Pumping Stations Design

For Infrastructure Master Program
Engineering Faculty-IUG

Lecture 1: Introduction to pumping stations

Dr. Fahid Rabah
Water and environment Engineering
frabah@iugaza.edu

1.1 Purpose and types of pumping stations

The main purpose of pumping stations is to transfer fluids from low points to higher points.

The main types of pumping stations are:

- Wastewater PS.
- Water PS.
- Sludge PS.
- Storm water PS.

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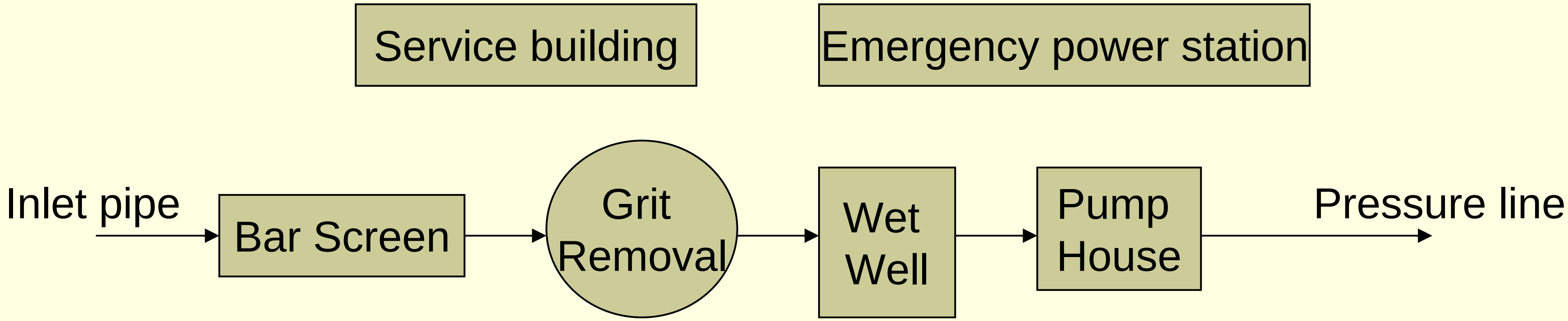

Typical Layout of Wastewater Pumping Station

Manual Bar Screen

Mechanical Bar Screen

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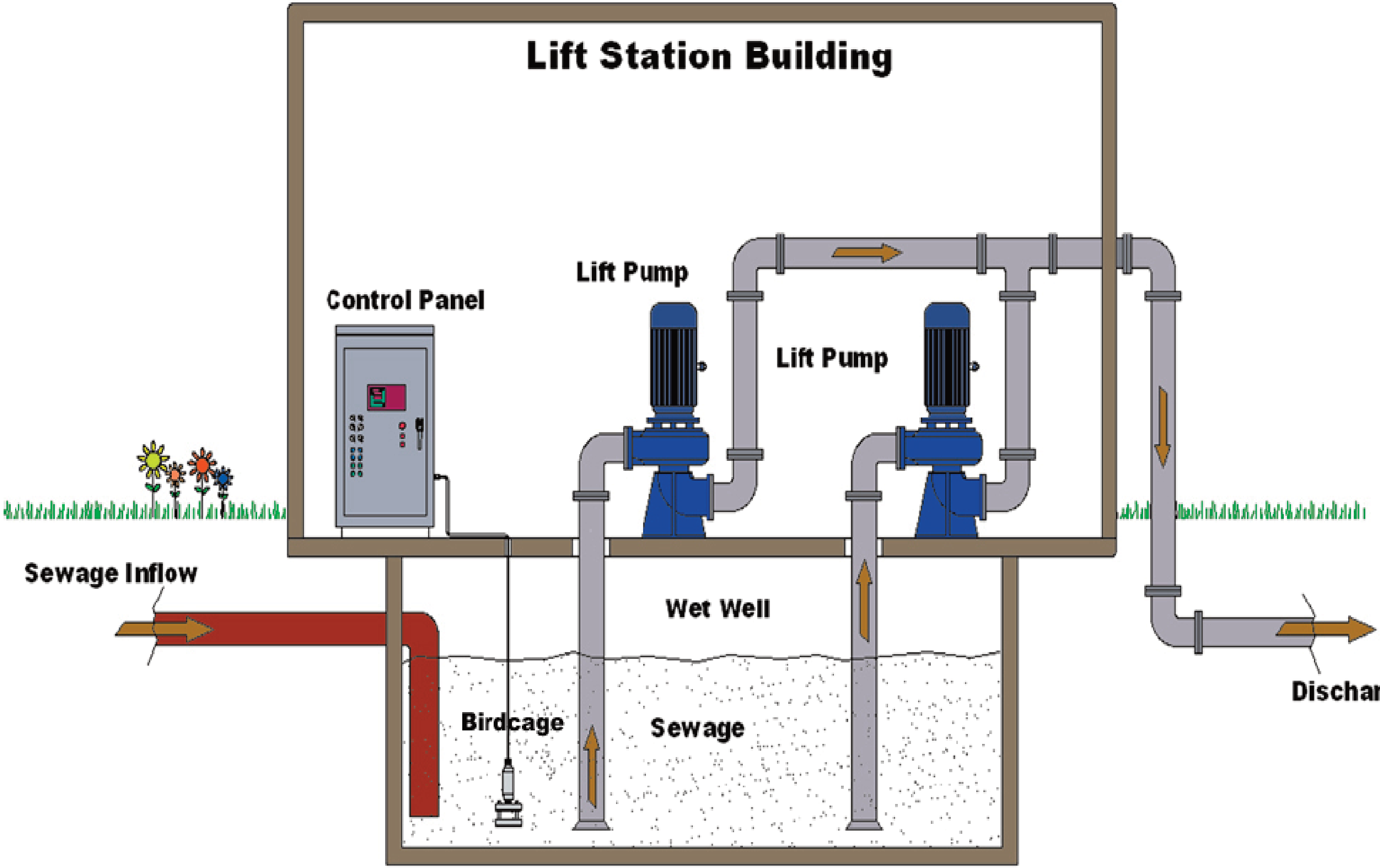
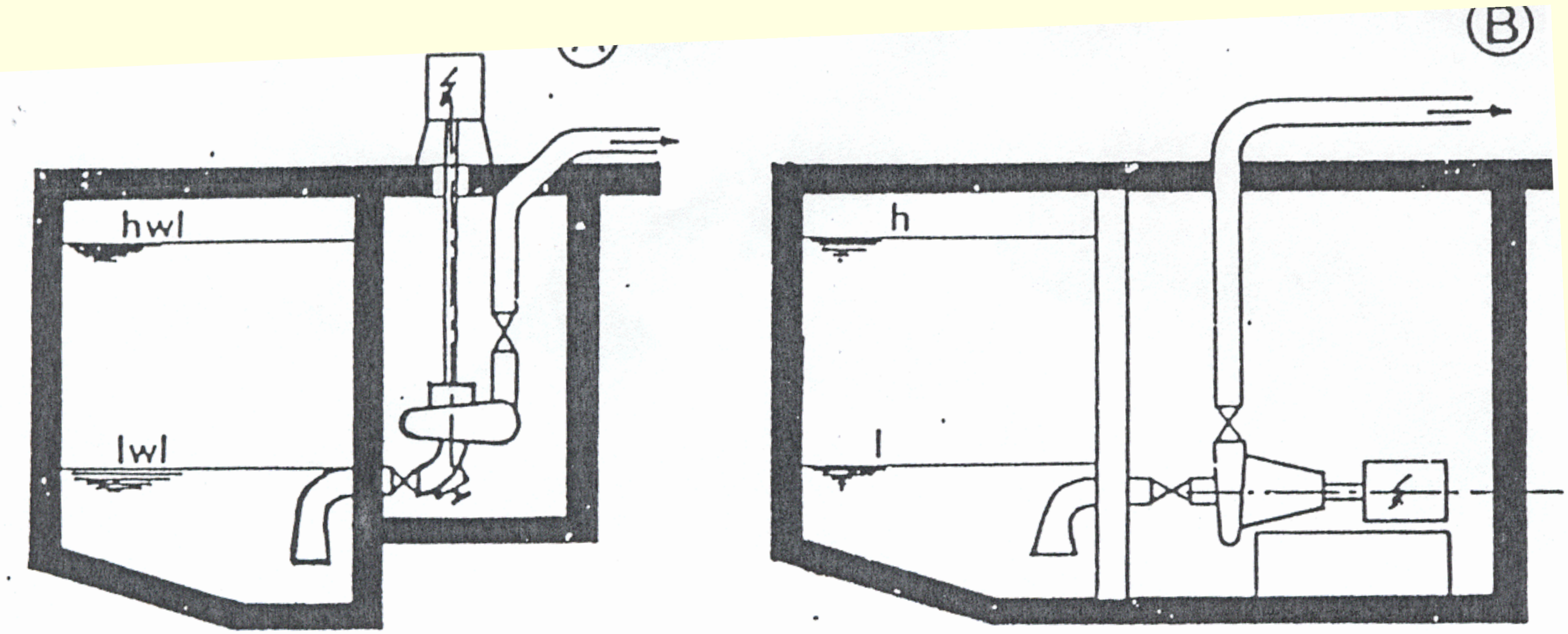
Grit Removal Channel



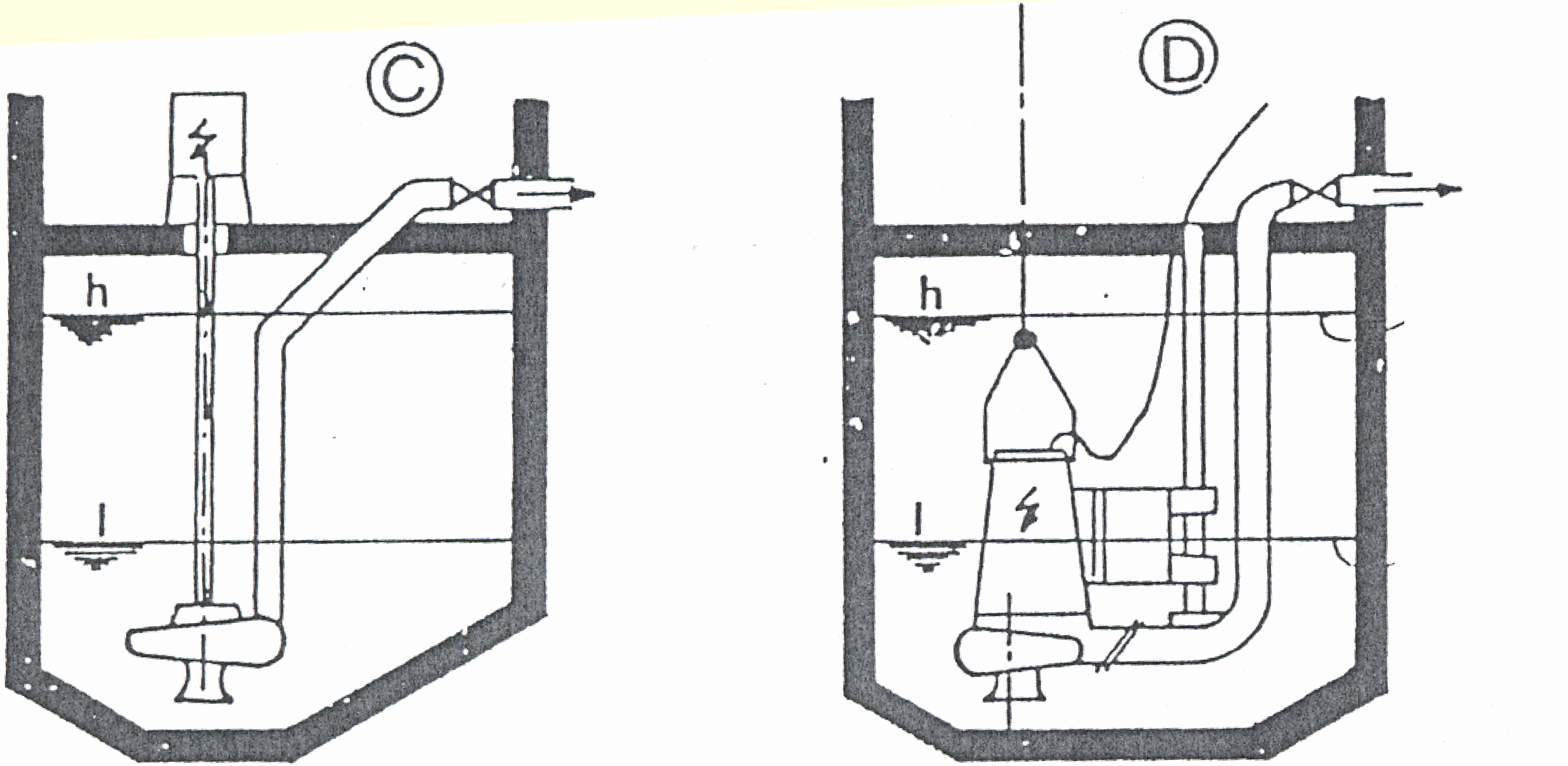
Wet well and Pump House



Dry, vertical execution
(long pump shaft)

Dry, horizontal execution
(large pump cellar)

Dry well Pump House



Wet, vertical execution

Submersible pumps (wet execution)

Wet well Pump House

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Pumping station Control Panel

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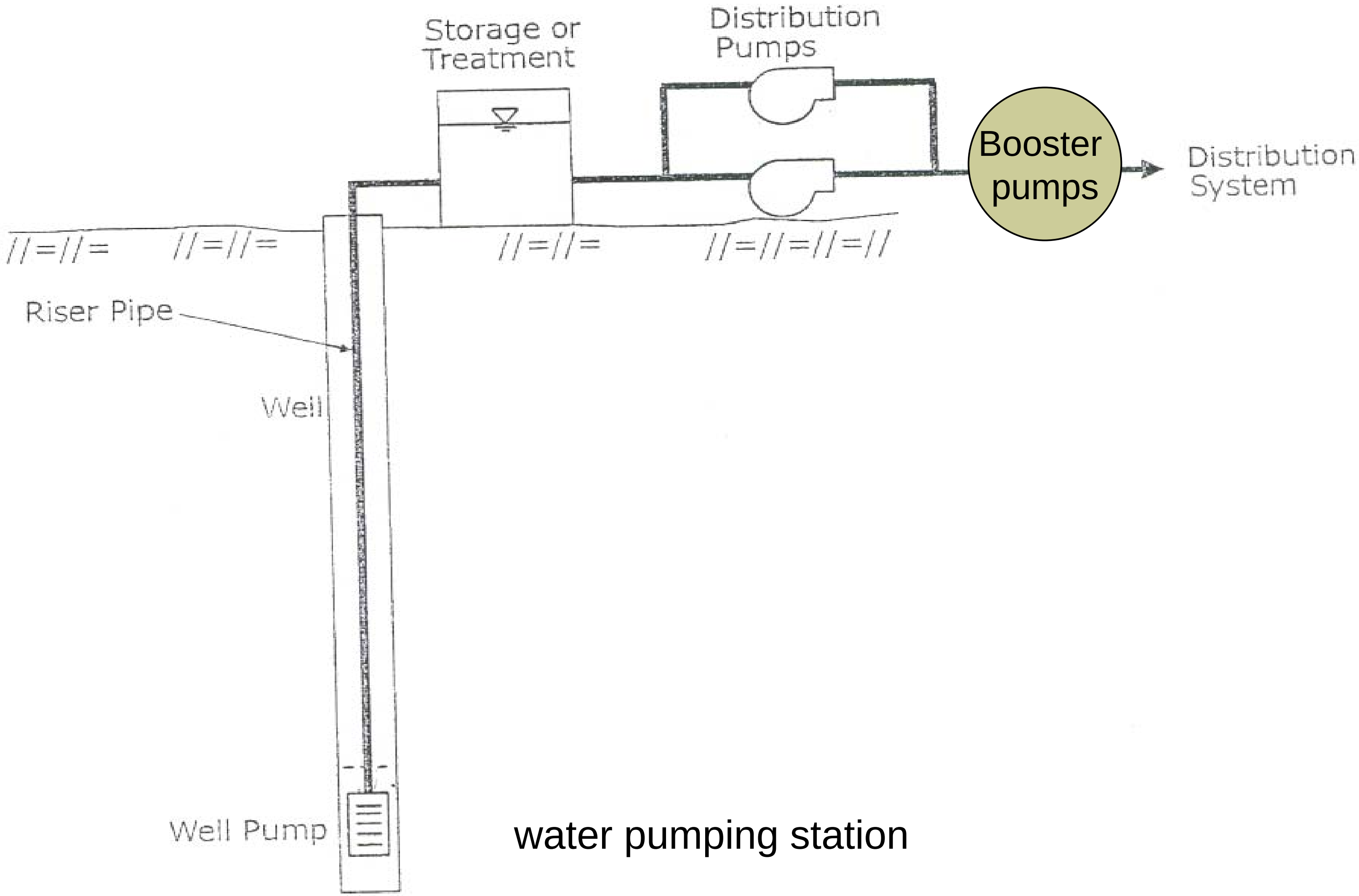

water pumping station

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water pumping station

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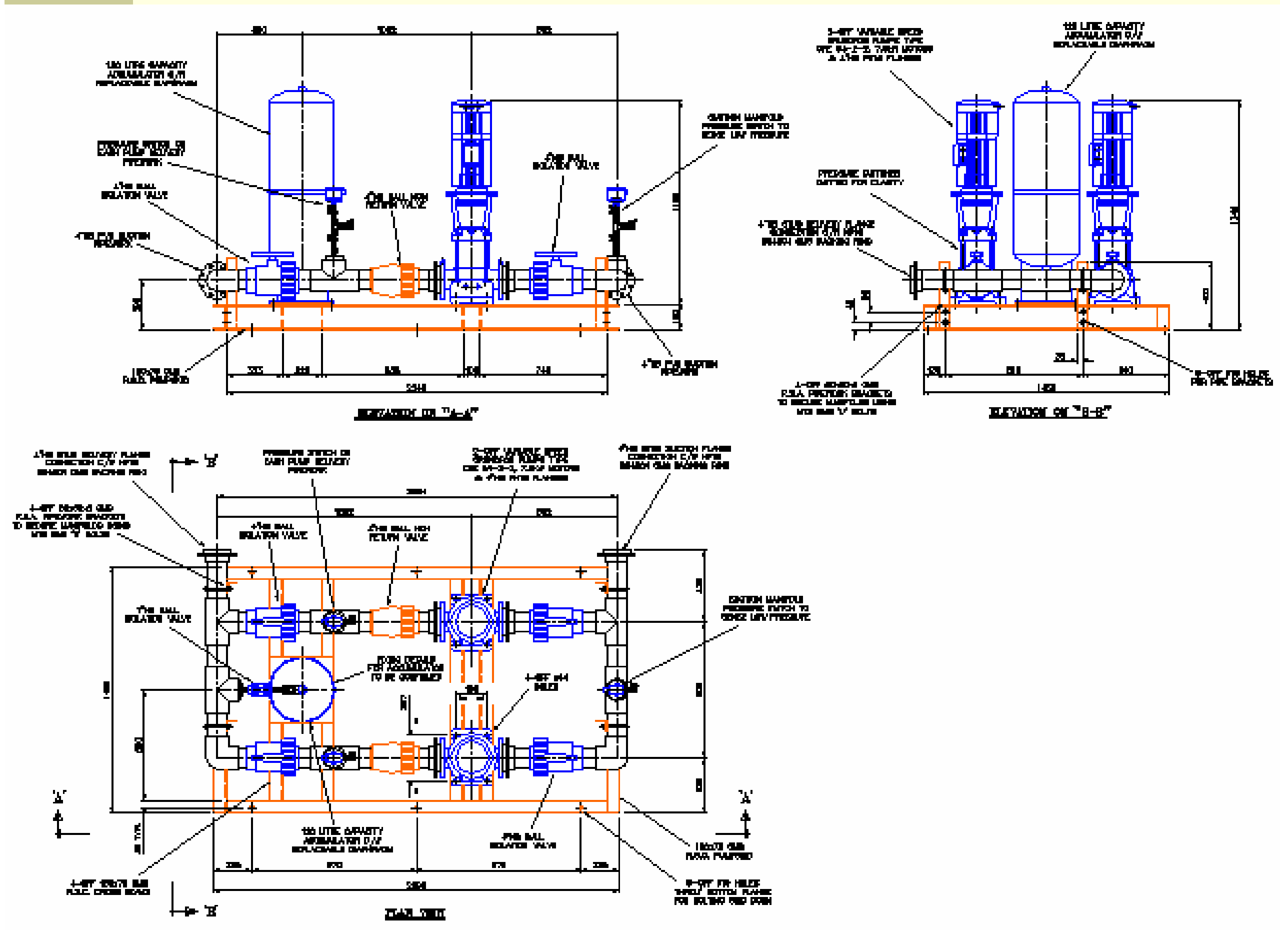

Booster pumping station

Inline Booster Pump

1.2 Types of pumps

Pumps are classified into two main categories:

- Kinetic pumps.
  - **Centrifugal pumps (radial, axial, mixed flow)**
  - **Turbine (vertical) pumps**
- Positive displacement pumps.
  - **Rotary pumps (screw, lobe)**
  - **Reciprocating pumps (plunger)**

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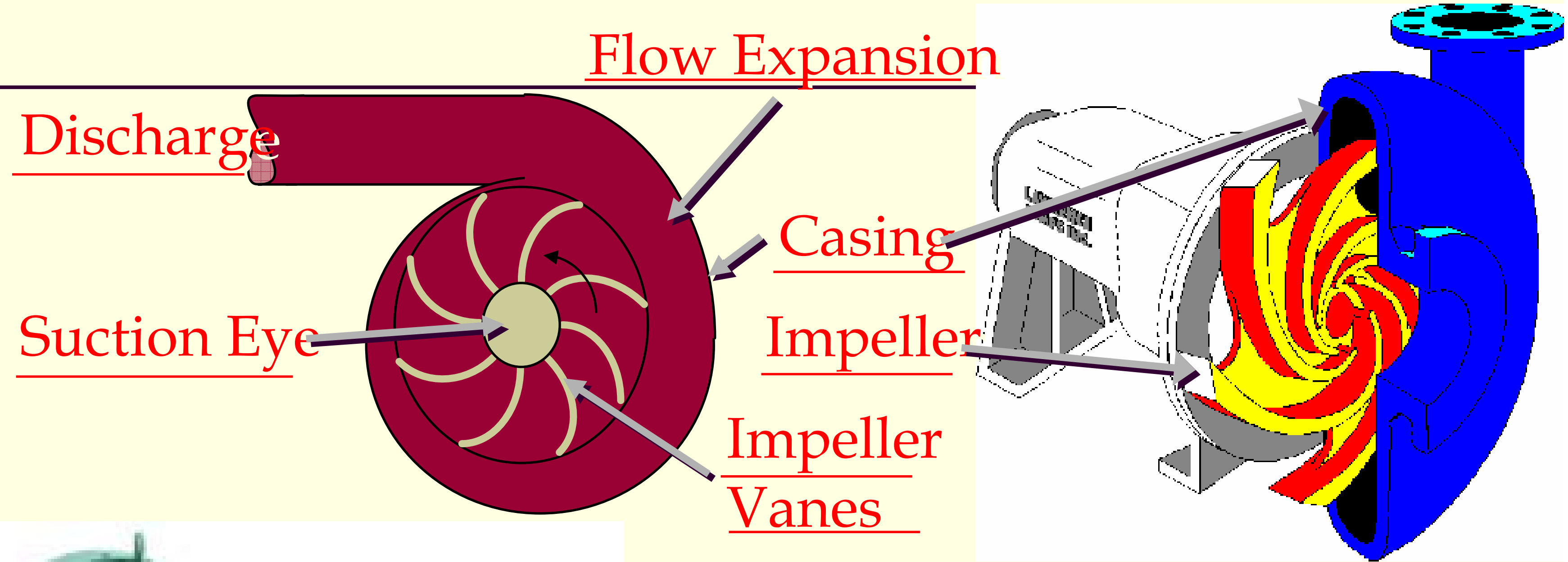

The common feature of all centrifugal pumps is the Volute

Kinetic Pumps: Centrifugal Pump

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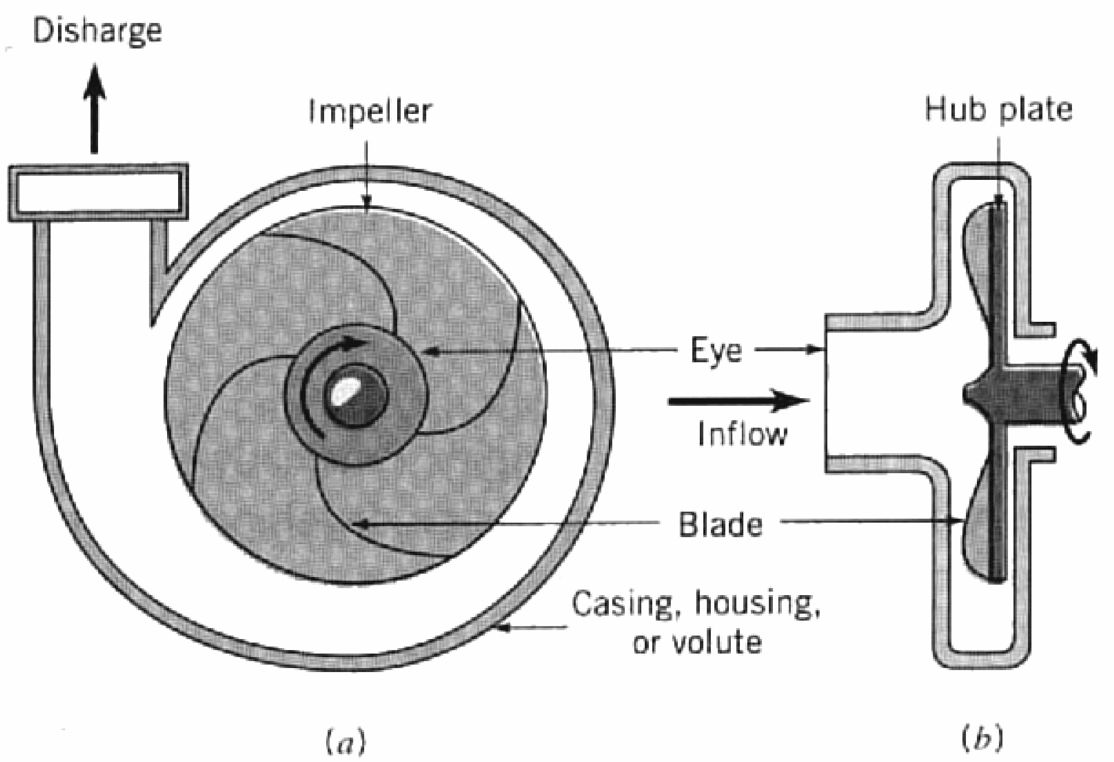

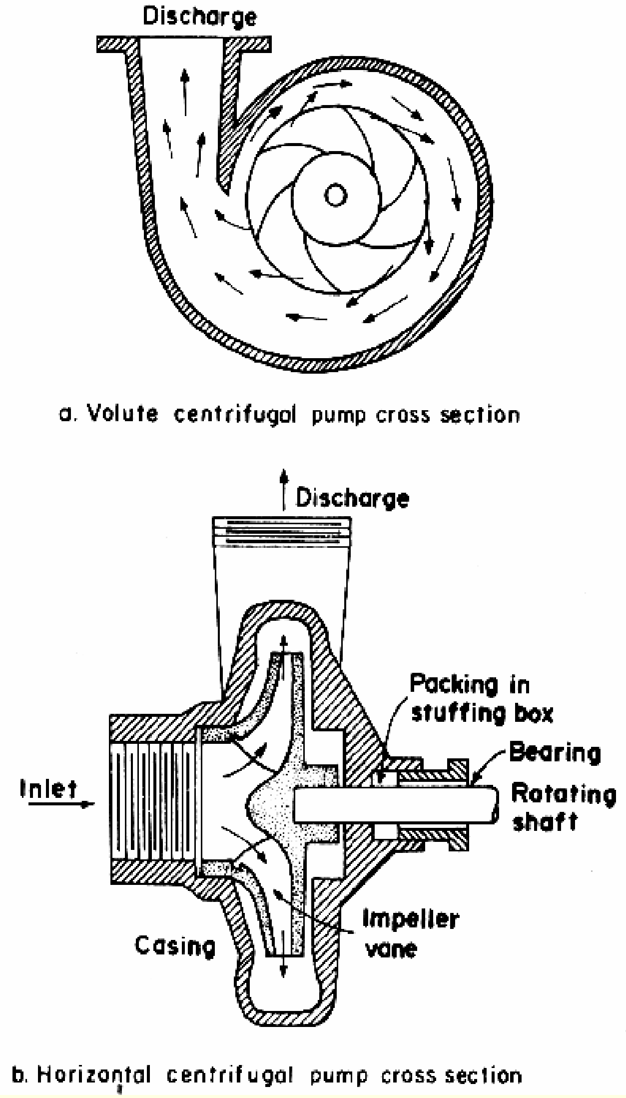

Kinetic Pumps: Centrifugal Pump

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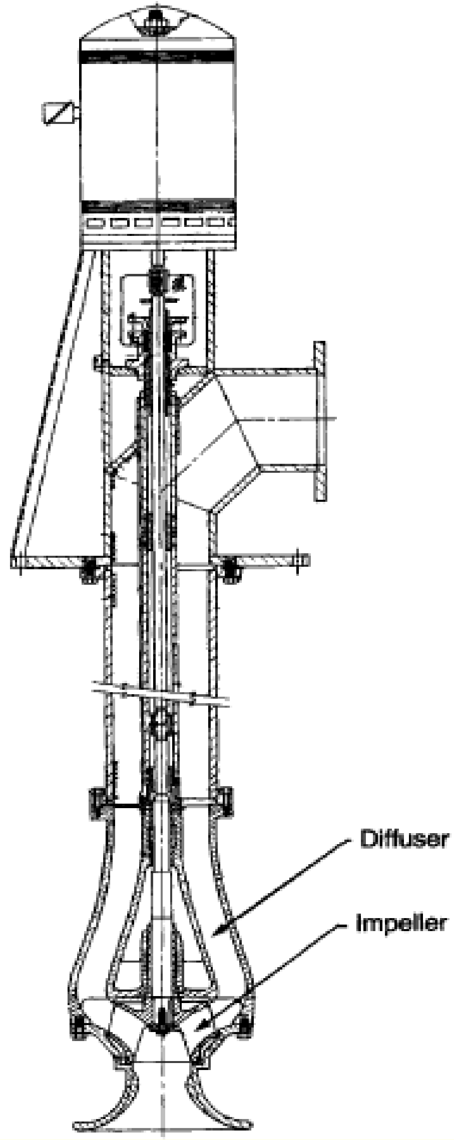

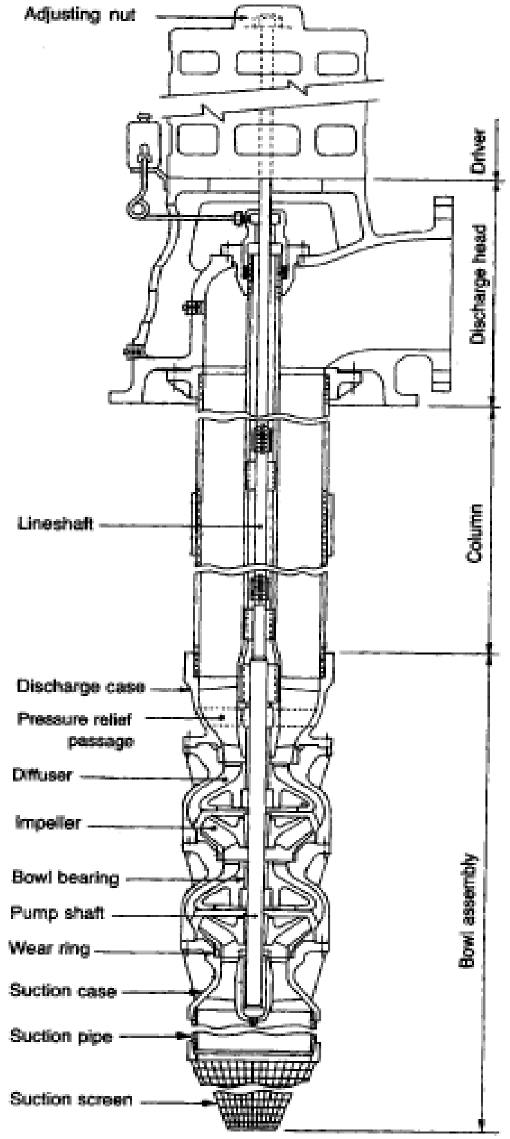

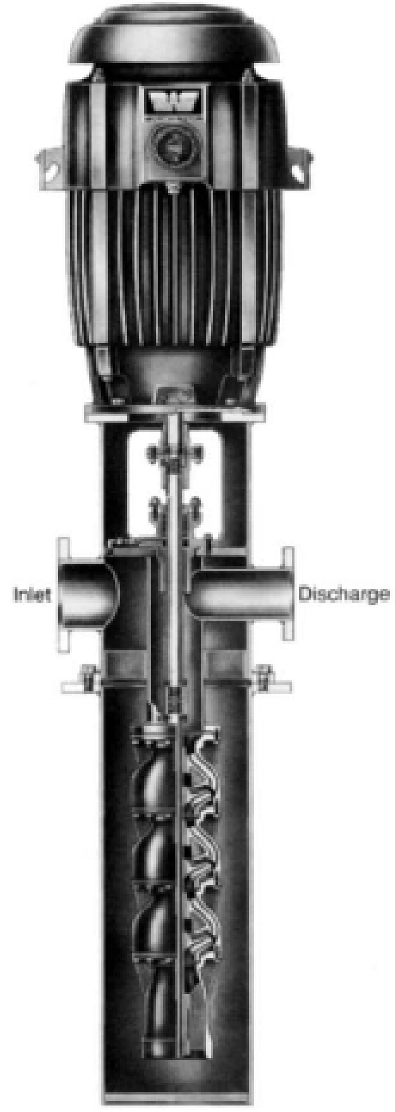

The vertical pumps are equipped by a diffuser instead the Volute

Kinetic pumps: Vertical Turbine Pump

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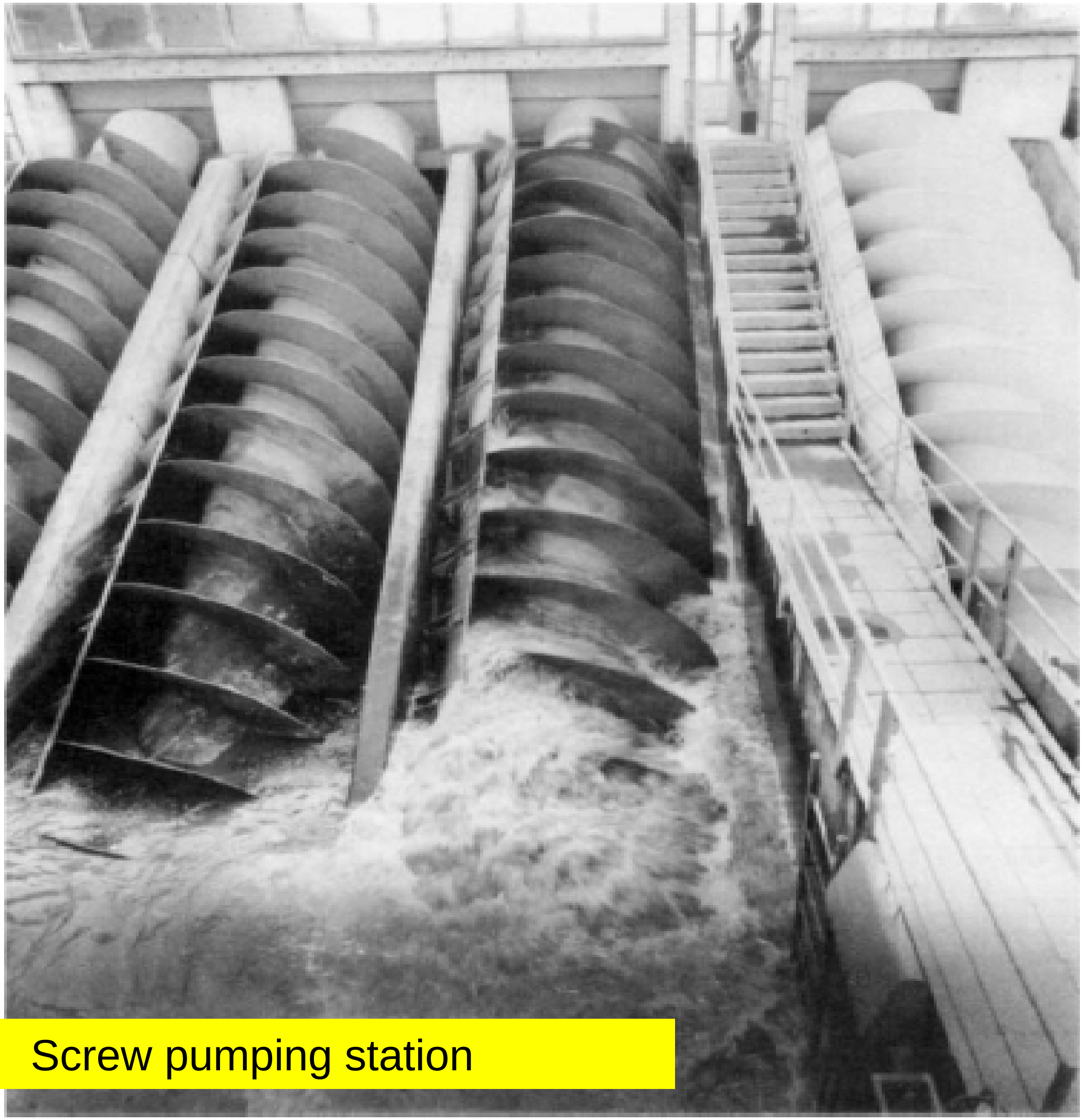

Rotary Pump: Screw pumping station

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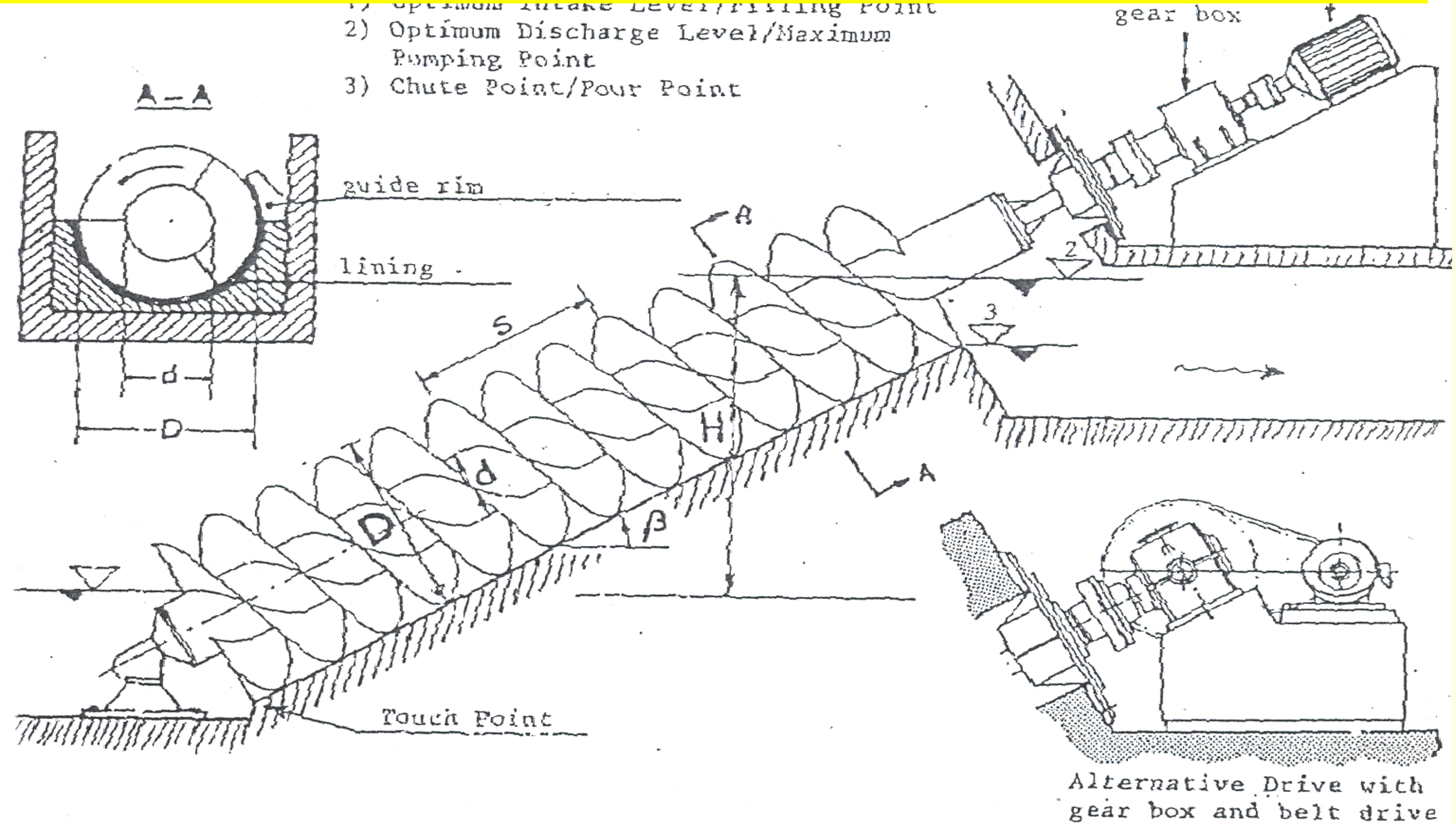
In the screw pump a revolving shaft fitted with blades rotates in an inclined trough and pushes the water up the trough.



Rotary Pump: Screw pumping station

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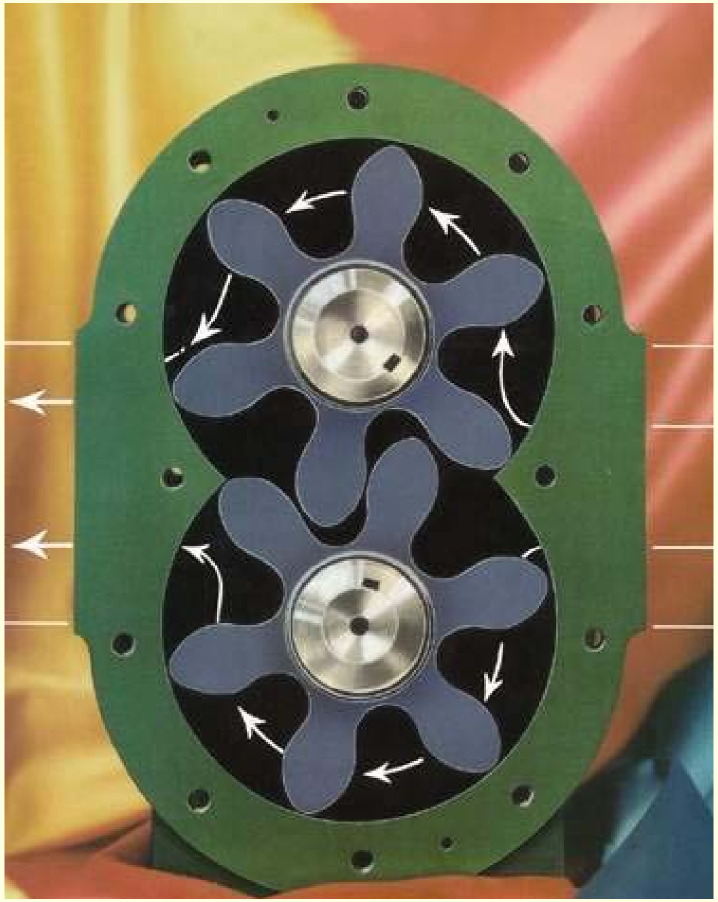
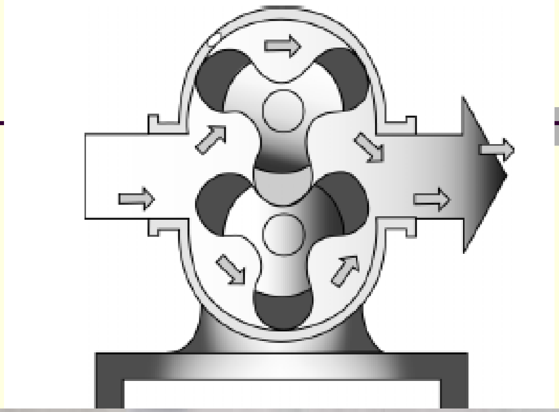
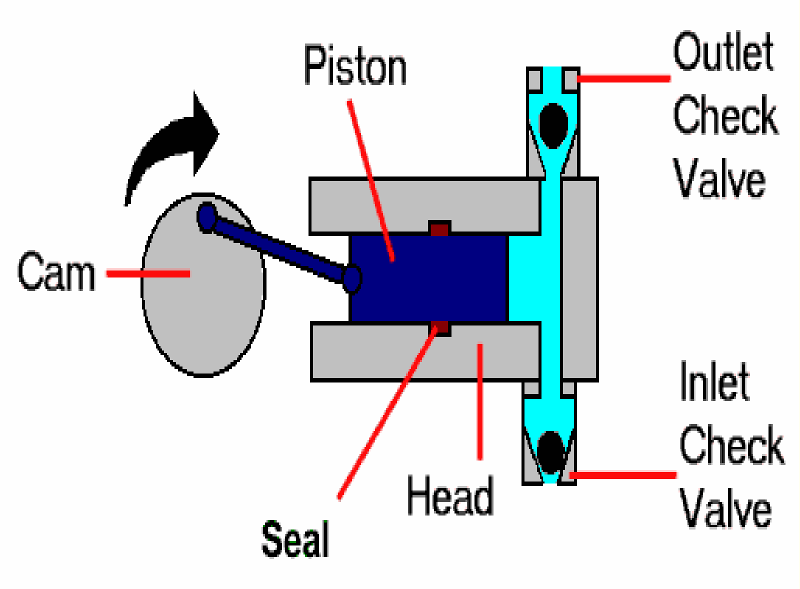
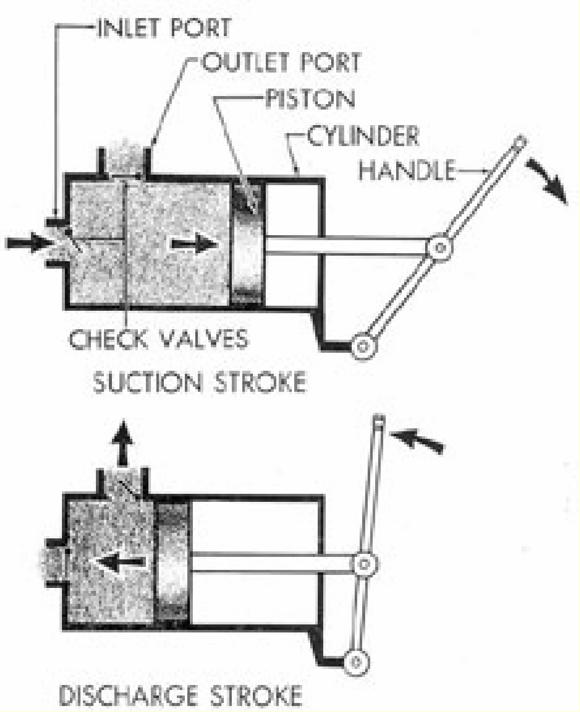




Rotary Pump: Lobe

- In the reciprocating pump a piston sucks the fluid into a cylinder then pushes it up causing the water to rise.





Reciprocating Pump: Screw pumping station

Pumping Stations Design

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Lecture 2: Pumping Hydraulics

Dr. Fahid Rabah
Water and environment Engineering
frabah@iugaza.edu

Lecture 2: Pumping Hydraulics

The main items that will be studied under the pumping hydraulics title are:

- ❑ Pumping pressure and head terminology
- ❑ Cavitation
- ❑ Pump characteristic curves
- ❑ Multiple pump operation
- ❑ Variable speed pumps
- ❑ Affinity laws
- ❑ Pump selection

2.1 pumping pressure and head terminology

- ❑ Understanding pumping pressure and head terminology is essential in pumping stations design.
- ❑ The pumping system consist mainly from :
  - The pump
  - A wet well containing the liquid to be pumped
  - A suction pipe connecting the wet well and the pump.
  - A delivery pipe
  - A delivery tank or a delivery point
  - Valves and fittings installed on the suction and delivery pipes
- ❑ There are two cases for the pumping system that depend on the location of the pump in reference to the wet well:
  - Case 1: the pump centerline is lower than the level of the water surface in the wet well.
  - Case 2 : the pump centerline is higher than the level of the water surface in the wet well.

The following is an explanation for pumping terminology for both cases.

Lecture 2: Pumping Hydraulics

2.1 pumping pressure and head terminology

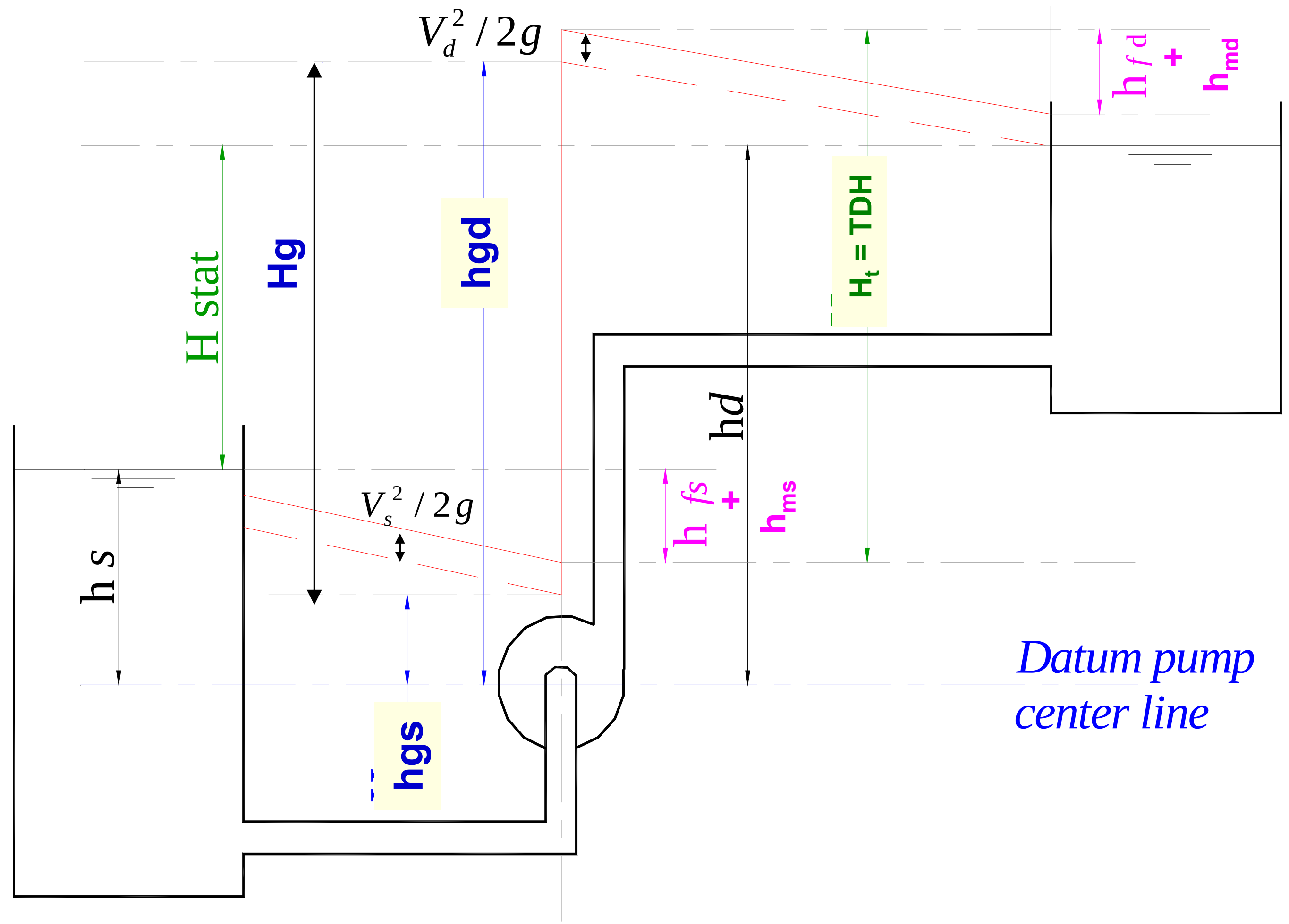


Figure 2.1 Case 1: Terminology for a pump with a positive suction head

2.1 pumping pressure and head terminology

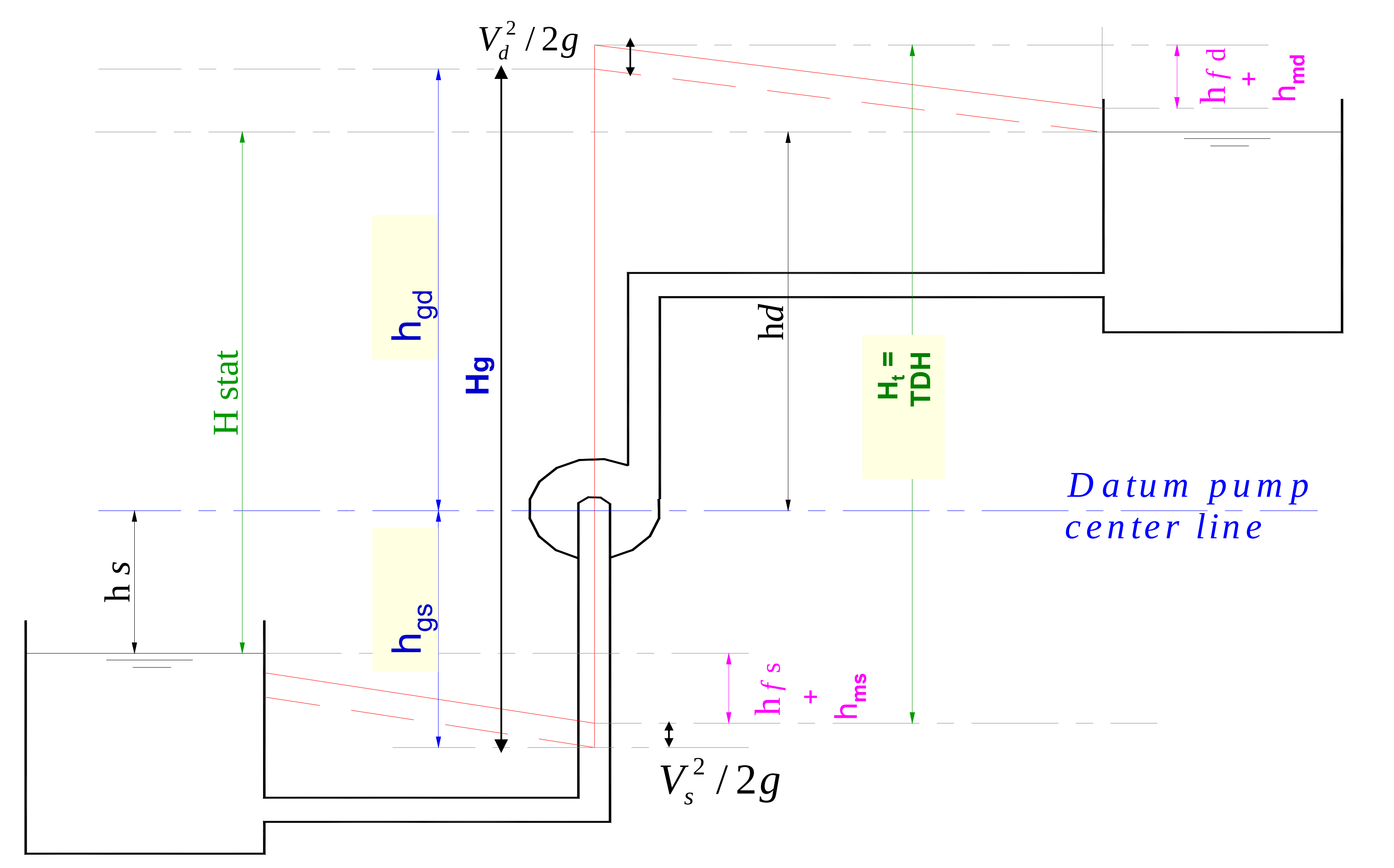


Figure 2.2 Case 2: Terminology for a pump with a negative suction head

2.1 pumping pressure and head terminology

Definition of pumping head terms:

The terms that appear in figures 2.1 and 2.2 can be defined as the following:

- **h_s** (*Static suction head):* It is the difference in elevation between the wet well liquid level and the datum elevation passing through the pump impeller center.

- **h_d** (*Static discharge head*): It is the difference in elevation between the discharge liquid level and the datum elevation passing through the pump impeller center.

- **H_{stat}** (*Static discharge head)*: It is the difference (or sum) in elevation between the static discharge and the static suction heads:

$$H_{stat} = h_d \pm h_s$$

Note: it is (-) when the pump center is below the wet well liquid level.
and (+) when the pump center is above the wet well liquid level

2.1 pumping pressure and head terminology

Definition of pumping head terms…. Continued:

- **h_{gs}** (*manometric (or gage) suction head):* It is the suction gage reading. it is also the height to which the water will rise in a manometer installed at the suction side of the pump.

- **h_{gd}** (*manometric (or gage) discharge head):* It is the discharge gage reading. It is also the height to which the water will rise in a manometer installed at the discharge side of the pump.

- **H_g** (*manometric (or gage) discharge head):* It is the increase in pressure head produced by the pump. It is calculated as follows:

$$\boxed{H_g = h_{gd} \pm h_{gs}}$$

Note: it is (-) when the pump center is below the wet well liquid level. and (+) when the pump center is above the wet well liquid level

2.1 pumping pressure and head terminology

Definition of pumping head terms…. Continued:

- $V_d^2/2g$ (*velocity head at the delivery side):* It is the kinetic energy in the liquid in the discharge pipe.

- $V_s^2/2g$ (*velocity head at the suction side):* It is the kinetic energy in the liquid in the suction pipe.

- h_{ms} (*minor losses at the suction side):* They are the losses in head (pressure) due to the eddies and turbulence created during liquid flow through valves and fittings in the suction pipe.

- h_{md}(*minor losses at the delivery side):* They are the losses in head (pressure) due to the eddies and turbulence created during liquid flow through valves and fittings in the delivery pipe.

2.1 pumping pressure and head terminology

Definition of pumping head terms…. Continued:

- **H_t** (*total dynamic head TDH):* It is the total head that the pump should deliver.

It can be determined using the following formula:

$$H_t = h_{gd} + \frac{V_d^2}{2g} - \left(h_{gs} + \frac{V_s^2}{2g}\right)$$ for **Case 1** ……….(1)

$$H_t = h_{gd} + \frac{V_d^2}{2g} + \left(h_{gs} - \frac{V_s^2}{2g}\right)$$ for **Case 2** ……….(2)

2.1 pumping pressure and head terminology

Definition of pumping head terms…. Continued:

- H_t (TDH) can also be calculated using Equations 3 or 4 :

$$H_t = H_{stat} + h_{fd} + \sum h_{md} + h_{fs} + \sum h_{ms} + \frac{V_d^2}{2g} \quad \text{..........(3)}$$

$$H_t = H_{stat} + H_{dynamic} \quad \text{..........(4)}$$

where

$$H_{dynamic} = h_{fd} + \sum h_{md} + h_{fs} + \sum h_{ms} + \frac{V_d^2}{2g}$$

These equation are valid for case 1 and 2 .

2.1 pumping pressure and head terminology

Definition of pumping head terms…. Continued:

- Friction losses (h_{fd} and h_{fs}) are calculated using Hazen Williams equation:

$$h_f = \frac{10.7}{D^{-4.87}}\left(\frac{Q}{C}\right)^{1.85}$$

Where:

h_f = friction losses due to friction, m
D = pipe diameter, m
Q = flow rate, m3/s
C = Hazen Williams friction coefficient
Values of " C" are given in hydraulic references. For example:
C = 150 for PVC pipes
= 145 for Steel pipes
Note: C for new pipes is larger than C for old pipes from the same material. And C is inversely proportional to h_f .

2.1 pumping pressure and head terminology

Definition of pumping head terms…. Continued:

- Minor losses (h_{md} and hm_{fs}) are calculated using the following equation:

$$\boxed{h_m = K \frac{V^2}{2g}}$$

Where:

h_m = minor losses, m
g = gravitational acceleration, m/s^2
V = liquid velocity, m/s
K = minor losses coefficient
Values of " K" are given in hydraulic references. For example:
K = 0.1 – 0.40 gate valve
= 0.30 for 45^o bend

2.1 pumping pressure and head terminology

Definition of pumping head terms…. Continued:

- **Power output of pumps :**

Power output of a pump is the useful energy delivered by the pump to the fluid. The power is calculated using the following formula:

$$P_p = \rho g Q H_t$$

Where:

P_p = Power output of the pump, Watt
H_t = the total dynamic head delivered by the pump, m
g = gravitational acceleration, m/s^2
Q = flow rate, m^3/s
ρ = fluid density, kg/m^3

2.1 pumping pressure and head terminology

Definition of pumping head terms…. Continued:

❑ Pump Efficiency :

Pump efficiency is defined as the ratio between power output of the pump to the power input supplied to the pump shaft:

$$\eta_p = \frac{P_p}{P_i} = \frac{\rho g Q H_t}{P_i}$$

Or

$$P_i = \frac{\rho g Q H_t}{\eta_p}$$

Where:

η_p **=** pump efficiency
P_p **=** pump output power, Watt
P_i **=** input power to the pump, Watt

P_i is the power supplied by the pump motor to the pump shaft and usually called the **brake power** (**bp**)

2.1 pumping pressure and head terminology

Definition of pumping head terms…. Continued:

❑ Pump motor Efficiency :

Pump motor efficiency is defined as the ratio between power output of the pump motor to the power input supplied to the pump impeller:

$$\eta_m = \frac{P_i}{P_m} \qquad \text{Or} \qquad P_m = \frac{P_i}{\eta_m}$$

Where:

η_m = pump motor efficiency
P_m = input power to the pump motor, Watt

P_m is the power supplied to the pump motor from an electricity source.

2.1 pumping pressure and head terminology

Definition of pumping head terms…. Continued:

- Overall System Efficiency :

Overall system efficiency is defined as the ratio between power output of the entire system to the power input supplied by the electricity source:

$$\eta_o = \frac{P_P}{P_m}$$

Or

$$\eta_O = \eta_m \eta_P$$

Where:

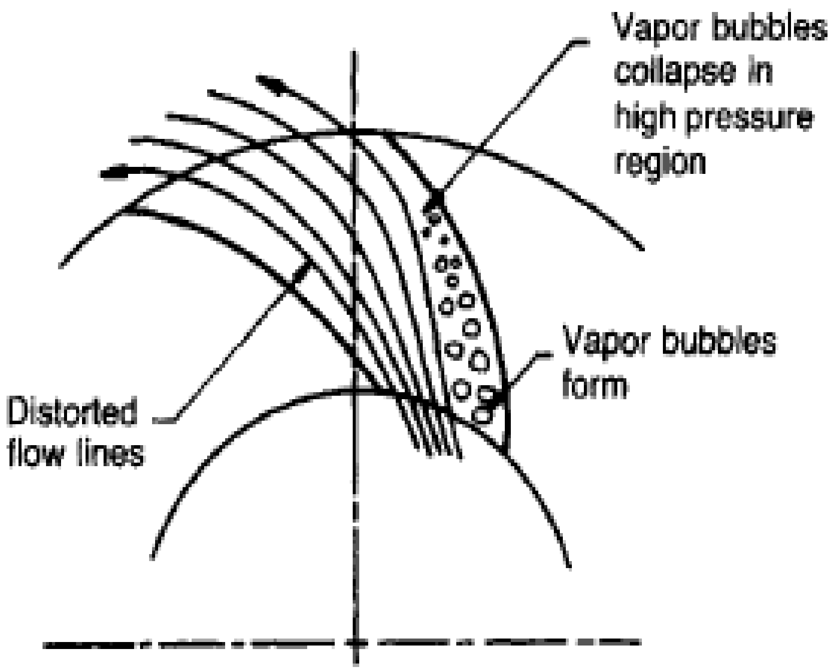
η_O = Overall system efficiency

2.2 Cavitation in pumping systems

- **Cavitation is the formation and collapse of vapor cavities within the pump.**
- **When Cavitation occurs in pumps it has the potential to cause:**
  1. **Performance degradation : loss of head, capacity and efficiency.**
  2. **Permanent damage due to erosion and mechanical failure of pump component and structures.**
  3. **Violet vibration and noise.**

- **Cavitation occurs when the absolute pressure at the inlet of the pump falls below the vapor pressure of the water (P_v). Under this condition, vapor bubbles form (water starts to boil) at the impeller inlet and when these bubbles are carried into a zone of higher pressure, they collapse abruptly and hit the vanes of the impeller.**

2.2 Cavitation in pumping systems



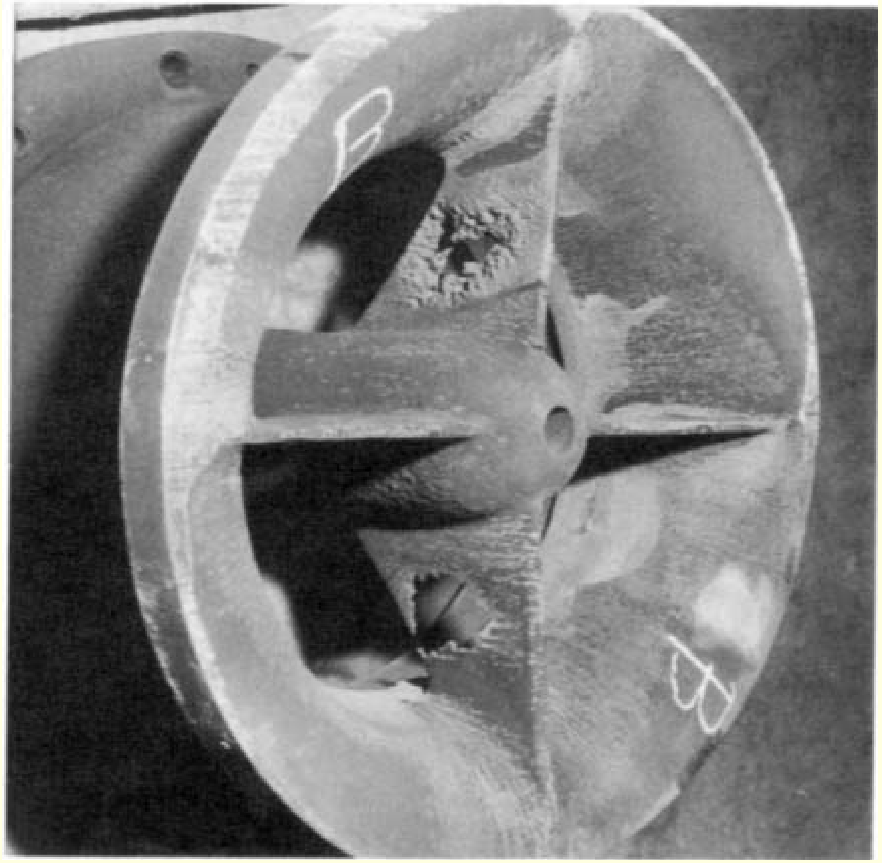


Figure 2.3 **Effect of Cavitation on pumps**

2.2 Cavitation in pumping systems

- **To avoid <u>Cavitation</u> the total pressure available at the pump inlet ($P_{inlet}/\rho g$) should be greater than the water vapor pressure ($PV/\rho g$) .**
In other words the difference between ($P_{inlet}/\rho g$) and ($P_{Vapor}/\rho g$) should be positive. This net head is called the net positive suction head (NPSH).

- **Knowing that the total pressure at the pump inlet is:**

$$\frac{P_{inlet}}{\rho g} = \frac{P_s}{\rho g} + \frac{V_s^2}{2g}$$

Then the NPSH mathematically is:

$$NPSH = \frac{P_s}{\rho g} + \frac{V_s^2}{2g} - \frac{P_{vapor}}{\rho g}$$

2.2 Cavitation in pumping systems

- **To avoid Cavitation the following condition should be maintained :**

$$(NPSH)_A > (NPSH)_R$$

- **$(NPSH)_A$ in the pumping systems shown in case 1 and case2 can be calculated using the following formula:**

$$(NPSH)_A = \mp h_s - h_{fs} - \sum h_{ms} + \frac{P_{atm}}{\rho g} - \frac{P_{vapor}}{\rho g}$$

Notes:

- **(+) is used if hs is above the pump centerline (datum).**
- **Usually a factor of safety of 1.5 m on (NPSH)A is used by subtracting 1.5m from the calculated value.**
- **Values of P_{atm} and P_{vapor} are given in Tables A-6 and A-8**

2.2 Cavitation in pumping systems

Table A-6. Atmospheric Pressure (SI Units)

Elevation above sea level (m)	Atmospheric pressure[a] kPa	Water column, mm	Mercury column, mm	Specific weight, γ, of air at 20°C, kN/m^3 [b]
0	101.3	10.33	760	1.18 E−2
500	95.6	9.74	717	1.11 E−2
1000	90.1	9.19	676	1.05 E−2
1500	84.8	8.64	636	9.87 E−3
2000	79.8	8.13	598	9.29 E−3
2500	73.3	7.47	550	8.53 E−3
3000	70.3	7.17	527	8.19 E−3
3500	66.1	6.74	496	7.70 E−3

[a] Storms commonly reduce atmospheric pressure by about 1.7%.

[b] At other temperatures and pressures, use $p_1 v_1/K_1 = p_2 v_2/K_2$ where p is pressure, v is volume, and K is degrees Kelvin (°C + 273).

2.2 Cavitation in pumping systems

Table A-8. Physical Properties of Water (SI Units)[a]

Temperature, °C	Specific weight[b], γ kN/m^3	Density, ρ kg/m^3	Bulk modulus of elasticity[c], K kPa	Dynamic viscosity, μPa · s	Kinematic viscosity, νm^2/s	Surface tension[d], σN/m	Vapor pressure, p_v kPa	Vapor pressure, p_v kPa *WC* m
0	9.805	9.998 E+2	1.98 E+6	1.78 E−3	1.79 E−6	0.0765	0.61	0.06
5	9.807	1.000 E+3	2.05 E+6	1.52 E−3	1.52 E−6	0.0749	0.87	0.09
10	9.804	9.997 E+2	2.10 E+6	1.31 E−3	1.31 E−6	0.0742	1.23	0.13
15	9.798	9.991 E+2	2.15 E+6	1.14 E−3	1.14 E+6	0.0735	1.70	0.17
20	9.789	9.982 E+2	2.17 E+6	1.00 E−3	1.00 E−6	0.0726	2.34	0.24
25	9.777	9.970 E+2	2.22 E+6	8.90 E−4	8.93 E−7	0.0720	3.17	0.32
30	9.764	9.957 E+2	2.25 E+6	7.98 E−4	8.00 E−7	0.0712	4.24	0.43
40	9.730	9.922 E+2	2.28 E+6	6.53 E−4	6.58 E−7	0.0696	7.38	0.76
50	9.689	9.880 E+2	2.29 E+6	5.47 E−4	5.53 E−7	0.0679	12.33	1.27
60	9.642	9.832 E+2	2.28 E+6	4.66 E−4	4.74 E−7	0.0662	19.92	2.07
70	9.589	9.778 E+2	2.25 E+6	4.04 E−4	4.13 E−7	0.0644	31.19	3.25
80	9.530	9.718 E+2	2.20 E+6	3.54 E−4	3.64 E−7	0.0626	47.34	4.97
90	9.466	9.653 E+2	2.14 E+6	3.15 E−4	3.26 E−7	0.0608	70.10	7.41
100	9.399	9.584 E+2	2.07 E+6	2.82 E−4	2.94 E−7	0.0589	101.33	10.78

[a] Adapted from Vennard and Street [1].

[b] Specific weight, γ, is force per unit volume. The relationship between γ, ρ, and the acceleration due to gravity, g, is $\gamma = \rho g$. See Table A-3 for the value of g.

[c] At atmospheric pressure.

[d] In contact with air. Note 1000 N/m = dynes/cm.

2.2 Cavitation in pumping systems

❑Cavitation constant:

The cavitation constant is defined as the ratio of $NPSH_i$ to the to H_t :

$$\sigma = \frac{(NPSH)_R}{H_t}$$

This constant is called Thoma's cavitation constant

Where:

σ = cavitation constant

$NPSH_R$ = required net positive suction head, m

H_t = total dynamic head, m

2.2 Cavitation in pumping systems

❑Cavitation constant …. continued:

The cavitation constant can also be calculated by the following formula :

$$\sigma = \frac{K \times n_s^{4/3}}{10^6}$$

Where:

K = a constant that depends on pump efficiency.

n_s = the pump specific speed

Typical values of K are: 1726, 1210, and 796, for pumps efficiency of 70%, 80% and 90%, respectively.

The pump specific speed can be calculated by the following formula:

$$n_s = n\, Q^{\frac{1}{2}} H_t^{\frac{-3}{4}}$$

2.2 Cavitation in pumping systems

❑Cavitation constant …. continued:

Where:

n = pump speed, revolution/s.
Q = pump flow rate, m^3/s

study examples 10.5 and 10.6 in the Pumping Station design Reference book.

❑ Prevention and control of cavitation:

- **To prevent cavitation keep the internal pressure of the pump above vapor pressure**
- **To solve an existing cavitation problem you may:**
  1. **Decrease the suction lift (hs)**
  2. **Decrease the suction losses (hs_f , hs_m)**
  3. **Reduce the impeller speed**
  4. **Change the pump or impeller**
  5. **Add a booster pump on the suction side**

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Lecture 3: Pumping Hydraulics

Dr. Fahid Rabah
Water and environment Engineering
frabah@iugaza.edu

Lecture 3: Pumping Hydraulics

The main items that will be studied under the pumping hydraulics title are:

- ❑ Pumping pressure and head terminology
- ❑ Cavitation
- ❑ Pump characteristic curves
- ❑ Multiple pump operation
- ❑ Variable speed pumps
- ❑ Affinity laws
- ❑ Pump selection

3.1 Piping System Curves

- Piping System is composed of :
  1. Suction piping and fittings
  2. Discharge piping
  3. Manifold
  4. Force main or rising main

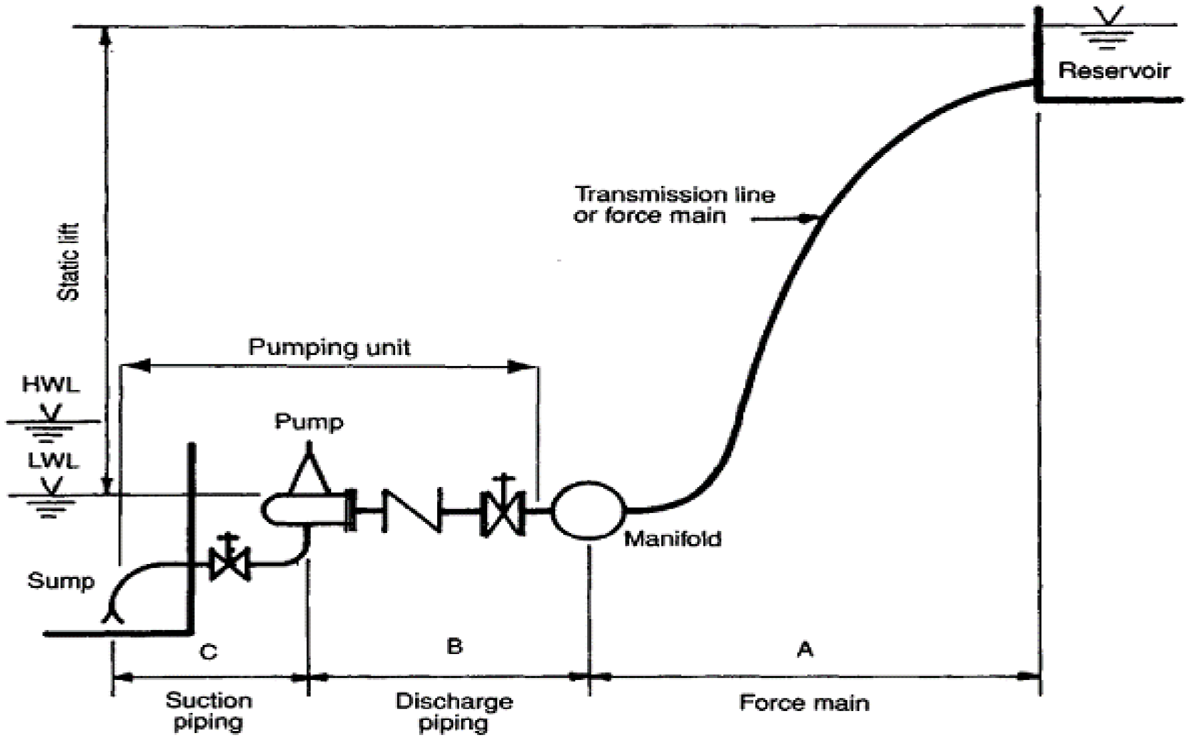


Figure 3.1:
Typical piping system curve

3.1 Piping System Curves

- Piping System curve:

  It is curve that relates the total dynamic head (H_t) with the discharge flow rate (Q) passing through the piping system. The equation relating H_t and Q is:

$$H_t = H_{stat} + H_{dynamic}$$

Notice that $H_{dynamic}$ is a function of Q in this equation.

$$H_{dynamic} = KQ^n$$

The system curve is drawn by assuming several values of Q and calculating the corresponding $H_{dynamic}$ values and H_t values.

3.1 Piping System Curves

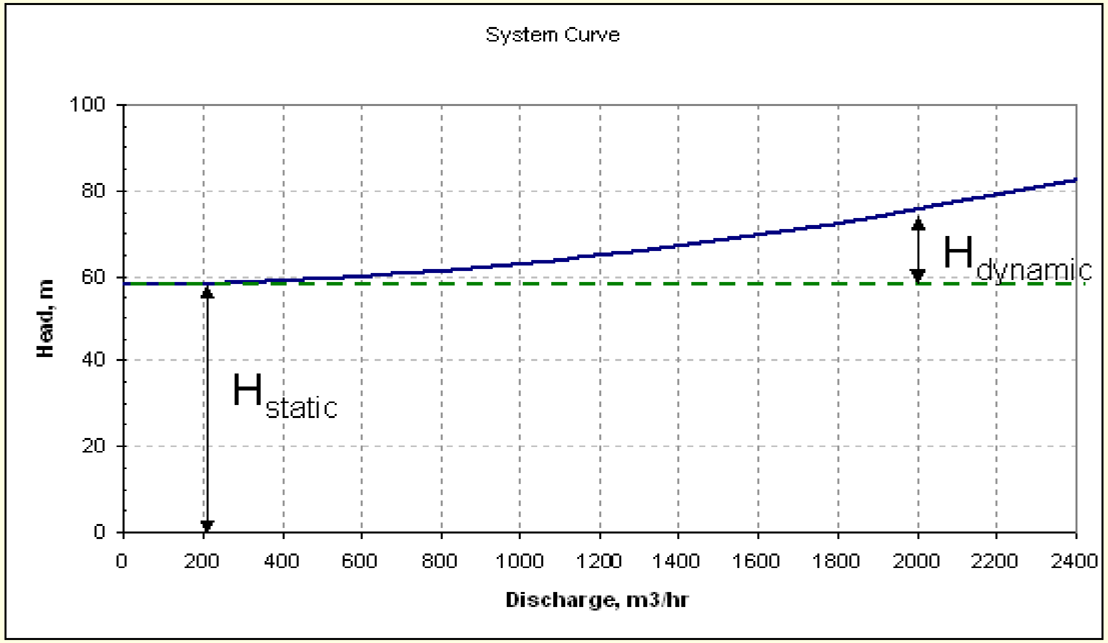


Figure 3.2: Typical piping system curve

Lecture 3: Pumping Hydraulics

3.1 Piping System Curves

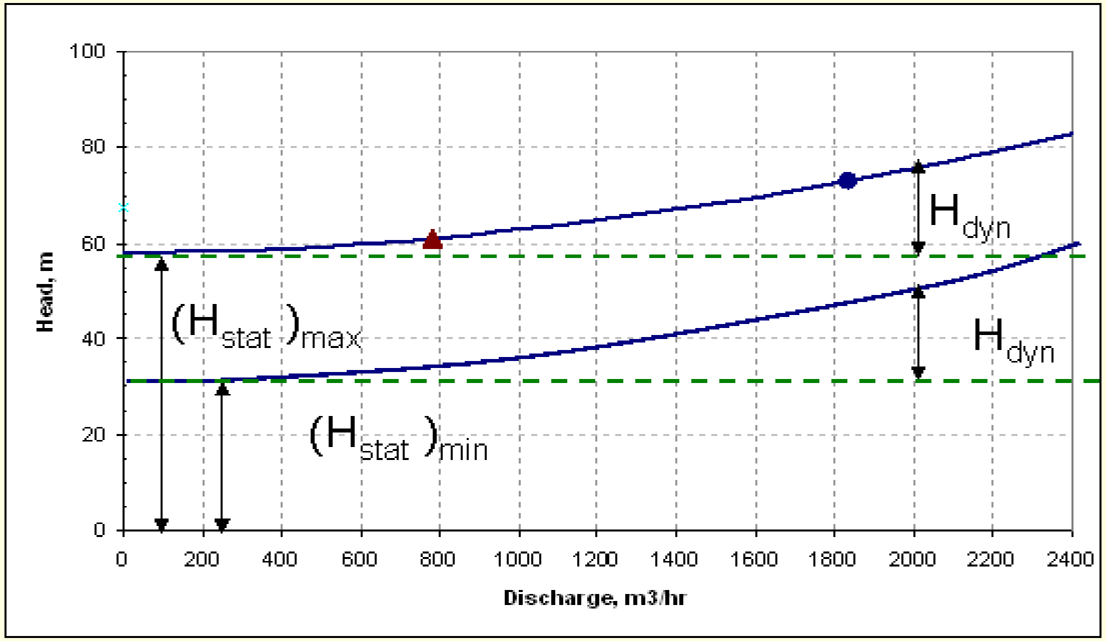


Figure3.3: Change of system curve during operation due to the change in static head

Note: the system curve starts at $(H_{stat})_{min}$ then with time it moves up during continuous pumping to $(H_{stat})_{max}$.

3.1 Piping System Curves

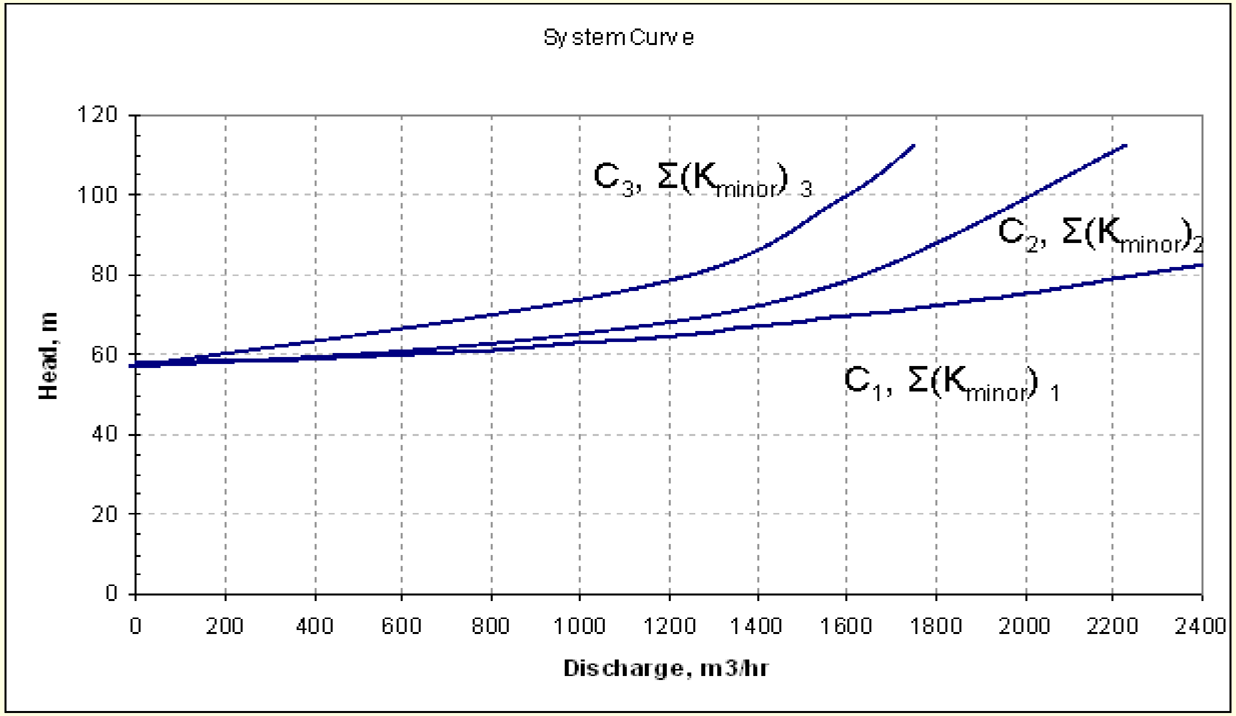


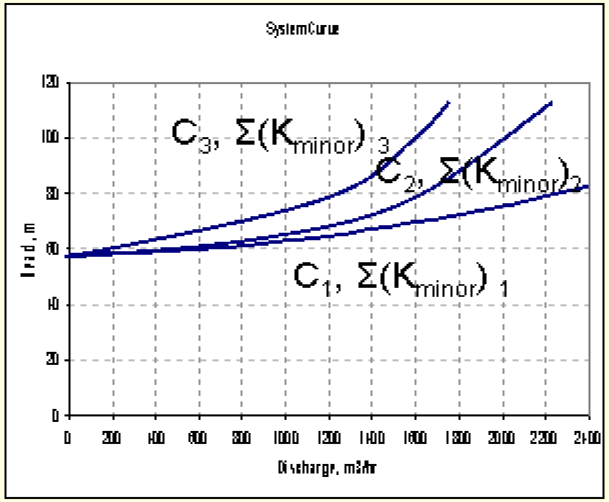
Figure 3.4: Change of system curve during operation due to the change in head losses

Lecture 3: Pumping Hydraulics

3.1 Piping System Curves

- The change in C (the coefficient of friction) of the system occurs in many cases such as:
  1. When the pipe becomes old the friction in the system increases (i.e $C_{old} < C_{new}$)
  2. When some of the flow is diverted from a pipe branch to another one.

- The change in ΣK_{minor} (the minor losses coefficient) of the system occurs in many cases such as:
  1. Opening or closing the valves installed on the system.
  2. When some of the flow is diverted from a pipe branch to another one.



Change of system curve during operation due to the change in head losses

3.2 Pump Characteristic Curves

- Pump characteristic curve :

  It is the curve that relates the head delivered by the pump (H_t) at several discharge flow rates (Q). This curve is given by the manufacturer of the pump

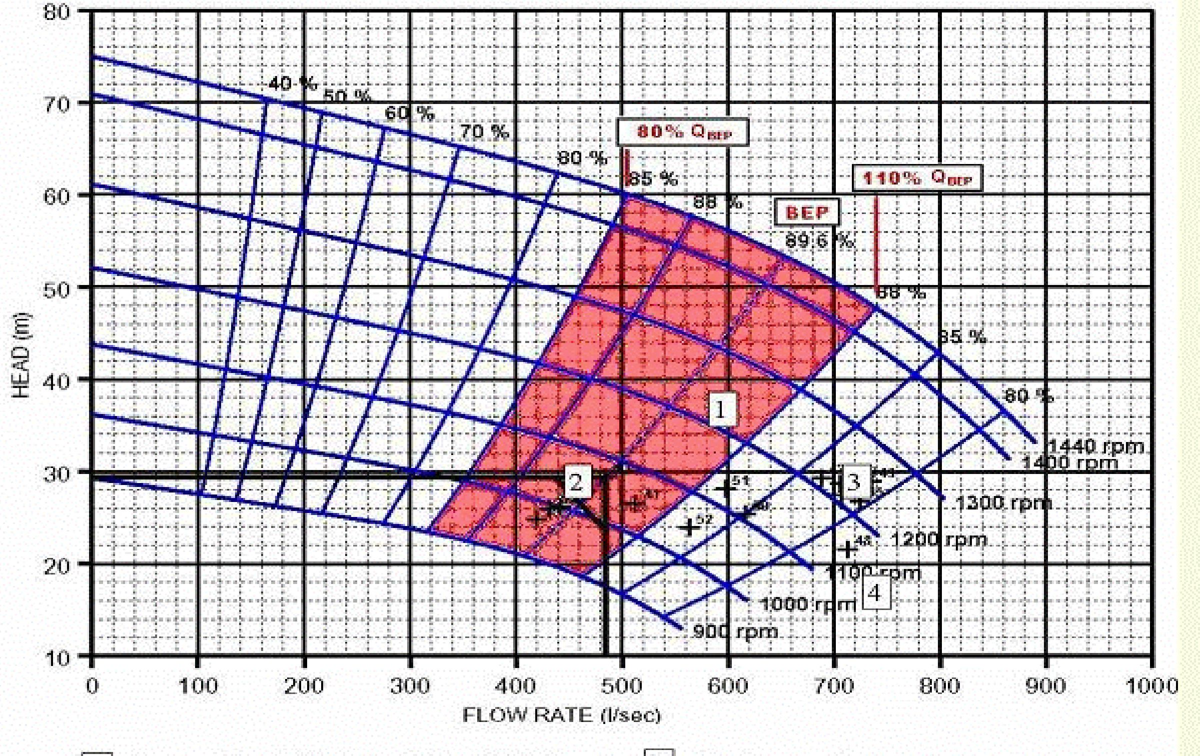


Figure 3.5: Typical pump characteristic curves curve

3.2 Pump Characteristic Curves

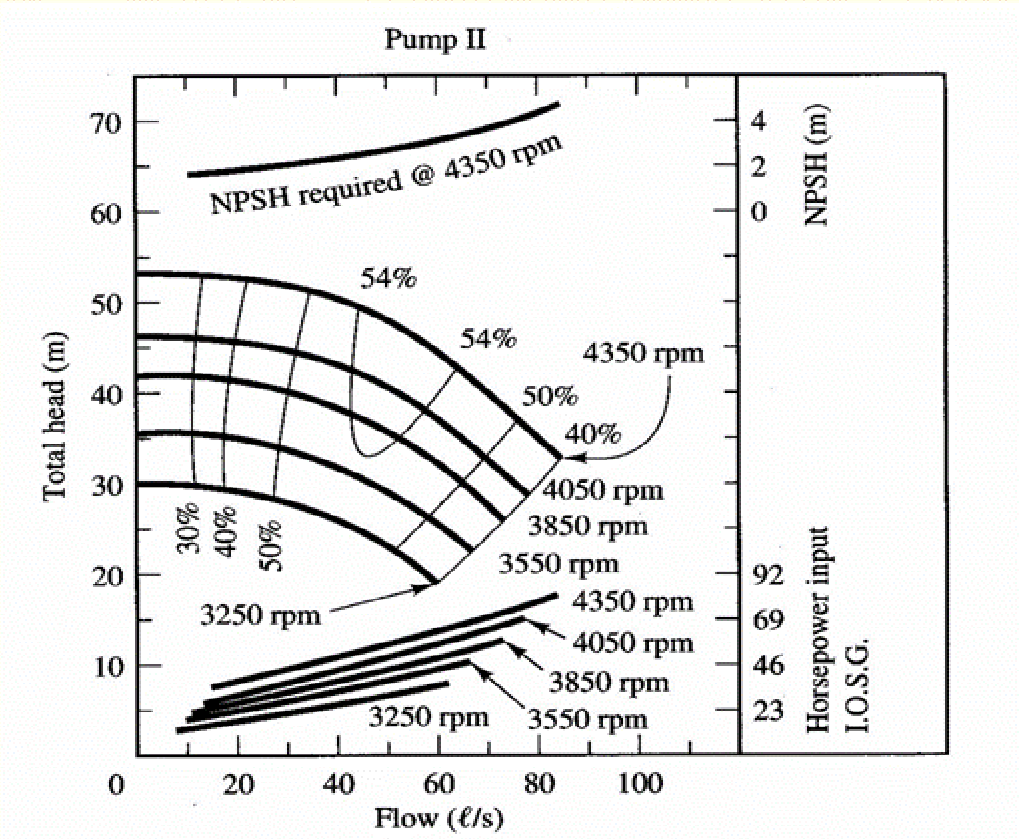


Figure 3.6: Typical pump characteristic curves curve

3.2 Pump Characteristic Curves

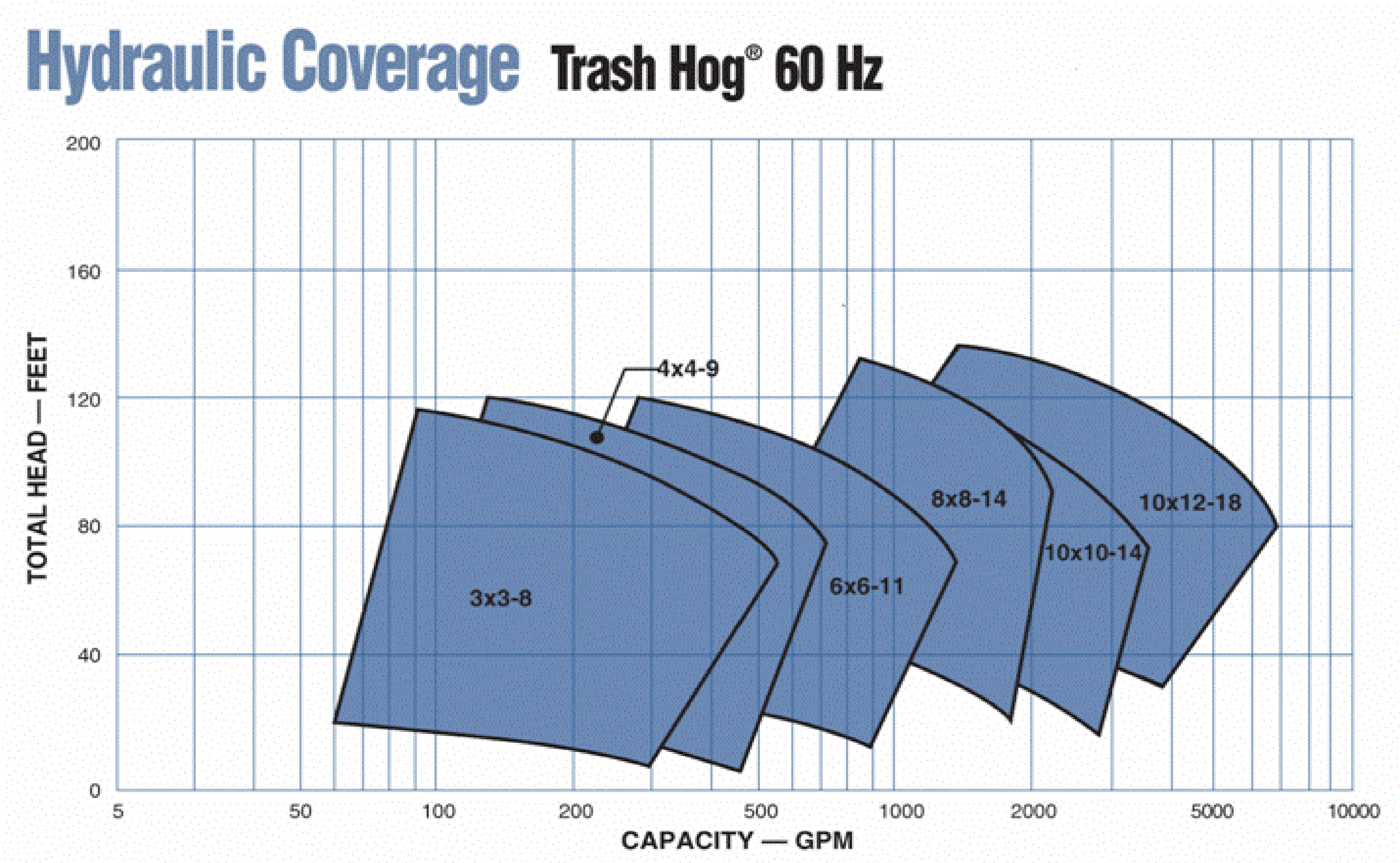


Figure 3.7 :Coverage range of different pump models given by the manufacturer

3.3 Interaction between Pump and system curves

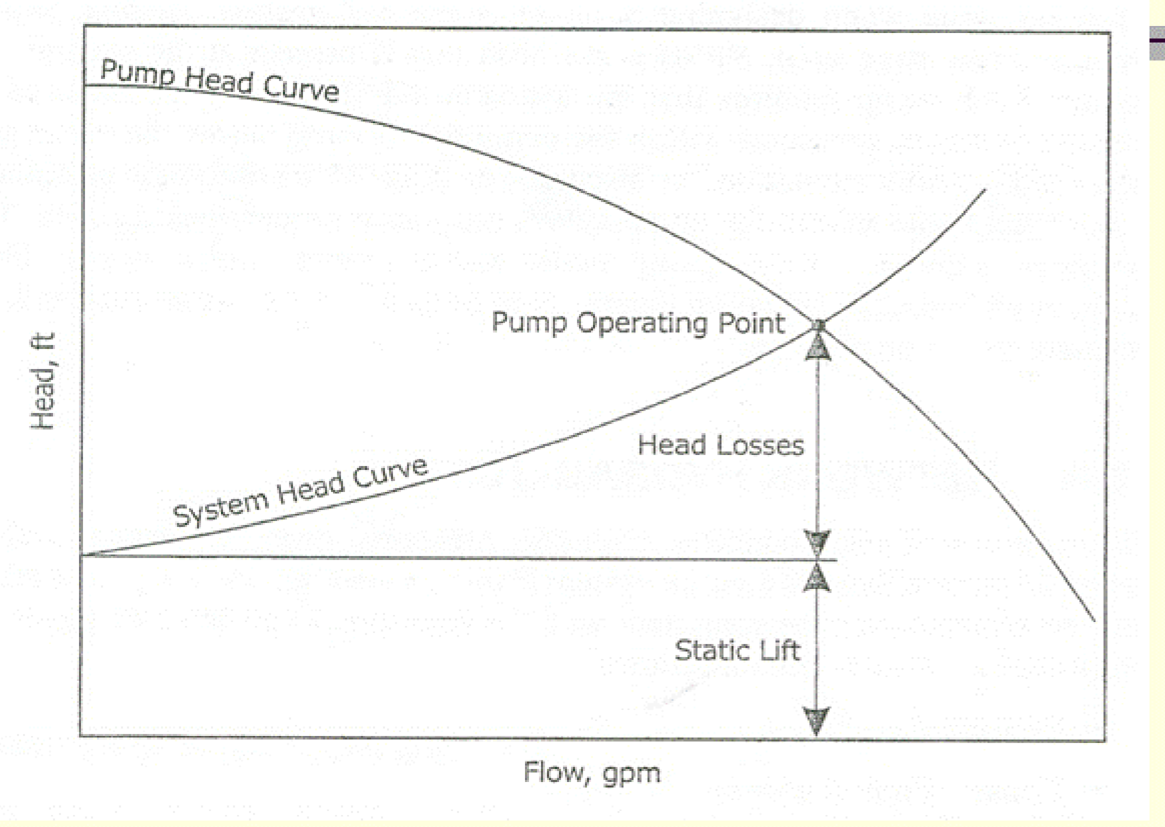


Figure 3.8: Interaction between pump and system curves

Lecture 3: Pumping Hydraulics

3.3 Interaction between Pump and system curves

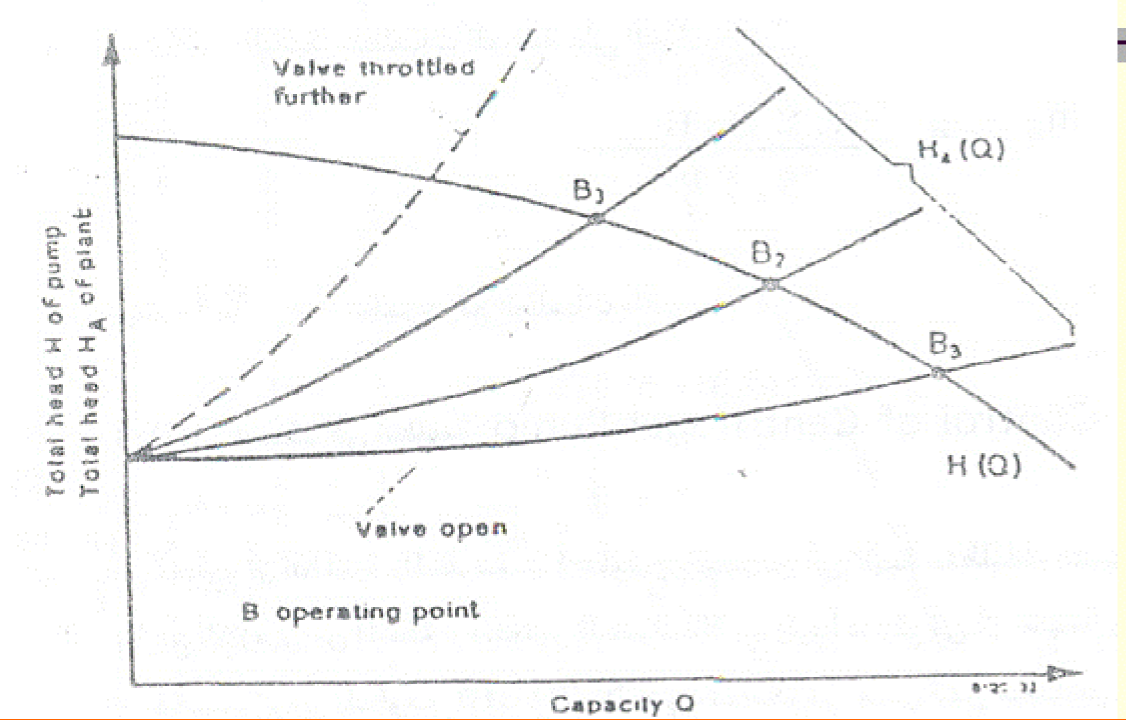
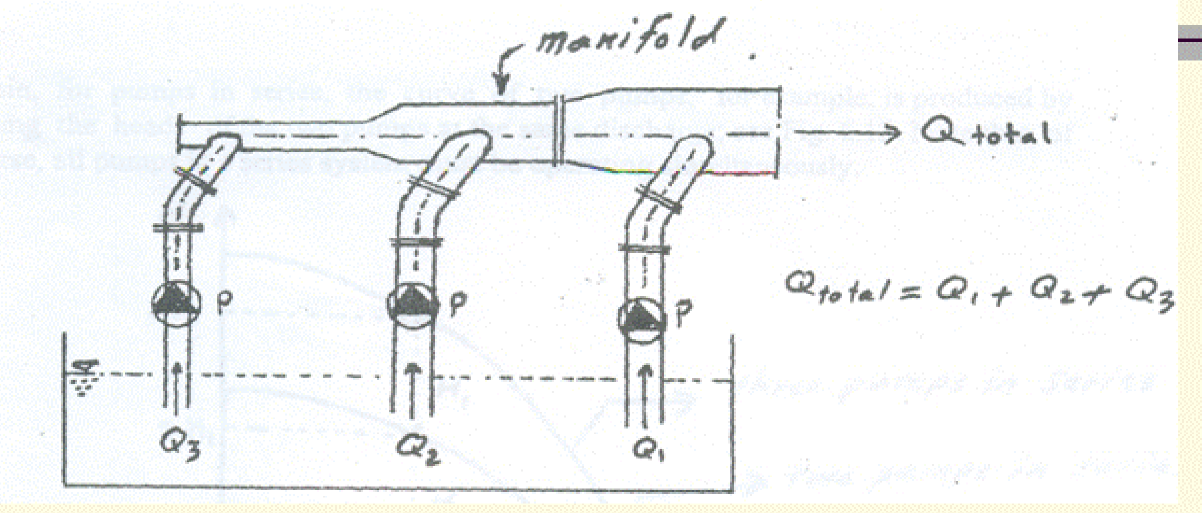


Figure 3.9: Interaction between pump and system curves

Note that the operating point of the pump changes with the change in the system curve. The changes in the system curve in this figure is due to the Change in minor losses.

3.4 Multiple Pump operation



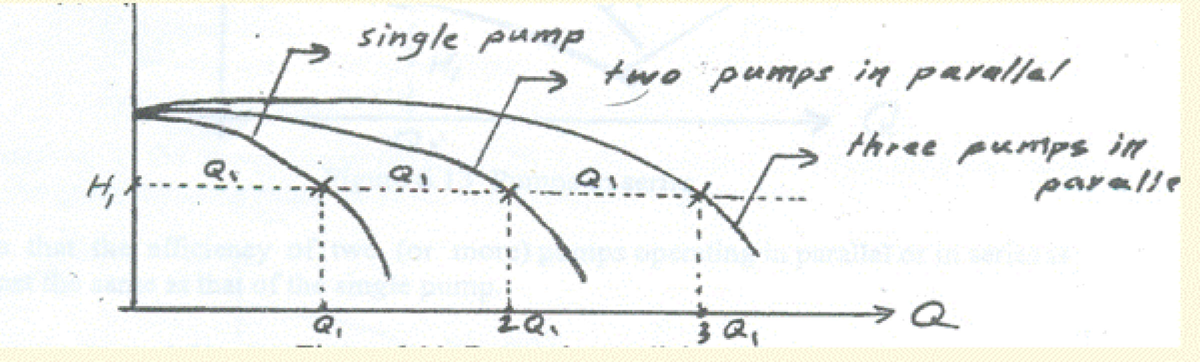
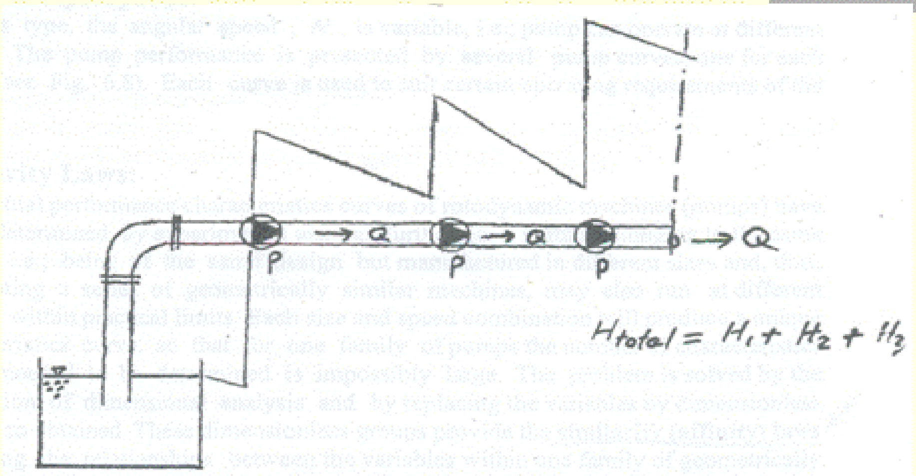


Figure 3.10 : pumps in parallel

3.4 Multiple Pump operation



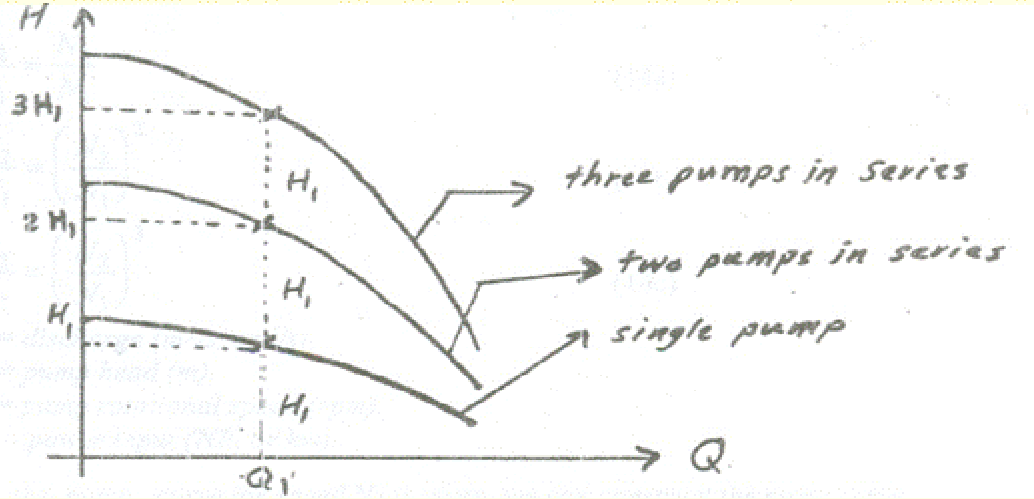


Figure 3.11pumps in series

3.5 Variable and constant speed pumps

- Pumps are classified into constant speed and variable speed pumps. The main difference between the two types is given hereafter:

Constant speed pump	Variable speed pump
Constant speed drive	**Variable speed drive**
Constant rotational speed of the impeller	**Variable rotational speed of the impeller**
One pump characteristics curve	**Many pump characteristics curves**
Operational range is limited	**Operational range is wide**

3.5 Variable and constant speed pumps

- Benefit of variable speed pumps:

1. In wastewater pumping stations:

- Energy savings as the pump interacts with the variations in Q and H.
- No storage is required (theoretically) so the wet well be Minimized.
- Short residence time of wastewater. This prevent odor production and minimize deposition of organic matter in the wet well.
- The soft and reduced number of starts increase the life of the pump.
- The variations in the flow coming out of pump station are gradual which protect wastewater treatment processes from sudden flow variations.
- The gradual start up and shut down protects the system from water hammer.

2. In water pumping stations:

- Energy savings as the pump interacts with the variations in Q and H.
- V/S booster pumps can provide constant pressure d during varying rates of water demand.

3.5 Variable and constant speed pumps

- disadvantages of variable speed pumps:
  - It requires more equipments and maintenance.
  - Usually the operators do not understand the operation of the V/S and turn the pump to the C/S operation mood.
  - V/S drives are less electrical efficiency than C/S drives.
  - The equipments used to regulate the pump speed are sensitive and needs special care.
  - V/S drives are more noisy than that of the C/S drives.

3.5 Variable and constant speed pumps

- Affinity laws (similarity laws) for variable speed pumps:
  - Variable speed pumps are operated at variable speeds. Consequently their performance characteristics change (i.e. Q,H, P). The performance of V/S pumps can be predicted Using the affinity laws.
  - If the V/S is operated at a rotational speed N1 and has an operation point of Q1,H1 and has a power input of P1, then, if the rotational speed is changed to N2 the other performance parameters can be calculated from the affinity laws:

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 3.1$$

$$\frac{H_1}{H_2} = \left[\frac{N1}{N2}\right]^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 3.2$$

$$\frac{P_1}{P_2} = \left[\frac{N_1}{N_2}\right]^3 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 3.3$$

Where:
Q= pumping flow rate ,m^3/s
N= pump rotational speed, rpm
H= pump head, m
P = power input to the pump, Watt

3.5 Variable and constant speed pumps

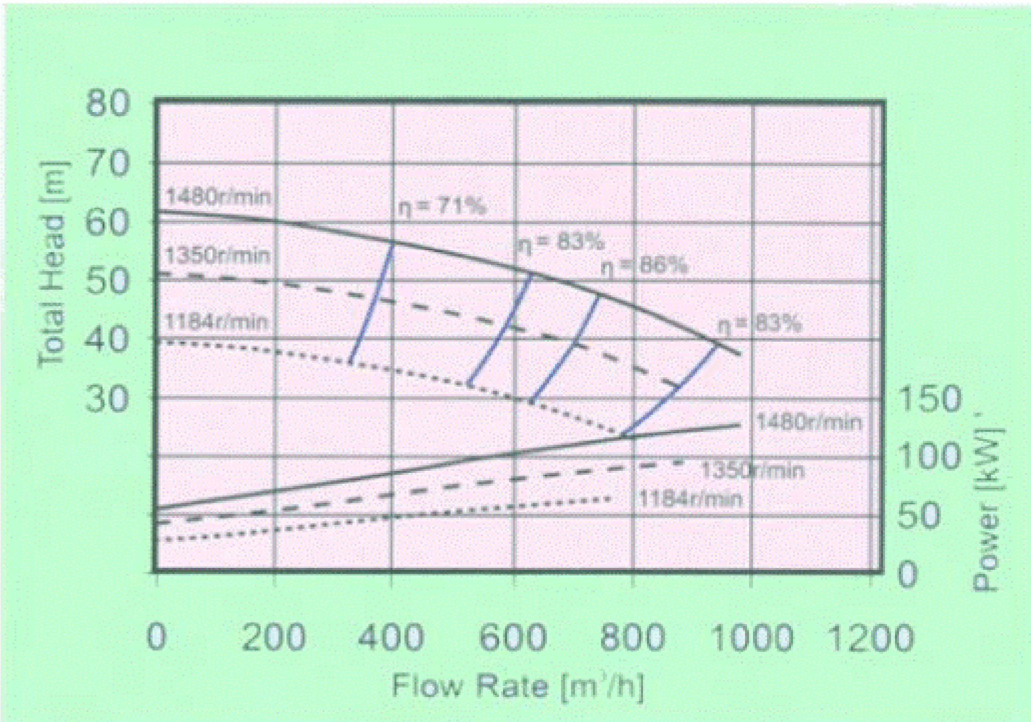


Figure 3.12: system curves of a variable speed pumps: same impeller diameter with variable rotational speed (1184,1350, 1480 rpm)

3.5 Variable and constant speed pumps

Example 10-2
Application of Affinity Laws

Problem: A pump with a normal operating speed of 705 rev/min and a 356-mm (14-in.) diameter impeller was tested at rotational speeds of 705, 625, 550, 450, and 350 rev/min. Using the head and capacity data collected at 705 rev/min, generate pump curves for rotational speeds of 625, 550, 450, and 350 rev/min and compare the curves generated with the measured data points.

	SI Units		**U.S. Customary Units**	
	Flow rate (m^3/h)	Head (m)	Flow rate (gal/min)	Head (ft)
705 rev/min				
	1596	5.91	7025	19.4
	1361	7.28	5994	23.9
	1140	8.66	5020	28.4

3.5 Variable and constant speed pumps

	Flow rate (m³/h)	Head (m)	Flow rate (gal/min)	Head (ft)
	1002	9.39	4410	30.8
	866	10.21	3550	33.5
	363	11.28	1600	37.0
	0	13.11	0	43.0
625 rev/min				
	1420	4.57	6250	15.0
	1136	6.25	5000	20.5
	908	7.32	4000	24.0
	681	7.92	3000	26.0
	454	8.63	2000	28.3
	227	9.45	1000	31.0
	0	10.36	0	34.0
550 rev/min				
	1249	3.51	5500	11.5
	943	5.12	4150	16.8
	772	5.82	3400	19.1
	454	6.55	2000	21.5
	363	6.80	1600	22.3
	0	7.92	0	26.0
450 rev/min				
	1022	2.44	4500	8.0
	795	3.35	3500	11.0
	568	3.96	2500	13.0
	227	4.72	1000	15.5
	0	5.43	0	17.8
350 rev/min				
	568	3.13	2500	7.0
	472	2.41	2080	7.9
	363	2.59	1600	8.5
	0	3.05	0	10.0

3.5 Variable and constant speed pumps

Solution: Plot the pump head-capacity (H-Q) data obtained at 705 rev/min (shown as solid circles in Figure 10-7) and draw a smooth pump curve (shown as a solid line) through the plotted points.

Develop H-Q curves for the other rotational speeds from the smooth pump curve by applying the affinity laws for flow rate and head simultaneously. For example, the flow rate and head values at the point on the 625-rev/min curve that corresponds to point a on the 705-rev/min pump curve, with coordinate values of 1250 m^3/h and 8.10 m (5504 gal/min and 26.6 ft), are determined as follows.

Using Equation 10-15

$$Q_2 = Q_1/(n_1/n_2) = 1250/(705/625) = 1108\,\mathrm{m}^3/\mathrm{h}$$

Using Equation 10-16

$$H_2 = H_1/(n_1/n_2)^2 = 8.10/(705/625)^2 = 6.37\,\mathrm{m}$$

So, point b with coordinates of 1108 m^3/h and 6.37 m (4875 gal/min and 20.9 ft) on the 625-rev/min curve *corresponds* to point a on the 705-rev/min curve. Enough other *corresponding points* are computed in the same manner to allow the entire curve to be drawn for 625 rev/min. The process is repeated for curves at 550, 450, and 350 rev/min, which pass through points c, d, and e, respectively. The curves are shown as dashed lines to indicate that they are derived mathematically from the curve at 705 rev/min. The measured values for head and capacity are plotted in Figure 10-7 as open circles. The correspondence between measured values and computed pump curves is excellent.

3.5 Variable and constant speed pumps

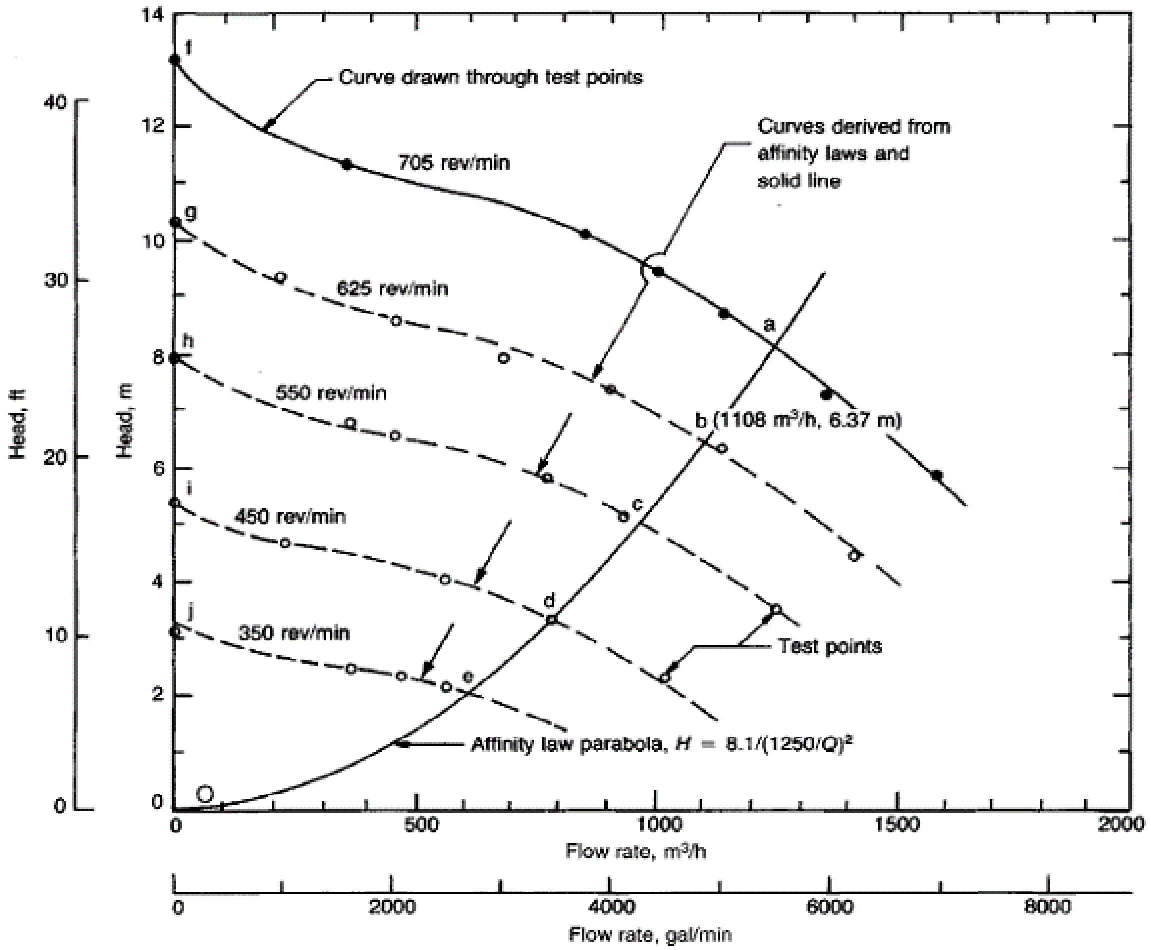


Figure 10-7. Pump curves developed from the affinity laws compared with test data.

3.5 Variable and constant speed pumps

Points a, b, c, d, and e lie on a parabola that passes through the origin. The equation for the affinity parabola can be found by solving Equations 10-15 and 10-16 simultaneously to eliminate n.

$$\left(\frac{Q_1}{Q_2}\right)^2 = \left(\frac{n_1}{n_2}\right)^2 = \frac{H_1}{H_2}$$

So $H_1 = H_2\,(Q_1/Q_2)^2$, which is a parabola through point O. Hence, all *corresponding points* lie on parabolas that pass through point O, the origin.

Lecture 3: Pumping Hydraulics

3.5 Variable and constant speed pumps

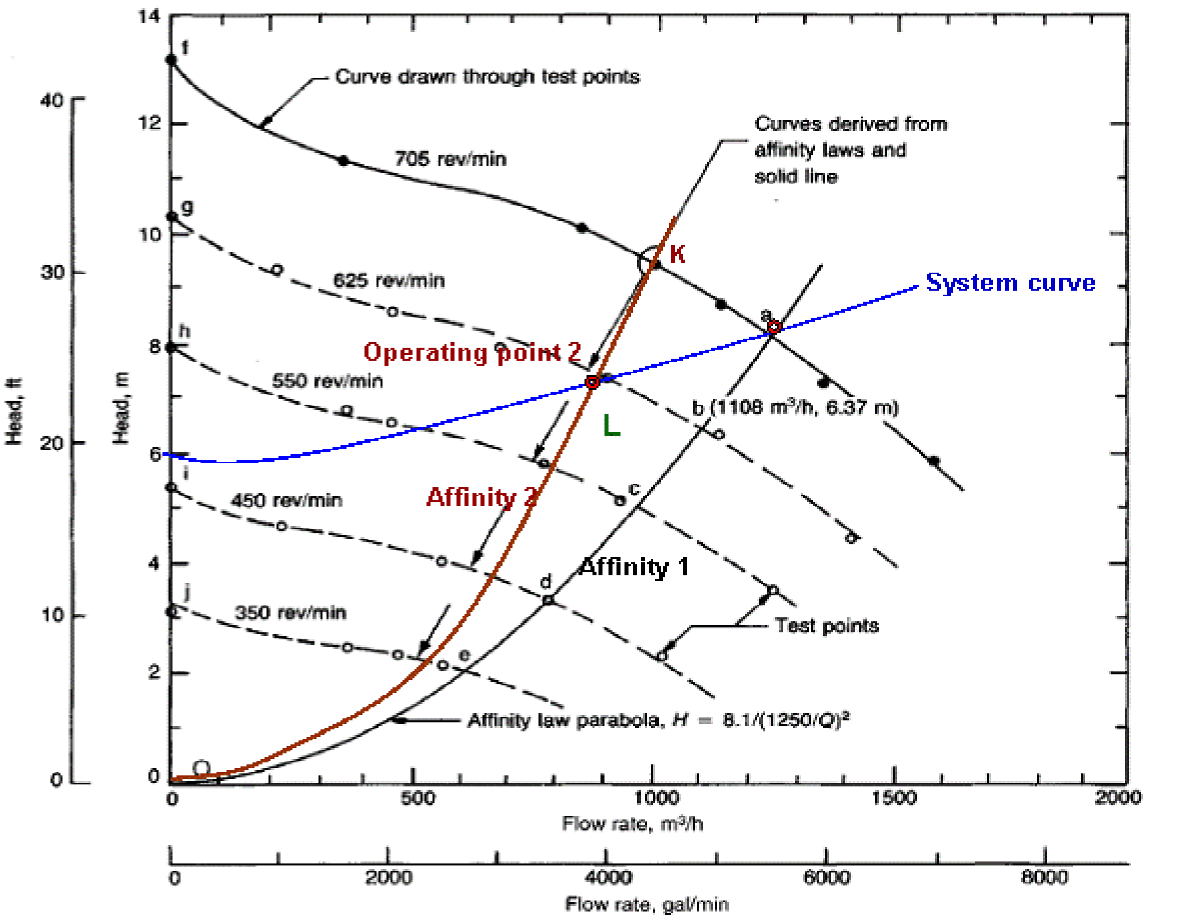


Figure 10-7. Pump curves developed from the affinity laws compared with test data.

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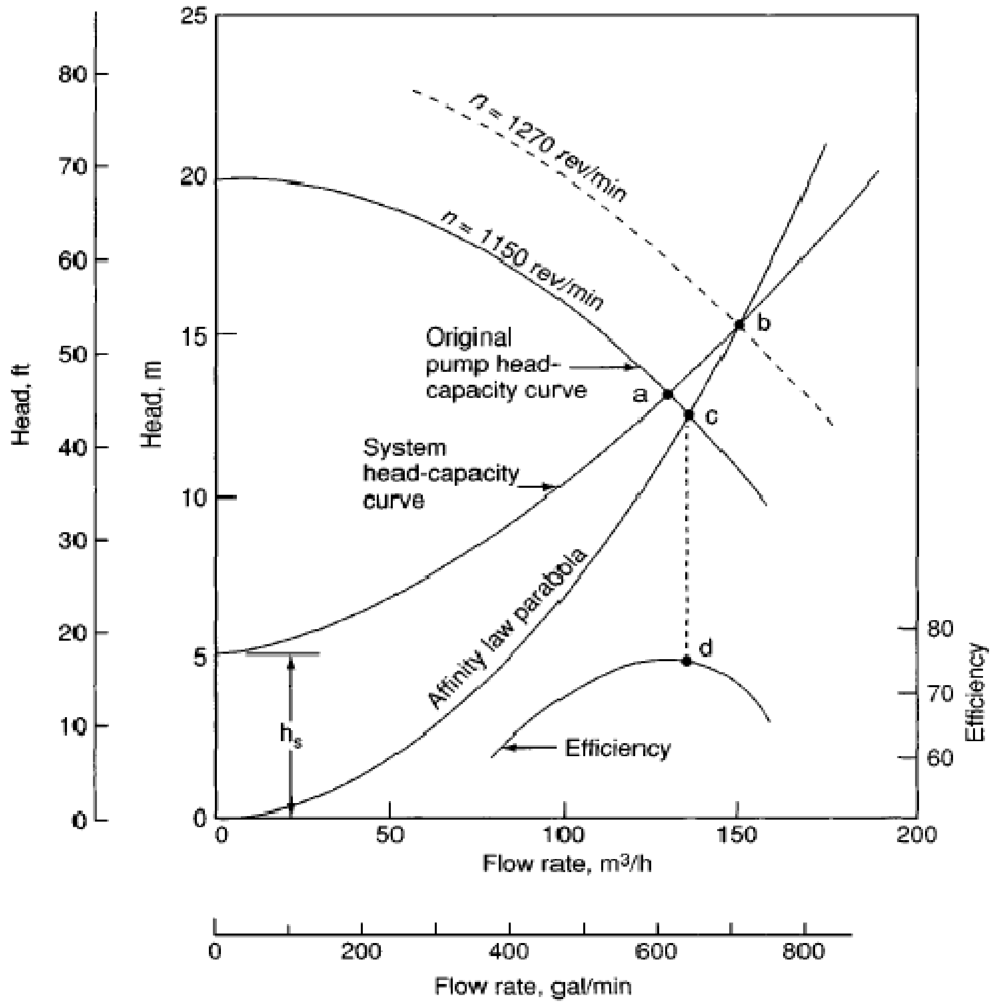
3.5 Variable and constant speed pumps

Example 10-8
Application of a Variable-Speed Pump

Problem: The pump curve shown in Figure 10-23 is for a pump operated at 1150 rev/min that pumps $130\,m^3/h$ (572 gal/min) at a head of 13.2 m (43.3 ft), which is shown as point a. If the pump must discharge $150\,m^3/h$ (660 gal/min) at a head of 15.5 m (50.8 ft), which is shown as point b, what is (1) the required new operating speed, and (2) the efficiency of the new operating point?

Solution: (1) *Operating speed.* Extend the system H-Q curve from point a to an intersection with the desired flow rate at point b. The new pump curve must pass through point b. Note that Equation 10-15 ($Q_1/Q_2 = n_1/n_2$) cannot be used to find the pump speed because points a

3.5 Variable and constant speed pumps



3.5 Variable and constant speed pumps

and b are not corresponding points (see Example 10-2 and the text following Equation 10-17). One way to solve the problem is to create an "affinity law parabola" by solving Equations 10-15 and 10-16 simultaneously to eliminate n:

$$\frac{Q_1^2}{Q_2} = \frac{n_1^2}{n_2} = \frac{H_1}{H_2}, \quad H_1 = H_2(Q_1/Q_2)^2$$

which defines a parabola passing through the origin and point b in Figure 10-23. Every point on the parabola is a corresponding point. Because point b corresponds to point c, Equation 10-15 can be used to find speed at point b after Q is scaled at point c.

SI Units modified to m³ / h

$Q_c = 136\ \mathrm{m^3/h}$ (by scaling)

From Equation 10-15 ($n_b = n_c Q_b / Q_c$),

$n_b = 1150\ (150/136) = 1270\ \mathrm{rev/min}$

U.S. Customary Units

$Q_c = 598\ \mathrm{gal/min}$ (by scaling)

$n_b = 1150\ (660/598) = 1270\ \mathrm{rev/min}$

By scaling H at points b and c, the speed can also be found from Equation 10-16

($n_b = n_c\sqrt{H_b/H_c}$)

$n_b = 1150\sqrt{15.4/12.6} = 1270\ \mathrm{rev/min}$

$n_b = 1150\sqrt{50.5/41.3} = 1270\ \mathrm{rev/min}$

(2) *Efficiency*. The efficiency of the new operating point is found by projecting a line from point c downward to point d on the efficiency curve, which gives an expected efficiency of 75%.

3.5 Variable and constant speed pumps

- Affinity laws (similarity laws) for constant speed pumps:

  - Constant speed pumps are operated at a constant speed. The manufacturer of C/S pumps produce them in families of pumps . Each family has the same constant rotational speed N but with different impeller diameters. The performance characteristics (i.e. Q,H, P) of each pump from the family change by the change of the impeller diameter. The performance of C/S pumps from the same Family can be predicted using the affinity laws.

  - If one C/S from the family has an impeller diameter D1 is operated at a and has an operation point of Q1,H1 and has a power input of P1, then, if the another pump from the family has an impeller diameter D2 the other performance parameters can be calculated from the affinity laws:

3.5 Variable and constant speed pumps

- Affinity laws (similarity laws) for constant speed pumps:

$$\frac{H_1}{H_2} = \left[\frac{D_1}{D_2}\right]^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 3.4$$

$$\frac{Q_1}{Q_2} = \left[\frac{D_1}{D_2}\right]^3 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 3.5$$

$$\frac{P_1}{P_2} = \left[\frac{D_1}{D_2}\right]^5 \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad 3.6$$

Where:
Q= pumping flow rate ,m^3/s
D= impeller diameter, m
H= pump head, m
P = power input to the pump, Watt

3.5 Variable and constant speed pumps

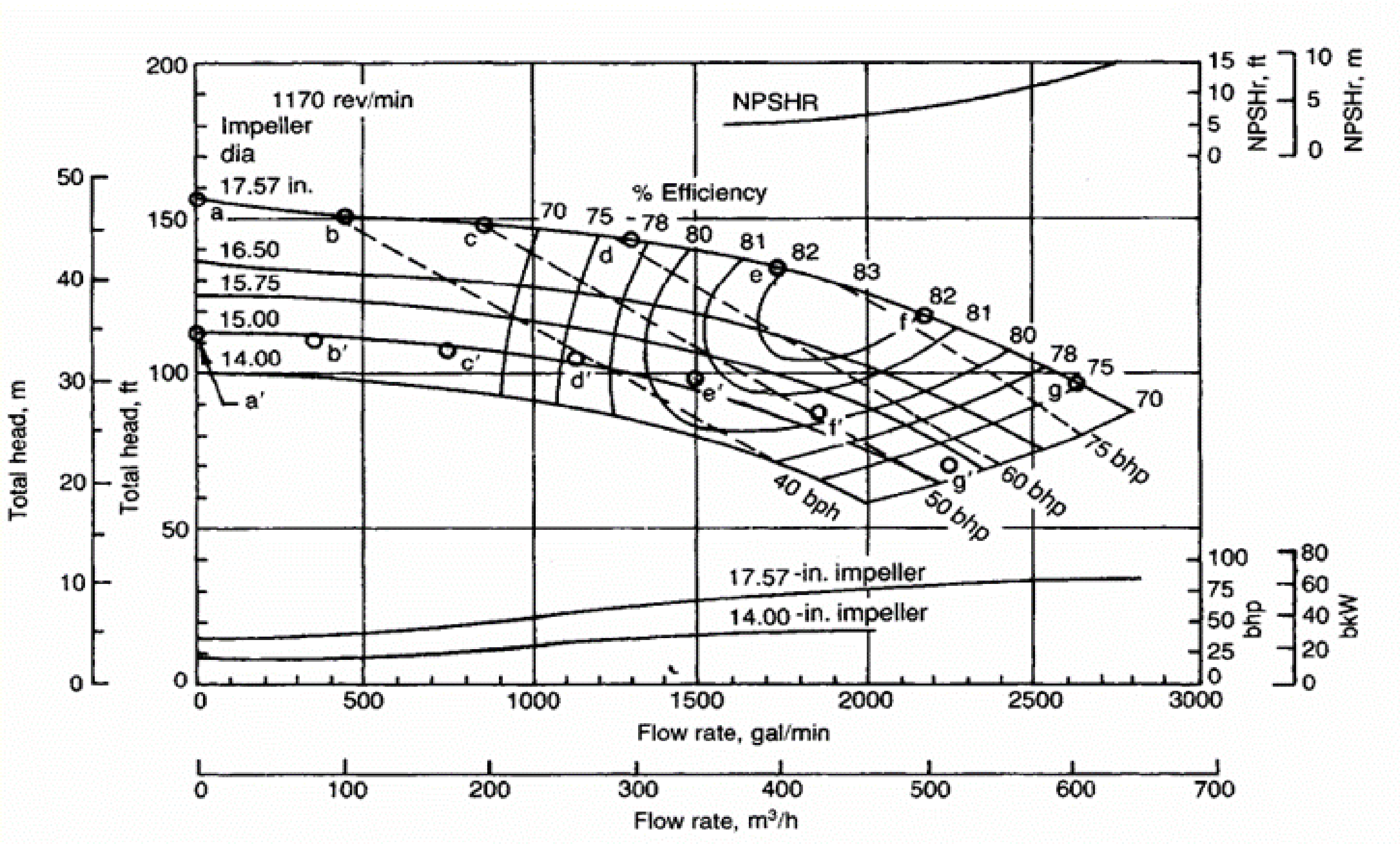


Figure 3.14: system curves of a family of constant speed pumps: N =1170rpm and D is variable

3.5 Variable and constant speed pumps

Example 10-3
Effect of Changes in Impeller Diameter and Speed

Problem: Data from Figure 10-5 for a pump with an impeller diameter of 0.4463 m (17.57 in.) operating at 1170 rev/min are tabulated as follows.

SI Units			U.S. Customary Units		
Point	Flow rate (m^3/h)	Head (m)	Point	Flow rate (gal/min)	Head (ft)
a	0	47.6	a	0	156
b	100	46.3	b	440	152
c	200	45.3	c	880	149
d	300	44.0	d	1320	144
e	400	41.0	e	1760	135
f	500	36.5	f	2200	120
g	600	29.8	g	2640	98

3.5 Variable and constant speed pumps

(1) Develop a new head-capacity curve for the same pump fitted with a new impeller 0.3810 m (15 in.) in diameter. (2) Compare the results with the test curve for the new impeller shown in Figure 10-5. (3) Find the increase in rotational speed that would be required for the new impeller to match the performance of the original impeller at 1170 rev/min.

Solution:

H-Q for the smaller impeller. Compute this using Equations 10-20 [$Q_2 = Q_1(D_2/D_1)$] and 10-18 [$H_2 = H_1(D_2/D_1)^2$]. Sample calculations for point d are

$Q_2 = 300\ (0.3810/0.4463) = 256\,\text{m}^3/\text{h}$ $\quad$ $Q_2 = 1320\ (15/17.57) = 1130$ gal/min

$H_2 = 44.0\ (0.3810/0.4463)^2 = 32.1\,\text{m}$ $\quad$ $H_2 = 144\ (15/17.57)^2 = 105\,\text{ft}$

Compare the results to Figure 10-5. The computed values of points a to g for the new 15.00-in. impeller diameter are summarized below.

SI Units				U.S. Customary Units			
		Head (m)				Head (ft)	
Point	Flow rate (m^3/h)	Computed	Test	Point	Flow rate (gal/min)	Computed	Test
a′	0	34.4	34.4	a′	0	113	113
b′	85	33.8	34.4	b′	376	111	113
c′	171	32.9	33.2	c′	751	108	109
d′	256	32.0	31.7	d′	1130	105	104
e′	341	29.9	29.0	e′	1500	98	95
f′	427	26.5	24.7	f′	1880	87	81
g′	512	21.3	—	g′	2250	70	—

3.5 Variable and constant speed pumps

Compare the computed with the tested heads in the last two columns or compare calculated points a′ to g′ in Figure 10-5 with the 15.00 curve. Minor differences (up to 0.6 m or 2 ft) are due to errors in plotting and scaling. Larger differences are due to the fact that actual losses in the pump are not considered in Equations 10-18 and 10-20.

Rotational speed for the trimmed impeller. Use Equation 10-16 ($n_2 = n_1\sqrt{H_2/H_1}$) at zero flow to estimate the new rotational speed required to obtain the original flow rate with the smaller impeller.

$$n_2 = 1170\sqrt{47.6/34.4} = 1376 \text{ rev/min} \qquad n_2 = 1170\sqrt{156/113} = 1375 \text{ rev/min}$$

Pumping Stations Design

For Infrastructure Master Program
Engineering Faculty-IUG

Lecture 4: Pumping Hydraulics

Dr. Fahid Rabah
Water and environment Engineering
frabah@iugaza.edu

Lecture 3: Pumping Hydraulics

The main items that will be studied under the pumping hydraulics title are:

- ❑ Pumping pressure and head terminology
- ❑ Cavitation
- ❑ Pump characteristic curves
- ❑ Multiple pump operation
- ❑ Variable speed pumps
- ❑ Affinity laws
- ❑ Pump selection

3.6 Pumps selection (see Chapter 25)

- Use of specific speed in pump type selection:

Specific speed is a term used to describe the geometry (shape) of a pump impeller.

Specific speed is defined as:

"The speed of an ideal pump geometrically similar to the actual pump, which when running at this speed will raise a unit of volume, in a unit of time through a unit of head."

Specific speed is calculated from the following formula:

$$N_s = \frac{n Q^{\frac{1}{2}}}{H_t^{\frac{3}{4}}}$$

Where:
N_s = specific speed (or type number)
n = pump rotational speed, rpm
H_t = total pump head, m
Q = pump discharge, m^3/s

3.6 Pumps selection

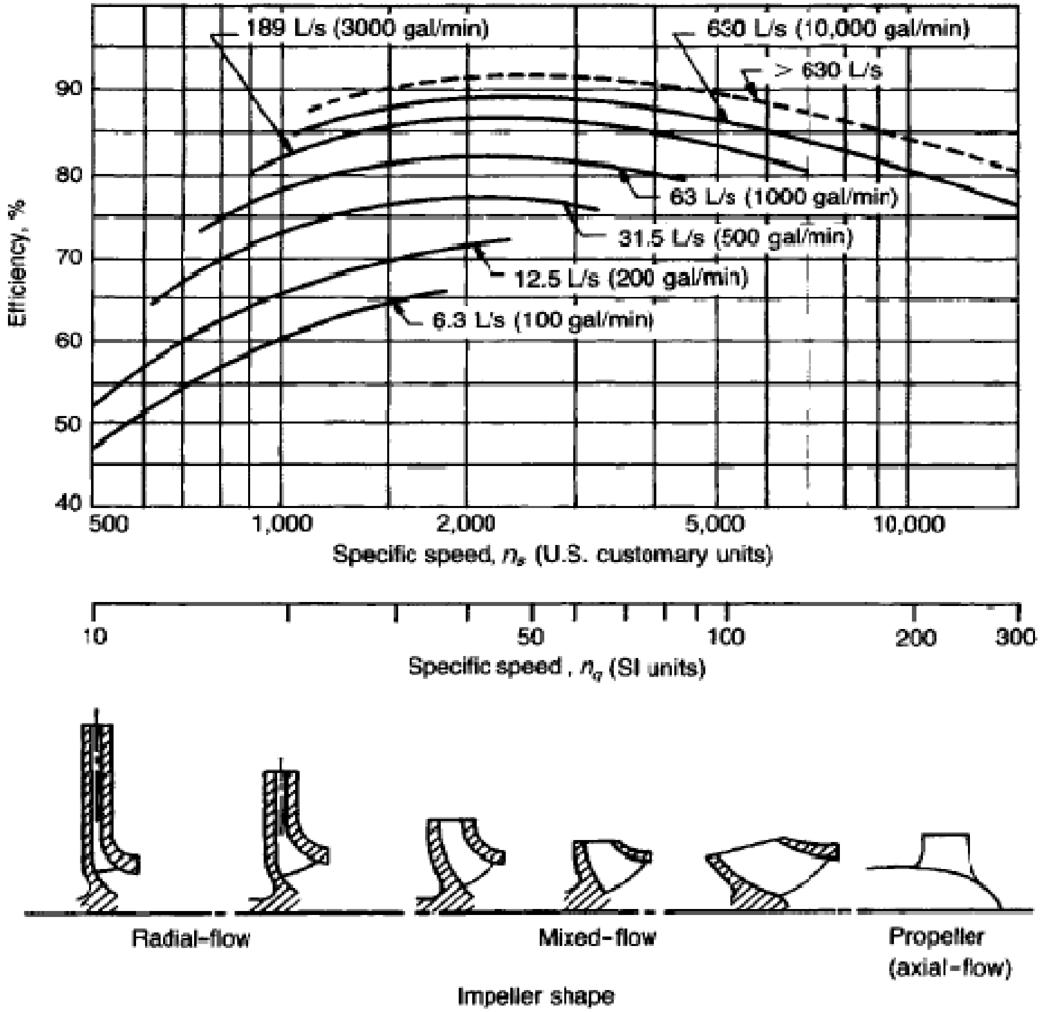


Figure 10-8. Pump efficiency as related to specific speed and discharge. After Flowserve Corp.

3.6 Pumps selection

- Use of specific speed in pump type selection:

Example:

A flow of 0.01 m^3/s a against a head of 20m. The pump is to be driven with An electric motor at a speed of 1750 rpm. What type of centrifugal pump should Be selected and what is corresponding efficiency?

Solution:

Calculate the specific speed of the pump.

$$N_s = \frac{n Q^{\frac{1}{2}}}{H_t^{\frac{3}{4}}} = \frac{1750(0.01)^{\frac{1}{2}}}{(20)^{\frac{3}{4}}} = 18.5$$

Using figure 10-8 the pump is radial flow with an expected efficiency of 62%

3.6 Pumps selection

- In the selection process the following important information is needed:

1. Type of the liquid to be pumped “ water, wastewater, sludge”.
2. Type of installation preferred“ submersible or dry”.
3. Pumps available in the market (KSB, FLYGT, ABS,...).

- selection of the specific pump needed :

1. Draw the system curve envelope and decide the operating point.
2. Decide the number of pumps needed (see table 3.1).
3. Use the catalog of the manufacturers to select a suitable pump curve.
4. Draw the pump characteristics curve over the system curve.
5. Select a pump so that the Operating range of is within 60% to 115% of Q at BEP.
6. Compare pumps of different manufacturers for more economical choice.

3.6 Pumps selection

Flygt

KSB

ABS

KSB

3.6 Pumps selection

Table 3.1 Recommended number of pumps in operation and their stand by.

Design flow m^3/h	No. of pumps needed	Standby pumps
Up to 160	1	1
160 -450	2 or 3	1
More than 450	3 to 5	2

Lecture 4: Water hammer phenomena (Hydraulic Transient)

4.1 General Introduction (See chapters 6 &7)

- What is water hammer?

Water hammer is a phenomena that occurs in pressurized pipe systems when the flowing liquid is suddenly (instantaneously) obstructed by (for example) closing a valve or by pump stoppage (failure). A laud noise similar to hammer knocking noise occurs due to the collision of the liquid mass with the obstruction body (valve or pump) and the internal walls of the pipe.

- What is water hammer scientifically?

Water hammer is a descriptive name of the phenomena but not scientific. The scientific name is hydraulic transient. Hydraulics Transient means temporary hydraulics since the phenomena last for a very short time. During this short time the flow in the pipe is disturbed and consecutive waves will be moving back and forth creating a very high pressure rise in the pipe.

4.1 General Introduction (See chapters 6 &7)

- What is the problem with water hammer?

Hydraulic transient (water hammer) may cause disasters in pressurized liquid systems such as:

1. Rupture of pipes and pump casings.
2. Vibration and high noise.
3. Excessive pipe displacements.
4. Vapor cavity formation.
5. Environmental pollution.
6. Life and economical losses.

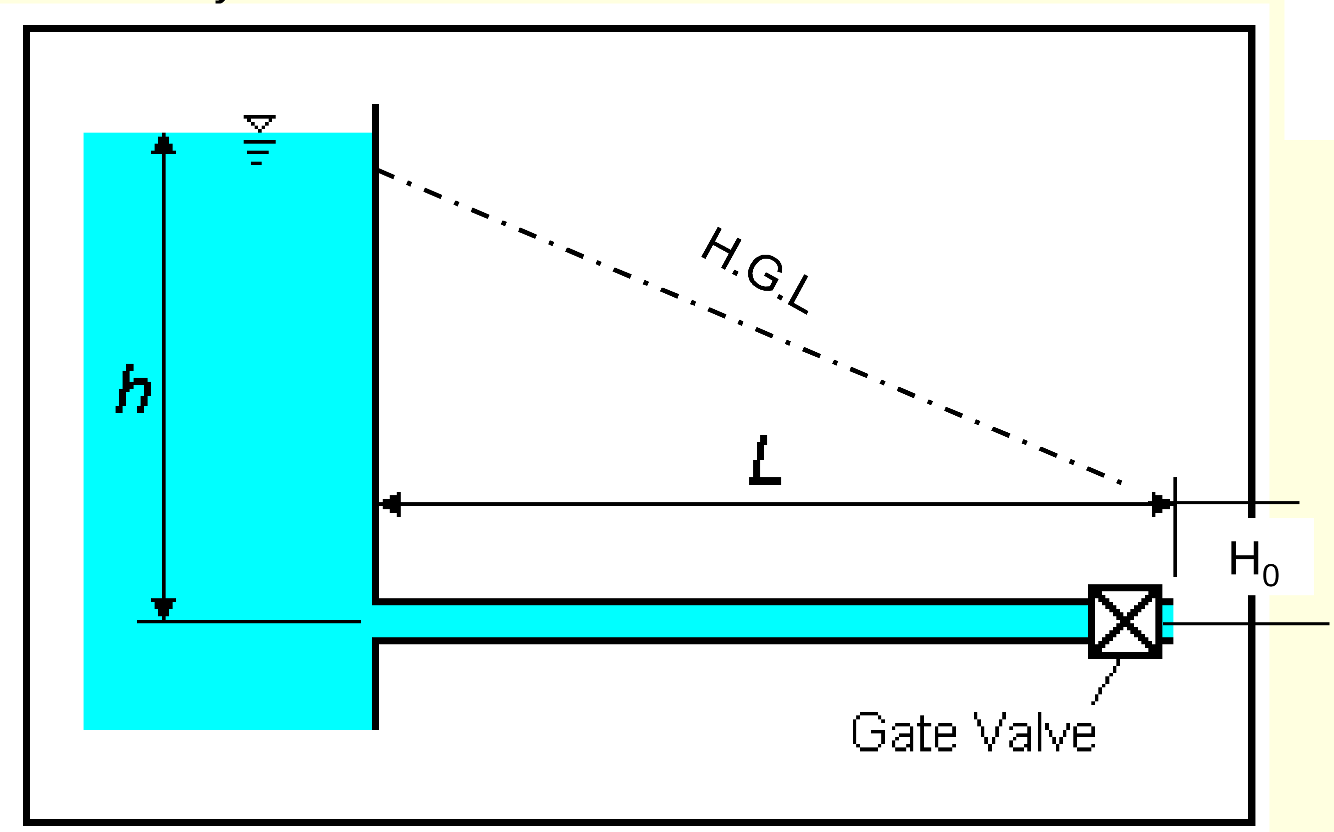
- When water hammer occurs?

The most common cases where water hammer occurs are:

1. Pump failure.
2. Pump start up.
3. Sudden closure a valve.
4. Failure of flow or pressure regulators.

4.2 Water hammer pressure surge

To illustrate the pressure waves resulting from the hydraulic transient we will discuss a tank, pipe, and valve system as an example. The figure below shows the normal operation of the system before the valve closure.



4.2 Water hammer pressure surge

What happens when the valve is closed instantaneously ?

- When a valve is closed instantaneously the liquid next to the valve comes to a halt.
- This liquid is then compressed by the liquid upstream which is still flowing.
- This compression causes a local increase in the pressure on the fluid.
- The walls of the pipe around the fluid are stretched by the resulting excess pressure
- A chain reaction then takes place along the length of the pipe, with each stationary
- element of fluid being compressed by the flowing fluid upstream.
- This chain reaction results in a pressure wave which travels up the pipe with a velocity “a “, which is called the celerity. (it may reach the speed of sound in water (1484 m/s).
- The pressure wave creates a transient hydraulic grade line (HGL) which is parallel to the steady flow HGL, but at a height above it.
- The resulting increase in pressure is due to the water hammer surge.
- When the pressure wave reaches the reservoir the pressure cannot exceed the water depth in the reservoir.

4.2 Water hammer pressure surge

What happens when the valve is closed instantaneously ?

- A wave of pressure unloading travels back along the pipe in the opposite direction.
- As the unloading wave travels down the pipe the flow is reversed, toward the reservoir.
- When this unloading wave hits the closed valve the flow is stopped and a drop in pressure occurs.
- A negative pressure wave now travels up the pipe toward the reservoir.
- The process described above now repeats itself for a negative wave.
- A cycle of pressure waves (positive - unloading - negative - unloading) now travels up and down the length of the pipe.
- Friction has the effect of slowly damping out the effect of the pressure waves.
- When the negative pressure wave travels along the pipe very low pressures may cause cavitation

4.2 Water hammer pressure surge

What are the factors affecting the wave speed and How it can be calculated ?

- The speed of pressure wave *"a"* depends on :
  - the pipe wall material.
  - the properties of the fluid.
  - the anchorage method of the pipe.
- For elastic pipes (steel, PVC,..) the wave speed is given by the following formula:

$$a = \sqrt{\frac{K/\rho}{1 + C\left(\frac{K}{E}\right)\left(\frac{D}{e}\right)}}$$

4.2 Water hammer pressure surge

What are the factors affecting the wave speed and How can it can be calculated wave ? Continued …….

Where:

a = elastic wave speed in water contained in a pipe, m/s

K =Bulk modulus of elasticity of the liquid, N/m^2 ,(K= $2.15X10^9$ N/m^2 for water)

E = Modulus of elasticity of pipe material, N/m^2

D = Inside diameter of the pipe, m

e = Wall thickness of the pipe, m

ρ = Density of the fluid , kg/m^3

C = Correction factor depending on the type of pipe restrain

= (1.25 -μ) for pipes anchored from one side only.

= $(1.0-\mu^2)$ for pipes anchored against axial movements.

= 1.0 for pipes with expansion joints throughout.

μ = Poisson's ratio of the pipe material

4.2 Water hammer pressure surge

What do we mean by instantaneous valve closure?

The time required for the pressure wave to travel from the valve to the reservoir and back to the valve is:

$$\boxed{t = \frac{2L}{a}}$$

Where:

L = length of the pipe (m)

a = speed of pressure wave, celerity (m/sec)

- If the valve time of closure is t_c , then
  - If $\boxed{t_c > \frac{2L}{a}}$ the closure is considered gradual
  - If $\boxed{t_c \leq \frac{2L}{a}}$ the closure is considered instantaneous

4.2 Water hammer pressure surge

What is the value of the pressure surge?

There are two cases for pressure surge:

1. Gradual closure of the valve:

In this case $t_c > \frac{2L}{a}$ and the pressure increase (Δ H) is:

$$\Delta H = \frac{LV_0}{gt}$$

Where:
V_0 = initial velocity of the fluid , m/s
g = gravitational acceleration, m2/s
t = closure time, s
L = pipe length, m
a = Wave speed, m/s

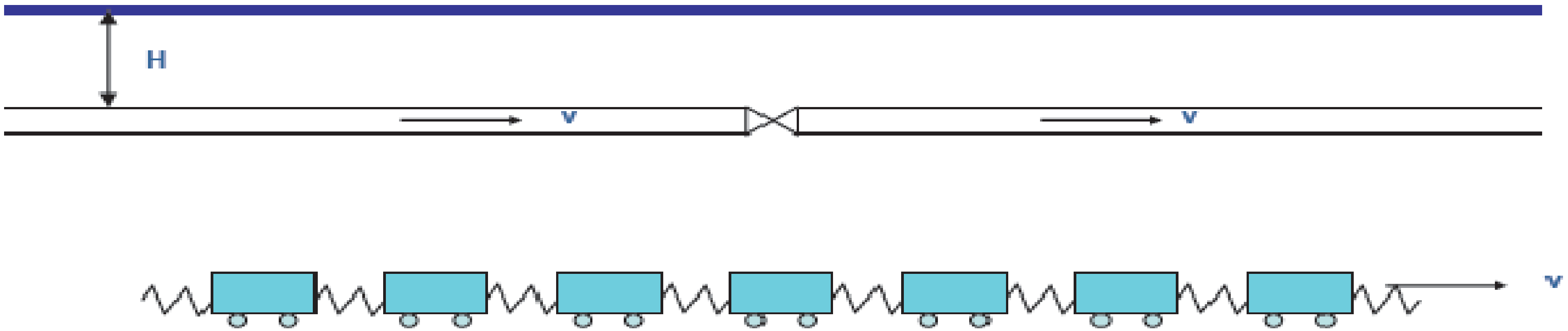
2. Instantaneous closure of the valve:

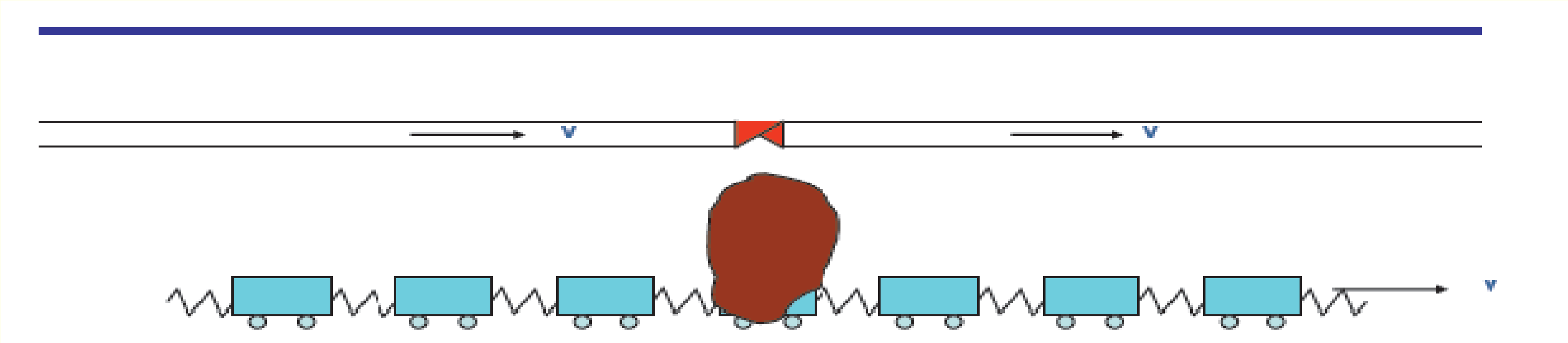
In this case $t_c \leq \frac{2L}{a}$ and the pressure increase (Δ H) is:

$$\Delta H = \frac{aV_0}{g}$$

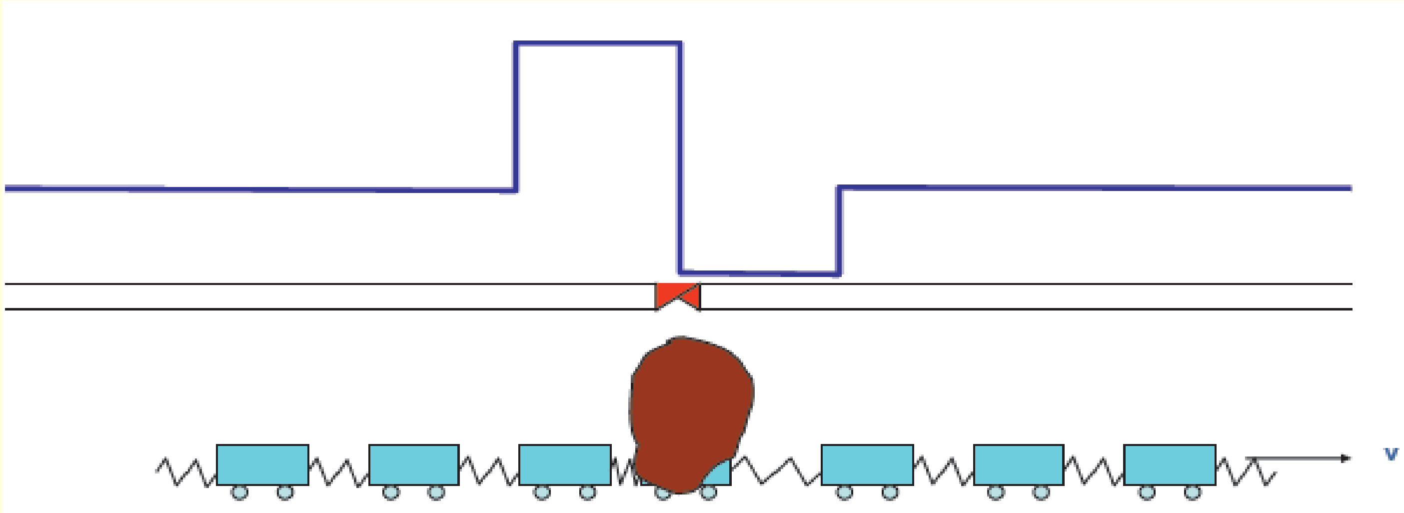
Note: that (Δ H) due to gradual closure is very small compared with the instantaneous (Δ H).

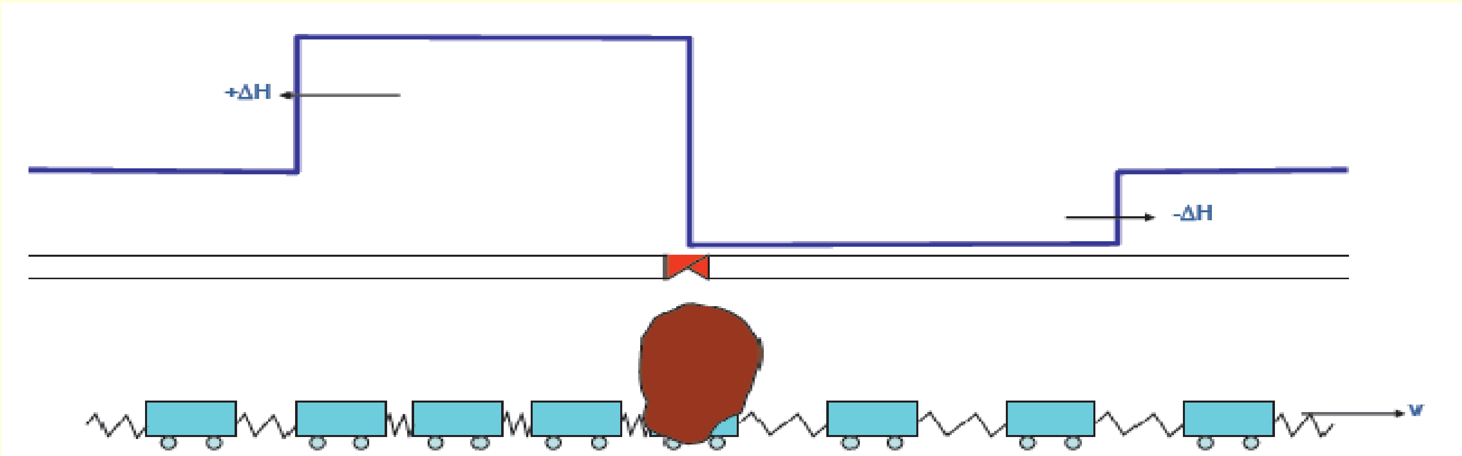
4.2 Water hammer pressure surge



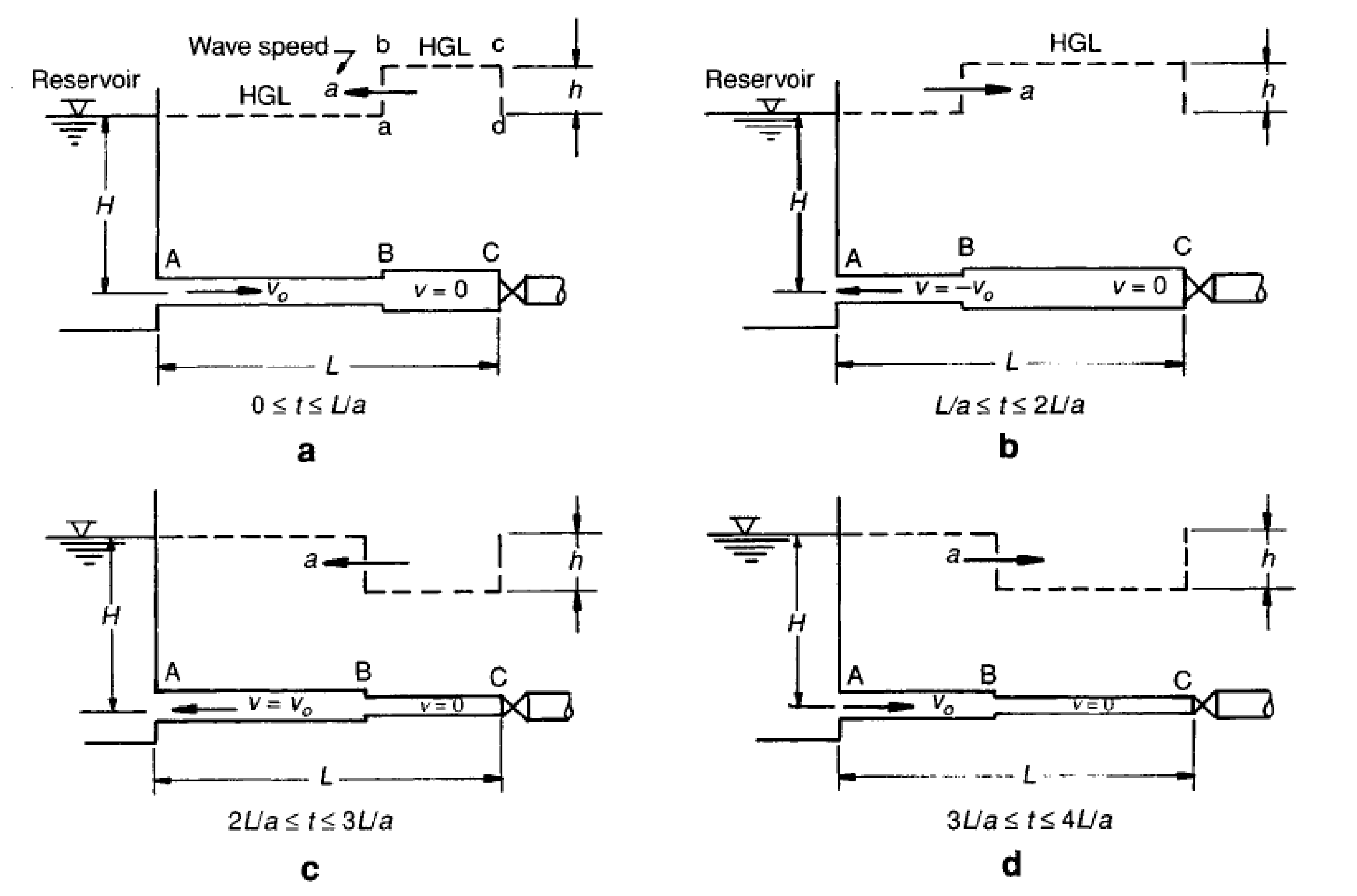


4.2 Water hammer pressure surge

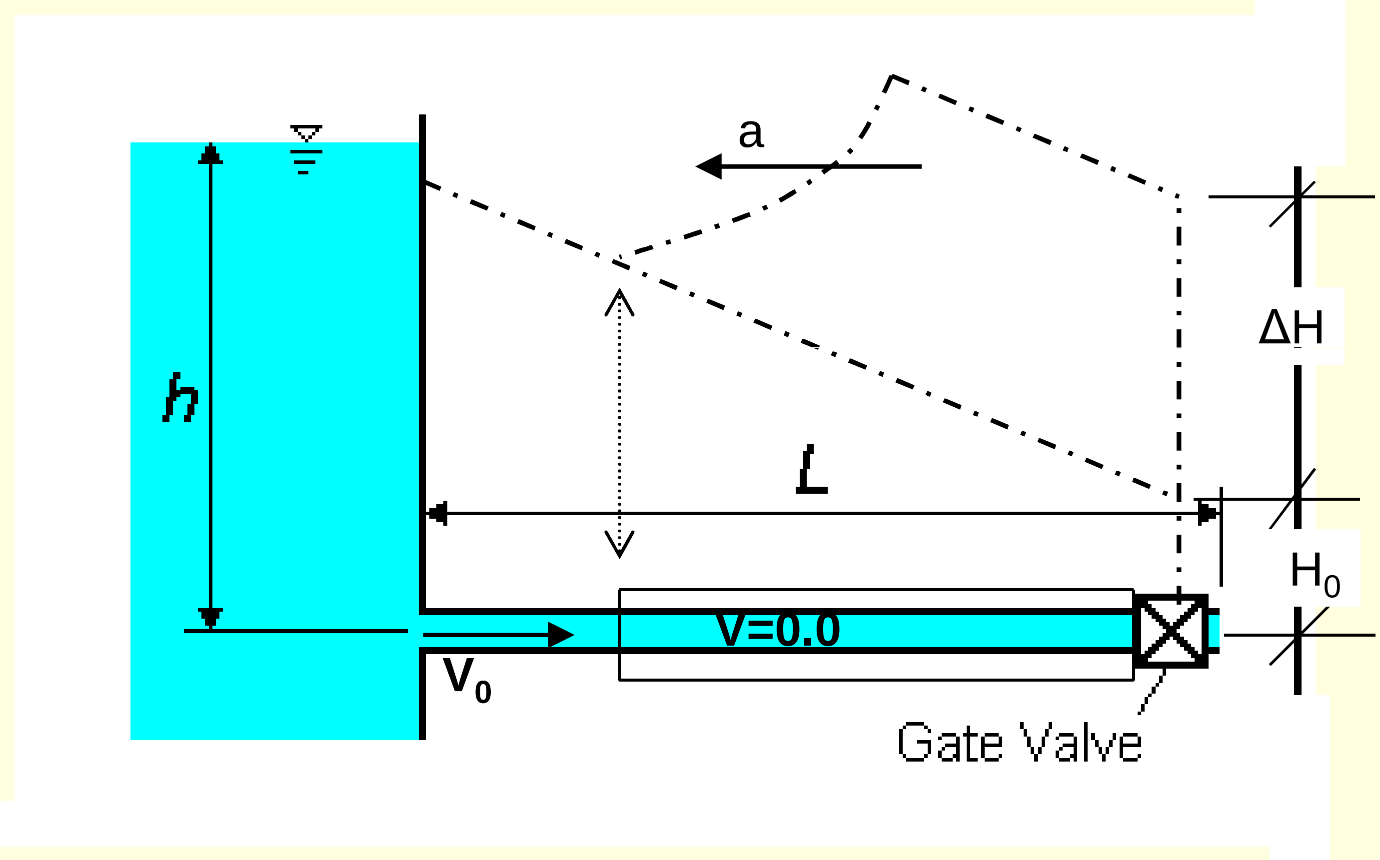




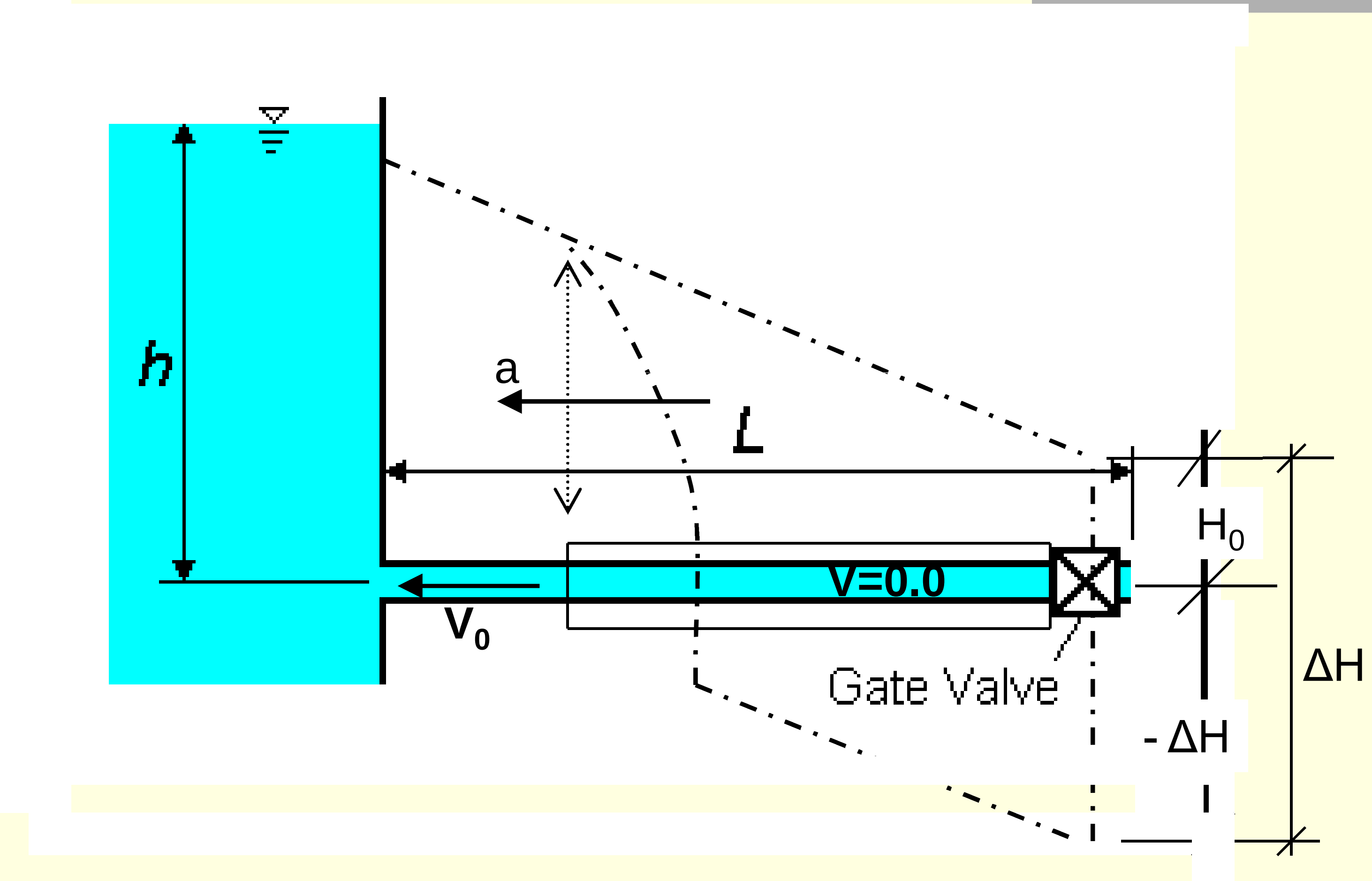
4.2 Water hammer pressure surge



4.2 Water hammer pressure surge



4.2 Water hammer pressure surge

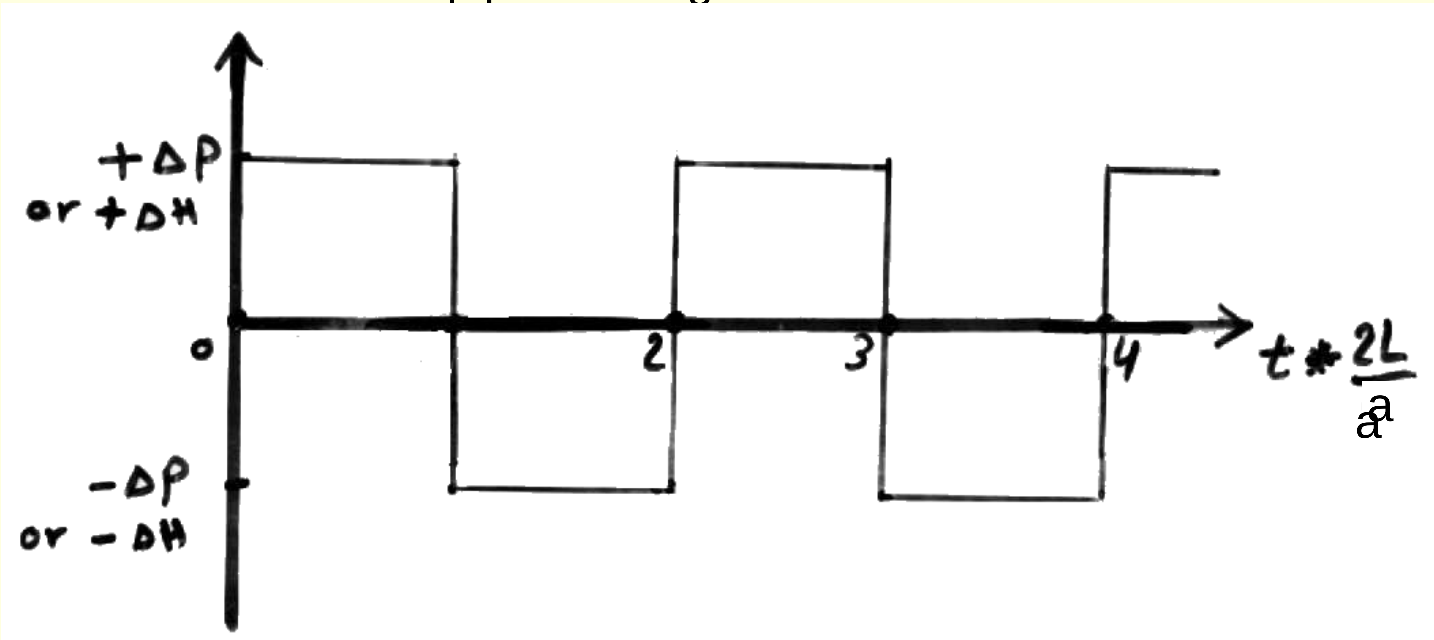


4.3 Time history of pressure wave due to water hammer):

The time history of the pressure wave for a specific point on the pipe is a graph that simply shows the relation between the pressure increase (ΔH, or ΔP) and Time during the propagation of the water hammer pressure waves. The following Figures illustrate the time history in the tank pipe system discussed previously.

4.3 Time history of pressure wave due to water hammer):

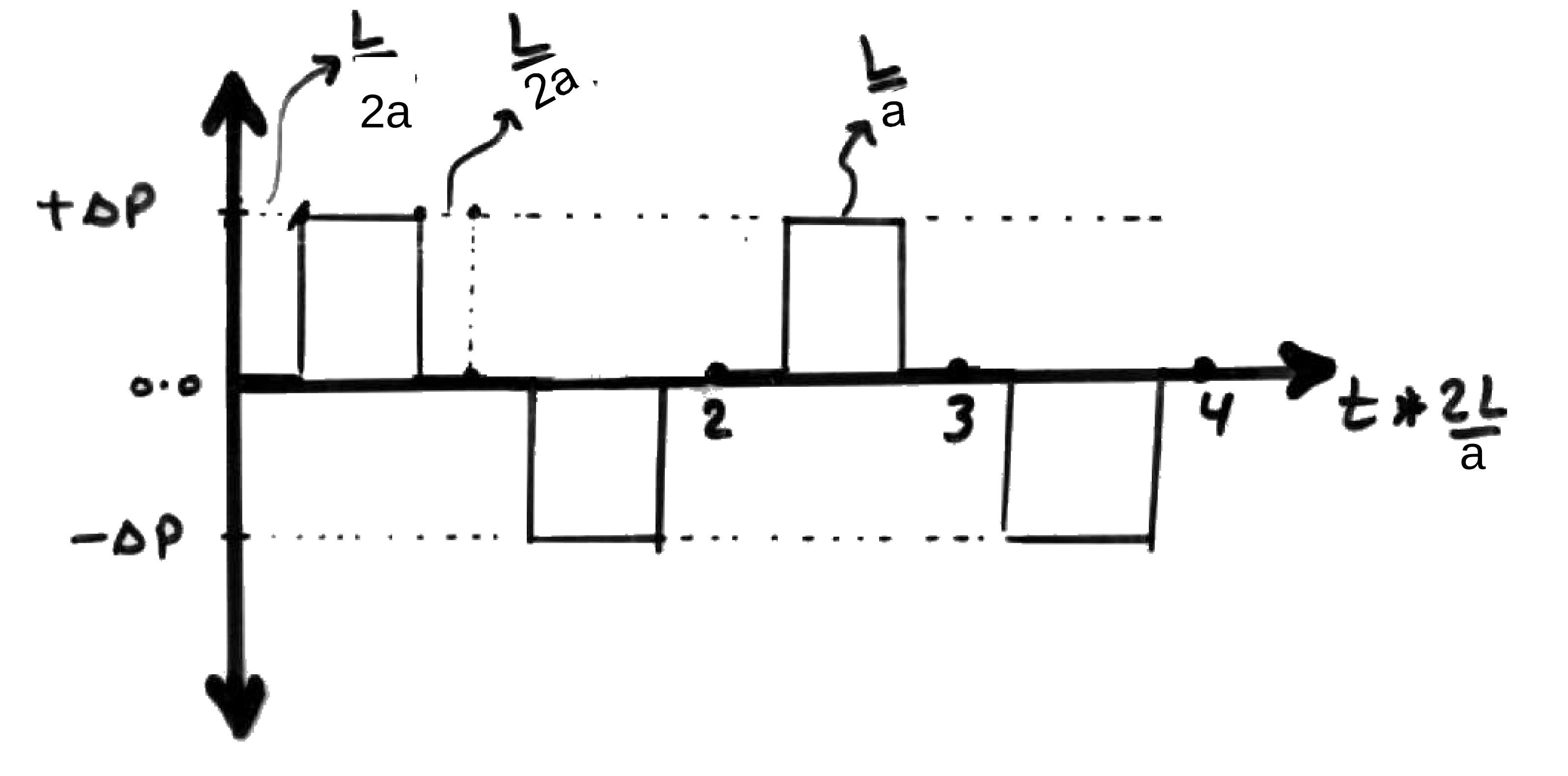
The figure below is the time history for a point "A" just to the left of the valve. The friction losses in the pipe are neglected.



Time history for pressure at point "A" (after valve closure)

4.3 Time history of pressure wave due to water hammer):

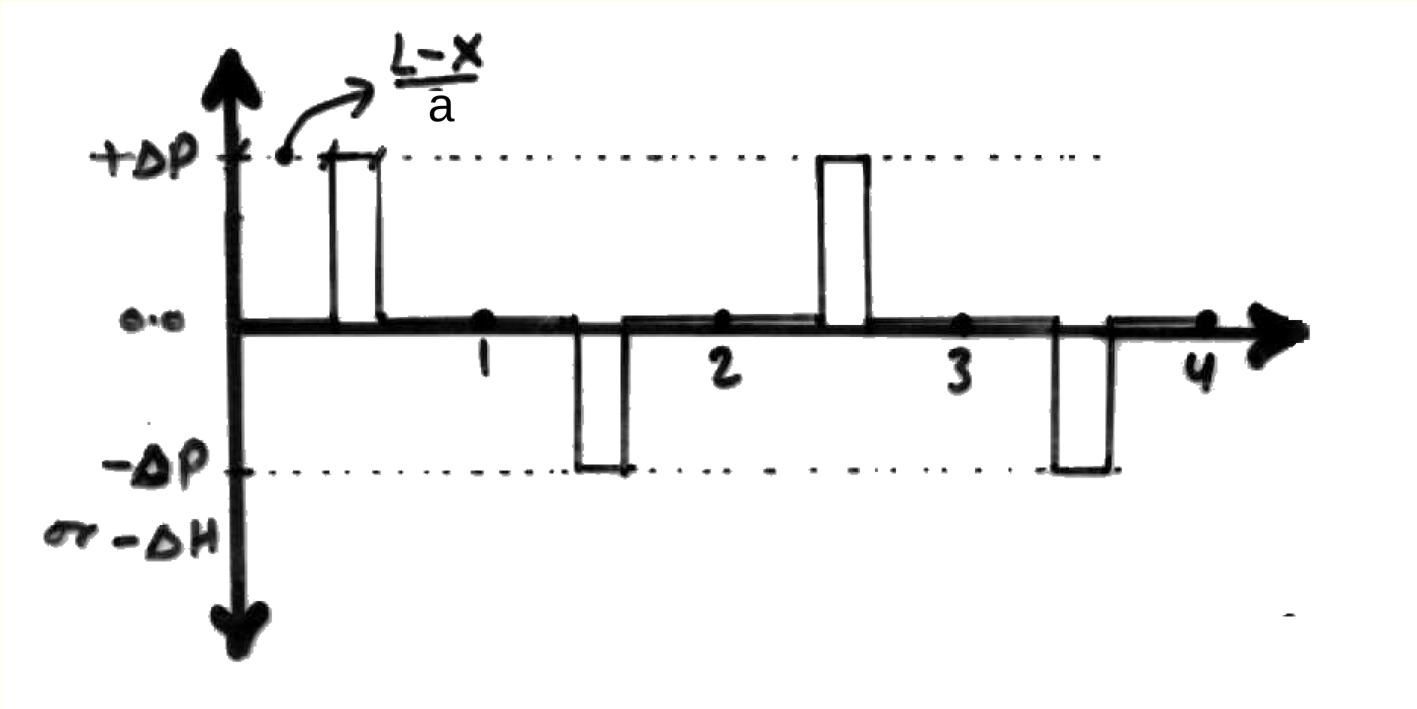
The time history for point ”M“ (at midpoint of the pipe). The friction losses in the pipe are neglected.



Time history for pressure at point “M” (after valve closure)

4.3 Time history of pressure wave due to water hammer):

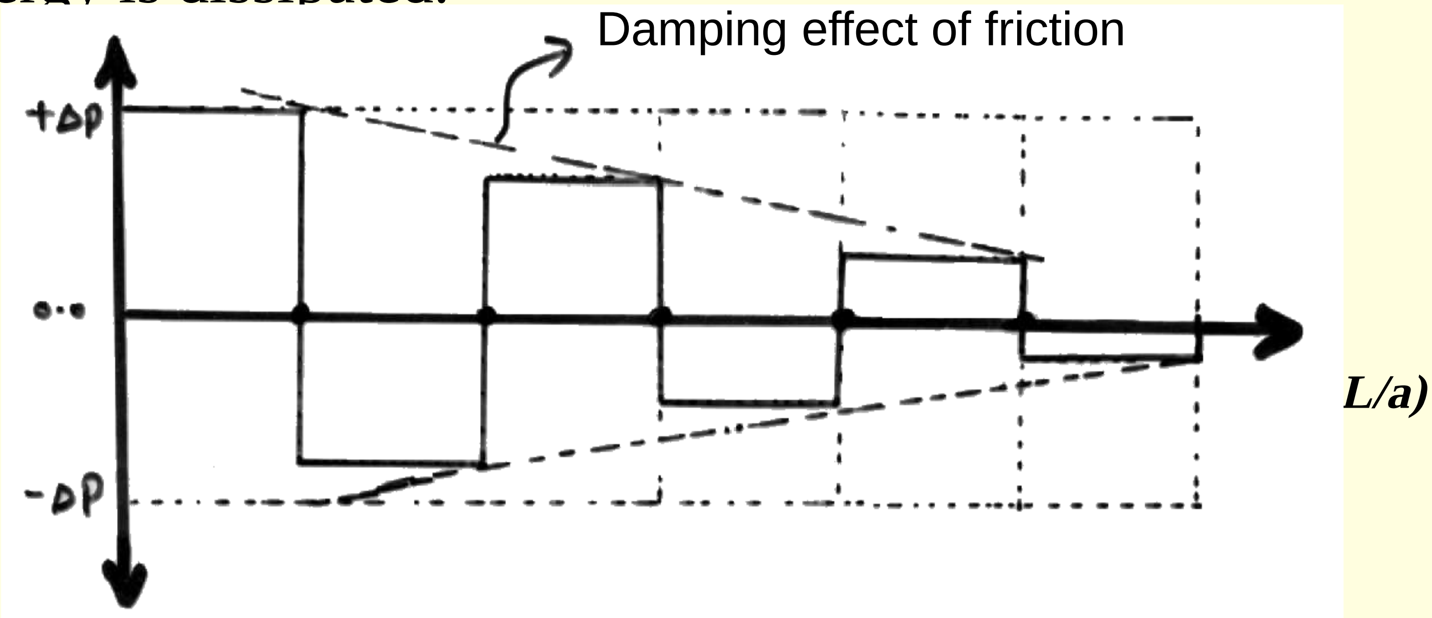
The time history for point B (at a distance x from the reservoir) The friction losses in the pipe are neglected.



Time history for pressure at point “B” (after valve closure)

4.3 Time history of pressure wave due to water hammer):

In real practice friction effects are considered and hence a damping effect occurs and the pressure wave dies out, i.e.; energy is dissipated.



Time history for pressure at point "A" (after valve closure) when friction losses are included.

4.4 water hammer in pumping mains

- Water hammer due to power failure:

As illustrated previously, one of the serious events that produce water hammer in delivery pipes of pumping stations is power failure.

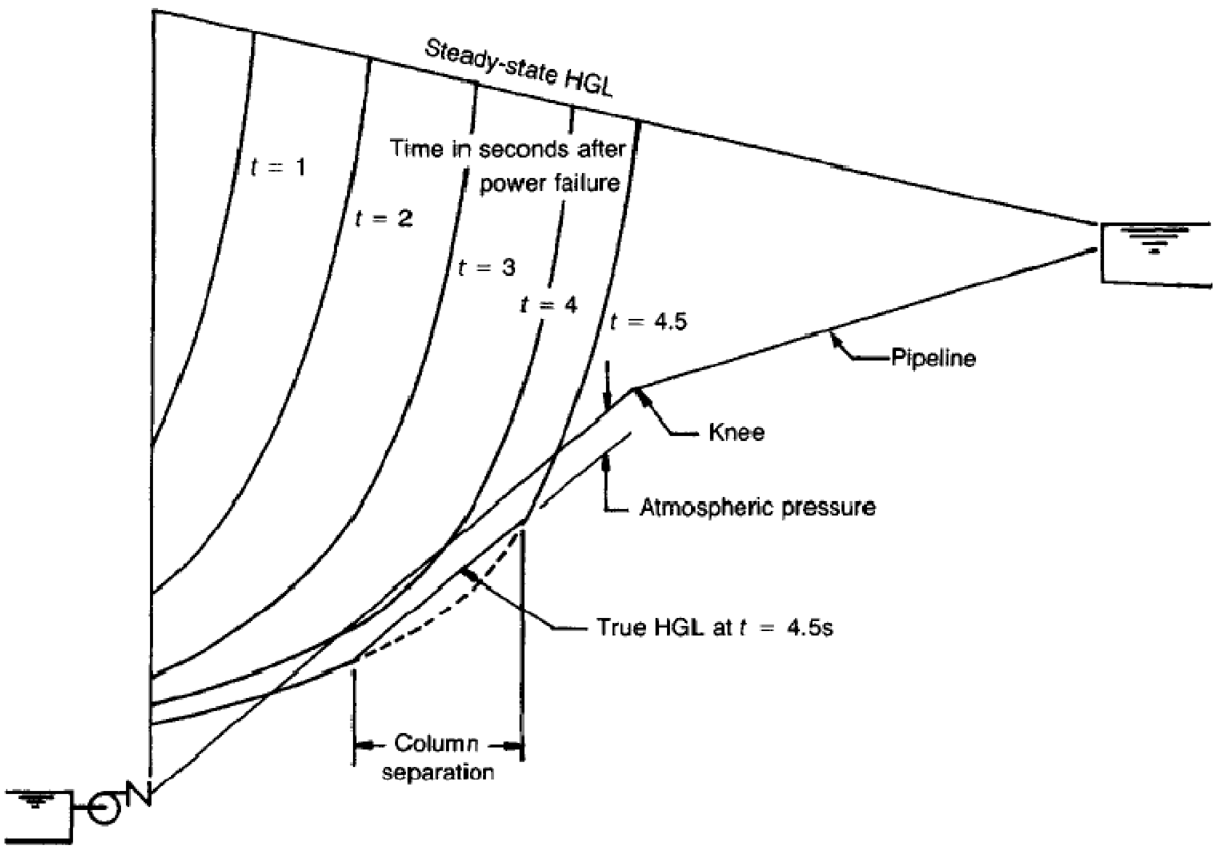


Figure 6-8. Successive hydraulic gradelines following power failure. Adapted from Watters [4, p. 271].

4.4 water hammer in pumping mains

- Column separation due to power failure:

  - What do we mean by column separation?

-After power failure a wave of negative pressure "down surge" occurs.

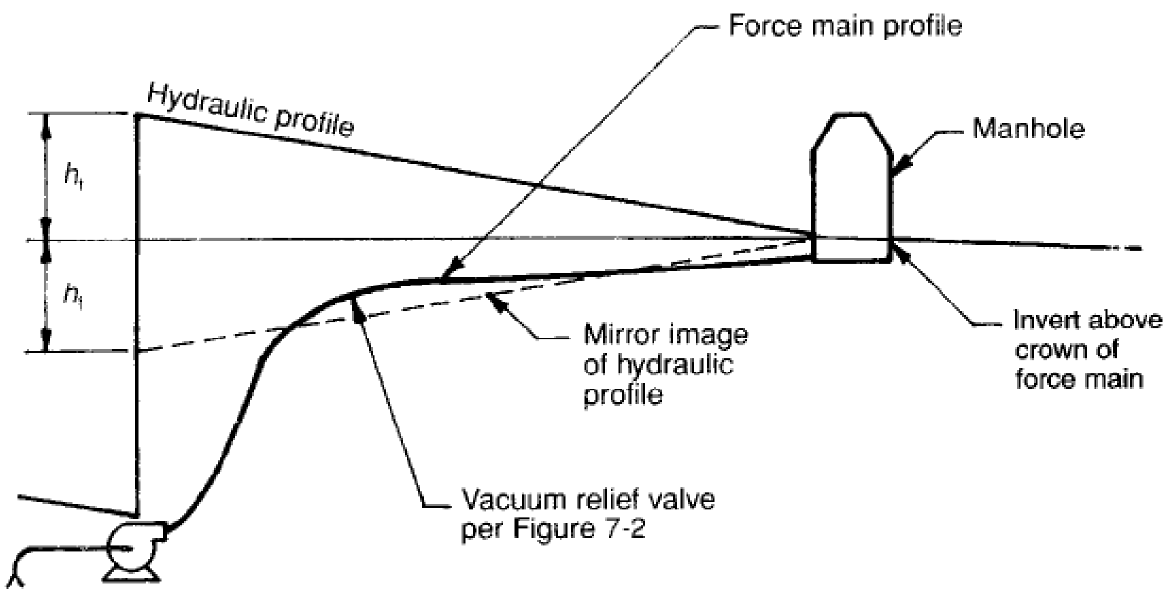
- If the hydraulic profile intersects the pressure pipe and falls below it for a pressure less or equal to the vapor pressure, water vapor starts to accumulate And large air pockets are formed.

-The air pocket separates the water column in the pipe into to parts, upstream column and down stream column. This phenomena is called column separation.

-Column separation is dangerous when the two water columns join again with huge collision when the reflected wave comes back. This collision produce an upsurge pressure that is much more than the first water hammer impulse calculated from the equation discussed previously $\boxed{\Delta H = \frac{aV_0}{g}}$.

4.4 water hammer in pumping mains

- Column separation due to power failure:
  - How to predict column separation?



To predict the occurrence of negative pressure and column separation due to water hammer draw a mirror image of the normal hydraulic profile . If the mirror HGL falls below the pipe for a distance “d “ = $h_{atmospheric} - h_{vapor}$, thin column separation will occur.

4.4 water hammer in pumping mains

- Graphical solution to calculate the water hammer surge after power failure:

A. water hammer without column separation:

The figure in the next slide illustrates the surge development in a pumping main after power failure. The following formulas are needed to understand the method:

$$\tan \Lambda = \frac{a}{gA_p}$$

: Λ (Lambda) is the angle of inclination of the surge line PR, A_p is the cross section area of the pressure line.

$$A_{0-1}$$

: the subscripts 0-1 means the situation at point A between time 0 and time 1, the same notation is used for point B. Time 1 in this example is = L/a.

$$C_p = \frac{Q_0}{2H_0} \tan \Lambda$$

: C_p is the pipe constant, (Q_0,H_0) is the operating point before pump failure.

If C_p is <0.50 then the maximum pressure at the pump is positive as this case. If Cp is > 0.50 then the maximum pressure at the pump is negative and column separation will occur as the next case explained in B.

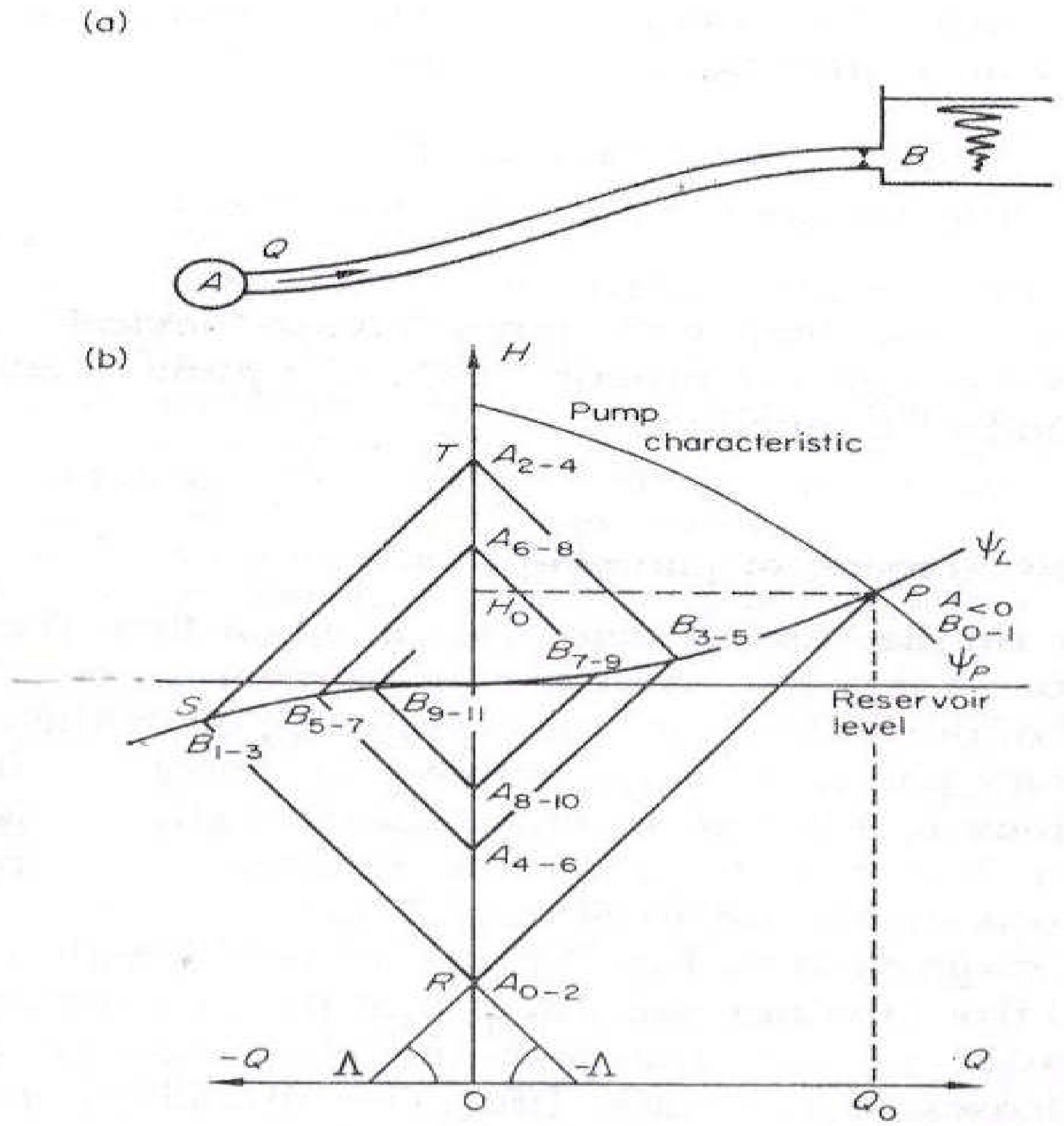

(a)
Q
A
B
(b)
H
Pump characteristic
T
A2-4
A6-8
H0
P
A<0
B0-1
ψL
ψP
B3-5
B7-9
S
B9-11
B5-7
B1-3
Reservoir level
A8-10
A4-6
R
A0-2
-Q
Λ
-Λ
Q
O
Q0

FIG. 7.1

4.4 water hammer in pumping mains

- Graphical solution to calculate the water hammer surge after power failure:

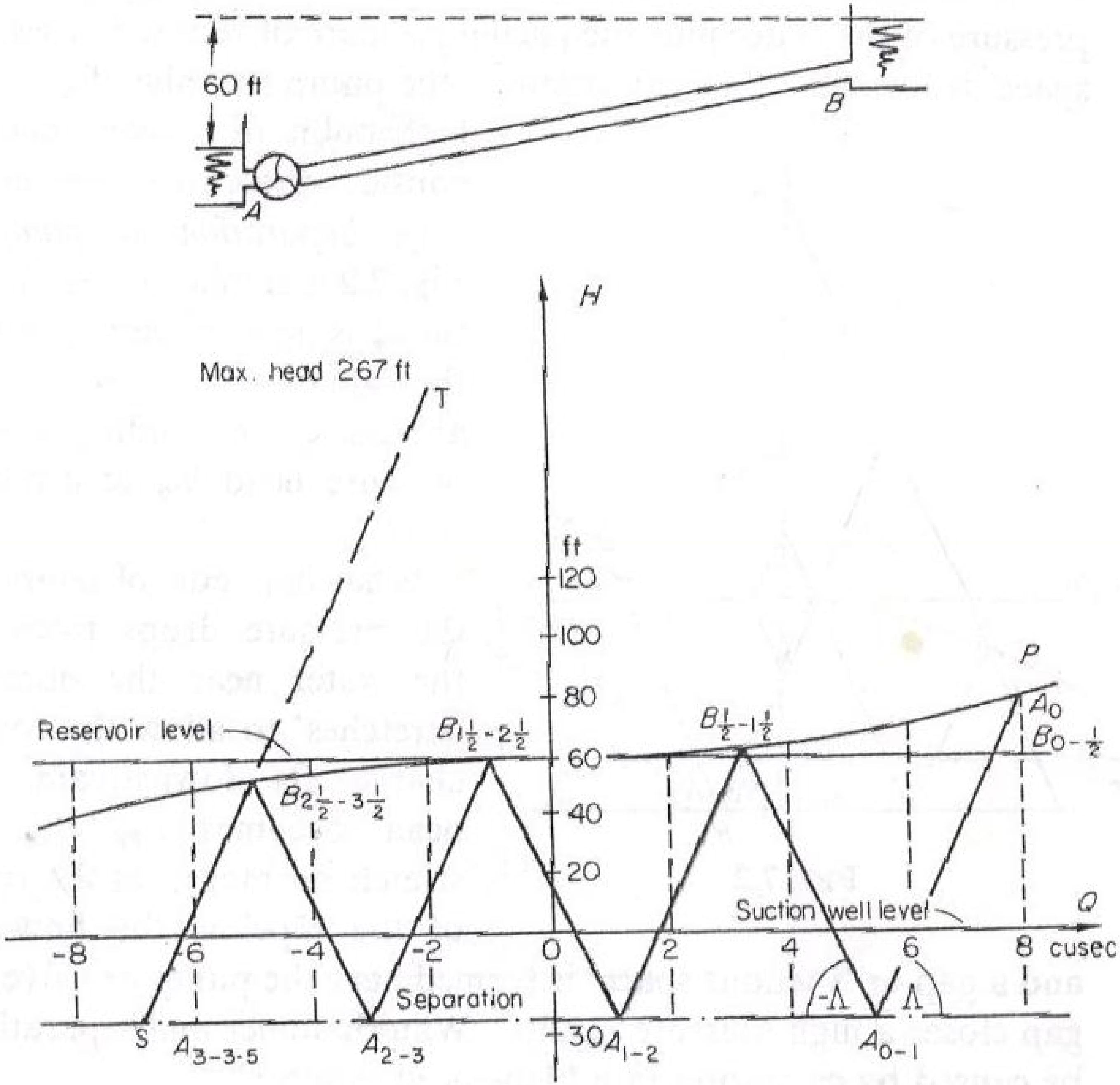
B. water hammer with column separation:

As indicated in Case A if Cp is > 0.50 then column separation occurs. The following example illustrates this case.

Example

Draw a surge diagram to obtain the maximum pressure head at the pump described below when the flow is suddenly stopped.

The pump is placed in a dry well adjacent to the suction well and 4 ft below the water level in the suction well, which is considered as datum. Separation occurs at a negative head of 30 ft. The pipe normally discharges 8 cusecs along a 21 in. diameter main which rises uniformly to a reservoir in which the water level is 60 ft above datum. The length of the main is 16 600 ft and the friction head (assumed proportional to velocity squared) is 19 ft at normal flow. $a = 3320$ ft/sec.



60 ft
A
B
H
Max. head 267 ft
T
ft
120
100
80
60
40
20
-30
Reservoir level
P
A_0
$B_{0-\frac{1}{2}}$
$B_{1\frac{1}{2}-2\frac{1}{2}}$
$B_{\frac{1}{2}-1\frac{1}{2}}$
$B_{2\frac{1}{2}-3\frac{1}{2}}$
Suction well level
Q
-8
-6
-4
-2
0
2
4
6
8 cusec
Separation
-Λ
Λ
S
$A_{3-3\cdot5}$
A_{2-3}
A_{1-2}
A_{0-1}

4.4 Prevention of water hammer in pumping mains

- Methods of preventing or minimizing the water hammer surge:

A. Control Tanks:
   - Open end surge tank or standpipe
   - one way surge tank
   - two way surge tank
   - Air champers (hydropneumatic tanks)

B. Control valves:
   - Vacuum relief valve
   - Air release valves
   - Air release and vacuum relief valves
   - surge relief valves

4.4 Prevention of water hammer in pumping mains

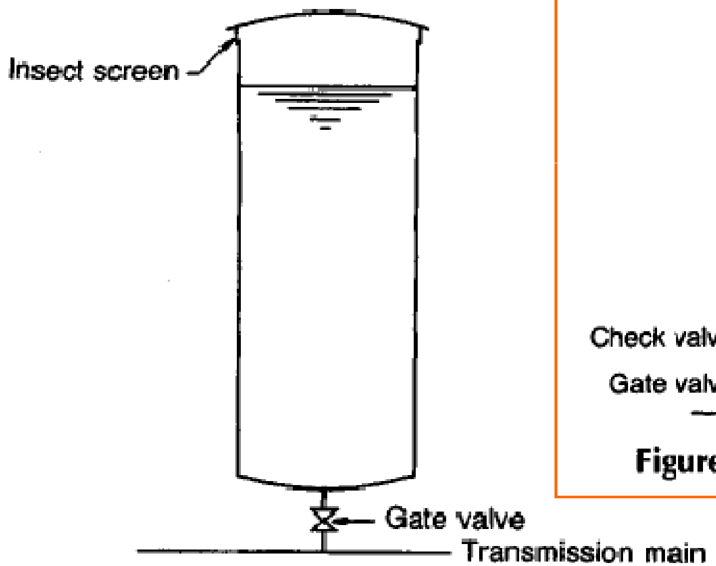


Figure 7-4. Open-end surge tank or standpipe.

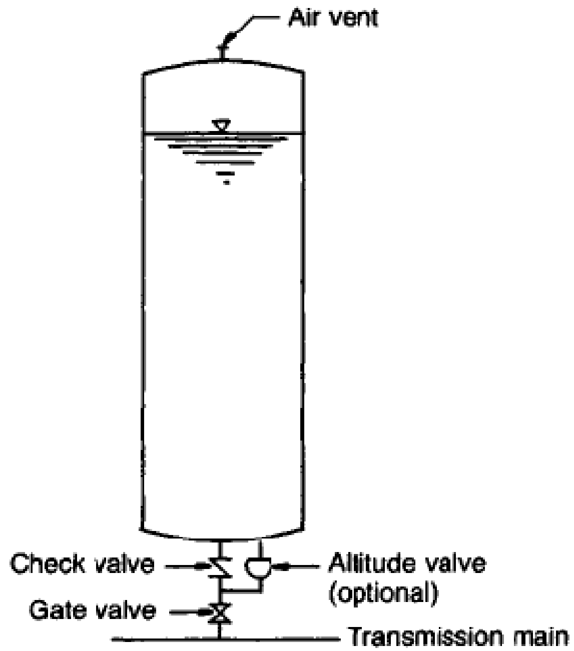


Figure 7-5. One-way surge tank.

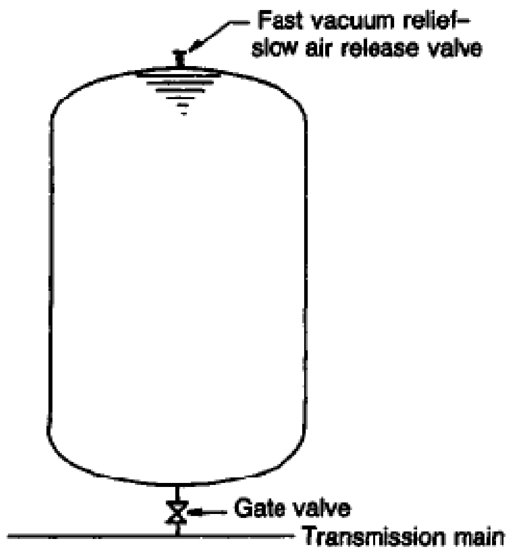


Figure 7-6. Two-way surge tank.

4.4 Prevention of water hammer in pumping mains

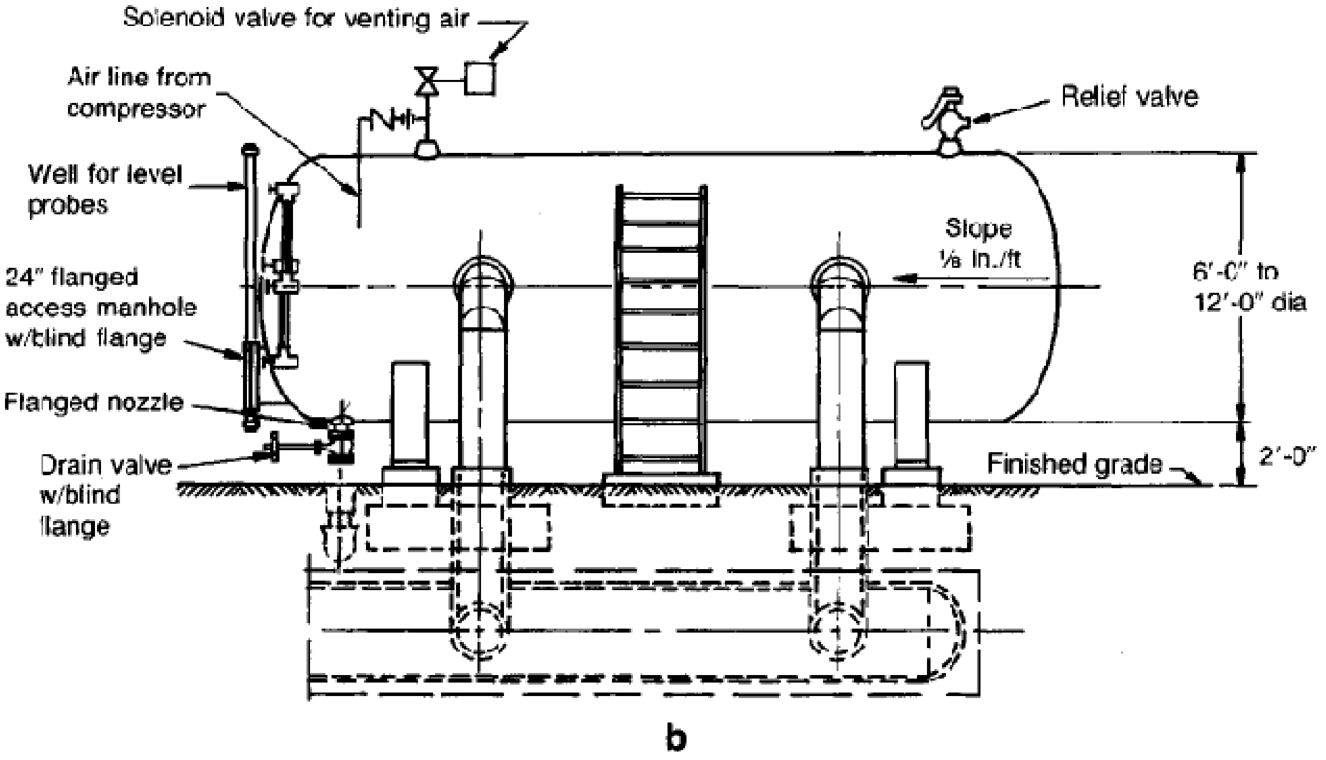


Figure 7-3. Horizontal hydropneumatic tank (air chamber) for clean water service. (a) End elevation; (b) side elevation.

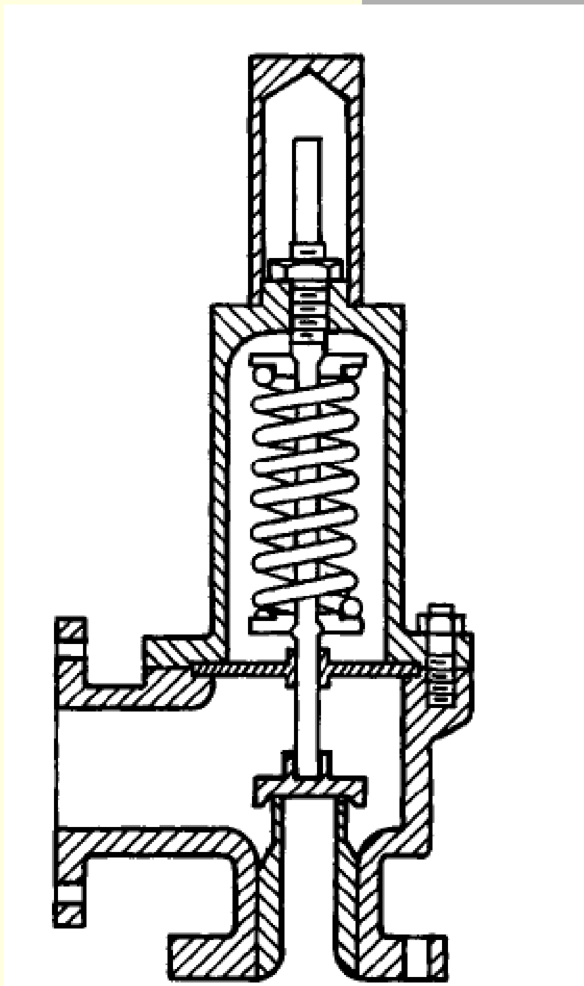
Horizontal Air chamber for water hammer prevention

4.4 Prevention of water hammer in pumping mains

Vertical Air chamber for water hammer prevention

Horizontal Air chamber for water hammer prevention

4.4 Prevention of water hammer in pumping mains



surge relief valve

4.4 Prevention of water hammer in pumping mains

Air release valve

4.4 Prevention of water hammer in pumping mains

Vacuum relief valve

Air /Vacuum relief valve

4.4 Prevention of water hammer in pumping mains

- Using Air champers to prevent water hammer surge:

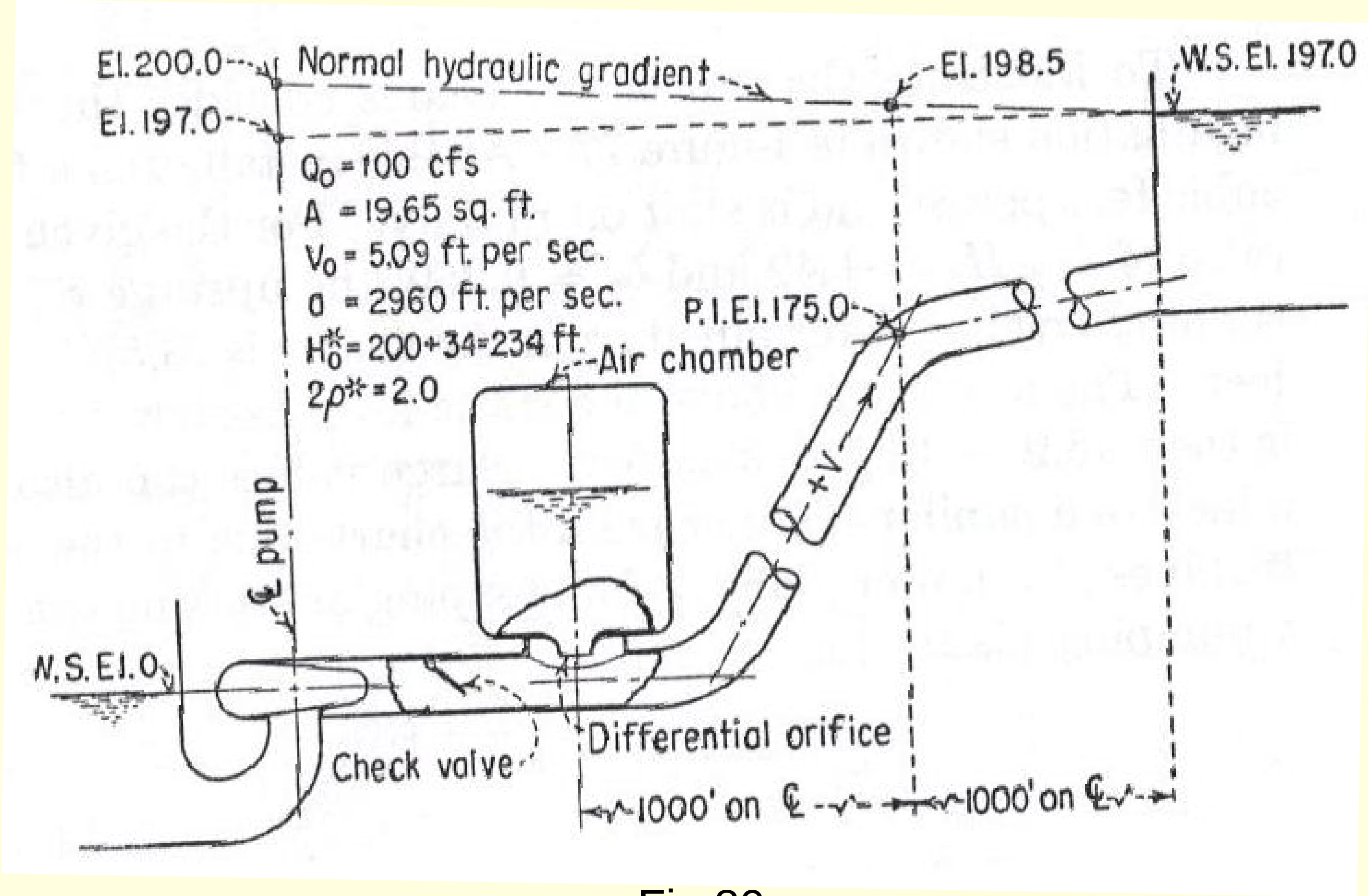


Fig 80

4.4 Prevention of water hammer in pumping mains

Example

Consider the pumping plant installation with an air chamber shown in Figure 80. It is desired to determine a chamber size such that the maximum upsurge in the discharge line adjacent to the pump will not exceed $0.43H_0^*$ and the maximum downsurge at the mid-length will not exceed $0.21H_0^*$. From the charts in Figure 83 it is found that these requirements are met by using the values $K = 0.3$ and $2C_0a/Q_0L = 21$ for which the maximum upsurge at the pump $= 0.27H_0^*$, the maximum downsurge at the mid-length $= 0.21H_0^*$, and the maximum downsurge adjacent to the pump $= 0.32H_0^*$. For this installation the pipe line friction loss for a flow Q_0 is about 3 feet. The differential orifice required at the chamber must then give a head loss for a flow of Q_0 into the chamber of $0.3 \times 234 - 3 = 67$ feet. With $2C_0a/Q_0L$ known, the initial volume of compressed air in the chamber C_0 is 709 cubic feet and the minimum volume for the whole air chamber C' as determined from Equation (78) is 1040 cubic feet.

- Charts to determine the size of the air champers

$$\rho^* = \frac{aV_0}{2gH_0^*}$$

K = losses percent in the air champer orifice

C_0 = *The initial compressed air volume in the air champer*

$$C' = \frac{C_0 H_0^*}{H_{min}^*}$$: air chamber volume

$$H_{min}^* = H_0^* - down\ surge\ adjacent\ to\ the\ pump$$

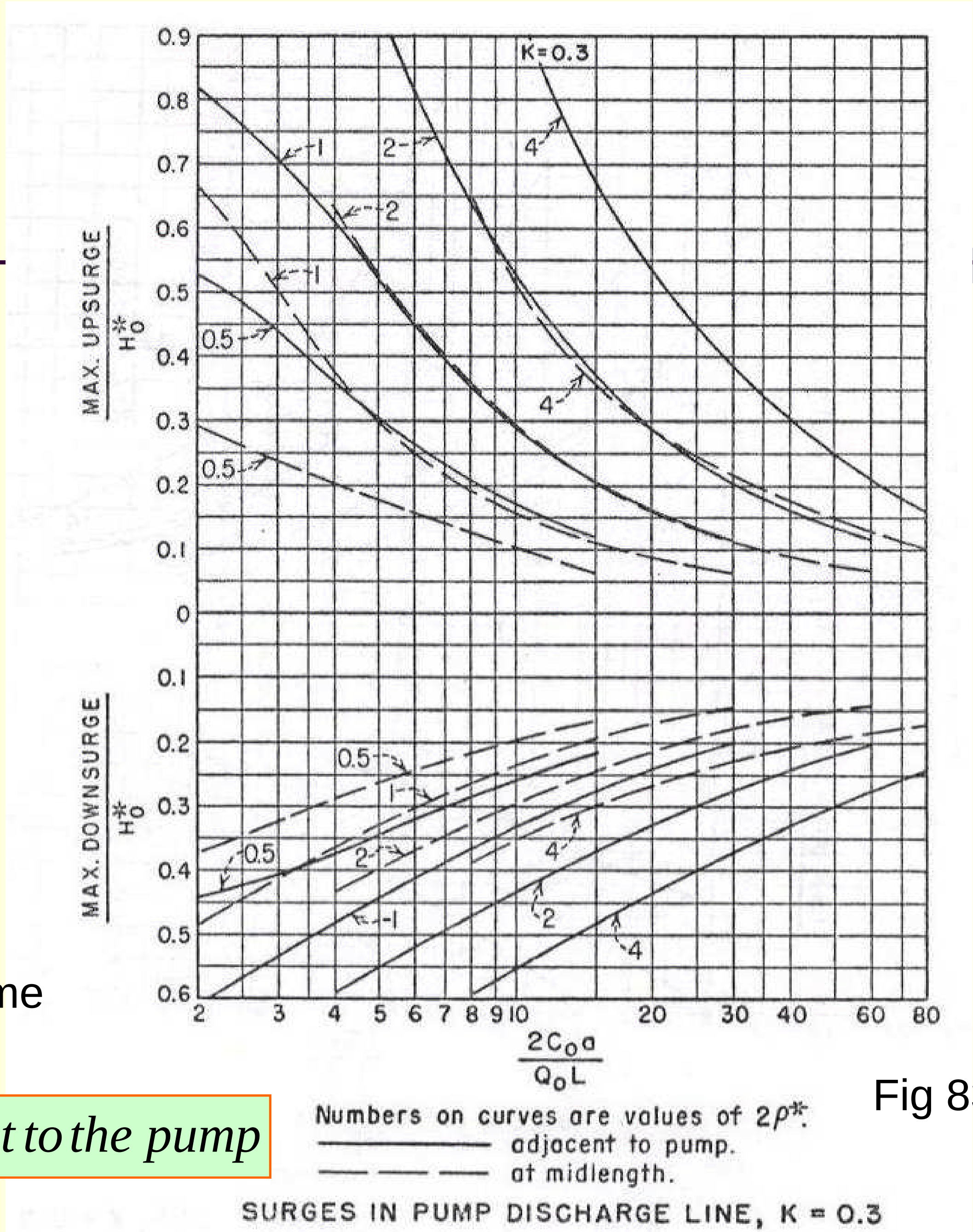


Fig 83

4.4 Prevention of water hammer in pumping mains

- Using computer software for water hammer analysis:

There are many computer software for water hammer analysis such as:

- Hammer (Bently company)
- HiTran
- AFT
- Pipe Expert

These software are not free , they are very expensive. However, you can Download DEMO versions and get good benefit out of it. You can also search For free software that I do not know.

5.1 General introduction

- Main components of WWPS:

  - Bar screen
  - Grit removal
  - Wet well or wet well + dry well
  - Electricity distribution and control room (MDB + PLC)
  - Transformer room
  - Stand by generator and its fuel tank
  - Guard room and its services (kitchen + showers and toilet)
  - Pressure pipes and control valves
  - Fence and landscaping

Pumping Stations Design

For Infrastructure Master Program
Engineering Faculty-IUG

Lecture 5: Design of wastewater pumping stations

Dr. Fahid Rabah
Water and environment Engineering
frabah@iugaza.edu

5.1 General introduction

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  - Pressure pipes and control valves
  - Fence and landscaping

Lecture 5: Design of wastewater pumping stations

5.2 Typical layout of WW pumping stations

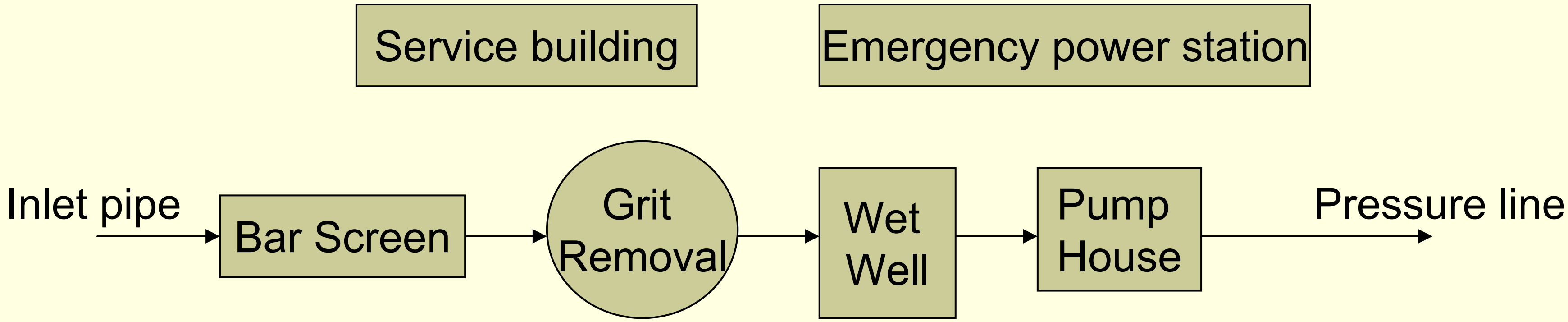


Fig 5.1 Typical Layout of Wastewater Pumping Station

5.2 Typical layout of WW pumping stations

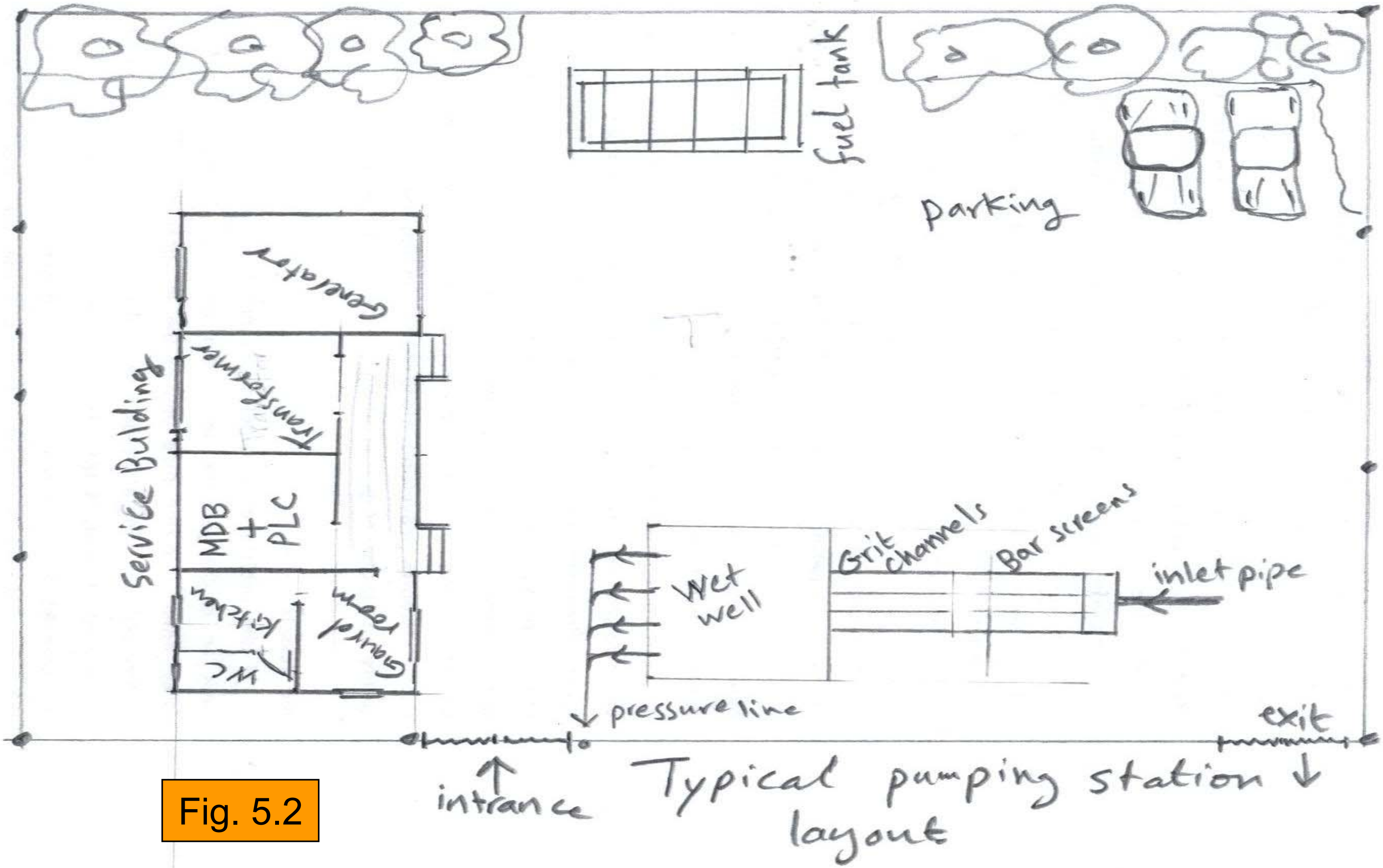


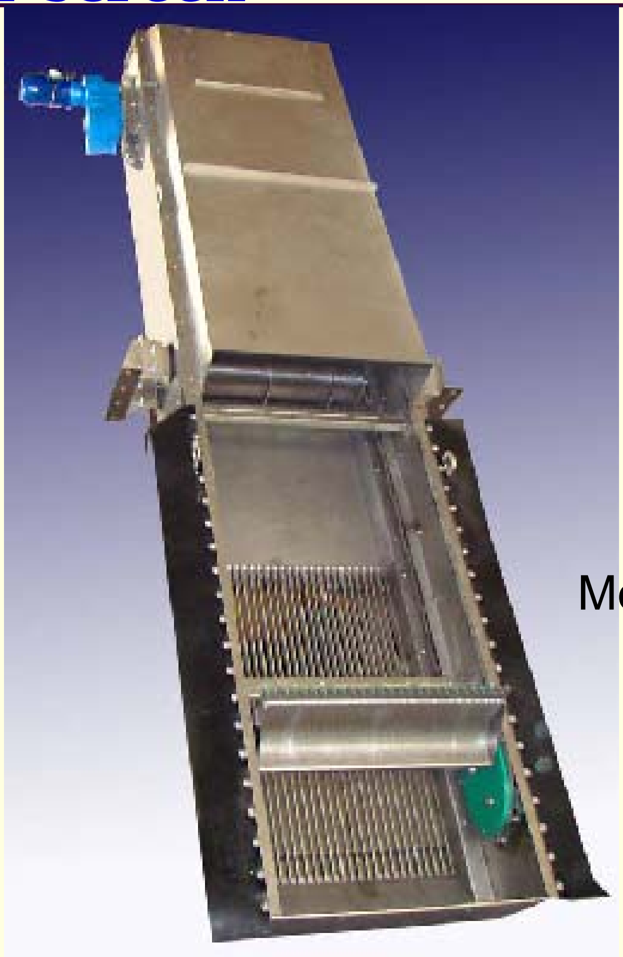
Fig. 5.2

5.3 main components of WW pumping stations

Bar screen

Fig. 5.3

Manual Bar Screen



Mechanical Bar Screen

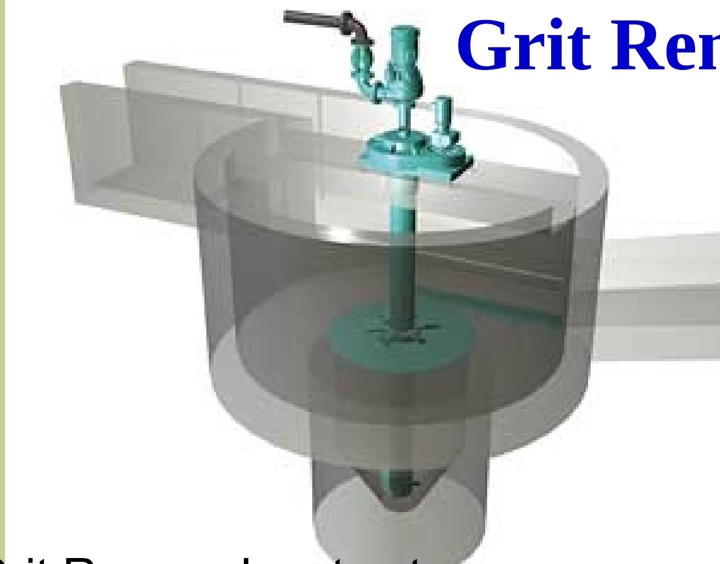
5.3 main components of WW pumping stations

Bar screen

Fig. 5.4 Mechanical Bar Screen

5.3 main components of WW pumping station

Grit Removal



Grit Removal vortex type

Fig. 5.5

5.3 main components of WW pumping station

Wet Well

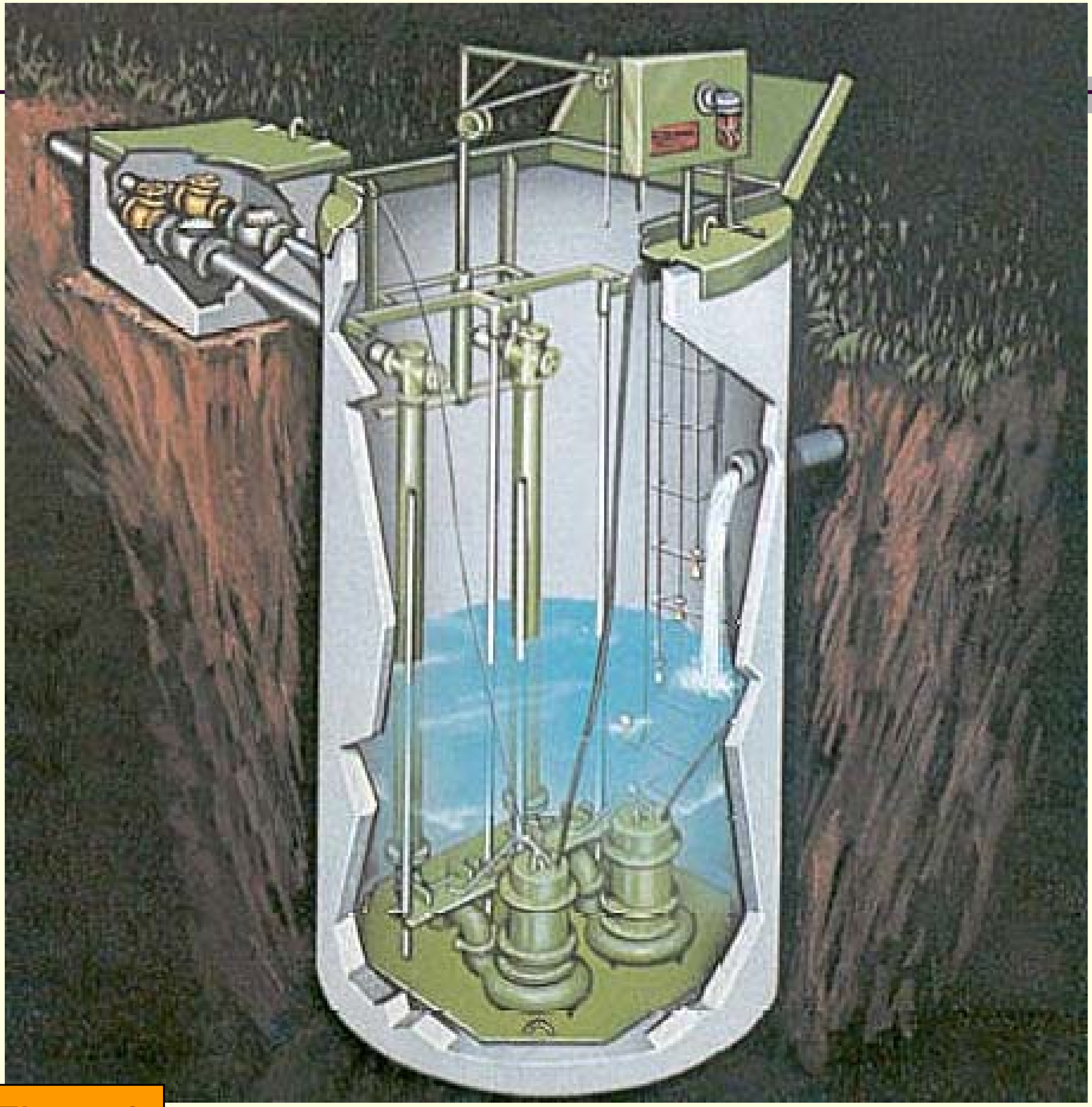


Fig. 5.6 Wet well Pump House

5.3 main components of WW pumping station

Fig. 5.7

5.3 main components of WW pumping station

Dry Well

Fig. 5.8 Dry room for dry pumps

5.3 main components of WW pumping station

Fig. 5.9 Dry room for dry pumps

5.3 main components of WW pumping station

Dry Well

Fig. 5.10 Dry room for dry pumps

5.3 main components of WW pumping station

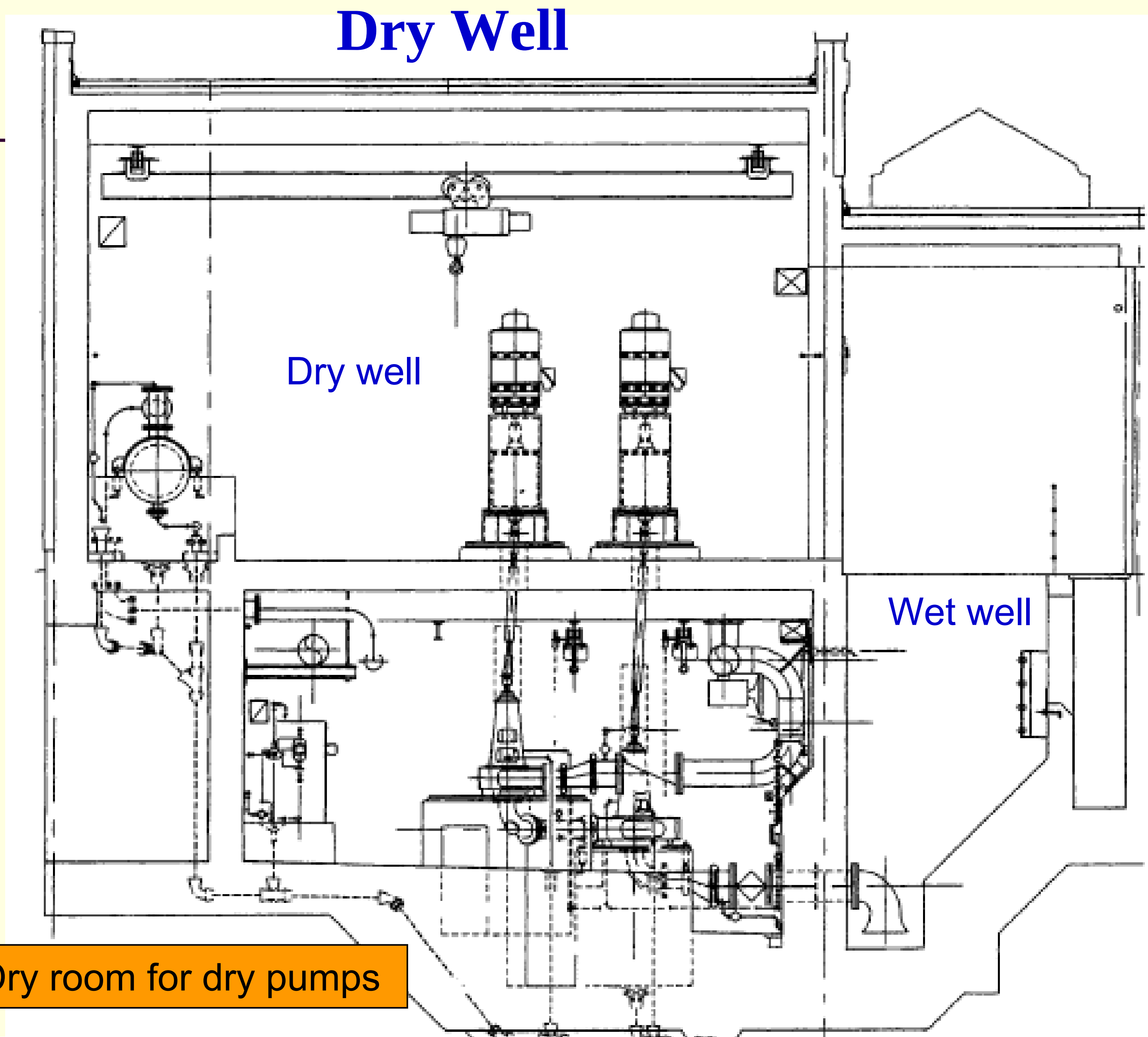


Fig. 5.11 Dry room for dry pumps

5.3 main components of WW pumping station

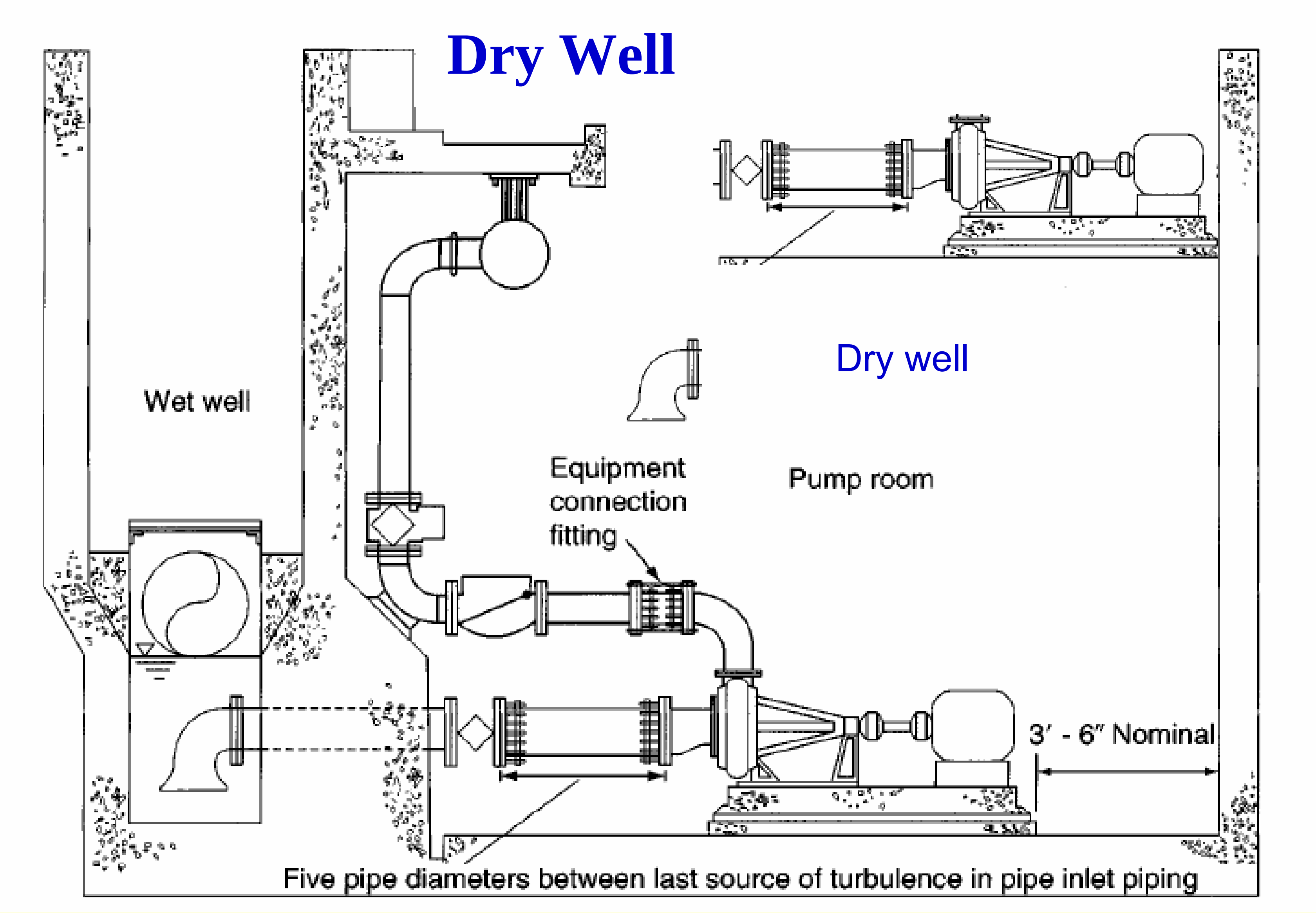


Fig. 5.12 Dry room for dry pumps

5.3 main components of WW pumping station

Dry Well

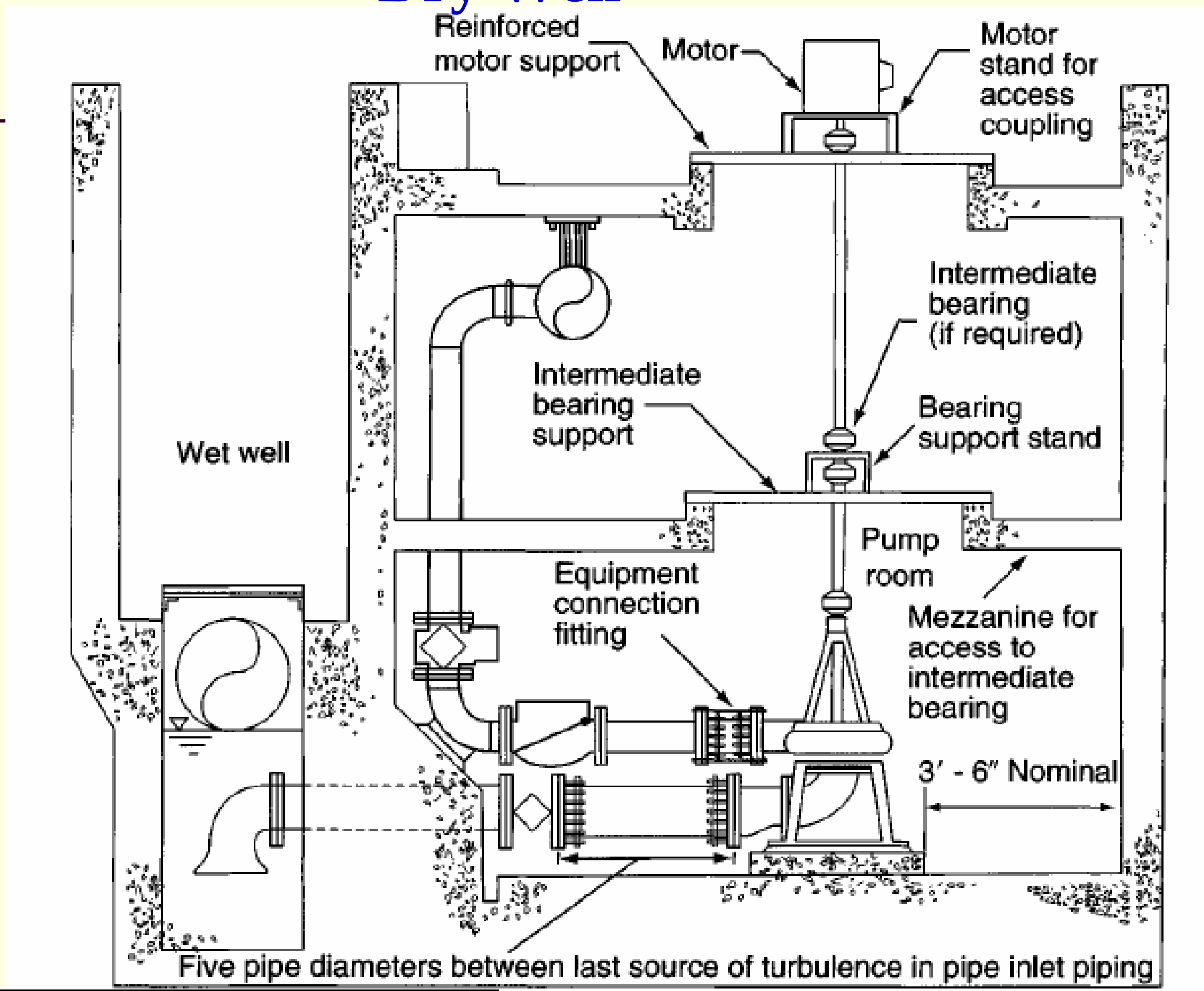


Fig. 5.13 Dry room for dry pumps

5.3 main components of WW pumping station

Fig. 5.14 Electricity distribution and control room (MDB + PLC)

5.3 main components of WW pumping station

Stand by Generator

Fuel tank

Stand by generator

Fig. 5.15 Standby generator and its fuel tank

5.3 main components of WW pumping station

Transformer

Transformer room

Fig. 5.15 Electricity Transformer

5.3 main components of WW pumping station

Service Building

Fig. 5.16 **Service Building**

5.3 main components of WW pumping station

Fig. 5.17 **Pumping station fence and gate**

5.3 main components of WW pumping station

Delivery Pipes , valves and manifold

Fig. 5.18 **Delivery Pipes , valves and manifold in WWPS**

5.4 Design of the main components of WWPS

- After the introduction to the main components of WWPS we will study the design of the following components:

  - Design of the general layout of the WWPS
  - Design of Bar screen channel
  - Design of Grit removal channels
  - Design of Wet well for submersible pumps
  - Design of Wet well and dry well for dry pumps
  - Design of the delivery pipes and the pressure line

5.4 Design of the main components of WWPS

5.4.1 Design of the general layout of the WWPS:

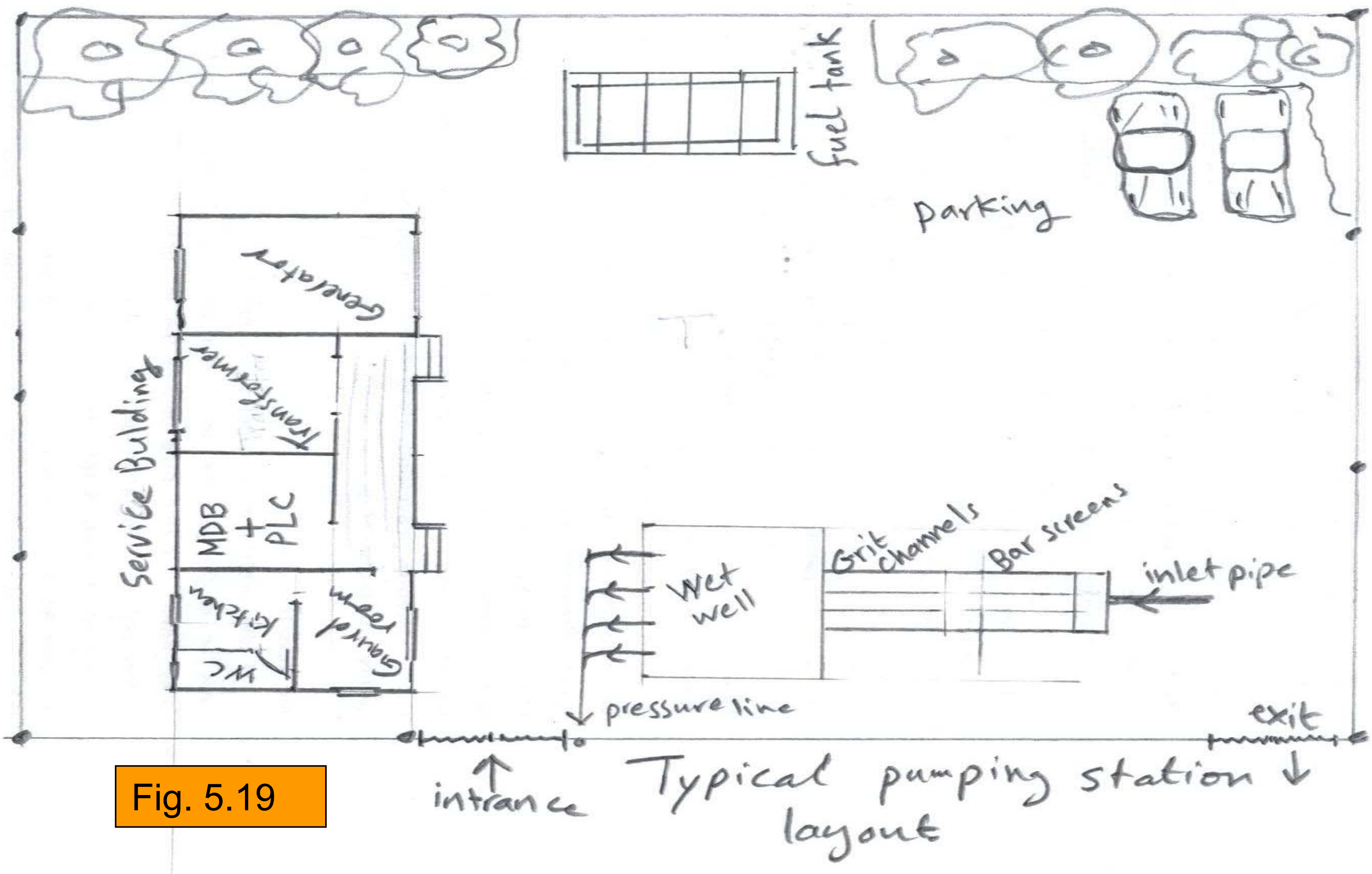


Fig. 5.19

5.4 Design of the main components of WWPS

5.4.2 Design of Bar screens

Screens are used in wastewater treatment for the removal of coarse solids. Screens are either manually or mechanical cleaned.

- **Bar spacing is in range of 2-5 cm**
- **The screen is mounted at an angle of 30-45**
- **Bars are usually 1 cm thick, 2.5 wide**
- **Minimum approach velocity in the bar screen channel is 0.45 m/s to prevent grit deposition.**
- **Maximum velocity between the bars is 0.9m/s to prevent washout of solids through the bars.**

- **Bar spacing is in range of 1.5-4 cm**
- **The screen is mounted at an angle of 30-75**
- **Bars are usually 1 cm thick, 2.5 wide**
- **Minimum approach velocity in the bar screen channel is 0.45 m/s to prevent grit deposition.**
- **Maximum velocity between the bars is 0.9 m/s to prevent washout of solids through the bars.**

5.4 Design of the main components of WWPS

5.4.2 Design of Bar screens

Design of the bar screen channel (Approach Channel)

The cross section of the bar screen channel is determined from the continuity equation:

$Q_d = A_c V_a$

$A_c = Q_d / V_a$

Q_d = design flow, m^3/s

A_c = bar screen cross section, m2

V_a = Velocity in the approach channel, m/s

Usually, rectangular channels are used, and the ratio between depth and width is taken as 1.5 to give the most efficient section.

The head loss through the bar screen

$$H_l = \frac{(V_b^2 - V_a^2)}{2g} \bullet \frac{1}{0.7}$$

H_l = head loss

V_a = approach velocity, m/s

V_b = Velocity through the openings, m/s

g = acceleration due to gravity, m/s^2

Example

A manual bar screen is to be used in an approach channel with a maximum velocity of 0.64 m/s, and a design flow of 300 L/s. the bars are 10 mm thick and openings are 3 cm wide. Determine:

The cross section of the channel, and the dimension needed
The velocity between bars
The head loss in meters
The number of bars in the screen

1. $A_c = Q_d/V_a = 0.3/0.64 = 0.47$ m2

$A_c = W \times 1.5W = 1.5\,W \times W$

W = 0.56 m, Depth (d) = 1.5 W = 0.84 m

$$\boxed{A_{net} = A_c \frac{S_c}{S_c + t_{bar}}}$$

= 0.84 x 0.56 (3/3+1) = 0.35 m^2

From continuity equation: $V_a A_c = V_b A_{net}$

V_b= 0.64 x 0.56 x 0.84/0.35 = 0.86 m/s < 0.9 m/s ok

3. Head loss: $$\boxed{H_l = \frac{(V^2{}_b - V^2{}_a)}{2g} \bullet \frac{1}{0.7}}$$

$$H_l = \frac{(0.86^2 - 0.64^2)}{2 \bullet 9.81} \bullet \frac{1}{0.7} = 0.024 \text{ m}$$

4. $n\, t_{bar} + (n-1)S_c = W$

n x 1 + (n-1) x 3= 56

n= 14.75 = 15

5.4.3 Design of rectangular Grit removal channel

Design a set of rectangular grit basins with proportional flow weir for a plant which has a peak flow of 80,000 m^3/day, an average flow of 50,000 m^3/day and a minimum flow of 20,000 m^3/day. Use three basins. Make the peak depth equal to the width. The design velocity (V_h) is 0.25 m/s, Vs = 0.021 m/s

Solution

The peak flow per channel will be 80,000/3 = 26,666 m^3/day = 0.31 m^3/s.

The average flow per channel will be 50,000/3= 16,666 m^3/day = 0.19 m^3/s.

The minimum flow per channel will be 20,000/3= 6,666 m^3/day = 0.077 m^3/s.

$A = Q/V_h$

$A_{peak} = 0.31/0.25 = 1.24\ m^2$.

The water depth D = W = 1.114

Take W = 1.10 m.

Then D = 1.13 m at peak flow

Take the channel depth = 1.50 m

This would provide a freeboard of 37 cm at peak flow.

$L = D\ (V_h/V_s)$

The length of the channel = 1.15 (0.25/0.021)

= 13.7 m

The weir must be shaped so that:

$Q = 8.18 * 10^{-6}\ wy^{1.5}$

_W_ = width of the proportional weir at depth y.

W (mm)	Y (mm)	Q (m^3/min)
120.50	280	4.62
76.70	691	11.4
66.90	909	15.0
59.90	1130	18.6

Note: $Y= Q/(V_h*\mathbf{W})$

5.4.3 Design of rectangular Grit removal channel

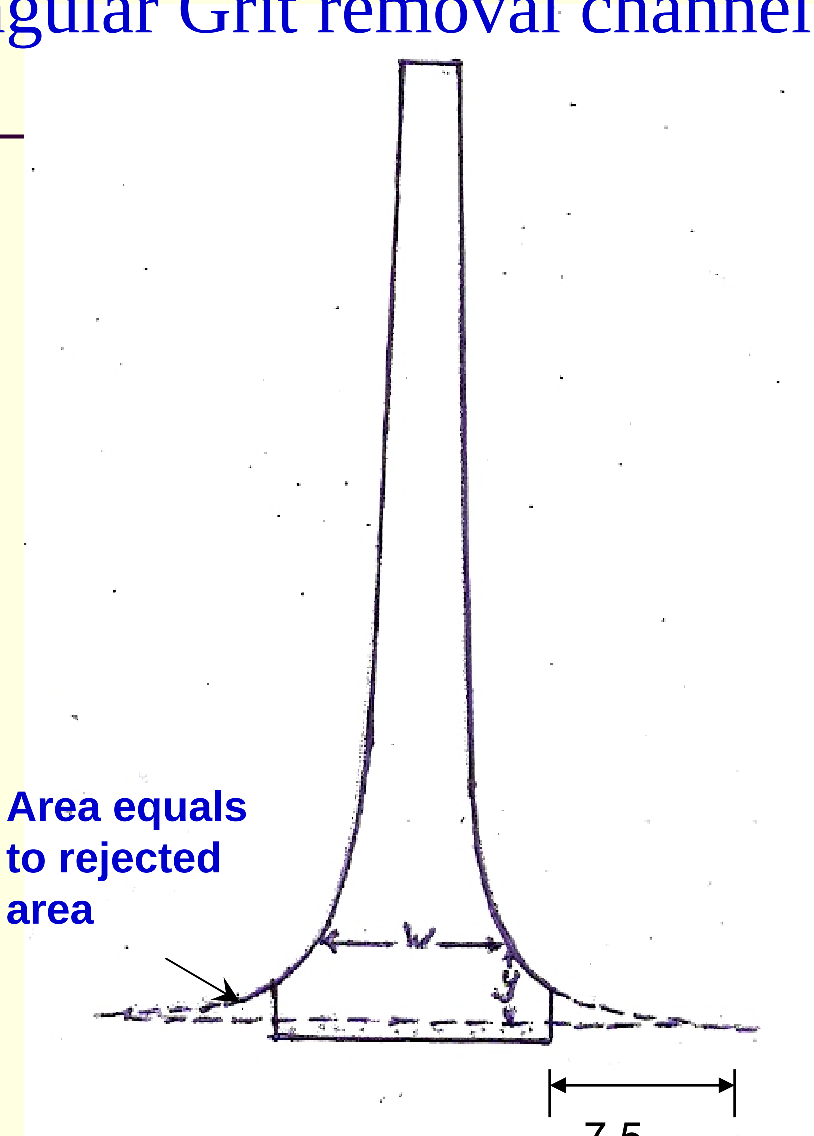


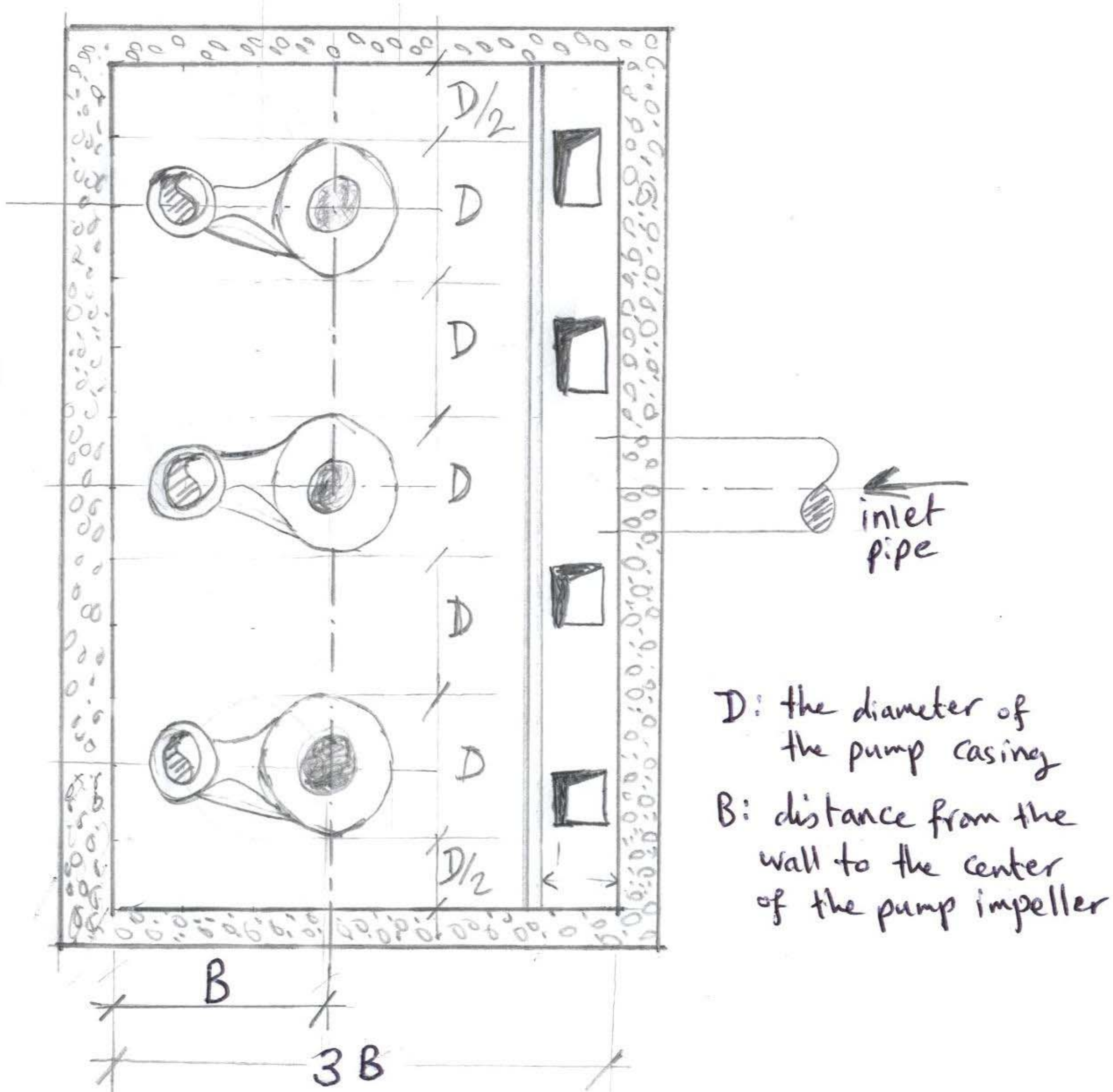
Fig 5.20 Proportional flow weir for use with rectangular grit chamber

5.4 Design of the main components of WWPS

5.4.4 Design of the wet well for submersible pumps

- Design of horizontal cross section of the wet well:

• The dimensions of the horizontal cross section of the wet well depends on the pump size and the recommended distance between the pumps.
• Figure 5.21 shows the recommended dimensions of the horizontal cross section of the wet well . Note that the diameter of the pump casing "D" is the dimension That decides the length of the wet well.
• The second needed distance is B, the distance between the wet well wall and the center of the pump. The distance B is necessary to decide the width of the wet well. The distances D and B is taken from the pump catalogs as those shown in figures 15.23 and 15.22.
• An inlet channel inside the wet well opposite to the inlet pipe is usually needed to As energy breaker to prevent turbulence in the wet well. The width of the channel is usually 50 to 70 cm and its height is equal to the inlet pipe diameter plus 30 cm.
• Figure 15.25 shows a horizontal plan of the wet well at a level above the concrete frame.



Horizontal section of the wet well Showing the typical dimenssions.

Fig. 5.21

5.4 Design of the main components of WWPS

5.4.4 Design of the wet well for submersible pumps

❑ Design of the vertical cross section of the wet well:

• Figure 5.24 shows a typical vertical cross section of the wet well.

• The LWL is taken from the manufacturer catalog of the pump. For example, the LWL for the KSB is indicated as the distance "R" in figure 15. 22.

• The HWL is calculated from the active volume as will be indicated later. The active volume is the volume between the LWL and the HWL.

• The HWL is taken as the level of the invert level of the inlet pipe as indicated in figure 15.22. The distance from the HWL to the top of the roof of the wet well depends on the ground level. It is usually preferred to take the level of the top of roof the of the wet well as 50 cm higher than the ground level.

• As shown in figure 15.22, a hoist and chain should be located above the pump to left it for maintenance. For this purpose a concrete frame should be built to hang the electrical hoist on it. The capacity of the hoist is decided according to the weight of each pump. The weight of the pump is taken from the manufacturer catalog as in figure15.23.

• A rectangular benching is needed to prevent solids from settling away from the pumps. The dimension of this triangle is shown in figure 15.24.

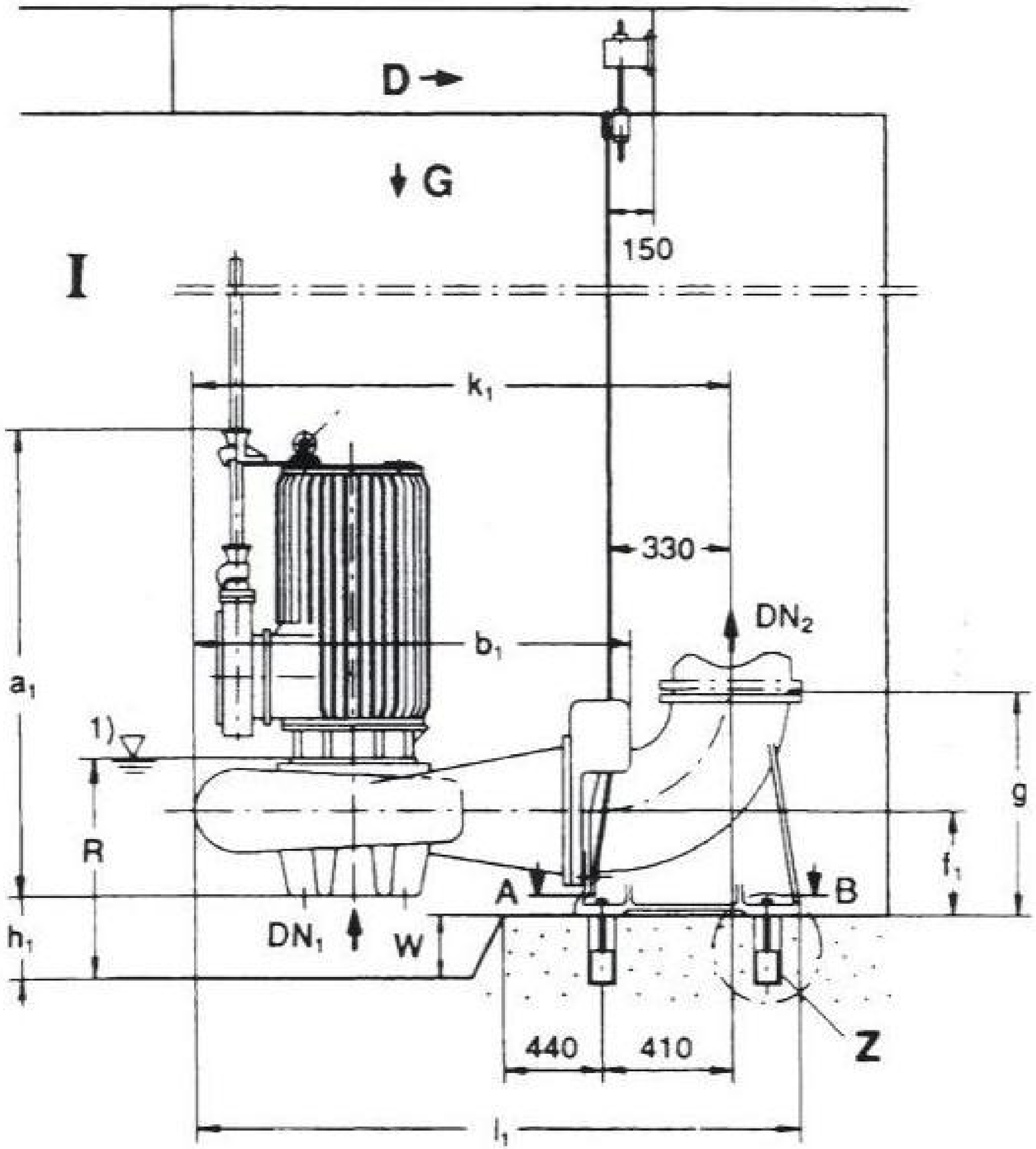
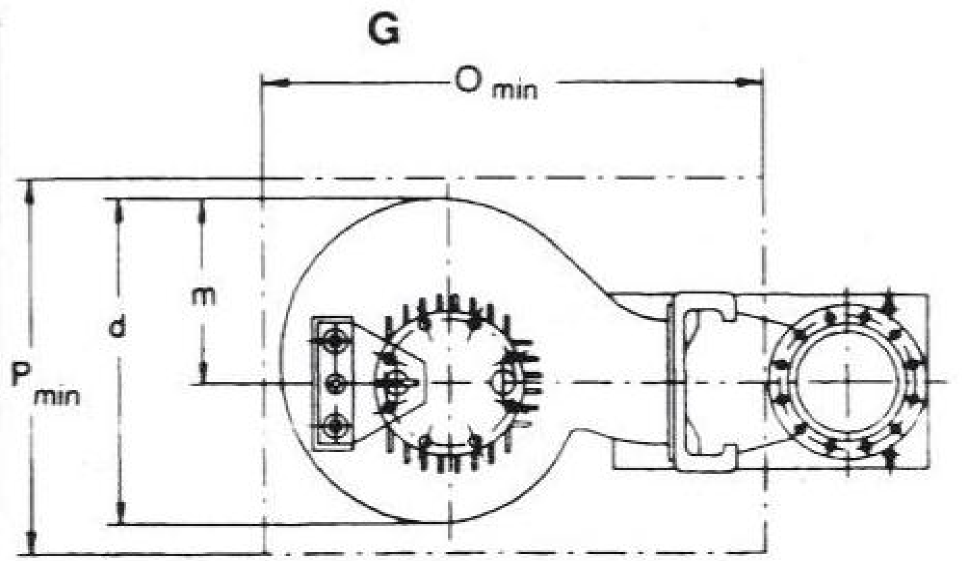


Fig. 5.22 Typical dimensions of a submersible KSB pump needed for the design of the wet well



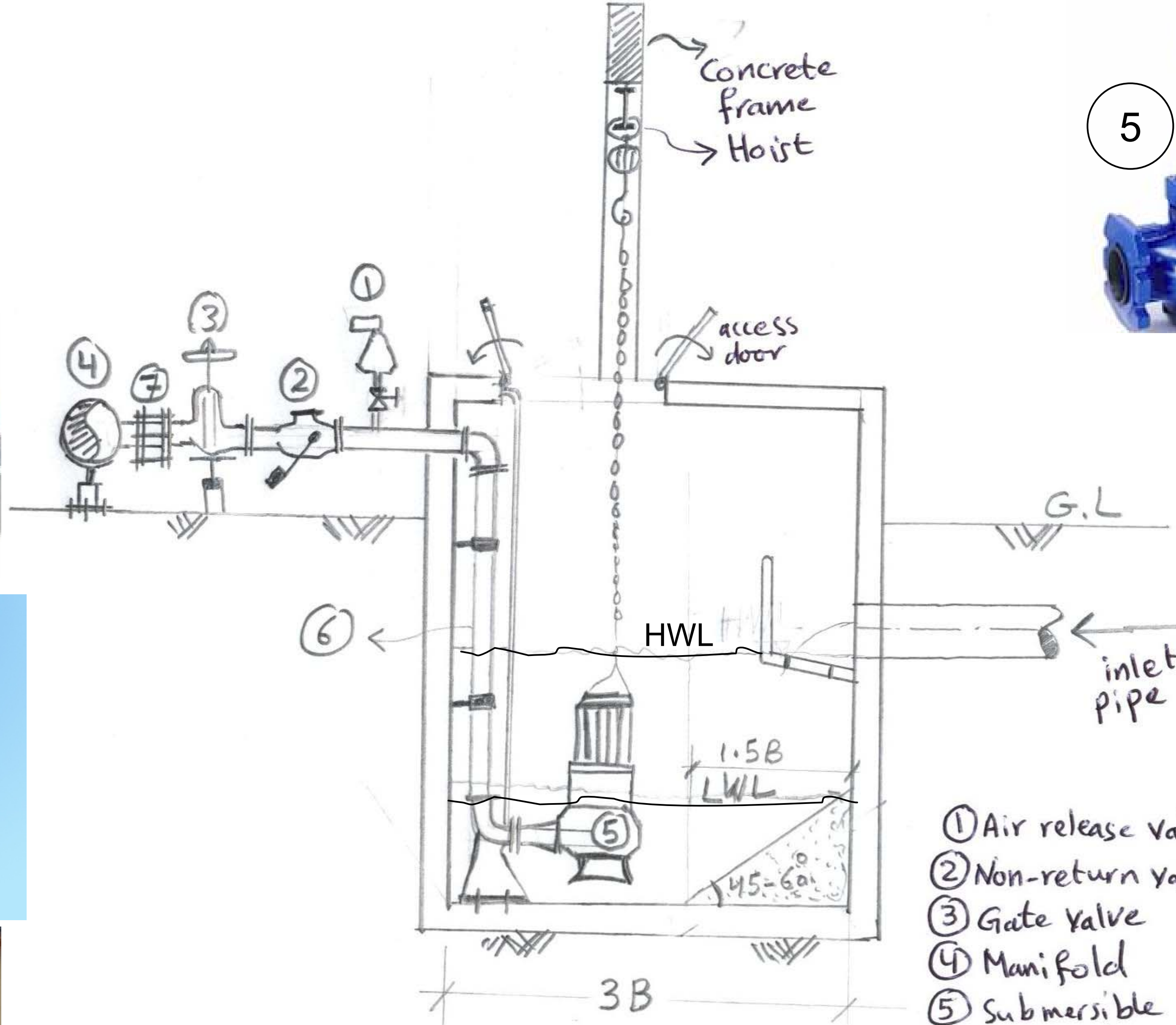
Flansch / Flange / Bride					
DN_2	g_f	K_f	D_f	Z_f	ø l_f
300	370	400	445	12	22

1) tiefster Ausschaltpunkt bei Automatikbetrieb
1) Lowest shut-off point for automatic operation
1) Point d'arrêt le plus bas en service automatique

	KRT ... / ... U/X/W		Pumpe / Pump / Pompe DN_1	DN_2		a_1	b_1	d	f_1	g	h_1	k_1	l_1	l_2	m	Fundament / Foundation / Fondation L	M	N	O	P	R	S	V	W	[kg] G
K	300-400 /	1104 1	300	300		1640	1285	830	440	840	350	1535	1760	—	480	120	250	—	1500	1060	900	420	65	180	1450
		1354 1			I																				1560
		1504 1																							1600
		2104				1875	1395					1645	—	1945									45		2100
		2504			II																				2250
		2804																							2450
		376 1				1335	1230					1465	1790	—									65		1000
		456 1																							1050
		646 1			I	1440																			1150
		806 1				1430	1265					1515	1740												1350
		1026				1640	1285					1535	1760												1500
K	300-500 /	646 1	300	300		1445	1200	790	440	840	350	1450	1670	—	440	120	250	—	1600	1200	850	420	65	180	1260
		806 1			I	1430	1245					1495	1720						1700						1470
		1026				1640	1265					1515	1740												1580
		1206																							1700
		1386				1878	1375					1625	—	1920					1800				45		2000
		1656			II																				2160
		2006																							2320

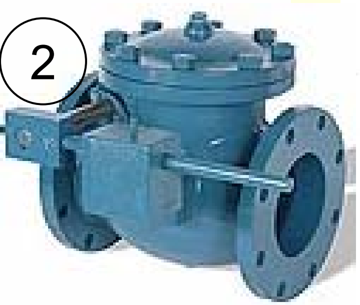
0W ... Cotes en mm

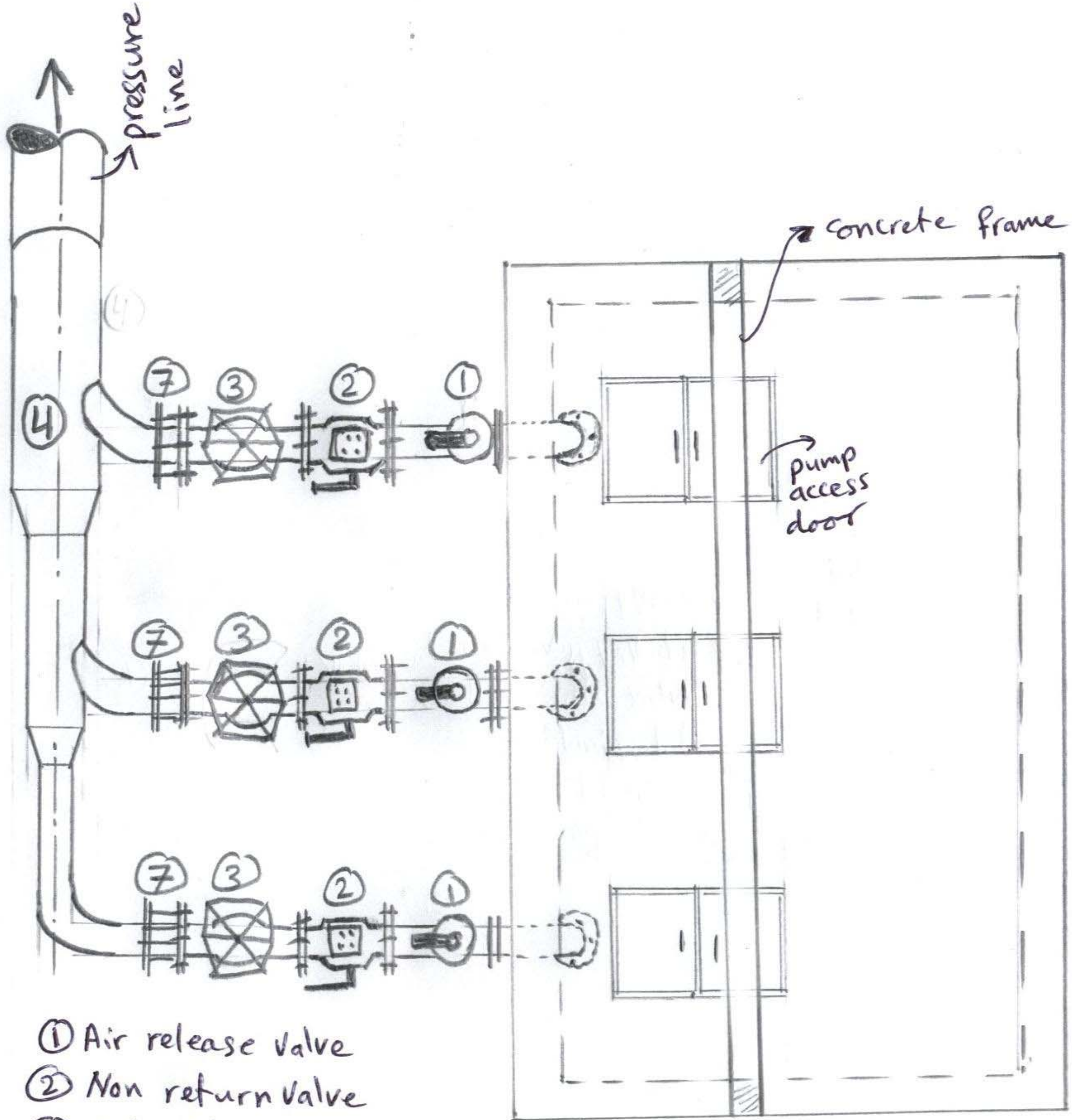
Fig. 5.23 Typical dimensions of a submersible KSB pump needed for the design of the wet well



(1) Air release valve
(2) Non-return valve
(3) Gate valve
(4) Manifold
(5) Submersible pump
(6) pump delivary pipe
(7) Dresser (flexiable) joint

Fig. 5.24 Vertical cross section in a wet well installation





① Air release Valve
② Non return Valve
③ Gate Valve
④ Manifold
⑦ Dresser (flexiable joint)

plan of the wet well

Fig. 5.25

Fig. 5.26

An example of a submersible pump installed in the wet well

Fig. 5.27 A photo at the top of a wet well showing delivery pipes and valves

5.4 Design of the main components of WWPS

5.4.4 Design of the wet well for submersible pumps

- Determination of the wet well active volume:

• The active volume in the wet well is the volume enclosed between the LWL and the HWL.
• This volume depends on many factors such as the number of pumps, the capacity of each pump and the raw WW influent flow rate.
• The active volume (V) when one pump is in operation is calculated from the following equation:

$$\boxed{V = \frac{Tq}{4}}$$

Where:
T: minimum cycle time between pump starts, min.
q : the pumping rate of a single pump in operation, m^3/min
V: active volume for one pump in operation, m^3

$$\boxed{\mathrm{T = time\ to\ fill + time\ to\ empty}}$$

5.4 Design of the main components of WWPS

5.4.4 Design of the wet well for submersible pumps

- Determination of the wet well active volume:

• The cycle time T depends on the pump type and the manufacturer recommendations.

• Typical value of T 6 minutes. This will lead to 10 starts of the pump in 1 hour. it is recommended to limit the number of pump starts to 12 as a maximum to protect the pump and to increase its effective life span.

•If more than a pump is used, an additional 15 cm is added to the height of the wet well for each pump other than the first.

5.4 Design of the main components of WWPS

5.4.4 Design of the wet well for submersible pumps

- Determination of the wet well active volume:

Example:
What is the active volume of a wet well that has 3 pumps working in parallel plus one standby pump. The capacity of one pump when operating alone is 14.7 m^3/min. Take T as 6 minutes. The area of the horizontal cross section of the wet well is 15 m^2 .

Solution:

V = (14.7*6)/4 = 22.05 m^3

Water height = (22.05)/15 = 1.47 m

Total height of active volume = 1.47 + 2*0.15 = 1.77 m

Total active volume = 1.77 * 15 = 26.55 m^3

5.4 Design of the main components of WWPS

5.4.4 Design of the wet well for submersible pumps

❑ Determination of the wet well active volume:

The pumps operation sequence can be set according to the active volume calculated in the previous example as indicated in the following figure.

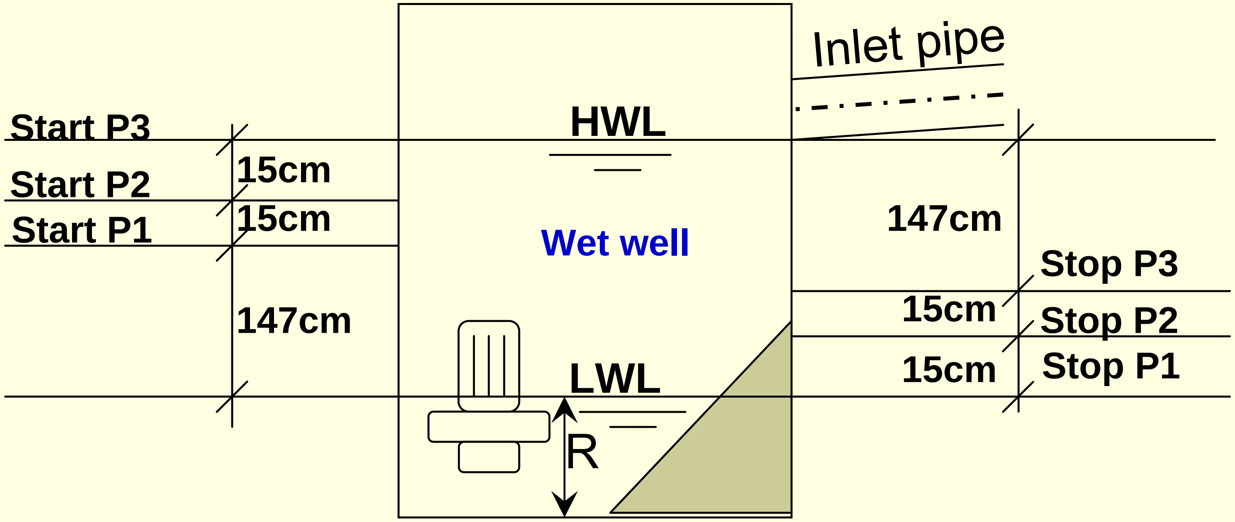


Fig. 5.28 Pumps operation sequence

5.4 Design of the main components of WWPS

Dry well installation

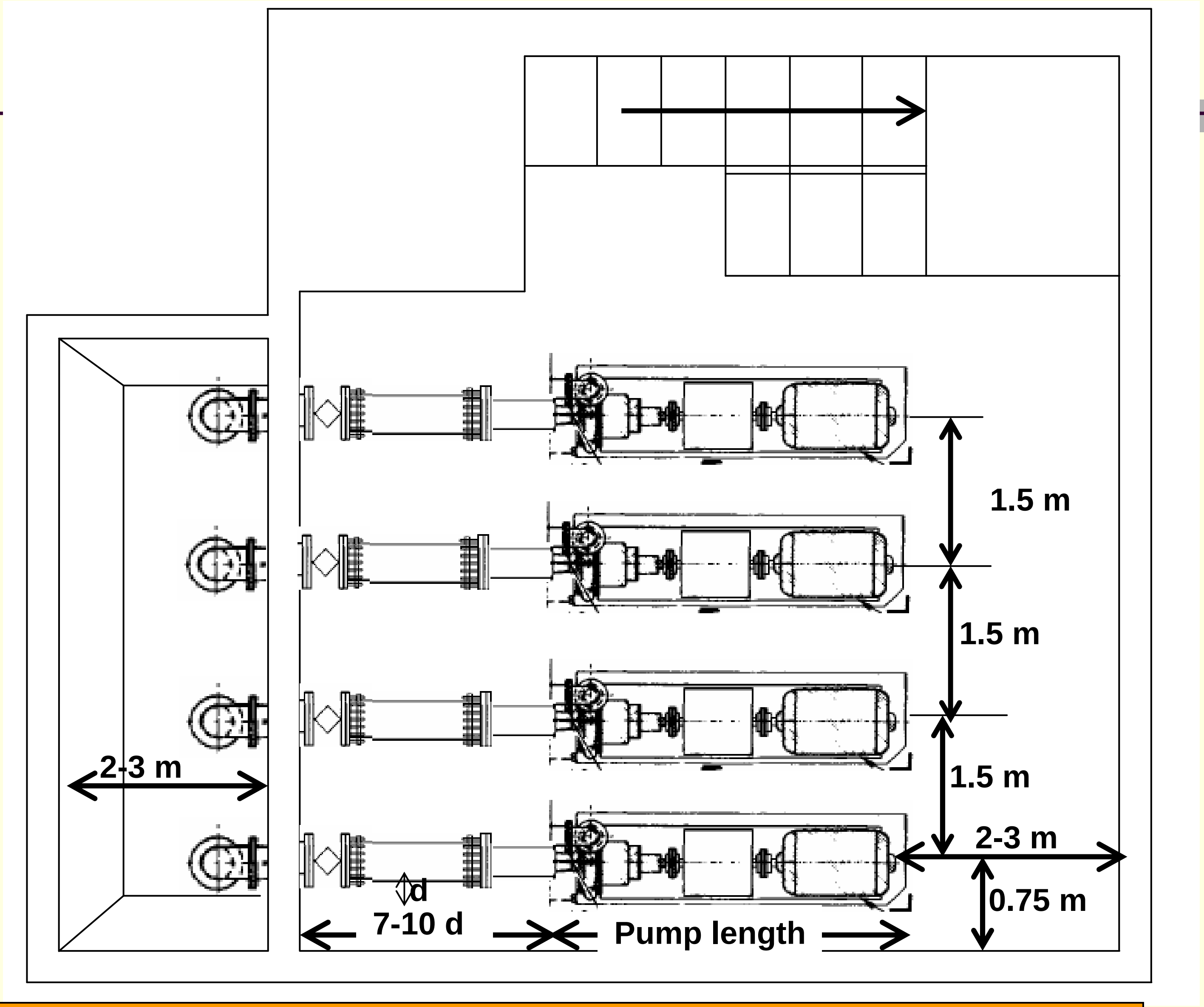


Fig. 5.29 Typical dimensions of the horizontal section of dry well installation

5.4 Design of the main components of WWPS

Dry well installation

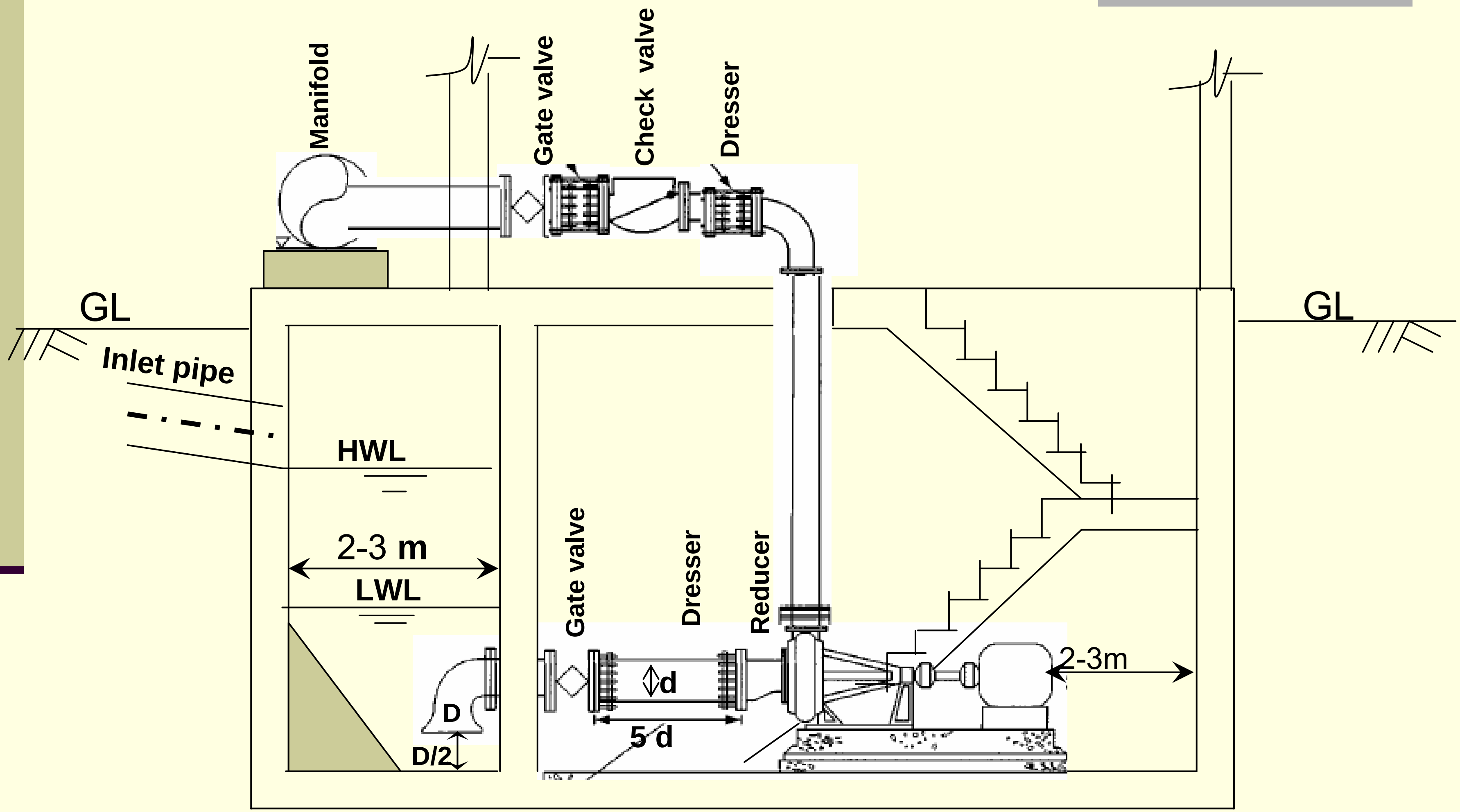


Fig. 5.30 Typical dimensions of the vertical section of dry well installation

5.4 Design of the main components of WWPS

5.4.5 Design of the wet and dry wells for dry pumps

- Figure 5.29 gives the typical dimensions of the the horizontal cross section of the dry and wet well installation.

- Figure 5.30 gives the typical dimensions of the the vertical cross section of the dry and wet well installation.

- The active volume and pump operation sequence is the same as for the wet wall installation as discussed in the previous example.

5.4 Design of the main components of WWPS

5.4.6 Design of the delivery pipes and the pressure line:

- pressure line (selecting the most economical diameter)
- delivery pipes
- Valves on the pressure line
- Thrust block

5.4 Design of the main components of WWPS

5.4.6 Design of the delivery pipes and the pressure line:

- Delivery pipes:

A delivery pipe is the pipe that connects the pump with the manifold. The diameter of this pipe is determined using the Continuity equation:

$$\boxed{D = \left[\frac{4Q_{pump}}{\pi V}\right]^{\frac{1}{2}}}$$

Where:

D = pipe diameter, m

Q_{pump} = discharge of one pump when operation alone, m^3/d

V = flow velocity, m/s

The velocity in delivery pipes is usually assumed in the range of 2-2.5 m/s. Q pump is the determined from the intersection between the system curve and The characteristic curve of a single pump in operation.

5.4 Design of the main components of WWPS

5.4.6 Design of the delivery pipes and the pressure line:

- The friction losses in the delivery pipe are calculated using the Hazen Williams equation :

$$h_f = \frac{10.7 * L}{D^{4.87}} \left[\frac{Q}{C} \right]^{1.852}$$

- The minor losses in the delivery pipe are calculated using the following equation :

$$h_m = k \frac{V^2}{2g}$$

- The material of the delivery pipes is usually steel, but some times UPVC pipes are used. The Thickness of the pipe wall is determined according to the pressure Exerted on the pipe, especially water hammer pressure.

- The valves and fittings installed on the delivery pipe are shown on Figures 15.24 and 15.25

5.4 Design of the main components of WWPS

5.4.6 Design of the Manifold:

- Manifold:

The manifold is the pipe that connects the delivery pipes with the main pressure line. As shown on Figure 15.25 the diameter of the manifold is variable diameter of this pipe is determined using the Continuity equation:

$$\boxed{D = \left[\frac{4Q}{\pi V}\right]^{\frac{1}{2}}}$$

Where:

D = pipe diameter (variable) according to the # of pumps in operation, m

Q = discharge (variable) according to the # of pumps in operation, m^3/d

V = flow velocity (variable) according to the # of pumps in operation, m/s

- The velocity in the manifold is usually assumed in the range of 1 to 2 m/s.

Q is Q_{pump} for the first segment (the smallest), and equals Q_{2pumps} for the second segment and Q_{3pumps} for the second third segment and so on.

Some designers us constant diameter of the manifold designed for Q_{3pumps} .

5.4 Design of the main components of WWPS

5.4.6 Design of the Manifold :

- The friction losses in the manifold pipe are calculated using the Hazen Williams equation :

$$h_f = \frac{10.7 * L}{D^{4.87}} \left[\frac{Q}{C} \right]^{1.852}$$

- The minor losses in the manifold pipe are calculated using the following equation :

$$h_m = k \frac{V^2}{2g}$$

- The material of the manifold pipes is usually steel, but some times UPVC pipes are used. The Thickness of the pipe wall is determined according to the pressure Exerted on the pipe, especially water hammer pressure.

5.4 Design of the main components of WWPS

5.4.6 Design of the main Pressure line (Rising main):

- Selecting the diameter:

The main pressure line is the pipe that connects the manifold with ponit of disposal. Its determined is determined using the Continuity equation:

$$\boxed{D = \left[\frac{4Q_{peak}}{\pi V}\right]^{\frac{1}{2}}}$$

Where:

D = pipe diameter (variable) according to the # of pumps in operation, m

Q_{peak} = discharge (variable) according to the # of pumps in operation, m^3/d

V = flow velocity at peak flow, m/s

- The velocity at peak flow in the main pressure line is usually assumed in the range of 1 to 2 m/s. However the velocity when one pump in operation should not be less than 0.60 m/s.

5.4 Design of the main components of WWPS

5.4.6 Design of the Manifold :

- The friction losses in the main pressure line are calculated using the Hazen Williams equation :

$$h_f = \frac{10.7 * L}{D^{4.87}} \left[\frac{Q}{C} \right]^{1.852}$$

- The minor losses in the main pressure line are calculated using the following equation :

$$h_m = k \frac{V^2}{2g}$$

- The material of the main pressure line is usually steel, but some times UPVC pipes are used. The Thickness of the pipe wall is determined according to the pressure Exerted on the pipe, especially water hammer pressure.

5.4 Design of the main components of WWPS

5.4.6 Design of the main Pressure line (Rising main):

- Selecting the most economical diameter for the main pressure line:

• As understood from the continuity equation, D is a function of V, so there are many combinations of D and V that satisfy the equation. Since we , as engineers, look for the most economical design we usually try to use the smallest pipe Diameter, Unfortunately, when D decreases, V increases, consequently the Power needed and the operation cost increases. So we need to select the most Optimum combination between D and V. This is done using the graph shown below (Figure 15.31).

• From figure 15.31, when we add up the capital cost and the operation cost we get the concave up curve. The point of inflection indicates the minimum total Cost. From this point draw a vertical line that will intersect the X-axis at the most Optimum diameter.

• Notice that the running cost paid along the life span of the pump station and the Pressure line so the values on figure 15.31 are the present worth value of the Running cost. The capital cost is paid in one payment at the beginning of the project.

5.4 Design of the main components of WWPS

5.4.6 Design of the main Pressure line (Rising main):

- Selecting the most economical diameter for the main pressure line:

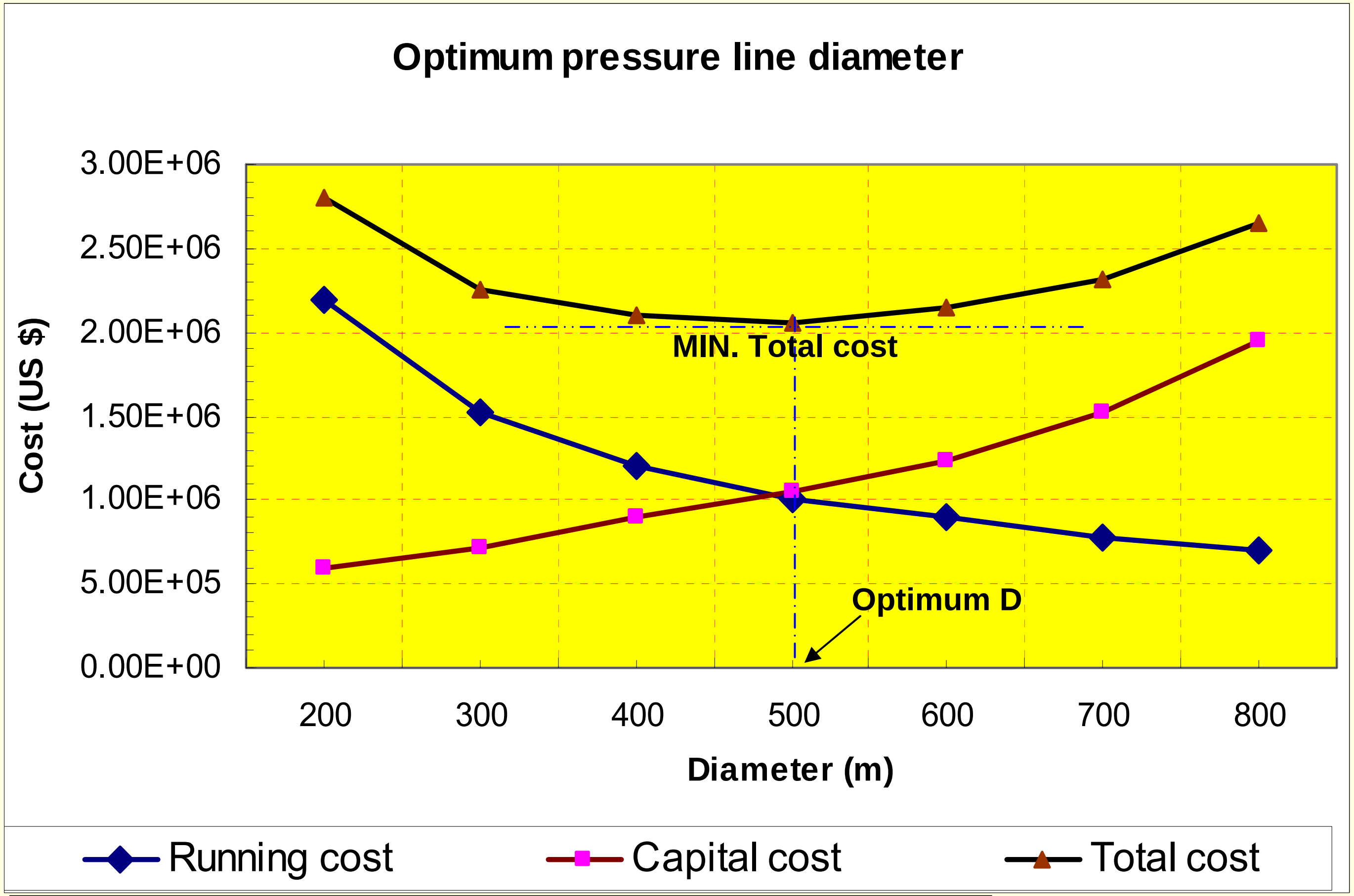


Fig. 5.31 Optimum pipe diameter

5.4 Design of the main components of WWPS

5.4.6 Design of the main Pressure line (Rising main):

- Valves and tanks installed on the main pressure line:
  - There are two main valves that are installed on the pressure lines:

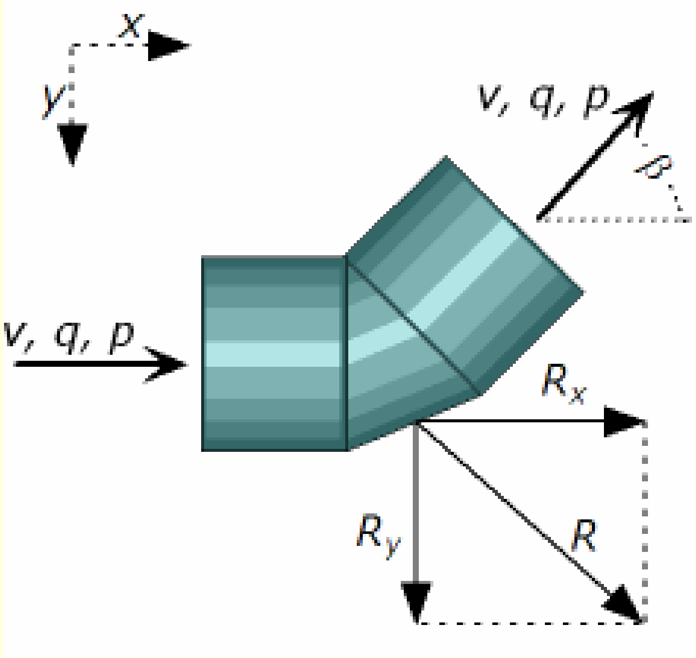
1. Air release /Vacuum valves are installed at:
   - All height points
   - long rising segments at intervals of 750 to 1000 m
   - long descending segments at intervals of 750 to 1000 m
2. Drainage valves are :
   - All low point
   - Long rising and flat pipes at intervals of 500 m
3. Pressure relief valves are installed at:
   - Low points along the pipe where high pressure is expected
   - locations related to water hammer waves such as before valves or after pumps. If other tools are used to control water hammer, these valves will be not necessary.
4. Surge tanks and Air champers are installed at:
   - Just after the pumps.
   - At high points a long the pipe to prevent column separation.

5.4 Design of the main components of WWPS

5.4.6 Design of the main Pressure line (Rising main):

- Thrust blocks installed on the main pressure line:

Thrust blocks are concrete blocks installed along the pressure lines at horizontal and bends vertical to protect the pipe from the water thrust force created due to The change in direction.



5.4 Design of the main components of WWPS

5.4.6 Design of the main Pressure line (Rising main):

- Thrust blocks installed on the main pressure line:

The resulting force due to mass flow and flow velocity can be expressed as :

$$\boxed{R_x = \rho\pi\left(\frac{d}{2}\right)^2 \bullet V^2(1-\cos\beta)}$$

$$\boxed{R_y = \rho\pi\left(\frac{d}{2}\right)^2 \bullet V^2 \sin\beta}$$

$$\boxed{R = (R_x^2 + R_y^2)^{\frac{1}{2}}}$$

Rx = resulting force in x-direction (N)
Ry = resulting force in y-direction (N)
R = Resultant of the x and y forces (N)
v = flow velocity , m/s
β = bend angle , degrees
ρ = fluid density, kg/m
d = internal pipe diameter , m

5.4 Design of the main components of WWPS

5.4.6 Design of the main Pressure line (Rising main):

- Thrust blocks installed on the main pressure line:

Resulting force due to Static Pressure:

$$R_x = P\pi\left(\frac{d}{2}\right)^2(1-\cos\beta)$$

$$R_{py} = P\pi\left(\frac{d}{2}\right)^2 \sin\beta$$

$$R_p = (R_{px}^2 + R_{py}^2)^{\frac{1}{2}}$$

R_{px} = resulting force due to pressure in x-direction ,N

R_{py} = resulting force due to pressure in x-direction ,N

Rp= resultant force on the bend due to pressure, N

P = gauge pressure inside pipe (Pa, N/m^2)

5.4 Design of the main components of WWPS

5.4.6 Design of the main Pressure line (Rising main):

See also chapter 4

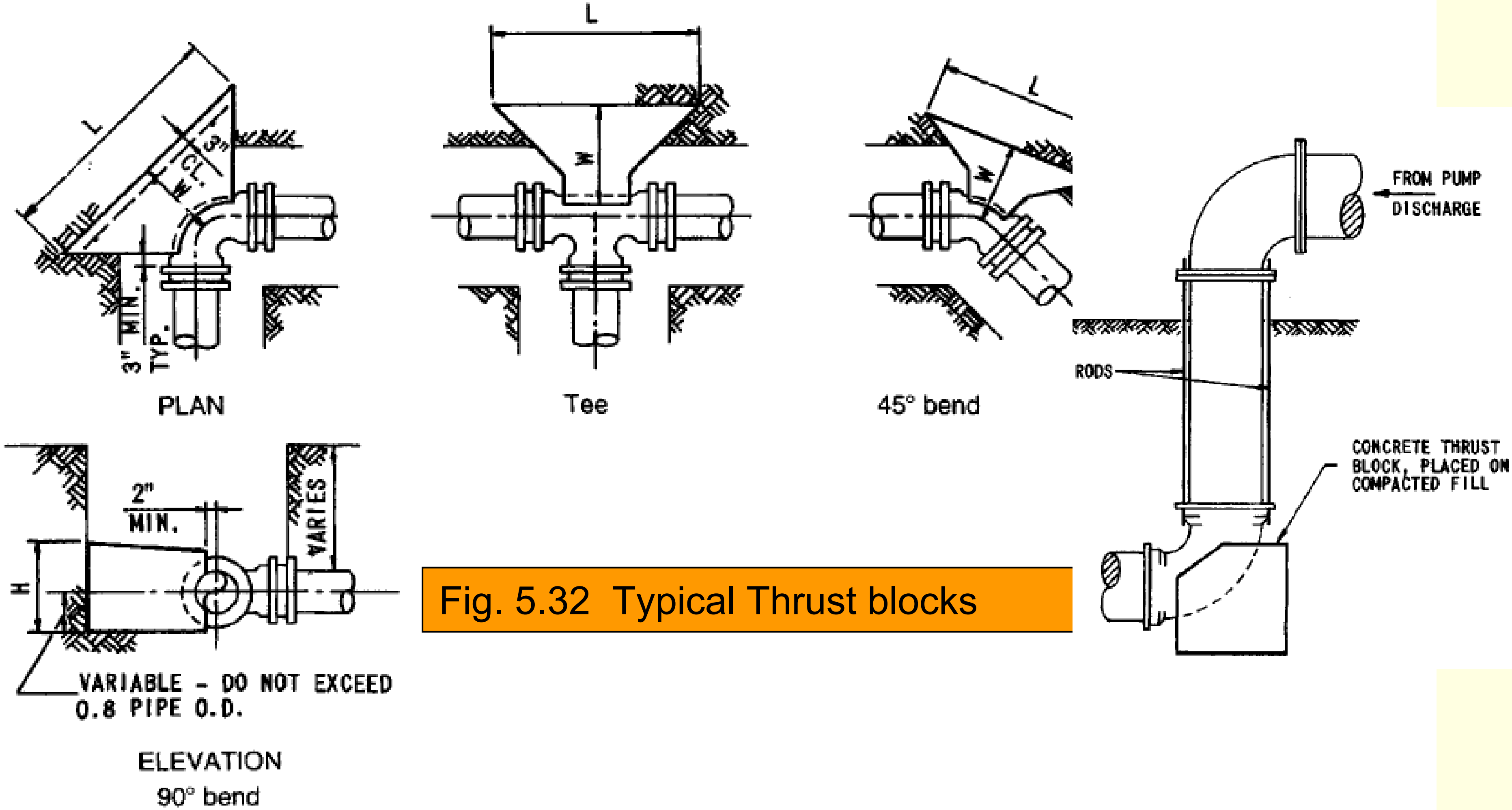


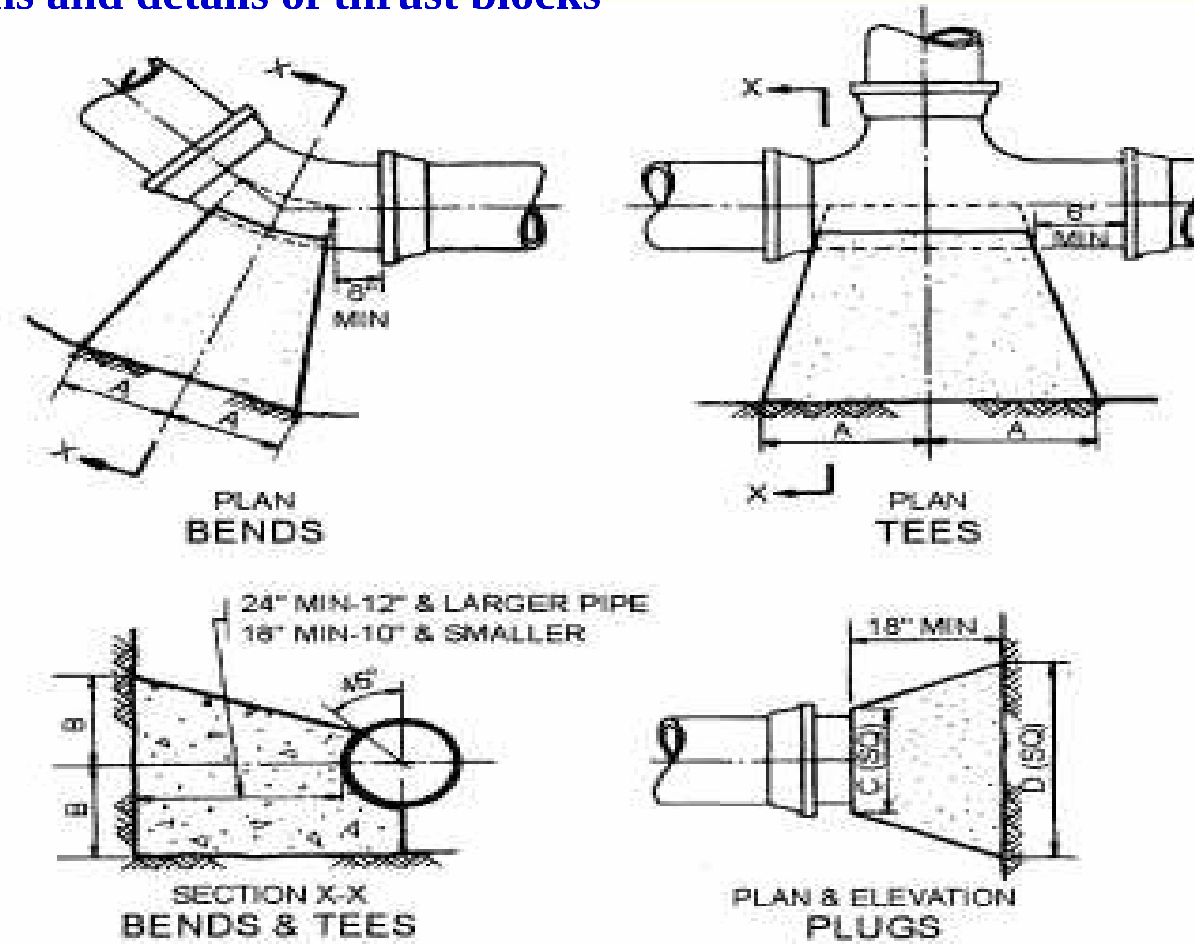
Fig. 5.32 Typical Thrust blocks

5.4 Design of the main components of WWPS

5.4.6 Design of the main Pressure line (Rising main):

Fig. 5.33 Thrust block under construction

Typical dimensions and details of thrust blocks



TYPE	SIZE	1/4 BENDS		1/8 BENDS		1/16 BENDS		TEES		PLUGS	
		A	B	A	B	A	B	A	B	C	D
TYPE I 4000 PSF SOIL	6"	8"	10"	6"	8"	3"	8"	8"	8"	10"	15"
	8"	12"	12"	8"	10"	5"	9"	9"	12"	12"	20"
	10"	16"	14"	10"	12"	6"	10"	11"	14"	14"	25"
	12"	19"	16"	12"	14"	8"	11"	14"	16"	16"	30"
	14"	23"	18"	14"	16"	10"	12"	16"	18"	18"	34"
	16"	26"	20"	16"	18"	11"	13"	18"	20"	20"	38"
TYPE II 2000 PSF SOIL	6"	16"	10"	9"	10"	6"	8"	10"	12"	10"	21"
	8"	22"	13"	12"	13"	8"	10"	13"	16"	12"	29"
	10"	26"	17"	14"	17"	10"	13"	16"	20"	14"	36"
	12"	29"	21"	16"	21"	11"	16"	18"	24"	16"	41"
	14"	35"	24"	19"	24"	12"	20"	22"	27"	18"	48"
	16"	38"	27"	21"	27"	12"	24"	24"	30"	20"	54"

Pumping Stations Design

For Infrastructure Master Program
Engineering Faculty-IUG

Lecture 6: Design of water supply pumping stations

Dr. Fahid Rabah
Water and environment Engineering
frabah@iugaza.edu

Lecture 6: Design of waster supply pumping stations

6.1 General introduction

- Main Types of water pumping stations :

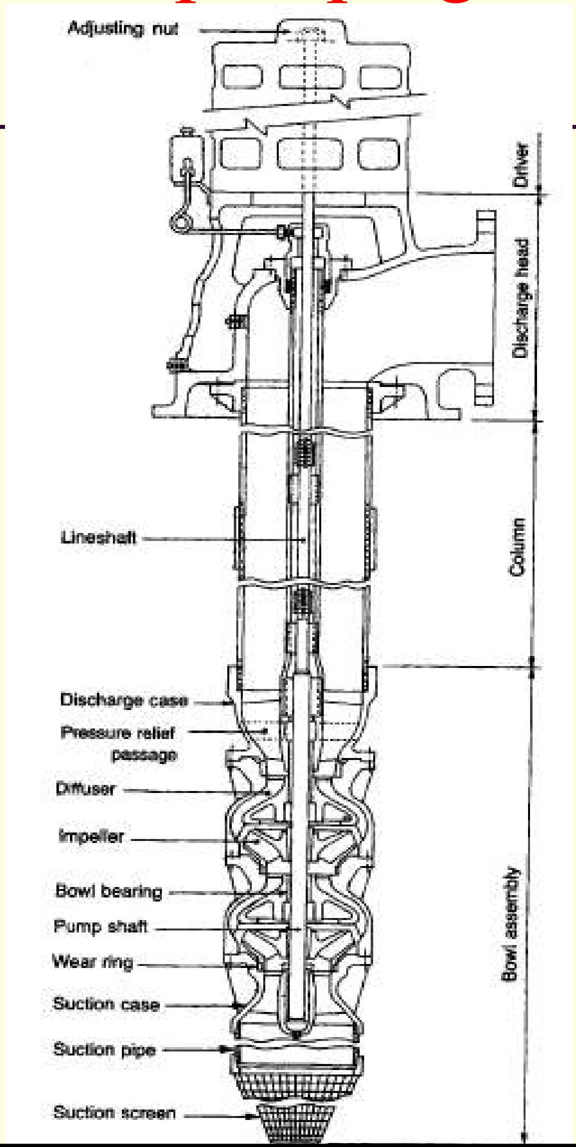
1. Wells pumping stations
2. Distribution pumping stations
3. Booster pumping stations
4. Surface water pumping stations

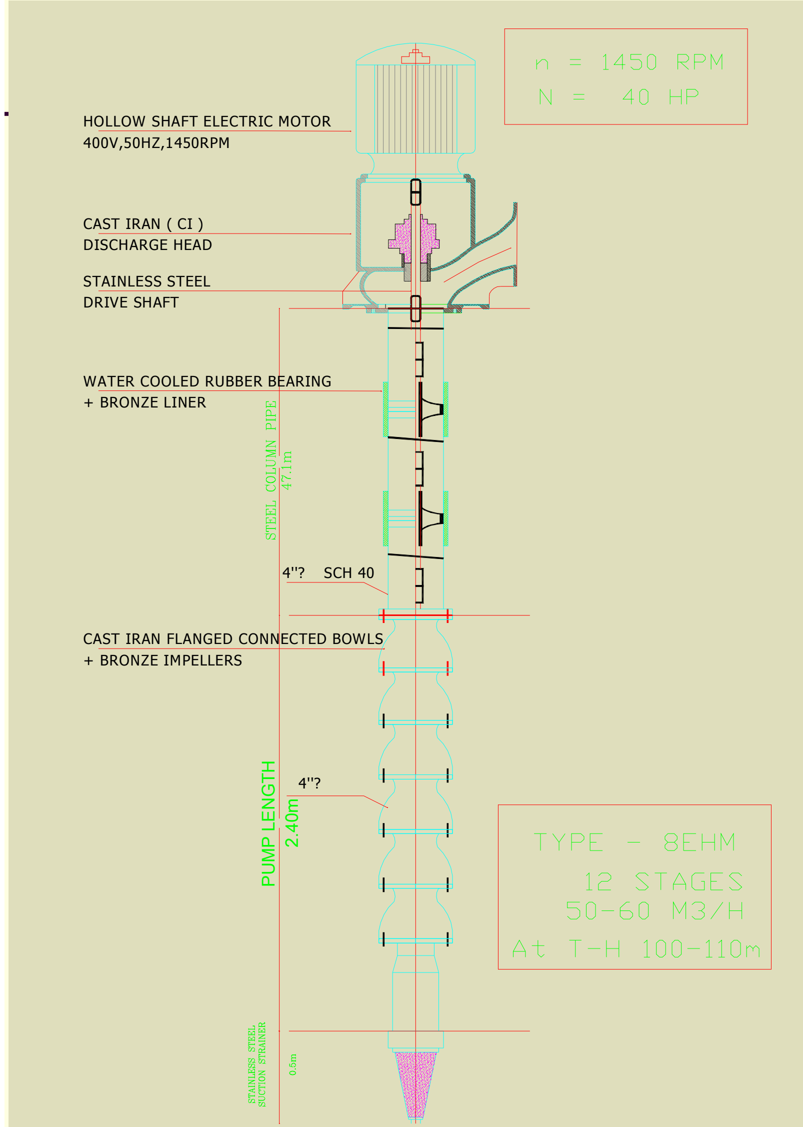
6.2 Well pumping stations

- Main components of well pumping stations :

1. Multi stage pumps (connected in series)
2. Suction shaft (pipe)
3. Vertical motor (above ground or submerged)
4. Delivery pipe
5. Valves
6. Cyclone
7. Surge vessel (air champer for water hammer protection)
8. Chlorination tank and chlorine injection pump.
9. Stand by generator and its fuel tank
10. Main electricity distribution panel and control
11. Service building.

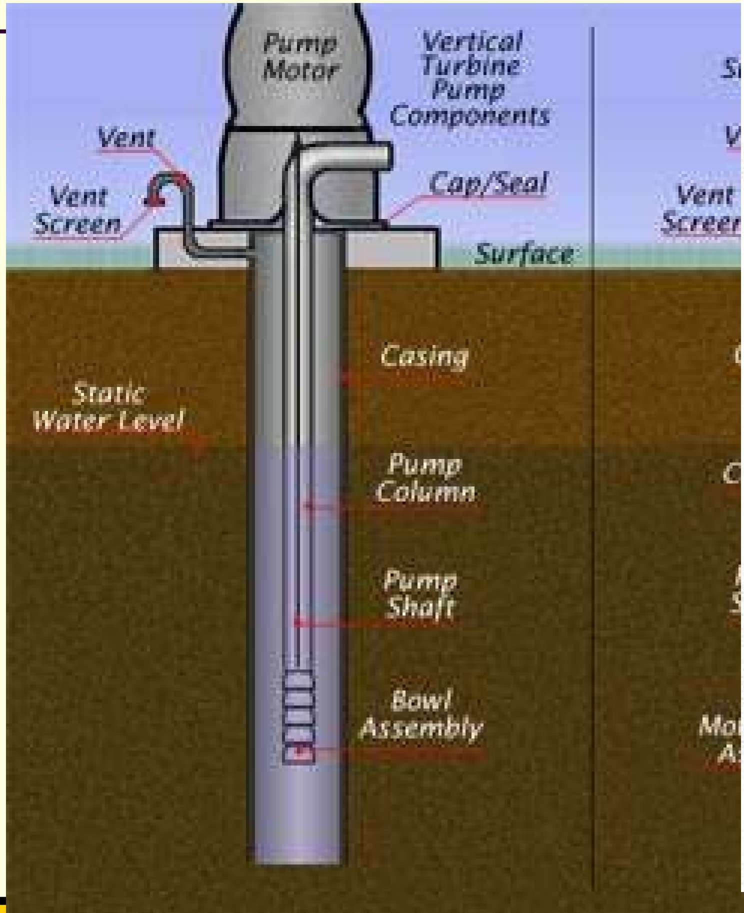
6.2 Well pumping stations





Multi stage pumps connected in series with a dry installation vertical motor

6.2 Well pumping stations



CONSTRUCTION FEATURES:

1. One-piece, stainless steel discharge head & NEMA faced mounting ring.
2. Built-in stainless steel check valve with elastomer ring for positive seal.
3. Sintered lead-free sleeve bearing with polypropylene sand slinger.
4. 304 Stainless Steel Pump shell threaded on both ends for easy attachment to discharge head & mounting ring.
5. Stainless steel suction screen covers a large round suction inlet.

Multi stage pumps connected in series with dry or submersible motor

6.2 Well pumping stations

Multi stage pumps connected in series with diesel motor

6.2 Well pumping stations

CONSTRUCTION FEATURES:

1. One-piece, stainless steel discharge head & NEMA faced mounting ring.
2. Built-in stainless steel check valve with elastomer ring for positive seal.
3. Sintered lead-free sleeve bearing with polypropylene sand slinger.
4. 304 Stainless Steel Pump shell threaded on both ends for easy attachment to discharge head & mounting ring.
5. Stainless steel suction screen covers a large round suction inlet.

Multi stage pumps connected in series- submersible motor

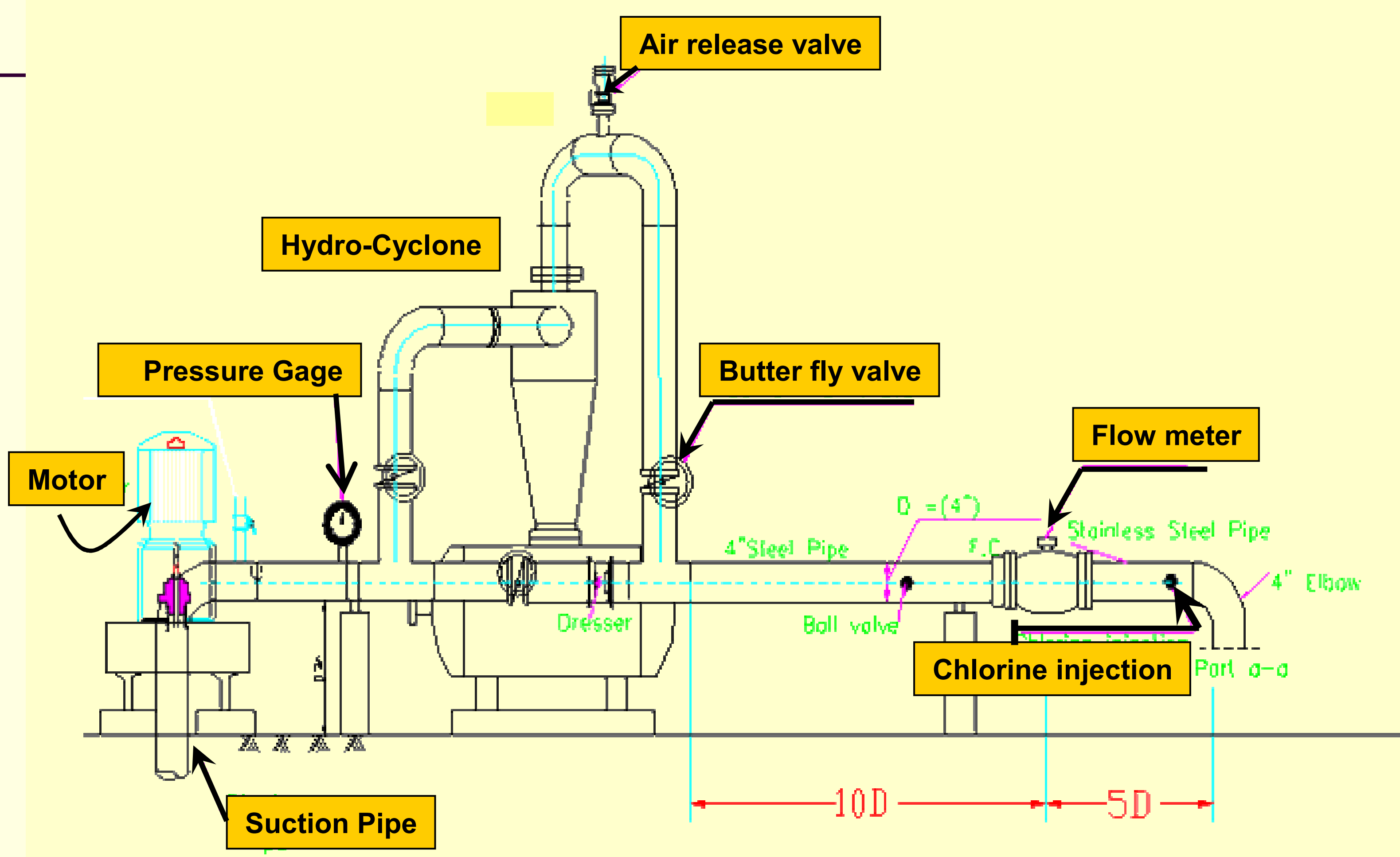
6.2 Well pumping stations

Vertical dry motor installation

6.2 Well pumping stations

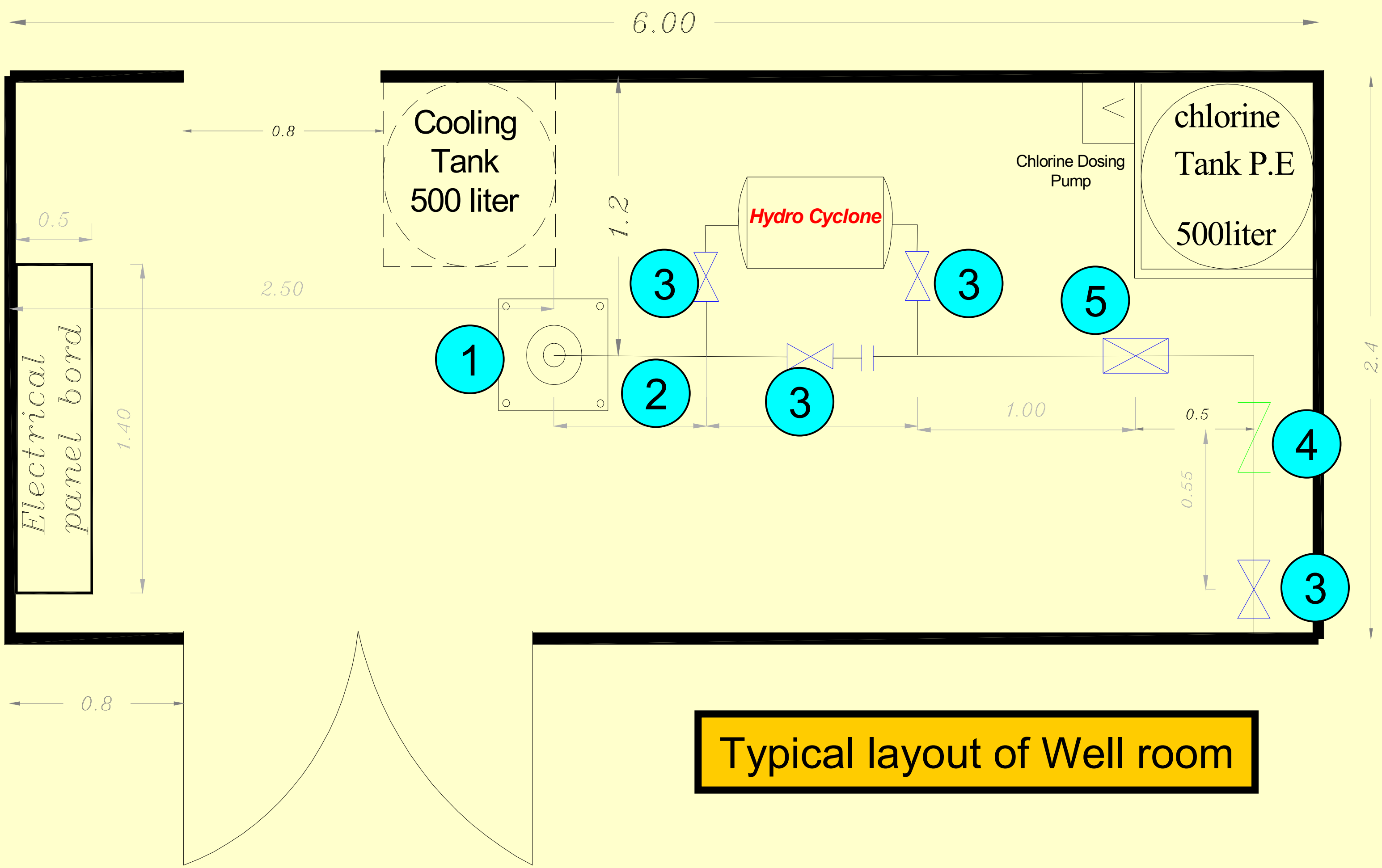
Submersible type motor installation

6.2 Well pumping stations



Typical piping and valves arrangement for well pumping stations

6.2 Well pumping stations

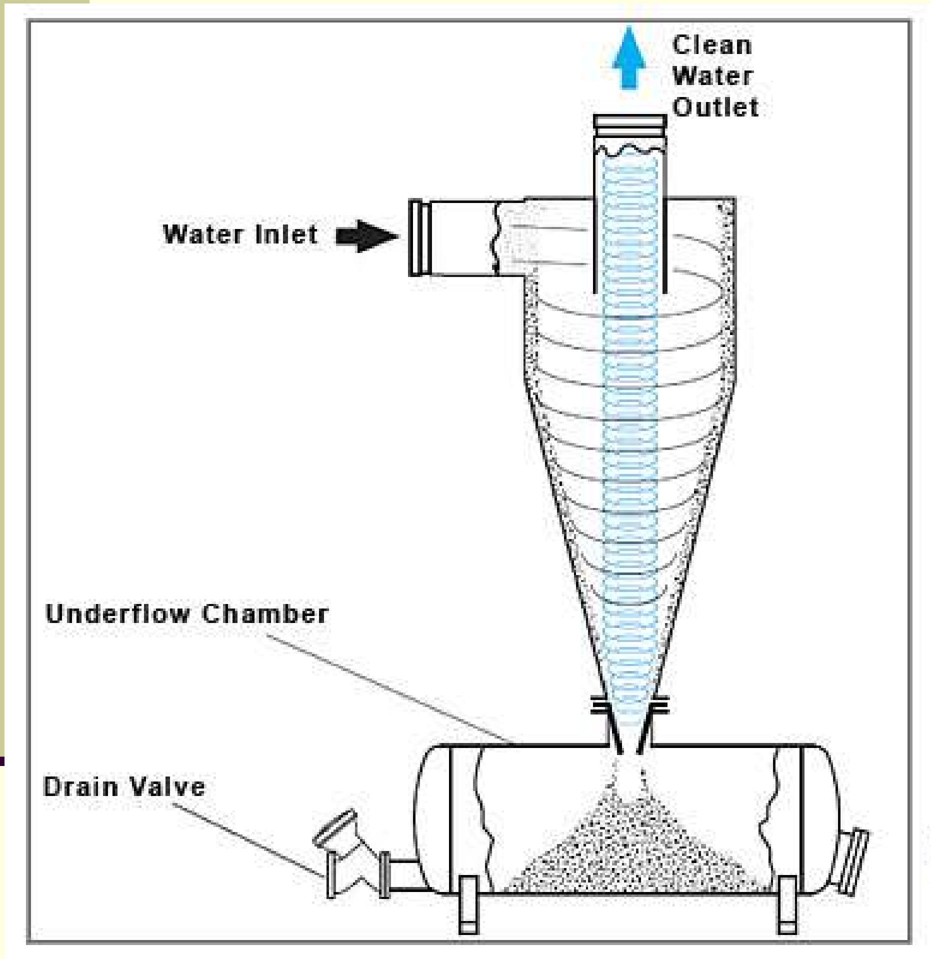


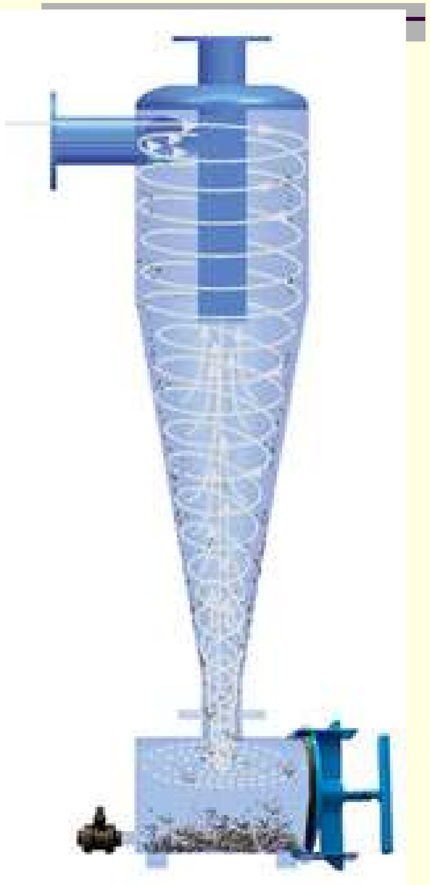
1 Pump well 2 Delivery pipe 3 Butterfly valve 4 Check valve 5 Flow meter

6.2 Well pumping stations

Delivery pipe , see the Hydro Cyclones and valves

6.2 Well pumping stations





Hydro Cyclone function

6.2 Well pumping stations

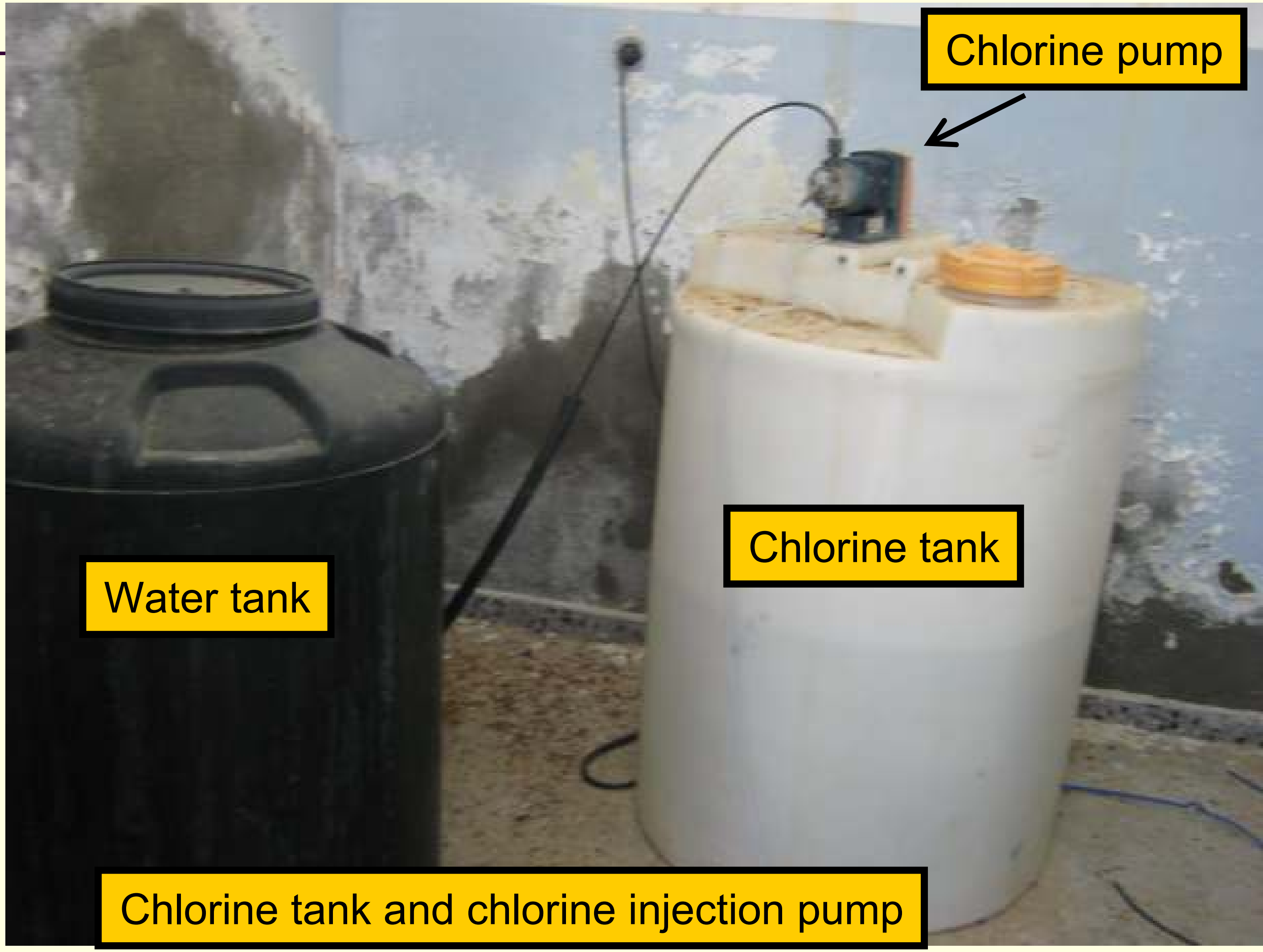
Delivery pipe , see the Hydro Cyclones and valves

6.2 Well pumping stations

Standby generator

6.2 Well pumping stations

6.2 Well pumping stations

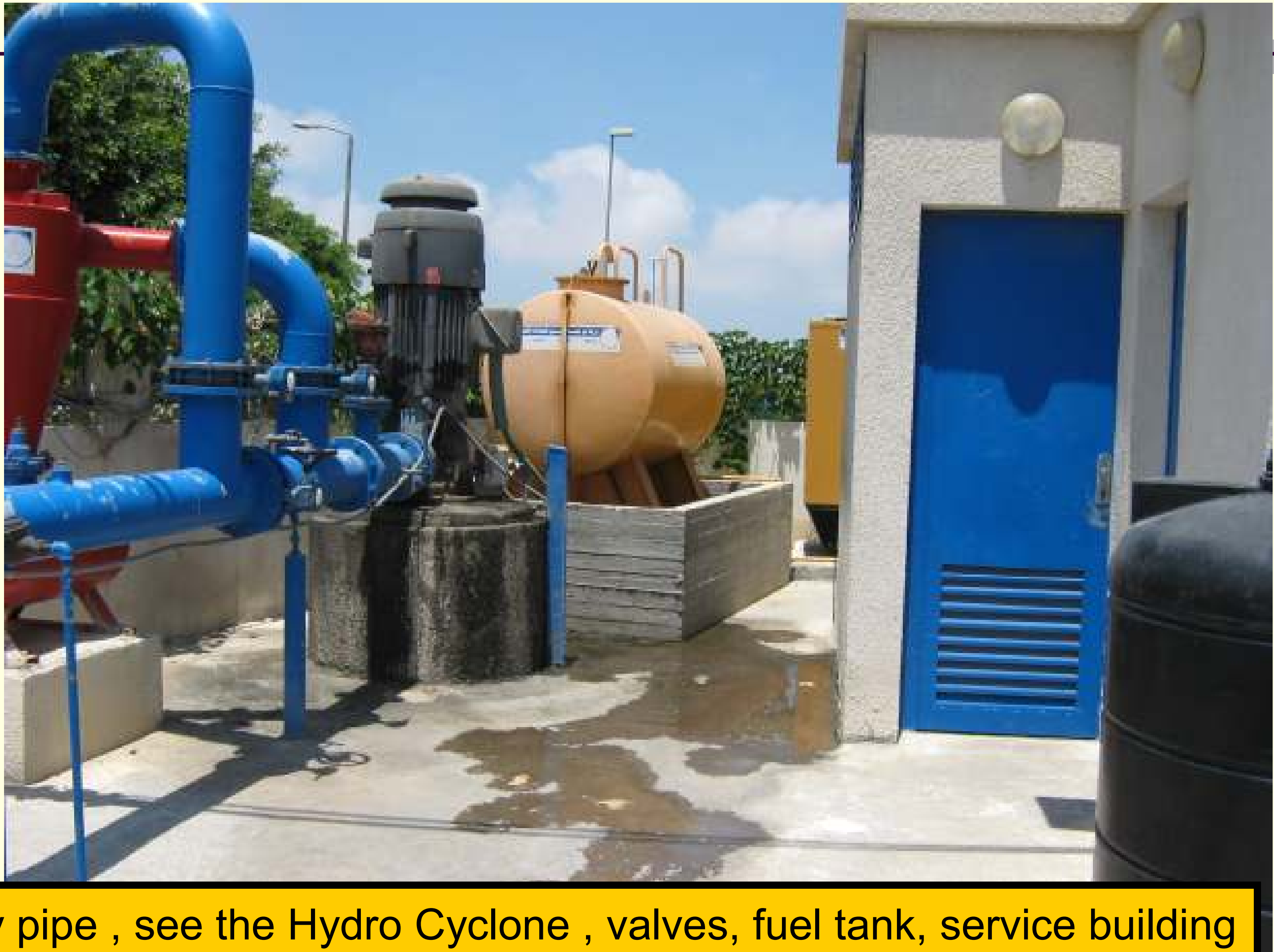


Chlorine tank and chlorine injection pump

6.2 Well pumping stations

chlorine injection point to the delivery pipe

6.2 Well pumping stations



Delivery pipe , see the Hydro Cyclone , valves, fuel tank, service building

6.2 Well pumping stations

Delivery pipe , Hydro Cyclone , valves, fuel tank, generator

6.2 Well pumping stations

6.2 Well pumping stations

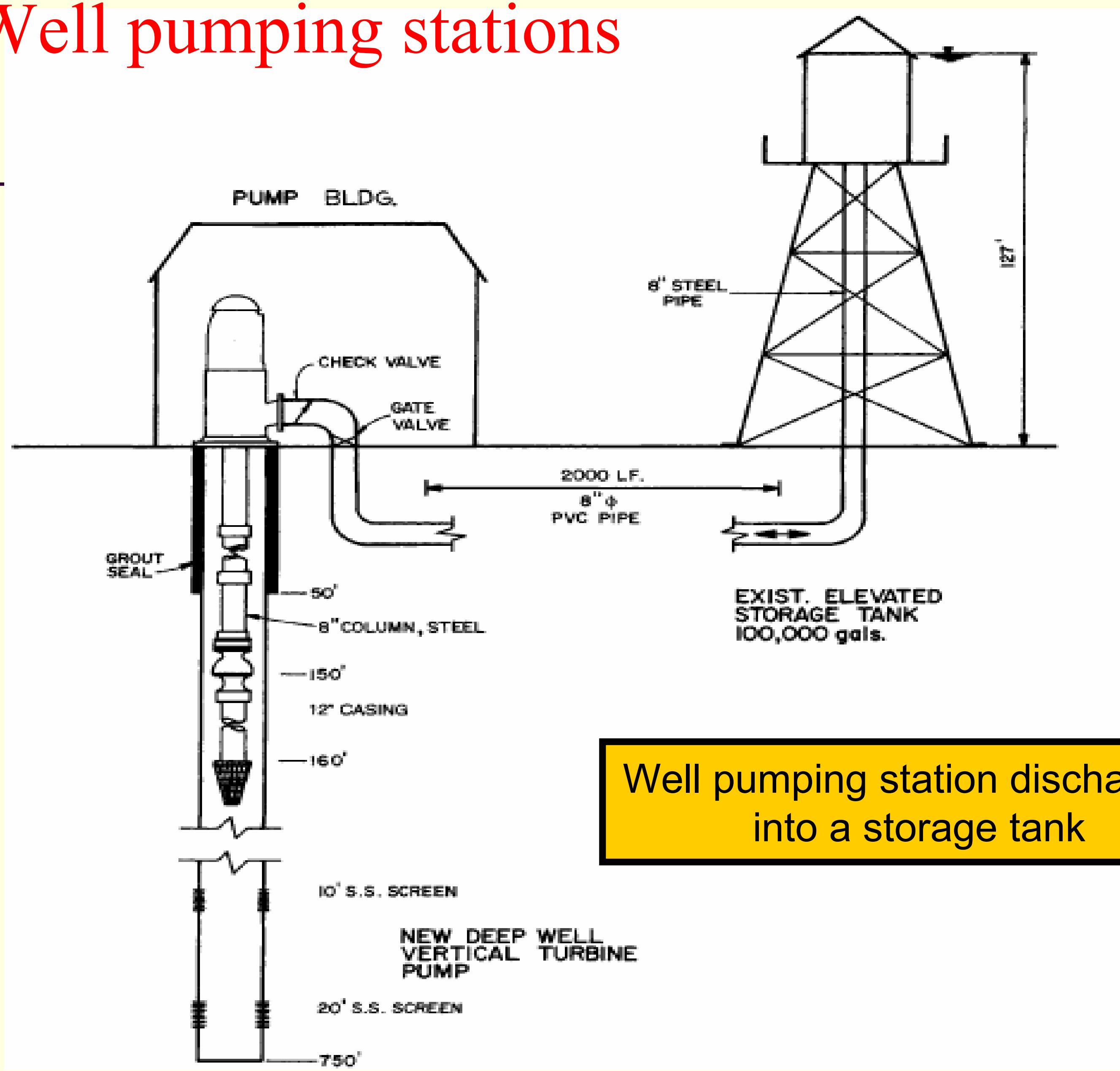
6.2 Well pumping stations

- Types of well pumping stations :

1. Direct pumping from well pumps to the distribution system
    - Most of Gaza wells are from this type
    - Not recommended for long distribution systems and for cases where more than one well pumping directly into the system. This system has two problems: back pressure between wells and lower discharge from each individual well.

2. Pumping from well pumps to a storage or distribution tank.
    - This system is recommended and there is a trend in Gaza to use this system
    - This system solves the problem of back pressure associated with the direct pumping discussed above. It also increase the pumping efficiency from each pump (lower head losses leading to more flat system curve and consequently more discharge)

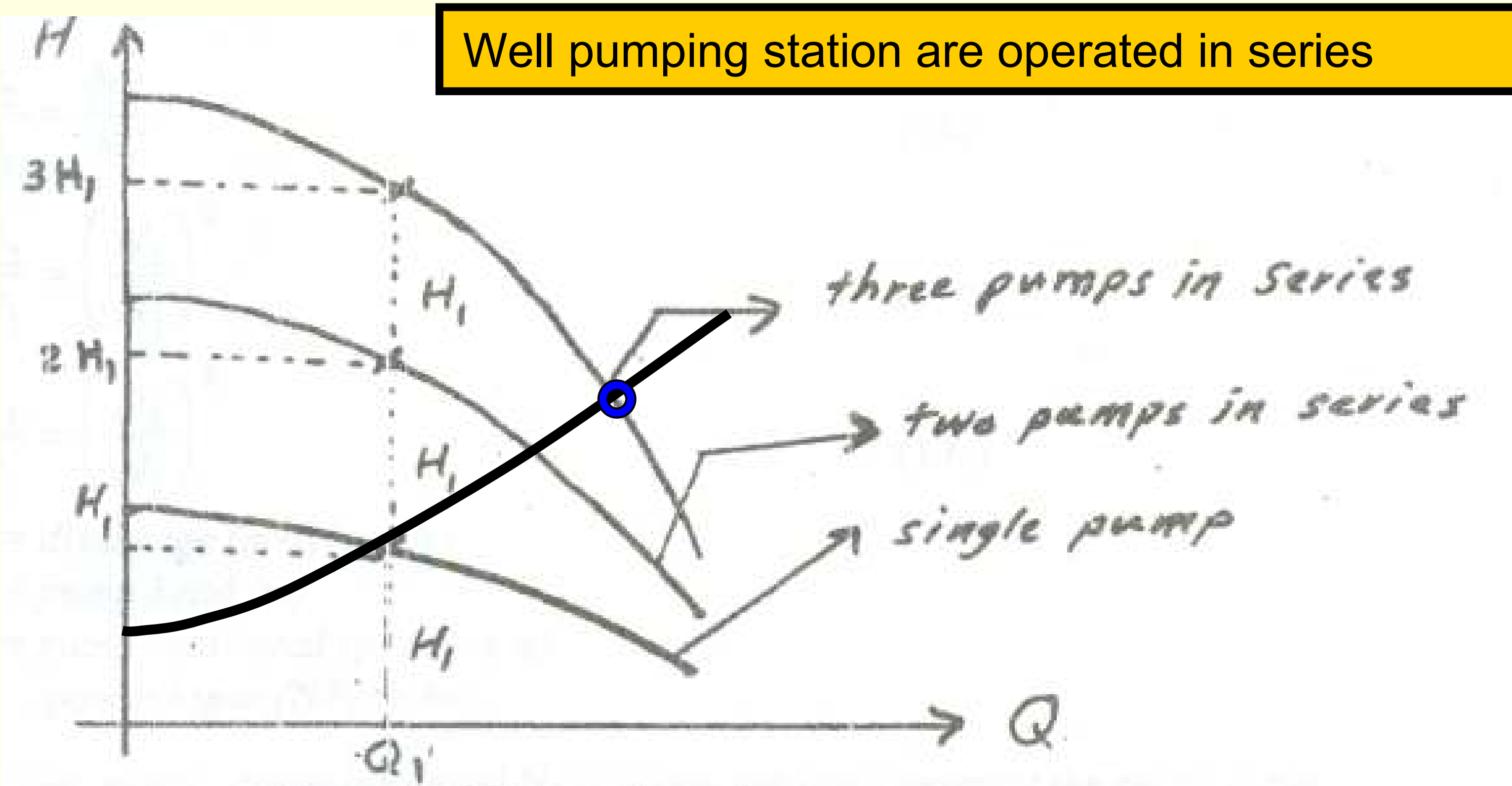
6.2 Well pumping stations



Well pumping station discharging into a storage tank

6.2 Well pumping stations

Well pumping station are operated in series



Note that for well pumps there is only one valid pump curve which is the curve of the total pumps in operation.

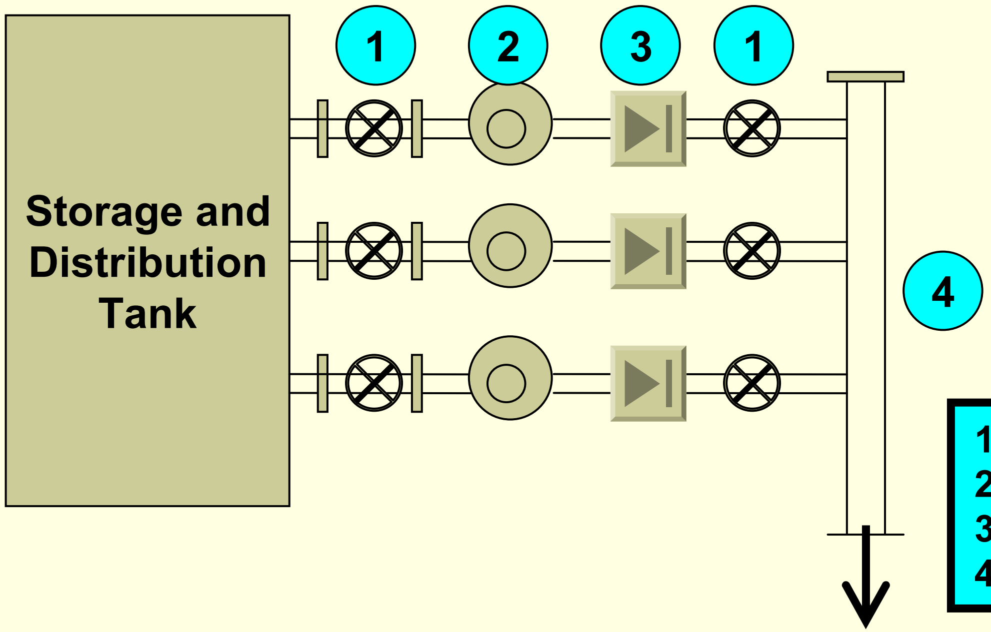
6.3 Distribution pumping stations

- Main components of distribution pumping stations :

1. Dry pumps (connected in parallel)
2. Suction shaft (pipe)
3. Storage and distribution tank
4. Delivery pipe
5. Valves
6. Surge vessel (air champer for water hammer protection)
7. Chlorination tank and chlorine injection pump.
8. Stand by generator and its fuel tank
9. Main electricity distribution panel and control
10. Service building.

6.3 Distribution pumping stations

- Typical layout of distribution pumping stations:

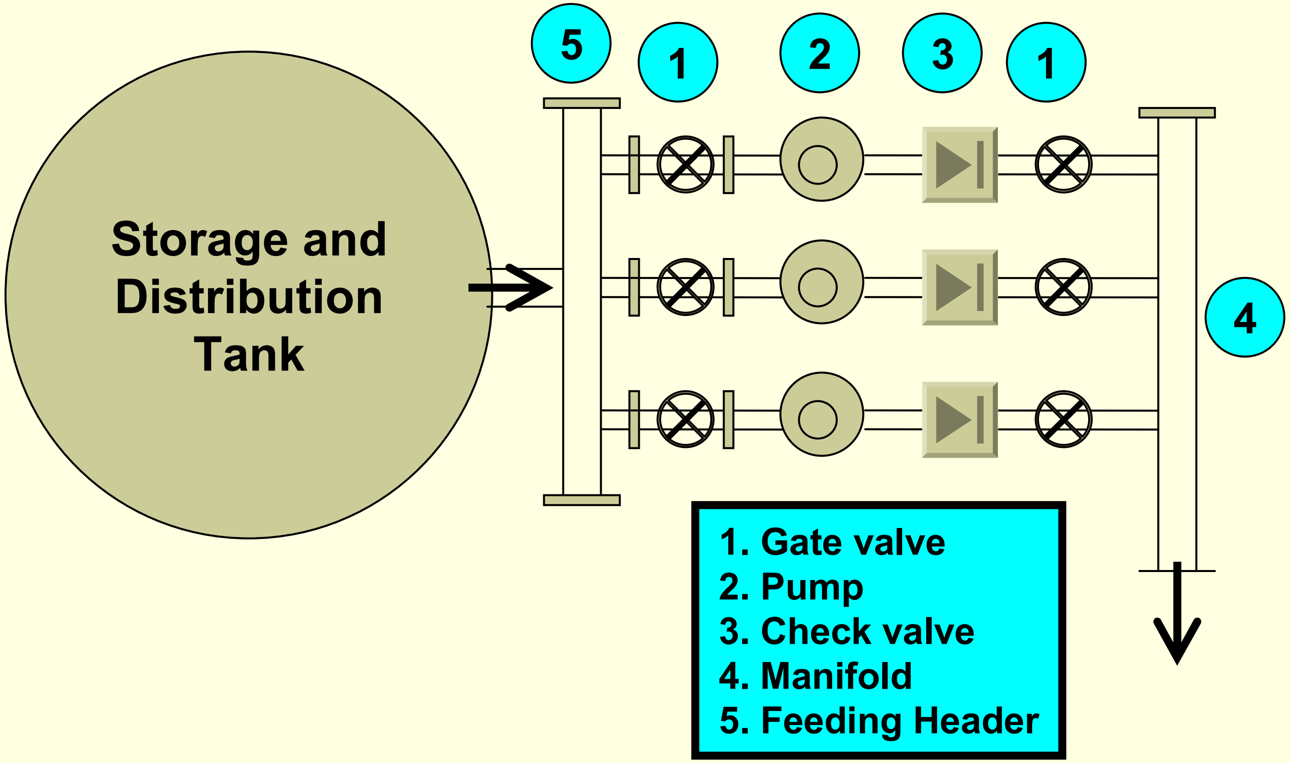


1. Gate valve
2. Pump
3. Check valve
4. Manifold

Distribution pumping station with rectangular storage and distribution tank

6.3 Distribution pumping stations

- Typical layout of distribution pumping stations:

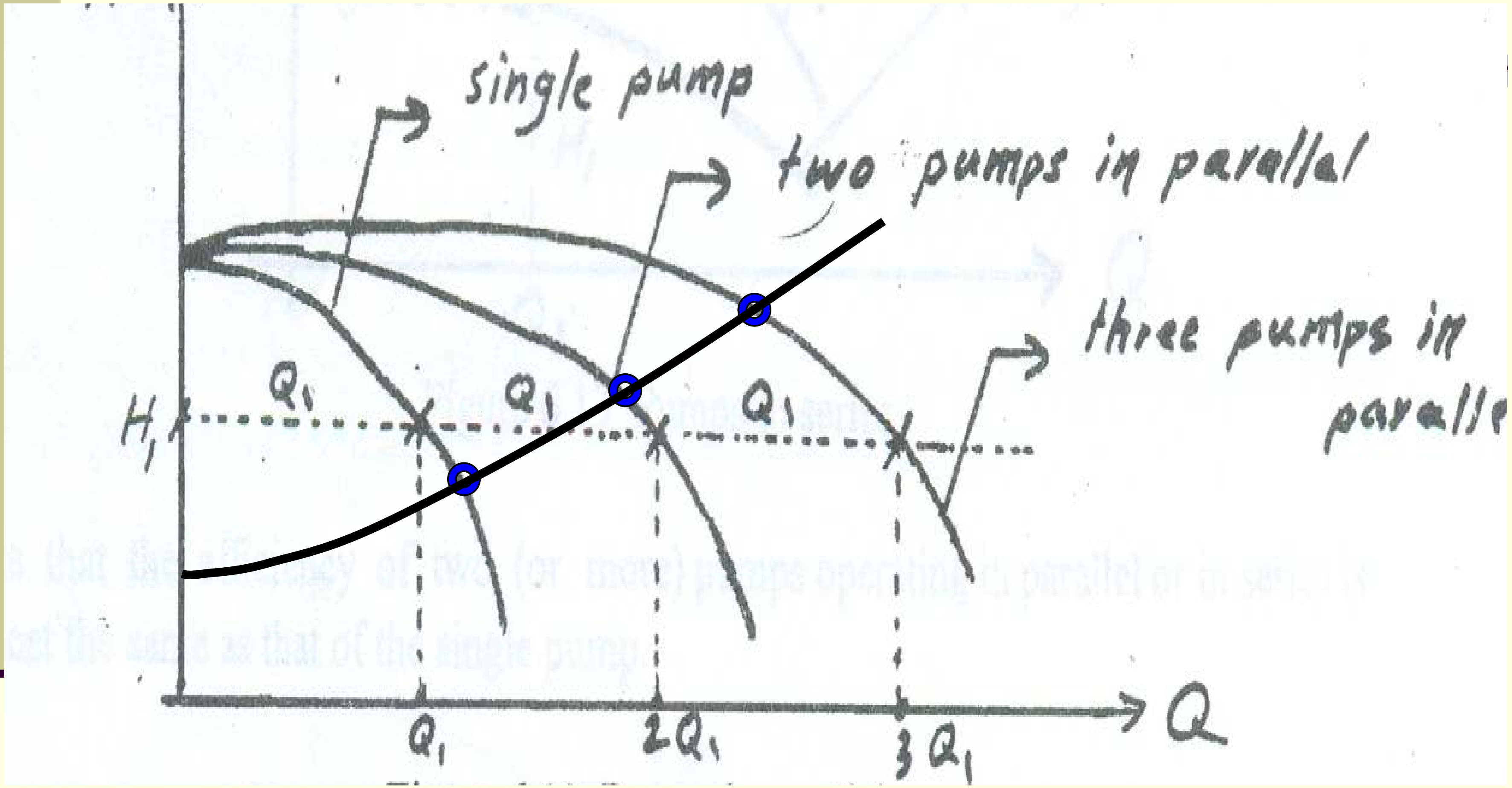


Distribution pumping station with circular storage and distribution tank

6. Design of water supply pumping stations

Distribution pumping station

6.3 Distribution pumping stations



Distribution pumping station are operated in parallel

6.3 Distribution pumping stations

- Control of distribution pumping stations :

1. Pressure switch at the discharge side of the pipe.
   If the pressure in the network increases above a preset value (for example 6 bar) the pumps will be shut down one after the other. The pressure on the delivery pipe increases at low demands when many connections are closed.

2. Level switch connected to the water distribution tank.
   If the level in the water distribution tank drops to the a pre assigned minimum level the pumps are shut off one after the other with a pre assigned intervals. The pumps will be started again one after the other when the water in the tank reaches a pre assigned level. An ultra-sound level detector is usually used for level detection.

6.4 Booster pumping stations

- Main components of booster pumping stations :

1. Dry pumps (connected in series)
2. Suction connection (pipe)
3. Delivery pipe
4. Valves
5. Stand by generator and its fuel tank (for offline large stations only)
6. Main electricity distribution panel and control
7. Service building (for offline large stations only).

6.4 Booster pumping stations

- Main types of booster pumping stations :

1. Inline booster pumps
2. Offline booster pumps

6.4 Booster pumping stations

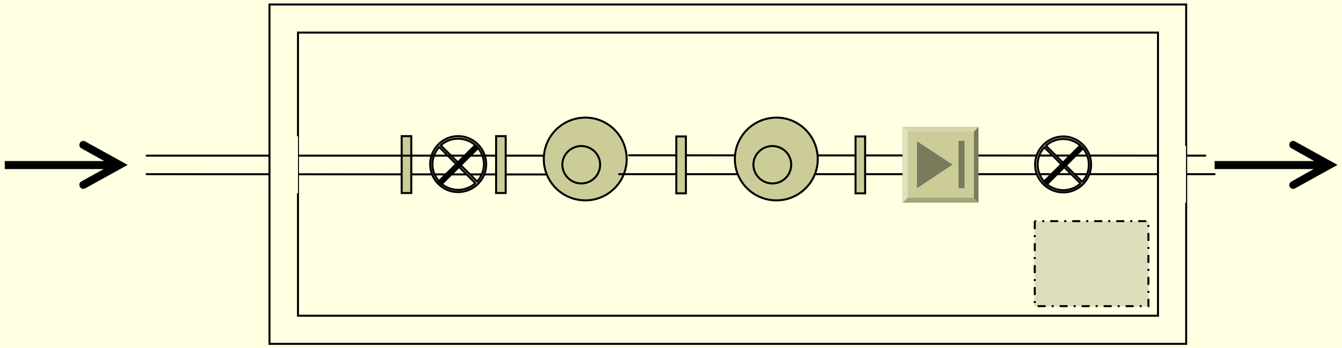
Typical inline Booster Pump

6.4 Booster pumping stations

Inline Package Booster Pump

6.4 Booster pumping stations

- Typical layouts of inline booster pumping stations:

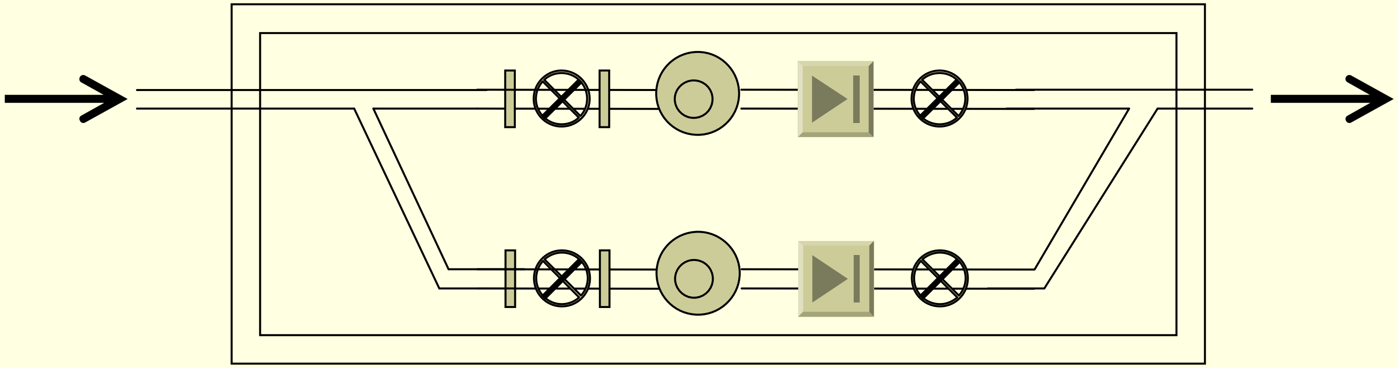


Inline booster pumping station - two pumps in series

This layouts is used when two or three pumps are able to deliver the required flow and head. If more than three pumps are needed or when the pumps are large, we should go to an offline booster pumping station.

6.4 Booster pumping stations

❑ Typical layouts of inline booster pumping stations:

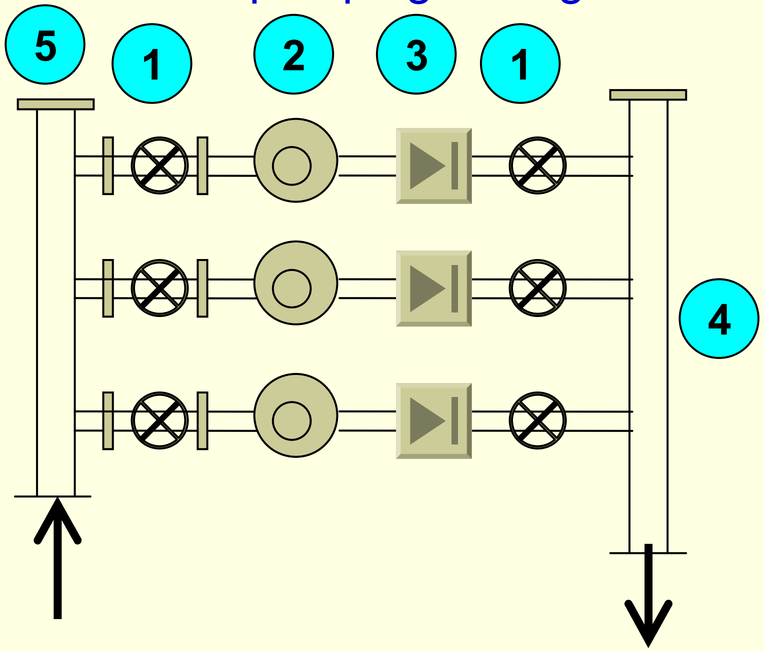


Inline booster pumping station - two pumps in parallel

This layouts is used when one pump is able to deliver the required head but not able to deliver the required flow. In this case we use two or more pumps in parallel. If more than three pumps are needed or when the pumps get large, we should go to an offline booster pumping station.

6.4 Booster pumping stations

- Typical layout of offline booster pumping arrangements:



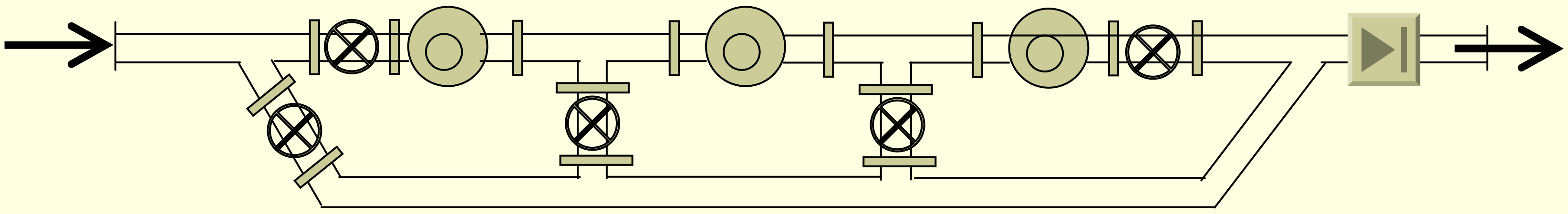
offline booster pumping station - three pumps in parallel

This configuration is used when one pump is able to give the required head but not able to give the required flow, so we need more than one pump in parallel to deliver the total flow.

Lecture 6: Design of waster supply pumping stations

6.4 Booster pumping stations

- Typical layout of offline booster pumping arrangements:



offline booster pumping station - three pumps in series

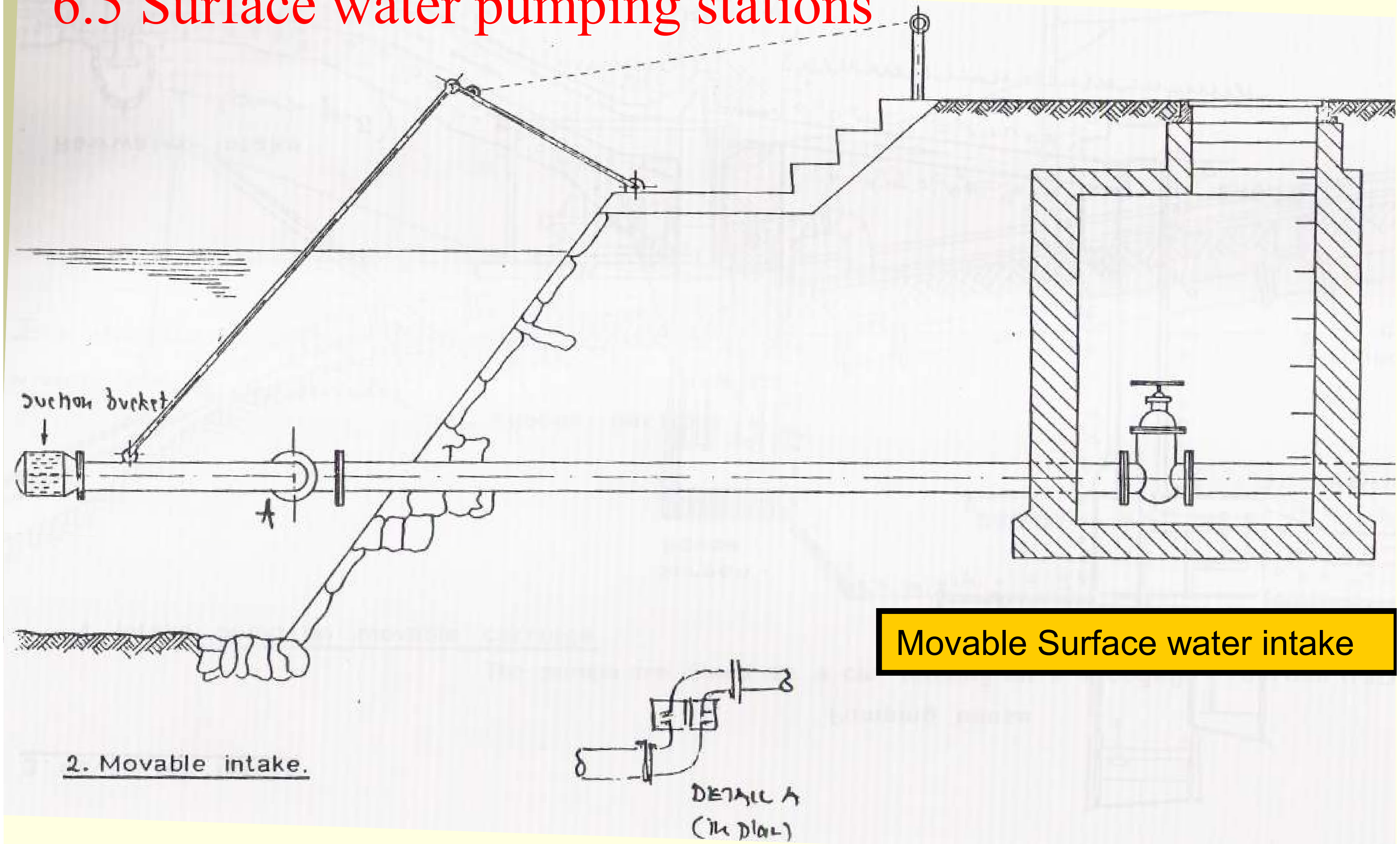
This configuration is used when one pump is able to give the required flow but not able to give the required head alone, so we need more than one pump in series to deliver the total head. Note that we use a bypass line to achieve flexibility in operation When one or more pumps are out for maintenance.

6.5 Surface water pumping stations

- Main components of surface pumping stations :

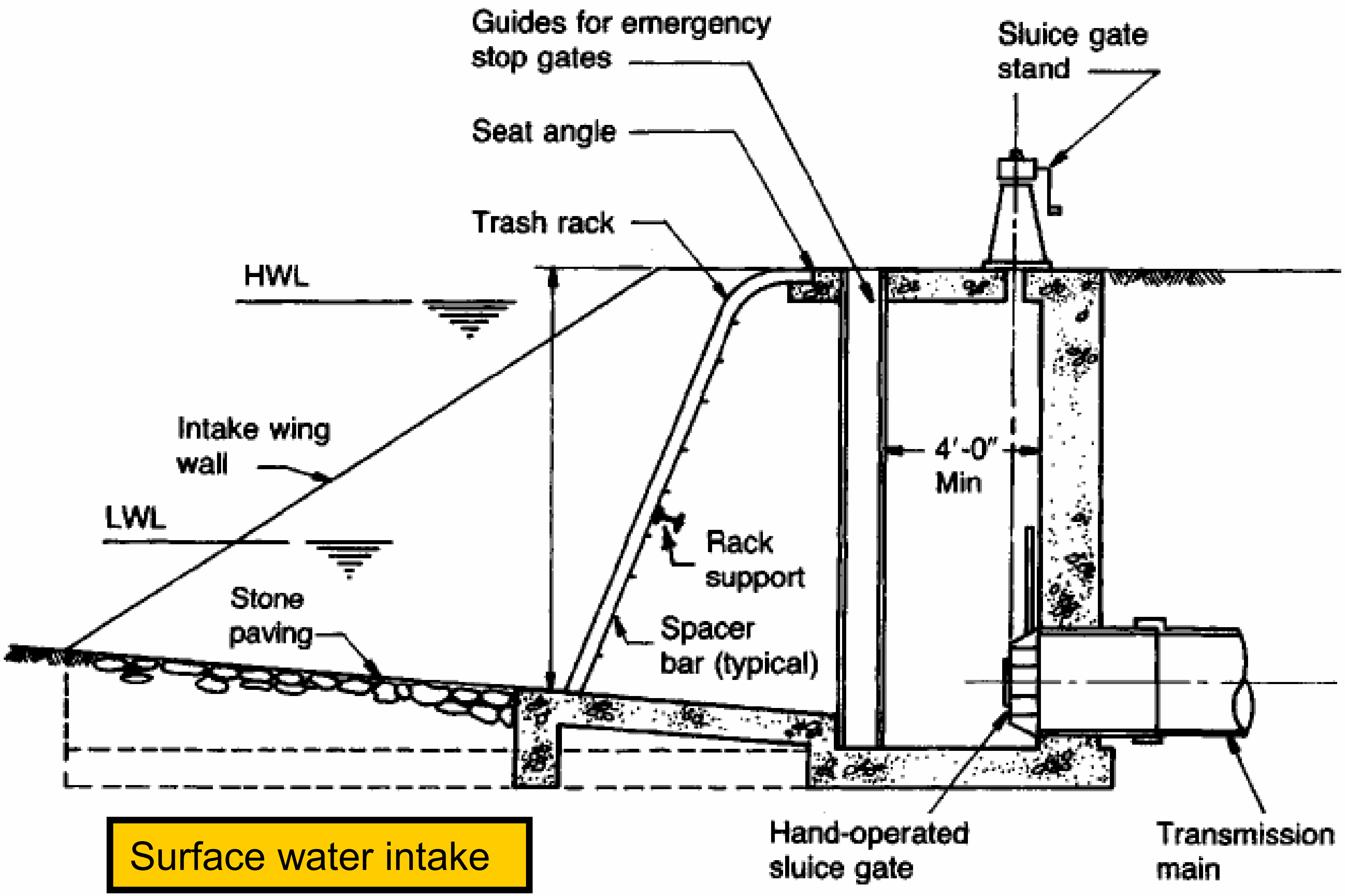
1. Water Intake structure
2. Submersible or dry pumps (connected in parallel)
3. Suction connection (pipe)
4. Delivery pipe
5. Valves
6. Stand by generator and its fuel tank
7. Main electricity distribution panel and control
8. Service building

6.5 Surface water pumping stations



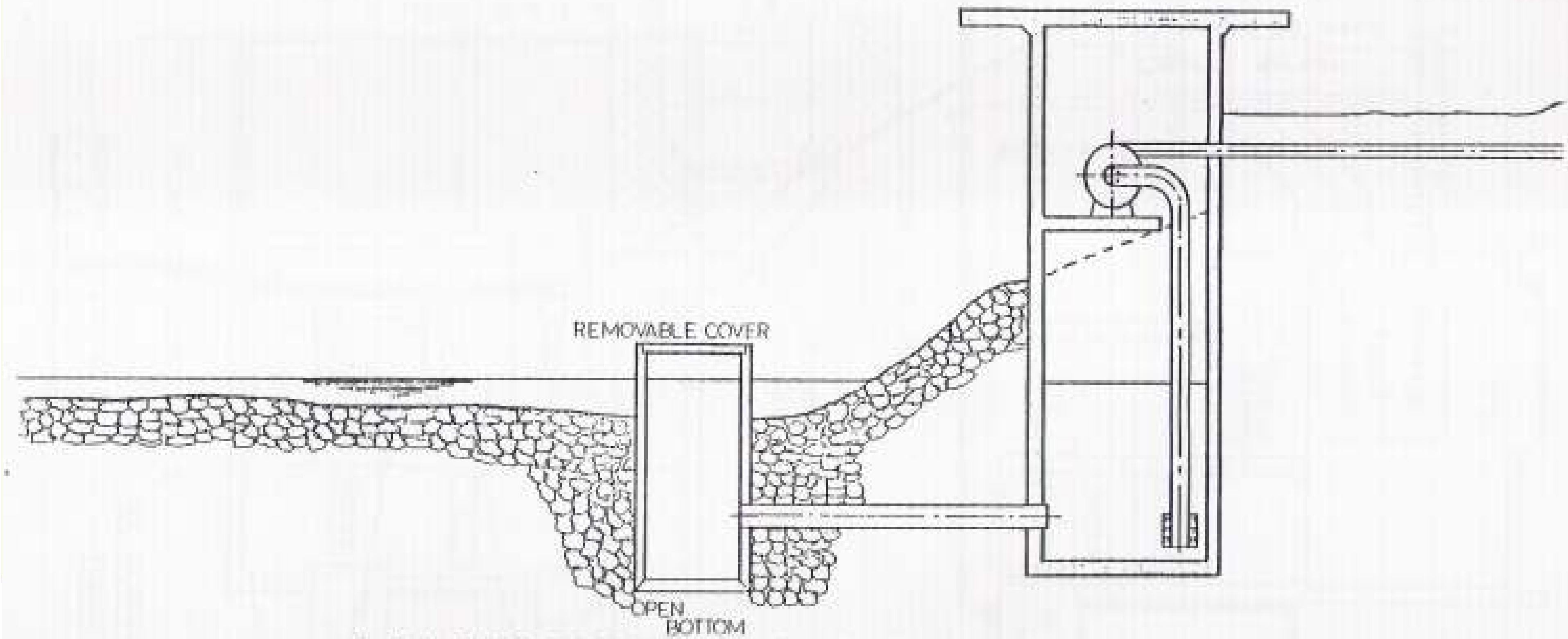
Movable Surface water intake

6.5 Surface water pumping stations



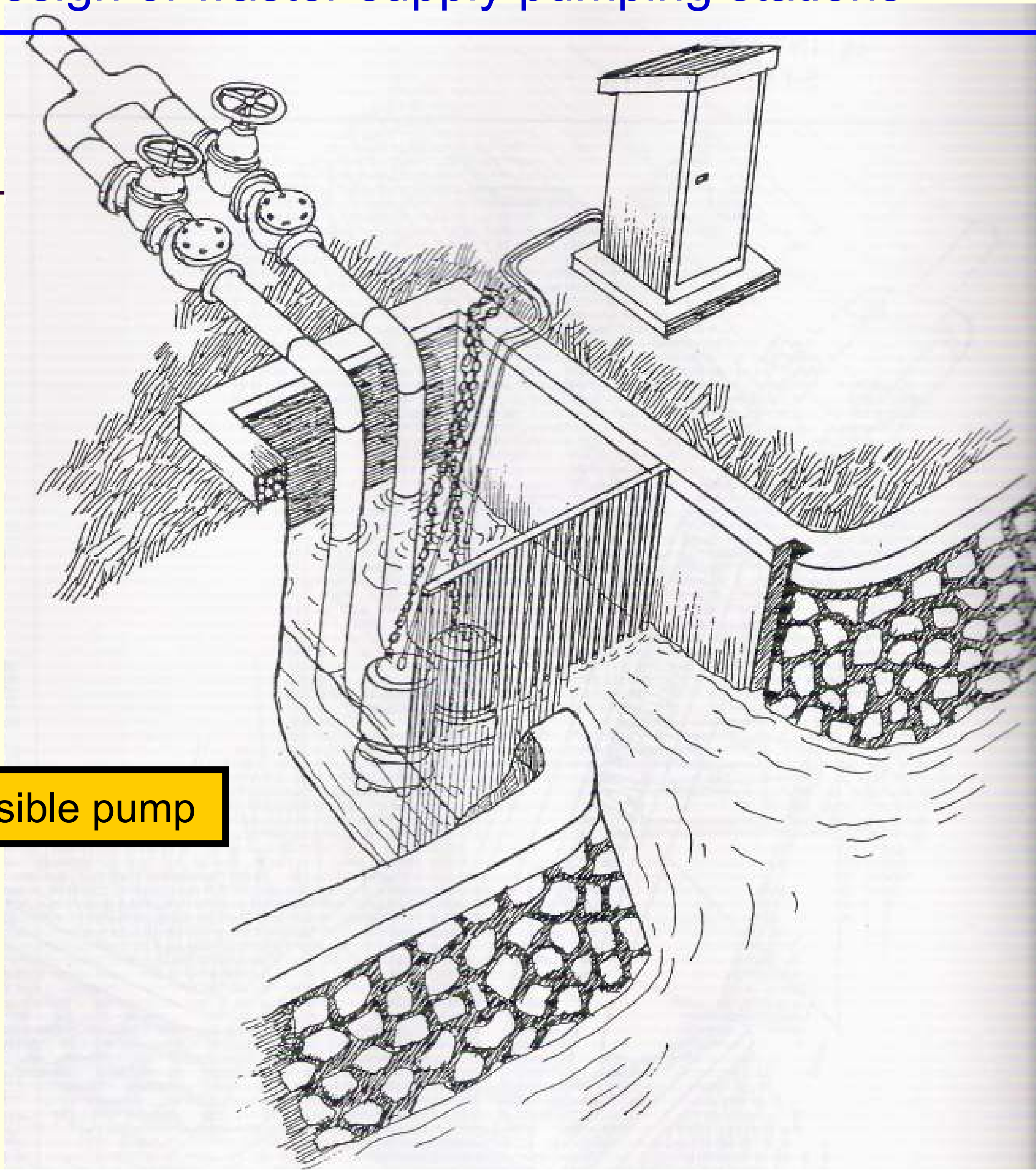
Surface water intake

6.5 Surface water pumping stations



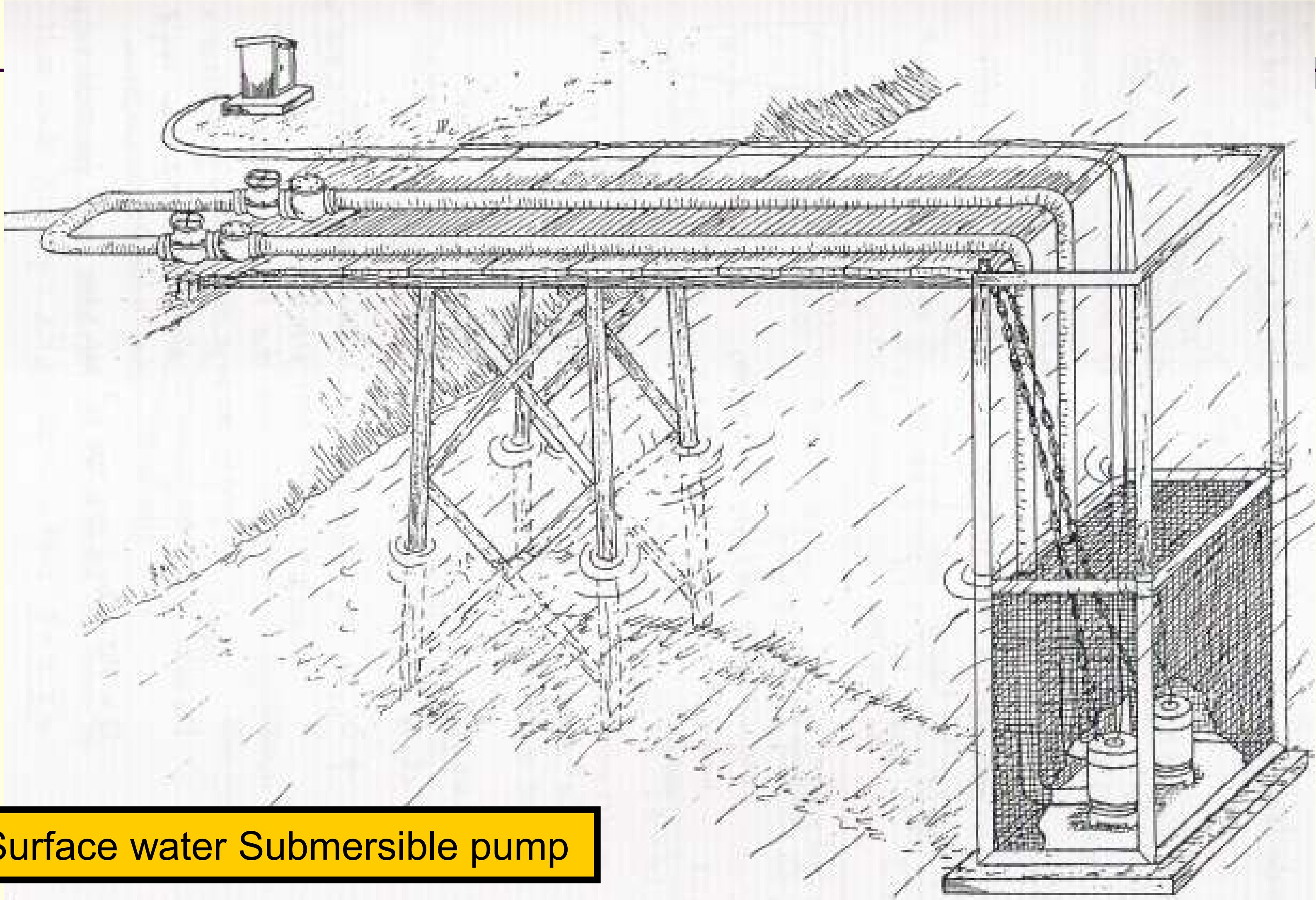
Dry pumps and Surface water intake

Lecture 6: Design of waster supply pumping stations



Surface water Submersible pump

6.5 Surface water pumping stations



Surface water Submersible pump

Pumping Stations Design

For Infrastructure Master Program
Engineering Faculty-IUG

Lecture 7: Design of sludge pumping stations

Dr. Fahid Rabah

Water and environment Engineering

frabah@iugaza.edu

Lecture 7: Design of sludge pumping stations

7.1 General introduction

- Sludge definition :

Sludge is a fluid composed of a mixture of water and solids.
Some of these solids are dissolved and some are suspended. The suspended solids are the most important part and their concentration is in the range of 1% to 10% according to sludge type.

- Types of sludge :

1. Primary sludge (solids concentration 3-5%)
2. Secondary sludge (solids concentration 1-1.5%)
3. Digested sludge (Solids concentration 6-7%)
4. Thickened sludge (Solids concentration 8-10%)

7.2 Hydraulics of Sludge pumping

- The sludge is a non-Newtonian fluid, so its flow properties is different from Newtonian fluids such as water and wastewater. Thus, the equations used to determine the hydraulics of water and wastewater doesn't apply for sludge.

- The main cause of classifying sludge as a non-Newtonian fluid is the presence of solids in the range of 1% and above.

- The hydraulics of sludge pumping is highly dependent on the concentration and properties of the solids in the sludge.

- The sludge flow is classified into laminar flow or turbulent flow.

7.2 Hydraulics of Sludge pumping

❑Laminar flow in sludge pumping systems:

Bingham equation is used for defining the hydraulics of sludge in the laminar range:

$$\frac{h}{L} = \frac{16 s_y}{3 D \rho g} + \frac{32 \eta V}{\rho g D^2}$$

Where,

h = headless in the pipe, m
L = pipe length, m
s_y = yield stress of the sludge, N/m^2
ρ = sludge density , kg/m^3
D = pipe diameter , m
g= gravitational acceleration,9.81 m/s^2
η= coefficient of rigidity, kg/(m.s)
V = flow velocity, m/s

7.2 Hydraulics of Sludge pumping

❑Laminar flow in sludge pumping systems:

To apply the Bingham equation the flow should be laminar.
Bingham defined the lower critical velocity (V_{lc}) to check for laminar flow. If the selected design velocity of sludge pumping is less or equal to V_{lc} then the flow is laminar, otherwise the flow is turbulent.

$$V_{lc} = \frac{1000\eta + 1000\sqrt{\eta^2 + s_y \rho D^2 / 3000}}{D\rho}$$

Note:

- **The values of s_y and η are taken from figures 19.2 and 19.3.**
- **The Bingham equations are valid for laminar flow only.**

Lecture 7: Design of sludge pumping stations

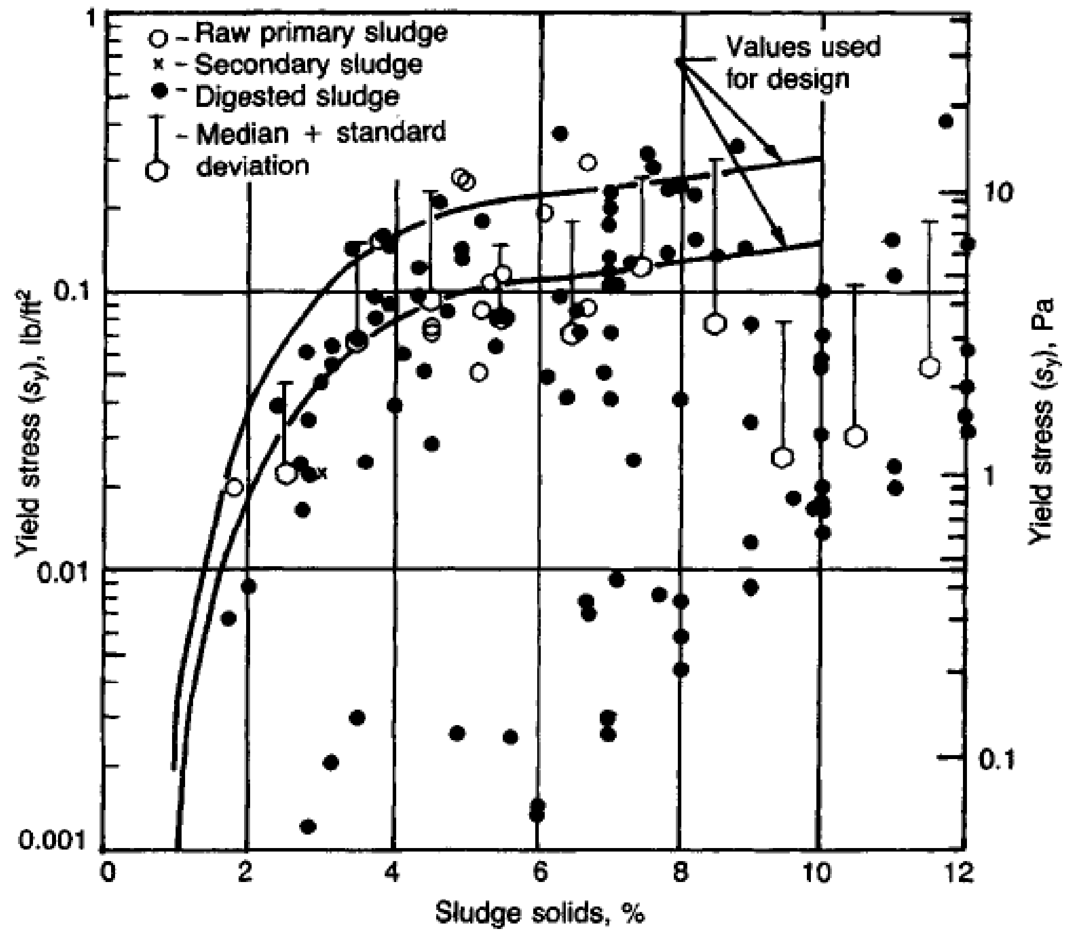


Figure 19-2. Yield stress versus sludge solids.

Lecture 7: Design of sludge pumping stations

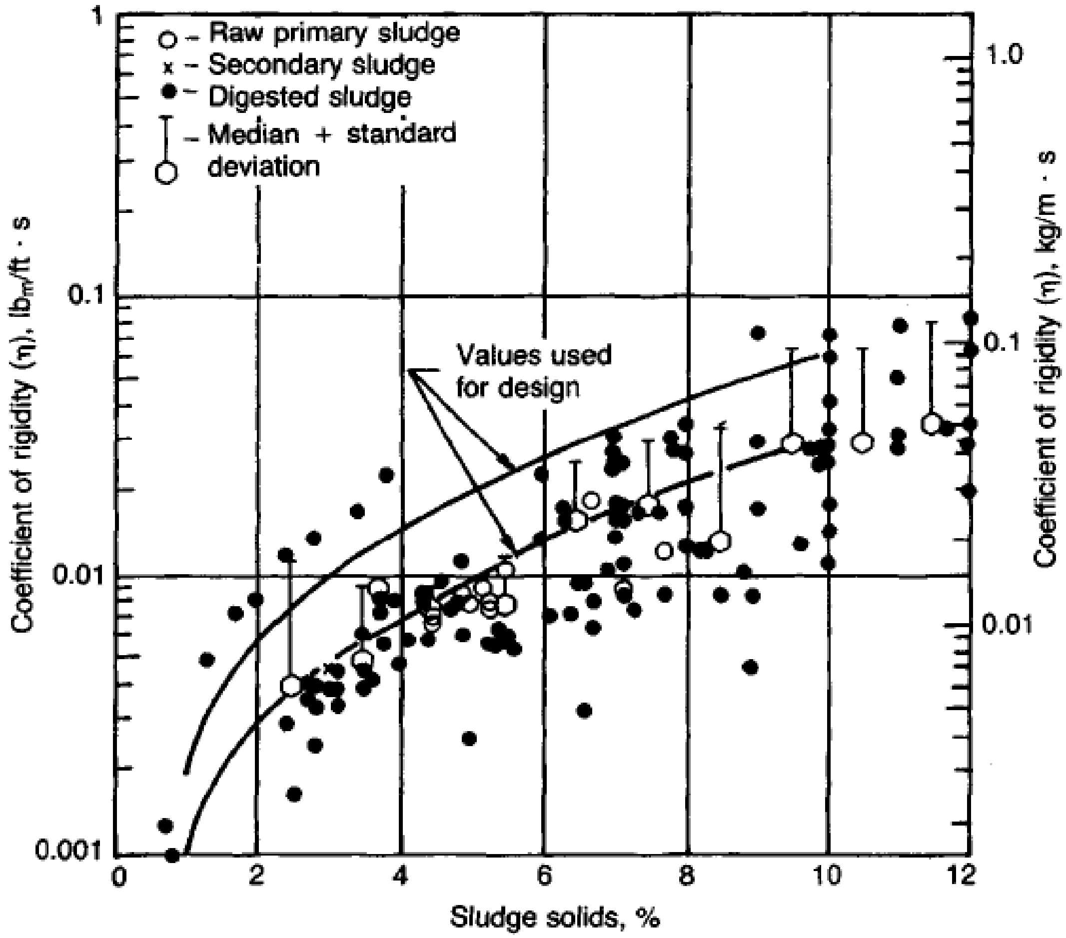


Figure 19-3. Coefficient of rigidity versus sludge solids.

Lecture 7: Design of sludge pumping stations

7.2 Hydraulics of Sludge pumping

❑Turbulent flow in sludge pumping systems:

Hanks and Dadia equation is used for defining the hydraulics of sludge in the turbulent range:

$$\frac{H}{L} = \frac{2fV}{gD}$$

or

$$\Delta p = \frac{2f\rho LV}{D}$$

Where,

H = headless in the pipe, m
L = pipe length, m
Δ p = pressure drop, N/m^2
ρ = sludge density , kg /m^3
D = pipe diameter , m
G = gravitational acceleration,9.81 m/s^2
f = friction factor
V = flow velocity, m/s

Lecture 7: Design of sludge pumping stations

7.2 Hydraulics of Sludge pumping

The friction coefficient (f) is found from figure 19.6 after calculating (H_e) and (R) from the following equations:

Hedstrom number: $$H_e = \frac{D^2 s_y \rho}{\eta^2}$$

Reynolds number: $$R = \frac{DV\rho}{\eta}$$

Where,

H_e = Hedstrom number , dimensionless
R = Reynolds number , dimensionless
s_y = yield stress of the sludge, N/m^2
ρ = sludge density , kg /m^3
D = pipe diameter , m
H = coefficient of rigidity, kg/(m.s)
V= flow velocity, m/s

7.2 Hydraulics of Sludge pumping

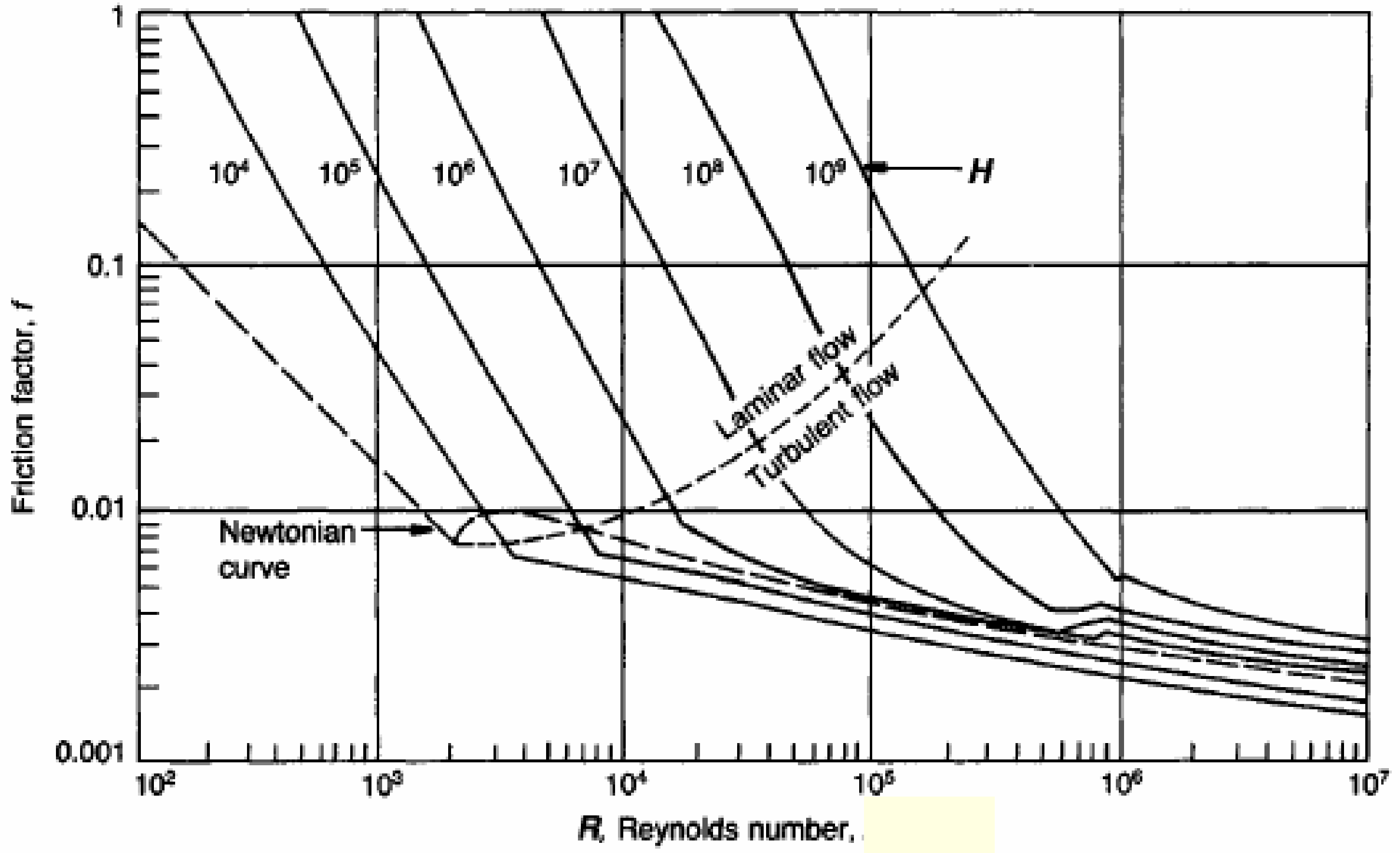


Figure 19-6. Theoretical friction factor for sludge analyzed as a Bingham plastic. After EPA [2].

Lecture 7: Design of sludge pumping stations

7.2 Hydraulics of Sludge pumping

- Approximate method for headless calculations

Figure 19.5 gives a the headless as a factor of the headless calculated for water or wastewater using the equation of Hazen Williams with a coefficient of Friction C=140.

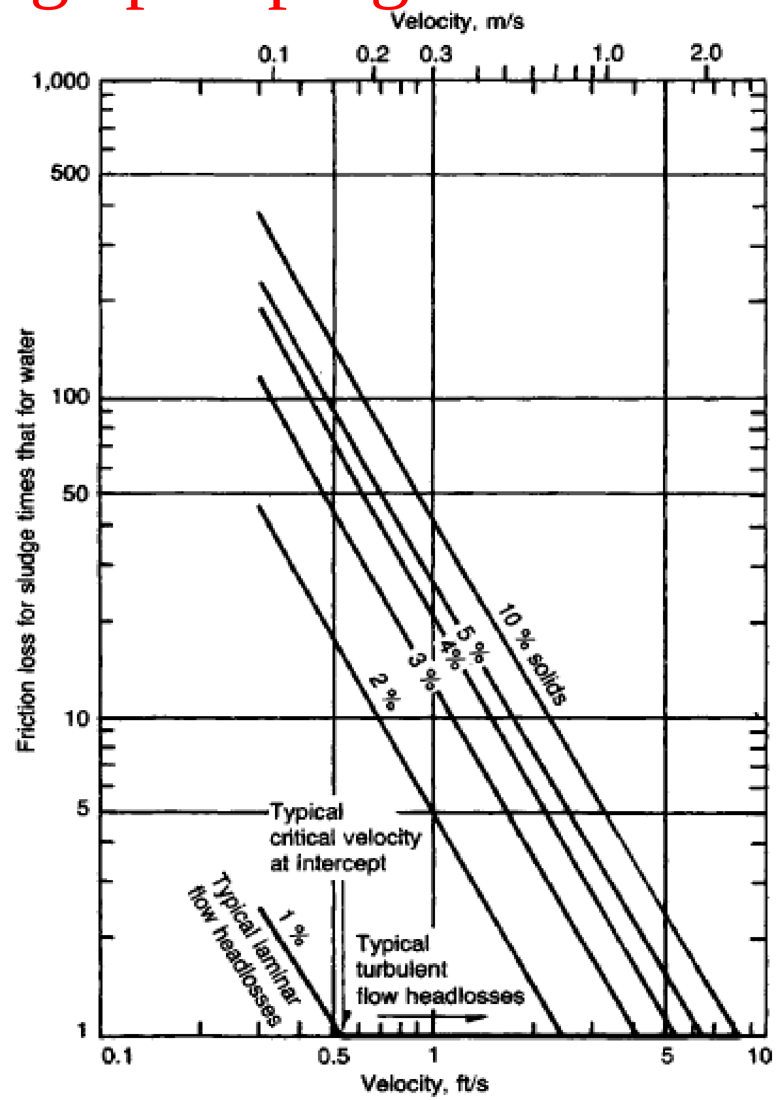


Figure 19-5. Recommended design curves. Headloss prediction for routine operation.

7.2 Hydraulics of Sludge pumping

- Typical sludge pumping system curve compared to water pumping system curve

Using the headless equations you can draw the system curve For sludge systems. Figure 19.10 is an example of a system curve for a 6 in pipe. Notice the difference between The system curve for water and For sludge having different Solids concentrations.

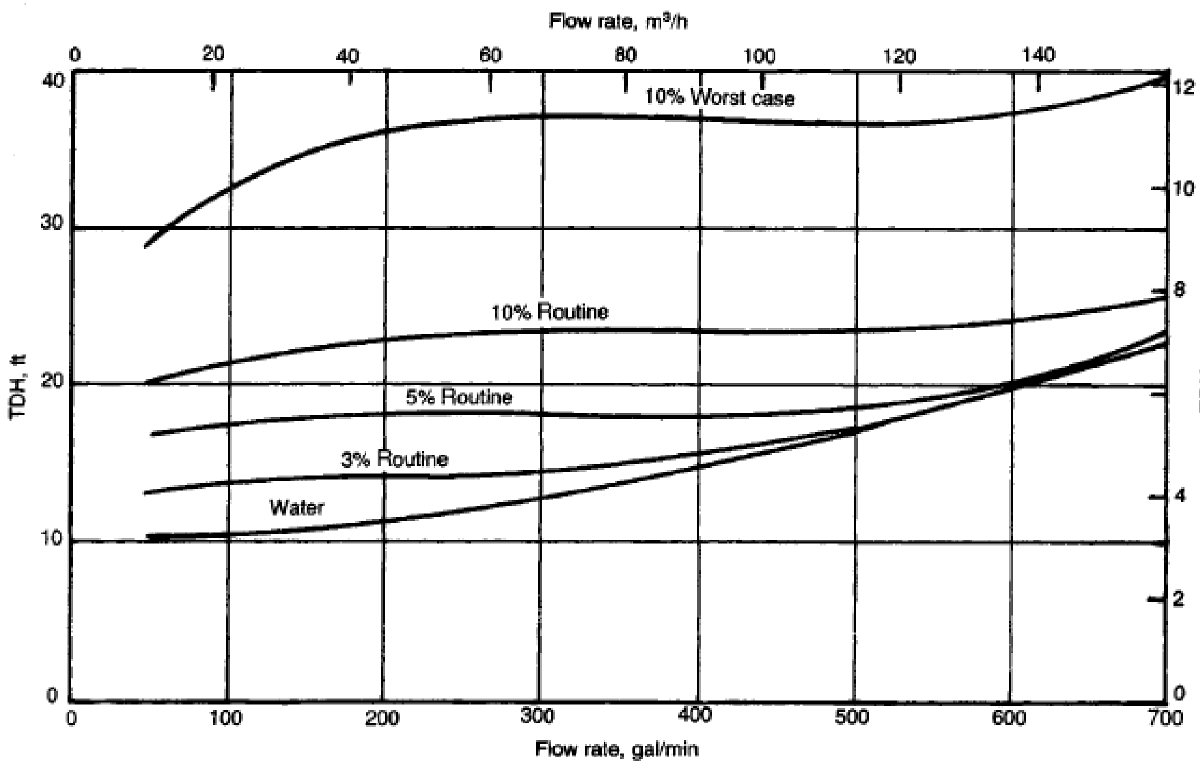


Figure 19-10. System H-Q curves for sludge in a 150-mm (6-in.) pipe.

7.2 Hydraulics of Sludge pumping

- Sludge pressure pipes sizing:
  - Max. design velocity is 1.8 m/s.
  - Min velocity is 0.6 m/s
  - Min. pipe diameter is 6 in.
  - Pump connections should not be less than 4 in.

7.3 Sludge pumping stations

- Sludge pumping stations types:

1. Submersible pumps
2. Dry pumps

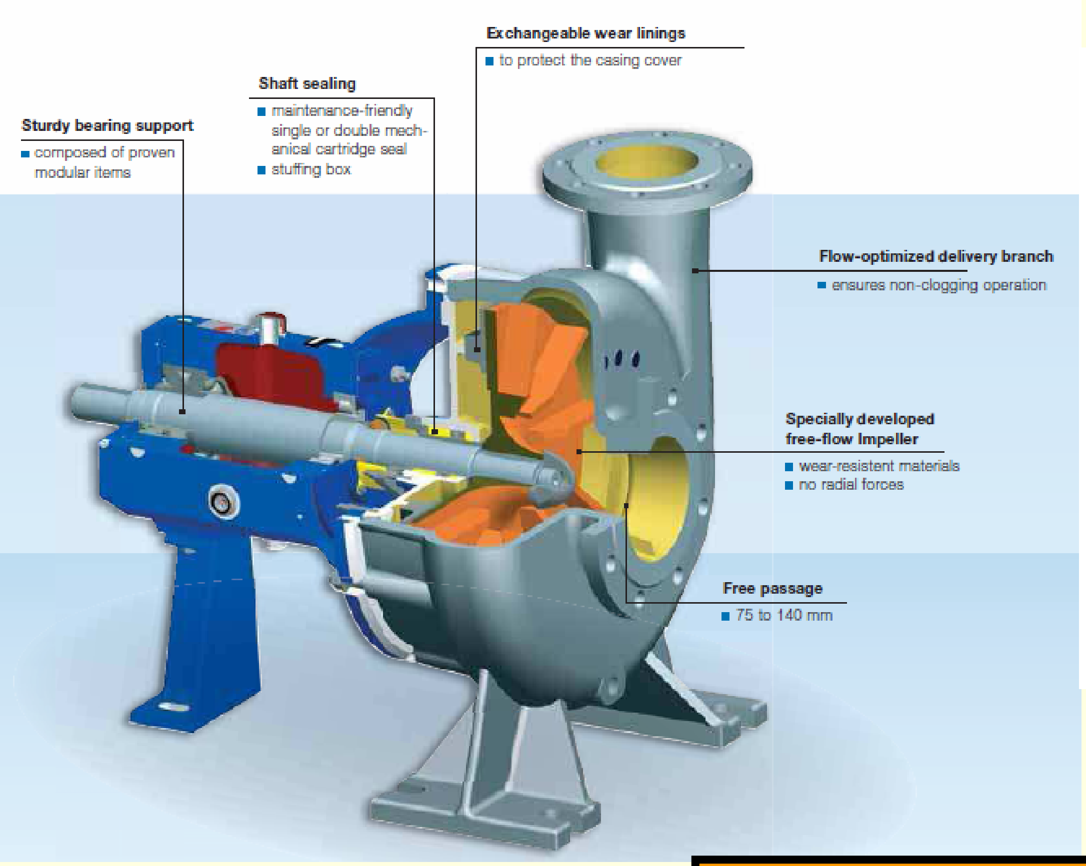
Submersible pumps and dry pumps are similar to wastewater pumping stations in terms of wet wells and dry wells and the piping layout and valves.

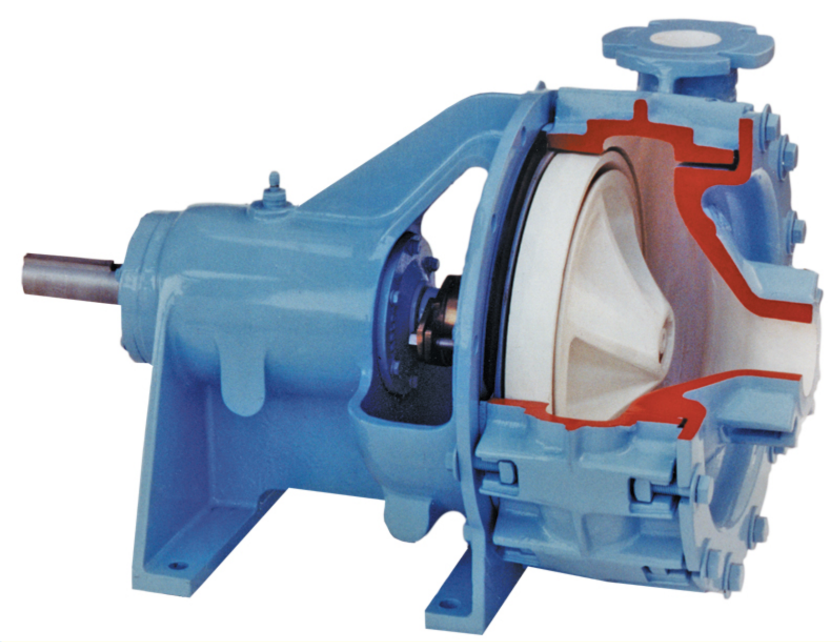
- Types of pumps used for Sludge pumping :

1. Centrifugal Vortex pumps
2. Diaphragm pumps
3. Rotary pumps
4. Progressive cavity pumps

7.3 Sludge pumping stations

- ❑ Sludge pumping stations types:

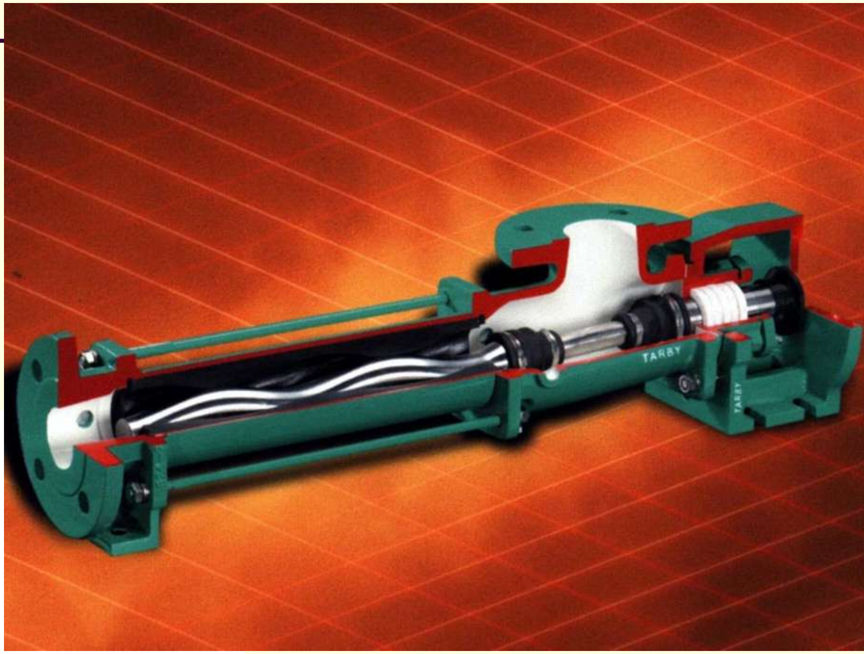




Centrifugal Vortex pump

7.3 Sludge pumping stations

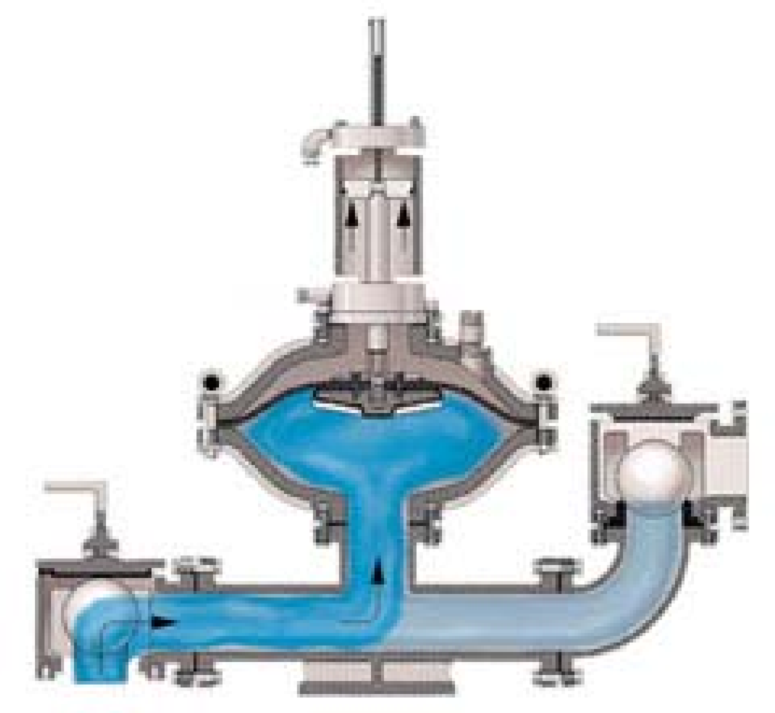
- Sludge pumps types:



Progressive cavity pump

7.3 Sludge pumping stations

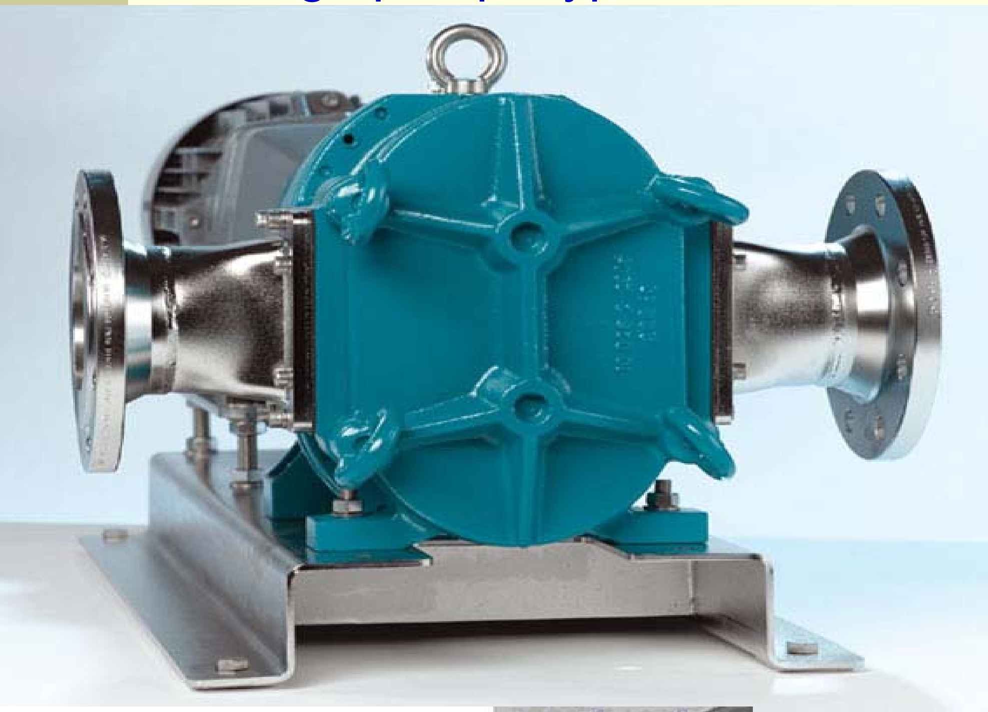
- Sludge pumps types:

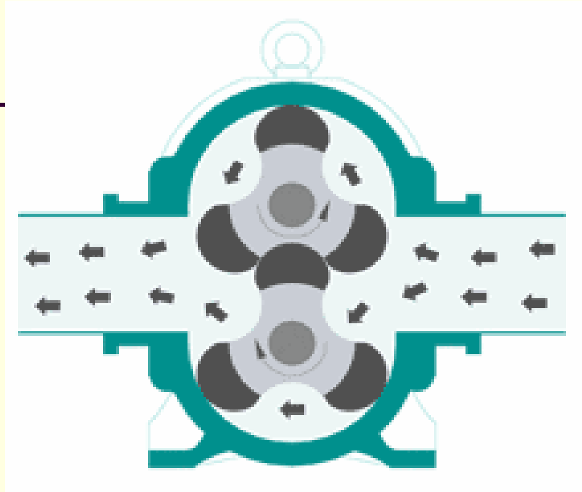


Diaphragm pump

7.3 Sludge pumping stations

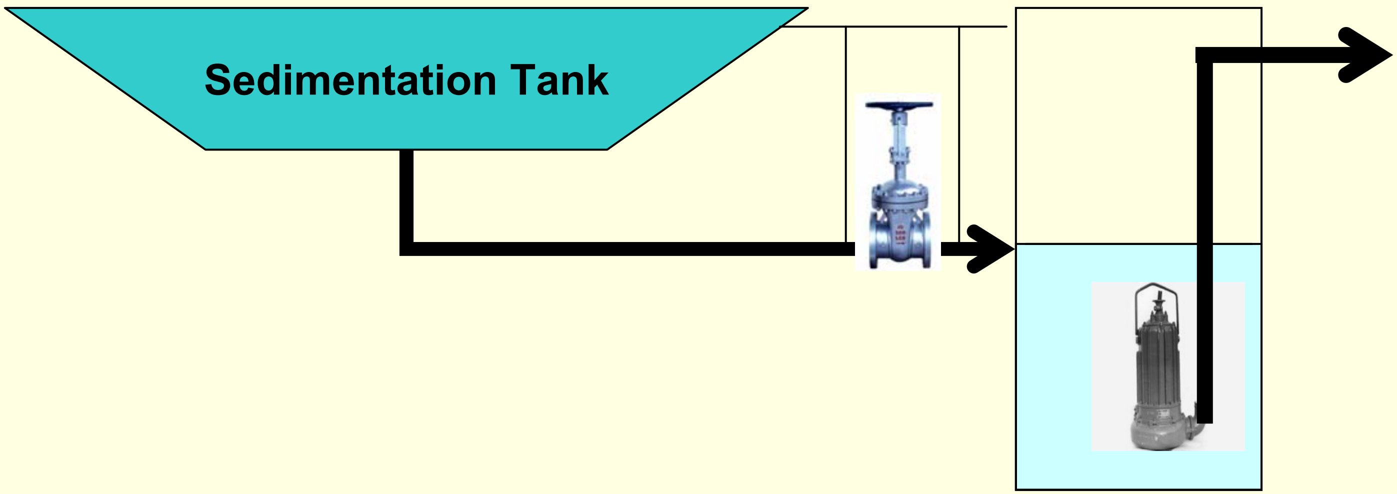
- Sludge pumps types:





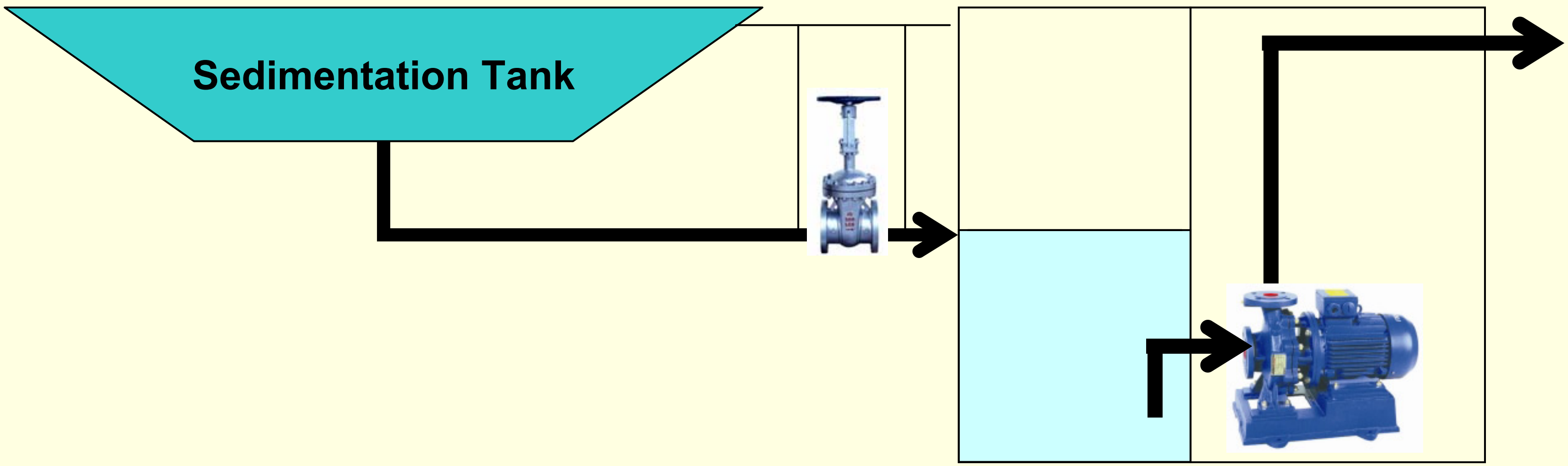
Rotary Lobe pump

7.4 Typical location of sludge pumping stations



Submersible sludge pumping station

7.4 Typical location of sludge pumping stations



Dry well sludge pumping station