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# THE PUMPING SYSTEM

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# The Pumping System

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## Bernoulli's Equation

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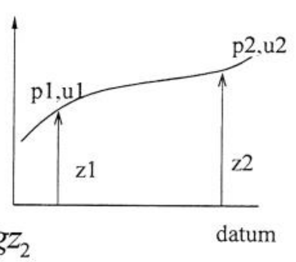
- Integration Euler Equation by assuming incompressible flow
- Relates the pressure, velocity and height along the flow direction under 'ideal' conditions.

$$\frac{p}{\rho g} + \frac{u^2}{2g} + z = \text{constant} = H$$

$$p + \frac{1}{2}\rho u^2 + \rho g z = \text{constant}$$

$$p_1 + \tfrac{1}{2}\rho u_1^2 + \rho g z_1 = p_2 + \tfrac{1}{2}\rho u_2^2 + \rho g z_2$$



## Energy/Hydraulic Gradient

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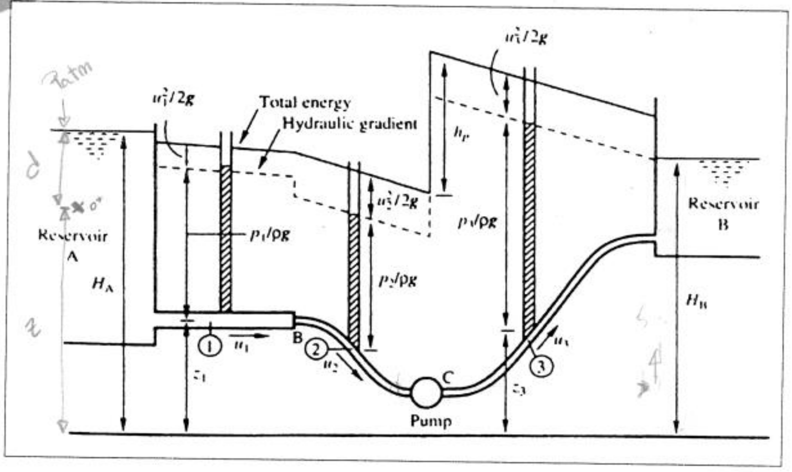


$\frac{P}{\rho g} + \frac{U^2}{2} + z = cte = H$

$P = \rho g d + P_{atm}$

$0^* \Rightarrow P_{gage}$, $U \approx 0$

$H = d + z$

It doesn't matter where is it in the reservoir, the total energy will be the same



## Energy Equation

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$$\frac{p_1}{\rho} + \frac{u_1^2}{2} + gz_1 + pumpinput = \frac{p_2}{\rho} + \frac{u_2^2}{2} + gz_2 + losses$$

Total energy at 1 + pump input = Total energy at 2 + losses

$$H_1 + \frac{u_1^2}{2g} + z_1 + pumphead = H_2 + \frac{u_2^2}{2g} + z_2 + headlosses$$

Total head at 1 + pump head = Total head at 2 + head losses

## A Simple System

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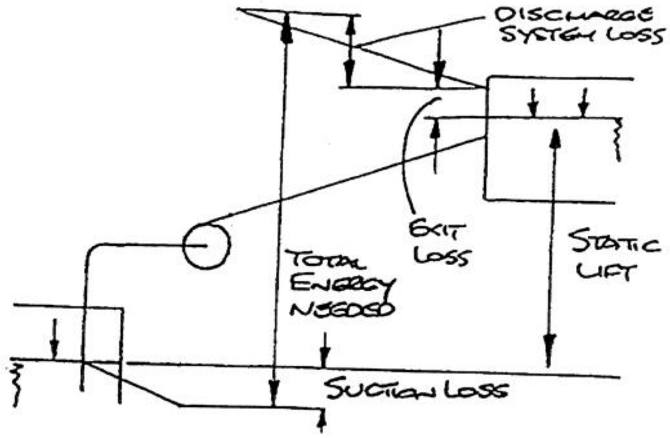




## System Characteristics

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System loss = Frictional loss + component loss

$$Headloss = \sum f_i \frac{l_i}{d_i} \frac{u_i^2}{2g} + \sum K_i \frac{u_i^2}{2g}$$

$$\Delta H_{sys} = K_{sys} Q^2$$

$$\boxed{H_{sys} = a + bQ^2}$$

a = static lift

# Operating Point

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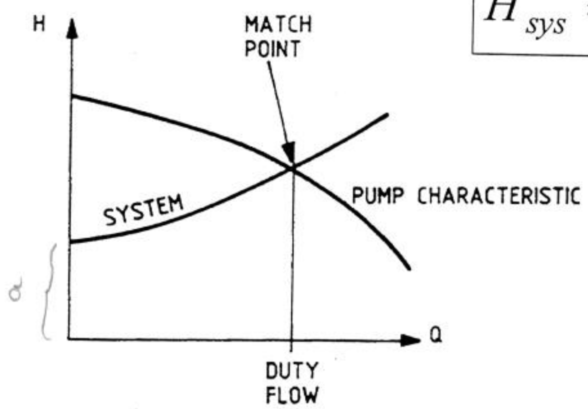
$$H_{sys} = a + bQ^2$$






# Cavitation Damage

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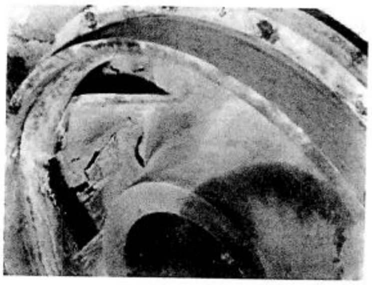

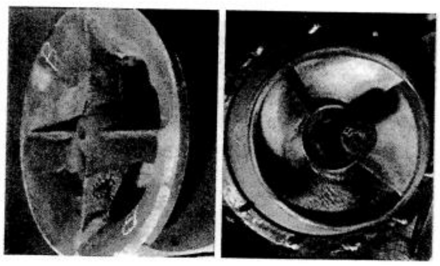

## Net positive suction head available (NPSHa)

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- NPSHa = Total pressure at inlet above the vapour pressure

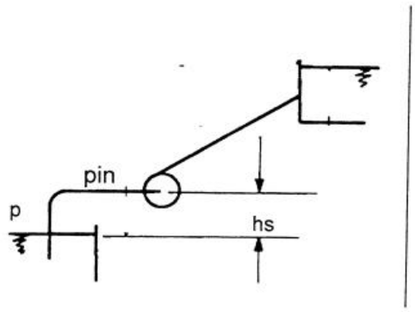
$$NPSHa = \frac{p}{\rho g} + \frac{p_{atm}}{\rho g} - h_s - h_f - \frac{p_v}{\rho g}$$

p = pressure at suction tank surface

hs = static lift to pump

hf = inlet loss

pv = vapour pressure




$$NPSHa = \frac{p_{in}}{\rho g} + \frac{p_{atm}}{\rho g} + \frac{u^2}{2g} - \frac{p_v}{\rho g}$$

$$NPSHa = a - bQ^2$$



## Cavitation performance of centrifugal pumps

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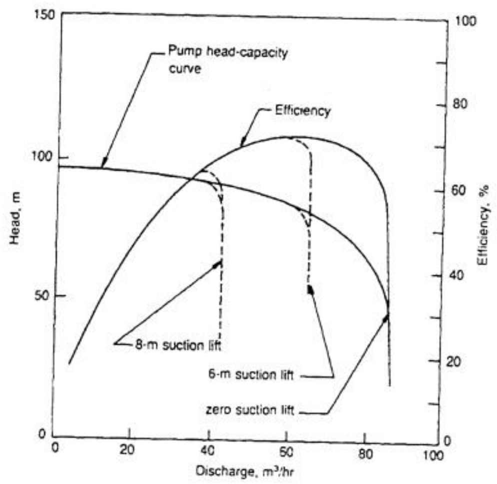
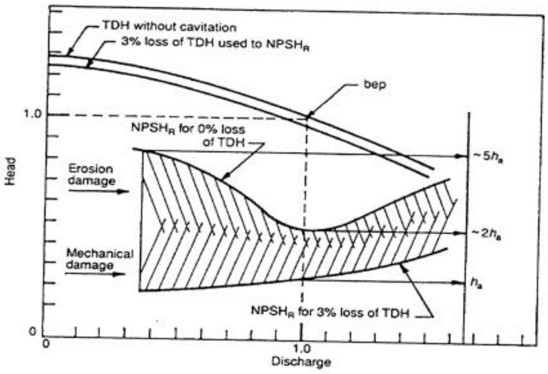


# NPSHr and NPSHa

- NPSHa > NPSHr
- 0.6m, 1.2NPSHr
- (1.5m, 1.35NPSHr)
- Consult manufacturer








# Cavitation

- Mechanism

  Pressure reduction - cavity (dissolved gas, vapour)

  Pressure rise - cavity implodes

- Effects

  Mechanical damage, noise, vibration, flow fluctuation, loss performance

- Cavitation in pumps

  Velocity increases inside impeller + blade profile effects

  Swirl and vortices increase cavitation problems

# Net positive suction head required (NPSHr)

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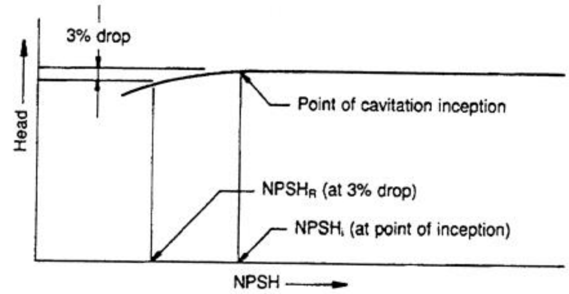


Test at constant speed and flow, by lowering suction pressure

**NPSHr defined at 3% head drop**



# Frictional loss

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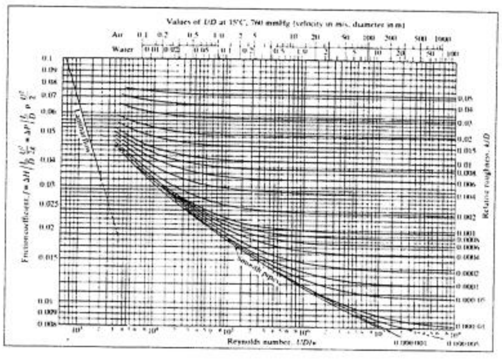
$$\Delta H_f = f \frac{l}{d} \frac{u^2}{2g}$$

f = friction factor
l = pipe length
d = pipe diameter (hydraulic dia)
u = average velocity

Re>4000 $f = \dfrac{0.25}{\left(\log_{10}\left(\dfrac{k}{3.7d} + \dfrac{5.74}{\mathrm{Re}^{0.9}}\right)\right)^2}$

Re<4000 $f = \dfrac{64}{\mathrm{Re}}$

$\left(Re = \dfrac{\rho V d}{\mu}\right)$



# Moody's Diagram

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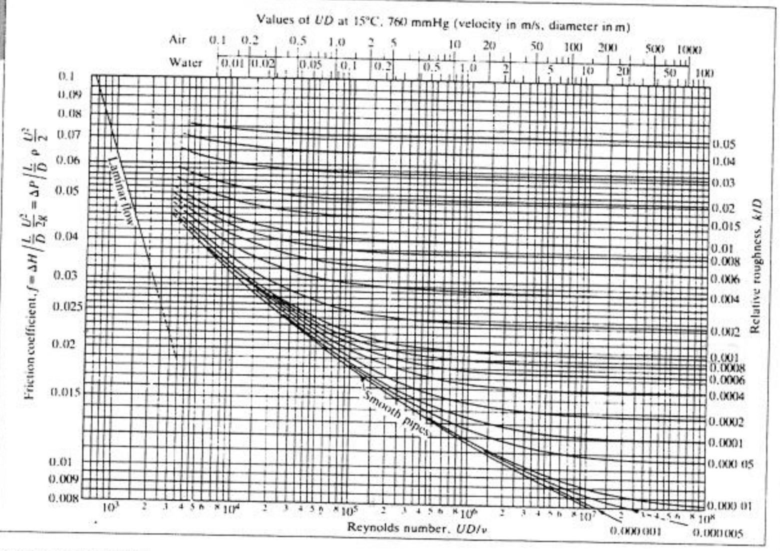
$$\cancel{hf = 4f\frac{L}{D}\frac{u^2}{2g}}$$

$$hf = f\frac{L}{D}\frac{u^2}{2g}$$






# Pipe Roughness

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| PIPE TYPE | r (mm) |
|---|---|
| New plastic and non-ferrous | 0.03 |
| Spun bitumen or cement lined ductile iron | 0.05 |
| Steel (uncoated) | 0.05 |
| Good ductile or cast iron | 0.05 |
| Galvanised steel | 0.15 |
| Precast concrete | 0.15 |
| Tuberculated water mains up to 20 years old | 1.5 - 15 |
| Tuberculated water mains up to 50 years old | 0.3 - 30 |



$POWER_{pump} = \rho \cdot g\, \Delta H \cdot Q$

$Q = \frac{\pi \phi^2}{4}$

KINETIC ENERGY => energy in the flow $= \underbrace{\frac{mv^2}{2}}_{} = 1000 \overbrace{\left(L\frac{\pi\phi^2}{4}\right)}^{volume} \cdot \frac{v^2}{2} = (J)$

MASS

$\rho$

If we "block" => 1seg => J/s = kW

## Resistance/Friction Losses

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There are several empirical formulae which relate the hydraulic gradient, i, to velocity (and hence flow):-

Blasius: $V = 2.87g^{4/7}.D^{5/7}.\nu^{-1/7}.i^{4/7}$

but only applicable to smooth pipes

Hazen-Williams: $V = 0.345C.D^{0.63}.i^{0.54}$

widely used in the past and still in the USA but only applicable in the intermediate range of flows and pipe sizes

Manning: $V = R^{2/3}.i^{1/2}/n$ $\quad R = A/P = D/4$ for pipes

only applicable to large diameter conduits



## Compressible Fluid

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Incompressible solution applies when

Mach number < 0.3

- Divide the pipe into sections
- Calculate the pressure drop in section 1 using upstream density
- Calculate density as the exit of section 1 use it as input to section 2
- Further divide the section into subsections if density change is too large
- Continue to the end

## Component loss

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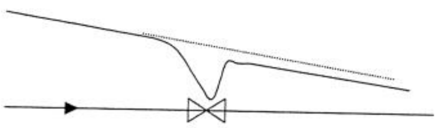
$$\Delta H_c = K\frac{u^2}{2g}$$



- Interference effect
- minimum pressure



## Valve Loss Coefficient

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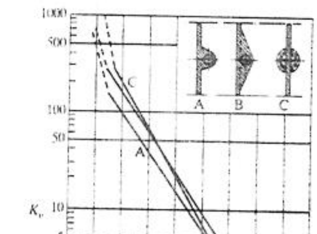


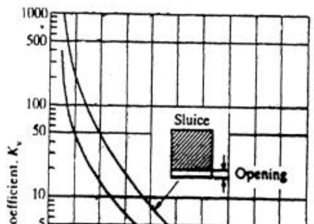

# Flow round a Bend

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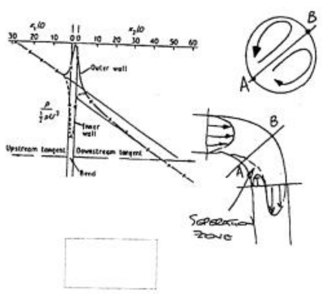


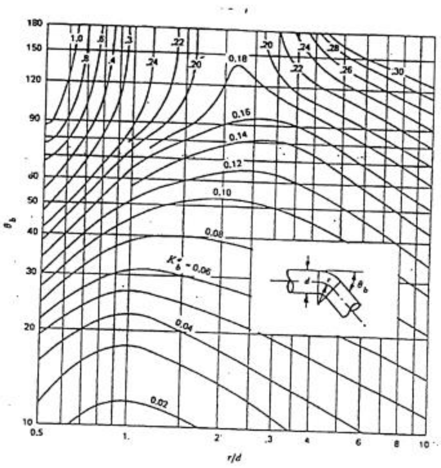




# Loss Coefficient of some common components

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| FITTING | k |
|---|---|
| Sharp edged entrance | 0.50 |
| Bellmouth entrance | 0.05 |
| Close radius 90 deg bend | 0.75 |
| Close radius 45 deg bend | 0.30 |
| Long radius 90 deg bend | 0.40 |
| Long radius 45 deg bend | 0.20 |
| Tee with flow in line | 0.35 |
| Tee with flow in branch | 0.80 |
| Reducer | 0.15 |
| Sudden contraction 3:1 | 0.45 |
| Sudden contraction 5:1 | 0.50 |
| Sudden enlargement 1:3 | 0.80 |
| Sudden enlargement 1:5 | 1.00 |
| ~~Bellmouth exit~~ | ~~0.20~~ |
| Gate valve fully open | 0.12 |
| Butterfly valve fully open | 0.45 |
| Reflux valve | 1.00 |

## Operating Point

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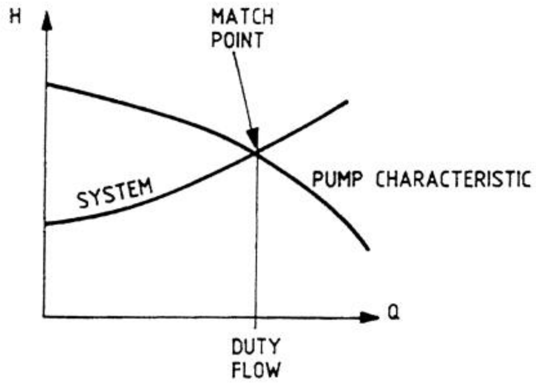




## Pump characteristic

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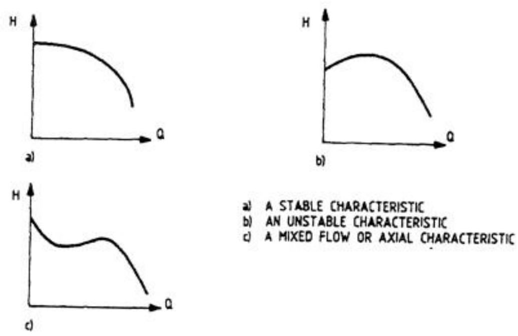
$$H_{pump} = H_{shut} + aQ + bQ^2$$



# Mixed flow pump

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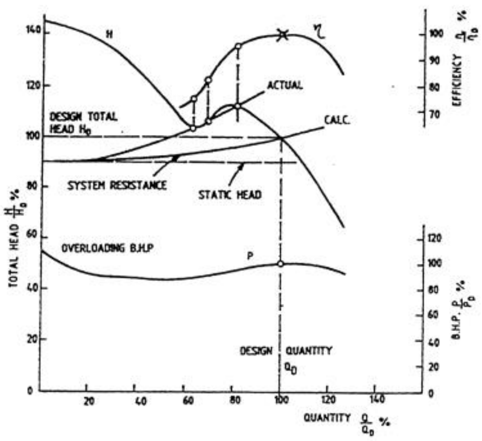




# Flow control - Throttling

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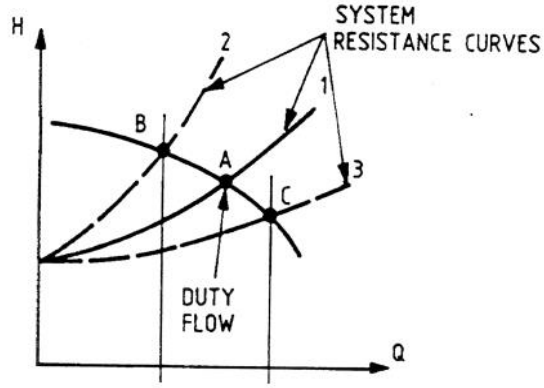

# Affinity Laws

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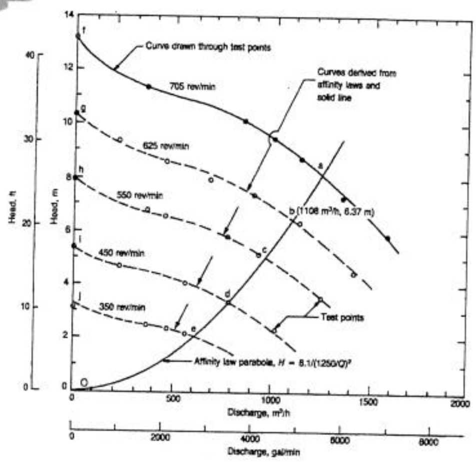


$$\frac{P}{\rho\omega^3 D^5} = \varphi\left(\frac{gH}{\omega^2 D^2}, \frac{Q}{\omega D^3}, \mathrm{Re}, \ldots\right)$$

$$\frac{Q_1}{Q_2} = \frac{N_1 D_1^3}{N_2 D_2^3}$$

$$\frac{H_1}{H_2} = \frac{N_1^2 D_1^2}{N_2^2 D_2^2}$$

$$\frac{P_1}{P_2} = \frac{N_1^3 D_1^5}{N_2^3 D_2^5}$$

For same pump, trimmed impeller

$$\frac{Q_1}{Q_2} = \frac{N_1 D_1}{N_2 D_2}$$

$$\frac{H_1}{H_2} = \frac{N_1^2 D_1^2}{N_2^2 D_2^2}$$

$$\frac{P_1}{P_2} = \frac{N_1^3 D_1^3}{N_2^3 D_2^3}$$



# Flow control - Variable speed

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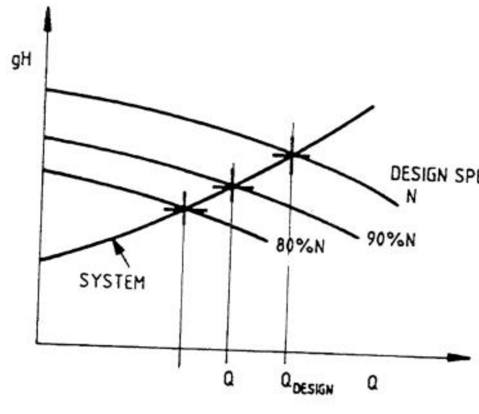
$$Q = Q_{norm}\left(\frac{N}{N}\right)$$

$$H = H_{norm}\left(\frac{N}{N_{norm}}\right)^2$$



# Comparison between control methods

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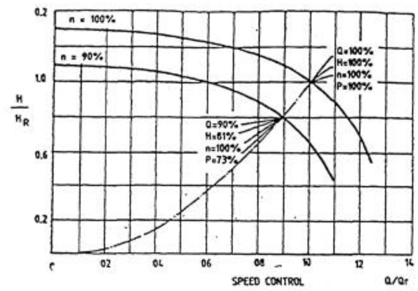


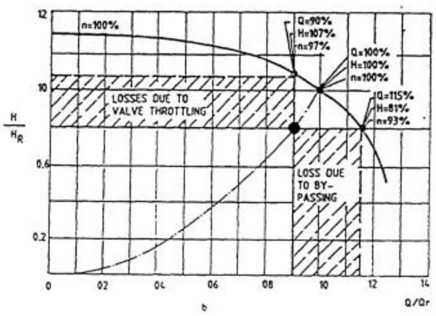


Is energy the only consideration?

The Pumping System P&PS, ADUUEA 05/09

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# Pumps in series

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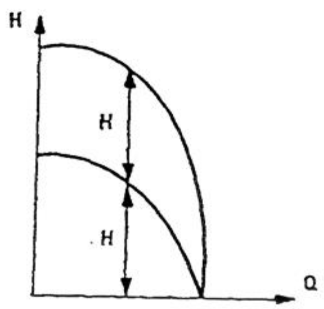
- Combined characteristics -Head doubled at the same flow
- Inlet pressure for the second pump is much higher

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## Pumps in parallel

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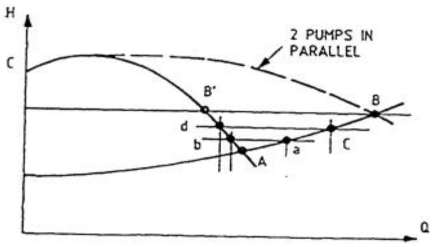
- Combined characteristic- Flow doubled for the same head
- Actual flow delivered not doubled
- Watch out for unstable characteristics






## Branched System

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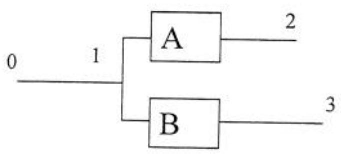


Pressure at 1 is the same whether it is calculated from 2 or 3

$$P_1 = P_2 - \Delta P_{12} = P_3 - \Delta P_{13}$$

$$\Delta P_{12} = K_A Q_{12}^2 \qquad \Delta P_{13} = K_B Q_{13}^2$$

Flow into junction (node) = Total flow out of junction (node)

$$Q_{01} = Q_{12} + Q_{13}$$

## A Branched System

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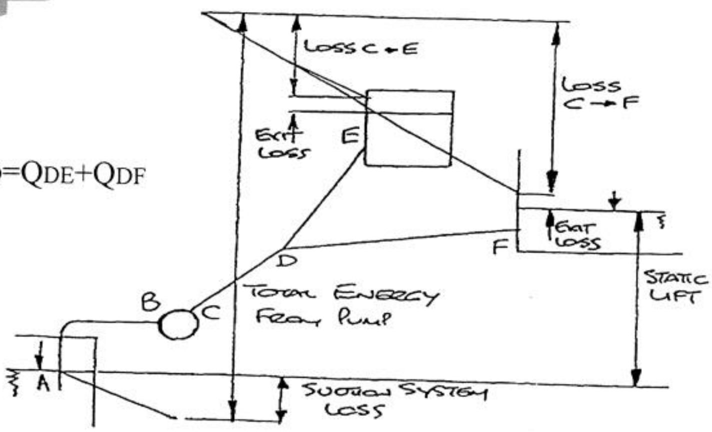
$Q_{BD}=Q_{DE}+Q_{DF}$






## A Methodology to Calculate System Characteristics for Branched System

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For a given flow rate

- Calculate the pressure at pump inlet
- Calculate the pressure at pump outlet
  - Assuming a flow split between each branch
  - Calculate pressure at the junction along each branch
  - Modify the flow split, Iterate till converge

Repeat for a different flow rate

# PUMP BASICS

## 1. BASIC PUMP TERMINOLOGY

| TERM | DESCRIPTION |
|---|---|
| Vapour pressure | The pressure at which a liquid turns to vapour at a given temperature. |
| Cavitation | The effect of liquid turning to vapour due to the surrounding pressure reducing to the liquid vapour pressure at some point within a pump (usually the inlet) |
| Impeller | The bladed, rotating element of a rotodynamic pump |
| Rotor | The rotating pumping element of a positive displacement pump |
| Inlet | The connection taking liquid into the pump (was known as suction) |
| Outlet | The connection taking liquid out of the pump (was known as discharge) |
| Power | Usually refers to the power consumption of the pump, but may refer to the rated power of the driver. Can be related to either mechanical or electrical power. |
| Efficiency | The ratio (usually expressed in percent) of water power produced divided by the input power of the pump |
| Water power | The amount of useful power imparted into the liquid. Equal to $\rho gHQ$. |
| Wire to Water Efficiency | The efficiency expressed as Water Power output divided by Electrical power consumed. |

## 2. PUMP TYPES

While the water industry is a relatively specialised activity, the number of pump types and applications which may be encountered are large and in order to understand the various pump types it is necessary to classify pumps in some way. There are many ways of doing this, but one widely accepted system is:

- Classified by Operating Principle
- Classified by Construction
- Classified by Application Features

Under each of the above headings, pumps could be broken down into the following categories, for example:

| | |
|---|---|
| **OPERATING PRINCIPLE** | |
| Rotodynamic Pumps | Single or multi-stage forms |
| Axial Flow | Open impeller with fixed / variable pitch |
| Mixed Flow<br>Radial Flow | Single or double suction, with or without self priming, with open or semi-open or closed impeller. In diffuser or, volute or double volute casing. |
| Peripheral | Also known as “side channel”, “vortex” or “turbine” with or without self-priming. |
| Positive Displacement Pumps | Reciprocating in simplex or multiplex forms. Piston pumps – single / double acting diaphragm pumps. |
| Rotary, single rotor | Vane, piston, progressive cavity, screw, peristaltic |
| Multi Rotor | Lobe, gear, screw, circumferential piston |
| **CONSTRUCTION FEATURES** | |
| Shaft orientation | Vertical, horizontal, inclined |
| Shaft support | Overhung, between bearings, cantilever |
| Connection branch | End suction, in-line, top connections, overshot |
| Drive linkage | Long coupled, close coupled, universal |
| Bearing type | Bush, sleeve, ball, roller |
| Lubrication system | Product, grease, oil, splash, mist, flooded |
| Enclosure | Submersible, submergible (immersible), dry pit |
| Seal type | Packed seal gland, mechanical, single, double |
| Seal-less type | Permanent magnet, eddy, submerged rotor, canned |
| Mounting | Bare shaft, portable, skid, trolley, static, baseplate, plinth. |
| Drive type | Engine, motor, electric, hydraulic, pneumatic |
| Material | Iron, steel, non-metallic, coated, lined, rubber |
| Construction | Cast, welded, pressure, moulded |
| Specification | API, ANSI, ISO, EN, Contract |

| APPLICATION | |
|---|---|
| Booster duty | Single, multi-stage, rotodynamic<br>Peripheral<br>Rotary P.D. |
| Chemical Process | All types |
| Irrigation | Single / multi-stage, end suction, vertical |
| Sewage | Dry well, submersible, non-clog, rotodynamic, grinder, macerator |
| Shallow water supply | End suction, rotodynamic or peripheral, axial / mixed flow (for high duties) sliding vane, submersible, portable (engine driven) |
| Sludge | Single stage, vertical, end suction, diaphragm, submersible |
| Surface Water | Horizontal single / multi stage rotodynamic, peripheral, reciprocating, submersible. |
| Water supply (Potable) | Single stage rotodynamic, axial, mixed flow, radial. |
| Water treatment | Rotary PD, proportioning, metering gear, lobe. |

# PRINCIPLES OF ROTODYNAMIC PUMPS

Pumps and compressors of the rotodynamic type provide energy to the fluid being passed continuously. Figure 1 illustrates some simple examples of pumps and compressors that are either axial or centrifugal in action. The principles that govern the transfer of energy in rotodynamic machines will be examined. The effects of density, viscosity and other fluid properties will be examined,

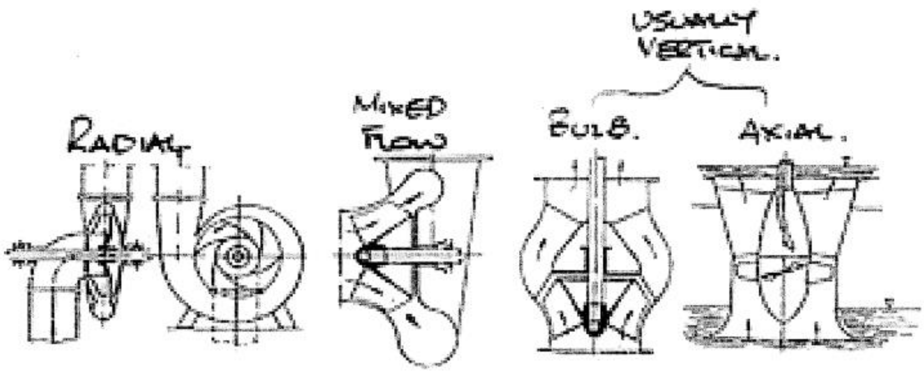


Figure 1 Centrifugal, mixed and axial machines

# 1. THE EULER EQUATION

This relationship is based on the concept that the energy change produced by a rotating element like an impeller is related to the change of angular momentum experienced by the fluid it is affecting. It is the main equation to be used in relating the energy change to the velocity diagrams describing the inlet and outlet flow distribution. The simplest machine to which Euler can be applied is the single stage centrifugal pump sketched in figure 2.

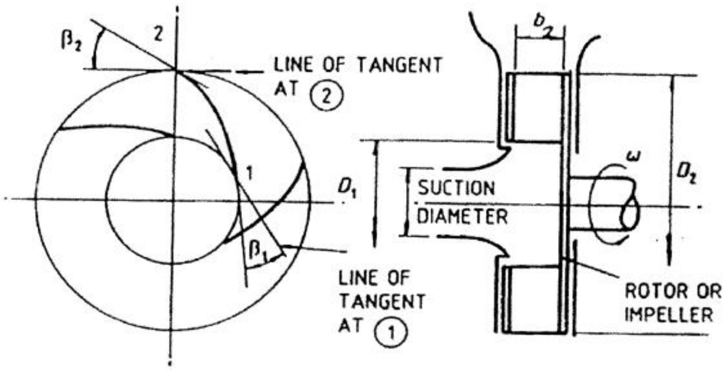


Figure 2 A simple centrifugal pump

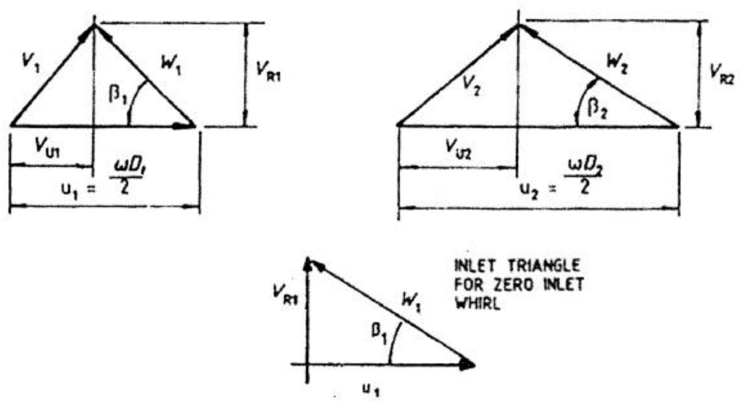


Figure 3 Velocity triangle for a centrifugal machine

Using the parameters shown in figure 3, the Euler equation states that the total energy change experienced by the fluid is given by the relation

$$gH_E = U_2 V_{U2} - U_1 V_{U1} \tag{1}$$

Equation 1 can be written for the special case that $V_{U1} = 0$, the zero whirl or design condition to read

$$gH_E = U_2 V_{U2} = U_2^2 - \frac{QU_2}{A_2}\cot\beta_2$$

where $A_2 = \pi D_2 b_2$

Thus for fixed geometry and speed

$$gH_E = K_1 - K_2 Q \tag{2}$$

Figure 4 illustrates how choice of outlet angle affects energy change and figure 5 shows the effect of this angle on the shape of the outlet triangle.

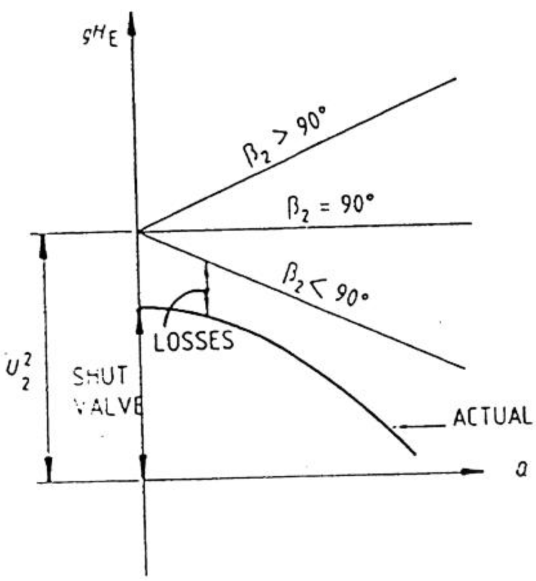


Figure 4 Effect of outlet angle on head flow characteristic

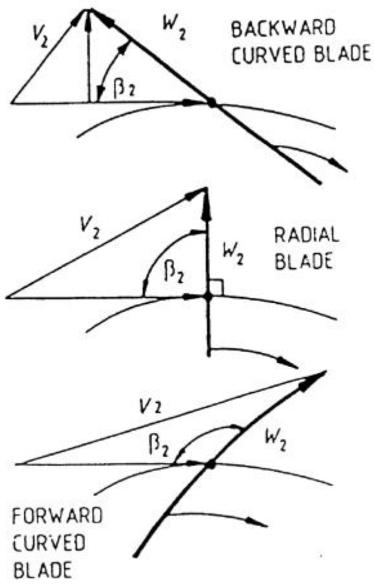


Figure 5 Effects of outlet angle on velocity triangle

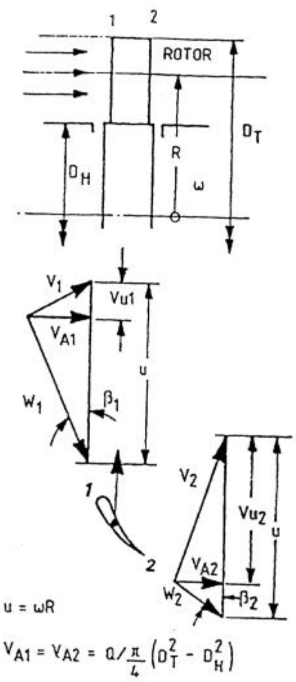


Figure 6 Velocity diagram for axial machine

Figure 6 illustrates an axial flow stage consisting of inlet guide vanes, rotor and outlet guide vanes. The Euler equation for this machine is written for one cylindrical flow surface at radius R. Since the peripheral speed U is the same for both inlet and outlet the simple equation becomes

$$gH_E = U(V_{U2} - V_{U1}) = U\Delta V_U$$

The above equation reduces for zero inlet whirl (the 'design' case for a stage without inlet guide vanes) to read

$$gH_E = UV_{U2} \qquad (3)$$

## 2. THE EFFECT OF FLUID PROPERTIES

A change in viscosity will affect the fluid pressure rise that can be produced by a machine with a corresponding change in the power demand from the driver. Figure 7 illustrates this for a pump

The effect of gas content on the performance of a pump is shown in figure 8. At low flow rates the pressure rise is negligible and at higher flows the rise falls away as the content percentage increases. This is due to a number of factors one of which is the inability of the blades in the impeller to expel the gas that accumulates in the passages. This effect will be discussed later in the course.

The effect of density change is to change the power input required by a pump, but in a compressor, the effect is related to the gas laws as it is related to pressure and temperature changes through the machine.

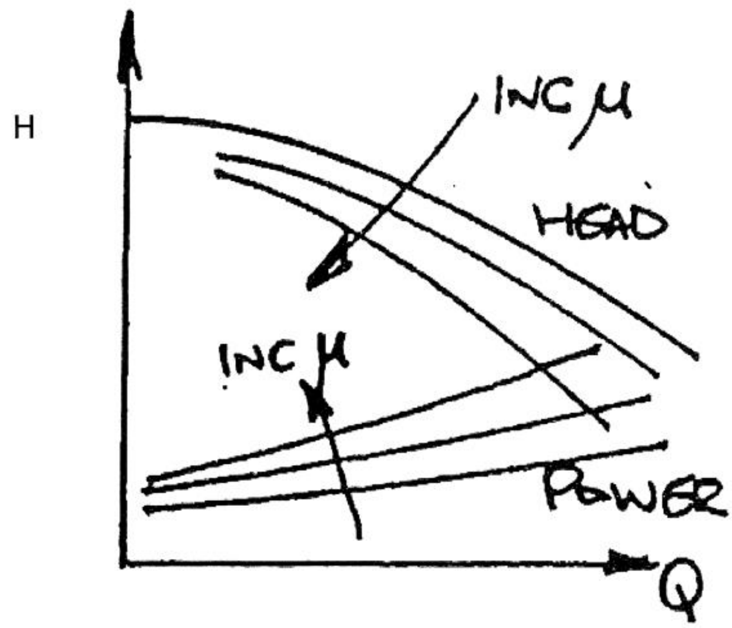


Figure 7 Effect of viscosity

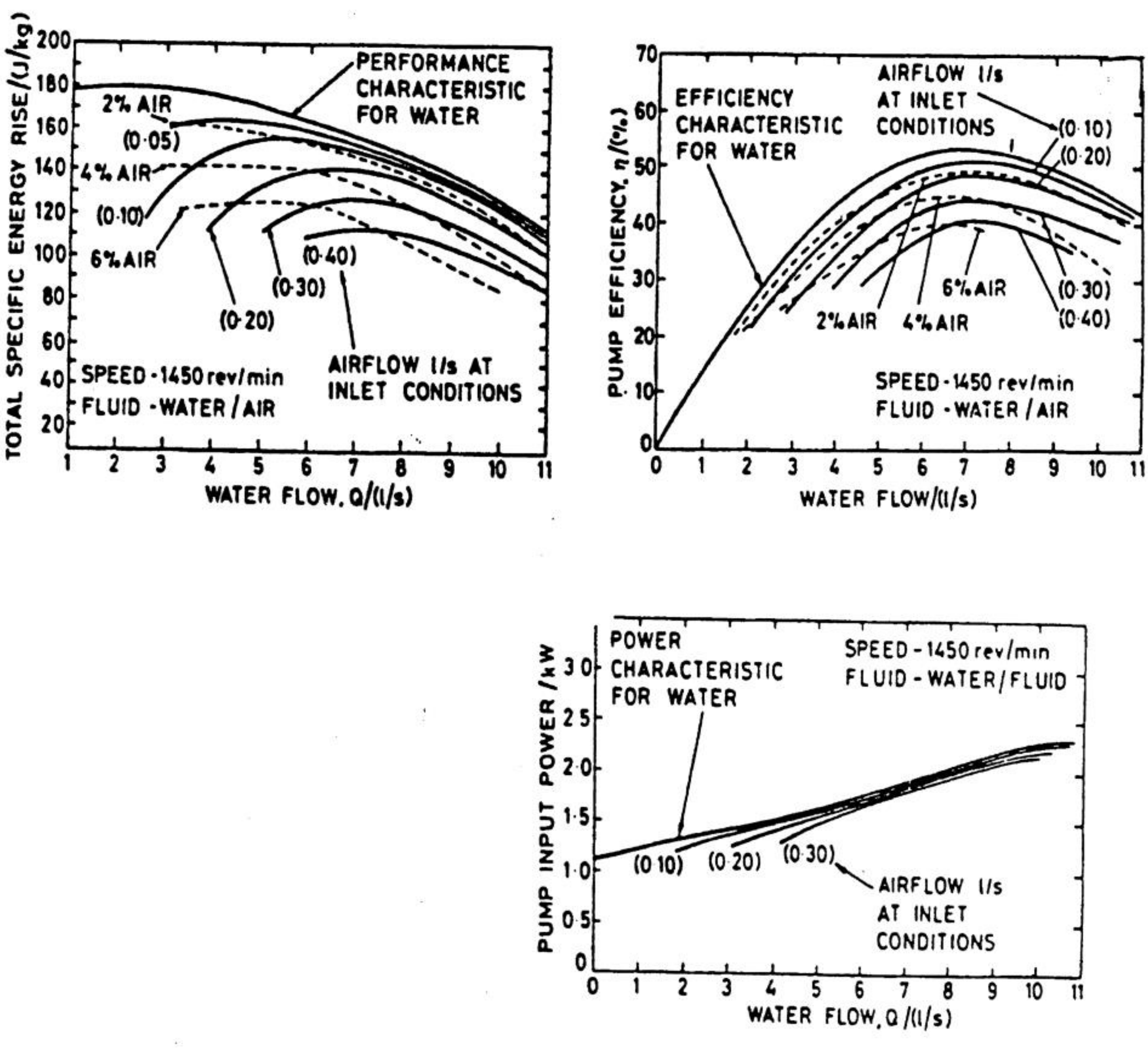


Figure 8 Effects of gas content

## 3.DIMENSIONLESS AND CONVENTIONAL RELATIONSHIP USED TO PRESENT PERFORMANCE DATA

### 3.1 Pumps

If a black box is used to represent a pump, a dimensional analysis will yield the relationship

$$\frac{P}{\rho\omega^3 D^5} = \Phi\left(\frac{gH}{\omega^2 D^2}, \frac{Q}{\omega D^3}, \mathrm{Re}\right) \quad (4)$$

wh

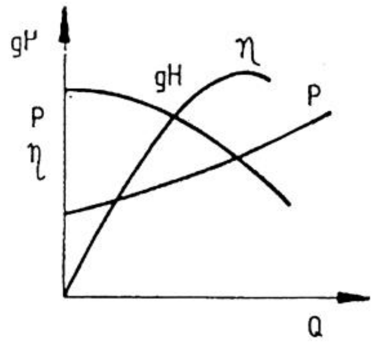


Figure 9 Typical Centrifugal pump characteristics

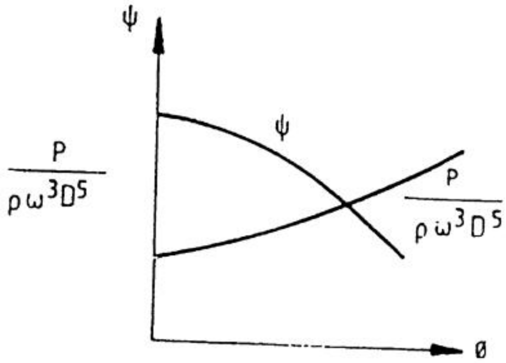


Figure 10 Non-dimensional characteristics

Figure 9 illustrates a typical centrifugal characteristic for a single speed, and figure 10 is a presentation of a family of pump characteristics using the nondimensional groups in equation 13. Figure 11 demonstrates how dynamically similar performance can be predicted using the dimensionless groups

$$\frac{gH}{\omega^2 D^2} = \text{Constant}$$

$$\frac{Q}{\omega D^3} = \text{Constant}$$

$$\frac{P}{\rho\omega^3 D^5} = \text{Constant}$$

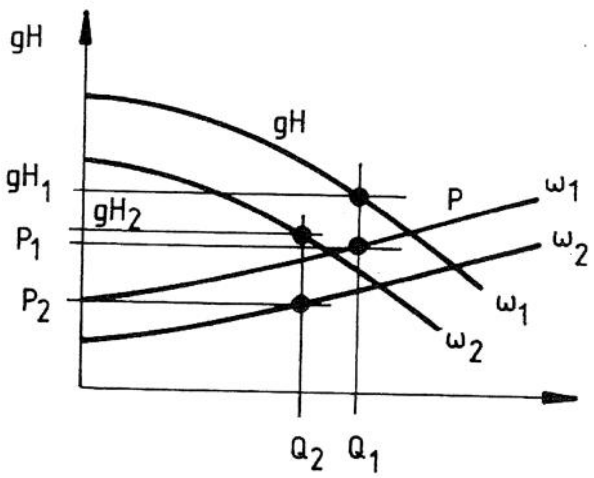


Figure 11 Performance prediction using dimensionless group

### 3.1.1 The Application of the Scaling Laws to 'Turning Down'

A common method of modifying the performance of a centrifugal pump is to reduce the outside diameter of the impeller a process known as 'turning down', and a 'cut' in American handbooks. The laws of similarity cannot be strictly applied since the flow geometry must change as the diameter of the impeller is reduced. Referring to the sketches figure 12 below it can be argued that for a small reduction in diameter from D1 to D2 (up to 10%) the outlet angle does not change significantly. If the speed remains constant the similar triangles lead to the conclusion that the relative velocity vectors are parallel as the peripheral speed reduces with D. This leads to the equations

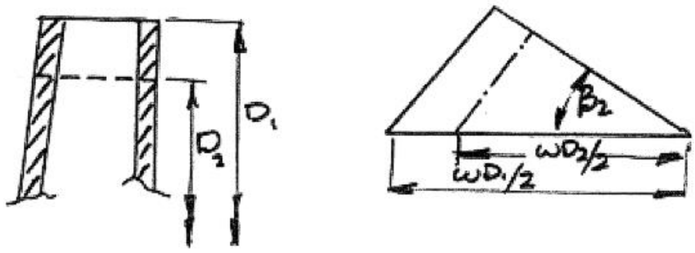


Figure 12 Turning down and velocity triangle

$$\frac{Q_1}{Q_2}=\frac{N_1D_1}{N_2D_2};\quad \frac{H_1}{H_2}=\left(\frac{N_1D_1}{N_2D_2}\right)^2;\quad \frac{P_1}{P_2}=\left(\frac{N_1D_1}{N_2D_2}\right)^3 \qquad (5)$$

For $N_1 = N_2$, equation 5 reduces to

$$\frac{D_1}{D_2}=\left(\frac{H_1}{H_2}\right)^{0.5},\quad \frac{D_1}{D_2}=\frac{Q_1}{Q_2},\ \frac{P_1}{P_2}=\left(\frac{H_1}{H_2}\right)^{1.5}$$

When cutting down the impeller, with N being constant these relations can be written as

$$H = K_hQ^2 \quad and \quad P = K_pQ^3 \qquad (6)$$

the constants $K_h$ and $K_p$ can be found from the operating conditions at the maximum diameter.

`As an example of the approach, Figure 13 will be used. The head is to reduce from 224.4 ft to 192.9 ft. Using the relations above, the predicted diameter is 15.125in and the flow rate is 3709 gpm with the power 215.5 hp.

Comments: In Figure 13, a , b, and c Point A are the actual test points and point B the predicted points assuming the efficiency does not change. The best efficiency flow rate is lower at points a and b, as figure 13 c shows, but the head and flow are very close.

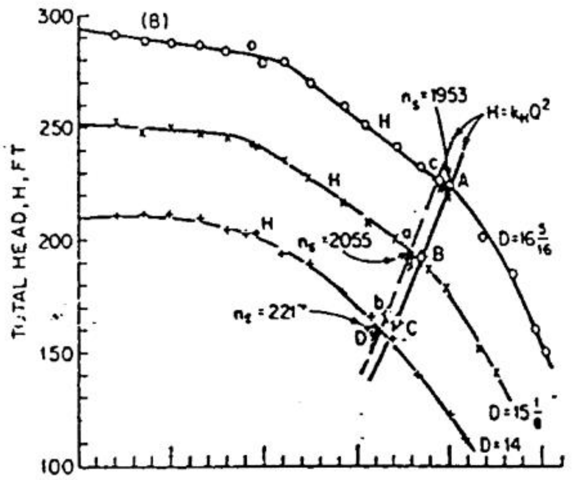


Figure 13a

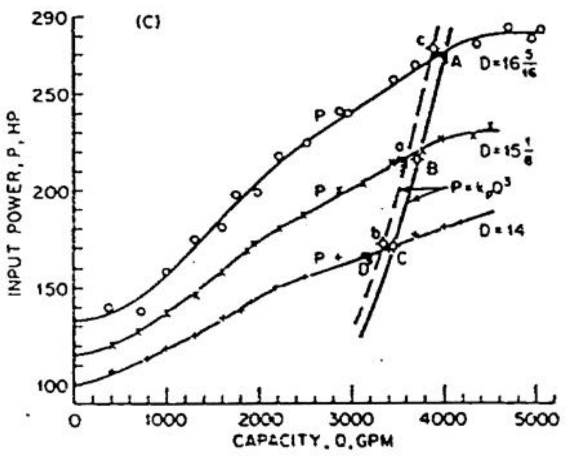


Figure 13b

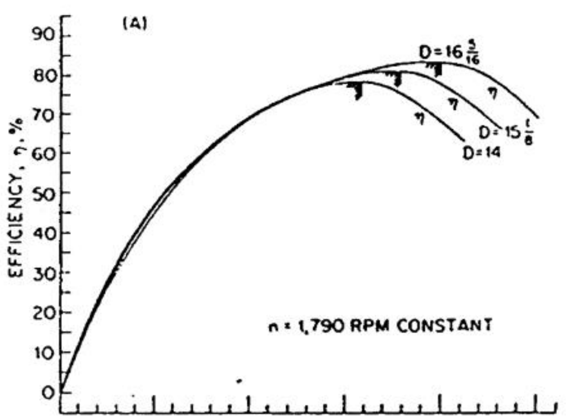


Figure 13c

As Karassik in the handbook points out, dynamic similarity is not satisfied and suggests that cutting takes place in small steps with a test run each time. Figure 14 taken from Stepannof (6) suggests a correction based on experience over many years that gives a more accurate prediction of diameter, and Figure 15 is an alternative correction based on specific speed by Rutschi (7).

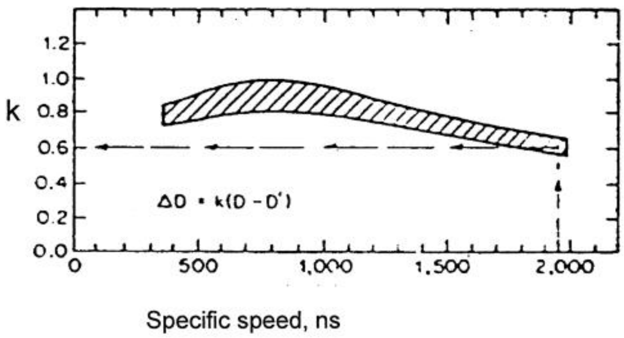


Figure 14 Correction due to Stapannoff

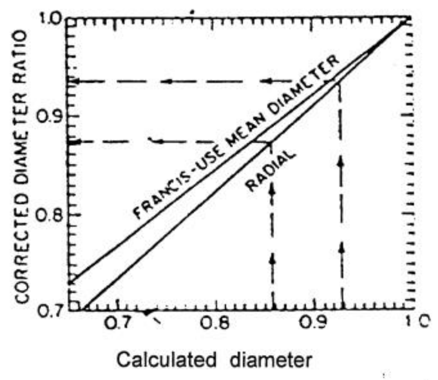


Figure 15 Correction due to Rutschi

The impeller can be turned off flush as in figure 16a or the blades turned down and the back plate and shroud left alone as in figure 16b (It is argued that the latter gives better flow control at the outlet from the impeller)

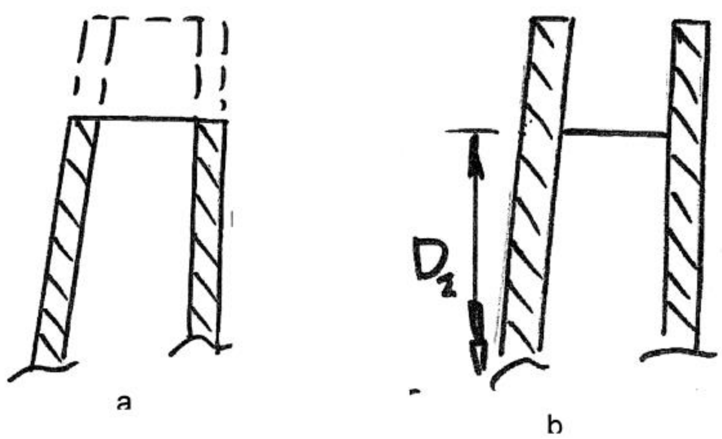


Figure 16 Alternative 'turn down'

### 3.1.2 Specific Speeds

An important group of parameters much used in the pump industry is the specific speed. It has been used a s an index related to the best efficiency flow and energy rise for many years in the form

$$N_S = \frac{N\sqrt{Q}}{H^{0.75}} \tag{7}$$

or in SI system, the characteristic number

$$k_S = \frac{\omega\sqrt{Q}}{(gH)^{0.75}} \tag{8}$$

Figure 17 is a well known chart used for classification.

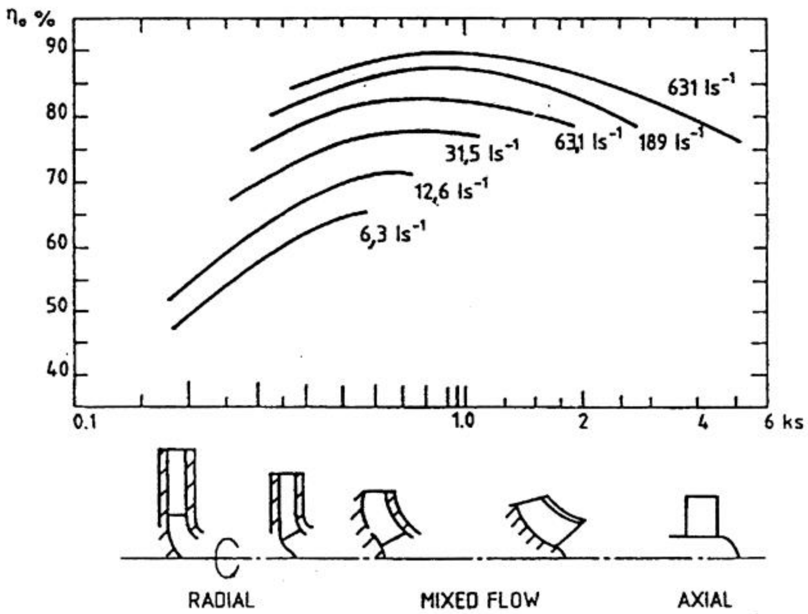


Figure 17 Variation of efficiency with flow and ks

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6. Stepannof A J — Centrifugal and Axial Flow Pumps 1957 John Wiley New York.

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# Pump Basics I

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**Mr Tony Salisbury**

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## Single Stage End Suction Pump

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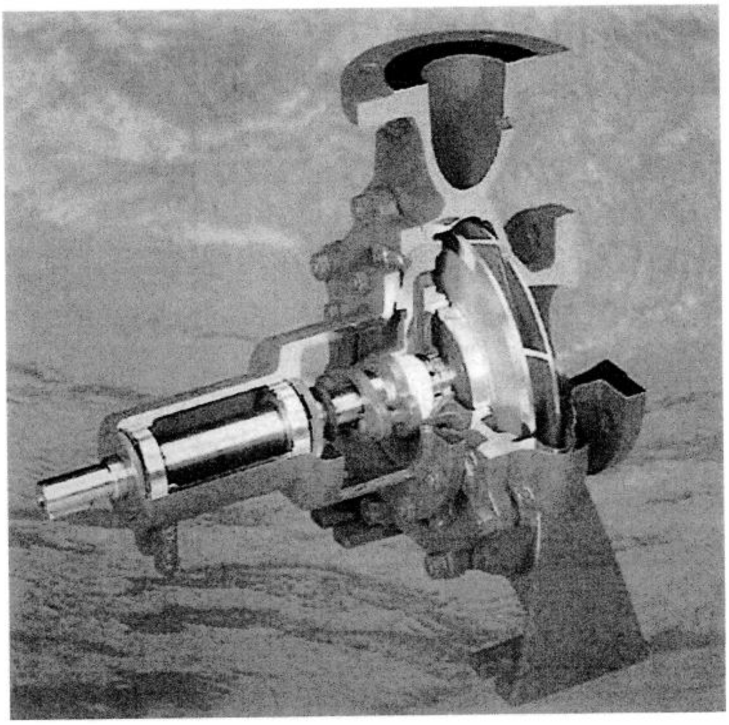

## Multistage Pumps

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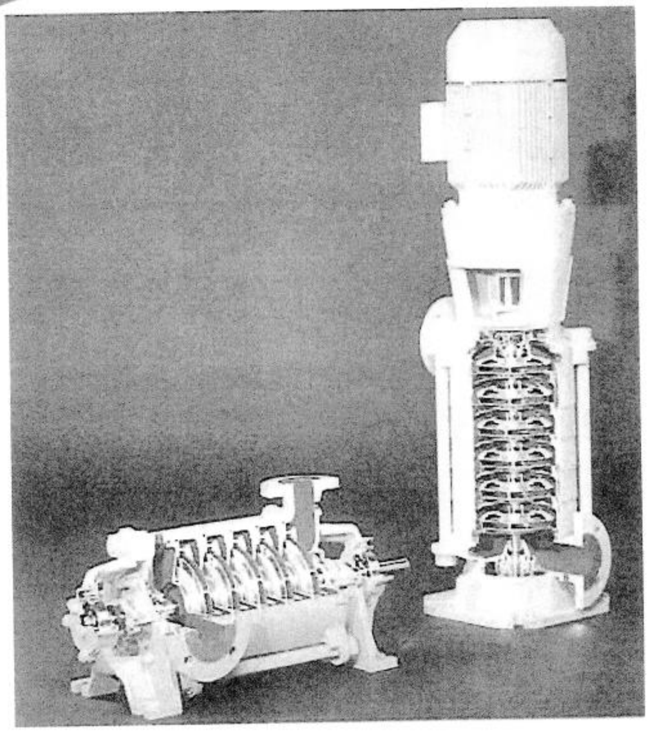



## Mixed Flow Pump

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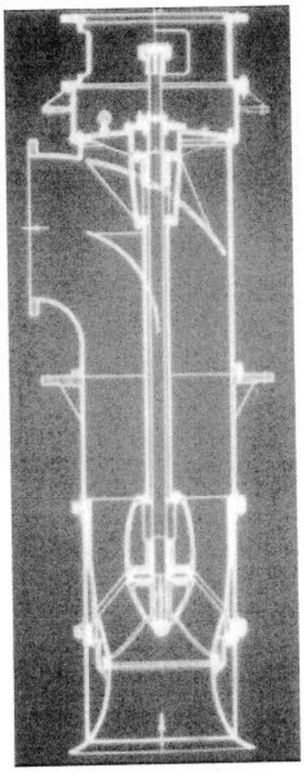

# Multistage Pump Installation

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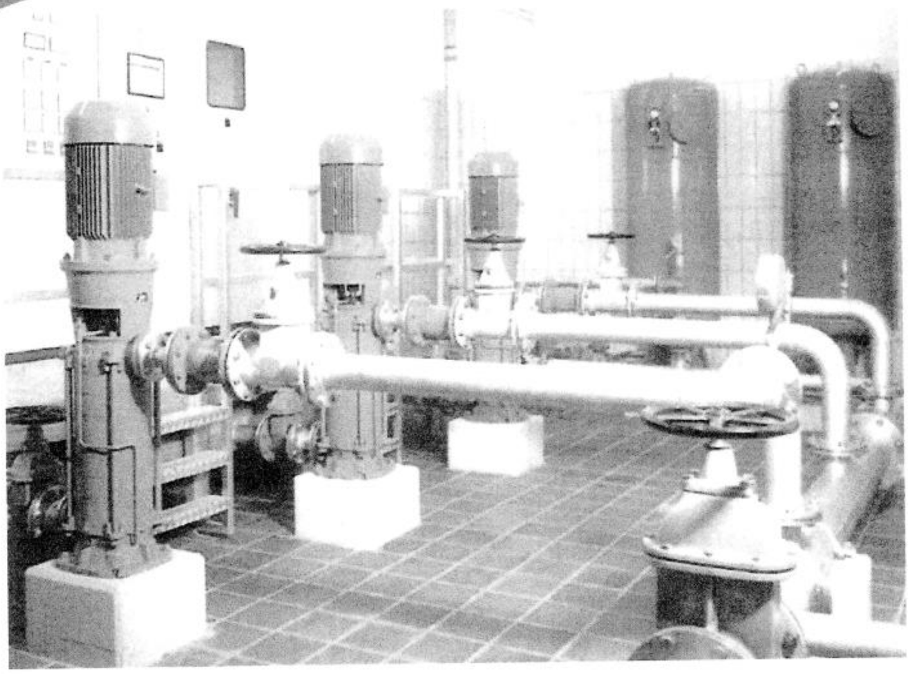



# Matching the Pump to the System

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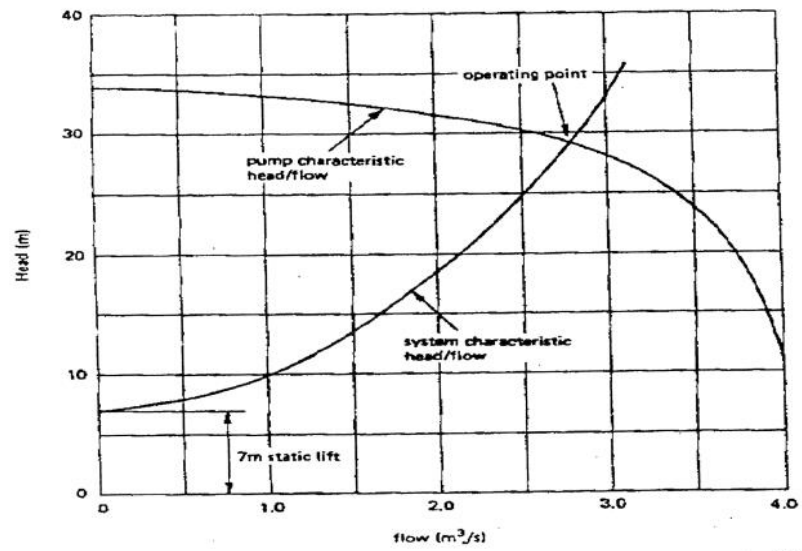


Pump and system characteristics

Ref 2

# The Euler Equation

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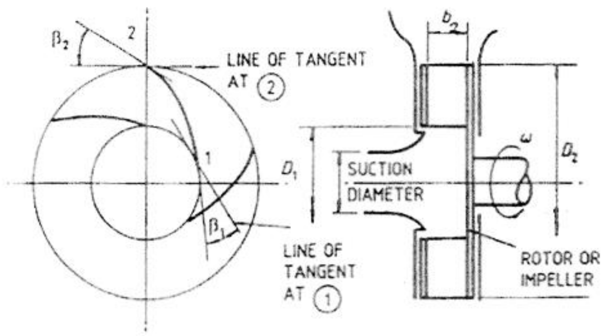
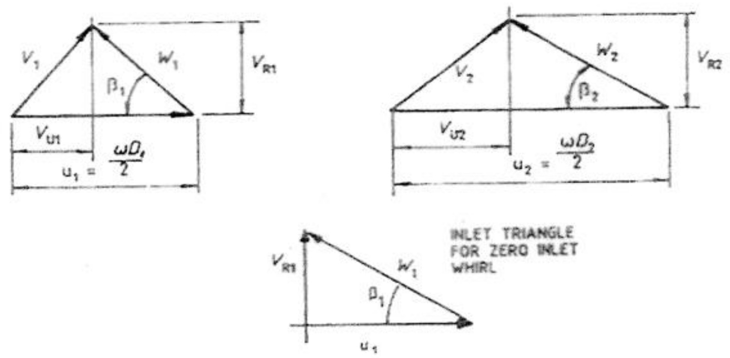



$$gH_E = U_2 V_{U2} - U_1 V_{U1}$$

↑ rot. velocity ⇒ ↓ impeller






# The Simple Centrifugal Pump

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For zero inlet whirl condition $V_{U1} = 0$

The Euler equation simplifies to:

$$g\,H_E = U_2 V_{U2}$$

$$H = U_2\,V_{U2}/g$$

For a given impeller $V_{U2}$ is related to $U_2$

$$H = f(\,U_2^2/g)$$

# The effect of varying $\beta_2$ on pump performance

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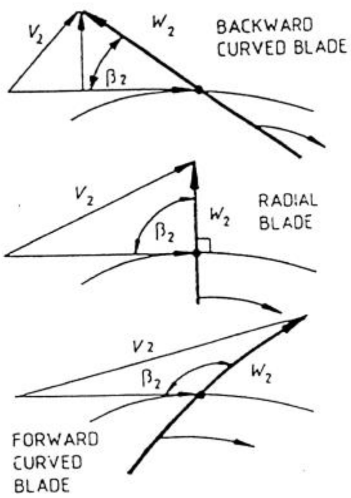


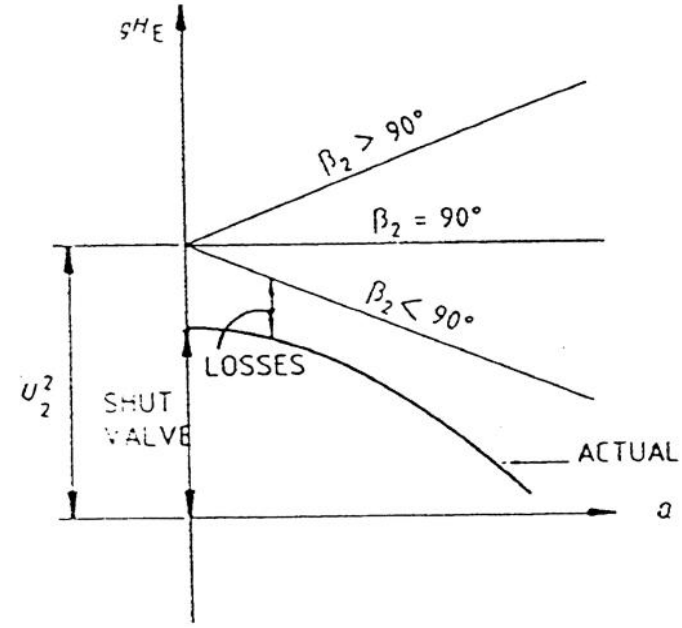




# For an Axial Flow Pump

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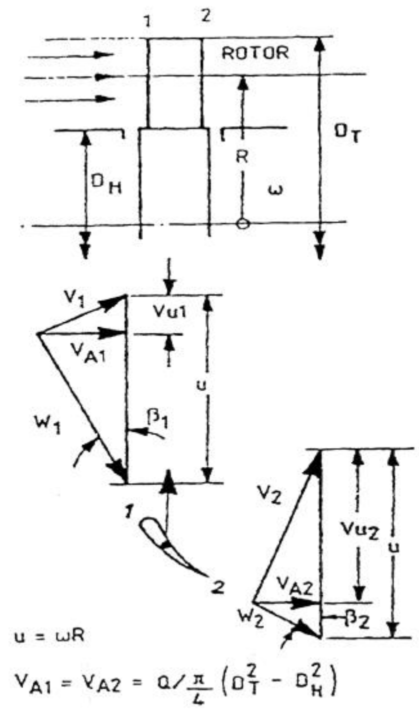
$$gH_E = UV_{U2}$$



# The effect of density on performance

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- The effect of **density change** is to increase the **power** as the density increases
- For a compressor, the effect is determined by gas laws



# The effect of viscosity on pump performance

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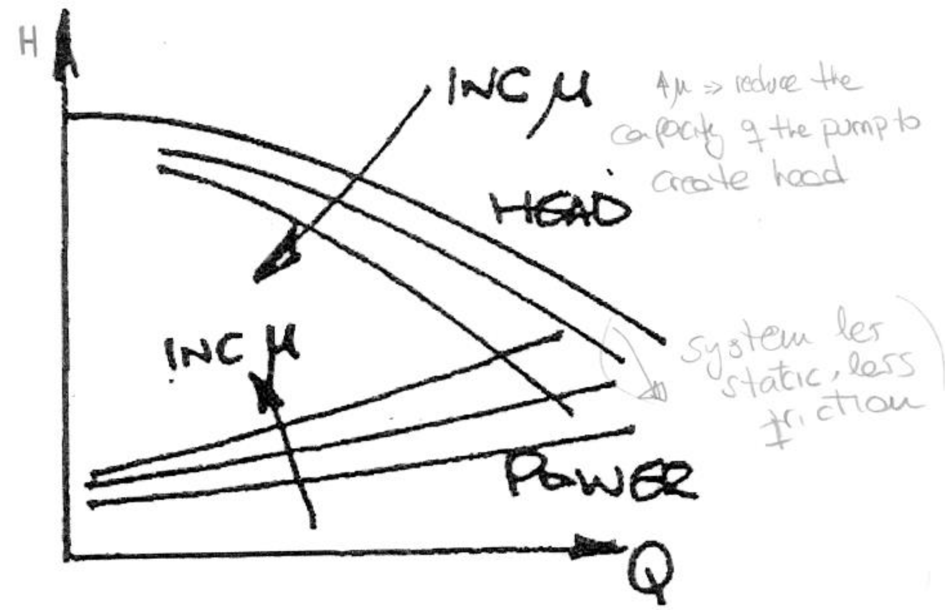

# The effect of gas content on pump performance

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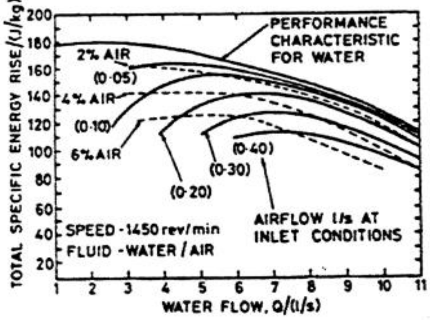
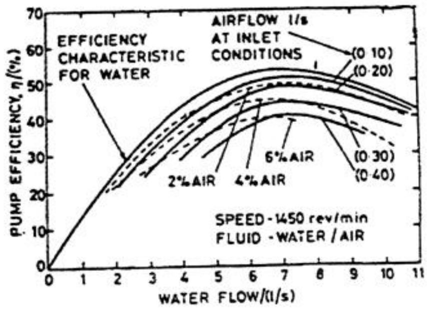
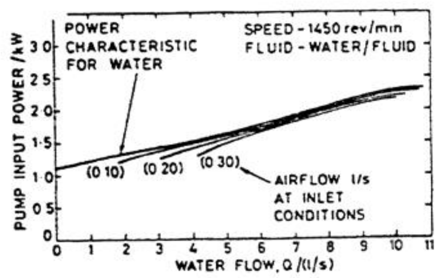
air in the water ⇒ 1-2%






# Scaling (Affinity) Laws

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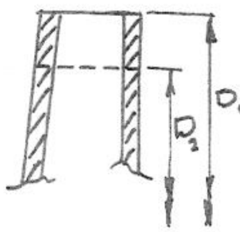
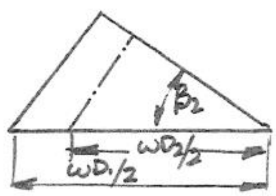
$$\frac{Q_1}{Q_2} = \frac{N_1 D_1}{N_2 D_2}; \quad \frac{H_1}{H_2} = \left(\frac{N_1 D_1}{N_2 D_2}\right)^2; \quad \frac{P_1}{P_2} = \left(\frac{N_1 D_1}{N_2 D_2}\right)^3$$

For constant N

$$\frac{D_1}{D_2} = \left(\frac{H_1}{H_2}\right)^{0.5}, \quad \frac{D_1}{D_2} = \frac{Q_1}{Q_2}, \frac{P_1}{P_2} = \left(\frac{H_1}{H_2}\right)^{1.5}$$

Affinity laws only works in a small range of diameter changes



# Prediction of the effect of change in D or ω for a pump

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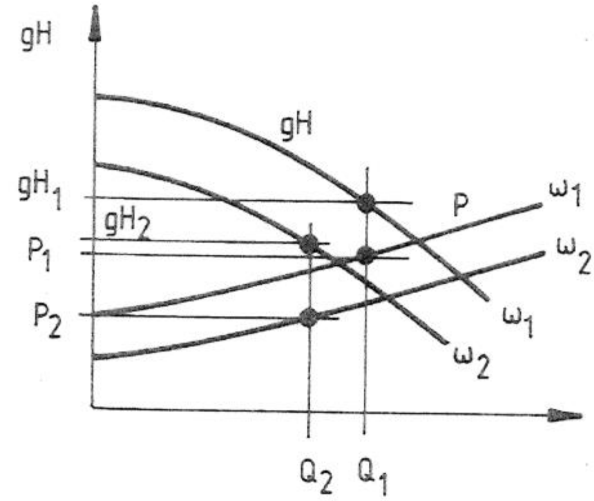
$$\frac{gH}{\omega^2 D^2} = \text{Constant}$$

$$\frac{Q}{\omega D^3} = \text{Constant}$$

$$\frac{P}{\rho\omega^3 D^5} = \text{Constant}$$






# Specific speed Ns or Type number, ks

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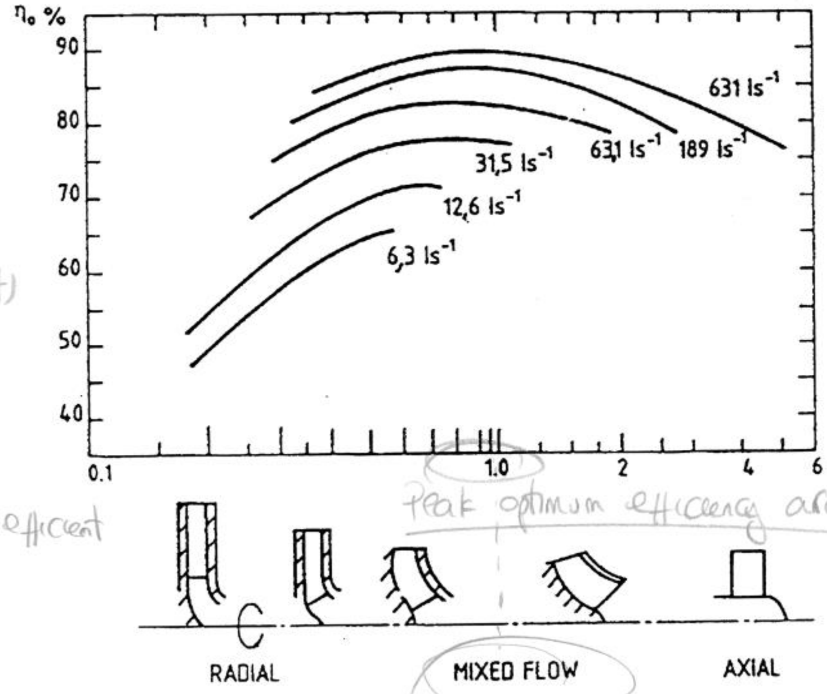
$$N_S = \frac{N\sqrt{Q}}{H^{0.75}}$$

Q, H decide from the system (project)

$$k_S = \frac{\omega\sqrt{Q}^{\,(m^3/s)}}{(gH)^{0.75}}$$

Generally, big pumps are more efficient than small pumps

↑efficiency ⇒ $k_S \approx 1$




Peak optimum efficiency around 1

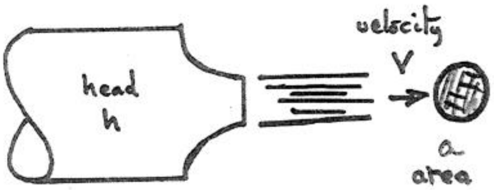
types of impeller.

Principles of Rotodynamic Pumps P&PS – ADWEA 05/09 www.cranfield.ac.uk

ks also known as shape number

NOZZLE OR MONITOR

(BERNOULLI)




$$Q = Cd(a)\sqrt{2gh}$$

$$(p = \rho g h)$$

Hence $Q = Cd\ a\sqrt{\frac{2p}{\rho}}$

# Terminology

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# PUMPING STATIONS - KEY ISSUES

Process Systems Engineering

- Where is the liquid coming from?
  What is the liquid to be pumped?
- Where is it going to?
- Site layout, ground conditions
- Where is the liquid relative to ground level?
- Pumping cost
- Relative importance of maintenance Vs. capital cost.
- What is the total lift of the liquid?
- How will the pumps be controlled? What does the process demand?

# PRESSURE

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- Pressure is usually expressed in "Gauge Pressure", which is the pressure above or below atmospheric pressure.
- Sometimes, pressure is expressed as "Absolute Pressure" which is relative to a perfect vacuum.
- Absolute Pressure = Gauge Pressure + Atmospheric Pressure



# VACUUM

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- The absence of pressure.
- A perfect vacuum is very difficult to achieve and the term is usually applied to a partial vacuum, being less than atmospheric pressure.
- Care is needed in understanding the description of vacuum and possible confusion with absolute pressure.

# DIFFERENTIAL PRESSURE & TOTAL HEAD

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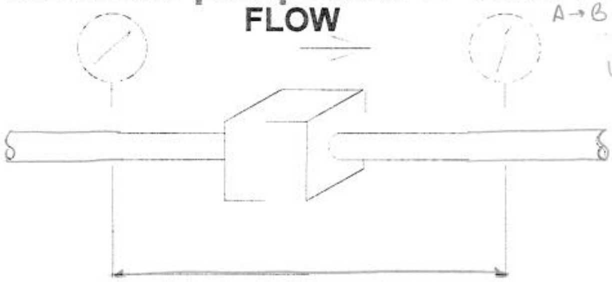
Process Systems Engineering

- The difference between the **Total Pressures** at the pump inlet and outlet at **Rated Flowrate**. It is a measure of the energy gained by the liquid when passing through the pump. For rotodynamic pumps, the **Differential Head** may be referred to. This is also called the **pump head or Total Head.**

**FLOW**

A→B changes in pressure and velocity head.

$V_s^2/2g$



$V_d^2/2g$

**DIFFERENCE IN PRESSURE OR HEAD**



# RATED PRESSURE

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- The outlet pressure at the guaranteed operating conditions of the pump

# HEAD

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- The height of a column of pumped liquid that would be supported by the pressure at a given point.
- Basically a measure of pressure, it is used mainly in hydraulics & the rotodynamic pump industries.
- The head generated by a rotodynamic pump is the same whatever the pumped liquid density, but the pressure is not.

$$HEAD = \frac{PRESSURE}{\rho g}$$





# VAPOUR PRESSURE

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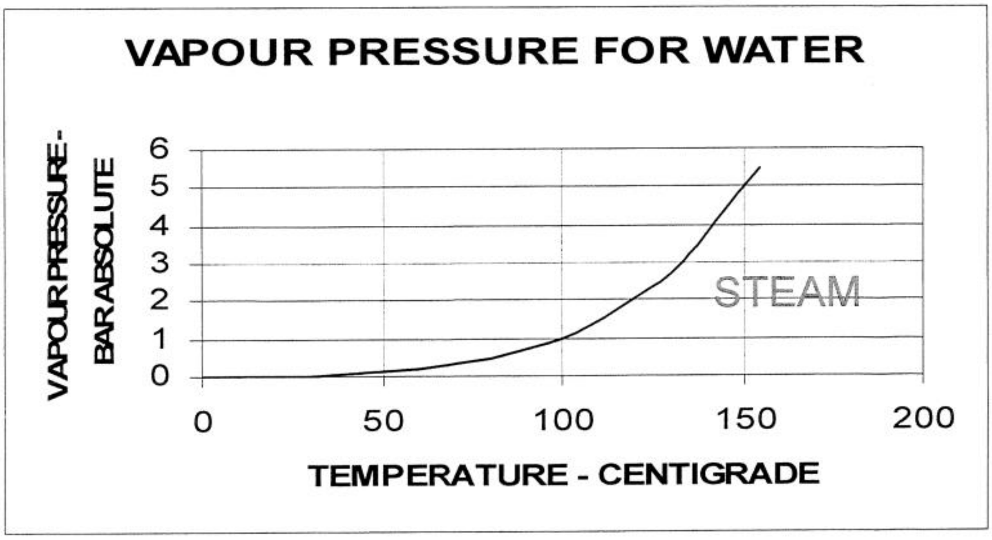
- The pressure at which conversion from the liquid to the vapour state takes place. The vapour pressure depends on the particular liquid and the temperature.



# PRESSURE AT PUMP INLET

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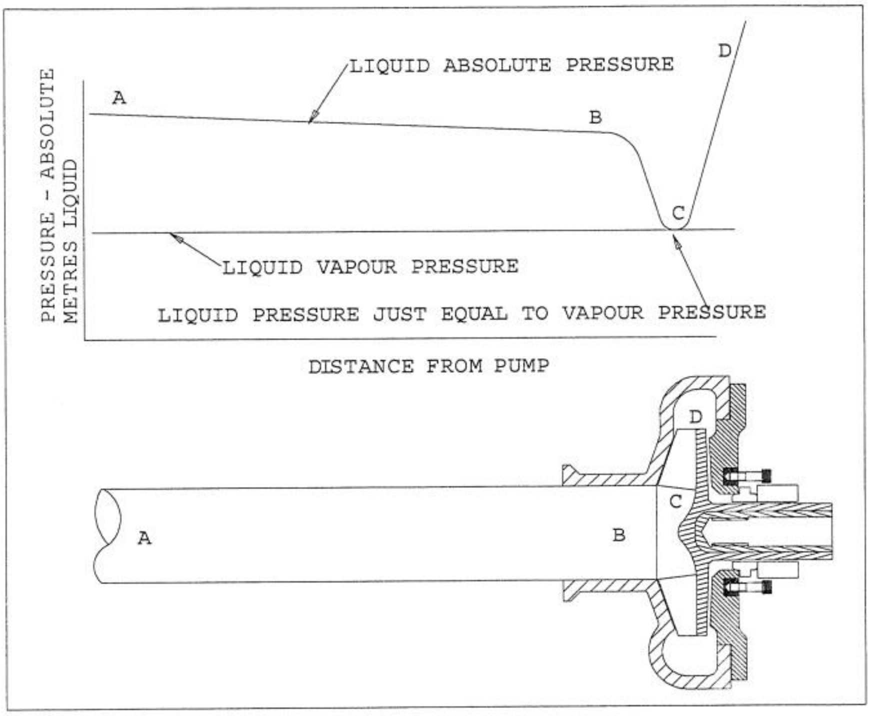




# CAVITATION

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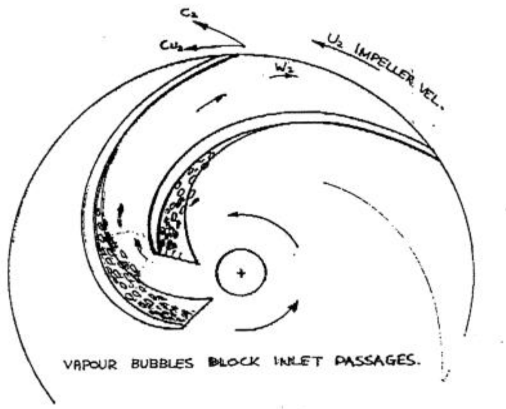


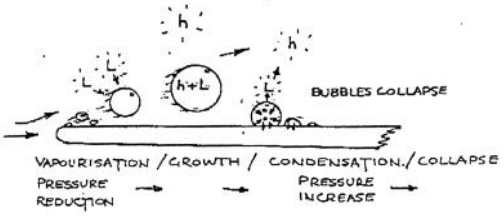

# POSITIVE DISPLACEMENT PUMP - TYPICAL ACTION

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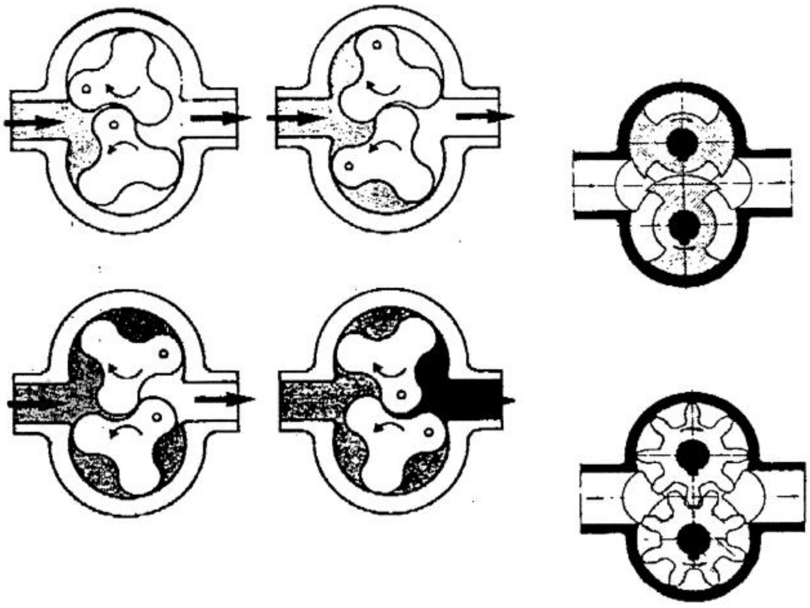



# ROTODYNAMIC PUMP INLET & OUTLET

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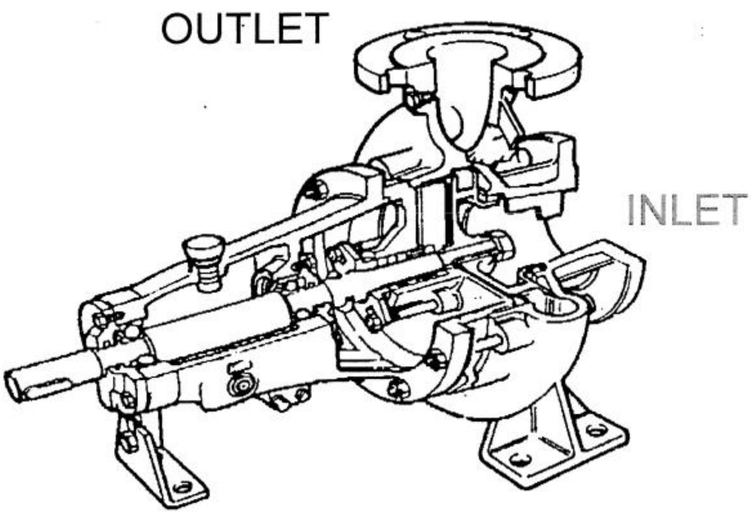
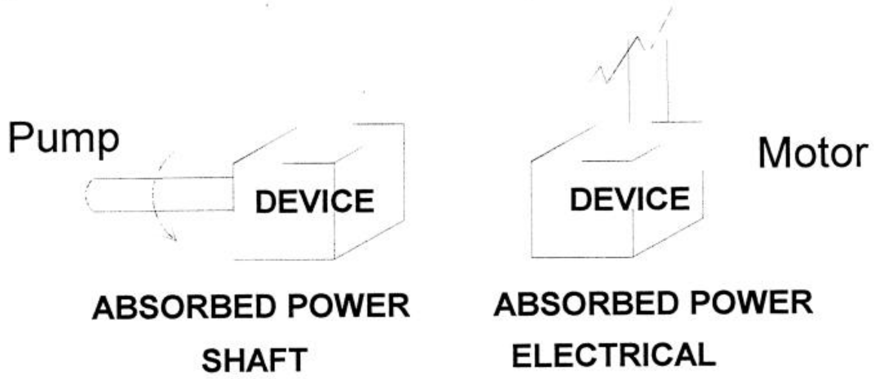


# ABSORBED POWER



- The energy used by the pump to add pressure and/or velocity to the flow of liquid.
- Typical units: Kilowatts, (Horsepower).






# WATER POWER



- The increase in the total energy of the liquid imparted by the pump.
- Also known as water horse power, even when expressed in kilowatts!
- Water power can be calculated from the formula:
- WATER POWER = FLOWRATE x HEAD x DENSITY x GRAVITY
- POWER IN WATTS, FLOWRATE IN $m^3/S$, TOTAL HEAD IN m, DENSITY IN
- $kg/m^3$, GRAVITY = 9.81 $m/s^2$

# EFFICIENCY



- The ratio of water power to the absorbed power, usually expressed as a percentage.
- it is a measure of how well the pump converts input energy into useful energy within the liquid.
- Pump efficiency is calculated using power absorbed at the drive connection of the pump shaft.
- Overall efficiency is calculated using power absorbed into the driver (usually an electric motor). Also known as "Wire to Water" efficiency.



# CLASSIFYING PUMPS

Cranfield UNIVERSITY
Process Systems Engineering

- Classified by Operating Principle
- Classified by Construction
- Classified by Application Features

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