Türbo San®

TS-EN-ISO 9905

PUMP HANDBOOK

K - Q
TSE-ISO-EN
9000

TS EN ISO 9001 : 2000

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FOREWORD

The Centrifugal Pumps Manual is prepared in order to meet the basic requirements of the technical people dealing with pumps and to enable them to reach easily the information they may need.

The Manual covers the basic concepts related with pumps, the structure of pumping installations and pumping stations, operational information about the maintenance and repairs of pumps, various basic engineering calculations and tables. We hope that this Manual will be helpful for technical people interested with pumps.

I thank Mr. Burhan SİLAHTAROĞLU our President of the Board, Mr: Sedat SİLAHTAROĞLU our General Manager, Prof. Dr. Erkan AYDER our Advisor, Mr. Ufuk KALMANOĞLU our Sales and Marketing Manager and all our staff members, those of the R&D and Technology Department being at the lead, for their support and contribution to the preparation of this book..

TÜRBOSAN TÜRBOMAKİNALAR
SANAYİ VE TİCARET A.Ş.

TÜRBOSAN HISTORY

Türbosan A.Ş. has been established in 1971.

1971	Irrigation and sprinkling pumps for agricultural purposes
1972	Norm pumps produced
1973	Multi-stage pumps produced
1974	EKS-E/K Axial and Mixed Flow pumps produced
1975	Double suction pumps produced
1978	Monopump produced
1979	CAP series sewage and sewer system pump produced
1980	DAC type submersible sewage and sewer system pump produced
1981	Motor driven and manual butterfly valve produced
1982	Non-return valve produced
1981	Vertical axis double suction pump produced
1982	Turbine (Banki type) produced
1982	Türbosan exports for the first time
1983	Process pumps produced
1983	Francis and Kaplan type turbines produced
1983	Screw type pump produced
1986	DAÇ-E/K tube type submersible pump produced
1987	DAS submersible pumps serial production
1988	Shredder blade pumps serial production
1990	Entering international markets
1991	Deep well submersible pumps introduced in the market
1991	Gasoline and self priming motor pumps introduced in the market
1993	ÇEP 700/710 double suction pump produced (Q_{max}: 11000 m³/h)
1994	ISO 9001 certificate received
1994	EKS-K 1200 Axial column pipe pump produced (Q_{max}: 12000 m³/h)
1995	Domestic sewage and sewer system stainless submersible pumps introduced in the market
1999	Smallest ÇEP 65/250 pump produced
2002	400 kW Cooling jacket sewerage submersible pump produced
2003	The biggest screw type pump produced (Q_{max}: 5000 m³/h)
2004	Light submersible pump produced
2004	Pump selection program prepared
2005	Three dimensional (3D) projects prepared
2005	Cooling jacket submersible pump series completed
2005	Double suction and Norm pump series pumps developed
2006	Pump handbook prepared
2006	Pump design analyses carried out by computers
2006	Submersible pump series developed
2006	Pump flow analyses performed by developed softwares (3D)
2007	Introduction of pump set groups.
2007	Fire fighting sets introduct to the market.

TABLE OF CONTENTS :

ANNEX - TABLES

Table : 1 Continuous Pipe Losses

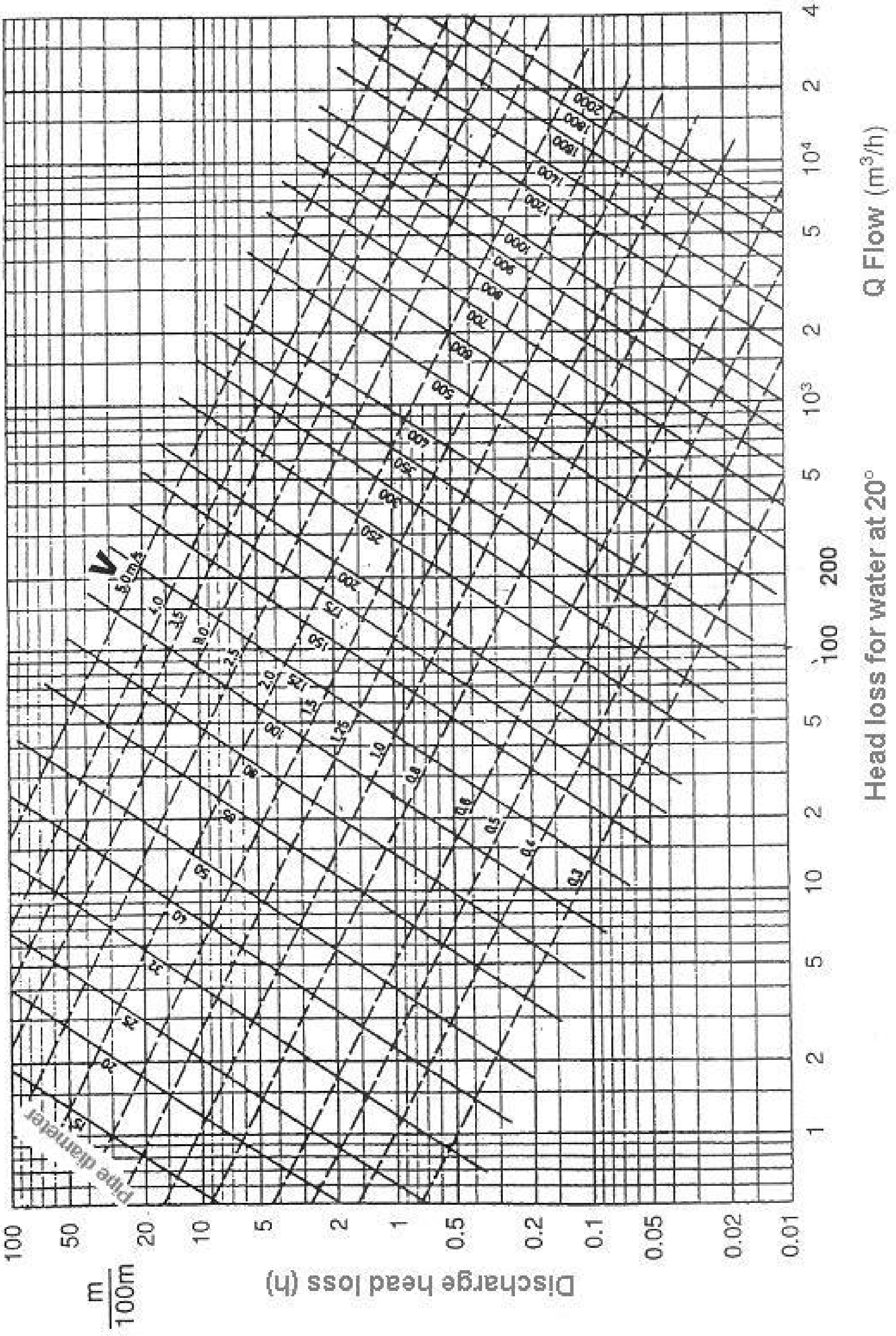
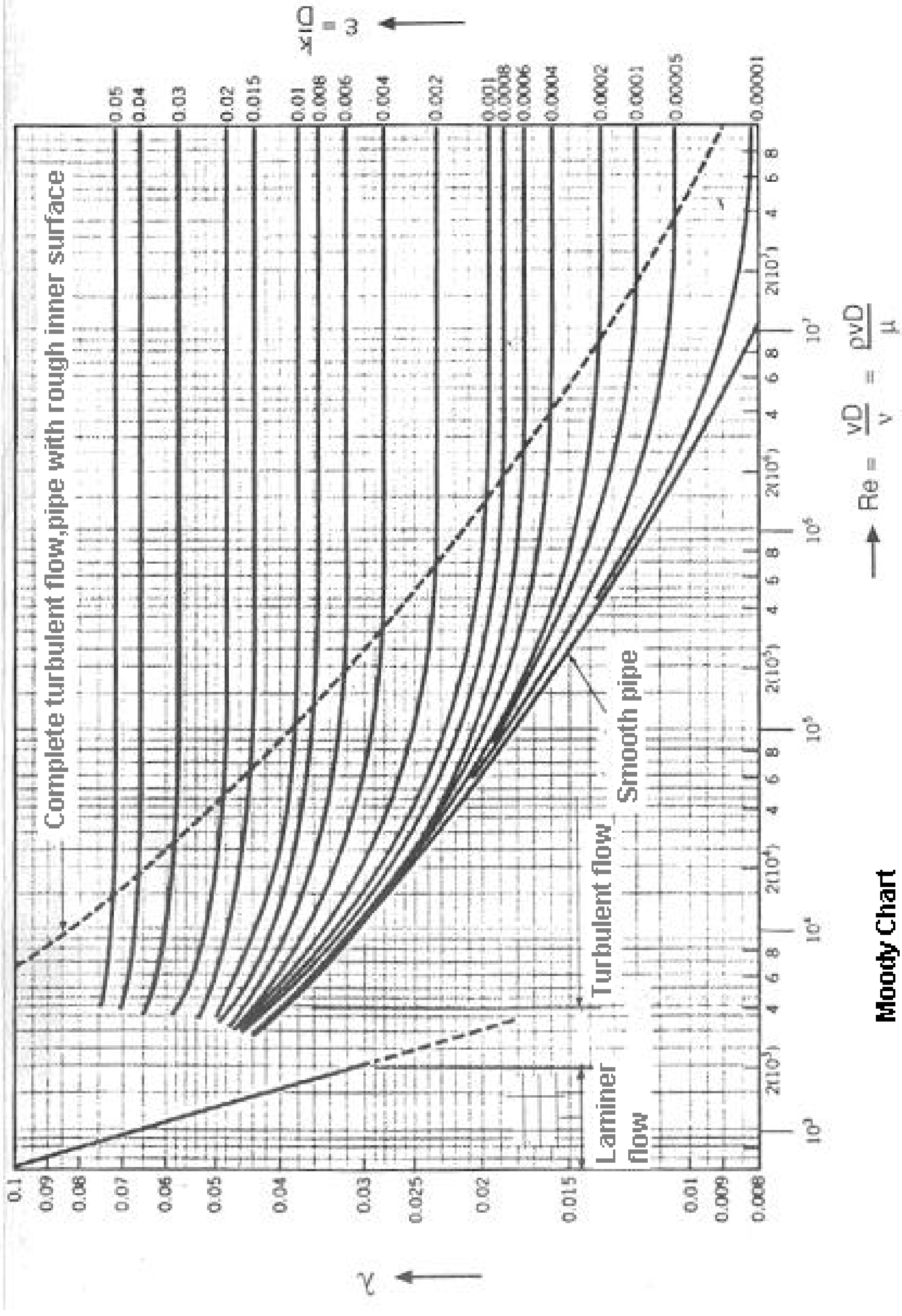


Table : 2 Moody Diagram



Moody Chart

Table : 3 Kinematic Viscosities and Densities of Fluids

Table 3: Kinematic viscosities and densities of fluids

Sıvı	Bağıl yoğunluk (SG)	Kinematik viskozite-ν (centistokes)	Sıcaklık t (°C)
Freon-12	1.33	0.27	21
Glycerin (100%)	1.26	648 / 176	21 / 38
(50% water)	1.13	5.29	20
Acetaldehyde	0.762	0.295	20
Glycol-propylene	1.038	52	21
Triethylene	1.125	40	21
Diethylene	1,12	32	21
Ethylene	1,125	17.8	21
Hydrochloric acid (31,5%)	1,05	1.9	20
Mercury	13,57	0,12/0,11	21 / 38
Phenol (Carbolic acid)	0,95 - 1,08	11,8/1,17	18 / 90
Soda silicate	40 Bome	79	38
	42 Bome	138	38
Toluene	0.866	0,68	20
Bromine	2,9	0,34	20
Mineral Oils SAE 10	0,880-0,935(16°C)	35 - 52	38
	"	18 - 25	54
SAE 20	"	52 - 87	38
	"	25 - 40/6-10	54/99
SAE 30	"	87 - 125	38
	"	40 - 55/10-13	54/99
SAE 40	"	125 - 206	38
	"	75/13 - 17	54-99
SAE 50	"	206-352/16-22	38/99
SAE 60	"	352-507/22-26	38/99
SAE 70	"	507-682/26-32	38/99
SAE 80	"	22000 (maks)	-18
SAE 90	"	173 - 325	38
		65-108/14-25	54/99
SAE 140	"	206-507/25-43	54/99
SAE 150	"	43 (min)	99
SAE 5W	"	1295 (max)	-18
SAE 10W	"	1295-2590	-18
SAE 20W	"	2590-10350	-18
SAE 75W	"	4.2	99
SAE 80W	"	7.0	99
SAE 85W	"	11.0	99
Diethylene glycol	1,12	32	21
Acetic acid (5% vinegar)	1.006	-	15
%10	1.014	1.35	15
%50	1.061	2.27	15
%80	1.075	2.85	15
Acetone	0.792	0.41	20
Methyl alcohol	0.79	1,04 / 0,74	0 / 15
Aluminum sulfate(%36)	1.055	1.41	20
Ammoniac	0.662	0.3	-18
Aniline	1,022 / 1,035	4,37 / 6,4	20 / 10
Jet fuel	0,72 - 0,84	7,9 / 1,1	-35 / 16
Calcium chloride %5	1.04	1.156	18
%25	1.23	4	16
Carbon tetrachloride	1.594	0,612 / 0,53	20 / 38
Carbon disulfide	1,293 / 1,263	0,33 / 0,30	0 / 20
Diethyl ether	0,714	0.32	20
Ethyl acetate	0,907 / 0,90	0,49 / 0,40	15 / 20
Ethylene bromide	2.18	0.787	20
Ethylene chloride	1.246	0.688	20
Ethyl bromide	1.45	0.27	20
Formic acid %10	1.025	1.04	20
%50	1.121	1.2	20
%80	1.186	1.4	20
Methyl acetate	0.93	0.44	20
Naphthalene	1,145	0.9	80
Sodium chloride %5	1,037 (4°C)	1.097	20
%25	1,19	2.4	16
Sodium hydroxide %20	1.22	4	18
%30	1.33	10	18
%40	1.43	24	18
Propyl alcohol	0,817 / 0,804	2,8 / 1,4	20 / 50
Ascetic acid anhydride	1.087	0.88	15
Butyric acid -n	0.959	2,4 / 1,6	0 / 20
Furfural	1.159	1.45	20
Chloroform	1,489 / 1,413	0,38 / 0,35	20 / 60
Diesel fuel No.2 D	0,82 - 0,95	2-6 / 1-4	38/54
3 D	"	6-12 / 4-6,8	38/54
4 D	"	30 / 13	38/54
5 D	"	87 / 35	50/71

Sıvı	Bağıl yoğunluk (SG)	Kinematik viskozite-ν (centistokes)	Sıcaklık t (°C)
Engine oil #10	0,88 - 094	34-72 / 18-25	38/54
#20	"	72-83 / 25-39	38/54
#30	"	75-119 / 39-55	38/54
Fuel - oil No.1	0,82 - 0,95	2,4 - 4,3	21
	"	2.7	38
No.2	"	3 - 7,4	21
	"	2,1 - 4,3	38
No.3	"	2,7 - 0,6	38
	"	2,1 - 4	54
No.5 A	"	7,4 - 26,4	38
	"	4,9 - 13,7	54
No.5 B	"	26,4 - 86,6	38
	"	13,6 - 67,1	54
No.6	"	97,4 - 660	50
	"	37,5 - 172	71
Fuel - oil (Ship)	1. (maks)	325 (maks)	50
	"	104 (maks)	71
Gasoline	0.74	0,88 / 0,71	16 / 38
	0.72	0.64	16
	0.68	0,46 / 0,40	16 / 38
Cut-off fluid (oil) #1	-	30-40/17-23	38 / 54
#2	-	40-46/23-26	38 / 54
Kerosene	0.78 - 0.82	2,7 - 3,8	20
	"	2	38
Ethyl alcohol	0.79 / 0.77	1,52 / 1,20	20 / 38
Benzene	0.90 / 0,88	1,0 / 0,74	0 / 20
Glucose	1.35 - 1.44	7700 - 22000	38
	"	880 - 2420	65
Honey	1.5	74	38
Milk	1.02 - 1.05	1.13	20
Corn starch solution (22 Baume)	1.18	32,1 / 27,5	21 / 38
Crude oil 48° API	0,79 / 0,76	3,8 / 1,6	16 / 54
40° API	0,825 / 0,805	9,7 / 3,5	16 / 54
32,6°	0,862 / 0,840	23,2 / 7,1	16 / 54
Sulfuric acid % 100	1.839	14.6	20
% 60	1,50	4.4	20
Sea water	1.03	1.15	16
Heat transfer oil	0.875	90 - 100	15.5
	0.82	4,6 - 5,1	100
	0.76	1,2 - 1,3	200
	0.725	0,79 , 0,85	250
	0.685	0.55	320
Hydraulic oil	0.875	400-600/28-46	0 / 40
Grape molasses A	1,40 - 1,46	280 - 5070	38
	"	150 - 1760	55
B	1,43 - 1,46	1410 - 13200	38
	"	660 - 3300	55
C	1,46 - 1,49	2630 - 55000	38
	"	1320 - 16500	55
Sugar syrup 60 Brix	1.29	50	21
	"	19	38
64 Brix	1.31	95	21
	"	32	38
68 Brix	1.34	216	21
	"	60	38
72 Brix	1.36	595	21
	"	139	38
76 Brix	1.39	2200 / 440	21 / 38
Crème	1.02	20	4
Caramel	1.2	400	60
Chocolate	1.1	17000	50
Tallow	0.92	9	100
Olive oil	0,912 - 0,918	43 / 24	38 / 54
Corn oil	0.923	39,6 / 23	38 / 54
Soya oil	0,93 - 0,98	35 / 20	38 / 54
Beer	1,02 - 0,04	1.1	4
Face cream (cosmetic)	1.4	7000	20
Shampoo	1.4	3500	20
Ketch-up	1.11	530	60
Orange juice	1.1	4500	3
Mayonnaise	1	5000	24
Printing ink	1, - 1,38	550 - 2200	38
	"	240 - 660	54
Varnish	0.9	313 / 143	20 /38
Solvent	0,8 - 0,9	6 - 12	20 / 38
Waste water (purified)	1.1	2000	20 / 38

Table : 4 Roughness heights of various materials

Pipe name and type			K (mm)
Cast pipe	Bitumen coated		0,20 - 0,30
	Without coated		0,25 - 0,50
	Concrete covered		0,025 - 0,15
	Old and used		1,20 - 3,0
Steel pipe	Galvanized		0,10 - 0,15
	Bitumen coated		0,05 - 0,10
	Concrete covered		0,025 - 0,10
	Black pipe		0,04 - 0,10
	Old and used		0,15 - 1,50
Concrete pipe	Smooth pipe		0,30 - 0,80
	Rough pipe		2. - 3
Glass fiber reinforced plastic			0.05
Asbestos cement pipe			0,03 - 0,12
Aluminum, copper, lead, brass		New	0.02
		Used	0.04
Hose	Cloth reinforced rubber		0,7 - 0,12
	Plain cloth		1.3
	Garden hose		0.75
Plastic pipe			0,03 - 0,10

Table : 5 Loss coefficient of local elements

	25	40	50	65	80	100	125	150	200	250	300	350	400	500	600	700	800	900	1000
Gate Valve (Threaded)	0.25	0.20	0.15	0.14	0.13	0.12	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10
Gate Valve (Flanged)	0.55	0.50	0.45	0.40	0.35	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30
Angle Valve - Vertical (Threaded)	5.00	3.00	2.00	1.60	1.30	1.00													
Angle Valve - Vertical (Flanged)	3.10	3.40	3.80	4.10	4.40	4.70	5.00	5.30	5.70	6.00	6.30	6.50	6.60						
Angle Valve - Angled (Flanged)	1.60	1.60	1.60	1.60	1.60	1.60	1.60	1.60	1.60	1.60	1.60	1.60	1.60						
Buterfly Valve (PN 16)	3.00	2.75	2.50	2.30	2.25	2.15	2.05	1.95	1.70	1.57	1.45	1.35	1.20	1.00	0.85	0.78	0.72	0.66	0.60
Globe Valve	0.07	0.07	0.06	0.06	0.05	0.05	0.05	0.05	0.04	0.04	0.04	0.04	0.04						
Valve	3.50	3.20	2.90	2.80	2.70	2.60	2.48	2.35	2.10	2.05	2.00								
Needle Valve	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80					
Tap	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15					
Check Valve Swing type (without balance weight) v=1m/s	3.800	3.400	3.100	3.080	3.040	3.000	2.97	2.95	2.90	2.90	2.90	2.90	2.90	2.80					
Check Valve Swing type (without balance weight) v=2m/s	1.70	1.55	1.40	1.40	1.40	1.40	1.37	1.35	1.30	1.30	1.30	1.25	1.20	1.10					
Check Valve Swing type (without balance weight) v=3m/s	1.10	1.00	0.90	0.90	0.90	0.90	0.87	0.85	0.80	0.80	0.80	0.75	0.70	0.60					
Check Valve Swing type (with balance weight) v=1m/s	9.50	8.50	7.75	7.70	7.60	7.50	7.43	7.38	7.25	7.25	7.25	7.25	7.25	7.00					
Check Valve Swing type (with balance weight) v=2m/s	4.25	3.88	3.50	3.50	3.50	3.50	3.43	3.38	3.25	3.25	3.25	3.13	3.00	2.75					
Check Valve Swing type (with balance weight) v=3m/s	2.75	2.50	2.25	2.25	2.25	2.25	2.18	2.13	2.00	2.00	2.00	1.88	1.75	1.50					
Check Valve Wafer type v=1m/s	29.00	13.00	5.00	4.90	4.76	4.60	3.65	2.70	2.45	2.05	2.30	2.65	1.80						
Check Valve Wafer type v=2m/s	12.00	9.00	8.00	5.00	4.47	4.30	3.35	2.40	2.30	2.00	2.20	2.50	1.80						
Check Valve Wafer type v=3m/s	9.00	7.00	7.00	5.20	4.80	4.30	3.40	2.50	2.25	2.00	2.10	2.35	1.90						
Check Valve Tilting type a=5^{o}	0.90	0.80	0.76	0.74	0.72	0.68	0.65	0.62	0.56	0.52	0.48	0.44	0.39	0.24					
Check Valve Tilting type a=15^{o}	2.40	2.35	2.3	2.25	2.15	2.0	1.92	1.85	1.7	1.60	1.5	1.35	1.2	0.7					
Foot Valve (with strainer)	1.70	1.55	1.40	1.37	1.35	1.30	1.25	1.20	1.10	1.10	1.00	0.97	0.95	0.90					
Check Valve Globe type	70.00	70.00	70.00	70.00	70.00	70.00	70.00	70.00	70.00	70.00									
Check Valve Wafer type (with disc, spring) v=1m/s	4.00	6.00	7.00	7.20	7.35	7.60	8.25	8.90	9.30	9.80									
Check Valve Wafer type (with disc, spring) v=2m/s	4.40	4.80	5.00	5.60	6.20	7.00	7.25	7.50	8.50	9.40									
Check Valve Wafer type (with disc, spring) v=3m/s	4.40	4.60	4.70	5.40	6.00	6.90	7.60	8.30	7.80	10.90									
Hydrostop v=2m/s	4.50	4.75	5.00	5.3	5.60	6	6.35	6.75	7.5	7.60	7.25	7.10	7						
Hydrostop v=3m/s	1.30	1.50	1.8	2.5	3.20	4	4.00	4.00	4	3.85	3.70	3.55	3.4						
Hydrostop v=4m/s	4.40	4.20	3.9	3.7	3.50	3	2.90	2.75	2.5	2.40	2.35	2.30	2.2						
Strainer	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80	2.80						
Submerged pipe	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00

	25	40	50	65	80	100	125	150	200	250	300	350	400	500	600	700	800	900	1000
Sump outlet rectangular	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Sump outlet cicular r/d=0.02	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28
Sump outlet cicular r/d=0.06	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15	0.15
Sump outlet cicular r/d=0.10	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09
Sump outlet cicular r/d>0.15	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.04
Suction pipe entry (conical)	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20
Suction pipe entry (tapered)	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Sump outlet (inclined) a=45^o	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80	0.80
Sump outlet (inclined) a=60^o	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70
Sump outlet (inclined) a=75^o	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60	0.60
Sump entry	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Muff	0.08	0.07	0.05	0.05	0.04	0.04													
Tapered muff d2/d1=0.3	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Tapered muff d2/d1=0.5	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40
Tapered muff d2/d1=0.7	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30
Tapered muff d2/d1=0.9	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10	0.10
Sudden area increase d1/d2=0.8	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13
Sudden area increase d1/d2=0.7	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26	0.26
Sudden area increase d1/d2=0.6	0.41	0.41	0.41	0.41	0.41	0.41	0.41	0.41	0.41	0.41	0.41	0.41	0.41	0.41	0.41	0.41	0.41	0.41	0.41
Sudden area increase d1/d2=0.5	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56	0.56
Sudden area increase d1/d2=0.4	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70
Sudden area increase d1/d2=0.3	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83	0.83
Sudden area increase d1/d2=0.2	0.92	0.92	0.92	0.92	0.92	0.92	0.92	0.92	0.92	0.92	0.92	0.92	0.92	0.92	0.92	0.92	0.92	0.92	0.92
Sudden area decrease d2/d1=0.8	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18	0.18
Sudden area decrease d2/d1=0.7	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25	0.25
Sudden area decrease d2/d1=0.6	0.32	0.32	0.32	0.32	0.32	0.32	0.32	0.32	0.32	0.32	0.32	0.32	0.32	0.32	0.32	0.32	0.32	0.32	0.32
Sudden area decrease d2/d1=0.5	0.38	0.38	0.38	0.38	0.38	0.38	0.38	0.38	0.38	0.38	0.38	0.38	0.38	0.38	0.38	0.38	0.38	0.38	0.38
Sudden area decrease d2/d1=0.4	0.42	0.42	0.42	0.42	0.42	0.42	0.42	0.42	0.42	0.42	0.42	0.42	0.42	0.42	0.42	0.42	0.42	0.42	0.42
Sudden area decrease d2/d1=0.3	0.46	0.46	0.46	0.46	0.46	0.46	0.46	0.46	0.46	0.46	0.46	0.46	0.46	0.46	0.46	0.46	0.46	0.46	0.46
Sudden area decrease d2/d1=0.1	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50	0.50
Standart elbow 90^o	0.69	0.63	0.57	0.55	0.52	0.51	0.47	0.45	0.42	0.40	0.39	0.37	0.36	0.36	0.36				

	25	40	50	65	80	100	125	150	200	250	300	350	400	500	600	700	800	900	1000
Standart elbow 90^o (Large elbow dia.)	0.37	0.34	0.30	0.29	0.28	0.27	0.26	0.24	0.22	0.22	0.21	0.20	0.19	0.19	0.19				
Standart elbow 45^o	0.37	0.34	0.30	0.29	0.28	0.27	0.26	0.24	0.22	0.22	0.21	0.20	0.19	0.19	0.19				
Elbow (elbow pipe) 90^o r=D	0.46	0.42	0.38	0.37	0.36	0.34	0.33	0.31	0.28	0.27	0.26	0.25	0.24						
Elbow (elbow pipe) 90^o r=2D-3D	0.28	0.25	0.23	22.00	21.00	0.20	0.19	0.18	0.17	0.17	0.16	0.15	0.14						
Elbow (elbow pipe) 90^o r=4D	0.32	0.29	0.27	0.26	0.25	0.24	0.23	0.22	0.20	0.19	0.18	0.18	0.17						
Elbow (elbow pipe) 90^o r=6D	0.39	0.36	0.32	0.31	0.30	0.29	0.28	0.27	0.24	0.23	0.22	0.21	0.20						
Standart elbow 180^o	1.15	1.05	0.95	0.92	0.90	0.85	0.82	0.78	0.70	0.68	0.65	0.63	0.60						
Serial connected elbow 90^o "U"	1.38	1.26	1.14	1.10	1.04	1.02	0.94	0.90	0.84	0.80	0.78	0.74	0.72						
Serial connected elbow 90^o "crosswise"	2.07	1.89	1.71	1.65	1.56	1.53	1.41	1.35	1.26	1.20	1.17	1.11	1.08						
Serial connected elbow 90^o "S"	2.76	2.52	2.28	2.20	2.08	2.04	1.88	1.80	1.68	1.60	1.56	1.48	1.44						
Elbow (welded pieces) a=30^o	0.18	0.17	0.15	0.15	0.15	0.14	0.13	0.12	0.11	0.11	0.11	0.11	0.10						
Elbow (welded pieces) a=45^o	0.35	0.33	0.29	0.28	0.27	0.26	0.24	0.22	0.21	0.21	0.21	0.21	0.20						
Elbow (welded pieces) a=60^o	0.58	0.53	0.48	0.46	0.45	0.43	0.41	0.39	0.35	0.35	0.34	0.34	0.33						
Elbow (welded pieces) a=75^o	0.92	0.85	0.76	0.74	0.71	0.68	0.64	0.60	0.56	0.55	0.54	0.53	0.52						
Elbow (welded pieces) a=90^o	1.38	1.25	1.14	1.08	1.05	1.02	0.95	0.90	0.84	0.83	0.81	0.80	0.78						
Elbow welded 90o (equal pipe dia.) "T"	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50	2.50
Elbow welded 90o (equal pipe dia.) "S"	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00	3.00
Elbow welded 90o (equal pipe dia.) "H"	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00	5.00
Flowmeter (short type venturi) d/D=0.3	21.00	21.00	21.00	21.00	21.00	21.00	21.00	21.00	21.00	21.00	21.00	21.00	21.00	21.00	21.00	21.00	21.00	21.00	21.00
Flowmeter (short type venturi) d/D=0.4	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00	6.00
Flowmeter (short type venturi) d/D=0.5	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00
Flowmeter (short type venturi) d/D=0.6	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70	0.70
Flowmeter (short type venturi) d/D=0.7	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30
Flowmeter (short type venturi) d/D=0.8	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20	0.20
Flowmeter standart orifice d/D=0.3	300	300	300	300	300	300	300	300	300	300	300	300	300	300	300	300	300	300	300
Flowmeter standart orifice d/D=0.4	85.00	85.00	85.00	85.00	85.00	85.00	85.00	85.00	85.00	85.00	85.00	85.00	85.00	85.00	85.00	85.00	85.00	85.00	85.00
Flowmeter standart orifice d/D=0.5	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00	30.00
Flowmeter standart orifice d/D=0.6	12.00	12.00	12.00	12.00	12.00	12.00	12.00	12.00	12.00	12.00	12.00	12.00	12.00	12.00	12.00	12.00	12.00	12.00	12.00
Flowmeter standart orifice d/D=0.7	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50	4.50
Flowmeter standart orifice d/D=0.8	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00	2.00
Rotameter	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00	10.00
Turbine type counter	7.50	7.50	7.50	7.50	7.50	7.50	7.50	7.50	7.50	7.50	7.50	7.50	7.50	7.50	7.50	7.50	7.50	7.50	7.50
Multiple leveled turbine type counter	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00
Volumetric counter	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00	15.00
Expansion joints, compansators	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00	4.00

Table : 6 Measuring units and conversions

US	: American unit
UK, imperial	: British Unit

6.1 Length

1 m = 10 dm = 100 cm = 1000 mm = 10^6 µm (micron) = 3,28 ft
1 inch (in.) = 1″ = 25,4 mm
1 foot (ft) = 12 inch = 0,3048 m
1 yard (yd) = 63 inch = 0,9144 m
1 nautical mile, international = 1852 m
1 mile (US statute mile) = 1609,344 m
1 angstrom = 10^{-10} m
1 fathom = 2 yard = 1,8288 m
1 rod = 5.5 yard = 20,117 m
1 chain = 22 yard 201,168 m
1 milli-inch (mil) = 0,001 inch

6.2 Area

1 m^2 = 100 dm^2 = 10^4 cm^2 = 10^6 mm^2
1 square inch (sq in, $inch^2$) = 6,4516 cm^2
1 square foot /sq ft, $foot^2$) = 929,03 cm^2
1 are (a) = 100 m^2
1 decare (da) = 1000m^2
1 hectare (ha) = 10.000m^2
1 township = 36 mil^2 (mile) = 93,2396
1 rod^2 (sg rd) = 25,29 m^2
1 rood = 40 rod^2 = 1011 , 714 m^2
1 acre = 4 rood = 4047 m^2
1 mil^2 (mile) = 2,59 km^2
1 $yard^2$ (yd^2, sq yd) = 0,8361 m^2

6.3 Volume

1 m^3 = 1000 dm^3 = 10^6 cm^3
1 liter (l) = 1 dm^3 = 1000 cm^3
1 cubic inch (cu in, $inch^3$) = 16,387 cm^3
1 cubic foot (cu ft, $foot^3$) 28,3168 dm^3
1 US gallon (dry) = 4,4048 dm^3
1 US quart (dry) = 1,101 dm^3
1 US pint (dry) = 0,5506 dm^3
1 cubic yard (cu yd, $yard^3$) = 0,764555 m^3
1 bushel (US) = 8 US gallon (dry) = 35,239 dm^2
1 register ton (RT) = 100 cu ft = 2,832 m^3
1 acre foot = 1233,48 m^3
1 gill (US) = 0,0001183 m^3
1 gill (KU) = 0,0001421 m^3
1 peck (US) = 0,00881 m^3

Special volumetric units for fluids

1 US gallon = 3,7854 dm^3 = 8 pints (US) = 4 quarts (US)
1 imperial gallon (UK gallon) = 4,546 dm^3
1 barrel (US) = 0,11924 m^3 = 31,5 gallon (US)
1 US barrel (crude oil unit) = 42 US gallon = 0,1593
1 pint (imperial pt.) = 0,5683 dm^3
1 pint (US, pt) = 0,4732 dm^3
1 quart (US, qt) = 2 US pint = 0,9464 dm^2
1 quart (imperial, qt) = 2 pint (imperial) = 1,1365 dm^3
1 fluid ounce (US) = 0,02957 dm^3
1 dram (US) = 3,6967 cm^3

6.4 Pressure

1 kgf/cm^3=1kp/cm^2=0,9806 bar=10m H_2O (4^oC)
1 mbar = 0,001 bar= 100Pa
1 kPa= 0,01 bar = 0.102 m
1 Mpa= 10 bar
1 m H_2O (4^oC)= 0,09806 bar
1 torr= 1 mm Hg (0^oC)= $1,333.10^{-3}$ bar
1 cm Hg (^oC)= 0,013332 bar
1 ft H_2O (4^oC)= 0,02989
1 inch H_2O (4^oC)= 2,49110 bar
1 inch Hg (^oC)= 0,03386 bar
1 psi (pound square inch, $lbf/inch^2$)= 0,06895 bar
1 lbf/ft^2=$0,4788.10^{-3}$ bar
1 ozf/in^2= 43,1 Pa
1 atmosphere (atm)(standard)= 1,01325 bar= 760 mmHg (0^oC)
1 atmosphere (atm)(standard)= 10,335 mH_2O (4^oC)= 1,0332 kgf/cm^2
1 atmosphere (atm)(standard)= 1,0332 kp/cm2= 14,696 psi
1 bar= 0,98692 atm (standard)= 750,06 mmHg (0^oC)
1 bar= 10,197 m H_2O(4^oC)= 1,0197 kg/cm^2)= 1,0197 kgf/cm^2
1 bar= 33,456 ft H_2O (4^oC)
1 technical atmosphere (at)= 1kp/cm2=1 kgf/cm2=10 m H_2O (4^oC)
1 kp/m^2= 1mm H_2O
0,102m= 1kPa
100kPa= 10,2mSS

6.5 Volumetric flow rate

1 m^3/h = 0,2778.10-3-m^3/s = 16,66 l/minute
1 l/minute = 0,06 m^3/h
1 dm^3/dak = 0,01667.10-3 m^3/s = 0,060 m^3/h
1 dm3/s = 0,001 m^3/s = 3.6 m^3/h
1 U.S. gallon per minute (U.S. gpm) = 630,9m^3/s = 0,2271m^3/h
1 imp gallon per minute (U.K. gpm) = 757,7 m^3/s = 0,2728 m^3/h
1 m^3/h = 4,403 U.S. gpm = 3,663 U.K gpm
1 cubic foot per second (ft3/s) = 0,02832 m^3/s = 101,94 m^3/h
1 cubic yard per second = 0,7646 m^3/s = 2752,4 m^3/h

6.6 Mass rate of flow

1 kg/min. = 0,01667 kg/s
1 t/h = 0,2778 kg/s
1 t/min = 16,67 kg/s
1 oz/s = 28,35 g/s
1 lb/s = 0,4536 kg/s
1 lm/min = 0,00756 kg/s
1 short ton per hour (shtn/h) = 0,2520
1 long ton per hour (ltn/h) 0,2822 kg/s
1 m^3/h= 16,66 lt/min
1 lt/min= 0,06 m^3/h

100 GPM= 22,71 m^3/h

6.7 Density

1 g/cm^3= 1000 kg/m^3
1 kg/dm^3 =1000 kg/m^3
1 pound per cubic foot (lb/cuft.,lb/ft3)= 0,0160 kg/dm^3
1 pound per cubic inch (lb/cu.in,lb/in3)= 27,68 kg/dm
1 slug / ft^3 = 515,38 kg/m^3
1 ounce per cubic inch (oz/cu.in) = 1,73 kg/dm^3
1 pound per gallon-US (lb/gallon) = 119,8 kg/m^3
1 pound per gallon-British (lb/gallon) = 99,79 kg/m^3
Relative density (SG, sg, sp, gr)
Is dimensionless and defined hereunder. The relative density is also equivalent to the unit density of the fluid expressed in g/cm^3.
S.G. = (the density of the fluid at defined operation temperature) / (the density of water at + 4°C temperature)
The density of water at + 4°C temperature = 1000kg/m^3
The relative density of water at + 4°C = 1.
(The density of a fluid with relative density of S.G = 1,16 is ρ = 1,16 g/cm^3)
The relation between the API grade at 60 °F (15,6 °C) temperature and the relative density :

$$\mathbf{SG=\frac{141.5}{131.5+^\circ API}}$$

6.8 Specific Weight

1 kgf /m^3 = 9,81 N/m^3
1 kp /m^3 = 9,81 N/m^3
1 lbt/ft^3 = 157,09 N/m^3

6.9 Force

1 N = 1 kg . m/s^2
1 dyne = 10-5 N
1 kg force (kgf) = 9,806 N
1 kilopond (kp) = 9,806 N
1 ounce – force (ozf) = 0,278 N
1 pound – force (lbf) = 4,448 N
1 lbf = 1 slug ft /s^2
1 poundal (pdl) = 0,13825 N
1 sthene (sn) 0,001 N

6.10 Power

$$1\ W = 1\ \frac{J}{s} = 1\ \frac{N\ .\ M}{s} = 1\ \frac{kg\ .\ M2}{s3}$$

1 kW = 1000 W
1 MW =106 W = 1000 kW
1 VA = 1W (görülen elektrik gücü)
1 VAr=1 W (reaktif elektrik güç
1 ft . lbf/s = 1,3558 W
1 BTU/s = 1055,06 W
1 BTU/h = 0,29307 W
1 HP (metrik) = 1 PS = 0,7355 kW
1 HP (elektrik) = 0,746 kW
1 HP (U.K) = 0,7457 kW
1 cat/h = 1,163 W
1 kcal/s = 4186,8 W

6.11 Energy

1 J = 1 N.m = 1 kg m^2/s^2
1 erg = 10-7 J
1 W. s = 1 J
1 kWs = 1000 J
1 kWs = 3,6 . 106 J 860 kcal
1kp . m = 9,81 J
1 cal = 4,1868 J
1 kcal = 4186,8 J = 4,1868 kJ
1 British Thermal Unit (BTU) = 1055 J = 1,055 kJ
1 ft . lbf = 1,3558 J
1 therm = 108 BTU = 105,5 MJ
1 therm = 1000 frigories = 4,1855

6.12 Mass

1 gram (g) = 0,001 kg
1 ton (t) = 1000 kg
1 slug = 14,594 kg
1 ounce (oz, oz av) = 28,35 g
1 pound (lb, lbm) (avoirdupois) = 0,4536 kg = 7000 grain = 256 dram = 16 oz av
1 pound (troy) = 0,3732 kg
1 stone (st) = 14 lb = 6,3503
1 short hundredweight (sh cwt) = 4 US quarters = 45,3592 kg
1 long hundred weight (l cwt) = 4 UK quarters = 50,8024 kg
1 hundredweight (cwt) = 50,8024 kg
1 short ton (shtn) = 907,19 kg = 2000 lb
1 long ton (ltn) = 1016,05 kg= 20 cwt
1 grain (gr) = 0,0648 g
1 pound mass (lbm) = 0,4536 kg
1 dram (dm,dm av) = 1,772g
1 metric karat (Kt) = 0,2 g
1 ounce (troy) = 31,10 g
1 pennyweight (dwt) = 1,555 g
1 gram (avoirdupois) = 1,7718 g
1 gram (troy) = 3,8879 g

6.13 Speed

1 km/h = 0,2778 m/s
1 foot per second (ft/s) = 0,00508 m/s
1 foot per minute (ft/min) = 0,9144 m/s
1 yard per second (yd/s) = 0,01524 m/s
1 yard per minute (yd/min) = 0,01524 m/s
1knot (international) = 0,5144 m/s = 1,852 km/h = 1 nautical mile / h = 1,1516 mph
1 mile per hour (mph) (US statute) = 1,6093 km/h = 0,447 m/s

6.14 Viscosity

Dynamic or absolute viscosities

1.Paç S = 1 N s/m^2 = 1 kg/s . m
1 Poise (P) = 0,1 Pa .s
1 Centipoise (cP) = 0,001 Pa . s
1 kgf.s/m2 = 9,81 Pa . s
1 lbm/ft . s = 1,4882 Pa . s
1 lbf.s/ft2 . = 47,88 Pa . s
1 slug/ ft . s = 47,88 Pa . s

Kinematic viscosity

1 stoke (St) = 1 cm^2 /s = 100mm^2/s = 10-4 m^2/s
1 Centistoke (cSt) = 1 mm^2/s = 0,01 St = 10-6 m^2/s
1 St = 100 cSt
1 ft2/s = 0,09290 m2/s 92903 Cst

Relations between the absolute (dynamic) viscosity and the kinematic viscosity
Absolute (Dynamic) Viscosity = Density x Kinematic Viscosity
$\mu = \rho \cdot \nu$
μ (Pa . s) =ρ (kg /m^3) . ν (m^2/s)
μ (Pa . s) =1000 ρ (g/cm^3) . ν (m^2/s)
μ (Pa . s) = 0,001 . ρ (g/cm^3) . ν(cST)
μ (cP) = ρ (g/cm^3) . ν(cSt)
μ (lbf . s/ft^2) = 0,03108 . ρ (lb/ft^3) ν (ft^2/s)
μ (cP) = 1488,16 . ρ (lb/ft^3) ν (ft^2/s)
μ (kgf . s/m^2) = 0,10109 . ρ (kg /m^3) . ν(cts)
μ (Pa . s) = 0,001 . ρ (kg /dm^3) . μ (cP)
μ (cP) = . SG . ν (cSt) , (SG = relative density)

6.15 Moment , Torque

1 N m=1kg m^2/s^2
1kgf m=9,807 n m
1foot-pound force(ft lbf)=1,356 N m
1inch-pound force(in lbf)=0,113 N m

6.16 Rotation Speed (n), Angular Speed (ω)

1radian/second (rad/s)=9,549 rpm=57,296 °/s=0,1592d/s
1 rpm = 60 revolutions/s

6.17 Water hardness

1GPG (grains/U S gallon)=0,01712 g/l

Table : 7 Alloy elements effects on steel properties

EFFECTS OF ALLOY COMPONENTS STEEL PROPERTIES

ALLOY COMPONENTS	HARDNESS	TENSILE STRENGHT	YIELD STRENGTH	ELONGATION	REDUCTION IN AREA	IMPACT RESISTANCE	ELASTICITIES	HIGH TEMP. RESISTANCE	COOL DOWN TIME	CARBON FORMATION	WEAR RESISTANCE	MALLABILITY	MARKING BILITY	OXIDATION TENDENCY	CORROSION RESISTANCE
Si	↑	↑	↑↑	↓	~	↓	↑↑↑	↑	↓	↓	↓↓↓	↓	↓	↓	—
Mn*	↑	↑	↑	~	~	~	↑	~	↓	~	↓↓	↑	↓	~	—
Mn**	↓↓↓	↑	↓	↑↑↑	~	—	—	—	↓↓	—	—	↓↓↓	↓↓↓	↓↓	—
Cr	↑↑	↑↑	↑↑	↓	↓	↓	↑	↑	↓↓↓	↑↑	↑	↓	—	↓↓↓	↑↑↑
Ni*	↑	↑	↑	~	~	~	—	↑	↓↓	—	↓↓	↓	↓	↓	—
Ni**	↓↓	↑	↓	↑↑↑	↑↑	↑↑↑	—	↑↑↑	↓↓	—	—	↓↓↓	↓↓↓	↓↓	↑↑
Al	—	—	—	—	↓	↓	—	—	—	—	—	↓↓	—	↓↓	—
W	↑	↑	↑	↓	↓	~	—	↑↑↑	↓↓	↑↑	↑↑↑	↓↓	↓↓	↓↓	—
V	↑	↑	↑	~	~	↑	↑	↑↑	↓↓	↑↑↑↑	↑↑	↑	—	↓	↑
Co	↑	↑	↑	↓	↓	↓	—	↑↑	↑↑	—	↑↑↑	↓	~	↓	—
Mo	↑	↑	↑	↓	↓	↑	—	↑↑	↓↓	↑↑↑	↑↑	↓	↓	↑↑	—
S	—	—	—	↓	↓	↓	—	—	—	—	—	↓↓↓	↑↑↑	—	↓
P	↑	↑	↑	↓	↓	↓↓↓	—	—	—	—	—	↓↓↓	↓↓↓	↓↓	↑↑

* Pearlite steel
** Austenitic steel

↑ Increase ↓ Decrease ~ No effect — Unimportant / Unknown

Table : 8 Scale coefficients

Coefficient	Symbol	Number
tera	T	10^{12}
giga	G	10^{9}
mega	M	10^{6}
kilo	k	10^{3}
hecto	h	10^{2}
deca	da	10
Deci	da	10^{-1}
centi	c	10^{-2}
milli	m	10^{-3}
micro	μ	10^{-6}
nano	n	10^{-9}
pico	p	10^{-12}

Table : 9 Greek alphabet characters

Letter	Name
Α α	Alfa
Β β	Beta
Γ γ	Gamma
Δ δ	Delta
Ε ε	Epsilon
Ζ ζ	Zeta
Η η	Eta
Θ θ	Theta
Ι ι	Iota
Κ χ	Kappa
Λ λ	Lambda
Μ μ	Mu
Ν ν	Nu
Ξ ξ	Xi
Ο ο	Omicron
Ρ ρ	Ro
Π π	Pi
∑ σ	Sigma
Τ τ	Tau
Υ υ	Upsilon
Φ φ	Phi
Χ χ	Chi
Ψ ψ	Psi
Ω ω	Omega

Table : 10 Water pH value change by temperature

Water temperature (°C)	0	25	50	75	100	150	200	250	300
Water pH value	7,5	7	7	6	6.1	5.9	5.7	5.6	5.6

Table : 11 Density and viscosity of several fluids

Fluid	Relative density (SG)	Kinematic viscosity-ν (centistokes)	Temperature t (°C)
Freon -12	1.33	0.27	21
Glycerin (%100)	1.26	648 / 176	21 / 38
(%50 water)	1.13	5.29	20
Acetaldehyde	0.762	0.295	20
Glycol - Propylene	1.038	52	21
Triethylene	1.125	40	21
Diethylene	1,12	32	21
Ethylene	1,125	17.8	21
Hydrochloric Acid (%31.5)	1,05	1.9	20
Mercury	13,57	0,12/0,11	21 / 38
Phenol (Carbolic Acid)	0,95 - 1,08	11,8/1,17	18 / 90
Soda Silicate	40 Baume	79	38
	42 Baume	138	38
Toluene	0.866	0,68	20
Bromine	2,9	0,34	20
Mineral Oils SAE 10	0,880-0,935(16°C)	35 – 52	38
SAE 20	"	52 – 87	38
SAE 30	"	87 – 125	38
SAE 40	"	125 – 206	38
SAE 50	"	206-352/16-22	38/99
SAE 60	"	352-507/22-26	38/99
SAE 70	"	507-682/26-32	38/99
SAE 80	"	22000 (maks)	-18
SAE 90	"	173 – 325	38
SAE 140	"	206-507/25-43	54/99
SAE 150	"	43 (min)	99
SAE 10W	"	1295-2590	-18
SAE 20W	"	2590-10350	-18
SAE 75W	"	4.2	99
SAE 80W	"	7.0	99
SAE 85W	"	11.0	99

Table : 12 Fuel-Oils' viscosity change by temperature

Temperature (°C)	Kinematic Viscosity - v (mm²/s)				
	Fuel - oil No				
	2	4	5 - light	5 – heavy	6
-20	9 - 30	100-2 000	550 - 12 000	20 000 - 100 000	-
-10	6 - 17	50 - 700	1400 -3 000	5 000 - 20 000	-
0	4,5 - 11	29 - 280	500 - 1000	1 400 - 5 000	7 000 - 200 000
10	3 - 7	17 -120	200 - 400	550 - 1 700	2 300 - 50 000
20	2,6 - 6	11 - 60	90 - 200	240 – 600	900 - 13 000
40	1,8 - 3,5	6 - 22	30 - 60	70 – 140	180 - 1500
60	-	3,5 - 11	13 - 25	30 – 50	65 - 350
80	-	2,4 - 6	7 - 13	14 – 22	26 - 100
100	-	1,8 - 4	4,3 - 7,5	8 – 12	14 - 42
120	-	-	3 - 5	5,2 - 7,3	8,5 - 21
140	-	-	2,2 - 3,5	3,6 - 4,8	5,4 - 12

Table : 13 Various fluids kinematic viscosity change by temperature

SIVI		Kinematic Viscosity -v (mm^2/s) Temperature (°C) -10	0	20	40	60	80	100
Crude Oil	SG=0,925	-	310	75	21	15	10	7
	SG=855	40	16	7	4.6	3.5	2.7	2.2
Kerosene, SG =0,79-0,81		-	4	2.3	1.7	1.3	1.1	0.9
Gasoline, SG=0,716		-	0.8	0.62	0.5	0.41	-	-
SG=0,784		-	1.05	0.81	0.66	0.55	-	-
SG=0680		-	0.54	0.43	0.35	0.3	-	-
Carbon tetrachloride		-	2.4	1.4	1	0.75	0.56	0.46
Glycerin		-	80 000	1 000	300	80	30	10
Mercury		0.15	0.15	0.13	0.12	0.1	0.1	0.09
Salt water (%20 NaCl) SG= 1,18		-	2.3	1.3	0.9	-	-	-
Treacle SG= 1,42		-	-	2 000	410	140	55	27
Ethyl alcohol		-	2.5	1.5	1	0.75	0.56	0.46
Freon – 22		0.21	0.2	0.2	0.19	-	-	-
Ammonia water		0.37	0.36	0.35	0.35	-	-	-
Carbon dioxide		0,10	0,10	0,10	-	-	-	-
Methyl chloride		0.29	0.28	0.26	0.25	0.24	-	-
Sugar solution	% 10	-	2.5	1.3	0.8	0.57	-	-
	% 20	-	3.5	1.8	1.1	0.75	-	-
	% 40	-	12	5.4	2.8	1.7	-	-
	% 60	-	170	45	16	8	-	-
	% 70	-	2500	350	85	30	-	-
	% 76	-	-	4 000	400	90	-	-
Dowtherma A SG= 1,056		-	-	4.6	2.9	2	1.4	1.1
Sulphur Dioxide		0,30	0.26	0.22	0.19	-	-	-

Table : 14 Mineral oil kinematic viscosities (SAE) change by temperature (S.G.=0,82-0,95)

SAE No	Kinematic Viscosity: v (mm^2/s) Temperature (°C) -15	-10	0	10	20	30	40	60	80	100
50	30 000	19 000	6 000	2 100	900	420	220	80	37	20
40	20 000	10 000	3 500	1 400	600	290	155	60	28	15
30	10 000	5 500	2 000	750	370	190	105	41	20	12
20W-40	8 000	3 500	1 400	650	305	170	105	43	23	14
20W-20	4 500	2 700	950	400	200	110	70	29	15	9
10W	2 000	950	400	200	110	70	40	20	11.5	7
10W-30	2 000	1 000	500	250	140	85	60	29	16	10
5 W -20	800	430	210	125	70	42	29	15	9	6

Table : 15 pH values of various drinks and foodstuffs

Liquid	pH Degree
Light drinks	2,0 -4,0
Lemon	2,2 - 2,4
Vinegar	2,4 - 3,4
Wine	2,8 - 3,8
Plum	2,8 - 3,0
Apple	2,9 - 3,3
Grapefruit	3,0 - 3,3
Orange	3,0 - 4,0
Strawberry	3,0 - 3,5
Grapes	3,5 - 4,5
Olive	3,6 - 3,8
Apricot	3,6 - 4,0
Tomato	4,0 - 4,4
Beer	4,0 - 5,0
Beet	5,2 - 5,6
Milk	6,3 - 6,6
Potable water	6,5 - 8,0
Egg yolk (fresh)	7,6 - 8,0
Detergent	9,0 - 14,0

Table : 16 pH values of various liquids according to normality values

	Liquid	Normality	Mass ratio (1) (%)	pH Value at 25 °C temperature
ACID	Hydrochloric acid	N	3.6	0.1
	"	0.1 N	0.36	1.1
	"	0.01 N	0.036	2.0
	Sulfuric acid	N	4.9	0.3
	"	0.1 N	0.49	1.2
	"	0.01 N	0.049	2.1
	Sulphur	0.1 N		1.5
	Oxalic acid	0.1 N		1.6
	Tartaric acid	0.1 N		2.2
	Citric acid	0.1 N		2.2
	Formic acid	0.1 N		2.3
	Lactic acid	0.1 N		2.4
	Ascetic acid	N		2.4
	"	0.1 N	6	2.9
	"	0.01 N	0.6	3.4
	Carbonic acid (saturated)		0.06	3.8
	Hydrogen sulfur	0.1 N		4.1
	Hydrocyanic acid	0.1 N		5.1
	Boric acid	0.1 N	0.2	5.2
BASE	Sodium hydro carbonate		0.42	8.4
	Borax	0.1 N	139	9.2
	Calcium carbonate (saturated)			9.4
	Ammonia water	0.01 N	0.017	10.6
	"	0.1 N	0.17	11.1
	"	N	1.7	11.6
	Sodium carbonate	0.1 N		11.6
	Sodium hydroxide	0.01 N	0.04	12.0
	"	0.1 N	0.4	13.0
	"	N	4.0	14.0
	Potassium hydroxide	0.01 N	0.056	12.0
	"	0.1 N	0.56	13.0
	"	N	5.6	14.0
	Lime (saturated)			12.4

1) : Mass ratio = the ratio of the defined substance in %, within the total mass of the solution

Table : 17 Physical properties of some gases under standard atmospheric pressure (p_{atm}, $ab_{absolute}$=1,01325bar)

Gas	Temperature (°C)	Density ρ (kg/m^3)	Dynamic Viscosity μ (Pa.s)	Gas Constant R (J/kg.K)	Specific temperature Ratio K
Air (standard)	15	1.225	$1{,}79.10^{-5}$	286.9	1,40
Carbon dioxide	20	1.83	$1{,}47.10^{-5}$	188.9	1,30
Hydrogen	20	0.0838	$0{,}884.10^{-5}$	4124	1.41
Methane	20	0.667	$1{,}10.10^{-5}$	518.3	1,30
Nitrogen	20	1.16	$1{,}76.10^{-5}$	296.8	1,40
Oxygen	20	1.33	$2.04.10^{-5}$	259.8	1,40
Butane	27	2.38	$0{,}75.10^{-5}$	143	1,09
Propane	27	1.79	$0{,}81.10^{-5}$	188	1,13
Natural gas (%85 methane)	15	L - 0,61 H - 0,64	$1{,}2.10^{-5}$ $1{,}1.10^{-5}$	459 470	1,32 1,31
Chlorine	0	3.21	$1{,}4.10^{-5}$	117.3	1,37
Carbon monoxide	0	1.25	$1{,}8.10^{-5}$	297.1	1,40
Helium	30	0.166	$1{,}97.10^{-5}$	2079	1,66
Water vapor	100	0.598	$1{,}2.10^{-5}$	-	-

Table : 18 Atmospheric pressure changes by altitude

Altitude z (m)	Atm. pressure, H_a (m)	(Torr)
0	10,33	760
100	10,20	751
200	10,08	742
300	9,97	733
400	9,85	724
500	9,73	716
600	9,62	707
700	9,50	698
800	9,40	690
900	9,30	682
1000	9,20	674
1500	8,60	655
2000	8,10	598

Table : 19 Evaporation pressure change of water by temperature

Temperature t (C^o)	Water evaporation pressure h_b (m.Su S.)	h_b (Torr)
0	0.062	4.58
5	0.089	6.54
10	0.125	9.20
20	0.283	17.50
30	0.432	37.70
40	0.752	55.20
50	1.257	92.30
60	2.031	149.20
70	3.177	233.10
80	4.829	354.60
90	7.149	525.40
100	10.330	760.00

Table : 20 Density of some metallic alloys and solid substances

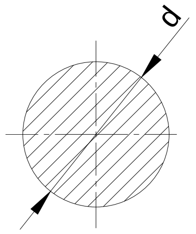
Material	Density (kg/m^3)
Aluminum	2700
Copper	8960
Nickel	8900
Zinc	7100
Tin	7310
Brass	8470
Grey cast iron	7200
Cast steel	7650
Lead	11350
Magnesium	1740
Platinum	21450
Silver	10500
Stainless steel	8020
Tungsten	19300
Uranium	18800
Carbon steel (Iron)	7860
Gold	19320
Chromium	7200
Aluminum bronze	7800

Material	Density (kg/m^3)
Silicone	2330
Sodium	970
Iron	7870
Titanium	4540
Manganese	7430
Molybdenum	10220
Cadmium	8640
Asphalt	1100 - 1500
Brick	1400 - 2200
Concrete	2700 - 3000
Clay	1800 - 2600
Gelatin	1270
Glass	2400 - 2800
Marble	2600 - 2840
Rubber	1100 - 1190
Sugar	1500
Water	1000

Table : 21 Some formulas used in engineering

21-1 Area calculations

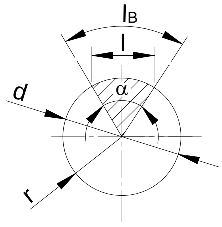
a- Circle



$$Perimeter = U = \pi \cdot d$$

$$Area = A = \frac{\pi \cdot d^2}{4}$$

b- Sector



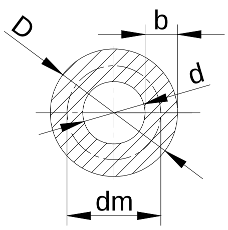
$$Area = A = \frac{\pi \cdot d^2}{4} \cdot \frac{\alpha}{360^0}$$

$$Area = A = \frac{l_B \cdot r}{2}$$

$$l = 2 \cdot r \cdot \sin\frac{\alpha}{2}$$

$$l_B = \frac{\pi \cdot r \cdot \alpha}{180^0}$$

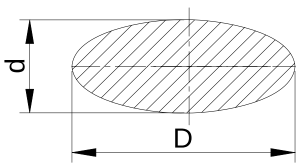
c- Circle ring



$$Area = A = \pi \cdot dm \cdot b$$

$$Area = A = \frac{\pi}{4} \cdot \left(D^2 - d^2\right)$$

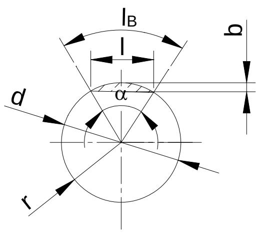
d-Ellipse



$$Perimeter = U \approx \frac{\pi}{2} \cdot \left(D + d\right)$$

$$Area = A = \frac{\pi \cdot D \cdot d}{4}$$

e- Sector



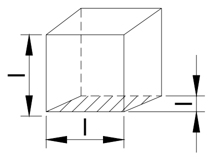
$$Area = A = \frac{\pi \cdot d^2}{4} \cdot \frac{\alpha}{360^0} - \frac{l \cdot \left(r - b\right)}{2}$$

$$Area = A = \frac{l_B \cdot r - l \cdot \left(r - b\right)}{2}$$

$$l = 2 \cdot r \cdot \sin\frac{\alpha}{2}$$

21-2 Volume calculations

a- Cube

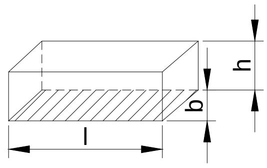


Total Surface Area =

$$\boxed{A_0 = 6 \cdot l^2}$$

$$\boxed{Volume = V = l^3}$$

b- Rectangular prism

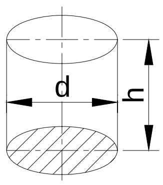


Total Surface Area =

$$\boxed{A_0 = 2 \cdot (l \cdot b + l \cdot h + b \cdot h)}$$

$$\boxed{Volume = V = l \cdot b \cdot h}$$

c- Cylinder



Total Surface Area=

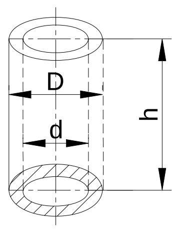
$$\boxed{A_0 = \pi \cdot d \cdot h + 2 \cdot \frac{\pi \cdot d^2}{4}}$$

Side Surface Area =

$$\boxed{A_M = \pi \cdot d \cdot h}$$

$$\boxed{Volume = V = \frac{\pi \cdot d^2}{4} \cdot h}$$

d-İçi boş silindir

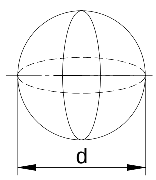


Total Surface Area =

$$\boxed{A_0 = \pi \cdot (D + d) \cdot \left[\frac{1}{2} \cdot (D - d) + h \right]}$$

$$\boxed{Volume = V = \frac{\pi \cdot h}{4} \cdot \left(D^2 - d^2 \right)}$$

e- Sphere



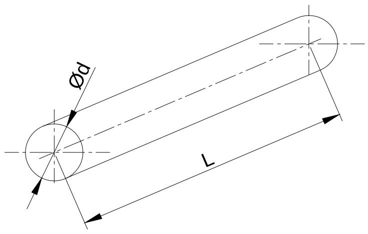
Outer Surface Area =

$$\boxed{A_0 = \pi \cdot d^2}$$

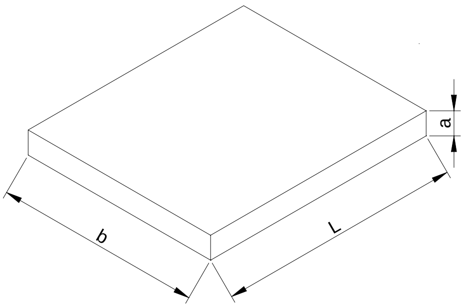
$$\boxed{Volume = V = \frac{\pi \cdot d^3}{6}}$$

21-3 Weight calculations

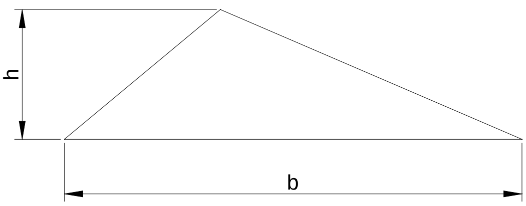
a- Round iron bar

	$\boxed{Area = A = \frac{\pi \cdot d^2}{4} \cdot l \cdot 7,85}$ *Example :* *Bar diameter Ød = Ø80mm* *Length L=1000mm* $A = \frac{\pi \cdot (8cm)^2 \cdot (100cm) \cdot 7,85}{4 \cdot 1000} = 39,46kg$

b- Sheet iron

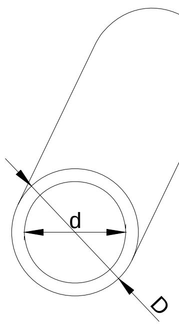
	$\boxed{Area = A = (a \cdot b \cdot L) \cdot 7,85}$ *Example :* *a=30cm , b=50cm , L=200cm* $A = \frac{(30cm \cdot 50cm \cdot 200cm) \cdot 7,85}{1000} = 235,5kg$

c- Triangle

	$Area = A = \frac{b \cdot h}{2}$

21-4 Development calculations

a- Sheet iron pipe development



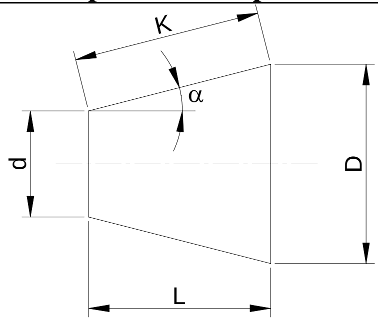
Width of the sheet iron to cut out:

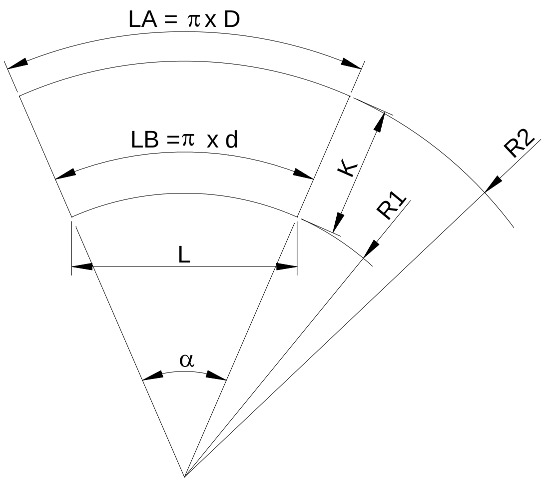
$$\boxed{U = \pi \cdot \left(\frac{D+d}{2} \right)}$$

Example :
Outer diameter D=Ø110cm
Inner diameter d=Ø90cm

$$U = \pi \cdot \left(\frac{110+90}{2} \right) = 315$$

b- Conic piece development calculation





$$\boxed{tg\alpha = \frac{D-d}{2L}}$$

$$\boxed{\cos\alpha = \frac{L}{2K}}$$

$$\boxed{K = \frac{L}{\cos\alpha}}$$

Conic piece development calculation :

$$L_A \cdot 180 = R_2 \cdot \pi \cdot \varphi$$

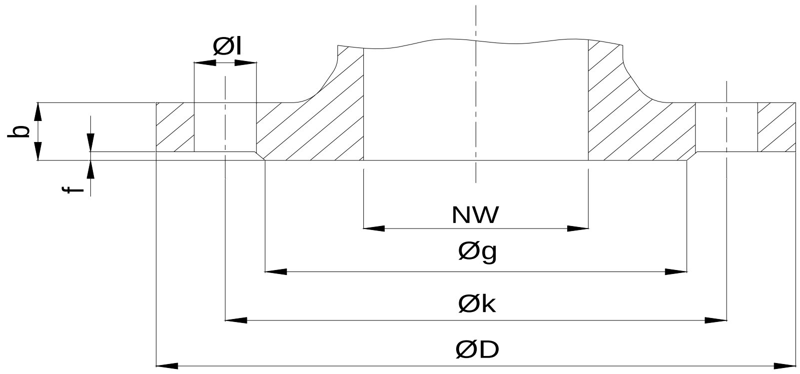
$$\boxed{L = 2 \cdot R_1 \cdot \frac{\sin\varphi}{2}}$$

$$\varphi^o = \frac{180 \cdot (D-d)}{K}$$

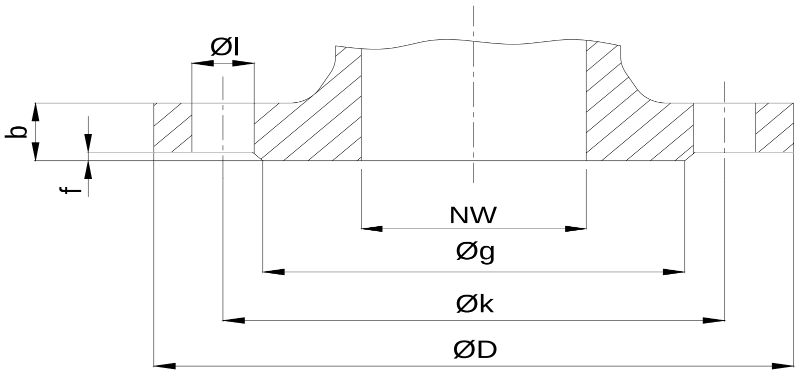
$$\boxed{R_2 = \frac{D \cdot 180}{\varphi^o}} \qquad \boxed{R_1 = \frac{d \cdot 180}{\varphi^o}}$$

Table : 22 Flange Norms

22.1 TS ISO 7005-2

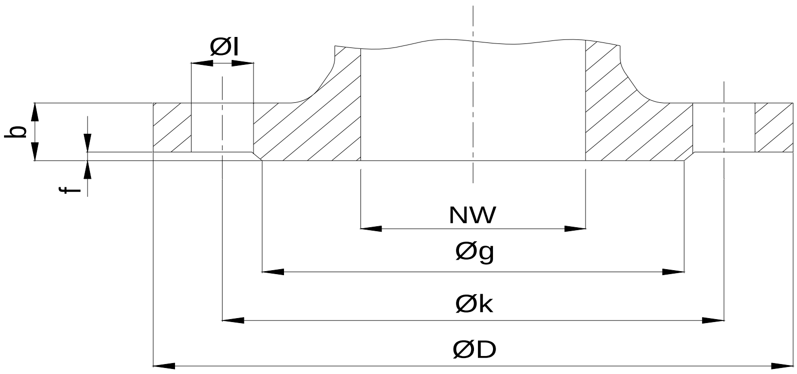


ISO PN6 FLANGE DIMENSIONS (TS ISO 7005-2)									
NOMINAL DIAMETER NW	FLANGE			NUT				g	f
	∅D	b	∅k	Piece	d		∅l		
					Metric	Inch			
40	130	16	100	4	M12	½	14	78	3
50	140	16	110	4	M12	½	14	88	3
65	160	16	130	4	M12	½	14	108	3
80	190	18	150	4	M16	5/8	19	124	3
100	210	18	170	4	M16	5/8	19	144	3
125	240	20	200	8	M16	5/8	19	174	3
150	265	20	225	8	M16	5/8	19	199	3
200	320	22	280	8	M16	5/8	19	254	3
250	375	24	335	12	M16	5/8	19	309	3
300	440	24	395	12	M20	¾	23	363	4
350	490	26	445	12	M20	¾	23	413	4
400	540	28	495	16	M20	¾	23	463	4
450	595	28	550	16	M20	¾	23	518	4
500	645	30	600	20	M20	¾	23	568	4
600	755	30	705	20	M24	7/8	26	667	5
700	860	32	810	24	M24	7/8	26	772	5
800	975	34	920	24	M27	1	31	878	5
900	1075	36	1020	24	M27	1	31	978	5
1000	1175	36	1120	28	M27	1	31	1078	5
1200	1405	40	1340	32	M30	$1^{1}/_{8}$	34	1295	5

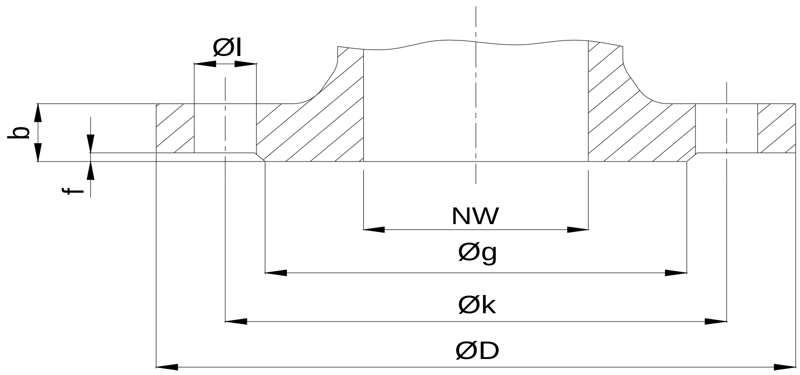


ISO PN10 FLANGE DIMENSIONS (TS ISO 7005-2)

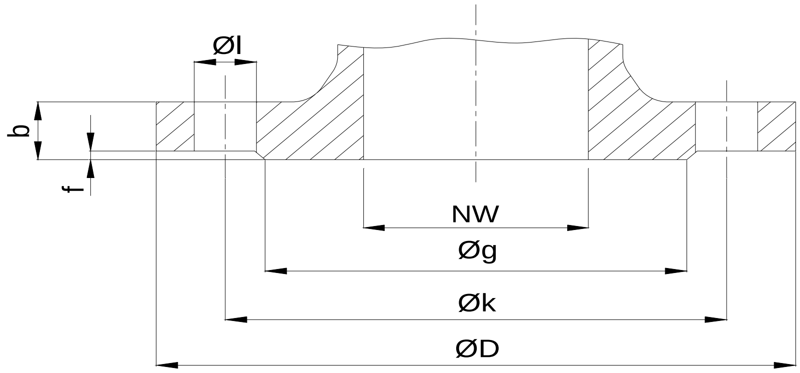
NOMINAL DIAMETER NW	FLANGE			NUT				g	f
	ØD	b	Øk	Piece	D		ØI		
					Metric	Inch			
40	150	18	110	4	M 16	5/8	19	84	3
50	165	20	125	4	M 16	5/8	19	99	3
65	185	20	145	4	M 16	5/8	19	118	3
80	200	22	160	8	M 16	5/8	19	132	3
100	220	24	180	8	M 16	5/8	19	156	3
125	250	26	210	8	M 16	5/8	19	184	3
150	285	26	240	8	M 20	3/4	23	211	3
200	340	26	295	8	M 20	3/4	23	266	3
250	395	28	350	12	M 20	3/4	23	319	3
300	445	28	400	12	M 20	3/4	23	370	4
350	505	30	460	16	M 20	3/4	23	429	4
400	565	32	515	16	M 24	7/8	28	480	4
450	615	32	565	20	M 24	7/8	28	530	4
500	670	34	620	20	M 24	7/8	28	582	4
600	780	36	725	20	M 27	1	31	682	5
700	895	40	840	24	M 27	1	31	794	5
800	1015	44	950	24	M 30	1 $^{1}/_{8}$	34	901	5
900	1115	46	1050	28	M 30	1 $^{1}/_{8}$	34	1001	5
1000	1230	50	1160	28	M 33	1 $^{1}/_{4}$	37	1112	5
1200	1455	56	1380	32	M 36	1 $^{3}/_{8}$	40	1328	5



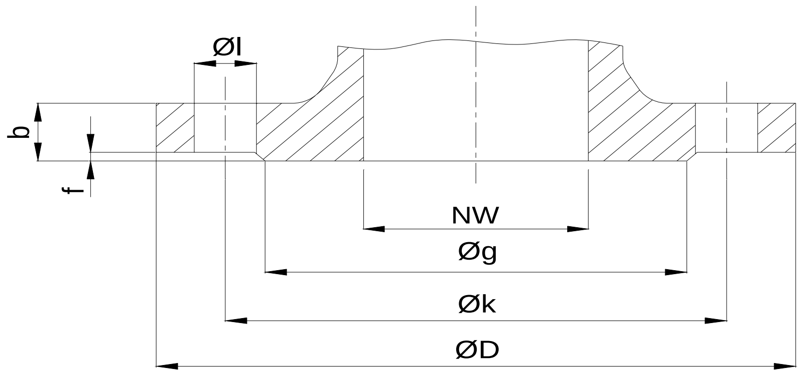
ISO PN16 FLANGE DIMENSIONS (TS ISO 7005-2)									
NOMINAL DIAMETER NW	FLANGE			NUT				g	f
	∅D	b	∅k	Piece	D		∅I		
					Metric	Inch			
40	150	18	110	4	M 16	5/8	19	84	3
50	165	20	125	4	M 16	5/8	19	99	3
65	185	20	145	4	M 16	5/8	19	118	3
80	200	22	160	8	M 16	5/8	19	132	3
100	220	24	180	8	M 16	5/8	19	156	3
125	250	26	210	8	M 16	5/8	19	184	3
150	285	26	240	8	M 20	3/4	23	211	3
200	340	30	295	12	M 20	3/4	23	266	3
250	405	32	355	12	M 24	7/8	28	319	3
300	460	32	410	12	M 24	7/8	28	370	4
350	520	36	470	16	M 24	7/8	28	429	4
400	580	38	525	16	M 27	1	31	480	4
450	640	40	585	20	M 27	1	31	548	4
500	715	42	650	20	M 30	$1^{1}/_{8}$	34	609	4
600	840	48	770	20	M 33	$1^{1}/_{4}$	37	720	5
700	910	54	840	24	M 33	$1^{1}/_{4}$	37	794	5
800	1025	58	950	24	M 36	$1\ ^{3}/_{8}$	40	901	5
900	1125	62	1050	28	M 36	$1\ ^{3}/_{8}$	40	1001	5
1000	1255	66	1170	28	M 39	$1^{1}/_{2}$	43	1112	5
1200	1485	-	1390	32	M 45	-	49	1328	5



ISO PN25 FLANGE DIMENSIONS (TS ISO 7005-2)									
NOMINAL DIAMETER NW	FLANGE			NUT				g	f
	∅D	b	∅k	Piece	D		∅l		
					Metric	Inch			
40	150	20	110	4	M 16	5/8	19	84	3
50	165	22	125	4	M 16	5/8	19	99	3
65	185	24	145	8	M 16	5/8	19	118	3
80	200	26	160	8	M 16	5/8	19	132	3
100	235	28	190	8	M 20	3/4	23	156	3
125	270	30	220	8	M 24	7/8	28	184	3
150	300	34	250	8	M 24	7/8	28	211	3
200	360	34	310	12	M 24	7/8	28	274	3
250	425	36	370	12	M 27	1	31	330	3
300	485	40	430	16	M 27	1	31	389	4
350	555	44	490	16	M 30	1 1/8	34	448	4
400	620	48	550	16	M 33	1 1/4	37	503	4
450	670	50	600	20	M 33	1 1/4	37	548	4
500	730	52	660	20	M 33	1 1/4	37	609	4
600	845	56	770	20	M 36	1 3/8	40	720	5
700	960	-	875	24	M 39	1 1/2	43	820	5
800	1085	-	990	24	M 45	-	49	928	5
900	1185	-	1090	28	M 45	-	49	1028	5
1000	1320	-	1210	28	M 52	-	56	1140	5
1200	1530	-	1420	32	M 52	-	56	1350	5

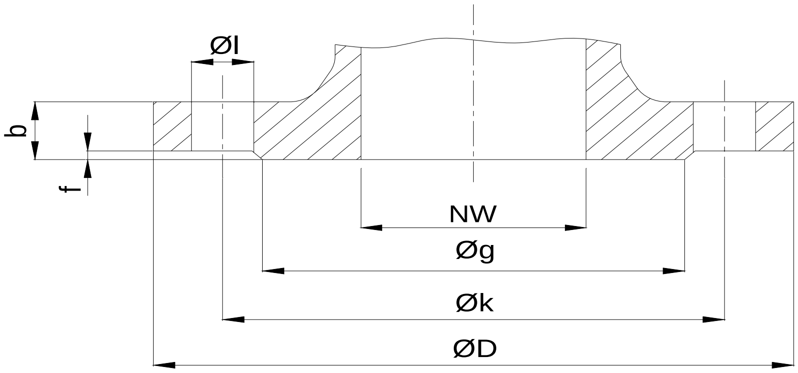


ISO PN40 FLANGE DIMENSIONS (TS ISO 7005-2)									
NOMINAL DIAMETER NW	FLANGE			NUT				g	f
	∅D	b	∅k	Piece	d		∅l		
					Metric	Inch			
40	150	20	110	4	M 16	5/8	19	84	3
50	165	22	125	4	M 16	5/8	19	99	3
65	185	24	145	8	M 16	5/8	19	118	3
80	200	26	160	8	M 16	5/8	19	132	3
100	235	28	190	8	M 20	¾	23	156	3
125	270	30	220	8	M 24	7/8	28	184	3
150	300	34	250	8	M 24	7/8	28	211	3
200	375	40	320	12	M 27	1	31	284	3
250	450	46	385	12	M 30	1 $^{1}/_{8}$	34	345	3
300	515	50	450	16	M 30	1 $^{1}/_{8}$	34	409	4
350	580	54	510	16	M 33	1 ¼	37	465	4
400	660	62	585	16	M 36	-	40	535	4
450	685	-	610	20	M36	-	40	560	4
500	755	-	670	20	M 39	1 ½	43	615	4
600	890	-	795	20	M 45	-	49	735	5

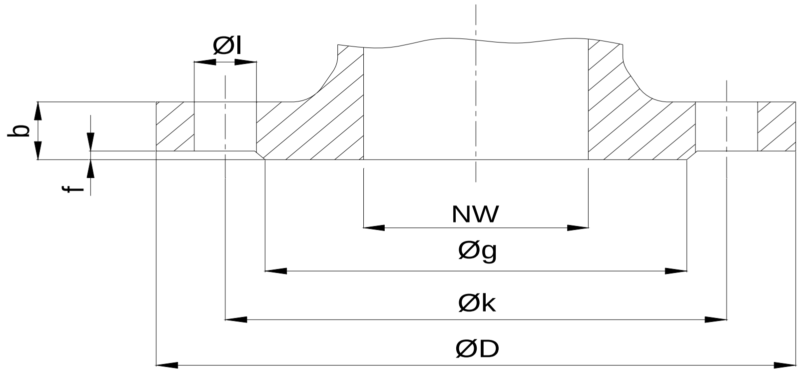


ISO PN50 FLANGE DIMENSIONS (TS ISO 7005-2)									
NOMINAL DIAMETER NW	FLANGE			NUT				g	f
	∅D	b (min.)	∅k	Piece	d		∅l		
					Metric	Inch			
40	155	20.5	114.5	4	M20	3/4	22	90	2
50	165	22.5	127	8	M16	5/8	18	106	2
65	190	25.5	149.5	8	M20	3/4	22	125	2
80	210	28.5	168	8	M20	3/4	22	144	2
100	255	32	200	8	M20	3/4	22	176	2
125	280	35	235	8	M20	3/4	22	211	2
150	320	36.5	270	12	M20	3/4	22	246	2
200	380	41	330	12	M24	7/8	26	303	2
250	445	48	387.5	16	M27	1	29.5	357	2
300	520	51	451	16	M30	1 $^{1}/_{8}$	32.5	418	2
350	585	54	514.5	20	M30	1 $^{1}/_{8}$	32.5	481	2
400	650	57	571.5	20	M33	1 $^{1}/_{4}$	35.5	535	2
450	710	60.5	628.5	24	M33	1 $^{1}/_{4}$	35.5	592	2
500	775	63.5	686	24	M33	1 $^{1}/_{4}$	35.5	649	2
600	915	70	813	24	M39	1 $^{1}/_{2}$	42	770	2
750	1095	76	997	28	M45	-	48	945	2

22.1 ANSI B 16-5 - 1981



PN20 (CLASS 150) FLANGE DIMENSIONS (ANSI B 16-5 - 1981)									
NOMINAL DIAMETER		FLANGE				HOLE		g	f
DN	NPS	∅D	NW (min.)	b	∅k	Piece	∅l		
15	1/2	90	22	11.5	60.5	4	16	34.9	2
20	3/4	100	28	13	70	4	16	42.9	2
25	1	110	34.5	14.5	79.5	4	16	50.8	2
32	$1\,^{1}/_{4}$	120	43.5	16	89	4	16	63.5	2
40	$1\,^{1}/_{2}$	130	49.5	17.5	98.5	4	16	73	2
50	2	150	62	19.5	120.5	4	20	92.1	2
65	$2\,^{1}/_{2}$	180	74.5	22.5	139.5	4	20	104.8	2
80	3	190	90.5	24	152.5	4	20	127	2
-	$3\,^{1}/_{2}$	215	103.5	24	178	8	20	139.7	2
100	4	230	116	24	190.5	8	20	157.2	2
125	5	255	143.5	24	216	8	22	185.7	2
150	6	280	170.5	25.5	241.5	8	22	215.9	2
200	8	345	221.5	29	298.5	8	22	269.9	2
250	10	405	276	30.5	362	12	26	323.8	2
300	12	485	327	32	432	12	26	381	2
350	14	535	359	35	476	12	30	412.8	2
400	16	600	410.5	37	540	16	30	469.9	2
450	18	635	462	40	578	16	33	533.4	2
500	20	700	513	43	635	20	33	584.2	2
600	24	815	616	48	749.5	20	36	692.2	2

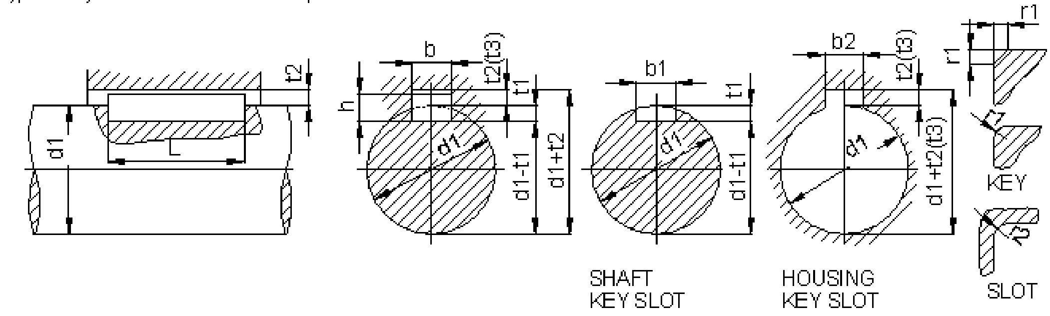


PN50 (CLASS 300) FLANGE DIMENSIONS (ANSI B 16-5 - 1981)									
NOMINAL DIAMETER		FLANGE				HOLE		g	f
DN	NPS	∅D	NW (min.)	b	∅k	Piece	∅l		
15	1/2	95	22	14.5	66.5	4	16	34.9	2
20	3/4	120	28	16	82.5	4	20	42.9	2
25	1	125	34.5	17.5	89	4	20	50.8	2
32	1 1/4	135	43.5	19.5	98.5	4	20	63.5	2
40	1 1/2	155	49.5	21	114.5	4	22	73	2
50	2	165	62	22.5	127	8	20	92.1	2
65	2 1/2	190	74.5	25.5	149	8	22	104.8	2
80	3	210	90.5	29	168.5	8	22	127	2
-	3 1/2	230	103.5	30.5	184	8	22	139.7	2
100	4	255	116	32	200	8	22	157.2	2
125	5	280	143.5	35	235	8	22	185.7	2
150	6	320	170.5	37	270	12	22	215.9	2
200	8	380	221.5	41.5	330	12	26	269.9	2
250	10	445	276	48	387.5	16	30	323.8	2
300	12	520	327	51	451	16	33	381	2
350	14	585	359	54	514.5	20	33	412.8	2
400	16	650	410.5	57.5	571.5	20	36	469.9	2
450	18	715	462	60.5	628.5	24	36	533.4	2
500	20	775	513	63.5	686	24	36	584.2	2
600	24	915	616	70	813	24	42	692.2	2

Table : 23 Key and key slot standards

KEY AND KEY SLOT STANDARTS

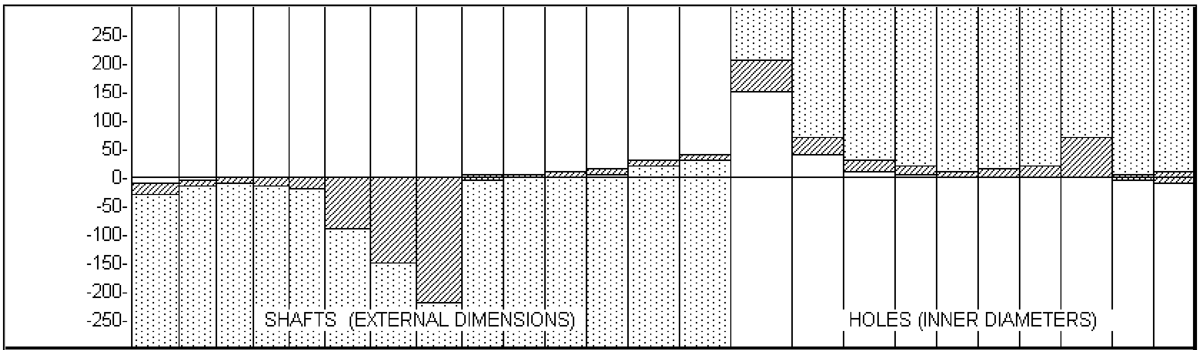
This file includes the key and key slots according to TS 147/9, DIN 6885/11 for those are used in Turbosan product. For other type of key refer to related TS 147/9, DIN 6885/11 norms.



KEY	Width: b		b	(mm)	3	4	5	6	8	10	12	14	16	18	20	22	25	28	32
			tolerance	h9	0.000 -0.025		0.000 -0.030				0.000 -0.043					0.000 -0.052			0.000 -0.062
	Thickness: h		h	(mm)	3	4	5	6	7	8	8	9	10	11	12	14	14	16	18
			tolerance	h11		0.000 -0.030					0.000 -0.090					0.000 -0.011			
Shaft diameter space: d1			exterior	(mm)	8	10	12	17	22	30	38	44	50	58	65	75	85	95	110
			interior	(mm)	10	12	17	22	30	38	44	50	58	65	75	85	95	110	130
SHAFT KEY SLOT	Width: b1		b1	(mm)	3	4	5	6	8	10	12	14	16	18	20	22	25	28	32
		Tolerance	Tight	P9		-0.012 -0.042			-0.015 -0.051			-0.018 -0.061				-0.022 -0.074			-0.026 -0.088
			Wide	N9		0.000 -0.030			0.000 -0.036			0.000 -0.043				0.000 -0.052			0.000 -0.062
	Depth: t1		t1	(mm)	1.8	2.5	3	3.5	4	5	5	5.5	6	7	7.5	9,	9	10	11
			tolerance	(mm)		+0.1								+0.2					
HOUSING KEY SLOT	Width: b2		b2	(mm)	3	4	5	6	8	10	12	14	16	18	20	22	25	28	32
		Tolerance	Tight	P9		-0.012 -0.042			-0.015 -0.051			-0.018 -0.061				-0.022 -0.074			-0.026 -0.088
			Wide	Js9		±0.015			±0.018			±0.022				±0.026			±0.031
	Depth:	Space: t2	t2	(mm)	1.4	1.8	2.3	2.8	3.3	3.3	3.3	3.8	4.3	4.4	4.9	5.4	5.4	6.4	7.4
			tolerance	(mm)		+0.1								+0.2					
		Normal: t3	t3	(mm)	0.9	1.2	1.7	2.2	2.4	2.4	2.4	2.9	3.4	3.4	3.9	4.4	4.4	5.4	6.4
			tolerance	(mm)		+0.1								+0.2					
Key Korner	r1		min.	(mm)	0.16		0.25			0.40					0.6				
			max.	(mm)	0.25		0.40			0.60					0.8				
Slot Korner	r2		min.	(mm)	0.08		0.16			0.25					0.4				
			max.	(mm)	0.16		0.25			0.40					0.6				

Lenght - L	Tolerance Key	Tolerance Slot	Weight (kg / 1000 piece)														
14	-0.2	+0.2	0.989	1.76	2.75	3.94											
16			1.13	2.01	3.14	4.52											
18			1.27	2.26	3.53	5.09	7.93										
22			1.55	2.76	4.32	6.22	9.67	13.8									
28			1.98	3.52	5.50	7.91	12.3	17.6	21.1								
36	-0.3	+0.3	2.54	4.52	7.06	10.2	15.8	22.6	27.1	35.6							
45				5.65	8.83	12.7	19.8	28.3	33.9	44.5	56.5						
56					11	15.8	24.6	35.2	42.2	55.4	70.3	87.0	106				
63						17.8	27.7	39.6	47.5	62.3	79.1	97.9	119	152			
70						19.8	30.8	44.0	52.8	69.2	88.0	109	132	169	192		
80							35.2	50.2	60.3	79.1	100	124	151	193	220	281	
90	-0.5	+0.5					39.6	56.5	67.8	89.0	113	140	170	218	247	317	407
100								62.8	75.4	98.9	126	155	188	242	275	352	452
110								69.1	82.9	109	138	171	207	266	302	387	497
125									94.2	124	157	194	235	302	343	440	565
140									106	138	176	218	264	338	385	492	633
160										158	201	249	301	387	440	563	723
180											226	280	339	435	495	633	814
200												311	377	484	550	703	904
220													414	532	604	774	995
250														604	687	880	1130
280															769	985	1270
320																1130	1440

Table : 24 Numerical values of most used tolerance classes



Nominal Dimension Area (mm)	f7	g6	h6	h7	h8	h11	h12	h13	j6	k5	k6	m6	r6	t6	B10	D9	F7	G7	H6	H7	H8	H11	J6	J7
6 _ 10	-13 -28	-5 -14	0 -9	0 -15	0 -22	0 -90	0 -150	0 -220	+7 -2	+7 +1	+10 +1	+15 +6	+28 +19		+208 +150	+76 +40	+28 +13	+20 +5	+9 0	+15 0	+22 0	+90 0	+5 -4	+8 -7
10 _14 14 _18	-16 -34	-6 -17	0 11	0 -18	0 -27	0 -110	0 -180	0 -270	+8 -3	+9 +1	+12 +1	+18 +7	+34 +23		+220 +150	+93 +50	+34 +16	+24 +6	+11 0	+18 0	+27 0	+110 0	+6 -5	+10 -8
18 _24	-20 -41	-7 -20	0 -13	0 -21	0 -33	0 -130	0 -210	0 -330	+9 -4	+11 +2	+15 +2	+21 +8	+41 +28		+244 +160	+117 +65	+41 +20	+28 +7	+13 0	+21 0	+33 0	+130 0	+8 -5	+12 -9
24 _30														+54 +41										
30 _40	-25 -50	-9 -25	0 -16	0 -25	0 -39	0 -160	0 -250	0 -390	+11 -5	+13 +2	+18 +2	+25 +9	+50 +34	+64 +48	+270 +170	+142 +80	+50 +25	+34 +9	+16 0	+25 0	+39 0	+160 0	+10 -6	+14 -11
40 _50														+70 +54	+280 +180									
50 _ 65	-30 -60	-10 -29	0 -19	0 -30	0 -46	0 -190	0 -300	0 -450	+12 -7	+15 +2	+21 +2	+30 +11	+60 +41	+85 +66	+310 +190	+174 +100	+60 +30	+40 +10	+19 0	+30 0	+46 0	+190 0	+13 -6	+18 -12
65 _ 80													+62 +43	+94 +75	+320 +200									
80 _100	-36 -71	-12 -34	0 -22	0 -35	0 -54	0 -220	0 -350	0 -540	+13 -9	+18 +3	+25 +3	+35 +13	+73 +51	+113 +91	+360 +220	+207 +120	+71 +36	+47 +12	+22 0	+35 0	+54 0	+220 0	+16 -6	+22 -13
100 _ 120													+76 +54	+126 +104	+380 +240									
120 _ 140	-43 -83	-14 -39	0 -25	0 -40	0 -63	0 -250	0 -400	0 -600	+14 -11	+21 +3	+28 +3	+40 +15	+88 +63	+147 +122	+420 +260	+245 +145	+83 +43	+54 +14	+25 0	+40 0	+63 0	+250 0	+18 -7	+26 -14
140 _ 160													+90 +65	+159 +134	+440 +280									
160 _ 180													+93 +68	+171 +146	+470 +310									
180 _ 200	-50 -96	-15 -44	0 -29	0 -46	0 -72	0 -290	0 -460	0 -720	+16 -13	+24 +4	+33 +4	+46 +17	+106 +77	+195 +166	+525 +340	+285 +170	+96 +50	+61 +15	+29 0	+46 0	+72 0	+290 0	+22 -7	+30 -16
200 _ 225													+109 +80		+565 +380									
225 _ 250													+113 +84		+605 +420									
250 _ 280	-56 -108	-17 -49	0 -32	0 -52	0 -81	0 -320	0 -520	0 -810	+16 -16	+27 +4	+36 +4	+52 +20	+126 +94		+690 +480	+320 +190	+108 +56	+69 +17	+32 0	+52 0	+81 0	+320 0	+25 -7	+36 -16
280 _ 315													+130 +98		+750 +540									
315 _ 355	-62 -119	-18 -54	0 -36	0 -57	0 -89	0 -360	0 -570	0 -890	+18 -18	+29 +4	+40 +4	+57 +21	+144 +108		+830 +600	+350 +210	+119 +62	+75 +18	+36 0	+57 0	+89 0	+360 0	+29 -7	+39 -18
355 _ 400													+150 +114		+910 +680									
400 _ 450	-68 -165	-20 -60	0 -40	0 -63	0 -97	0 -400	0 -630	0 -970	+20 -20	+32 +5	+45 +5	+63 +23	+166 +126		+1010 +760	+385 +230	+131 +68	+83 +20	+40 0	+63 0	+97 0	+400 0	+33 -7	+43 -20
450 _ 500													+172 +132		+1090 +840									

1_ This table is taken from TSE 1845 (DIN 7154) and includes mostly using tolerance class in Turbosan products.
2_ Ø100 mm tolerance areas specifed on the above chart.
3_**ATTENTION !** When part tolerance measuring part temperature must be neither be cold nor too hot.
Temperature should be between $+15^{\circ}C$~$+30^{\circ}C$. Especially for diameters above Ø100 mm. Thermal expansion is getting more important.

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ALPHABETIC INDEX

PART 1

TERMS, DEFINITIONS AND FUNDAMENTAL FORMULAS

This section covers terms, symbols and related formulas and explanations used in this handbook.

Terms	Symbol	Unit	Definitions
Flow rate, volumetric	Q	m^3/s, lt/s lt/min, m^3/h	The volume of the fluid delivered by the pump or the pipe in the unit of time
Total flow rate	Q_T		The volume of the fluid delivered by the pump or the pipe in the unit of time
Flow rate – mass	m	kg/s, kg/min	The mass of the fluid delivered by the pump or the pipe in the unit of time
Geometric pump head	H_g	m	Difference of altitude between the liquid levels of suction and discharge reservoirs
Static pump head	H_{st}	m	Potential energy transferred by the pump per unit weight of the pumped liquid
Pump total head	H_m	MAY, mSS	Total energy provided by the pump per unit weight of the pumped liquid
Net positive suction head	NPSH	m	Difference between the total head of the fluid at the pump inlet flange and vapor pressure head
Power – shaft	N_e	kW	Shaft power
Power – hydraulic	N_0	kW	Power transferred to the fluid by the pump impeller
Power – motor	N_m	kW	Power absorbed by the electric motor or driving unit
Efficiency – hydraulic	η_h	(-)	Ratio of the useful power of the fluid to absorbed power by the impeller
Efficiency – general	η_g	(-)	Ratio of the useful power of the fluid to the shaft power
Rotation speed	n	rpm	Rotational speed (revolution per minute)
Pressure	p	Pascal (Pa)	Fluid force acting perpendicularly on the surface of unit area
Density	ρ	kg/m^3	Mass of the fluid per unit volume
Flow velocity	V	m/s	Velocity
Gravitational acceleration	g	m/s^2	Gravitational acceleration
Vertical distance	z	m	Vertical distance between the considered point and defined reference plane
Specific weight	$\gamma = \rho.g$	N/m^3	Weight of the fluid per unit volume
Total minor losses	$\sum_{A-B} h$	m	Total head loss occuring inside the valves, fittings and pipe specials
Friction coefficient	λ	(-)	Coefficient used in the calculation of the head losses in the straight pipes and can be obtained from Moody diagram
Reynolds number	Re	(-)	Ratio of the inertia force to the viscous force
Relative surface roughness	ε	(-)	Ratio of the absolute average roughness to the pipe diameter
Roughness	k	mm	Absolute average roughness of the surface
Diameter	D	m	The diameter of the pipe
Minor loss	h_{local}	m	Head loss occuring inside the valves, fittings and pipe specials
Friction	$h_{continuous}$	m	Head loss of the fluid inside the straight pipes
Suction specific speed	S_q	(-)	Cavitation specific speed of the pump
Dynamic viscosity	μ	Pa.s	Dynamic viscosity
Kinematic viscosity	ν	m^2/s, stokes	Kinematic viscosity

PART 2

PUMPS AND RELATED BASIC TERMS

Pump, by its most general definition, a pump is a machine that transforms the mechanical energy into hydraulic energy. In other words, energy level of liquid, passing through the pump, is increased. The mechanical energy that is available at the shaft of the driving unit is converted into the hydraulic energy by the pump. Pumps are used for the following purposes:

- To deliver the liquid from lower to higher level,
- To pump the liquid from the suction tank with low pressure to the discharge tank with high pressure,
- To increase the flow rate of the liquid flowing from the suction reservoir located at the higher altitude than the discharge reservoir. In this application, pump compensates the friction losses of the piping system.

Pumps are driven by electric motors, diesel engines, turbines, etc. The system consisting of a suction pipe, pump, delivery pipe is called the pumping system. The total head loss, occuring inside the suction pipes, delivery pipe fittings, valves and pipe specials, is a function at the flowrate and is called head loss characteristic of the pumping system or system head characteristic.

2.1 Flow Rate (Q)

The flowrate is the volume of the liquid delivered by the pump in the unit of time. If the amount of the liquid is expressed as mass instead of volume, it is called mass flowrate.
The flowrate is the main independent variable of the pump and pumping system.

2.2 Pump Total Head (H_m)

The total head of the liquid is composed of potential energy, kinetic energy and pressure energy per unit weight of the liquid. The pump total head is the amount of the total energy transferred to the unit weight of the pumped liquid between pump inlet and outlet:

$$H_m = (\frac{p_b}{\rho g} + z_b + \frac{V_b^2}{2g}) - (\frac{p_e}{\rho g} + z_e + \frac{V_e^2}{2g})$$

where indexes e^i indicates the pump exit and inlet respectively.
In other words,

$\frac{Pe - Pi}{99}$: difference of pressure head between exit and inlet of a pump

Ze-Zi: difference of altitude between exit and inlet of a pump

$\frac{Ve2 - Vi2}{2g}$: difference of velocity head of liquid between outlet and inlet of a pump

Delivery tank
Suction tank
Reference plane

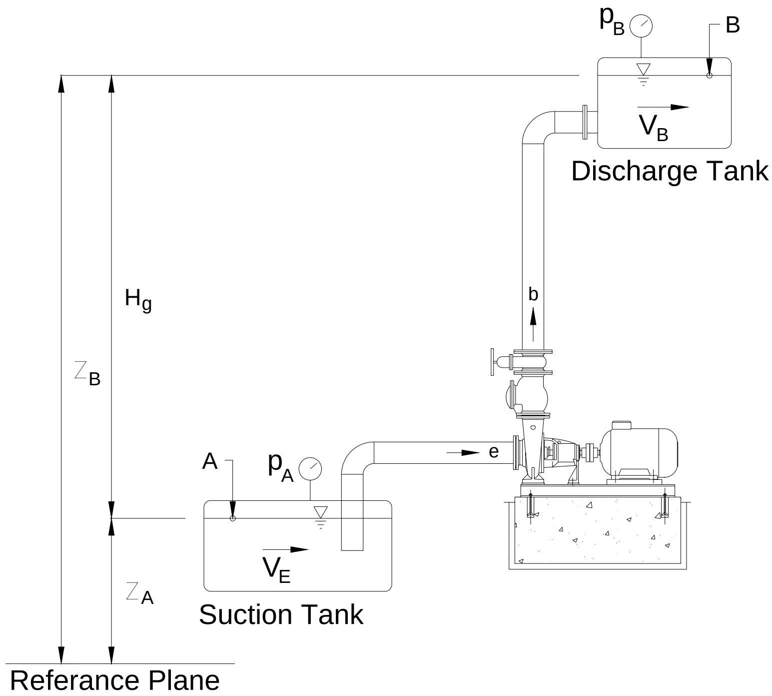


Figure 2.1 A Typical Pumping System

Now let's introduce the geometrical characteristic of the pumping system, given in Fig. 2.1, into the total head term.
The sum of pressure and potential energy may be defined as static energy. The static energy increase per unit weight of the fluid is called the "static head" of the pump:

$$H_{st} = (\frac{p_B}{\rho g} + z_B) - (\frac{p_A}{\rho g} + z_A)$$

Where the indexes A and B indicate the free surfaces of the fluid at the suction and delivery tank respectively.
The difference in altitude (Z_B-Z_A), is the geometric head of the pump.

$$H_g = z_B - z_A$$

If the altitude of fluid free surface in delivery tank is lower than the one of the suction tank, the geometric head of the pump becomes negative. Similar situations may happen for the static head as well. In such situations, pump increases flowrate of the fluid by compensating the friction losses of the piping system.
If the same pressure (for example atmospheric pressure) acts on the free surfaces of the fluid in the suction and delivery tanks, (P_A=P_B=P_{atm}), the static and geometric heads are the same:

H_{st}=H_g

If the suction pipe at the pump inlet and delivery pipe at the pump exit have the same diameter, the total head of the pump becomes:

$$H_m = (z_b - z_e) + (\frac{p_b - p_e}{\rho g}) + \cancelto{0}{(\frac{V_b^2 - V_e^2}{2g})}$$

Its valve can be directly determined by using the pressure gauge (manometer) readings and it is therefore called also as manometric head.

If the diameters of the suction pipe at the pump inlet (Db) and delivery pipe at the pump exit are different, the average fluid (velocities, V_b and V_e, rewuired) for the calculation of total head are calculated by using the pump flowrate.

After designing the pumping system, pump total head is calculated by using following equation:

$$H_m = H_g + (\frac{p_B - p_A}{\rho g}) + \sum_{A-B} h$$

Where the term $\sum_{A-B} h$ is the total head loss of the fluid occurring between point (A) and (B) in figure located free fluid surfaces of the suction and discharge tanks respectively.

2.1 This term does not include the loss occurring inside the pump since it is covered by pump efficiency.
If the same pressure acts on the free fluid surfaces of suction and delivery tanks, pump total head is calculated by the following equation:

$$\boxed{H_m = H_g + \sum_{A-B} h}$$

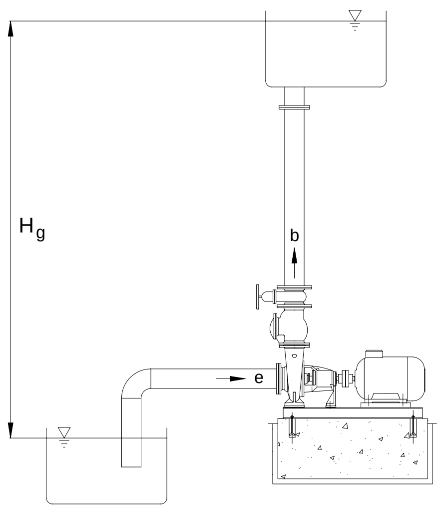


Figure 2.2 Pumping System with open tanks (reservoirs)

It is useful to remember the following remark at this point:

Someone should always keep in mind that atmosphere pressure is nearly equal to 1 kg/cm^2 or 1 bar or 10,33 mWC or 760 mmHg.
1 atmosphere pressure is equivalent to about 1 kg/cm^2 or 1 bar or 10,33 mSS or 760 mmHg height.

2.3 Effect of the density of the pumped liquid on the pump total head and pump outlet pressure

The pump total head is not a function of the pumped liquid density. The pump total head can be measured by using the U-tube manometer in terms of meter liquid column and it does not change with the similar kinematic viscosity. If the kinematic viscosity of the pumped liquid is larger than 20 mm^2/s (kinematic viscosity of water at 20^o C is equal to 1 mm^2/s or 1cSt), pump total head does not remain the same.

The pump exit pressure is a function of the pumped liquid density and is calculated approximately by
$P_{exit} = P_{inlet} + \delta g H$

For viscous liquids whose kinematic viscosity is above 600 cSt, the use of centrifugal pumps is not adequate on account of the excessive drop in the efficiency and extreme increase in the power. Here gear pumps with positive displacement are suitable.

2.4 Power Definitions

a) Theoretical Power (Hydraulic Power) : N_0

The power that the fluid receives *from the pump* irrespective of several energy losses occurring in the pump is called "theoretical power" and is referred to as N_0. The power received (acquired) by the pumped liquid is

$N_0 = \rho\ g\ Q\ H_m = \gamma\ Q\ H_m$

In order to compensate the losses occurring inside the pump a power more than N_0 should be applied to the shaft of the impeller. As it stands, N_0 is the useful power of the pump.
In this expression (ρ) is the density of the fluid (kg/m^3); (g) is the gravitational acceleration (m/s^2); (Q) is the flow rate of the pump (m^3/s); (H_m) is the pump total head (m) and (η_g) is the overall efficiency (nondimensional)of the pump.

b) Effective Power, Shaft Power: N_e

The power which must be applied to the shaft of the pump and drawn from the driving unit (e.g. electrical motor) is called the "effective power" or the "shaft power" of the pump and is expressed by N_e . The shaft power may be calculated in kW or HP using the formulas given in the following table:

Power Formula	Unit	Power Unit
$N_e = \dfrac{\rho g Q H_m}{1000\,\eta_g}$	Q: m^3/s H_m:m ρ: kg/m^3(for water 1000) g:m/s^2 η_g: dimensionless	Ne:kW

$N_e = \frac{\rho Q H_m}{102 \eta_g}$	Q: liter/s H_m: m ρ: kg/dm^3 (for water 1) η_g: deminsonless	Ne:kW
$N_e = \frac{\rho g Q H_m}{75\, \eta_g}$	Q: liter/s H_m: m ρ: kg/dm^3 (for water 1) η_g: nondimensional	Ne:HP

Moreover it should be remembered that the following relationship exists between HP and KW

$Ne_{BG} = Ne_{kW}$ *x 1.36*

c) Power drawn by the Electrical Motor from the Mains: N_s

$$N_s = \frac{N_e}{\eta_m} \qquad N_s = \frac{\rho \cdot Q \cdot H_m}{102 \cdot \eta_s} \qquad \eta_s = \eta_m \cdot \eta_p$$

η_s= system efficiency (efficiency of the system consisting of electric motor and pump)
η_m= motor efficiency
η_p= pump efficiency

- The power drawn by the motor from the mains (N_s), is greater than the shaft power (N_e) due to the losses occurring inside the motor (coils, frictions, heating etc.).
- The power drawn by the motor from the mains (N_s) is obtained by dividing the effective power (shaft power) (N_e) to the motor efficiency (η_m). The motor efficiency is given in the catalogs of motor manufacturers.

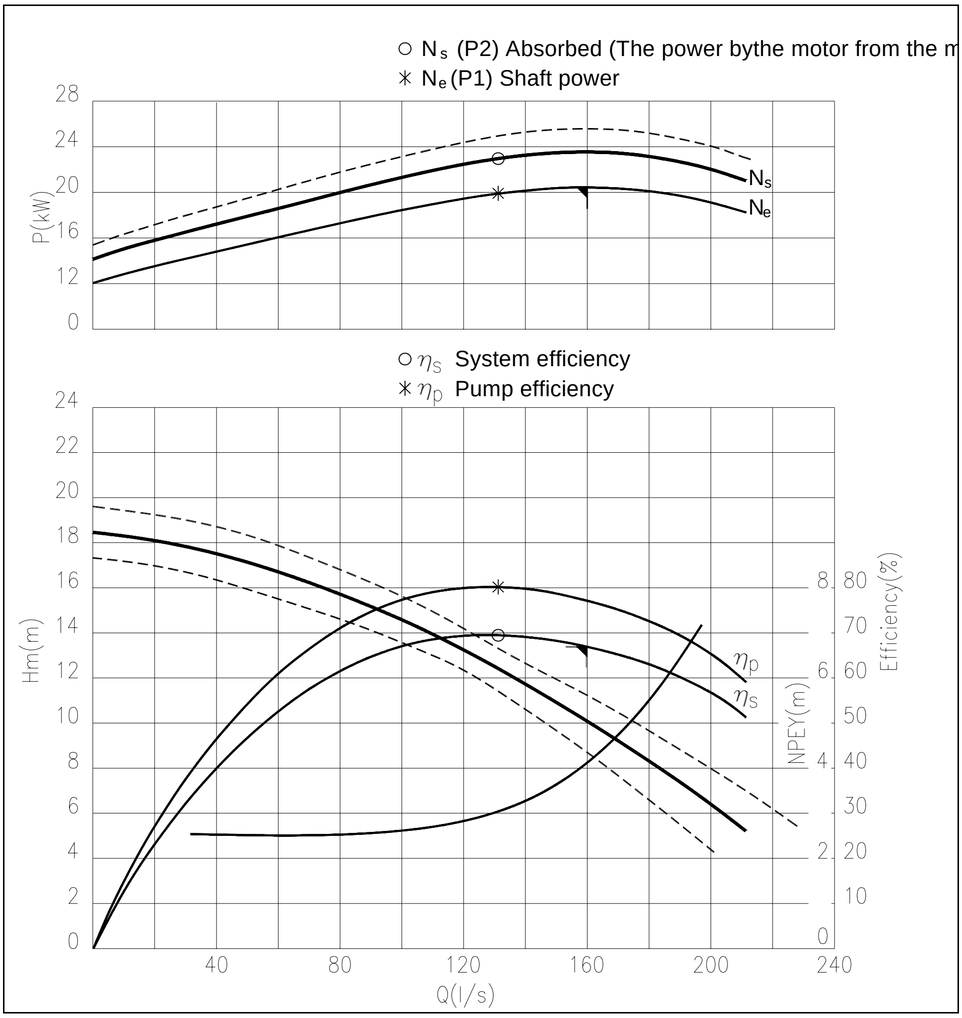


Figure 2.3 Pump and System Performance Curves

When choosing the electrical motor to drive the pump, the power of the motor must be somewhat higher than the shaft power as the pump may be run for flow rates above the labeled values. How high should the motor power be depends on the power of the motor to be selected and on the operating conditions and must be calculated by the formula $N_m = N_s \, x \, k$ where the value of (k) is found from the following table in connection with the motor power.

Motor power (kW)	k (coefficient)
1,1 < 4	1,30
5,5 < 37	1,25
45 < 250	1,20
315 >	1,10 ~ 1,15

2.5 Pump Efficiency and System Efficiency : η_p

The ratio of the power *drawn* from a pump to the power *supplied* is called “overall efficiency” or “total –global- efficiency and is indicated by (η_p), or by (η). This is calculated by the following formula:

$$\eta_p = \frac{N_0}{N_e}$$

The efficiency of the system consisting of a pump and motor is calculated with the following formula, where the efficiency of the drive unit (the motor) is included in the definition:

$$\eta_{g-sis} = \frac{N_0}{N_s}$$

Therefore,

$\eta_{g-sis} = \eta_p x \eta_m$ (system efficiency = pump efficiency x motor efficiency)

2.6 Specific Speed

Specific speed determines the type of the pump impeller (radial, mixed flow or axial). Therefore, the term specific speed is related to the pump designer. It is calculated by using the data of the design point characteristics.

Specific speed is defined in the literature by many ways. Three most referred definitions are given in the following table:

	Specific speed formula	**Units**	**Specific Speed Unit**
$\mathbf{n_q}$	$\frac{n\sqrt{Q}}{H^{3/4}}$	n: rpm H: m Q: m^3/s	Rpm
$\mathbf{n_s}$	3,65 . n_q		rpm
$\mathbf{n_s^1}$	$\frac{n\sqrt{Q}}{(gH)^{3/4}}$	n: rad/s Q: m^3/s g: m/s^2 H:m	(-)

Impeller types according to specific speed values are indicated in the following Figure 2.4. As the specific speed of axial impellers is higher than that of radial impellers, axial impellers are called high speed impellers and centrifugal impellers are called low speed impellers. These ranges will have different values for different definitions of the specific speed.

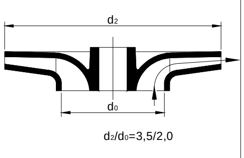
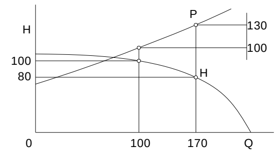
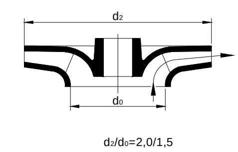
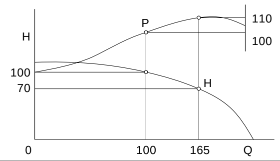
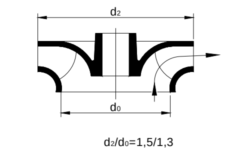
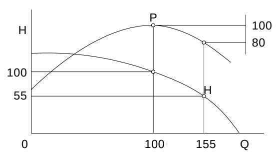
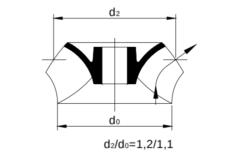
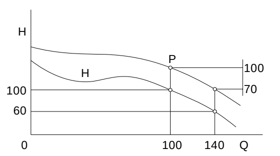
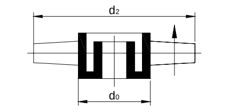
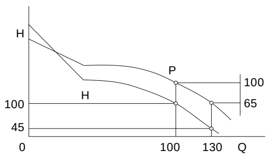
	IMPELLER TYPE	CHARACTERISTICS	Speed - nq (rpm)	EXAMPLE PUMP
CENTRIFUGAL			10 - 30	NORM 40/200 KOT 80
CENTRIFUGAL			30 - 50	NORM 125/250 CEP 400/400
HELIKOCENTRIFUGAL			50 - 80	NORM 300/400 CEP 300/315
MIXED FLOW			80 - 150	EKS-K 400 DAC-K 600
AXIAL			135 - 120	EKS-E 1200 DAC-E 700

Figure 2.4 Examples of Pump Impellers and Characteristic Curve Types for Different Specific Speeds

2.7 Calculating Pump Shaft Diameter

The shaft of the pump is chosen and sized in accordance with the moment it will convey, the number of revolutions, the material quality and the characteristics of the liquid to be pumped. The diameter of the shaft is calculated by using the following equation:.

$$Ne = \frac{M.\omega}{75}$$ Ne : shaft power (HP)

$$\omega = \frac{\pi.n}{30}$$ M : moment (kg.m)

$$d = \sqrt[3]{\frac{16.M}{\pi.\tau}}$$ ω : angular speed (1/s)

n : rpm

τ : shear strength of the shaft material (kg/cm^2)

d : shaft diameter (cm)

The shaft diameter is obtained by the formula $d = \sqrt[3]{\frac{360000.Ne}{\tau.n}}$.

According to the type of the material, shear strength values (τ) are as follows:

For Ç-1040 $\tau = 450$ kg/cm^2
For Ç-1050 $\tau = 600$ kg/cm^2
For AISI 304 $\tau = 750$ kg/cm^2
For AISI 420 $\tau = 1150$ kg/cm^2

When the diameter of the shaft is found, other dimensions of the shaft are formed by taking into account rabbets necessary for production ease and safety. For details see books on strength of materials and machine elements.

Shaft moment $M = T \cdot W$

$$M = 9549.3 \frac{N_e}{n} \qquad M = N_m, \quad N_e = kW, \quad n = rpm$$

$$M = 716.2 \frac{N_e}{n} \qquad M = kg \cdot m, \quad N_e = HP, \quad n = rpm$$

2.8 Pump Data Form

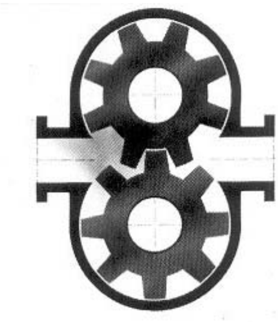
When making a pump request, the following pump data sheet must be filled out and sent to the pump manufacturer.

TürboSan A.Ş.	PUMP DATA SHEET		DATE :
CUSTOMER :			
	PUMP TECHNICAL INFORMATION		
1	MEDIUM		
	MEDIUM TEMPERATURE	°C :	
	MEDIUM DENSITY	gr/cm^3 :	
	VAPOUR PRESSURE	Bar :	
	DYNAMIC VISCOSITY	cP :	
	NPSHav	m :	
2	CAPACITY	m^3/h :	
	TOTAL HEAD	m :	
	SELECTED PUMP		
3	PUMP MODEL		
	SPEED	rpm	
	EFFICIENCY	%	
	WHP	kW	
	POWER ABSORBED	kW	
	$NPSH_{Pump}$	m	
	CONSTRUCTION		
4	VERT. / HORIZ.	V / H	
	SUCT. (Single /Double)	S / D	
	STAGE (Single /Multi)	S / M	
	ROTATION	CW /CCW	
	DISCH. FLANGE NORM		
	SUCT. FLANGE NORM		
	SEAL : MECH. (Single/Double)	S / D	
	SEAL : PACKING		
	MATERIALS		
5	CASING		
	IMPELLER		
	SHAFT		
	WEAR RING		
	STUFING BOX		
	SHAFT SLEEVE		
	COLUMN PIPE		
	BEARING HOUSING		
	BEARINGS		
	LUBRICATION		
6	INTERMEDIATE/LOWER BEARING LUB.		
	UPPER (AXIAL) BEARING LUB.		
	DRIVER		
7	DIESEL / ELECTRIC	D /E	
	PHASE	ph	
	VOLTAGE	V	
	FREQUENCY	Hz	
	NOMINAL POWER	kW	
	SPEED	rpm	
	PROTECTION CLASS	IP	
	INSULATION CLASS		
	MANUFACTURER / ORIGIN		
	WEIGHT	kg	
REMARKS : Any additional data can be made available on request			

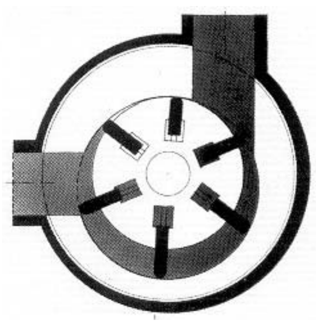
Figure 2.5 Pump Data Sheet

PART 3

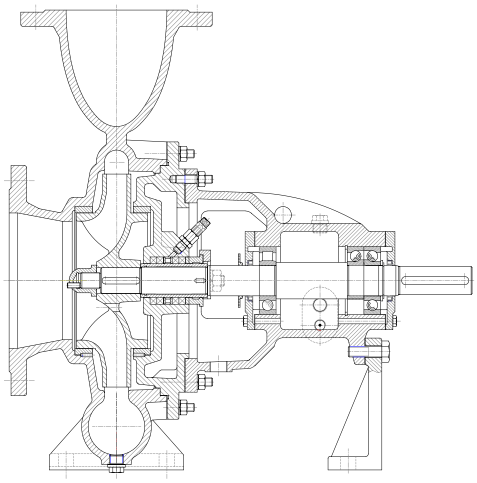
Classification of Pumps
Pumps may be classified in many different ways according to the type of the energy transfer occurring inside the pump, flow direction, impeller number etc. Among all this classification the most important one is the one made according to type of the energy transfer:
a) Volumetric pumps
b) Rotodynamic pumps



Gear pump



Sliding vane rotary pump



a. *Volumetric pumps*

b. *Roto-dynamic pump*
(Turbosan norm 150/315 pump)

3.1. Volumetric Pumps (Pumps with Positive Displacement)

Volumetric pumps have one or several volumes since the amount of these volumes changes during the operation the fluid is drawn into volume, compressed and forced out. As these volumes are filled and discharged during the operation, the pumping of the fluid is intermittent and the connection between the suction and the delivery parts is not continuous. Since these pumps push the fluid to the tank, the fluid pressure in the discharge tank may theoretically reach very high values. A safety valve should be placed in order to prevent the explosion of the discharge tank.

Piston pumps, gear pumps, sliding vane rotary pumps, and diaphragm pumps available in the market are examples of volumetric pumps. Volumetric pumps are preferred for their very high pressure values.

In these pumps, the change of pump head by the flow rate is a vertical line passing theoretically through Q flow rate, as shown in Figure 3.1. The increase in the pump RPM moves this line to high flow rate value, i.e. to the right side, and the decrease in the pump speed moves this line to low flow rate value, i.e. to the left side.

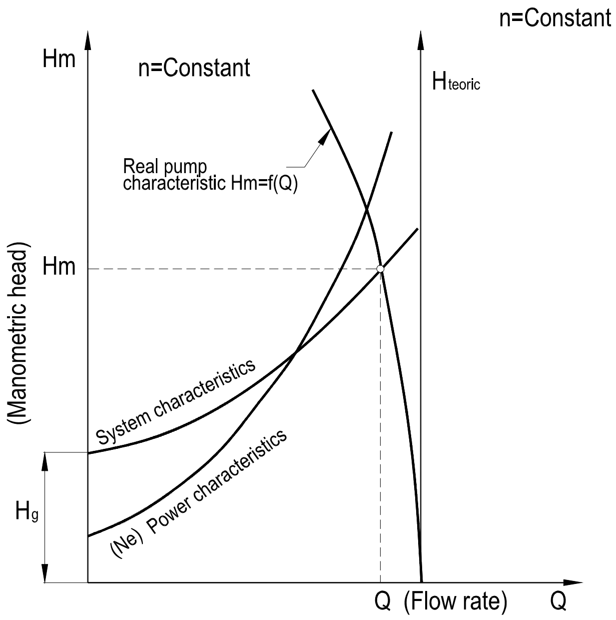


Figure 3.1. Characteristics of a Volumetric Pump

The theoretical characteristic in the form of a vertical line, bends towards low flow rate values as the pressure increases and takes the form of a curve in reality. The reason thereof is the leaking fluid between the stationary and moving parts in high pressure values.

3.2 Rotodynamic Pumps

In rotodynamic pumps the fluid flows continuously between the suction and discharge parts. The momentum of the flow changes within the pump impeller which generates a pressure difference. Centrifugal, mixed flow and axial flow pumps mentioned in Part 2 are within this group. The characteristic curves of this type of pumps are considerably different from that of volumetric pumps as explained in Part 2 and the pump head of rotodynamic pumps is a function of the flow rate passing through the pump. (Figure 3.2).

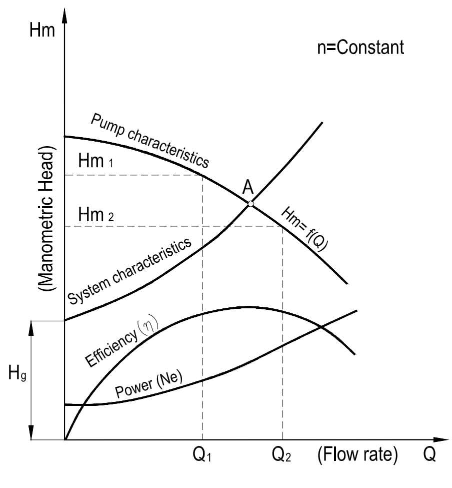
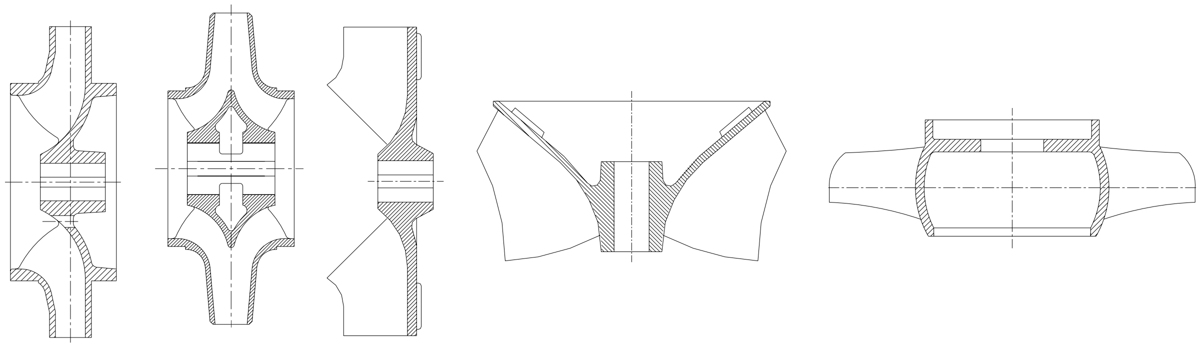


Figure 3.2 Rotodynamic pump characteristics

3.2.1 Classification of Rotodynamic Pumps according to Flow Type (Impeller Type)

The most fundamental classification of rotodynamic pumps is the one according to flow. The specific speed of the fluid at the targeted operating point calculated by using the pump total head, the flow rate and the rpm values defines the impeller type that the pump would operate with highest efficiency (design point).

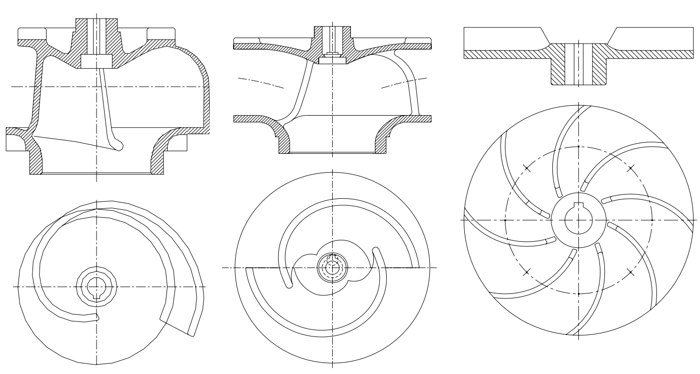


Centrifugal impeller	Double suction centrifugal impeller	Open impeller	Mixed flow impeller	Axial impeller

Figure 3.3 Rotodynamic pump impeller types

As explained in Part 2, pump impellers may be classified as centrifugal, mixed flow and axial. An example of each type is shown in Part 2, Figure 2.4. Besides, the Figure 3.3 gives some examples of rotodynamic pump impellers.

In rotodynamic pumps, there are impellers with wide spaces used for pumping fluids containing solid particles. Clogging problem does not occur in such impellers. They may bedesigned as torque-flow pumps, single vane or two or three vane impellers (Figure 3.4)



a) single vane b) double vane c) torque-flow

Figure 3.4 Sewage pump impellers

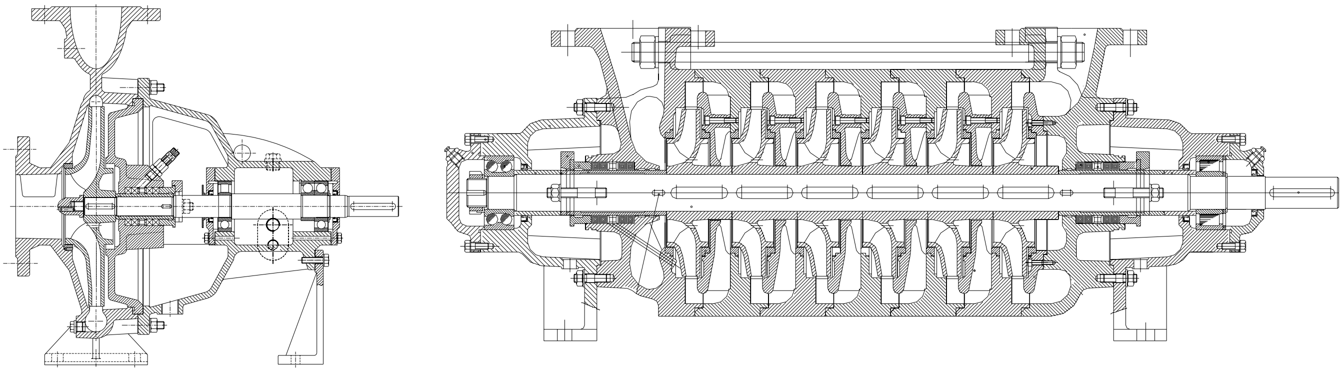
3.2.2 Classification of Rotodynamic Pumps according to their Structural Characteristics

Rotodynamic pumps may be classified in several forms according to their structural properties:

1) According to the number of impellers:

a) single stage

b) multi-stage



a) Single stage (*Türbosan Norm 80/315 pump*)

b) Multi stage *(Türbosan YKP 65x6)*

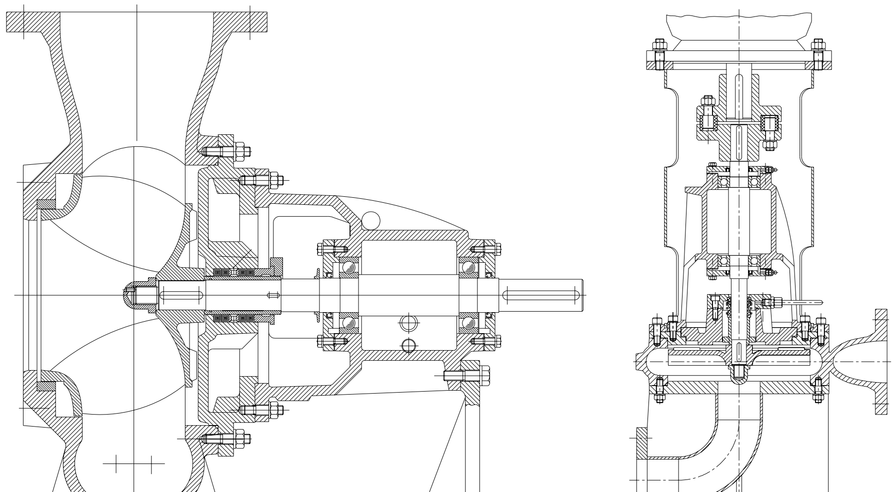
Figure 3.5 (a) Single stage and (b) Multi stage Centrifugal Pump

2) According to the position of the pump axis:

a) horizontal axis

b) vertical axis

c) in-line



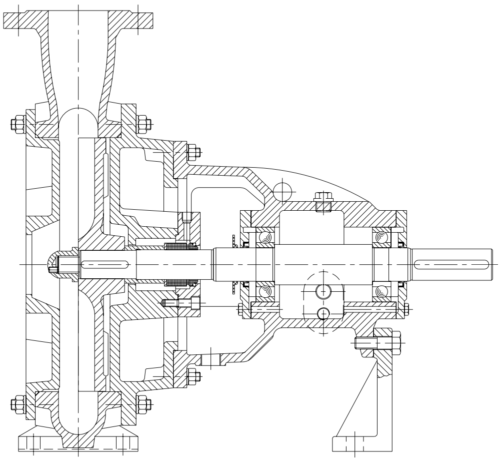
a) Horizontal axis (*Türbosan Çap 200/315F-HK pump*)

b) Vertical axis (*Türbosan Çap*

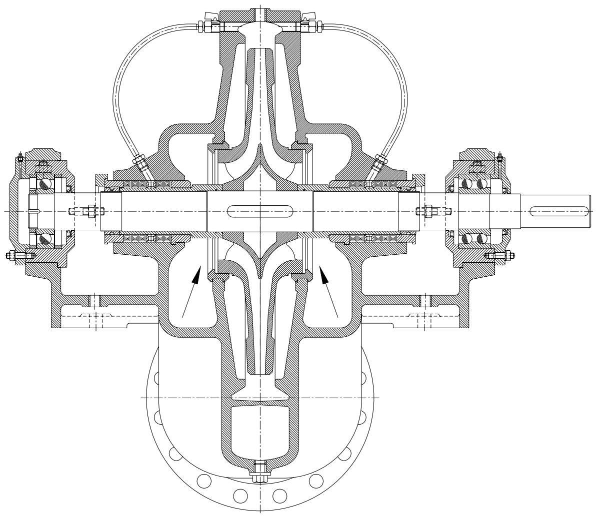
80/250V-VM pump)

Figure 3.6 a) horizontal axis b) vertical axis

3) According to the number of suction flanges :
   a) single suction
   b) double suction



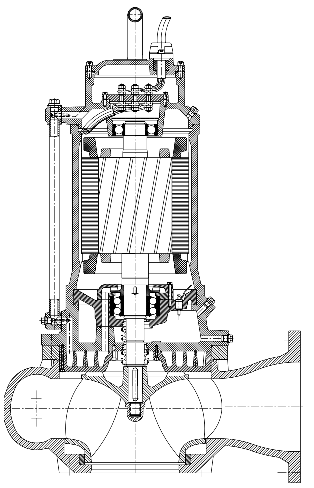
a) Single Suction Pump
(*Türbosan Çap 80/315V-HK pump*)



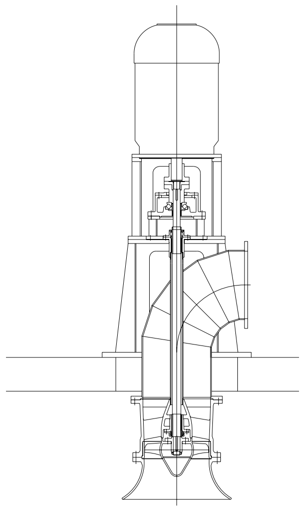
b) Double Suction Pump
(*Türbosan Çep 200/690 pump*)

Figure 3.7 a) single suction pump and b) double suction pump

4) According to the operation position:
   a) submersible pump
   b) deep well pump



a) Submersible Pump
(*Türbosan Dac-Y 200/315F pump*)



b) Axial Flow Pump
(*Türbosan Eks-K 400 pump)*

Figure 3.8 a) submersible pump and b) axial flow pump

5) According to the type of the fluid pumped:

a) hot oil pump
b) slurry pump
c) circulation pump

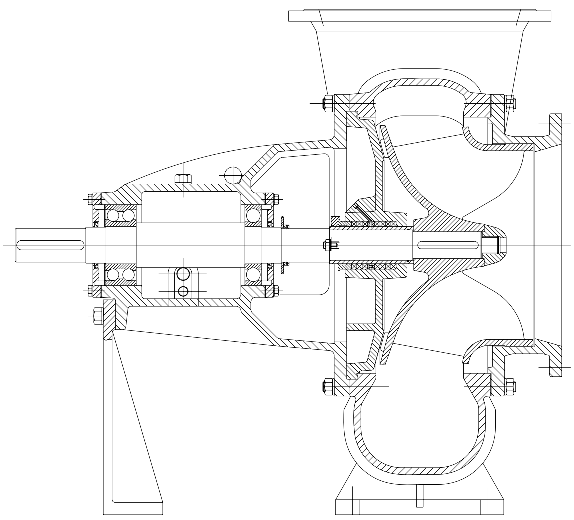


Figure 3.9 Sludge pump
(Türbosan CAP 600/600 pump)

3.3 Single Stage Volute Centrifugal Pumps (NORM)

Covers low and medium pressure single stage volute centrifugal pumps, conforming to TS EN ISO 9905 and EN 733 (DIN 24255)-EN 22858 (DIN 24256). Principal fields of usage are agricultural irrigation and sprinkler installations, potable water installations, cooling water and circulation circuits and heavy industries like iron and steel production plants.

Flow rate (L/s)	4 - 550
Hm (mWC)	5 - 90
Discharge diameter (inches)	50 - 300

3.4 Double Suction Centrifugal Pumps (CEP)

Covers split case horizontal/vertical shaft double suction centrifugal pumps conforming to TS EN ISO 9905. Among utilization fields are irrigation and potable water installations, refineries, oil pipeline installations, cooling water and circulation circuits and heavy industries like iron and steel production plants.

Flow rate (L/s)	30 - 3000
Hm (mSS)	8-175
Discharge diameter (inches)	65-700

3.5 Multi Stage Centrifugal Pumps (KOT-KAT)

Covers medium and high pressure multi stage centrifugal pumps where axial thrust force is compensated by a balance disk.

Principal fields of utilization, besides those of NORM pumps, are fire extinguishing and washing installations, booster set installations and coal exploitations.

Flow rate (L/s)	4 – 200
Hm (mWC)	20 – 400
Discharge diameter (inches)	50 - 250

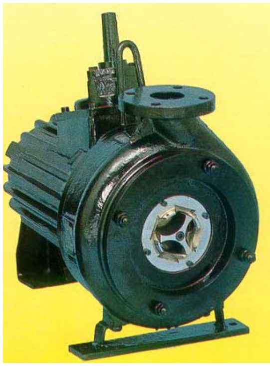
3.6 Submersible Sewerage Pumps – Water Coolant (DAS/DAC-S)

Designed to operate under water; with DAS type single DAÇ-S type double mechanical seals. Mounting alternatives are duck foot bend type, horizontal/vertical sump type (DAS only) or hose connection type. Standard motors are 380V 50Hz.

	DAS	DAÇ-S
Flow rate (L/s)	2 – 80	5 – 700
Hm (mWC)	4 – 40	5 – 60
Motor power (kW)	1.5 – 11	11 – 355
Discharge diameter (mm)	Ø50 – Ø150	Ø80 – Ø400
Solid substance (mm)	Ø35 – Ø120	Ø40 – Ø200

3.7 Submersible Pump with Shredder Blades (PARPO)

Designed in a manner to operate under water, particularly for the discharge of household waste water without obstructing small diameter pipes. Blades in front of the wheel smashes into pieces solid materials in the water like nylon, plastic etc. and fibrous substances and then pumps them. Built with single mechanical seal and standard motors 380V 50Hz.



Flow rate (L/s)	1 – 13
Hm (mSS)	3 – 43
Motor power (kW)	1.5 – 4
Discharge diameter (mm)	Ø40 – Ø50

3.8 Submersible Sewage Pumps – Oil Cooling (DAC-Y)

As they are internally oil cooled, they are designed in a manner to operate continually, dry or wet, without any damage to the motor and the mechanical seals in the oil tank. Mounting choices are slide type, hose connection and vertical dry system. 380V and 50Hz are standard but different voltage and frequency is also produced on demand. In addition to this group, DAC-SC Cooling Jacket Submersible Pumps are also produced.

	DAÇ – Y
Flow rate (L/s)	5 – 700
Hm (mWC)	5 – 60

Motor power (kW)	11 – 355
Discharge diameter (mm)	Ø80 – Ø400
Solid substance (mm)	Ø40 – Ø200

3.9 Sewage and Waste Water Pumps (CAP)

Turbosan CAP Series pumps are used for pumping fluids consisting of a mixture of solid- liquid substances like suspensions used in various processes, muddy waters, sewage fluids etc. Torque flow impellers, double blade and single blade impellers are used in CAP pumps. Primary fields of utilization are pumping industrial sewerage, purification installations, pumping muddy and solid substance containing liquids, pumping liquid with fibrous solid contents and viscous liquids, draining and discharge works, heat reclaiming systems.

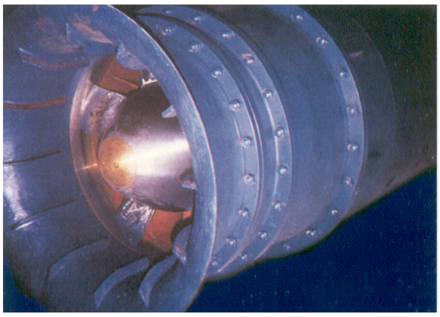
Flow rate (L/s)	2 – 700
Hm (mWC)	5 – 60
Motor power (kW)	1,5 - 400
Discharge diameter (mm)	Ø50 – Ø400
Solid substance (mm)	Ø35 – Ø200

3.10 Volute Process Type Pumps (PRO)

Covers low and medium pressure single stage volute chemical process pumps with Teflon or mechanical seal. Primary usage field is the pumping of low viscosity industrial liquids (like gasoline, oil, acids, bases etc.) in several industries.

Flow rate (L/s)	4 – 300
Hm (mWC)	5 – 70
Discharge diameter (mm)	40 – 250

3.11 Mixed Flow and Axial Flow Pumps (EKS-K/E)



Tube type submersible versions (DAÇ-K/E) of mixed/axial flow high flow rate and low pressure column pumps (EKS-K/E) are also manufactured.
Installations of potable water, irrigation water, cooling water and draining and fish farms are among usage fields.

Flow rate (L/s)	140 – 5.000
Hm (mWC)	1- 50
Column-tube diameter (mm)	300-1200

3.12 Tube Type Submersible Pumps (DAC-E/K)

The DAC-E/K series pumps are tube type mixed flow, high flow rate, low pressure submersible pumps operating under water.
Pumping drainage waters, irrigation, cooling water, circulation water in fish culture are the main utilization areas.

Flow rate (L/s)	140 – 3.000
Hm (mWC)	1 - 50
Tube diameter (mm)	300 - 1200

3.13 Screw (Archimedean) Sewage Pumps

Gearbox driven screw type pumps designed on Archimedes principle, where lower bearings are automatically lubricated by a special grease pump. Used in different flow rates for high rate sewerage without complex control panels. Helicoids diameter up to 2300mm, outer diameter finish machined by lathe.

Flow rate (L/s)	50 – 1600
Hm (mWC)	3 - 10
Tube diameter (mm)	400 - 2300

Figure 3.10 Complete Pumping Station built by Turbosan

3.14 Submersible Pumps :

Submersible pumps are designed for underwater operation. Pump and motor operate as completely immersed in tanks or reservoirs under the water.

Submersible pumps are mainly divided in 2 groups :

a) Deep well submersible pumps: 4", 6", 8", 10" … multi stage, narrow and long pumps used for pumping clean water.
b) Submersible sewage pumps: used for pumping clean and waste waters containing large size solid particles. Their capacity and power range are very wide.

Turbosan submersible waste water and sewage pumps are manufactured under 5 groups:

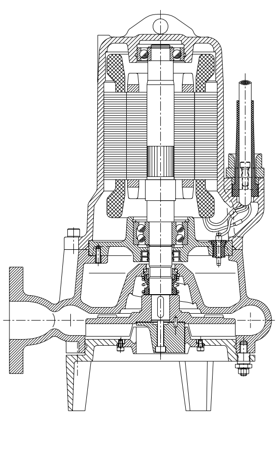
1- DAS: water cooled submersible pumps
2- DAC-SC: cooling jacket submersible pumps
3- DAC-Y: oil cooled submersible pumps
4- PARPO: water cooled submersible pumps with smashing blades
5- DAC-E/K: water cooled axial and mixed flow submersible pumps

1) The cooling system of DAS pumps:
   DAS type pumps are cooled externally by the liquid they are immersed in or they pump. For adequate cooling of the pump, it should be fully immersed in water. No dry running.

2) The cooling system of DAC-SC pumps:
   A special cooling jacket (casing) is externally added to the motor body. The circulation of the coolant fluid within this jacket is ensured via the special design of the impeller. The fluid moving in the jacket cools the motor by dispelling the heat, irrespective of mounting type. Cloggings are prevented by the special design of cooling channels.

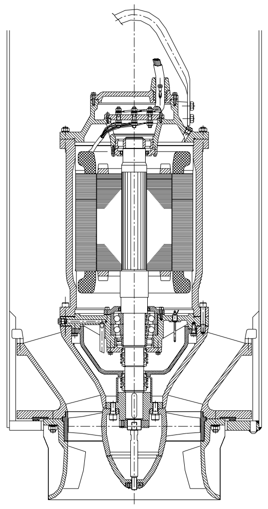
3) The cooling system of DAC-Y pumps:
   The motor of the submersible pump is cooled by electrically non conductor oil, good conductor of heat, which travels between the coils of the motor and absorbs the heat therein. The cooling system consists of a small size heat exchanger pump within the submersible pump itself.
   Our DAC-SC and DAC-Y type pumps are designed for underwater as well as dry operation.

4) PARPO type shredder sewage pumps are used for pumping waste waters and liquids containing solid sewage and short fibrous particles in populated areas and industrial installations. It prevents pipe clogging because of its shredding ability.



PARPO Shredder System: a system of shredder blades made of hardened and quality steel is located in front of the pump impeller. The jaws of the fixed blade retain solid substances. The rotating blade attached to the center of the impeller revolves together with the impeller and its cutting teeth slice the substances held by the fixed blade. Materials with long fibers and hard substances like bones may not be cut.

5. DAC-E/K : axial and mixed flow submersible pumps are designed in a manner to operate in a tube and under water. A steel pipe or a concrete block may be used as tube. Since DAC-E/K pumps occupy small space and does not require large size building, the initial investment cost is moderate. Tube type submersible pump mounting and dismounting is easy. Pumps with higher flow rate and small pump head are used in pumping draining waters, larger flow rates for agricultural purposes and in waste water purification installations



Submersible Pump Parts Characteristics :

Motor : the electric motors of Turbosan DAC series submersible pumps run with 3-phase 380V AC. The insulation is F class and the protection is IP68 class. H class insulated and different frequency (60Hz) or different voltage motors may be produced on special order.

Shaft sealing : Shaft sealing between the pump and the motor is ensured via double mechanical seals running in an oil chamber (single mechanical seal up to 11 kW).

Bearings: the motor rotor is centered by heavy duty type roller bearings at the top and bottom. These bearings are chosen in a manner to meet the radial and axial loads originating from the motor and the pump. In DAC-Y type, all roller bearings run in the cooling oil. In DAS and DAC-SC type submersible pumps roller bearings are lubricated by grease.

Motor Overheat Protection System: submersible motors are protected against overheating by thermistors. Thermistor ends are carried to surface by a cable and are marked (T) on the connector. The thermistor system is attached to the special Turbosan STR Protection Relay.

Water Leakage Protection System: an electrode (sensor) system is available for emergency warning in case of water leakage from the mechanical seal by time or in case of water leaked in motor body or oil chamber from exchanger, washers, oil plugs etc.

Cables : In submersible pumps H07RN-F type, rubber coated, flexible thin cable conduit, multi-wire cables resisting to liquid sewage are used. Generally 10 meter cable is provided as attached to the pump. **Do not carry the pump by holding the cable**.

Volute : Normally with large cross section and concentric outlet. Designed in a manner not to catch substances of any form and size passing through the impeller and obstruct the system.

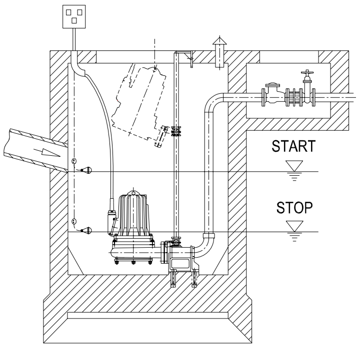
Pump Impeller: Different impeller models are used in submersible pumps.

- SINGLE BLADE Double Angled Non-Clog Impellers : impeller type capable of letting large size solid substances, with high capacity and compliant power characteristics (not forcing the motor at low altitudes).
- DOUBLE BLADE Impellers : used generally in large size pumps. Balanced and vibration free operation on account of rotational symmetry. General efficiency high and with compliant power characteristics. Wide flow channels in the impeller enable the pumping of solid substances.

- VORTEX Impellers : there are no closed channels in this type of impeller. The free torque flow formed by the revolution of the liquid mass by the impeller makes the pumping. Because of this feature it may easily pump large size solids and long fibrous materials. Reciprocally its general efficiency is considerably low and may only used in low powers.
- The dynamic balance of pump impellers is made in accordance with standards.

Submersible Pumps Mounting Types :

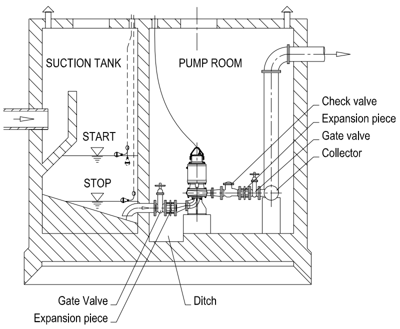
1. **Wet Mounting on Fixed Installations (Duck Foot Bend Automatic Coupling)**



This is a practical and economical mounting type for fixed installations. Consists of a duck foot bend attached with a guide rail (two parallel pipes, galvanized) and a coupling flange connected to the pump. The coupling equipment and the discharge pipe system should be mounted during the construction, when the suction reservoir is dry. Automatic coupling may be applied to all types.

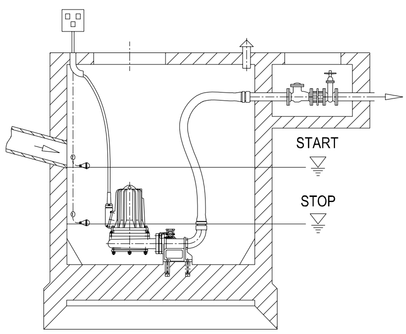
Operation : the coupling flange is fixed to the guide rail. The pump is lowered along the guide rail by means of a chain. The own weight of the pump fits it to its location when it touches the duck foot bend. The pump is taken up by pulling the chain. No stud attachment is necessary for mounting and dismounting the pump.

2. **Dry Mounting**



Used for DAC-SC cooling jacket and DAC-Y oil cooled pumps. As these types of pumps are capable of self cooling they may operate continuously out of water. Thus the benefits of dry operation such as installation maintenance add to the advantage of wet operation such as low space occupation and operating under difficult conditions. In dry mounting the suction reservoir is separated from the pump chamber by a wall. The floor of the pump room is dry and repairs and maintenance may be carried there during the operation. Since pumps are placed on solid concrete foundation vibrations are minimized. Carrying elbows are located under the pumps. One valve and valve disassembly part must be available in the inlet of the pump. A small draining pump must be utilized in the pump room in order to evacuate leaking waters.

3. **Vertical Hose Connection Mounting**



Used in suction reservoirs and pits where the floor is even. The pump must freely sit on the operation ground. The pump is lowered to the reservoir and lifted by a chain. Used for small size pumps.

* A valve, non-return valve disassembly part or an elastic spacer must be available in the pumping lines of all mounting forms. .

PART 4

OPERATIONAL PERFORMANCES OF PUMPS

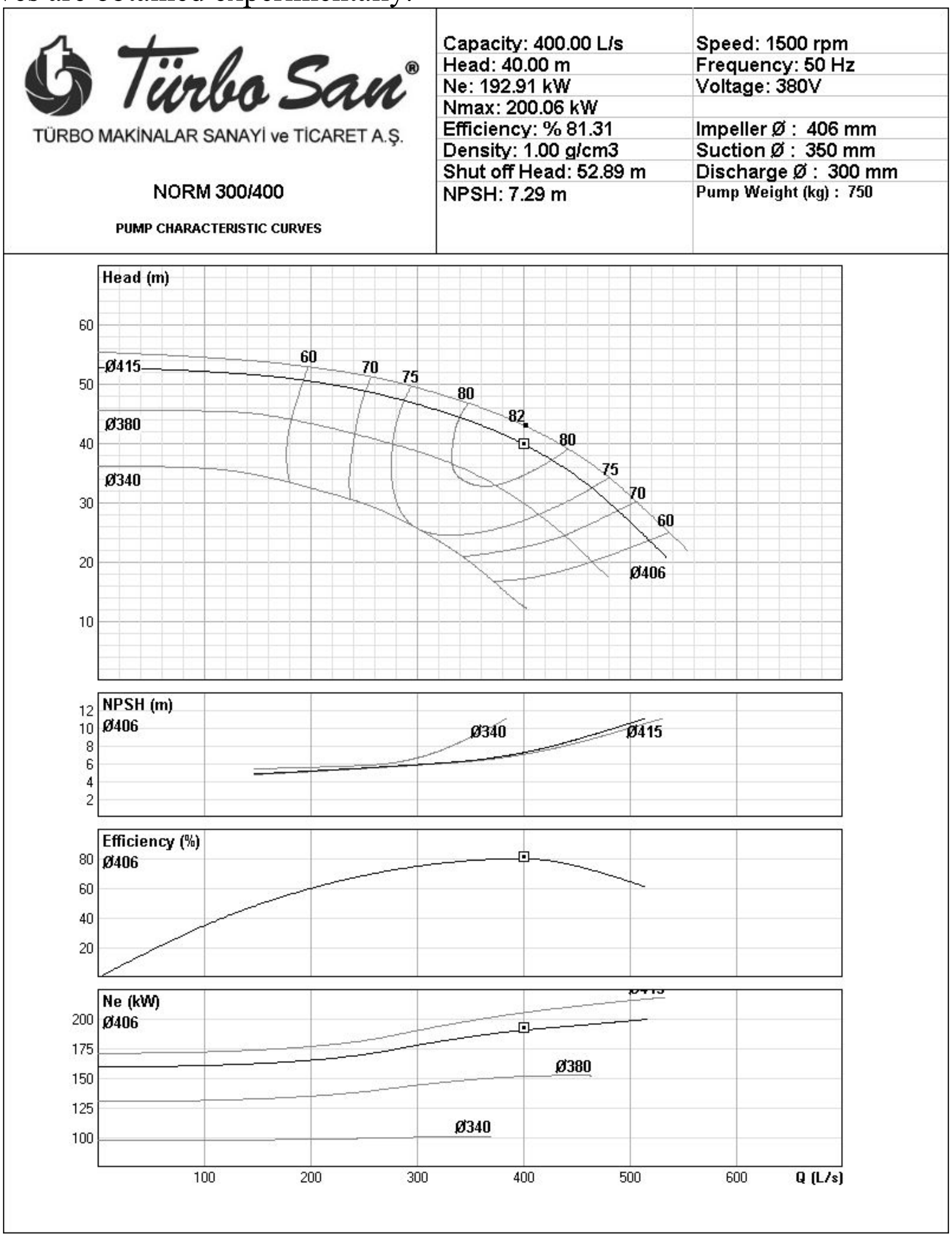
4.1 Characteristic Curves

The characteristic curves of pumps show the following variation for given RPM

Those are a family of curves demonstrating the changes of the characteristics of a pump with any structural character in any number of revolutions.

1. The variation of the pump head by the flow rate : $H_m = f(Q)$
2. The variation of the pump efficiency by the flow rate : $\eta_g = f(Q)$
3. The variation of the pump shaft power by the flow rate : $N_e=f(Q)$
4. The variation of the pump net positive head –NPSH- by the flow rate : NPSH=f(Q)

These curves are obtained experimentally.



Picture 4.1 Characteristic curves of a Pump

4.1.1 Optimum Operation Point of a Pump

The point where the pump operates with the highest efficiency is called the optimum operation point (or design point) of the pump. The optimum operation point of the pump whose characteristics are given in *Picture 4.1* is H_m= 42 m, Q= 400 lt/s. At that point the pump runs with 82% efficiency. Values indicated on the label of the pump are the pump total head and the flow rate at optimum operation point. From the standpoint of power economy the pump must run at or close to the optimum operation point.

4.2 Adjustment of Pump Characteristics

Pump characteristic curves are given for constant RPM as explained above. These curves do not change as long as the rpm of the pump or the diameter of the impeller are not modified.

The effect of the rpm and the impeller diameter on the pump characteristic may be obtained by using similarity rules. The pump head varies directly with the square of the diameter rate. Similarly, it also varies directly with the square of the RPM . The flow rate of the pump varies directly with the RPM ratio and the cube of the impeller diameter ratio.

$$\boxed{\frac{H_m}{H_{m'}}=\left(\frac{n}{n'}\right)^2 \qquad \frac{H_m}{H_{m'}}=\left(\frac{D}{D'}\right)^2} \qquad \boxed{\frac{Q}{Q'}=\frac{n}{n'} \qquad \frac{Q}{Q'}=\left(\frac{D}{D'}\right)^3}$$

The hydrodynamic characteristics of the pumping system define the operation point of the pump.

4.3 H-Q characteristic modified by an orifice at the discharge side

The performance characteristics of a pump may be modified by an orifice placed at the discharge flange of the pump. In such application the total head of the pump is reduced as much as the local loss caused by the orifice. Therefore the modification of the total head variation of the pump with orifice by the flow rate is steeper than that of the pump without orifice. Moreover the efficiency of the pump with orifice is also reduced.

4.4 Startup moments, startup and run down times of centrifugal pumps

It may be important from the standpoint of the operation of the pumping station to know the startup moments, startup and shutdown times of centrifugal pumps.
The startup moment of a pump is calculated by the formula:

$$M_k=\frac{P}{\omega}$$

Where P, is the power of the pump at the time of startup and ω is its angular speed. The moment value starts from zero and when the pump arrives to its operation point it reaches the optimum RPM and moment value. The moment at the shaft of the machine driving the pump is greater than the moment required by the pump until the pump attains its operation point and the difference in between ensures the acceleration of the revolving masses.
The variation of the startup moment of a pump depends on the characteristics of the installation at the time of startup and may be classified under 4 headings:
a. Starting against a closed valve: The valve remains closed until the pump attains the nominal speed, then it is opened and the required operation point is reached. This case is shown in the following H-Q and M_k-n graphs. When the valve is closed (between O-A) the

hydraulic moment of the pump increases by the square of the rotation speed. The original startup moment where n=0 occurs because of static frictions and may have considerably high values. Static friction generally causes original startup moments of 5 to 10% of the nominal moment value. Between A and B the pump operates at fixed rpm and its moment increases or decreases in function of the form of the pump power curve. (Figure 4.2)

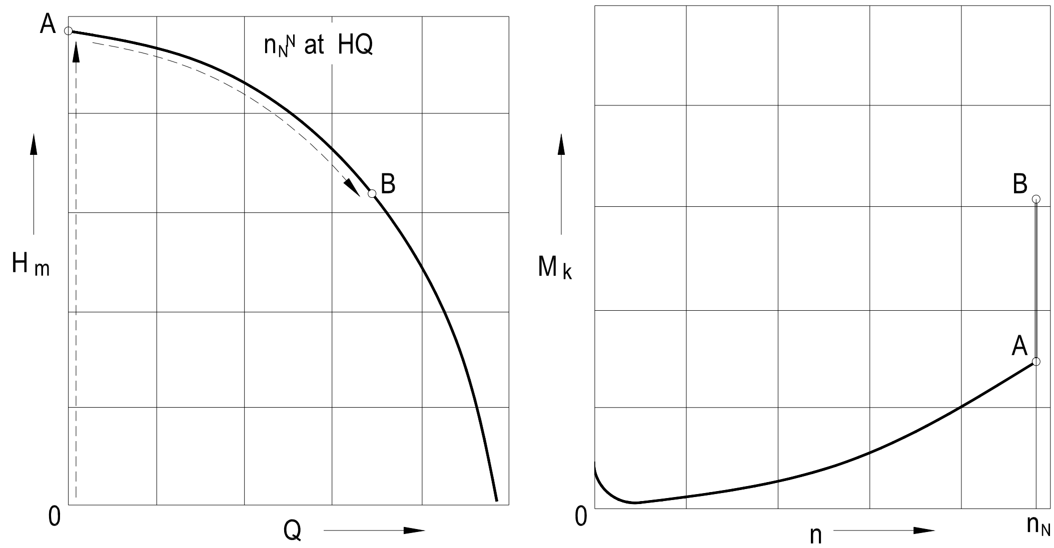


Figure 4.2 Closed Valve Case

b. Non-return Valve System: in this case there exists a static load at the delivery tank side of the non-return valve of the delivery pipe. The pump head is equal to the pressure at point A on the other side of the non-return valve. Until this point the pump operates against the closed valve. After this value, fluid flows towards the delivery tank and the required operation point (point B) is reached when the pump attains its nominal speed. The moment variation with RPM may be considered as linear between the points A and B. (Figure 4.3)

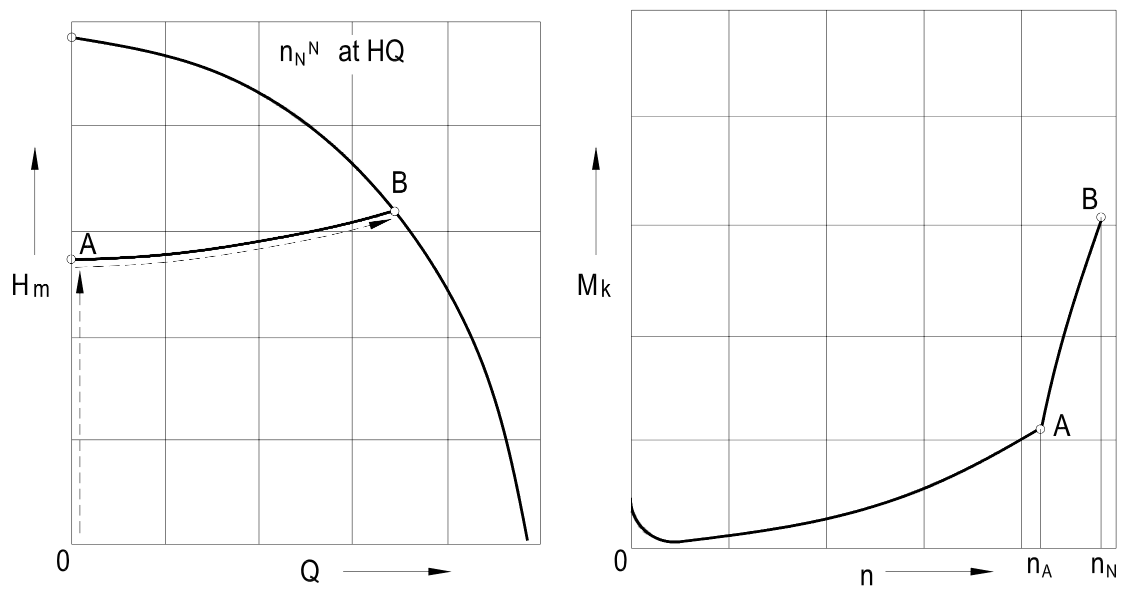


Figure 4.3 Non-return Valve System

c. Starting with an Open Valve Against a Dynamic System Head: H-Q and M_k-n variations occurring in case of startup with open valve with very short discharge pipe line are shown in the following. For considerably long discharge pipe the time required for accelerating the water is much longer than the startup time of the pump. In this case a similar phenomena that of Figure 4.4 occurs

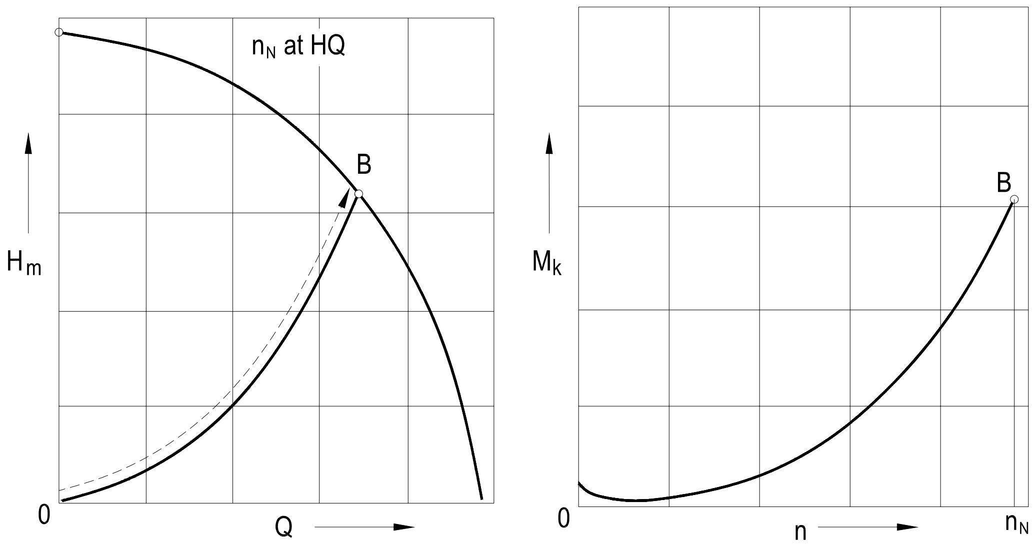


Figure 4.4 Open Valve Against Dynamic System Head System

d. Open Valve and an empty delivery pipeline: If the time required for filling the pipe line with water is longer than the startup time, at the beginning the pump runs on the O-A curve shown below. The moment variation in region A-B is closely related with the power curve of the pump. (Figure 4.5)

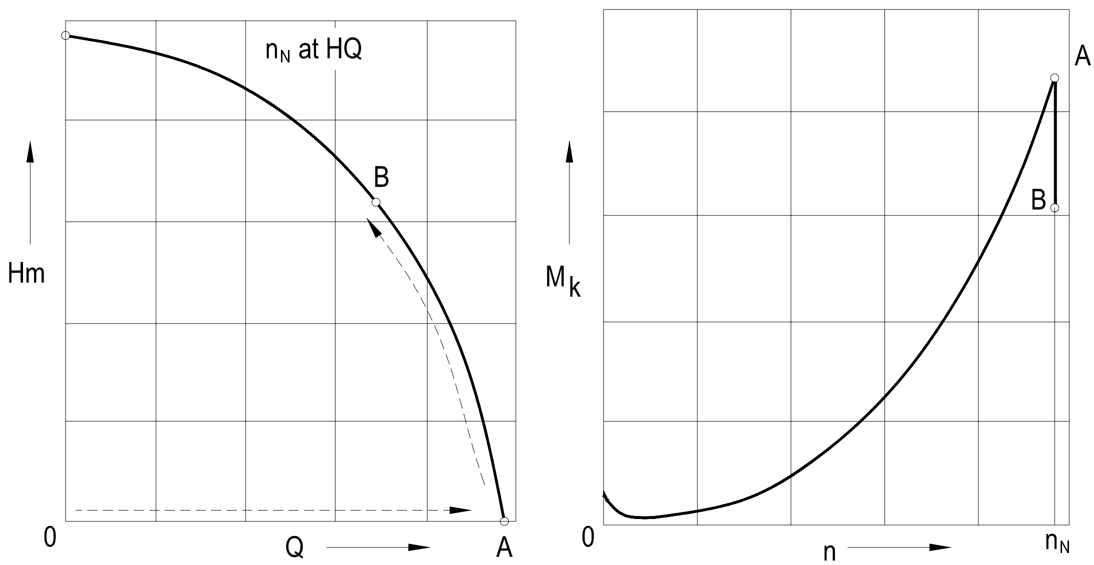


Figure 4.5 Open Valve and an Empty Delivery Pipeline

e. Startup Moment (Torque) : Centrifugal pumps are started up against a closed valve. The relationship between the moment at the startup and RPM is shown in Figure 4.6.

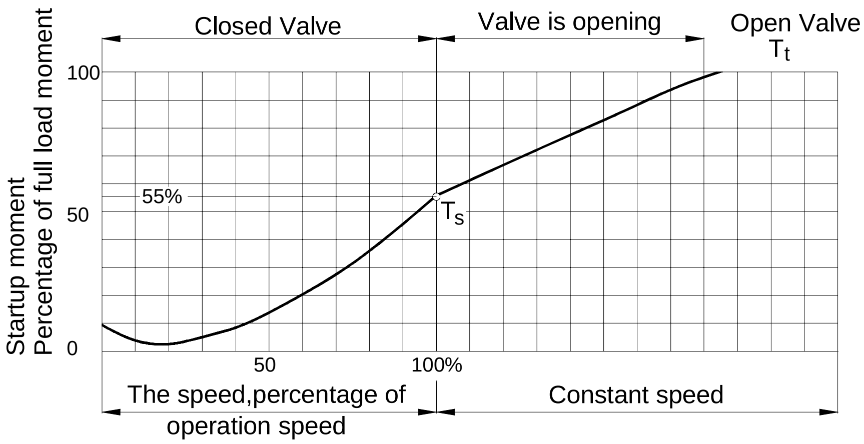


Figure 4.6 Centrifugal (Norm) Pumps Startup Torque

Axial pumps are started up against an open valve. Figure 4.7 compares moment characteristics of axial pumps in case of open and closed valve startups.

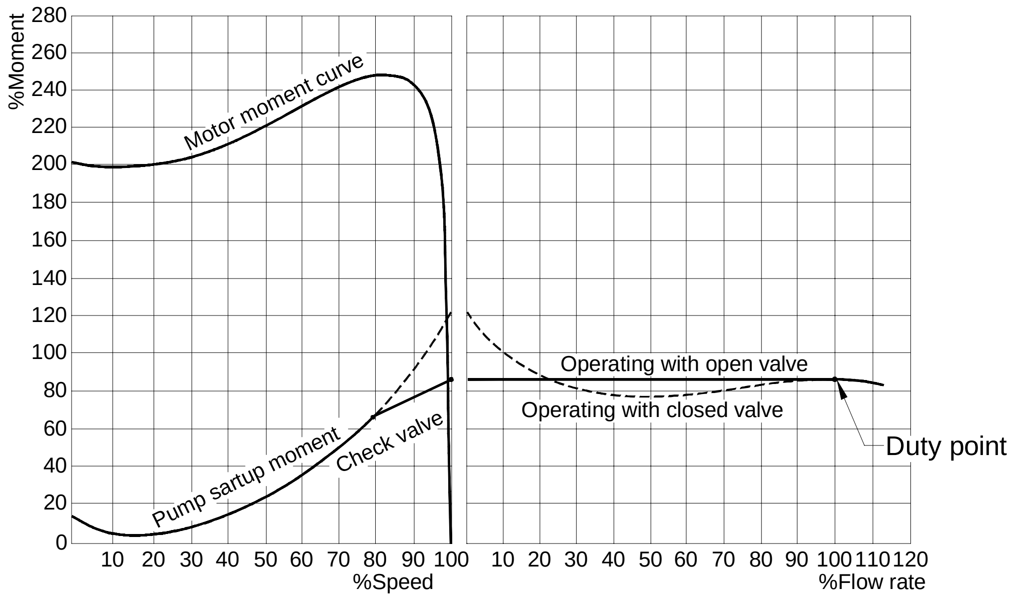


Figure 4.7 Axial (EKS) Pump and Motor, Operation and Startup Moment Characteristics

4.5 Determination of Pump Startup and Rundown Times

The difference between the startup moment of the pump M_k and the pump shaft moment M_m accelerates the rotating parts (mainly impeller):

$$M_m - M_k = J\,\alpha$$

In this expression J is the moment of inertia of the rotating parts and α is the angular acceleration of the pump shaft. The difference between the moments causes the variation of the rotational speed by the time:

$$\frac{d\omega}{dt} = \frac{M_m - M_k}{J}$$

The variations of M_m and M_k with respect to RPM are graphically described hereinafter. The above-mentioned differential equation may be graphically integrated:

$$\frac{\Delta n}{\Delta t} = \frac{30}{\pi} \frac{M_m - M_k}{J}$$

For this purpose the n axis is divided into finite pieces of Δn and the start-up time may be calculated by the following formula

$$t = \sum_{i=1}^{k} t_i = \frac{\pi\, J}{30} \sum_{i=1}^{k} \frac{\Delta n_i}{(M_m - M_k)} = \frac{\pi\, J}{30} \left(\frac{\Delta n_1}{M_m - M_k} + \frac{\Delta n_2}{M_m - M_k} + ... + \frac{\Delta n_k}{M_m - M_k} \right)$$

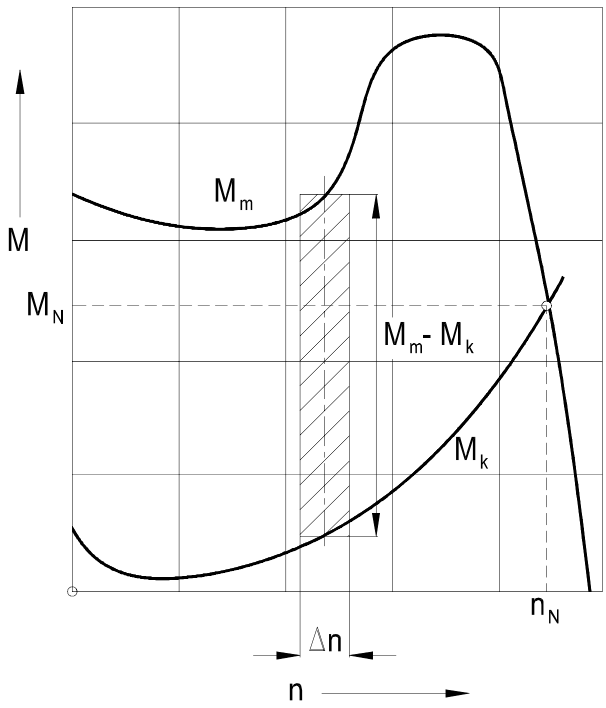


Figure 4.8 Determination of the Pump Start-up Time

4.6 Minimum Flow Rate

All the power of the pump driving unit may not be transferred to the fluid because of the losses which are explained previously. This energy translated into heat increases the temperature of the pump body and the pumped fluid.

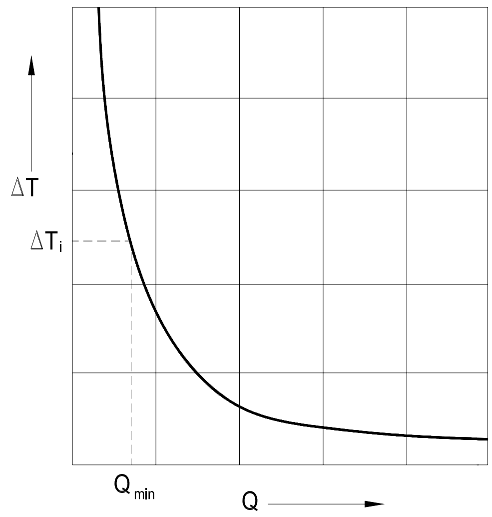


Figure 4.9 Minimum Flow Rate

Figure 4.9 shows the variation of the temperature increase in relation to the flow rate. In case the allowed temperature increase is ΔT_i , the minimum flow rate corresponding thereto is called Q_{min} and the pump should not be operated at lower flow rate than Qmin.

Whereas the vaporisation risk is low because of the high pressure at the outlet of the pump, the temperature increase does not create an important problem. However the situation is more critical in the low static pressure regions within the pump. Therefore some precautions may be necessary.

4.7 Optimum Operation Range of Pumps

The flow rate range where centrifugal pumps may continuously operate should be in between 70% and 120% of the design flow rate. (Figure 4.10)
Pumps operated beyond the region shown in Figure 4.10 may create undesired hydraulic problems, besides the high power consumption.

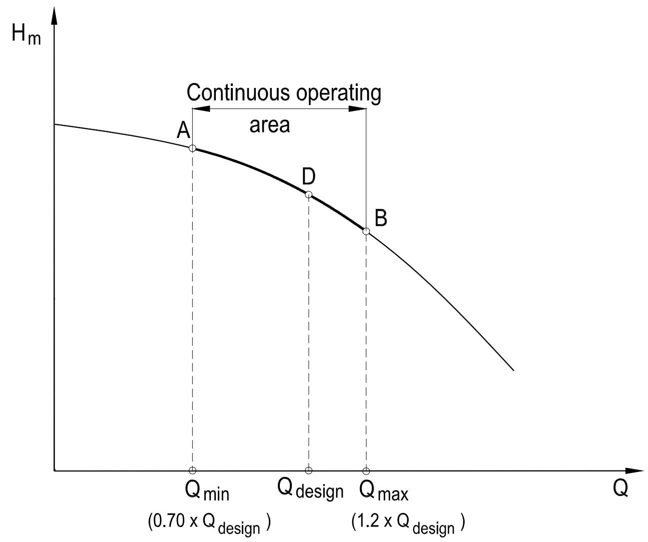


Figure 4.10 Recommended Operation Range

4.8 Problems Caused by the Off-Design Operation

The H_m value of the pumping system must be calculated correctly. This is possible by the correct calculation of the system characteristics, minor loss and pipe friction loss. As an example; in case the pump head is taken as 70 mWC, instead of 60 mWC adding 10m as safety factor the following cases may occur:

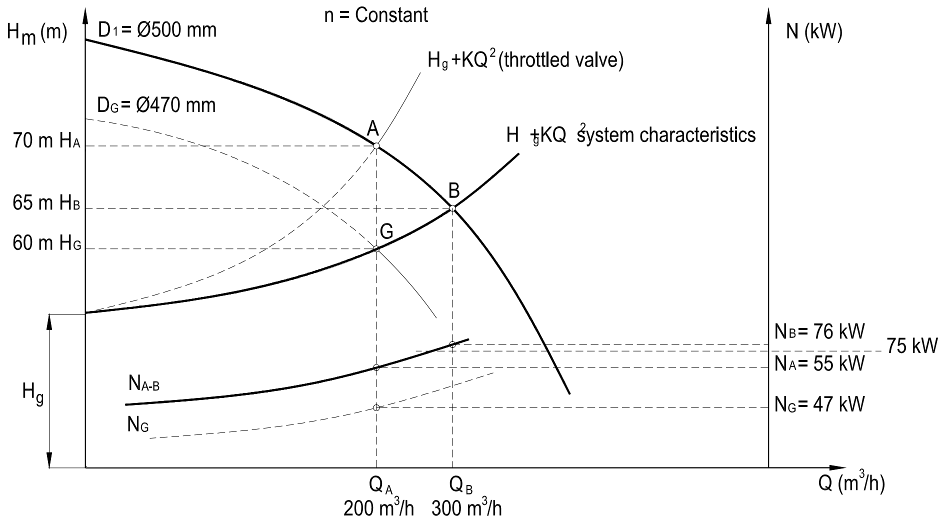


Figure 4.11 Off-Design Operation

1- A motor pump (pump+motor) with the following characteristics would be purchased: 200 m^3/h, 70 mSS, 75 kW.
2- The pump characteristic curve will meet the true system characteristic at point B. Values at that point would be read as; Q= 300 m^3/h, H_m= 65 mWC
3- The power that the pump will consume at the true point G would be N_B= 76 kW instead of N_G= 47 kW as the pump operates at point B. (We assume that the efficiency of the pump remains the same at points G, A and B, i.e. 70%). At point B the pump will consume 29 kW more power. The annual cost of this difference is = 29 x 0,2 $/Kwh x 365 days x 12 hours/day = 25404 $/year.
4- Moreover as the motor power is 75 kW, it will result in higher current (ampere) value. It will burn out shortly after if we would run it at point B.
5- Since the flow rate is high, if we wish to run it at the requested flow rate point Q_A (200 m^3/h) we have to turn down the valve and come to points Q_A and H_A.
6- At this point the power drawn by the pump would be N_A = 55 kW. Now the pump may be safely run. Nonetheless at this point the following values would be valid: Q=200 m^3/h , H_m=70 mSS. The pump draws 8 kW in excess. The annual cost thereof is = 8 x 0,2 $/Kwh x 365 days x 12 hours/day = 7008 $/year.
7- The solution is to modify the pump in a manner to run it in the correct operation point G: The impeller diameter of D_1 (Ø500) must be lathed out to D_G (Ø470), thereby reducing it to true operating values.

4.9 Axial Thrust and Balancing Methods

Axial thrust is originated from the pressure difference between the two sides of the impeller. It pushes the impeller and the attached parts (like the shaft) towards low pressure region. Axial thrust occurs due to the pressure difference between low pressure region impeller inlet and high pressure region at the back side of the head of the impeller. This thrust may attain very high values particularly in multi stage pumps and must be balanced.

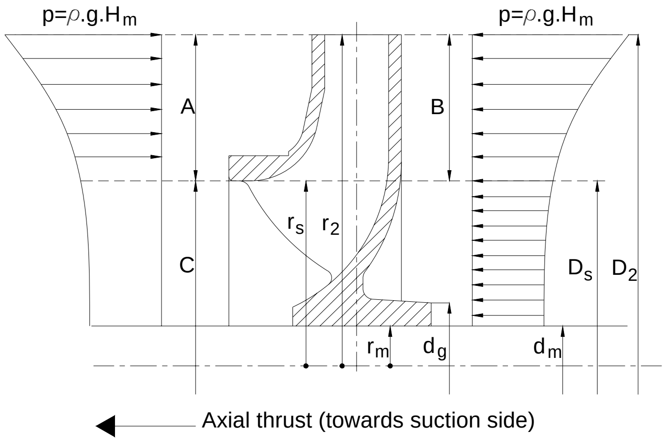


Figure 4.12 Distribution of Pressure on the Front and Rear Side of the Impeller

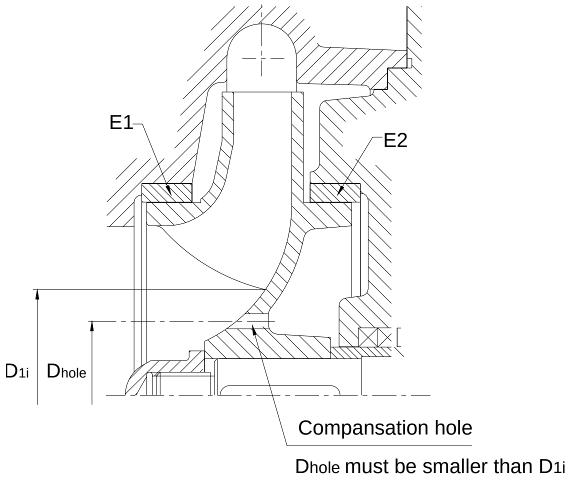
As shown in Figure 4.12, pressure forces created on the face A and the backside B of the impeller because of pressure distribution, compensate each other. However, because of the vacuum on the face of the impeller, the pressure on the C ring is not balanced and these forces push the impeller and the shaft towards the suction side. For simple approach, the value of the thrust may be calculated by the following formula:

$$P = \pi \times \left(\frac{D_s^{\,2} - d_m^{\,2}}{4} \right) \cdot p$$

Where the p is the pressure corresponding to the pump head, without taking into consideration the leakage losses and may be considered as $\mathbf{p = \rho.g.H_m}$ **.**

Following are some measures taken in pump designs in order to balance the axial force:

In single stage NORM type centrifugal pumps, the pump total head is not very high and the total axial thrust remains between 1000 – 2000 N. Forces at this level may be balanced by balancing holes to be opened on the back side of the impeller. Besides, single row rigid ball journal bearings are capable of carrying axial loads of such value. If the axial thrust is higher, then one of the ball bearings (generally the bearing at the rear of the shaft) must be assisted with a roller bearing or a ball bearing carrying axial loads.



E1, E2 : Wear Rings

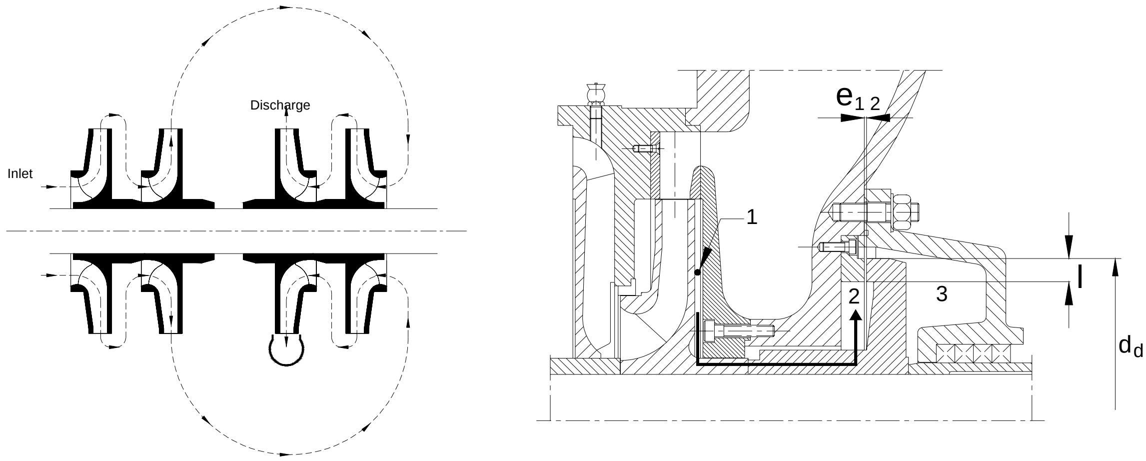
Figure 4.13 Axial thrust balanced by compensation holes

The number of holes to be opened on the impeller may be between 3 and 8 and should not weaken the hub. Although it varies with the the size of the pump, the diameter of the holes may be Ø5 to Ø10 mm. Impellers with balancing holes must be reinforced by wearing rings (Figure 4.13).

Double Suction (Entry) pumps : impellers are backing each other. Therefore the axial thrust created in each impeller balances the other and axial thrust does not occur. In other terms, there are no axial thrust and balancing issues in double suction (entry) pumps.

Wortex (Open) impellers : the axial load compensation is made by balancing blades. The number of blades is 4~16 and the blade height is 4~7 mm.

Multi stage (KAT) pumps : Axial thrust is very important for multistage pumps. Whereas pressures are high the axial thrust may attain very high values (expressed in tons). Different balancing methods must be considered according to the size of the axial thrust. If the force is not very high, it may be balanced via balance holes opened to the impeller and special bearings to support axial loads. As forces increase, it becomes necessary to proceed towards more complicated construction types as shown in Figure 4.14. It is also possible to ensure balance in Figure 4.14 by reversing the flow directions and the impellers. By arranging the same number of impellers shown in figure 4.14 it is possible to balance the thrust. Another way is the use of balance disc as shown in Figure 4.14. The balance disc works as follows:

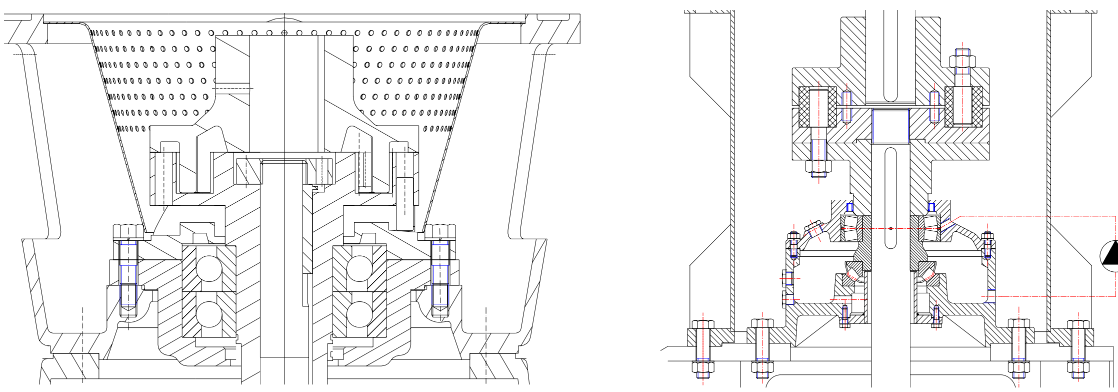


(a) *(b)*

Figure 4.14 Balancing the axial thrust in multi-stage pumps

The disk forms two disc compartments after the last stage with very well polished surface: one attached to the body, the other attached to the shaft. The pressurized water coming from the compartment (1) enters to the compartment (3) which is connected to compartment (2) and acts on the balance disc opposite to the thrust and compensates the force. In order to create the required pressure difference the compartment (3) is connected to the suction of the first stage of the pump. The balance disc surface and the opposite surface must be perfectly worked out and sized in a manner to compensate the axial thrust. The pump shaft may move back and forth axially together with impellers and balance disc within certain tolerances. Since the delivery and the suction parts of the pump are connected via compartments (1), (2) and (3) and intermediate connection pipe, a flow leakage happens. In order to keep the leakage at minimum level and ensure adequate balance, the size of the balance disc is practically as follows: largest disc diameter is $D_d \leq (0,7 \sim 0,8)\,.D_2$. ($D_2$: impeller outer diameter): This value gives good results particularly in pumps where the D_1/D_2 rate of the impeller is between 0,4 and 0,5. Disc ring width (I) may be the 8 to 10% of the disc diameter and the $e_{1\,2}$ width may be 0,4 to 0,8 times. In such balancing leakages may vary from 3 to 6%. However the simplicity of the construction and the self balancing of the thrust according to pressure differences make the balance discs widely used.

In vertical pumps EKS and deep well pumps, the compensation of the axial thrust is ensured via heavy duty type special bearings (thrust bearing) placed on the upper end of the shaft.



a) b)

Figure 4.15 EKS Pump bearing examples

PART 5

SYSTEM CHARACTERISTICS AND HEAD LOSS IN PUMP INSTALLATIONS

5.1 System Characteristics

The right hand side of the following equation, mentioned in Subsection 2.2, is the characteristics of the pumping system and consists of the difference of altitude between the liquid levels of suction and discharge reservoirs. The difference of pressure head acting on the free liquid surfaces of suction and discharge tanks and the total head loss occurring in pipe systems:

$$H_m = H_g + (\frac{p_B - p_A}{\rho g}) + \sum_{A-B} h$$

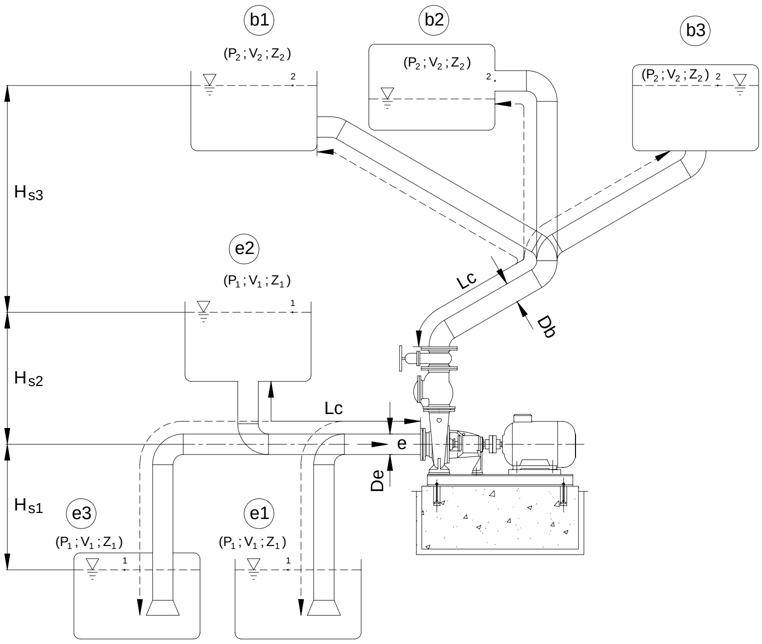


Figure 5.1 Different pumping system configuration

For the different pumping system given in Figure 5.1 these terms can be calculated in the following way for the system from e3 (suction tank) to b1 (delivery reservoir):

(1) The term H_g, geometric head is equal to $H_g = H_{s1} + H_{s2} + H_{s3}$.

(2) The term $(\frac{p_B - p_A}{\rho g})$ is calculated as, $(\frac{p_2 - p_1}{\rho g}) = \frac{0 - p_1}{\rho g} = -\frac{p_1}{\rho g}$ taking into consideration pressures acting on the free liquid surfaces of the tanks. It should noticed here that all pressures referred to herein are gage pressures.

(3) The term $\sum_{A-B} h$ contains the sum of the friction loss and minor loss in suction and delivery pipes.

H_m calculations for different pumping systems are given in Subsection 5.11.

5.2 System Characteristics, System Efficiency

The first two terms of the following equation defining the system characteristics are not function of the flow rate of the pumping system but the third term is a function of the flow rate. Therefore the system characteristic which is the sum of these three terms is a function of the flow rate.

$$H_m = H_g + \left(\frac{p_B - p_A}{\rho g}\right) + \sum_{A-B} h$$

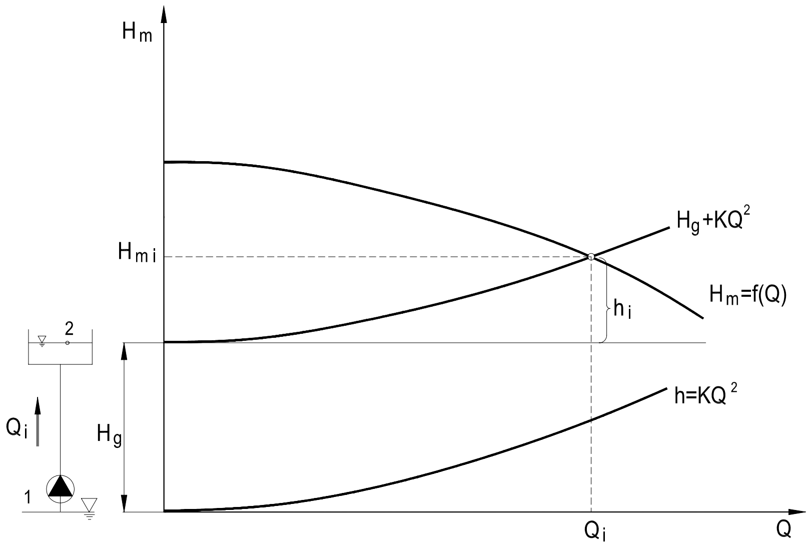


Figure 5.2 Pumping System and System Characteristics

In the pumping system indicated in Figure 5.2 the term H_g is the difference of altitude between the liquid levels of suction and discharge reservoirs.

Since tanks are open to atmosphere, the atmospheric pressure acts on the surface of the liquid. Therefore p_B and p_A pressure values are zero ($p_A = p_B = 0$)

The variation of the term $\sum_{A-B} h$ indicating the total loss in function of the flow rate is a parabola, as explained later on, and can be shown as $h = KQ^2$.

For this example, the system characteristics is equal to the sum of the loss parabola and the altitude level difference between the free liquid surfaces in tanks. In graphical terms, this addition op is made by moving up the loss parabola $h = KQ^2$ by the distance H_g.

If the tanks are not open to the atmosphere, the value of the difference of static pressuse heads between suchtion and discharge tanks is added to H_g. As explained in Subsection 2.2, the value obtained is the static head (H_{st}). In this case the system characteristic is obtained by moving the loss parabola up to the calculated static head.

Since the purpose of the pumping system is to deliver the liquid from the suction tank to the tank**,** the efficiency of the pumping system may be defined as the ratio of the useful pump power output $\left(H_m - \sum_{A-B} h\right)$ to the power transmitted to the liquid in the pump:

$$\eta_{system} = \frac{H_m - \sum_{A-B} h}{H_m}$$

5.3 Head Loss: Friction Loss, Minor Loss

System loss expressed by the term $\sum_{A-B} h$ are the sum of friction loss created in suction and delivery pipes and minor loss:

$$\boxed{\sum_{A-B} h_{cfrictions} + \sum_{A-B} h_{\min or} = \sum_{A-B} h = h}$$

It is the amount of the energy lost per unit weight of the pumped liquid.

5.3.1 Friction Loss

Friction loss occurs in long pipes where the flow direction or velocity does not change.

Friction loss in pipes may be expressed by the following relation:

$$\boxed{h_{friction} = \lambda \frac{L}{D} \cdot \frac{V^2}{2g}}$$

In this expression;

(1) the term *λ* is the friction coefficient and is the function of the Reynolds number and the relative surface roughness of the pipe and may be found from the Moody diagram given in the Annex Table 1. However first the Reynolds number (Re) and the relative surface roughness of the pipe (ε) must be calculated. The Reynolds number is calculated by the following formula:

$$\mathrm{Re} = \frac{VD}{\nu}$$

In this expression the term V, is the velocity of the fluid in the pipe (m/s); D, is the diameter of the pipe (m) and *ν*, is the kinematic viscosity (m^2/s). The kinematic viscosity of water at 15^0C is $11{,}416.10^{-7}$ m²/s. The kinematic viscosity of other fluids is given in Annex Table 3.

The relative surface roughness of the pipe is calculated by the expression.

$$\varepsilon = \frac{k}{D}$$

In this expression the term k is the roughness of the pipe material (m); D is the diameter of the pipe (m). The roughness of pipes made from different materials is given in Annex Table 4.
(2) the term L is the pipe length (m),

(3) the term D is the pipe diameter (m),

(4) the term V is the velocity of the fluid in the pipe (m/s),

(5) the term g is the gravitational acceleration (m/s^2)**.**

(6) the term A is the cross-sectional area (m^2)**.**

(7) the term Q is the flow rate (m^3/sn).

If the following relations are substituted in side the friction loss equation

$$V = \frac{Q}{A} = \frac{4Q}{\pi D^2}$$

and

$$\frac{V^2}{2g} = \frac{1}{2g} \cdot \frac{16 \cdot Q^2}{\pi^2 D^4} = 0{,}0826 \cdot \frac{Q^2}{D^4}$$

friction loss equation

$$\boxed{h_{cfriction} = 0{,}0826 \cdot \lambda \cdot \frac{L \cdot Q^2}{D^5}}$$

In this equation L and D are expressed in meters and Q is ^{3}/sec.

Example (1) : plotting of the system characteristics

Suction pipe diameter = Ø150 mm
Suction pipe length = 10 m
For steel pipe λ = 0,023

Discharge pipe diameter = Ø125 mm
Discharge pipe diameter = 160 m

$$h_{friction} = 0{,}0826 \cdot \lambda \cdot \frac{L \cdot Q^2}{D^5} \qquad (Q = m^3/sn,\ L = m,\ D = m)$$

$$h_{friction(suction)} = 0{,}0826 \cdot 0{,}023 \cdot \frac{10 \cdot Q^2}{(0{,}15)^5} = 251 \cdot Q^2$$

$$h_{friction(discharge)} = 0{,}0826 \cdot 0{,}023 \cdot \frac{160 \cdot Q^2}{(0{,}125)^5} = 9960 \cdot Q^2$$

$$h_{friction} = (251 + 9960) \cdot Q^2$$

$$\boxed{h_{friction} = (10211) \cdot Q^2}$$

If in this formula the term Q is expressed in m³/sec than $h_{friction\ is}$ obtained in m.

Calculated $h_{friction}$ values are shown in the following table for different Q values :

Q(m³/sec)	0	0,0055	0,014	0,022	0,034	0,044	0,056
$H_{frictions}$	0	0,4	1,97	5	11,4	20,2	31,5
Q(m³/h)	0	20	50	80	120	160	200

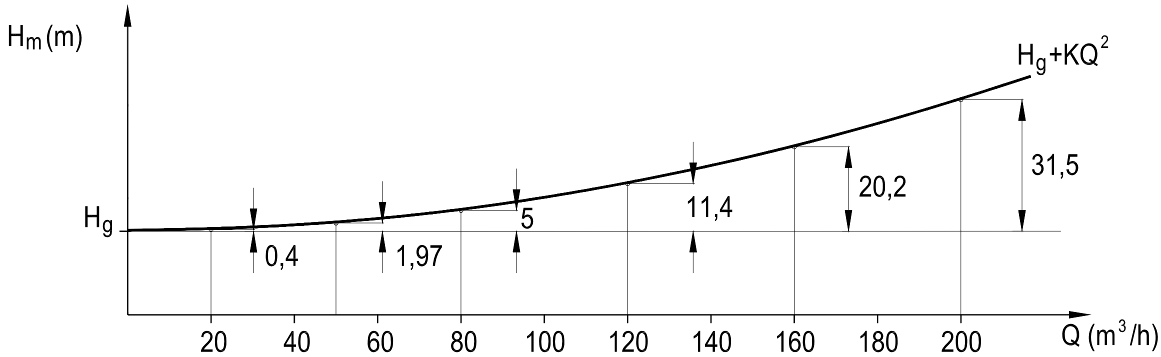


Figure 5.3

The friction loss in pipes may be calculated by using figure 5.4. The value of the flow rate is found on the horizontal axis and a vertical line passing through is drawn. The intersection point of this line with the related pipe diameter is found. Vertical coordinate of this point gives the friction loss for the 100 m lenghts of the pipe (h_{100}). Therefore the continuous head loss for the total pipe length L will be calculated by the following formula:

$$\boxed{h_{friction} = K \cdot h_{100} \cdot \frac{L}{100}}$$

For pre-used steel pipes it is advisable to take *K= 1,25*.

Example (2): Let's calculate the friction loss occuring inside the pipe of 150 mm diameter and 400 m length for flow rate equal to 175 m^3/h. Table 2 would give the value of $h_{100} = 5m$ and the friction loss occurring in the pipe is calculated as follows:

$$H_{friction} = 1{,}25 \cdot 5 \cdot \frac{400}{100} = 25\ m$$

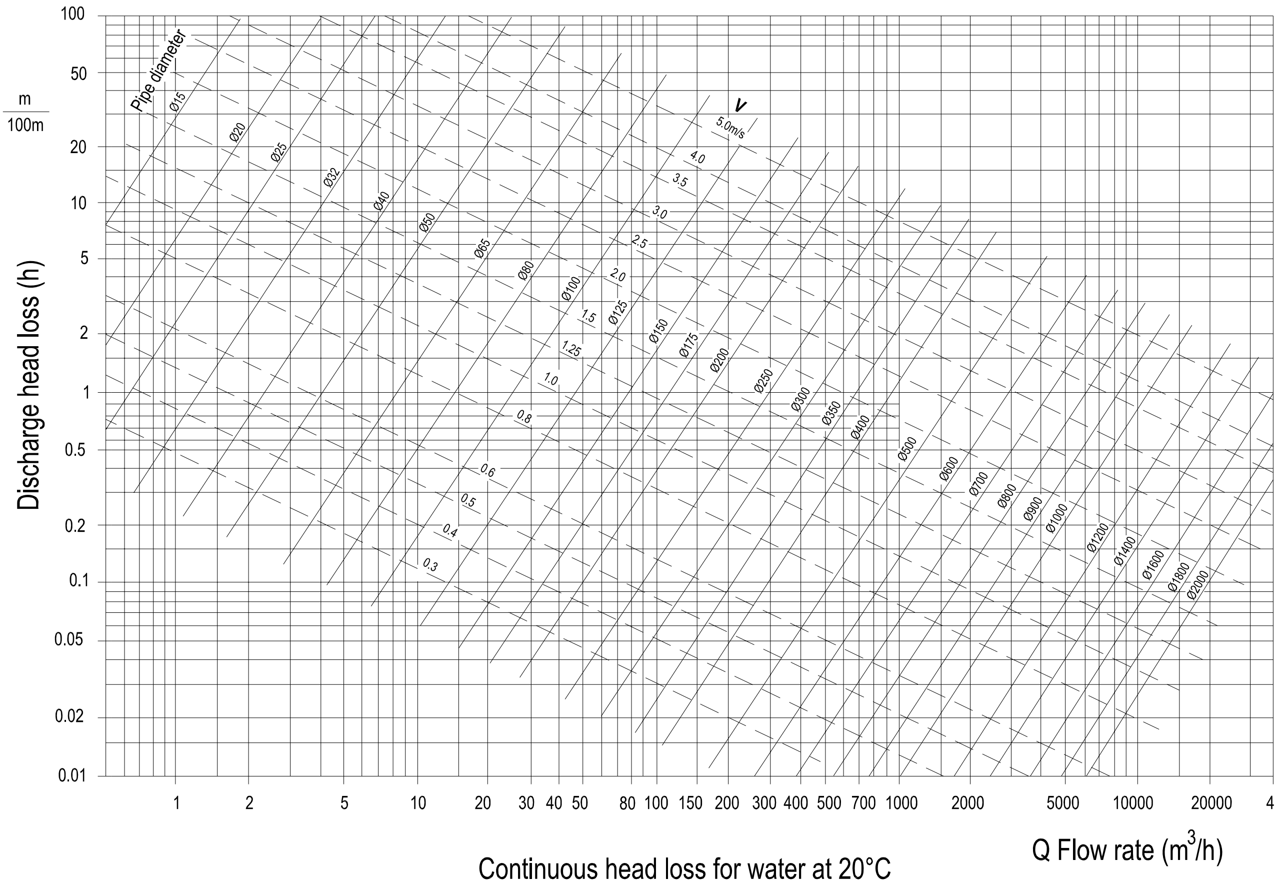


Figure 5.4

5.3.2 Minor Loss

Minor loss is caused by system elements where the flow changes its direction or velocity like reservoir inlets, reservoir outlets, elbows, check valves, filters, valves etc. and are expressed by

$$h_{\min or} = K \cdot \frac{V^2}{2g}$$

If we substitute flow rate instead of velocity, we obtain

$$\boxed{h_{\min or} = 0{,}0826 \cdot K \frac{Q^2}{D^4}} \quad (Q = m^3/h,\ D = m)$$

We see that minor loss increases with the second power of the flow rate. The friction loss

$$h_{friction} = 0{,}0826 \cdot \lambda \cdot \frac{L \cdot Q^2}{D^5}$$

Thus, the fact that total loss also increases wit the second power of the flowrate explains why the change of the loss curve (h=KQ²) indicated in Figure 5.1 has a parabolic shape.

All the minor loss coefficient K, of local elements must be taken from Annex 5 and minor loss must be calculated individually. The summation of these values would give the minor loss curve of the system.

5.4 Determining the Pipe Diameter

It is advisable that in pumping systems the velocity of the fluid not to exceed 4 m/sec and not to be less than 0,8 m/sec. Generally the pumping system is designed as if the velocity of the fluid in the discharge pipe would be 3,5 m/sec and the suction pipe speed 1 m/sec. The following table gives the recommended velocity of the fluid for different pumping systems. The flow rate is calculated by using these values and the following formula:

Fluid velocity in the pipe V (m/sec)		
Water purification installation	Watering and Draining	Industrial Water
0,8 ~ 1,5	1,5 ~ 3,0	1,5 ~ 2,5

Q, flow rate = (m^3/h); A, pipe cross section area = (m^2); V, fluid velocity= (m/sec)

$$\boxed{Q = A \cdot V}$$

If the flow rate is given then the diameter of the pipe is calculated by the following formula:

$$D = \sqrt{\frac{4 \cdot Q}{\pi \cdot V}}$$

Determining the pipe diameter in the pumping system is an optimization problem. Choosing small diameter pipes would decrease the initial investment cost. Looking into friction loss expression we notice that the head loss is inversely proportional to the pipe diameter. Therefore using of small diameter pipes would increase the operational cost. Consequently it is necessary to optimize these two criteria. This optimization should be made in line with the preferences of the investor.

5.4.1 Formulas used in calculating H_m

Pump Total Head Formula	Units	Pump Total Head Unit
$H_m = H_g + \frac{P_A - P_B}{\rho . g} + \sum h$	H_g: m P_A: Pascal P_B: Pascal ρ: kg/m^3 (for water 1000) g: m/s^2 h: m	H_m: m

System Loss Formula	Units	System Head Unit
$h = \sum h_s + \sum h_y$	h_s: m h_y: m	h: m
$h = \sum \lambda \frac{L}{D} \cdot \frac{V^2}{2g} + \sum K \cdot \frac{V^2}{2g}$	L: m D: m V^2: m/s	h: m

	g: m/s^2 K: (-)	

Pipe Friction Loss Formula	**Units**	**Pipe Continuous Losses Unit**
$\sum h_s = \lambda \frac{L}{D} \cdot \frac{V^2}{2g}$	L: m D: m V^2: m/s g: m/s^2	h_s: m
$\sum h_s = 6.4 \cdot 10^6 \cdot \lambda \cdot \frac{L}{D^5} \cdot Q^2$	L: m D: mm Q: m^3/h	h_y: m

Local Head Loss Formula	**Units**	***Local Head Loss Unit***
$h_y = K \cdot \frac{V^2}{2g}$	V^2: m/s g: m/s^2 K: (-)	h_y: m
$h_y = 6400 \cdot K \cdot \frac{Q^2}{D^4}$	D: mm Q: m^3/h K: (-)	h_y: m

5.5 Pump and Installation Common Operation Point

In case a pump hose characteristics are defined in the installation described in Figure 5.5, the point of intersection of the system characteristics curve $(H_g + KQ^2)$ and the pump characteristics curve H_m=f(Q) will be the pump operation point. In Figure 5.5 the operation point of the pump is shown by the index (i). The flow rate crossing the pump is Q_i and the pump head is H_{mi}. By deducting H_g from H_{mi} value we get the energy spent for the pipe system losses per the unit weight of the fluid crossing the pipe.

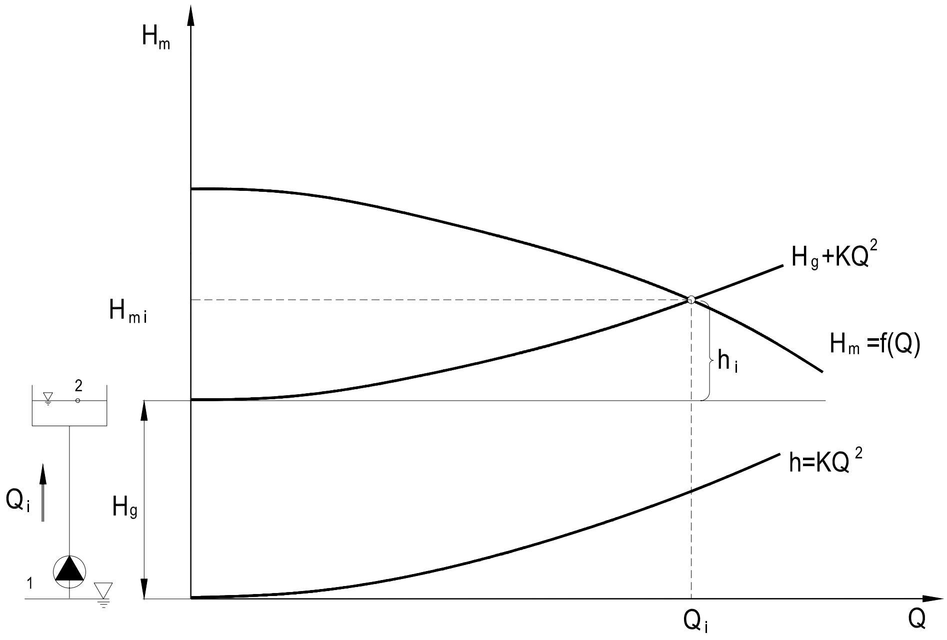


Figure 5.5 Pump and Installation Common Operation Point

Example (3) : Pumping System Calculation

The flow rate of the pump in Figure 5.6 hereinafter is 650 m^3/h and H_g =35m. Suction and delivery pipes are galvanized.

Suction :
Suction pipe diameter = Ø300
Suction pipe length = 8 m
Bottom clack with filter = Ø350 1 piece
Gate valve = Ø300 1 piece
90^o elbow = 1 piece
Vertical conic = Ø300-Ø250 1 piece

Discharge:
Delivery pipe diameter = Ø250
Discharge pipe length = 200 m
Gate valve = Ø250 1 piece
90^o elbow = 3 pieces
Check valve = Ø250 1 piece
Reducer = Ø200/Ø250 1 piece
Dismantling = Ø250 1 piece

Head losses in the system is calculated as follows;

As explained before, the formula giving the pump head is :

$$\boxed{H_m = H_g + (\frac{p_B - p_A}{\rho g}) + \sum_{A-B} h_{friction} + \sum_{A-B} h_{\min or}}$$

Considering that the suction and discharge tanks are open to atmosphere *(p_A=p_B)*, this formula will become $H_m = H_g + \sum_{A-B} h_{friction} + \sum_{A-B} h_{\min or}$. Now let’s calculate each term of this formula:

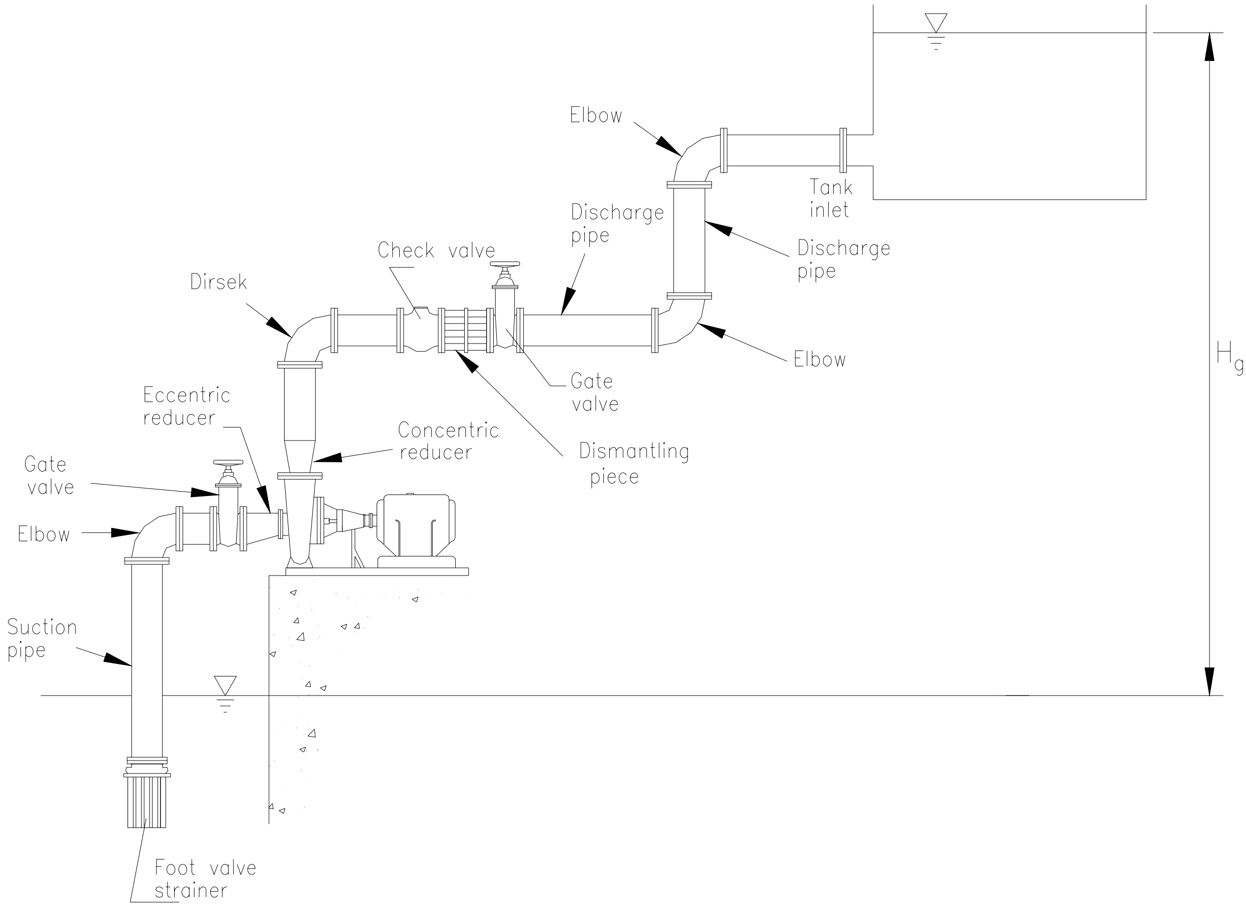


Figure 5.6

H_g, is the water free surface elevation difference between the tanks in the installation. and is equal to H_g = 35 m.

The term $\boldsymbol{h_{friction}}$ is the sum of friction loss occurring in suction and discharge pipes.

If this term is calculated for the suction pipe,

In order to calculate the friction loss coefficient (λ) we need the Reynolds number (Table-1) and the relative roughness.

Reynolds Number is given by the formula: $\mathrm{Re} = \dfrac{VD}{\nu}$, and

V is the velocity of the fluid in the pipe, D is the pipe diameter and ν is the kinematic viscosity of the fluid. The velocity within the pipe is obtained by the flow rate and the cross section area of the suction pipe:

$$V = \frac{Q}{A} = \frac{4Q}{\pi D^2} = \frac{4\,.650/3600}{\pi\ 0{,}3^2} = 2{,}55\ m/\sec$$

Then the Reynolds number is calculated as following:

$$\mathrm{Re} = \frac{VD}{\nu} = \frac{2{,}55\,.\,0{,}3}{11{,}416\,10^{-7}} = 670112$$

The roughness for galvanized pipe is given as 0,12 mm in the Annex Table: 4. Therefore the relative surface roughness is:

$$\varepsilon = \frac{k}{D} = \frac{0.12 \cdot 10^{-3}}{0{,}3} = 0.0004$$

Then by using these values and the Moody diagram given in Annex Table: 1, the friction loss coefficient is obtained as 0,0165. Consequently the friction loss in the suction pipe is:

$$h_{friction\,(suction)} = \lambda \frac{L}{D} \cdot \frac{V^2}{2g} = 0{,}0165 \frac{8}{0{,}3} \frac{2{,}55^2}{2\,.\,9{,}81} = 0{,}14\ m$$

Similar calculations for the discharge pipe are performed:

$$V = \frac{Q}{A} = \frac{4Q}{\pi D^2} = \frac{4\,.650/3600}{\pi\ 0{,}25^2} = 3{,}68\ m/\sec$$

$$\mathrm{Re} = \frac{VD}{\nu} = \frac{3{,}68\,.\,0{,}25}{11{,}416\ 10^{-7}} = 805886$$

$$\varepsilon = \frac{k}{D} = \frac{0{,}12 \cdot 10^{-3}}{0{,}25} = 0{,}00048$$

And the friction loss coefficient is 0,017. Consequently the friction loss in the discharge pipe is:

$$h_{friction\,(delivery)} = \lambda \frac{L}{D} \cdot \frac{V^2}{2g} = 0{,}017 \frac{200}{0{,}25} \frac{3{,}68^2}{2\,.\,9{,}81} = 9{,}38\ \text{m.}$$

The total friction loss of the system is $\sum h_{friction} = 0{,}14 + 9{,}38 = 9{,}52$.

Minor loss, h_{minor}, is calculated for suction pipe in the following way: First, velocity of the liquid is calcluted:

$$V = \frac{Q}{A} = \frac{4Q}{\pi D^2} = \frac{4\,.650/3600}{\pi\ 0{,}3^2} = 2{,}55\ m/\sec$$

Minor loss in the suction pipe is:

$$h_{\min or(suction)} = K_S \cdot \frac{V^2}{2g} + K_D \cdot \frac{V^2}{2g} + K_{SV} . \frac{V^2}{2g} + K_{DR} . \frac{V^2}{2g} = (K_S + K_D + K_{SV} + K_{DR}) \frac{V^2}{2g}$$

$$= (2{,}25 + 0{,}29 + 0{,}20 + 0{,}02) \frac{2{,}55^2}{2.9{,}81} = 0{,}92\ m$$

Then in order to obtain the minor loss in the discharge pipe, the liquid velocity is first calculated:

$$V = \frac{Q}{A} = \frac{4Q}{\pi D^2} = \frac{4\,.650/3600}{\pi\ 0{,}25^2} = 3{,}68\ m/\sec$$

Minor loss at the discharge pipe is as follows:

$$h_{\min or(delivery)} = K_{DR} \cdot \frac{V^2}{2g} + 3 \cdot K_D \cdot \frac{V^2}{2g} + K_{CV} \cdot \frac{V^2}{2g} + K_{SP} \cdot \frac{V^2}{2g} + K_{SV} \cdot \frac{V^2}{2g} + K_{HG} \cdot \frac{V^2}{2g}$$

$$= (K_{DR} + 3 \cdot K_D + K_{CV} + K_{SP} + K_{SV} + K_{HG})\frac{V^2}{2g}$$

$$= (0{,}04 + 3 \cdot 0{,}29 + 0{,}8 + 0{,}05 + 0{,}22 + 1)\frac{3{,}68^2}{2.9{,}81} = 2{,}05\ m$$

Therefore the sum of the minor loss is $\sum h_{\min or}$ = (0,92+2,05) = 2,97 m. When we replace the friction loss occurring in suction and discharge pipes, the minor loss and the elevation difference in the following formula we obtain the pump total head (H_m).

$$H_m = H_g + \sum_{A-B} h_{friction} + \sum_{A-B} h_{\min or} = 35 + 9{,}52 + 2{,}97 = 47{,}49\ \text{ m}$$

It means physically that the pump installed to the system transfers an energy to the fluid for a value of 47,49 m and 35 m of this is spent to lift the fluid from the suction tank to the discharge tank and the remaining 12,49 m is spent for losses. Therefore, the term H_m at the right side of above equation represent the pump total head. Consequently the following figure shows the pump (NORM 200/400) to realize this operation and the operation point.

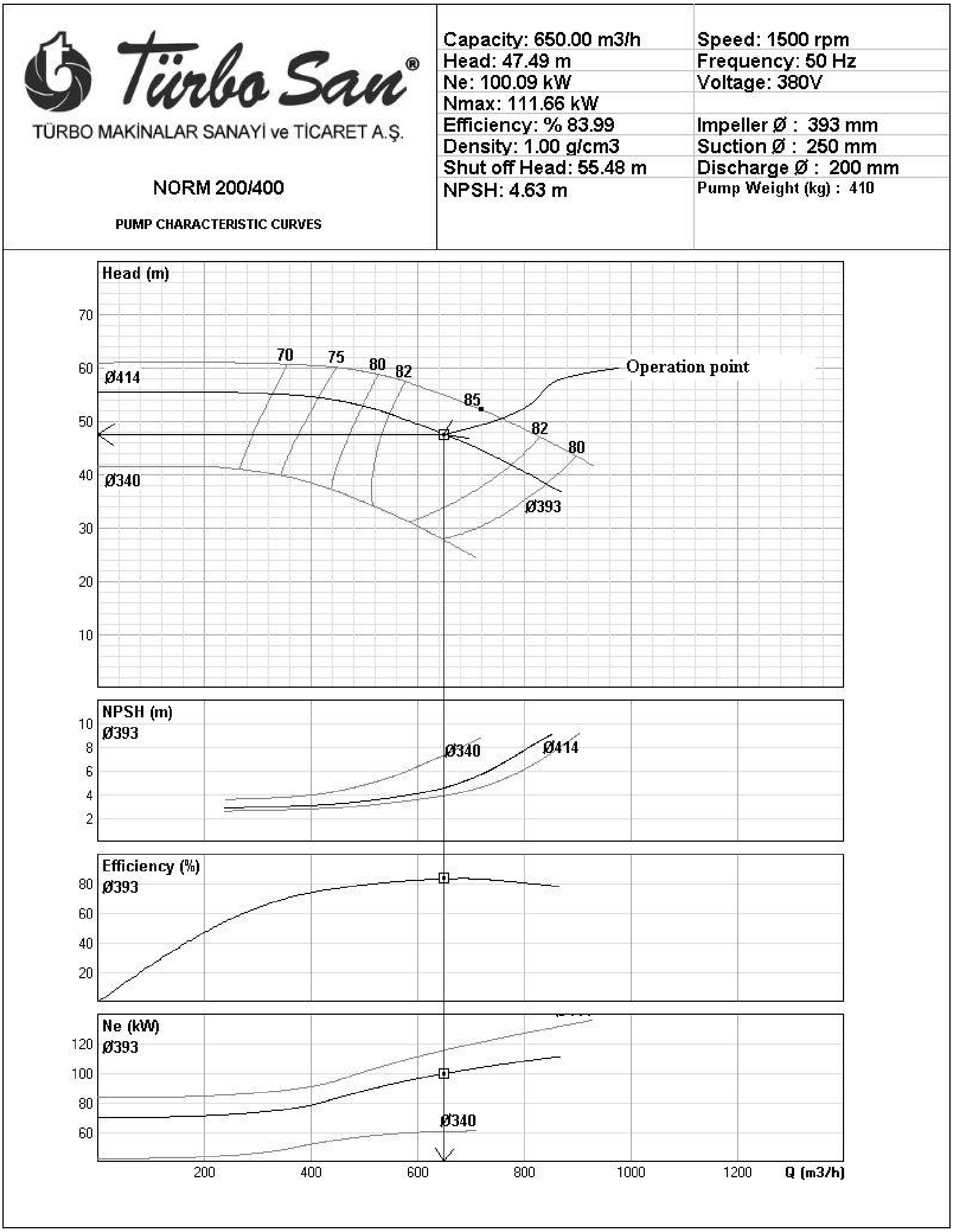


Figure 5.7 Characteristic Curve of the Pump chosen according to Calculation in Example 3

5.5.1 System Loss Increase and its Influence on the Pump Total Head

Although the intersection point of the original system characteristic curve and the pump characteristic curve is A, the (h) curve will become (h') on account of the change (increase) of loss characteristics of some elements on the piping system and the operation point will become A'. The pump total head H_m will increase by $\triangle H_m$ and the flow rate will decrease by $\triangle Q$.

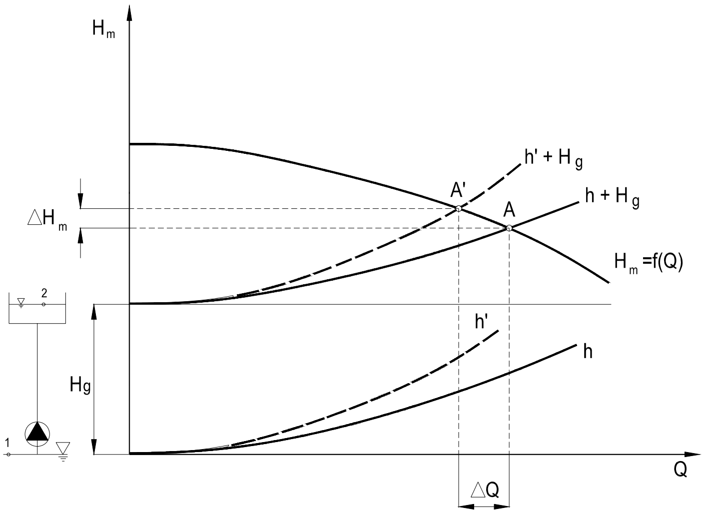


Figure 5.8 Effect of System Loss on the Pump Total Head

5.5.2 Variable Elevation Head and its Influence on the Pump Operation Point

The variation of the liquid levels of the suction and delivery reservoirs in time will result in the variation of the system characteristics. For example let's assume that in the following system the water level of the discharge tank changes between the maximum and minimum values. System characteristics corresponding to these levels are given in the following H_m-Q graph. Thus, when the water level in the tank will be at the minimum level the pump will operate at point A. As the water level will increase this the operation point will remain above the curve H_m-Q and move towards the point A': When the pump is operating at point A' the system flow rate will be Q_{min}, at point A it will be Q_{max}. These variation should be taken into consideration when determining the motor power for selected pump.

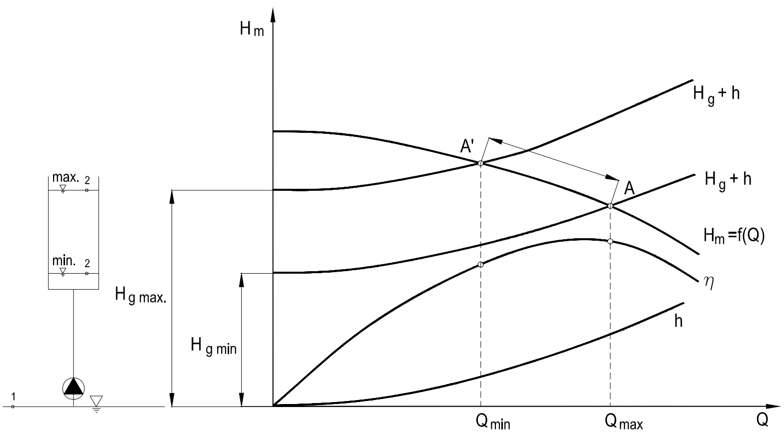


Figure 5.9 Elevation head influencing the Pump Operation Point

5.5.3 Valve Throttling

In the event the pump discharge valve is slightly trottled a new pipe characteristic $h= KQ^2$ will form for each valve opening between the fully open valve position (A) and the very slightly open position valve (A_n). Therefore each point of intersection of these characteristics with the curve H_m-Q becomes a new operation point for the pump**.**

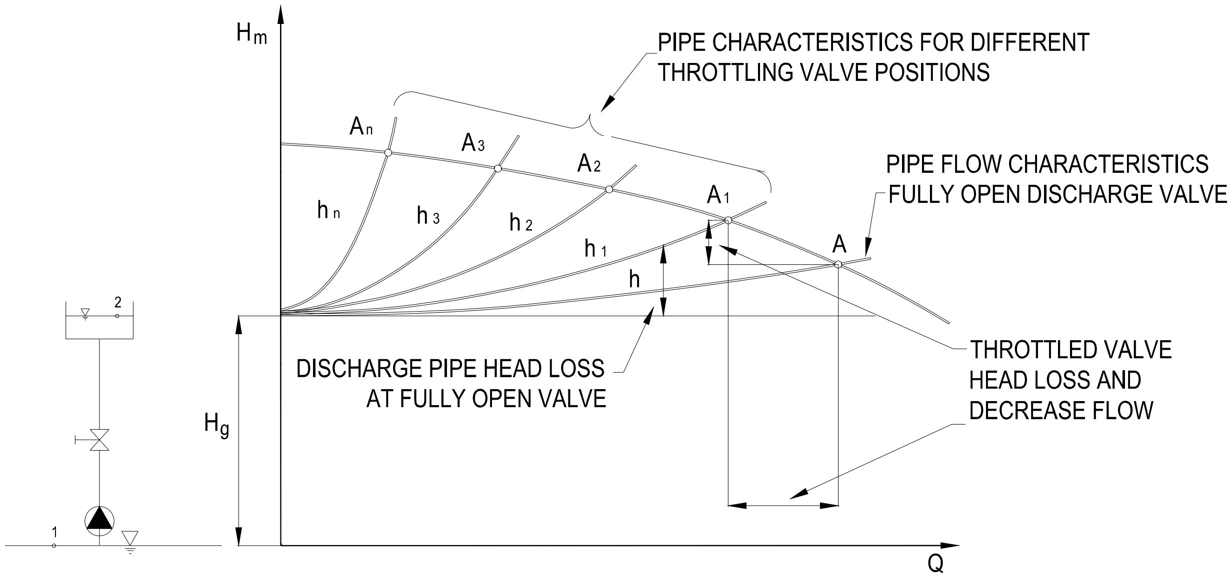


Figure 5.10 Influence of Valve Throttling on the Pump Operation Point

5.6 Adjustment of the pump operating point by modifying the RPM:

There are two independent variables for the adjustment of the pump operation point: the flow rate and the rpm. The flow rate is adjusted by modifying the system characteristics and the pump shifts to the new operation point as explained in Subsection 5.5. Another way of adjusting the operation point is to change the RPM of the pump. In such event the characteristic curve of the pump is modified and the pump adjusts its operation point in accordance with system characteristics in a manner to remain on the new curves.

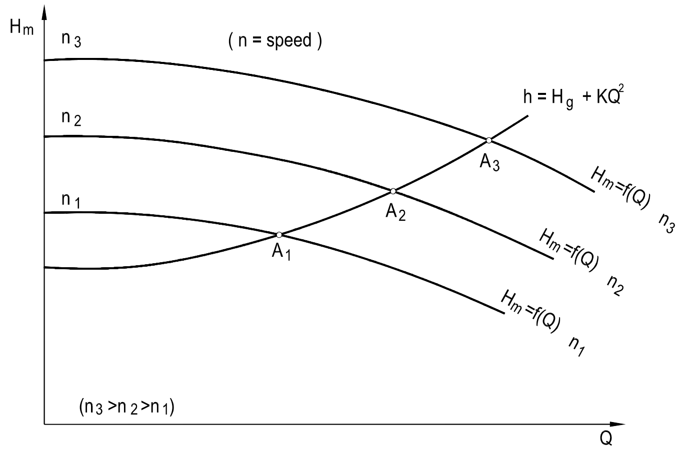


Figure 5.11 Pump Operation Point adjusted by modifying the rpm

5.7 Parallel and Series Operation of Pumps

5.7.1 Parallel Operation

Parallel operation are used to obtain high flow rates and meeting the variable flow rate requirements by a number of pumps. The main target is to increase the system flow rate.

When two pumps are connected in parallel the total flow rate is the sum of the flow rates of individual pumps at the same head.

In case of parallel connected pipe lines exist in the pipe system, the system characteristic is obtained by adding the individual pipe line system characteristics in the Q axis.

These cases are explained in detail in the following subsections.

5.7.1.1 Three identical Pumps Running in Parallel

In case three pumps with the same H_m-Q characteristics are connected in parallel and pump in the same installation, the total pump characteristic is obtained by using the characteristics of one individal pump, as explained hereunder:

a) As shown in the following figure a line parallel to Q axis is drawn on the graph H_m-Q.
b) The flow rate values of this line intersecting the H_m-Q curve of each pump remaining on the same horizontal line are added. The total Q value thus obtained is the flow rate value of a point on the total pump characteristic curve. The H_m value of this point is the intersection of the horizontal line with H_m axis. Thus a point relating to a total pump characteristic is obtained (the point A). As parallel pumps are identical in this example, the addition of pump flow rates on the Q axis is equivalent to the multiplication of the flow rate of a single pump by three.
c) This operation is repeated for horizontal lines drawn for different H_m values for any number of points as requested and the pump characteristic curve is obtained.

The point of intersection of the total pump characteristics with the system characteristic $h=H_g+KQ^2$ is the common operation point of the pump and the system (the point A). The flow rate value of the common operation point (Q_A) is equal to the flow rate crossing the system. The H_m value of the intersection point equals to the sum of losses occurring in the system and the static head. Therefore this value is equal to the pump head.

In order to find out the operation point of each pump, a line parallel to the (Q) axis passing from the common operation point is drawn. The $Q_{A'}$ ve H_{mA} values of this line corresponding to the point of its intersection with the H_m-Q characteristic curve of each pump are the operation point of each pump. As pumps in this example are identical the same $H_{mA'}$ and $Q_{A'}$ values are obtained for each pump. In case of non identical pumps, pumps will operate with the same H_m value but with different Q value.

In the event two of the three pumps are operating, the point of intersection of the total characteristics of two pumps as explained hereinbefore with the system characteristics curve

will be B. Each single pump will operate in point B' and will pump the $Q_{B'}$ flow rate with pump head value of $H_{mB'}$.
If a single pump is running in the same system the pump characteristic curve and the system characteristics curve will intersect at point C and the point of operation of the pump will be that point C. In order to determine the power of the motors the Q_C and H_{mC} values pertaining to point C must absolutely be taken into consideration.

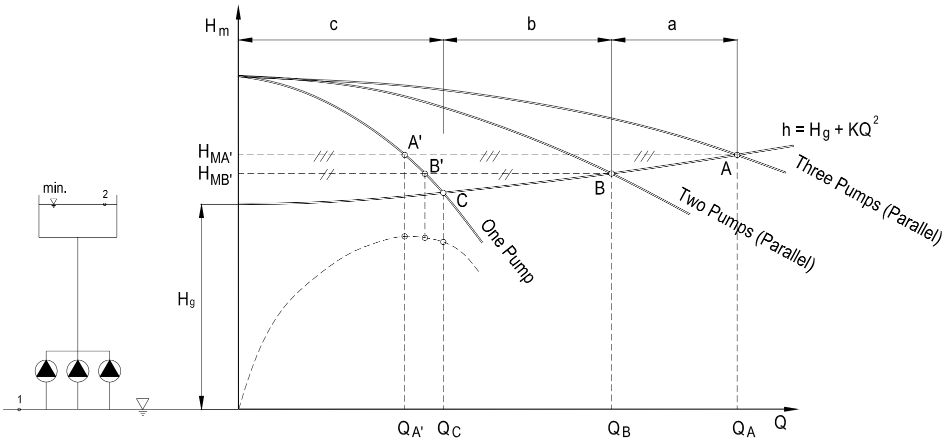


Figure 5.12 Three identical Pumps in Parallel Operation

5.7.1.2 Two different Pumps in Parallel Operation

In case two pumps with different H_m-Q characteristic, the total pump characteristic curve is obtained as explained above.
The point of intersection of the system characteristics with the total pump characteristics (point A) is the common operation point. The system flow rate is Q_A. The operation point of each single pump is the intersection point the horizontal line passing from the point A with the characteristic of each individual pump. Therefore, as indicated in the following Figure, these points are C' for the pump 1 and B' for the pump 2.

If the pumps are running individually the operation point of the pump 1 will be C and that of the pump 2 will be B. In the determination of the motor powers of pumps, individual running situation must be taken into consideration..

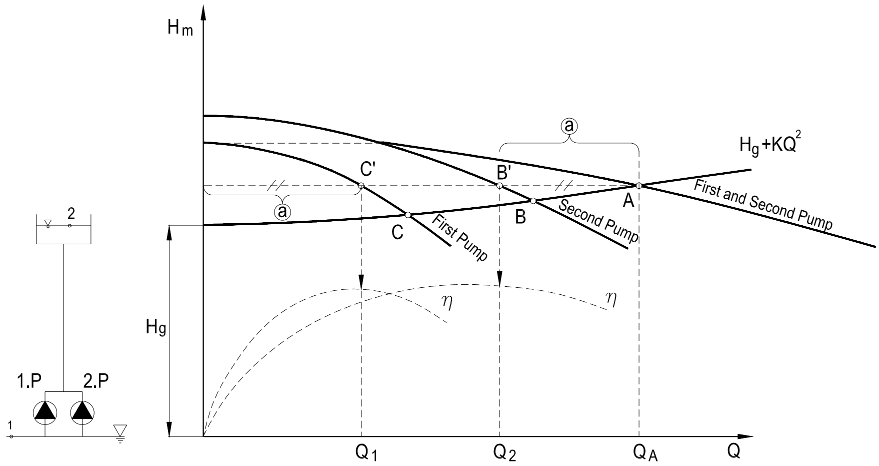


Figure 5.13 Two different Pumps in Parallel Operation

5.7.1.3 Volumetric Pump and Centrifugal Pump running in Parallel

H_m-Q characteristic of a volumetric pump is shown in the following figure. As explained before, it is a vertical line. In the event a centrifugal pump is connected in parallel to the volumetric pump, the total pump characteristic will be obtained by on the horizontal axis, as explained hereinbefore. This is equivalent to moving forward the H_m-Q characteristic curve of the centrifugal pump on the Q axis for the value of the flow rate of the volumetric pump.

The operation point of pumps connected in parallel with the system will be the point A. The volumetric pump will operate at point C and the centrifugal pump at point B.

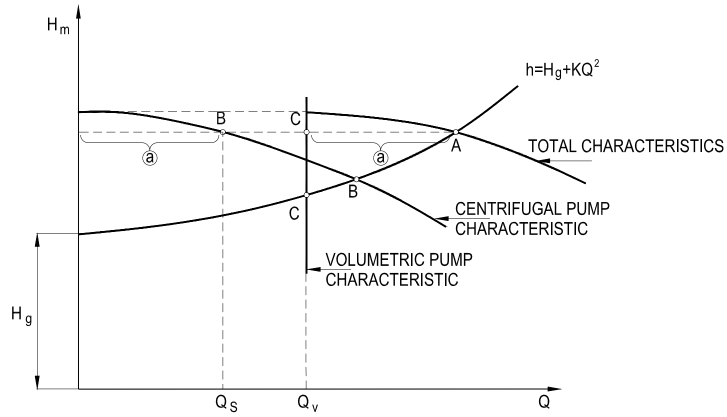


Figure 5.14 Volumetric Pump and Centrifugal Pump running in parallel

5.7.1.4 Characteristics of Piping System Connected in parallel

If the system characteristic for each branch is known, the total system characteristic will be obtained by using the similar method that is applied for pumps running in parallel. In other terms, the total system characteristic is obtained by adding the H-Q characteristics curves of the parallel connected pipe systems on the horizontal axis. In the following Figure, the total characteristic of pipe lines whose individual system characteristics are h_1 and h_2 and running in parallel are added on the horizontal line and obtained as the curve shown by h_1+h_2.

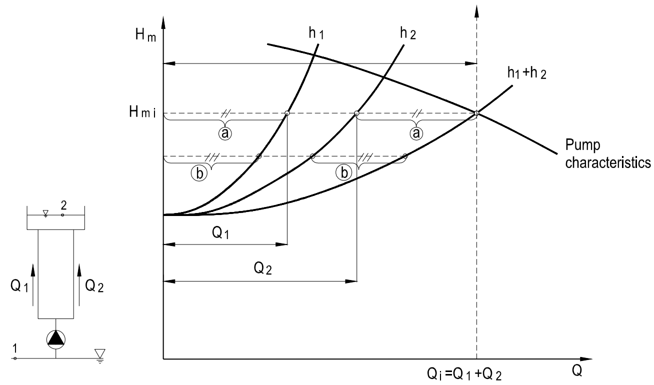


Figure 5.15 Characteristic of piping systems connected in parallel

In case pump characteristics are drawn over these characteristics, the common point of operation of the pump and the system will be the point A. The flow rate of each pipe line is found by a line drawn from the point A in parallel to Q axis. The Q value of the point where this line intersects with the h_1 characteristic will give the Q_1 flow rate passing through the pipe line No: 1 and the Q value of the point of intersection with the h_2 characteristic will give the Q_2 flow rate passing thorug the pipe line No: 2.

If the flow rate needed in the installation is Q_i and considering that a single pipe installation (h_1 or h_2) is in question, the value of the H_m should be considerably high in order to allow the flow rate Q_i to cross any of the characteristic loss curves.

In the event parallel pipe installation is built, a part of Q_i will be provided as Q_1 and the other as Q_2 to be distributed between the pipes according to their resistance.

This type will be obtained either by building a parallel line to the existing installation or by making an optimization between building single line in the original investment and building a parallel pipe line for the operation point A.

5.7.1.5 Some important Points about Parallel Connections

Effect of the Pump Characteristics Form on the Operation Point

During single operation a pump with steep characteristics will intersectthe system characteristic curve at point A, whereas when running in parallel with an identical pump the pump total characteristics curve will meet the system curve at point B. The operation point of each pump will be B'.

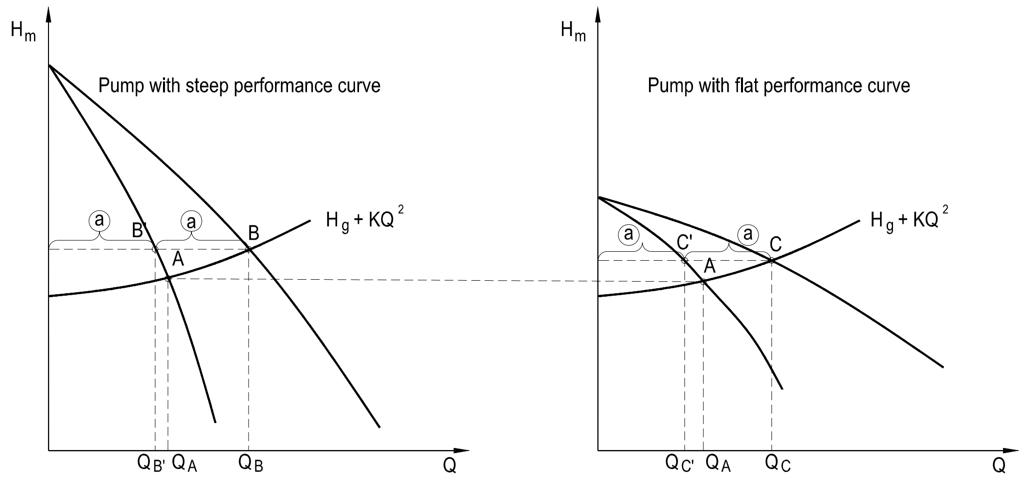


Figure 5.16 Effect of the Form of Parallel connected Pump Characteristics on the Operation Point

Whereas a pump with flatter characteristics running in single mode will meet the curve H_g+KQ^2 at point A, it will meet the common characteristic curve H_g+KQ^2 at point C when running in parallel with its identical pump. The operation point of each single pump will be C'.

When pumps with two different characteristics are pumping at point A, the flow rate to be obtained with two pumps with steep characteristics connected in parallel will be lower than that with flatter characteristics.

The Effect of the Piping System Characteristics Form on the Operation Point

When pumps are connected in parallel, the system flow rate will not equal the double of the flow rate of the single pump. In parallel operation, the system flow rate is set by the system characteristics.

In the following figure on the left side the characteristic of the pipe system is horizontal. Therefore the flow rate provided by parallel connected pumps is about the double of the single pump. The piping system characteristic of the figure at the right side is vertical. The flow rate provided by parallel connected pumps is almost equal to the flow rate of the single pump.

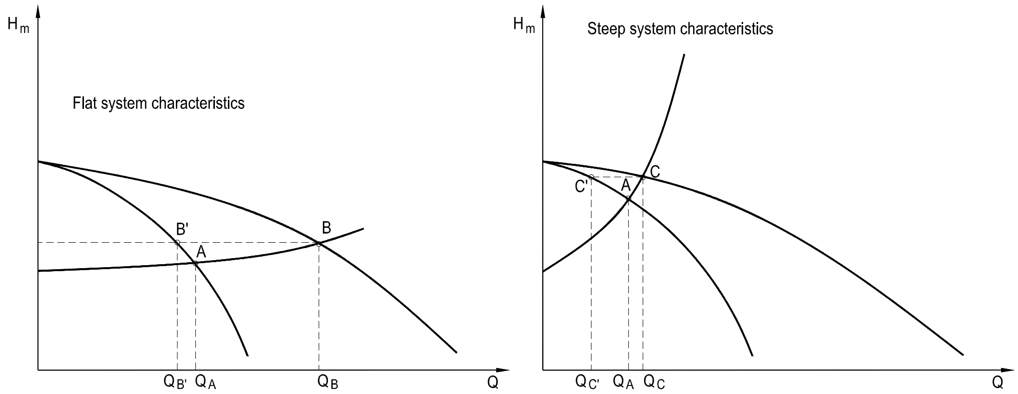


Figure 5.17 System Flow Rate Changing by the Form of the System Characteristics in Parallel Connected Pumps

5.7.2 Serial Connection

When pumps connected in serial, the total pump characteristic is obtained by adding the H_m-Q curves of the pumps on the vertical axis–H_m axis.

If the pipe system contains serial connected pipe lines, the system characteristic is calculated by the classical method. In case pipe diameters are different along the line, velocity variations should be taken into consideration.

These topics are explained in detail in following sub-sections.

5.7.2.1 Pumps Running in Serial

In case two pumps with different H_m-Q characteristics are connected in parallel and deliver in the same installation, the total pump characteristic is obtained as follows:

a) As shown in Figure 5.18 hereunder, a line parallel to H_m axis is drawn on the graph H_m-Q.
b) The H_m values of the points where this line intersects the H_m-Q curve of each pump on the same vertical line are added. The total H_m value obtained is is the pump head value of a point on the pump characteristics curve The Q value of that point is at the point where the vertical line intersects the Q axis. Thus a point pertaining to the total pump characteristic is obtained (point A).
c) This operation is repeated for vertical lines drawn in different Q values for a requested number of points and the total pump characteristic curve is obtained.

The point of intersection of the total pump characteristic and the H_g+KQ^2 system characteristic is the common operation point of the pump and the system (Point A). The flow rate of the common operation point (Q_A) equals to the flow rate crossing the system. The H_m value of the intersection point is equal to the sum of the losses occurring in the system and the static head between the tanks. This value is provided by the pumps Therefore this value is equal to the sum of pump heads.

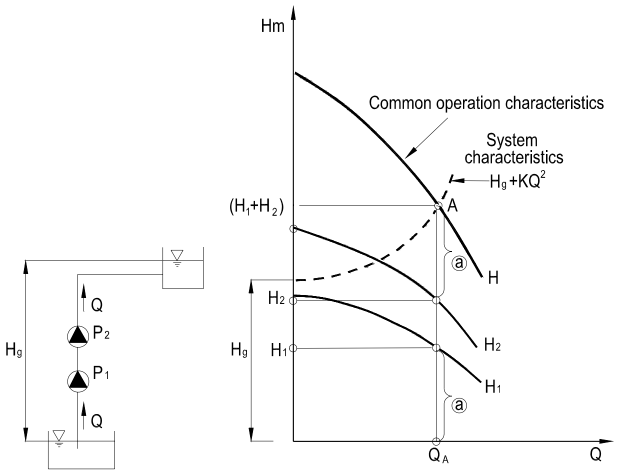


Figure 5.18   Total Characteristics of Serial connected Pumps

In order to find out the common operation point of each pump, a line passing from the common operation point and parallel to pump head axis (H_m) is drawn. The Q ve H_m values corresponding to point of intersection of this line with the H_m-Q characteristic curve of each pump are the operation points of each pump.

5.7.2.2 Characteristics of Serial connected Pipe System

The system head loss characteristics are obtained by adding the head loss terms (i.e. pipe characteristics added vertically) relating to the line which forms the serial connected.

5.8 Some Important Points

5.8.1 Evaluation of Flow Rate Increasing Methods

The effects of the different system characteristics of parallel and serial connected pumps are studied hereinbelow. The Figure 5.19 compares the values of the total flow rate provided by two identical pumps when operated in serial and parallel connection under vertical system characteristics and the value of a single pump. Two pumps serial connected provide more flow rate than parallel connected.

The Figure 5.20 studies the case where the pipe characteristic is flatter. In this case parallel connected pumps provide higher flow rate.
Therefore the system characteristics should absolutely be taken into account when calculating the flow rate.

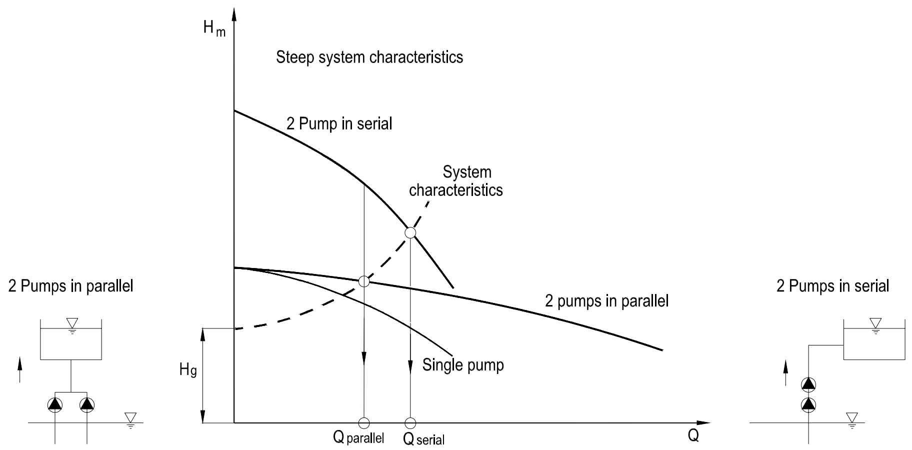


Figure 5.19 Line with Steep System Characteristics where Pipe Losses are High

When two or more pumps run together in pipe systems with steep characteristics, the value of the total flow rate never increases by the number of pumps.

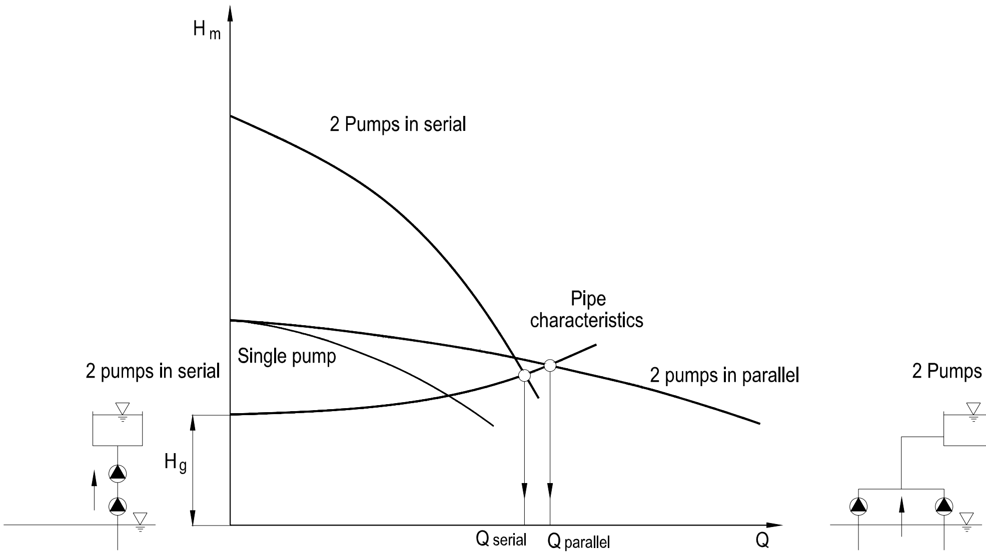


Figure 5.20 Line with Flatter System Characteristics where Pipe Losses are Low

5.9 Pump Running in Complex Networks

Methods for finding out the operation point of some special networks are given hereinbelow.

5.9.1 Single Pump supplying Two Tanks

a- Junction Point at Pump Level

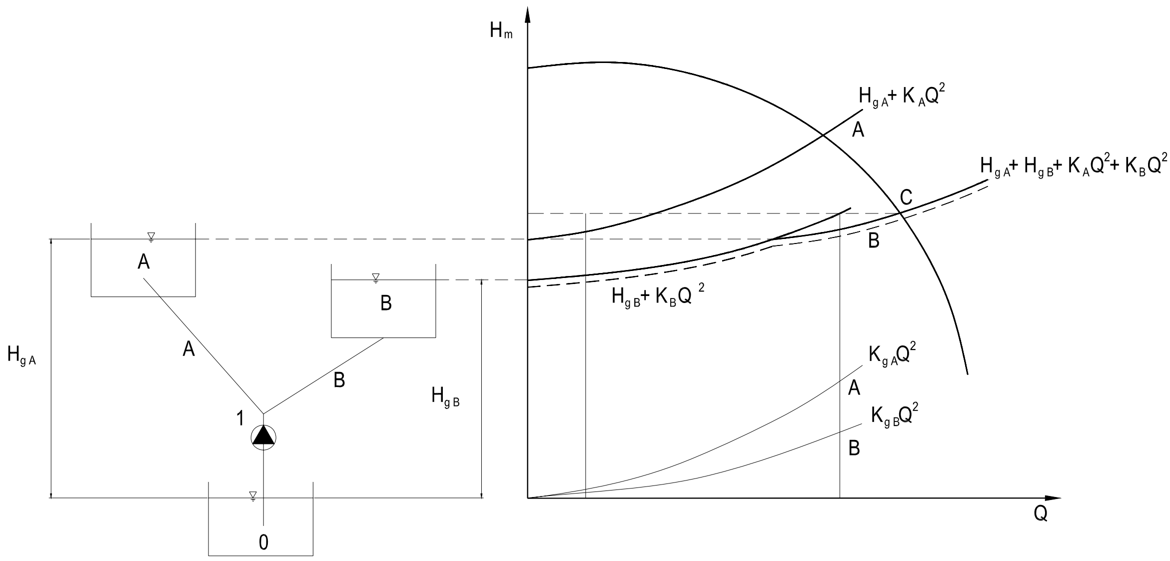


Figure 5.21 Single Pump Supplying Two Tanks – Junction Point at Pump Level

b- Junction Point above Pump Level

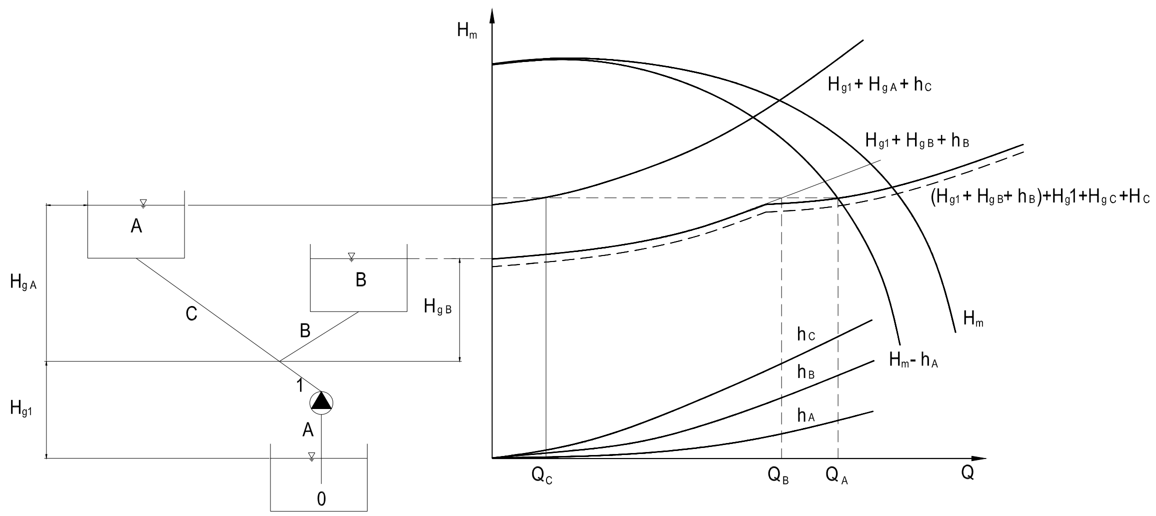


Figure 5.22 Single Pump Supplying Two Tanks – Junction Point above Pump Level

5.9.2 Parallel Operation of Two Pumps with Suction from Different Tanks

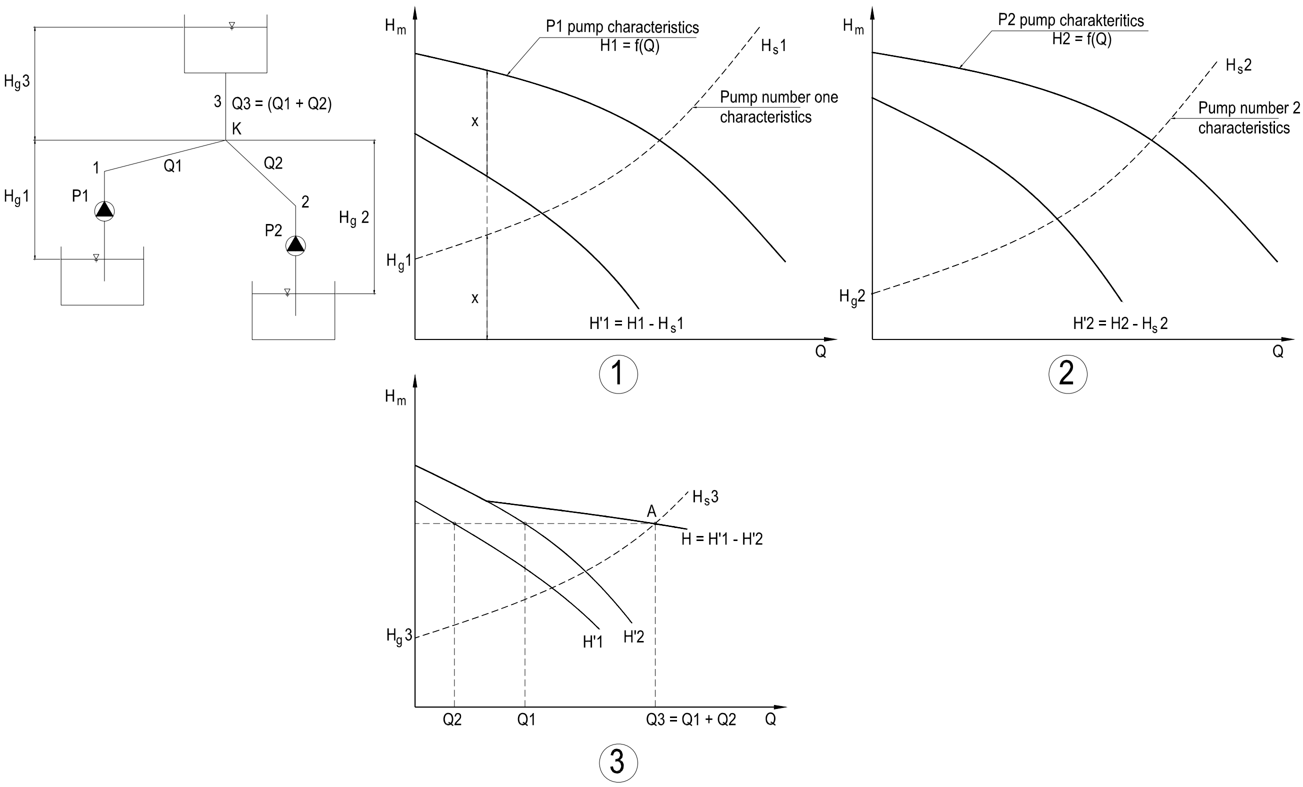


Figure 5.23 Parallel Operation of Two Pumps with Suction from Different Tanks

5.9.3 Parallel Discharge to Two Tanks – One Line with Additional Pump

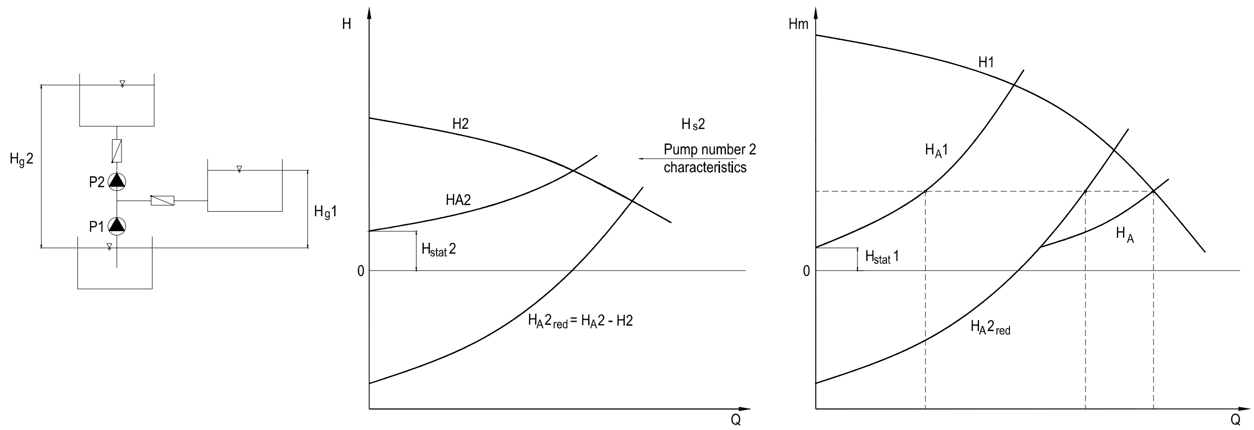


Figure 5.24 Parallel Discharge to Two Tanks

5.9.4 Circulation Circuits

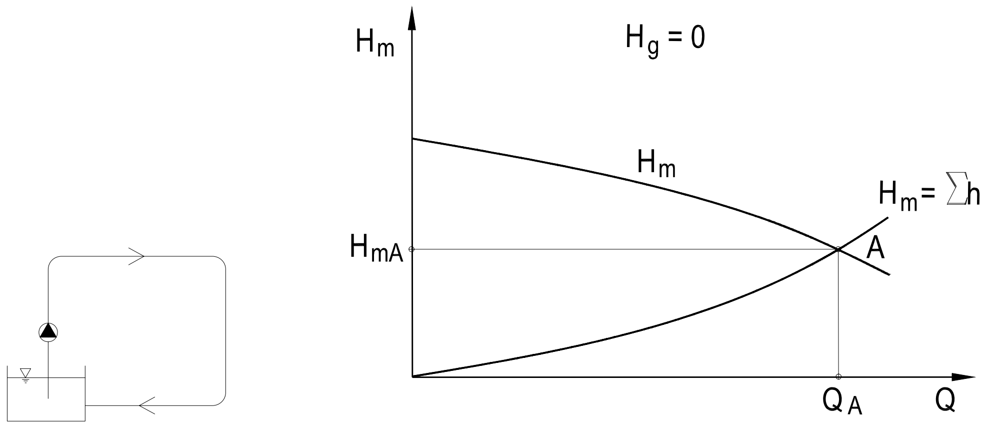


Figure 5.25 Circulation Pump

5.9.5 Installation with Negative Pump Head

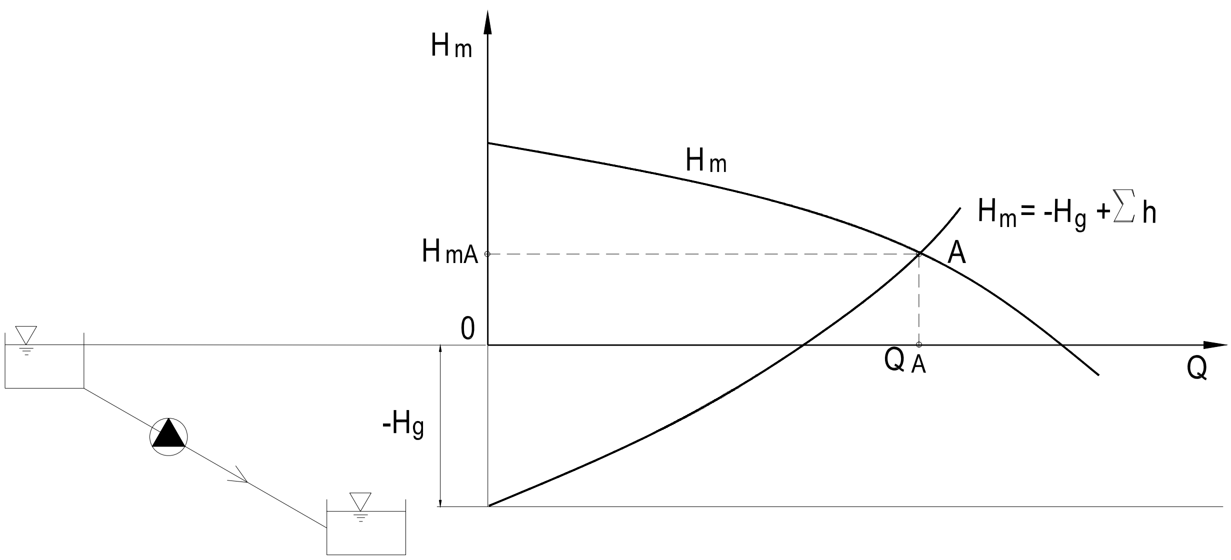


Figure 5.26 Installation with Negative Pump Head

5.10 Installations with Unstable Pump Characteristics

As shown in the following Figure, in case a pump capable of running with a pump head higher than that of the closed valve runs in a system where the difference of water level is variable such as pressure tanks, pressure water tanks, shipyard docks, drainage channels, the operating point will change in relation with the level difference. Whereas a single operation point (A_1) exists for the case A, there will be two operating points (B_1, B_2) for the case B and in such event the flow rate and the pump head will change between these two values, creating thereby undesired flow fluctuation within the system. This fact should be taken in consideration when designing the system.

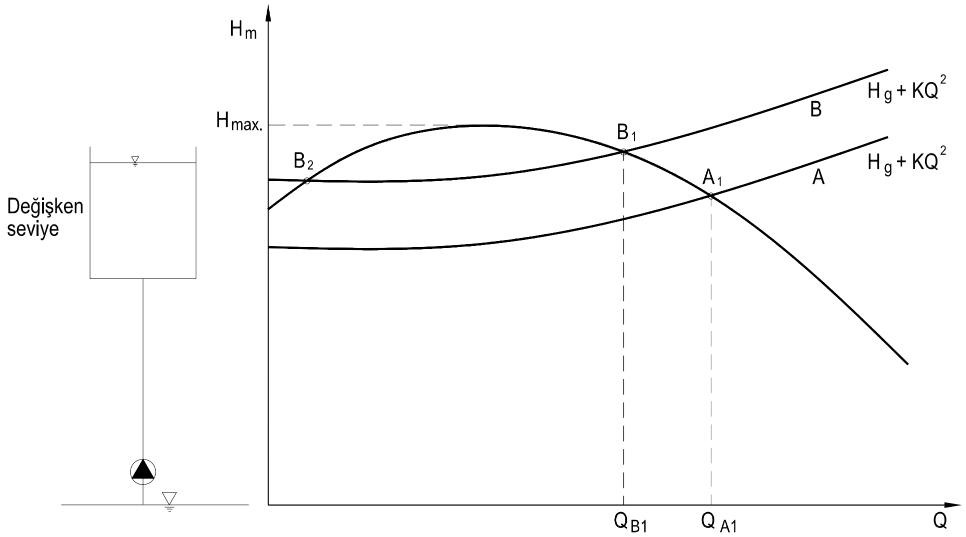


Figure 5.27 Installations with Unstable Pump Characteristics

5.11 Pump Total Head and Head Loss Examples in Pumping Systems

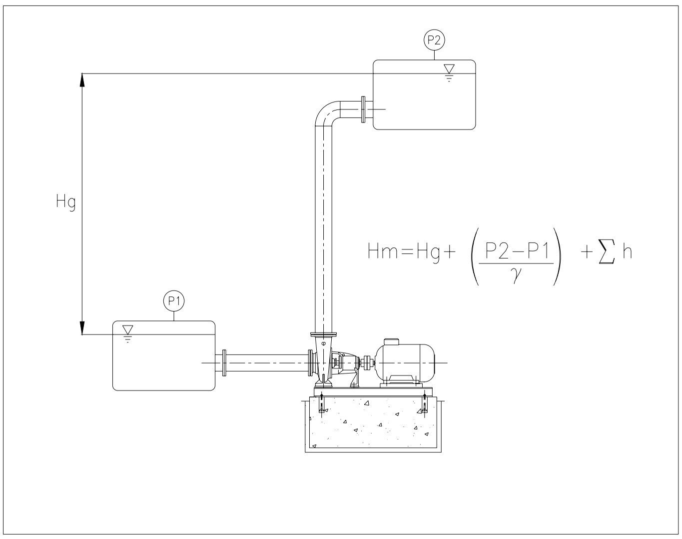


Figure 5.28 System with Pressure in Suction and Delivery Reservoirs

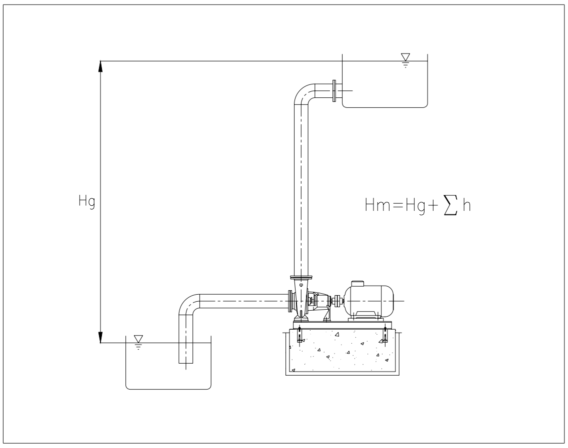


Figure 5.29.a

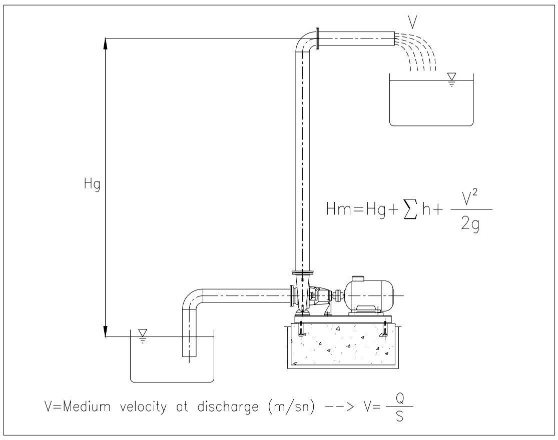


Figure 5.29.b

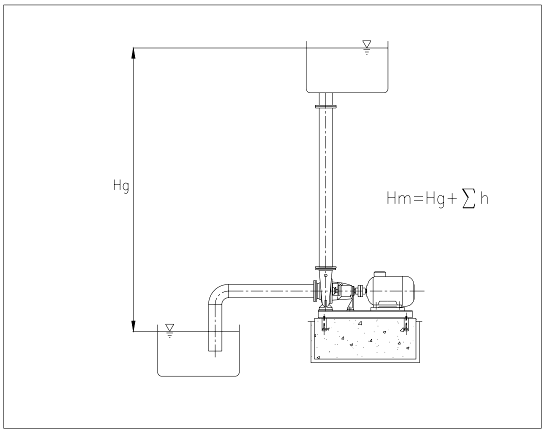


Figure 5.29.c

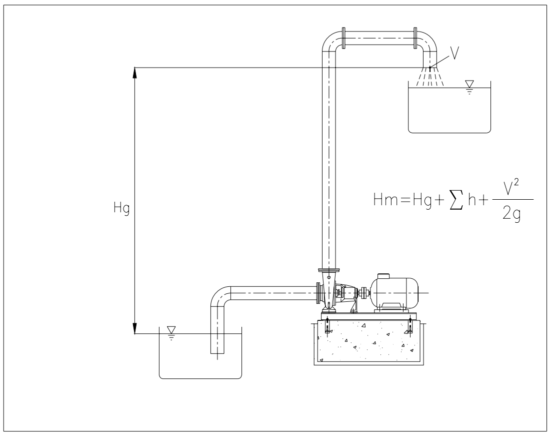


Figure 5.29.d

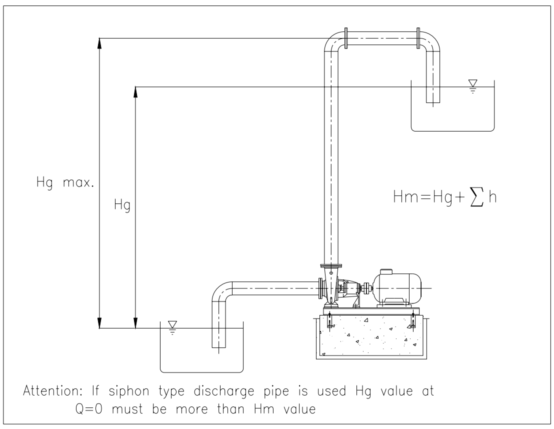


Figure 5.29.e

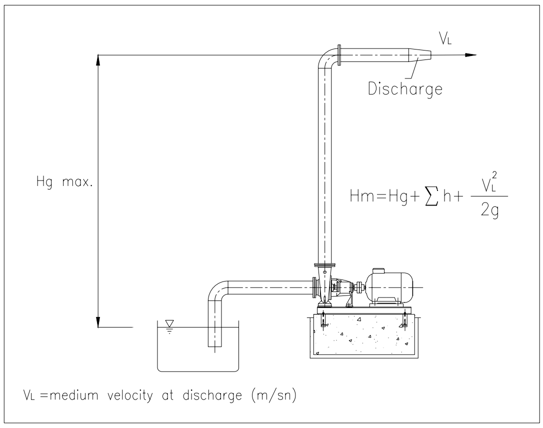


Figure 5.29.f

Figure 5.29 Systems with Open Air Type Suction and Delivery Reservoirs

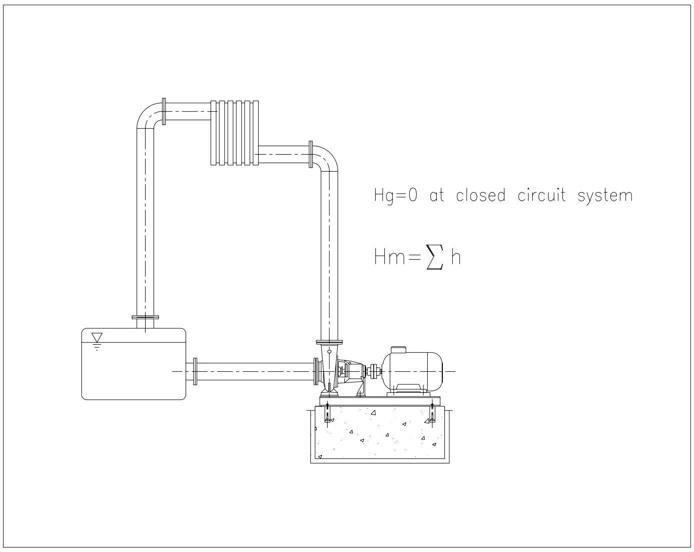


Figure 5.30 Closed Circuit Circulating System

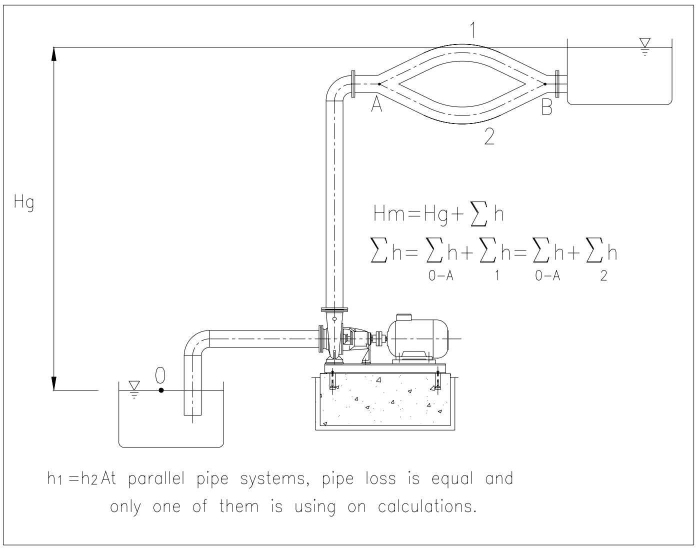


Figure 5.31 System with Parallel Pipe on the Delivery Line

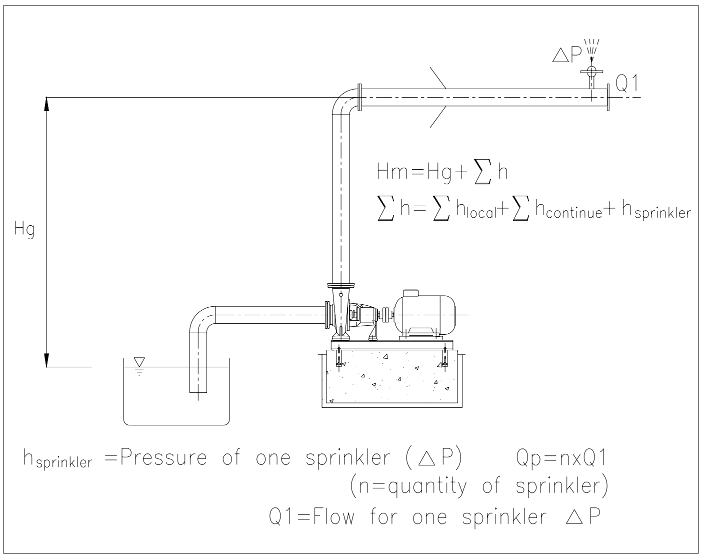


Figure 5.32 Sprinkling System with Sprinkler on the Delivery Line

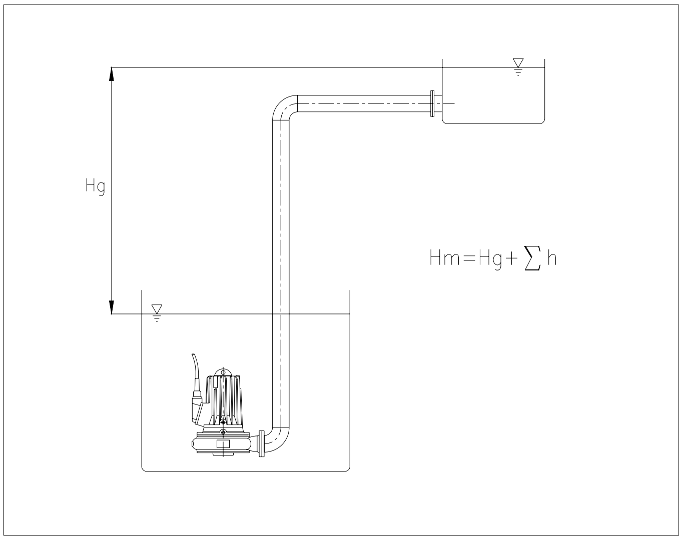


Figure 5.33 System with Submersible Pump

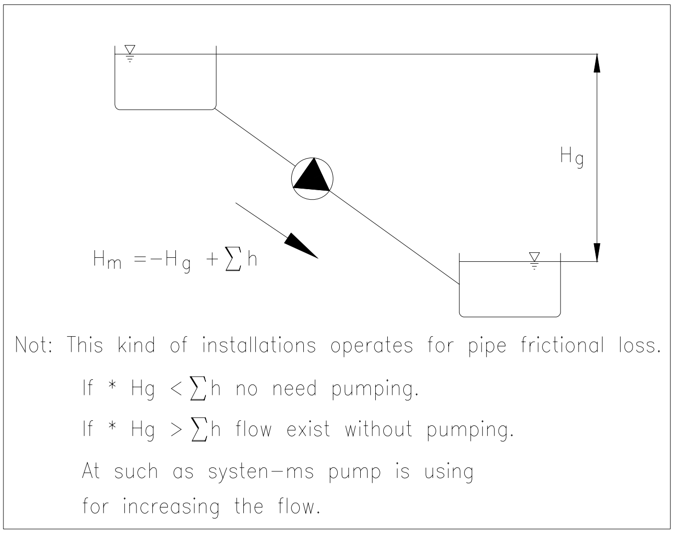


Figure 5.34 System where the Delivery Reservoir is Lower than the Suction Reservoir

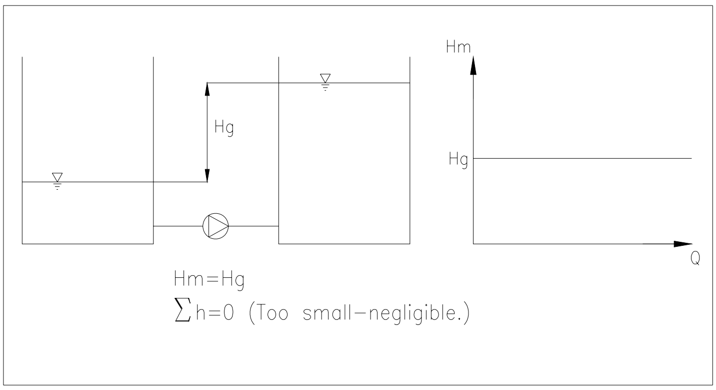


Figure 5.34.a System for Pumping from one Tank to Another

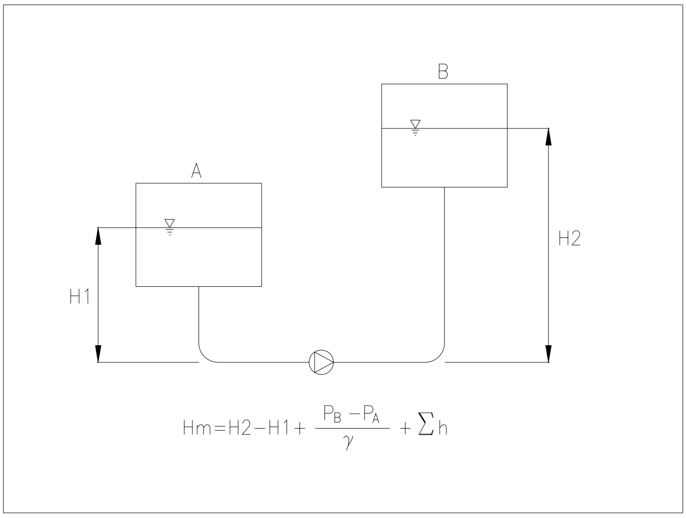


Figure 5.34.b   System for Pumping from one Tank to Another

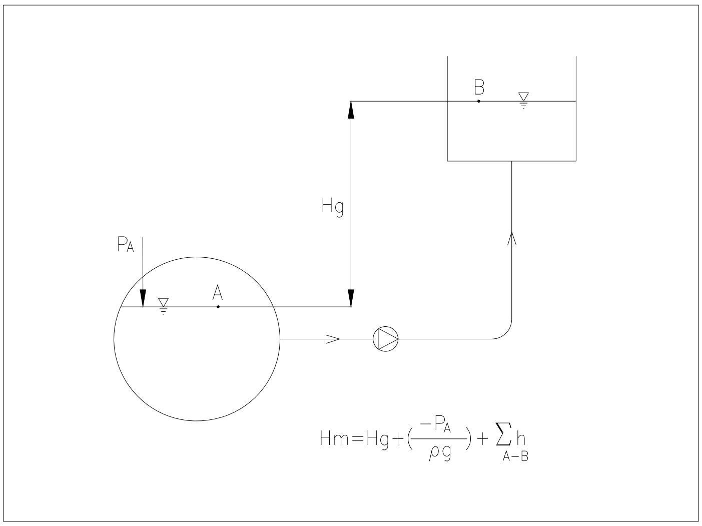


Figure 5.35   Pressure in the Suction Pipe (Tank)

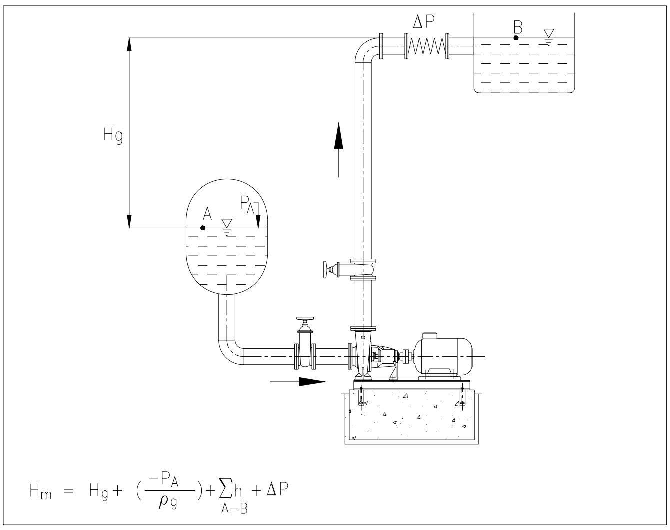


Figure 5.36   Pressure in the Suction Pipe and Exchanger on the Delivery Line;

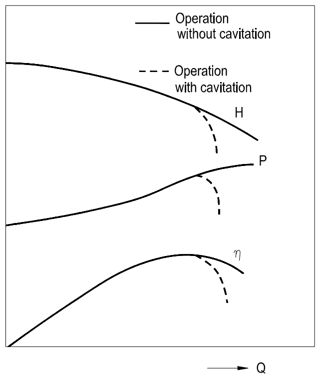
PART 6

PHYSICAL EVENTS DETERMINING THE RANGE OF OPERATION OF PUMPS – CAVITATION AND SURGE

The operation range of a pump is limited with physical events called cavitation and surge. Cavitation determines the highest flow rate of the pump and the surge determines the lowest flow rate. Both concepts are closely related with the pump design and operating conditions. By its results cavitation inflicts damage to the pump. Surge is the fluctuation of the flow direction in the pumping system and is related with the system as regards its consequences.

6.1 Cavitation

6.1.1 Introduction



In the event during the movement of the fluid within the pump, the absolute pressure of the fluid falls below the vapor pressure of the fluid (determined by the atmosphere conditions of the pump location) small gas bubbles are formed in the fluid. This gas bubbles generally containing the fluid vapor and air –on account of air dissolved in the fluid- are drifted away by the stream and, when reaching the region where the pressure is higher than the vapor pressure, they collapse and disappear. The vacuity thus formed is rapidly filled up by the fluid. This fact momentarily creates very high pressures on the walls. The signs and the effects of the cavitation may be listed as follows:
a) very obvious vibrations strong knocks in the pump.
b) important and very rapid fall downs in the pump characteristics curve.
c) rapid and important worn-outs of the walls and formation of porous appearance on the walls on account of spontaneous and huge pressures. In case cavitation stays for a long time the pump may become unusable. For that reason cavitation is an unwanted event.

Cavitation depends both on the pump design as well as operating conditions. The suction head of the pumps is one of the important conditions together with operating conditions. The most important factor limiting the pump head lift is the cavitation. A pump conforming to suction conditions must be provided. If necessary, a piping installation modification and a pump type change must be made such as double entry pump instead of axial pump.

As the specific speed increases the pump becomes more vulnerable to cavitation. The pump RPM is an important factor with respect to specific speed. Although the pump with lower RPM among two type pumps providing the same flow rate and pump total head can run without cavitation, this event may be witnessed in the pump with higher RPM.

When calculating the specific speed of double suction pumps, the flow rate is divided into two since the impeller has two inlets. Thus the specific speed falls down to $n_{s1} = n_s \cdot \left(\frac{1}{2}\right)^{0,5}$. As pumps with lower specific speed have higher suction head as compared to higher specific speed pumps, double suction pumps are preferred for high flow rates.

6.1.2 Net Positive Suction Head : NPSH

Seen from its physical aspect the NPSH is, to certain extent, a criteria defining how the total head value of the fluid at the pump suction flange is far from cavitation, i.e. how much above the pressure head level equivalent to the vapor pressure (p_b) of the fluid and is defined as follows:

$$NPSH = \left(\frac{p_e}{\rho g} + \frac{V_e^2}{2g}\right) - \frac{p_b}{\rho g} = \frac{p_e - p_b}{\rho g} + \frac{V_e^2}{2g}$$

p_e = static pressure measured at the pump reference level (Pa) (absolute) ($1bar = 10^5\,Pa$)

V_e = the velocity of the fluid at the cross section of the pump suction pipe (m/s)

p_b = the vapor pressure of the fluid (Pa) (absolute)

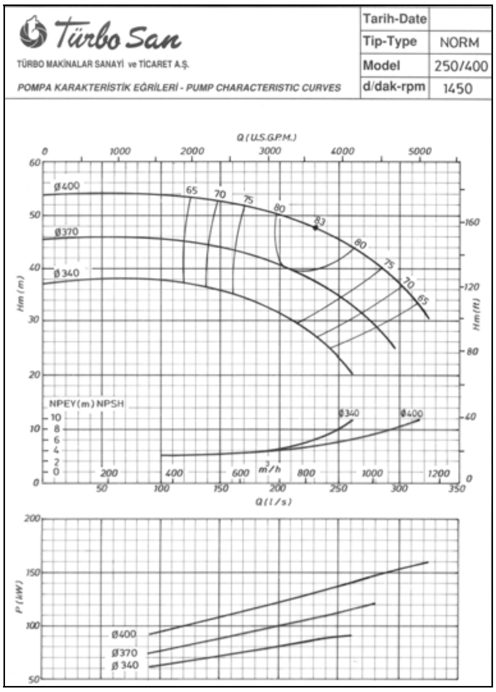
ρ = the density of the fluid (kg/m^3)

g = the gravitational acceleration ($9{,}81 m/s^2$)

The "pump reference level" referred to in NPSH calculation is indicated in Figure 6.7.

6.1.3 Required Net Positive Suction Head: NPSH$_P$

This is the net positive suction head where the pump runs without cavitation. NPSH$_P$ values are set by special cavitation tests and are indicated in the catalogs of pump producers. Figure 6.1 shows the change of the NPSH$_p$ (NPSH (m)), by the flow rate for a NORM 250/400 type pump produced by Türbosan this curve may also be called the cavitation curve.



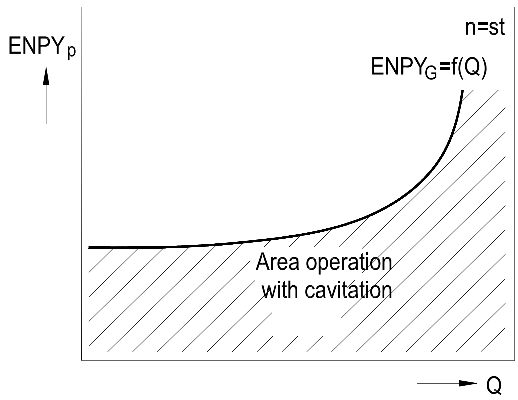


Figure 6.1 Cavitation Characteristics of Türbosan Norm 250/400 Model Pump

6.1.4 Net Positive Suction Head of the Installation: $NPSH_i$

This is defined as follows, taking into consideration the characteristics of the pumping system:

$$NPSHi = H_A - h_s - h_b - h$$

In this formula,

H_A(m): is the absolute value of the pressure head acting on the fluid open surface the reservoir of the installation where the pump will make the suction. In case the suction tank is open to the atmosphere, it is equal to the value of the atmospheric pressure head in meter water column prevailing under the pump operating conditions. The value of the atmospheric pressure changes with respect to the altitude from the sea level. This change must absolutely be taken into consideration.

h_s (m): is the difference of level between the pump impeller suction flange axis (pump suction inlet axis) and the fluid surface in the suction reservoir. The reference determining the direction of the static suction lift is the surface level of the fluid in the suction tank, and;

a) in case the pump is below the reference level than the value of the static suction lift will be negative ($-h_s$). (NEGATIVE STATIC SUCTION LIFT)
b) in case the pump is above the reference level than the value of the static suction will be positive ($+h_s$). (POSITIVE STATIC SUCTION LIFT)

h_b (m): is the value of the vapor pressure of the pumped liquid in meter water colums prevailing under the pump operating conditions and must be selected from the annexed table , taking into consideration the air temperature.

h (m): is the total value of the head losses, in meter water column, occurring in the pumping system suction pipe, on account of several reasons (tank outlet, continuous losses, elbows, valves, clacks, filters etc.).

All pressures mentioned herein are absolute pressures.

As seen from the $NPSH_i$ formula above, in order to increase the suction lift (h_s) losses in the suction pipe and fluid velocity at the suction pipe must be reduced to the extent possible. Therefore, in most pumps the inlet diameter is selected larger than the exit diameter. Besides, items creating local losses such as clacks, filters, valves, elbows, T connections must be avoided as much as possible in the suction pipe.

NPSH is always a positive value and the unit is a meter. Two examples are given for the calculation of the NPSH. in the following

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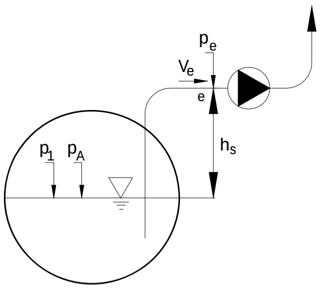

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$$NPSH_P = \left(\frac{p_e - p_b}{\rho \cdot g}\right) + \frac{V_e^2}{2g}$$

$$NPSH_i = H_A - h_s - h_b - h$$

$$(H_A = \frac{p_A + p_1}{\rho \cdot g})$$

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Example 1:

$p_A = 1{,}01\,bar = 10{,}1\,m = 1{,}01 \cdot 10^5\,Pa$

$p_1 = 0{,}3\,bar = 3\,m = 0{,}3 \cdot 10^5\,Pa$

$p_b = 0{,}021\,bar = 0{,}21\,m = 0{,}021 \cdot 10^5\,Pa$

$\rho = 1000\,kg/m^3$; $V_e = 2\,m/s$

$h_s = 2\,m$; $h = 3\,m$

$NPSH_i = \dfrac{(1{,}01 + 0{,}3 - 0{,}2 - 0{,}021 - 0{,}3) \cdot 10^5}{1000 \cdot 9{,}81}$

$NPSH_i = 8{,}04\,m$

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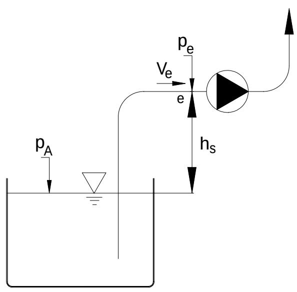

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$$NPSH_P = \left(\frac{p_e - p_b}{\rho \cdot g}\right) + \frac{V_e^2}{2g}$$

$$NPSH_i = H_A - h_s - h_b - h$$

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Example 2:

$p_A = 1{,}01\,bar = 10{,}1\,m = 1{,}01 \cdot 10^5\,Pa$

$p_b = 0{,}021\,bar = 0{,}21\,m = 0{,}021 \cdot 10^5\,Pa$

$\rho = 1000\,kg/m^3$; $V_e = 2\,m/s$

$h_s = 2\,m$; $h = 3\,m$

$NPSH_i = \dfrac{(1{,}01 - 0{,}2 - 0{,}021 - 0{,}3) \cdot 10^5}{1000 \cdot 9{,}81}$

$NPSH_i = 4{,}98\,m$

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($NPSH_p$ is determined from the catalog of the manufacture)

In order to have a pump running without cavitation the following condition should be satisfied:

$$NPSH_i > NPSH_P$$

Here the $NPSH_P$ is the NPSH value given in the catalog where cavitation starts and is established by specially conducted experiments.

Figure 6.2 compares the operation points in the event two installations, one with negative and the other with positive static suction lift, deliver water to tanks with the same level by using the same pump. NPSH curves show that whereas the installation with negative suction lift runs with cavitations, the installation with positive suction lift runs without cavitation.

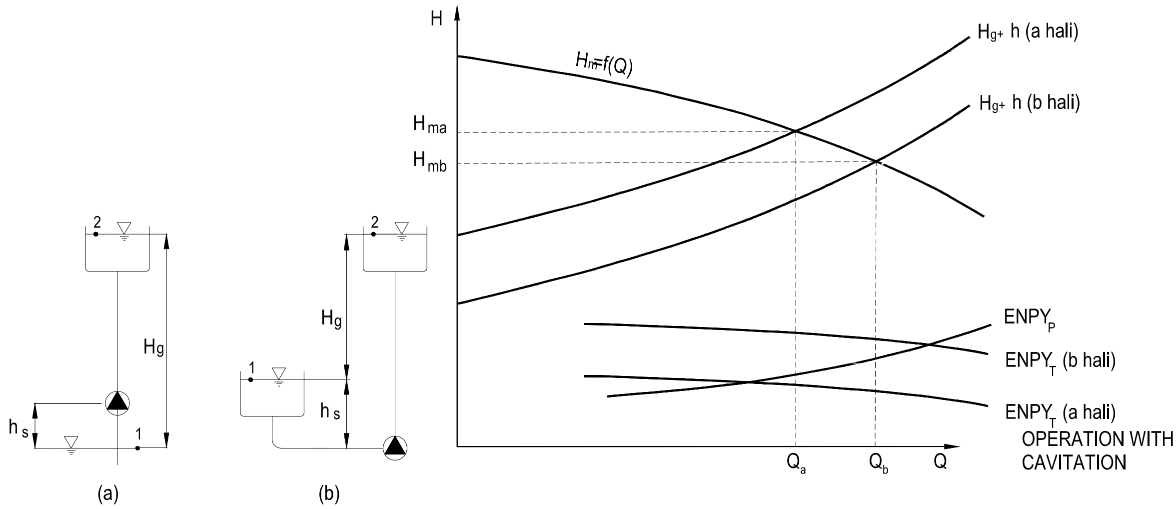


Figure 6.2 Determining the Operation Points of Pumps with Positive and Negative Static Suction Lift

Following are examples for the determination of the $NPSH_i$ value in pumping systems:

Example 1 – Effect of the Pumped Water Temperature on $NPSH_i$

Let's assume that in a pumping system built in a location where the atmospheric pressure equals 10,13 meter water column, operating with a suction lift (h_s) of 4m and with the flow rate Q, the head loss in the suction pipe (h_k) equals 0.5m water column.

a) Assuming that the water temperature is 0°C the attached table gives the vapor pressure as h_b= 0,258 m and thus

   $NPSH_i = 10{,}13 - 4 - 0{,}258 - 0{,}5 = 5{,}37$ m.

b) Without modifying any other feature in the pumping system, let's assume that the water temperature increases to 80°C. In this case the vapor pressure in the attached table reads h_b= 4,958 m and thus
   $NPSH_i = 10{,}13 - 4 - 4{,}958 - 0{,}5 = 0{,}672$ m.

Consequently if a pump with $NPSH_P$=5.0 m value is present in the system,

In case of (a), the pump will run without cavitation since $NPSH_i > NPSH_P$.
In case of (b), the pump will go into cavitation since $NPSH_i < NPSH_P$. The impeller inlet will suffer cavitation damage and the pump will be out of operation well before its economic lifetime.

Example 2 – Effect of Erroneous Determination (With Safety Factor) of the Pump Total Head

We assume that the pump total head value (H_m*) in any installation calculated for a requested flow rate of Q^* is taken as H_{m1} by adding some safety allowances on account of several concerns (Figure 6.3).

The positive suction head value for the installation with Q* value is $NPSH_i$*. We assume that a pump with this value is ordered to the manufacturer.

When preparing his offer, the manufacturer will take the following values for selecting the pump: H_{m1}; Q^* and $NPSH_i$* .

The $NPSH_P$ value of the pump selected in a manner to meet the condition $NPSH_i > NPSH_P$ for running without cavitation is called $NPSH_P$. In this case running without cavitation will be ensured when $NPSH_i^* > NPSH_P$.

When the pump determined by the manufacturer will be commissioned it will run with the pump head H_m* corresponding to the position without **safety factor (!)** in the system. Thus the flow rate delivered by the pump will be Q_2, which is higher than the value of Q*. Therefore at this point the values of the pump will be Q_2, Hm* and $NPSH_{P2}$.
Simultaneously the $NPSH_i$ value of the installation will decrease on account of the increasing h_k value, depending on the flow rate change. In this event the inequality

$$NPSH_{T2} > NPSH_{P2}$$

may not be valid and the pump may run with cavitations. Moreover,

(a) since the pump operation point is drifted from the point (*) to point (2), the pump will drain the power at point (2) instead of point (*), as it would be noticed from the curve P= f(Q). Thus power consumption and operational costs will increase.

(b) as a result of pump entering into cavitation, the operating point will be the region (3) instead of point (2) and the power drained will remain within the region (3). Besides, spare part consumption, repair and maintenance works and related production losses will increase on account of higher radial and axial forces because of cavitation wearing.

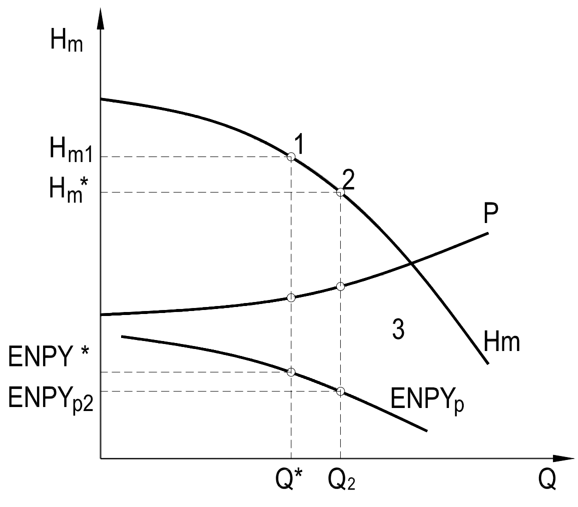
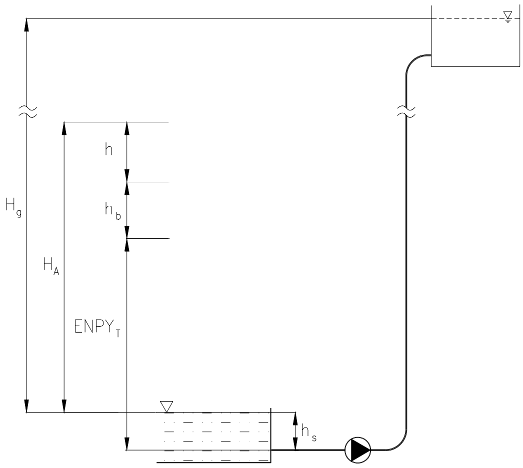


Figure 6.3 Pump Total Head erroneously Defined

Let's calculate the NPSH$_i$ values of few installations as example:

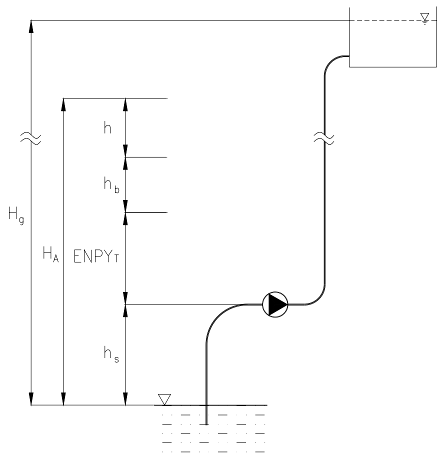
(a) NPSH$_i$ Calculation in an Installation with Positive Static Suction Lift



$$NPSH_i = H_A - (-h_s) - h_b - h$$

Figure 6.4 Installation with Positive Suction Lift

(b) NPSH$_i$ Calculation in an Installation with Negative Static Suction Lift



$$NPSH_i = H_A - h_s - h_b - h$$

Figure 6.5 *Installation with Negative Static Suction Lift*

6.1.5 Suction Specific Speed: S$_q$

The suction specific speed of a pump (also called suction specific number of revolutions, cavitation specific speed) is expressed with the formula:

$$S_q = n\frac{\sqrt{Q}}{H_{sv}^{3/4}}$$

where n is RPM of the pump (rpm), Q is the flow rate of the pump (m³/sec) and H_{sv} is calculated as:

$H_{sv} = \dfrac{p_A - p_b}{\rho g} - h_s$	p_A = atmospheric pressure (Pa) ($1\,bar = 10^5\,Pa$) p_b = fluid absolute vapor pressure (Pa) ρ = density of the liquid pumped (kg/m^3) g = gravitational acceleration ($9{,}81\,m/s^2$)

and the unit is (m).

In end suction centrifugal pumps the S_q varies between 150-200 rpm**, at the optimum operation point.** The average value of S_q may be considered as 154 rpm in single entry centrifugal pumps and as 219 rpm in double entry centrifugal pumps. Using these value as and above-mentioned formulas, the pump static suction lift necessary for cavitation free operation of the pump may be set approximately.

In addition to this approach, the literature contains also the Thoma cavitation factor which reads:

$$\sigma = \frac{NPSH_P}{H_m}$$

In preliminary calculations the maximum static suction lift of the pump may be set by the formula $h_{smax} = H_A - h_b - h - \sigma H_m$ by using the Thoma cavitation factor.
Thoma cavitation factor depends on the specific speed. Therefore it is a value depending on the net positive suction head, the pump flow rate and the RPM. Researchers (Wislicenus, Karassik and Watson) claim that, for centrifugal pumps the $NPSH_P$ value may be calculated by the following formulas, based on experimental results:

a) for single entry centrifugal pumps (Q:m^3/s, n: rpm)

$$NPSH_P = 12{,}2 \cdot 10^{-4} \cdot n^{4/3} \cdot Q^{2/3}$$

for n= 1000 rpm $NPSH_P = 12{,}2 \cdot Q^{2/3}$

for n= 1500 rpm $NPSH_P = 21 \cdot Q^{2/3}$

for n= 3000 rpm $NPSH_P = 53 \cdot Q^{2/3}$

b) for double entry centrifugal pumps (Q:m^3/s, n: rpm)

$$NPSH_P = 7{,}7 \cdot 10^{-4} \cdot n^{4/3} \cdot Q^{2/3}$$

for n= 750 rpm $NPSH_P = 5{,}25 \cdot Q^{2/3}$

for n= 1000 rpm $NPSH_P = 7{,}7 \cdot Q^{2/3}$

for n= 1500 rpm $NPSH_P = 13{,}22 \cdot Q^{2/3}$

for n= 3000 rpm $NPSH_P = 33{,}33 \cdot Q^{2/3}$

These formulas are for general reference and the $NPSH_P$ value must be provided from pump manufacturers. An $NPSH_P$ curve cavitation tests are formed for a pump.
The relation between the Thoma cavitation factor and the Suction Specific Speed is $S_q = \dfrac{n_q}{\sigma^{3/4}}$.

This formula is only valid for the design flow rate and the H_m values of the pump. It should be remembered that it can not be used for obtaining the $NPSH_P$ curve. The necessary net positive suction head is established by the following formula:

$$NPSH_P = \left(\frac{n\sqrt{Q}}{S_q} \right)^{4/3}$$

6.1.6 Calculating the $NPSH_P$ when Hot Water and Hydrocarbons are Pumped

The reference characteristics of pumps are measured by water in normal conditions. In case a pump delivers water with different temperature or another liquid the $NPSH_P$ characteristics change. If the water temperature is above 40^0C the $NPSH_P$ changes. High water temperature reduces the $NPSH_P$ value. US Hydraulics Institute proposes the utilization of the following correction graph for calculating the effect of the temperature and the type of the liquid on the $NPSH_P$.

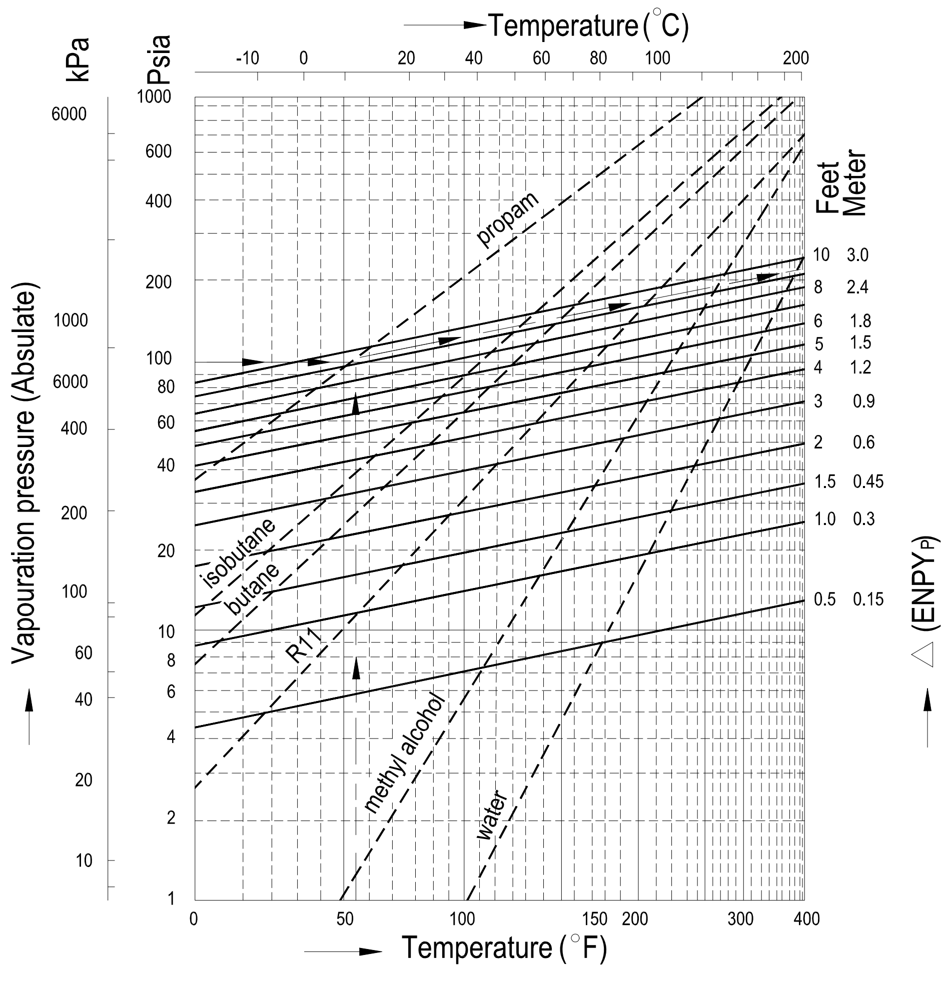


Figure 6.6 $NPSH_P$ change in case Pumps deliver Hot Water or Different Liquids

Two different families of curves exist in the said graph, for temperature correction and for liquid type. The difference to occur in the value of $NPSH_P$ is read on the vertical axis on the right side and the $NPSH_P$ value of the pump is modified for the value read on the graph.

6.1.7 Cavitation Control and Static Suction Lift Calculation

It is important to calculate the maximum static suction lift where a pump in an installation will run without cavitation. For this purpose, first the operation point and therefore the $NPSH_P$

value of the pump is defined. The maximum $NPSH_İ$ value necessary for a cavitation free operation of the pump will be

$NPSH_T = NPSH_P$

Since the NPSH value of the installation is defined as

$NPSHi = H_A - h_s - h_b - h$,

by writing $NPSH_P$ instead of $NPSH_i$ and making the necessary adjustments we obtain the maximum suction lift where the pump will operate cavitation free is

$$(h_s)\max = H_A - NPSH_P - h_b - h$$

6.1.8 Reference Plan

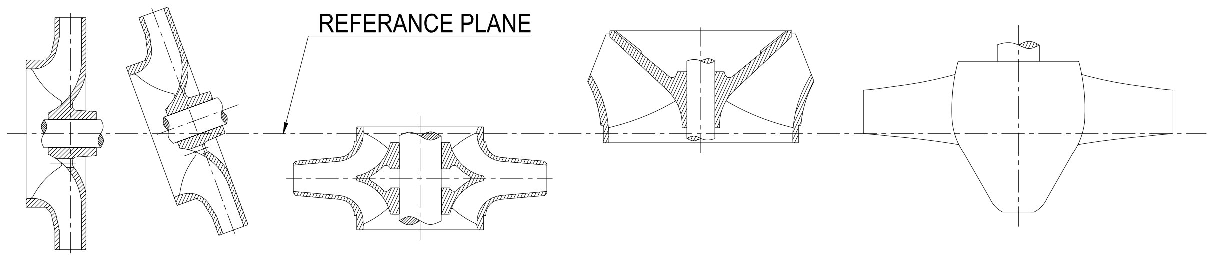


Figure 6.7 Reference planes for NPSH value

$NPSH_i$ value is expressed with respect to a plane defined as the NPSH plane (Figure 6.7). The NPSH reference plane is a horizontal plane crossing the center set by the circle passing from the outer edges of the impeller inlet. For vertical operation double entry pumps the NPSH plane is defined with respect to the upper impeller inlet.

6.1.9 Factors affecting the Cavitation

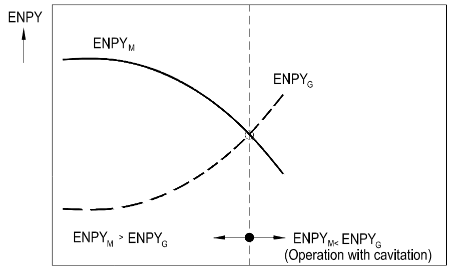
Parameters affecting the cavitation are clearly visible from the definition of the cavitation:

$$NPSHi = H_A - h_s - h_b - h_k \geq NPSH_P$$

(a) $NPSH_P$:T is totally related with the design of the pump and the operator has nothing to do in this respect. The low value of $NPSH_P$ means low risk for cavitation.

(b) Head losses in the suction pipe (h_k): The lower value increases the value of $NPSH_İ$. Therefore cavitation free operation requisite is achieved more easily. Consequently efforts must be exerted in order to minimize head losses in the suction line to the extent possible. For this purpose minor loss elements must be as much as possible avoided and elements with lower loss coefficient must be chosen. Head losses in the suction pipe are proportional to the square of the flow rate. Therefore head losses will rise in larger flow rates and the cavitation risk of the pump will increase. Since increasing the RPM of the pump will increase the flow rate, losses will increase on account of velocity in the suction pipe and the risk of cavitation will be higher.

(c) Vapor pressure of the liquid (h_b): is the function of the temperature of the liquid and the atmospheric pressure. Temperature increases augment the vapor pressure and therefore the risk of cavitation. Higher altitudes from the sea level decrease the vapor pressure, thereby increasing the value of $NPSH_I$. But if in this case the free surface of the suction tank is open to air the value of H_A decreases. The reduction in the H_A value being higher than the decrease in the h_b value, the risk of cavitation increases.



Air and other gases dissolved in the liquid are released before vaporization starts in the regions of the pump where the pressure decreases and may trigger the cavitation. Cavitation starts a bit later in gas freed liquids.

(d) Static Suction Lift (h_s): higher suction lifts increase the risk of cavitation.

(e) Pressure acting on the free liquid surface in the suction tank (H_A): increase in the pressure reduces the risk of cavitation.

(f) Speed: high values increase the cavitation risk.

(g) Flow rate : as high flow rates increase fluid velocity in the suction pipe, the risk of cavitation also increases.

(h) Gases and air dissolved in the air: higher amounts increase the risk of cavitation.

(ı) Liquid temperature: higher temperatures increase the risk of cavitation.

(i) Pressure in the suction reservoir: higher pressures decrease the risk of cavitation.

6.1.10 Cavitation Erosion

As a result of the wear out, holes and spongy appearances occur in the walls of the pumps running under the effect of the cavitation. The resistance of the pump material against cavitation depends on various parameters such as the composition of the material, the production method, the surface roughness, the thermal process etc.

It is possible to increase the lifetime of the impeller, in other words to lower the value of the $NPSH_P$, by choosing materials resisting to cavitation.

The following table lists several examples of materials by decreasing order of resistance to cavitation. The relative material loss is defined as the approximate loss of material for several substances with respect to cast iron (for the same liquid pumped). Besides these, very different and special materials are also used. The cavitation erosion decreases when the surfaces of the pump hydraulic elements are coated with rubber.

Material	Loss of material after 2 hours of operation under cavitation (mg)	Relative material loss
Special alloy stainless steel, cast	5	0,05
Aluminum bronze	10	0,1
Stainless steel, cast (AISI 304-1.4317)	20	0,25
Bronze and copper alloys (CuSn10)	80	0,55
Cast steel (GS 45)	110	0,6
Nodular cast iron (GGG 40)	170	0,8
Cast iron (GG 20)	225	1,0

Cavitation suffered Impeller

6.1.11 Measures against Cavitation

Measures to minimize the risk of cavitation, with respect to engineers designing the pump installation and the operator may be enumerated as follows:

(a) to choose a pump with lower $NPSH_P$ value

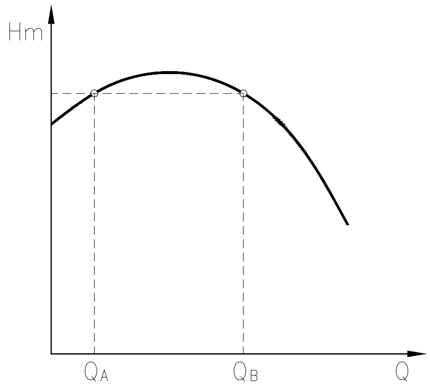
- Pump specially designed against cavitation
- Double suction pump
- Pump with low rpm

(b) to take measures related with the installation which will increase the $NPSH_I$ value

- To reduce the minor and friction head losses in the suction pipe (to choose elements with low local loss coefficient , to reduce flow velocity by large diameter suction pipes, to reduce the length of the suction pipe, to choose pipes with low surface roughness)
- To reduce the static suction lift
- To evacuate gases dissolved in the liquid
- To increase the suction pressure of the pump (to apply pressure on the suction tank or to by-pass the delivery pipe)
- To ensure adequate flow rate to the pump

6.2 Instability and Surge

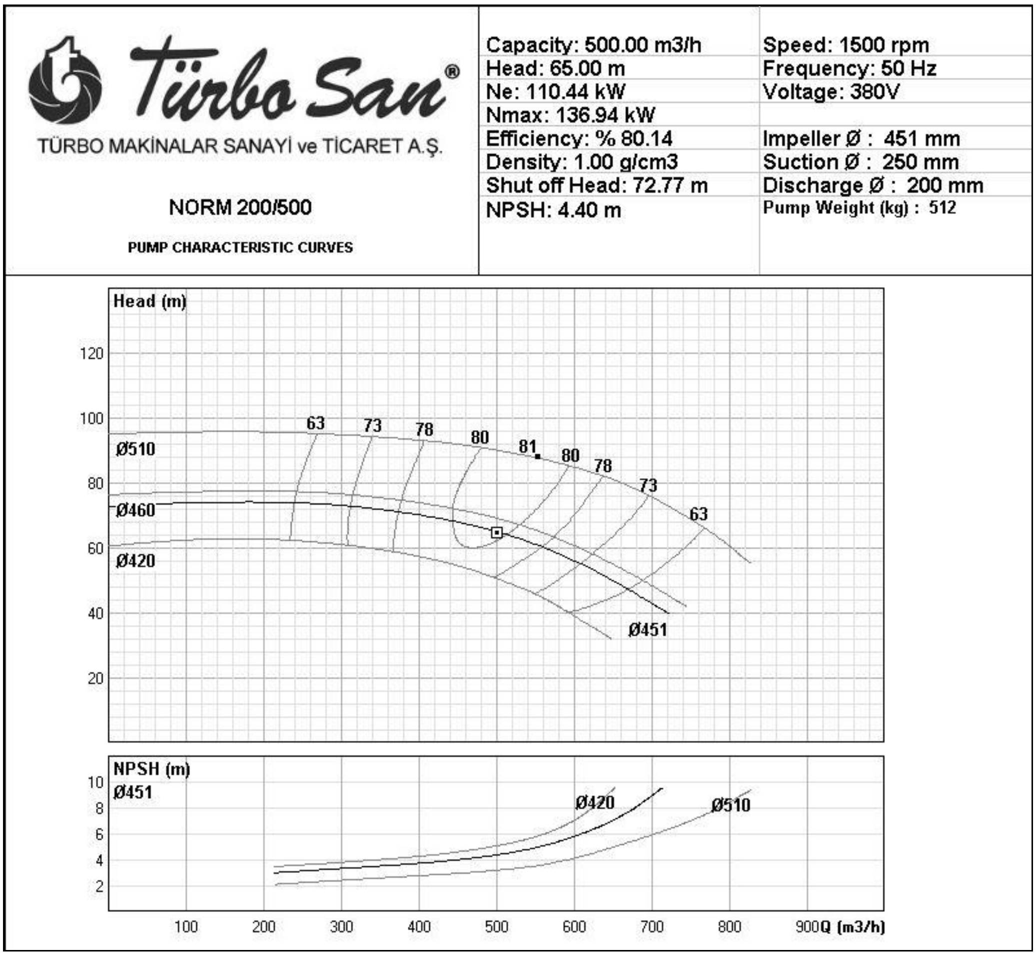
(a) Instability: instability occurs in connection with the H_m-Q characteristics when the pump runs under lower flow rate near to closed valve flow rate. In the event characteristics pass from maximum the pump operates by wavering between two flow rate values having the same pump head. This fact generates undesired flow fluctuations and vibrations in the system.



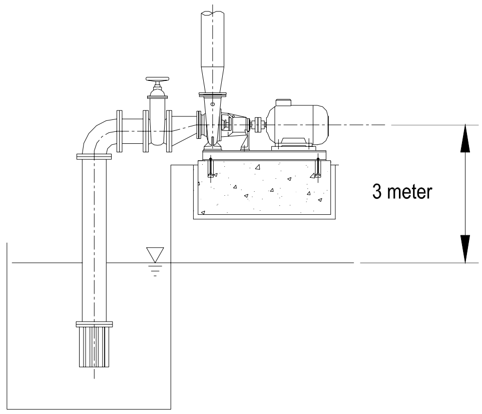
(a) **Surge:** in this case the flow direction of the liquid in the pumping installation changes alternatively. The liquid flows some time from the suction tank to the delivery tank, some time vice versa.

6.3 Example 1: Cavitation control of the pump:

Let’s check whether the NORM 200/500 pump whose characteristic curves, installation layout and the suction pipe characteristics are given hereinbelow, operates under cavitation or not:



Suction Line Data :



Flow rate (Q)	: 500 m³/h = 0,139 m³/s
H_m	: 65 mWC
Suction pipe diameter	: Ø300 = 0,3 m
Suction pipe length ℓ	: 6 m
Static suction lift h_s	: 3 m
Gate valve Ø300	: 1 piece, K= 0,25
Foot valve Ø400	: 1 piece, K= 2,25
Elbow 90° Ø300	: 1 piece, K= 0,45
Vertical cone Ø300 / Ø250	: 1 piece, K= 0,1
Liquid temperature	: 20°C
Altitude	: 100 m

In order to run cavitation free :

$$\mathbf{NPSH_i \rangle NPSH_P}$$

$NPSH_P$ is the catalog value of the pump. The selected pump being Türbosan NORM 200/500 (1500 rpm) the $NPSH_P$ for the operation point given is 3.1 meters.

$$\mathbf{NPSH_i = H_A - h_b - h_s -} \sum h$$

$$NPSH_i = 10{,}2 - 0{,}283 - 3 - \sum h$$

$$\sum h = \sum h_{friction} + \sum h_{\min orl}$$

$$\sum h = \lambda \frac{L}{D} \cdot \frac{V^2}{2g} + \sum K \cdot \frac{V^2}{2g} \qquad V = \frac{4 \cdot Q}{\pi \cdot D^2} = \frac{4 \cdot 0,139}{\pi \cdot 0,3^2} = 1,96 \text{ m/s}$$

$$\sum h = 0,025 \cdot \frac{6}{0,3} \cdot \frac{(1,96)^2}{2 \cdot 9,81} + (0,25 + 2,25 + 0,45 + 0,1) \cdot \frac{(1,96)^2}{2 \cdot 9,81}$$

$$\sum h = 0,10 + 0,6$$

$$\boxed{\sum h = 0,7 \text{ m}}$$

$$NPSHi = 10,2 - 0,283 - 3 - 0,7$$

$$\boxed{NPSH_i = 6,22 \text{ m}}$$

Consequently, as 6,22 > 3,1 meters the Norm 200/500 pumps runs without cavitation.

PART 7

MODIFYING PUMP PERFORMANCE CHARACTERISTISCS

Each pump has its characteristics curve and the pump operates on these curves. These curves are defined for a constant RPM. They do not change unless the RPM or the diameter of the pump impeller are modified (by machining). The same pump has different characteristics curves for different RPM.

As well known in the literature and as proven experimentally and theoretically, when pump performances are expressed by dimensionless coefficients, similar pumps have the same characteristics curves. In more clear terms, whatever the impeller diameters and the number of revolutions of similar pumps are, operation points defined with dimensionless coefficients remain on a single curve. Dimensionless coefficients are:

Flow coefficient (on the horizontal axis): $\phi = \dfrac{Q}{nD^3}$

Pressure coefficient (on the vertical axis): $\psi = \dfrac{gH_m}{n^2 D^2}$

Since efficiency is dimensionless, its definition and curve do not change. The power coefficient is defined as follows:

Power coefficient (on the vertical axis): $\kappa = \dfrac{N_e}{\rho_{n^3} D^5}$

The Figure 7.1 hereunder shows the H_m-Q characteristic curves of a Türbosan CEP 350/400 type pump at different RPM. There are 3 curves in the Figure. These characteristic curves are reduced to a single curve by using the above-mentioned dimensionless coefficients.

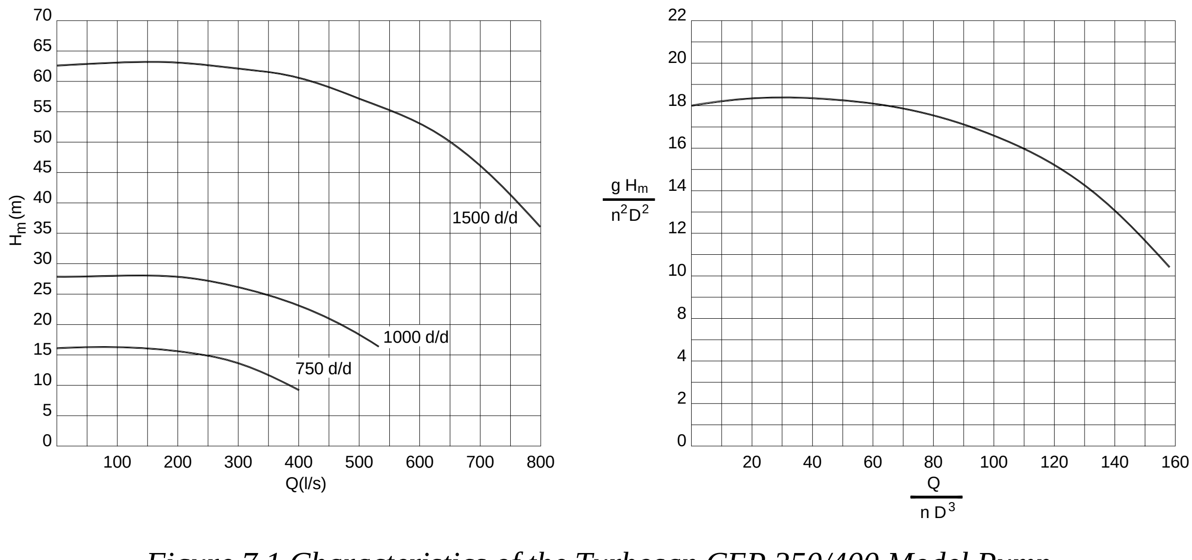


Figure 7.1 Characteristics of the Turbosan CEP 350/400 Model Pump

The pressure coefficient head of pumps operating at the same flow coefficient would be the same. Two similar pumps are running with the same flow coefficient, their operation point are called as similar. In this situation efficiencies would also be the same.

By using dimensionless coefficients as mentioned hereinbefore, we notice that the similar operation points of a pump are located on parabolas on the (gH-Q) plane.

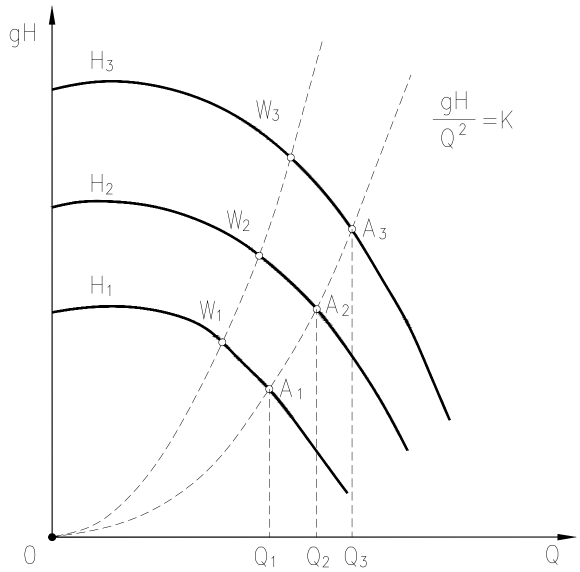


Figure 7.2 Similar Operation Points shown on (gH-Q) Plane

As it would be noticed from the definition of the dimensionless flow coefficient, since the impeller outlet diameter will not change in case the same impeller is used, this number will remain constant whenever the flow rate will increase commensurate with the rpm increase.

Here the term similar pumps means such pumps having the same specific speed value. In other terms, pumps whose all geometric dimensions are in constant ratio are similar pumps. A pump is similar to itself. The new pump created by lathing the impeller of a pump may be considered, to certain extent, similar to its original state.

The following subsections describe how the pump performance curves are modified when the number of revolutions of the pump and the diameter of the impeller is reduced by lathing.

7.1 Changing Pump Impeller Number of Revolutions

Whenever the RPM of a pump changes the new operation point of the pump can be found by using the above-mentioned dimensionless numbers. In case the former and the new operating points are expressed in dimensionless coefficients, the operating point corresponding to the new RPM of the pump is similar to the former operation point. In other terms

$$\frac{Q}{nD^3} = \frac{Q'}{n'D'^3} \qquad \frac{gH}{n^2D^2} = \frac{gH'}{n'^2D'^2}$$

In these expressions (′) indicates the new operation point. Since the impeller diameter remains the same

$$\frac{Q}{n}=\frac{Q'}{n'} \qquad Q'=\frac{n'}{n}Q \qquad \boxed{\frac{Q}{Q'}=\frac{n}{n'}}$$

$$\frac{H}{n^2}=\frac{H'}{n'^2} \qquad H'=\left(\frac{n'}{n}\right)^2 H \qquad \boxed{\frac{H}{H'}=\left(\frac{n}{n'}\right)^2}$$

As it would be noticed from these expressions the flow rate increases directly proportional with the RPM and the pump head increases proportional square of RPM.

Taking into consideration these similarity rules and by using the known characteristics curve of a pump known with respect to a definite number of revolutions, characteristics curves for different number of revolutions can be obtained. *****

The effective power of the pump at the new operation point is established by the same way by equalizing the power coefficients.

$$\frac{N_e}{\rho n^3 D^5}=\frac{N_e'}{\rho n'^3 D'^5}$$

Taking into consideration the density of the fluid and the constant impeller diameter of the pump,

$$\frac{N_e}{n^3}=\frac{N_e'}{n'^3} \qquad N_e'=\left(\frac{n'}{n}\right)^3 N_e \qquad \boxed{\frac{N_e}{N_e'}=\left(\frac{n}{n'}\right)^3}$$

Thus the effective power at the new operating point increases with the third power of the RPM.

7.2 Changing the Diameter of the Impeller

The effect of the pump impeller diameter on the pump performance curves may be found with the same approach described in the previous sub-section. (n=constant)

$$\frac{Q}{nD^3}=\frac{Q'}{n'D'^3} \qquad \frac{gH}{n^2D^2}=\frac{gH'}{n'^2D'^2} \qquad \frac{N_e}{\rho n^3 D^5}=\frac{N_e'}{\rho' n'^3 D'^5}$$

In these expressions (′) indicates the operating conditions at the new operation point. Since the number of revolutions of the pump remain the same

,

$$\frac{Q}{D^3}=\frac{Q'}{D'^3} \qquad Q'=(\frac{D'}{D})^3 Q$$

$$\frac{H}{D^2}=\frac{H'}{D'^2} \qquad H'=\left(\frac{D'}{D}\right)^2 H$$

we obtain $\frac{N_e}{D^5}=\frac{N_e'}{D'^5} \qquad N_e'=\left(\frac{D'}{D}\right)^5 N_e$

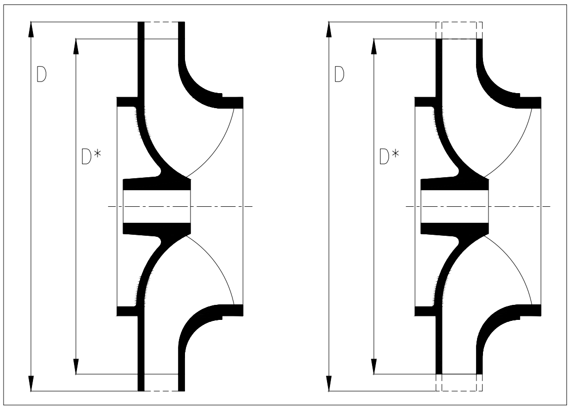


Figure 7.3 Lathing the Impeller of a Multi-stage Pump

If the pump impeller is centrifugal type, then there are some changes in the formulas. In centrifugal pumps, the width of the impeller close to the impeller outlet is constant. The lathing process reduces the impeller diameter in this region. According to similarity theory the flow rate number is obtained by dividing the flow rate value with the amount of (outlet area x speed). Since in the impeller of the centrifugal pump the outlet width is constant in the vicinity of the outlet, the area equal to (outlet width x impeller outlet diameter x 3,14169) substitutes the impeller outlet diameter. Therefore the flow rate value of the new position when the impeller of a full centrifugal pump is lathed is obtained by the following formula:

$$Q' = (\frac{D'}{D})^2 Q$$

In general, the efficiency of the pump with reduced impeller diameter is a bit lower than that of the original impeller. In multi-stage pumps only the blades should be lathed (Figure 7.3). Moreover in order to calculate the effect of the impeller diameter lathing on the operation point of the pump the notation defined in Figure 7.4 and the following formula are used:

$$\boxed{\sqrt{\frac{a}{b}} = \frac{D}{D'}}$$

Besides, the Figure 7.5 shows the performance curve of a Turbosan Norm 250/400 type pump with different impeller diameters.

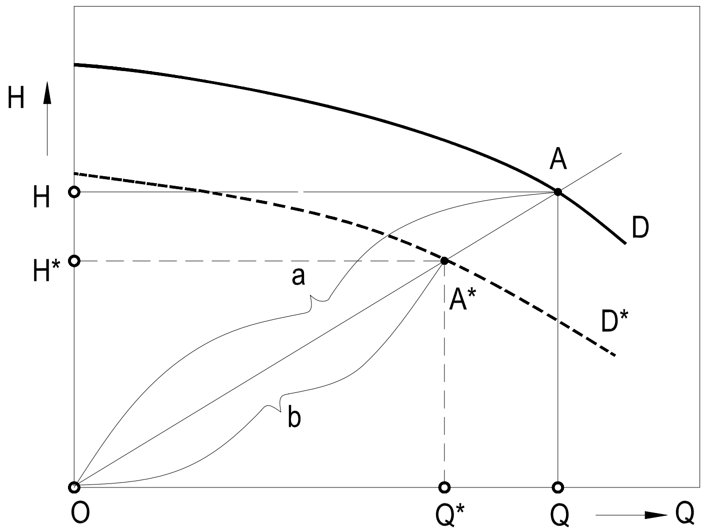


Figure 7.4. Finding the New Operation Point by reducing the Impeller Diameter

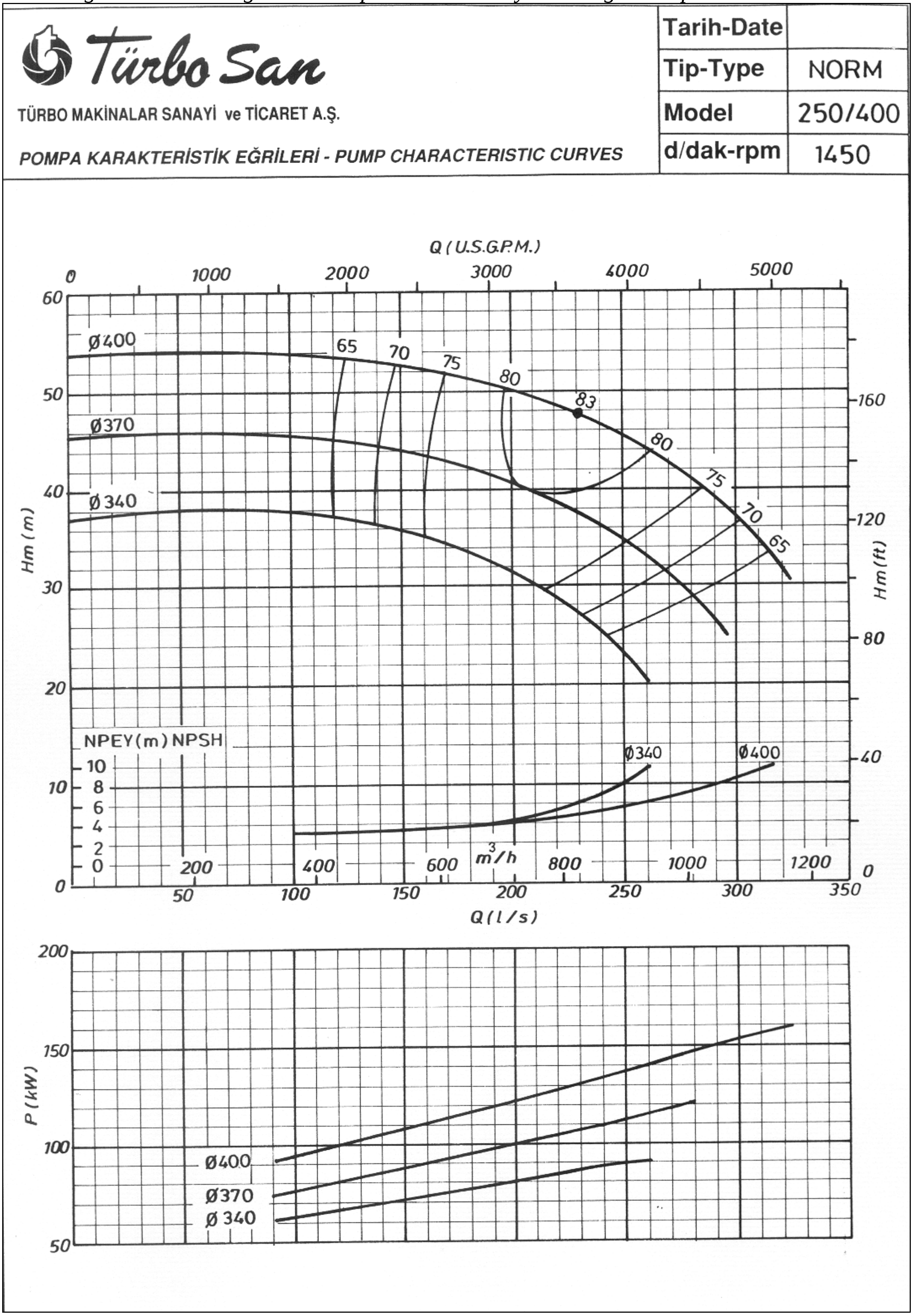


Figure 7.5 The Performance Curve of a Pump with Constant Revolution but Different Impeller Diameters

PART 8

PUMPING VISCOUS LIQUIDS AND PUMP SELECTION

8.1 Viscosity Particulars

Viscosity may be defined as the generated resistance to relative movement between adjacent layers. Viscosity has two different definitions as dynamic (absolute) and kinematic.

(a) Dynamic Viscosity (μ):
The dynamic viscosity is defined as the ratio of the shear stress acting on fluid particle within the flow area to the velocity gradient acting on that fluid particle. For Newtonian fluids like air, water this ratio is constant. The unit of the dynamic viscosity is ($N\ s\ /m^2$ = Pa.s). In literature the term "centipoises" (cP) is used as the viscosity unit and corresponds to 10^{-3} Pa.s (1 cP=10^{-3} Pa.s). In liquids the dynamic viscosity decreases by the temperature whereas it increases in gases. The dynamic viscosity of water and some other liquids is given in the Annex.

(b) Kinematic Viscosity (ν)
Is defined as the ratio of the dynamic viscosity to the density of the fluid and it has a unit of m^2/s. The kinematic viscosity of water at 20^0C temperature is 1,002 .10^{-6} m^2/s. In literature the term "centi-Stokes" (cSt) is also used as kinematic viscosity unit and corresponds to 10^{-6} m^2/s.
(1 cSt=1 mm^2/s=10^{-6} m^2/s)

The relation between dynamic and kinematic viscosity is: $\nu = \dfrac{\mu}{\rho}$ and the dynamic viscosity of water at 20°C is 1 cP and the kinematic viscosity is 1 cSt.

Viscosity has an influence on the flow rate the least and the efficiency the most. Motor powers are substantially increased on account of decreasing efficiency.

Using centrifugal (rotordynamic) pumps is not economically and technically suitable for use for liquids whose kinematic viscosity is above 600 cSt, since the efficiecny falls down and the motor power increases substantially. Volumetric pumps (with positive displacement) should be used instead of centrifugal pumps.

If the kinematic viscosity of the fluid is below 20mm^2/s (cSt) the flow rate and the pump head do not change unless the RPM is changed (see Figure 8.2).

8.2 Conversion of Characteristics Curves of Centrifugal Pumps for Pumping Viscous Liquids

Unless specified otherwise, pump characteristics shown in catalogs are those measured with water. This sub-section will explain how the new characteristics of a pump will be obtained in the event the pump will drain a liquid more viscous than water. The change in performance characteristics is schematically shown in Figure 8.1. Different approaches in the literature are summarized as follows:

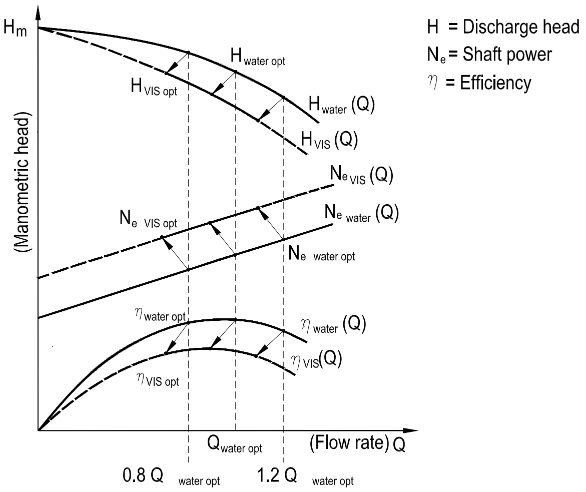


Figure 8.1 Effect of Viscosity to the Pump Performance

Approach Proposed by the American Hydraulics Institute:

This method has been experimentally tested by using volute type pumps with impeller diameters varying between 50 – 200 mm and with specific speed $\eta_q = 20(l/\min)$. The said method has been included in "ISO 9906 - Rotordynamic pumps – Hydraulic performance acceptance tests- Grades 1 and Grades 2" and defined as a standard method. The same standard has been included in Turkish Standards as "ISO 9906 - Rotordynamic pumps – Hydraulic performance acceptance tests- Grade 1 and Grade 2".
In this approach, performance curves experimentally obtained by using water are transformed for the viscous fluid for the 0.6, 0.8, 1.0 and 1.2 times the optimum flow rate (Q_{opt}). Therefore the fundamental principle of the method is to find the coefficients for flow rate, pump head and efficiency corrections. There are six correction factors in the method: one for each flow rate and efficiency and one for each of the four flow rates as above-mentioned for the pump head. Following are the steps to find these correction factors:

(1) Q_{opt}, H_{opt} and η_{opt} values relating to the optimum operation point of the pump are obtained from the pump characteristics for water.
(2) The multiples of the Q_{opt} value for 0.6, 0.8 and 1.2 ($Q_{0.6}$, $Q_{0.8}$, $Q_{1.2}$) are calculated and the values $H_{0.6}$, $H_{0.8}$ and $H_{1.2}$ and $\eta_{0.6}$, $\eta_{0.8}$ and $\eta_{1.2}$ values are defined from the performance characteristics of the pump for water.
(3) The Q_{opt} value is read on the horizontal line of Figure 8.2 and a vertical line is drawn from that value. The same diagram indicates also the pump head curves (lines). The intersection point of the vertical line drawn from the Q_{opt} value with the H_{opt} pump head is found.
(4) Viscosity lines are also available on the same graph. A horizontal line is drawn from this intersection point and its intersection with the viscosity value of the viscous fluid is found.
(5) From this point of intersection a vertical line is drawn and the correction factors k_Q, k_η, $k_{H0.6}$, $k_{H0.8}$, k_{Hopt}, $k_{H1.2}$ are obtained from the upper graph.
(6) The change of the pump head and efficiency values by the flow rate, in case the pump will drain viscous fluid are set by using the flowing formulas:

$$Q_{vis} = k_Q \, Q_{water} \qquad H_{vis} = k_H \, H_{water} \qquad \eta_{vis} = k_\eta \, \eta_{water}$$

Here the (vis) index shows the viscous fluid and, (water) index shows the water.

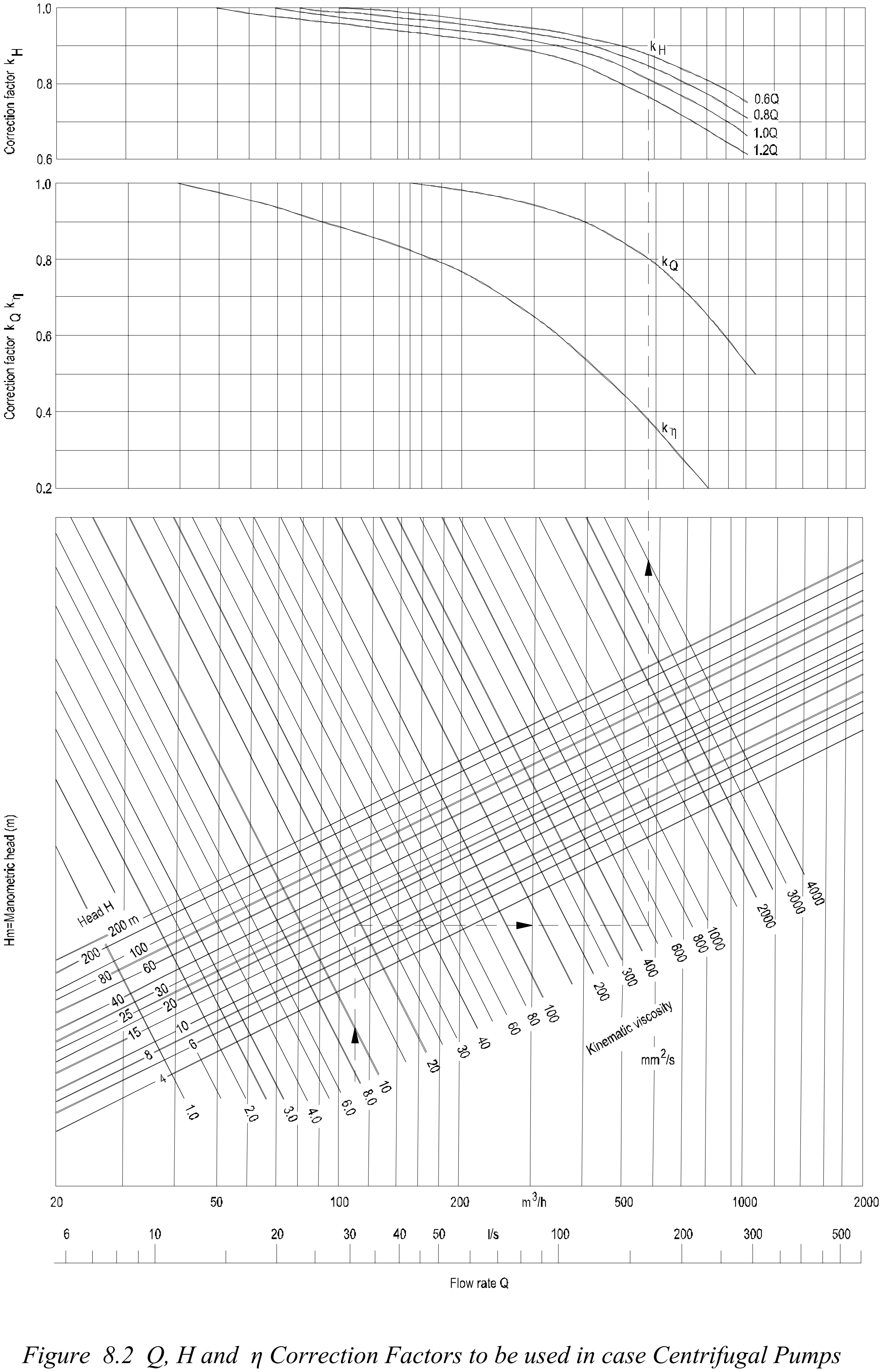


Figure 8.2 Q, H and η Correction Factors to be used in case Centrifugal Pumps mentioned in ISO 9906 drain Viscous Liquids

8.3 Choosing the Right Pump ensuring the Required Flow Rate and Pump Head in Case Viscous Fluids are mupmed

The purpose of this sub-section is to explain how to choose the pump to pump a viscous liquid, by using the characteristics of pumps whose characteristics are given with respect to water.

The pump head of the pump for viscous fluid for the required operation point is H_{vis}, the flow rate is Q_{vis} and the efficiency is η_{vis}. Our aim is to find out the equivalents of these values when the pump will pump water. This operation consists of the following steps:

1. The value of Q_{vis} is found on the horizontal axis of Figure 8.2 and a vertical line is drawn from that value. The same graph indicates also the pump head curves (lines). The intersection point of the vertical line drawn from the Q_{vis} value with the H_{vis} pump head is found.
2. From this point of intersection a horizontal line is drawn and the intersection point with the viscosity value of the viscous fluid.
3. From this point of intersection a vertical line is drawn and the correction factors k_Q, k_η, $k_{H0.6}$, $k_{H0.8}$, k_{Hopt}, $k_{H1.2}$ are obtained from the upper graph.
4. The change of the pump head and efficiency values by the flow rate, in caseof viscous fluid are set by using the flowing formulas:

   $Q_{water} = Q_{vis} / k_Q \qquad H_{water} = H_{vis} / k_H$
5. The pump is selected from the catalog. The efficiency value from the catalog is calculated by the following formula and the shaft power is set.

   $\eta_{vis} = \eta_{water}\ k_\eta$

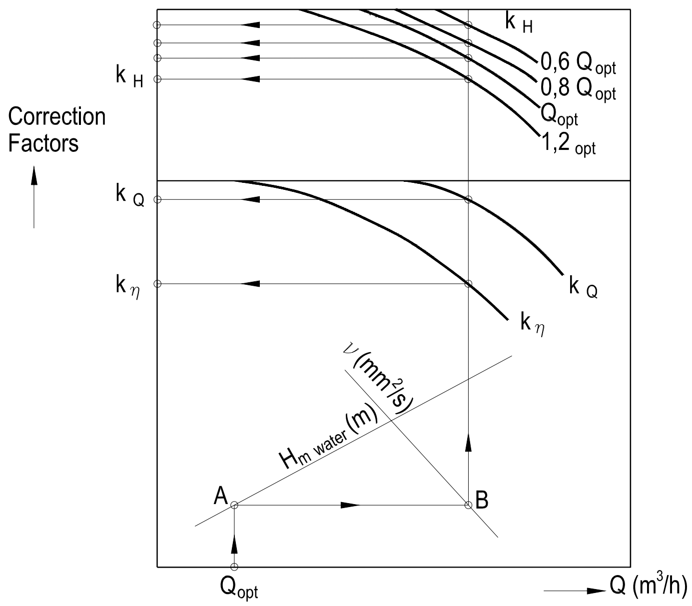


Figure 8.3 Determining Viscosity Correction Factors

According to these values, the operation curve for viscous liquid is obtained on the pump performance curve set for pumping water.

The new performance curve to be obtained by this method is an approximate working curve. Real values may only be obtained through experiments with the viscous liquid.

8.4 EXAMPLES

8.4.1 Example 1 – Pump Characteristics for Water Converted for Pumping Viscous Liquid

Calculating the pump performance for a viscous liquid:

a) The operation values of a centrifugal pump (water at 20°C temperature) :
   Flow rate Q : 300 m³/h
   Pump head Hm : 40 mSS
   Pump Efficiency η : %80
   Speed n : 1500 rpm

   The pump chosen according to these data is the centrifugal pump with horizontal shaft Turbo-Norm 150/400.

b) The new performance curve of the pump will be obtained in accordance with the viscous liquid values given hereunder. The pump rpm is assumed to be constant. The kinematic viscosity of the mineral oil SAE 20 at 20°C temperature is ν : 200 mm²/s, density ρ : 900 kg/m³ (0,9 gr/cm³).
   Using the table in Figure 8.2 we find the correction values k_η, k_Q, k_H for the viscous liquid. Q_{vis}, H_{vis}, η_{vis}, P_{vis} values for 4 different operation points for the viscous liquid are calculated and the performance curve is drawn (Figure 8.4). Said values are given hereunder.

Liquid	**Units**		**0,6**	**0,8**	**1**	**1,2**
Water	Q (m³/h) flow rate		180	240	300	360
	H_m (m) pump head		47	44	40	33
	η (%) productivity		68	77	80	73
	Ne (kW) power drawn by the pump		33,8	37,3	41	44,3
Correction factors	Q_{water} = 300 m³/h	k_Q	0,97			
	H_{msu} = 40 mSS	k_η	0,70			
	ν = 200 cSt	k_H	0,98	0,96	0,94	0,91
Viscous fluid	Density ρ_{vis} (gr/cm³)		0,9 (900 kg/m³)			
	$Q_{vis} = k_Q \cdot Q_{water}$ (m³/h)		174,6	232,8	291	349,2
	$H_{mvis} = k_H \cdot H_{mwater}$ (m)		46	42,2	37,6	30
	$\eta_{vis} = k_\eta \cdot \eta_{su}$ (%)		47,6	53,9	56	51,1
	$Ne_{vis} = \dfrac{Q_{vis} \cdot Hm_{vis} \cdot \rho_{vis}}{102 \cdot \eta_{vis} \cdot 3{,}6} (kW)$		41,3	44,6	47,9	50,2
n = 1500 rpm = constant						
Water density = 1000 kg/m³ = 1 kg/dm³ = 1 gr/cm³						
For water at 20°C $\mu = \nu$ = 1 cP = 1 cSt						

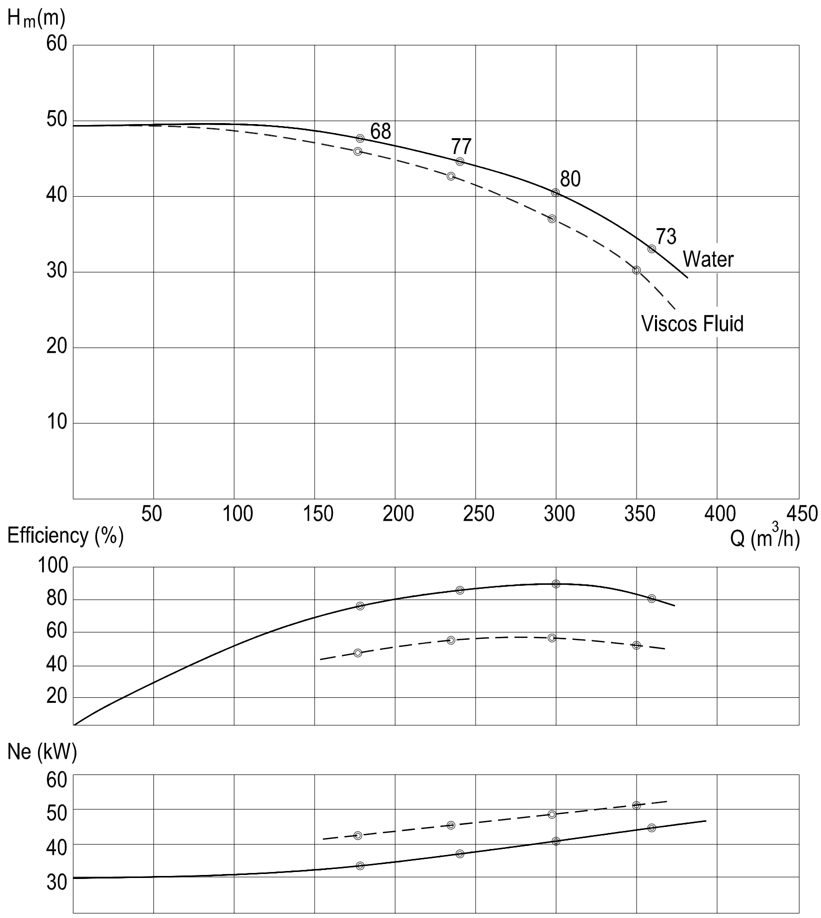


Figure 8.4 Türbosan Norm 150/400 Pump Characteristic Curves and their Reduction for a Fluid with a Viscosity $\nu = 200\ cSt$

8.4.2 Example 2 : Choosing the Pump for the Viscous Liquid

1- Data relating to pumping viscous liquid is known. Now first we have to calculate the pump that will drain water under the same conditions. In order to find out the necessary correction coefficients from the Figure 8.2 a vertical line from the Q value of the graph is drawn and moving to the right or left from the intersection point of the discharge curve H, its intersection with the viscosity curve is set. Then the intersection points of the vertical line from this second point with the curves K_H, K_Q and $K\eta$ are found.

Q= 170 m³/h, H= 32 m, Viscosity= 210 cSt. Specific weight ρ=0,9 gr/cm³. Following are the values obtained from the Figure 8.2:

K_Q = 0,94 $K\eta$ = 0,64 K_H = 0,92 (for Q= 1)

The values of the pump for water must be as follows:
Q_{water}= 170/0,094 = <u>181 m³/h</u> H_{water}= 32/092 = <u>34,7 m</u>

The size of the pump to be purchased must be defined pursuant to these values.
Now let's choose the pump which will drain a liquid with kinematic viscosity 210 cSt, specific weight 0,9 gr/cm^3 with a flow rate of Q=170 m^3/h and whose pump head would be 32m.

For this purpose we must reduce these values of the pump to the state where the liquid to be pumped is water. The following values are obtained from the table of Figure 8.2 for Q=170 m^3/h:

$$k_Q = 0{,}94$$
$$k_\eta = 0{,}64$$
$$k_H = 0{,}92$$

Therefore in case this pump would drain water the values would read:

$$Q_{water} = 170 / 0{,}94 = 181 \quad \text{m}^3\text{/h}$$
$$H_{water} = 32 / 0{,}92 = 34.7 \ \text{m}$$

If the efficiency in case of operation with water was 79%, the efficiency when operating with the viscous liquid would be 0,79 x 0,64 = 0,50 and the motor power would be :

$$Ne = \frac{Q_{vis} \cdot H_{vis} \cdot g}{102 \cdot 3{,}6 \cdot \eta_{vis}} = \frac{170 \cdot 32 \cdot 0{,}9}{102 \cdot 3{,}6 \cdot 0{,}5} = 26{,}7 \ \text{kW}$$

PART 9

BOOSTER SETS

This section supplies descriptions for booster sets, calculating the minimum tank volume for submersible pumps and the plans for typical pumping stations in order to help the operator and the project designer.

9.1 Booster Set

Those are with single or multiple pump systems erected in order to meet the water requirements of installations and buildings (Figure 9.1).

Fields of utilization of booster sets:

a) Supply potable water
b) Garden irrigation
c) Water supply vision for fire extinguishing systems
d) Supply process water
e) Villas, apartment houses and residences
f) Hospitals, schools, offices
g) Hotels and holiday villages

Figure 9.1 Booster Set

9.1.1 Defining the Flow Rate of Booster Sets For Domestic Water Use (Q)

Following is the practical way to calculate the necessary flow rate :

$$Q = \frac{D_s \cdot B \cdot Q_g \cdot K}{1000}$$

Q: Booster set flow rate (m^3/h)

D_s : Number of apartments (number of families)

B: Number of persons residing in one apartment (4 persons) (in general 4 persons live in an apartment in Turkey)

Q_g : Daily water consumption per capita (liters/day) (average consumption in Turkey is advised to read 150-200 liters/day. This value changes according to living standards) (Table 9.4)

K: Simultaneous utilization factor (see Table 9.1)

Number of houses	Simultaneous utilization factor (K)
Up to 4 apartment house	0,65
5-10 apartment house	0,45
11-20 apartment house	0,40
21-50 apartment house	0,35
51-100 apartment house	0,30
More than 100 apartment house	0,25

Table 9.1

For example flow rate necessary for a residence building of 22 flats is calculated practically as follows:

$$Q = \frac{22 \cdot 4 \cdot 150 \cdot 0{,}35}{1000} = 4{,}62 \ \text{m}^3/\text{h}$$

For a compound of i.e. 120 houses the quantity of water necessary is

$$Q = \frac{120 \cdot 4 \cdot 150 \cdot 0{,}25}{1000} = 18 \ \text{m}^3/\text{h}$$

This value is considerably higher than the first example. The system may be chosen by two different ways:

1st choice: system consisting of 2 pumps of 18 m^3/h flow rate each with a lower value of ($H_{m\ alt}$).
2nd choice: system consisting of 3 pumps of 9m^3/h flow rate each (DIN 1988)

In both options one pump is a stand-by pump in the system.

9.1.2 Calculating the Booster Set Pressure (H_m)

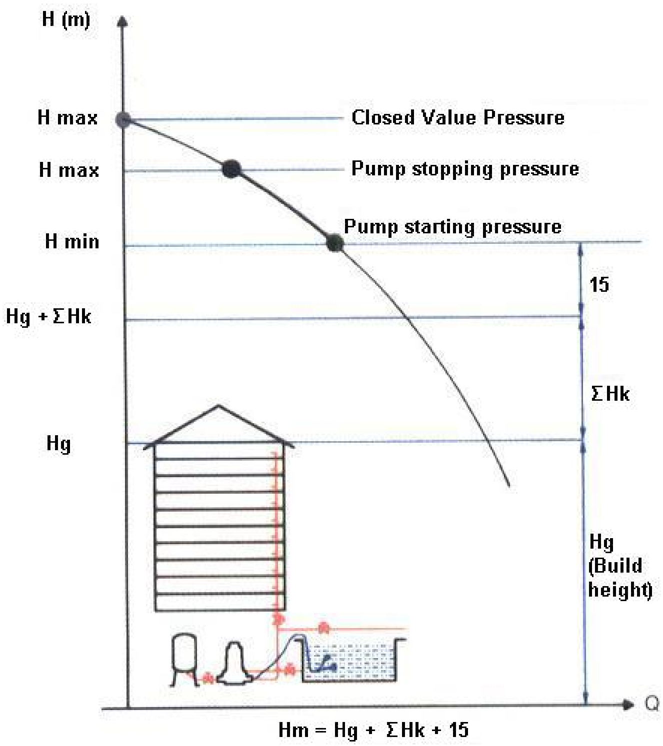


Figure 9.2 Booster Sets *Pump Head*

The booster sets pressure may be calculated with the following formula:

$$H_m = h_g + h_k + 15 \ \ mWV + 7{,}5 \ \ mWC + h_{special}$$

$H_m (H_{\min})$: Total pump head (mWC), the pump operation pressure called minimum pressure.

h_g = coefficient x 3m
h_k = total pressure losses in the installation (mWC)
$h_k = H_g \cdot (0{,}20 - 0{,}25)$
Generally accepted as a value between the 20-25% of the building height.
15 mWC = flow pressure
7,5 mWC = loss of pressure in water meter (if installed)
$h_{special}$ = pressure losses pertaining to special equipment (water filters, whirlpool etc.) if available

Besides, it is advisable to take the $H_{\max}$ value (the maximum pressure of the booster set– cutoff pressure) as $H_{\max} = H_{\min} + (10m)$.

Let's calculate the pressure of a booster set necessary for a 10 storey residence building including the basement.

Each house is equipped with a water-meter.

$H_m = 30 + (30 \cdot 0{,}25) + 15 + 7{,}5$
$H_{m\min} = 60$ mWC pump operation pressure – minimum pressure
$H_{m\max} = H_{m\min} + 10 \ m$
$H_{m\max} = 60 + 10 = 70$ mWC

Besides the Table 9.2 may also be referred to for the selection of a booster set for a residence building.

Number of storeys	Booster set pressure Hm (bar)	Number of flats	Flow rate Q (m^3/h)	Number of flats	Flow rate Q (m^3/h)
1	2,4	1	0,4	31	6,5
2	2,8	2	0,8	32	6,7
3	3,1	3	1,2	33	6,9
4	3,5	4	1,4	34	7,2
5	3,8	5	1,5	35	7,4
6	4,1	6	1,6	36	7,6
7	4,5	7	1,7	37	7,8
8	4,8	8	2	38	8
9	5,2	9	2,2	39	8,2
10	5,5	10	2,3	40	8,4
11	5,9	11	2,4	41	8,6
12	6,2	12	2,5	42	8,8
13	6,6	13	2,7	43	9
14	6,9	14	2,9	44	9,3
15	7,3	15	3,2	45	9,5
16	7,6	16	3,4	46	9,7
17	7,9	17	3,6	47	9,9
18	8,3	18	3,8	48	10,1
19	8,6	19	4	49	10,3
20	9	20	4,2	50	10,5
21	9,3	21	4,4	51	10,7
22	9,7	22	4,6	52	10,9
23	10	23	4,8	53	11,1
24	10,4	24	5	54	11,3
25	10,7	25	5,3	55	11,6
26	11,5	26	5,5	56	11,8
27	11,4	27	5,7	57	12
28	11,7	28	5,9	58	12,2
29	12,1	29	6,1	59	12,4
30	12,4	30	6,3	60	12,6

Table 9.2 Practical Booster Set Selection According to Number of Storeys and Flats

9.1.3 Determining Pressure Limits (Zoning)

In pipe installations for domestic water the static water pressure should never exceed 50 mWC at any point of the installation.
Otherwise pressure reducers must be used or zoning must be applied to the installation (Figure 9.3).
Generally zoning is made by 2 separate booster sets in buildings with more than 10 storeys. In such event individual booster sets must be selected for each area (Figure 9.3a)
In case there are more than two pressure zones in the building one single booster set may be used. However for the minimum pressure zone a “pressure reducer” must be mounted (Figure 9.3b). The initial investment cost is low for a pressure reduced system but the power consumption is high.

As an example let's choose a booster set for a residence building of 15 storeys and 60 flats:
In buildings with more than 10 storeys zoning must be made with two separate booster sets. Accordingly first, the apartment house is divided into two pressure zones.

1st Zone (Minimum pressure zone) = storeys 1-8 ..... 32 flats
2nd Zone (Maximum pressure zone) = storeys 9-15 .... 28 flats

<u>1st Zone</u>

Calculating the necessary pressure (H_m)

$$H_{m\min} = h_{geo} + h_k + 15m + 7{,}5\ \ m$$
$$H_{m\min} = (8 \cdot 3) + (24 \cdot 0{,}25) + 15 + 7{,}5\ \ m$$
$$H_{m\min} = 52{,}5\ \ m$$
$$H_{m\max} = 52{,}5 + 10$$
$$H_{m\max} = 62{,}5\ \ m$$

Calculating the necessary flow rate (Q)

$$Q = \frac{32 \cdot 4 \cdot 150 \cdot 0{,}35}{1000}$$
$$Q = 6{,}72 m^3 / h$$

Characteristics of the pump required for the first zone should be:

$$Q = 6{,}72\ \ m^3 / h$$
$$H_m = 52{,}5\ \ mWC$$

<u>2nd Zone</u>

Calculating the pressure necessary (Hm)

$$H_{m\min} = (15 \cdot 3) + (45 \cdot 0{,}25) + 15\ \ m + 7{,}5\ \ m$$
$$H_{m\min} = 78{,}75\ \ mWC - 78\ \ mWC$$
$$H_{m\max} = 78 + 10 = 88\ \ mWC$$
$$H_{m\min} = 78\ \ mWC$$

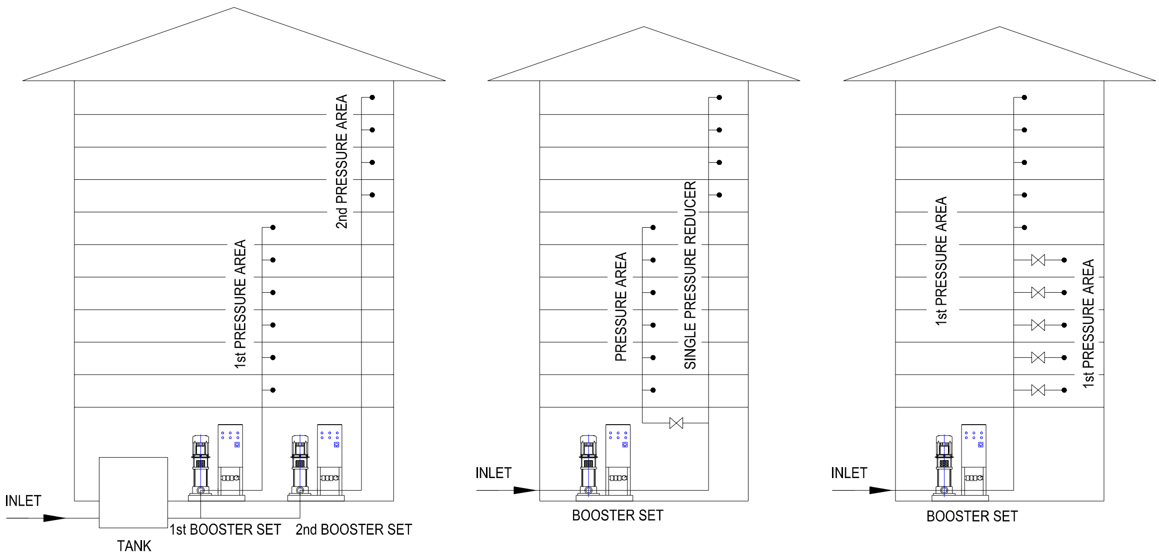
Calculating the necessary flow rate (Q)

$$Q = \frac{28 \cdot 4 \cdot 150 \cdot 0{,}35}{1000}$$
$$Q = 5{,}88\ \mathrm{m^3/h}$$

Characteristics of the pump required for the second zone should be:

$$H_m = 78\ \ \mathrm{mWC}$$
$$Q = 5{,}8\ \ \mathrm{m^3/h}$$

<u>Note :</u> As in the first two storeys the pressure is more than 30m (3 bar) it is advisable to fix a pressure reducer in order to avoid possible problems in the hot water circuits of water heaters.



(a) *(b)* *(c)*

Figure 9.3 Determining Pressure Limits (Zoning)

9.1.4 Determination of the Booster Set Connection Type

The main criteria for determining the connection type is the speed of the fluid (water). The fluid speed is recommended to be between 1 and 3,5 m/sec. If the speed of the fluid is selected within this range:

a) The noise formation in the installation is prevented to great extent.
b) Excessive pressure increases called water hammer may not happen.
c) As long as the pipe diameter will decrease losses will increase and the pump head will go up. Thus the pump motor will also become bigger and the energy expenses will increase. All selections and calculations must be envisaged for optimum power consumption.

The Figure 9.4 shows Türbosan MAX and MCK series pumps transformed into booster set.

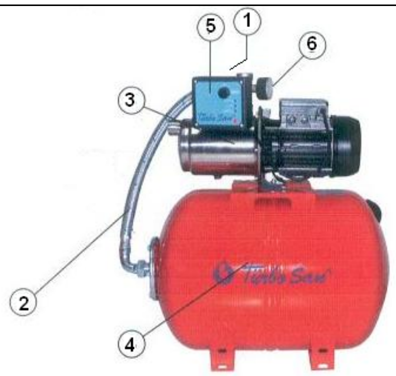
	**Components of MAX and MCK Booster Set:** 1- 5-way connection 2- Flexible hose 3- Pump 4- Tank (50 lt) 5- Pressure switch 6- Manometer

Figure 9.4 Booster Set with Türbosan MAX and MCK Series Pump

Besides, the Figure 9.5. shows an exemplary booster set installation example.

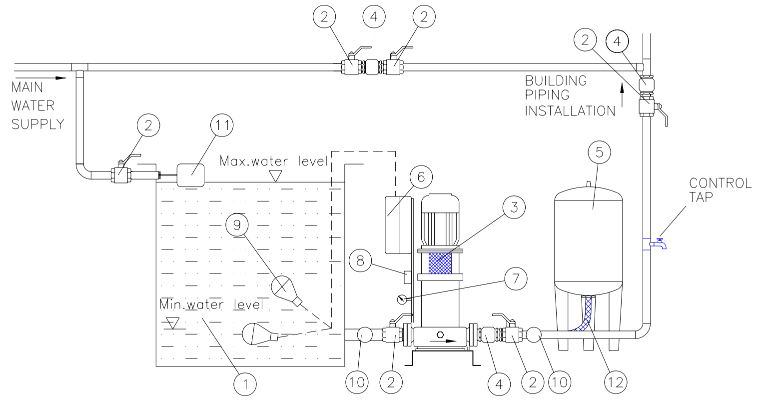
	**Components of a Booster Set Installation:** 1- Water tank 2- Ball valve 3- Pump 4- Check valve 5- Expansion tank 6- Control panel 7- Pressure switch 8- Manometer 9- Float 10- Suction discharge collector 11- Level Float 12- Flexible hose

Figure 9.5 Türbosan Booster Set Installation Example

9.1.5 Components Necessary for Booster Sets System

1- Pump : systems with 1, 2, 3 pumps
2- Panel : the control panel arranges the functions of the booster set and protective automation system preventing dry running
3- Expansion Tank: membrane type, odor free, full hygienic, placed on the pressure line.
4- Chassis : monobloc with rubber footings (to absorb vibrations)
5- Valves : ball valves in suction and discharge lines
6- Check valve : to prevent backflow to discharge line
7- Pressure switch : for minimum and maximum pressure values
8- Level switch: to shut off automatically the booster set when the water level in the tank drops and to restart
9- Impurity strainer :
10- Collector : piping installation in multi pump booster sets connecting the suction and discharge line (installation) to the booster set
11- Expansion Joint: if necessary for the system
12- Pressure reducing valves and safety vents : if necessary for the system
13- Single pump booster set : one check valve or suction bottom valve and a connector with valve at the outlet of the pump

9.1.6 Membrane Type Expansion Tank

- Expansion tanks with adequate volume should be used in the booster set.
- Tank membranes should be replaceable, inodorous in water with complete hygienic characteristics.
- Outer surfaces of tanks should be corrosion resistant, epoxy electrostatic furnace dried painted.
- Tanks should be compressor filled with dry air or nitrogen with adequate pressure. The tank pressure should be 10% lower than the operation pressure (Halt) (operation pressure x 0,90)
- Tanks should be connected to the pressure collector of the booster set or to a suitable part of the installation with a valve or preferably with a flexible hose.

- From the standpoint of system economy and useful lifetime, it is advisable to use large volume tanks. Tanks smaller than 300 lt. should not be used.
- Following values are generally proposed for the volume of the tank to be used in the booster set (DIN 1988 B1.5). Besides, the volume can also be set by calculation.
- The greater the tank volume the lower the on/off cycle of the booster set, thereby the lifetime of the pump/motor is extended and the power consumption is reduced.

Booster Set Maximum Flow Rate Q_{max} (m^3/h) (Single pump flow rate)	**Volume of the Membrane tank (lt)**
1-3	50
4-6	150
6-10	200
10-14	300
15-20	500
20-30	750
30-50	1000

Table 9.3 Determining the Volume of the Expansion Tank

- The main purpose of the expansion tank is:
  1- To limit the switch on/off number of the pump,
  2- To absorb the pressure shocks to occur in the installation,
  3- To store pressurized water for consumption,
- Calculating the volume of the membrane type tank:

  $$V_d = 33 \cdot Q\max \frac{(H_{üst} - 1)}{(H_{üst} - H_{alt}) \cdot S}$$

  V_d = Volume of the expansion tank (m^3)

  $Q_{\max}$ = The flow rate of one pump

  S = Number of on/off, is the switch on/off number of the booster setwithin one hour. Advised to be about 30 times per hour.

  H_{min}, H_{max} :booster set pressures (Bar). Pressure difference may be 2 or 3 bars.
- Example :

  $Q_{\max} = 4\ m^3/h$, $H_{\max} = 5\ bar$

  $$Vd = 0{,}33 \cdot 4 \cdot \frac{(5+1)}{2 \cdot 30} = 0{,}132\ m^3 = 132\ liters$$

  Consequently a tank with 150 liters volume should be selected.

9.2 Water Consumption of Installations

Pumps are needed for pumping several waters in installations. In order to set the requirements it is necessary first to calculate the flow rate (water quantity). Following information will be useful for in this respect.

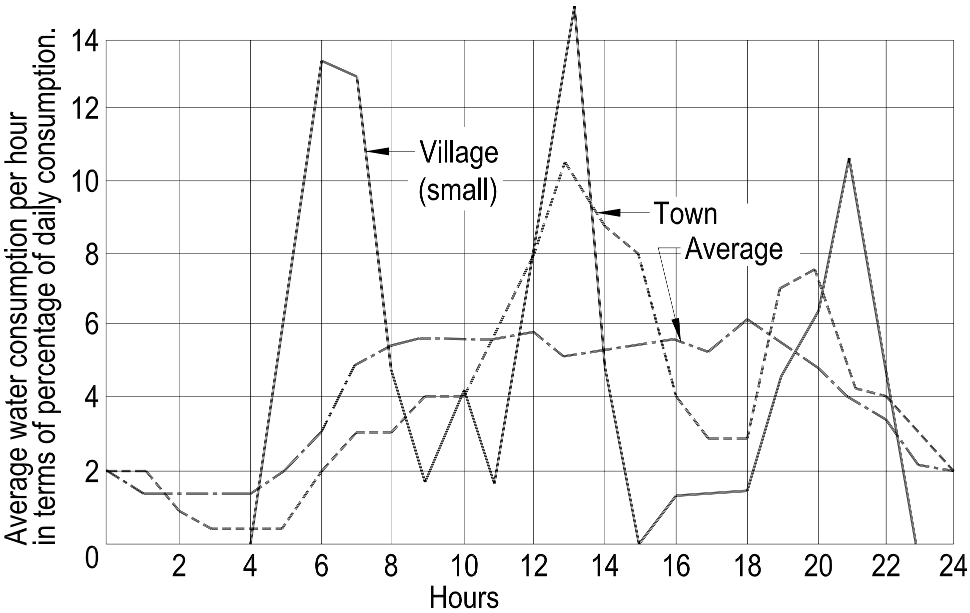


Figure 9.6 Variations in Hourly Water Consumption

Operation	Water quantity (Liters)
Drinking, cooking, bathing, per day	20-30
Laundry, per day	10-15
W.C. one flush (flush tank)	10-15
Home bath	150-200
Home shower	50-70
Watering the garden 1 m2 per day	1,5-3,0
Green vegetable watering m2 per day	5-10
Cattle, per animal/day	40-60
Sheep, goat, per animal/day	8-10
Car washing, private	200-300
Bus washing	80-100
Schools, per student/day	5
Campings,	
Per capita, including meals	100
Hospitals and residences, per capita/day	100-650
Hotels	
Full day accommodation per person/day	100-300
Public baths	
Per person, including cleaning	300
Turkish bath	Up to 700
Public swimming pools m3/per day	500
Towns, per capita/day	60-80
Cities, up to 50.000 people, per capita/day	80-120
Cities, more than 50.000 people, per capita/day	120-200

For maximum consumption the double and for hourly consumption %15% of above figures should be taken. Fire hydrant per second	5-10

Table 9.4 Per capita Water Consumption in Several Daily Life Activities

Device	Flow rate (lt/min)
Washbasin	10
Bathtub	12
Whirlpool	20
Shower	12
Toilet with flush tank	7
Washing machine	12
Dishwasher	8
½” cock	20

Table 9.5 Water Consumption for some Device Types

WATER CONSUMPTION FOR RESIDENCES AND INDIVIDUALS

Daily water consumption of an individual	: 190 liters
Annual water consumption of residences	: 400.000 liters

Daily water consumption of a house (approximate);

Bathing	: 115 liters
Dishwasher	: 110 liters
Dishwashing, manual	: 75 liters
Washing machine	: 84 liters
Toilets	: 26 liters
Teeth brushing	: 4 liters
Drink water	: 4-8 liters

Table 9.6 Water Consumption according to Individuals

PART 10

DESIGN, OPERATION AND MAINTENANCE OF PUMPING INSTALLATIONS

10.1 Safety

Pumps connection to the feeding system is subject to certain rules. Rules relating to the operation of the system where the pump will be connected are behind the scope of this manual.
In order to avoid breakdowns and expensive maintenance costs it is imperative to abide by the rules of the manual.

ATTENTION	Failure to abide by the rules herein may result in accidents causing bodily injuries, damages to the environment and to the machinery.

10.2 Safety Rules for the Operator

- Hot or cold parts in pumps must be reduced to safe levels according to the temperature of the fluid, in order to avoid possible accidents.
- The coupling is under protection by the coupling cover.
- Never hold rotating parts when the pump is running.
- Turbosan A.Ş. may not be held responsible in case of accidents risks which may accrue from safety equipments like the coupling cover, if the operator provided the pump only from Turbosan A.Ş. and other equipment from another supplier.
- All rotating parts must be guarded.
- Electric motor cooling fan must be protected by guard.
- Evacuation and draining measures should be taken in order to avoid flooding in case of an accident.
- The pump should only be run within its operational limits.
- Do not touch the pump when hot or parts which may become hot during the operation, like bearings etc.
- Do not subject the pump to sudden temperature changes.
- Do not exceed the density values of the fluid as explained in operating values.
- Do not adjust the flow rate by the suction valve.
- Do not run the pump dry.

ATTENTION	The mounting, commissioning, repair and maintenance of the pump must be carried by the authorized services of Turbosan and electrical works should be executed by certified electricians. TURBOSAN A.Ş. is not responsible for any damages incurred by unauthorized or uncertified persons.

10.3 Preventing Pump Failures

A proper mounting prevents pump failures and avoids excessive maintenance costs. A majority of breakdowns originate from the improper alignment of the pump shaft and the drive motor. Failures in the pump and motor bearings, rapidly worn out couplings result from alignment errors.

In order to avoid breakdowns and the resulting expensive maintenance costs, following precautions are advisable when mounting the pump to its place.

- After mounting the pump and the driving motor on the base frame, alignment must be controlled. Leveling should be corrected by shims and pieces of sheet metal. Angular and misalignments faults which may happen during the transport and use should be corrected.
- Concrete mortar should not be laid in the base frame before completing the fine adjustment.
- Another control must be made after laying the mortar and strongly securing the anchoring deviations.
- Pump settings should be controlled after suction and discharge pipes are connected.
- After running the pump at the actual operating temperatures make a final check on alignment.
- Any improper setting to be discovered during above-mentioned controls must absolutely be corrected.

ATTENTION	Responsibility accruing from the use of non genuine parts is fully incumbent upon the operator.

10.4 Storage and Carriage

ATTENTION	• Inside of the pump must be emptied completely from the pumped liquid. • Working safety rules must be observed. • The load should never stay on the air (sling) when transporting and the transport must be made by adequate means. • The load should be transported in the form of parts with reasonable dimensions and weights. • Pump dimensions and weights are indicated in the catalogs. • Flange holes and bolts on the pump are not for transport purposes. • Pump documentation should accompany the transport.

10.4.1 Storage

- If the pump will not be immediately used it should be stored in a dry and fix environment under closed area. If open air storage is required the pump should be covered against external effects.
- During storage the suction and discharge ends must remain closed in order to prevent the introduction of foreign substances in the pump.
- In case the pump is with mechanical seal and intended for long time storage, the pump shaft must be manually rotated for 2-3 turns each week, in order to prevent the formation of spot adhesions and the deterioration of the leak proof surface, as the surfaces of the mechanic seal remains under pressure.
- No water should remain in the pump to avoid freezing hazard.

ATTENTION	Pumps should be protected against frost and necessary measures must be taken accordingly.

10.4.2 Transport

- Türbo-NORM and ÇEP series pumps must be carried as shown in (Figure 10.1).

10.4.3 Protection of Pumps

- The pump must be kept in a manner to be suitably held for transport. It should never be moved by pulling out or pushing.
- The pump should not be kept under such atmospheric and chemical conditions which may cause damage.

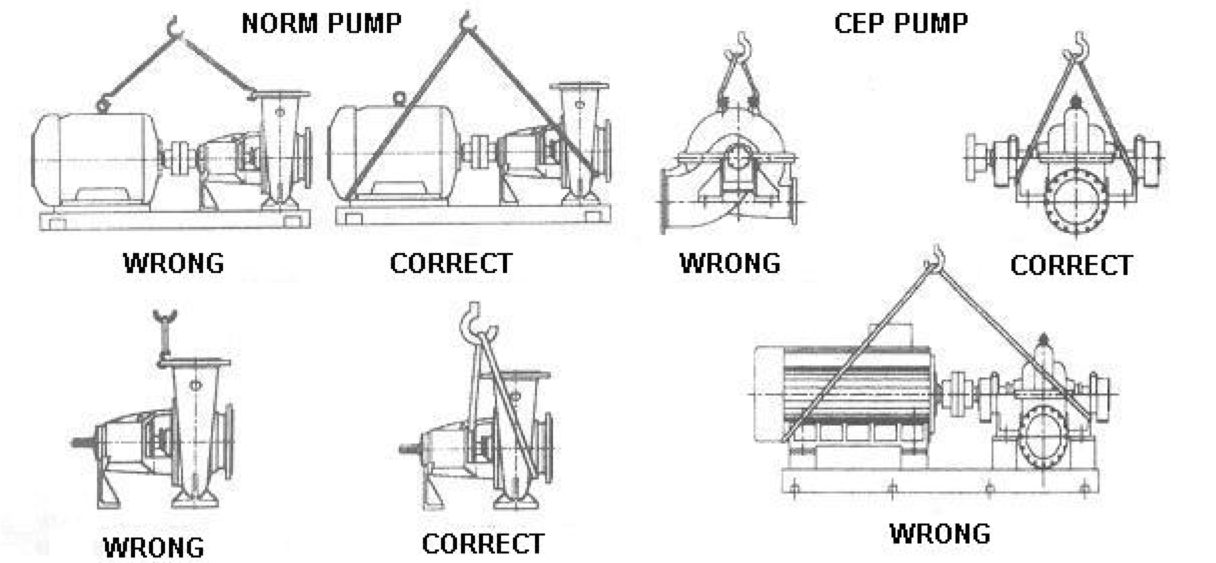


Figure 10.1 Ways of Carrying the Pumps

10.5 Pump Foundation and Base Plate

In case the produced pump is placed and connected in conformity with the installation, it will impeccably serve for several years without any problem. The pump base should have such a weight to provide rigid support to the base plate and should be capable of absorbing normal tensions and shocks during the operation. A concrete base sitting on the ground is the most suitable. If the pump must be places in a building or on a steel construction, the most suitable places are the upper part of beams or the location most close to main columns or walls. Wooden or unsupported floors generally curve, causing the bending of the pump base frame and spoiling the settings.

ATTENTION	The pump should be mounted on an even and vibration-free base plate. In case of doubt the base frame should be placed onto vibration absorber feet.

As to the pump base plate, a commercial saying notes that, “THERE EXISTS NO FULLY RIGID BASE FRAME”. Due to the weight of the pump and the driving motor and the

distribution of this load on the base frame, **all base frame suffer certain transformation during the transport and operation**. Therefore, whatever perfect are the settings of the pump and the driving motor made at the factory; **it is imperative to make realignment at the mounting site** (Figure 10.2).
Stretchings occurring on the suction and discharge pipes to be attached to the pump causes bents on the bearing body. As a result, shaft frictions and wearing increase and even the shaft may break. Therefore pipes must be supported suitably and stretchings in pump pipes should be prevented.

The concrete mortar must be laid beneath the base frame, always before pipes are connected. **Pipe connections must start from the pump side, never in other direction.**

Before fixing the base frame on the concrete base, the pump is placed and first settings are made. At least 48 hours must pass after laying the mortar and then the fixation nuts are tightened. Now the motor can be set.

In order not to subject the bearings to undesired loads, care should be taken to lay the pump well horizontally. For easy check of the level, the horizontal and vertical alignment of the suction and discharges flanges are verified with a small water level. This is particularly important in pumps with oil lubrication where bearings should be equally lubricated.

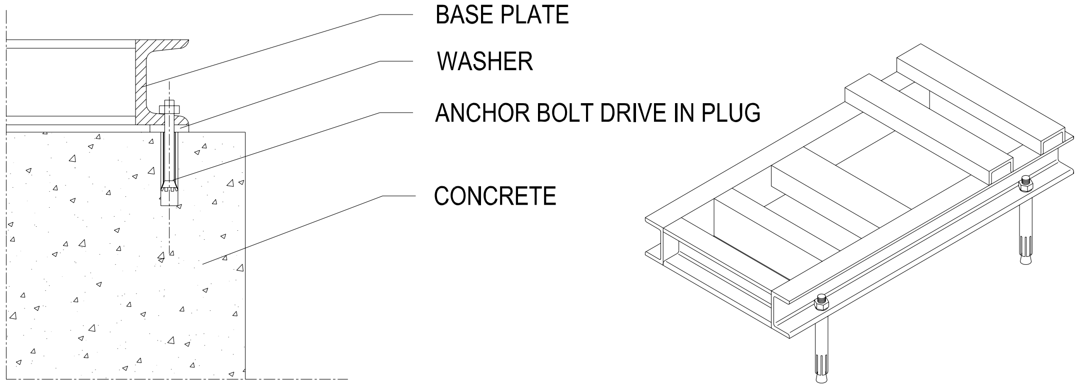


Figure 10.2 Detailed Connection of the Base frame on the Ground

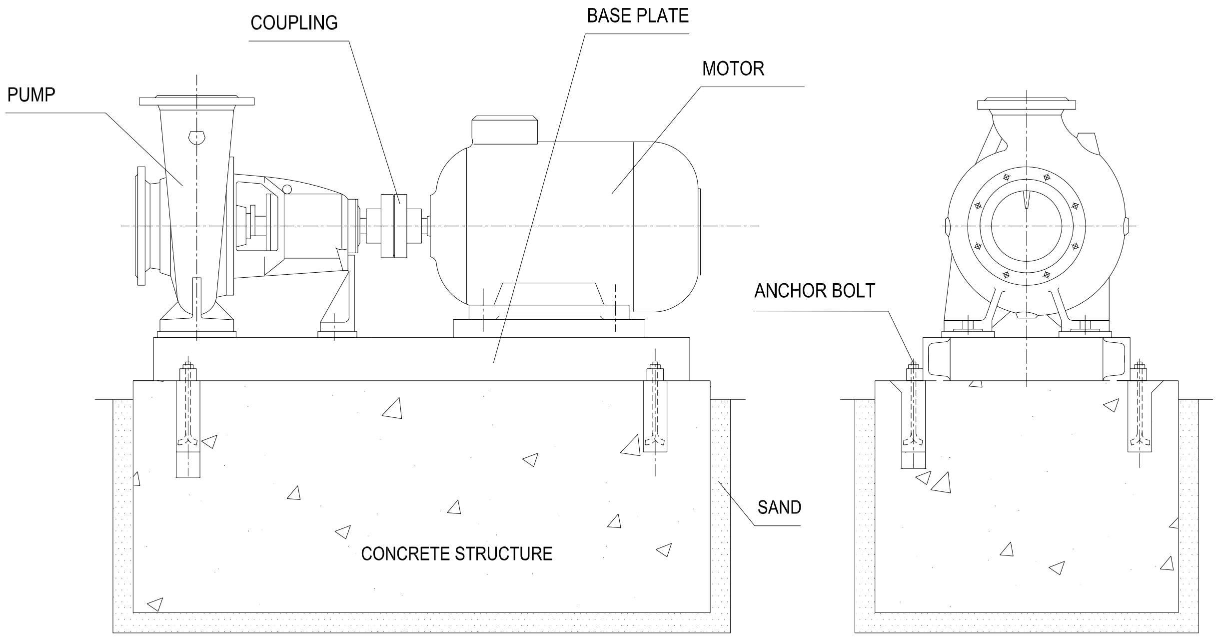


Figure 10.3 General Application

10.6 Coupling Alignment

- The most important factor in the problem-free running of the pump and motor group is the properly aligned coupling. The principal cause of many problems such as vibrations, noise, bearing heating, excess loading is the unadjusted or wrong adjusted coupling. Therefore the coupling alignment should be carried perfectly and checked regularly.
- Elastic coupling should not be considered as a tool to correct a bad alignment. Elastic coupling does not remedy the shaft misalignment between the pump and the motor nor remove the excessive misalignment.
- The “coupling alignment” is ensuring the motor and pump revolution axis are on the same line. If the pump is ordered with motor and base frame, it is shipped with coupling settings carried. However this setting may easily be altered during the transport, mounting and fitting the installation. Therefore, without relying on factory settings, it is absolutely necessary to re-make the coupling alignment after the mounting of the pump and motor to its place.
- Two metal pieces with true edges (like metallic ruler, gage etc.) and a precision calipers are needed for coupling alignment (special tools must be used for very fine and exact alignments).

10.6.1 Making the Coupling Alignment:

a) Angular alignment
b) Linear Alignment

a) In order to check the angular alignment the distance between the two parts of the coupling is reciprocally measured in horizontal and vertical plane. Spaces measured between these four points should be equal (Figure 10.4 a-b).
b) In order to check the parallelism setting a gage with true edges is pressed on one of the coupling components in parallel to the shaft and the position of the gage is observed with respect to the other component. The gage must touch both components at the

same time and all along its edges. This test must be carried in two locations on horizontal and vertical planes (Figure10.4 c-d).

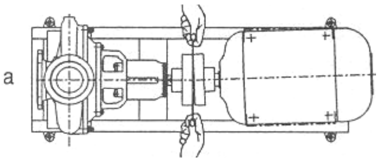


Figure 10.4 a
Angular Error Correction on Horizontal Plane

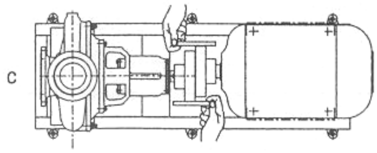


Figure 10.4 c
Linear Alignment Correction on Horizontal Plane

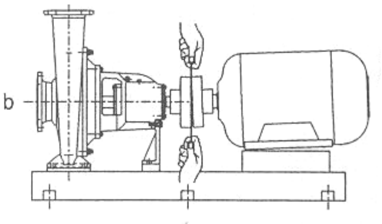


Figure 10.4 b
Angular Error Correction on Vertical Plane

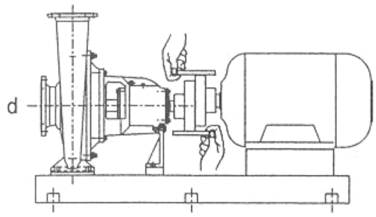
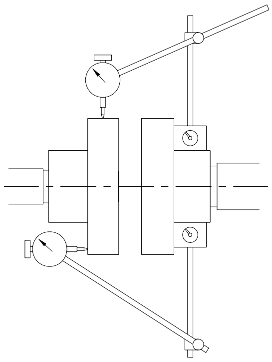


Figure 10.4 d
Linear Alignment Correction on Vertical Plane

Linear Alignment



Angular Alignment

Figure 10.4 Checking Settings with Comparator

10.6.2 Correcting Alignment Errors:

Alignment errors may happen on horizontal and/or vertical plane(s). Errors on vertical plane may be remedied by placing thin metal pieces on the bottom of the pump or motor footings whereas errors on horizontal plane are made up by slipping the pump or the motor horizontally availing of the empty spaces on the attachment holes.

ATTENTION	Do not disassemble the COUPLING COVER when the pump is running.

- Provided to comply with the coupling dimensions, the distance between two parts must be 2-6 mm.
- Couplings must be protected by a closed coupling case.

- All settings must be checked after any modification, because any alignment carried in one direction may affect the alignment on the other direction.
- The ends of the pump shaft and motor shaft should never touch each other.

ATTENTION	• The rotation direction of the pump should be observed. • The rotation direction of pumps is clockwise, looking to the pump from the motor side (except special requests)

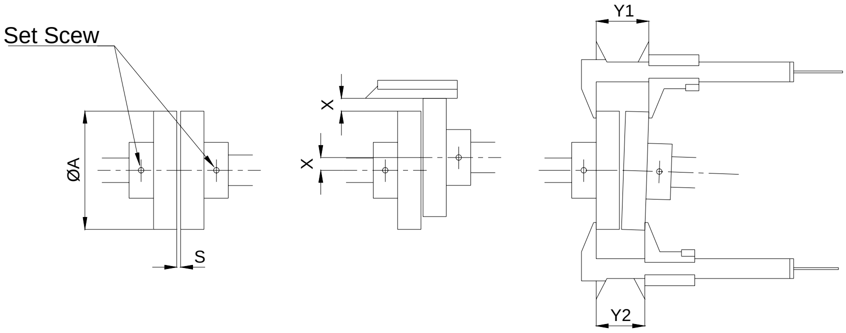


Figure 10.5 Coupling Alignment

Coupling (ØA mm)	Slack (S mm)	Parallel (X mm)	Angular (Y mm) = Y1-Y2
125-140	3-4	0.15	0.15
160-180	3-4	0.15	0.15
200-250	3-4	0.20	0.20
280-315	4-5	0.20	0.20
400	4-5	0.25	0.25

Table.1 Recommended Coupling Alignment Measures

10.7 Installation Design

10.7.1 Pipe Line and Matters of Importance

- Flow direction must be observed.
- The pump head should be able to balance the frictions of the pipe line designed flow rate and local losses.
- Tightness materials should not protrude (flange gaskets etc.)
- Pipe connections should be supported in a manner not to suffer any tension and the load and pre-tensions of the pipe line should not be conveyed to the pump.
- Precautions must be taken for removing tensions which may occur because of the temperature effects and compensators or similar devices should be used.
- Sudden section changes and direction deviations should be avoided.
- Axial misalignments in connecting elements should be avoided.
- The maximum suction lift of the installation must be calculated.
- The flow speed at the suction pipe should not exceed 2,5 m/s.
- The water hammer effect of the installation must be calculated and necessary measures should be taken.
- Air lock formation in the installation should absolutely be prevented. Irregular suction conditions which may be created by axially misaligned connection elements or similar divergent surfaces.
- Pump operation with minimum loss in the suction line should be targeted.

10.7.2 Flow Rate Alignment

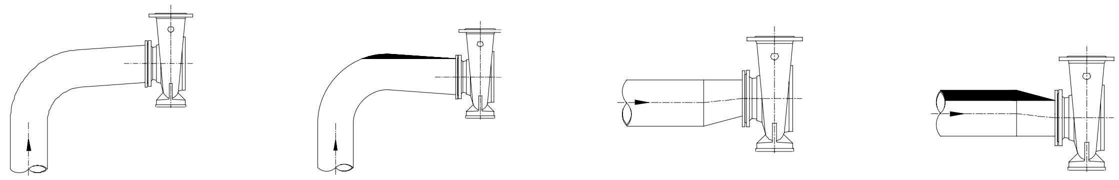
A valve fixed to the outlet of the pump may regulate the flow rate as required to flow through the pipe line. During this alignment water hammerings must be avoided and pressure changes should not be created by suddenly altering the valve positions. Flow rate alignments should never be carried by the suction valve.

10.7.3 Leakages

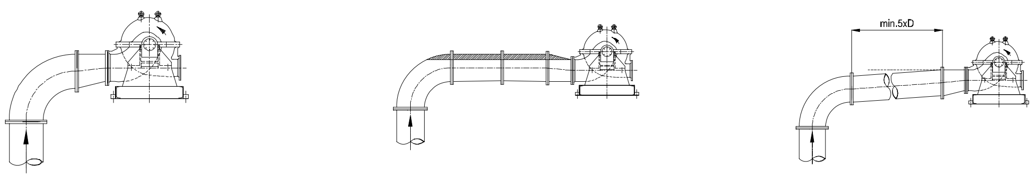
Standard tightness elements should be used in order to remedy leakages.

10.7.4 Pressure Control

Pressure meters should be used and their values should be recorded in the pump inlet and outlet in order to check the pressure remaining within adequate range (Figure 10.6).



Good Bad Good Bad



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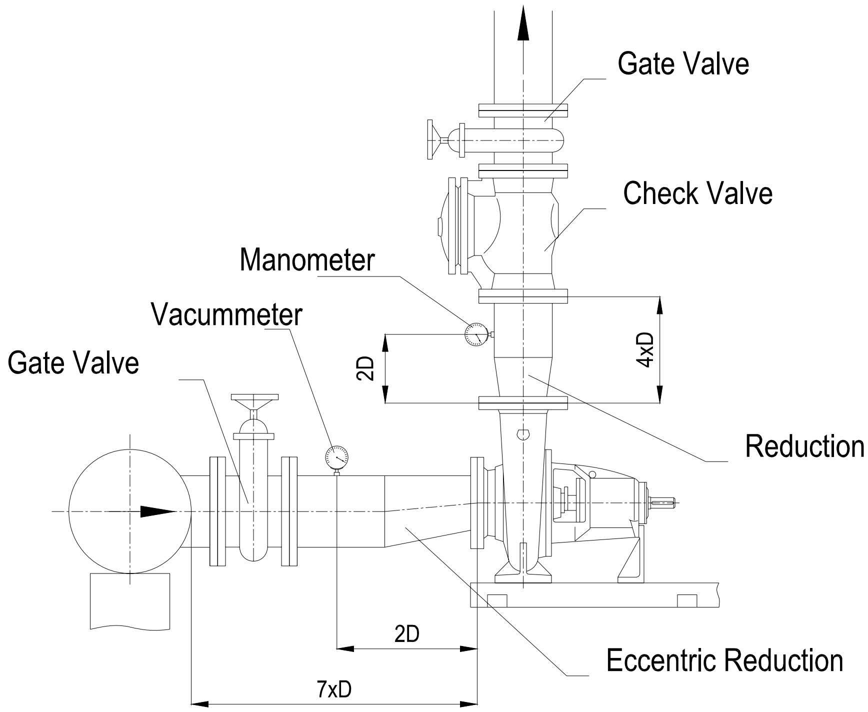


Figure 10.6

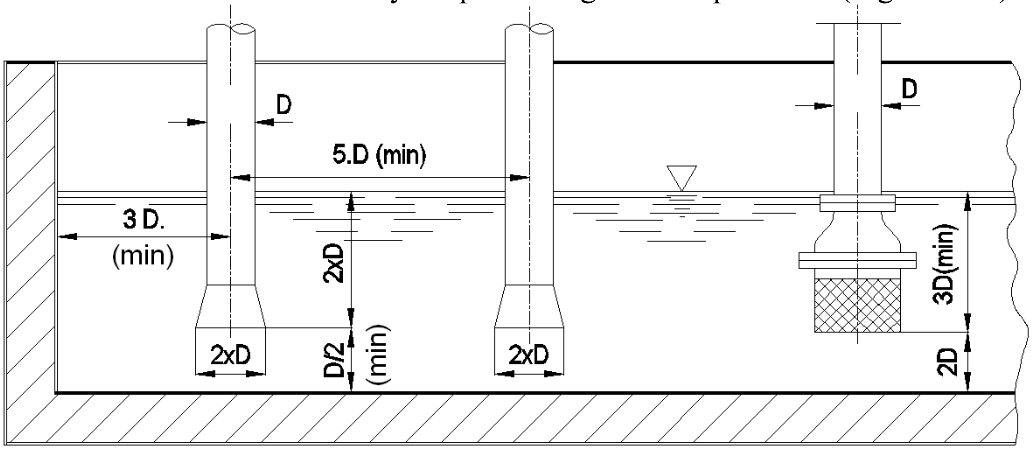
10.8 Problems originating from the Suction Pipe

10.8.1 Importance of the Suction in Pumps

In many cases centrifugal pumps can not adequately perform their function. The reason is the air penetrating into the pump on account of following reasons:

1- The diameter of the suction pipe is too small.
2- The suction pipe is not sufficiently immersed in water.
3- Both cases co-exist.

A vortex occurs if the speed is too high in the entry of the suction pipe and the air carried thereby enters into the pump. The pump starts to make noise and vibrates. In short time the shaft may break or other failures happen. If the quantity of air sucked the pump is excessive the pump starts to run dry and binding may occur.
These difficulties may be overcome by choosing a larger diameter suction pipe. Low speed and sufficient immersion is a warranty for preventing such air problems (Figure 10.7).



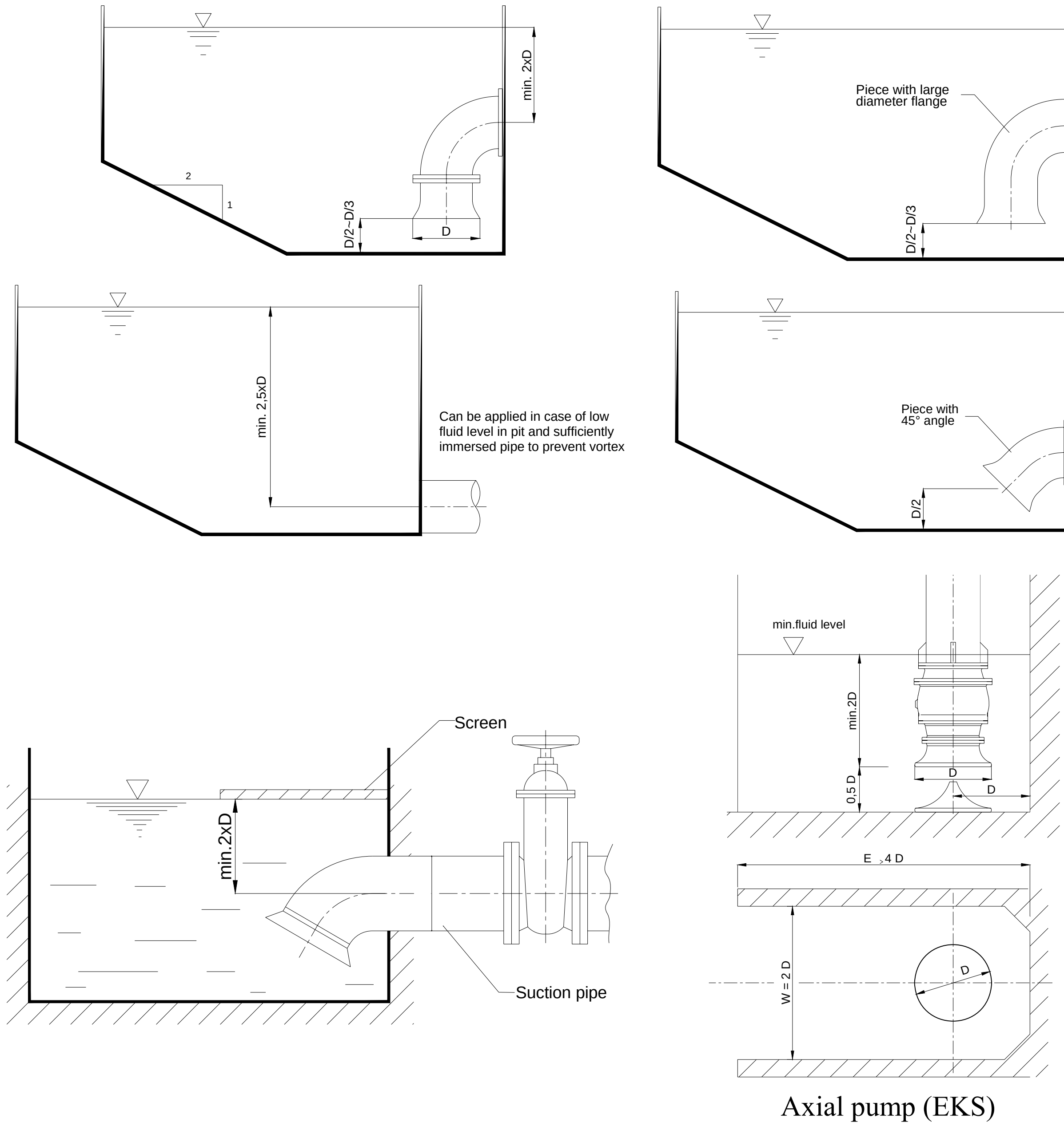


Figure 10.7

10.8.2 Importance of the Suction Pipe Diameter

The majority of the problems faced in centrifugal pumps arise from inadequate suction conditions. Some of these problems are caused by insufficient diameter of the suction pipe. The diameter of the suction pipe should never be less than the diameter of the suction inlet, even in most cases, it should be one size bigger.

If the friction losses in the suction pipe are substantial the pump does not only overwork but also may be subject to cavitation even may discontinue pumping.

Inappropriate speed distribution increasing friction losses results generally from the decomposition of air or vapor. If elbows and T-junctions are close to the suction inlet the situation becomes more complicated. Uneven distribution of the flow or vapor decomposition prevents the fluid to feed the impeller regularly. This fact impairs the hydraulic balance and generates vibrations, possibly cavitations and excessive shaft bents. As a result the shaft may break off and the bearings may prematurely break down.

Unless suction conditions are extremely good, speeds in the short and straight suction pipe should not exceed 2,5 m/s. If suction pipes are long and contain connection parts the speed should stay between 1,5 and 1,8 m/s. If precautions are not taken, elbows in the suction may cause problems. If attaching elbows to the pump suction flange is necessary, this must absolutely be in VERTICAL position. In case the location is restricted or the elbow has to be attached other than vertically, a straight pipe should be placed between the pump flange and the elbow the length being at least two times the diameter size.

If there is no straight pipe before the pump in the suction line as mentioned before, the liquid will build up towards the outer side of the elbow due to centrifugal force and will not be equally distributed to the inlets of double suction impeller. Such an imbalance will cause noisy and unsatisfactory operation of the pump (Figure 10.8).

ATTENTION	• The diameter of the suction pipe should never be less than the diameter of the suction inlet, even in most cases, it should be one size bigger.

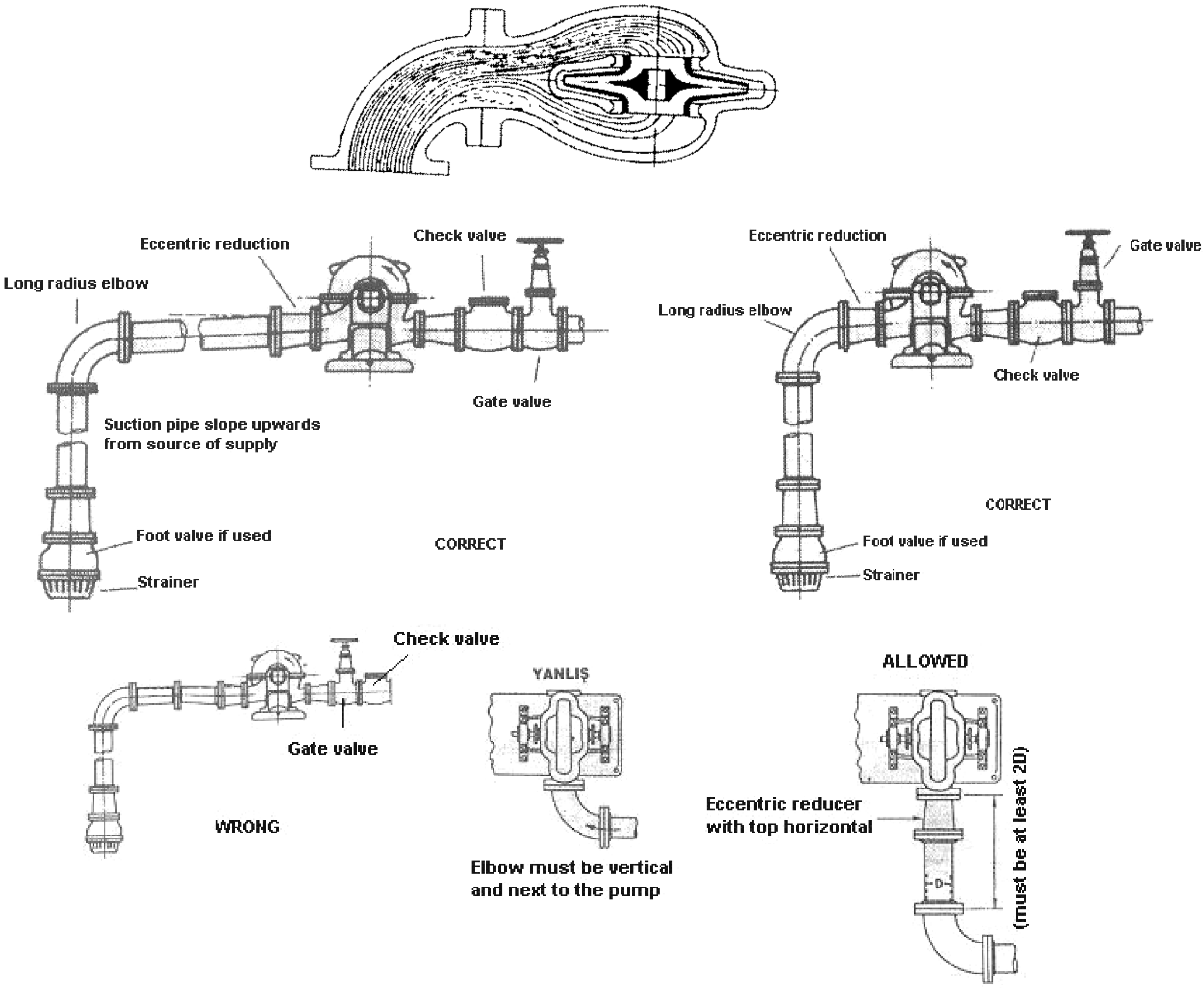


Figure 10.8 Mount Types of Centrifugal on the Installation

10.8.3 Operating with air in the suction line

Centrifugal pumps must never be allowed to run with the mixture of liquid and gas. Otherwise severe mechanical problems will occur, the pump shall be nonproductive and its lifetime will be shortened. Even if a small amount of air exists in the pumped liquid, the flow rate and therefore the efficiency will be drastically reduced. 2% free air in the liquid will reduce the flow rate by 10% and 45 free air by 43%.
In addition to this considerable reduction in the efficiency and waste of power, the pump will operate with noise and vibration. Air with foam is one of the most important reasons of shaft breaks. Besides, the air in the liquid will make it difficult the original suction of the pump and accelerate the corrosion considerably.

The introduction of air into the pumped liquid happens generally on account of holes in the suction pipe and inadequate fitting of seals.

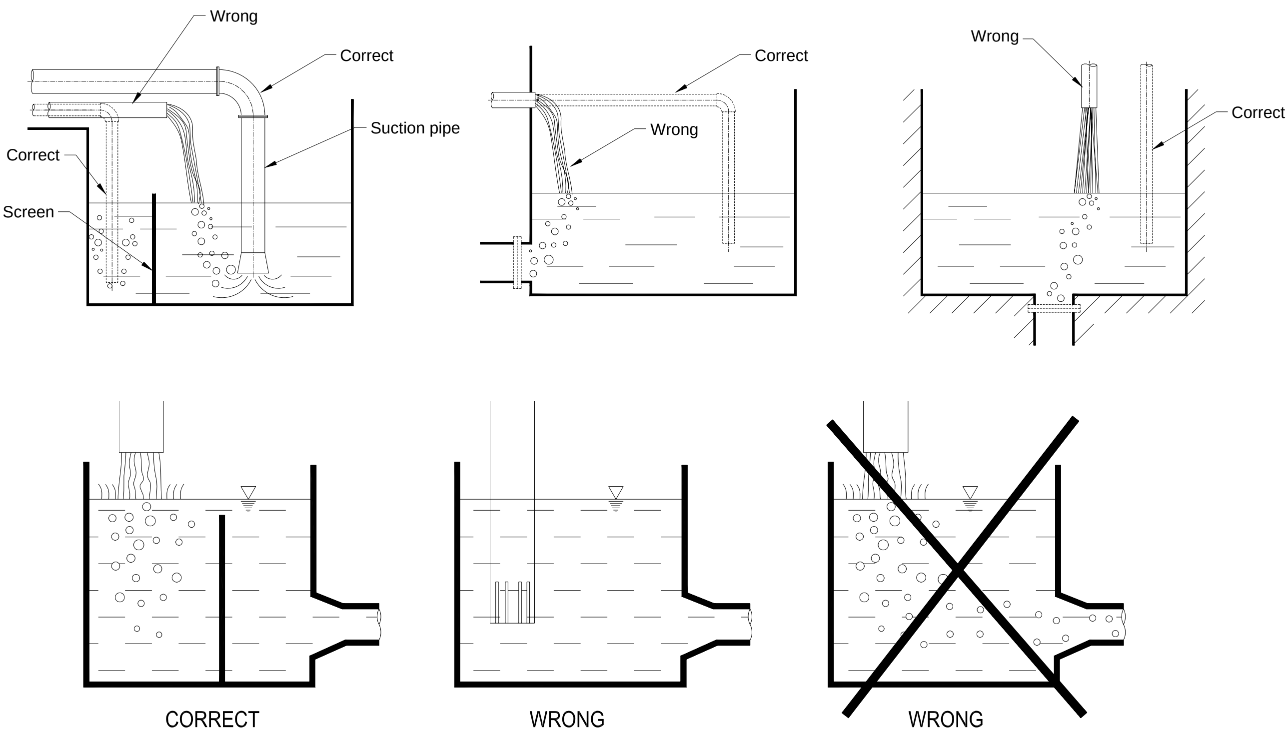
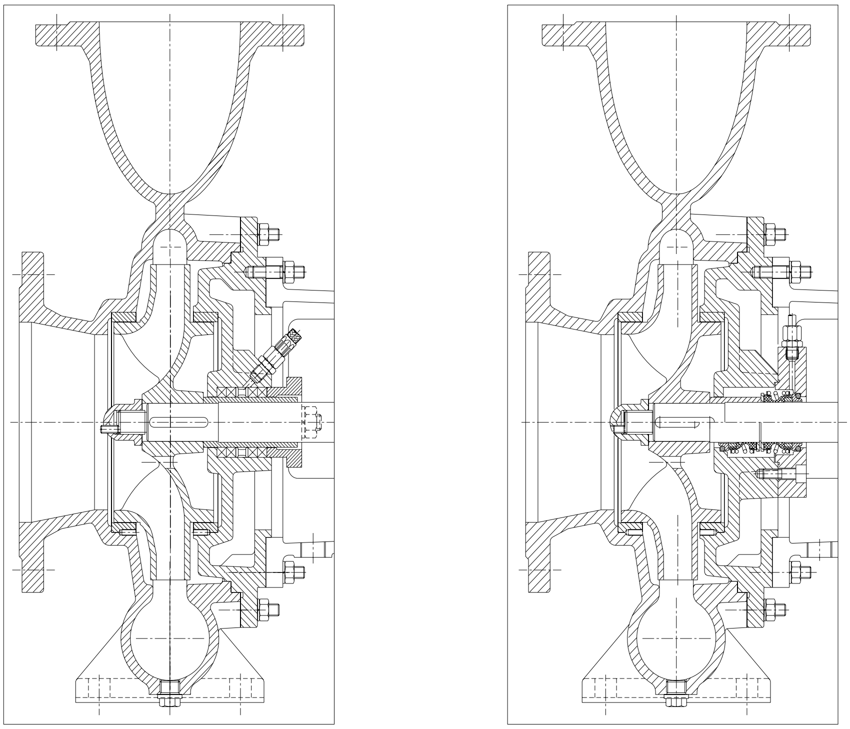


Figure 10.9 Situations in the Suction Tank

10.9 Shaft Sealing

The stuffing box is the part of centrifugal pump to ensure the tightness. Mechanical and packing seals are the two ways to provide the tightness. Both items act as a bearing on the shaft sleeve and require lubrication. Packing seals can not offer full tightness. Some quantity of water must always drip out for the lubrication of seal packs (Figure 12).
The choice of the lubricating material for the stuffing box depends on the type, the temperature and the pressure of the liquid pumped. Obviously, if the lubrication characteristics of the fluid pumped are weak or if the fluid contains abrasive particles, the seals will suffer severe damages. In case of external lubrication, the lubricating liquid must be carefully selected. In most cases water is good enough as lubricating liquid. If the use of water is inconvenient grease, oil or another liquid with good lubrication properties and without any chemical effect must be used.

ATTENTION	
	• Never operate the pump during servicing the packing. • Never operate pump without packing being lubricated.



Soft Packing Mechanical Seal

Figure 10.10 Seal Applications in Centrifugal Pumps

10.9.1 Maintenance of Seals

- Damaged and worn-out seals on the stuffing box should be removed without forcing and in a manner not to harm the housing surfaces.
- The parts of the two-piece lantern ring placed between the seals should be taken out separately, by pulling out with a steel wire hook or a special tool.
- After removing the seal at the rear of the housing, the latter should be thoroughly cleaned and the hole feeding the cooling/lubricating water to the trundle should be checked clean.

10.9.2 Preparing Seals

- Seal cutters should be used in the preparation of the seals. If cutting devices are not available the seal should be first wound on the relevant circumference in adequate turns and then cut correctly with a knife as seen in the picture (Figure 10.11).
- When placing the seals on the housing, care should be paid for mounting them in a manner to have the joints laid with 90° difference in consecutive seals.

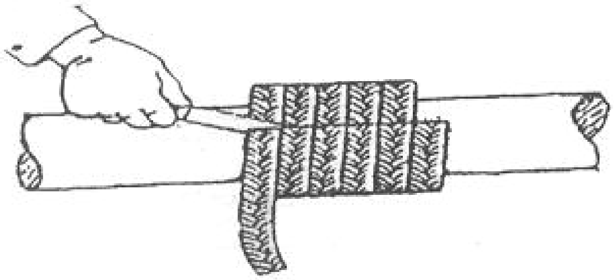


Figure 10.11 Cutting Seals

ATTENTION	
	• Seals with asbestos are not used in Turbosan pumps

10.9.3 Packing Problems

In order to ensure the lubrication and prevent heating in seal boxes using normal package type sealing sufficient outward must occur outwards. Only in the following cases feeding external lubrication fluid becomes necessary:

a) if the internal pump pressure applied to the stuffing box is negative
b) if the pumped liquid contains suspended solids
c) if the pumped liquid is very volatile, therefore has poor lubricating properties

In case negative pressure is formed in the stuffing box , it is necessary to insert a liquid obstacle in between, in order to prevent the air suction of the pump, to ensure its priming and to avoid seizing of the rotating parts. The sealing liquid is conveyed to the stuffing box via an external pipe and distributed in between packages by means of the trundle. A part of this liquid is sucked in the pump and the remaining part drips out of the stuffing box. In this case the sealing liquid ensures lubrication and prevents air suction. If the pumped liquid contains solid particles in suspension leakages from the stuffing box must be minimized. Because solid particles accumulate on packages and abrade the shaft. In this occurrence it is advisable to feed the stuffing boxes externally by clean water. The pressure of this water should be at least 0,7 kg/cm^2 higher than the pressure in the stuffing box. Thus a flow from the stuffing box towards the pump is ensured thereby preventing the introduction of solid particles into the stuffing box.

10.9.4 Mechanical Sealing

Mechanical sealing should be used depending on the characteristics of the liquid. The physical and chemical properties of the liquid should be clearly indicated in the selection of the mechanical sealing. Mechanical sealings not conforming to the liquid loose very shortly their tightness abilities. Figure 10.10 shows the mounting type of mechanical sealings in Norm type pumps.

LIQUID TYPE	MG1 MECHANICAL SEALING STANDART MATERIAL COMBINATION
Waste water, Sewage	Q1Q1PGG
Cold water	BVPGG
Hot water	AQ1EGG
Light chemicals	Q1Q1VGG-A(B)Q1VGG

Definition of Materials used in Mechanical Sealings:

Surfaces:

Hard		Soft	
Q1	Silicone Carbide *)**)	A	Carbon antimony impregnated
V	Ceramics (AL-Oxide)	B	Carbon resin impregnated *)**)
U3	Tungsten Carbide *)		

Bellows :

V	FKM (Viton)
E	EPDM *)**)
P	NBR
X4	HNBR

*) conform to KTW and WRC approval
**) conform to FDA approval

Metal parts and springs: G 1.4571 stainless steel

Regarding sealings to be used in special liquids, manufacturers of pumps and mechanical sealings should absolutely be consulted. In our products we use CARTEX type mechanical sealings newly developed for special liquids and with large span of utilization (Figure 10.12).

Figure 10.12 Cartex type Mechanical Sealing Applications

Mechanical seal Mounting:
Except special liquids, Turbosan A.Ş. generally uses MG1 series mechanical sealings. Mounting of MG1 Mechanical Sealing is easy and is shown in the following figures.

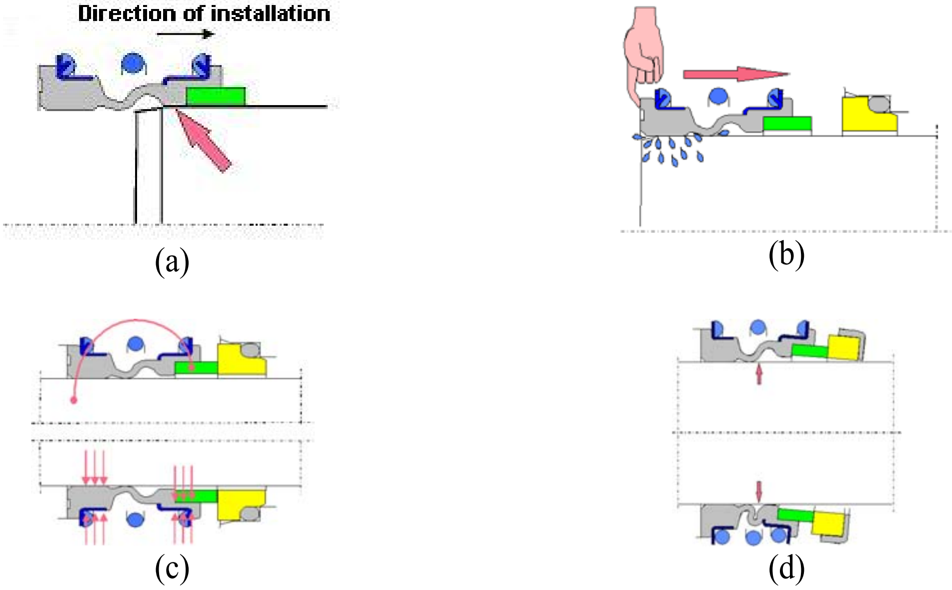


Figure 10.13 Mounting of MG1 Series Mechanical Sealing

Mechanical sealings are not sensible to axial and radial misalignments. Cleanness rules should be observed when mounting and dismounting and care should be shown for tension free mounting (Figure 10.13.(d))

10.10 Operation and Extra Operation Rules

10.10.1 Safety Rules

ATTENTION	• The electric panel is not within the responsibility of Turbosan and should be manufactured according te EN safety and other related instructions. These rules should be made only by experts and trained staff.

ATTENTION	• Rotation direction should be checked only when pump is filled with the fluid. • Movements should be slow in pumps filled with hot fluid in order to reduce tensions and sudden temperature changes. • Flow approved in constant rpm number must only be adjusted by valves of the discharge side. Valves at the suction side should be always open. Cavitation risk should always be considered. • If the pump shaft can not be rotated manually the pump should not be started up. During mounting the pump was subject to relative tensions. In such event all nuts and pump settings are loosened and pipe carriers are re-adjusted. All connecting parts are re-tightened without any tension.

10.10.2 Controlling Bearings

• The bearings of pumps connected to the installation whose suction and discharge lines are filled up should be filled with oil up to the indicator level before start-up.

• As the oil viscosity will increase in cold environments bearings should be filled up slowly and with patience.

ATTENTION	• During fill ups air bubbles in the oil increase the oil level. Thus bearings, therefore pumps, not adequately filled up by oil will have their lifetime shortened.

10.10.3 Pre-operation Controls

Before power supply the following checks must be performed.

- Are all pipes connected properly and tight?
- Is the valve on the discharge side closed?
- Is the valve before the suction inlet fully open?
- Is the motor ready to use?
- Is motor rotation direction correct? (may be seen by a short trial)
- Are clutches axially aligned?
- Is the pump air completely discharged?
- Is the pump shaft rotating easily?

10.10.4 Commissioning

Following may be observed:

- The pump should be started up by persons mentioned in Item 2.
- Close the valve on the discharge side.
- Fully open the valve on the suction side.
- Start the motor.
- Open the discharge valve as fast as possible.
- Observe the pressure on the discharge side. If it does not attain the value required with respect to the relevant rpm, stop the motor and purge the motor air.
- Adjust the valve on the discharge side and reach the pressure required for the given rpm.
- Motor started for the first time should be run in the operation point as long as possible.
- Do not adjust by the suction valve.

10.10.5 Checks during the Operation

Following matters should be observed when operating:

- The speed and the requested pump head should be controlled.
- Vibration free operation requirements should be met.
- The flow at the suction inlet should be checked.
- The temperature of the bearings should be measured by a thermometer and should not exceed 70°C.
- The cooling liquid supplied to sealings should be checked
- The quantity of coolant escaping from seals in the form of dew or vapor should be around 0.1-0.2 m^3/hour. Negligible leakages in the form of drippings are allowed.
- Sealing liquid or seal box should not be directly touched as they may become hot.

ATTENTION	• In case of increasing leakages from sealings the pump must be forthwith stopped and sealing must be added.

10.10.6 Non Operating Conditions

- Before stopping the pump, valves on the discharge side should be shut down. They may remain closed after shutting the pump off. The fluid in the pump must be drained off against frost risk by opening the blind flange at the bottom of the volute (the same risk is also valid for the system).

ATTENTION	• The safety of explosive, toxic, red-hot, crystal structured liquids should be assured with respect to human and environmental health. The same rule also applies for the draining of fluids from the blind flange at the bottom of the pump. For draining back the pumping line a by-pass line should be used instead of the pump and the suction line.

10.10.7 Bearings

Bearings are one of the most important constituents affecting the lifetime of rotating machines. Following are the principal factors having an effect on the life of bearings:
• the quality of the bearing,
• mounting - dismounting,
• lubrication,
• operating conditions (dust, humidity, temperature, load, vibrations)
• storage

As it would be noticed all factors, except the bearing quality, are under the control of the operator. Above-mentioned points should be carefully observed with respect to bearings which are assuming one of the sensitive and critical duties among machine components.

10.11 Pump Operated with Slightly Open Valve

When designing an installation pumps are generally chosen with respect to future expectations rather than the maximum capacity or actual requirements of the installation. In such case pumps should be equipped with valves in order to operate in existing conditions. If the pump is continually run with closed or slightly opened valves, the lifetime may be shortened and repair-maintenance costs may considerably increase. In case of slightly opened valve a very small percentage of the operating flow rate passes through the pump. The difference between the pump intake power and the power transmitted to the fluid by the pump is transferred to the pumped fluid as heat. The pump body may remain short in transferring this energy into the environment and the temperature of the fluid and the pump may attain dangerous levels. In the event the pump operates under nearly closed valve another problem arises. In slightly open valve case hydraulic radial push forces can not be balanced and this may cause abnormal bending of the shaft. The pump operates with noise and excessive vibrations. As a result of these unbalanced forces the shaft may break off and bearings may fail.
If it becomes necessary to extremely turn down the valve on the discharge pipe when the by-pass pipe is fully open, the true pump total head and the flow rate of the system should be revised and new values should be selected.
The pump should never operate continuously with slightly open valve, particularly when pumping liquids containing abrasive materials. Abrasion increases considerably when running with turned down flow rates and the impeller may be destroyed within few weeks or months (Figure 10.14).

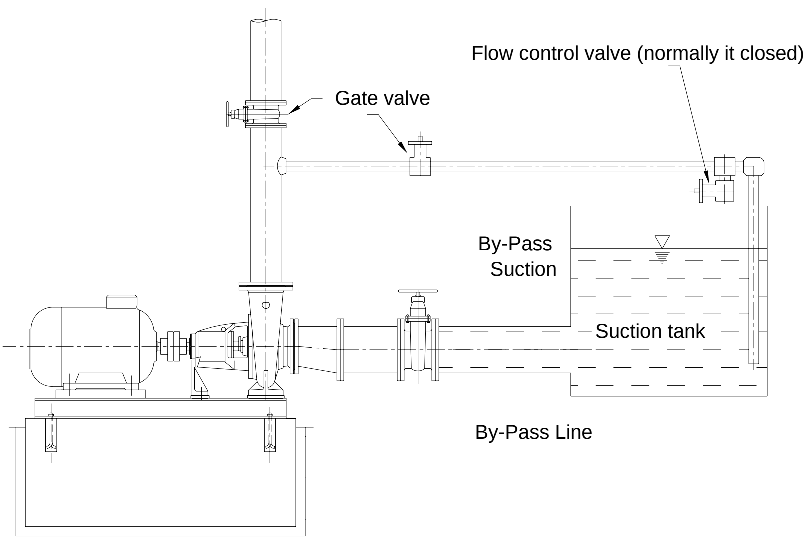


Figure 10.14 By-Pass Line

ATTENTION	• Do not run the pump for a long time when the valve is shut or turned down.

10.12 Lubrication of Pumps

In Turbosan pumps bearings are designed in a manner to allow grease or oil lubrication and easy maintenance. (see Table.3 Oil List for Pumps)

10.12.1 Grease Lubricated Bearings

It is recommended to use Lithium Soap Class 3 grease (such as Shell Alvania R3 or equivalent) in these bearings. This type of grease may be used in several places within broad temperature range and is resistant to oxidation. It is suitable for use in hot, cold, dry, wet, clear and dirty environments.

Oil seals at bearing covers prevent humidity and dirt to penetrate the bearings (Figure 10.15). Grease in bearings should be filled up after about 3 months (or 2000 hours) operation. Frequent greasing cause excessive heating in bearings and shortens the lifetime. In many cases less frequent greasing should be made.

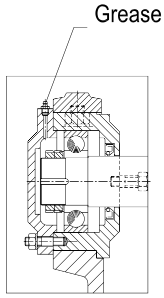


Figure 10.15 Grease Lubricated CEP Bearing

10.12.2 Oil Lubricated Bearings

It is recommended to use No: 46 oil (i.e. Shell Tellus 46 or equivalent) in these bearings. This type oils display high resistance to wearing and oxidation. They are not affected by air or humidity, offer protection against rust and foaming and can be used in low temperatures. (Figure 10.16)

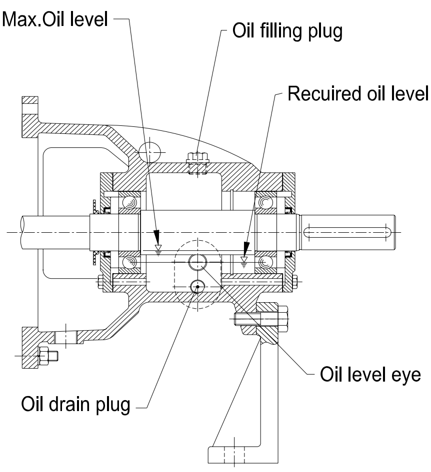


Figure 10.16 Oil Type NORM Bearing

10.13 Bearings

Following matters should be observed for extending the lifetime and ensuring problem-free operation of bearings:

10.13.1 Cleanness

Cleanness is the number one condition in bearings. 90% of bearing damages result from dirt. Following matters should be carefully observed:
1- **ABSOLUTELY** do not open the bearing unless necessary.
2- Wash your hands. **Wipe out** the dirt, burrs and grease on the tools.
3- Spread a clean sheet of paper on the workbench you intend to work on. Lay the tools you use and clean as well as the bearing only on this **clean sheet of paper.**
4- Cover the bearings, bearing bodies and the shaft to be fitted with the bearing with a **CLEAN** piece of cloth.
5- **Do not take out** the new bearings from their special package until the fitting, if you will not use them immediately.
6- Before re-mounting the shaft and bearing bodies on their place **clean with a solvent.**

10.13.2 Taking apart Bearings with Care

1- When fitting or taking apart a bearing use only a **bushing** or a **puller** touching the inner ring of the bearing. (The only exception is some double entry pumps where bearings are taken apart by the bearing body).
2- Never apply force on balls and cages. **Force should be applied on rings** only**.**
3- Do not directly knock on bearings. **Use** a bushing with flat front or a properly fitting **puller.**
4- If the bearing body is used to pull out the bearing, the pull should be straight and never **hammer** the bearing or the shaft**.** As both rings are attached, hits will be transmitted to balls and destroy the bearing.

ATTENTION 	• The bearing temperature in bearings should not exceed 70°C.

10.13.3 Checking the Bearings and the Shaft

1- Check bearings thoroughly. If any bent, dot, crack, pitting exists never use the bearing and discard it. Bearings must be fully perfect.

2- Hand rotated bearing must turn easily and silently. If any seizing and noise is noticed do not use the bearing and discard it.

3- If any doubt exists with respect to the perfection of the bearing do not use it. A new bearing costing few thousand Liras will save you from severe loss of time and pump damage. During each maintenance and repair works in important and critical places change the bearings.

4- Check the shaft. Shaft contacts with the bearing should be burr free and clean. Any burrs should be removed and polished. Shaft dimensions should match the given tolerances. Shaft rabbets should be straight and should not exceed dimensions.

10.13.4 Checking New Bearings

You must be sure to have chosen the right type and dimensions for bearings. For example fixed bearing and slant bearings with the same dimension fit the pump perfectly. However slant bearings are not suitable for forces of both directions and carry the axial force towards one direction. If the bearing has protecting covers check their genuineness. Refer to pump catalogs and special user manuals of bearings in order to choose the right bearing.

10.13.5 Careful Placement

1- Slightly oil the shaft surface and the seat of the bearing.
2- If protective covers exist check proper directions. Slant bearings should face the proper direction on the pump. If double bearings are used, mounting should be made in a manner to have correct faces juxtaposed. Assembly forms are different for each model. Therefore catalogs and maintenance books should be consulted for special pumps.
3- Forces should be applied to bearings perpendicularly. There should be no tightening on the shaft. The sleeve used for pushing the bearing must have a clean surface, be perpendicular and touching only the inner ring.
4- Press the bearing so that the shaft will lean fully against its rabbet. The shaft rabbet is the leaning surface of the bearing and also assures the straightness of the bearing.
5- Assure that the safety washer is properly mounted, its straight side facing the bearing. Check the tightness of shaft nuts.
6- Oil as indicated in instruction books and in accordance with directives.

10.13.6 Recommended Oil Change Interval

Pump speed	Oiling time
1000 rpm	4400 running hours
1500 rpm	3000 running hours
3000 rpm	2000 running hours

10.14 Bearing Temperatures and Measuring

- Bearing temperatures are generally estimated by manual checks. Nevertheless the human hand is not a precision thermometer and may erroneously report danger. What we call “hot” is relative to individuals and changes between 50° C and 55° C. The upper operation limit in bearings is indeed the temperature where the lubricant remains deficient and carbonization starts. Up to 70° C bearing temperatures are largely safe. This is the maximum operation temperature desired for bearings. Because the lubricant flow is expected to be better at this temperature.

- Unless water cooled, all bearings operate somewhat above the environmental temperature. Heat in bearings is generated by the friction of rolling balls and ring seats. Some part of the heat may be transmitted to the bearings by the shaft. The quantity of the heat distributed may vary according to the cooling area of the bearing seats, the movements of the environmental air and temperature. When this temperature is set once it remains constant until one of the variables is changed and a stationary operating temperature is formed. A stationary temperature is not a risk sign unless it exceeds the maximum limit of the oiling agent, whatever felt by the human hand. The temperature must be exactly **measured by a thermometer** and properly recorded.

- Violent and sudden increase in temperature is a sign of danger and the reason must be absolutely investigated. Some grease must be added and the temperature must be checked. If the temperature does not fall no more grease should be added. The unit should be checked for extra loads such as wrong clutch alignment, inadequate seal placing etc.

- Nonetheless the temperature increase may not result from a contingent bearing failure or excessive load. The temperature of the liquid pumped or warm summer months may cause the environmental temperature increase. The heat transfer caused by increasing temperature of the liquid may be reduced to minimum by circulating cooling water in bearings externally. Bearing temperatures may also increase because of wrong oiling.

- Excessive oil in bearings increases the rotation resistance and the temperature. Taking out surplus oil decreases the bearing temperature.

- Sometimes excessive bearing heat is noticed in first started up pumps. This fact originates from oil seals and not from the bearing itself. When oil seals are run in, the temperature falls down to normal.

10.15 Matters to Observe when Starting up the Pumps for the First Time

❖ Before starting up a new pump for the first time all the factors contributing to the successful operation of the pump should be carefully checked.

a) The pump and the motor shaft should easily rotate manually.

b) Check the motor electrical connections are made correctly.
c) Check the rotation direction of the motor is correct.
d) Check the pump frame, foundation, anchorage connections and pipe assemblies.
e) Never start up a pump which can not be easily rotated manually, before taking corrective measures.
f) Seals, intake and cooling water pipes should be placed and connected to ensure adequate flow and checked.
g) After all these controls fully open the suction valve of the pump to ensure the filling of the pump with the liquid. If there is no liquid in wearing rings and seals sufficient for oiling and cooling, these elements cause very effective breaking.
If the pump is totally filled up by the liquid there is enough liquid for oiling. Check if the pump is full or not through the appropriate plug or filling connections.
h) Start up the pump and open the discharge valve as quick as possible. Check the pressure at the discharge side. If the pump can not reach the operation pressure value in the corresponding rpm stop the motor, purge the air and remake the necessary controls.
ı) The pump starting up for the first time should be run as close as possible to its operation point and for an adequate period of time.
j) Pumps should not be run with closed valve for a long time, because in such event all the energy supplied to the pump will forthwith transformed into heat and increase the seizing risk.

ATTENTION !	• In order to reduce failures to minimum and extend the lifetime, the pump should not be stopped and re-started frequently.

10.16 Typical Failures in Centrifugal Pumps and Causes

FAULTS	POSSIBLE CAUSES	
1) The pump does not deliver or low flow rate	1.1	The friction loss (resistance) of the installation is above the calculated value. (The total pump head of the system is more than the rated height of the pump)
	1.2	The suction head is high.
	1.3	The impeller, the volute delivery line may be stuck.
	1.4	The rotation direction is wrong.
	1.5	The suction pipe is not enough immersed in the liquid.
	1.6	Air in the suction pipe.
	1.7	Air leaks in through suction pipe connections.
	1.8	The total static height of the installation is more than the pump head.
	1.9	Discharge line clogged
	1.10	Low rpm speed
	1.11	Air leaks through seal
	1.12	Suction line not filled up with water prior to start-up
	1.13	Bottom valve and filter clogged
	1.14	Parallel running pumps are not suitable for parallel operation
	1.15	The diameter of the suction pipe is too small
2) Pump discharge head insufficient	2.1	Air leaks through suction pipe connections or bottom valve
	2.2	Low rpm speed
	2.3	The rotation direction is wrong
	2.4	The fluid viscosity increased
	2.5	The pump total height is higher than the calculated height
	2.6	If parallel operating pumps are available, they are not suitable for parallel operation in the system
	2.7	Impeller worn out or clogged
	2.8	Excessive internal leakage in the pump
	2.9	Discharge line clogged
3) Pump stops after operating for a while	3.1	Suction line not filled up with water prior to start-up
	3.2	Compressed air blocked in the suction line
	3.3	Air leaks in suction line
	3.4	When pump starts to deliver, the water level drops at suction side and air is sucked together with the liquid
4) Motor draws excessive power	4.1	Motor rpm higher than the usual
	4.2	The rotation direction is wrong
	4.3	The pump total height of the system is lower or higher than the rated height (wrong pump total height is the most important factor9
	4.4	The viscosity and the specific weight of the liquid is high
	4.5	Impeller is clogged
	4.6	Pump shaft is bent or friction between rotating and fixed parts
	4.7	Wear ring highly worn out
	4.8	Seal too stuck, shaft too tight
	4.9	Coupling setting too bad
	4.10	Disintegrated bearings
5) Pump vibrates and runs noisily	5.1	Air pockets in the pump and suction pipes. Wrong suction piping
	5.2	Suction lift too high (the fluid evaporation pressure drops at the suction side, starting the cavitation)
	5.3	Bottom valve and filter too narrow and insufficient or partly clogged, causing excessive load loss in the suction line
	5.4	The orifice of the suction pipe is not enough immersed in water
	5.5	Pump operates lower or higher than the rated flow rate
	5.6	Impeller is clogged
	5.7	Pump shaft misaligned with the motor shaft or pump shaft bent
	5.8	Bearings excessively worn out, badly lubricated, bearings disintegrated or excessively heated
	5.9	Imbalanced or out of balance impeller
	5.10	Frictions between rotating and fixed parts
	5.11	Pump foundation not enough rigid
	5.12	Suction and discharge collector connections not rigid

6) Bearings frequently worn out	6.1	Pump shaft misaligned with the motor shaft or pump shaft bent
	6.2	Misalignment in bearing assembly
	6.3	Imbalanced or out of balance impeller
	6.4	Inadequate lubrication or pump runs without oil
	6.5	Balancing holes not opened in the impeller or inadequate. Axial thrust impossible to balance.
	6.6	Bearing types do not conform to operating conditions
	6.7	Incorrect coupling setting
	6.8	Pump foundation not enough rigid
7) Temperature rise in the pump casing	7.1	Flow rate conveyed by the pump much less than the normal flow rate or pump does not deliver at all.
	7.2	If pumps are parallel connected, they are not suitable for parallel operation with each other.
	7,3	Frictions between rotating and fixed parts
	7.4	Bearings seize or shaft bent
	7.5	Axial thrust impossible to balance.
	7.6	Sealings too much tightened
	7.7	Pump foundation not enough rigid, pump operating with vibrations
	7.8	Pump closed or runs with very slightly open valve
8) Pump disintegrate bearings, breaks shafts quite often and motor is burnt out	8.1	Suction problem: suction lift too high or faulty suction installation
	8.2	Not enough water
	8,3	Incorrect coupling setting
	8.4	Incorrect calculation of the system's pump total head. The pump does not run on values as ordered
	8.5	Pump operates far right from the curve, in the inefficient area
	8.6	When stopped, the motor turns the other way and the shaft nut is undone.
	8.7	Pump bearing assembly incorrect
	8.8	Pump shaft bent
	8.9	Pump shaft diameter not suitable
	8.10	Wearing ring gap too much increased
	8,11	Selected bearing is wrong
	8.12	No oil in bearing housing
	8.13	Balancing holes not opened in the impeller or balancing blades too big or do not exist
	8.14	Large size solid substances stuck between the impeller blades
	8.15	Pump is very often switched on and off
	8.16	Selected pump size wrong (low power)
9) Pump turns the main switch on	9.1	Selected switch not commensurate with the necessary current (small switch)
	9.2	Motor does not match the pump
	9,3	Motor defective
	9.4	Abnormal fluctuations in the mains voltage
	9.5	Pump blocked

Table 2

10.17 Pump Oil List

Oiled Parts: Bearings

OIL PROPERTIES

A) Lubrication Type	Oil bath
Bearing temperature	> 50° C
Manufacturer	SHELL
Oil type	Tellus T68
Density	0,877 kg/dm^3
Flashing point	230° C
Yield point	- 36° C
Viscosity (20° C)	173 mm^2/s
Alternative oils	Mobil DTE Light / HH
First change	300 hours
Change frequency	3000 hours
Maximum change time	6 months

Bearing temperature	< 50° C
Manufacturer	SHELL
Oil type	Tellus T46
Density	0,872 kg/dm^3
Flashing point	210° C
Yield point	- 39° C
Viscosity (20° C)	110 mm^2/s
Alternative oils	Mobil DTE Light / HH
First change	300 hours
Change frequency	5000 hours
Maximum change time	12 months

Total oil quantity	Q	
Norm Pump Type	Kg	Liters
32/250; 40/200; 40/250; 50/200; 50/250; 657200; 80/160;	0,1	0,11
50/315; 65/250; 65/315; 80/200; 80/250; 80/315; 100/200; 100/250; 100/315; 125/200; 125/250	0,2	0,26
80/400; 100/400; 125/315; 125/400; 150/250; 150/315	0,3	0,36
150/400; 200/315	0,3	0,34
150/500; 200/400; 250/315	0,5	0,52
200/500; 250/400; 250/500; 250/450; 300/315; 300/400; 300/500	0,9	0,90

B) Lubrication Type	**Grease**
Viscosity (40°C)	80 cst
Thickness level	3
Use temperature	-20° C - +140° C
Long term use temperature	75° C
Change frequency	1.500 – 2.000 hours
Grease type	ARCANOL MULTI 3 L71V
Alternatives	SKF LGMT3; SHELL ALVANIA EP GREASER

Attention: First change and change frequency depends upon the operating conditions and the real running time.

PART 11

PUMP ACCEPTANCE TESTS

A set of standards defines the tests and measuring methods to determine the pump performance and to check the warranties offered by manufacturers. They are DIN, ISO and TS standards. The ordering customer and the manufacturer agree on one of these standards and check whether the operating point warranted by the manufacturer is realized within the tolerance limits set by the agreed standard. Following are the main aspects of some of these standards.

11.1 ISO Standards

Two standards named,

> ISO 2548-Centrifugal, mixed flow and axial pumps - Code for acceptance tests - Grade C
> and
> ISO 3555-Centrifugal, mixed flow and axial pumps - Code for acceptance tests - Grade B

covering the acceptance tests applied for radial, mixed flow and axial pumps have been unified into one single standard in 1996, named

> "ISO 9906- Rotodynamic pumps-Code for hydraulic performance test for acceptance-Grades 1 and 2"

ISO 2548, ISO 9906 corresponds to Grade 2; and ISO 3555, ISO 9906 corresponds to Grade 1. Grade 1 is more precise as compared to Grade 2. New tolerance factors have been added into ISO 9906 and pumps assuring the guarantee point according to ISO 2548 and ISO 3555 are also ensuring the guarantee point according to ISO 9906.

Here we will consider the main points of the ISO 9906 standard.

11.1.1 ISO 9906 Standard

First, terms are described and definitions are made in the standard. Thereafter guarantee matters are taken up. Data is supplied with respect to the application of tests, analysis of test results, flow rate measurements, measuring the pump total head, the measurement of the pump rpm, the measurement of the pump shaft power and cavitation tests. Here we will explain the guarantee matters and the guarantee conditions and how results are evaluated are as to whether the results meet the guarantee terms or not.

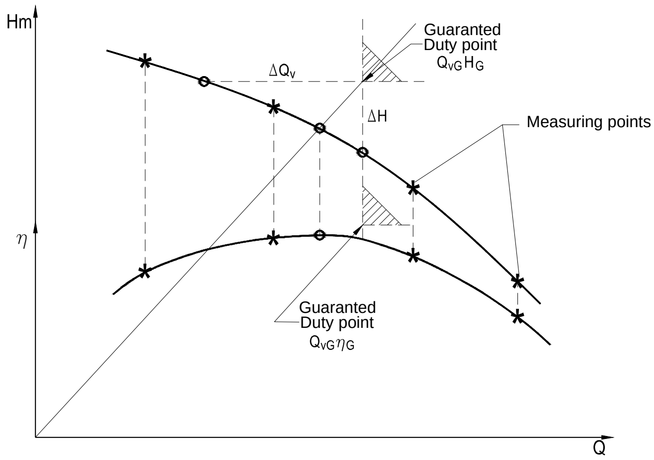
Guarantee Matters

A guarantee point is defined by the warranted flow rate Q_G and the warranted pump head H_G corresponding to that flow rate. The manufacturer must warrant that the H(Q) performance curve measured under defined conditions and at the defined rpm (or sometimes at frequency and tension values) crosses the tolerance area around the warranted point as defined according to Table 11.1.

Value	Grade 1 %	Grade 2 %
Flow rate	±4,5	±8
Pump head	±3,0	±5
Pump efficiency	-3,0	-5

Table 11.1. Values relating to Tolerance Factors (TS EN ISO 9906)

The performance of serial manufactured pumps with less than 10 kW inlet power selected with respect to the typical performance curves as indicated in the catalogues may vary. The tolerance factor for such pumps is defined under the standard.



Picture 11.1 Guarantee Control of Flow Rate, Height and Efficiency

Different tolerance ranges may also be defined by mutual agreement. Besides, the following values may be defined with regard to defined conditions and rpm number:

1. Pump efficiency η_G or general efficiency together with the pump drive unit η_{grG}
2. Net positive suction head at warranted flow rate (NPSH)

at the flow rate as defined in Picture 11.1.

Moreover, multiple warranted points and the flow rates and necessary NPSH values at these points may also be warranted under special agreements. However this fact may require the acceptance of broader tolerance ranges upon mutual agreement.

The highest shaft power value may be warranted for the warranted flow rate or operation range.

Unless otherwise expressed in the agreement, warranted values are valid for the following conditions:

a) If the physical or chemical properties of the liquid are not indicated, the warranted operation points are valid for the pump running clean and cold water.
b) The relationship between the warranted values with respect to running clean water and using fluids with different conditions should be indicated in the agreement to be made.

c) Warranties are valid only for pumps tested with procedures and test arrangements as indicated in this international standard.
d) The pump manufacturer is not liable for the determination of the warranted point.

Measuring Tolerances allowed in Tests

Tolerance values as a sum of systematic tolerances resulting from measuring instruments, measuring methods and random tolerances are given in Table 11.2 and such values should be met during tests.

Value	Grade 1	Grade 2
Flow rate	±2,0	±3,5
rpm	±0,5	±0,2
Rotation moment	±1,4	±3,0
Pump total height	±1,5	±5,5
Drive intake power	±1,5	±5,5
Pump shaft power (as calculated from the shaft moment and the rpm)	±1,5	±5,5
Pump shaft power (as calculated from the driving unit power and the motor efficiency)	±2,0	±4,0

Table 11.2. Allowed Measuring Tolerance Values

Test results converted into data based on indicated rpm and density
All data obtained in a number of r.p.m. (n) differing from the specified rpm (n_{sp}) must be çonverted into data based onto specified rpm (n_{sp}). In order to have this conversion effected, the difference between n and n_{sp} and the characteristics of the test liquid and the specified liquid must meet the following conditions:

a) Measuring the flow rate, the pump total head and pump shaft power: $0,5\ n_{sp} \leq n \leq 1,20 n_{sp}$
b) Measuring the efficiency: $0,8\ n_{sp} \leq n \leq 1,20 n_{sp}$
c) Measuring $NPSH_G$: $0,8\ n_{sp} \leq n \leq 1,20 n_{sp}$ ve $0,5\ Q_{opt} \leq Q \leq 1,2\ Q_{opt}$

If these conditions are met, the necessary reduction may be carried out by the following formula:

$$Q_T = Q\frac{n_{sp}}{n} \qquad H_T = H(\frac{n_{sp}}{n})^2 \qquad P_T = P(\frac{n_{sp}}{n})^3 \frac{\rho_{sp}}{\rho} \qquad \eta = \eta_T$$

$$NPSH_T = NPSH\frac{n_{sp}}{n}$$

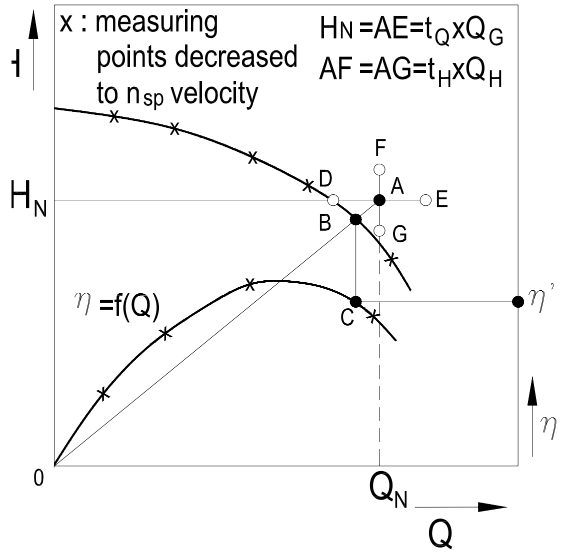
Values with T index in above formulas are values reduced to n_{sp} number of rpm

Meeting Guarantee Conditions

The guarantee condition prescribed for any value described is deemed met whenever the value measured through tests pursuant to this international standard is found to be within the tolerance limits defined in the previous sub-section. Other points located on the H-Q curve are not to be covered by the guarantee, unless these points and their tolerances are not identified in the agreement.

Let's explain this operation in more details. First the results of all measurings are reduced to the number of rpm as defined hereinbefore. The modifications of the pump head and the efficiency with respect to the flow rate are marked on a scaled graph. The performance curves of the pump are obtained by drawing curves through points marked for each value.

Taking H and Q values of the guarantee point as reference, and the rpm and the pump head tolerance values $\pm t_Q$, $\pm t_H$ defined in Table 11.1. a tolerance cross is drawn with $\pm t_Q\ Q_G$ and $\pm t_H\ H_G$ values (Picture 11.2).



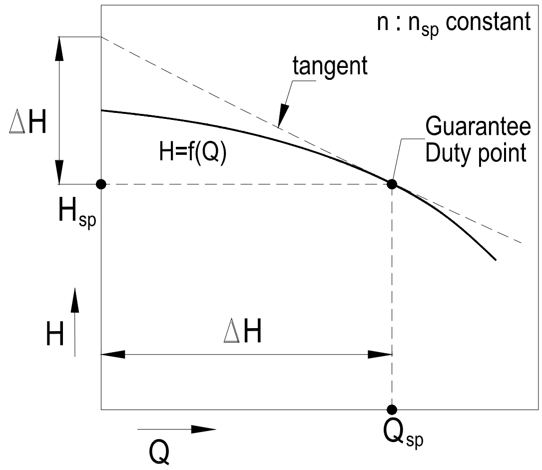
Picture 11.2 Fulfilling Guarantee Conditions

If the pump performance curve H-Q intersects the lines of this cross or at least passes through one of its corners, guarantee conditions relating to H-Q variation.

With regard to efficiency, the point of intersection of the η-Q graph with the perpendicular line drawn from the point where the line joining the guarantee point in the graph H-Q with the origin cuts the H-Q curve is determined. If the value of the efficiency thus found is equal to or more than the value η_G $(1-t_\eta)$, the guarantee condition for the guarantee is deemed to have been met.

11.2 DIN Standard (DIN 1944)

Acceptance test methods defined in DIN 1944 are very different from ISO 9906. Applicable tolerances are established in relation with the inclination of the H-Q characteristic declared at rated r.p.m. number n_N of the pump at the guarantee point. The inclination of the tangent to the H-Q curve at the guarantee point (Q_{sp}, H_{sp}) is taken into consideration as the inclination of the pump characteristics (Picture 11.3):



Picture 11.3 Guarantee Point

Inclination = $\frac{Q_{sp}}{H_{sp}}\frac{\Delta H}{\Delta Q}$

Acceptance test conditions are explained in Table 11.3. In order not to exceed the measuring tolerances given in Table 11.3 the deviation of the values read during the measuring from the average values of the corresponding point should not exceed the following margins:

a) for n rotating speed: ±%5 of the average speed
b) for H pump head: ±%1of the average pump head
c) for P pump shaft power: ±%1,5 of the average shaft power.

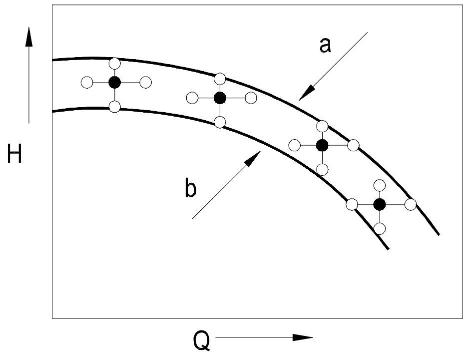
The following conditions should be met in order to reduce the test results obtained in a number of rpm differing from the warranted rpm $n_{sp,}$ to the warranted rpm n_{sp}:

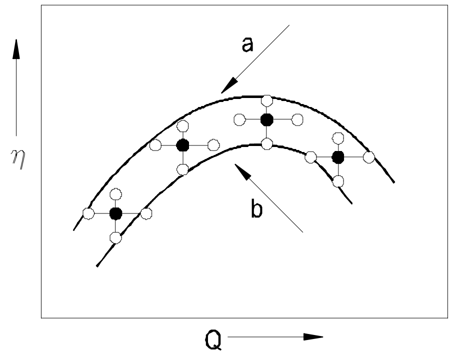
a) for H-Q characteristics n should remain between $0.5n_{sp}$ and 1.05 n_{sp}.
b) for η-Q characteristics n should remain between $0.9n_{sp}$ and 1.05 $n_{sp.}$.

In this case the measured values are reduced to warranted speed n_{sp} by dint of similarity rules.

$$Q_T = Q\frac{n_{sp}}{n} \qquad H_T = H(\frac{n_{sp}}{n})^2 \qquad P_T = P(\frac{n_{sp}}{n})^3 \qquad \eta = \eta_T$$

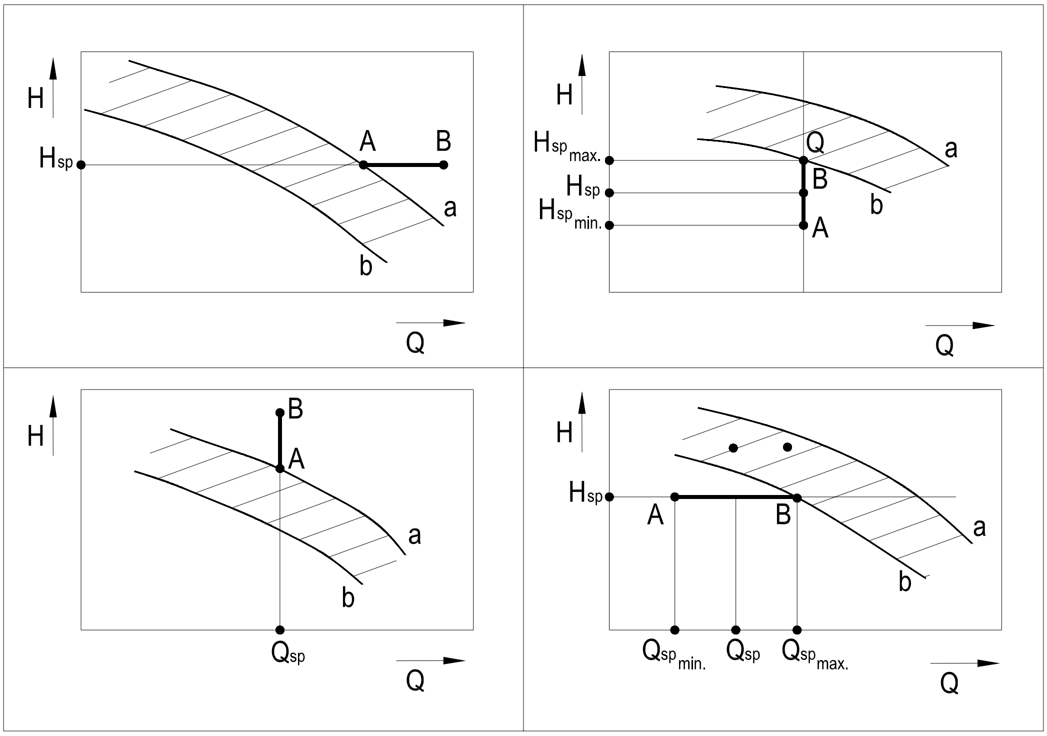
The determination of upper and lower limits of the characteristic curves with respect to measuring tolerances given in Table 3 are shown in Picture 11.4.





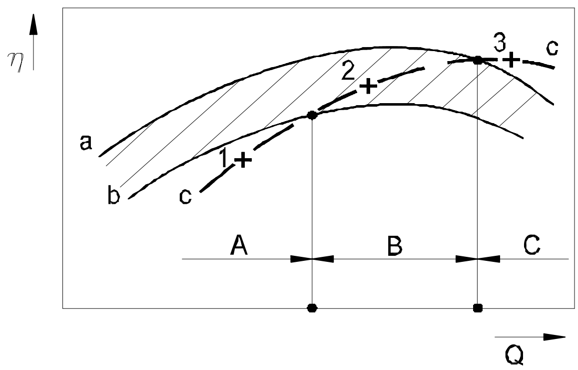
Picture 11.4 Tolerance Band

To check the warranted values Q_{sp} and H_{sp} the tolerance values given in Table 3 are marked on the H-Q graph and tolerance curves are formed both for Q and for H values. The tolerated performance band should touch or cross at least one of the tolerance curves thus formed (Picture 11.5).



Picture 11.5 Guarantee realized

The guarantee remains assured as long as the tolerated efficiency region (region A) measured for the control of the efficiency guarantee remains above the warranted efficiency curve, the warranted efficiency curve falls within or touches the tolerated efficient region measured (region B). Otherwise (region C), the efficiency guarantee conditions are deemed as being not fulfilled (Picture 11.6).



Picture 11.6 Guarantee not fulfilled

11.3 TS Standards

Standards No TS ISO 2548 and TS ISO 3555 published by the Turkish Standards Institute in 1969 with respect to pump acceptance tests, have been abrogated by the Standard No TS EN "Rotodynamic pumps-hydraulic performance test for acceptance-Grades 1 and 2" which entered into force in 2002.

The Standard No: TS EN 9906 has been drawn up on the basis of ISO Standard No: 9906. Therefore, as there are big similarities with ISO 9906, it will not be duplicated here.

Table 11.3. DIN 1944 Acceptance Test Standard

<table>
<tr><th rowspan="2"></th><th colspan="3">Conformity Grade</th></tr>
<tr><th>III</th><th>II</th><th>I</th></tr>
<tr><td>Guarantee conditions for pumps with steep H-Q characteristics
$\frac{Q_N}{H_N}\left|\frac{dH}{dQ}\right| > 0.2$</td><td>The Q flow rate value of the pump measured at acceptance tests with n_N rated rpm, H_N rated pump head and a sufficient NPSH should be between 0.95 to 1.15 times the warranted flow rate Q_N</td><td colspan="2">The Q flow rate of the pump measured at acceptance tests with n_N rated rpm, H_N rated pump head and a sufficient NPSH should be between
0.95 to 1.10 (II)   0.95 to 1.05 (I)
times the warranted flow rate Q_N</td></tr>
<tr><td>Guarantee conditions for pumps with slant H-Q characteristics
$\frac{Q_N}{H_N}\left|\frac{dH}{dQ}\right| \leq 0.2$</td><td>The H pump head value of the pump measured at acceptance tests with n_N rated rpm, H_N rated pump head, Q_N rated flow rate and a sufficient NPSH should be between 0.99 to 1.03 times the warranted pump head H_N</td><td colspan="2">The H pump head value of the pump measured at acceptance tests with n_N rated rpm, H_N rated pump head, Q_N rated flow rate and a sufficient NPSH should be between
0.99 to 1.02 (II)   0.99 to 1.01 (I)
times the warranted pump head H_N</td></tr>
<tr><td>Number of warranted operating point</td><td>1</td><td colspan="2">1 or more</td></tr>
<tr><td>Warranted Efficiency
Note: each efficiency value not defined as warranted efficiency shall be evaluated only as additional information.</td><td>No warranted efficiency. The agreed pump inlet shaft power or the drive motor power should not be exceeded with respect to agreed pump head H.</td><td colspan="2">The efficiency value of the pump measured in acceptance tests with n_N rated rpm and the rated NPSH value should be equal to or greater than the warranted efficiency value H_N.
The efficiency may be warranted ;
a) for a single warranted operating point
b) for a warranted flow rate span
as weighted average,
as arithmetic average,
as planimetric average.</td></tr>
<tr><td>Acceptance Test Methods</td><td colspan="3">With cold water in the test station or with fluid defined at full or reduced rpm or on the installation where the pump will run over the operating conditions</td></tr>
<tr><td>Tolerances allowed in measurings</td><td></td><td></td><td></td></tr>
<tr><td>Flow rate f_Q</td><td>±%3.0</td><td>±%2.0</td><td>±%1.5</td></tr>
<tr><td>Pump head f_H (or specific energy f_Y)</td><td>±%2.0</td><td>±%1.5</td><td>±%1.0</td></tr>
<tr><td>Shaft power f_P</td><td>±%2.0</td><td>±%1.5</td><td>±%1.0</td></tr>
<tr><td>Efficiency f_η</td><td>not defined</td><td>±%3.0</td><td>±%2.0</td></tr>
</table>

PART 12

NOISE EMISSION IN PUMPS AND PUMPING STATIONS

12.1 General

The principal reason of the noise in pumps and pumping stations is the vibrations caused by the flow. These vibrations occur anywhere the flow is separated. Considering the flow structure at pump impeller outlet, there are high energy and low energy fluid zones in every region between two blades. These zones generate periodic pressure fluctuations at the impeller outlet and therefore noise. Another source of noise is the flow separations around the volute tongue arising in operating conditions beyond the design or in case of incorrect design of the tongue. This section does not deal with noises originating from the motor driving the pump and the pump is assumed mechanically perfect (mechanically noiseless). In other terms, only flow originated noises are taken up here.

A summary of definitions and concepts related with noise and noise measurements are given hereinbelow:

Sound is defined as pressure fluctuations on the air. The human ear is capable of hearing sounds with frequency between 16 Hz and 16 000 Hz.

Pressure fluctuations generating the sound may have different forms:

Tone: Pressure variation with single frequency and in sinusoidal form.
Periodic sound: Mixture of basic tons and its harmonics.
Noise: All audible sounds containing tones with different frequencies.

Sound pressure p, the difference of pressure change caused by the sound source with respect to atmospheric pressure value. The intensity of the sound heard by ears is related with the frequency of the sound. Low frequency sounds must have higher sound pressure values in order to be heard under the same intensity with higher frequency (1.000 Hz and up) sounds. The human ear feels the sound pressure variations in a logarithmically changing form. In other words, the sound pressure increase from 1 microbar to 2 microbar is felt almost the same way as increase from 20 to 40 microbar or from 40 to 80 microbar. Thus the **sound pressure level** L is defined as,

$$L_p = 20 \log \left(\frac{p}{p_0} \right)$$

In this formula p is the sound pressure in microbars, $p_0=2.10^{-4}$ is the reference sound pressure in microbars. L_p, is obtained in dB as the sound pressure level of the (p) sound pressure.

When different sound pressure levels are added together, the loudest sound determines the total sound pressure level.

Sound Power and Sound Power Level: sound power is the quantity of the sound energy emanating from the sound source and the level of such power is called sound power level calculated by the formula: $L_w = 10 \log_{10} \frac{P}{P_0}$, where P is the sound power (W) and P_0 is the reference sound power of 10^{-12} W.

Sound Intensity and Sound Intensity Level: sound waves originating from a sound source of W power passing through the A area gives the sound intensity per unit area as I=W/A. The level of the sound intensity is calculated by the relation $L_I = 10 \log_{10} \frac{I}{I_0}$, where I_0 is taken as 10^{-12} W/m^2.

Sound Level, is the sound pressure level obtained as weighted with respect a definite curve. The sound pressure level at each frequency band is taken with a definite weight and the total sound pressure level is obtained. Weigh curves are named as A, B, C, D. Measurings carried out by using these curves are called sound level measurings. Measuring instruments that execute the weighing operation internally and showing the sound level after the measuring are called sound level-meters. The unit of sound level is dBA, dBB, dBC according to the weighing curve used.

12.2 Measuring Noise

Standards DIN 45635, DIN 45633, ISO 3744, TS 2373, ISO 3740 define noise measurements. The standard no. TS 2782 is about the mechanical vibrations of machines.

In general, the purpose of noise measurings is to discover the source of the noise or to determine the level of noise at particular point. Noise measurements are performed in the medium where the noise is located or in specially prepared test rooms.

The section 24 of the Standard No DIN 45635 which relates to noise measuring takes up matters related with pumps. The said section defines the characteristics of the measuring volume and measuring points in centrifugal pumps noise measurements.

The most performed measurement is the determination of the sound level. The results of measurings carried by using sound level-meters can be measured as dBA, dBB, dBC or dBD by using weighing curves. If the purpose is to determine the noise level, to compare it with standards or to investigate whether they are harmful for health, then measuring the sound level with simple level-meters research is sufficient. However if the measuring is carried out for determining which component of the machine generates the noise, then the harmonic frequency distribution should be identified. Special microphones are used for this purpose and the electrical signals obtained at the output are evaluated through spectrum analysis. The choice of such microphones is a matter of expertise.

12.3 Noise Sources

Noises in pumps have two causes: flow originated and mechanic originated. The reasons of flow originated noises may be enumerated as:

1. periodic pressure fluctuations caused by jets and consecutive path flow regions formed at the impeller outlet on account of finite number of blades,
2. In case the pump has vaned diffuser, the interaction of periodic pressure fluctuations formed at the outlet of the impeller with the diffuser blades,
3. Friction of water with solid walls, vortexes coming out of border layer,
4. Cavitation.

Mechanic originated noises result from vibrations caused by the rotating walls touching each other, such as bearings and sealings.

Similar noises are also present in electric motors driving the pump. For example, the cooling fan of the electric motor generates flow originated noise and the bearings generate mechanical originated noises.

12.4 Pumps Noise Levels

The following formulas are proposed in the literature for calculating the noise levels of pumps with different structures. The main parameter in these formulas is the shaft power:

- Volute or collector type pumps

$$L_{wA} = 71 + 13.5 \log \frac{P}{P_0} \qquad 4\ kW \le P \le 2000\ kW$$

- Pumps with vaned return diffusers

$$L_{wA} = 83.5 + 8.5 \log \frac{P}{P_0} \qquad 4\ kW \le P \le 20000\ kW$$

- Axial pumps

$$L_{wA} = 21.5 + 10 \log \frac{P}{P_0} + 57 \frac{Q}{Q_{opt}} \qquad 10\ kW \le P \le 1300\ kW \quad \text{and} \quad 0.77 \le Q/Q_{opt} \le 20000\ kW$$

In these formulas

L_{wA} : is the sound power level in dB
P : is the pump shaft power in kW
P_0 :1 kW
Q/Q_{opt} : is the ratio between the flow rate at the measuring point and the optimum flow rate

These formulas give a noise level somewhat higher than the effective level.

Besides, statistical researches demonstrate that about 10^{-9} to 10^{-6} times the pump shaft power is transformed into noise energy. These values demonstrate a distribution around 30 dB.

12.5 Matters to be Considered for reducing Noise

The pump must be chosen correctly in order to run at maximum efficiency point in the installation it will operate. The operation of the pump with a flow rate lower or higher than its maximum efficiency point will result in increased noise level. Other important matters are :

1. Pump operation outside the cavitation region
2. Low flow speeds preferred in suction and discharge pipes
3. Mechanically correct assembly of the pump with the installation and the drive unit
4. Vibration dampening elements placed between the pump base frame and the concrete foundation
5. Pump attached on vibration absorbers
6. Attaching elements with good noise characteristics used in the installation
7. Sudden diameter changes avoided as much as possible in the installation
8. Large diameter elbows used
9. Pump and pipe connections made with flexible expansion elements

In case these precautions do not suffice to reduce the noise level, the resulting noise may be eliminated by reflection or by using noise absorbing elements. For such applications the character of the noise must be known.

12.6 Environmental Criteria

Maximum noise level values allowed in the Noise Control Regulations are given in the following table.

Noise Source	Noise LEVEL LpA(dB)A
Universal Lathe	95
Concrete Pump	115
Compressor	115
Pump (200 kW -1500 rpm)	110
Tractor	120
Electrical Welding Machine	85
Pneumatic Hammer	90-105
Living Room	55-65
Vehicles	75-85
Military Aircraft	110-140

Data relating to noise levels of electric motors are given in the following table.

NOISE LEVEL				
MOTOR POWER	**SOUND PRESSURE LEVEL (d B)***			
	Pump		**Pump and Motor**	
KW	**1450 rpm**	**3000 rpm**	**1450 rpm**	**3000 rpm**
0.55	54	56	61	63
0.75	54	56	61	65
1.1	56	58	63	66
1.5	57	59	65	69
2.2	59	61	66	70
3	60	62	68	72
4	61	63	70	74
5.5	62	64	71	81
7.5	63	65	72	82
11	67	70	73	83
15	69	72	74	83
18.5	70	73	75	84
22	71	75	76	84
30	74	77	78	91
37	75	78	79	92
45	76	79	80	93
55	77	80	81	93
75	78	81	82	94
90	79	82	83	94
110	80	83	85	95
132	81	84	86	95
160	82	-	87	96
200	83	-	88	-
250	85	-	89	-

*

Approximate values measured 1m away from the pump in a open area above the sound reflecting surface without sound protective barrier (ISO 3744)

PART 13

MONITORING PUMP OPERATING CHARACTERISTICS AND CONJECTURAL – PREVENTIVE MAINTENANCE

Some operating conditions require the continuous monitoring and recording of the pump operating conditions. This monitoring may be for the purposes of evaluating the system performance as well as for anticipated estimation of the breakdowns which may occur.

Such systems have generally a central unit called PC or PLC equipped with a CPU (processor). This central unit contains a data collection card which transforms the electrical signals coming from the measuring sensors into digital data. These data translated into digital form are processed in the central unit by special softwares as required and monitored. The main components of this kind of system are shown in Figure 13.1.

Pressures in the suction and delivery pipes of the pump are translated into electrical signals (volts or milliamperes) by pressure transducers and then translated into digital data by the A/D card and conveyed to the computer. Besides, the analog outputs of the flow-meter, moment-transducer, optical tachometer and electrical power meter are translated into digital values by the A/D card and transmitted to the computer. A program running on the computer transforms these values into requested units by means of calibration curves and displays on the screen. In such a system all characteristics curves of the pump is displayed in real time mode.

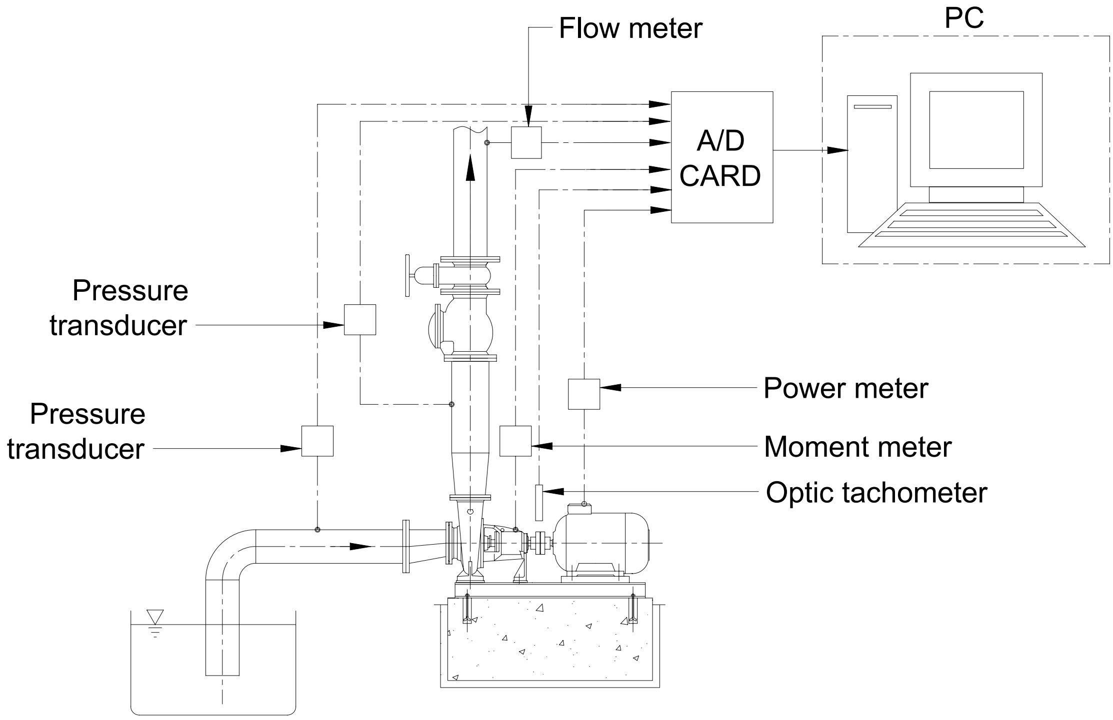


Figure 13.1 System where Pump Operating Conditions are Monitored

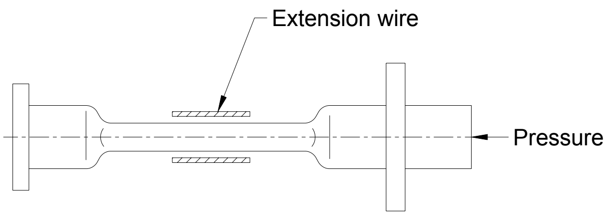
13.1 Pressure Measurings

The purpose of measuring the pressure in pumps is to define the pump total head, $NPSH_G$ and the flow rate. Pressure gages may be classified as liquid column, mechanical and electrical output types. Among these types, the water column manometers and mechanical manometers (Bourdon tube manometers) are analog instruments and do not generate electrical outputs. Thus such analog devices are not suitable for utilization in the measuring systems to be taken up here. However, these analog instruments are widely used in the industry on account of their practicality.

Pressure sensors called as pressure transducers or transmitters in the literature, generating electrical output as tension (volt) or as current intensity (milliamperes) may be classified according to their operation principle as follows.

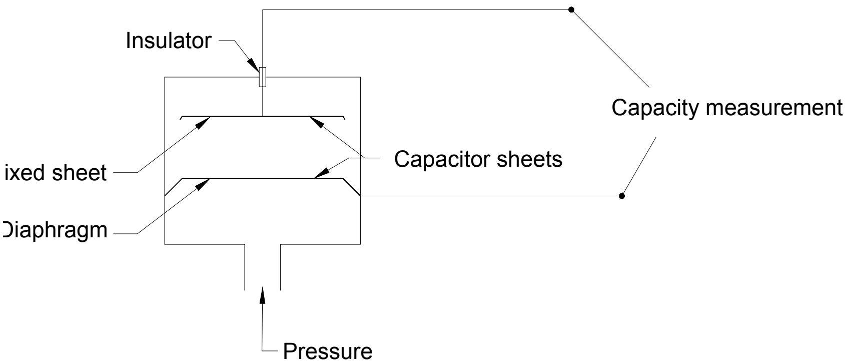
- Strain gage pressure-transducers
- Capacitive type pressure-transducers
- Piezoelectric type pressure-transducers

Strain gage pressure-meters work according to the measurement of the resistivity change occurring as a result of the elongation of gages fixed on the elastic surface affected by the pressure.



Picture 13.2 Strain Gage Pressure-meter

Capacitive type pressure-meters work according to the principle of measuring the capacity created by the variation of the distance between the first sheet of the capacitor where the pressure acts upon and changes its form and the second sheet.



Picture 13.3 Capacitive Type Pressure-meter

Piezoelectric type pressure-meters use crystals such as quartz, barium, titanium. These crystals generate electrical tension when affected by pressure. Piezoelectric type pressure-meters work on the principle of measuring the value of this tension. Such type pressure-meters can not measure the true value of the pressure but measure pressure changes.

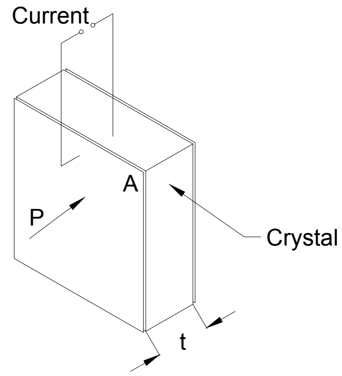


Figure 13.4 Piezoelectric Type Pressure-meter

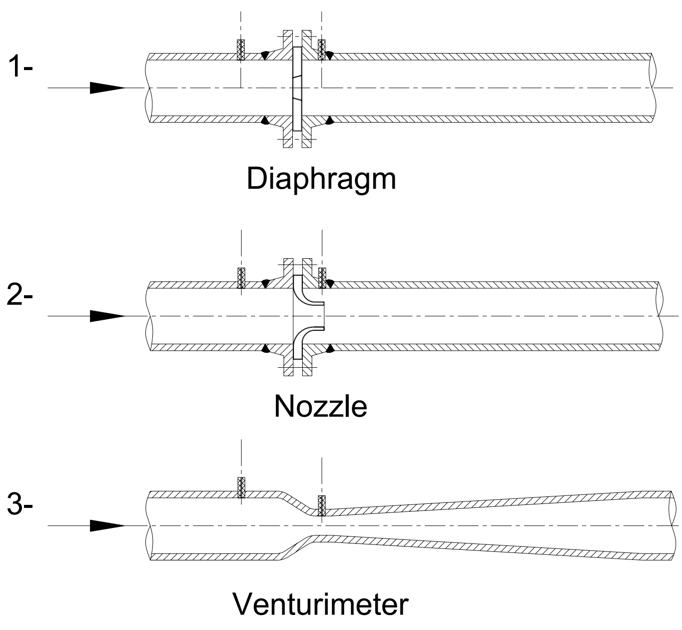
It is important to remember that such type of pressure-meters must be periodically calibrated.

13.2 Flow Rate Measurings

The mass rate of the flow or the volumetric flow rate of the fluid passing through the pump may be measured by very different methods, as explained hereinafter:

13.2.1 Measuring the Flow Rate by Diaphragm (orifice), Nozzle and Venturi Meter

Flow rate measurements by diaphragm (orifice), nozzle and venturi meter are explained in detail in the Standard No: ISO-5167. The following diagrams show schematically how the diaphragm, nozzle and venturi meter are placed in the measuring system and the points from which pressure measurements must be carried out. Pressure values measured are used for calculating the flow rate by using the formulas and the correction factors described in the standard.



Picture 13.5 Measuring Flow Rate by Diaphragm, Nozzle and Venturi Meter

13.2.2 Magnetic Flow-Meters

They are the ISO standard and most widely used flow-meters. The difference of tension that the fluid in the magnetically uniform field creates between the electrodes is directly proportional with the speed of the fluid. The ratio coefficient is the function of the calibration constant, which includes the conductivity effect depending on the intensity of the magnetic field, the distance between the electrodes and the type of the fluid.
Magnetic flow-meters may reach rated ambiguities of 0,1 % or better and may be used as calibration flow-meters. They are suitable for all type of conductive fluids, i.e. dirty, multi-phase, chemicals, foodstuff and liquid metals. They are capable of precise measurings even under very slow flow velocities. The fluid which flow rate will be measured must be conductor and the minimum applicable conductivity has to be 1 μS/cm. They may not be used for measuring the flow rate of petroleum, oil derivates, many vegetable oils and chemicals.

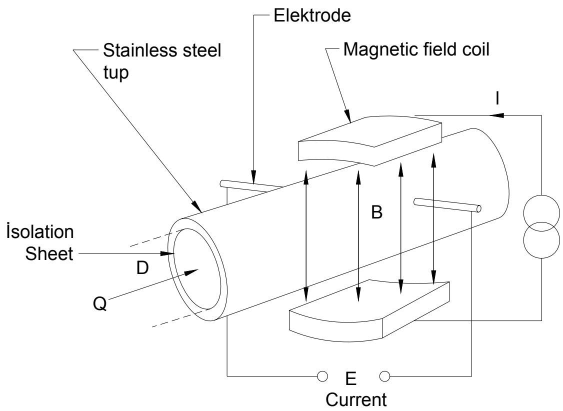


Figure 13.6 Magnetic Flow-meter

13.2.3 Ultrasonic Flow-meters

Ultrasonic flow-meters are the most widespread type of flow-meters after magnetic devices. Advantageous because of their portability, but disadvantageous because of their high price due to their complex electronic structure. the flow rate is measured by the Doppler principle. The frequencies of the sound waves generated by an ultrasonic power source change when passing through the moving medium. The modification of the sound wave frequencies is proportional to the flow rate.
0,5 % rated ambiguity values may be attained when measuring the flow rate of all fluids (liquid or gas) by ultrasonic flow-meters. They are very suitable for multi-phase flows. Highly prices, they may be broken down easily.

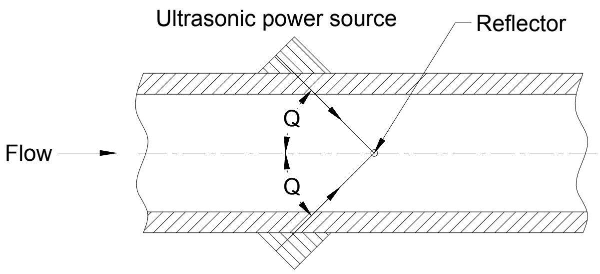
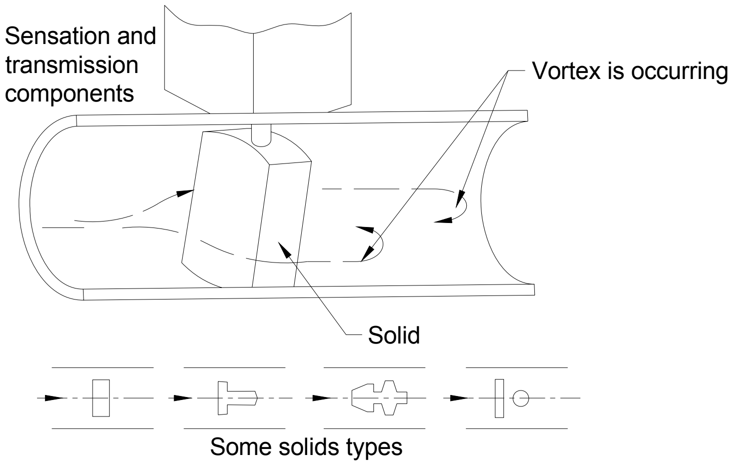


Figure 13.7 Ultrasonic Flow-meter

13.2.4 Vortex Flow-meters

The frequency of the vortexes occurring behind a blunt object placed in the flow space is proportional to the flow rate. Although the ambiguities of vortex flow-meters are higher than that of magnetic and ultrasonic flow-meters, they are much cheaper and shorter.



Picture 13.8 Vortex Flow-meter

13.2.5 Measuring Flow Rate by Weir in Open Channels

One of the methods for measuring the flow rate in open channels with rectangular cross section is the use of weir. The amount of height increase of the water flowing over the weir is measured and the flow rate passing through the channel is calculated. Following matters should be taken into consideration when measuring the flow rate by the weir method:

- The weir plate must be made of stainless material and should be perpendicularly fixed on the bottom and the side walls and in a manner to have its symmetry axis coinciding with the symmetry axis of the channel. The space between the weir plate and the channel surface should be leak proof.
- The flow in the channel should not be waving.
- The bottom inclination of the channel should be less than 5/1000 and should remain constant all along the channel.
- The average water velocity in the channel should be less than 0.5 m/s.
- The position of the water rising height measurement point behind the weir:

The position of the weir and the additional devices that must be put in the channel for measuring the flow rate by weir are shown in Figure 13.9. The weir shape may be 90^0 triangular, rectangular with narrowed side and rectangular without narrowed side as indicated in Figure 13.10.

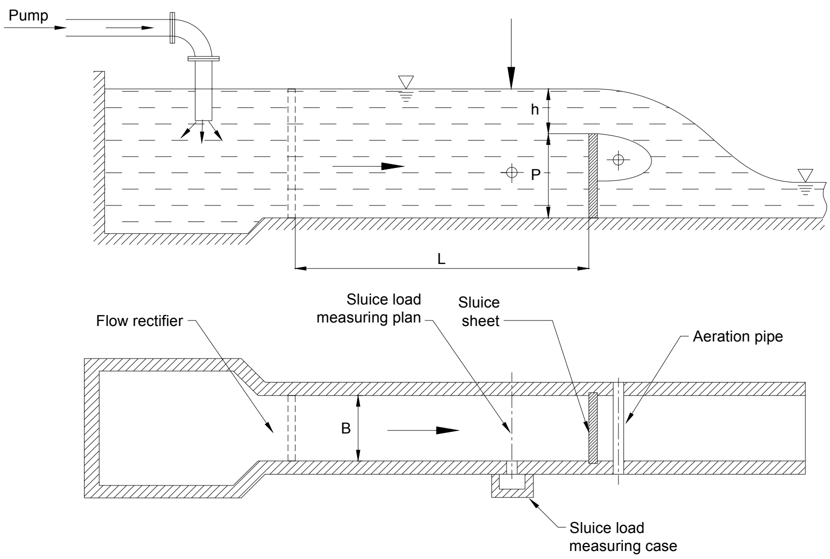


Figure 13.9 Diagram of the Channel Structure lodging the Weir

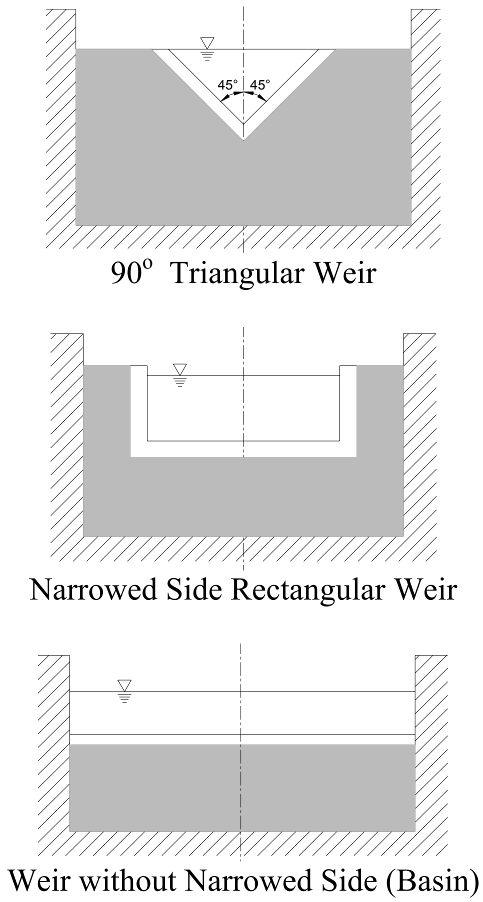


Figure 13.10 Weir Sections

13.3 Power Measurings

May be carried out by two ways, electrically and mechanically:

In electrical power measuring the simplest method is to use the electricity meter. The value read in the meter for a definite length of time (i.e. 5 kWh) is divided to the corresponding period to obtain the power drawn from the mains by the system consisting of the pump motor and the electricity meter. There are some difficulties in monitoring such a system continually. Another way of measuring the electric power is to measure the tension and the current intensity of the power line entering the motor. As the current and the tension are not in the same phase, these values are multiplied by cos φ and the power value is so obtained. The pump shaft power may be calculated from the value thus obtained by taking into account the efficiency of the electric motor. There are power analyzers that measure and indicate all these values and simultaneously translating them into electric signals for transmission to the data collection card.

In the mechanical determination of the power, the most widely applied method is to attach in series a moment transducer on the shaft located between the electric motor and the pump. The value of the moment measured by the moment transducer is multiplied by the angular speed of the shaft and the shaft power of the pump is obtained. Since the transducer output and the angular speed signal may directly transferred to the data card, the measuring and recording is continuous.

13.4 RPM Measurings

Measuring the rpmmay be done by two ways: optically and mechanically. In the optical method, the reflection frequency of the light emitted by an optical tachometer reflected by a reflector fixed on the shaft is determined and the number of r.p.m. is obtained. In mechanical tachometers the shaft of the tachometer is let touch the rotating shaft. The tension produced by the generator in the mechanical tachometer is commensurate with the number of r.p.m. Some moment transducers are also capable of giving the output simultaneously in r.p.m.

13.5 Temperature Measurings

Thermo-elements are widely used for temperature measurements. The tension value formed by thermo-elements proportionally with the temperature are strengthened and displayed by an indicator.

13.6 Vibration Measurings

Vibration measurings are carried out by using vibration sensors called accelerometers. The signal output from vibration sensors is evaluated and interpreted by a spectrum analyzer. These instruments are in different sizes with different functions. The basic function of the spectrum analyzers is to carry the FFT analyses of the signals sent by vibration sensors. The main purposes of vibration measurings may be classified as periodic measurings for forewarning maintenance and measurings for discovering the source of the breakdown. Pumps have vibration charts under normal operating conditions. Values obtained in periodic measurings for forewarning maintenance purposes are compared with the vibration charts. If there is a difference, measurings are made for discovering the source of the breakdown.

13.7 Conjectural Maintenance

The method of timely intervention by tracking the amplitude of the vibrations of the areas of the pump where different breakdowns are coming out according to their frequencies, as of their occurrence is called conjectural-preventive maintenance.

Vibration measurings are made by a spectrum analyzer attached to an accelerometer placed on the inspected region of the pump. The electrical signal coming from the accelerometer is processed in the spectrum analyzer and its FFT is calculated. As a result of this operation the frequency-amplitude variations of the vibrations are obtained.
Each pump has its own vibration signal which is the function of several parameters. This signature must be measured in connection with the structure of the pump after its manufacture and kept. Later on, periodical measurings are compared with the vibration signature of the pump and conjectural maintenance may be carried out. The pump operating point should be the same when carrying out these measurings; otherwise different values will be obtained since the flow generated noise will be different in different valve openings.
The type of breakdowns which may occur in a pump and which may be identified through vibration measurings may be in the form of, imbalance, coupling misalignment, loose chassis, weak anchoring, ball bearing breakdowns, cavitation, and electric motor breakdowns.
The frequency spectrum, time wave graph, phase graph are obtained by the measurings performed.

PART 14

WATER HAMMER

14.1 General

The pressure increase or decrease generated in the pressurized fluid lines on account of momentum variations due to the changes of the flow velocity is called water hammer. For instance in a pipe line where the velocity is 3 m/s the pressure increase occurring as a result of sudden stop of the flow will be about 36 bars. This value added to the operating pressure under normal conditions may cause the pipe to blow off. In the event a closed valve in the system is opened, the flow velocity of the water will increase from the still value generating a pressure drop in the line and if necessary measures are not taken, collapsing may occur in the pipe line. Therefore water hammer calculations must be made and necessary precautions must be taken in pipe lines.

14.2 Water Hammer Formation and Development

Water hammer is caused by events such as turning down, closing down or opening a valve in a pipe line; starting up, shutting down or changing the r.p.m. of the pump; closing down a non-return valve etc.

The Figure 14.1 illustrates a system consisting of a valve, pipe and a reservoir. In case the valve is completely closed down the water flow velocity in the system (V_0) will fall down to null at the valve cross section, thereby generating a pressure surge in the system which will move forth in the pipe towards the reservoir at a sound speed (a). When this surge will attain the reservoir all the pipe line is subject to a pressure above the operating value ($p+\Delta p$). At that time there is no flow in the pipe. The pressure surge (compressive surge) will reflect as expanding surge, i.e. a flow towards the reservoir will occur in the pipe and when the expanding surge will attain the cross section of the closed valve all the pipe line will be at the operating pressure (p). The expanding surge will reflect from the valve as expanding surge and the pressure values inside the pipe will be ($p-\Delta p$). It should be noted that the ($p-\Delta p$) value can not be below the absolute vacuum value. When the expanding surge will attain the reservoir it will be reversed, i.e. reflected as pressure surge and when it will reach the cross section of the valve the system will be equivalent to closed valve position and the cycle will repeat. In reality this fact will develop in progressively diminishing pattern of the hammer effect (Δp) on account of frictions. The study of the development of the water hammer demonstrates that facts are identical to each other in $4L/a$ time periods where (L) is the length of the pipe (frictions neglected) and therefore the (T) period of the water hammer is $4L/a$. In case the valve is not fully but partially closed down, similar events occur as well.

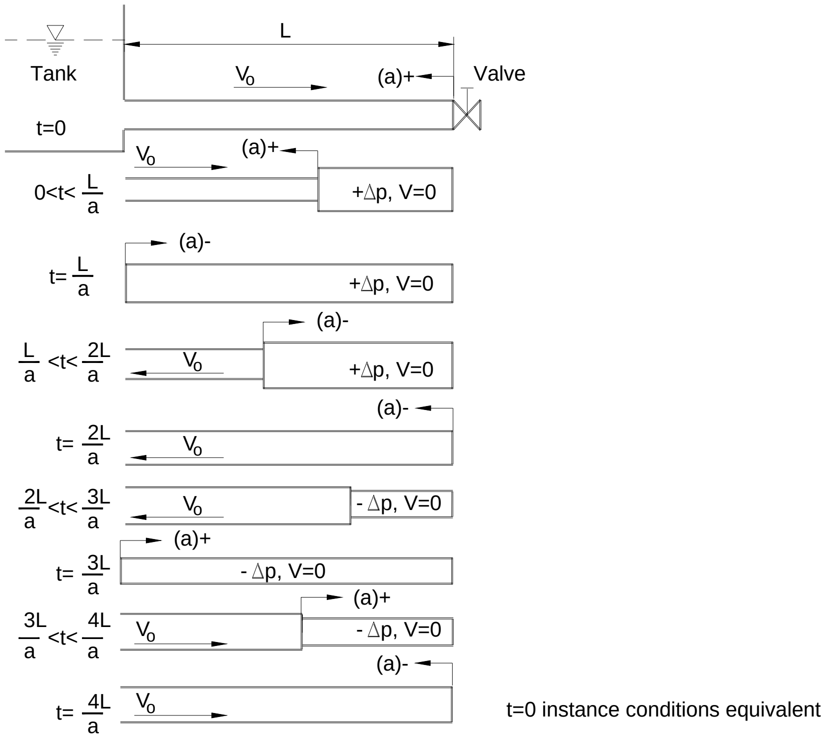


Figure 14.1 Water Hammer Variation by Time in Reservoir-Pipe-Valve System

The variation by time of the water hammer at definite points (a, b, c and d) of a reservoir-pipe-valve system is shown in Figure 14.2.

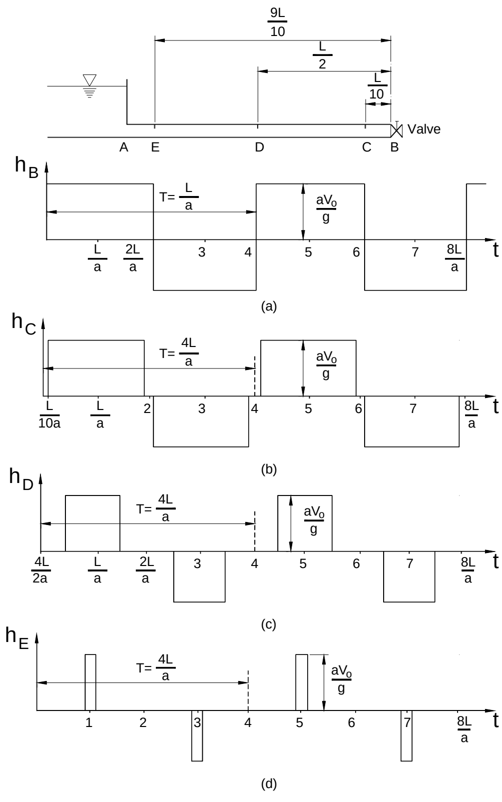


Figure 14.2 Development of Water Hammer in some Points of the Pipe Line in a Reservoir-Pipe-Valve System

In the event the valve is located between two reservoirs and is suddenly closed down fully, the development of the water hammer occurring in the system is shown in Figure 14.3.

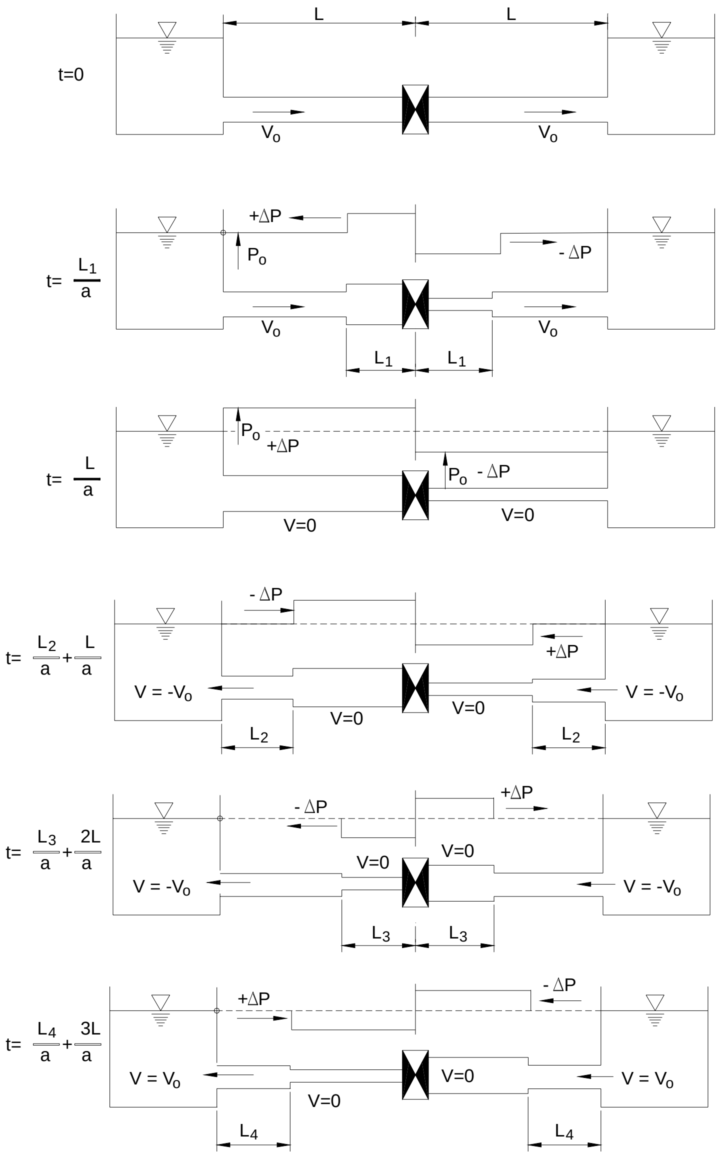


Figure 14.3 Water Hammer Development when Valve is Located between Two Reservoirs and when Suddenly and Fully Closed Down

Valve Closing Extent and Time

Valve closing down according to the closing time (τ) is called as follows:

$\tau = 0$	complete close down
$\tau \leq 2L/a$	fast close down
$\tau > 2L/a$	slow close down

The closing extent of a valve may be full or partial. In partial close down the speed does not fall down to zero, as it is the case in full close down, and is reduced to a definite V value.

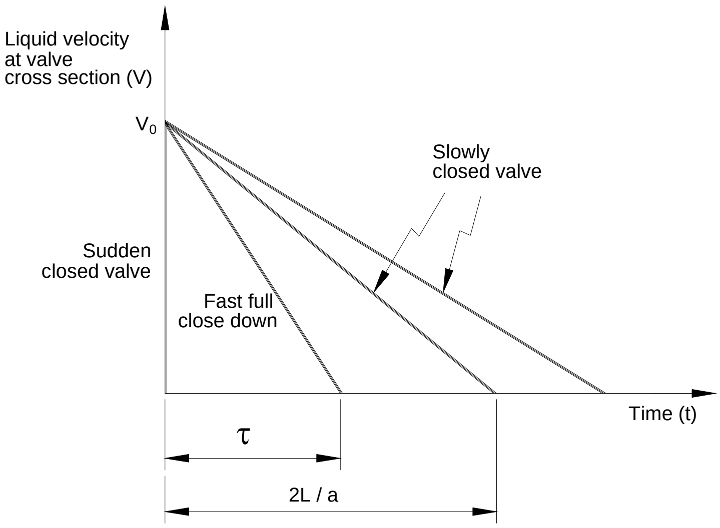


Figure 14.4 Valve Close down Extent and Time

Calculating Water Hammer Intensity

The value of the water hammer taking place in case of quick and complete close down is calculated by the following formula:

$$\Delta p = \rho \, a \, \Delta V$$

In this formula Δp (Pa) represents the pressure increase, ρ (kg/m^3) represents the density of the liquid, ΔV (m/sn) amount of velocity decrease at valve cross section. The value of this pressure increase in terms of water column height (m) is obtained as hereunder:

$$\Delta H = \frac{a \, \Delta V}{g}$$

In case of full close down $\Delta V = V_0$.

When the close down is slow, the value of ΔV is the difference between the speed V_0 at the beginning of closing at the valve cross section and the flow speed at the moment 2L/a at valve cross section. Assuming that the closing down operation is linear, the value of the water hammer in terms of water column height is calculated by the following formula:

$$\Delta H = \frac{2 \, L \, \Delta V}{g \tau}$$

Sound Speed

The speed of the sound in a pipe containing water is calculated by the formula

$$a = \sqrt{\frac{1}{\rho(\frac{1}{K} + \frac{D}{Es})}}$$

where a is the propagation speed (m/s) of the sound, ρ is the density (kg/m^3) of the fluid, K is the volumetric compression module (Pa) of the fluid, D is the bore (m) of the pipe, s is the wall thickness (m) of the pipe and E is the elasticity module (Pa) of the pipe material. In preliminary calculations, the sound speed for steel pipes may be taken as 1200 m/s.

14.3 Pump Installations fitted with Non-return Valve

When the electric power will fail in a pump-non-return valve-pipe system shown in Figure 14.5 the water will tend to flow in the pipe towards the suction tank but the non-return valve at the outlet of the pump will shut off and will not allow this return. Supposing that the non-return valve will shut off suddenly, an expanding surge will be formed moving towards the tank and the pipe pressure will decrease by aV_0/g. The Figure 14.5 illustrates the time variation of the pressure values at cross section B caused by the event.

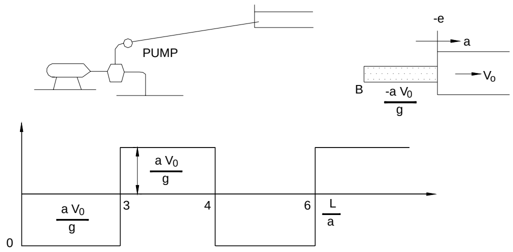


Figure 14.5 Water Hammer in Pump Installations with Non-returnValve

14.4 Controlling Water Hammer and Precautions

The cheapest solution as a precaution for protecting the pipe lines from the water hammer is to increase the inertia of the rotating parts. In order to provide such solution a flywheel with suitable size may be fixed onto the shaft. Thus the pump stopping in case of power failure will be retarded and the slow shut off of the valve will be simulated. As a consequence, the effect of the water hammer will diminish.

The most widely used water hammer protection method in Turkey is to use an air surge tank. As seen in Figure 14.6, an air tank is placed on the discharge pipe after the pump and the non-return valve. There is a compressed air at the top of the tank. When the electric power is cut and the non-return valve is closed down suddenly, the pump is substituted by the air tank which starts to feed the system. During this feeding the pressure in the air tank will decrease. This will result in slow reduction of the water velocity in the discharge pipe. When the flow towards the non-return valve will start in the pipe, the water entering into the tank will re-compress the air and during this process the speed of the water will be gradually reduced. Because of friction this process will continue as 4 to 5 periods with gradually diminishing intensity and then fades out. In order to increase these friction losses a nozzle is placed at the outlet of the tank in a manner to increase the flow losses in the direction of the tank inlet. Another way of increasing losses is to fix a non-return valve and a small bore bypass pipe at the outlet of the tank. The non-return valve will not allow flows towards the tank inlet and the flow will enter the tank by crossing the small bore pipe which has a high resistance. This will cause high pressure losses and provide shorter time for dampening the water hammer.

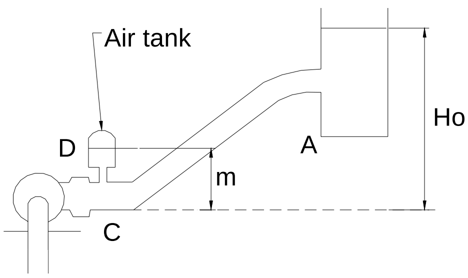


Figure 14.6 Using Air Tank

PART 15

PUMP START-UP

In our days pumps are most widely driven by electric motors. Internal combustion engines, gas and steam turbines are used for the purposes of driving pumps in areas where electric power is not available.

15.1 Starting-up Pumps with Electric Motor

On account of various advantages, squirrel cage asynchronous electric motors are preferred for driving pumps. Compared to other motor types, squirrel cage motors are easy to manufacture, highly safe to operate and require seldom maintenance. Besides these advantages, speed adjustment units which emerged by technical developments and changing the motor r.p.m. by modifying the frequency paved the way for squirrel cage asynchronous electric motors to replace direct current motors and slip ring asynchronous motors. For these reasons the utilization percentage of squirrel cage asynchronous motors raised up to 80-90%.

15.1.1 Number of rpm of Electric Motors

The synchronous speed of single and three-phase electric motors are calculated through the mains frequency (f) and the even number of poles (p) according to the following formula:

$$\boxed{n = \frac{60 \times f}{p}}$$

Accordingly, the rpm number of electric motors with respect to the number of poles and electric frequency are given in the following table:

Number of Poles	**50 Hz rpm**	**60 Hz rpm**
2	3000	3600
4	1500	1800
6	1000	1200
8	750	900
10	600	720
12	500	600
14	428	514
16	375	450

Table 15.1 Electric Motors rpm

True motor speeds are 1 to 8% lower than synchronous speeds on account of slippage. For example the rpm number of 15 kW and 4-pole Gamak brand motor is 1460.

15.1.2 Environmental Conditions for Electric Motors

Single and three-phase motor performances are indicated with reference to operation at 1000 meters altitude and 40°C temperature. When run in other altitudes and environment temperature, the rated powers of standard motors manufactured in class F insulation change as per following rates.

Ambient Temperature	°C	40	45	50	55	60
Rated Power	%	100	95	90	85	80

Altitude	m	1000	2000	3000	4000
Rated Power	%	100	95	90	80

Table 15.2 Electric Motors Performances modified by Temperature and Altitude

If both the environmental temperature and the altitude change simultaneously, the rated power is multiplied by the coefficients of temperature and altitude in order to obtain the new power.

15.1.3 Insulation Class

When measured in accordance with IEC 60034-1 standards (F) insulation class motors allow maximum coil temperature increase of 105°C at 40°C environment temperature, with 10°C safety margin.

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Figure 15.1 Electric Motors Insulation Classes

15.1.4 Protection Levels

Protection levels of electric motors are given in Table 15.3.

IDENTIFICATION NUMBER	1.Number: Level of protection against touching and penetration of foreign substances	2.Number: Level of protection against water penetration
0	Without protection	Without protection
1	Protection against large size foreign substances	Protection against vertical dropping water

2	Protection against medium size foreign substances	Protection against slant dropping water
3	Protection against small size foreign substances	Protection against sprinkling water
4	Protection against granulated foreign substances	Protection against water spills
5	Protection against dust accumulation	Protection against water jets
6	Protection against dust penetration	Protection against flood
7	---	Protection against Immersion
8	---	Protection against staying under water

Table 15.3 Protection Levels of Electric Motors

In accordance with TS 3209, EN 60529 and IEC 60034-5, the protection level is expressed with the initials of the word “International Protection” IP and by two digit numbers as indicated in Table 15.4.

<table>
<tr><th rowspan="2">Sign</th><th>First Number</th><th>Second Number</th></tr>
<tr><th>Against solid substances</th><th>Against liquid substances</th></tr>
<tr><td>IP 54</td><td>Full protection against inadvertent touch of parts in the casing which move or are under tension.</td><td>Protection against water spills on the motor from any direction.</td></tr>
<tr><td>IP 55</td><td rowspan="2">Protection against dust accumulation susceptible of causing damage. Although dust penetration is not fully stopped, penetration of dust in a quantity capable of affecting the operation of the motor is prevented.</td><td>Protection against damage to the motor by a water spout towards the motor from any direction</td></tr>
<tr><td>IP 56</td><td>Protection against water penetration in the motor on account of storm on ship’s deck or water jet.</td></tr>
<tr><td>IP 68</td><td>Full protection against dust penetration</td><td>Protection against continuous immersion in water</td></tr>
</table>

Table 15.4 Protection Levels

Electrical units measured (V: tension; I: intensity) and the formulas for calculating motor input and output powers are given in Table 15.5.

Current type	Input Power kW	Output Power	
		kW	HP
Direct Current	{ EMBED Equation.3 }	{ EMBED Equation.3 }	{ EMBED Equation.3 }
Single Phase	{ EMBED Equation.3 }	{ EMBED Equation.3 }	{ EMBED Equation.3 }
Three Phase	{ EMBED Equation.3 }	{ ΕΜΒΕΔ Εθυατιον.3 }	{ EMBED Equation.3 }

Table 15.5 Electric Motors Input and Output Powers

Table 15.6 summarizes the connection type and the way of starting-up motors in normal (continuous) operation with respect to line tension and stator winding type.

<table>
<tr><td rowspan="3">Line (mains) Tension (V)</td><td colspan="4">Stator winding type as indicated on the motor label</td></tr>
<tr><td colspan="2">220 V (Δ) / 380 V (λ), 220V (Δ)</td><td colspan="2">380 V (Δ), 380 / 660 V</td></tr>
<tr><td>Normal operation</td><td>Start-up</td><td>Normal operation</td><td>Start-up</td></tr>
<tr><td>220</td><td>Δ</td><td>Direct</td><td>-</td><td>-</td></tr>
<tr><td>380</td><td>λ</td><td>Direct</td><td>Δ</td><td>Direct or λ /Δ</td></tr>
<tr><td>660</td><td>-</td><td>-</td><td>λ</td><td>Direct</td></tr>
</table>

Table 15.6 Motor Connection Type

15.1.5 Motor Start-Stop Numbers (Start-up Frequency)

As the motor start-up current is greater that the nominal current at the time the pump is started up by asynchronous electric motor, the coils of the motor are excessively heated up. This fact also damages the pump shaft, the impeller and the ball bearings. Therefore the number of motor start-up should not exceed a definite number.

Motor Rated Power	**≤5,5**	**7,5-15**	**18,5-30**	**37-45**	**55-75**	**90-132**	**160-200**	**250-315**	**≥400**
Maximum number of start-up per hour	**18**	**12**	**10**	**8**	**6**	**5**	**4**	**3**	**2**

Table 15.7 Start-up Frequency

15.2 Pumps Start-up by Internal Combustion Engines

The Figure 15.2 shows the characteristics curves of an internal combustion engine. The region marked (a) is the continuous operation area of the engine. If pumps are driven by an internal combustion engine the pump operation point must remain within the continuous operation area of the internal combustion engine. The definition curve of a pump suitable for operation with an engine which characteristics curve is indicated in Figure 15.2 is given in Figure 15.3.

{ EMBED AutoCAD.Drawing.15 }

Figure 15.2 Characteristics Curves of an Internal Combustion Engine

{ EMBED AutoCAD.Drawing.15 }

Figure 15.3 Characteristics Curve of a Centrifugal Pumi

Example:

The motor pump characteristics curve of a centrifugal pump coupled with a diesel engine is given in the Figure. This pump is operated as directly coupled with an electric motor with 1500 rpm and is requested to produce a total head of H_m =10 mWC at Q=7 lt/s flow rate. Let's see if it is possible, and if so, how the impeller diameter should be changed:

Solution:

As it would be noticed from the motor pump characteristics curves (I) in Figure 15.5,the motor pump rpm increases and the curve goes up by the motor power/rpm characteristics, as the power drawn will decrease by the reduced flow rate.

The first thing to do is to use the rpm values on the curve (I) and to assume that the pump is the same as itself, and to draw the H_m =f (Q) characteristics curve (II) for 1.500 r.p.m. by using Rateau theorems.

Whereas in the formulas,

{ EMBED Equation.3 } and { EMBED Equation.3 }

the rpm number on each point on the (I) curve is different and although the n'=1500 rpm value is taken the same for each point, the (n) values are read on the curve and the H'm and Q' values will be calculated for 1.500 rpm (For example, for 1710 rpm. Q=11lt/s, Hm=10,6 mWC; for 1500 rpm. Q'=(1500/1710). Q'=0,877. 11=9,65 lt/s, H'm= $(1500/1710)^2$. Hm=0,769. 10,6=8,16 mWC). The points on the II curve corresponding to points on the curve { EMBED Equation.3 }are indicated on the figure. When the H'm= f (Q') curve is drawn for n'= 1500 r.p.m constant, we notice that the point A pertaining to Q'=7 lt/s and H'm=10mWC values remain below the curve. Therefore, as the number of rpm can not be modified, it is possible for the same pump to produce the said characteristics at 1.500 rpm, by reducing the impeller diameter.

In order to find the modifications to be made on the impeller outlet diameter, the pump should be assumed identical to itself and the Rateau theorems should be used.

As n' =1500 rpm is constant and the modifications in b_2 and { EMBED Equation.3 }$_2$ are negligible for small diameter changes, we can take

{ EMBED Equation.3 } and { EMBED Equation.3 }

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Figure 15.4 Effect of Diameter Ratio on Characteristics

As it would be noticed the modification is { EMBED Equation.3 }. Whereas { EMBED Equation.3 }lt/s and { EMBED Equation.3 }mWC, therefore { EMBED Equation.3 } will remain constant.

Thus the points of intersection of the lines to be drawn from the origin zero of the { EMBED Equation.3 } axis with the curves { EMBED Equation.3 } will correspond to each other. This fact is schematically indicated particularly in Figure 15.4.

In order to find the point { EMBED Equation.3 } of the point { EMBED Equation.3 } on the curve{ EMBED Equation.3 } it is sufficient to draw the ray line joining the point O and{ EMBED Equation.3 }.

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Figure 15.5 Shifting from Motor Pumps Characteristics to other Characteristic Curves - Example

If this operation is performed on Figure 15.5, for the point { EMBED Equation.3 } we find { EMBED Equation.3 }lt/s and { EMBED Equation.3 }mWC.

As values are checked;
{ EMBED Equation.3 }

{ EMBED Equation.3 }
{ EMBED Equation.3 }

We notice that ratios remain constant within the curve accuracy and readable scaled values.
Since { EMBED Equation.3 } or { EMBED Equation.3 }therefore { EMBED Equation.3 } ; { EMBED Equation.3 }

In other terms, whenever n=1500 rpm in order to get the values { EMBED Equation.3 }lt/s, { EMBED Equation.3 }mWC the outlet diameter of the impeller should be reduced by lathing out the original diameter for 3%.

Similarly, by utilizing Rateau theorems the { EMBED Equation.3 }characteristic curve (III) at the fixed speed of 1.500 rpm may be drawn out. If calculations are made according to values obtained from the curve (II),

for { EMBED Equation.3		for { EMBED Equation.3 } diameter :	
n= 1500 r.p.m.		n=1500 rpm	
{ EMBED	{ EMBED	{ EMBED	{ EMBED Equation.3
4	11,80	4.0,944=3,77 lt/s	11,80.0,944=11,14 mWC
6	11,5	5,66>>	10,85>>
7	10,9	6,61>>	10,3>>
8	10,1	7,55>>	9,5>>
9	8,9	8,5>>	8,4>>
10	7,4	9,44>>	6,98>>
11	5,7	10,38>>	{ EMBED Equation.3
12	3,4	1,32 >>{ EMBED	3,21>>

Table 15.8 Pump Characteristics Change by Small Impeller Diameter Modifications

The new characteristics curve (III) is drawn out according to D_2 diameter. This way of computing generates results very close to reality for small diameter changes. Nonetheless this practice will prove wrong for bigger diameter modifications, as the self similitude of the pumps will disappear and the specific speed will change.

15.3 Using Frequency Converter

In variable flow rate systems, like heating systems, where the heating level is modified (for saving energy) by changing the flow rate of the fluid circulated in the system the flow rate of the pump is controlled by modifying the number of revolutions of the pump by changing the frequency of the electric current. Thus the pump operation point will remain within the high efficiency region for all flow rate values, as shown in Figure 15.6. In case the pump flow rate is adjusted by means 0f a valve, the pump will run with low efficiency.

- Flow rate changed by modifying the rpm. by a frequency converter

x Flow rate changed by reducing valve

{ EMBED AutoCAD.Drawing.15 }

Figure 15.6 Using Frequency Converter

Frequency increase is not very much welcomed in systems with frequency converter as it may overheat the motor. When the system is designed at design stage the frequency value must be drawn up with regard to frequency reduction.

Motor Start-up Current

Motor start-up current valves are given in the following table for DOL and { EMBED AutoCAD.Drawing.15 } start-up.

DOL	$6 \times I_n$
{ EMBED AutoCAD.Drawing.15 }	$2 \times I_n$

Starting current values of electric motors :

1) Direct (Δ) starting :
   Starting Current : 6 x In > 5 seconds

2) Star – Delta starting :
   Starting current : 2 x In >15 seconds

In : Current at nominal power (Ampere)

{ EMBED AutoCAD.Drawing.15 }

Figure 15.7 Pump Performance Curve Modification with respect to the Number of RPM in a System with Four Pumps run with a Frequency Converter

Following are the formulas used for translation

{ EMBED Equation.3 } { EMBED Equation.3 } { EMBED Equation.3 } { EMBED Equation.3 }

RPM **Flow rate** **Pump total head** **Motor power**

15.4 Failures in Three-phase Motors and Solutions

FAILURES IN THREE-PHASE MOTORS AND SOLUTIONS		
FAILURES	**POSSIBLE CAUSES**	**SOLUTIONS**
Motor does not start; no magnetic hum.	1- Fuse taken out or blown out. 2- Thermic fuse blown out. 3- Unsuitable or loose cable connections 4- No contact in the switch.	1- Check fuse. 2- Check thermic fuse. 3- Check cable connections. 4- Check the switch.
Motor does not start; magnetic hum exists, thermic fuse blows out	1- Wrong cable connections. 2- Mains voltage low. 3- Relay fails. 4- Mechanical jamming in the motor. 5- Motor runs with two phases.	1- Check cable connections. 2- Identify cause and remedy. 3- Change the relay. 4- Check motor bearings. 5- Identify cause and remedy..
Motor starts up and runs, but shortly after the thermic fuse blows out.	1- Mains voltage low. 2- Thermic fuse fails. 3- Motor under excessive load. 4- Capacitor fails	1- Identify cause and remedy.. 2- Check the thermic fuse. 3- Check motor compatibility with load. 4- Check the capacitor and connections.
Motor impossible to start-up.	**Motor runs idle :** 1- No mains voltage. 2- Mains voltage low. 3- Motor runs with two phases. 4- Loose connections.	 1- Identify cause and remedy. 2- Identify cause and remedy. 3- Identify cause and remedy. 4- Tighten up the connections.

	5- Wrong connections 6- Mechanical jamming in the motor **Motor under load, in addition to above:** 7- Motor under excessive load. 8- Relay fails. 9- Timer switch period too short.	5- Check the connections. 6- Check motor bearings. 7- Check motor compatibility with load. 8- Change the relay. 9- Modify the timer switch period.
Noisy operation	1- Loose parts.(footing, belt pulley etc.) 2- Fan blades broken or bent. 3- Motor bearings down 4- Motor coupling wrong. 5- Ball bearing tension spring down. 6- Fan contacts. 7- Motor runs with two phases. 8- Loose connections.	1- Tighten up the connections. 2- Change the fan. 3- Check motor bearings. 4- Check and rectify the motor coupling. 5- Change the ball bearing tension spring. 6- Prevent contacting. 7- Identify cause and remedy. 8- Tighten up the connections.
Excessive heating up	1- Mains voltage low. 2- Motor under excessive load 3- Motor runs with two phases. 4- Fan broken. 5- Bearings down. 6- Environment temperature too high. 7- Motor air suction jammed. 8- Short circuit in windings.	1- Identify cause and remedy. 2- Check motor compatibility with load.. 3- Identify cause and remedy. 4- Change the fan. 5- Check motor bearings. 6- Use special motor. 7- Identify cause and remedy. 8- Call the service.

Table 15.9

*The motor should absolutely be grounded; connection surface to the mains should be large and made via suitable cable shoes in order to prevent short circuits.

PART 16

STANDARDS RELATING TO PUMPS

16.1 EU Standards

Norm pump standards (DIN 24255) EN 733 and EN 22858
Acceptance test standards ISO 9906
Balance standards ISO 1949
Sealings standards EN 12756 (DIN 24250)
DIN 24260 Pump Installations
DIN 1944
Standards relating to pump body and impeller materials
Flange Standards
Connection Pipes Standards
Noise Measuring Standards
Flow Rate Measuring Standards

16.2 CE Sign (Safety measures)

INTRODUCTION

Within the context of works of harmonization with the technical regulations of the European Union, the Ministerial Decree promulgated in 29 April 1997 set forth which ministry or governmental institution will carry out which harmonization works. As a result of these works the said institutions have completed to great extent the technical legislation harmonization works falling within their purview.

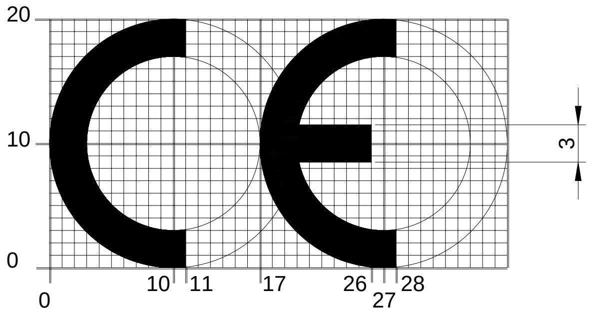
Upon the entry into force of directives relating to the CE sign in Turkey in the form of regulations, products covered by the said regulations both imported and manufactured in Turkey will not be marketed in Turkey without the CE sign.

CE SIGN

The CE sign is a sign of compliance with the European Union New Approach Directives evidencing the healthiness and the safety of the product onto which it is affected with respect to humans, animals and environment. Criteria relating thereto are provided in the New Approach Directives of the European Union (EU). Any product falling within the context of one or more New Approach Directive(s), actually numbering about 20, but not carrying the CE sign is impossible to be introduced in the EU markets. The production of the manufacturers in compliance with the EU harmonized standards relating to directives is very important for evidencing the compliance with New Approach Directives.

The CE sign is not a quality brand or a product brand by itself. Therefore it may not be used in lieu of brands like TSE, VDE, TÜV, GS. The CE sign indicating compliance with basic requirements provided in New Approach Directives and therefore to safety criteria, is at the same time the indication of the product performance characteristics and used together with the product certification brands.

The “CE” compliance logo is as shown hereinbelow. The minimum size is 5 mm. The CE sign should be affixed onto the product or, if not possible, on the packing.



TECHNICAL FILE

The manufacturer is obliged to prepare the technical file upon the completion of all the procedures related with his product as explained hereinbefore. A copy of the technical file is kept with the manufacturer himself and another copy should remain available with the importer or the resident authorized representative for being presented when demanded by public authorities.

Documents prescribed in EU New Approach Directives, test reports and some other documentation is required in the Technical Files relating to the CE sign. Documents suggested to be contained in the Technical File are as follows:

1. General definition of the product,
2. Drawings related with the design and production, diagrams of the components, parts, circuits diagrams and information about the operation procedure,
3. The list of standards used in the manufacture of the product,
4. In cases where standards could not be applied, solutions accepted for harmonization with basic safety requirements prescribed by the relevant directives,
5. Design results etc.
6. Test documents and reports,
7. EU Type Inspection Document,
8. Quality Assurance System Document,
9. Declaration of Conformity.

Above-mentioned documents may differ from one directive to another. This fact may also change according to the Module chosen by the manufacturer within the context of directives with respect to the CE sign.

HARMONIZED STANDARDS

Harmonized Standards are the European Standards prepared by the European Union Standardization Organizations CEN, CENELEC and ETSI and agreed upon y the member organizations (EN, HD).

DECLARATION OF CONFORMITY

The Declaration of Conformity is drawn up by the manufacturer himself. The Declaration of Conformity is a statement of the manufacturer to the effect that the production he made is in compliance with the requirements related with the product. Information about the EU directives and standards relating to the compliance of the product should be mentioned in the Declaration of Conformity. Moreover, the type and the model numbers of the products declared to be compliant must also be indicated.

The full name and the signature of the person filling in the declaration and the place and date of issue must be endorsed at the foot of the Declaration of Conformity. The Declaration of Conformity should be made ready before the introduction of the product in the market, together with the Technical File.

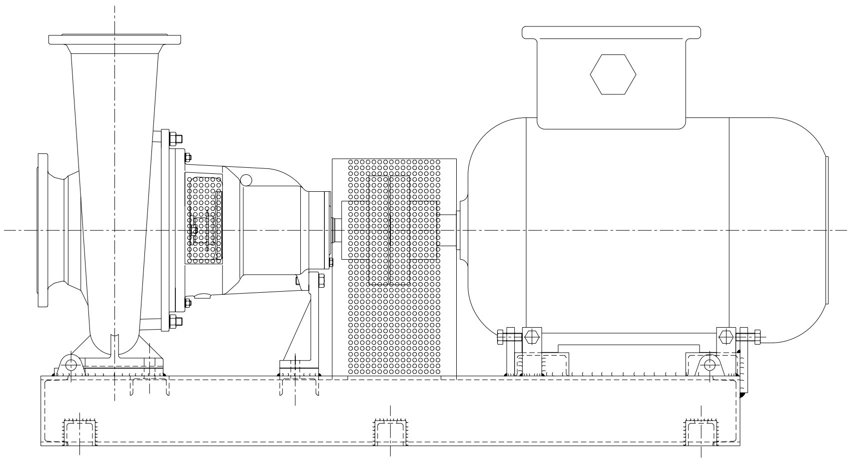


Figure 16.1

BUREAU VERITAS

CE

Energy & Process

CERTIFICATE OF ATTESTATION

B.V. Job Ref: IDD.059.05.D21

Project/Installation: Türbosan's Consultancy Of 98/37/EC Machinery Directive and 73/23/EEC LVD Directive For Horizantal and Vertical Clean and Waste Water Pumps / Yatay ve Düşey Milli Temiz ve Pis Su Pompaları için 98/37/EC Makina Direktifi ve 73/23/EEC Alçak Gerilim Direktifine Göre CE Markalama İçin Yapılan Danışmanlık

B.V. No: TUR_5507

Inspection ordered to B.V. by (1):
Türbosan Türbomakinalar San. Ve Tic. A.Ş.

Ref.of the Order to B.V.: IDD.059.05.D21

Supplier: Türbosan Türbomakinalar San. Ve Tic. A.Ş.

Sub-supplier, if any: NIL

Copies to: Türbosan Türbomakinalar San. Ve Tic. A.Ş., BV Master File

(1) the addressee of the original of this report ;

Description of the Supply / Subject of inspection:
Consultancy Of 98/37/EC Machinery Directive and 73/23/EEC LVD Directive For Same As Declaration Of Conformity's Trade Mark / Üreticinin uygunluk deklarasyonunda belirtilen 98/37/EC Makina Direktifi ve 73/23/EEC Alçak Gerilim Direktifi gerekliliklerine uygunluğun sağlanmasına yönelik teknik danışmanlık :

Pump Types / Pompa Tipleri : Horizantal and Vertical Clean and Waste Water Pumps / Yatay ve Düşey Milli Temiz ve Pis Su Pompaları : NORM, ÇEP, KOT/KAT/YKP, ÇAP, KAÇ, TÜR-TRY , HMS,VMS, TÜR – Y/S , PROSES,EKS – E/K

This report covers the whole of the supply: x oYES (no more inspection planned) NO (part of the supply still to be inspected)

Scope of the B.V. Survey:
Consultancy of Preparing Technical File, Reviewing Test Reports and Application and Settlement Of CE Directive's Rules / Teknik Dosya hazırlanmasına danışmanlık, Test Raporlarının İncelenmesi ve CE markalama direktifi kurallarının uygulanmasının sağlanması

This supply complies with the following applicable document (s): (4)
BV Rules / BV Kuralları
98/37/EC Machinery Directive / 98/37/EC Makina Direktifi
73/23/EEC LVD Directive / 73/23/EEC Alçak Gerilim Direktifi
EN 292-1, EN 292-2, EN 1050, EN 12162, EN 23661, EN 25199, EN 9905, EN 9906, EN 9908, EN 12873, EN 809, EN 12639, EN 294, EN 349, EN 733, EN 20858, EN 12576

(2) and only for parts of the document(s) which concern the certification or the relevant service provided by Bureau Veritas.
List of enclosures: NIL
(or reference to enclosed list)
The report is valid together with enclosures (if listed). Only pages of enclosures (or parts of pages) which are signed/stamped, are considered as part of this report.

Marking and Stamping on the items: CE Marking / CE Markalama
Particulars or comments: THE UNDERSIGNED REPORT THAT IS ABOVE MENTIONED SUPPLY WAS INSPECTED IN CONFORMITY WITH 98/37/EC MACHINERY DIRECTIVE AND 73/23/EEC LVD DIRECTIVE

Date of Issuance: 23.06.2005

Issued by:
Name: Bülent SÜZER
Sign:

Validated by:
Name: ~~İnci~~ DOĞUSAN
Sign:

17 BUREAU VERITAS 1828 IND TURKEY

Location of inspection: Türbosan Türbomakinalar San. Ve Tic. A.Ş. / İSTANBUL

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BUREAU VERITAS

CE

Energy & Process

CERTIFICATE OF ATTESTATION

B.V. Job Ref: IDD.059.05.D21

Project/Installation: Türbosan's Consultancy Of 98/37/EC Machinery Directive, 73/23/EEC LVD Directive and 89/336/EEC EMC Directive For Submersible Clean Water, Waste Water and Sewage Pumps / Dalgıç Temiz Su, Pis Su ve Kanalizasyon Pompaları için 98/37/EC Makina Direktifi, 73/23/EEC Alçak Gerilim Direktifi ve 89/336/EEC Elektromanyetik Uyumluluk Direktifine Göre CE Markalama İçin Yapılan Danışmanlık

B.V. No: TUR_5443

Inspection ordered to B.V. by (1):
Türbosan Türbomakinalar San. Ve Tic. A.Ş.

Ref.of the Order to B.V.: IDD.059.05.D21

Supplier: Türbosan Türbomakinalar San. Ve Tic. A.Ş.

Sub-supplier, if any: NIL

Copies to: Türbosan Türbomakinalar San. Ve Tic. A.Ş., BV Master File

(1) the addressee of the original of this report ;

Description of the Supply / Subject of inspection:
Consultancy Of 98/37/EC Machinery Directive, 73/23/EEC LVD Directive and 89/336/EEC EMC Directive For Same As Declaration Of Conformity's Trade Mark / Üreticinin uygunluk deklarasyonunda belirtilen 98/37/EC Makina Direktifi, 73/23/EEC Alçak Gerilim Direktifi ve 89/336/EEC Elektromanyetik Uyumluluk Direktifi gerekliliklerine uygunluğun sağlanmasına yönelik teknik danışmanlık :

Pump Types / Pompa Tipleri : Submersible Clean Water, Waste Water and Sewage Pumps / Dalgıç Temiz Su, Pis Su ve Kanalizasyon Pompaları : DAS, PARPO, DAÇ, DAÇ-E/K, DAP

This report covers the whole of the supply: x oYES (no more inspection planned) NO (part of the supply still to be inspected)

Scope of the B.V. Survey:
Consultancy of Preparing Technical File, Reviewing Test Reports and Application and Settlement Of CE Directive's Rules / Teknik Dosya hazırlanmasına danışmanlık, Test Raporlarının İncelenmesi ve CE markalama direktifi kurallarının uygulanmasının sağlanması

This supply complies with the following applicable document (s): (4)
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98/37/EC Machinery Directive / 98/37/EC Makina Direktifi
73/23/EEC LVD Directive / 73/23/EEC Alçak Gerilim Direktifi
89/336/EEC EMC Directive / 89/336/EEC Elektromanyetik Uyumluluk Direktifi
EN 292-1, EN 292-2, EN 1050, EN 12162, EN 23661, EN 25199, EN 9905, EN 9906, EN 9908, EN 60034-9, EN 60529, EN 55011, EN 61000-6-4, EN 61000-3-2
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Date of Issuance: 23.06.2005

Issued by:
Name: Bülent SÜZER
Sign:

Validated by:
Name: Yakup UÇAR
Sign:

Location of inspection: Türbosan Türbomakinalar San. Ve Tic. A.Ş. / İSTANBUL

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СИСТЕМА СЕРТИФИКАЦИИ ГОСТ Р
ГОССТАНДАРТ РОССИИ

1784795

ПРИЛОЖЕНИЕ

К сертификату соответствия № РОСС TR.AE41.B02262

Перечень конкретной продукции, на которую распространяется действие сертификата соответствия

код ОК 005 (ОКП) / код ТН ВЭД СНГ	Наименование и обозначение продукции, ее изготовитель	Обозначение документации, по которой выпускается продукция
36 3100	Насосы динамические:	Техническая документация изготовителя
36 3100 8413 70 819 0	- насосы центробежные Типа NORM, CAP;	
36 3100 8413 70 659 0	- насосы двухступенчатые Типа CEP;	
36 3100 8413 70 899 0	- насосы многоступенчатые Типа KAT;	
36 3100 8413 70 210 0	- насосы погружные Типа DAS - DAC - PARPO.	
	ИЗГОТОВИТЕЛЬ: «TURBOSAN TURBO MAKİNALAR SAN. ve TİC. A.Ş.» Gümüşsuyu Cad., Hastane Yolu No:1, 34020 Topkapı- İstanbul, Турция	

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ГОССТАНДАРТ РОССИИ

СЕРТИФИКАТ СООТВЕТСТВИЯ

№ РОСС TR.AE41.B02262

Срок действия с 03.08.2007 по 02.08.2008

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ОРГАН ПО СЕРТИФИКАЦИИ рег. № РОСС RU.0001.10AE41
ПРОДУКЦИИ И УСЛУГ ООО "ТУЛЬСКИЙ ЦЕНТР СЕРТИФИКАЦИИ, МОНИТОРИНГА И СОДЕЙСТВИЯ ЗАЩИТЕ ПРАВ ПОТРЕБИТЕЛЕЙ"
юридический адрес: 300012, г.Тула, ул.Мира, 54, почтовый адрес:300026 г.Тула, пр.Ленина,102, тел. /факс 35-19-47, 35-21-85, e-mail tulatest@mail.ru

ПРОДУКЦИЯ Насосы динамические, перечень типов см. Приложение бланк № 1784795
по технической документации изготовителя
Серийный выпуск

код ОК 005 (ОКП): 36 3100

СООТВЕТСТВУЕТ ТРЕБОВАНИЯМ НОРМАТИВНЫХ ДОКУМЕНТОВ
ГОСТ 6134-87 П.п. 1.2, 2.4, 3.8-3.10, 3.10а

код ТН ВЭД России: 8413

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СЕРТИФИКАТ ВЫДАН «TURBOSAN TURBO MAKİNALAR SAN. ve TİC. A.Ş.»
Gümüşsuyu Cad., Hastane Yolu No:1, 34020 Topkapı- İstanbul, Турция, тел. +90 212 567 95 50, факс +90 212 612 47 41

НА ОСНОВАНИИ Протоколы сертификационных испытаний № 89/11-07, № 89/12-07, № 89/13, № 89/14-07 от 25.06.2007, ИЦ ЗАО «СПЕКТР-К», рег. № РОСС RU.0001.21MM02, 156019, г. Кострома, ул. П.Щербины, 9

ДОПОЛНИТЕЛЬНАЯ ИНФОРМАЦИЯ Знак соответствия по ГОСТ Р 50460 наносится на изделие и в сопроводительной технической документации.
Схема сертификации 3.

М.П.

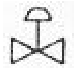
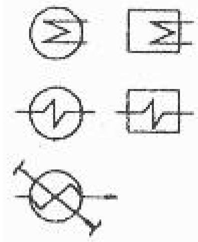
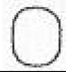
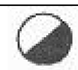
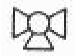
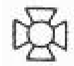
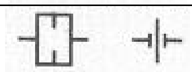
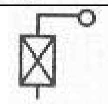
Руководитель органа _____ А.В.Сизов
подпись / инициалы, фамилия

Эксперт _____ А.В.Евплова
подпись / инициалы, фамилия

Сертификат имеет юридическую силу на всей территории Российской Федерации

16.3 Signs and Symbols

SYMBOL	DEFINITION	SYMBOL	DEFINITION
	Flanged pipe		Automatic pressure valve
	90° Flanged elbow		Safety cock
	45° Flanged elbow		Counterweight safety cock
	Flanged plug		Non-return valve general
	Flanged cover		Swing type non-return valve
	Flanged reducer		Straight lift type non-return valve
	Flanged eccentric reducer		Corner lift type non-return valve
	Flanged T elbow		Ball non-return valve
	90° Flanged footing elbow		Bottom valve
	Y pipe		Pressure-meter
	Flange		Flow-meter
	Compensators general		Thermometer
	U compensator		Temperature measuring point
	Twin offset compensator		Speedometer
	Bellow compensator		Vibrometer
	Valves general		Pump general
	Non-return valve		Centrifugal pump
	Gate valve		Gear pump
	Butterfly valve		High pressure pump
	Flow rate or pressure regulating valve		Vacuum pump, compressor
	Solenoid valve		Variable flow rate compressor, vacuum pump
	Motor valve		Electric motor general

SYMBOL	DEFINITION	SYMBOL	DEFINITION
	Differential pressure controlled diaphragm valve		Exchanger
	Straight spherical valve		Filter
	Corner type spherical valve		Dirt trap
	3-way spherical valve		Tank
	Pressure reducing straight valve		Diaphragm tank
	Pressure reducing corner valve		Coil
	Straight cock		Steam trap
	Corner type cock		Light signal
	3-way cock		Sound signal
	4-way cock		Orifice
			Bleeder, ventilation cock

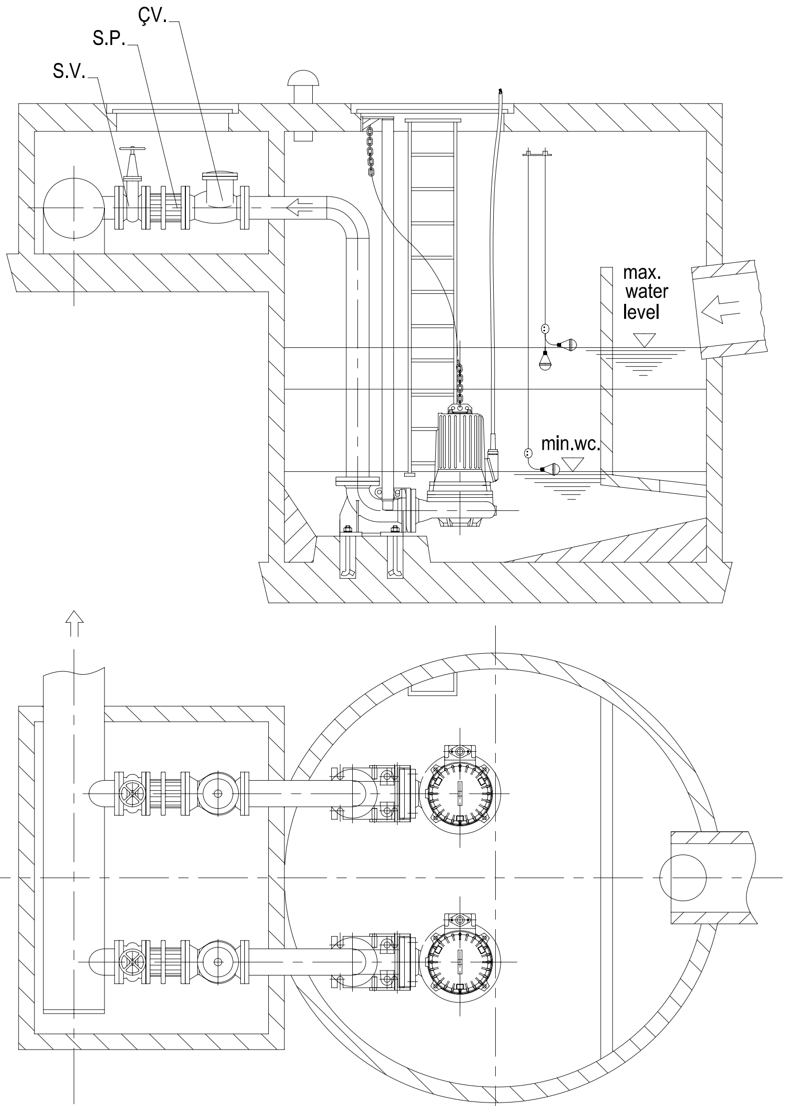
PART 17

PUMP TYPE STATIONS

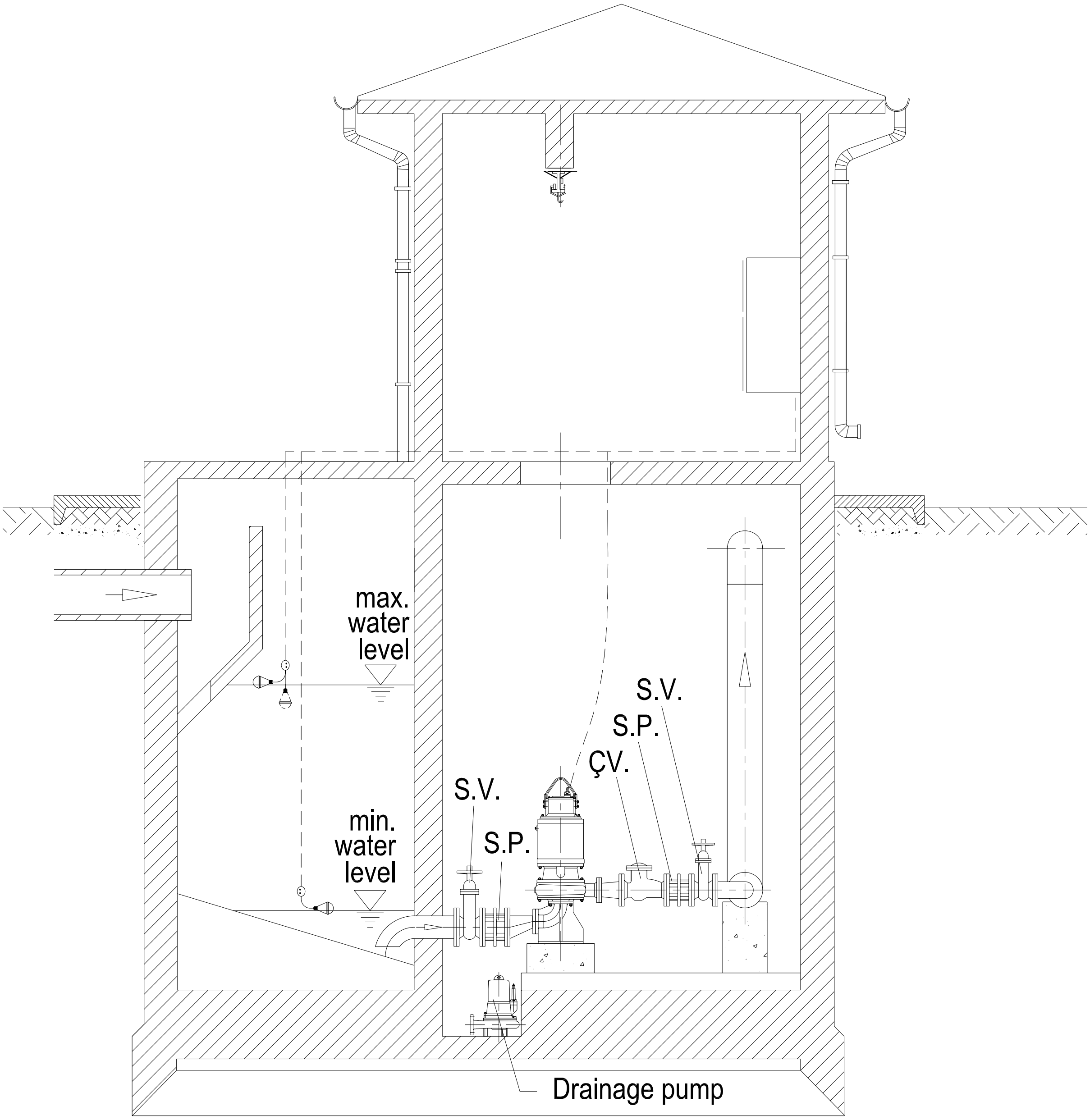
Typical mounting projects of several pump stations are indicated hereinbelow.

Abbreviations used in pictures:

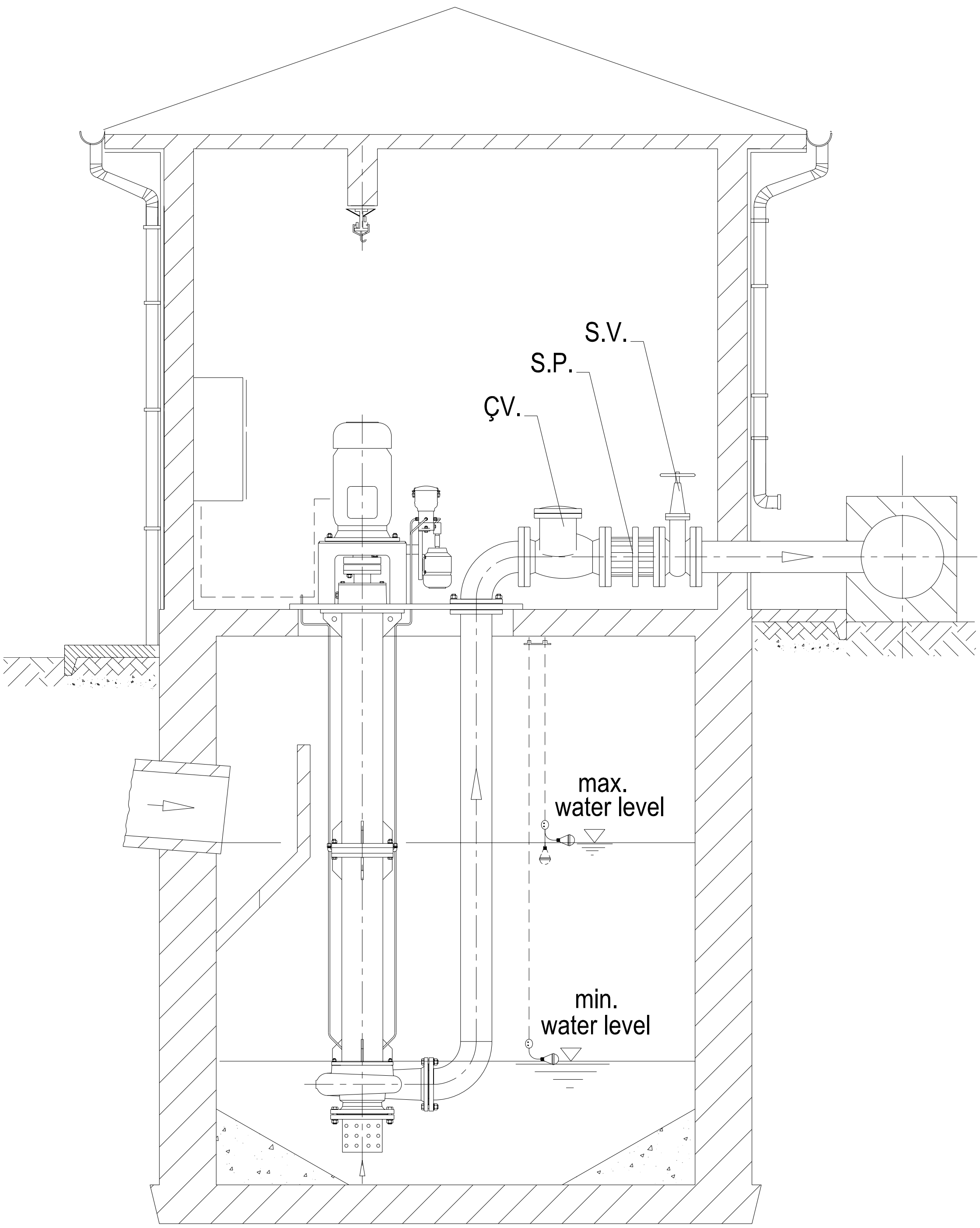
K.V: Butterfly valve,
S.V: Gate valve,
ÇV: Non-return valve,
S.P: Mounting and dismounting element, expansion part
D.K: Vertical cone
K.P: Coaxial conic part
FV: Flap valve



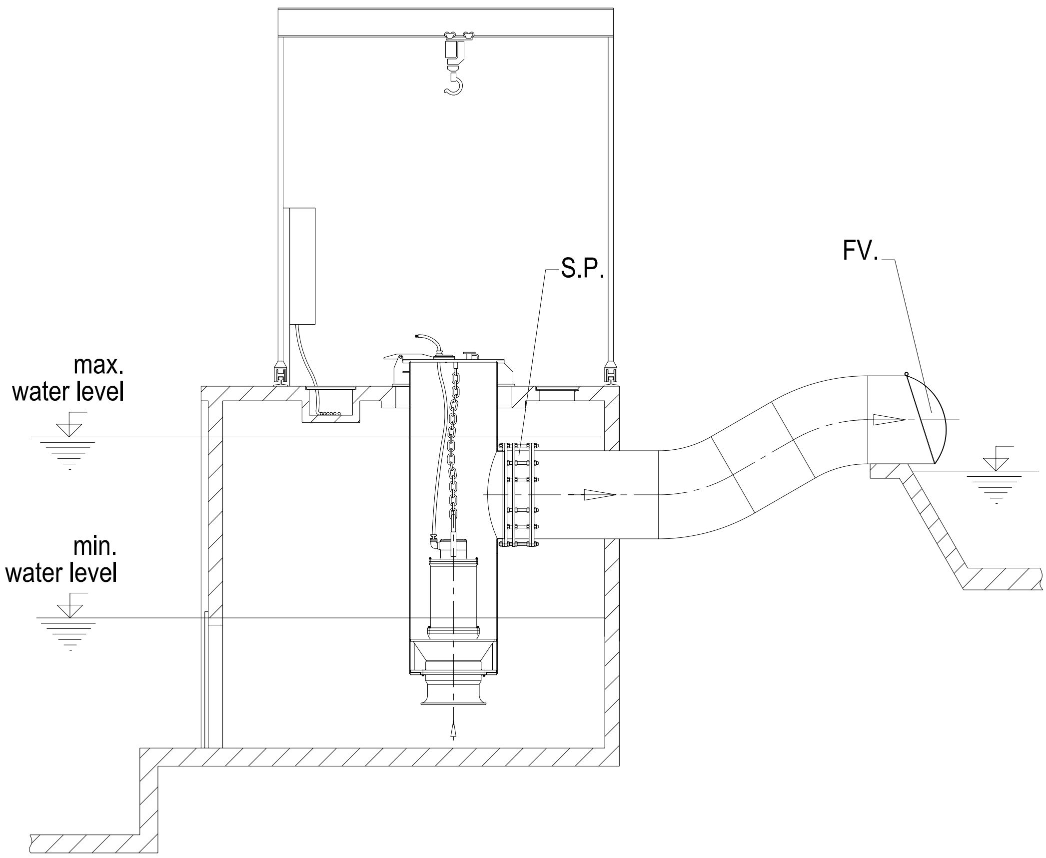
Picture 17.1 Wet Type Submersible Pump (Türbosan DAC, DAS) Station



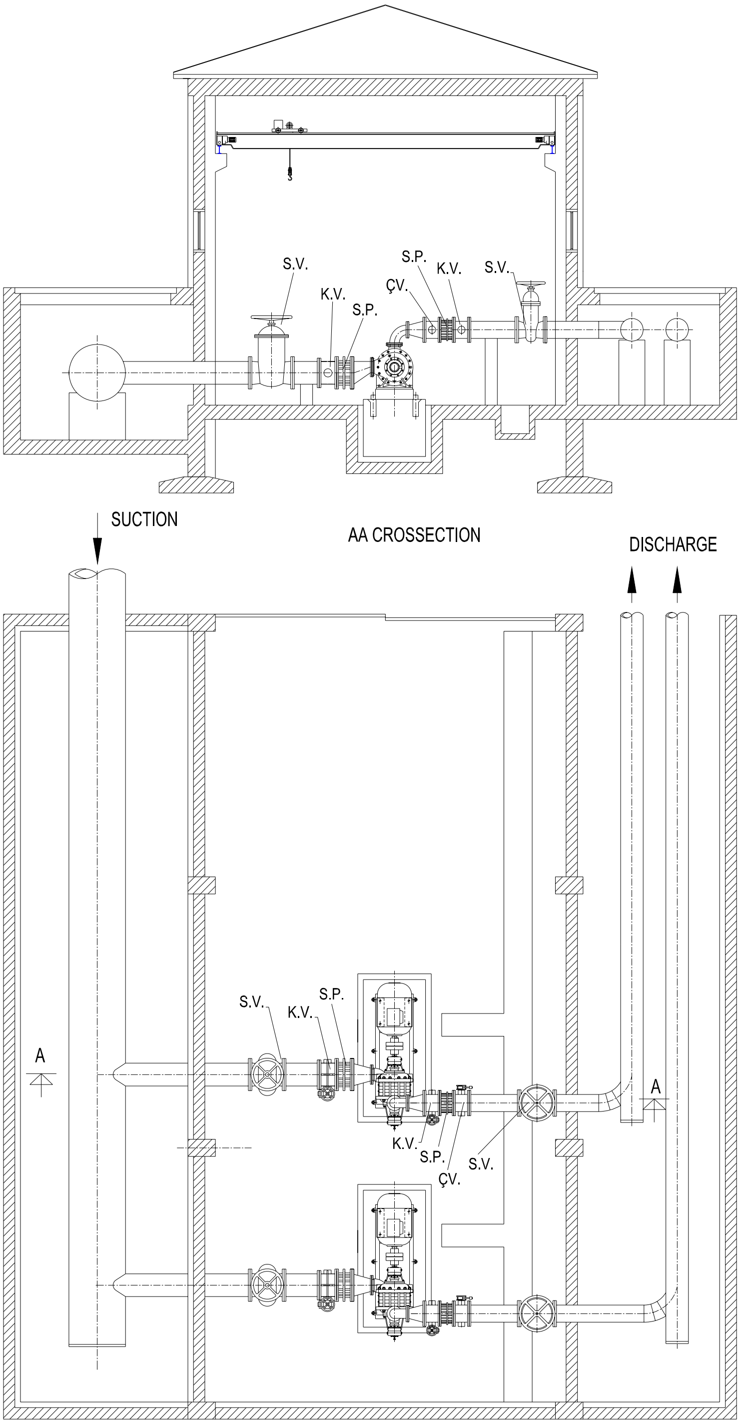
Picture 17.2 Dry Type Cooling Jacket Submersible Pump (Türbosan DAC-SC) Station



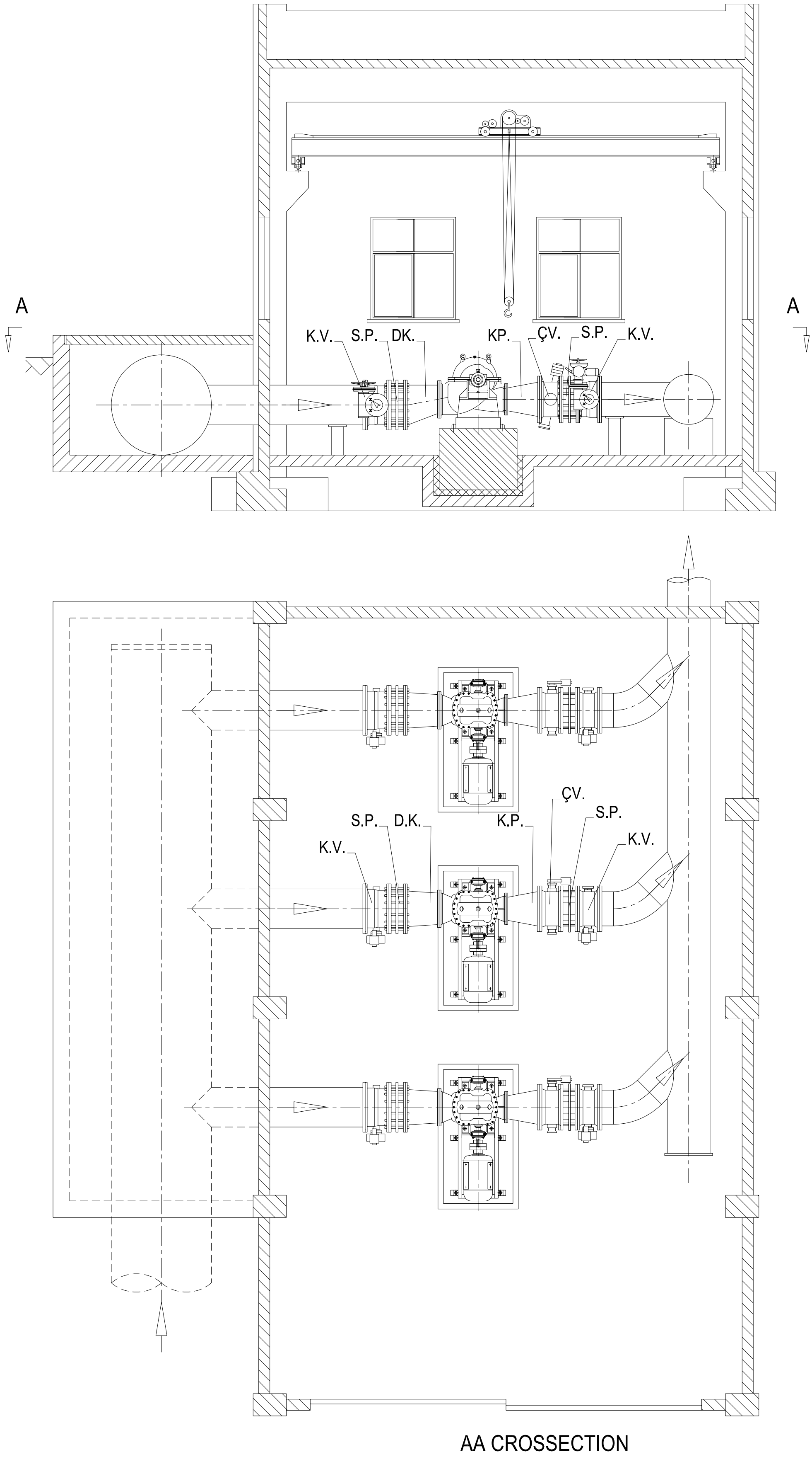
Picture 17.3 Wet Type Column Pipe Sewerage Water Pump (Türbosan ÇAP-VY) Station



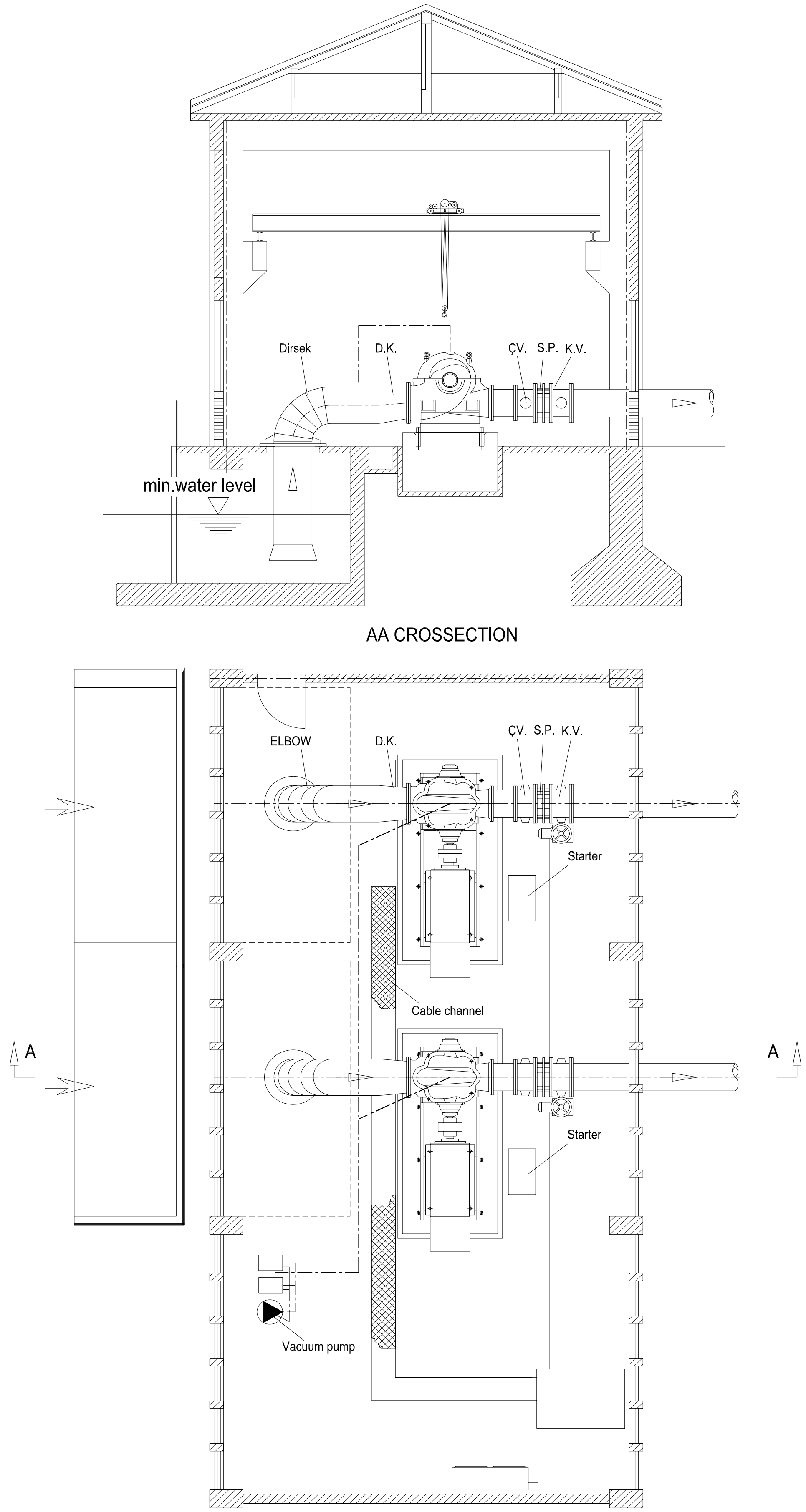
Picture 17.4 Dry Type Tube Submersible Pump (Türbosan DAC E/K) Station



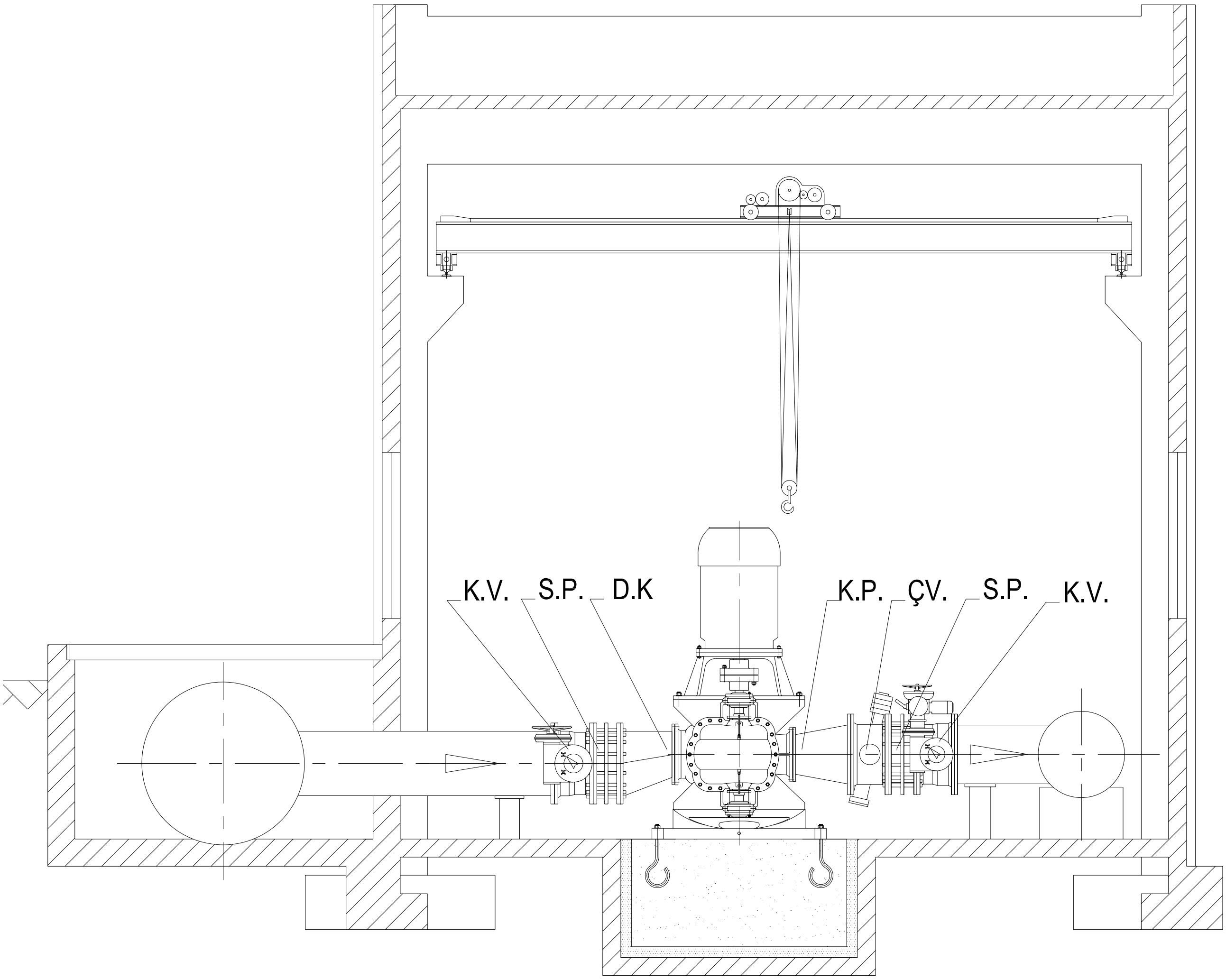
Picture 17.5 Dry Type Multi-stage Pump (Türbosan KOT,KAT) Station



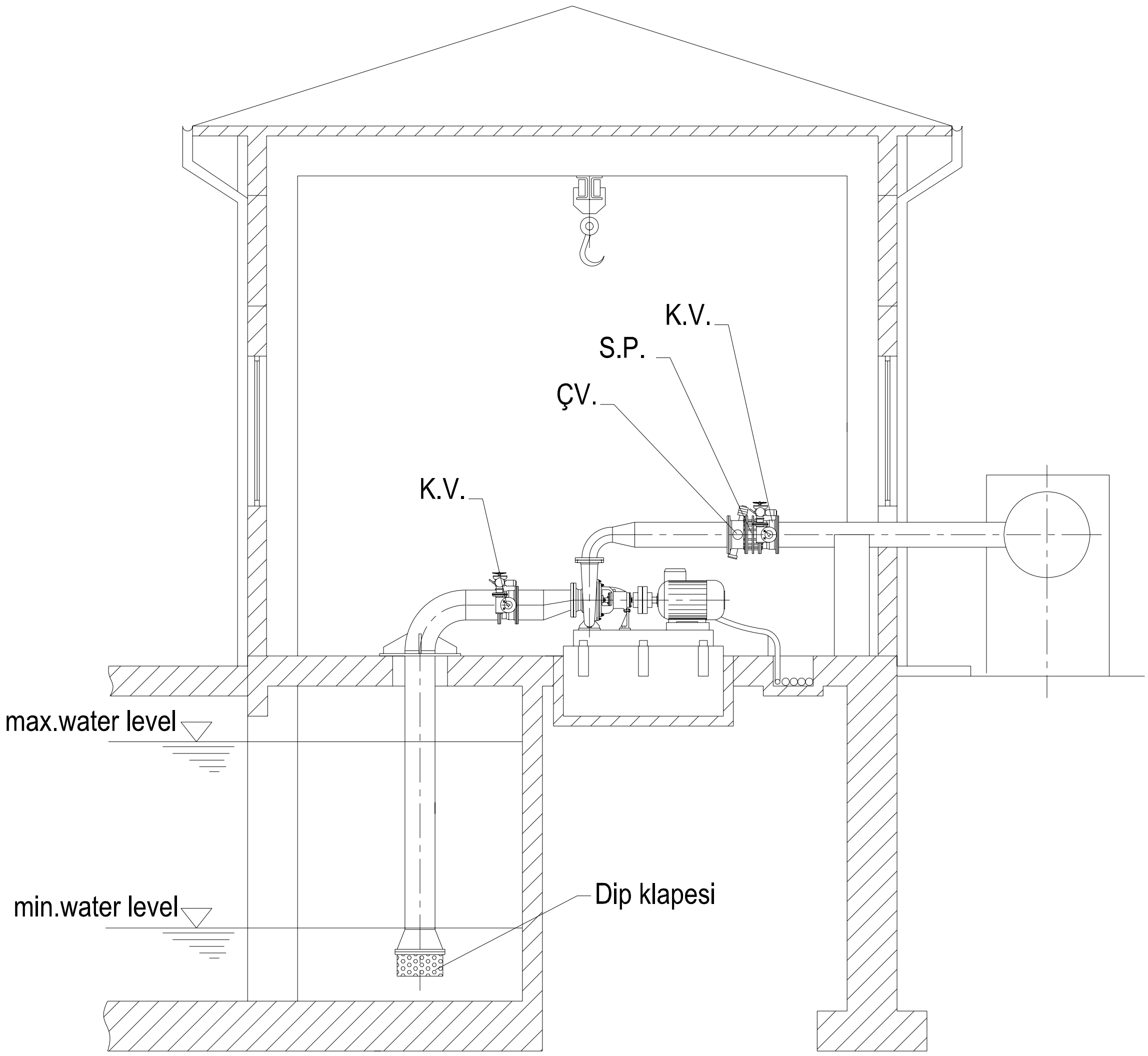
Picture 17.6 Dry Type, Horizontal Shaft, Double Suction Pump (Türbosan ÇEP) Station



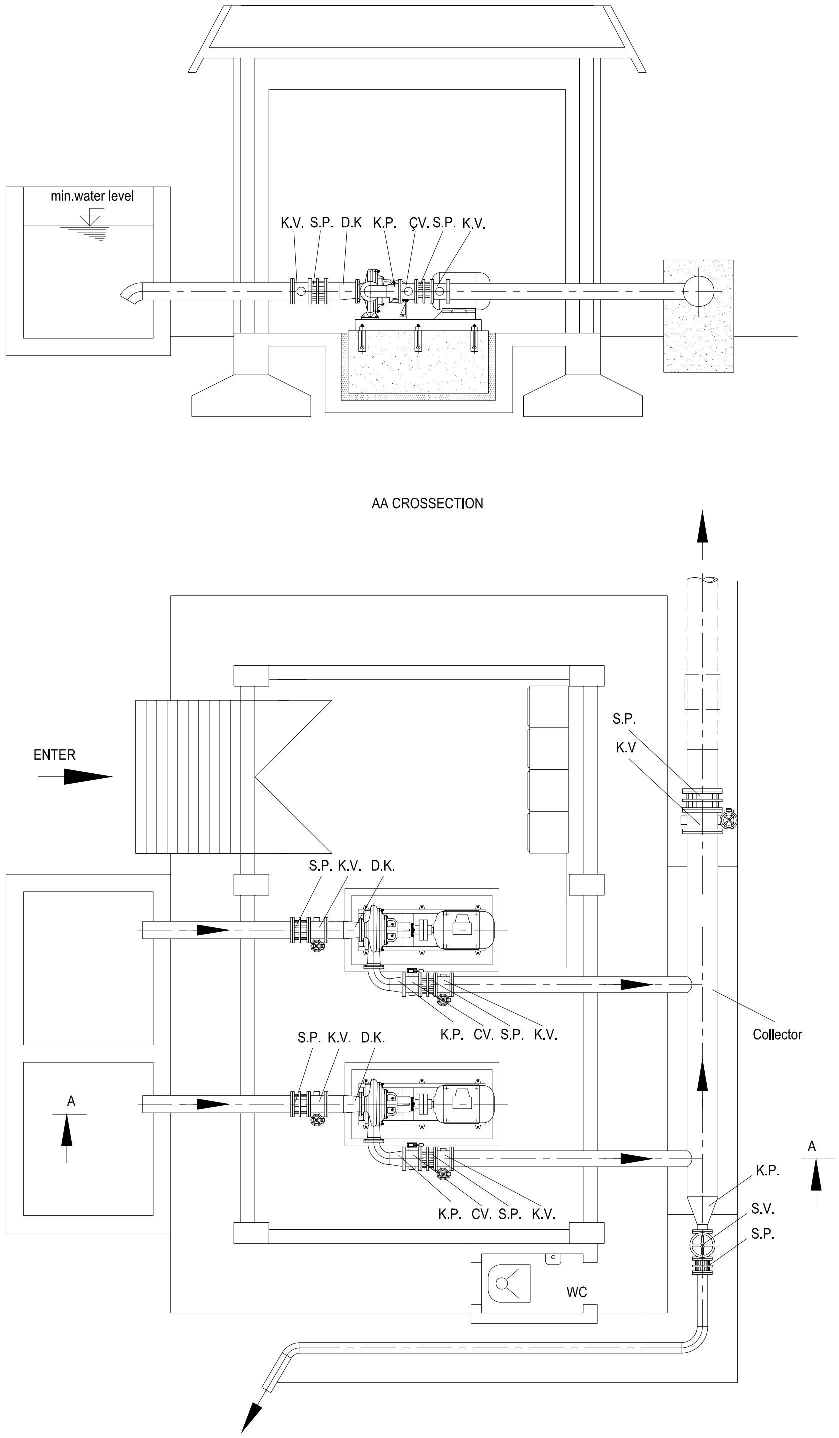
Picture 17.7 Dry Type, Double Suction Vacuum Pump (Türbosan ÇEP) Station



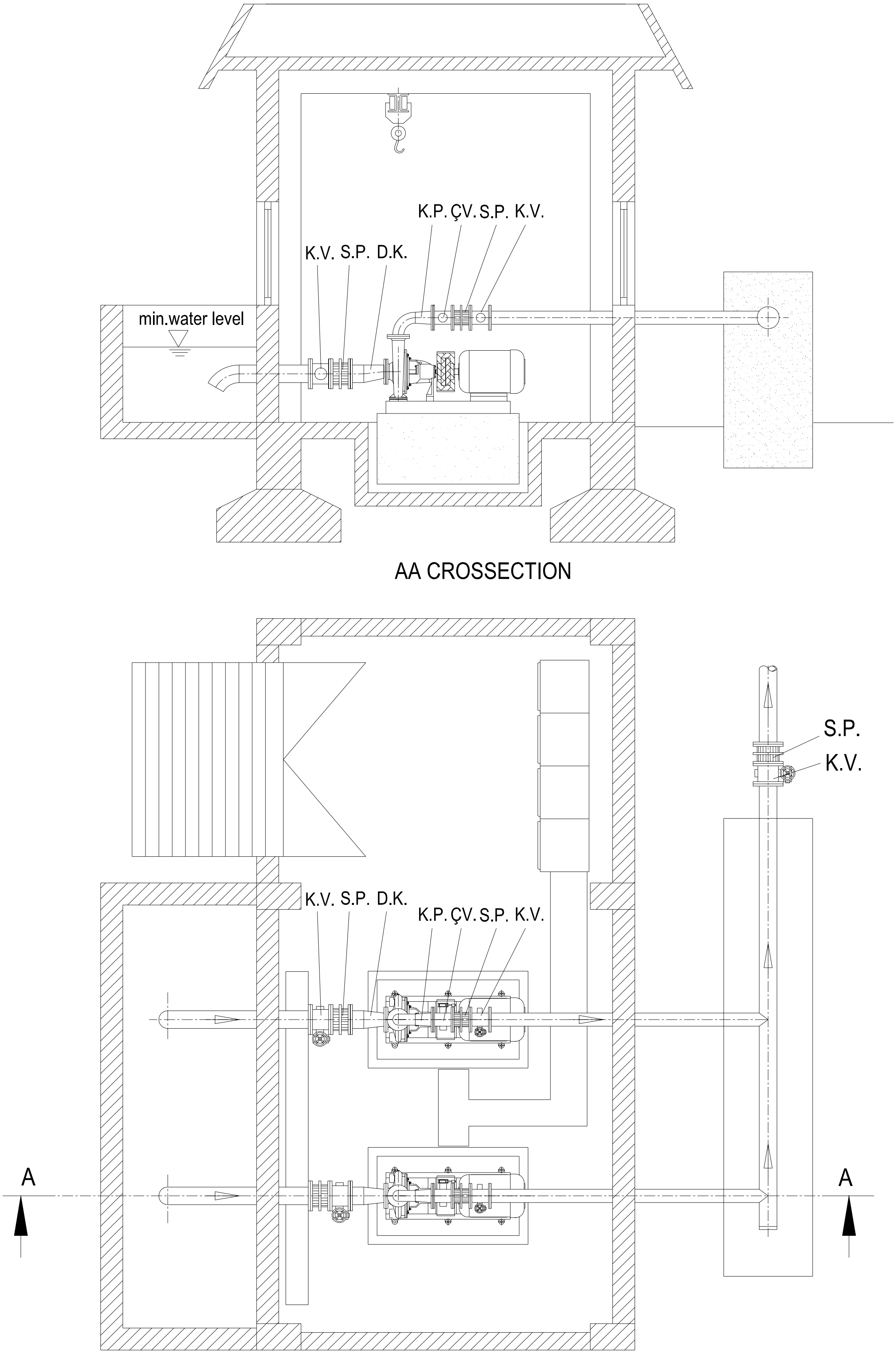
Picture 17.8 Dry Type, Vertical Shaft, Double Suction Pump (Türbosan ÇEP-D) Station



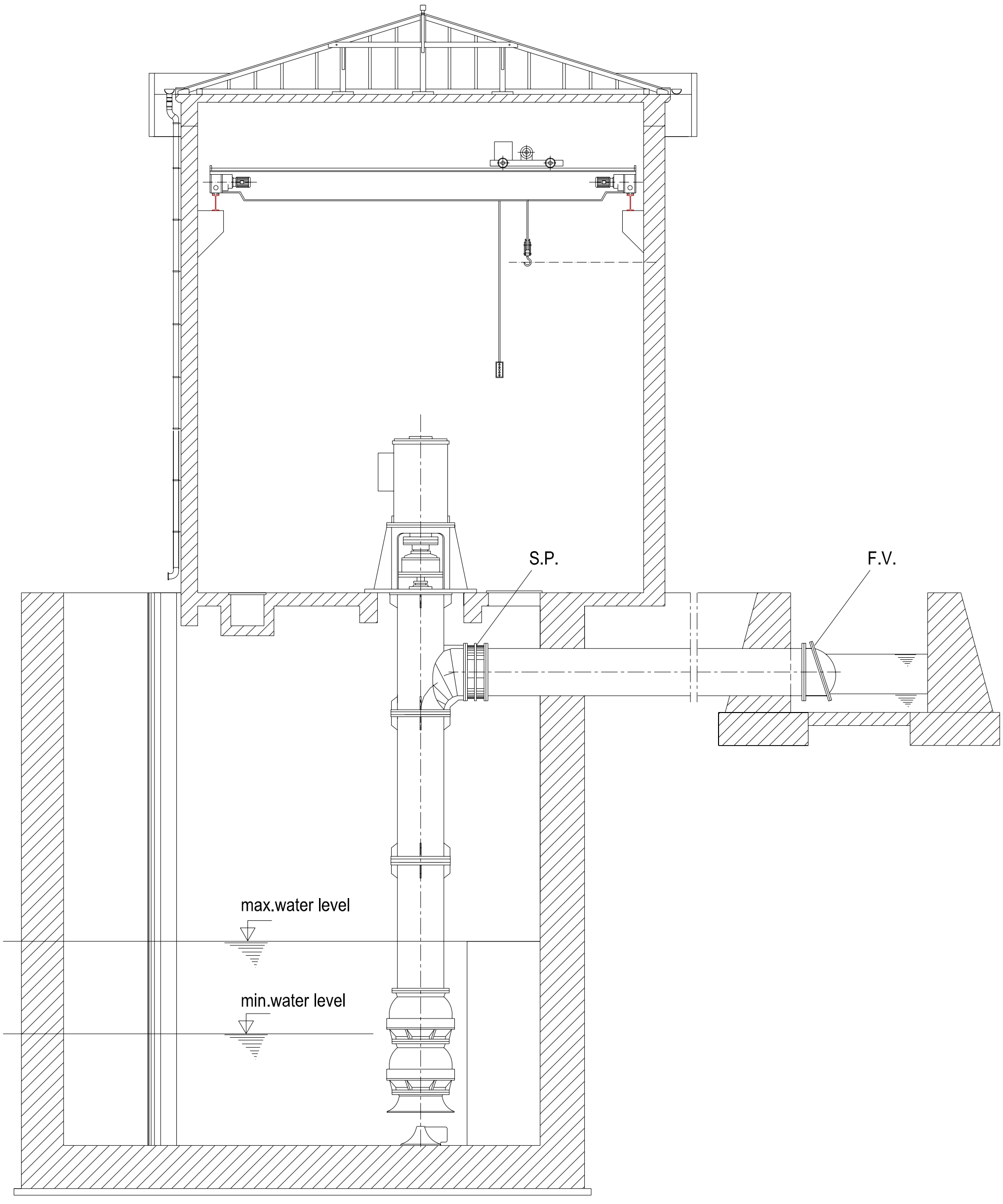
Picture 17.9 Dry Type, Horizontal Shaft Centrifugal Pump (Türbosan NORM) Station



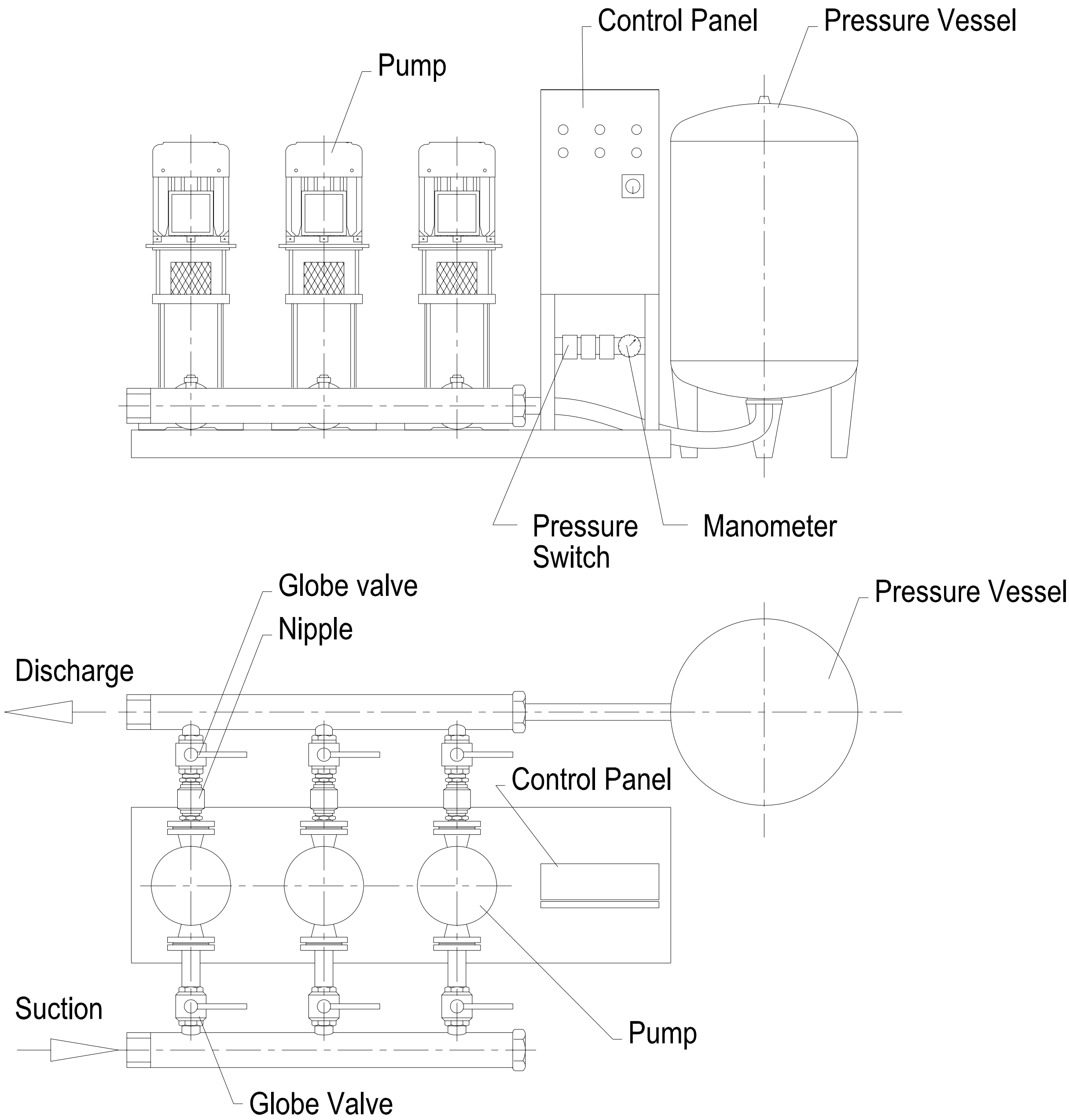
Picture 17.10 Dry Type Horizontal Shaft Centrifugal Pump (NORM) Station w/ Suction Tank



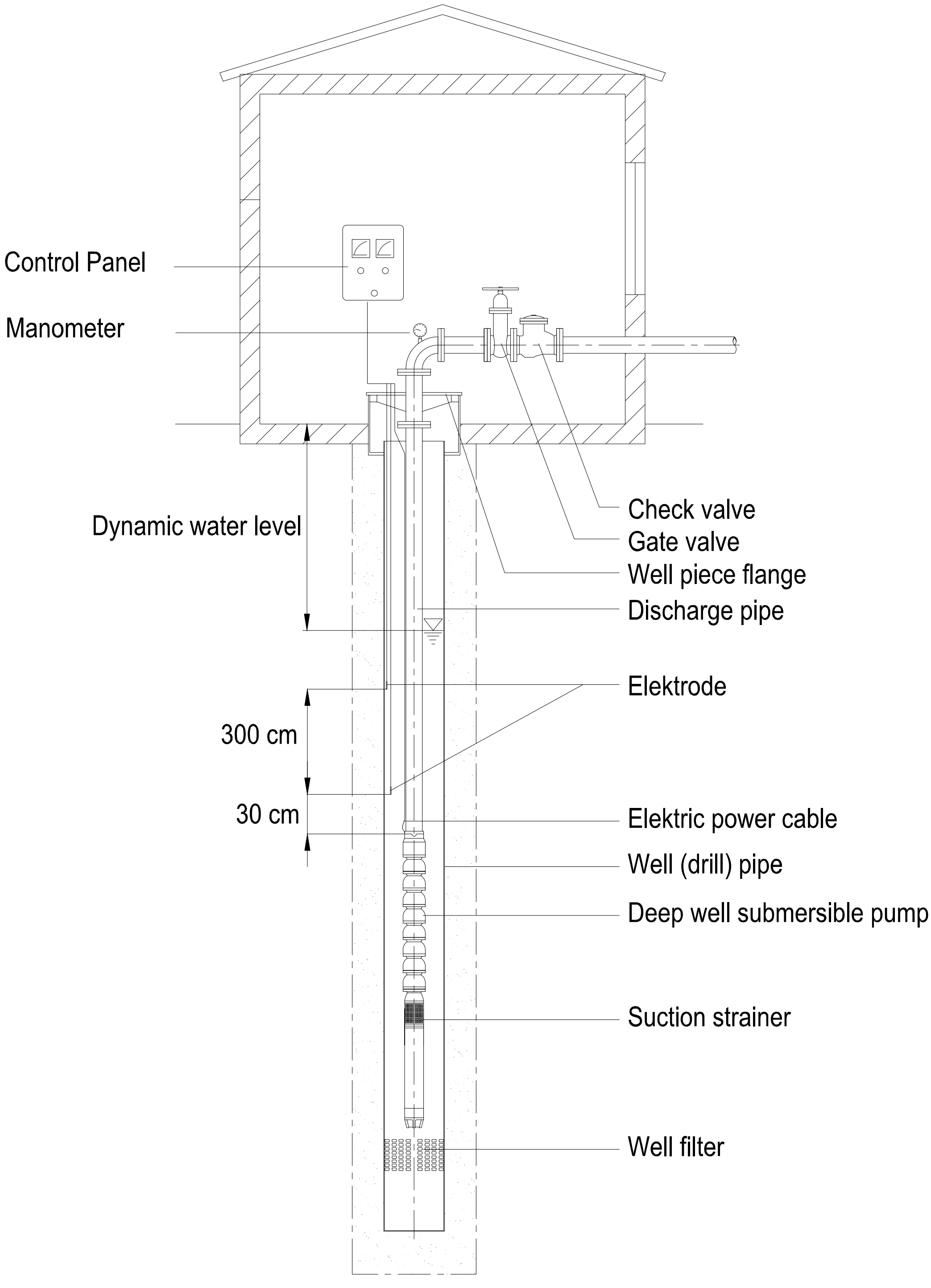
Picture 17.11 Dry Type Horizontal Shaft Centrifugal Pump (NORM) Station w/ Suction Tank



Picture 17.12 Axial, Dry Type, Vertical Shaft Pump (Türbosan EKS) Installation



Picture 17.13 Türbosan Hydrophore Pump Installation



Picture 17.14 Wet Type Vertical Shaft Deep Well Submersible Pump Station

Dynamic Water Level : The water level that remains constant when water equal to the well output is pumped.

Note : For a efficient operation of submersible pumps, the wells should be dug in conformity with required techniques and should be clean.

PART 18

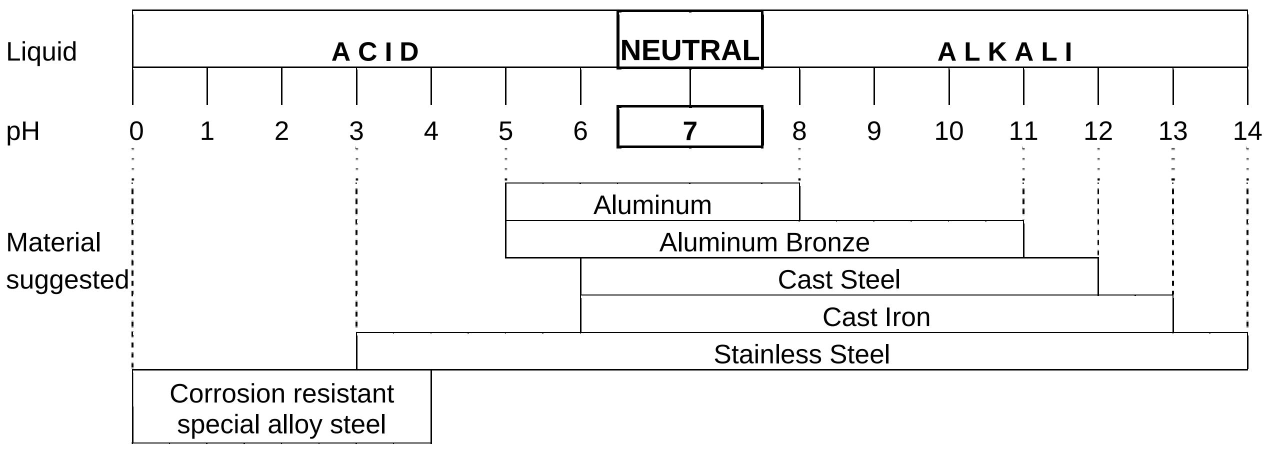
CHOOSING PUMP MATERIAL ACCORDING TO LIQUID CHARACTERISTICS

Pumps convey liquids of very different characteristics. The study summarized in British Standards Institute DD 38 states that pumps are used for conveying about 220 types of liquids with different particularities and indicates which materials should be used for manufacturing adequate pumps for such liquids.

For choosing the correct material for the pump all the characteristics and the chemical composition of the liquid to be pumped must be fully known. The following values of the liquid to be pumped should be taken into account when choosing the pump material:

- the pH value
- the temperature
- the concentration
- the purity
- the flow speed
- the amount of dissolved gas (oxygen, carbon dioxide, hydrogen, sulfite etc.)
- the nature and the quantity of the solid substances,
- the quantity of dissolved salts (calcium carbonate, chlorine, sulfate etc.)
- the abrasive characteristic of the liquid and the size of solid particles.

The main criterion in choosing the material is the corrosive particulars of the liquid. The pH values of the liquids are the sign of the corrosive characteristics. The following table lists the materials suggested according to the pH values of liquids. Water at 25^{o}C temperature and chemically pure is neutral and its pH value is 7.



As, in general, the costs of highly corrosion resistant materials are high, the material is chosen by considering also the material and production costs, the expected lifetime of the pump, the maintenance expenses, the operation safety, the time required for providing spare parts. The Table 18.1 illustrates several materials suitable for various liquids.

Table 18.1 Pump materials to be chosen according to the particulars of the liquid

Material Code	**ASTM number**	**Description of the Material (pump parts in contact with the liquid)**
1	-	Bronze
2	-	Cast Iron, Sphero Cast, Low Alloy Steel
3	A 216 – WCB	Carbon Steel
4	A 217 – C5 A 743 – CA15 A 743 – CB 30 A 743 – CC 50	Chrome Steel (5% to 28% Cr)
5	A 743 – CF – 8 A 743 – CF – 8M A 743 – CN – 7M	Austenitic Steel
6	-	Nickel Alloy Steels
7	A436 Type 2 A439	Austenitic Cast Iron
8	-	Nickel – Copper Alloy
9	-	Nickel

Fluid	**Chemical Formula**	**Pump Material Code**
Alcohols		1
Aluminum sulfate	$Al_2(SO_4)_3$	5,6,8
Ammonia water	NH_4OH	2
Ammonium bicarbonate	NH_4HCO_3	2
Ammonium phosphate	$(NH_4)_2HPO_4$	2,5,6,8
Ammonium chloride	$NH_4 Cl$	5,6,8
Ammonium nitrate	$NH_4 NO_3$	2,5,6,8
Ammonium sulfate	$(NH_4)_2SO_4$	2,5,6
Ammonium sulfate	$(NH_4)_2SO_4$	1,5,6
Aniline	C_6H_7N	1,2
Aniline hydrochloride	$C_6H_5NH_2HCl$	6
Arsenic (ortho)	C_2AsO_4 ½ H_2O	5,6
Acetaldehyde	C_2H_4O	2
Acetate solvent		1,2,5,6
Ascetic anhydride	$C_4H_6O_3$	5,6
Ascetic acid	$C_2 H_4 O_2$	5,6
Acetone	C_3H_6O	1,2
Asphalt		2,4
Waste water		1,2
Copper ammonium acetate		2,5,6
Copper chloride	$Cu Cl_2$	6
Copper nitrate	$Cu(NO_3)_2$	5,6
Copper sulfate	$Cu SO_4$	5,6

Fluid	**Chemical Formula**	**Pump Material Code**
Barium chloride	$Ba\ Cl_2$	2,5,6
Barium nitrate	$Ba\ (NO_3)_2$	2,5,6
Benzene (Benzol)	C_6H_6	1,2
Gasoline		1,2
Benzoic acid	$C_7H_6O_2$	5,6
Beer		1,5
Beer yeast		1,5
Boric acid	H_3BO_3	5,6
Butyric acid	$C_4H_8O_2$	5,6
Butane	C_4H_{10}	1,2,3
Zinc chloride	$Zn\ Cl_2$	5,6
Zinc sulfate	$Zn\ SO_4$	1,5,6
Sea water		1,2
Diphenyl	$C_6H_5.C_6H_5$	2,3
Distilled water		1,5
Tallow		2
Enamel		2
Ethylene dichloride	$C_2H_4Cl_2$	1,5,6,8
Phenol	C_6H_6O	1,5,6
Formaldehyde	CH_2O	1,5,6
Formic acid	CH_2O_2	5,6
Phosphoric acid	$H_3\ PO_4$	5,6
Photo/film developing liquid		5,6
Fuel - oil		1,2
Furfural	$C_5H_4O_2$	1,2,5,6
Kerosene		1,2
Glucose		1
Glycerol	$C_3H_8O_3$	1,2
Silver nitrate	$Ag\ NO_3$	5,6
Crude oil		1,2,3
Mustard		1,5,6
Heptane	C_7H_{16}	1,2
Hydrofluoric acid	HF	3,8
Hydrofluosilicic acid	H_2SiF_6	1,8
Hydrogen peroxide	H_2O_2	5,6
Hydrogen sulfite	H_2S	5,6
Hydrochloric acid	HCl	6
Hydrocyanic acid	HCN	2,5,6
Calcium bisulfate	$Ca\ (HSO_3)_2$	5,6
Calcium chlorate	$Ca(ClO_3)_2.2H_2O$	5,6
Blood		1

Fluid	Chemical Formula	Pump Material Code
Kaolin		2,3
Carbolic acid	C_6H_6O	2,5,6
Carbon bisulfide	$C\ S_2$	2
Carbon hypochlorite	$Ca\ (OCl)_2$	2,5,6
Carbon tetrachloride	$C\ Cl_4$	1,2
Carbonic acid	$CO_2 + H_2O$	1
Tar		2,3
Tank supply water		2
Ketch-up		1,5,6
Lime water	$Ca\ (OH)_2$	2
Chlorinated water		5,6
Chloro-benzene	C_6H_5Cl	1,5
Chloroform	$CH\ Cl_3$	1,5,6,8
Cresol (meta)	C_7H_8O	2,4
Chromic acid	$Cr_2O_3 + H_2O$	1,5,6
Lead		2,3
Lead acetate	$Pb(C_2H_3O_2)_2.3H_2O$	5,6,8
Lead tetra ethyl	$Pb\ (C_2H_5)_4$	1,2
Sulfur	S	1,2,5,6
Sulfur chloride	S_2Cl_2	2
Lactic acid	$C_3H_6O_3$	1,5,6
Liquor		2,3,5,6,8
Lithium chloride	$LiCl$	2
Mineral water		1,5,6
Mineral oil		1,2
Magnesium chloride	$Mg\ Cl_2$	5,6
Magnesium sulfate	$Mg\ SO_4$	2,5,6
Manganese sulfate	$Mn\ SO_4\ .\ 4H_2O$	1,2,5,6
Manganese chloride	$Mn\ Cl_2\ .\ 4H_2O$	1,5,6
Maya		1
Methyl chloride	CH_3Cl	2
Methylene chloride	$CH_2\ Cl_2$	2,5
Fruit		1,5,6,8
Fruit juice		1,5,6,9
Naphtha		1,2
Naphthenic		2,4,5,6
Turpentine		1,2
Starch	$(C6H_{10}C_5)_X$	1
Nitric acid	HNO_3	4,5,6
Oxalic acid	$C_2H_2O_4.2H_2O$	5,6
Sugar beet pulp		1,5,6

Fluid	Chemical Formula	Pump Material Code
Sugar beet juice		1,5
Paraffin		1,2
Grape molasses		1
Picric acid	$C_6 H_3 N_3 O_7$	5,6
Pyrogallic Acid	$C_6H_6O_3$	5,6
Potash		1,5,6,7,8
Potassium bicarbonate	$K_2Cr_2O_7$	2
Potassium hydroxide	K OH	2,4,5,6,7,8,9
Potassium carbonate	K_2CO_3	2
Potassium chlorate	$K\ Cl\ O_3$	5,6
Potassium chloride	K Cl	1,5,6,8
Potassium nitrate	$K\ NO_3$	2,4,5,6
Potassium cyanide	K CN	2
Potassium sulfate	$K_2\ SO_4$	1,5,6
Pyridine	C_5H_5N	2
Pyridine sulfate		5
Propane	$C_3\ H_8$	1,2,3
Vegetable water		1,5,6,8
Cellulose acetate		5,6
Liquid soap		2
Vinegar		1,5,6,8
Citric acid	$C_6H_8O_7 + H_2O$	1,5,6
Cyanogen	$(CN)_2$ gas	2
Soda (carbonate)	$Na_2\ CO_3$	2
Sodium bicarbonate	$Na\ H\ CO_3$	2,5,6,7,8
Sodium bisulfate	$Na\ H\ SO_4$	5,6
Sodium hydroxide	Na OH	2,4,5,6,7,8
Sodium hydrosulfite	$Na_2S_2O_4.2H_2O$	5,6
Sodium hypochlorite	NaOCl	5,6
Sodium hyposulfite	$Na_2S_2O_3.5H_2O$	5,6
Sodium carbonate	$Na_2\ CO_3$	2,5,6,8
Sodium chlorate	$Na\ Cl\ O_4$	5,6
Sodium meta phosphate	$Na_4P_4O_{12}$	1,5,6
Sodium nitrate	$Na\ CO_3$	2,4,5,6
Sodium cyanide	Na CN	2
Sodium sulfate	Na_2SO_4	1,5,6
Sodium sulfite	Na_2S	2,5,6
Sodium sulfite	$Na_2\ SO_3$	1,5,6
Soya oil		1,2,5,6,8
Water		1,2
Sulfuric Acid	H_2SO_4	5,6

Fluid	Chemical Formula	Pump Material Code
Sulfurous	H_2SO_3	1,6,7
Milk		5
Alum (chromium)	$CrK(SO_4)_2.12H_2O$	5,6
Wine		1,5
Sugar		1,5,6,7
Tannic liquor		1,2,5,6,8
Tannic acid	$C_{14}H_{10}O_9$	1,5,6,8
Tartaric acid	$C_4H_6O_6 . H_2O$	1,5,6,9
Toluene (toluol)	C_7H_8	1,2
Trichlorethylene	C_2HCl_3	1,2,5
Glue		1,2
Saltwater	NaCl	1,2,5,6,7,8
Varnish		1,2,5,8
Lubricating oil		1,2
Olive oil		1,2

Note : Data supplied in this table is intended for general information. Please consult relevant material and chemical documentation for detail.

Table 18.2 Widely used materials and equivalent standards

DEFINITION	DIN / EN 17007	DIN	ASTM
Carbon steel	1.0501		SAE C 1040
Sheet metal	1.0120		
Cast iron	0.6020 0.6025	GJL-200 (GG 20) GJL-250 (GG 25) EN1561	A 48 Class 40-B
Nodular cast iron	0.7040	GJS-400-15 (GGG 40) EN1563	A 536 Gr. 60-40-18
Cast steel	1.0619	GP240GH (GS-C 25) EN 10213-2	A 216 Gr. WCB
Cast steel	1.0443	GS - 45	A 27 : Gr65 – 35 A 216 – 75 : Gr WCA
Chromium steel	1.4008	G-X 8 Cr 14	A 743-87 (CA-15)
Chromium nickel steel cast	1.4308	G-X6 Cr Ni 18.9	A 351-75 Grade CF8
Chromium nickel molybdenum steel cast	1.4408	G-X6 Cr Ni Mo 18.10	A 351-75 Grade CF8M
Chromium nickel molybdenum steel cast	1.4404	G-X2 Cr Ni Mo 18.10	A-167-74 Type 316L
Chromium molybdenum steel cast	1.4138	G-X120 Cr Mo 29.2	---
Chromium steel	1.4021	X20 Cr 13	A 276 Type 420
Chromium nickel steel	1.4301	X5 Cr Ni 18.9	A 276 Type 304
Chromium nickel molybdenum steel	1.4401	X5 Cr Ni Mo 18.10	A 276 Type 316
Duplex stainless steel cast	1.4462	X 2 CrNiMoN 22-5-3	A 276 Type 2205
Duplex stainless steel cast	1.4517	X 3 CrNiMoCuN 26-6-33	A 351 CD-4-MCu
Alloy cast iron (Ni-Resist D2)	0.7660	GGG-NiCr202	A 439-62
Bronze casting	2.1050.01	G-Cu Sn 10	B 584 C 90700
Bronze casting	2.1052.01	G-Cu Sn 12	B 427 C 908 B 427 C 917
Bronze casting	2.1090.01	G-Cu Sn 7 Zn Pb	B 584 C 93200
Aluminum bronze	2.0975.01	G-Cu Al 10 Ni	B –148-74:955-958
NiHard2	0.9620	G-X260NiCr42	