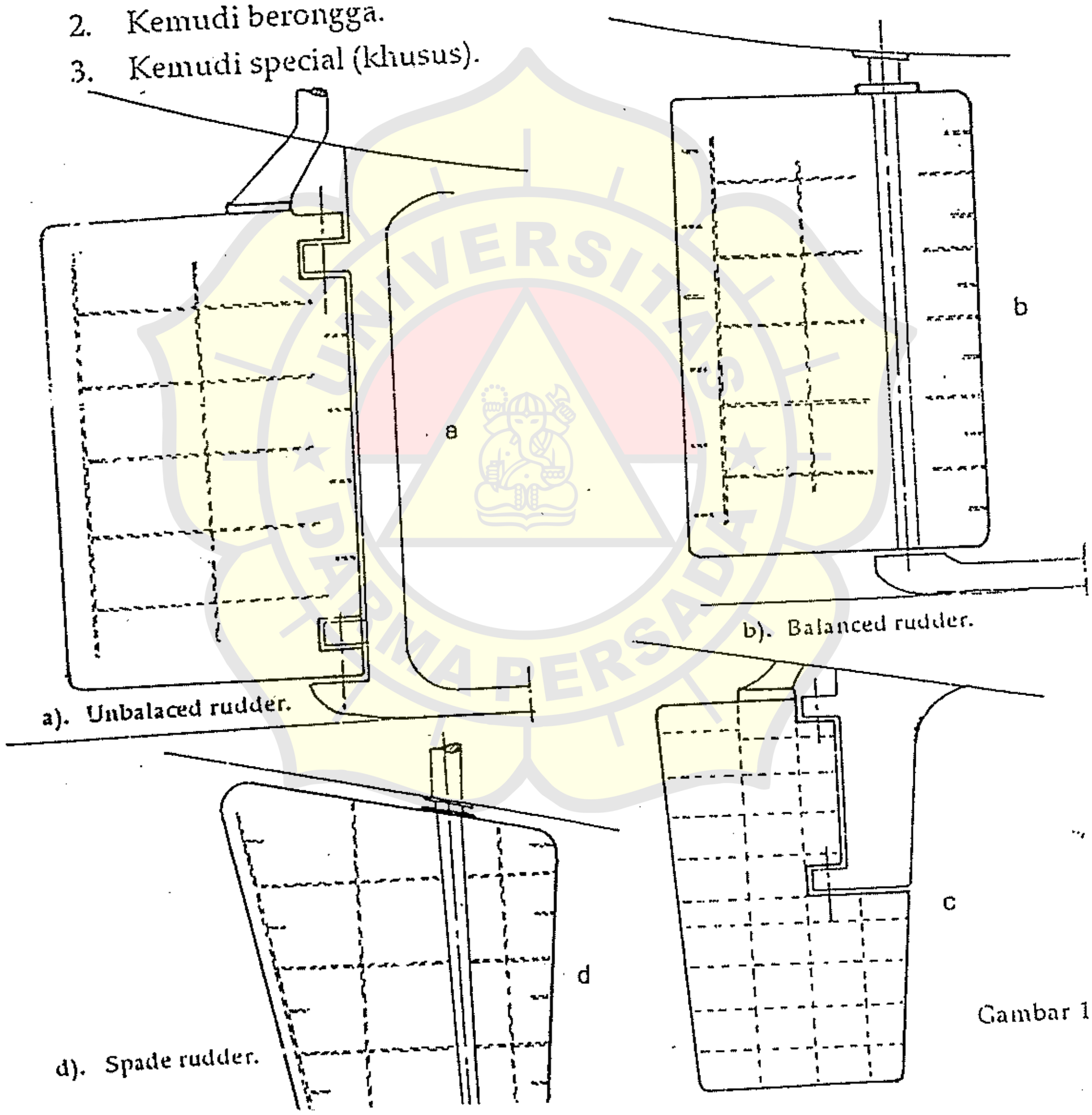
Oleh Van Lammeren ditetapkan batasan-batasan λ = h/b sebagai berikut :

Type kapal dan kemudi	h/d
1. kapal barang 1 baling-baling dan kapal penumpang semuanya dengan kemudi balansir.	1,8
2. Kapal pantai 1 baling-baling dengan kemudi balansir.	1,15
3. Kapal tunda 1 baling-baling dan kapal pandu.	1,75

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.

Gambar 130.

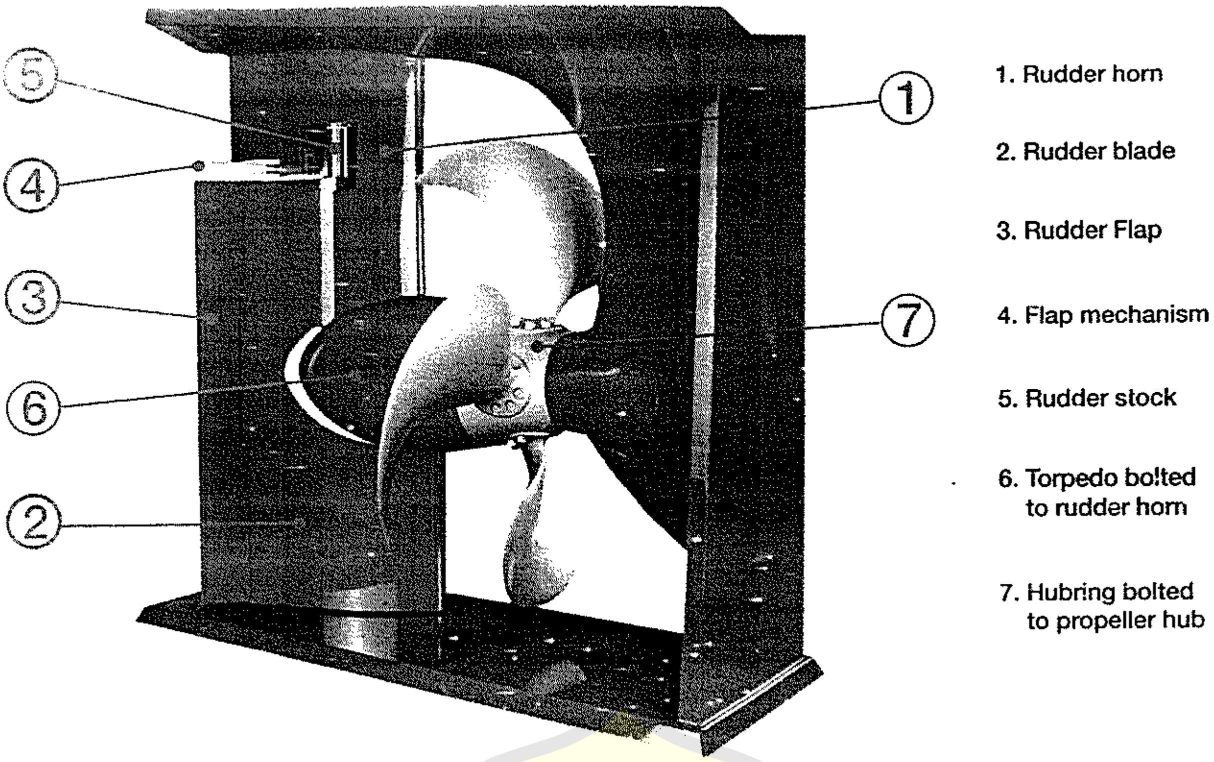


Fig. 1 The Propac Rudder.

testing facilities Modelbasin Marin and Vienna confirm the expected gain in power with the Propac Rudder.

Single-screw vessels

Model tests show highest gain for full hull form vessels such as tankers, bulk carriers and general cargo vessels. A gain of at least 5 % has been achieved due to the presence of the torpedo (Figures 2 and 4).

Twin-screw vessels

Propulsion tests on various twin-screw vessels, with the integrated torpedo fitted on the rudder, have shown a 2-5 % power saving. The rudder trunk and geometry can further increase this saving by another 1-5 %.

Manoeuvring performance

One requirement during the development of the Propac Rudder was to obtain similar or better manoeuvring performance than with conventional rudders. Model tests carried out so far indicate that the total area of the Propac Rudder often may be about 10 % larger than the rudder are of an all-movable rudder. The Propac Rudder can be delivered with or without a flap mechanism. ■

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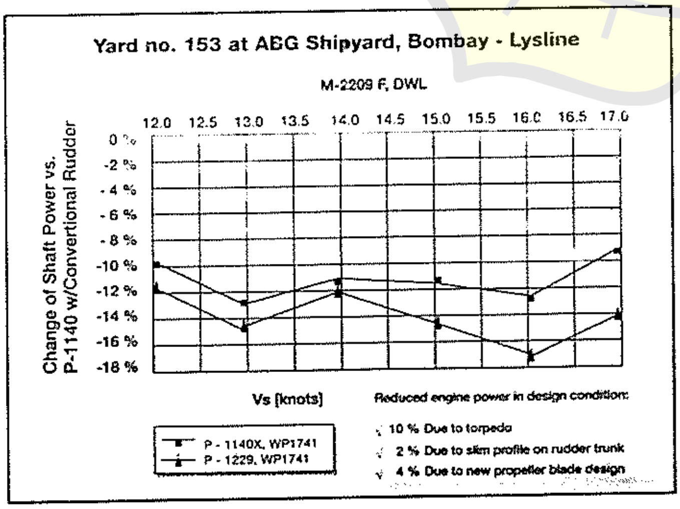


Fig. 4 Total power savings (%) for prototype Propac Rudder.

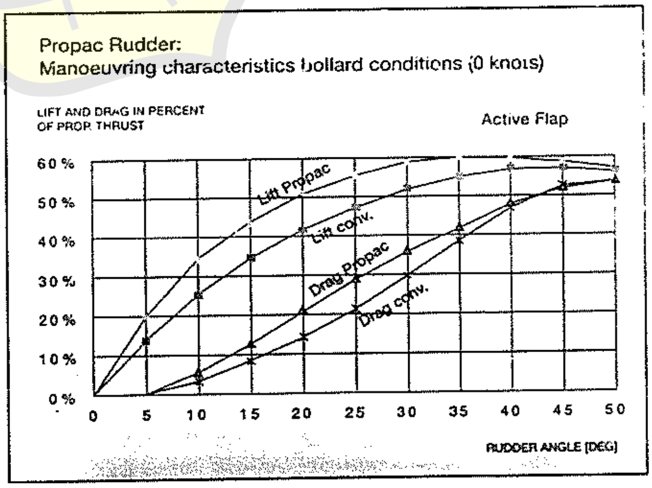
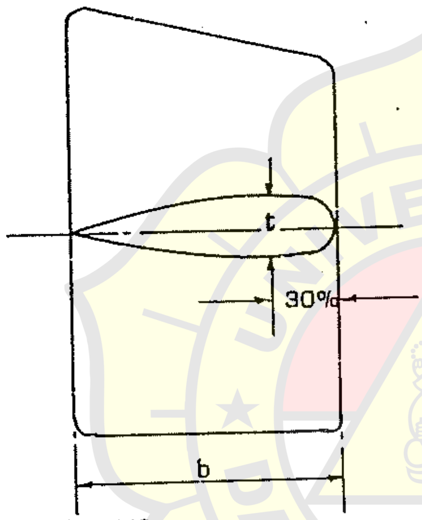


Fig. 5 Manoeuvring characteristics.

4. Untuk semua kapal dengan 2 baling-baling dengan-kemudi biasa.	1,5
5. Untuk kapal-kapal 2 baling-baling dengan kemudi-setengah balansir.	1,1
6. Untuk kapal-kapal dengan 2 baling-baling dengan-dua kemudi.	2,2

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



Gambar 142.

Berdasarkan praktek yang dilakukan, koefisien tebal plat profil kemudi :
$Ct = t/b$ terletak dalam batas-batas : 0,18 - 0,22.
Tetapi untuk kemudi setengah menggantung pada kapal besar hanya Ct mencapai 0,5.
Untuk kemudi biasa (tak balansir) untuk twin screw diambil batas-batas:

$$Ct = 0{,}15 - 0{,}18$$

Untuk setengah balansir :

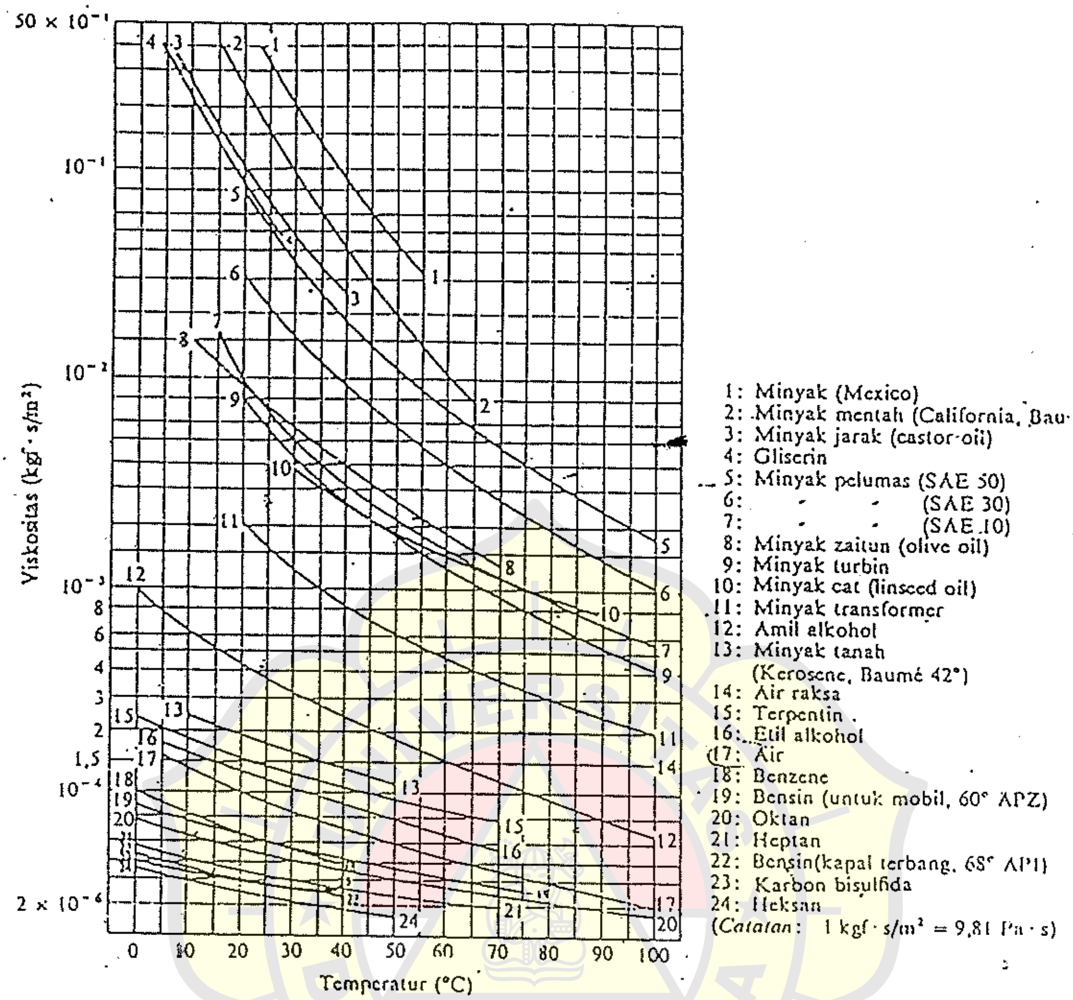
$$Ct = 0{,}18 - 0{,}22$$

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

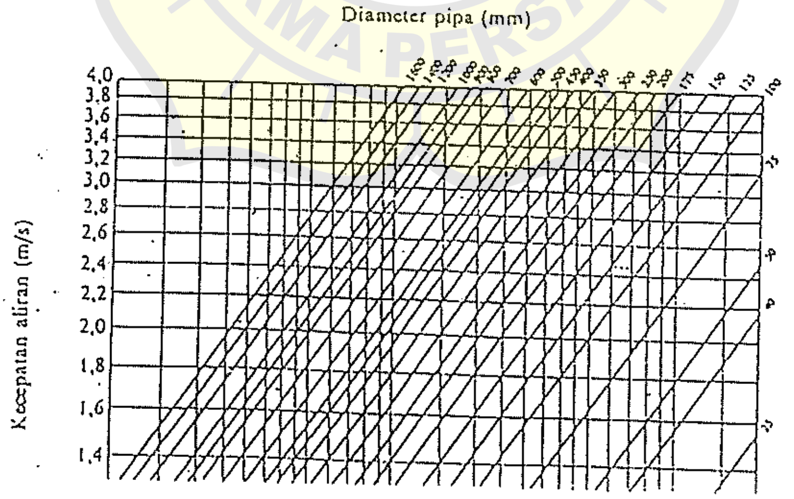
Untuk menghindari getaran dianjurkan supaya jarak maximum penampang kemudi yaitu 30% lebar profil, dihitung dari permukaan depan.

Koefisien kompensasi dihitung dengan rumus pendekatan yang menghasilkan perhitungan moment putar yang sangat kecil di poros, sehingga memperkecil kekuatan motor penggerak kemudi serta pengeluaran energi untuk merubah letak kemudi.

<u>Viskositas</u>



(a) Viskositas μ



2 SPESIFIKASI

2.1 Spesifikasi Pompa

Dalam memilih suatu pompa untuk suatu maksud tertentu, terlebih dahulu harus diketahui kapasitas aliran serta head yang diperlukan untuk mengalirkan zat cair yang akan dipompa.

Selain dari pada itu, agar pompa dapat bekerja tanpa mengalami kavitasi, perlu ditaksir berapa tekanan minimum yang tersedia pada sisi masuk pompa yang terpasang pada instalasinya. Atas dasar tekanan isap ini maka putaran pompa dapat ditentukan.

Kapasitas aliran, head, dan putaran pompa dapat ditentukan seperti tersebut di atas. Tetapi apabila perubahan kondisi operasi sangat besar (khususnya perubahan kapasitas

Tabel 2.1 Data yang diperlukan untuk pemilihan pompa.

No.	Data yang diperlukan	Keterangan
1	Kapasitas	Diperlukan juga keterangan mengenai kapasitas maksimum dan minimum.
2	Kondisi isap	Tinggi isap dari permukaan air isap ke level pompa. Tinggi fluktuasi permukaan air isap. Tekanan yang bekerja pada permukaan air isap. Kondisi pipa isap.
3	Kondisi keluar	Tinggi permukaan air keluar ke level pompa. Tinggi fluktuasi permukaan air keluar. Besarnya tekanan pada permukaan air keluar. Kondisi pipa keluar.
4	Head total pompa	Harus ditentukan berdasarkan kondisi-kondisi di atas.
5	Jenis zat cair	Air tawar, air laut, minyak, zat cair khusus (zat kimia), temperatur, berat jenis, viskositas, kandungan zat padat, dll.
6	Jumlah pompa	
7	Kondisi kerja	Kerja terus-menerus, terputus-putus, jumlah jam kerja seluruhnya dalam setahun.
8	Penggerak	Motor listrik, motor bakar torak, turbin uap.
9	Poros tegak atau mendatar	Hal ini kadang-kadang ditentukan oleh pabrik pompa yang bersangkutan berdasarkan instalasinya.
10	Tempat instalasi	Pembatasan-pembatasan pada ruang instalasi, ketinggian di atas permukaan laut, di luar atau di dalam gedung, fluktuasi temperatur.
11	Lain-lain	

Tabel 2.2 **Jumlah kebutuhan air maksimum per orang per hari menurut kelompok jumlah penduduk.**

Jumlah penduduk (satuan: 10,000 orang)	Kebutuhan air (l/orang · hari)
Kuang dari 1	150–300
1–5	200–350
5–10	250–400
10–30	300–450
30–100	350–500
Lebih dari 100	Lebih dari 400

(c) Konsumsi harian rata-rata

Angka ini akan diperlukan untuk menghitung konsumsi energi listrik serta biaya operasi dan pemeliharaan. Besarnya dapat ditaksir sebagai berikut:

(Konsumsi harian rata-rata) =
(Konsumsi harian maksimum) × 0,7 (untuk kota kecil atau sedang), atau
0,8 (untuk kota besar atau kota industri

(d) Konsumsi tiap jam maksimum

Konsumsi ini merupakan kebutuhan puncak dalam jangka 1 tahun, di mana akan terjadi laju aliran maksimum pada sistem distribusi air. Jadi angka ini penting untuk menentukan ukuran pipa dan sistem distribusi yang akan direncanakan. Adapun cara menaksirnya adalah sebagai berikut:

(Konsumsi per jam maksimum) =
(Konsumsi harian maksimum/24) × 1,5 (untuk kota kecil atau sedang), atau
1,3 (untuk kota besar atau kota industri)

(e) Pompa penyadap dan penyalur

Pompa yang dipakai untuk menyadap air baku dari sumber serta mengalirkannya ke instalasi penjernihan disebut pompa penyadap (intake). Adapun pompa yang dipergunakan untuk mengalirkan air bersih dari penjernihan ke tandon distribusi disebut pompa penyalur.

Kapasitas pompa ini dapat ditaksir sebagai berikut:

1) Jumlah air yang disadap = (Konsumsi harian maksimum) × (1,1 sampai 1,15)
Faktor perkalian sebesar 1,1 sampai 1,15 tersebut di atas diambil untuk mengimbangi kebocoran pipa atau pemakaian air kerja di pusat penjernihan.
2) Jumlah air yang disalurkan = (Konsumsi harian maksimum) + (α)
Di sini (α) adalah jumlah air yang harus ditambahkan untuk mengganti kehilangan karena bocoran antara pusat penjernihan dan reservoir distribusi.
3) Fluktuasi jumlah air dan dasar penentuan jumlah pompa.

Pompa penyadap dan pompa penyalur biasanya bekerja tanpa fluktuasi aliran yang cukup berarti. Pada umumnya pompa-pompa ini bekerja dengan beban penuh. Adapun jumlah pompa yang diperlukan untuk memenuhi jumlah air yang dipompa dapat ditentukan menurut Table 2.3.

(f) Pompa distribusi

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

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

$$\eta_{ad} = \frac{L_{ad}}{L_s} \qquad (2.$$

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

$$N = L_{ad} = \frac{mk}{k-1}\,\frac{P_s Q_s}{6120}\left[\left(\frac{P_d}{P_s}\right)^{(k-1)/mk} - 1\right], \quad (\mathrm{kW}) \qquad (2.$$

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) dinyata 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

$$L_{ad} = \frac{mk}{k-1}\,\frac{P_s Q_s}{60000}\left[\left(\frac{P_d}{P_s}\right)^{(k-1)/mk} - 1\right], \quad (\mathrm{kW}) \qquad (2.$$

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

Sebagai contoh, dari Tabel 2.7 terbaca bahwa untuk kompresi 1 tingkat sa 7 kgf/cm^2 (g) atau 8,033 kgf/cm^2 abs, diperlukan daya sebesar 4,7074 kW. Ini dipe dari Pers. (2.21) dengan mengambil harga $k = 1{,}4$ dan $m = 1$. Daya sebesar 4,707 tersebut juga akan diperlukan untuk kompresi 2 tingkat tanpa pendingin antara. Na jika digunakan pendingin antara maka daya yang diperlukan menjadi sebesar 4 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 se berikut. Seandainya untuk sebuah kompressor 2 tingkat yang memampatkan menjadi 7 kgf/cm^2 (g) diperlukan daya poros sebesar 5,4 kW, maka dengan daya

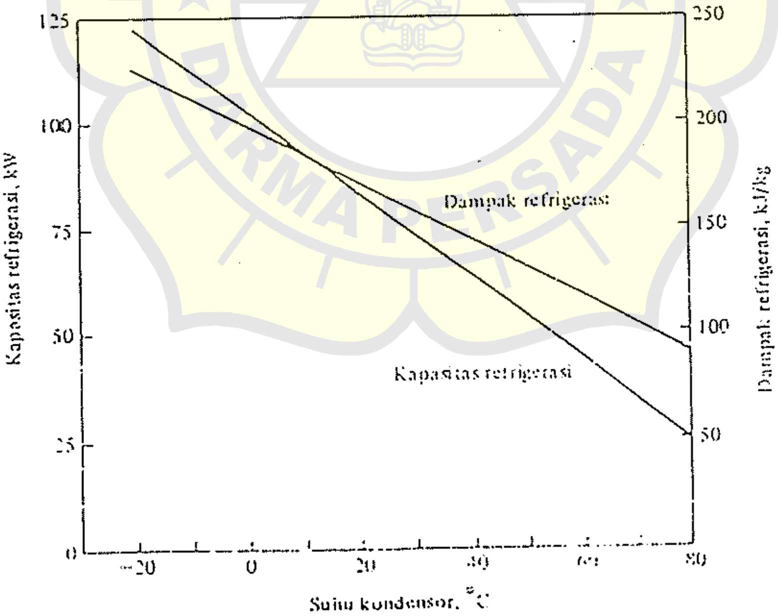
19

LAMPIRAN

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.



[illegible] 11-10 Dampak refrigerasi dan kapasitas refrigerasi untuk kompresor ideal dengan refrigeran

Mula-mula perlu ditentukan jumlah limpasan keseluruhan dari air hujan di tanah pertanian dengan rumus

$$Q = 10fRA10 \tag{2.2.a}$$

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

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

$$Q_r = \frac{Q}{24 \times 3600 \times D} \tag{2.2.b}$$

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

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

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

Tabel 2.7 Curah hujan total dan koefisien limpas total.

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

Tabel 2.8 Faktor distribusi limpasan dari curah hujan tunggal.

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

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

(5) Pengairan tanah pertanian

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

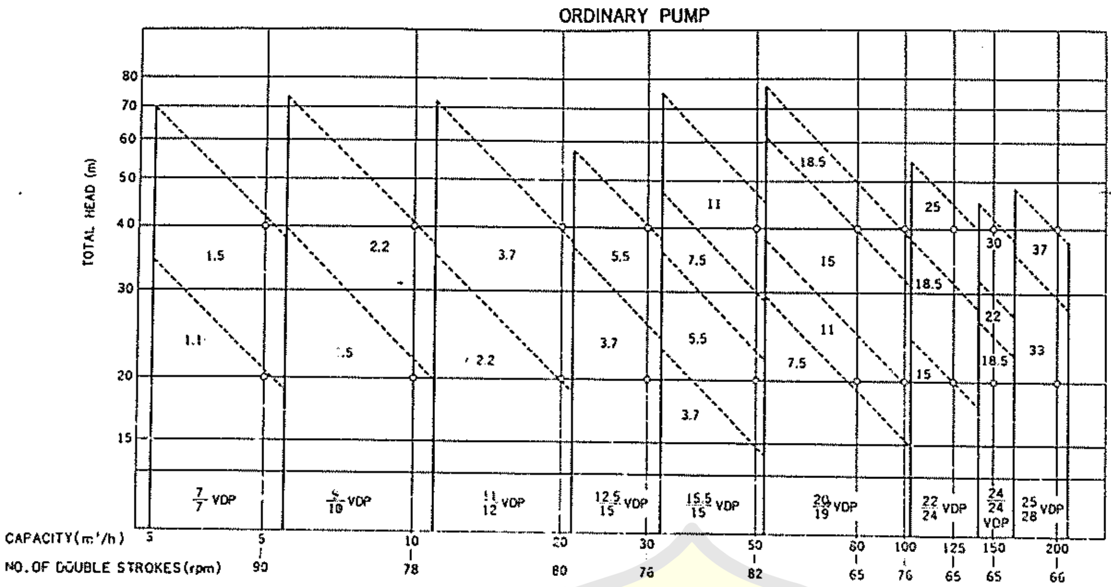
(a) Pengairan sawah

1) *Keperluan air*

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

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

PERFORMANCE CHART



EX

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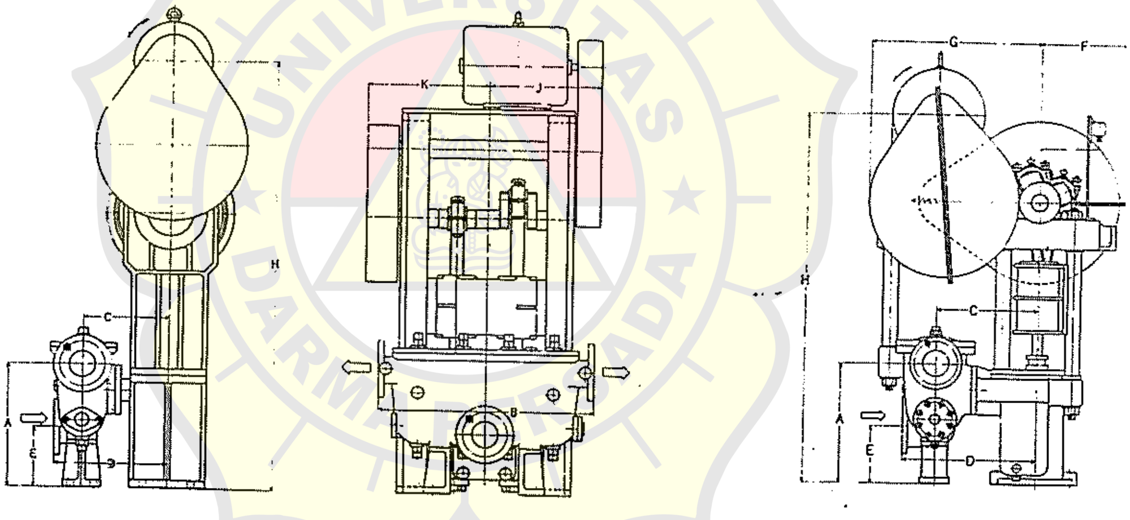
DIMENSIONS



TYPE	PUMP CYL. DIA	LENGTH OF STROKE	BORE SUCTION	BORE DELIVERY	A	B	C	D	E	F	G	H	J	K	M	N	O	P	
$\frac{7}{7}$ VDP	70	70	50	40	257	510	205	275	122	—	—	928	265	285	—	—	—	100	1
$\frac{9}{10}$ 〃	90	100	65	50	330	570	225	300	160	—	—	1148	305	325	—	—	—	60	1
$\frac{11}{12}$ 〃	110	120	80	60	350	〃	250	330	175	—	—	1292	340	365	—	—	—	〃	1
$\frac{12.5}{15}$ 〃	125	150	100	80	415	670	290	385	210	—	—	1525	360	390	—	—	—	55	1
$\frac{15.5}{15}$ 〃	155	150	125	100	455	810	330	440	212	—	—	1612	405	445	—	—	—	70	1
$\frac{17}{17}$ 〃	170	170	150	125	520	1100	380	520	245	355	695	1650	455	490	140	145	480	135	
$\frac{20}{19}$ 〃	200	190	150	125	555	1280	480	630	265	400	800	1757	600	600	160	180	570	180	-
$\frac{22}{24}$ 〃	220	240	200	150	620	1540	490	665	300	475	990	2105	635	660	175	205	670	〃	-
$\frac{24}{24}$ 〃	240	240	200	150	〃	〃	〃	〃	〃	〃	〃	〃	〃	〃	〃	〃	〃	〃	-
$\frac{25}{28}$ 〃	250	280	200	200	690	1630	525	720	350	〃	965	2101	700	715	〃	210	730	220	-

ECIPROCATING PUMPS

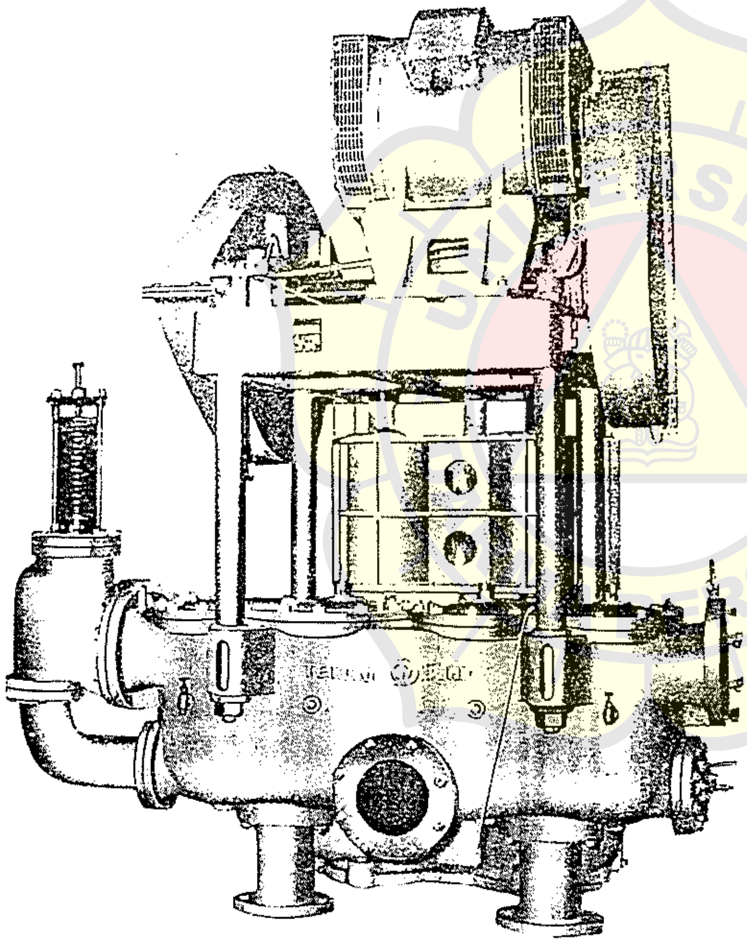
ERTICAL

MOTOR DRIVEN

DUPLEX • DOUBLE ACTING

TYPE

VDP



APPLICATION

- BILGE
- FIRE
- STRIPPING
- FRESH WATER TRANSFER
- F.O./L.O. TRANSFER

TEIKOKU MACHINERY WORKS, LTD.

OSAKA, JAPAN

DIRECT-ACTING STEAM PUMPS

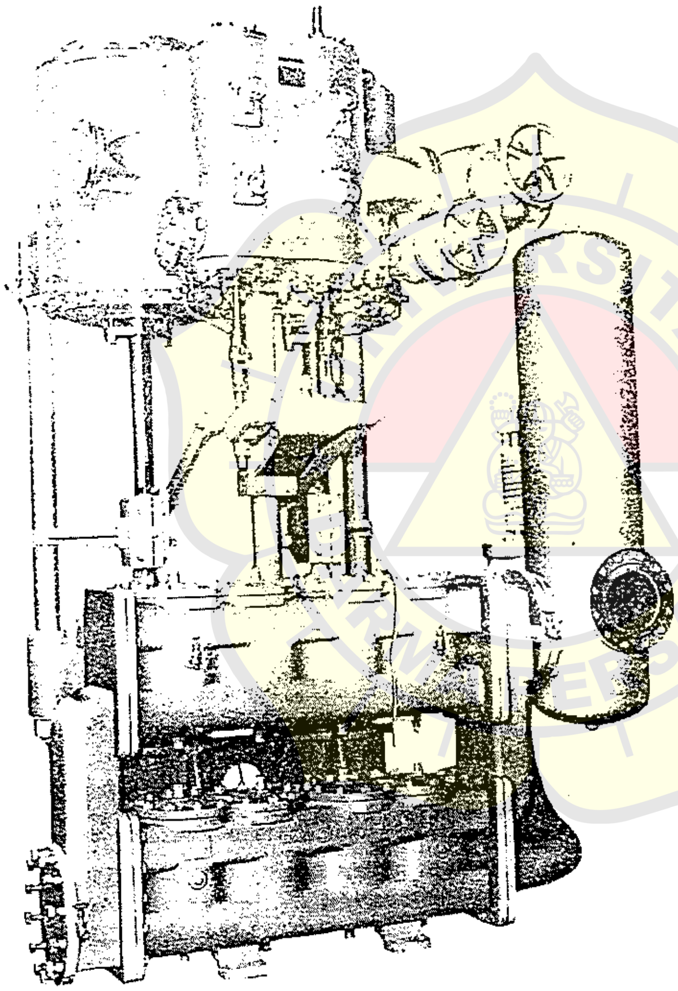
VERTICAL

DUPLEX · DOUBLE ACTING

TYPE
VTW
VTW-S
VTW-SK

APPLICATION

- ORDINARY SERVICE
- CARGO OIL STRIPPING
- TANK CLEANING
- F.O./L.O. TRANSFER



TYPE VTW-SK

TEIKOKU MACHINERY WORKS, LTD.

OSAKA, JAPAN

EXPLANATION ON PERFORMANCE CHART

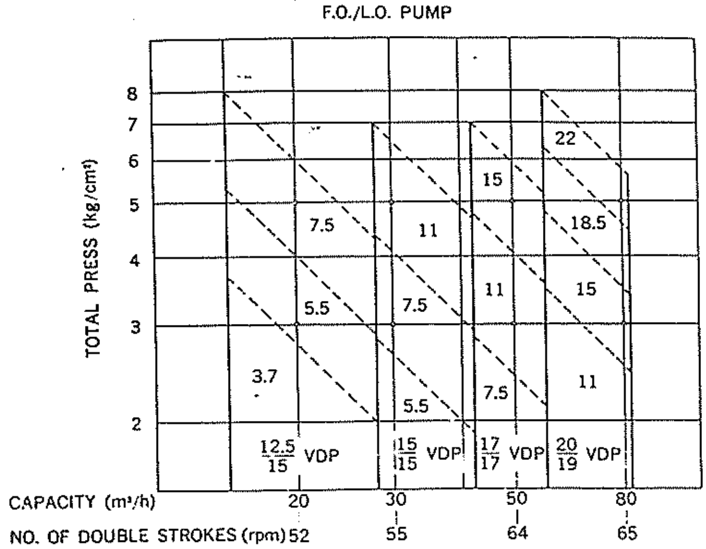
"○" shows the standard specified point of head against

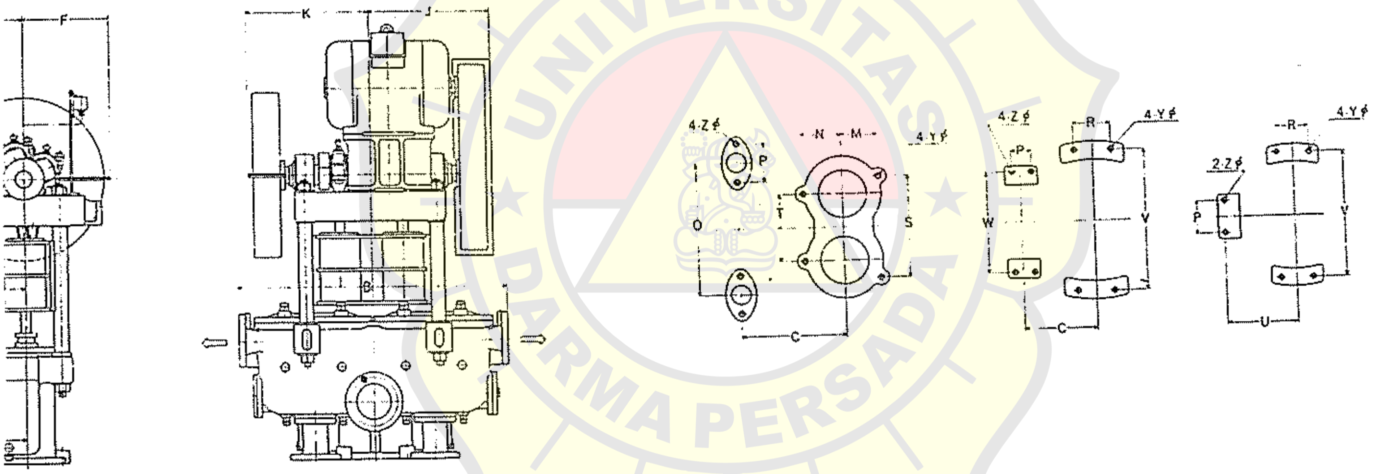
capacity range delivered by an identical type is divided by bold

ıse the service specific point will come on the dividing range ne, it is recommendable in principle to select the smaller type joining ones in the performance chart.

ɔers of double strokes corresponding to any other capacity than ıdard can be obtained by calculation proportioning to the number le strokes corresponding to the standard capacity.

he case of F.O. or L.O. pump, capacity, total pressure and output are based on the oil whose viscocity is 260 cst (1000 Redwood NO. 1).



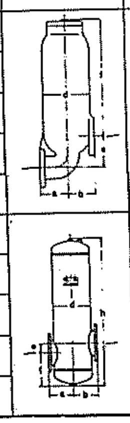


UNIVERSITAS DARMA PERSADA

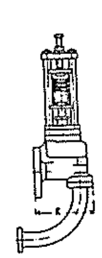
UNIT ; mm

	P	R	S	T	U	V	W	Y	Z	WEIGHT(kg) PUMP (CAST IRON LIQUID END)	AIR CHAMBER d	AIR CHAMBER l	AIR CHAMBER a	AIR CHAMBER b	AIR CHAMBER e	AIR CHAMBER f	ESCAPE VALVE g
-	100	110	—	—	265	345	—	19	15	350	106	—	80	70	90	344	138
-	60	130	—	—	—	400	260	〃	19	370	128	—	90	90	120	410	145
-	〃	160	—	—	—	450	〃	23	〃	445	160	—	150	110	〃	457	180
-	55	180	—	—	—	490	330	〃	〃	700	194	—	140	105	160	510	285
-	70	190	—	—	—	550	400	〃	〃	〃	272	—	160	155	180	675	360
30	135	—	380	280	—	—	—	27	23	1510	356	1500	240	240	50	270	〃
70	180	—	460	300	—	—	—	〃	27	2200	〃	〃	〃	〃	〃	〃	395
70	〃	—	580	400	—	—	—	33	〃	2883	〃	1650	230	230	100	300	462.5
〃	〃	—	〃	〃	—	—	—	〃	〃	〃	〃	〃	〃	〃	〃	〃	〃
30	220	—	660	480	—	—	—	〃	〃	3540	378	1850	270	270	〃	325	505

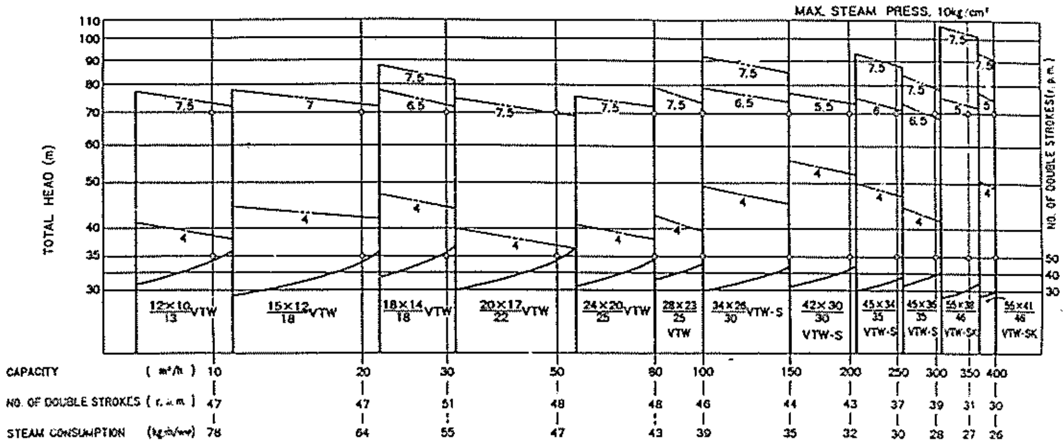
AIR CHAMBER



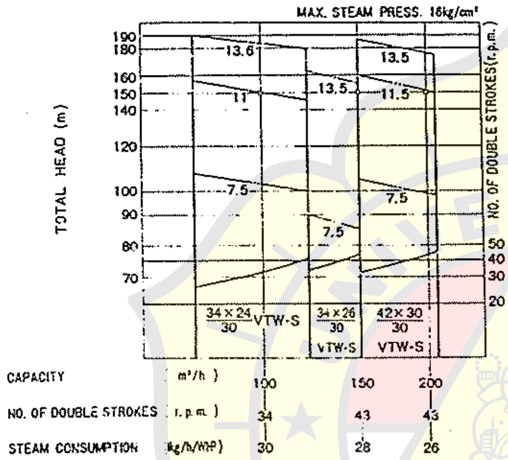
ESCAPE VALVE



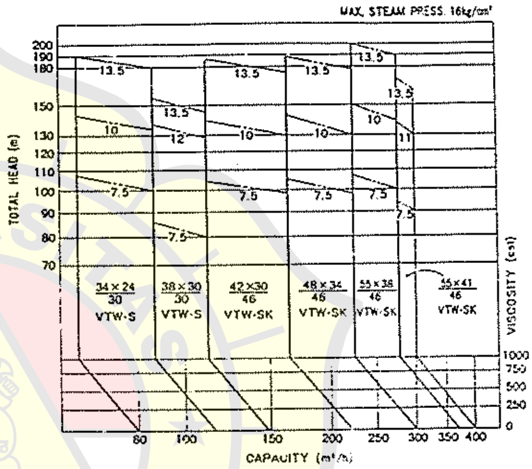
ORDINARY PUMP



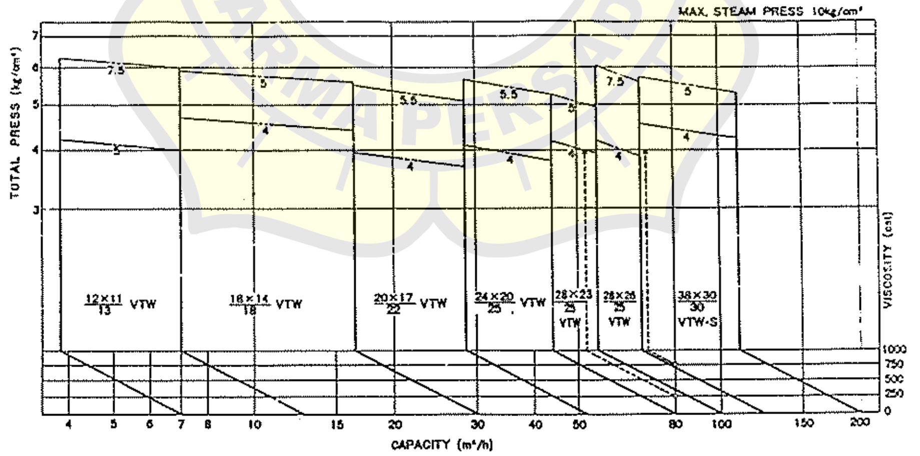
TANK CLEANING PUMP



CARGO OIL STRIPPING PUMP



FUEL OIL/LUBRICATING OIL PUMP



EXPLANATION ON PERFORMANCE CHART

Mark "O" shows the standard specified point of head against capacity. The capacity range delivered by an identical type is divided by bold lines. In case the service specific point will come on the dividing range chart line, it is recommendable in principle to select the smaller type from adjoining ones in the performance chart. The figures beneath the chain line indicate an effective steam pressure (difference between steam and exhaust pressure in slide valve box.)

The rate of steam consumption entered in the chart shows the value for standard capacity against total head 70 meters under the steam pressure 8.5 kg/cm² for general pump and against total head 150 meters under the steam pressure 14.5 kg/cm² for tank cleaning pump. Please pay attention to the fact that the size of L.O. or F.O. pump and stripping pump must be selected according to the viscocity as well as the capacity and total head or pressure of the liquid pumped.

For example, for the specification of capacity 80 m³/h and total pressure 4kg/cm², the results become as shown in the following table.

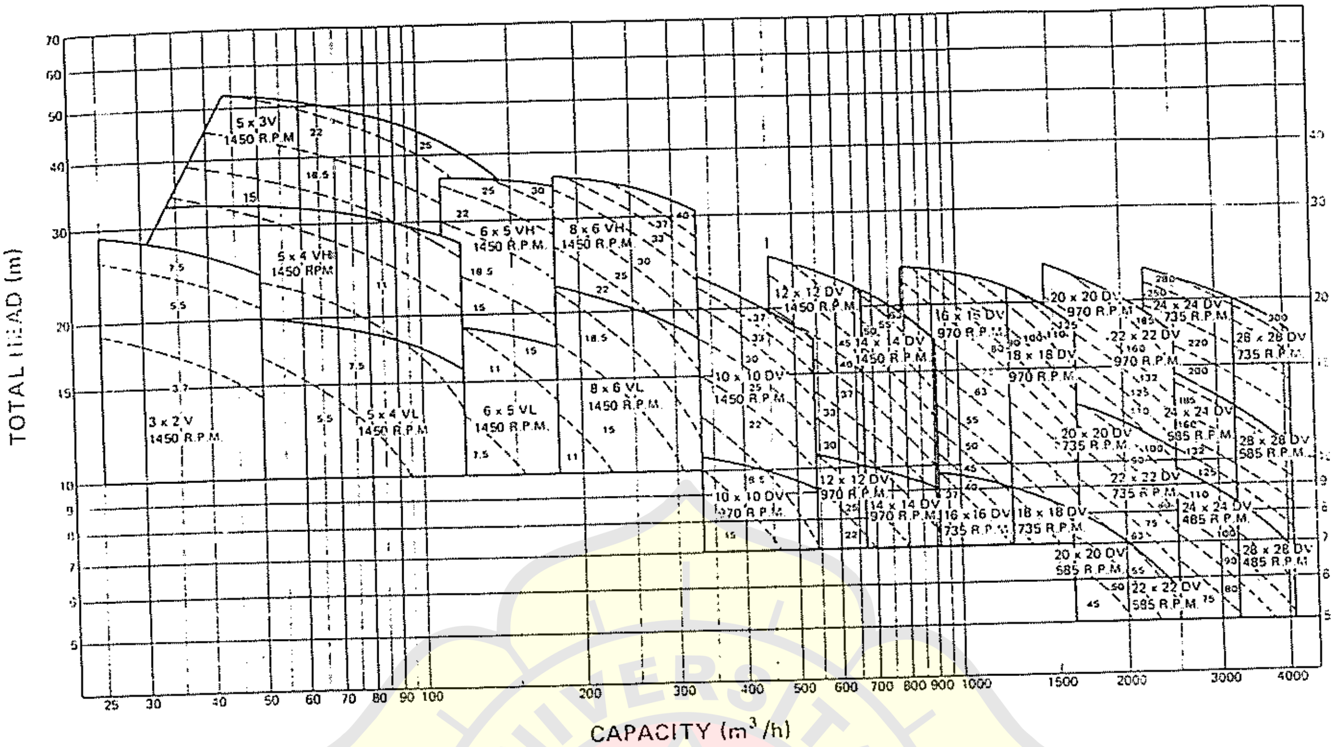
Viscocity	Type	Effective working steam pressure
250 cst	28×23/25 VTW	4 kg/cm²
750 cst	38×30/30 VTW-S	3.3 kg/cm²

CENTRIFUGAL PUMPS

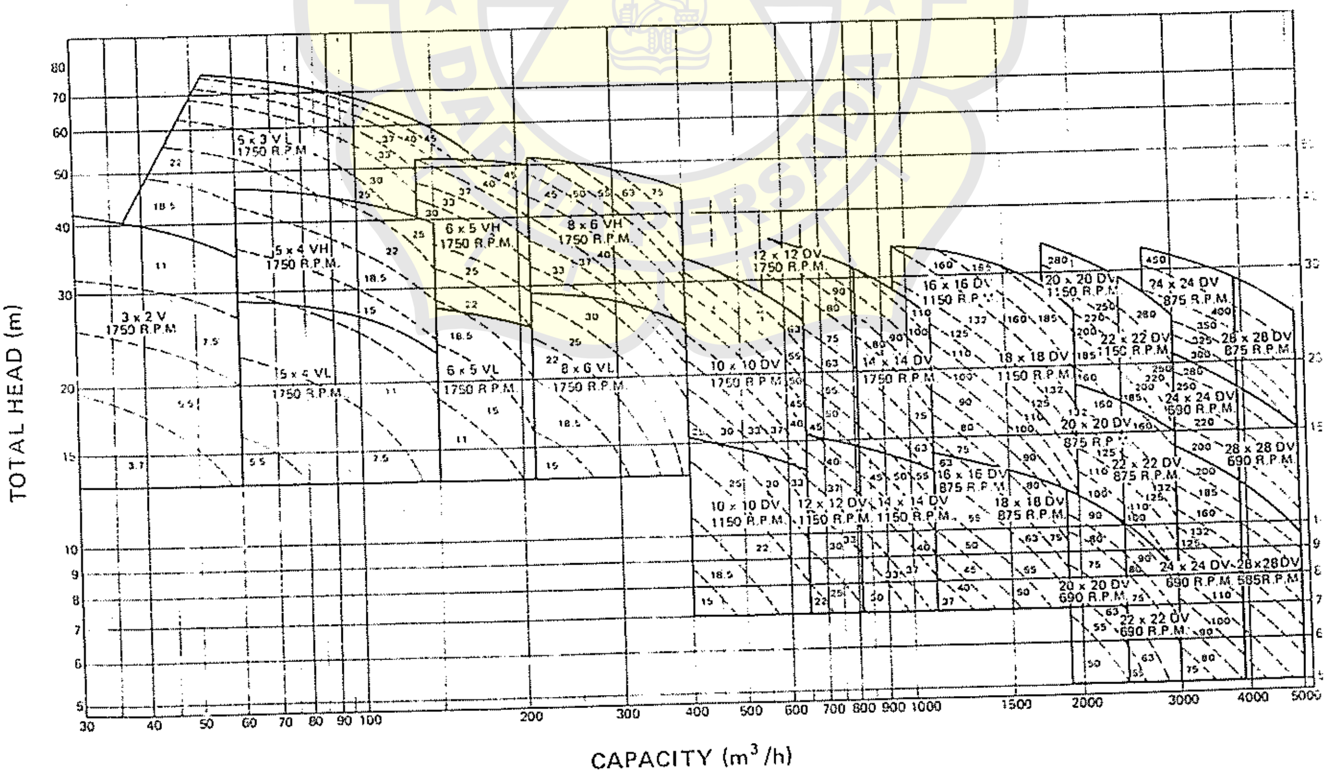
TYPE	CAPACITY (m^3/h)	HEAD (m)	APPLICATION
DOLPHIN	8 – 5,000	13 – 75	Cooling Fresh/Swa Water, S.W. Service, General Service, Ballast
VCS	30 – 350	13 – 60	
VCD	300 – 2,500	13 – 55	
2VCS	30 – 500	30 – 140	Fire & General Service Bilge & Ballast
VCB	35 – 600	25 – 125	
VCSE, 2VCSE, 2VCDS	2 – 160	18 – 145	Condensate Extraction Drain Transfer
VCD-KS, -KSL -KR, -KRL	350 – 12,000	5 – 8	Condenser Circulating
TVC, 2TVC	80 – 1,100	2.5 – 7kg/cm^2	Engine Forced Lubrication
MS	2 – 130	5 – 65	Fresh Water, Potable Water, Sanitary, Fuel Valve Cooling, Hot Water Circulating
2SL, 3SL, 4SL	2 – 85	70 – 280	Feed Water, Emergency Fire Extinguish
DL	130 – 1,100	15 – 65	Cooling Fresh/Sea Water, Ballast
BF	2 – 80	25 – 65	Boiler Water Circulating, Exhaust Gas Economizer Circulating

Performance Ranges

50Hz



60Hz

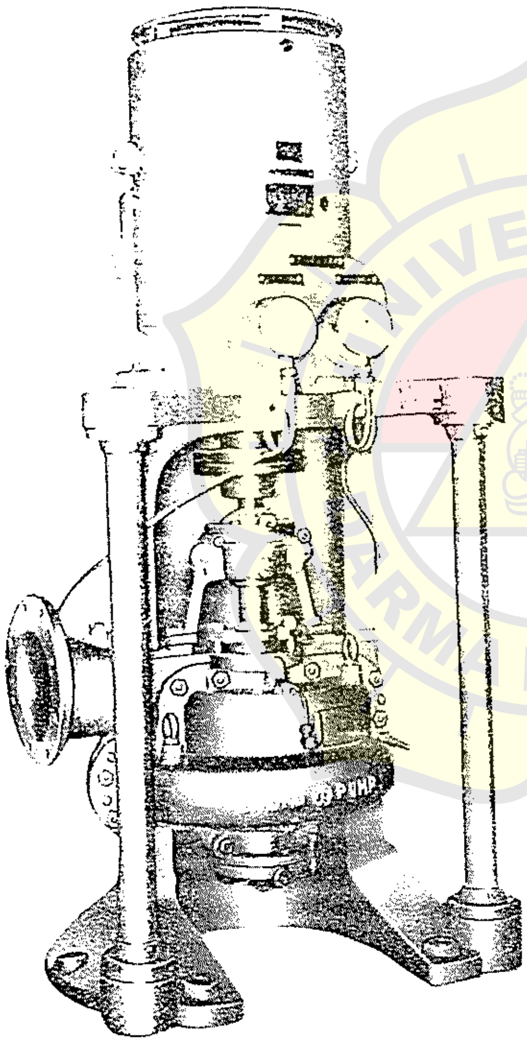


CENTRIFUGAL PUMPS

TYPE

VERTICAL

SINGLE STAGE
SINGLE SUCTION
VERTICALLY SPLIT



APPLICATION

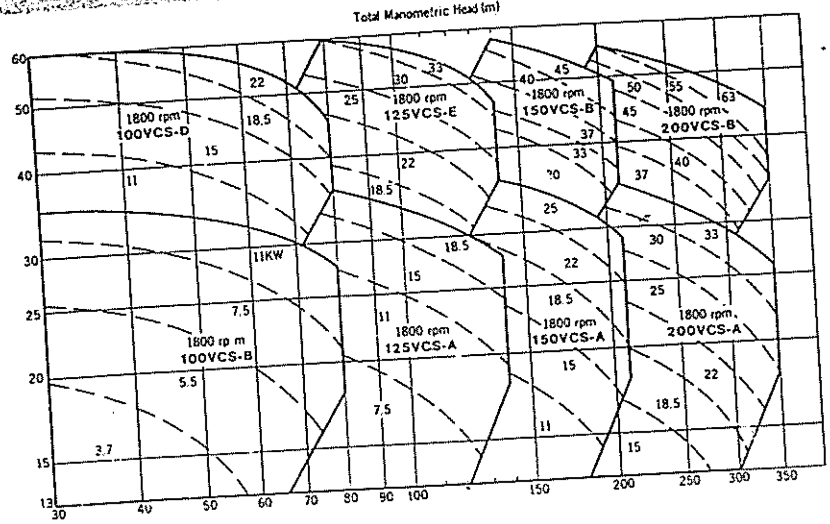
- COOLING FRESH WATER
- COOLING SEA WATER
- SEA WATER SERVICE
- BALLAST & ETC.

UNIVERSITAS DARMA PERSADA

TEIKOKU MACHINERY WORKS, LTD.

OSAKA, JAPAN

PERFORMANCE CHART



EXPLANATION ON PERFORMANCE CHART

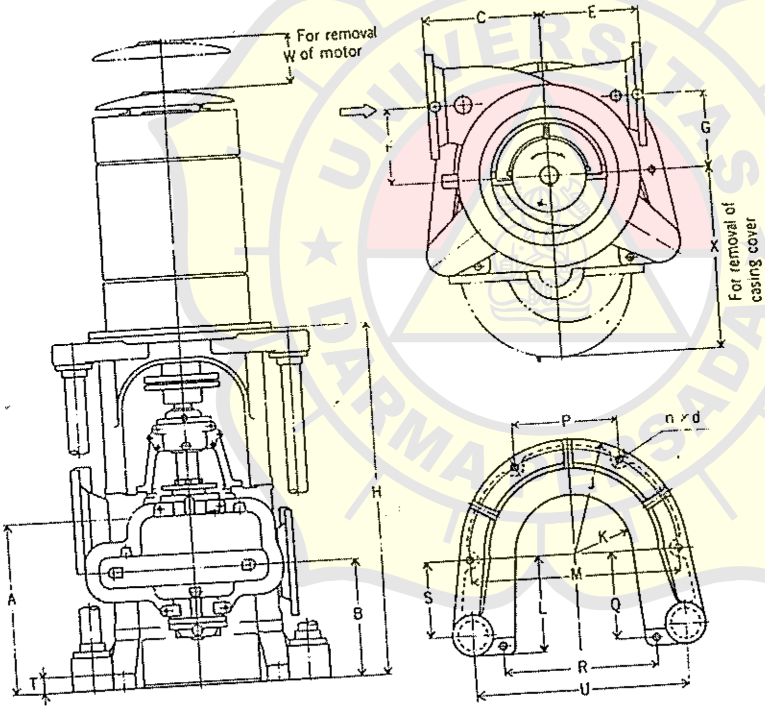
In selecting the size of a pump pattern, if the required specified point of Q-H falls just on the boundary line in the performance chart, please select the smaller size of the nominal bore of the pattern from the adjoining ones.

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

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

Ex. In case, the specified capacity, total head and speed are $125m^3/h$, 30m and 1,750 rpm respectively,
Select 125 VCS-A from the adjoining patterns of 125 VCS-A, 125 VCS-E and 150 VCS-A.
Capacity of the driver, 18.5 KW.

DIMENSIONS



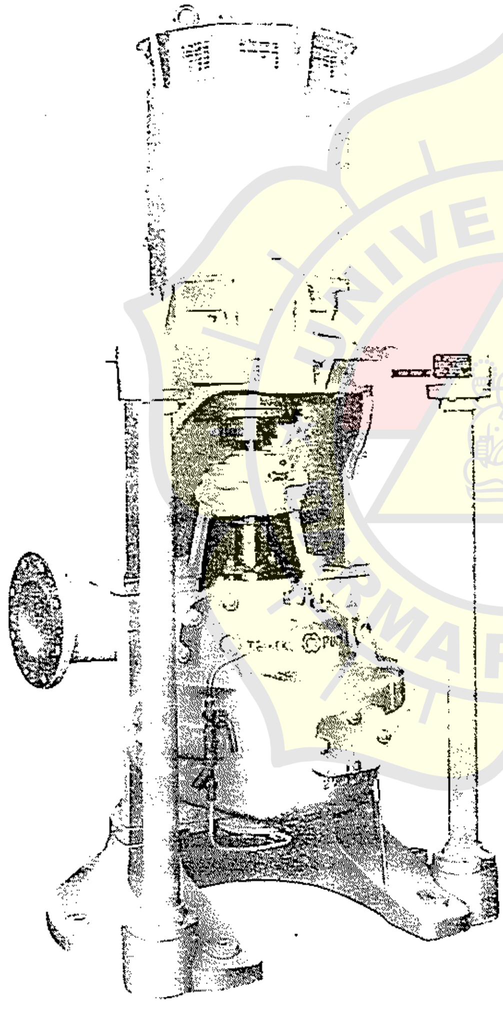
TYPE	MOTOR KW	MOTOR rpm	NOMINAL BORE SUC	NOMINAL BORE DEL	A	B	C	E	F	G	H	J	K	L	M	P	Q	R	S	T	U	nxd	W	X	WEIGHT (kg) PUMP FC CASING	WEIGHT (kg) PUMP BC CASING
100 VCS-B	3.7~11	1800	100	100	357	270	250	220	180	180	790	280	150	260	510	255	235	420	-	30	-	6x23	110	330	218	218
100 VCS-D	11~22				377	290	280	250	210	210	850	"	"	"	"	"	"	"	-	"	-	"	"	400	228	248
125 VCS-A	7.5~18.5		125	125	374	270	250	220	200	200	797	"	"	"	"	"	"	"	-	"	-	"	"	340	248	251
125 VCS-E	18.5~33				394	290	290	260	230	230	880	"	"	"	"	"	"	"	-	"	-	"	140	420	268	275
150 VCS-A	11~25		150	150	456	326	310	270	210	210	901	340	190	280	630	315	255	470	-	"	-	"	110	360	361	371
150 VCS-B	30~45				493	373	340	300	220	220	1061	"	"	"	"	"	"	"	-	"	-	"	140	500	417	427
200 VCS-A	15~33		200	200	486	326	320	280	230	230	953	"	"	"	"	"	"	"	220	"	650	"	"	490	422	427
200 VCS-B	37~63				485	325	350	300	"	"	1022	"	"	"	"	"	"	"	"	"	"	"	"	520	462	472

CENTRIFUGAL PUMPS

TYPE PVC

VERTICAL

TWO STAGE
SINGLE SUCTION
VERTICALLY SPLIT



APPLICATION

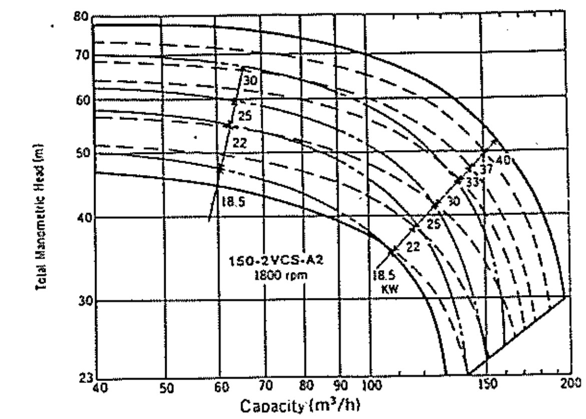
- COOLING FRESH WATER
- COOLING SEA WATER
- FIRE & GENERAL SERVICE
- BILGE & BALLAST & ETC.

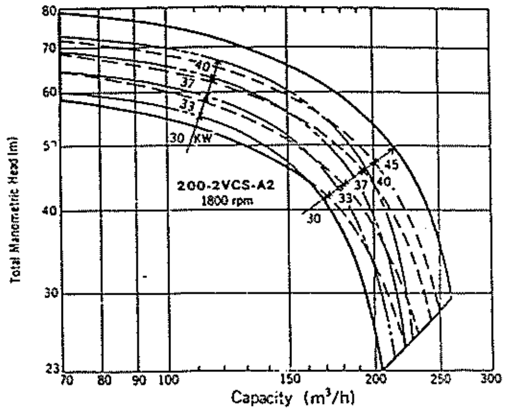
UNIVERSITAS DARMA PERSADA

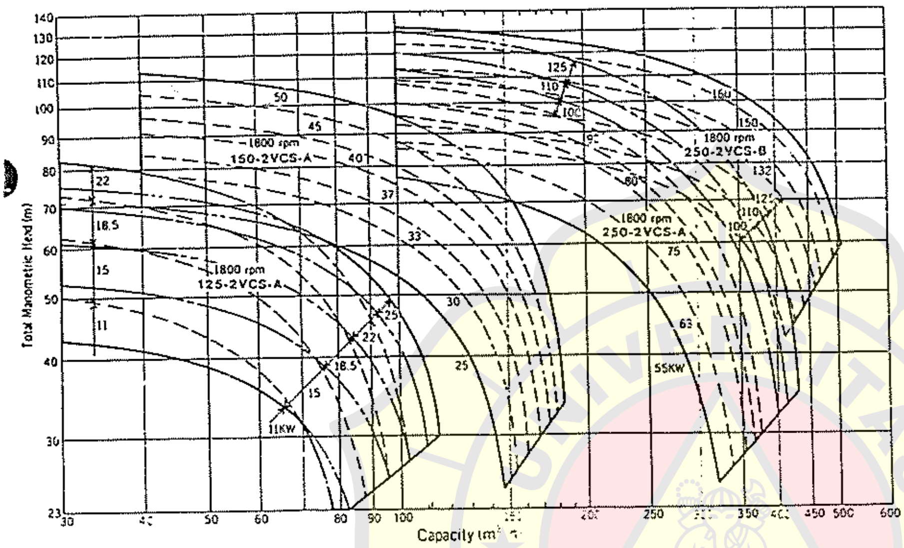
TEIKOKU MACHINERY WORKS, LTD.

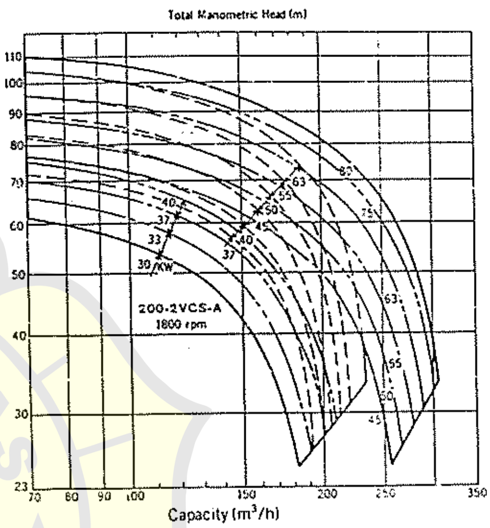
OSAKA, JAPAN

PERFORMANCE CHART





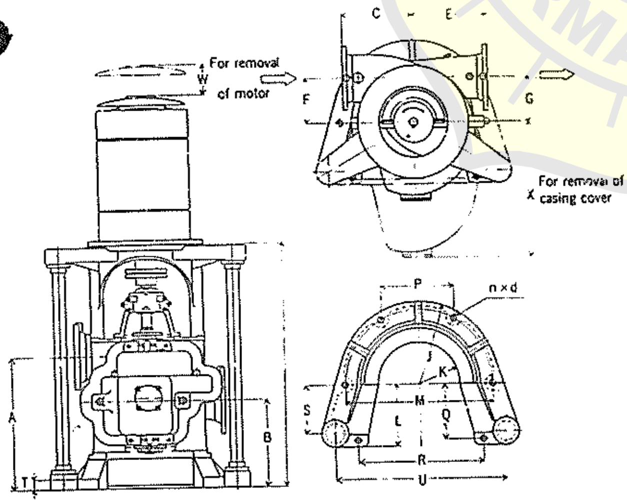




In selecting the size of pump pattern, if the required specified point of Q-H falls just on the boundary lines in the performance chart, please select the smaller size of the nominal bore of the pattern from the adjoining ones.

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

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



TYPE	MOTOR KW	MOTOR rpm	NOMINAL BORE SUC.	NOMINAL BORE DEL.	A	B	C	E	F	G	H	J	K	L	M	P	Q	R	S	T	U	n×d	W	X	WEIGHT (kg) PUMP FC CASING	WEIGHT (kg) PUMP BC CASING
125-2VCS-A	11 ~ 25	1800	125	125	538	370	320	320	180	180	1117	340	190	280	630	315	255	470	220	30	650	6x23	140	570	602	612
150-2VCS-A	18.5~50		150	150	556	366	350	300	220	230	1109	"	"	"	"	"	"	"	"	"	"	"	"	"	705	720
200-2VCS-A	30 ~ 80		200	200	638	417	"	370	"	220	1191	400	205	340	740	370	310	560	270	34	780	6x27	"	620	762	772
250-2VCS-A	55~125		250	250	730	465	400	430	245	245	1298	"	"	"	"	"	"	"	"	"	"	"	"	720	902	907
250-2VCS-B	90~160				750	485	"	"	"	"	1354	450	250	440	840	420	410	630	355	"	930	8x27	176	850	965	975

TYPE

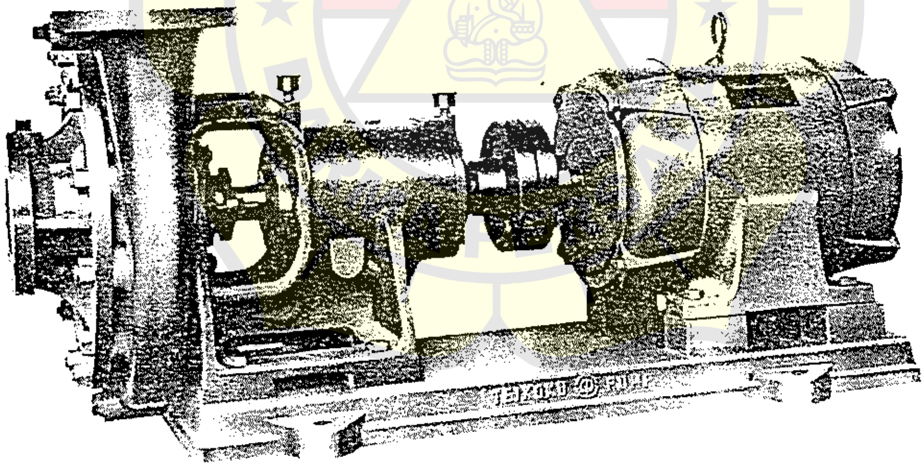
HORIZONTAL

SINGLE STAGE

SINGLE SUCTION

APPLICATION

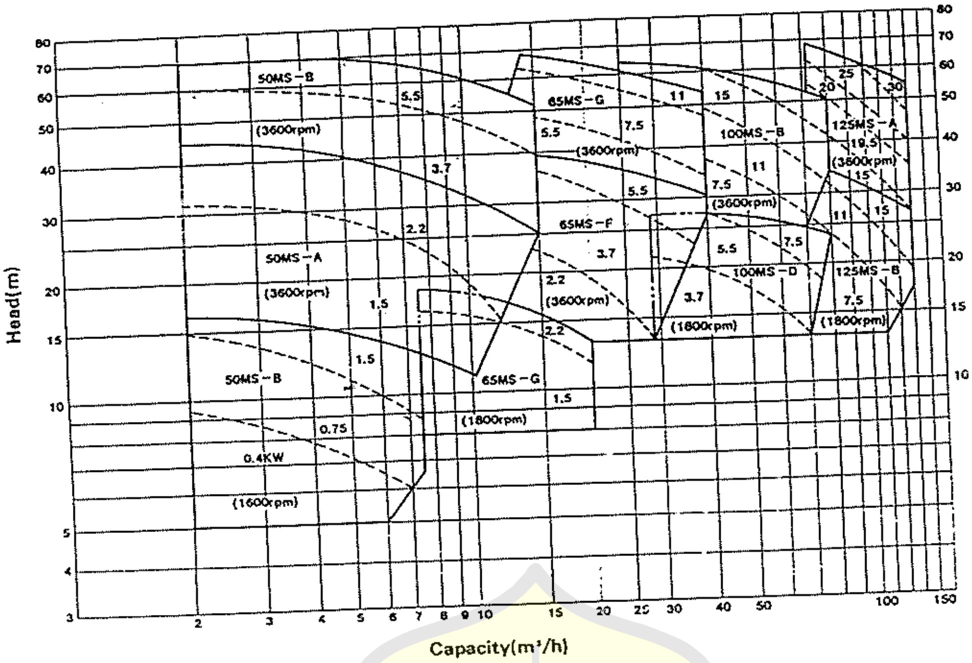
- FUEL VALVE COOLING
- FRESH WATER
- POTABLE WATER
- SANITARY
- HOT WATER CIRCULATING & ETC.



TEIKOKU MACHINERY WORKS, LTD.

OSAKA, JAPAN

PERFORMANCE CHART



EXPLANATION ON PERFORMANCE CHART

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

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

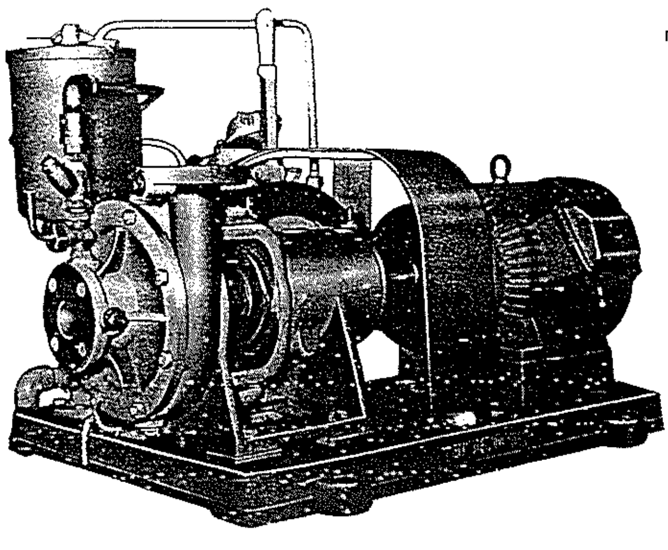
Ex. In case, the specified capacity, total head and speed are 30 m³/h, 15 m and 3,450 rpm, respecitively
Select 50 MS-B from between the adjoining patterns of 50 MS-B and 65MS-F. capacity of driver, 3.7 KW.

SELF PRIMING DEVICE

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

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

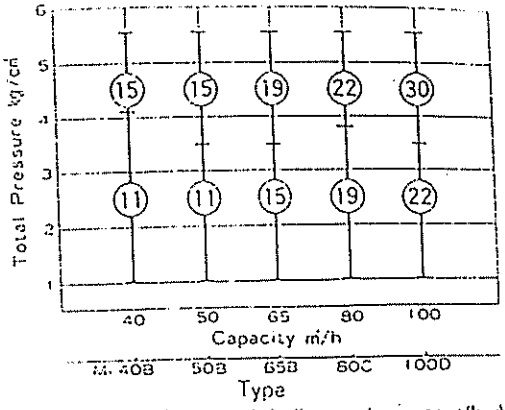
The primer is driven from the main pump shaft through combination coupling and friction pulley. The engagement an disengagement of the pulley are controlled automatically by means of a mechanism which is subjected to the discharg pressure developed by the main pump.



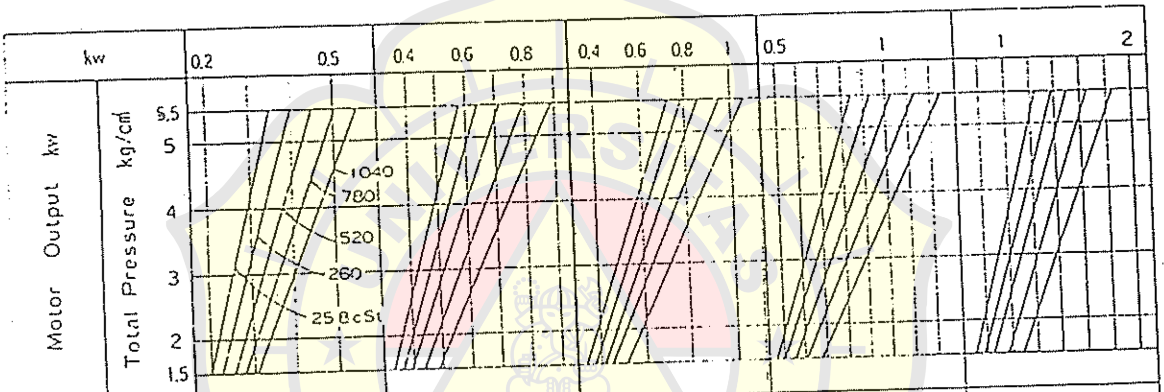
The primer ceases operation automatically on the acc mplishment of the priming of the main pump and remain idle during the main pump is in service.

If the air breaks into the main pump for some reaso resulting in going down of the discharge pressure develo ed by the main pump, the primer begins to work aut matically and the cycle recommences.

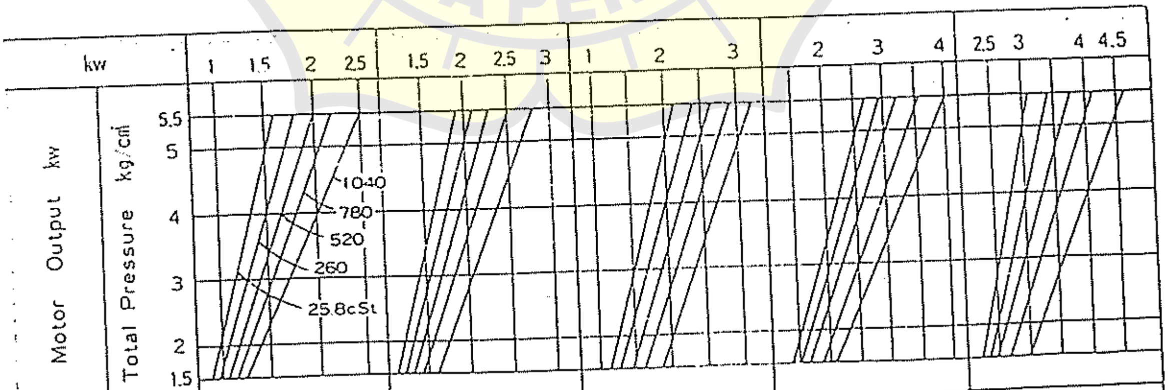
M Type PERFORMANCE CHART



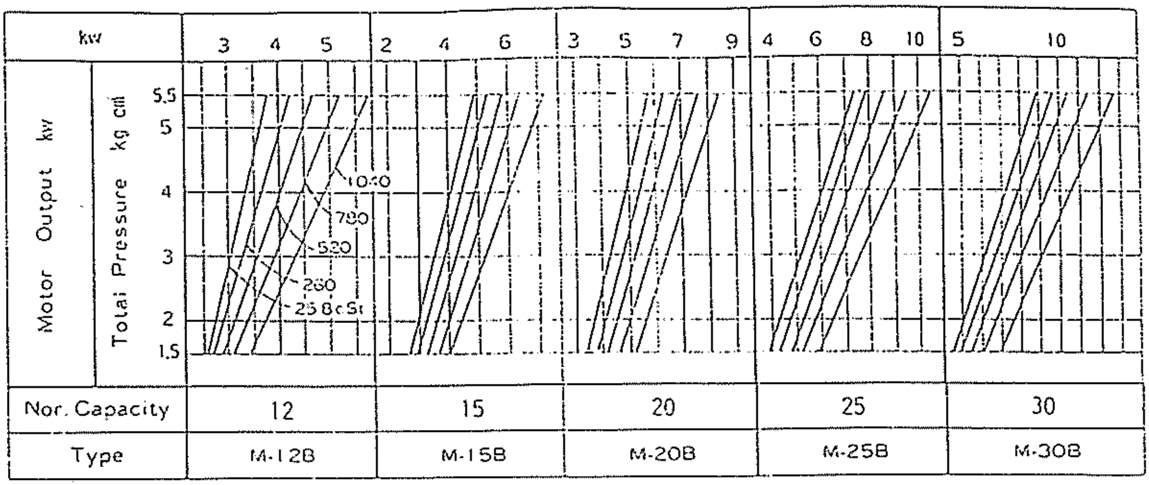
The number in ◯ mark indicates the output(kw) of the motor when 260cSt oil is used.

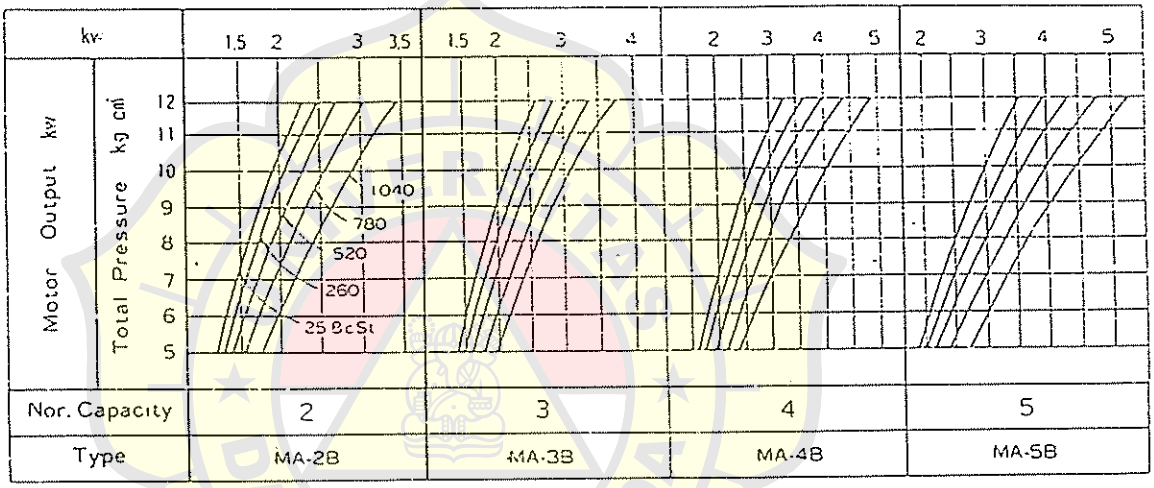


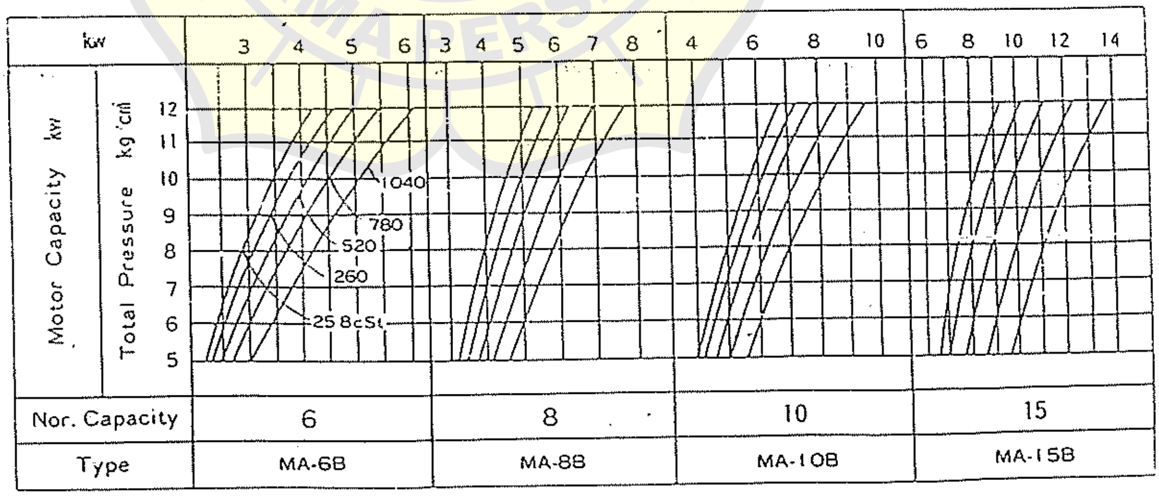
Nor. Capacity	0.5	1	1.5	2	3
Type	M-05B	M-1B	M-2B	M-2B	M-3B



Nor. Capacity	4	5	6	7.5	10
Type	M-4B	M-5B	M-6B	M-7.5B	M-10B







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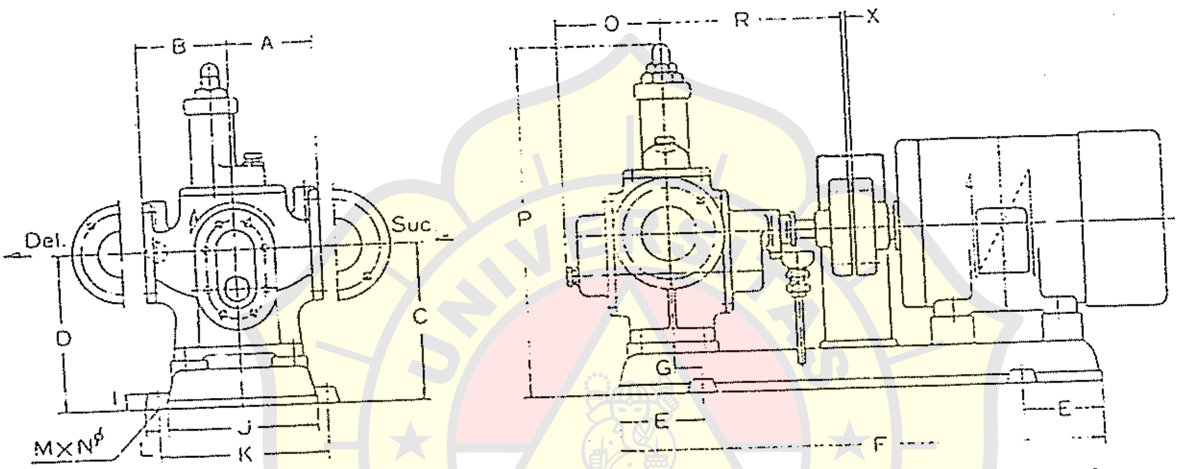
Type



Dimensions—mm

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