Mech. Power → Hydraulic Pump → Hydraulic Power

$T \cdot \omega$ → Hydraulic Pump → $P \cdot Q$

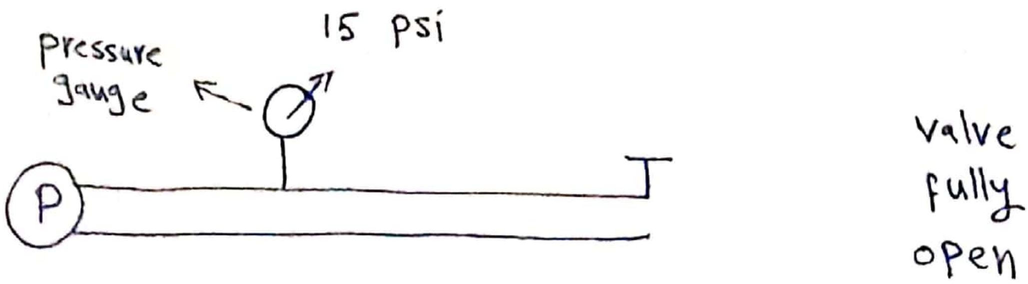
- Positive displacement
  - Reciprocating
  - Rotary
    - gear
    - vane
    - piston
- Dynamic pumps
  - Centrifugal
  - Axial

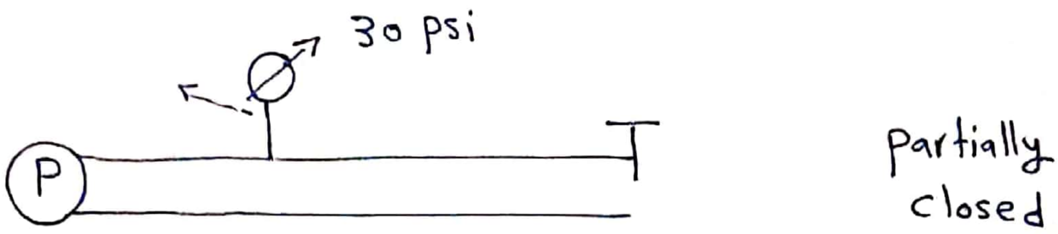
IMP:

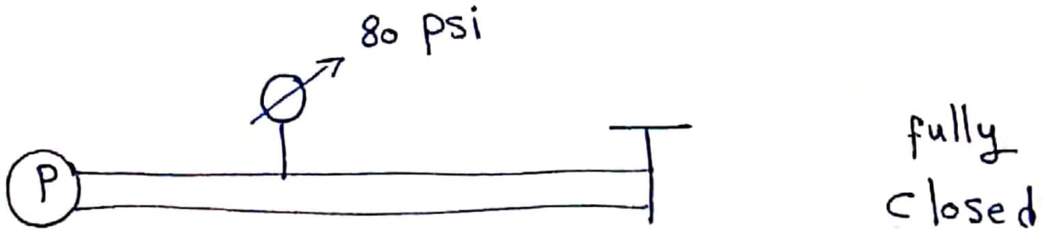
pumps Create flow, not pump pressure

→ pressure is to overcome : - elevation difference
- resistance
- Loads

→ To make it more clear :







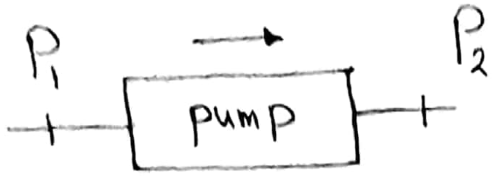
→ As resistance increase [valve closes more]

↳ pressure reading rise

→ which indicates that pump pressure is result of resistance.

Extra notes :

* Pump Pressure :



$$\text{pump pressure} = \underbrace{P_2}_{\text{delivery side}} - \underbrace{P_1}_{\text{suction side}}$$

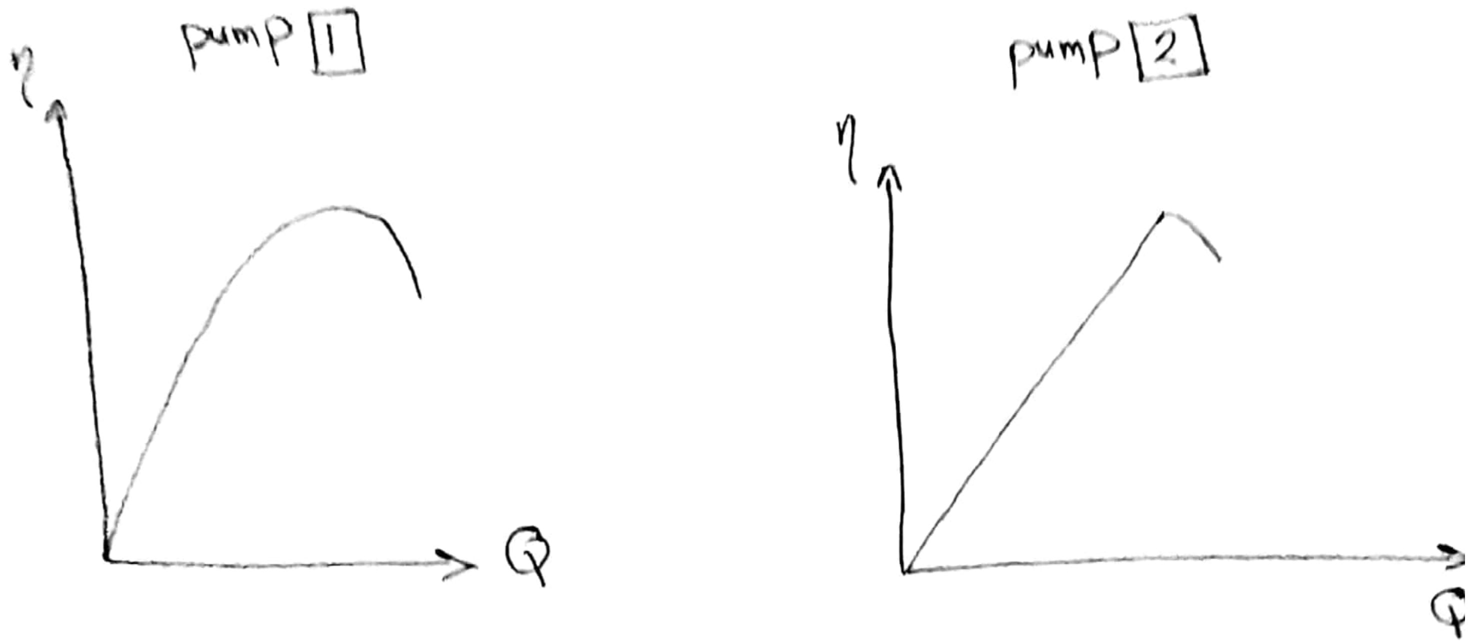
* Prymover that drives pump
  → Electric motor
  → Diesel engine
  → gas turbine
  → Steam turbine

← لازم يكون عندي أكتر من Prymover

← دة في حالة ان الكهربا قطعت بس واحد منهم

→ If we have 2 pumps having same max efficiency.

↳ which to choose?



→ we will choose pump having a flatter curve

→ because if there is change in operating conditions

[1] if η-curve is flat → small change in efficiency

[2] if η-curve is steep → high or considered change in efficiency

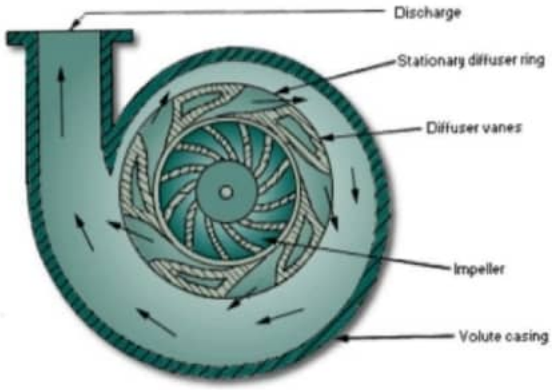
→ So, we will choose pump [1]

→ Components :

① impeller

② Casing

③ diffuser [optional]



→ Contribution to produced head :

impeller → 70 - 80 %

diffuser and casing → The rest

→ when to use diffuser ?

→ when spacing between impeller and casing is relatively high.

Figure 20 shows the **Contribution** of **Each Section** of the **Pump** to the **Total Head** developed by the **Pump**.

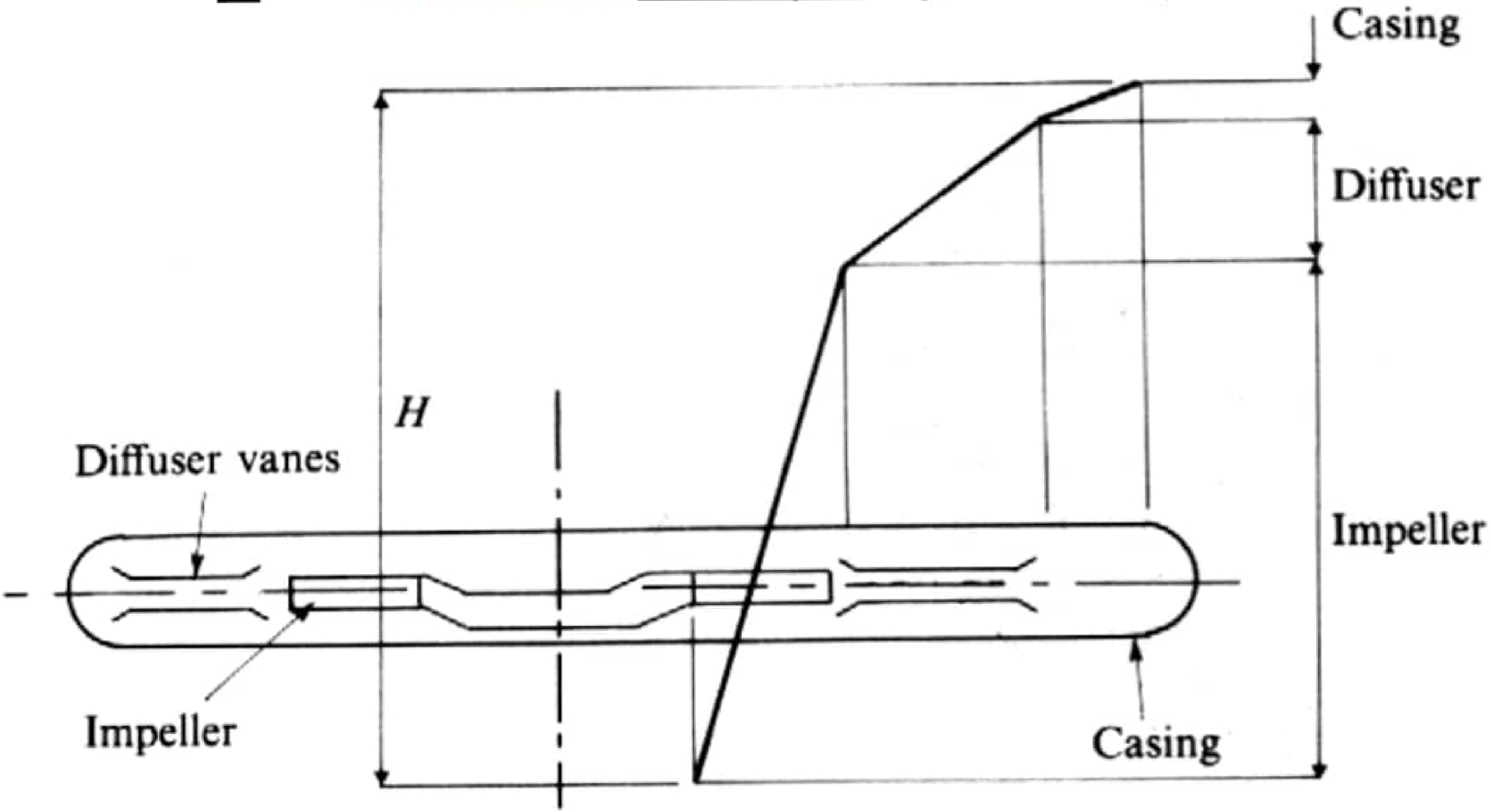


Figure 20 Head Rise across **a Centrifugal Pump**

→ no diffuser

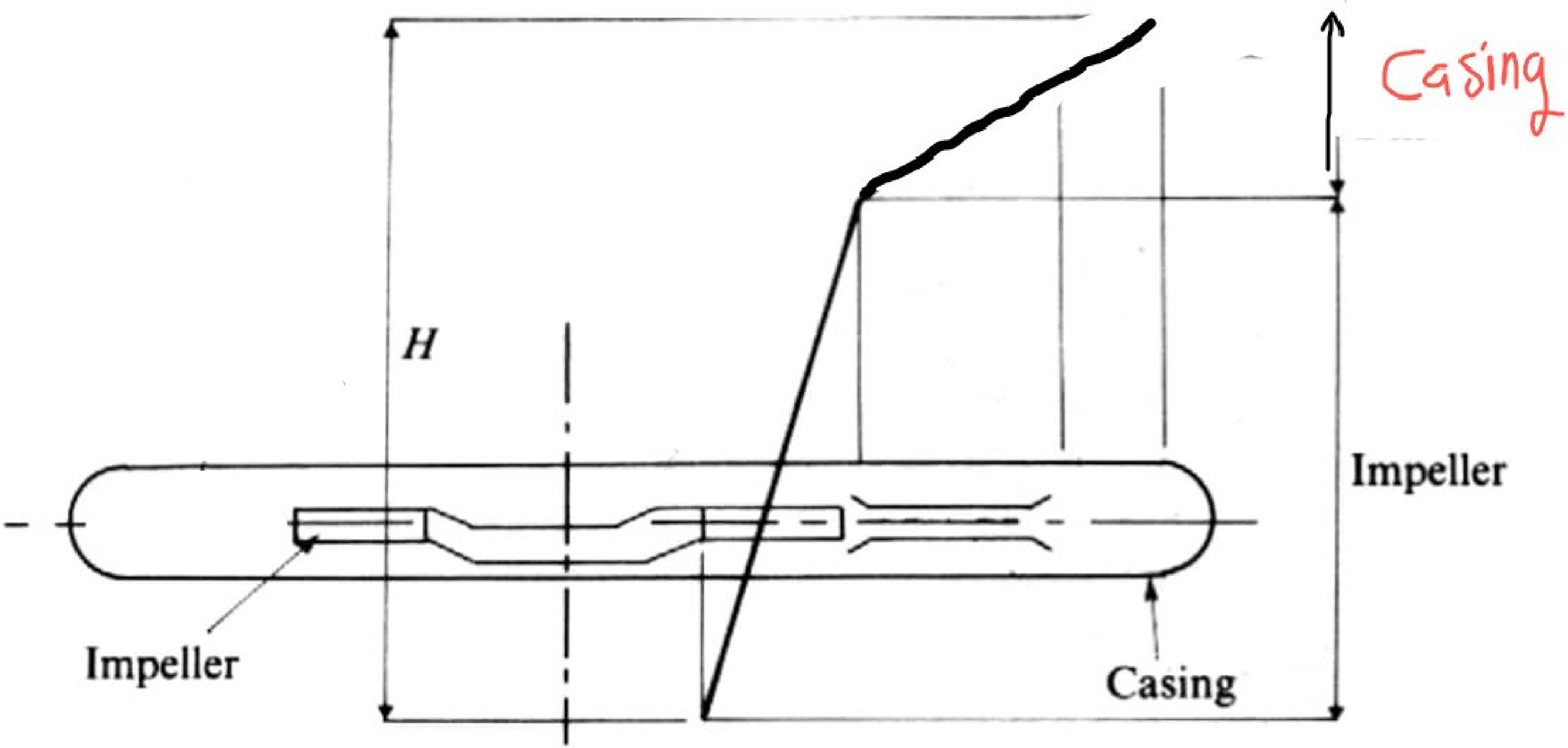
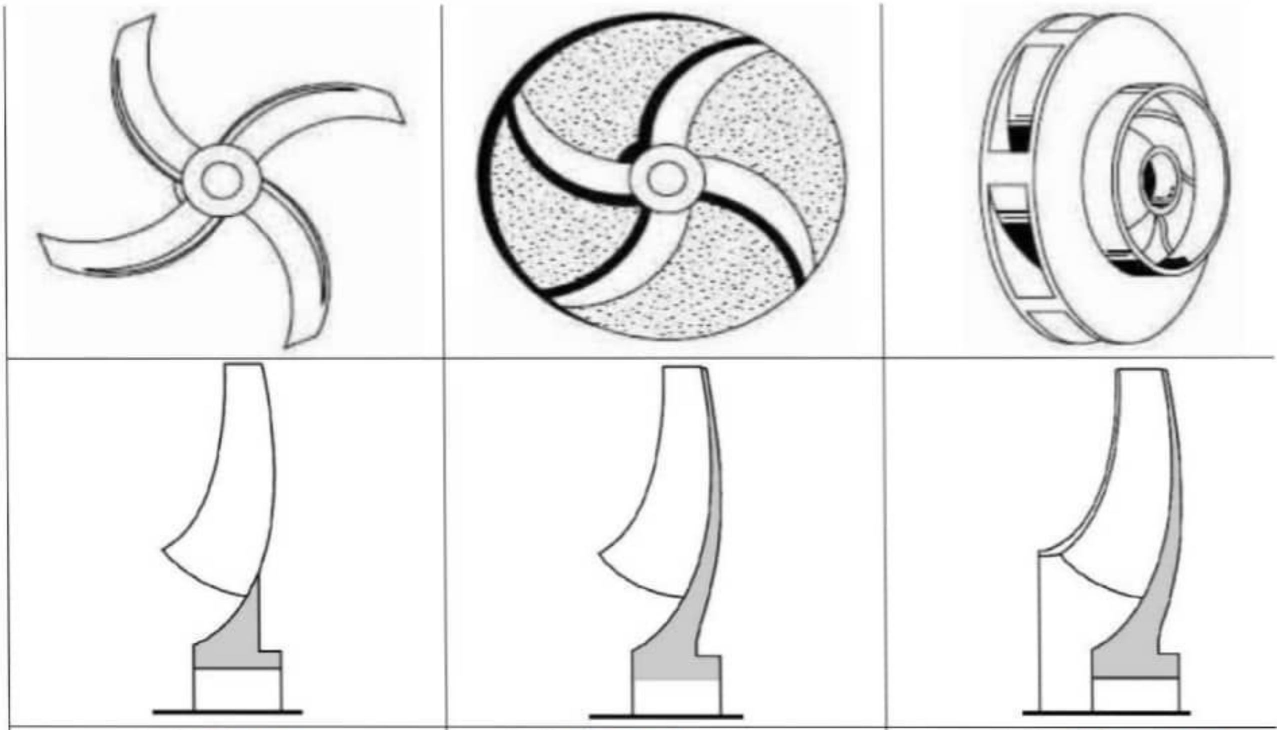


Figure 20 **Head Rise** across a **Centrifugal** Pump

Impeller

→ <u>Shape:</u>



(a) Open | (b) Semi-Open | (c) Closed

Closed type: → for very pure and clean liquids

Produce higher head
Has higher efficiency compared to open type

open type → for dirty liquids

→ **ImP:**

$$\eta_o = (\eta_i \; \eta_c)\, \eta_m \, \eta_v$$

$$= \eta_{hyd}$$

→ **in case:** $\eta_v = 100\ \%$

$$\boxed{\eta_o = \eta_{hydr}\, \eta_m}$$

#

→ Very important :

→ if we have different values for

η_o , η_m , η_v ...

→ which one is η_o ?

1. 80 %
2. 70 %
3. 90 %
4. 85 %

→ Choose Lowest Value [70 %]

سؤال امتحان
مهم جدا

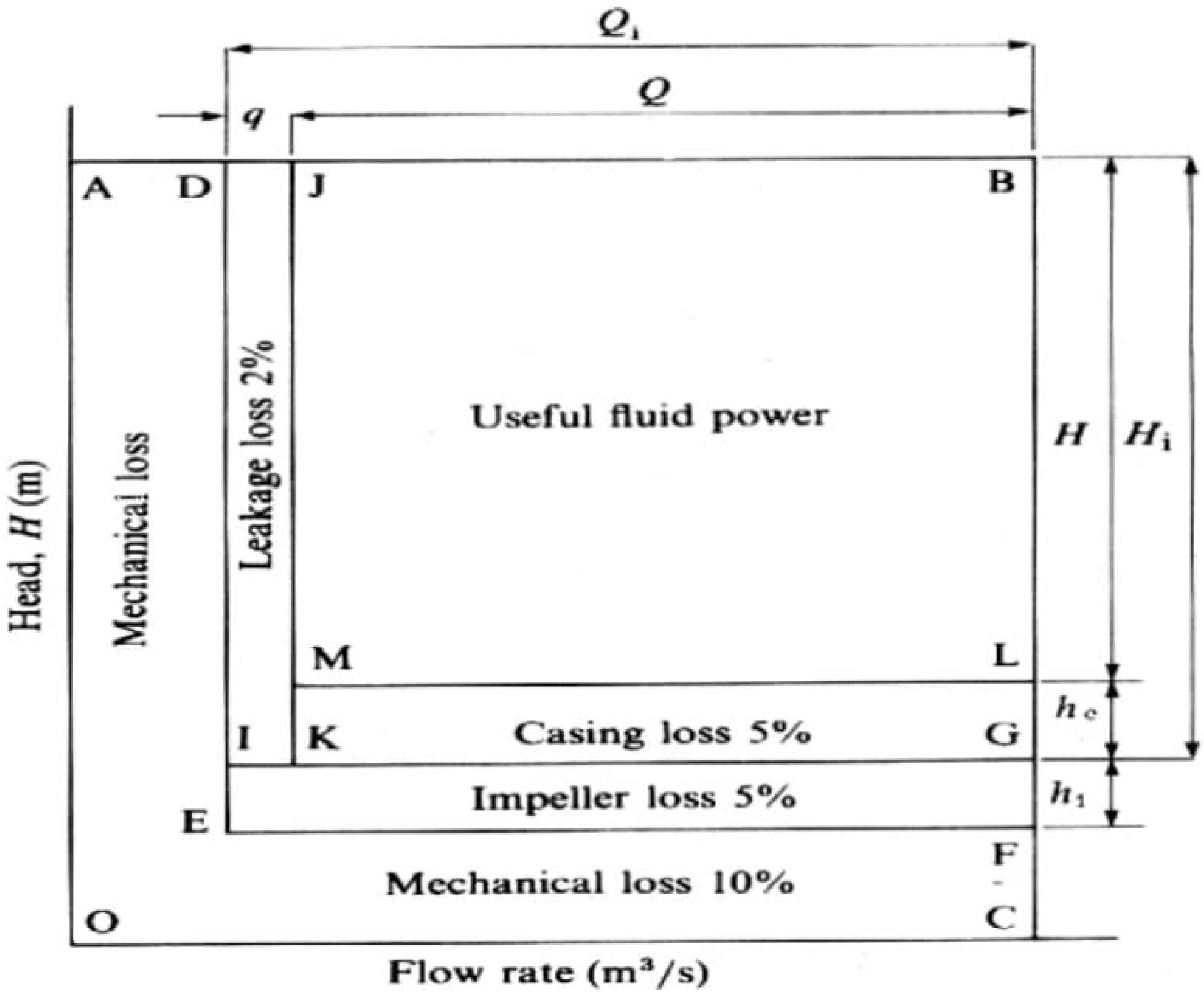
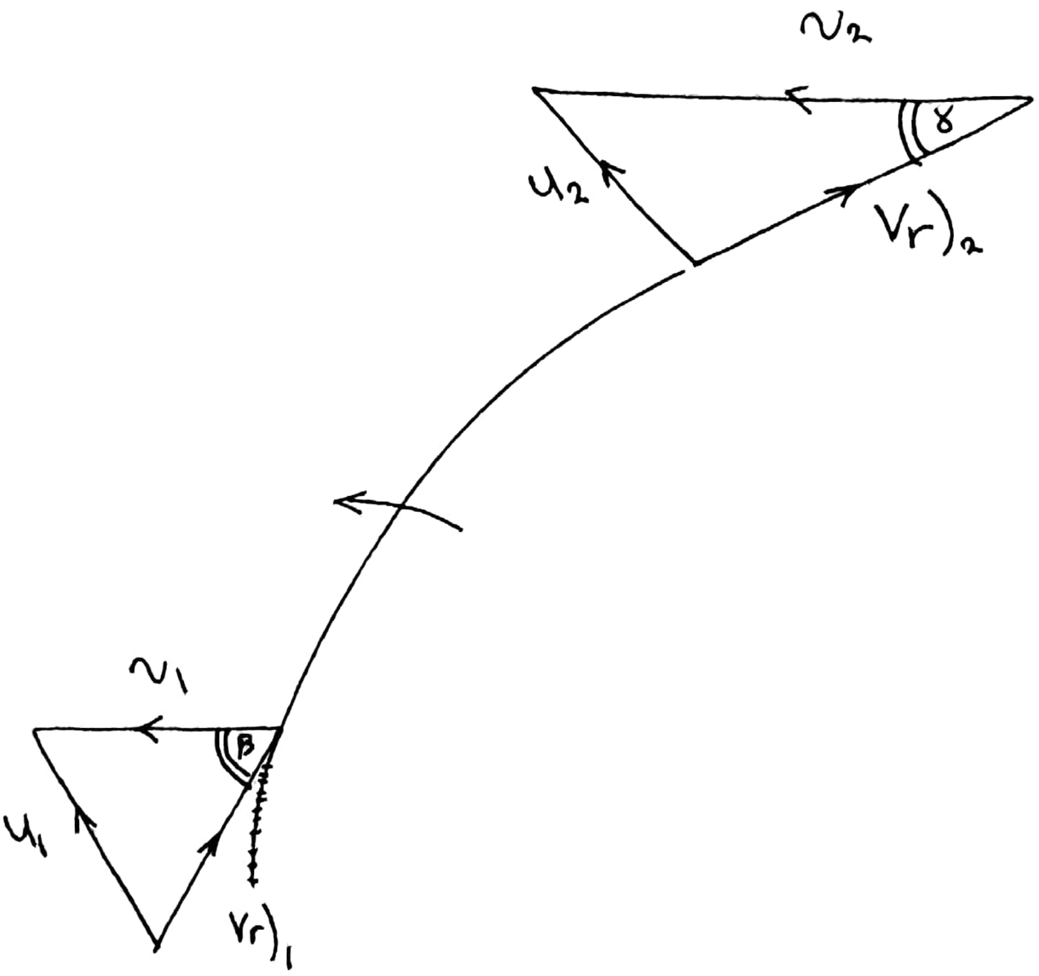


Figure 8 Losses in a Centrifugal Pump

→ Complete velocity triangles :

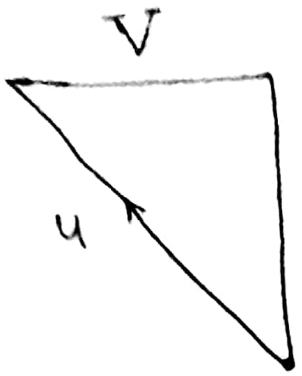


γ : blade angle at outlet

β : blade angle at inlet

Scanned with CamScanner

→ Components of absolute velocity

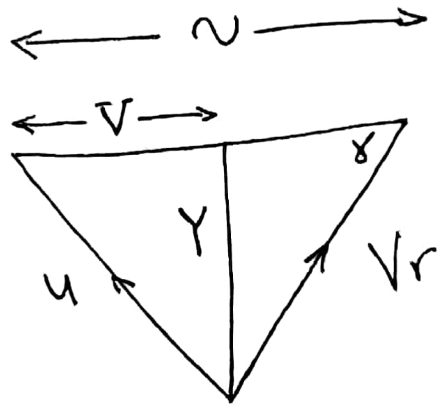


Y : flow velocity

V : whirl component

→ So, for example:

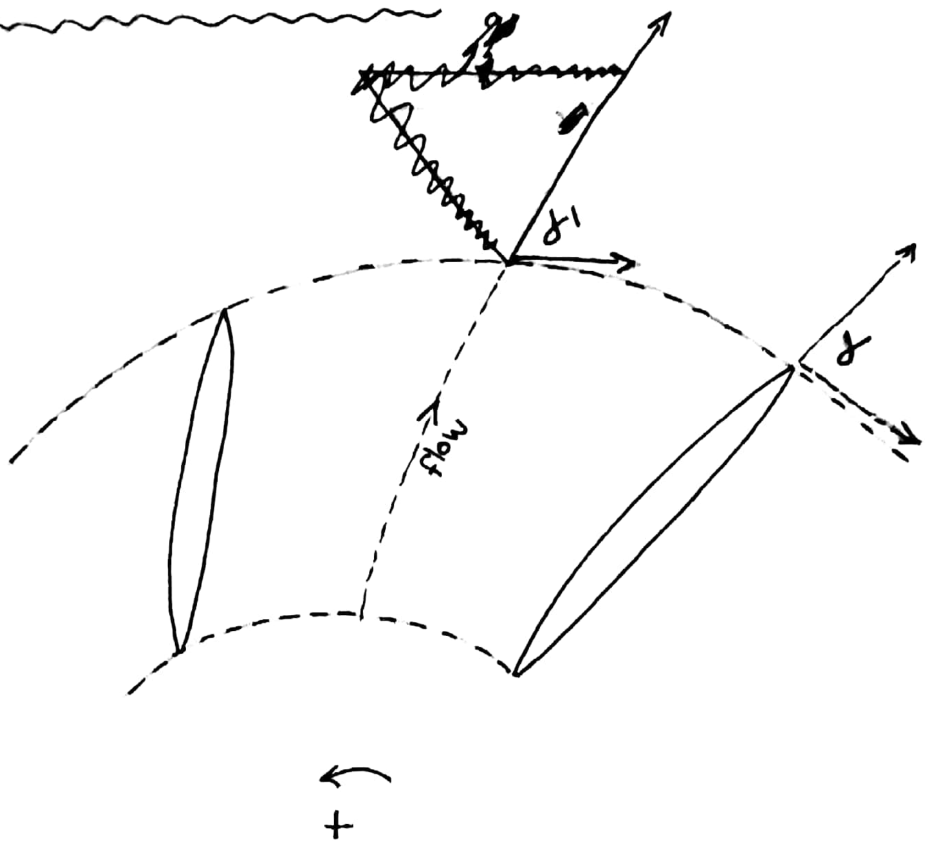
↳ if we draw complete velocity triangle at outlet



$$\rightarrow \tan \gamma = \frac{Y}{\nu - V}$$

Very IMP;

→ to be more precise :

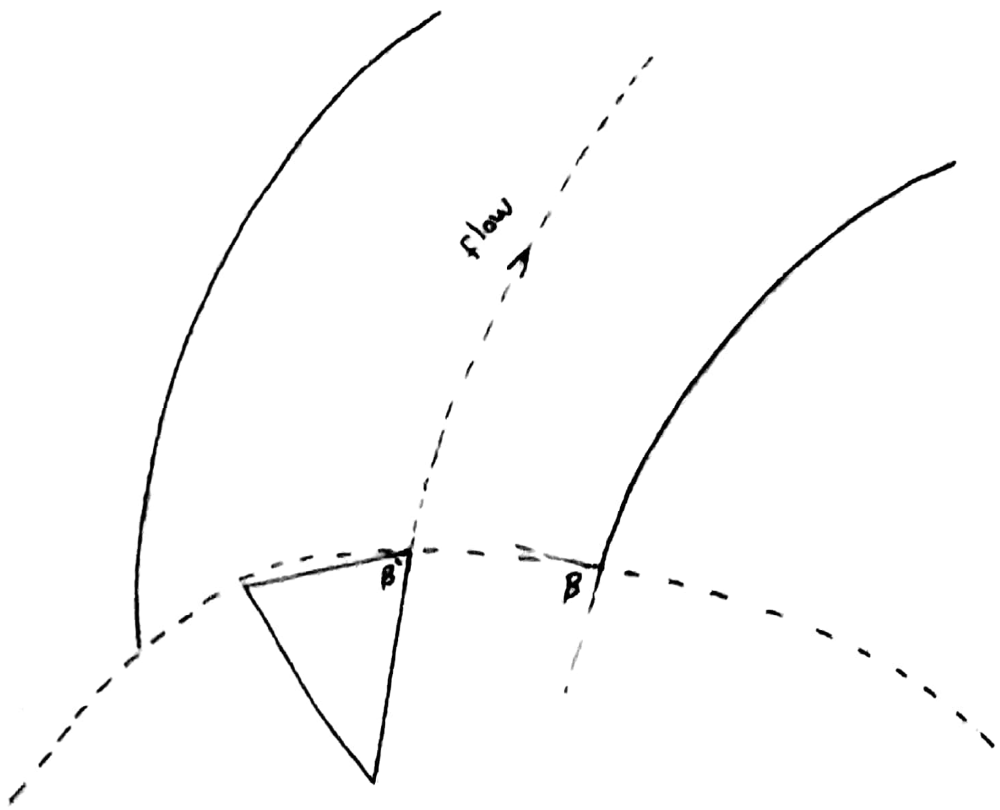


outlet Blade angle "γ" : angle which blades makes with tangeant at outlet.

Relative flow angle ; angle which flow makes with tangeant at outlet

→ for best design : $\gamma = \gamma'$ [default]

→ <u>Same for inlet :</u>

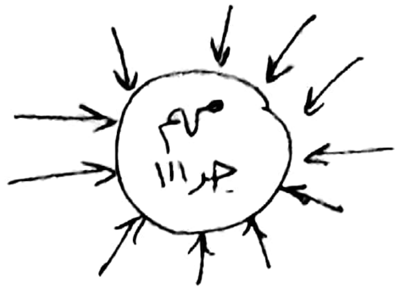


β : angle which blade makes with tangeant at inlet

β' : angle which flow makes with tangeant at inlet

<u>for Best design :</u> $\beta = \beta'$ [default]

→ According to slides :



Best pump design ;

- → $\gamma = \gamma'$ [no-fluid slip]
- → $\beta = \beta'$ [no-shock condition]

Euler Head:

$$E = \frac{v_2 V_2}{g}$$

$$E = \frac{v_2 \left[v_2 - x \right]}{g}$$

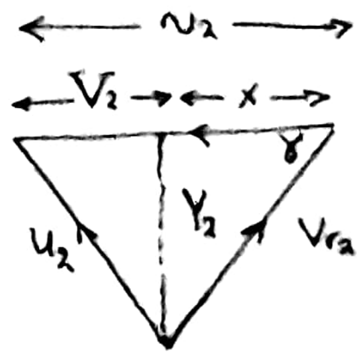
$$E = \frac{v_2 \left[v_2 - Y_2 \cot \gamma \right]}{g}$$

$$E = \frac{1}{g} v_2 \left[v_2 - \frac{Q}{A} \cot \gamma \right]$$

$$\boxed{E = K_1 - K_2 Q}$$ #

→ linear relation between E and Q

→ out vel. triangle



$$\tan \gamma = \frac{Y_2}{x}$$

$$x = \frac{Y_2}{\tan \gamma}$$

$$\boxed{x = Y_2 \cot \gamma}$$

Losses:

① slip Loss at outlet

② shock Loss at inlet

③ friction loss inside pump

① slip Loss,

$$E_N = \sigma_s E$$

② shock Loss

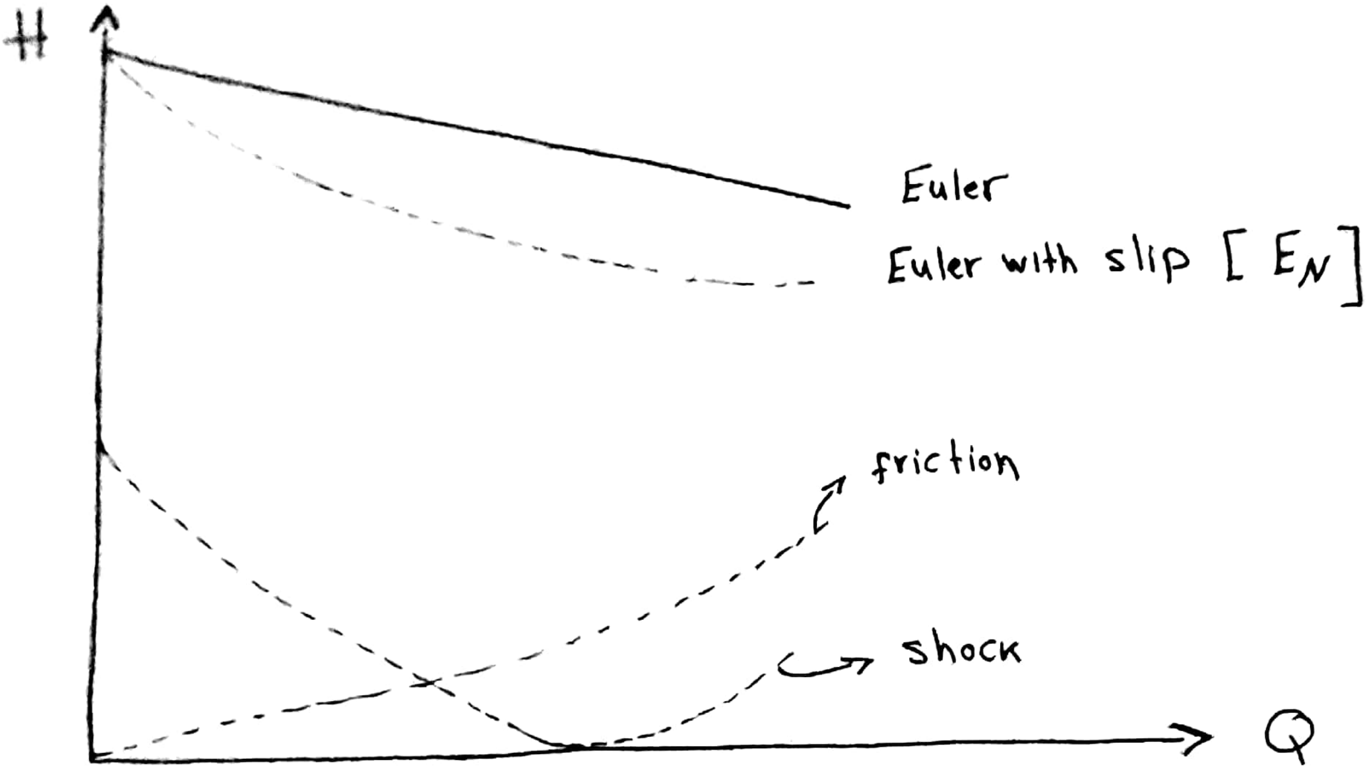
$$h_{shock} = K [Q - Q_D]^2$$

Q_D: design flow rate

③ friction Loss

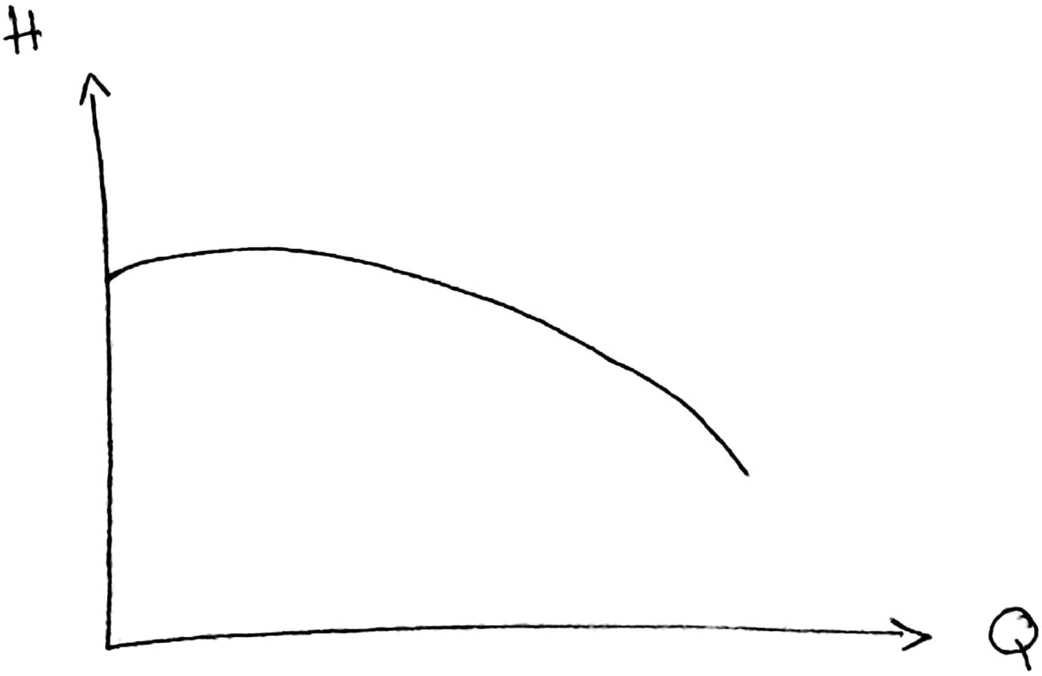
$$h_L = K Q^2$$

→ Now Let's represent them on graph.



subtract friction + shock from E_N

↓ get



→ IMP:

→ at design flow rate

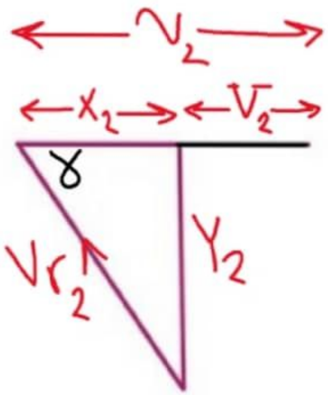
$\hookrightarrow$ shock loss = zero

$$h_{shock} = K[Q - Q_D]^2$$

if $Q = Q_D$

$\hookrightarrow h_{shock} = 0.0$

$Q_D \rightarrow$

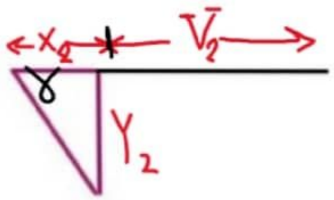


$\rightarrow$ Let γ = const

$\rightarrow Q \downarrow \rightarrow Y \downarrow$

$\bar{V}_2 + X_2 = v_2$

$Q \downarrow \rightarrow$



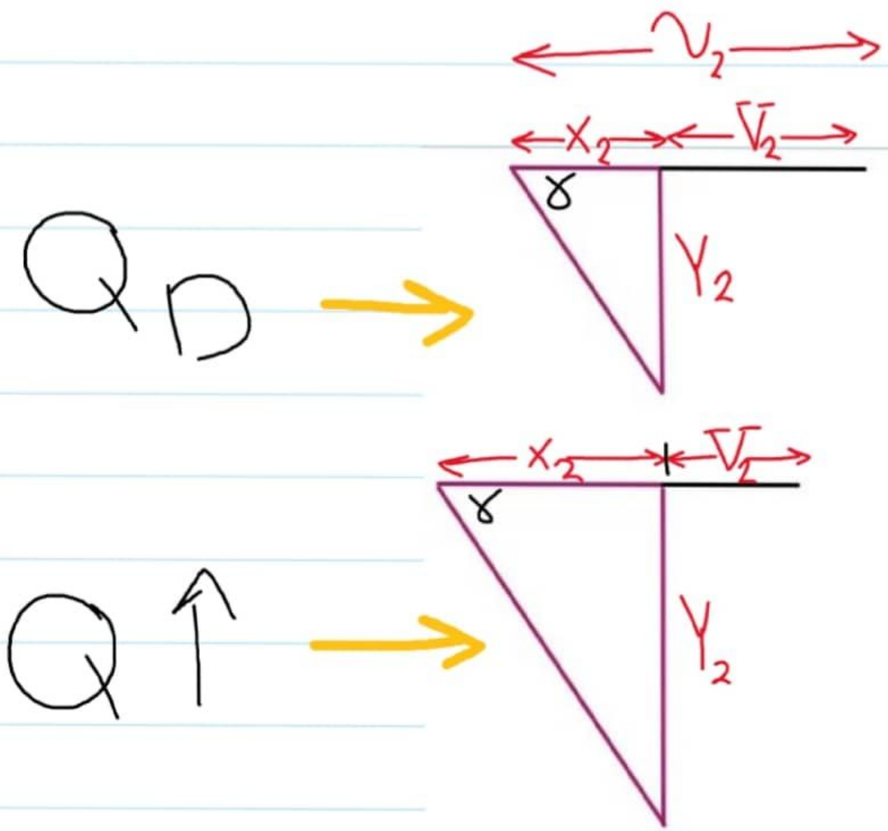
$\boxed{v_2 = C} \rightarrow$ if $X_2 \downarrow \rightarrow \bar{V}_2 \uparrow$

Effect of flow variation on exit velocity

Case 1 : flow increased over design Q

Effect of flow variation over exit velocity

Case 2 : flow less than design Q



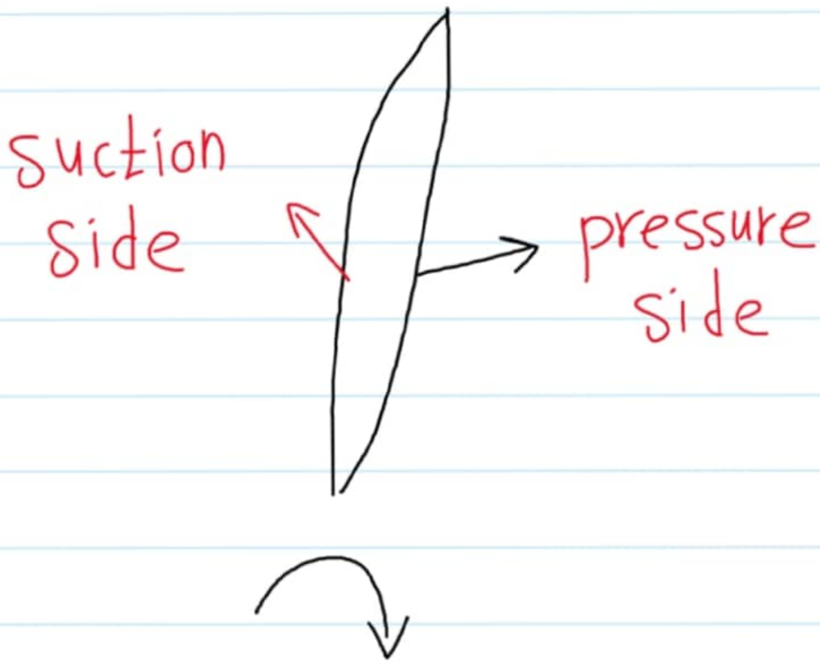
Let $\gamma = \text{const}$

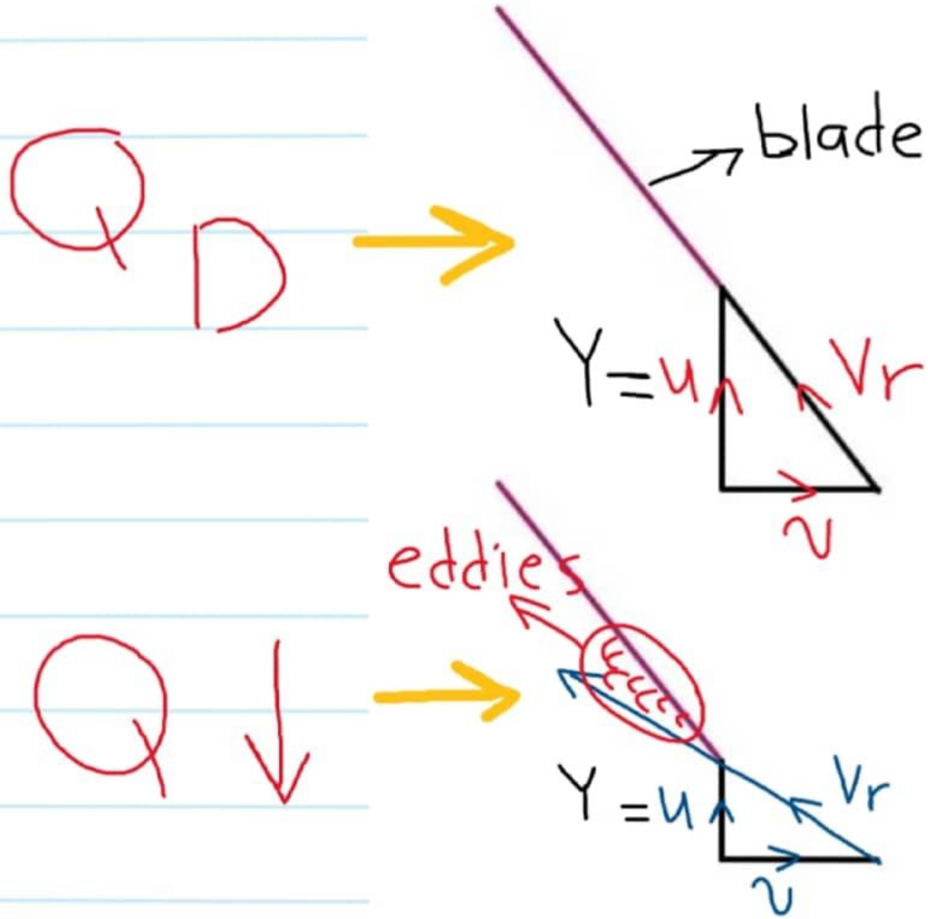
as $Q \uparrow \longrightarrow Y_2 \uparrow$

$v_2 = X_2 + \bar{V}_2 = \text{Const}$

as $X_2 \uparrow \longrightarrow \bar{V}_2 \downarrow$

→ **Blade:**





as $Q\downarrow \rightarrow Y\downarrow$

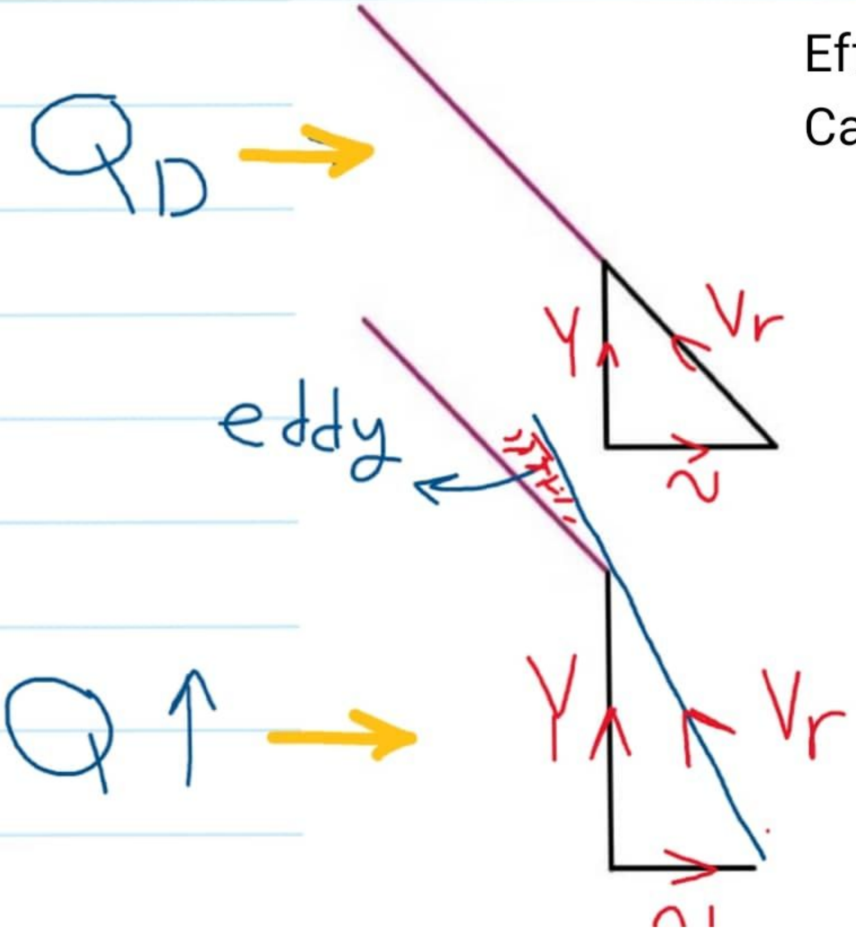
$\downarrow$

result in eddies on suction side

Effect of flow variation on inlet velocity
Q lower than design flow rate

Effect of flow variation on inlet velocity

Case 2 : flow above design flow rate

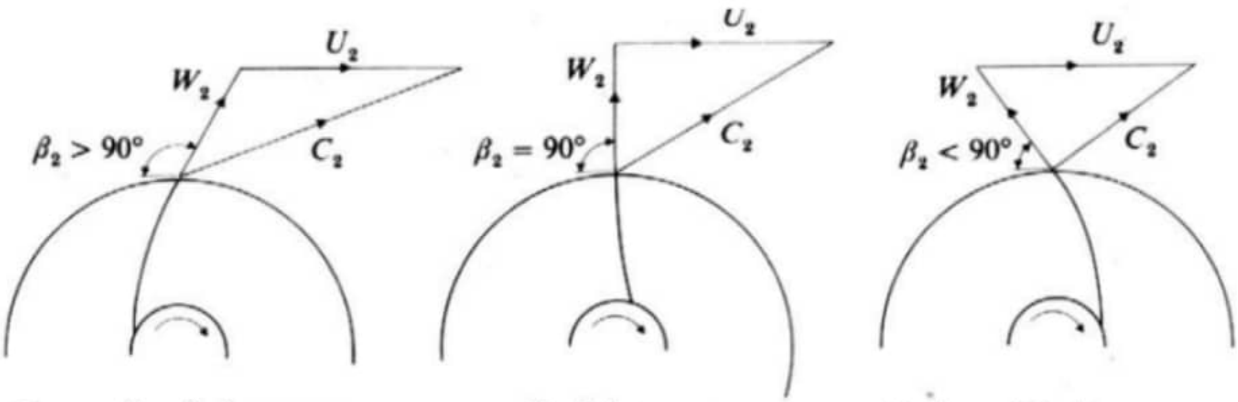


as $Q \uparrow \longrightarrow Y \uparrow$

$\downarrow$

eddies on pressure side

Types of blades :



Forwarding-facing vanes | Radial vanes | Backward-facing vanes

Forward $\longrightarrow$ exit blade angle $> 90°$

Radial $\longrightarrow$ / / / $= 90°$

Backward $\longrightarrow$ / / / $< 90°$

$\longrightarrow$ Now, Let's see effect of exit angle on $\underbrace{\text{ideal}}_{\text{Euler}}$ head.

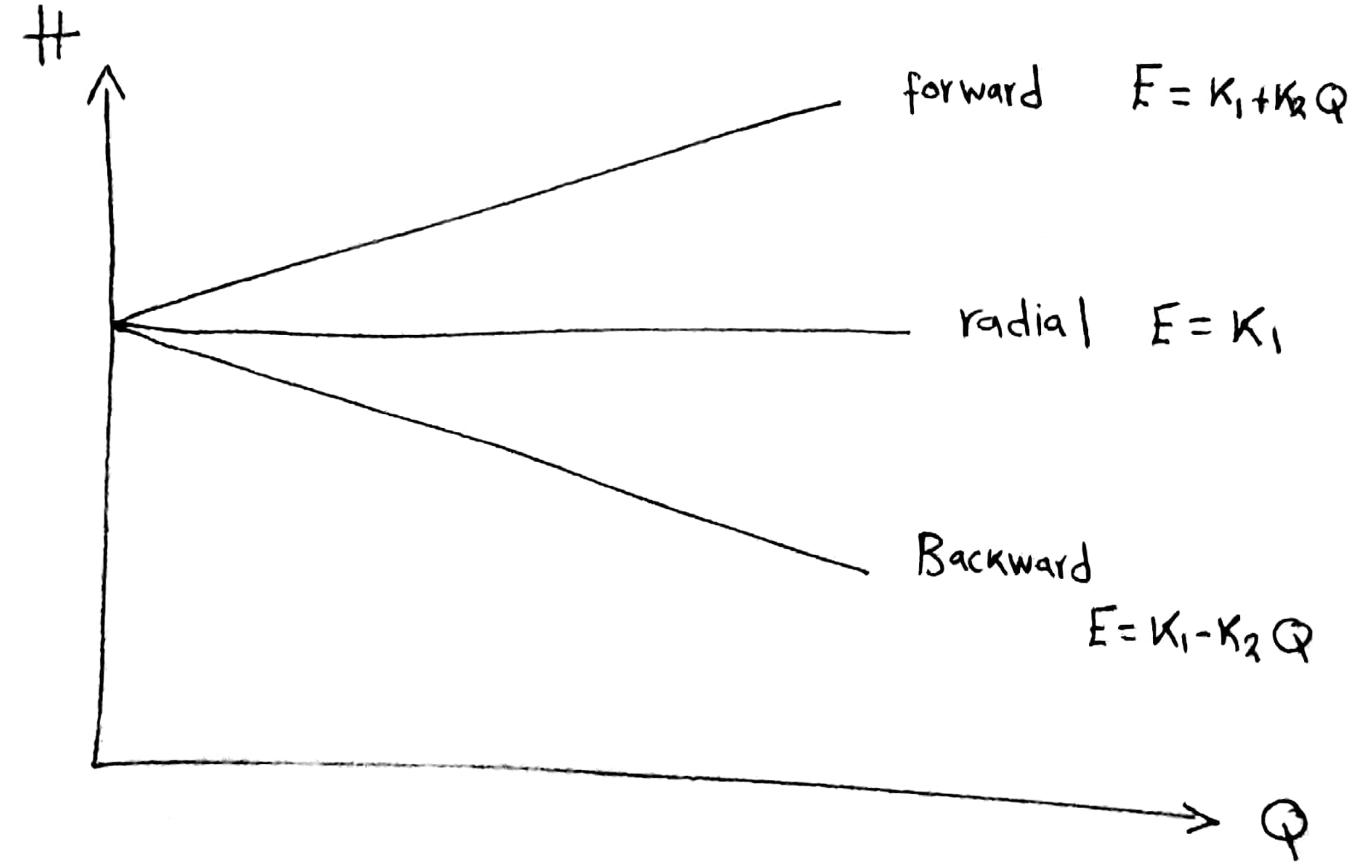
$$\text{Euler Head} = K_1 - K_2 Q$$

$$K_2 \propto \cot(\gamma) \rightarrow \text{exit blade angle}$$

if forward : $\gamma > 90 \longrightarrow \cot\gamma$ $-ve$

if radial : $\gamma = 90 \longrightarrow \cot\gamma = \text{zero}$

if Backward : $\gamma < 90 \longrightarrow \cot\gamma = +ve$

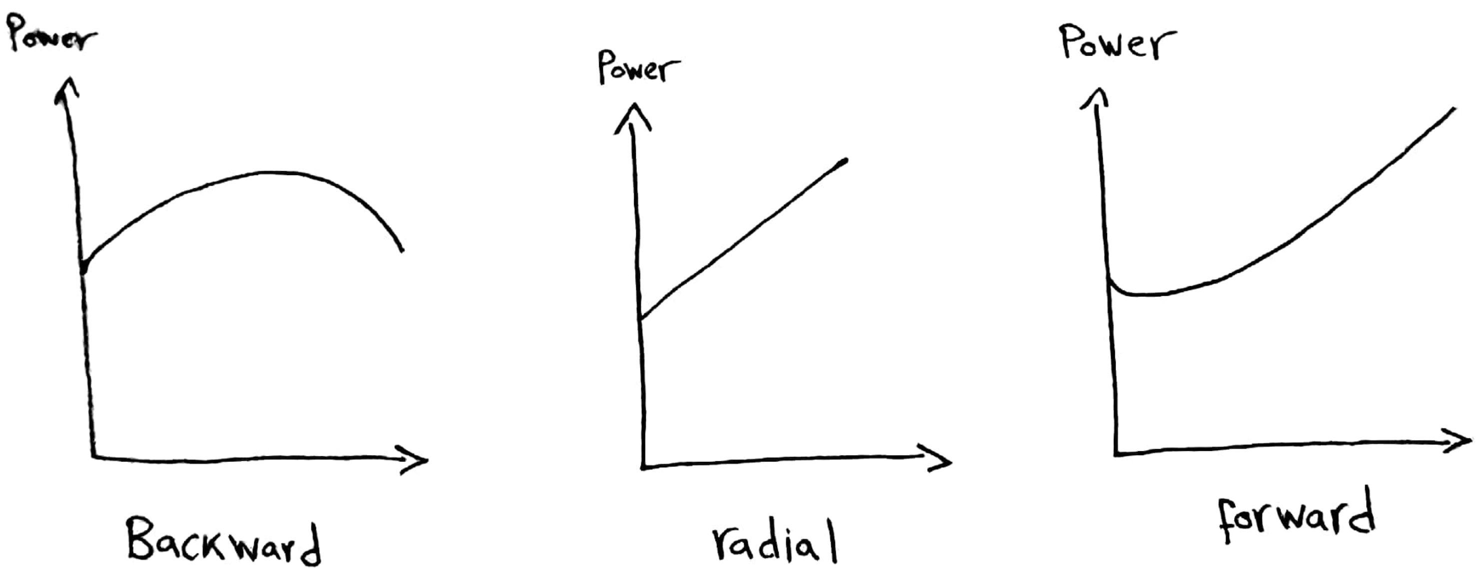


→ from Last figure :

→ is forward curved the best option ?

Answer :

→ Let's study Power - Q curve for each type.



if operating point changed :

① Backward → small change in Power → self - limiting characterestic

② radial and forward → Considered increase in Power
↳ So, we use big motor to withstand load.

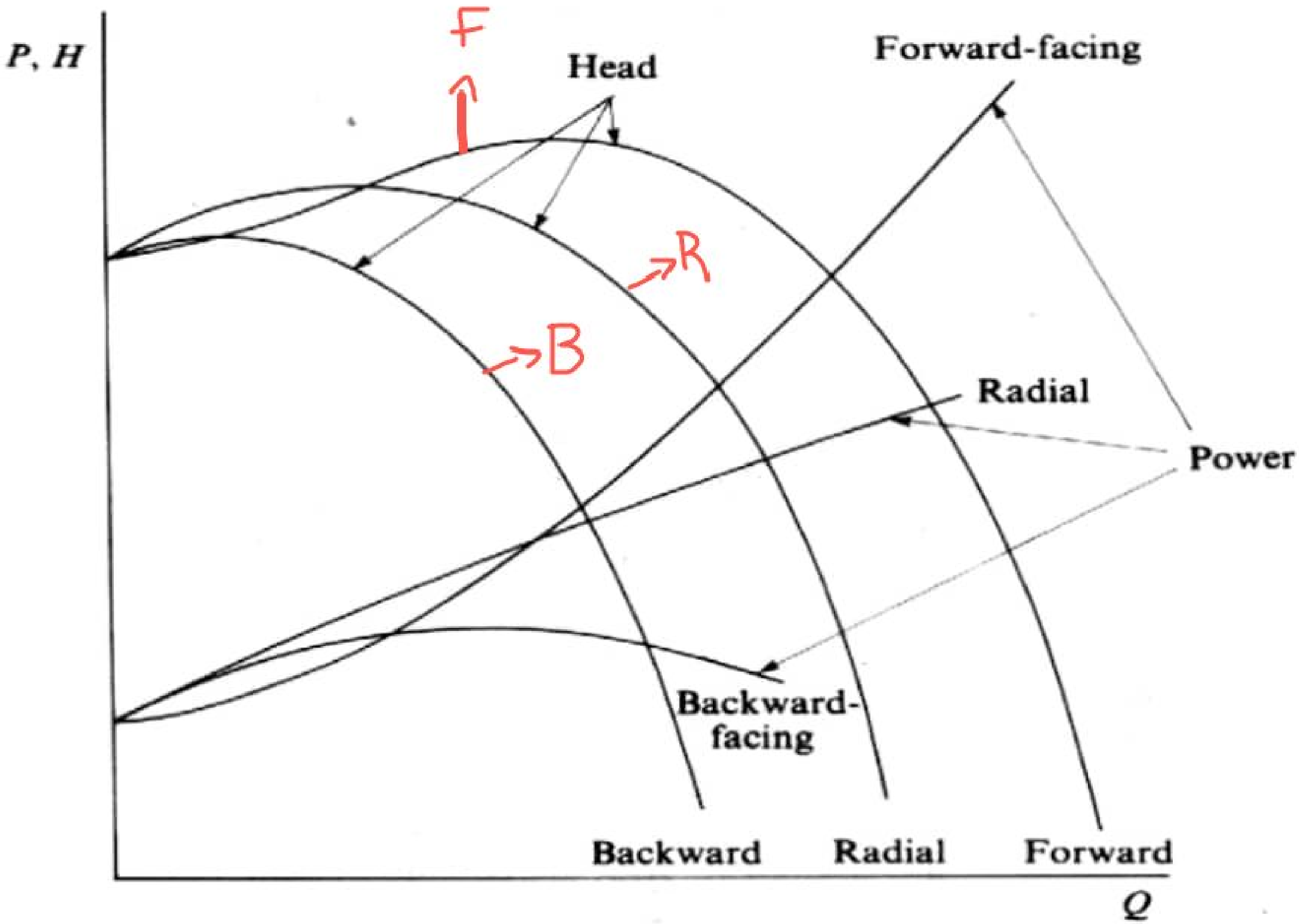
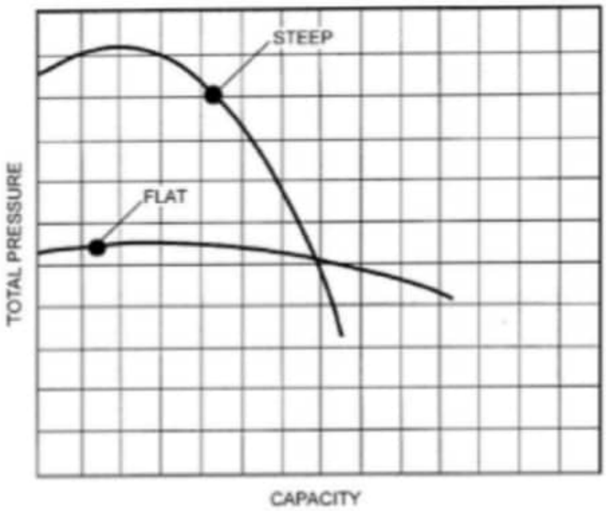
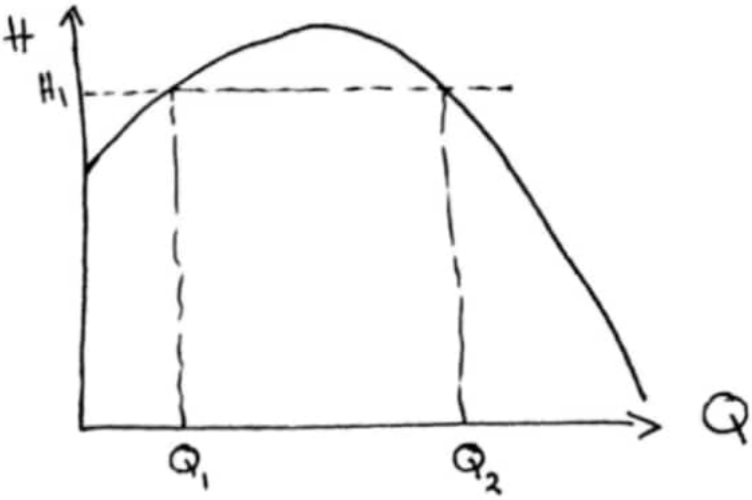


Figure 14 Actual Characteristics
for Varying Blade Outlet Angle

→ flat vs steep (H-Q) curve → which is best



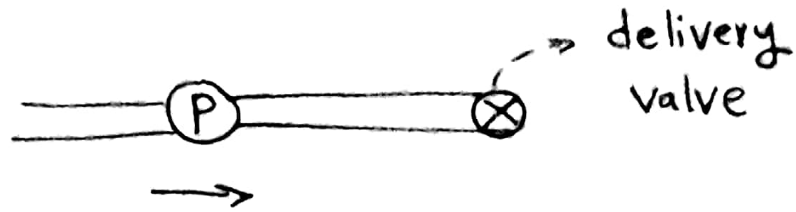
Steep curve is unstable :



at $H = H_1$ → there are two values for Q

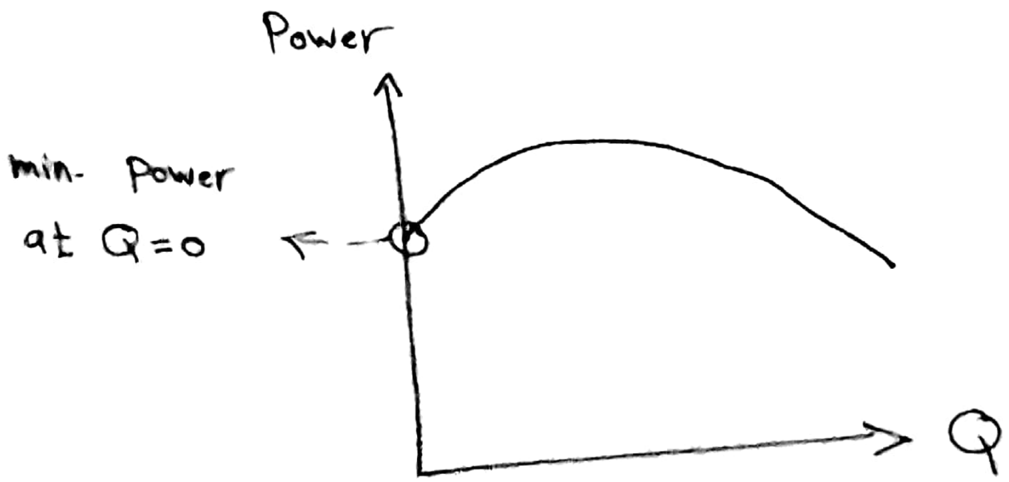
→ unstable operation

Important:



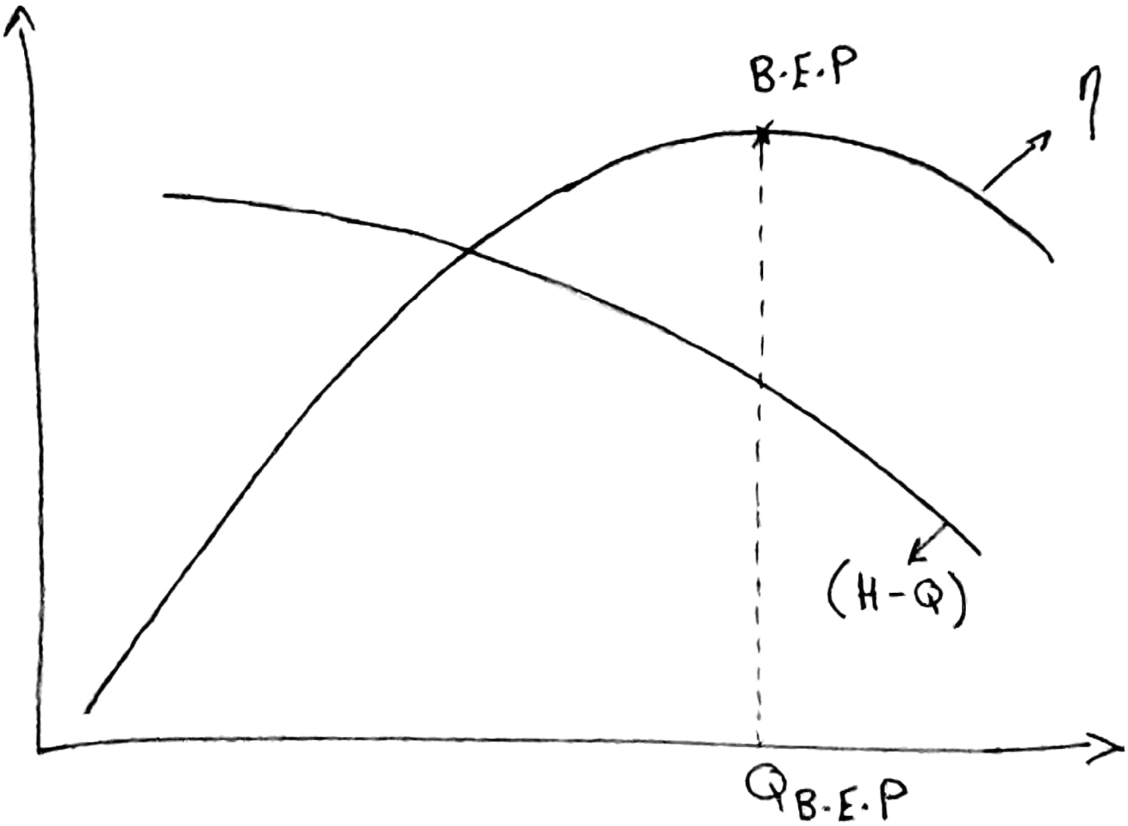
at start: delivery valve should be closed

→ So, motor work on min. load



Note: → This for backward curved.

→ Recommend Range for operation :



→ **Recomended** : 60 % $Q_{B.E.P}$ → 120 % $Q_{B.E.P}$

< 60 %

← Low flow → Excessive noise

> 120 %

← higher flow → Cavitation expected

→ inside this recomended range

↳ there is a preferred operating range.

[80% → 110%]

→ **why?**:

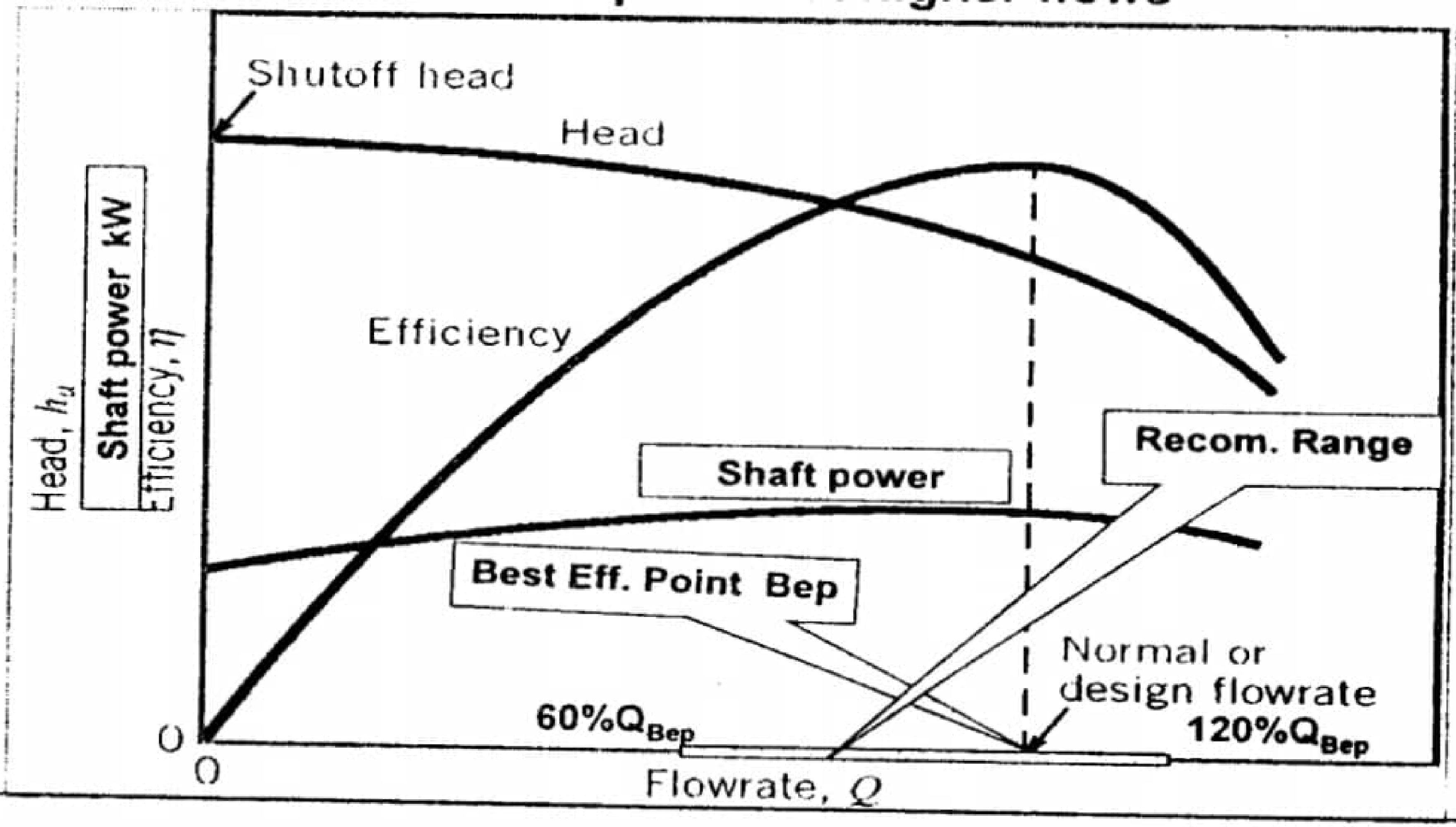
because in this range,

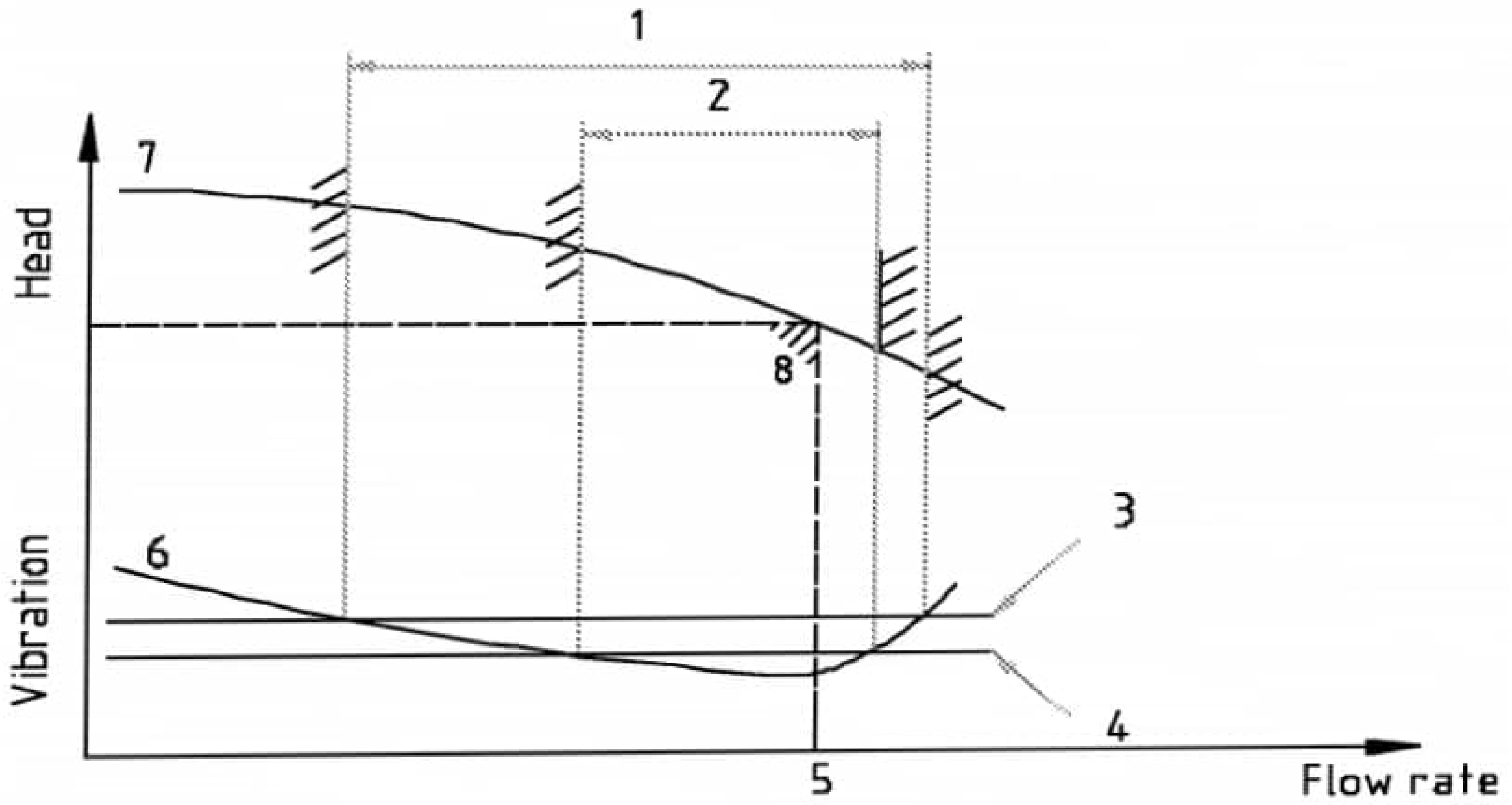
↳ η- curve almost flat

↳ so, small change in efficiency

Recommended Operating range: 60-120% of Q Bep

Excessive Noise & vibrations at lower flows
Cavitation expected at higher flows





Key

1. Allowable operating region of flow
2. Preferred operating region of flow
3. Maximum allowable vibration limit at flow limits
4. Basic vibration limit
5. Best efficiency point flowrate
6. Typical vibration vs. flowrate curve
7. Head-flowrate curve
8. Best efficiency point head and flowrate

Relationship between Flow and Vibration

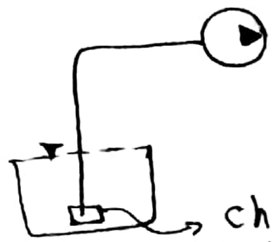
Some Notes:

→ when using a strainer:

↳ the total opening area of holes must equal the cross-section area of pipe.

→ if suction level is below pump level:

↳ we use check valve to ensure that suction line is full of Liquid



→ How to distinguish between suction and delivery lines in pump?

↳ suction line has larger diameter to decrease losses to avoid cavitation.

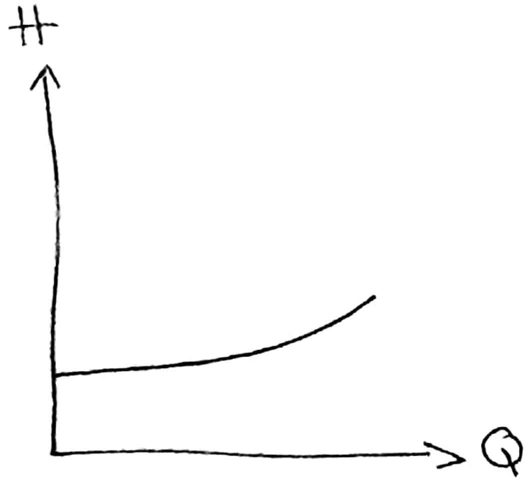
→ Some Notes on system Curve :

① $$H = h_{st} + KQ^2$$

neglect minor Loss : $$K = \frac{0.8\, fL}{g d^5}$$

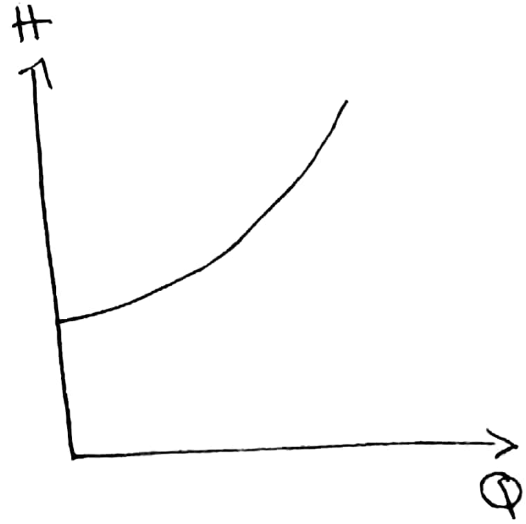
if we have long pipeline → $K \uparrow$

if we have short pipeline → $K \downarrow$



$K \downarrow$

[flat curve]



$K \uparrow$

[steep curve]

→ effect of variating suction tank level on system curve.

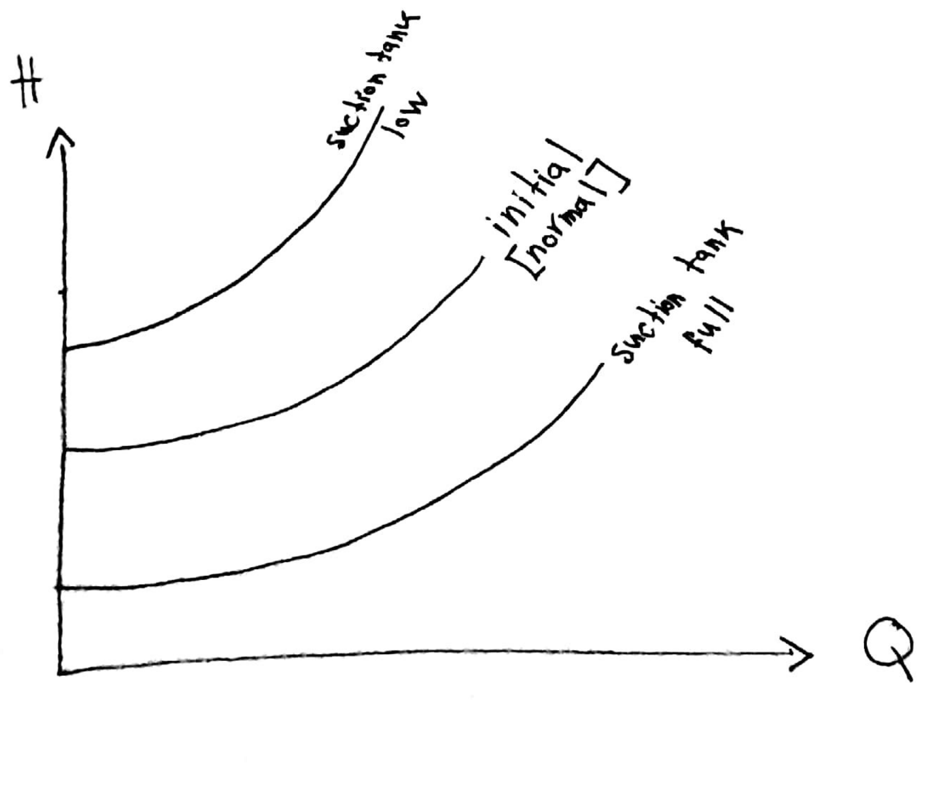
Answer:

$$H_{st} = h_{sd} - h_{ss}$$

→ for same h_{sd}:

as $h_{ss} \uparrow \longrightarrow h_{st} \downarrow$

as $h_{ss} \downarrow \longrightarrow h_{st} \uparrow$



→ effect of variating delivery tank level on system curve.

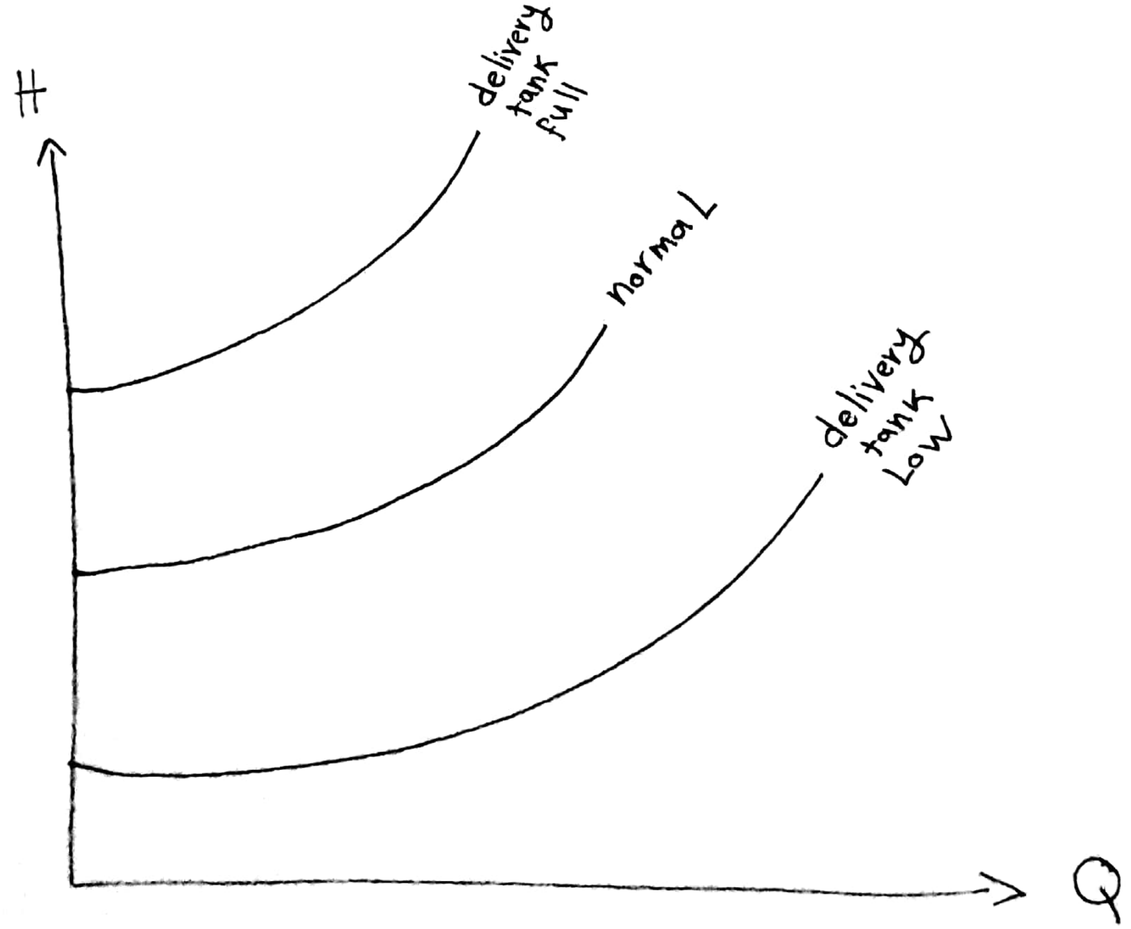
Answer,

$$H_{st} = H_{sd} - H_{ss}$$

→ for same h_{ss} :

as $h_{sd} \uparrow \longrightarrow h_{st} \uparrow$

as $h_{sd} \downarrow \longrightarrow h_{st} \downarrow$



→ what will happen to pump flow rate during operation ?

Answer:

→ at start: both suction and delivery tanks have same level

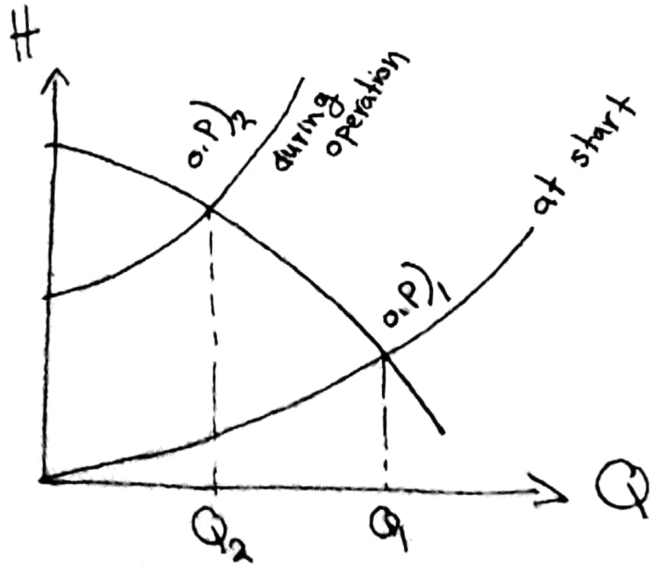
↳ ∴ $h_{st} = 0.0$

→ after operation, suction tank level decrease

↳ and delivery tank level increase

↳ So, $h_{st} \uparrow$

as $h_{st} \uparrow \rightarrow$ resistance $\uparrow \rightarrow Q \downarrow$



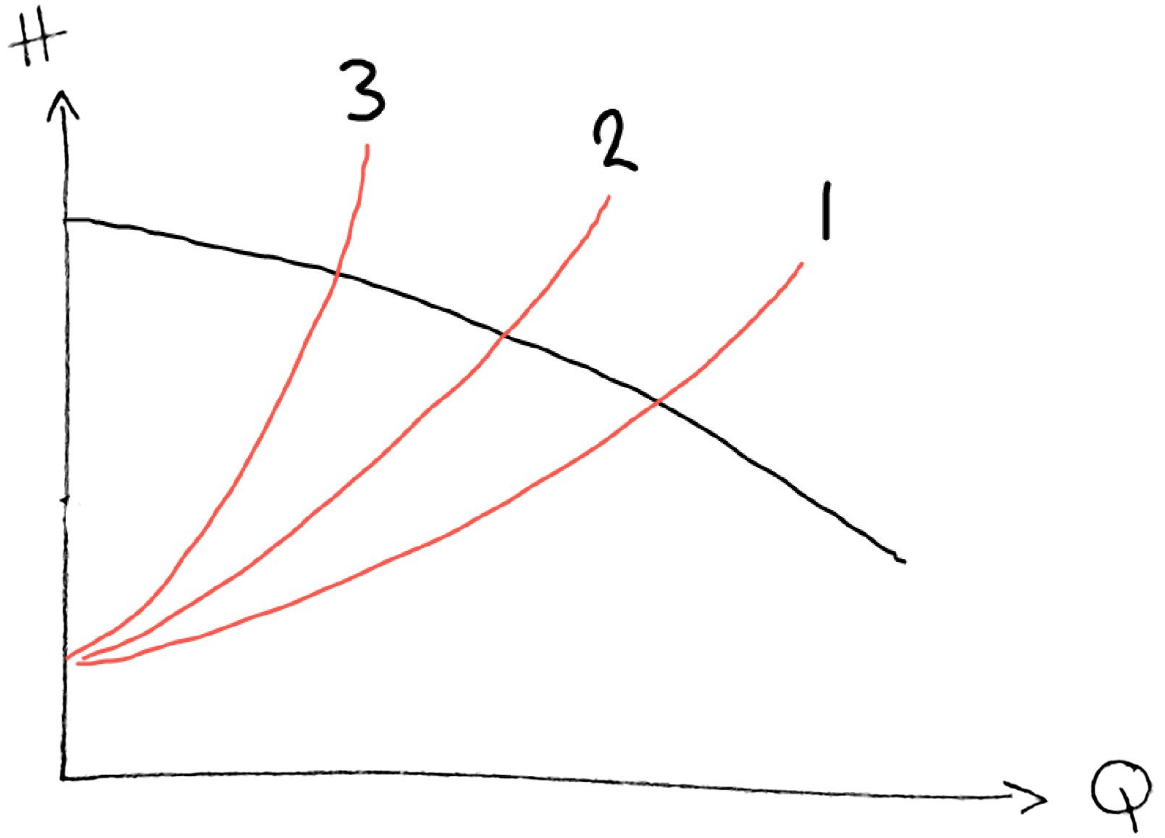
$\rightarrow Q_2 < Q_1$

→ draw system curve in case of:

① delivery valve 90% open

② delivery valve 50% open

③ delivery valve 10% open

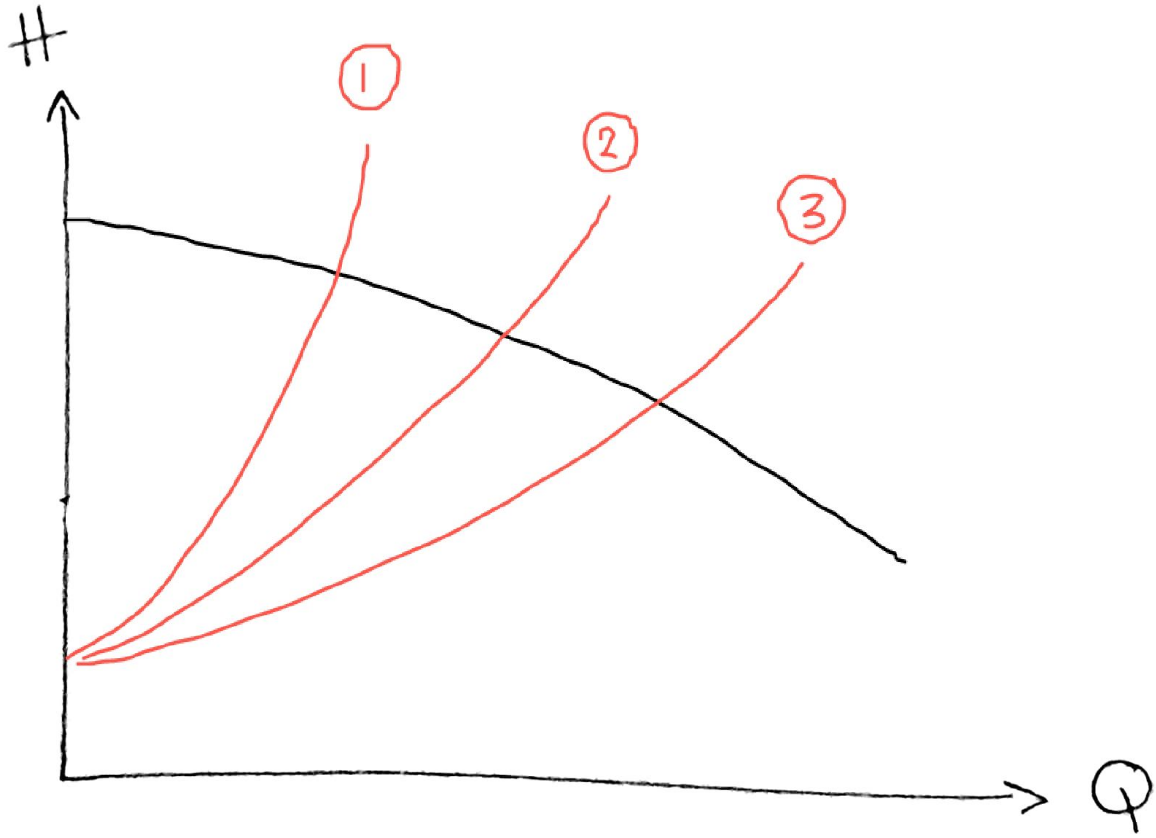


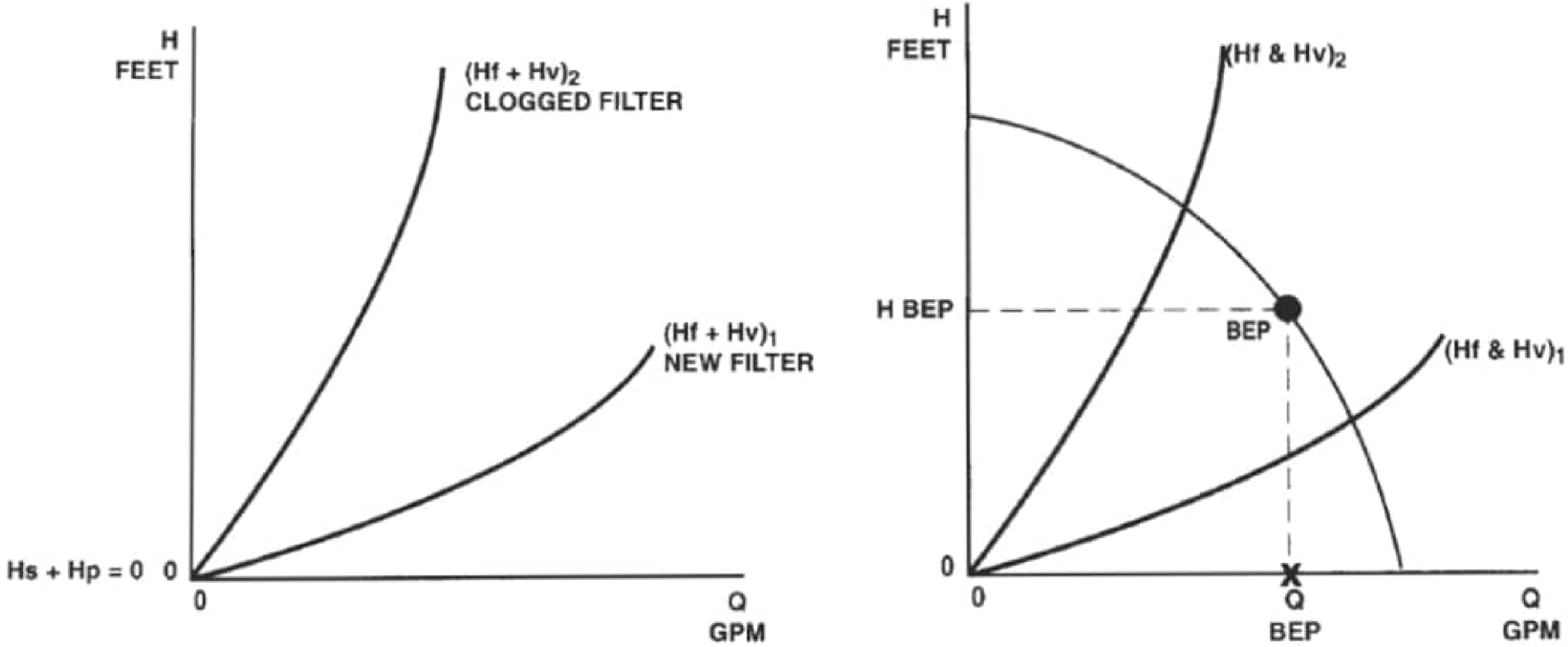
→ draw system curve in case of:

① delivery valve 90% closed

② delivery valve 50% /

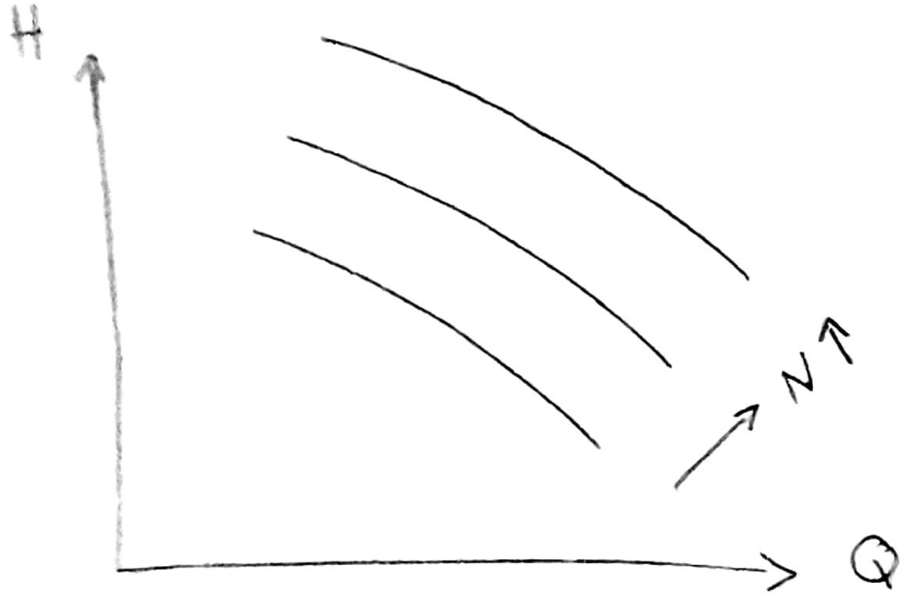
③ delivery valve 10% /



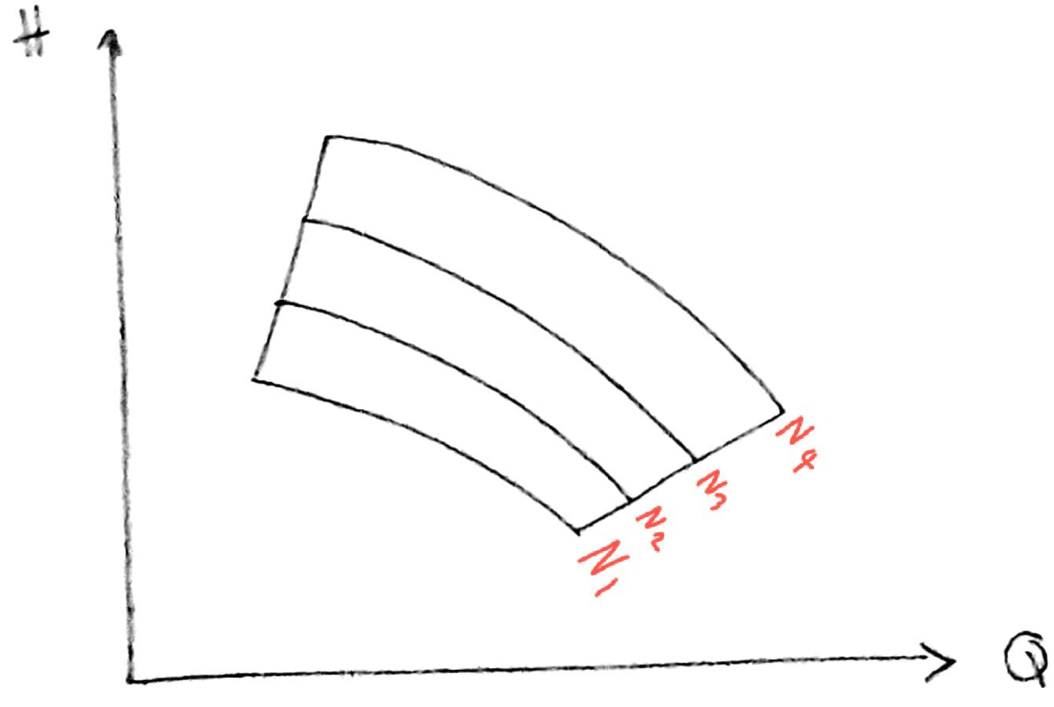


Effect of New and Clogged Filters on the System Curve

→ How [H - Q] curve change with N ?



→ Draw head-capacity envelope for variable-speed pump



→ important Note :

→ Recall :

system curve :

$$H_p = h_{st} + KQ^2$$

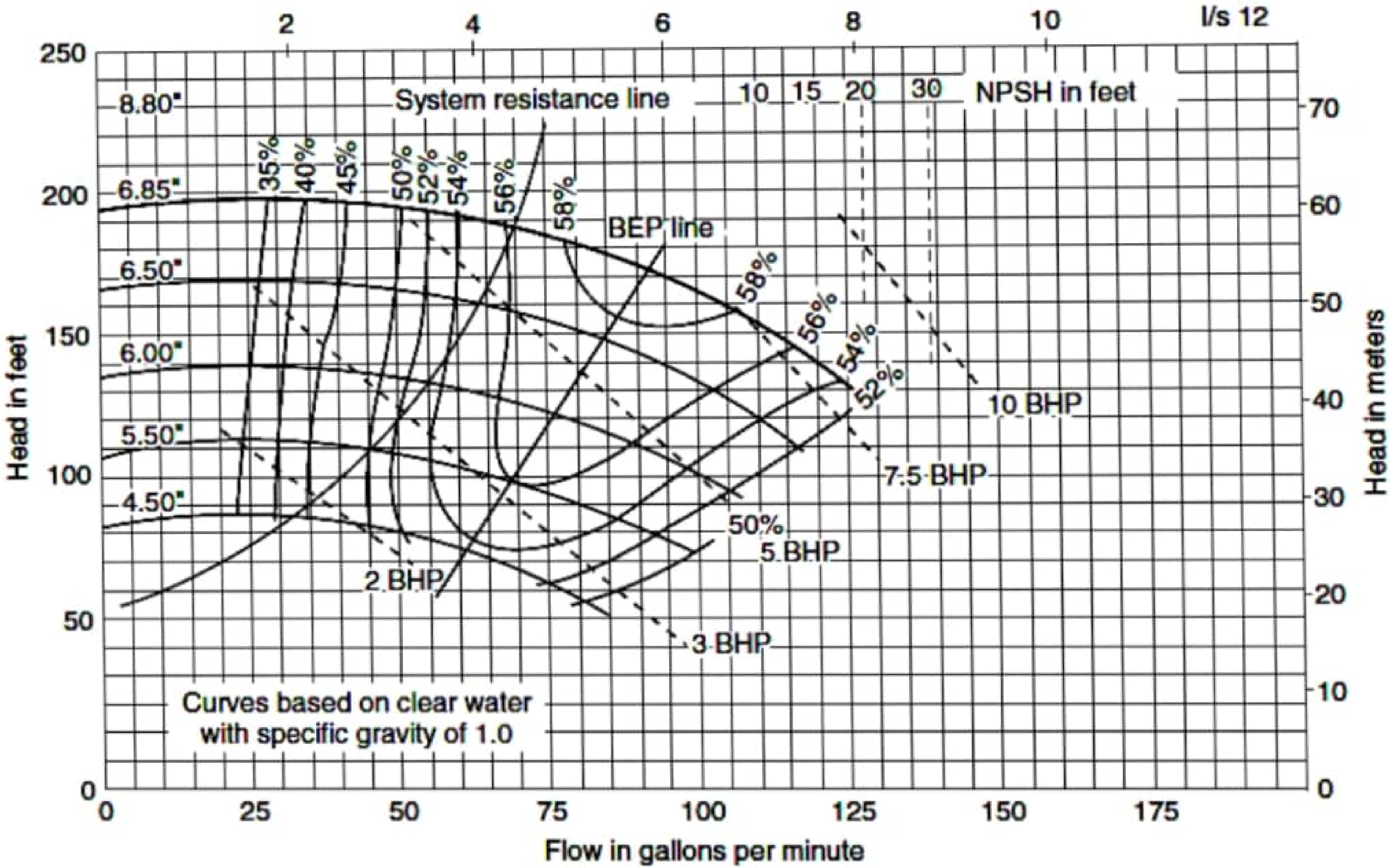
→ in Case $h_{st} = 0.0$

↳ system curve will be coincident on iso-η curve

→ and pass by origin

→ if $h_{st} \neq 0.0$:

↳ system curve and iso-η curve will not be coincident



Iso-Efficiency Curves

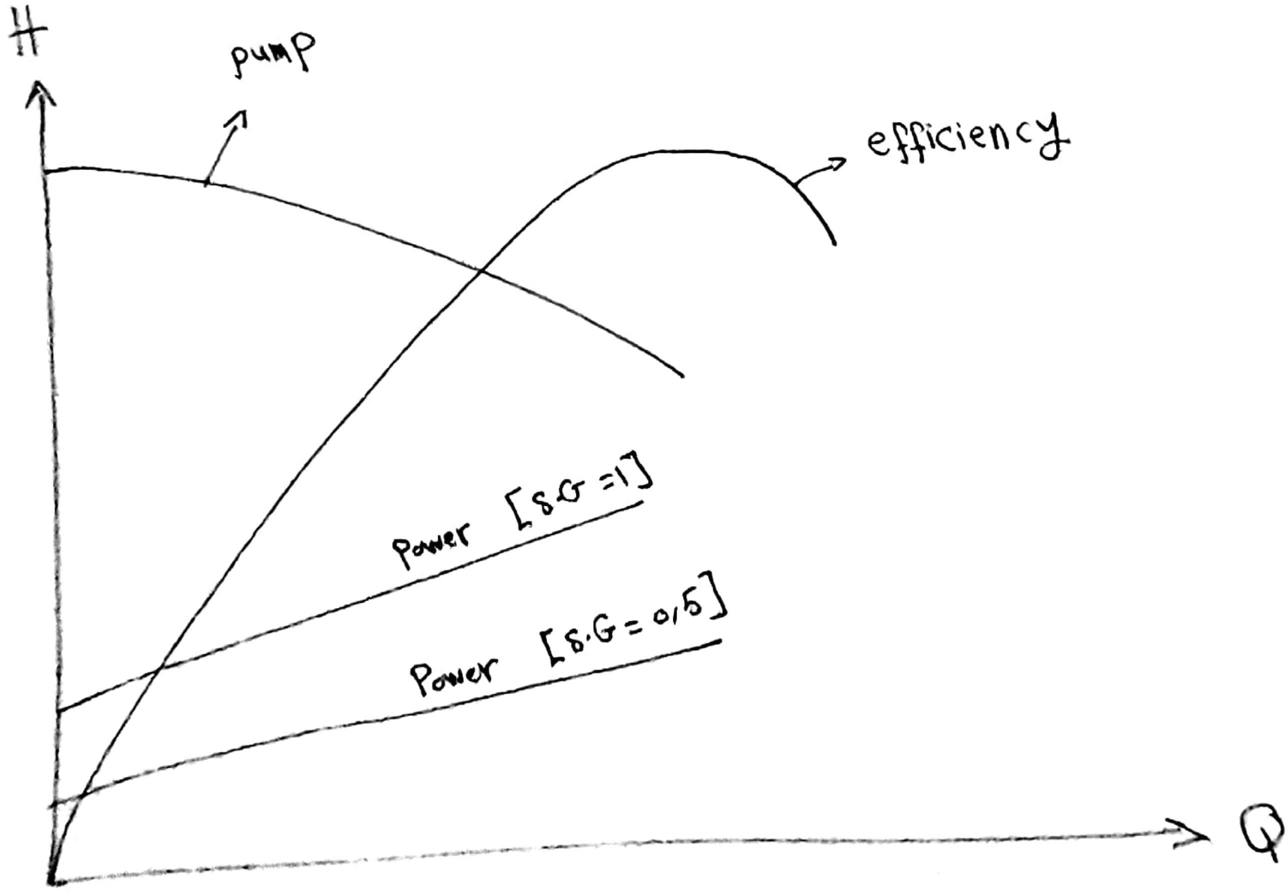
- effect of density :

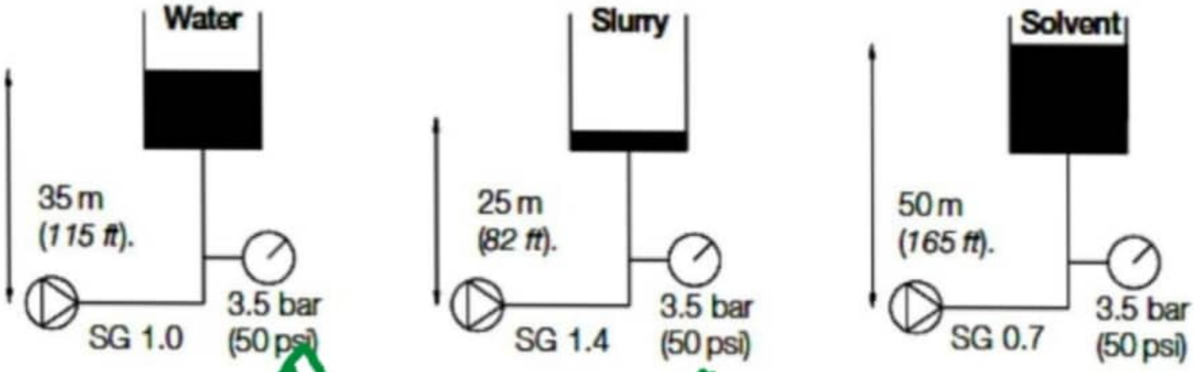
→ show the effect of changing density on performance curves.

→ pump curve [same]

→ η - curve [same]

→ power curve [changes]





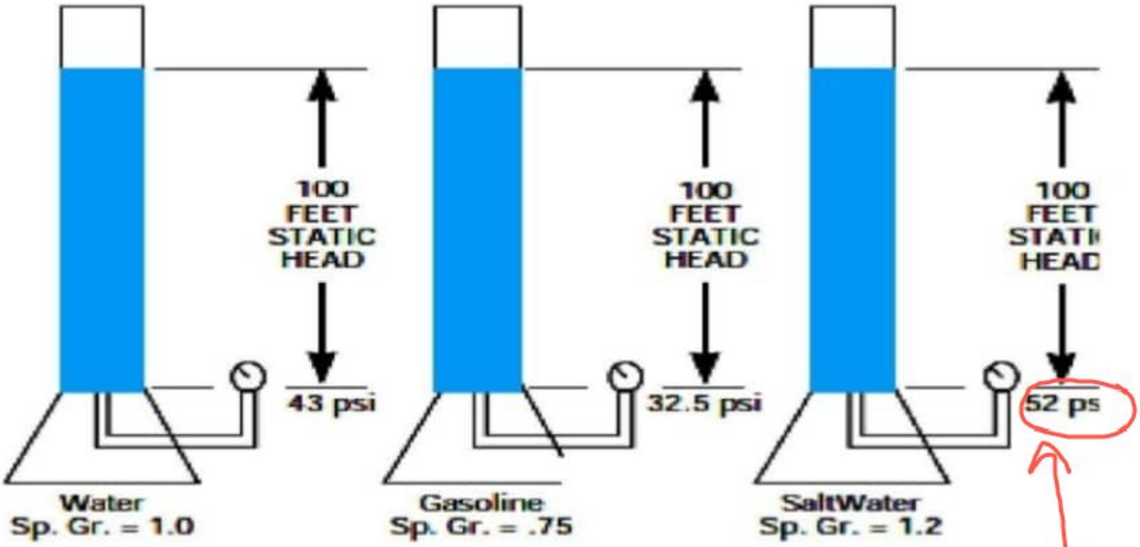
→ for the same pump pressure

$$h = \frac{P}{\rho g}$$

so as $\rho \uparrow \longrightarrow h \downarrow$

→ <u>from Last figure</u> ;

Solvent will be pumped to highest ~~the~~ elevation as it has low density.



→ for same static lift : "h"

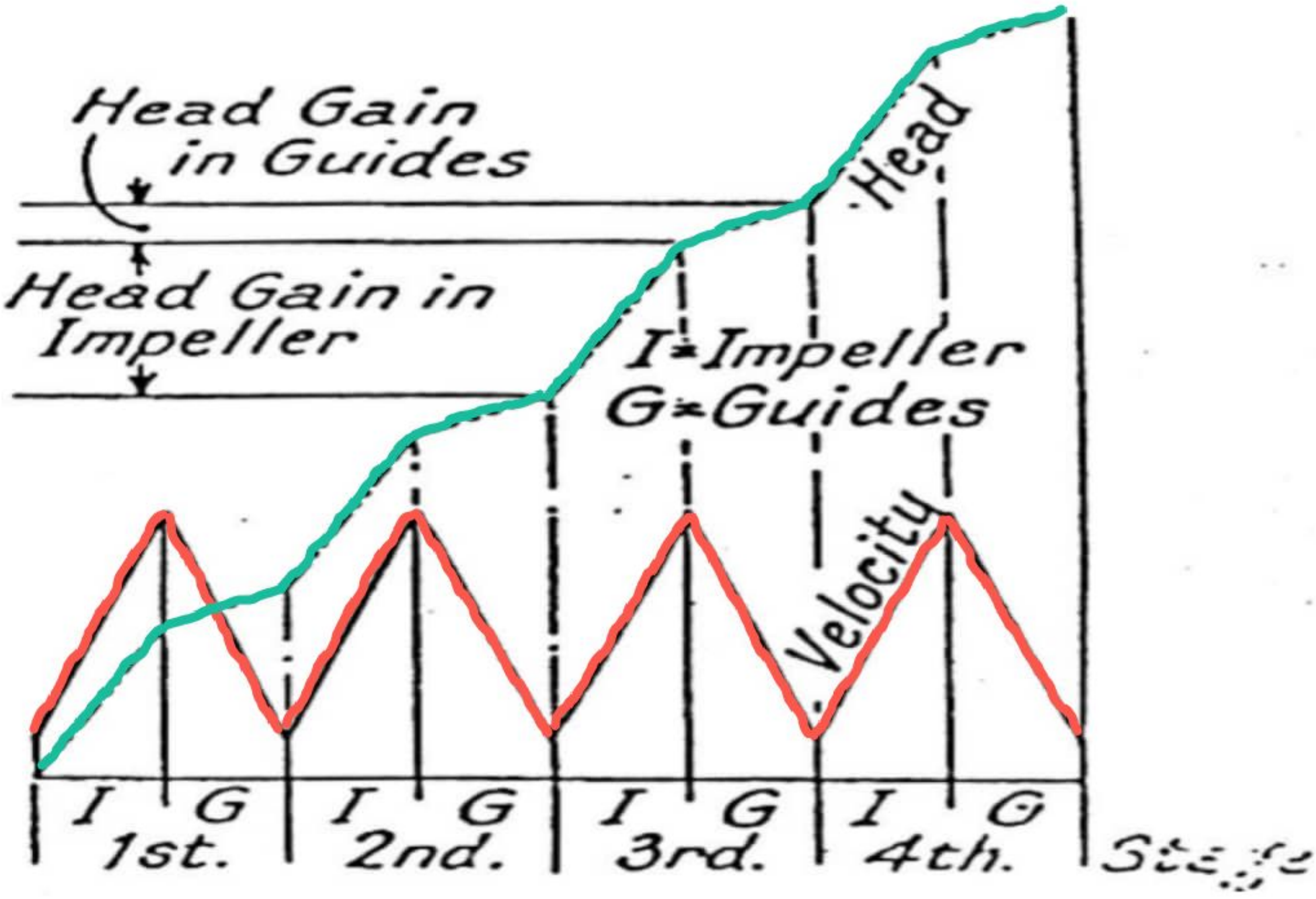
$$P = \rho g h$$

$$\text{as } \rho \uparrow \rightarrow P \uparrow$$

→ from last figure :

Salt water will need higher pressure than others.

→ velocity and pressure distribution for multi-stage pump.



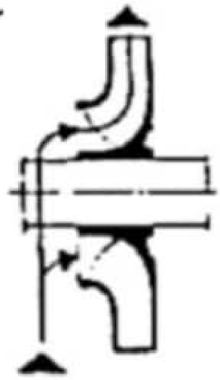
— velocity distri

— Pressure

→ How to determine number of stages?

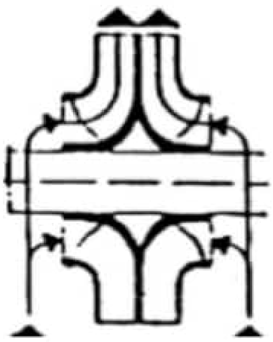
first → determine whether it's single or double suction

then → determine how many times the liquid is compressed in each suction path.



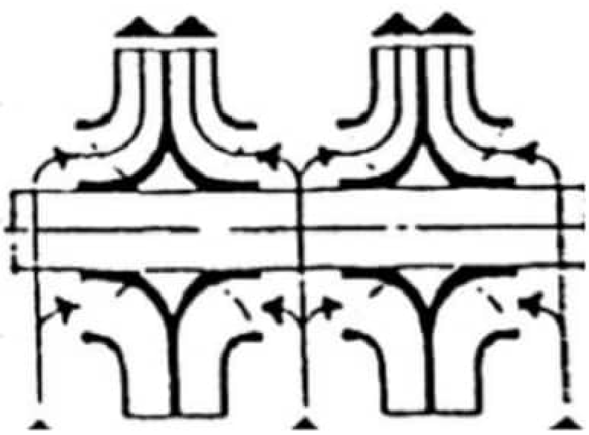
→ single suction

→ single stage



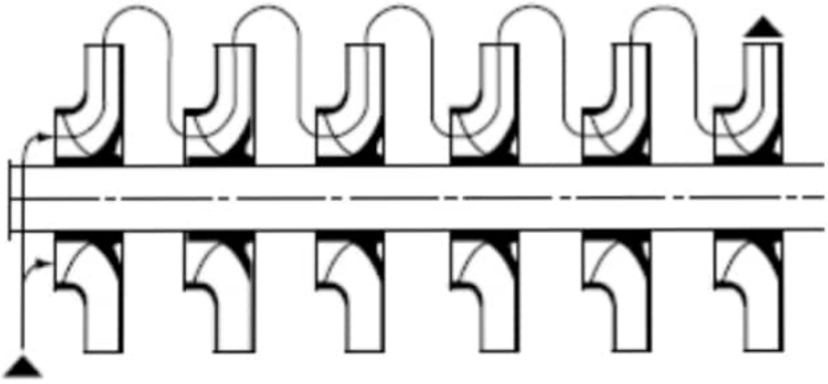
→ Double suction

→ single stage

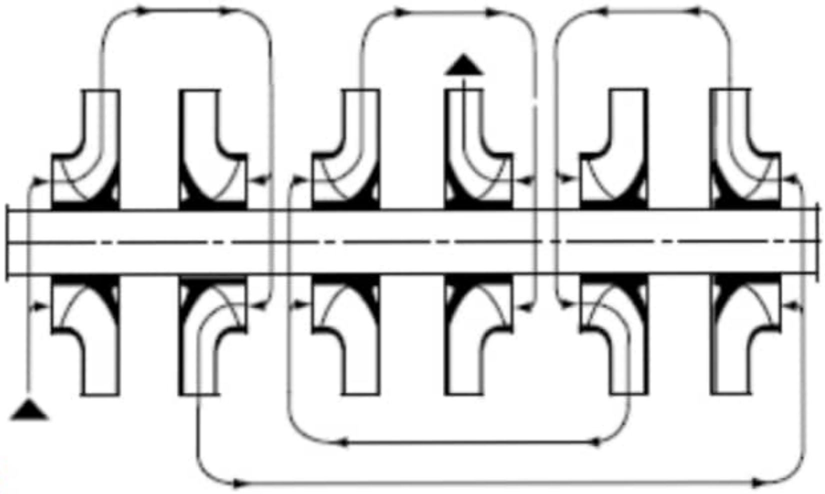


→ Quadraple suction

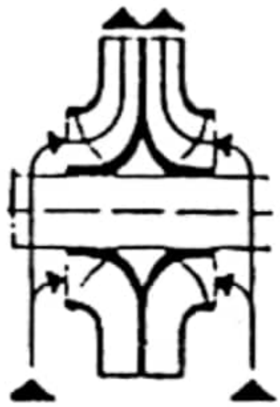
→ single stage



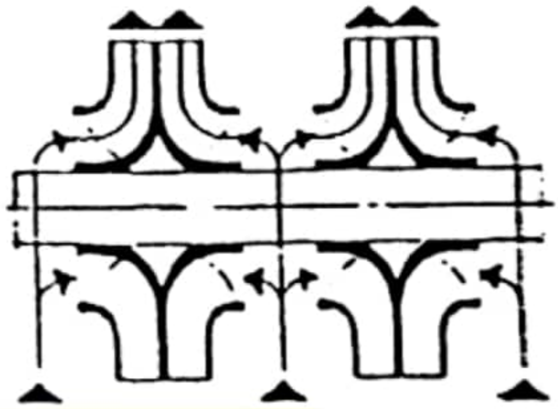
6-Stage



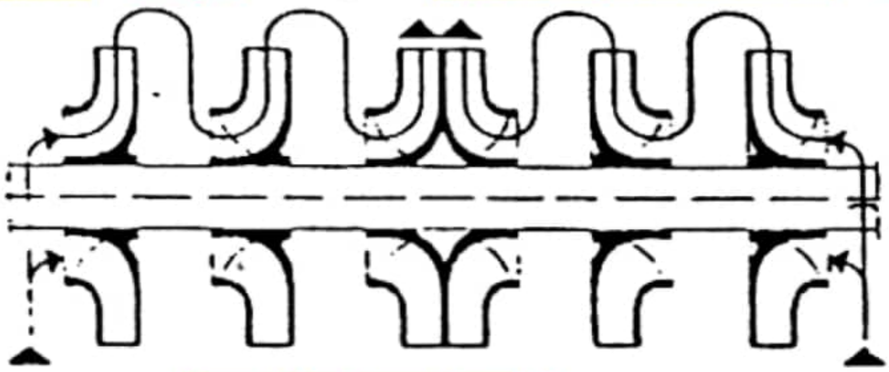
6-Stage, Back to Back



Double Suction, Single Stage



Quadruple Suction, Single Stage



Double Suction, 3-Stage

Different Arrangements for Multi-Stage, Multi-Suction Impellers

Very important :

→ assume that one pump gives $Q = 20 \frac{m^3}{s}$

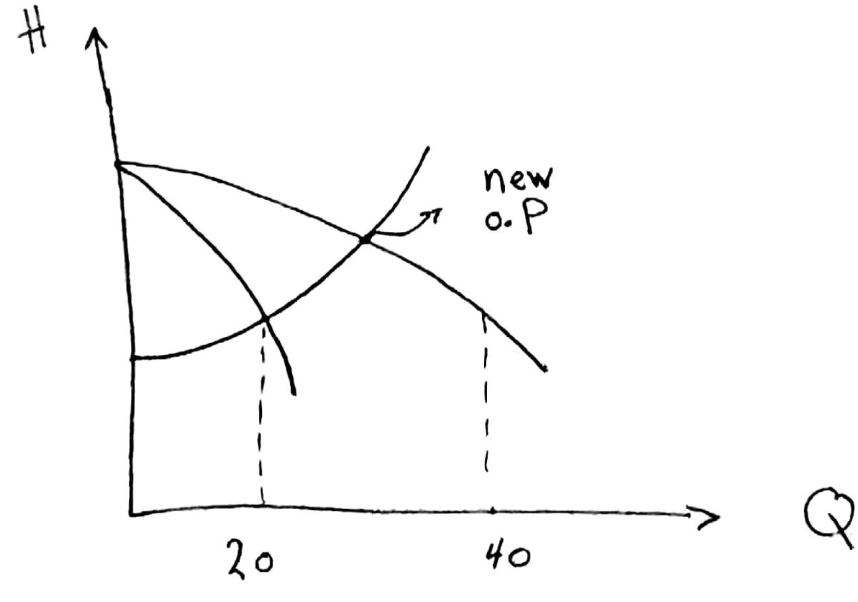
↳ if we put 2 identical pumps

in

↳ will Q be doubled ?

Answer :

No :



→ the new flow rate will be

less than double.

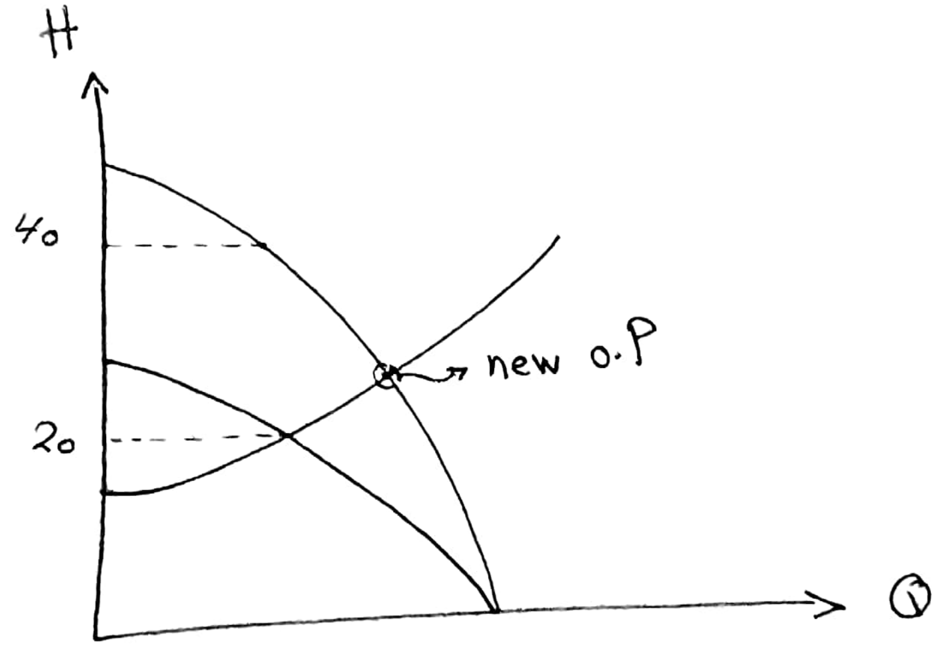
→ assume one pump gives $h = 20\,m$

↳ if 2 identical pumps connected in series

↳ will head be doubled ?

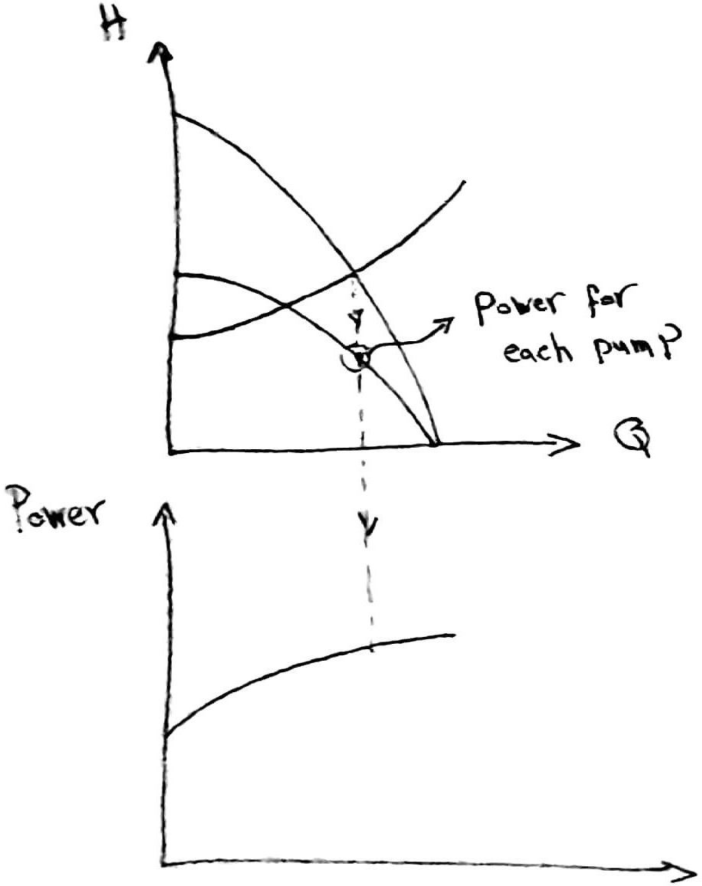
Answer:

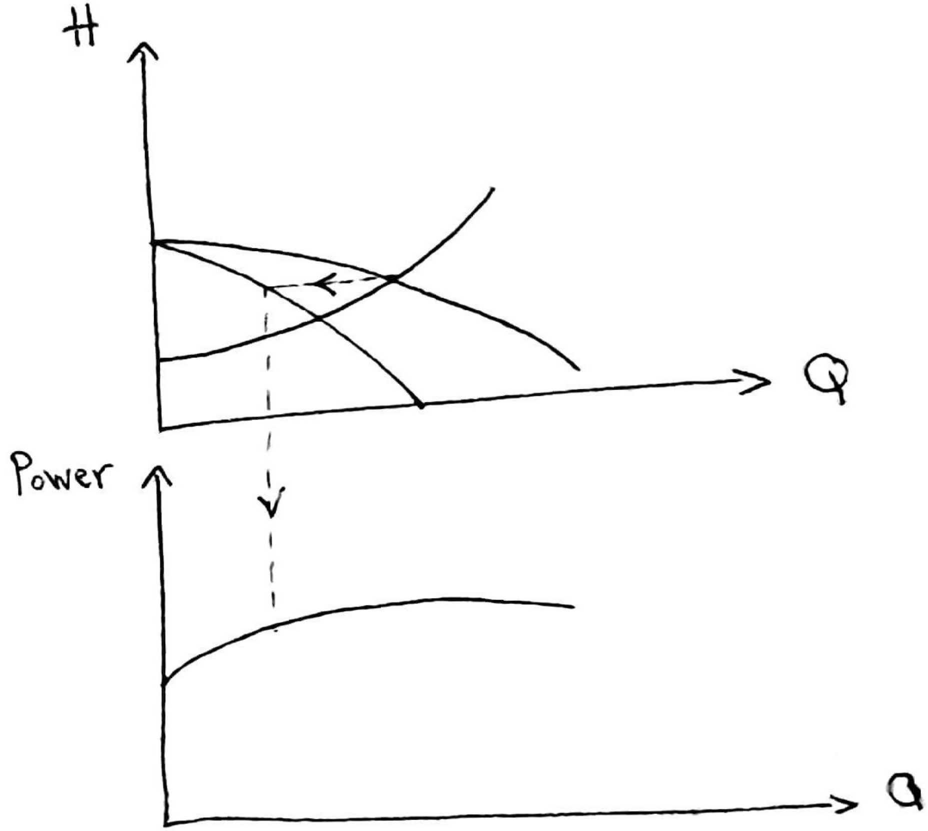
No :



→ new operating head will be less than double.

→ assume you have 2 pumps [identical] to give same h, Q,
↳ are they parallel or series?





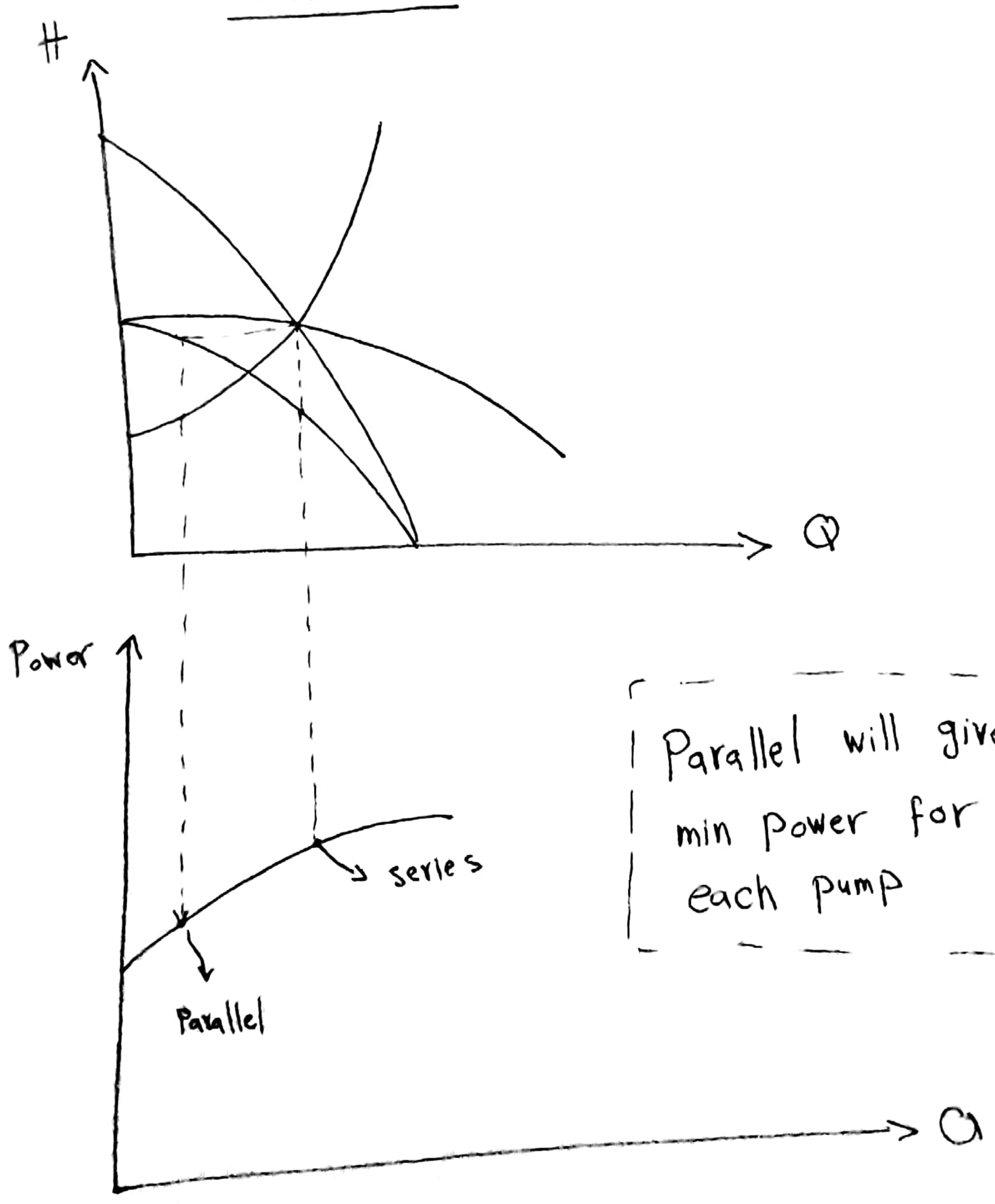
→ power consumed by one pump is lower in case of parallel

→ see next page to show both cases on same graph.

→ Assume you have 2 identical pumps to give same h, Q

↳ will they be parallel or series?

Answer:



Parallel will give min power for each pump

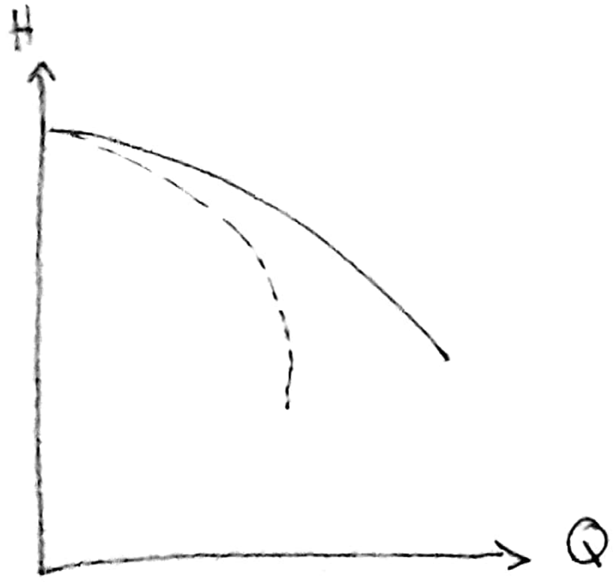
→ Signs of Cavitation :

1- drop in [H-Q] and [η-Q] curves

2- noise and vibration

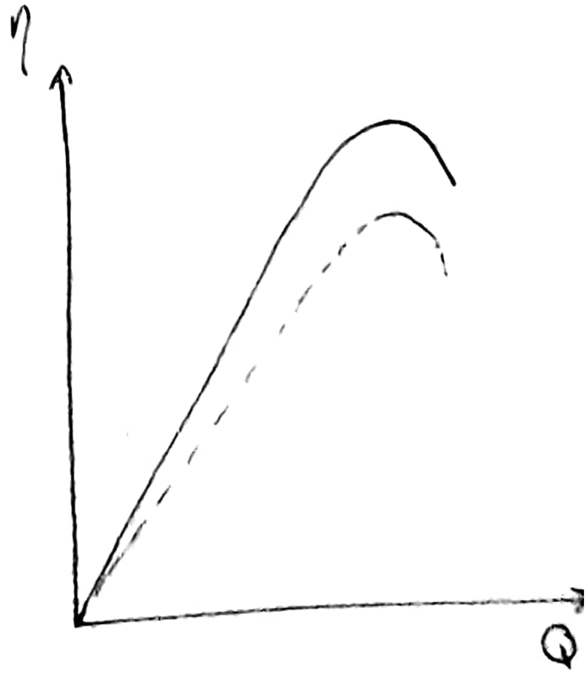
3- pitting on blade surface

→ Sign ① :



— before

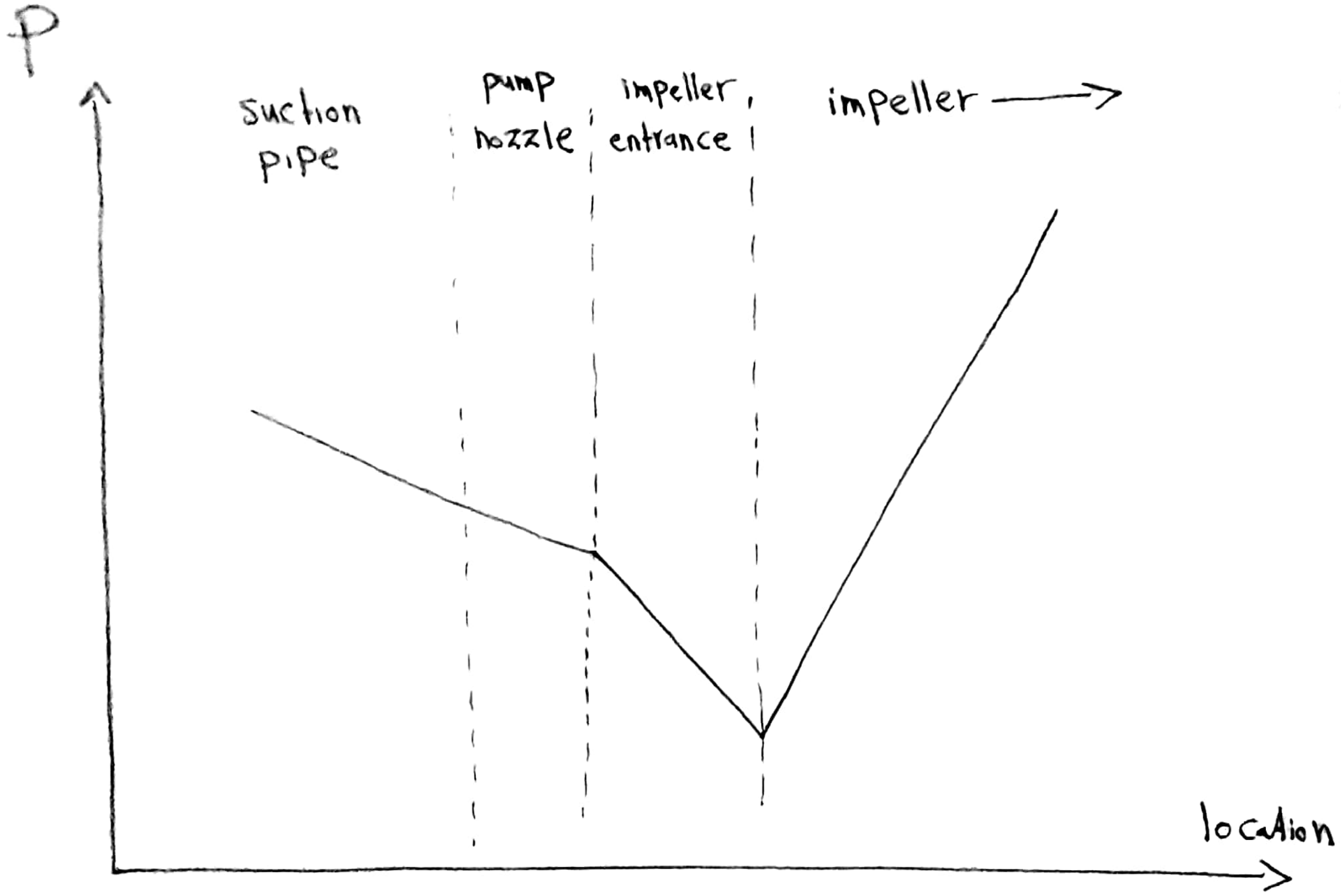
---- after



— befor

--- after

→ pressure gradient along pump suction :



slide 38

→ <u>important</u>:

→ Can we put valves on suction side of the pump ?

<u>Answer</u>:

Yes, but only for isolation [maintenance]

↳ not for controlling flow

→ <u>Recall</u>:

$$h_{min} > h_{vap} - h_{atm}$$

→ to check cavitation for pumping station, will we check it in summer or winter ?

↳ vapor pressure is directly proportional to Temp.

↳ So, if it is checked safe in summer

↳ it will of course will be safe in winter

→ when we placed pumping station in Alex

↳ no cavitation

→ but when placed in Aswan

↳ Cavitation occurs

↳ why ?

Answer :

→ Temperature is higher in Aswan

↳ So, Vapor pressure is higher

→ So, it's more exposed to cavitation.

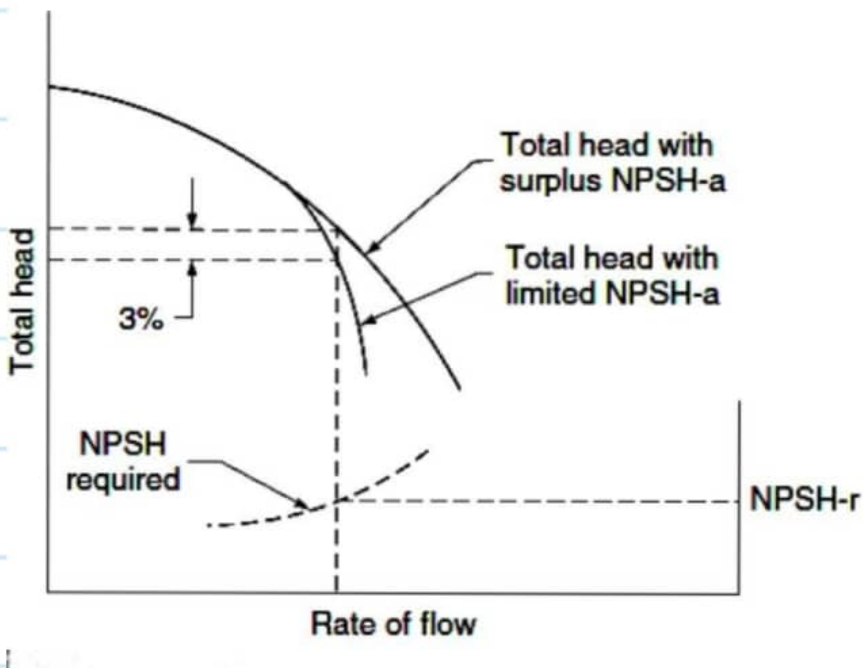
→ How to measure NPSH)$_{Required}$?

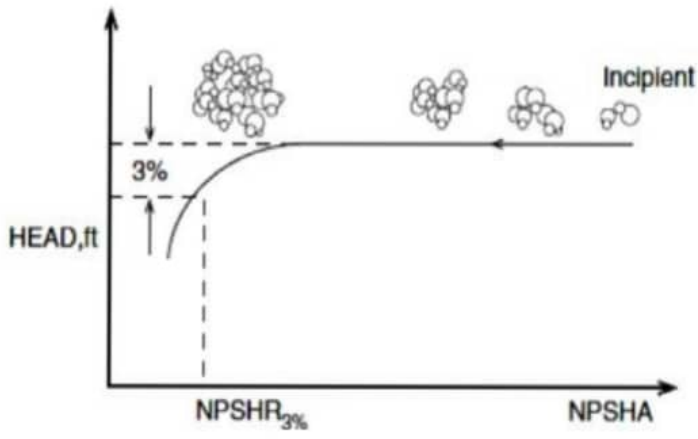
→ first, we reduce the energy before pump by means of throttling.

→ then, we measure the net positive suction head.

→ when a reduction in normal head with a value of 3 %

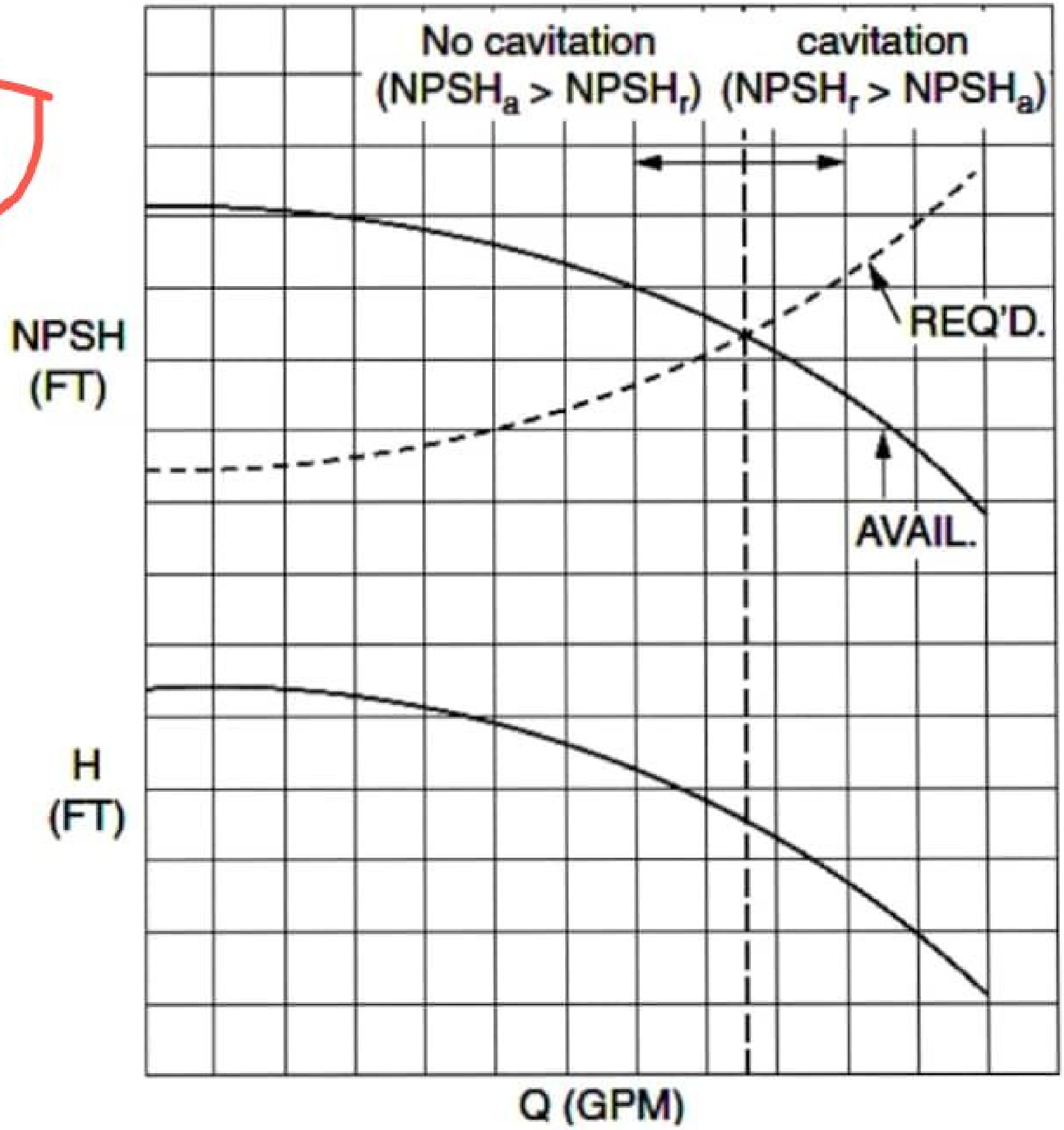
↳ the corresponding NPSH will be NPSH)$_{required}$





Scanned with CamScanner

سؤال امتحان مهم جدا



Variation of NPSH$_a$ and NPSH$_r$ with Flow through the System

→ How to avoid cavitation?

1- install pump with lowest position w.r.t suction tank level.

2- install pump at nearest position to suction tank.

3- reduce sources of eddy losses in suction line.

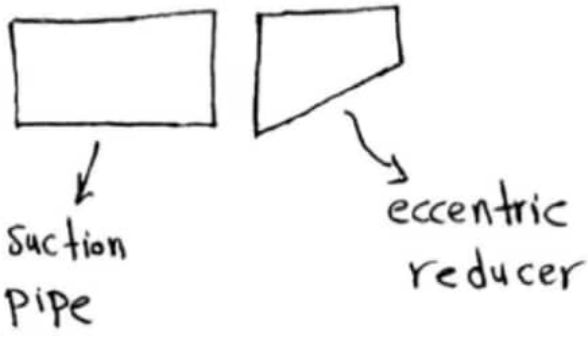
4- use of eccentric reducer

5- using of inducer

6- reducing fluid temperature

→ let's talk about number 4, 5, 6 in some detail.

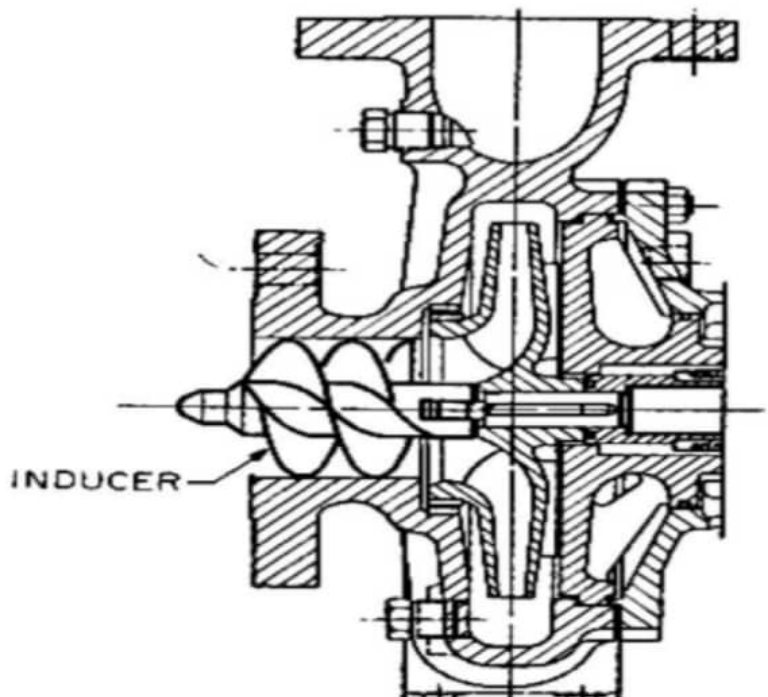
(4) eccentric reducer:



→ to prevent air entrapement in suction line.

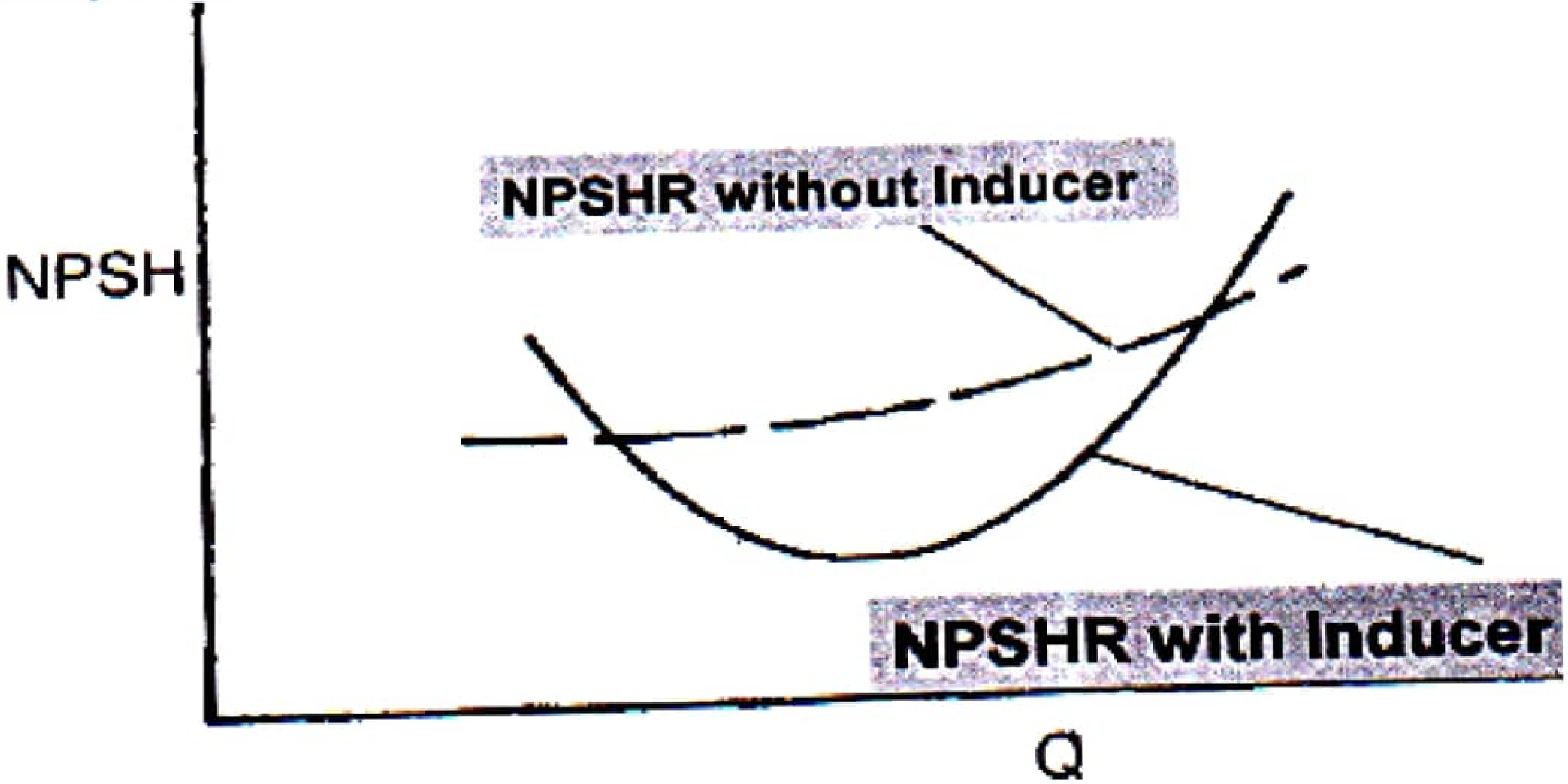
(5) Booster pump:

integrated booster pump used to increase suction pressure at pump inlet



Inducers are used to **Reduce $NPSH_r$**

Conventional Inducer is designed to lower the NPSHR value of the main pump in the range of the duty point, but they only allow a limited operating range of the pump.



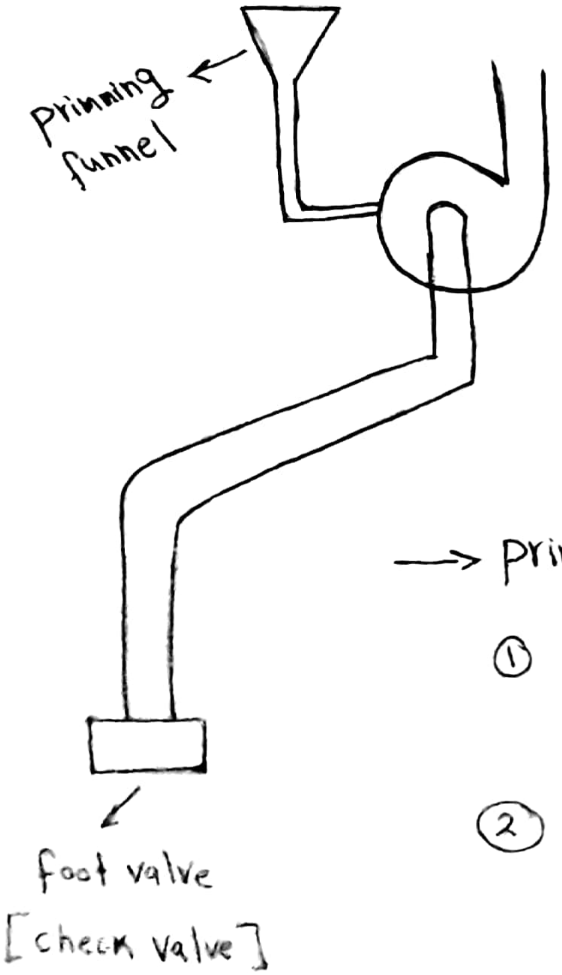
6 reduce liquid temperature :

→ temperature must be lowered to decrease vapor pressure to avoid cavitation.

→ but, it also shouldn't be too much lowered because this will result in higher pressure drop.

Primming :

Small pumps :

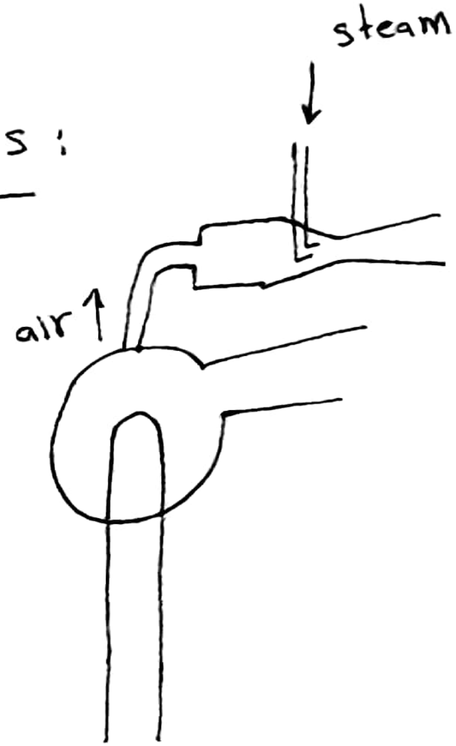


→ primming made by ;

① foot valve at pipe inlet.

② filling system through primming funnel

Large pumps :



→ by steam ejector

Wearing rings :

→ wearing rings provide a leakage Joint between impeller and casing

→ in case of Small pumps [due to wear]

↳ to restore original clearance between any two mating parts.

① buid new wearing parts by welding or metal spraying

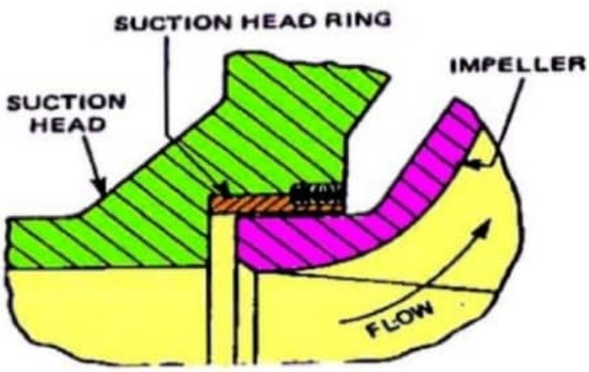
② buy new parts.

→ in case of Large pumps :

→ We use impeller and casing rings to reduce wear

→ types of rings :

Flat type :



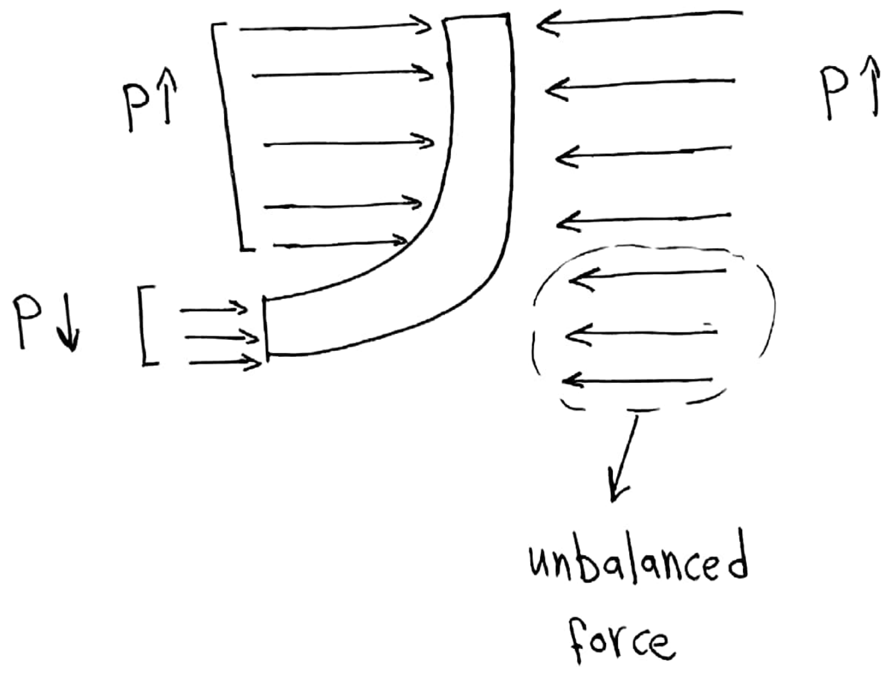
L - type :

→ L- type : - Low leak
- high friction

Axial thrust :

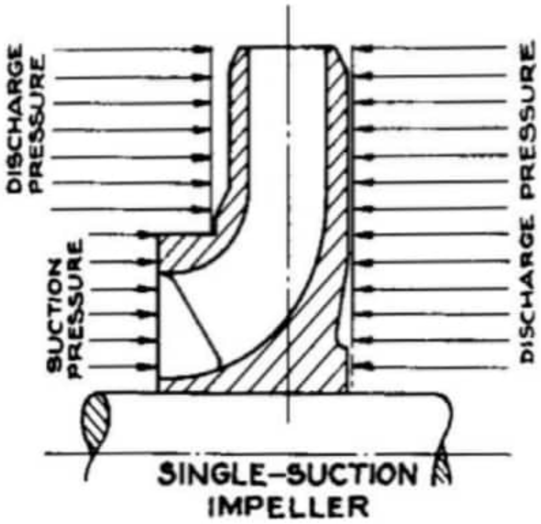
→ Axial thrust is summ of unbalanced forces acting on impeller

→ to show the idea :

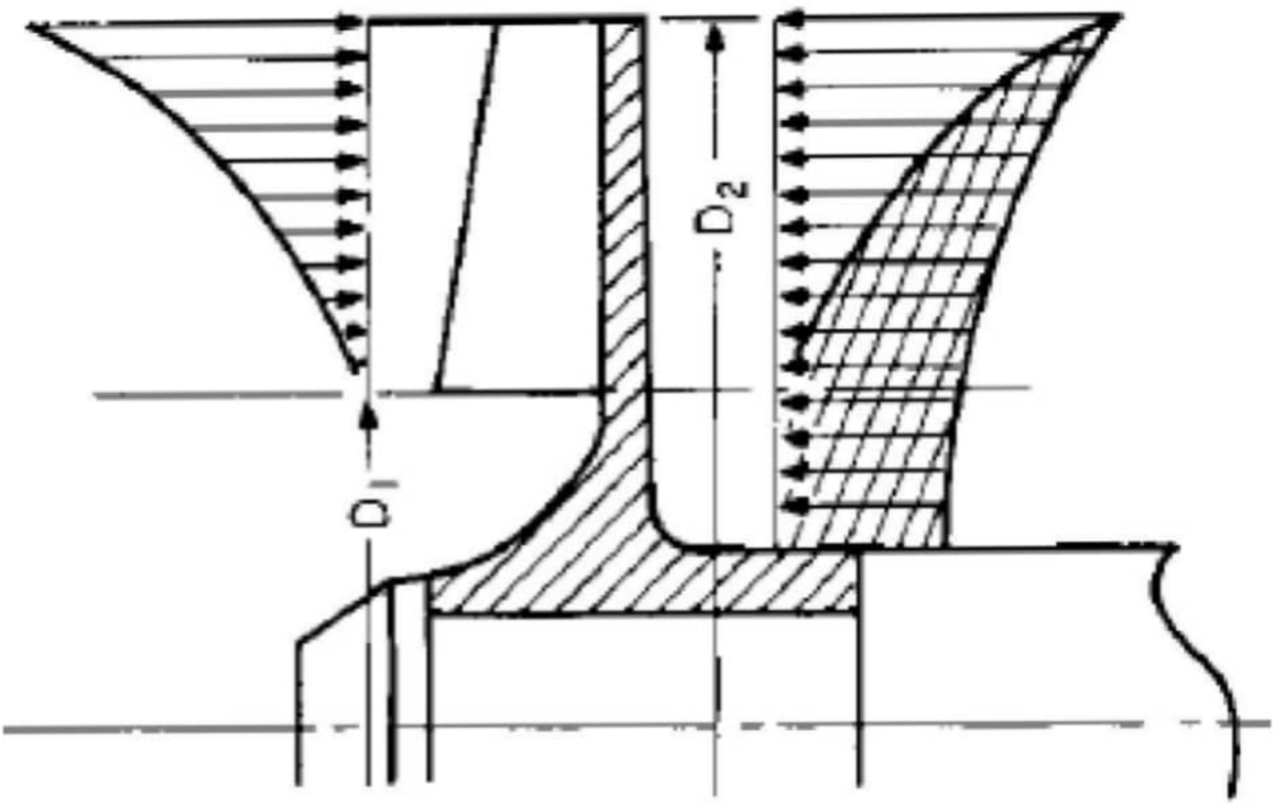


→ Pressure distribution :

→ Closed :



→ Semi-open :



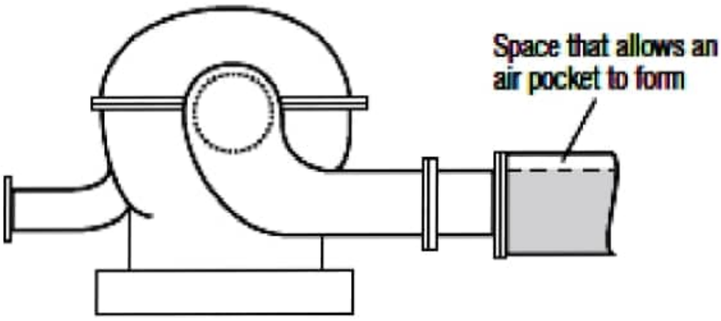
Scanned with CamScanner

How to eliminate axial thrust?

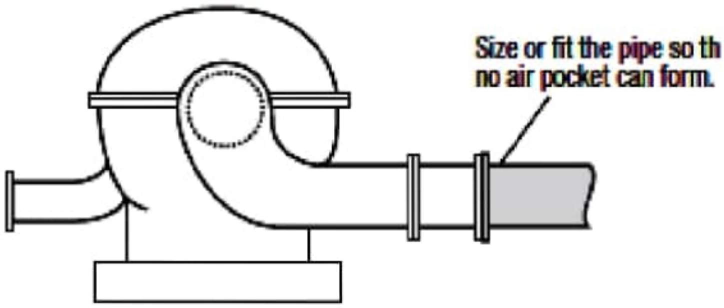
1. using pump-out vanes
2. balancing holes
3. double-suction impeller
4. opposite impellers
5. using hydraulic balancing devices
   - → balancing drum

→ Let's clarify each one

Incorrect

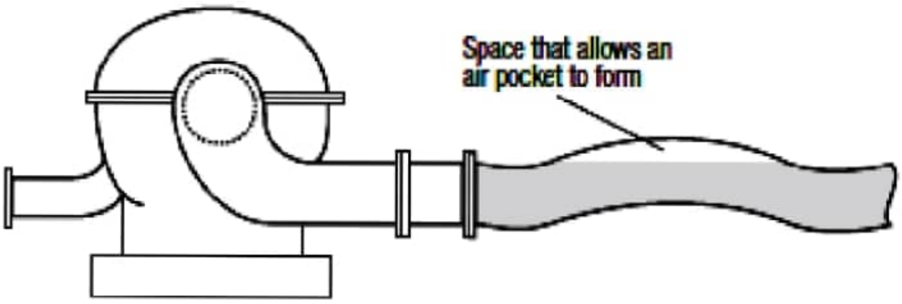


Correct

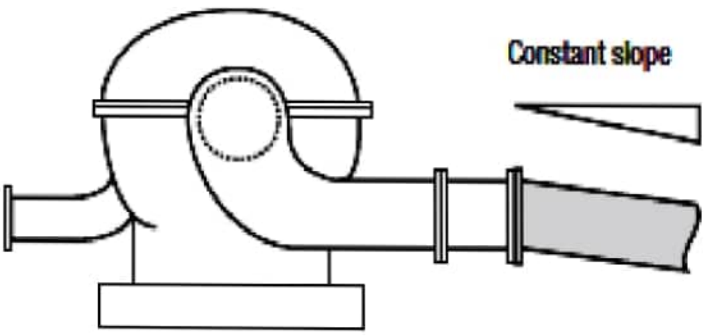


Common Pipe Configuration Problems and How to Correct Them

Incorrect

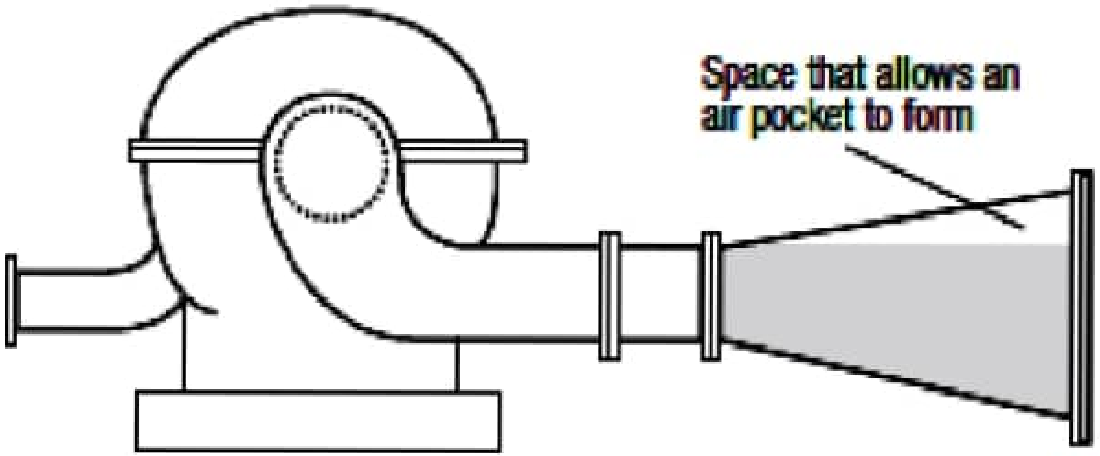


Correct

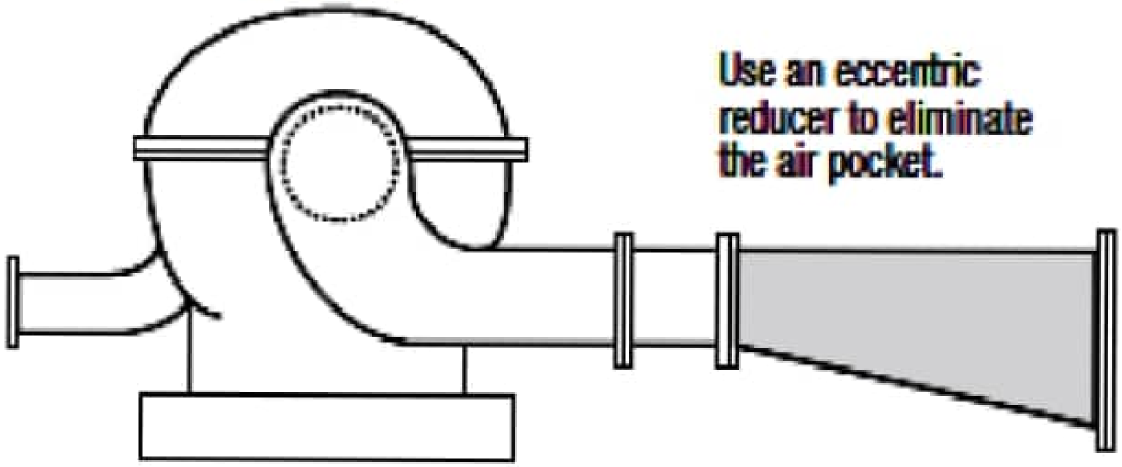


Common Pipe Configuration Problems and How to Correct Them

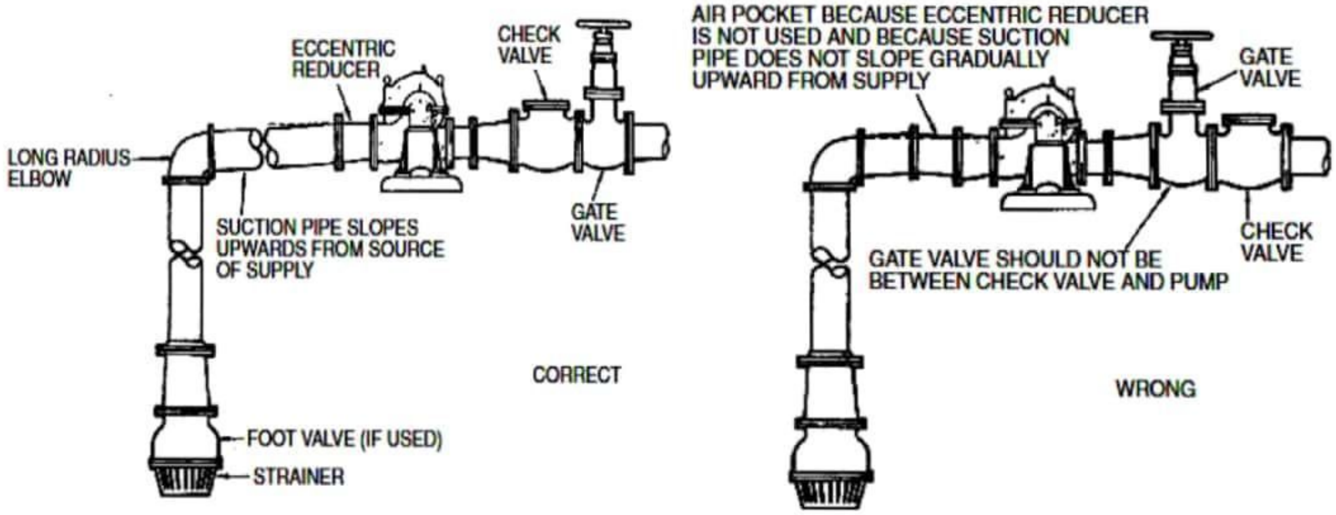
Incorrect



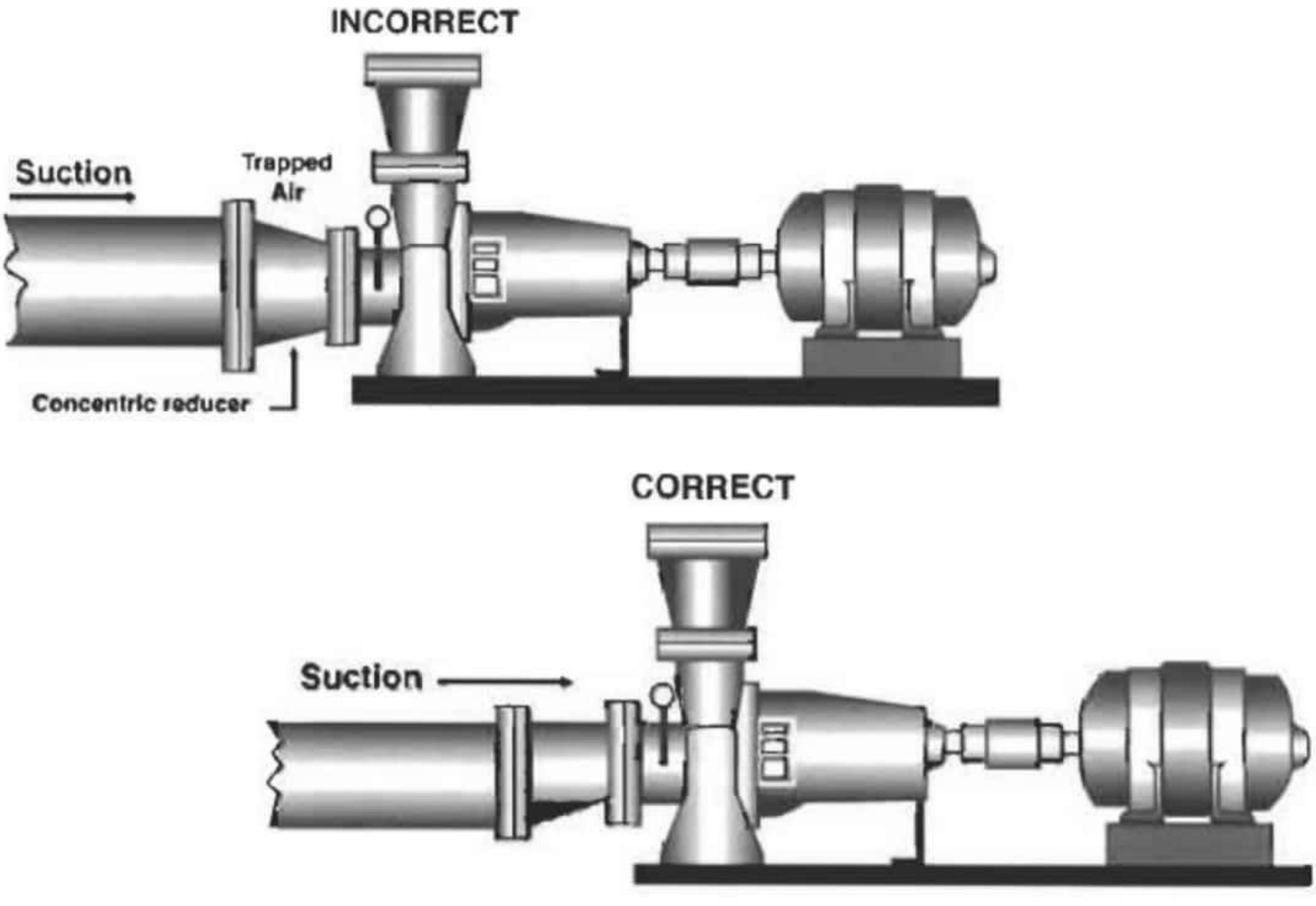
Correct

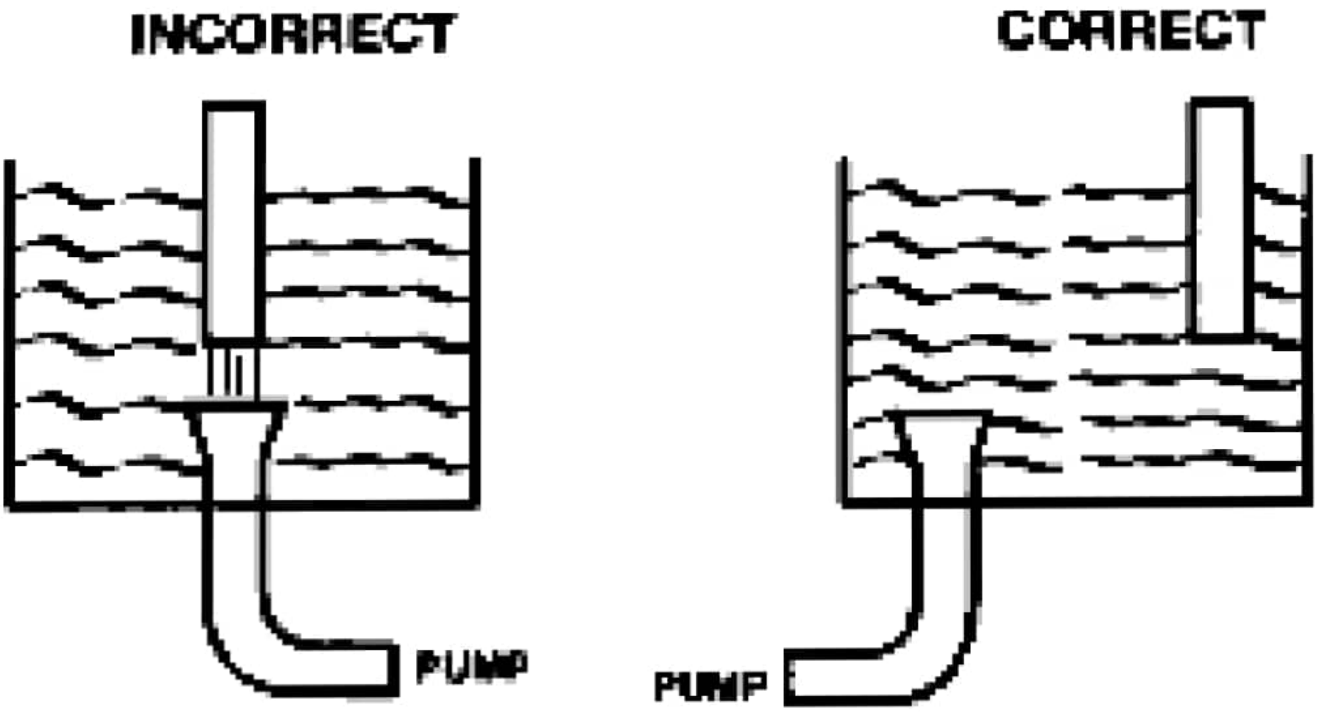


Common Pipe Configuration Problems and How to Correct Them

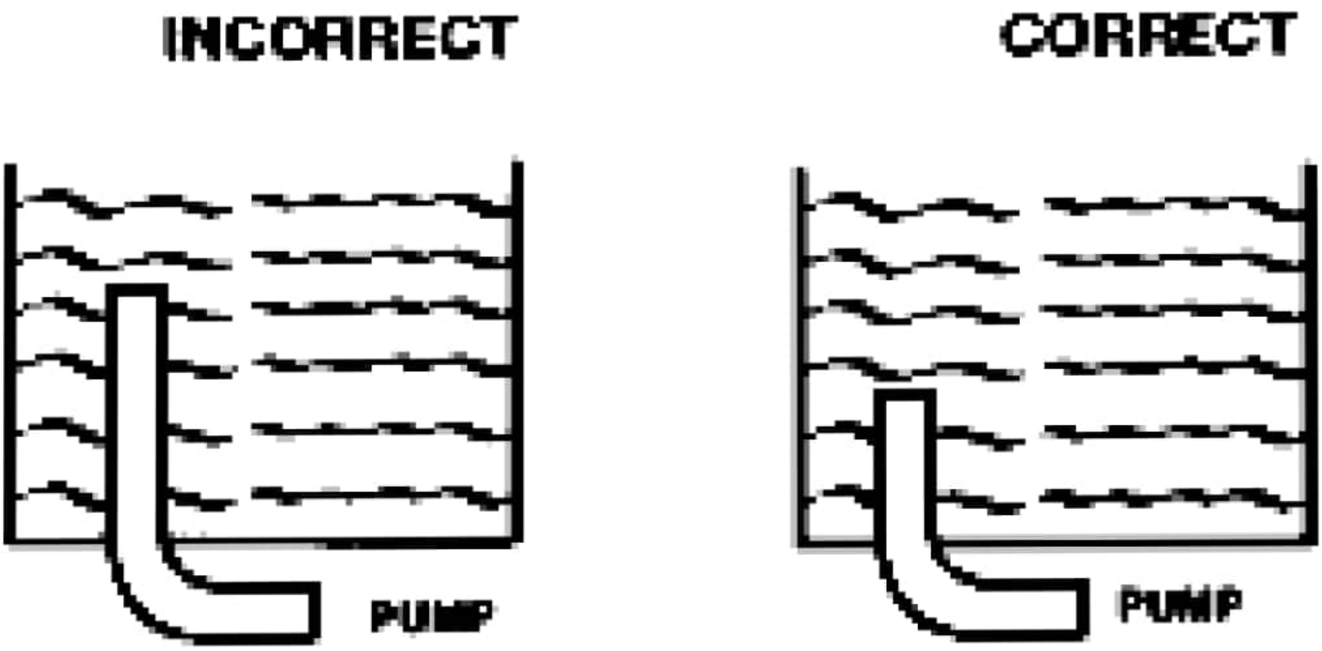


Suction pipe should have an upward slope towards pump and have eccentric reducer

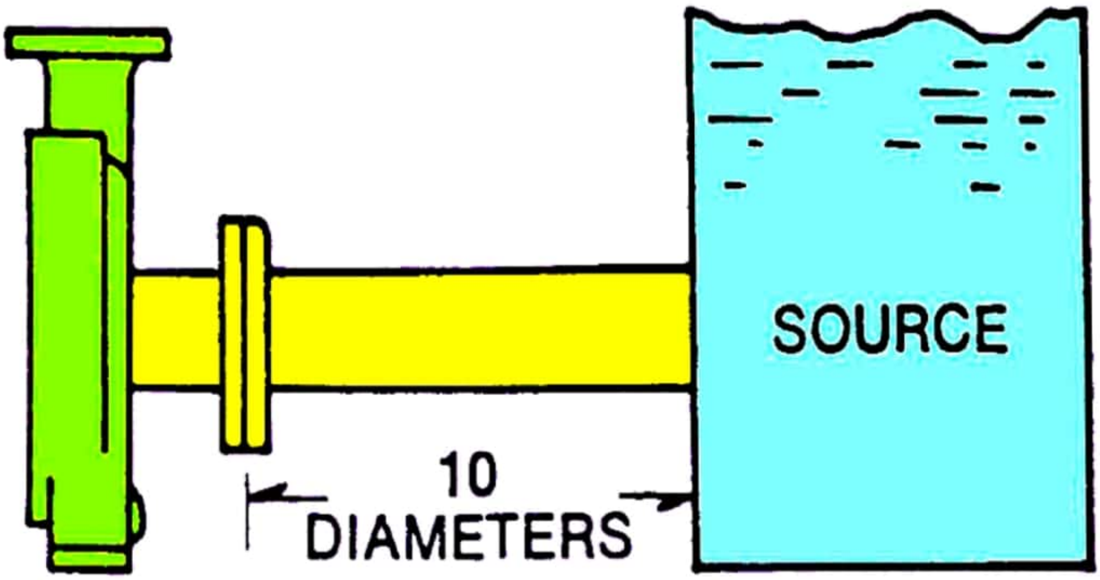




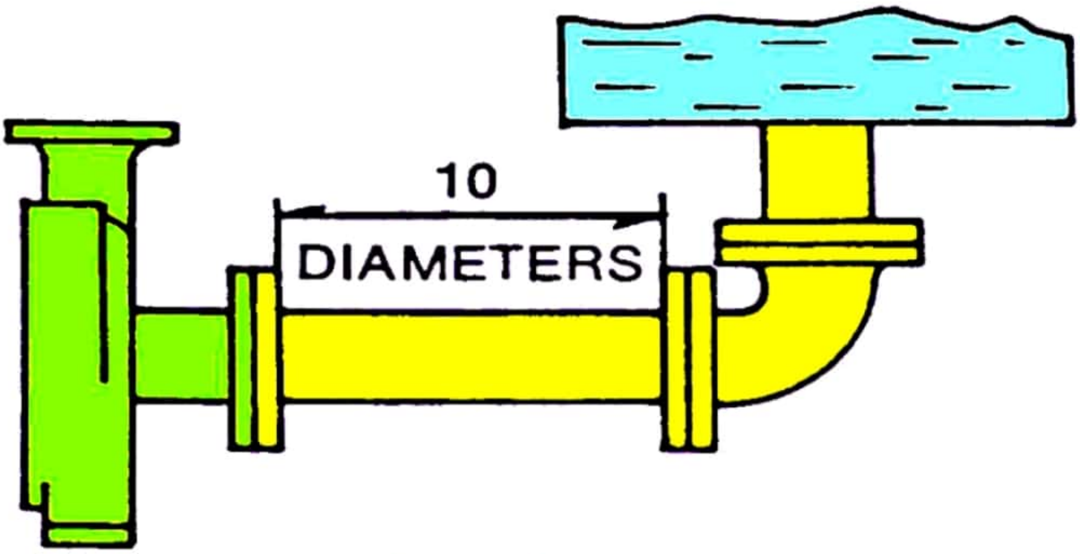
The **in-Flow Pipe should** **not** **Cause**
Interference with the **Drain** **Pipe**



Drain Pipe Design must Respect
Proper Submergence

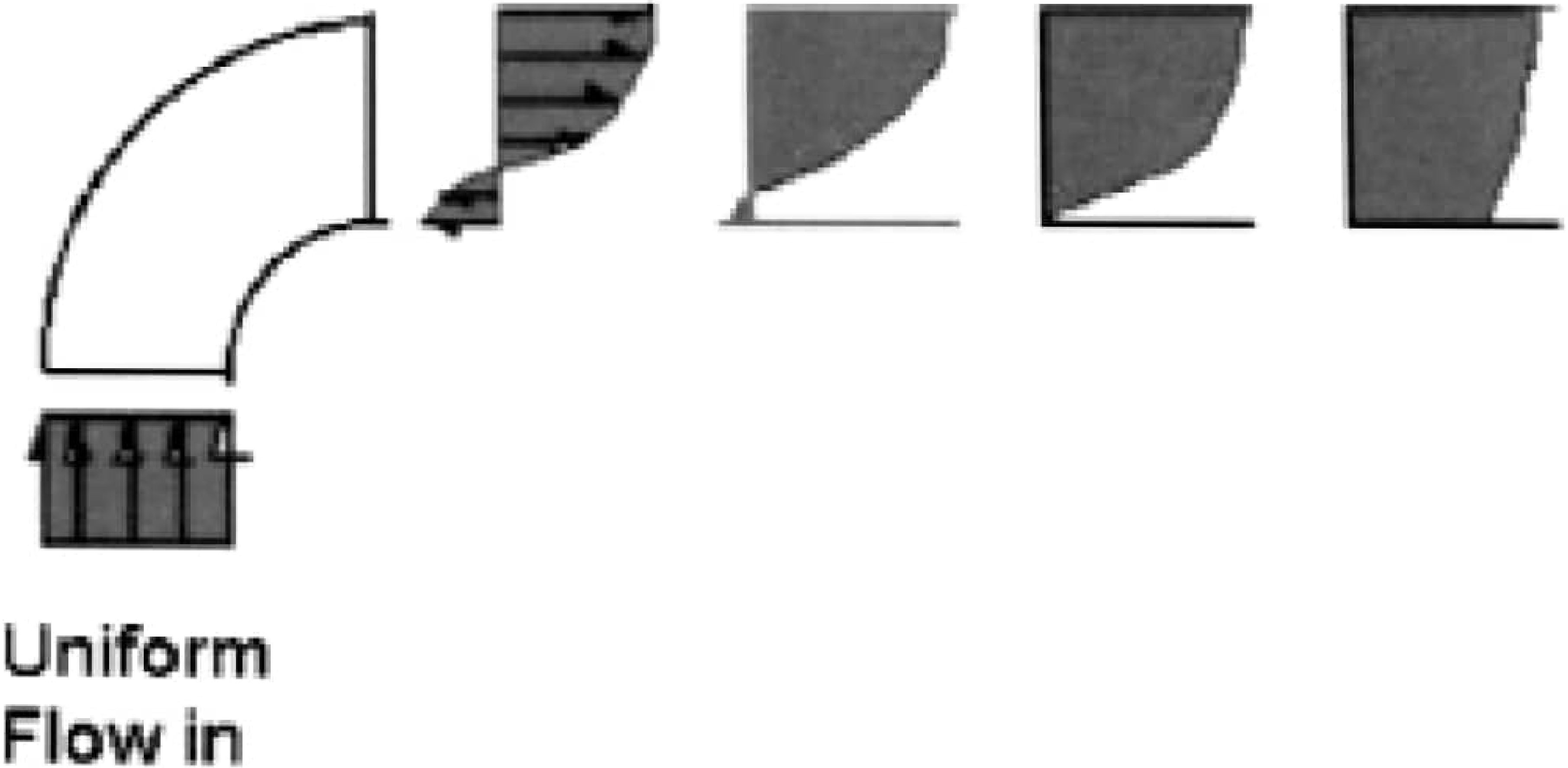


Ideal Piping Configuration
with a **Minimum of 10 Diameters** of **Straight Pipe**
between the **source** and the **Pump Suction**.

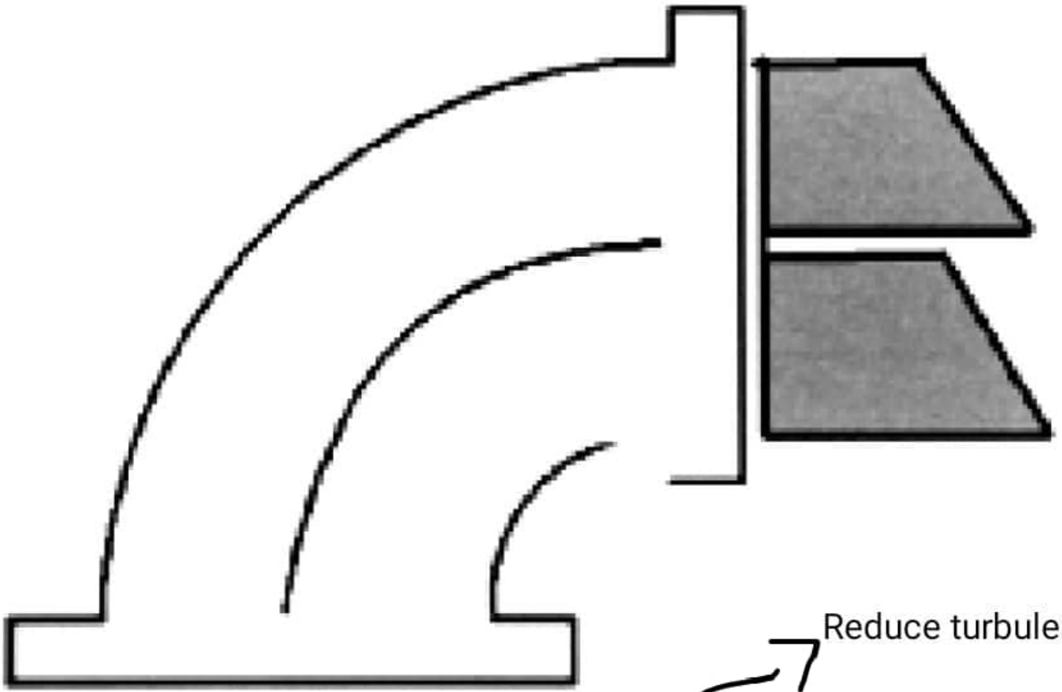


Elevated Source:
Preferred Piping to be in **One Plane,**
with a **Minimum of 10 Diameters** of **Straight Pipe**
between the **Elbow** and the **Pump Suction**.

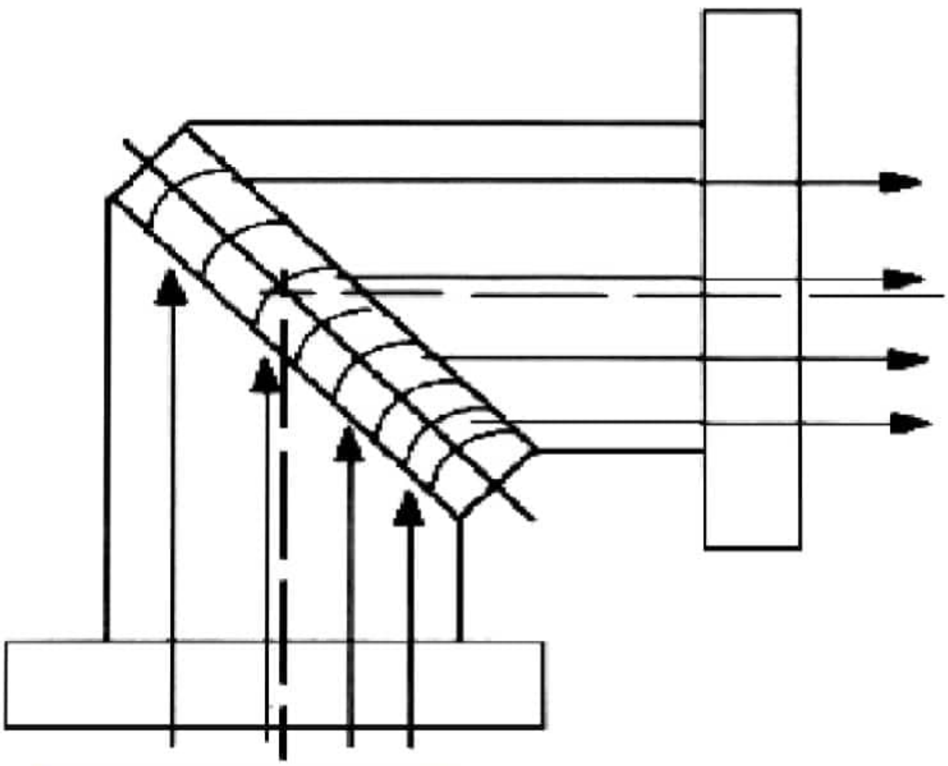
Distorted flow mixes out with distance



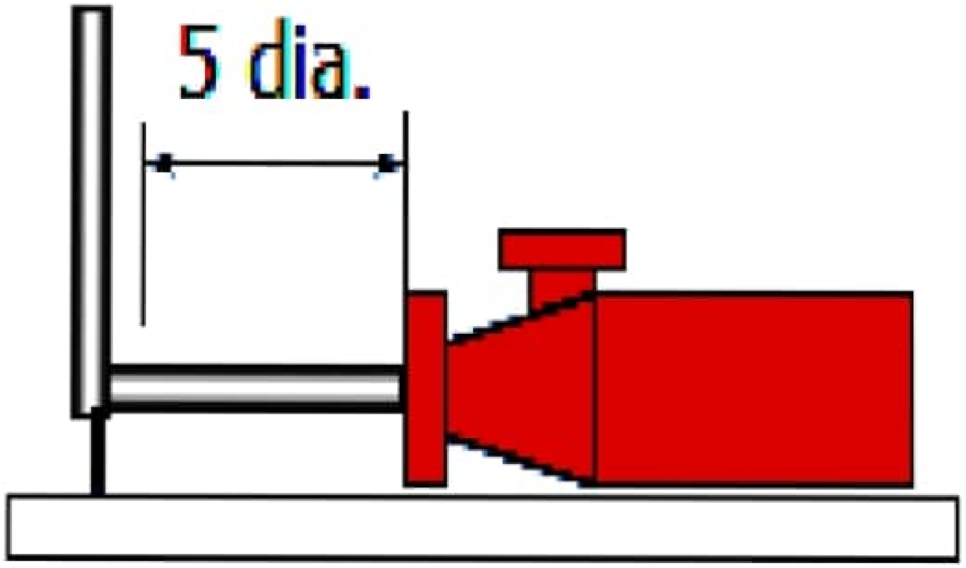
Straight Lengths are **necessary** to **Allow Flow** to **Stabilize** into a **more Uniform Pattern**

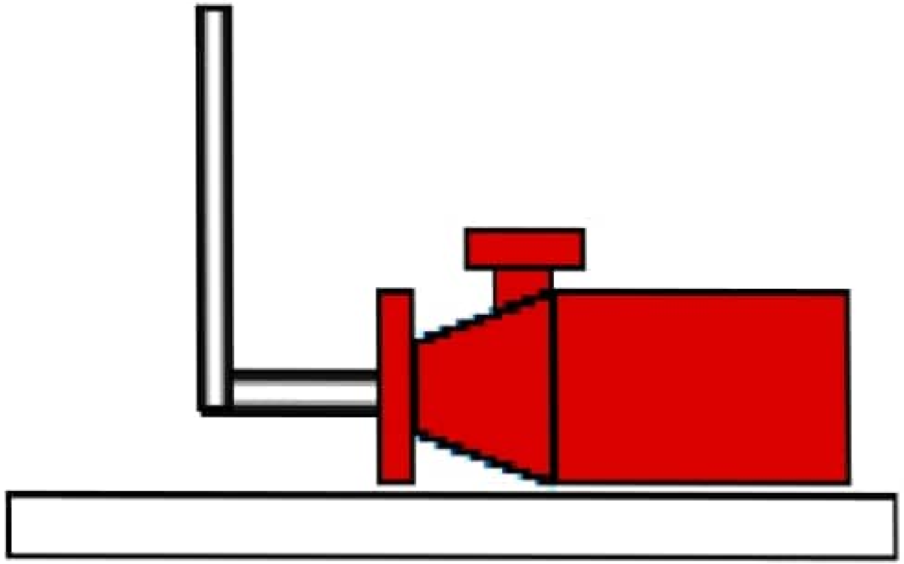


Central Splitters

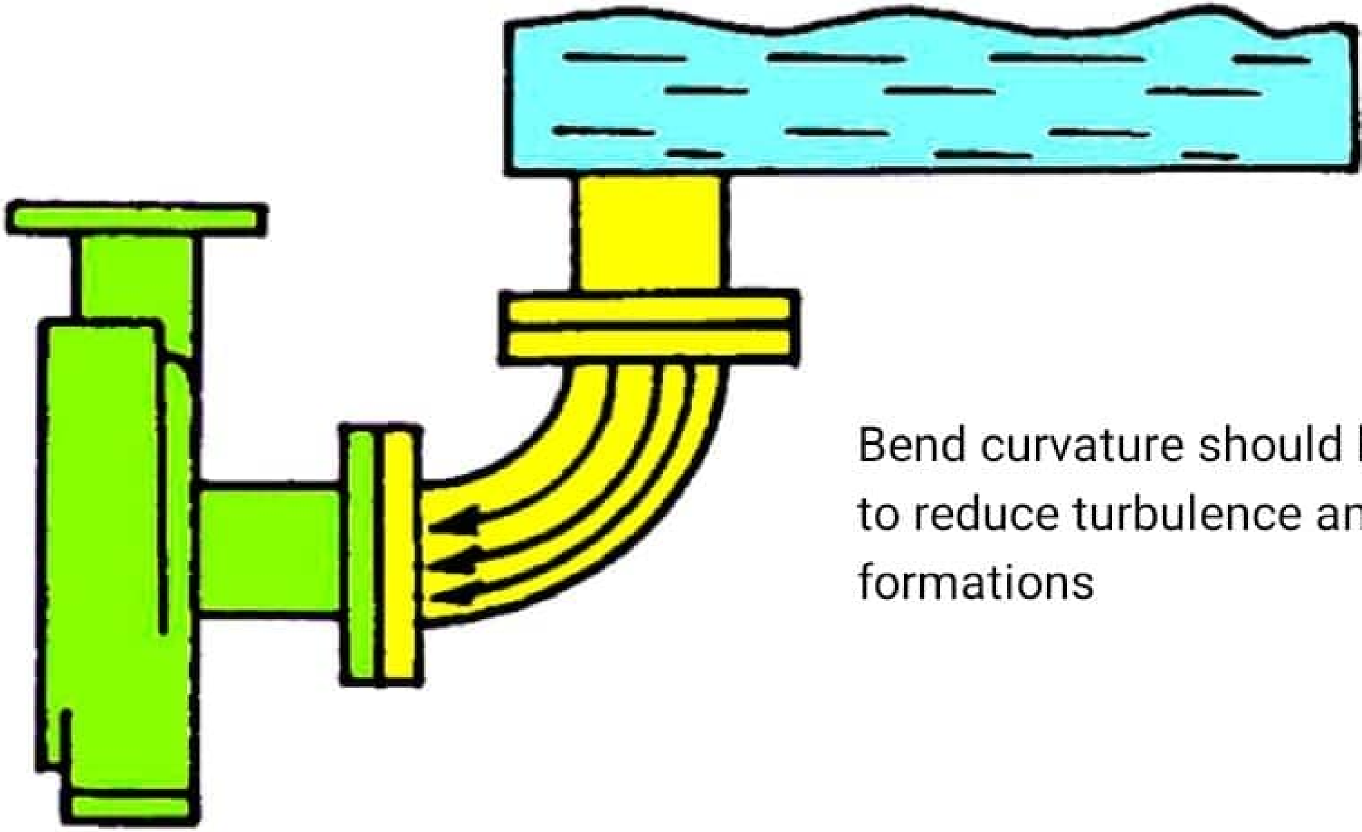


Cascade Turning Vanes Eliminate Flow Distortion in the Outlet Flow

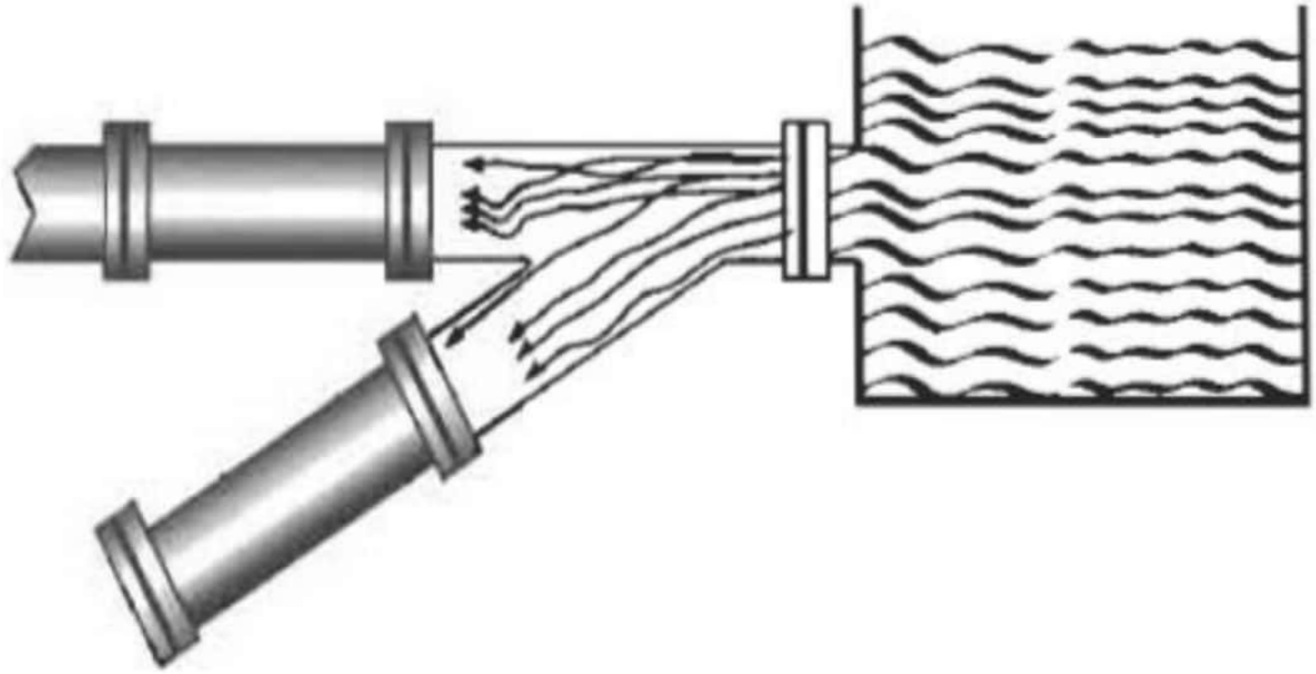




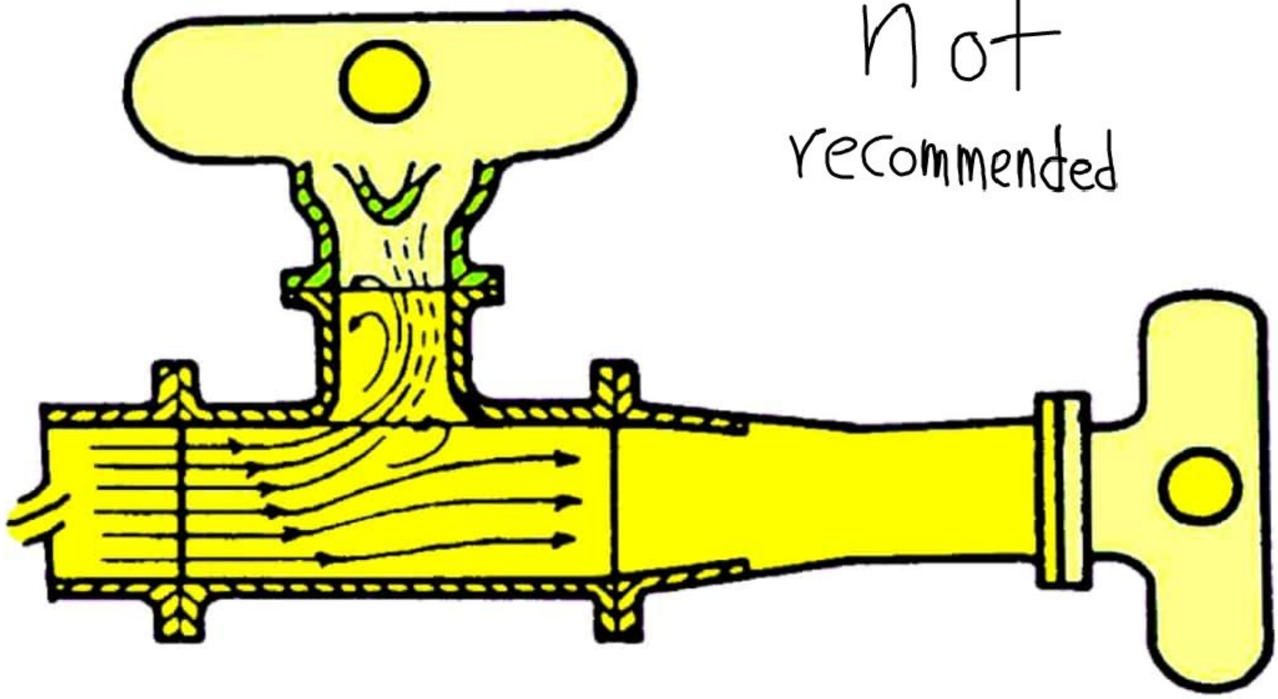
RIGHT	WRONG
1- Pipe Supported	1- Pipe Weight hangs on Pump Flange
2- Length of Suction Piping Allows Even Impeller Loading	2- Short Suction Pipe results in Uneven Impeller Loading



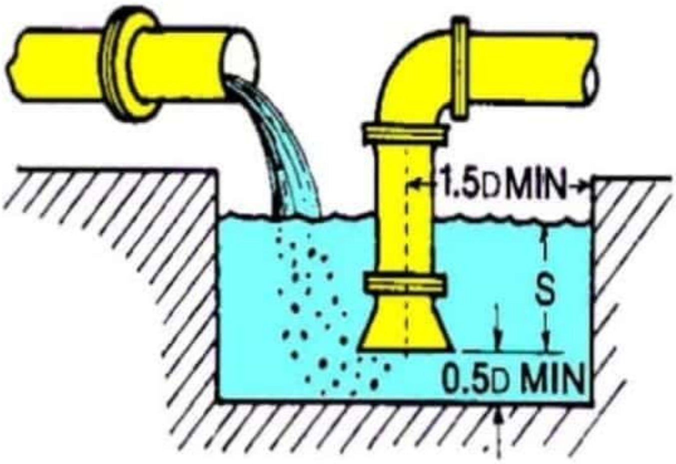
Bend curvature should be large enough to reduce turbulence and prevent void formations



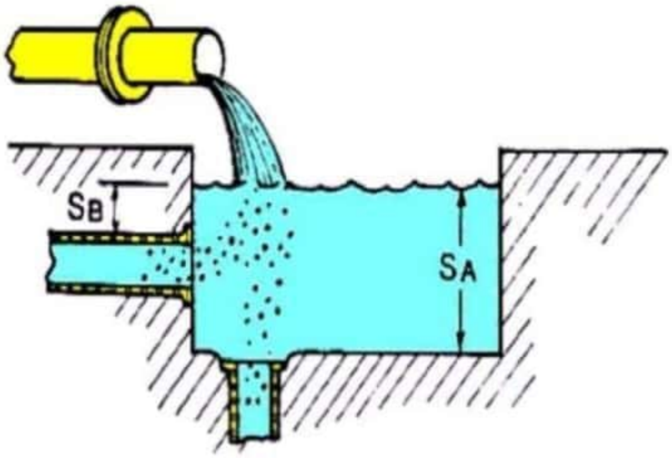
Use a "Y" Branch and not a "T" Branch.
This will **reduce Turbulence**



not
recommended



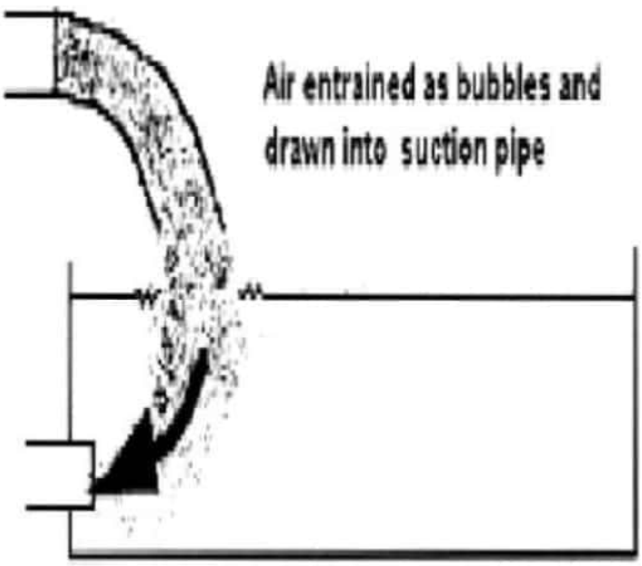
Correct

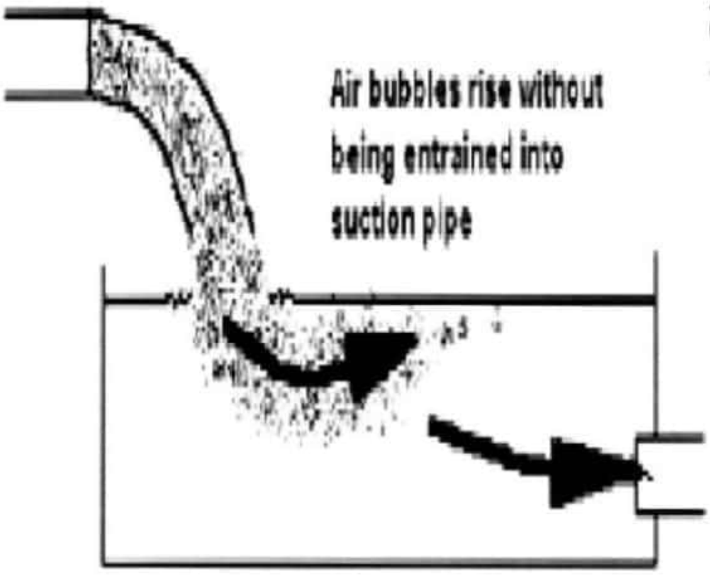


wrong

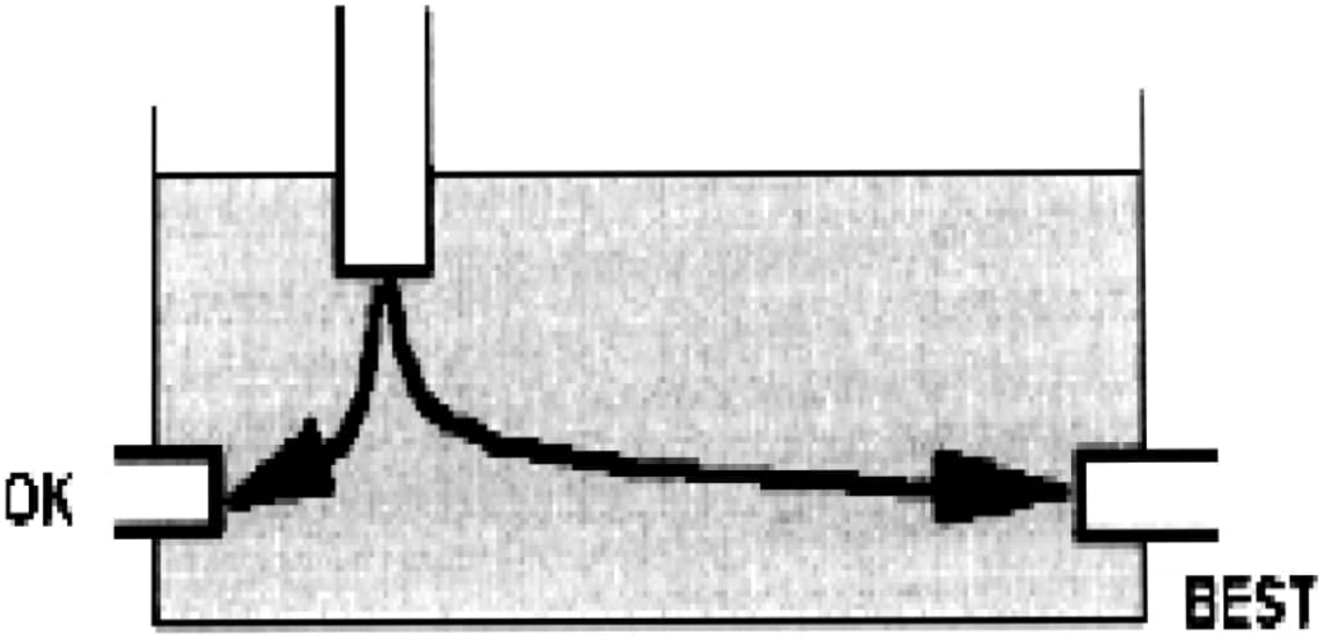
Solution : we should separate
The two lines by distance

Free falling will result in air bubbles
into suction line

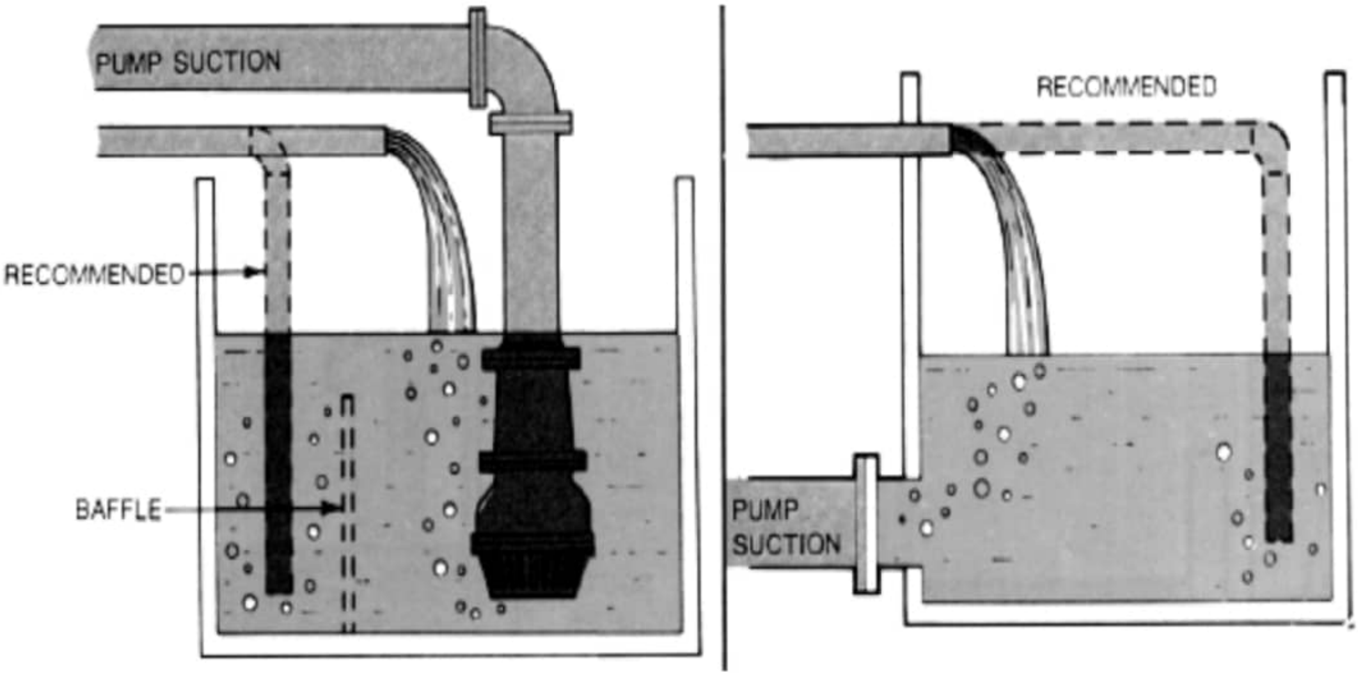




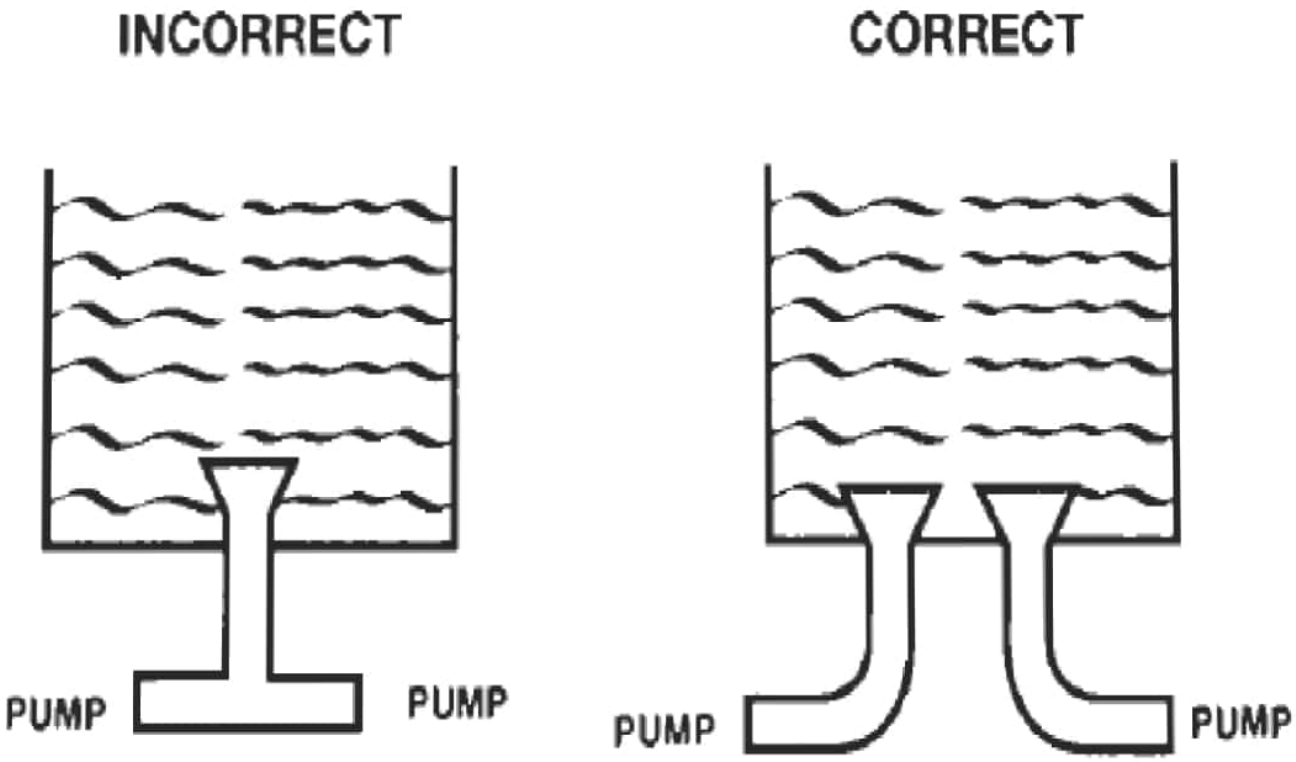
Good and Bad Suction Pipe Location When Surface Aeration is Present

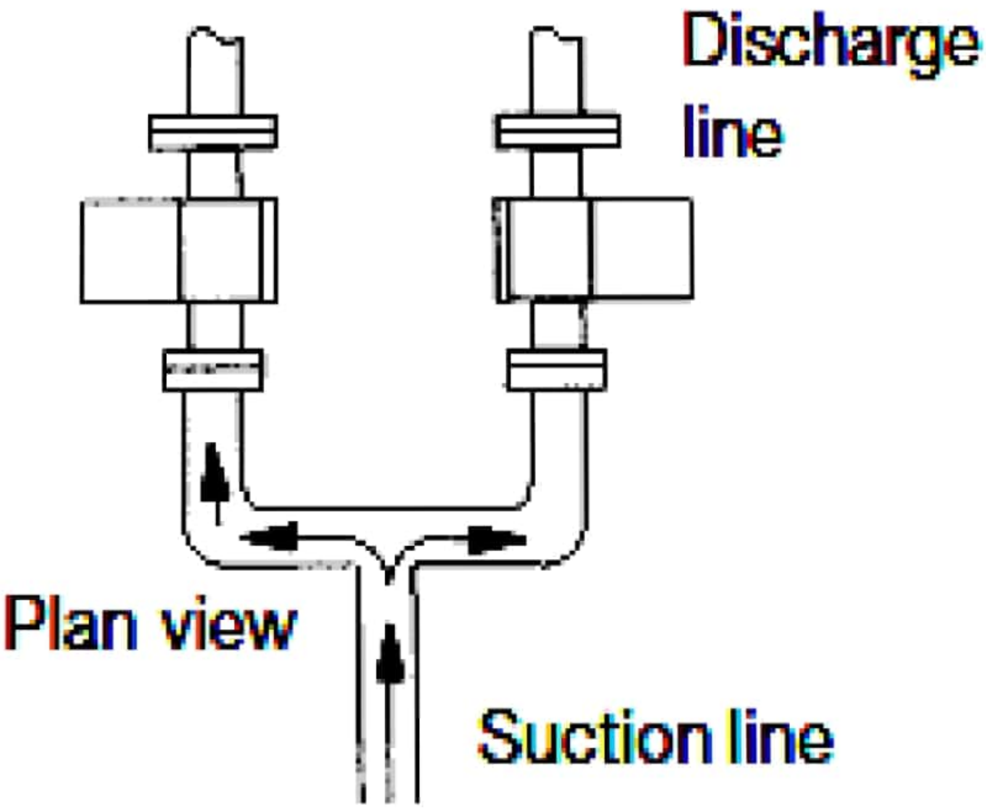


Preferred Sump Feed Arrangement is
Below Liquid Level to **Avoid Air Entrainment**

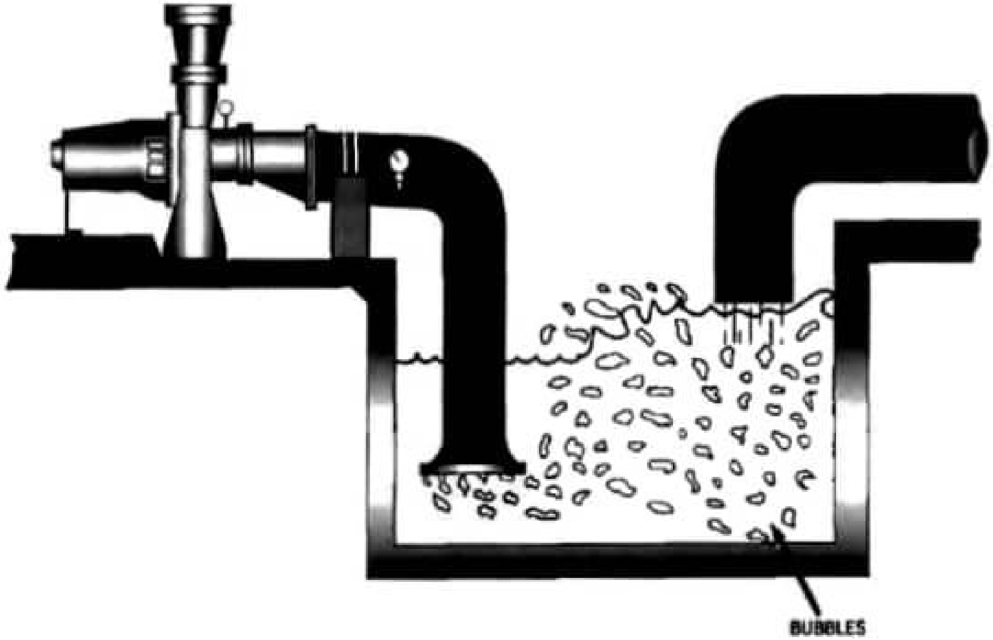


Return Lines Back to **a Pump Suction Vessel**
should **not** be allowed to **Free-Fall** into the **Vessel**.

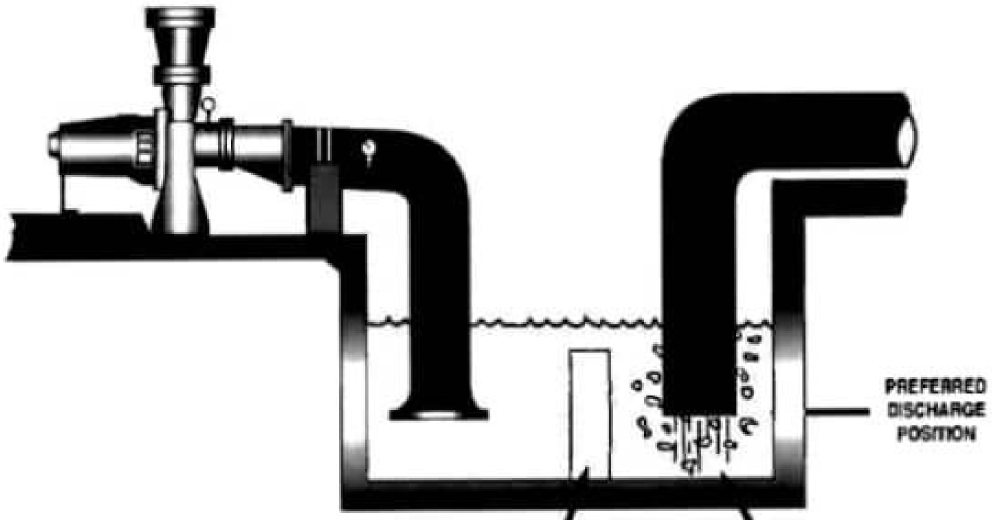


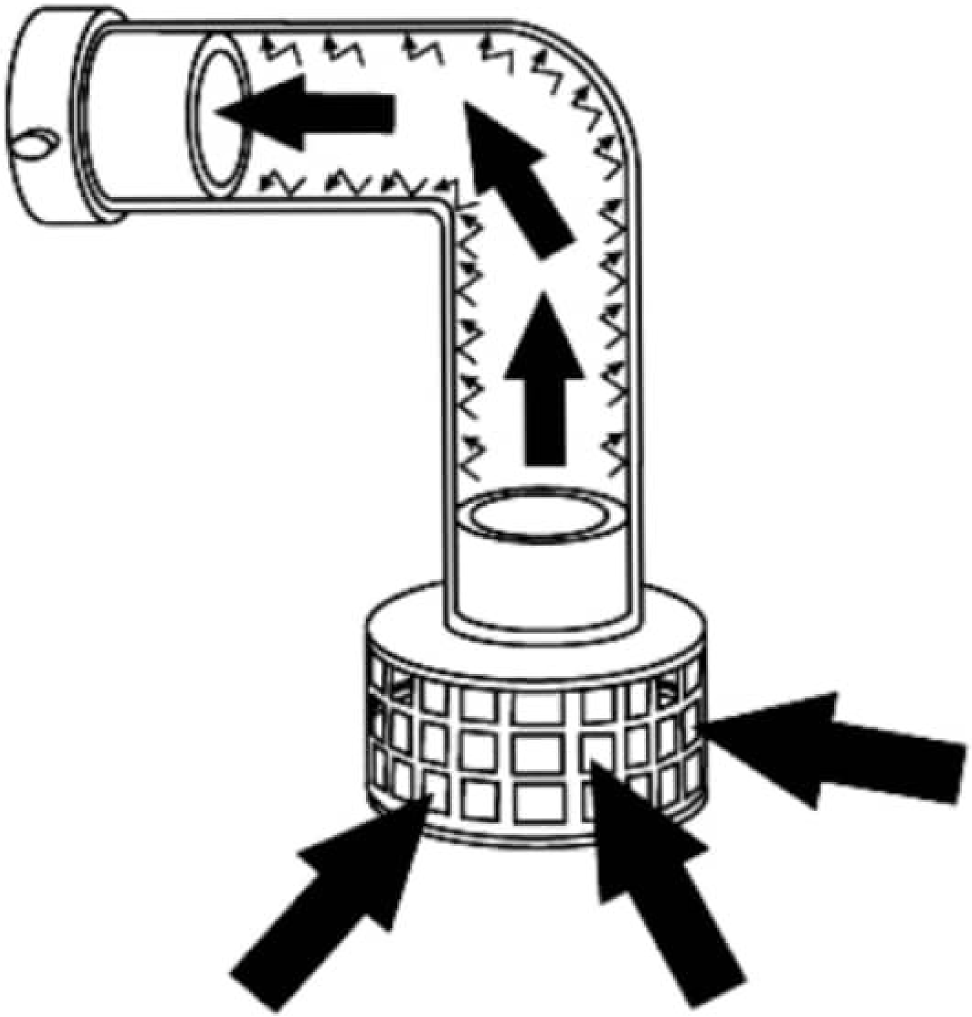


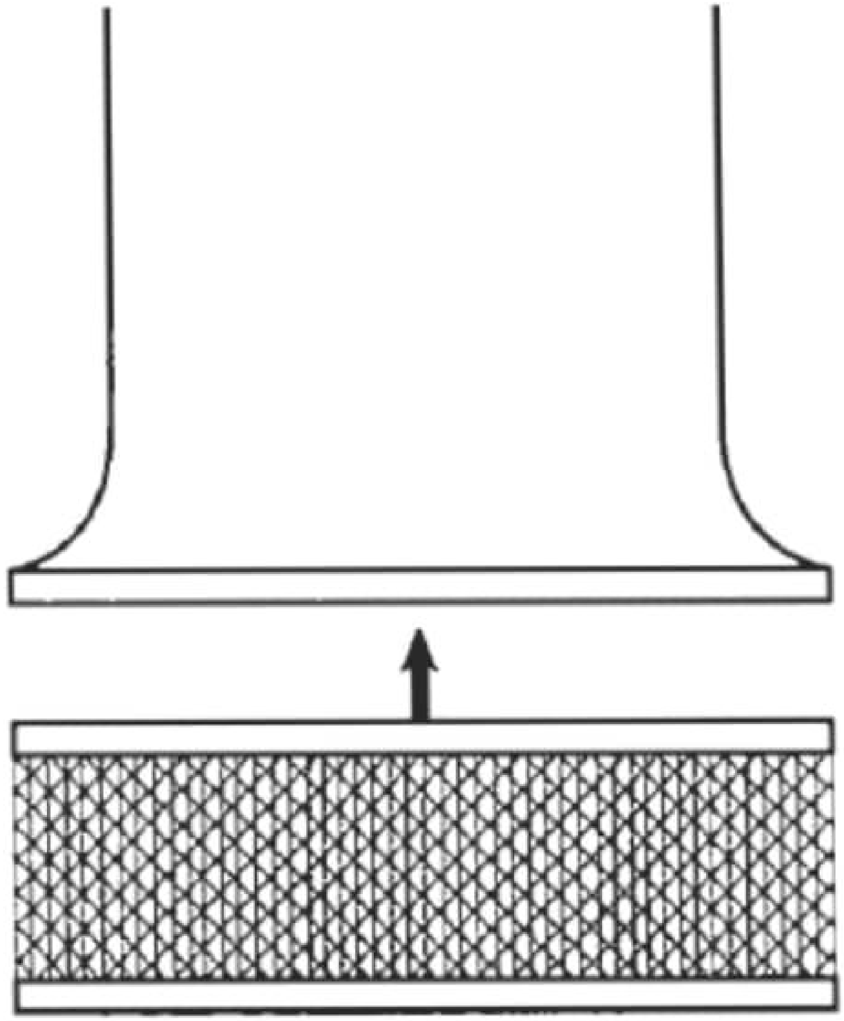
Avoid Common Suction Lines



Baffle plates used to
Reduce turbulence







Basket Strainer: Screen Area should be **4 X Area** of the **Entrance Pipe.**

Avoid Tight Mesh Screens because **They Clog Quickly**