TECHNICAL APPENDIX

ESPA

ESPA

Index

Unit Equivalences
(Conversion factors)

MEASUREMENTS:	TO CONVERT	INTO	MULITIPLY BY
LENGTH	Inches	Millimetres	25,401
	Feet	Metres	0,3048
SURFACE AREA	$Inches^2$	cm^2	6,4516
	$Feet^2$	m^2	0,0929
VOLUME	$Inches^3$	Litres	0,01638
	$Feet^3$	Litres	28,3205
	Gallons (USA)	Litres	3,785
	Gallons (Imperial)	Litres	4,5454
FLOW RATE	g.p.m (USA)	m^3/h	0,2271
	g.p.m (IMP.)	m^3/h	0,2727
PRESSURE	Pounds per sq. $inch^2$	Kg/cm^2	0,0703
	Bar	kg/cm^2	1,0197
	Atmospheres	Kg/cm^2	1,033
	Kilo Pascal	metros c.a	0,10197
	Kilo Pascal	Kg/cm^2	0,010197
WEIGHT	Pounds	Kg	0,4536
	Ounces	Kg	0,02834
POWER	Steam Power (SP)	Watts	736
	Horse Power (HP)	Watts	746
	CV	HP	0,98644
TEMPERATURE	Fahrenheit	Centigrade	$°C = \frac{5 \times (°F - 32)}{9}$

MEASUREMENTS:	TO CONVERT	INTO	MULITIPLY BY
LENGTH	Millimetres	Inches	0,0394
	Metres	Feet	3,2808
SURFACE AREA	cm^2	$Inch^2$	0,155
	m^2	$Feet^2$	10,7639
VOLUME	Litres	$Inch^3$	61,024
	Litres	$Feet^3$	0,03531
	Litres	Gallons (USA)	0,2642
	Litres	Gallons (IMP.)	0,22
FLOW RATE	m^3/h	g.p.m (USA)	4,4033
	m^3/h	g.p.m (IMP.)	3,66703
PRESSURE	Kg/cm^2	Pounds per sq. $inch^2$	14,2247
	kg/cm^2	Bar	0,9806
	Kg/cm^2	Atmospheres	0,968
	metres c.a	Kilo Pascal	9,8067
	Kg/cm^2	Kilo Pascal	98,005
WEIGHT	Kg	Pounds	2,2046
	Kg	Ounces	35,285
POWER	Watts	Steam Power (SP)	0,00136
	Watts	Horse Power (HP)	0,00134
	HP	CV	1,0139
TEMPERATURE	Centigrade	Fahrenheit	$°F = \frac{9 \times °C}{5} + 32$

Basic Concepts

FLOW (Q): Volume of fluid raised by the pump in a unit of time; it is independent of the variable unit weight when pumping fluids with a viscosity that is greater than that of water.

ATMOSPHERIC PRESSURE (P_a): Force exerted by the atmosphere per unit of surface area.

RELATIVE OR EFFECTIVE PRESSURE (P_r): This is average pressure in relation to atmospheric pressure. Manometers measure positive pressures. Vacuum gauges measure negative pressures.

ABSOLUTE PRESSURE (P_{abs}): This is the pressure above absolute zero (perfect vacuum).

$$P_{abs} = P_a + P_r$$

VAPOUR PRESSURE (VAPOUR TENSION) (T_v): This is the pressure at which a liquid at a certain temperature is in equilibrium with its gaseous form (vapour).

DENSITY: This is the mass of a substance per unit of volume.

UNIT WEIGHT (γ): This is the weight of a substance per unit of volume.

Unit weight = Density x Gravity

EFFECT OF UNIT WEIGHT: A pump is capable of raising liquids of varying unit weight, for example water, alcohol, sulphuric acid etc, to the same height. Only the discharge pressure and the absorbed power will be modified, in direct proportion to the unit weight.

SUCTION HEIGHT (H_a): This is the geometric height measured between the lowest level of fluid and the pump shaft (see diagram below).

DISCHARGE HEIGHT (H_i): This is the geometric height measured from the pump shaft to the maximum level to which the fluid is raised (see diagram below).

TOTAL GEOMETRIC HEIGHT (H_t):

$$H_t = H_a + H_i$$

FRICTION LOSSES (P_c): This is the level of loss caused by friction when the liquid passes through the pipes, valves, filters, curves and other accessories.

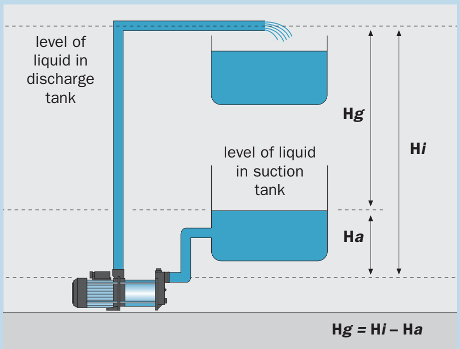
TOTAL MANOMETRIC HEIGHT (H_m): This is the total height (differential pressure) that the pump has to surmount. It corresponds to the equation:

$$Hm = Ht + Pc + \frac{10}{\gamma}(P_1 - P_2)$$

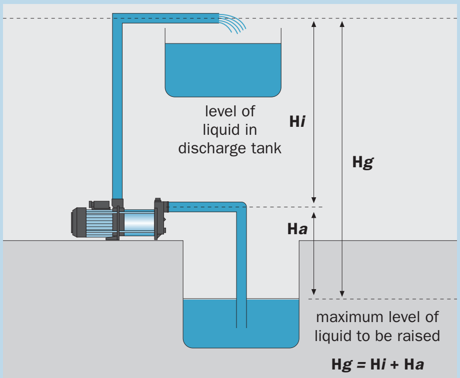
P_1 : Pressure in the discharge tank
P_2 : Pressure in the suction tank

If pumping takes place between open tanks at the same pressure (ambient pressure) as is NORMALLY the case, the value P_1-P_2= 0.
It is advisable to calculate separately the manometric suction height to ascertain whether the pump is capable of problem-free suction.

Inflow Installation



Suction Installation



Power and Output

(P1) POWER ABSORBED BY THE NETWORK
Consumption of power or active power

Single-phase engines

$$Kw = \frac{U \cdot I \cdot \cos\varphi}{1000}$$

Tri-phase engines

$$Kw = \frac{\sqrt{3} \cdot U \cdot I \cdot \cos\varphi}{1000}$$

(P2) NOMINAL ENGINE POWER
The main power supplied by the engine

Single-phase engines

$$Kw = \frac{U \cdot I \cdot \cos\varphi \cdot \eta_m}{1000}$$

Tri-phase engines

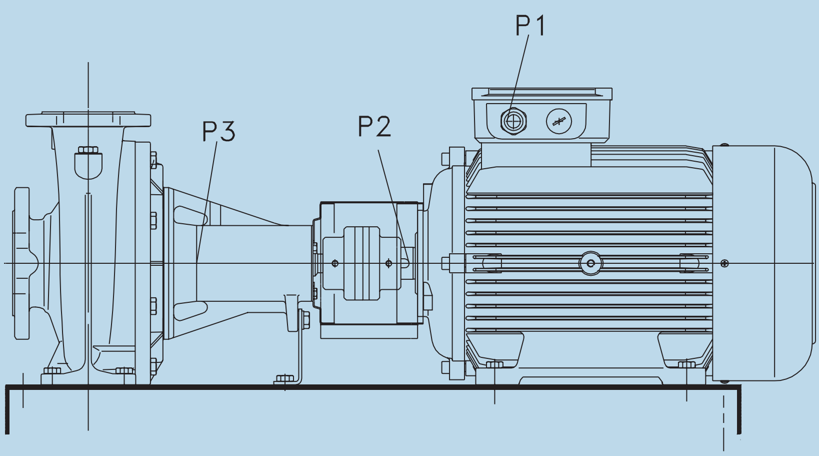
$$Kw = \frac{\sqrt{3} \cdot U \cdot I \cdot \cos\varphi \cdot \eta_m}{1000}$$

(P3) POWER ABSORBED BY THE PUMP SHAFT
To determine the operating conditions

$$Kw = \frac{Q \cdot H \cdot \gamma}{367 \cdot \eta_h} \qquad CV = \frac{Q \cdot H \cdot \gamma}{270 \cdot \eta_h}$$

Whereby:
- U : Operating voltage in V.
- I : Current in the stator in Amp.
- $\cos\varphi$: Output factor
- η_m : Engine output

- Q : Flow m^3/h
- H : Manometric height in metres liquid column
- η_h : Hydraulic output in %
- γ : Volume weight in kg/dm^3



Pipes

The selection of pipe diameters is a technical and economic decision that should be taken bearing in mind that friction losses should not be to high, so as to avoid excessive consumption of energy.
The size of the suction and discharge apertures of the pumps does not only give us an idea of the minimum size of the pipes. Dimensioning must be done so that speeds do not exceed the following values:

Suction pipe: 1,8 m/s
Discharge pipe: 2,5 m/s

The flow rate is important for economic use and duration of the discharge system.

- Speeds below 0.5 m/s normally give rise to sediments.
- Speeds above 5 m/s may give rise to abrasions.

The speeds of the liquid n the pipes is determined by the following formulas:

$$V = \frac{21{,}22 \times q}{D^2} \quad \text{or} \quad V = \frac{354 \times Q}{D^2}$$

Wherein the values represent the following:
V : speed in m/s — q : flow rate in l/m
D : diameter in mm — Q : flow rate in m^3/h

Equivalence between pipes

Equivalences between pipes enable data on other pipe systems to be obtained

Consistent diameter: Friction loss is directly proportionate to the square of the flow rate:

$$\frac{Pc}{Pc_1} = \frac{Q^2}{Q_1^2}$$

Constant flow: Friction loss is inversely proportionate to the fifth power of the diameter of the pipe:

$$\frac{Pc}{Pc_1} = \frac{D_1^5}{D^5}$$

Constant flow rate: The circulation rate is inversely proportionate to the section of the pipes:

$$\frac{V}{V_1} = \frac{S_1}{S}$$

Constant friction losses: The squares of flow rates are proportionate to the fifth power of the pipe diameters:

$$\frac{Q^2}{Q_1^2} = \frac{D^5}{D_1^5}$$

Equivalent Friction Losses

Based on the last equation, the table below associates the equivalences between pipes of different diameters.

	inch	1/2	3/4	1	1 1/4	1 1/2	2	2 1/2	3	4	5	6
inch	mm	13	19	25	32	38	50	64	75	100	125	150
1	25	3,7	1,8	1								
1 1/4	32	7	3,6	2	1							
1 1/2	38	11	5,3	2,9	1,5	1						
2	50	20	10	5,5	2,7	1,9	1					
2 1/2	64	31	16	8	4,3	2,9	1,6	1				
3	75	54	27	15	7	5	2,7	1,7	1			
4	100	107	53	29	15	10	5,3	3,4	2	1		
5	125	188	93	51	26	17	9	6	3,5	1,8	1	
6	150	297	147	80	40	28	15	9	5,5	2,8	1,6	1
7	175	428	212	116	58	40	21	14	8	4 2	3	1,4
8	200	590	292	160	80	55	29	19	10,9	5,5	3,1	2

OBSERVATIONS
The area of piping with the larger diameter is smaller than the total area of piping with a smaller diameter.
The circulation rate of the liquid in the pipe with the larger diameter is greater than the rate in the pipes with a smaller diameter.

Friction Losses

Friction losses in accessories: Equivalent length of straight piping in metres.

Pipe Diameter	25	32	40	50	65	80	100	125	150	200	250	300	350	400	500	600	700
Curve 90°	0,2	0,3	0,4	0,5	0,7	1	1,2	1,8	2	3	5	5	6	7	8	14	16
Elbow 90°	0,3	0,4	0,6	0,7	0,9	1,3	1,7	2,5	2,7	4	5,5	7	8,5	9,5	11	19	22
Diffuser cone	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5
Foot valve	6	7	8	9	10	12	15	20	25	30	40	45	55	60	75	90	100
Stop value	4	5	6	7	8	9	10	15	20	25	30	35	40	50	60	75	85
Gate Valve Open	0,5	0,5	0,5	0,5	0,5	0,5	1	1	1,5	2	2	2	2,5	3	3,5	4	5
Gate Valve $^3/_4$ Open	2	2	2	2	2	2	4	4	6	8	8	8	10	12	14	16	20
Gate Valve $^1/_2$ Open	15	15	15	15	15	15	30	30	45	60	60	60	75	90	105	120	150

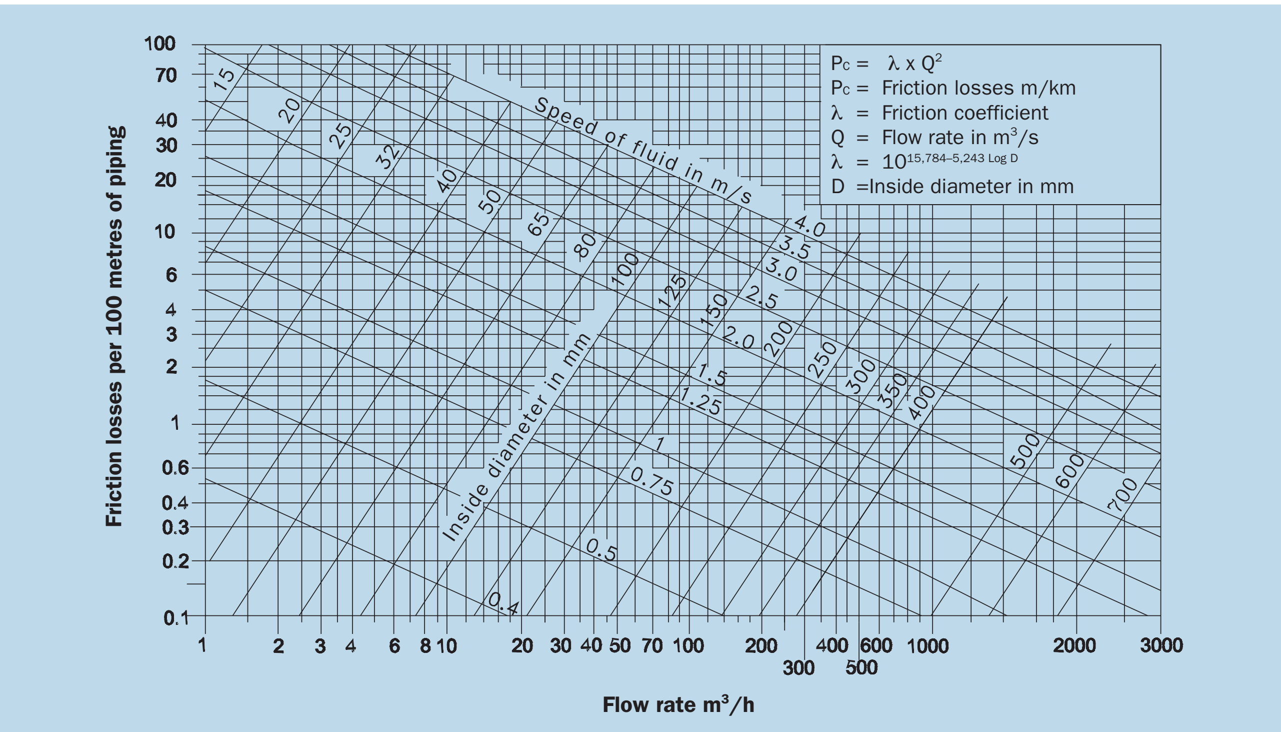
Approximate values, varying according to the quality of the accessories.

Manufacturers of valves that provide the kv, enable us to determine friction losses; it is of the utmost importance to use valves with a high flow coefficient in order to minimise friction losses.

The flow-rate coefficient kv is the flow rate of water in m^3/h that produces a friction loss of 1 kg/cm^2 when passing through a completely open valve.

Friction losses in cast iron piping

Diagram to determine the friction loss and speed of fluid in accordance with the flow rate and interior diameter of the piping.



Correction coefficients for other piping

PVC	0,6
Forged iron	0,76
Seamless steel	0,76

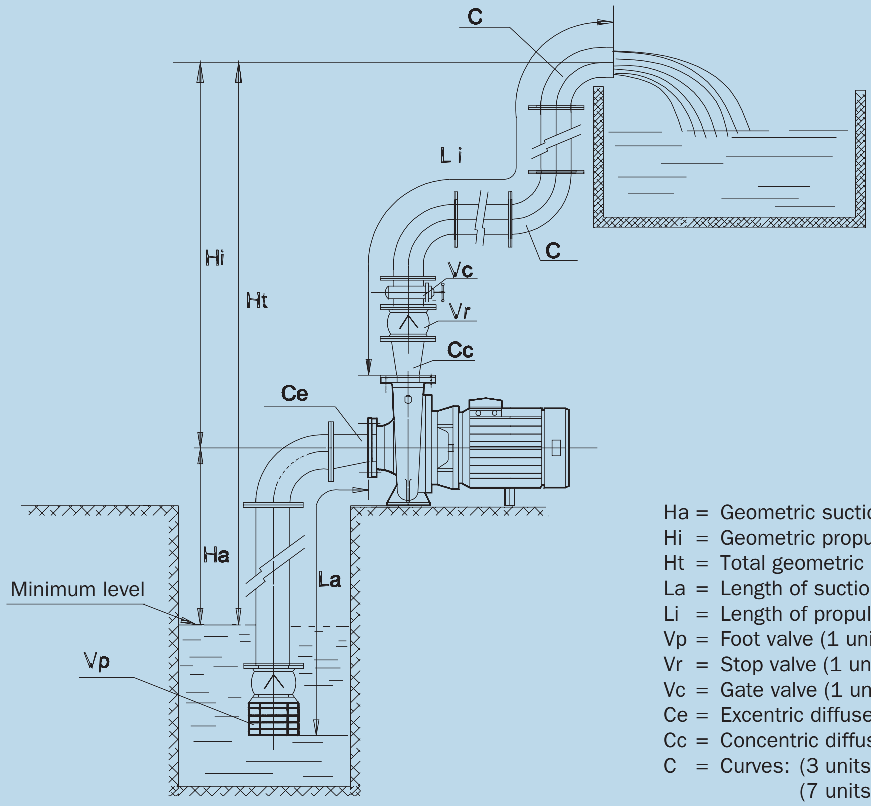
Cement asbestos	0,80
Cement (smooth walls)	0,80
Earthenware	1,17

Worn out wrought iron	2,10
Iron with rough-cast walls	3,60

Calculation of manometric height

Practical example

150 m^3/h needs to be raised from a well to a tank situated at a greater height above sea level.
The pumping conditions are as follows:



Ha = Geometric suction height (3 metres)
Hi = Geometric propulsion height (34 metres)
Ht = Total geometric height (37 metres)
La = Length of suction (8 metres)
Li = Length of propulsion (240 metres)
Vp = Foot valve (1 unit)
Vr = Stop valve (1 unit)
Vc = Gate valve (1 unit)
Ce = Excentric diffuser cone (1 unit)
Cc = Concentric diffuser cone (1 unit)
C = Curves: (3 units) suction
(7 units) propulsion

Let us calculate the diameter of the piping based on the formula:

$V = \dfrac{354 \text{ x } Q}{D^2}$ and this produces for speeds of 1.8 and 2.5 m/s

$Da = \sqrt{\dfrac{354 \text{ x } Q}{1{,}8}}$ 172 mm in diameter, the nearest commercial diameter is 200 mm.

$Di = \sqrt{\dfrac{354 \text{ x } Q}{2{,}5}}$ 146 mm in diameter, the nearest commercial diameter is 150 mm.

Once the diameters for the cast iron piping have been determined, we can ascertain the friction losses in accordance with the tables.

Suction piping 200 mm in diameter for 150 m^3/h approximately 1%.
Discharge piping 150 mm in diameter for 150 m^3/h approximately 4%.

Manometric suction height

Geometric height		3 metres
Equivalent length		
Length of piping	8 metres	
Foot valve (Equivalent)	30 metres	
Curves of 90° (3x3)	9 metres	
Diffuser cone	5 metres	
Total	**52 metres**	
Friction losses 52 metres x 1 %		0,52 metres
Total manometric suction height		**3,52 metres**

Manometric discharge height

Geometric height		34 metres
Equivalent length		
Length of piping	240 metres	
Diffuser cone	5 metres	
Stop valve	20 metres	
Gate valve	1,5 metres	
Curve of 90° (7x2)	14 metres	
Total	**280,5 metres**	
Friction losses 280.5 metres x 4%		11,22 metres
Total manometric discharge height		**45,22 metres**

TOTAL MANOMETRIC HEIGHT = SUCTION GEOMETRIC HEIGHT FRICTION LOSSES + DISCHARGE GEOMETRIC HEIGHT FRICTION LOSSES

Therefore:

Manometric height = 3,52 + 45,22 =	**48,74**	
Safety margin (+5%)	**2,44**	
Total	**51,18**	**metres**

FHF / FHN Catalogue

Electrical pump model FHF 80-200 with an impeller of 290 mm in diameter capable of raising 150 m^3/h at 52,5 metres is suitable.

OBSERVATION

Assuming that the pump only works at 49 metres and the requisite NPSH is 4.3 metres, but the pump selected has a suction capacity of approximately 5.5 metres and since it is only necessary to have a suction capacity of 3.52 metres in our case, we have an ample safety margin.

NPSH

For a pump to function correctly it must fulfil the following condition: the available NPSH must be greater than the required NPSH.
As a preventive and safety measure it is advisable to add 0.5 m to the value of the NPSH required, making:

$$\mathbf{NPSH_D \geq NPSH_R + 0{,}5\ m}$$

When the pump operates with excessive suction, a low suction pressure develops at the entrance of the pump; the pressure decreases until it can create a vacuum and the liquid turns into vapour. This formation of bubbles that blocks the entrance of the impeller gives rise to the **cavitation** process which causes serious problems in the mechanical parts of the machine. Common defects caused by cavitation are pitting, vibrations and noises. Severe cavitation is normally accompanied by excessive noise and damage to the pump; moderate cavitation is not capable of producing more than a small reduction in capacity, height, output and premature wear and tear.

NPSH (*Net Positive Suction Head*) or positive net height of aspiration is the difference bet-ween the pressure of the liquid in relation to the shaft of the impeller and the pressure of steam from the liquid at the pumping temperature.

There are two types of NPSH:

Available NPSH: This is specific to the installation and is deduced independently of the type of pump by applying the conservation of energy principle between the free surface of the liquid and suction:

$$NPSH_d = \frac{10\ P_a}{\gamma} - H_a - P_{ca} - \frac{10\ T_V}{\gamma}$$

Required NPSH: This is a characteristic of the pump that has to be advised by the manufacturer, and corresponds to the formula:

$$NPSH_r = Hz + \frac{V_a^2}{2\ g}$$

Suction capacity of a pump if the $NPSH_r$ is known

The basic formula that represents the correct suction operation of a pump is as follows:

$$10\ P_a/\gamma \geq H_a + P_{ca} + H_z + V_a^2/2g + 10\ T_v/\gamma$$

$$10\ P_a/\gamma - 10\ T_v/\gamma - H_z \geq H_a + P_{ca} + V_a^2/2g$$

$$NPSH_R = H_z + V_a^2/2g$$

$$H_z = NPSH_R - V_a^2/2g$$

$$10\ P_a/\gamma - 10\ T_v/\gamma - NPSH_R + V_a^2/2g \geq H_a + P_{ca} + V_a^2/2g$$

Finally:

$$\mathbf{H_a + P_{ca} \leq 10\ P_a/\gamma - 10\ T_v/\gamma - NPSH_R}$$

Wherein the values represent the following:

H_a: Geometric suction height in metres.
It is a good sign when the level of the liquid is below the shaft of the pump and a bad sign when it is above.

P_a: Atmospheric pressure or pressure in the suction tank, in kg/cm^2.

P_{ca}: Friction losses during suction (piping, valves, curves and accessories, etc.) in m.

T_v: Vapour tension of the liquid at the pumping temperature, in kg/cm^2.

γ: Total weight of the liquid, in kg/cm^2.

$V_a^2/2g$: Dynamic height corresponding to the speed of entry of the liquid into the pump, in m/sec.

H_z: Minimum pressure necessary in the zone immediately before the impeller vanes, in m.

Practical example

The same pump selected in the practical example for calculating the manometric height (page 9) is used. The aim is to work with water at 60° C and at a height of 600 m above sea level. The manometric suction height data calculated produces:

T^a: 60 °C
T_v: 0,2031 kg/cm^2
γ: 0,9831 kg/dm^3

P_a = 10,33 – 600/900 = 9,66 mca

The $NPSH_r$ obtained from the corresponding curve of the FHF 80-200/209 (2900 rpm) pump in ESPA's technical catalogue has a value of 3.85 m.

$$H_a + P_{ca} \leq 10\ P_a/\gamma - 10\ T_v/\gamma - NPSH_R$$

$$3 + 0{,}46 \leq 9{,}66/0{,}9831 - 2{,}031/0{,}9831 - 3{,}85$$

$$3{,}46 \leq + 3{,}91$$

In summary, the pump will work in the installation without any problems, even though the values have been adjusted.

Vapour tension depends on the temperature of the liquid and the situation above sea level; this requires the following table to make the correct calculation:

Vapour tension and total weight of the water based on temperatures

t °C	Tv Kg/cm^2	γ Kg/dm^3	t °C	Tv Kg/cm^2	γ Kg/dm^3	t °C	Tv Kg/cm^2	γ Kg/dm^3
0	0,0062	0,9998	92	0,7710	0,9640	122	2,1561	0,9414
10	0,0125	0,9996	94	0,8307	0,9625	124	2,2947	0,9398
20	0,0238	0,9982	96	0,8942	0,9611	126	2,4404	0,9381
30	0,0432	0,9955	98	0,9616	0,9596	128	2,5935	0,9365
40	0,0752	0,9921	100	1,0332	0,9583	130	2,7544	0,9348
50	0,1258	0,9880	102	1,1092	0,9568	135	3,192	0,9305
60	0,2031	0,9831	104	1,1898	0,9554	140	3,685	0,9260
70	0,3177	0,9777	106	1,2751	0,9540	145	4,237	0,9216
75	0,3931	0,9748	108	1,3654	0,9525	150	4,854	0,9169
80	0,4829	0,9718	110	1,4609	0,9510	155	5,540	0,9121
82	0,5234	0,9705	112	1,5618	0,9495	160	6,302	0,9073
84	0,5667	0,9693	114	1,6684	0,9479	165	7,146	0,9023
86	0,6129	0,9680	116	1,7809	0,9464	170	8,076	0,8973
88	0,6623	0,9667	118	1,8995	0,9448	175	9,101	0,8920
90	0,7149	0,9653	120	2,0245	0,9431	180	10,225	0,8869

$$T_v\ (\text{m.c.l.}) = T_v\ (\text{kg/cm}^2) \times 10/\gamma$$
$$T_v\ (\text{m.c.a.}) = T_v\ (\text{kg/cm}^2) \times 10$$

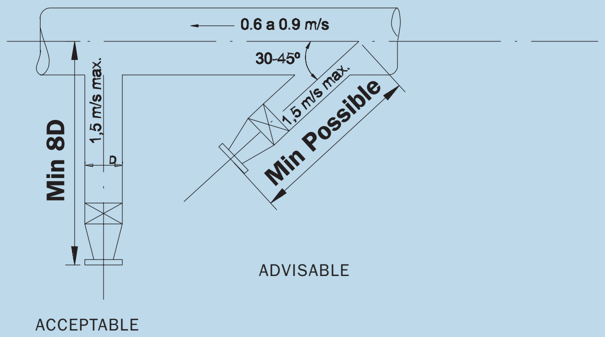
Effect of atmospheric pressure on altitude

It may be determined using the following formula:

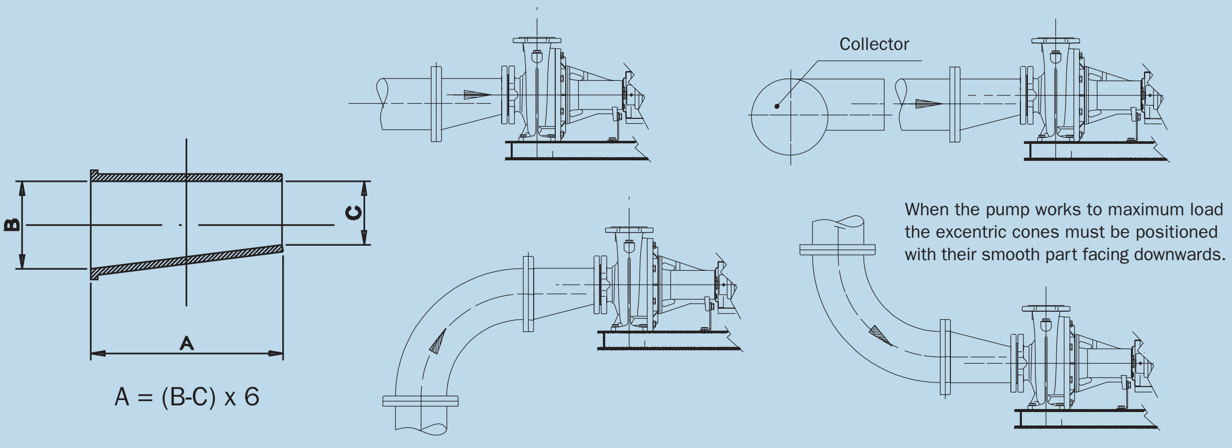
$$P_a\ (\text{m}) = 10{,}33 - \text{Altitude (m)} / 900$$

Suction Design

Correctly dimensioned suction piping and design has a positive effect on the smooth running of the pump. It is advisable to restrict the speeds in a suction pipe to 1.8 m/s if the flow rate is uniform. In a collector from which two or more pumps are suctioned, it is not advisable to have a speed in excess of 0.9 m/s for the main fluid. For the lateral connections at an angle of 30° to 45°, the speed of the fluid for the main line may be increased to 1.5 m/s.

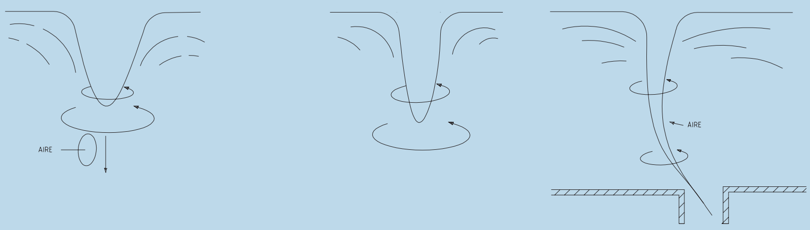


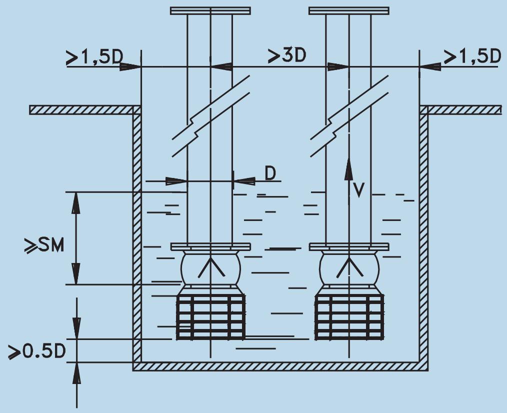
When the diameter of the suction aperture of the pump is lower than the diameter of the suction piping, it is necessary to install an excentric diffuser cone with the straight side in the upper part of the piping, the straight side is positioned downwards when the supply source is above the pump.



Vortex formation in the suction tank

It is frequently necessary for a pump to suction from a tank with the suction piping submerged a minimal distance.



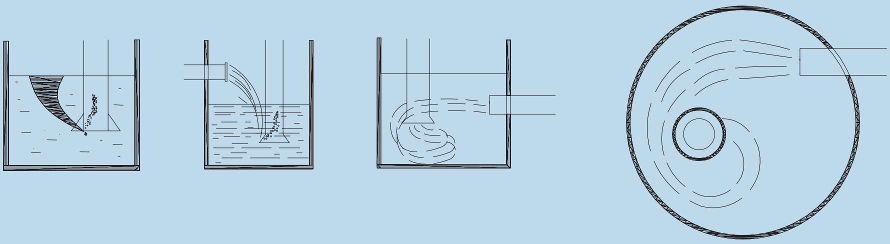


To prevent vortex formation, minimum submergence is necessary, expressed by:

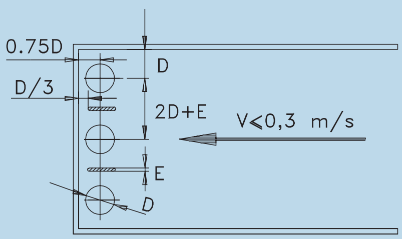
$S_m = V^2/2g + 0,1$

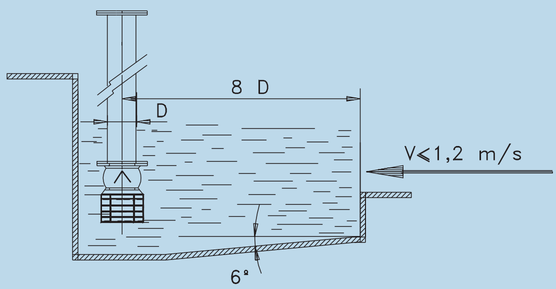
S_m : Minimum submergence (m)
V : Velocity suction (m/s)
g : Gravity (9.8 m/s^2)

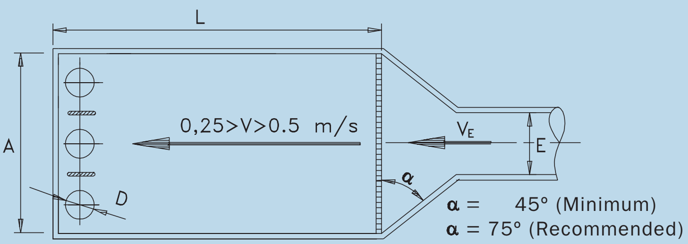
If the supply or return pipe to the well discharges radially above the level of the liquid, there is a danger of air entering and generating speeds that will adversely affect the smooth running of the pump.
If it is not possible to achieve the required height of liquid, the installation of brattices, anti-vortex plates, separators and appropriate speeds may provide the solution to the majority of these problems.



Abrupt changes of section between the entrance and the well must be avoided. The change must be gradual, with 45° coning, which in this case must be the speed in the well below 0.3 m/sec.
The design based on a small-sized pipe directly connected to the well with pumps situated near the entrance is not recommended at all. In this case, the current must produce major changes in direction to reach all the pumps. Neither is it advisable to centre the pumps in the well since it produces large areas of vortexes in the rear part.







A/E	1	1.5	2	4	10
L	3D	6D	7D	10D	15D
V_E	0.3	0.6	1.2	1.8	2.4

Pressure Equipment

Design of the Pressure Sets in accordance with the Basic Regulations of the new Technical Building Code (art. 3 of the LOE) applicable to Spain.

SUPPLIES		TYPES OF HOUSING/FLOW RATES									
		A	**L/S**	**B**	**L/S**	**C**	**L/S**	**D**	**L/S**	**E**	**L/S**
KITCHEN	LAUNDRY	1	0,2	1	0,2	1	0,2	1	0,2	1	0,2
	SINK	1	0,2	1	0,2	1	0,2	1	0,2	1	0,2
	DISHWASHER					1	0,2	1	0,2	1	0,2
OFFICE	TAP							1	0,15	1	0,15
LAUNDRY	TAP			1	0,2	1	0,2	1	0,2	1	0,2
COMPLETE BATHROOM	WC	1	0,1			1	0,1	1	0,1	2	0,2
	WASHBASIN	1	0,1			1	0,1	1	0,1	2	0,2
	BATH					1	0,3	1	0,3	2	0,6
	BIDET					1	0,1	1	0,1	2	0,2
WASHROOM	WC			1	0,1			1	0,1	1	0,1
	WASHBASIN			1	0,1			1	0,1	1	0,1
	SHOWER			1	0,2			1	0,2	1	0,2
TOTAL APPLIANCES-L/S		**4**	**0,6**	**6**	**1**	**8**	**1,4**	**12**	**1,95**	**16**	**2,55**

NOTE: A different type of study is required for installations with flushing.
NOTE: They should be designed in such a way that the sets will not start up if the network pressure is sufficient. They should consist of two sets of equipment to operate alternately, with pumps of identical characteristics mounted in parallel. They should be accompanied by membrane pressure tanks connected to enough devices to gauge the pressure of the installation for it to be stopped and started automatically.

1. Flow to be pumped according to the type and number of dwellings

NUMBER OF DWELLINGS	TYPE OF DWELLING				
	A	**B**	**C**	**D**	**E**
	Total flow rate of the pump(s) in m^3/h				
0 - 10	1,5	2,1	3	3,6	4,5
11 - 20	2,4	3,6	5,1	6	7,5
21 - 30	3,6	4,5	6,6	8,4	10,8
31 - 50	5,4	9	10,8	13,2	16,8
51 - 75	9	13,2	15	17	19,2
76 - 100	12	16,2	17,4	19,2	
101 - 150	15	18	19,2		

NOTE: The number of pumps to be installed in a conventional group, excluding reserve pumps, will depend on the total flow rate of the group. 2 pumps will be put in place up to a flow rate of 10 l/s (36/s m^3/h), 3 pumps up to 30 l/s (108 m^3/h) and 4 pumps for flow rates greater than 30 l.

2. Calculation of the pressure

Opening pressure: Geometric height + total friction losses for the installation + pressure required at the most unfavourable point.
Shut-off pressure: Opening pressure + 15 to 30 metres.

MINIMUM STARTING PRESSURE: This is obtained by adding 15 metres to the geometric height from the minimum water level or base of the pumps until the roof of the highest floor that has to be supplied plus friction losses.

$$P_b = H_a + H_g + P_c + P_r$$

Where:

P_b = Minimum pressure of starting, H_a = Suction height, H_g = Geometrical height, P_c = Friction loss, P_r = Residual pressure.

NOTE: Friction losses must be fixed at about 10 – 15% of geometrical height.

MAXIMUM SHUT-OFF PRESSURE: The shut-off pressure will be between 15 and 30 metres greater than the starting pressure. The maximum pressure at the point of consumption may not exceed 5 Kg/cm^2.

3. Capacity of the tank according to the type and number of dwellings

TANK OR ACCUMULATOR	TYPE OF DWELLING				
	A	**B**	**C**	**D**	**E**
	COEFFICIENT				
WITH INJECTORS	40	50	60	70	80
MEMBRANE, WITH COMPRESSOR	15	18	20	23	26

The volume of the tank will be equivalent to or greater than the result produced by multiplying the coefficient by the number of dwellings. The installation of injectors for operating pressures in excess of 8 kg/cm^2 is not recommended.

4. Break-pressure tank

In accordance with the Technical Code for

Construction (art. of the Law for Regulating Construction), applicable in Spain, in front of the pressure group (in suction) it is obligatory to include a BREAK-PRESSURE TANK with the following capacity, calculated according to the requisites of UNE standard 100.030:2.005:

$$V = Q \times t \times 60$$

Where: V = Volume (l), Q = Flow rate (l/s),
t = Time (15 – 20 minutes)

Pressure groups with regulatable action:
They will not require the auxiliary supply tank.
They should include a device which causes the closure of suction and shutdown of the pump in cases of collapse of the supply pipes.

Calculation of the pressure group

Flow rate

1. Let us calculate the flow rate installed and the number of supplies per dwelling using the following table:

SUPPLY	FLOW RATE L/S	SUPPLY	FLOW RATE L/S
KITCHEN SINK	0,2	WASHBASIN	0,1
OFFICE	0,15	WC WITH TANK	0,1
AUT. WASHING MACHINE	0,2	BIDET	0,1
DISHWASHER	0,2	BATH	0,3
KITCHEN SINK HOTEL	0,3	SHOWER	0,2
WASTE OUTLETS	0,2	URINARY FAUCET	0,05
FLUSHERS	1,25-2	AUTOMATIC URINAL	0,1

2. The simultaneity coefficient of a dwelling may be determined using the formula

$$K = \frac{1}{\sqrt{n-1}}$$

n: Number of supplies per dwelling

3. The economic flow rate installed for a dwelling will be:

Economic flow rate = K x Flow rate installed

4. Let us calculate the simultaneity coefficient for all the dwellings using the formula:

$$K_v = \frac{19 + N}{10\,(N + 1)}$$

N: Total number of Dwellings

5. The total flow rate required to supply all the dwellings will still be determined by

Total flow (L/S) =
Number of dwellings x economic Flow Rate x K_v

Tanks

Volume of the tank

$$V_d = k \frac{Q_m}{3N} \times \frac{P_p + 1}{P_p - P_a}$$

Where:
k = 0,33 (for membrane drums).
k = 0,45 (for galvanised drums with compressor).
k = 1 (for galvanised drums with injector).

and:

kW	N
$P_2 \leq 2{,}2$	30
$2{,}2 > P_2 \leq 5$	25
$5 < P_2 \leq 20$	20
$20 < P_2 \leq 100$	15

Total volume

$$V_u = 0{,}8\, V_d \times \frac{P_p - P_a}{P_p + 1}$$

Wherein the values represent the following:
V_d : Volume of tank in m^3
V_u : Total volume of the tank in m^3
Q_m : Average flow rate (Qa + Qp)/2 in m^3/h
Q_a : Opening pressure flow in m^3/h
Q_p : Shut-off pressure flow in m^3/h
P_p : Shut-off pressure in/cm^2
P_a : Opening pressure in kg/cm^2
N : Frequency of start-ups/hour

The pre-charge of air in the tank affects the tank volume and total volume.

Speed control provides energy saving, reduces space, prevents premature waste and water hammer.

Calculation of pressure equipment requires a detailed study in order to calculate the water requirements for:

Housing estates	Schools
Barracks	Hospitals
Irrigation	Commercial establishments
Markets	Public swimming pools
Industrial plants	Purifying plants
Hotels	Office buildings

Fundamental Relationships of Centrifugal Pumps

Speed variation

When the speed changes in line the constant impeller diameter, the flow, pressure and power vary simultaneously in accordance with the **laws of similarity or affinity** based on the following expression.
The flow rate of a pump increases or decreases proportionately to the increase or decrease in speed.

$$Q_1 = Q \cdot \frac{n_1}{n}$$

The manometric height increases or decreases with the square of speed.

$$H_1 = H \cdot \left(\frac{n_1}{n}\right)^2$$

The absorbed power increases or decreases with the cube of velocity.

$$P_1 = P \cdot \left(\frac{n_1}{n}\right)^3$$

The NPSH is proportionate to the square of the speed variation.

$$NPSH_{r1} = NPSH_r \cdot \left(\frac{n_1}{n}\right)^2$$

These relationships do not precisely coincide if the speed relationship is greater than 2.
Neither do they coincide if the suction conditions are inadequate.
Speed variation is the most efficient way of varying the characteristics of a pump subject to variable functioning conditions.
In those cases where the aim is to increase the speed of a pump, it is advisable to consult the manufacturer beforehand since the increase in speed may be restricted for the following reasons:

- Mechanical resistance of the shaft and the bearings since the power increases.
- Resistance to the pressure from the body of the pump since the pressure also increases.
- Modification of the suction power from the pump since the latter is not maintained proportionately to the increased flow rate.

Variation in line with the diameter of the impeller

Let us assume that the speed is set. When the diameter of the impeller is changed, the tangential speed changes proportionately and with it the flow rate, height and power in accordance with the following expression:

Flow rate $$Q_1 = Q \cdot \left(\frac{D_1}{D}\right)^2$$

Manometric height $$H_1 = H \cdot \left(\frac{D_1}{D}\right)^2$$

Absorbed power $$P_1 = P \cdot \left(\frac{D_1}{D}\right)^3$$

These relationships may only be applied if negligible changes in the diameter of the impeller (decrease up to 15 - 20% of the maximum diameter) and the vanes take place.
This is only feasible for radial impellers and for some mixed-flow impellers. In pumps with a diffuser only, the vanes are mechanised to the new diameter.
Although it has been assumed that the output is constant in all cases, it is nevertheless specified that the decrease in output is negligible for low-speed pumps, but pumps with substantial specific speeds are subject to substantial decreases in output.
It is not possible to reduce the diameter of impellers with side channels.
It is advisable to reduce the diameter of the impeller gradually and to test the pump in order to ascertain whether the desired effects have been achieved.

Calculation of the total volume of a pumping well

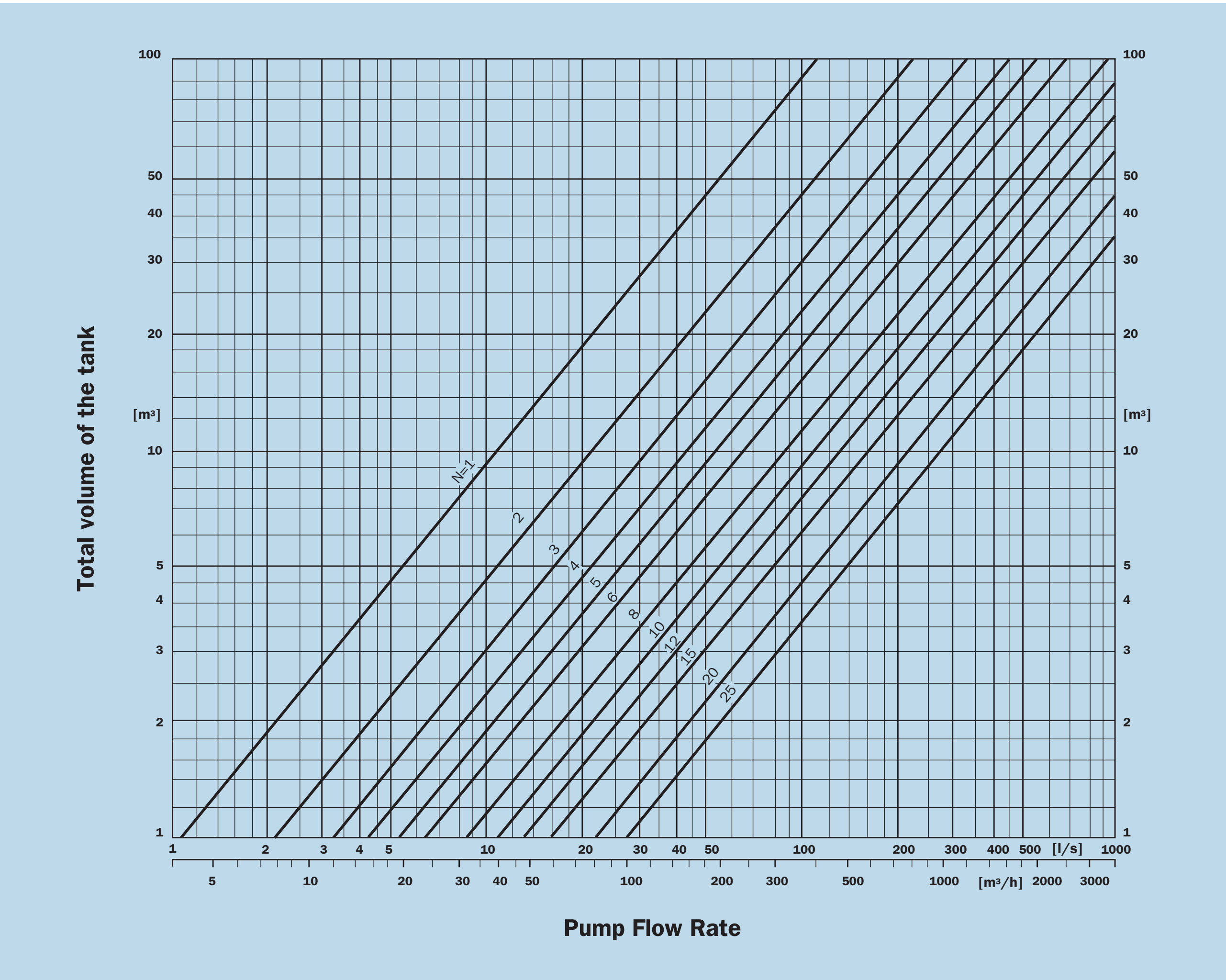
The worst case scenario for the calculation is obtained when the incoming flow rate is equal to the centre of the flow rate of the pump.
The minimum volume of water in the well depends on the frequency of start-ups by the engine per hour and on the largest flow rate of the pump in operation, so that:

$$V_u = \frac{Q}{4 \cdot N}$$

Whereby:
V_u : Total volume (m^3).
Q : Flow rate of the pump (m^3/h).
N : Frequency of start-up (start-ups/hour)

kW	N
0 – 5	15
5 – 20	13
20 – 100	11
100 - 400	10

The size of the pumping well must be sufficient for the total volume and for the pumps to function without hydraulic perturbations during suction (see pages 13 and 14), and taking into consideration the differences in the stop-start levels of the various units. The start-up frequency will be lower when two or more pumps work alternatively.



Nozzle tips and jets of water

Exit of water through an opening is deduced by the formula:

Flow rate: $Q = V \cdot S$ $\quad Q = K \cdot S \cdot \sqrt{2gH}$

Speed: $V = K \cdot \sqrt{2gH}$

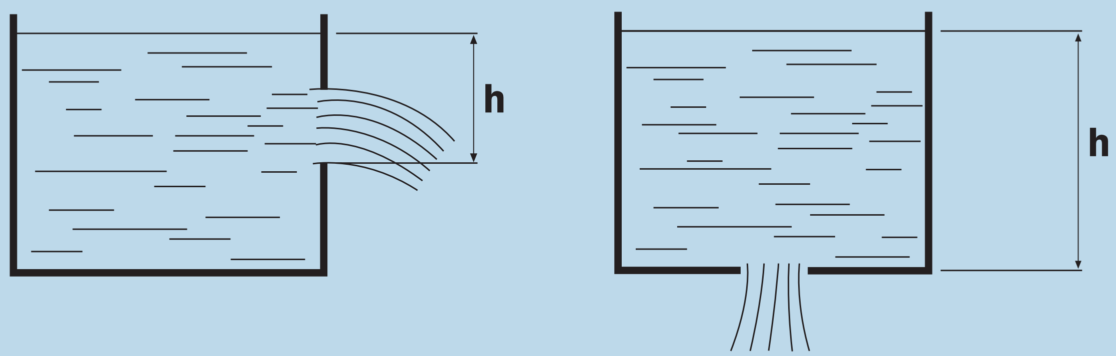
Wherein the values represent the following:

- Q : Flow rate in m^3/h
- V : Speed in m/s.
- S : Surface area of the opening m^2
- H : Load on the opening in metres
- g : Acceleration of gravity (9,81 m/s^2)
- K : Exit coefficient ≈ 0,62

If the opening is circular, practical consumption is approximately 62% of the theoretical.

For K = 0,62 we have the simplified formula

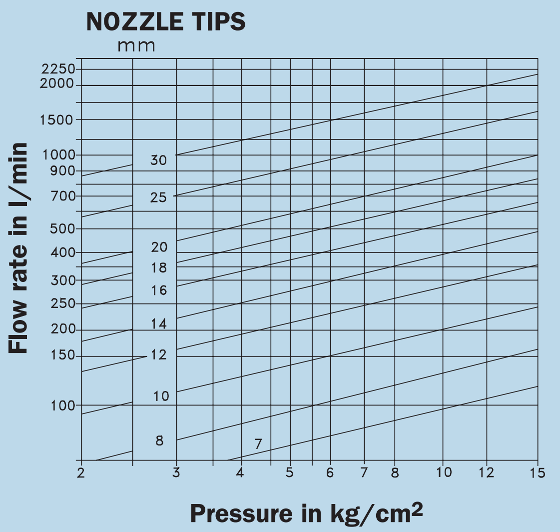
$Q(m^3/h) = S\ (cm^2) \times \sqrt{H\ (m.c.a.)}$

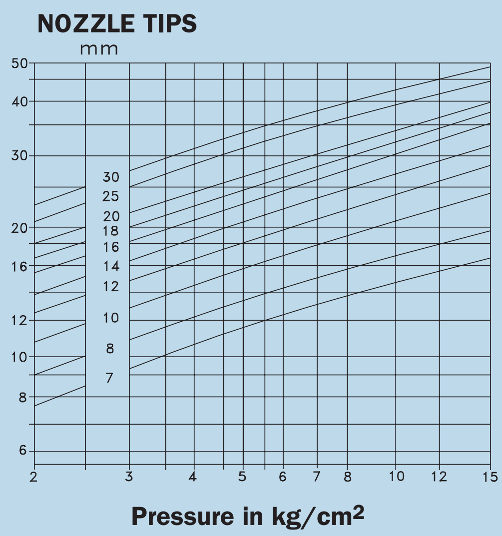


In the specific case of **jets of water** with conical polished nozzles, we take a discharge coefficient of 0.97 into consideration to determine the flow rates gushing forth in accordance with the pressures, using the formula:

$Q(l/min) = 0{,}64D^2\ (mm) \cdot \sqrt{H\ (kg/cm^2)}$

The ranges are obtained for an incline of 30° and a low wind.





Pumping of viscous liquids

The characteristic curves of the pumps are based on water and have a kinematic viscosity of approximately 1 cSt. An increase in viscosity has a noticeable effect since it is necessary to apply correction factors to the flow rate, height and output of the pump, and to the conditions of viscous fluid, in order to ascertain the characteristics that are equivalent to water.

- There is no noticeable decrease in the flow rate and height below 43 cSt.
- Power increases from 4.3 cSt upwards.
- When friction losses during suction increase, pumps with the requisite low NPSH must be used.
- The correction factors obtained in the charts are generally sufficiently accurate for application.

Limitation of the charts

- Only use them for open or closed radial type impellers, never with mixed-flow or axial impellers.
- In multi-phase pumps the height of the impeller must be used for the calculations; its accuracy is affected owing to additional losses between phases.
- In pumps with double-entry impellers, half of the flow rate must be used in the calculations.
- When pumping liquids with a high level of viscosity it is advisable to study the operating costs to assess whether different types of pumps are more economical due to the considerable loss of output generated by centrifugal pumps.
- The correction factors are only applicable to homogenous Newtonian liquids; they are not applicable to gelatinous liquids, paper pulp, fluids with solids or fibres etc.

Practical example

- If the flow rate and height of the viscous fluid are known, enter them on the chart and ascertain the correction factors.
- We use this data to determine the corresponding values for water and then select the pump.
- Using the characteristic curve for water, apply the corresponding correction factors to obtain the new values for the viscous fluid.

Specify a pump for raising 150 m^3/h of viscous fluid at a height of 28.5 mca. Viscosity 200 cSt, volume weight 0.9 kg/dm^3.
The curve 1.0 x Q is used to determine the correction factor for the height.

$$f_Q = 0{,}95 \qquad f_H = 0{,}91 \qquad f_\eta = 0{,}62$$

We calculate the values for water with these factors.

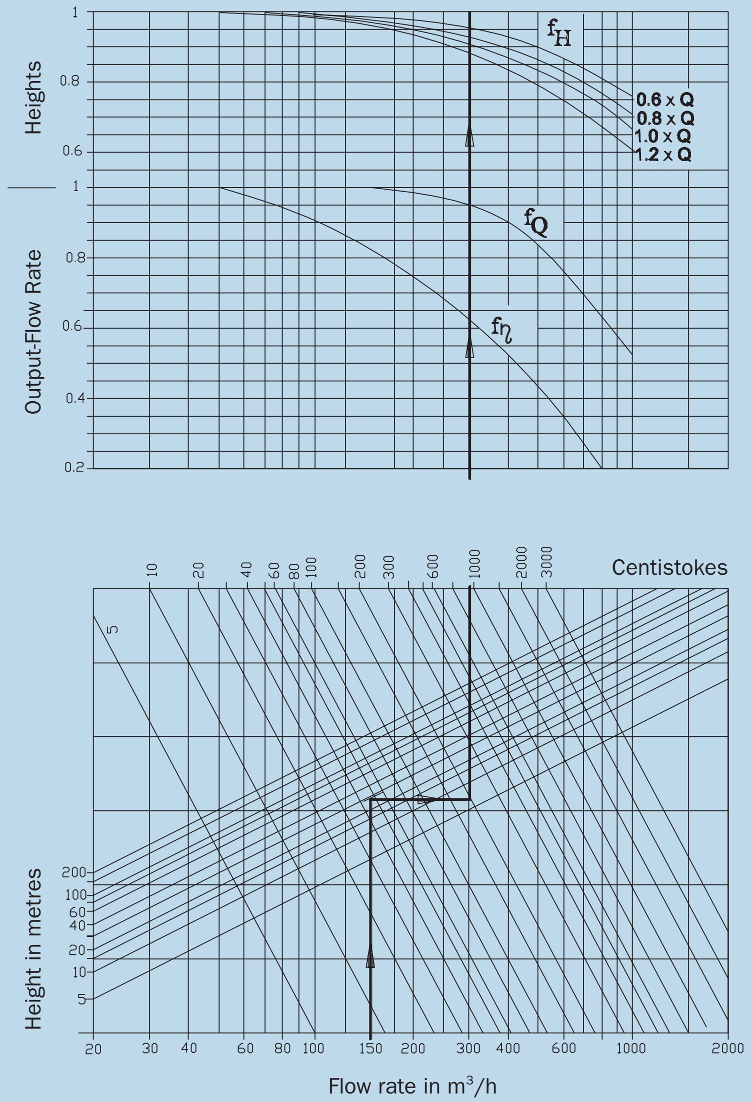
$$Q = \frac{150}{0{,}95} = 158\ m^3/h$$

$$H = \frac{28{,}5}{0{,}91} = 31{,}3\ \text{mca}$$

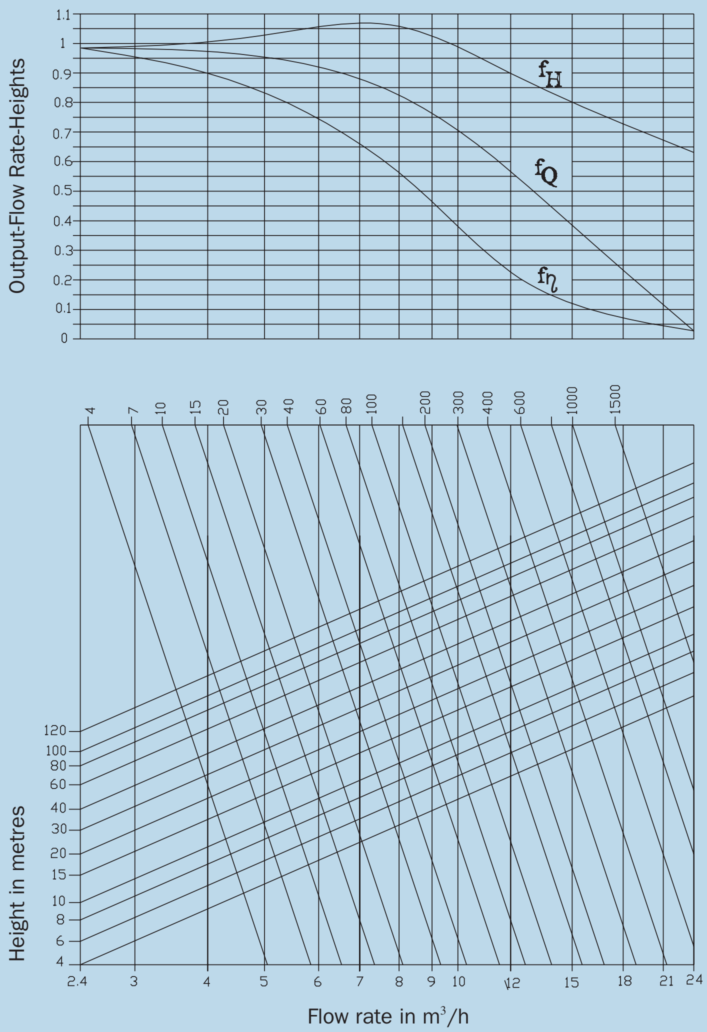
On the basis of these values we select pump model FLH 80-160 with a diameter of 173 mm at 2,900 rpm; we determine the values for the flow rate, height and output from the characteristic curve with water.
Applying the various correction factors, we obtain the new operating conditions for the viscous fluid.
We summarise the calculations in the following chart.

		0,6 Q	0,8 Q	1 Q	1,2 Q
WATER	FLOW RATE (Q)	95	126	158	190
	HEIGHT (H)	37,6	35	31,3	26,9
	OUTPUT	71	78	81	78
VISCOSITY CENTISTOKES		200			
CORRECTION FACTORS	f Q	0,95			
	f H	0,955	0,925	0,91	0,88
	f η	0,62			
VISCOUS LIQUID	Q v	90	120	150	180
	H v	35,9	32,4	28,5	23,7
	η v %	41	48,4	50	48,4
	Volume weight (kg/dm^3)	0,9			
	Absorbed power (CV v) $CV_V = \frac{Qv \times Hv \times \gamma}{270 \times \eta v}$	24,5	26,77	26,5	29,3

CORRECTION FACTORS (according to the example)



CORRECTION FACTORS



Conversion of viscosities

For the purpose of calibrating viscosities, the following factors provide an approximate conversion between viscosities:

SSU = CENTITSTOKES x 4,62
SSU = REDWOOD 1 (NORMAL) x 1,095
SSU = REDWOOD 2 (ADMIRALTY) x 10,87
SSU = SAYBOLT FUROL x 10
SSU = ENGLER DEGREES x 34,5
SSU = PARLIN SECONDS CUP Nº 15 x 98,2
SSU = PARLIN SECONDS CUP Nº 20 x 187,0
SSU = SECOND FORD CUP Nº 4 x 17,4

$$\text{DYNAMICVISCOSITY (CENTISTOKES)} = \frac{\text{KINEMATIC VISCOSITY (CENTIPOISES)}}{\text{UNIT WEIGHT}}$$

CENTISTOKES = SSU x 0,21645

THE TEMPERATURE HAS A CONSIDERABLE EFFECT ON VISCOSITIES AND UNIT WEIGHT

Water hammer

A water hammer is defined as excess pressures produced in the pipes before any modification of the circulation rate of the fluid that passes through them (opening or closure of a valve, start-up or standstill of a pump, etc.) and as a result of the modification of the kinetic energy of the fluid in motion.

If the pump shuts down, the water hammer primarily manifests in a depression followed by excess pressure.

The standstill time T is the time that has passed between the energy shut-off, opening or closure of the valve and the moment at which the circulation rate of the liquid is cancelled. The Mendiluce formula permits us to make an approximate calculation of the stillstand time:

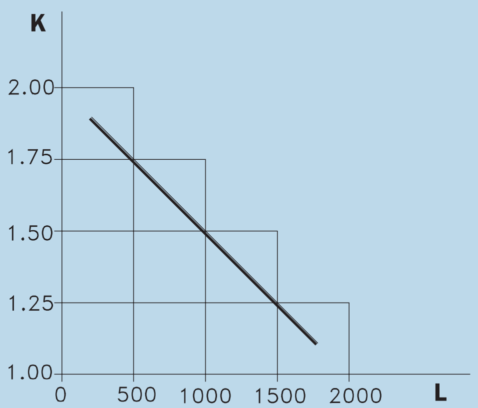
$$T = C + \frac{K \cdot L \cdot V}{g \cdot H_m}$$

Whereby:
L : Length of conduction (m).
V : Speed of the liquid (m/s).
g : Speed of gravity (m/s^2).
H_m : Manometric height (mca).

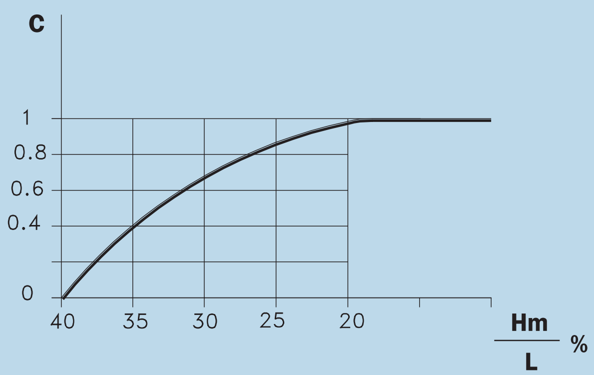
Great care must be exercised when calculating the water hammer for inclines in excess of 50%, with exclusive application of the Allievi formula being advisable since standstill is very rapid in these cases.
We wish to emphasise that the manometric height that is applied in the calculation of T must be measured immediately upstream of the pump and must therefore take the depth of the water level into consideration when it is raised by means of pumps submerged in wells. L. Allievi deduced that the water hammer is a fluctuating phenomenon that spreads along the piping at a speed of:

$$a = \frac{9.900}{\sqrt{48 + K_1 \cdot \frac{D}{e}}}$$

Whereby:
a : Speed of propagation (m/s)
D : Diameter of the piping (mm.)
e : thickness of the piping in mm.



The coefficient K principally represents the effect of the inertia of the motor-pump set and its values vary according to the length of discharge.



The C coefficient is a result of our experience and is according to the incline (Hm/L)

Calculation of K_1 :

$$K_1 = \frac{10^{10}}{E}$$

Where
E: coefficient of elasticity of the piping (kg/m^2).
Practical values of K_1:

Steel	0,5
Cast iron	1
Cement	5
Fibrous cement	5,5
Polyester	6,6
P.V.C.	33,3

Hydraulic studies recommend using the following formulas to calculate excess pressure:

For $L < \frac{a \cdot T}{2}$ (brief discharge),

Michaud formula $$\Delta H = \frac{2 \cdot L \cdot v}{g \cdot T}$$

For $L > \frac{a \cdot T}{2}$ (long discharge),

Allievi formula $$\Delta H = \frac{a \cdot v}{g}$$

For all discharge, even when

$$L > \frac{a \cdot T}{2}$$

is fulfilled and the **Allievi** formula must therefore be applied, if conduction is followed in the circulatory direction of the water, there will always be an intermediate point that will fulfil

$$L_c = \frac{a \cdot T}{2} \quad \text{(critical length)}$$

and this will produce

$$L_c < \frac{a \cdot T}{2}$$

with the application of the **Michaud** formula being mandatory in this area.

The maximum pressure achieved by discharge will be equal to the sum of static pressure or geometric height, with the maximum excess pressure + ΔH.

H max = Hg + ΔH

The minimum pressure will be the difference between static pressure or geometric height and minimum excess pressure – ΔH.

H min = Hg – ΔH

In long as well as short discharges, the water hammer may attain a value that is greater than static pressure and consequently produce a depression in the piping, below the atmospheric pressure, with a potential rupture of the liquid vein. It should be added that the pipes are generally very capable of resisting depressions in the region of 1 kg/cm^2, which far exceeds anything that would actually occur in practice.

Protection against the water hammer

The water hammer may be reduced or prevented by means of systems designed for this purpose, such as:
Flywheels
Surge shafts
Air reservoirs
Bladder dampers
Safety valves
Vents
Stop valves
Stop valves with differential by-pass
Anti-surge stop valve
Moderately by means of static starters or speed variators

Selection of power cables

We must take the following factors into consideration to determine the type of power cable.

- The maximum admissible intensity for copper conducts with EPDM insulation according to the Low Voltage Regulations.
- Maximum fall in voltage which must not exceed 3% of the value of the nominal voltage.
- cos φ 0,85
- Ambient temperature 40 °C.

We use the following formulas for the calculation.

Single-phase Current

$$S = \frac{2 \cdot L \cdot I \cdot \cos\varphi}{C \cdot \Delta U}$$

Three-phase Current (Direct Start-up)

$$S = \frac{\sqrt{3} \cdot L \cdot I \cdot \cos\varphi}{C \cdot \Delta U}$$

Three-phase Current (Star-delta starting)

$$S = \frac{2 \cdot L \cdot I \cdot \cos\varphi}{\sqrt{3} \cdot C \cdot \Delta U}$$

Wherein the values represent the following:

S	:	Section of the cable in mm^2.
I	:	Nominal Intensity of the engine in Amp.
L	:	Length of the cable in metres.
cos φ	:	Power factor under full load.
ΔU	:	Fill in line voltage, 3%. E.g.: For 230 V = 6.9 V For 400 V = 12 V
C	:	Electric conductivity (56 m/mm^2 for Cu and 34 m/mm^2 for Al).

Maximum Admissible Intensity for a TRIPOLAR OR TETRAPOLAR CABLE HO7RNF model or similar (according to R.B.T.)

Section (mm)	1,5	2,5	4	6	10	16	25
In. Max. (Amp.)	17	25	34	43	60	80	105

Section (mm)	35	50	70	95	120	150	185
In. Max. (Amp.)	130	160	200	250	290	335	385

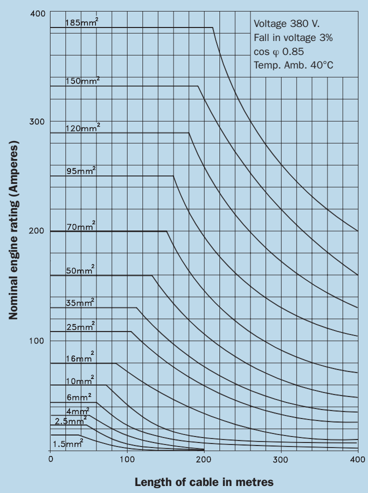
The rise in temperature caused by the electric current must not give rise to a temperature in the conductor that is in excess of the temperature permitted by the insulation, namely 90°C. The following correction factors will be used for ambient temperatures above 40°C.

Temperature °C	15	20	25	30	35	40	45	50
Correction Factor	1,22	1,18	1,14	1,1	1,05	1	0,95	0,9

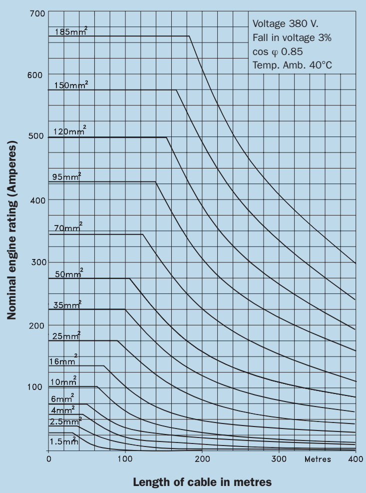
The cables are equally affected by other causes, cables directly exposed to the soil (factor 0.9), cables installed inside a tube, to the air or embedded (factor 0.8), grouping of various cables, etc.

DIRECT START-UP

Auto-transformer - Stator resistance
Static starter - Speed variator



STAR-DELTA STARTING



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