People's Democratic Republic of Algeria

Ministry of higher education and scientific research

University of Mustapha Stambouli

Faculty of science and technology

Mechanical engineer department

Course handout

Turbomachinery

Master 2 Academic Mechanical Engineering

Specialty: Mechanical Construction

Prepared by: Abed Mourad

Academic year: 2024/2025

Preface

Turbomachinery plays a fundamental role in various industries, including energy production, aerospace, and fluid transport. It encompasses a wide range of machines, such as turbines, compressors, and pumps, which facilitate the conversion and transfer of energy. The efficient operation of these machines is crucial for optimizing performance, reducing energy consumption, and ensuring reliability in industrial applications.

This course is designed to provide a comprehensive understanding of turbomachinery principles, covering key topics such as classification, performance analysis, and design considerations. It introduces students to fundamental concepts, including the laws of similarity, pump and turbine operation, cavitation, and system coupling. Through theoretical explanations and practical examples, the course aims to bridge the gap between theoretical knowledge and real-world applications.

Structured into multiple chapters, this course progresses from general turbomachinery principles to more advanced topics, such as centrifugal pump sizing and hydraulic turbines. It is primarily intended for mechanical engineering students specializing in energy systems but can also serve as a valuable reference for professionals seeking to deepen their knowledge in the field.

Each chapter is designed to reinforce learning through quizzes, exercises, and application-based examples. By the end of this course, students will be equipped with the analytical tools necessary to understand, evaluate, and optimize turbomachinery performance in various engineering contexts.

The chapters are as follows:

Chapter 01: Generalities on Turbomachines

Chapter 02: Turbomachinery Similarity Laws

Chapter 03: Pumps

Chapter 04: Cavitation in pumps

Chapter 05: Couplings of centrifugal pumps

Chapter 06: Sizing of centrifugal pumps

Chapter 07: Hydraulic turbines

Table of contents

Objectives

This course will introduce basic ideas of turbomachinery and the basic equations that govern the performance of turbomachinery.

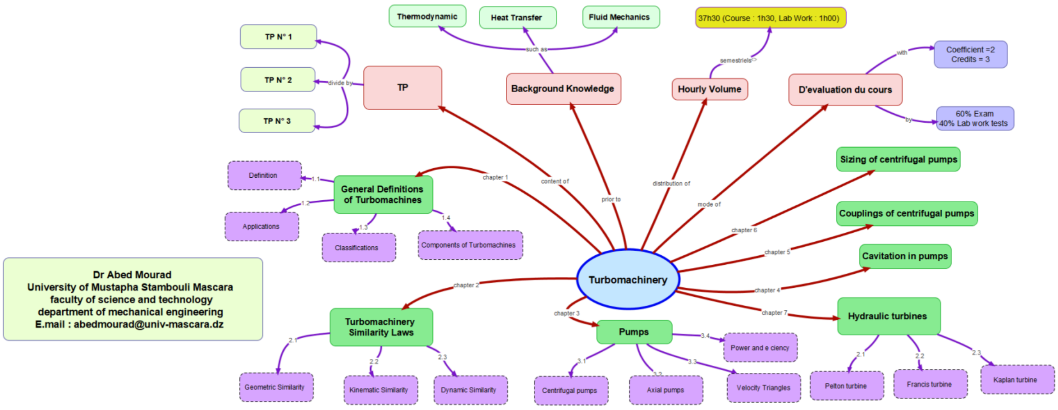
Course Objectives

Upon successful completion of the *course*, the student will be able to:

- Classify the turbo-machines based on various factors.
- Compare the features and working of various turbomachines
- Memorize the basic principles, governing equations and applications of turbo machine.
- Understanding operational characteristics of turbomachines.
- Apply the concepts of energy transformation in turbo machines
- Analyse the performance of Hydraulic pumps and turbines
- Design and evaluate the critical parameters involved in power generation
- Evaluate the performance of turbine.

Introduction

This course will provide a framework to discuss different kinds of turbomachinery through a unified approach. The material presented is intended for undergraduate and graduate students apart from professional engineers in the industry engaged in the analysis and development of turbomachinery. Coverage begins with the fundamental concepts. This is followed by the similarity rules. The course will be concluded with a discussion on *CFD* in the design and analysis of turbomachinery.



I Pre-requisites

The prerequisites for a turbomachinery course typically include a solid foundation in the following areas:

- **Basic Thermodynamics**: Understanding the principles of energy transfer, heat work, and the laws of thermodynamics.
- **Fluid Mechanics**: Knowledge of fluid properties, flow dynamics, and the behavior of fluids in motion.
- **General Engineering**: Familiarity with statics, dynamics, and basic engineering concepts.

II Pre-requisites Test

Quiz 1: Thermodynamics

Define the first law of thermodynamics.

Quiz 2: Thermodynamics

What is enthalpy, and how is it related to internal energy?

Quiz 3: Fluid Mechanics

Explain the difference between laminar and turbulent flow.

Quiz 4: Fluid Mechanics

Derive the Bernoulli equation for an incompressible fluid.

Quiz 5: General Engineering

What is the purpose of a free body diagram?

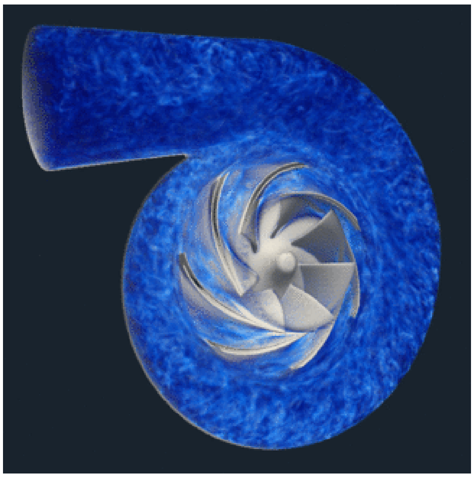
Quiz 6: General Engineering

Explain the concept of a control volume and its significance in engineering analysis.

III Chapter 01: Generalities on Turbomachines

1. Definition of Turbomachines

Generally speaking, a turbomachine is defined as a device which makes it possible to give or remove energy from a fluid by the dynamic action of a rotating element called the rotor. The prefix turbo comes from the Latin turbinis which means ¨which turns¨ or in rotation *1*[1].



Flow in a turbomachine

History

⊕ Extra

Historically turbomachines were introduced in 1822 by french mining engineer Claude Burdin

Note

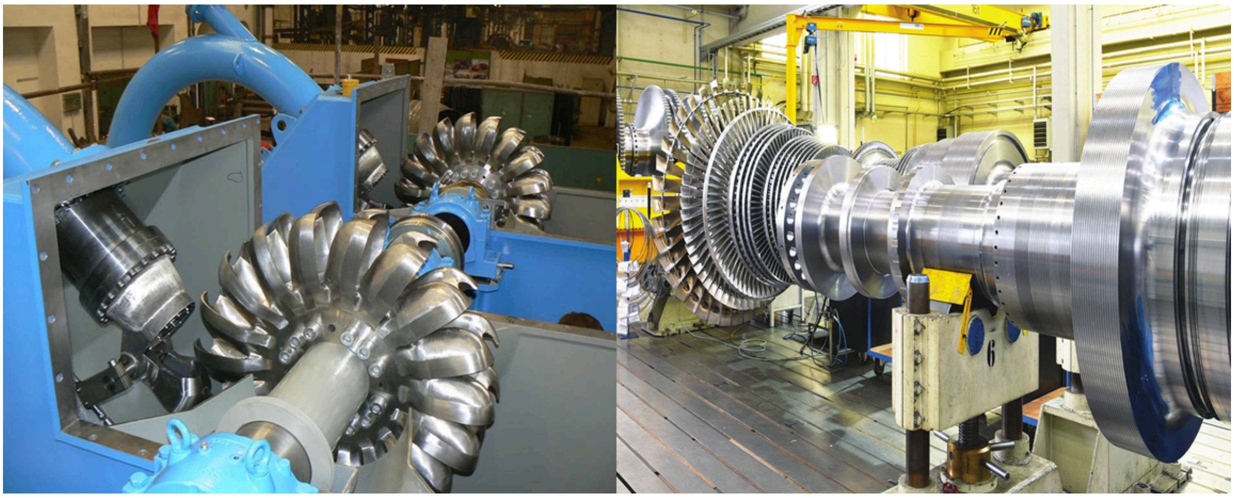
Turbomachines form an important category of devices that transform energy through the use of a fluid.

2. Applications

Turbomachines are found in a large number of applications requiring energy transfer.

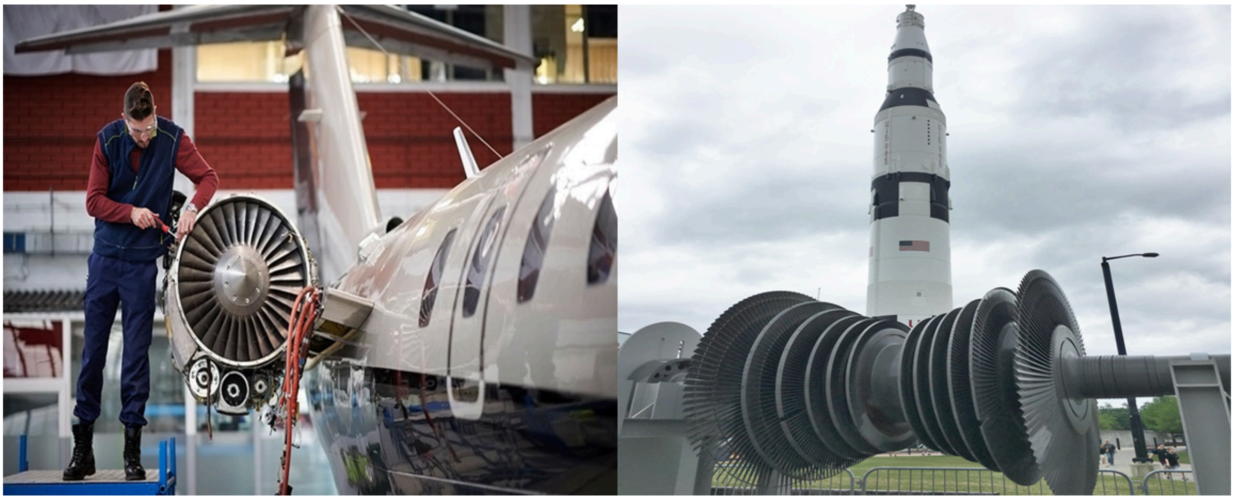
Essentially, there are three types of applications:

- Electricity production (gas turbine, steam turbines, hydraulic turbines)



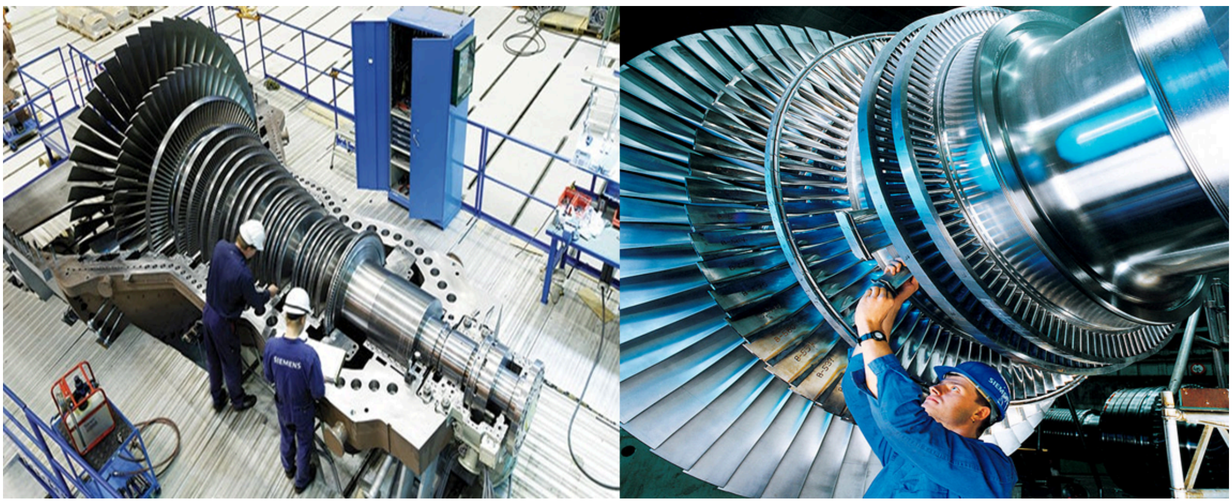
Turbine for electricity production

- Propulsion (aviation gas turbine, locomotive compressors, ship gas turbine)



Turbines for propulsion

- Heavy industry (centrifugal compressors, turbo-compressor for diesel engine, turbine steam, gas turbine, pumps and fans).



Turbines for heavy industry

3. Classification of Turbomachines

Many criteria are used to classify turbomachines, the most important are as follows:

3.1. The direction of energy transfer

Depending on the direction of energy transfer turbomachines are divided into two main categories

Power Consumer Turbos: Turbomachines which provide energy to the fluid (enthalpy). In this group, we find compressors, fans, and pumps.

Power Supplier Turbos: Turbomachines from which energy is removed from the fluid to use as mechanical work. In this case, we then speak of turbines.

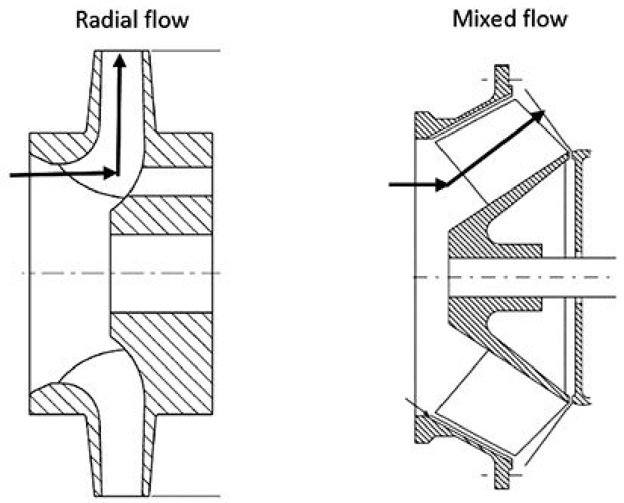
3.2. The main direction of the flow

According to the main direction of the flow relative to the axis of rotation of the machine we have

Axial flow turbomachines in which the direction of flow is parallel to the axis of rotation of the machine.

Radial or centrifugal flow turbomachines in which a significant part of the flow at the inlet or outlet is in the direction normal to the axis of rotation or radial.

Mixed flow turbomachines in which a turbine can be seen as a cross design between an axial and a radial turbine as it holds the characteristics of both.



The main direction of the flow relative to the axis of rotation of a turbomachine.

3.3. The type of installation

Depending on the type of installation: There are two types

Embedded turbomachines such as centrifugal pumps, gas turbines, etc., where the fluid circulates inside conduits;

Free-flow turbomachines such as wind turbines, airplane or ship propellers.

3.4. The nature of the fluid

Depending on the nature of the fluid : The fluid can be compressible or incompressible

The compressible fluid undergoes variations in its density which must be taken into account especially if they are important .

The incompressible fluid does not undergo almost no variation in its density .

3.5. The function of the turbomachine

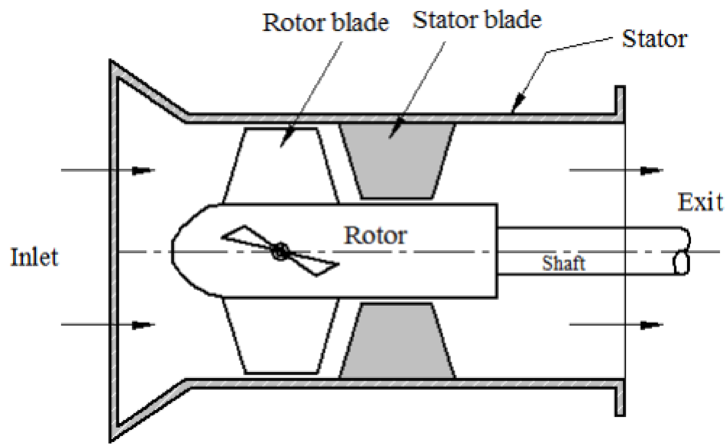
Depending on the function of the machine if it involves transforming the energy of a fluid into mechanical energy or vice versa.

If the transformation takes place from mechanical energy to hydraulic energy (fluid energy), the machine is said to be a **driving machine**. A pump, a fan, a compressor,…

If the transformation takes place from hydraulic energy to mechanical energy, the machine is said to be **receiving machine**. A hydraulic turbine, wind turbine, etc.

4. Components of Turbomachines

A turbomachine is essentially made up of three distinct organs



Basic turbomachine components component

4.1. Rotor

Rotor is a rotating element carrying the rotor blades or vanes. Rotor is also known by the names runner, impellers etc. depending upon the particular machine. Here energy transfer occurs between the flowing fluid and the rotating element due to the momentum exchange between the two.

4.2. Stator

Stator is a stationary element carrying the guide vanes or stator blades. Stator blades are also known by guide blades or nozzle depending upon the particular machine. These blades usually control the direction of fluid flow during the energy conversion process.

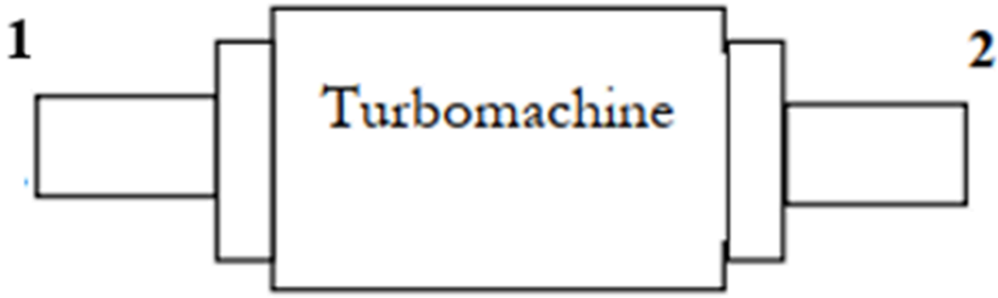
4.3. Shaft

Shaft is transmitting power into or out of the machine depending upon the particular machine. For power generating machines, it may call as output shaft and for power absorbing machines; it may called as input shaft.

5. General problem of Turbomachines

Consider a turbomachine schematized in the Figure with points 1 and 2 being respectively the inlet and outlet of a turbomachine. Two cases of fluid flow can be observed:

- flow where the fluid is incompressible
- flow where the fluid is compressible



Turbomachine

Reminder

To write the general relation of work in turbomachine we need first to understand the *Bernoulli's principle* here is a video to help you understand it

(see Bernoulli's Equation)

5.1. Case of a flow where the fluid is incompressible

The general relation for unit of mass is written:

$$W_{12} = \frac{(P_2 - P_1)}{\rho} + \frac{1}{2}(V_2^2 + V_1^2) + g(Z_2 - Z_1) \tag{1.1}$$

W_{12}: Represents the work exchanged with the outside environment.

$\frac{(P_2 - P_1)}{\rho}$: Represents the change in potential energy due to the change in pressure.

$\frac{1}{2}(V_2^2 + V_1^2)$:Represents the change in kinetic energy.

$g(Z_2 - Z_1)$: Represents the variation in potential energy due to the altitude variation.

5.2. Case of a flow where the fluid is compressible

This is the case for gases , air and water vapor . Let's consider a steam turbine with

State 1: pressure P1, mass volume V1, temperature T1, speed C1

State 2: pressure P2, mass volume V2, temperature T2, speed C2

The difference altitude (Z2-Z1) is neglected .

the general relation becomes:

$$(Q + W)_{12} = (H2 - H_1) + \frac{1}{2}(V_2^2 + V_1^2) \tag{1.2}$$

$(Q + W)_{12}$: Represents the thermal and mechanical energy exchanged with the outside environment.

$(H_2 - H_1)$: Represents the change in enthalpy.

$\frac{1}{2}(V_2^2 + V_1^2)$: Represents the change in kinetic energy.

If we assume that there is no heat exchange between the fluid which passes through the machine and the external environment.

the relationship then becomes:

$$W_{12} = (H_2 - H_1) + \frac{1}{2}(V_2^2 + V_1^2) \tag{1.3}$$

if W is positive, the fluid has received work; the machine is power consumer Turbomachine.

if W is negative the machine is power supplier Turbomachine

IV Did you understand the basics ?

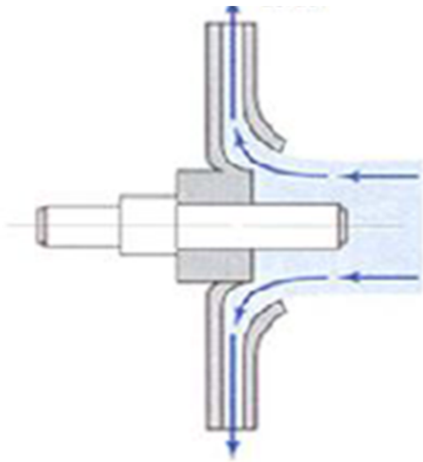
Quiz 1: The direction of energy transfer

What are the different classes of turbomachines depending on the direction of energy transfer ?

- **A** Power Consumer Turbos
- **B** Axial flow turbomachines
- **C** Power Supplier Turbos
- **D** Embedded turbomachines

Quiz 2: Turbine type

What is the name of the turbine shown in the figure?

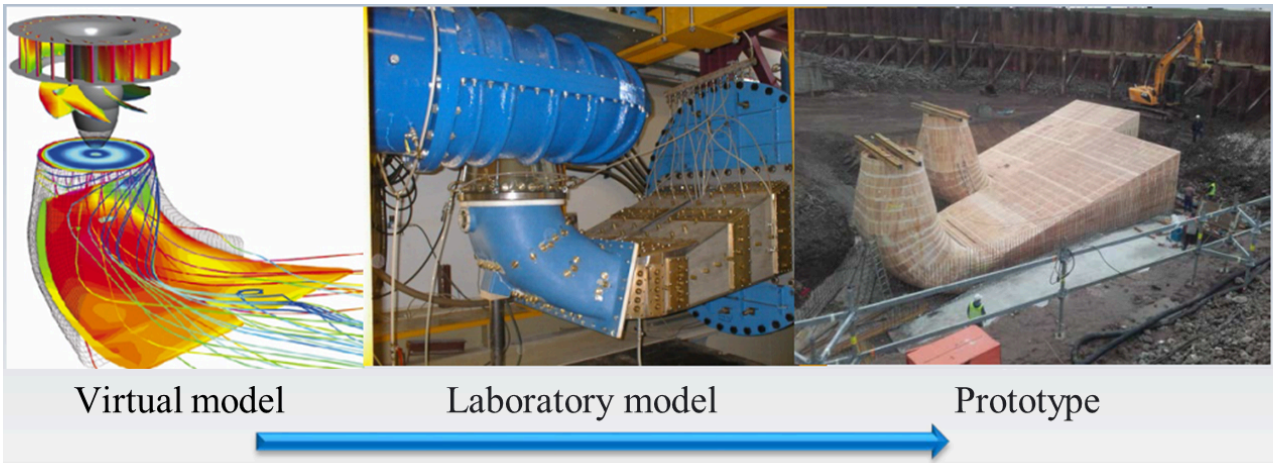


V Chapter 02: Turbomachinery Similarity Laws

Introduction

In the field of turbomachines, the objective of similarity is to compare the performances between so-called similar machines, or between a scale model and a prototype.

This comparison or analysis is carried out in terms of non-dimensional variables



1. Why we use turbomachinery similarity laws ?

The fundamental specification of a turbomachine concerns:

- the flow
- the nature of the fluid (gas, liquid)
- the quantity of work to be exchanged
- the state at entry

From these specifications we can:

- select the type of machine (axial, radial, mixed);
- carry out a preliminary sizing and estimate its performance.

Dimensional analysis provides theoretical support for the first stage of design "preliminary sizing". It incorporates dimensionless numbers whose value can be interpreted as part of the DNA of each device.

The richness of the formalism of dimensionless numbers allows the generation of families of so-called similar machines, the prediction of their performance and a guide to building prototypes.

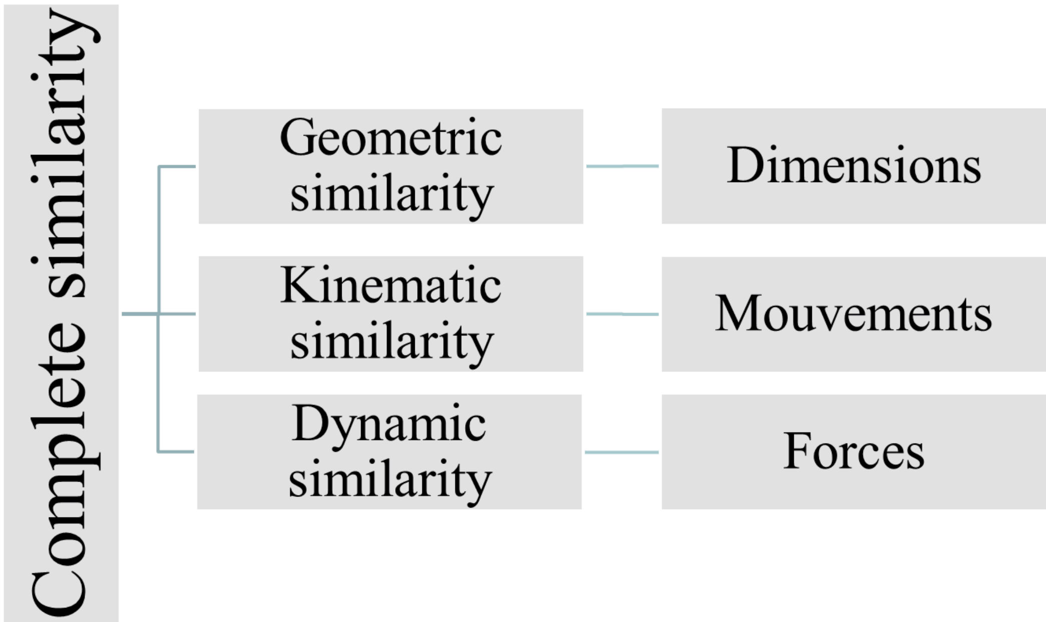
To carry out an agile design, we aim at the transport of known characteristics of one turbomachine towards another. To do this we need:

- the respective dimensions are to scale
- the speed ratios are equal at all points (the current lines are similar)
- the ratios of inertia/other forces are equal

2. Definition

Similarity is the basis so that the study of phenomena on scale models or models can be extrapolated towards the problem analyzed, called the prototype.

The similarity occurs if there is a constant relationship for a quantity (length, force, etc.) between homologous points of the model and the prototype. In fluid mechanics we distinguish three types of similarity: geometric, kinematic and dynamic 2[2].



similarity laws

Note

Similarity in turbomachines implies that we can correspond a point p1 of the prototype at a time t1 with a point p2 of the model at a time t2. Such points are called homologous points.

Reminder

In addition to an analogy in form between a model and a prototype, formal principles must be respected to bring the results obtained on a model back to the real case.

3. Geometric similarity

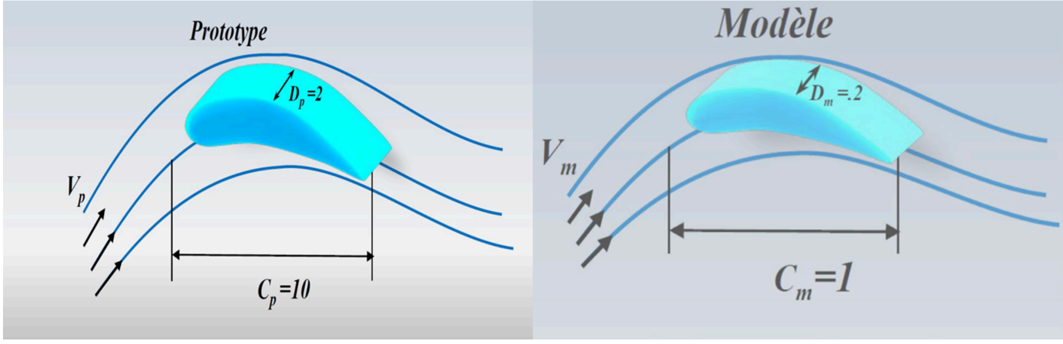
Az Definition

Geometric similarity deals with the geometric scale ratio, i.e. what concerns the L dimension

A model and a prototype are geometrically similar if all dimensions have the same scale ratio λ L

$$\frac{L_m}{L_p} = \lambda_L = \text{const} \tag{2.1}$$

L_m, L_p : Length of the model and the prototype, respectively.



geomteric similarity

⚠ Warning

Although conceptually simple, this similarity can be seen faced with practical difficulties to fully satisfy it For example, the similarity for the relative roughness of a surface is not easy to achieve, or even to reproduce on scale the dimensions at certain places of a turbomachine, such as the separation between the blade and the casing, proves very difficult.

To overcome these weaknesses, empirical corrections, often confidential, are applied by manufacturers to transpose the data obtained during tests on the model to the prototype

4. Kinematic similarity

Az Definition

Kinematic similarity refers to similarity of motion. A model and a prototype are kinematically similar if the trajectories of homologous particles are geometrically similar, ending up in homologous positions in homologous times.

Kinematic similarity treats the temporal scale ratio λ T through ratios of the derived variables, velocity and acceleration. SO :

Two flows are kinematically similar if the speed ratios λ V (accelerations λ a) are identical at homologous points 3[3].

$$\frac{v_m}{v_p} = \frac{\left(\frac{L_m}{T_m}\right)}{\left(\frac{L_p}{T_p}\right)} = \left(\frac{L_m}{L_p}\right)\left(\frac{T_p}{T_m}\right) = \lambda_v \Longrightarrow \lambda_v = \left(\frac{\lambda_L}{\lambda_T}\right) \tag{2.2}$$

$$\frac{a_m}{a_p} = \frac{\left(\frac{L_m}{T_m^2}\right)}{\left(\frac{L_p}{T_p^2}\right)} = \left(\frac{L_m}{L_p}\right)\left(\frac{T_p^2}{T_m^2}\right) = \lambda_a \Longrightarrow \lambda_a = \left(\frac{\lambda_L}{\lambda_T^2}\right) = \left(\frac{\lambda_v}{\lambda_T}\right) \tag{2.3}$$

Note

Kinematic similarity adds the fundamental dimension Time [4] *Principles of Turbomachinery, 2nd Edition*.

Warning

For kinematic similarity to take place, geometric similarity must then exist. But, geometric similarity does not guarantee kinematic similarity!

5. Dynamic similarity

Definition

Dynamic similarity implies that the forces between homologous points of the model y of the prototype have the same ratio of forces acting on them.

$$\frac{F_m}{F_p} = \frac{M_m L_m T_m^{-2}}{M_p L_p T_p^{-2}} = \left(\frac{M_m}{M_p}\right)\left(\frac{L_m}{L_p}\right)\left(\frac{T_m}{T_p}\right)^{-2} = \lambda_F \rightarrow \lambda_F = \lambda_M \lambda_a \tag{2.4}$$

Note

If the mass distribution is similar between the model and the prototype. So, geometric similarity leads to mass similarity.

Since the forces are proportional to the accelerations, in the previous case the kinematic similarity implies dynamic similarity.

In dynamic similarity ,we are interested in the ratio of the inertial force $\boldsymbol{Fi}$ to the viscous forces $\boldsymbol{F}$ visq , pressure , $\boldsymbol{F}$ press , gravitational $\boldsymbol{F}$ grav , and elastic $\boldsymbol{F}$ elas

The action of these forces will lead the acceleration of a fluid described by Newton's 2nd law [5] *Fluid Dynamics and Heat Transfer of Turbomachinery1st Edition*.

$$\sum \vec{F} = \vec{F}_{\text{grav}} + \vec{F}_{\text{press}} + \vec{F}_{\text{visq}} + \vec{F}_{\text{elas}} = m\vec{a} \tag{2.5}$$

If we notice that the inertial force, F_i, acts in the opposite direction to the acceleration $F_i = -m\vec{a}$, we can write

$$\vec{F}_{\text{grav}} + \vec{F}_{\text{press}} + \vec{F}_{\text{visq}} + \vec{F}_{\text{elas}} + \vec{F}_i = 0 \tag{2.6}$$

By considering only the modules of these forces, a (pseudo) dimensionless equation can be obtained by taking the inertial force as a reference. Notably:

$$\frac{F_{\text{grav}}}{F_i} + \frac{F_{\text{press}}}{F_i} + \frac{F_{\text{visq}}}{F_i} + \frac{F_{\text{elas}}}{F_i} + 1 = 0 \tag{2.7}$$

These non-dimensional ratios (or the inverse F_i/F) of the force modules have been associated with the work of scientists and abbreviations have been introduced which recall their names

$$\underbrace{\frac{F_{\text{grav}}}{F_i}}_{\text{Froude(Fr)}} + \underbrace{\frac{F_{\text{press}}}{F_i}}_{\text{Euler(Eu)}} + \underbrace{\frac{F_{\text{visq}}}{F_i}}_{\text{Reynolds(Re)}} + \underbrace{\frac{F_{\text{elas}}}{F_i}}_{\text{Mach(Ma)}} + 1 = 0 \qquad (2.8)$$

These numbers represent **relative intensities of forces**.

The importance of these relative intensities lies in the fact that we can write dimensionless equations describing the motion of a fluid and that these are UNIVERSAL

> 💬 Note
>
> Universal means that they are valid independently of the properties of the fluid: (μ, ρ), space (L), flow characteristics (pressure p, acceleration $\vec{a}$, elasticity E), and the gravitational field $\vec{g}$

Name	Symbol	Equation	Interpretation	Application
Archimedes Number	Ar	$\frac{g\ell^3\rho_f(\rho-\rho_f)}{\mu^2}$	buoyancy forces / viscous forces	in nature and force convection
Drag Coefficient	Cd	$\frac{D}{\frac{1}{2}\rho U^2 A}$	drag force / inertia effects	Aerodynamics, hydrodynamics, note this coefficient has many definitions
Eckert Number	Ec	$\frac{U^2}{C_p\Delta T}$	inertia effects / thermal effects	during dissipation processes
Euler Number	Eu	$\frac{P_0-P}{\frac{1}{2}\rho U}$	pressure effects / inertia effects	potential of resistance problems
Froude Number	Fr	$\frac{U}{\sqrt{gl}}$	inertia effects / gravitational effects	open channel flow and two phase flow
Grashof Number	Gr	$\frac{\beta\Delta Tg\ell^3\rho^2}{\mu^2}$	buoyancy effects / viscous effects	natural convection
Mach Number	M	$\frac{U}{c}$	velocity / sound speed	Compressibility and propagation of disturbance

Prandtl Number	Pr	$\frac{\nu}{a}$	viscous diffusion / thermal diffusion	Prandtl number is fluid property important in flow due to thermal forces
Reynolds Number	Re	$\frac{\rho U \ell}{\mu}$	inertia force / viscous force	in most fluid mechanics issues

Some dimensionless parameters encountered in Turbomachines

Conclusion

Similarity can be defined as "All dimensionless numbers have the same values for the model and prototype".

VI Did you understand the basics of Turbomachinery Similarity Laws ?

Quiz 1: Similarity rules

What is the use of similarity rules?

Quiz 2

Geometric Similarity in Turbomachinery refers to:

- Machines having the same color and shape
- Machines having the same type of fluid flowing through them
- All corresponding dimensions in the machines having the same ratio
- Machines having the same manufacturer

Quiz 3

The Reynolds number is significant in turbomachinery because it:

- Determines the color of the machine
- Represents the ratio of inertial forces to viscous forces
- Indicates the type of fluid used
- Is always a constant value for all machines

VII Chapter 03: Pumps

Introduction

Pumps are essential components in a wide range of industrial and engineering applications, playing a crucial role in moving fluids through mechanical processes. Whether it's for transferring water in municipal systems, circulating coolant in power plants, or facilitating chemical processes in refineries, pumps are an indispensable part of modern infrastructure.

In this chapter, we will explore the fundamental principles that govern pump operation, focusing on key concepts such as fluid dynamics, energy conversion, and mechanical design. You will be introduced to the various types of pumps, including centrifugal, positive displacement, and specialized pumps, each suited for different types of flow and pressure conditions.

By the end of this chapter, students will have a comprehensive understanding of:

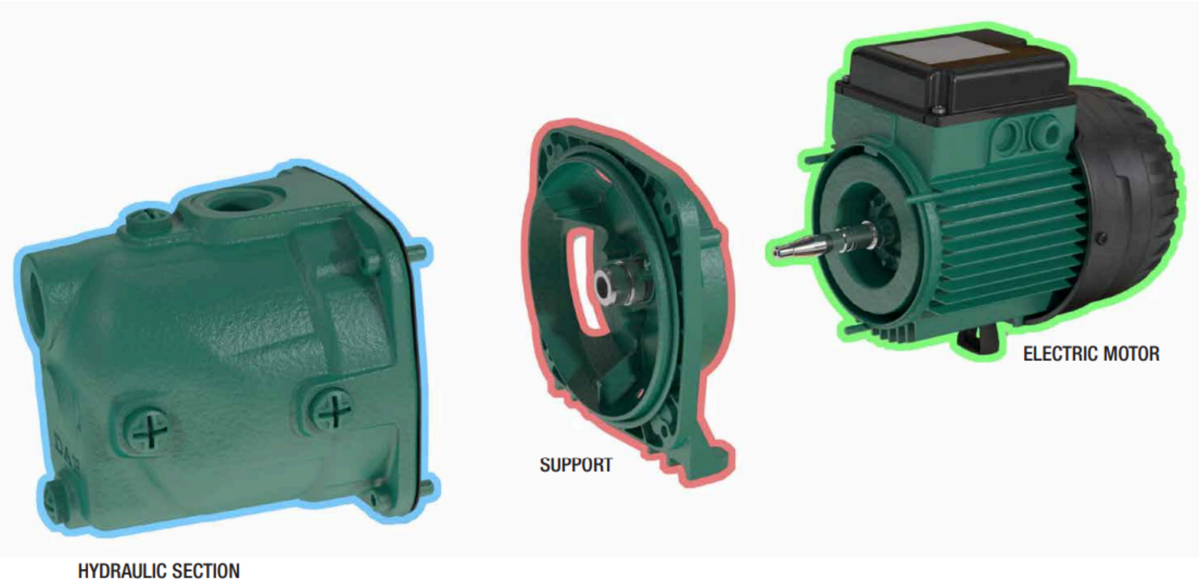
- The classification and operation of different types of pumps
- The performance characteristics that dictate pump efficiency and selection
- Key design considerations for optimizing pump performance in various applications.

The content of this chapter will build on prior knowledge from previous chapters, linking core fluid mechanics principles with practical turbomachinery applications. Pumps not only serve as a critical link between mechanical systems and fluid management, but they also represent a significant area of innovation in energy efficiency and sustainability.

1. Pumps

Az Definition

The pump is mechanical device which conveys liquid from one place to another place. It can be defined as a hydraulic machines which converts the mechanical energy into hydraulic energy(pressure energy of liquid)



BASIC ELEMENTS OF PUMP

All water pumps are composed of two basic selections: Electric motor and hydraulic section.

The support is utilized to secure the pump to the base and prevent it from moving

Note

The purpose of pumps may be increasing the pressure energy, imparting kinetic energy, lifting and circulating or extracting liquids etc.

Note

The pump is power absorbing machines.

The power can be supplied to the pump by a prime motor like an electric motor, an internal combustion engine or turbine.

Pumping means addition of energy to a liquid to move it from one place to the another, and this done by means of piston, plunger, impeller, propeller, gears or screw depending upon types of pumps.

1.1. Applications

The pumps are used for various applications in various fields as follows:

1. Thermal engineering:
    - To feed water in to the boiler
    - To circulate the water in condenser
    - To circulate lubricating oil in the proper place in various machines
2. Agriculture and irrigation:
    - To lift water from deep well
    - To convey water from one place to another
3. Chemical industries:
4. Municipal water works and drainage system
5. Hydraulic control systems

1.2. Classification

The pump can be classified according to principle by which the energy added to the fluid and their design features as follows:

According to Principles of Operation and Design Characteristics

According to shape of impeller and casing

According to type of impeller

According to working head

According to number of entrances to the impeller

According to specific speed of pump

a) Principles of Operation and Design Characteristics

- Positive displacement pump: These pumps operate on the principle of a definite quantity of liquid is discharged or displaced due to the positive or real displacement of working element like piston, plunger, gears, etc..

1. Reciprocating pump:
    1. Piston pumps.
        - Single cylinder single acting or double acting.
        - Double cylinder single acting or double acting.
    2. Plunger pumps
    3. Bucket pump-Hand pumps
2. Rotory pumps:
    1. Gear pumps
    2. Vane/lobe pumps
    3. Screw pumps

- Roto-dynamic pump: These pumps operate on the principle of the rise in pressure energy of liquid by the dynamic action of liquid. The dynamic action of liquid is carried by a revolving wheel which has curved vanes on it. This wheel is known as an impeller.

1. Radial flow pump or Centrifugal pump: In this pump, addition of energy to the liquid occurs when the flow of liquid in the radial path.
    - Single stage
    - Multi-stage
2. Axial flow pump: In this pump, the addition of energy to the liquid occurs when the flow of liquid in axial direction.
3. Mixed flow pump: In this pump, addition of energy to the liquid occurs when the flow of liquid in axial as well as radial directions.

- Other type of pumps: This types of pumps does not belongs to the category ofpositive displacement or roto-dynamic type pumps as follows:

1. Jet pump
2. Air lift pump

b) According to type of impeller

1. Closed or shrouded impeller:
    - The vanes are covered with metal side plates (shrouds) on both sides.
    - Wear is reduced.
    - Long-life performance with full capacity.
    - High efficiency.
    - This type of pump is used when the liquid to be pumped is pure and free from debris.
2. Semi-open impeller:
    - In this single plate (shroud) on the backside.
    - This pump handles liquids containing fibrous materials like paper pulp, sugar molasses and sewage water etc.
3. Open impeller:
    - Impeller vanes do not contain shrouds (side plate) on either side.
    - Vanes are open on both sides.
    - Handle abrasive liquids like a mixture of water-sand, pebbles and clay etc.

c) According to working head

1. Low head centrifugal pump: head up to 15m.
2. Medium head centrifugal pump: head between 15 to 40m.
3. High head centrifugal pump: head above 40m.

d) According to number of stages

1. Single stage centrifugal pump:
    - Only one impeller.
    - Can not produce sufficient high-pressure head efficiently.
    - Mostly used for lower head and lower discharge.

2. Multi-stage centrifugal pump:
    - Two or more identical impellers mounted on the same shaft in series or on different shafts in parallel.
    - Two types (1). Impeller in series & (2). Impeller in parallel
    - Produce a high-pressure head (use impeller in series).
    - Discharge a large quantity of liquid (use impeller in parallel).

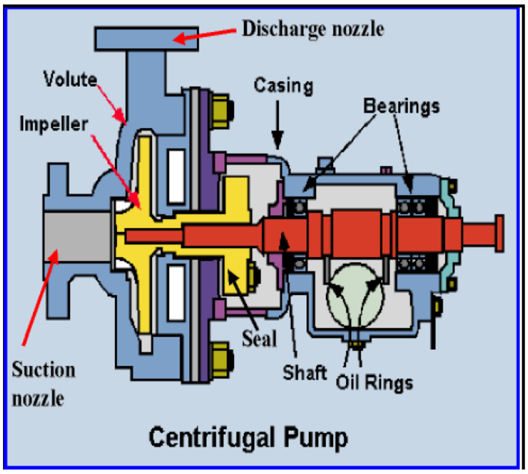
e) According to number of entrances to the impeller

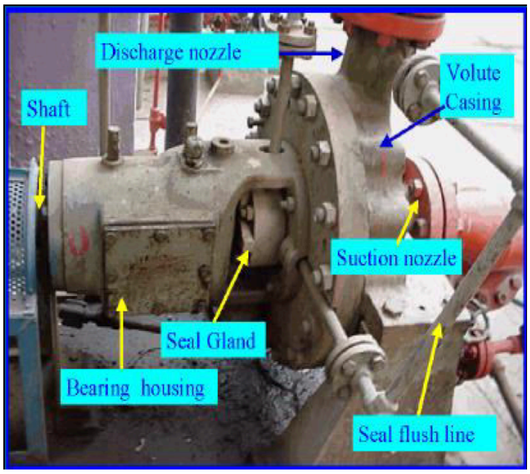
1. Single suction pump: Water is entered from one side of impeller.
2. Double suction pump: Water is entered from both side of impeller.

f) According to specific speed of pump

1. Low specific speed pump: specific speed in range 10-80 Example: Radial flow pump
2. Medium specific speed pump: specific speed in range 80-160 Example: Mixed flow pump
3. High specific speed pump: specific speed in range 160-500 Example: Axial flow pump

1.3. Main Parts of PUMP





geomteric similarity

a) Casing:

It is an air tight passage surrounding the impeller and its designed in such a way that the kinetic energy of liquid coming from impeller is converted into pressure energy before the delivery pipe

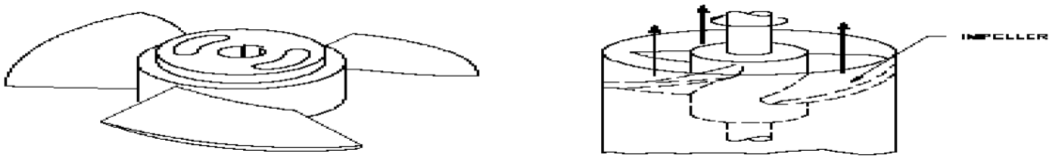
There are basically three types of casing

1. Volute (spiral) casing
2. Vortex (whirlpool) chamber casing
3. Volute casing with guide blades

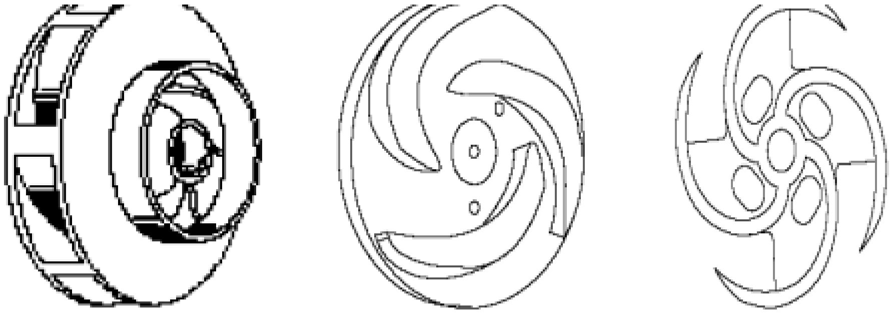
b) Impeller:

It is rotating element of centrifugal pump. It consists of finite number of backward-curved vanes. The number of vanes normally 6 to 12 in the impeller. Impeller is mounted on shaft which is coupled with the shaft of electric motor. In impeller the kinetic energy of liquid is increased.

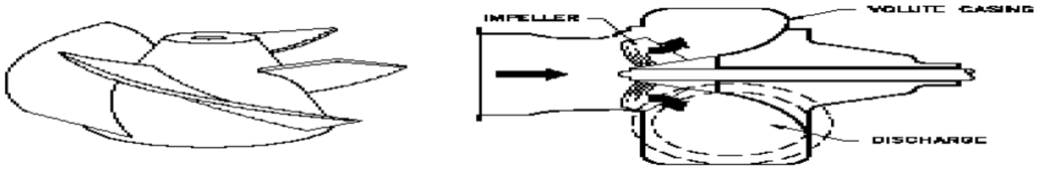
1. Different types of impeller for different type of flow is shown in below



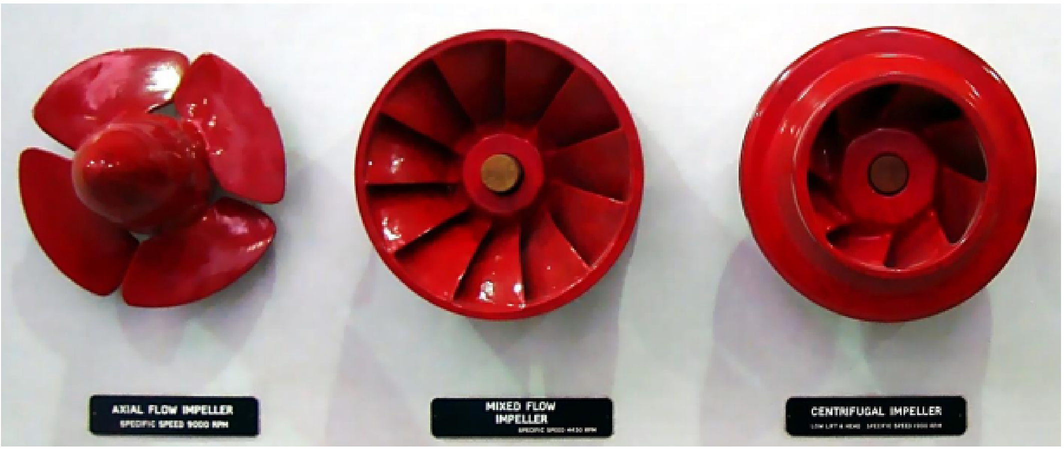
Axial flow



Radial flow



Mixed flow



Example impellers for the three types of rotodynamic pumps

c) Suction pipe with foot valve and a strainer

- Suction pipe: It is pipe whose one end is connected to the inlet of the pump and other end dips in to water in a liquid sump.
- Foot valve: It is a non return is essentially for all types of rotodynamic pumps. It helps in allowing the liquid to enter into pump in upward direction only and does not allow the liquid to move downwards.
- Strainer: The strainer is essential for all type of pumps. It protects pump against foreign material passes through the pump, without strainer pump may be chocked.

d) Delivery Pipe:

A pipe whose one end is connected to the pump and other end delivers at a required height is called delivery pipe.

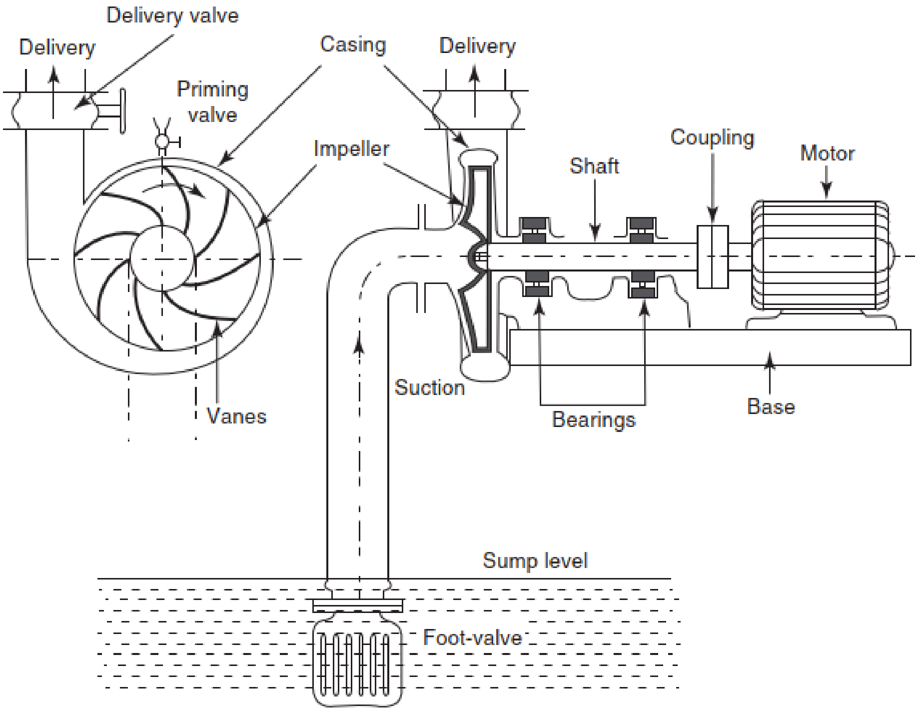
2. Centrifugal pumps

A centrifugal pump is a versatile type of turbomachine, used in a wide range of applications from small decorative water features to large-scale industrial processes. The smallest centrifugal pumps can lift water just 20 cm with minimal power, while large pumps used in city water supply, and oil wells can consume up to 1–1.5 MW per stage. This subchapter covers the construction, components, and classifications of centrifugal pumps, along with key operational concepts like minimum starting speed, net positive suction head (NPSH), cavitation, and priming.

Az Definition

A centrifugal pump is a turbomachine employed for lifting liquid from a lower level to a higher level. The input of the centrifugal pump is the mechanical energy in the form of torque on its rotating shaft. The output of the pump is the increased energy of the liquid in the form of the pressure or momentum (or both) of the liquid being lifted. Thus, the centrifugal pump converts mechanical energy into energy of the liquid.

Centrifugal pumps handle a variety of liquid chemicals, different types of oils, slurries, etc. However, instead of the general term "liquid," the term "water" is used in this chapter to explain the functions of the pump.



Schematic view of a centrifugal pump installation.

S. No.	Part	Functions
01	Motor	To provide power to the centrifugal pump
02	Shaft	To transfer the power from the motor to the impeller
03	Bearings	To support the shaft and to provide smooth running
04	Impeller	To impart energy to the water

05	Casing	To collect water from the periphery of the impeller and to carry it to the delivery flange (also to act as a diffuser to convert kinetic energy to pressure energy, if so designed)
06	Delivery pipe	To deliver water from the pump to some tank
07	Suction pipe	To draw water from the sump to the impeller
08	Foot-valve	1. To act as a non-return valve 2. To prevent any solid particle of bigger size from entering the suction pipe and impeller
09	Priming valve	To facilitate the process of priming (i.e., filling the impeller and casing with water)

Parts of a centrifugal pump installation and their functions

2.1. Advantages of Centrifugal Pumps

1. The flow process is continuous and not periodic.
2. The parts of centrifugal pumps are few in number.
3. The parts of centrifugal pumps are simple, their manufacture requires less care, and hence they are less costly.
4. The maintenance inventory of centrifugal pumps is very much limited; hence, the cost of inventory is also less.
5. The hydraulic efficiency of centrifugal pumps is very high.
6. The volumetric efficiency of centrifugal pumps is very high.
7. The mechanical efficiency of centrifugal pumps is very high.
8. The overall efficiency of centrifugal pumps is high.
9. The operating cost of centrifugal pumps is low.
10. Because of the purely rotary motion, there are no vibrations and higher speeds are possible.
11. Large volumetric flow rates are possible in centrifugal pumps.
12. Control of the flow of fluid is very easy in centrifugal pumps.

The only drawback is that in centrifugal pumps, very high pressure ratios are not possible.

2.2. Classification of Centrifugal Pumps

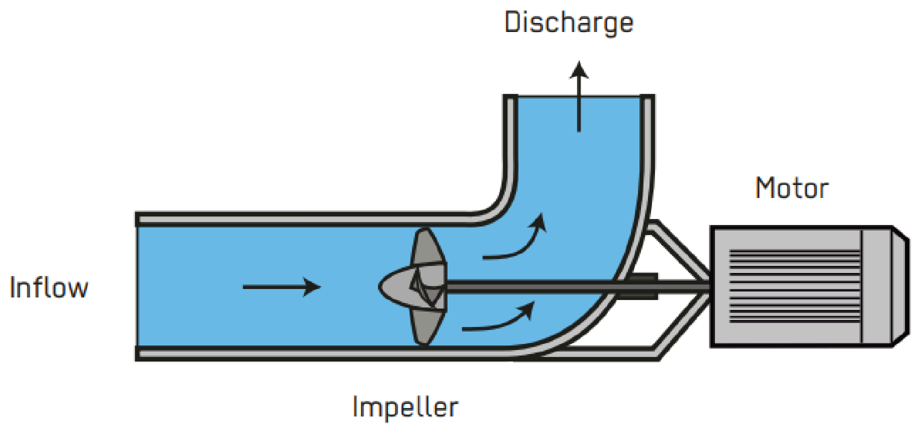
Centrifugal pumps can be classified based on different criteria:

S. No.	Basis of Classification	Different Types
1	Specific speed	Low, medium, and high specific speed pumps
2	Head	Low-, medium-, and high-head pumps
3	Type of casing	Volute, diffusion volute, and vane-less diffusion pumps
4	Type of impellers 1	Single-entry impeller, double-entry impeller Closed-, semi-open-, and open-type impellers
5	No. of impellers	Single-stage pumps, multi-stage pumps (2, 3, 4, ... stages)
6	Direction of flow	Radial flow type, mixed flow type, axial flow type pumps
7	Type of material	Cast iron pumps, alloy steel pumps, stainless steel pumps

Classification of centrifugal pumps

3. Axial pumps

An Axial Flow Pump is a large diesel or electric motor driven pump capable of moving large volumes of water at relatively low heads. The most common uses for Axial Flow Pumps are for clearing water from flooded areas, lifting large amounts of flow within a treatment plant or water system, or for agricultural purposes.



An Axial Flow Pump

3.1. Strengths and Weaknesses

+	Can pump large flow rates
+	Typically run at low speed, so less wear
-	Not possible to pump to high pressures
-	Better performance with individual discharge headers than combined discharge headers
-	Should not be used with a closed discharge valve
-	Need large depths of water in the suction pit to meet submergence requirements

Strengths and Weaknesses

4. Velocity Triangles

Az Definition

A velocity triangle with reference to a turbomachine is the representation of three important Velocities as the sides of a triangle. The three velocities are as follows:

1. The peripheral velocity of the rotor blade, U.
2. The relative velocity of the fluid with respect to the rotor blade Vr.
3. The absolute velocity of the fluid V.

$$\overrightarrow{V} = \overrightarrow{V_r} + \overrightarrow{U} \tag{3.1}$$

These above velocities are subscripted to indicate their positions with respect to the blades: Subscript "1" is for the inlet position and subscript "2" is for the outlet position.

The following points serve as the guidelines in understanding the velocity triangles:

1. All the velocities are vector quantities. Each velocity has a magnitude and a direction

2.The blade velocity U is invariably tangential to the circular path of the blade, with positive direction in the direction of rotation. The choice of the point of the tangent can be anywhere on the circular path, but it is taken at the uppermost point to keep U horizontal, as shown in Fig.

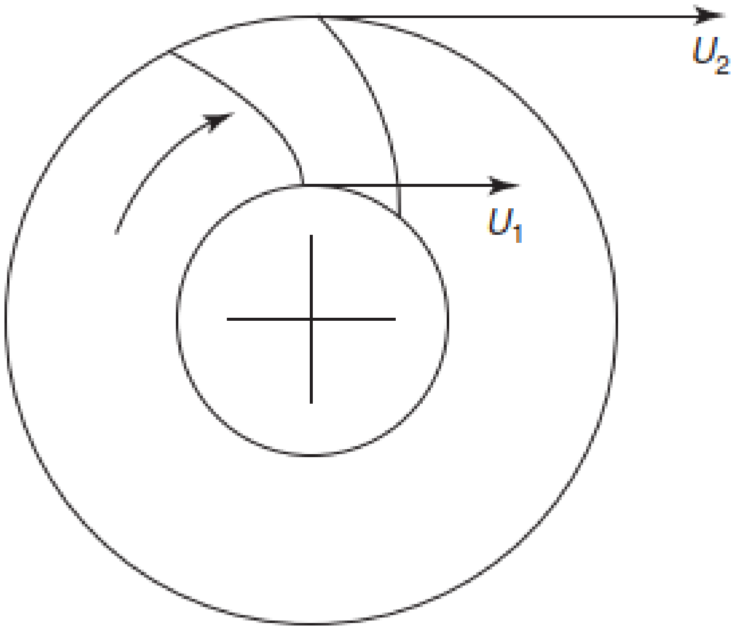
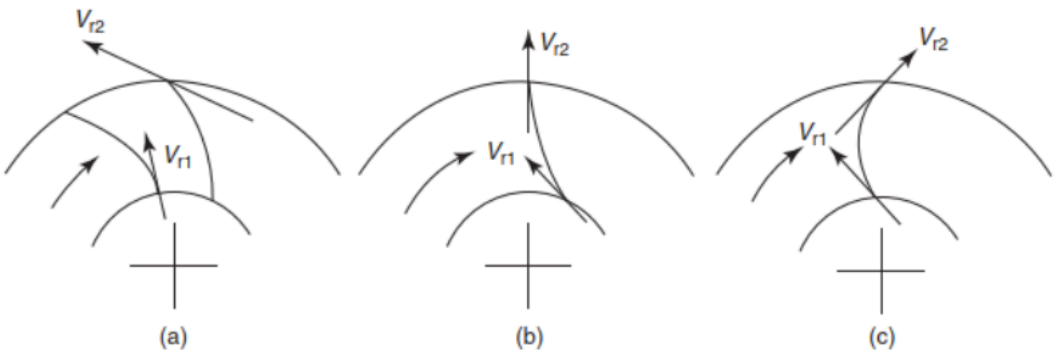


Image 1 Blade velocities.

3. The instantaneous relative velocity of the fluid (Vr) with respect to the blades is always tangential to the stream line of the flow of the fluid in the passage between the blades of the rotor. Vr is tangential to the blade profile.



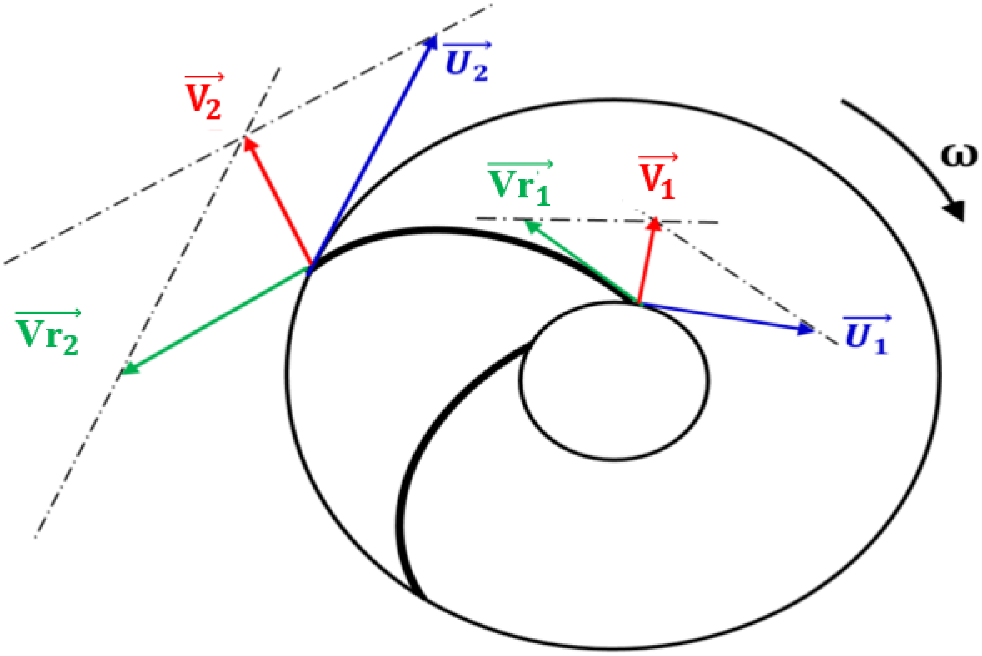
Relative velocities: (a) Blades are bent backwards. (b) Blades are radial at the outlet.(c) Blades are bent forward.

⚠ Warning

The three velocities form a triangle only when they satisfy the condition V=U+Vr. This means that the fluid smoothly glides over the blade surface without impact or separation.

4.1. Velocity Triangles representation representatio

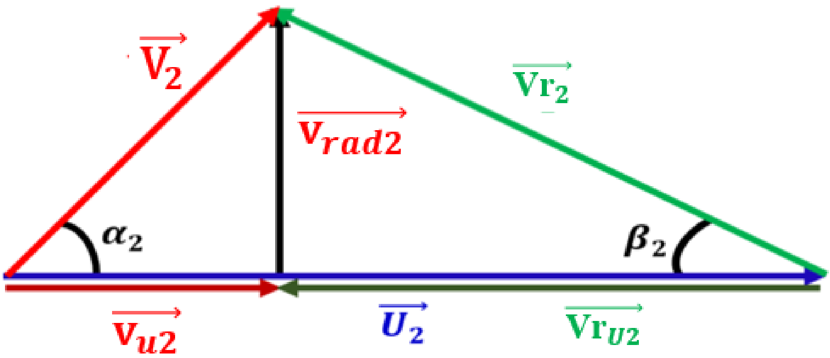
The following figure shows a wheel of a turbomachine on which the velocity vectors are drawn (at the input "index 1" and at the output "index 2")



Velocity Triangles

The angle α (angle of stalling) is formed by the velocities U and V

the angle β (angle of construction) is formed by the velocities U and Vr.



Velocity Triangles

In what follows, two more components of absolute velocity must be involved:

- A radial component:

$$V_{rad} = V \bullet sin\alpha \tag{3.2}$$

- A circumferential component:

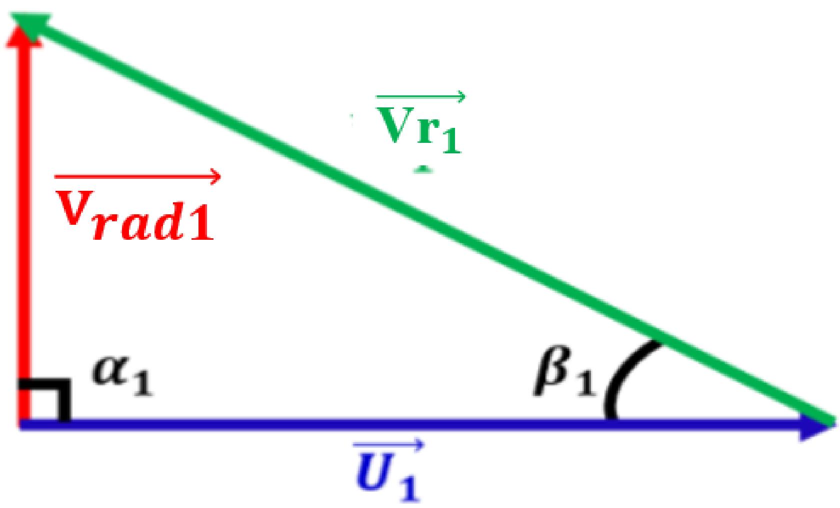
$$V_u = V \bullet cos\alpha \tag{3.3}$$

The component Vrad can be determined using the continuity equation:

$$V_rad = \frac{Q_v}{S} = \frac{Q_v}{\pi Db} \tag{3.4}$$

For a radial inlet turbomachine, the absolute speed is perpendicular to the drive speed and equal to its radial component since the tangential component is zero.

(V = Vrad ,α=90°).



Velocity Triangles

4.2. Euler's Theorem

The starting point for studying turbomachinery is **Euler's equation**. This equation can be easily derived from the principle of conservation of angular momentum or moment of momentum. Specifically, we consider a one-dimensional steady flow in the rotor of a turbomachine with uniform conditions at the inlet and outlet, denoted by indices 1 and 2, respectively.

"**The moment of momentum equation**:"

$$M = Q_m(r_2v_2 - r_1v_1) \tag{3.5}$$

The elegant mathematical form of Euler's equation requires some modifications to make it easier to use.

In turbomachinery,$r \cdot v = r \cdot C_u$

Equation (3.5) becomes:

$$M = Q_m(r_2C_{u2} - r_1C_{u1}) \tag{3.6}$$

The power absorbed by the pump is determined by:

$$P = M \cdot \omega = Q_m(r_2C_{u2}\omega - r_1C_{u1}\omega) \tag{3.7}$$

Given that the tangential velocity U can be expressed as U=r·ω equation (3.7) can be rewritten as:

$$P = Q_m(C_{u2}U_2 - C_{u1}U_1) \tag{3.8}$$

The power absorbed by the pump can also be determined as:

$$P = Q_m \cdot g \cdot H_{th} \tag{3.9}$$

By equating equations (3.8) and (3.9), we obtain **Euler's equation**:

$$H_{th} = \frac{U_2C_{u2} - U_1C_{u1}}{g} \tag{3.10}$$

For radial-inlet turbomachinery, $C_{u1} = 0$ $\alpha_1 = 90°$. Consequently, Euler's equation simplifies to:

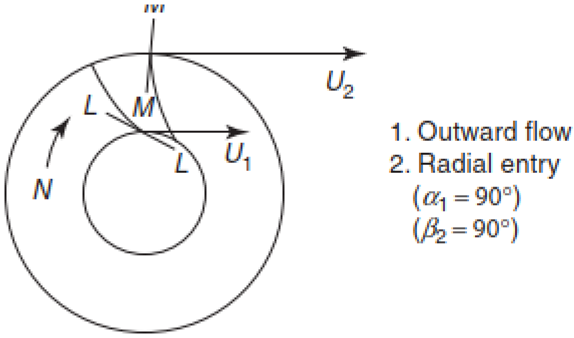
$$H_{th} = \frac{U_2C_{u2}}{g} \tag{3.11}$$

4.3. Practice

The purpose of the following examples is only to practice the drawing of velocity triangles. The utility of velocity triangles is yet to be discussed. However, it can be said that the study of turbomachines becomes very simple when velocity triangles are clearly understood.

Example 01

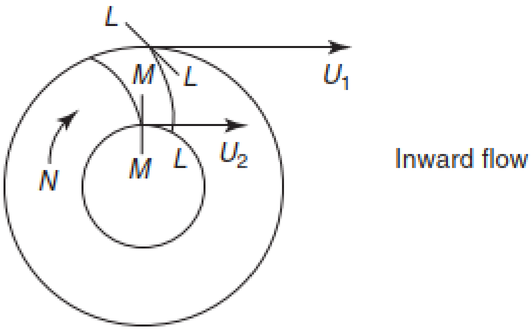
A rotor is shown in Fig. 3.14. The entry of fluid is radial and the flow is outward. The blades are radial at the outlet. Draw representative velocity triangles.



Rotor of Example

Example 02

A rotor is shown in Fig. 3 The entry of fluid is at an angle and the flow is inward. The blades are radial at the inner diameter. Draw representative velocity triangles.



Rotor of Example

5. Power and efficiency

5.1. Energy of Fluids

The fluids have different forms of energy: pressure energy, kinetic energy, potential energy, and thermal energy. The dimensions of the different forms of the energy are now considered as follows:

Pressure Energy

Kinetic Energy

Kinetic Energy

Potential Energy

Thermal Energy or Enthalpy

A fluid can have different forms of energy as discussed above and its total energy is the sum of the individual forms of energy.

Essentially, all the forms must be in the same dimensions to make it possible to add them together.

Total energy = Pressure energy + Kinetic energy + Potential energy

a) Pressure Energy

The pressure energy of a fluid at a pressure p (s N/m^2) and at a specific volume v (m^3/kg) is given by the product p v.

This expression is also written in the form p/ρ where ρ is the density, kg/m^3, whenever required.

b) Kinetic Energy

The kinetic energy of a fluid moving at a velocity of V m/s is given by $V^2/2$.

c) Potential Energy

The potential energy of a fluid at a height z (above a given datum level) is given by z g.

d) Thermal Energy or Enthalpy

It is a usual practice to refer to the energy of a gaseous fluid (combustion gas, steam, or air) in terms of its "enthalpy."

Enthalpy is a combined effect of pressure and temperature of a gas. For compressible fluids, enthalpy can be taken as a function of temperature. Changes in this form of energy, therefore, are calculated as $\Delta h = Cp \,.\, \Delta T$

> 💬 Note
>
> The equation $\Delta h = Cp \,.\, \Delta T$ above holds good when the specific heat cp remains constant.
>
> When the fluid is air, the specific heat can be taken as constant over the ranges of temperature changes that are usually encountered in turbomachines.
>
> When the fluid is steam, the enthalpies have to be obtained from the steam tables.

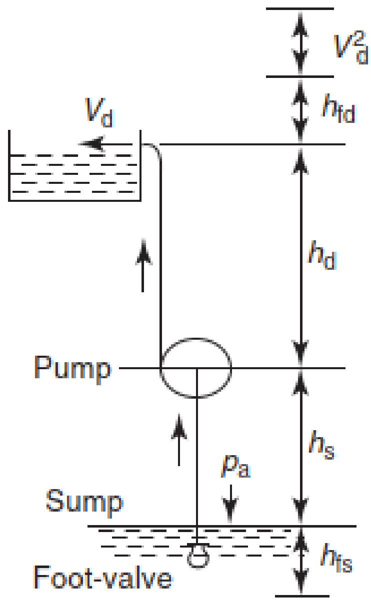
e) Practice

Examples

Calculate the energy of a stream of water at a pressure of 200 kPa, flowing at 8 m/s at an elevation of 5 m above a given datum. Take the density of water as 1000 kg/m^3.

5.2. Different Heads of Centrifugal Pumps

The study of a centrifugal pump is naturally with respect to the pumping of liquids. Therefore, the unit of energy, in general, is a column of liquid, that is, meters of liquid. Hence, the output of the centrifugal pump is in terms of meters of liquid or head units. With this background, various heads associated with an installation of a centrifugal pump are identified and these heads are as follows (shown in Fig). All the heads are in meters of liquid or meters of water.



Schematic layout of a centrifugal pump showing various heads.

1. h_s is suction head, that is the vertical distance from the sump level to the centerline of the pump.
2. h_d is delivery head, that is the vertical distance from the centerline of the pump to the exit point of the delivery pipe.
3. h_{fs} is frictional loss of head in the suction pipe, including the loss due to the Foot-valve, other pipe fittings such as bends, etc.

   h_{fd} is frictional loss of head in the delivery pipe, including the loss due to the delivery valve, other pipe fittings, etc.
4. $V_d^2/2g$ is loss of head equivalent to the exit velocity.

Further, a head known as "manometric head, Hm" is defined as the head measured across the inlet and outlet of the pump.

The manometric head represents the output of the pump. (The name might have been due to a manometer connected across the inlet and outlet of the pump, when the head generated was much less, sometime during the evolution of the pumps.)

Now the pressure gauges are being mounted, one each at the inlet and outlet of the pump. The difference in their readings, suitably converted, is the manometric head.

Because the pump is required to work against the total of the static head, friction head, and delivery velocity head, one can write:

$$H_m = h + h_f + \frac{V_d^2}{2g} \tag{3.12}$$

$$H_m = h_s + h_d + h_f s + h_f d + \frac{V_d^2}{2g} \tag{3.13}$$

The head developed by the impeller is given by the Euler turbine equation, where the whirl component Vu1 is taken as zero. Hence:

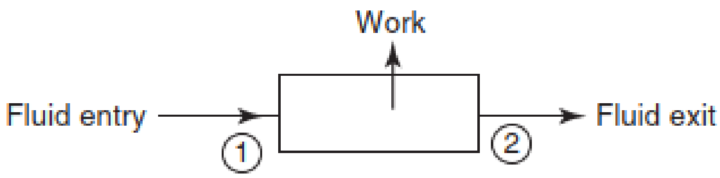
$$H_E = \frac{W_E}{g} = h + h_f + \frac{U_2 V_{u2}}{g} \tag{3.14}$$

The difference between Hm and HE is the loss due to fluid friction, wall friction, eddies, etc. in the impeller.

a) Application of the First Law of Thermodynamics

The first law of thermodynamics gives rise to the steady flow energy equation (SFEE) with a set of assumptions. These assumptions realistically hold good to a turbomachine because for a turbomachine, the inlet

or outlet conditions do not vary over the time and there is no depletion or accumulation of mass in the machine as the process is continuous. The heat transfer from or to a turbomachine is taken as negligible.



Schematic block diagram of a turbomachine.

Considering the turbomachine shown in Fig, one can write:

$$\text{Work output of the turbomachine} = \text{Total fluid energy at inlet } - \text{Total fluid energy at outlet} \quad (3.15)$$

$$\Longrightarrow W = E_1 - E_2 \quad (3.16)$$

On the basis of unit mass flow rate,

$$W = e_1 - e_2 \quad (3.17)$$

$$= \left(h_1 + \frac{V_1^2}{2g_c} + z_1\frac{g}{g_c}\right) - \left(h_2 + \frac{V_2^2}{2g_c} + z_2\frac{g}{g_c}\right) \quad (3.18)$$

$$= (h_0)_1 + z_1\frac{g}{g_c} - (h_0)_2 + z_2\frac{g}{g_c} \quad (3.19)$$

Also,

$$W = (h_1 - h_2) + \frac{V_1^2}{2g_c} - \frac{V_2^2}{2g_c} + (z_1 - z_2)\frac{g}{g_c} \quad (3.20)$$

the power communicated to the fluid or useful power

When the flow rate is $\dot{m}$ kg/s, the corresponding power of the machine is given by:

$$P\left(\frac{\mathrm{J}}{\mathrm{s}}\right) = W\left(\frac{\mathrm{J}}{\mathrm{kg}}\right) \times \dot{m}\left(\frac{\mathrm{kg}}{\mathrm{s}}\right) \quad (3.21)$$

All the units are J/kg in the above equations. The quantity on the left-hand side, W, is the “specific work” of a turbomachine. It is the work interaction of 1 kg of fluid while flowing over the rotor of the turbomachine.

When the flow rate is $\dot{m}$ kg/s, the corresponding power of the machine is given by $P\left(\frac{\mathrm{J}}{\mathrm{s}}\right) = W\left(\frac{\mathrm{J}}{\mathrm{kg}}\right) \times \dot{m}\left(\frac{\mathrm{kg}}{\mathrm{s}}\right)$

where P is in watts. W and P are positive when the fluid has higher energy at the inlet than at the outlet.

These are “power machines” producing motive power to run other machines. Mathematically, the equations hold good for work-absorbing machines, such as pumps, fans, compressors, etc. The fluid energy at the outlet of these machines is more than the energy at the inlet and the specific work becomes negative in the equation. This signifies that power is the input to such turbomachines.

When the fluid is incompressible (namely, liquids), the enthalpy terms are replaced by more appropriate terms of pressure energy.

In such cases, the density (ρ) is employed instead of specific volume. The density (ρ) is invariable $(\rho_1 = \rho_2)$ over a large range of pressures.

The expression for specific work when the fluid is incompressible becomes:

$$W = \left(\frac{p_1}{\rho} + \frac{V_1^2}{2g_c} + z_1\frac{g}{g_c}\right)\left(\frac{p_2}{\rho} + \frac{V_2^2}{2g_c} + z_2\frac{g}{g_c}\right) \tag{3.23}$$

Also:

$$W = \frac{p_1 - p_2}{\rho} + \frac{V_1^2 - V_2^2}{2g_c} + (z_1 - z_2)\frac{g}{g_c} \tag{3.24}$$

Simplified further, we get:

$$W = \frac{1}{\rho}\left(p_1 + \rho\frac{V_1^2}{2g_c} - p_2 - \rho\frac{V_2^2}{2g_c}\right) + (z_1 - z_2)\frac{g}{g_c} \tag{3.25}$$

And:

$$W = \frac{1}{\rho}(p_0 - p_0') + (z_1 - z_2)\frac{g}{g_c} \tag{3.26}$$

The power is given by:

$$P = W \times \dot{m} \tag{3.27}$$

Now, since the fluid is incompressible, the head energy terms can also be used. However, it is suggested that one has to be more cautious in using the expression “head-generated” in place of “power-generated” or power output. The terms are somewhat opposite to each other in their signs. If the “head” is the output, obviously the power or work is the input. This corresponds to the turbomachine as a power-consuming machine, using mechanical power and “lifting” or pumping the liquid through a height or the “head” of that liquid. If the “head” is the input, then power or work is the output. This corresponds to a turbine, worked by the liquid head and giving out mechanical power as output. Using H (for head) in place of W, the signs of the quantities are changed, and Eq. (1.5) becomes

$$H = \frac{p_2 - p_1}{w} + \frac{V_2^2 - V_1^2}{2g} + (z_2 - z_1) \tag{3.28}$$

All units are meters of liquid.

this equations can be written as:

$$gH = \frac{p_2 - p_1}{\rho} + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1) \tag{3.29}$$

In continuation:

$$W = gH \tag{3.30}$$

$$P = W \times \dot{m} \tag{3.31}$$

$$P = \rho Q_v g H \tag{3.32}$$

$$P = \frac{\mathrm{N}}{\mathrm{m}^3} \times \mathrm{m}^3/\mathrm{s} \times \mathrm{m} = \frac{\mathrm{m} \cdot \mathrm{N}}{\mathrm{s}} = \frac{\mathrm{J}}{\mathrm{s}} = \mathrm{W} \tag{3.33}$$

where Q is the volume rate of flow of the liquid(m^3/s) through the turbomachine.

In Eqs. (1.2) to (1.7), subscripts 1 and 2 represent the inlet and outlet points of the turbomachine. Thus, z_1 and z_2 represent the heights of the inlet and outlet, respectively, above a common datum. Equations (1.2)--(1.7) represent the general form. There are a few instances where z_1 and z_2 are really different, but in a large number of cases, z_1 and z_2 are equal.

5.3. Different Efficiencies of a Centrifugal Pump

Any efficiency, in general, can be written as output divided by input. It can also be written as Efficiency= (Input - Losses) / Input

a) Mechanical Efficiency

The mechanical efficiency (η_m) has been given as η_m= Rotor power / Shaft power

$$\eta_m = \frac{P_r}{P_s} \tag{3.34}$$

Here, the shaft power is the input power supplied by the motor. Due to mechanical losses at the bearings, the impeller or rotor gets slightly less power. The mechanical efficiencies of the pumps are of the order of 98–99%. Higher capacity pumps have higher mechanical efficiencies.

b) Volumetric Efficiency

The hydraulic efficiency (η_v) has been given as

$$\eta_v = \frac{\dot{m}}{\dot{m} + \Delta \dot{m}} \tag{3.35}$$

where m is the mass flow rate, or equivalently, the volume flow rate, that is, the designed and intended flow rate to be available at the delivery end of the pump and Δm is the extra flow rate that goes through the impeller, absorbing extra power.

The extra flow rate is the sum of the leakages at the glands and the return flow that occurs between the casing and the shrouds of the impeller, from its outlet end to the suction end.

The values of volumetric efficiencies are of the order of 94–96%.

c) Hydraulic Efficiency or Manometric Efficiency

The hydraulic efficiency (η_h) has been given as

$$\eta_h = \frac{P}{P_n} = \frac{wQH}{wQH_n} = \frac{H}{H_n} \tag{3.36}$$

where H is the head that is finally overcome by the pump, namely, the manometric head Hm given by

$$H = H_m = h + h_f + \frac{V_d^2}{2g} \tag{3.37}$$

The denominator Hn in Eq. is the head developed by the impeller. This is equivalent to the specific work as per the Euler turbine equation:

$$H_n = H_E = \frac{W_E}{G} = \frac{U_2 V_{U2}}{g} \tag{3.38}$$

The difference between Hm and HE is due to (a) the fluid friction because of viscosity, (b) fluid-rotor friction, and (c) non-ideal flow of liquid in the impellers. Hence

$$\eta_h = \frac{gH_m}{U_2 V_{U2}} \tag{3.39}$$

The hydraulic and manometric efficiencies are the same

$$\eta_{mano} = \frac{gH_m}{U_2 V_{U2}} \tag{3.40}$$

The manometric efficiency depends on the viscosity of liquids being pumped, the surface finish of the blades

and shrouds, etc. The usual values of manometric efficiencies are of the order of 85–94%.

d) Overall Efficiency

The overall efficiency has been given as

$$\eta_o = \eta_h * \eta_V * \eta_m \tag{3.41}$$

the overall efficiency can also be written as

$$\eta_o = \eta_{mano} * \eta_V * \eta_m \tag{3.42}$$

The overall efficiency is also known as the gross efficiency or actual efficiency.

6. Quiz: Practice Exercise

A water treatment plant uses a booster pump to transfer water from a storage tank at Point A to a filtration tower at Point B.

- **Point A:** The storage tank is located 5 meters below the pump's installation level, with water at a pressure of PA=1.2 bar.
- **Point B:** The filtration tower is located 25 meters above the pump, with water exiting at a pressure of PB=1 bar.
- The pump discharges water at a flow rate of Qv=5000 L/h through a pipeline with an internal diameter D=25 mm.
- The pipe has a total head loss due to friction Hf=6.0 m.
- The pump is powered by an electric motor with a system efficiency of η=85%.

Given:

- Water density: $\rho = 1000\,\mathrm{kg/m^3}$
- Gravitational acceleration: $g = 9.81\,\mathrm{m/s^2}$

Quiz

Calculate the **mass flow rate** of water delivered by the pump.

Quiz

Determine the **velocity of water** in the pipeline.

Quiz

Compute the **total head** (Ht) required by the pump, considering static, pressure, frictional, and velocity heads.

Quiz

Using the total head, calculate the **power required by the pump**.

Quiz

Calculate the **electrical power consumption** of the pump.

Quiz

If the pump operates for 8 hours daily, calculate the **total energy consumption (in kWh)** for one week.

Conclusion

In this chapter, we delved into the principles and operation of pumps, focusing on **centrifugal pumps** and **axial pumps**, two critical types used in a wide range of applications. The discussion included an analysis of **velocity triangles**, a key tool for understanding fluid dynamics within pumps, and their role in predicting pump performance.

We also examined the concepts of **power and efficiency**, highlighting their importance in designing and optimizing pump systems for maximum performance and minimal energy loss.

VIII Did you understand the basics about Pumps

Quiz 1: Exercise 01

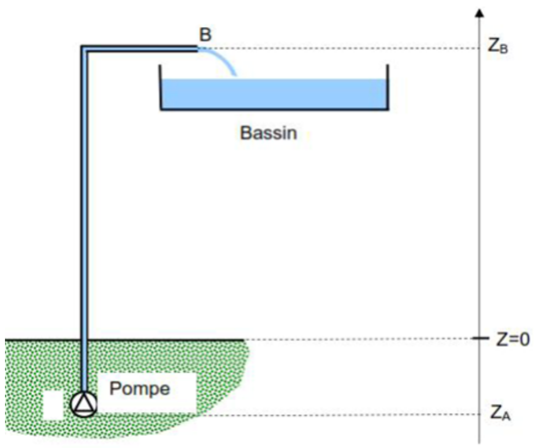
We aim to fill a basin by pumping water from the groundwater. For this, a submersible pump is used, which draws water from point A, located at an altitude $Z_A = -26\,\mathrm{m}$. The pressure at point A is $P_A = 2\,\mathrm{bar}$.

The water discharged by the pump is conveyed through a circular pipe with an inner diameter $d = 31\,\mathrm{mm}$. The water exits at point B, located at an altitude $Z_B = 30\,\mathrm{m}$, with a volumetric flow rate $q_v = 2772\,\mathrm{L/h}$. The pressure at point B is $P_B = 1\,\mathrm{bar}$.

The pump is powered by an electric motor. The efficiency of the pump-motor system is $\eta = 80\%$.

Assume that the fluid is ideal and the inlet velocity equals the outlet velocity ($V_A = V_B = V$)

- Water density: $\rho = 1000\,\mathrm{kg/m^3}$
- Gravitational acceleration: $g = 9.81\,\mathrm{m/s^2}$.



Quiz

1. Calculate the mass flow rate (q_m) of the pump.

Quiz

2. Determine the flow velocity (V) of water.

Quiz

3. Using Bernoulli's theorem, calculate the power supplied by the pump (P_pump).

Quiz

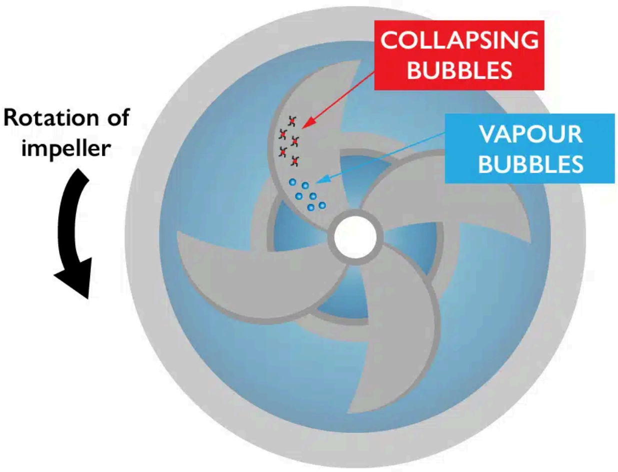
4. Calculate the electrical power consumed $P_{\text{electrical}}$

IX Chapter 04: Cavitation in pumps

Introduction

Pump cavitation is a persistent issue that affects many industries around the world. Today we will explore its causes, effects, and best mitigation strategies.

Our goal is to equip you with the knowledge needed to safeguard your pumping systems against the destructive forces of cavitation.



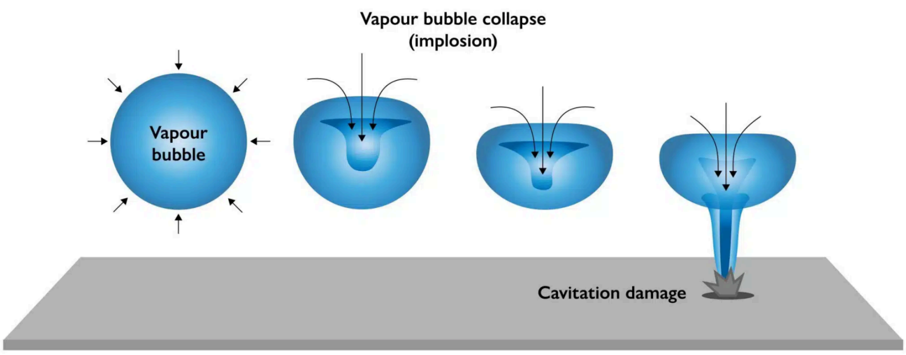
Cavitation in pumps

1. What is Pump Cavitation?

Az Definition

Pump cavitation is defined as the formation and collapse of vapour bubbles within a liquid due to low pressure conditions.

Its impact can have detrimental effects on pump performance, efficiency, and longevity. Therefore, understanding the mechanics of cavitation is crucial for maintaining the reliability and effectiveness of pumping equipment.



Cavitation in pumps

2. What Causes Pump Cavitation?

2.1. Vapour Pressure

As temperature increases, so does the vapour pressure of the liquid. When the pressure at the inlet of a pump falls below the vapour pressure of the liquid being pumped, it creates conditions conducive to cavitation.

This occurs because the liquid is no longer able to withstand the reduced pressure, causing it to vaporise and form bubbles.

2.2. Inlet Conditions

Improper inlet conditions can significantly contribute to the occurrence of cavitation.

Low pressure at the pump inlet, often referred to as insufficient Net Positive Suction Head (NPSH), can lead to cavitation by causing the liquid to vaporise as it enters the pump.

Additionally, air entrainment (where air is mixed with the liquid) can exacerbate cavitation problems. Air pockets within the liquid can collapse violently, triggering cavitation even under conditions where the liquid's vapour pressure is not exceeded.

2.3. Liquid Properties

The properties of the liquid being pumped, such as viscosity and temperature, play a significant role in determining its susceptibility to cavitation.

Higher viscosity fluids are less prone to cavitation due to their greater resistance to vapour bubble formation and collapse. Conversely, lower viscosity fluids are more susceptible to cavitation, as they are more likely to vaporise under reduced pressure conditions.

2.4. Pump Design

Inadequate suction piping or excessive restrictions in the inlet flow path can create areas of low pressure, promoting cavitation.

Similarly, improper impeller design or clearance can lead to uneven flow patterns and pressure differentials within the pump, increasing the likelihood of cavitation occurrence.

Finally, pumps operating at off-design conditions (such as low flow rates or high speeds) may also experience cavitation due to the deviation from their optimal operating parameters.

3. What are the Effects of Pump Cavitation?

3.1. Mechanical Damage

Cavitation-induced pressure fluctuations within a pump can wreak havoc on its internal components, leading to extensive mechanical damage.

As vapour bubbles collapse near solid surfaces, they generate intense shock waves and high-speed liquid jets. These forces can cause erosion, pitting, and surface deterioration on impeller blades, pump casing, and other critical components.

Over time, this erosion can compromise the structural integrity of the pump and reduce its operational lifespan.

3.2. Performance Degradation

Beyond causing mechanical damage, cavitation can also impair the overall efficiency and performance of a pump system.

The formation and collapse of vapour bubbles disrupt the smooth flow of liquid through the pump, leading to flow instabilities and hydraulic inefficiencies.

As a result, the pump's capacity to deliver the desired flow rate and pressure is compromised, leading to reduced system performance.

3.3. Noise and Vibration

One of the most noticeable effects of cavitation is the generation of noise and vibration within the pump system.

As vapour bubbles collapse with tremendous force, they produce audible acoustic emissions that manifest as loud noise levels.

Moreover, cavitation-induced vibrations can propagate through the pump structure and surrounding piping, causing mechanical stress and fatigue.

4. How to Avoid Pump Cavitation

4.1. Suitable Pressure

Maintaining appropriate pressure levels within the pump system is critical for preventing cavitation. Ensuring that the pump operates within its designated pressure range helps avoid situations where the pressure at the inlet falls below the vapour pressure of the fluid.

This can be achieved through proper system design, including the use of pressure relief valves, throttling devices, and pressure sensors to regulate and monitor system pressure levels.

4.2. Inlet Improvements

Optimizing inlet conditions is crucial for preventing cavitation at the suction side of the pump.

This involves implementing strategies to maintain adequate Net Positive Suction Head (NPSH) and minimise air entrainment.

Proper suction piping design, including sufficient diameter and length, helps ensure a smooth and uniform flow of liquid into the pump, reducing the risk of pressure drops and cavitation.

Installing inlet devices such as vortex breakers and strainers can also help improve flow stability and remove entrained air, further mitigating cavitation risks.

4.3. Liquid Management

Managing liquid properties effectively can significantly reduce the likelihood of cavitation occurrence.

Controlling temperature to maintain the fluid within its operating range helps regulate vapour pressure and minimise cavitation risk.

Measures such as filtration and degassing can also help remove impurities and entrained air from the liquid, improving its cavitation resistance.

Finally, adjusting fluid viscosity through additives or temperature control can also mitigate cavitation effects by altering the fluid's behaviour under varying operating conditions.

4.4. Proper Pump Selection

Engineers must carefully consider factors such as flow rate, head, fluid properties, and operating conditions when choosing a pump.

It's essential to ensure that the selected pump can operate within the specified pressure and flow range without exceeding the fluid's vapour pressure.

Additionally, choosing pumps with robust design features and materials resistant to cavitation-induced damage can further enhance system reliability and longevity.

Conclusion

Pump cavitation is a critical issue that can significantly impact the performance, efficiency, and lifespan of pumping systems. Understanding its causes—such as low inlet pressure, improper design, and unfavorable liquid properties—is essential for mitigating its effects.

Cavitation can lead to mechanical damage, performance degradation, and excessive noise and vibration, all of which reduce the reliability and operational life of pumps. By optimizing pump design, maintaining proper inlet conditions, and selecting appropriate liquids for pumping, industries can effectively safeguard their systems from cavitation's destructive impact.

Proactive measures and regular maintenance are key to ensuring pump longevity and efficiency in the face of this common challenge.

X Did you understand the basics of Cavitation in Pumps ?

Quiz 1

Which of the following is a primary cause of pump cavitation?

- High suction pressure
- Low vapour pressure
- Low inlet pressure
- High flow rate

Quiz 2

What is the main effect of cavitation on pump performance?

- **A** Elevated noise levels
- **B** Improved flow rate
- **C** Mechanical damage to components
- **D** Reduced energy efficiency

Quiz 3

Which of the following can help prevent pump cavitation?

- Increasing pump speed
- Maintaining suitable inlet pressure
- Lowering fluid viscosity
- Using larger pump impellers

Quiz 4

What forms during cavitation in pumps due to low pressure?

- Solid particles
- Air pockets
- Liquid droplets
- Vapour bubbles

Quiz 5

Which liquid property is less prone to cavitation?

- High viscosity
- High density
- Low viscosity
- Low temperature

XI Chapter 05: Couplings of centrifugal pumps

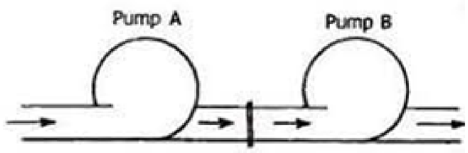
Introduction

The procedure of designing a pump starts from the specification of the pump in terms of the required flow rate and the head to be developed. The two factors give rise to the specific speed of the pump. Hence, the specific speed becomes the important parameter in the design of a pump.

However, there may be values of the head and flow rates that result in specific speeds which are outside the limits. If attempts are made to design pumps with such values of specific speeds, the designs may not be alright. Instead of such designs, it is useful to manipulate the design by using multiple units of standard (or available) designs. Two such schemes are the pumps-in-series and pumps-in-parallel.

1. Pumps-in-Series

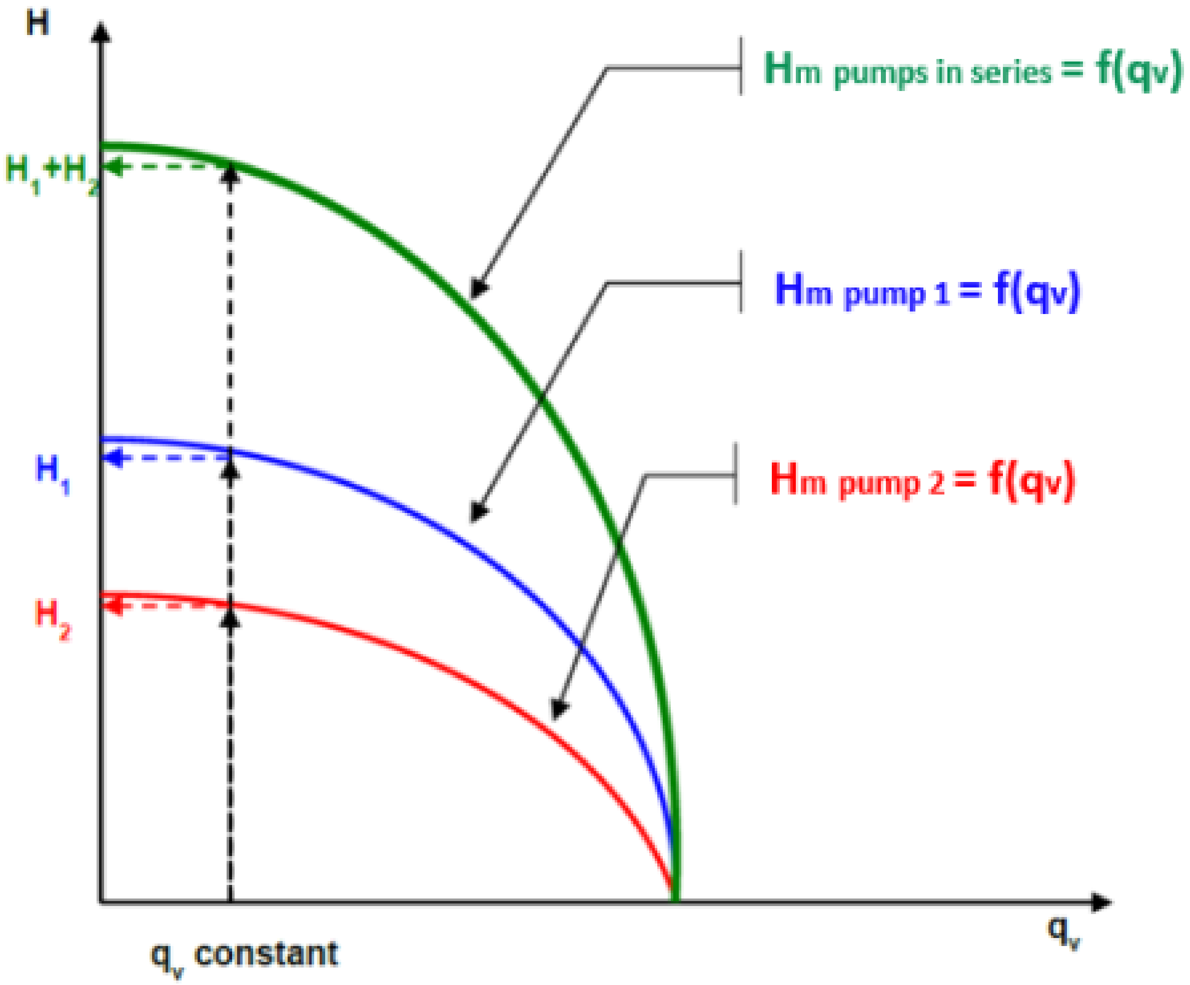
This method is to meet the requirements of high heads and is shown in Fig. The figure shows then schematic layout



the characteristics of two identical pumps in series. The resulting head, curve y, represents the values that are twice the head of a single pump, curve x. Pumps of different heads can also be operated in series, with a condition that the rated flow-capacities must be equal.

In such a case, the heads are additive. In the case of pumps-in-series, individual pumps cannot be stopped or withdrawn from the total system.

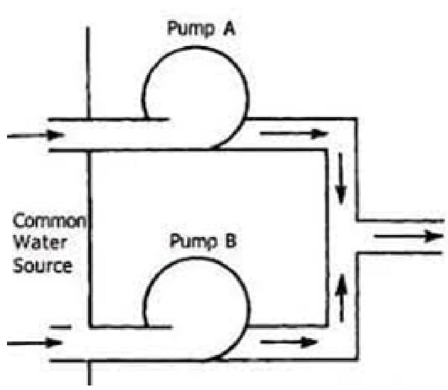
Multi-stage pumps with several impellers mounted on the same shaft and outlet from one impeller leading to the inlet of the next impeller are actually pumps-in-series. These are suitable for applications such as pumping of water to overhead tanks of high-rise buildings, boiler-feed pumps, deep-well pumps, pumps for oil rigs, etc.



(a) Schematic view of two identical pumps in series;(b) Characteristic of combined operation.

2. Pumps-in-Parallel

This method is to meet the requirements of large flow-rates and is shown in Fig.

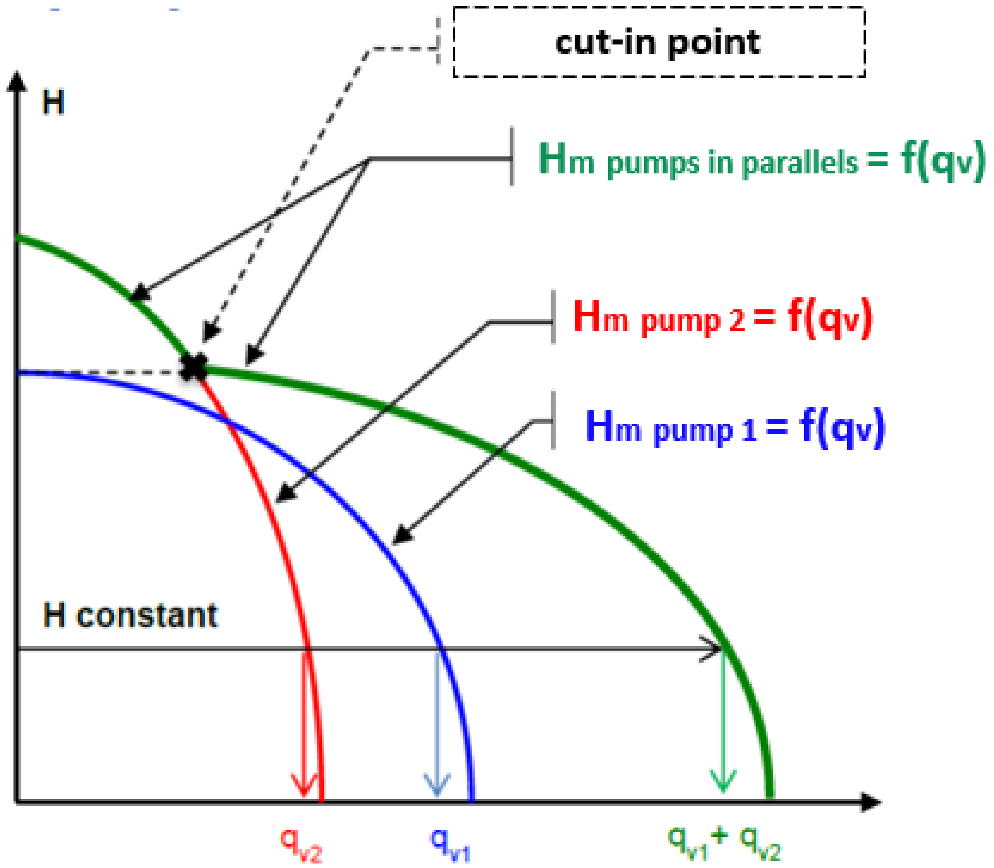


The schematic layout shows two identical pumps operating in parallel. The characteristic of the combined operation is also shown in Fig.

The resulting flow rate, curve y, represents the values that are twice the flow rate of a single pump, curve x.

Pumps of different flow rates can also be operated in parallel, with a condition that the heads must be equal.

The flow rates in such a case are additive. Individual pumps can be stopped or withdrawn from the system. The other pumps can continue the pumping.



(a) Schematic view of two identical pumps in parallel.(b) Characteristic of combined operation.

3. Conclusion

Pumps-in-series and pumps-in-parallel provide practical solutions for meeting specific design requirements:

- **Pumps-in-Series** increase the total head while maintaining the same flow rate, ideal for high-head applications like high-rise buildings and deep wells. All pumps must operate together.
- **Pumps-in-Parallel** increase the total flow rate while maintaining the same head, suitable for large capacity demands. Individual pumps can operate independently.

These configurations allow engineers to achieve desired performance efficiently without relying on impractical custom designs.

XII Did you understand the basics of Couplings of Centrifugal Pumps ?

Quiz 1

A number of centrifugal pumps are available, with specifications of $H_m = 10\,\mathrm{m}$

$Q = 50\ \mathrm{l/s},\ H_m = 10\,\mathrm{m},\ \text{and}\ N = 960\,\mathrm{rpm}$

Calculate the number of pumps required to meet a demand of $H_m = 30\,\mathrm{m}$ of water and a flow rate of 150 l/s Sketch the arrangement

Quiz 2

Pumps-in-series are primarily used for:

- Increasing flow rates
- Increasing head
- Reducing pressure
- Reducing pump speed

Quiz 3

What are the characteristics of pumps-in-parallel?

- **A** Additive flow rates
- **B** Increased total head
- **C** Suitable for large capacity demands
- **D** Independent pump operation

Quiz 4

What condition must be met for pumps-in-parallel operation?

- Identical head
- Identical flow rate

- Identical pipe diameter
- Identical specific speed

Quiz 5

What is a drawback of pumps-in-series?

- Limited flow rates
- Inability to operate independently
- Reduced head capacity
- Increased pipe friction

Quiz 6

What happens to the flow rate when pumps are operated in parallel?

- It remains constant
- It decreases
- It halves
- It doubles

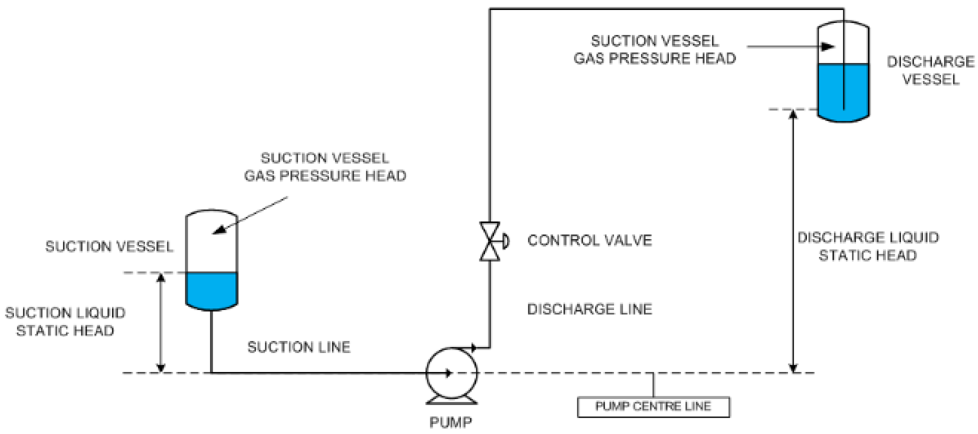
XIII Chapter 06: Sizing of centrifugal pumps

1. Basic Concepts

to choose a pump we need to know 2 basic values:

- The flow rate of liquid the pump is required to deliver
- The total differential head the pump must generate to deliver the required flow rate

Other key considerations for pump sizing are the net positive suction head available (NPSHa) and the power required to drive the pump.



Pump System Diagram

1.1. Flow Rate (Q)

The flow rate of liquid the pump is required to deliver.

Usually, the flow rate of liquid a pump needs to deliver is determined by the process in which the pump is installed.

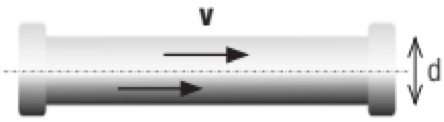
This ultimately is defined by the mass and energy balance of the process.

$$Q = S.v \tag{6.1}$$

S : area of the pipe $\pi.(D/2)^2[m^2]$.

v: speed of liquid (water) in the pipe $[m/s]$

Q: the quantity of liquid (water) flowing trough the pipe in a certain timespan $[m^3/s]$.



Pipe

1.2. Total Differential Head (H)

The total differential head the pump must generate to deliver the required flow rate

Essentially, the total differential head is made up of 2 components. The first is the static head across the pump and the second is the frictional head loss through the suction and discharge piping systems.

Total differential head = static head difference + frictional head losses

a) Frictional Head Losses

The total frictional head losses in a system are comprised of the frictional losses in the suction piping system and the frictional losses in the discharge piping system.

Frictional head losses = frictional losses in suction piping system + frictional losses in discharge piping system

The frictional losses in the suction and discharge piping systems are the sum of the frictional losses due to the liquid flowing through the pipes, fittings and equipment.The frictional head losses are usually calculated from the Darcy-Weisbach equation using friction factors and fittings factors to calculate the pressure loss in pipes and fittings.

Darcy-Weisbach equation:

$$\text{Frictional head losses} = (f \times \frac{L}{D} + K_{\text{fittings}})\frac{u^2}{2g} \tag{6.2}$$

Where:

f=pipe friction factor

L=pipe length

D=pipe diameter

$K_{fittings}$ =fittings factor

u=liquid velocity

g=acceleration due to gravity

In order to calculate the frictional head losses you therefore need to know the lengths and diameters of the piping in the system and the number and type of fittings such as bends, valves and other equipment.

b) Static Head Difference

The static head difference across the pump is the difference in head between the discharge static head and the suction static head.

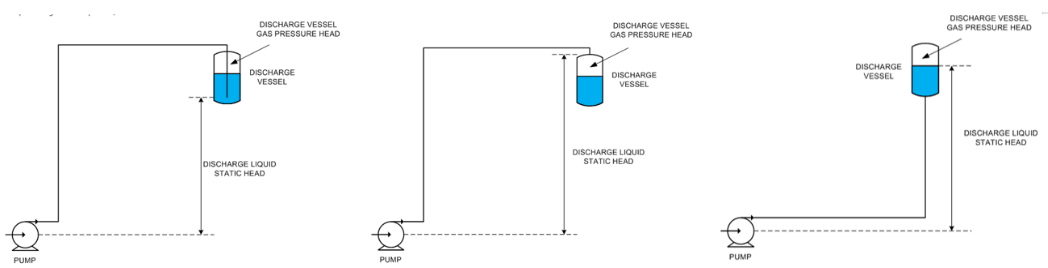
Static head difference = discharge static head – suction static head

i) Discharge Static Head

The discharge static head is sum of the gas pressure at the surface of the liquid in the discharge vessel (expressed as head rather than pressure) and the difference in elevation between the outlet of the discharge pipe, and the center line of the pump.

Discharge static head = Discharge vessel gas pressure head + elevation of discharge pipe outlet – elevation of pump center line

The discharge pipe outlet may be above the surface of the liquid in the discharge vessel or it may be submerged as shown in these 3 diagrams. (*Pump Discharge Above Liquid Surface/ Submerged Pump Discharge Pipe/ Discharge Pipe Enters The Bottom Of The Vessel)*



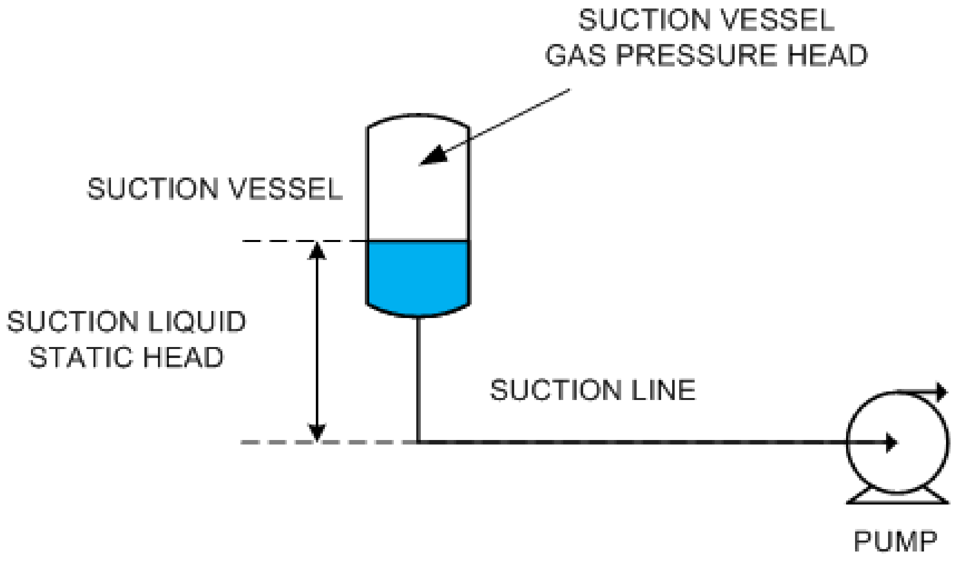
The discharge pipe outlet

ii) Suction Static Head

The suction static head is sum of the gas pressure at the surface of the liquid in the suction vessel (expressed as head rather than pressure) and the difference in elevation between the surface of the liquid in the suction vessel and the center line of the pump.

Suction static head = Suction vessel gas pressure head + elevation of suction vessel liquid surface – elevation of pump center line

Note: gas pressure can be converted to head using: Gas head = gas pressure ÷ (liquid density x acceleration due to gravity)



Pump Suction

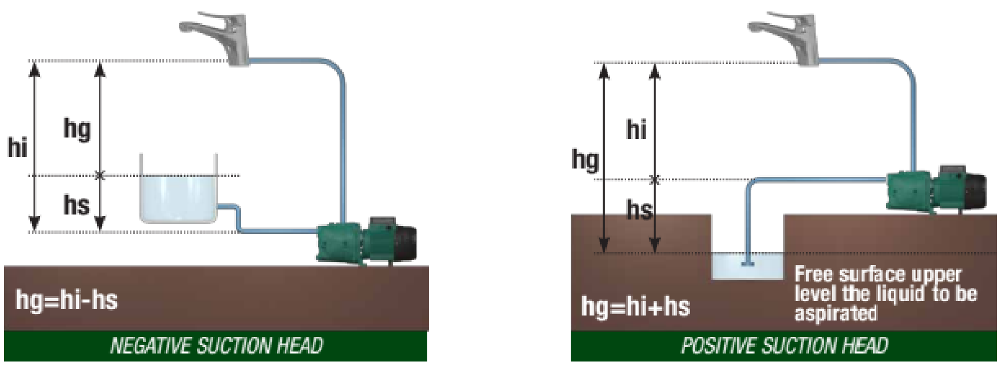
1.3. Pump Power

Pumps are usually driven by electric motors, diesel engines or steam turbines. Determining the power required is essential to sizing the pump driver.

Pump power = flow rate x total differential head x liquid density x acceleration due to gravity ÷ pump efficiency

2. Net Positive Suction Head (NPSH)

Anyone who has sized a pump should be familiar with the term net positive suction head (NPSH). Simply put, pump NPSH is the excess head (or pressure) exerted on the pump's suction that keeps the liquid from boiling.



NEGATIVE SUCTION HEAD VS POSITIVE SUCTION HEAD

NPSH stands for *Net Positive Suction Head* and is a measure of the pressure experienced by a fluid on the suction side of a centrifugal pump. It is quoted as a *Head* (in metres) rather than as an actual pressure (Pa) because 'head' is a fluid-independent property: a pump will lift different fluids to the same height irrespective of their densities.

NPSH is defined as the total head of fluid at the centre line of the impeller less the fluid's vapour pressure. The purpose of NPSH is to identify and avoid the operating conditions which lead to vaporisation of the fluid as it enters the pump – a condition known as*flashing*. In a centrifugal pump, the fluid's pressure is at a minimum at the eye of the impeller. If the pressure here is below the vapour pressure of the fluid, bubbles are formed which pass on through the impeller vanes towards the discharge port. As the bubbles of vapour are transported into this higher pressure region, they can spontaneously collapse in a damaging process called *cavitation*(Figure). The repeated shock waves produced by this process can be a significant cause of wear and metal fatigue on impellers and pump cases.

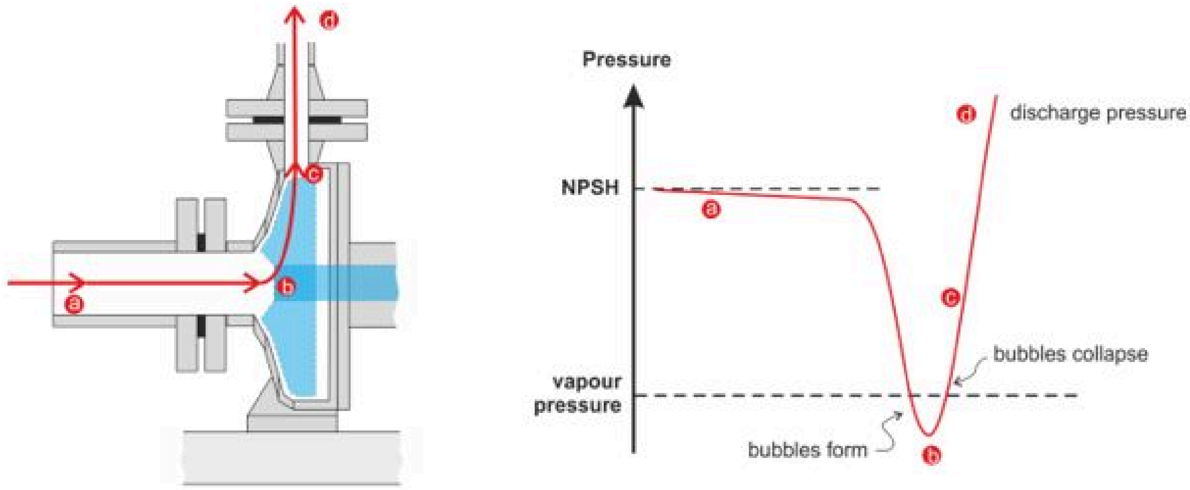


Figure 1. Pressure gradient through a centrifugal pump experiencing cavitation: fluid enters the pump (a); pressure drops below vapour pressure at impeller (b), pressure rises as fluid passes out to discharge (d) and bubbles condense and collapse (c).

NPSH is normally considered in two forms: NPSH-R (NPSH Required) and NPSH-A (NPSH Available).

2.1. Net Positive Suction Head Required (NPSHr)

NPSH-R is a pump property.*Net Positive Suction Head Required* is quoted by pump manufacturers as a result of extensive testing under controlled conditions. NPSH-R is a minimum suction pressure that must be exceeded for the pump to operate correctly and minimise flashing and cavitation.

a) How is NPSH-R measured?

Manufacturers test pumps under conditions of constant flow and observe the discharge pressure (differential head) as NPSH (the suction pressure) is gradually reduced. Tests are usually performed with water at 20°C. NPSH-R is defined as the value at which the discharge pressure is reduced by 3% because of the onset of cavitation (Figure). NPSH-R is sometimes shown as $NPSH_3$ or $NPSH_{3\%}$to highlight this fact. For multistage pumps, only the first stage is taken into consideration for determining the 3% pressure drop.

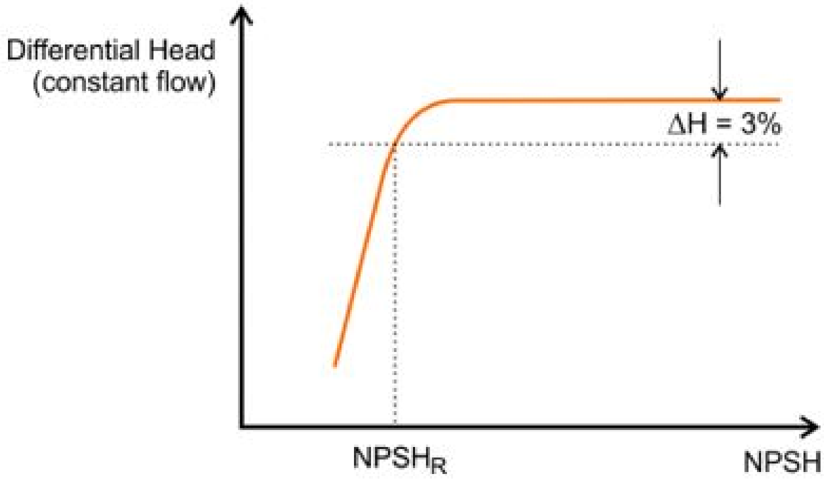


Figure 2. Determining NPSH-R for a given flow.

2.2. Net Positive Suction Head available (NPSHa)

The net positive suction head available (NPSHa) is the difference between the absolute pressure at the pump suction and the vapour pressure of the pumped liquid at the pumping temperature.

It is important because for the pump to operate properly, the pressure at the pump suction must exceed the vapour pressure for the pumped fluid to remain liquid in the pump.

To ensure that the pump operates correctly the net positive suction head available (NPSHa) must exceed the net positive suction head required (NPSHr) for that particular pump. The NPSHr is given by the pump manufacturer and is often shown on the pump curve.

Normally, a safety margin of 0.5 to 1m is required to take account of this and other factors such as:

- The pump's operating environment – is the temperature constant?
- Changes in the weather (temperature and atmospheric pressure).
- Any increases in friction losses that may occur occasionally or gradually during the lifetime of the system.

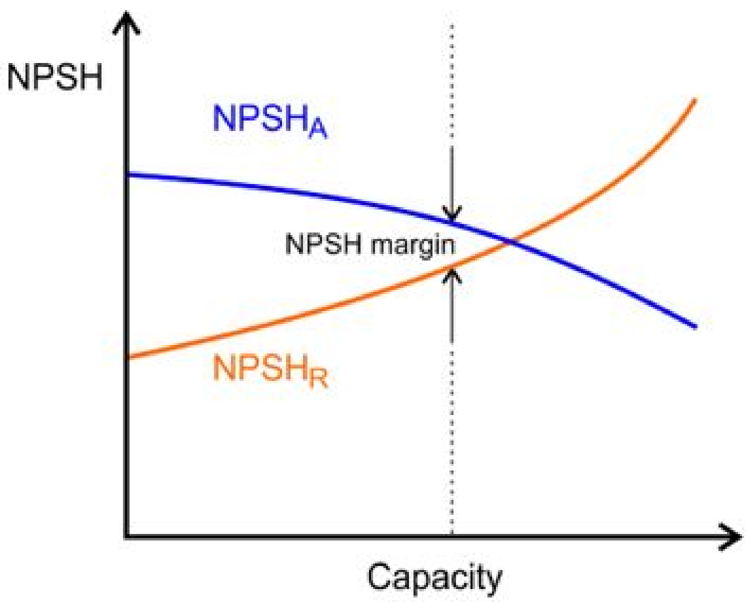


Figure 3. Variation of NPSH-R and NPSH-A with capacity (discharge flow)

a) How is NPSH-A calculated?

NPSH-A is calculated from the suction side configuration taking into account friction losses and the vapour pressure of the pumped fluid:

$$NPSHA = \frac{P_{\text{atm}} - P_{\text{vapor}}}{\rho g} + h_s - h_f \tag{6.3}$$

Where:

- P_{atm}= Atmospheric pressure at the pump suction (Pa)
- P_{vapor}= Vapor pressure of the fluid (Pa)
- ρ = Fluid density (kg/m³)
- g = Gravitational acceleration (9.81 m/s²)
- h_s = Static head in the suction line (meters)
- h_f = Frictional head loss in the suction line (meters)

3. How to Read Pump Curves for Centrifugal Pumps?

Centrifugal pumps are widely used in various industries, from water supply to chemical processing. A pump curve is a graphical representation of a centrifugal pump's performance characteristics, which includes flow rate, head, and power consumption. Understanding pump curves is essential in selecting the right pump for an application and optimizing its performance. In this article, we will discuss how to read pump curves for centrifugal pumps and how to use them to choose the right pump for your application.

3.1. What is a Pump Curve?

A pump curve is a graphical representation of a centrifugal pump's performance characteristics. It shows the relationship between the flow rate (Q), head (H), and power consumption (P) of a pump at different operating points. The pump curve is generated by testing the pump in a laboratory and plotting the data on a graph.

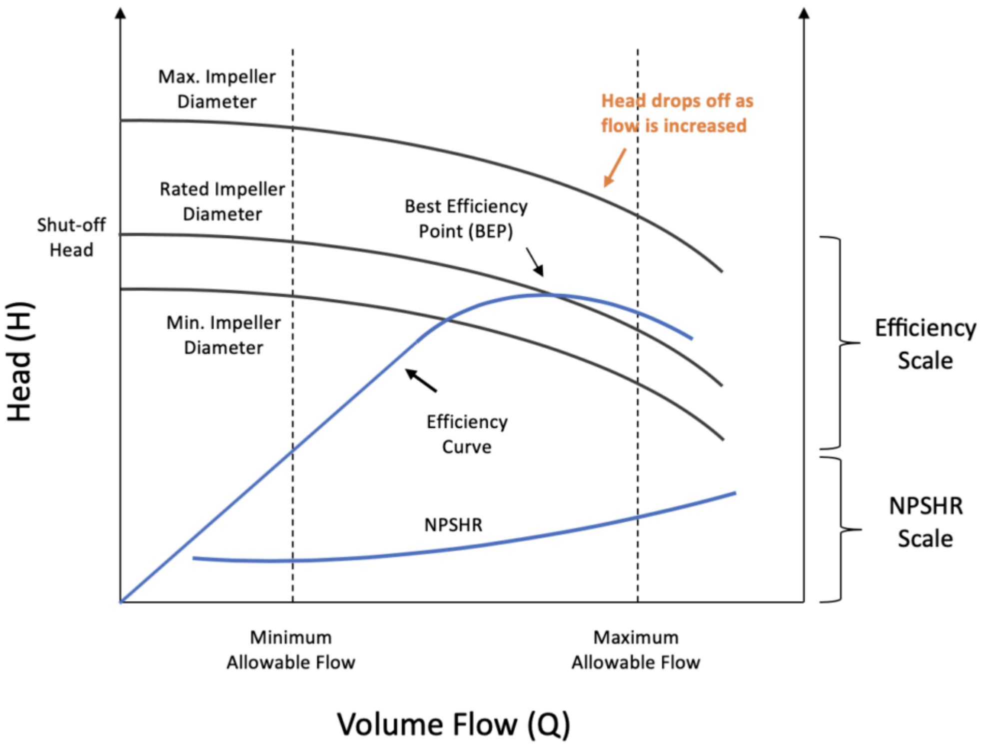


Image 2 Pump Curve

3.2. Pump Curve Components

A typical pump curve consists of the following components:

a) Head-Flow Curve (H-Q Curve)

The flow rate and head are the two primary parameters that define a pump's performance. The pump's head is directly proportional to its impeller's rotational speed and diameter.

The head-flow curve (H-Q curve) shows the relationship between the pump's head and flow rate at a constant rotational speed. The H-Q curve is typically plotted with head on the y-axis and flow rate on the x-axis. The curve shows that as the flow rate increases, the head decreases due to fluid friction and turbulence.

b) Head-Flow Curve (H-Q Curve)

The flow rate and head are the two primary parameters that define a pump's performance. The pump's head is directly proportional to its impeller's rotational speed and diameter.

The head-flow curve (H-Q curve) shows the relationship between the pump's head and flow rate at a constant rotational speed. The H-Q curve is typically plotted with head on the y-axis and flow rate on the x-axis. The curve shows that as the flow rate increases, the head decreases due to fluid friction and turbulence.

c) Power Curve (P-Q Curve)

The power curve (P-Q curve) shows the relationship between the pump's power consumption and flow rate at a constant rotational speed. The P-Q curve is typically plotted with power consumption on the y-axis and flow rate on the x-axis. The curve shows that as the flow rate increases, the power consumption increases due to increased fluid friction and turbulence.

d) Efficiency Curve (E-Q Curve)

The efficiency curve (E-Q curve) shows the relationship between the pump's efficiency and flow rate at a constant rotational speed. The efficiency curve is typically plotted with efficiency on the y-axis and flow rate on the x-axis. The curve shows that as the flow rate increases, the efficiency decreases due to increased fluid friction and turbulence.

3.3. Pump Curve Interpretation

Interpreting the pump curve is essential in selecting the right pump for an application and optimizing its performance. Here are some key points to keep in mind when interpreting the pump curve:

a) Best Efficiency Point (BEP)

The Best Efficiency Point (BEP) is the point on the H-Q curve where the pump operates at maximum efficiency. The BEP is the most economical operating point and should be targeted when selecting the pump.

b) Operating Range

The operating range is the range of flow rates and heads over which the pump can operate without damaging the impeller or motor. The operating range is limited by the pump's minimum and maximum flow rates and heads.

c) Identify the shut-off head and the maximum flow rate of the pump.

The shut-off head is the maximum head that the pump can generate at zero flow. It is the point where the pump curve intersects the vertical axis of the graph. The maximum flow rate is the maximum flow rate that the pump can deliver at zero head. It is the point where the pump curve intersects the horizontal axis of the graph.

d) Understand the pump's efficiency at different flow rates.

The efficiency of the pump varies with the flow rate. The efficiency curve is usually represented on the pump curve graph as a line or a curve. The efficiency curve shows the pump's efficiency at different flow rates. The efficiency of the pump is usually highest at the BEP and decreases as the flow rate increases or decreases from the BEP.

e) Net Positive Suction Head (NPSH)

The Net Positive Suction Head (NPSH) is the difference between the fluid's pressure at the pump suction and the vapor pressure of the fluid. The NPSH is a critical parameter in selecting the pump and should be greater than the pump's NPSH requirement to prevent cavitation.

3.4. Interpreting Pump Curve Shapes

Different pump curve shapes indicate different pump characteristics. The following are the most common pump curve shapes:

a) Steep Curve

A steep curve indicates that the pump can produce high head at low flow rates. This type of pump is suitable for applications that require high pressure.

b) Flat Curve

A flat curve indicates that the pump can produce high flow rates at low heads. This type of pump is suitable for applications that require high flow rates.

c) Steep Drop-Off Curve

A steep drop-off curve indicates that the pump is susceptible to cavitation at low flow rates. This type of pump requires a high NPSH to operate efficiently.

3.5. Tips for Pump Curve Analysis

Here are some tips for analyzing pump curves:

- Always use the pump curve provided by the manufacturer for accurate performance evaluation.
- Consider the system's friction losses when determining the system curve.
- Ensure that the pump operates at or near its best efficiency point (BEP) to minimize energy consumption and operating costs.
- Avoid operating the pump at low flow rates as it can cause excessive wear and tear on the impeller and reduce the pump's efficiency.
- Ensure that the required NPSH is available at the pump's suction to prevent cavitation.

Selecting the Pump

When selecting a pump, the following factors should be considered:

- Flow rate requirement
- Head requirement
- Operating range
- Fluid properties (viscosity, density, temperature, etc.)
- NPSH available
- Efficiency requirement

By plotting the system curve (the relationship between the flow rate and head required by the system) on the pump curve, the operating point can be determined. The pump selected should operate at or near the BEP for maximum efficiency and longevity.

Optimizing Pump Performance

To optimize the pump's performance, the following factors should be considered:

- Operating at or near the BEP for maximum efficiency
- Adjusting the impeller diameter or speed to match the system requirements
- Minimizing fluid friction and turbulence by reducing pipe diameter, increasing pipe roughness, or reducing fluid velocity
- Maintaining the pump and system to prevent wear and damage

4. Example : How To Size A Pump ?

Scenario:

A small town needs to size a centrifugal pump for its water distribution system with consideration for Net Positive Suction Head Available. The system must supply water to various elevations within the town, considering both residential and commercial demands.

Detailed Requirements:

System Layout:

1. The water source is a well with a static water level of 30 meters below ground level.
2. The highest point in the distribution system is at an elevation of 100 meters above sea level.
3. Pipeline Information: The system includes various pipes with different diameters totaling 1500 meters in length with Darcy friction factor f=0.02

   The pipeline is divided to 3 sections there is one section before the pump and after the pump there are two sections.

   Pipe diameters:

   1. Pipe diameters:

      300 mm (0.30 m).

      200 mm (0.20 m).

      150 mm (0.15 m).

   2. Fittings:

      - Elbows: 4 standard 90° elbows K=0.3 (2 in the first section, 2 in the last section).
      - Reducers: 2 with K=0.5
        - From 0.30 m to 0.20 m. (first section)
        - From 0.20 m to 0.15 m. (third section)

4. Flow Requirements:
   - Flow Rate (Q): The town requires a flow rate of 500 m³/h (cubic meters per hour)
   - Pressure at Discharge Point: 300 kPa (gauge pressure).
5. Fluid Properties:

   Water at room temperature (assume density = 1000 kg/m³, viscosity = 1 cP, vapor pressure at 20°C = 0.023 bar).

Tasks:

1. Calculate the Total Dynamic Head (TDH):
   - Static Head Calculation: Determine the elevation head from the well to the highest point.
   - Frictional Head Loss Calculation: Use the Darcy-Weisbach equation for head loss due to friction in pipes.
2. Calculate the Net Positive Suction Head available (NPSHa)
3. Select a pump from the presented options.

   Let's assume we have three candidate pumps with the following specifications:

Pump	**Flow Rate (m³/h)**	**TDH (m)**	**Efficiency (%)**	**NPSHr (m)**
Pump A	450–550	390–410	78	30
Pump B	480–520	400–420	82	28
Pump C	500–600	405–430	80	35

Pump option

Solution:

1. **Total Dynamic Head**

- Static Head (elevation change) Calculation:

The static head is the difference in elevation between the well and the highest point in the system.

$\text{Static Head} = \text{Elevation of highest point} - \text{Static water level (below ground)}$

\[\text{Static Head} = H_s = 100 - (-30) = 100 + 30 = 130 \, \text{m}\tag{6.5}

- Frictional Head Loss (Darcy-Weisbach Equation)

  Step 1: Velocity in each pipe section

  Flow rate (Q): $500\,\mathrm{m^3/h} = \frac{500}{3600} = 0.13889\,\mathrm{m^3/s}$

  For each section: $v = \frac{Q}{A}$

  - Pipe 1 (Diameter = 0.30 m):

    $A_1 = \frac{\pi(0.30)^2}{4} = 0.07069\,\mathrm{m}^2, \quad v_1 = \frac{0.13889}{0.07069} = 1.96\,\mathrm{m/s}$
  - Pipe 2 (Diameter = 0.20 m):

    $A_2 = \frac{\pi(0.20)^2}{4} = 0.03142\,\mathrm{m}^2, \quad v_2 = \frac{0.13889}{0.03142} = 4.42\,\mathrm{m/s}$
  - Pipe 3 (Diameter = 0.15 m):

    $A_3 = \frac{\pi(0.15)^2}{4} = 0.01767\,\mathrm{m}^2, \quad v_3 = \frac{0.13889}{0.01767} = 7.86\,\mathrm{m/s}$

  Step 2: Frictional Loss for Each Pipe Section

  Using Darcy-Weisbach:

  $h_f = f \cdot \frac{L}{D} \cdot \frac{v^2}{2g}$

  - Pipe 1 (Length = 500 m, Diameter = 0.30 m):

    $h_{f1} = 0.02 \cdot \frac{500}{0.30} \cdot \frac{(1.96)^2}{2\cdot 9.81} = 0.02 \cdot 1666.67 \cdot 0.196 = 6.53\,\mathrm{m}$
  - Pipe 2 (Length = 500 m, Diameter = 0.20 m):

    $h_{f2} = 0.02 \cdot \frac{500}{0.20} \cdot \frac{(4.42)^2}{2\cdot 9.81} = 0.02 \cdot 2500 \cdot 0.499 = 24.95\,\mathrm{m}$
  - Pipe 3 (Length = 500 m, Diameter = 0.15 m):

    $h_{f3} = 0.02 \cdot \frac{500}{0.15} \cdot \frac{(7.86)^2}{2\cdot 9.81} = 0.02 \cdot 3333.33 \cdot 3.15 = 210.00\,\mathrm{m}$

  Step 3: Minor Losses

  For fittings (elbows and reducers):

  $h_m = K \cdot \frac{v^2}{2g}$

  1. Elbows

  For standard 90° elbows: K=0.3

  1. First Section (2 elbows):
     - Velocity in the first section ($v_1 = 1.96\,\mathrm{m/s}$).
     - Minor loss for elbows: $h_{\text{elbows},1} = 2 \cdot K \cdot \frac{v_1^2}{2g} = 2 \cdot 0.3 \cdot \frac{(1.96)^2}{2\cdot 9.81} = 0.6 \cdot 0.196 = 0.118\,\mathrm{m}$
  2. Last Section (2 elbows):
     - Velocity in the last section ($v_3 = 7.86\,\mathrm{m/s}$).
     - Minor loss for elbows: $h_{\text{elbows},2} = 2 \cdot K \cdot \frac{v_3^2}{2g} = 2 \cdot 0.3 \cdot \frac{(7.86)^2}{2\cdot 9.81} = 0.6 \cdot 3.15 = 1.89\,\mathrm{m}$

2. Reducers

For reducers: K=0.5

1. First Reducer (0.30 m to 0.20 m):
   - Velocity in the second section ($v_2 = 4.42\,\mathrm{m/s}$).
   - Minor loss: $h_{\text{reducer},1} = K \cdot \frac{v_2^2}{2g} = 0.5 \cdot \frac{(4.42)^2}{2 \cdot 9.81} = 0.5 \cdot 0.998 = 0.499\,\mathrm{m}$
2. Second Reducer (0.20 m to 0.15 m):
   - Velocity in the third section ($v_3 = 7.86\,\mathrm{m/s}$).
   - Minor loss: $h_{\text{reducer},2} = K \cdot \frac{v_3^2}{2g} = 0.5 \cdot \frac{(7.86)^2}{2 \cdot 9.81} = 0.5 \cdot 3.15 = 1.575\,\mathrm{m}$

Total Minor Losses

$h_m = h_{\text{elbows},1} + h_{\text{elbows},2} + h_{\text{reducer},1} + h_{\text{reducer},2}$

$h_m = 0.118 + 1.89 + 0.499 + 1.575 = 4.082\,\mathrm{m}$

Frictional Head Loss:

$h_f = h_{f1} + h_{f2} + h_{f3} + h_m$

$h_f = 6.53 + 24.95 + 210.00 + 4.082 = 245.562\,\mathrm{m}$

Total Dynamic Head (TDH):

$\mathrm{TDH} = \text{Static Head} + h_f + \text{Discharge Pressure Head}$

$\mathrm{TDH} = 130 + 245.562 + 30.58 = 406.14\,\mathrm{m}$

2. **Net Positive Suction Head Available (NPSHa)**

$NPSHa = \frac{P_{\text{atm}} - P_{\text{vapor}}}{\rho g} + h_s - h_{f\,\text{suction}}$

- Atmospheric Pressure (Patm):

  Standard atmospheric pressure: $P_{\text{atm}} = 101325\,\mathrm{Pa}$.
- Vapor Pressure (Pvapor):

  Given: $P_{\text{vapor}} = 0.023\,\mathrm{bar} = 2300\,\mathrm{Pa}$.
- $\frac{P_{\text{atm}} - P_{\text{vapor}}}{\rho g} = \frac{101325 - 2300}{1000 \cdot 9.81} = \frac{99025}{9810} = 10.1\,\mathrm{m}$
- Water Density (ρ):

  ρ=1000 kg/m3.
- Gravitational Acceleration (g):

  $g = 9.81\,\mathrm{m/s}^2$.
- Static Head (hs) in the suction side:

  the elevation from the water source to the pump: hs=30m.
- Suction Pipe Friction Losses

  $h_{f,\text{suction}} = h_{f1} + h_{\text{minor},1}$

  From previous calculations:

  $h_{f1} = 6.53\,\mathrm{m}$

  $h_{\text{minor},1} = h_{\text{elbows},1} + h_{\text{reducer},1} = 0.118 + 0.499 = 0.617\,\mathrm{m}$

  $h_{f,\text{suction}} = 6.53 + 0.617 = 7.147\,\mathrm{m}$

NPSHa Calculation

Substitute into the formula:

NPSHa = 10.1 + 30 - 7.147= 32.953 m

The calculated NPSHa of **32.95 m** is sufficient for most pumps. Now we can compare this to the pump's NPSH requirement to confirm the selection.

Steps for Pump Selection

Pump Requirements:

- Total Dynamic Head (TDH): **406.14 m**
- Flow Rate (Q): **500 m³/h**
- NPSHa: **32.95 m**

Evaluation :

Pump	**Flow Rate (m³/h)**	**TDH (m)**	**Efficiency (%)**	**NPSHr (m)**
Pump A	450–550	390–410	78	30
Pump B	480–520	400–420	82	28
Pump C	500–600	405–430	80	35

Pump optionS

1. **Flow Rate Match:** All three pumps handle the required flow rate of **500 m³/h**.

2.**TDH Match:**

- Pump A: TDH range is **390–410 m** (sufficient).
- Pump B: TDH range is **400–420 m** (sufficient).
- Pump C: TDH range is **405–430 m** (sufficient).

3.**NPSHa vs. NPSHr:**

- Pump A: NPSHa (32.95)>NPSHr (30) → **Safe**.
- Pump B: NPSHa (32.95)>NPSHr (28) → **Safe**.
- Pump C: NPSHa (32.95)<NPSHr (35) → **Not Safe**

4.**Efficiency:** Pump B has the highest efficiency (**82%**) among the options.

Recommended Pump

Pump B is the best choice as it:

- Meets the TDH and flow rate requirements.
- Has the highest efficiency.
- Its NPSHr (28 m) is safely below the NPSHa (32.95 m).

XIV Did you understand the basics of Sizing of Centrifugal Pumps?

Quiz 1

What are essential parameters for sizing a pump?

- **A** Flow rate
- **B** Total differential head
- **C** Suction pipe length
- **D** Pump efficiency

Quiz 2

What is the primary purpose of calculating pump power?

- To design suction pipes
- To size the pump driver
- To determine fluid viscosity
- To optimize flow rates

Quiz 3

Which factor does NOT affect the total differential head of a pump?

- Vapour pressure
- Frictional losses
- Pipe diameter
- Static head

Quiz 4

What does NPSHa stand for in pump sizing?

- Net Positive Suction Height Available
- Neutral Pumping Suction Head Adjustment
- Net Positive Suction Head Available
- Net Pump Suction Head Absolute

XV Chapter 07: Hydraulic turbines

Introduction

Hydraulic turbines are power-producing turbomachines, using water as the fluid. The water has to be available at a reasonable height or head, in fairly large quantities so that some economically feasible power projects may be developed.

Basically there are three types of turbines: Pelton, Francis, and Kaplan turbines, named after their designers.

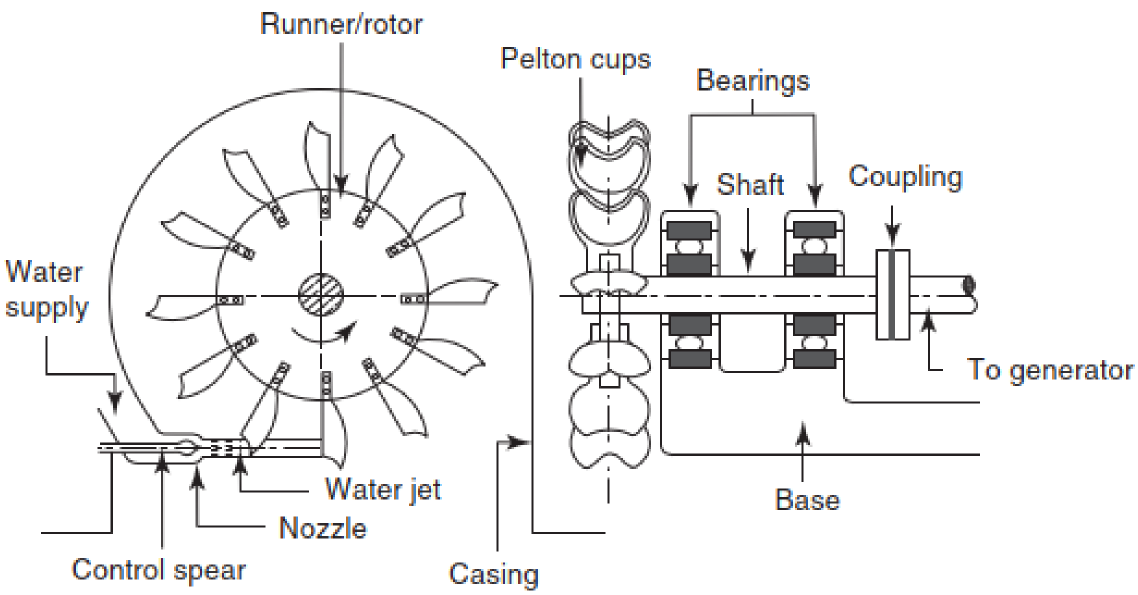
1. The Pelton turbine is an impulse turbine, with tangential flow, for high-head applications.
2. The Francis turbine is a reaction turbine, with radial or mixed flow, for medium head applications.
3. The Kaplan turbine is a reaction turbine, with axial flow, for low-head applications.

$$H = \frac{p_2 - p_1}{w} + \frac{V_2^2 - V_1^2}{2g} + (z_2 - z_1) \tag{7.1}$$

1. Pelton Turbine

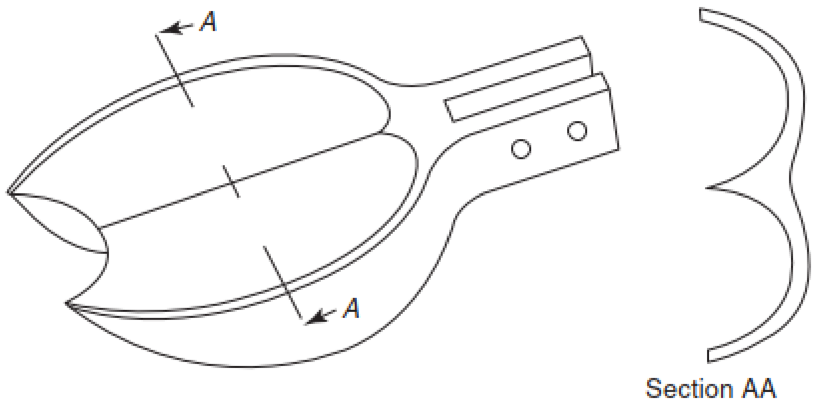
The Pelton turbine belongs to the range of the low specific speeds (5 to 70) and the range of high heads (150 m of water and above). It is an impulse-type turbine.

A Pelton turbine setup is shown in Fig. Some of the terms used are the shaft, the rotor, the nozzle, the jet, etc., which are shown in the figure.



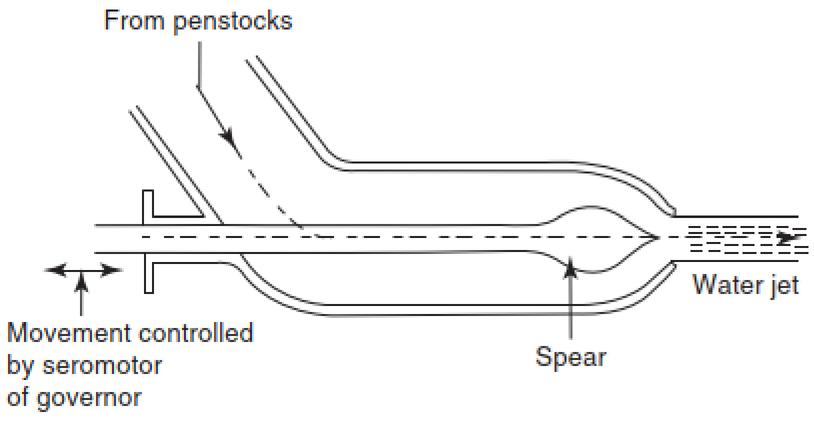
Schematic layout of a Pelton turbine.

The blades or vanes of the rotor in the case of Pelton turbine are the “Pelton double cups” or “buckets,” as shown in Fig. These double cups are mounted on the periphery of a circular disk and together they form the rotor of the Pelton turbine.



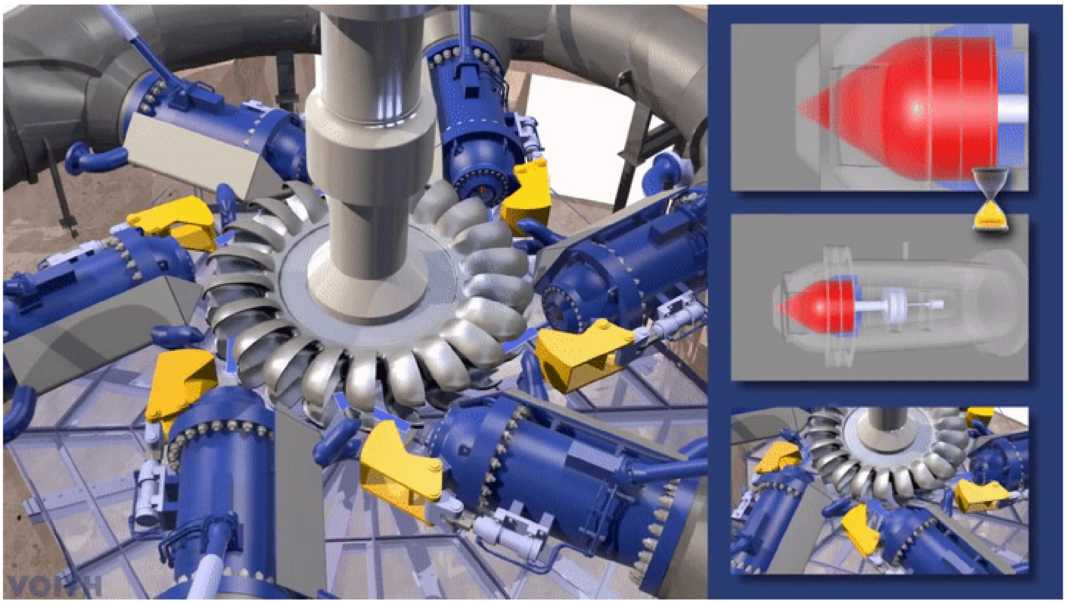
Pelton double cup.

The water, supplied from the head-works to the power house through the penstocks (steel pipes), is led to these buckets in the form of a high-speed jet issued from a nozzle. The kinetic energy of water jet is transferred to the series of the buckets (and to the rotor) that come in succession in the line of the jet as the rotor rotates. The jet gets divided into two equal halves by the jet splitter of the double cup, with each half striking the cups on either side.



Nozzle of Pelton turbine.

In multi-jet Pelton turbines, water is led around the rotor into the identical nozzles equally spaced around the periphery of the rotor. The spear assemblies are also identical in all the nozzles. Their movement is controlled by the same source, so that all the jets are equally controlled.



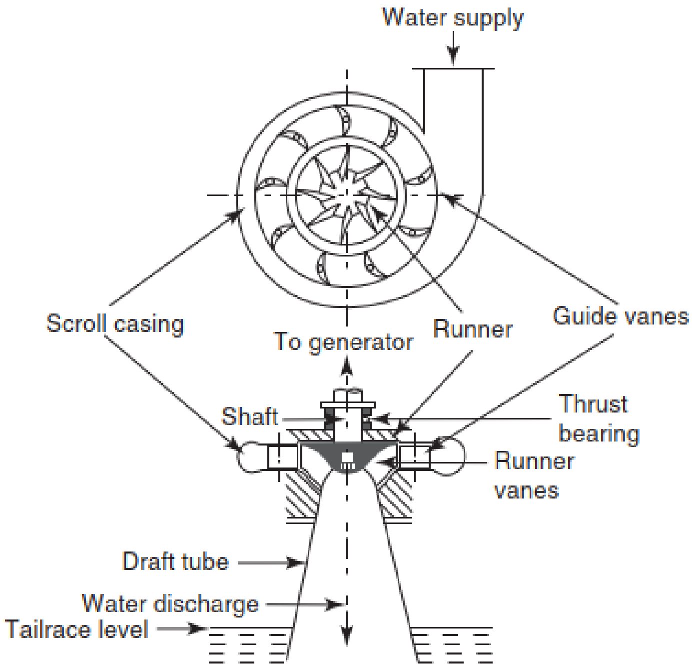
Pelton Turbine

2. Francis Turbine

The Francis turbine is a reaction turbine suitable for a medium range of specific speeds (60–300) and a medium range of heads (50–250 m).

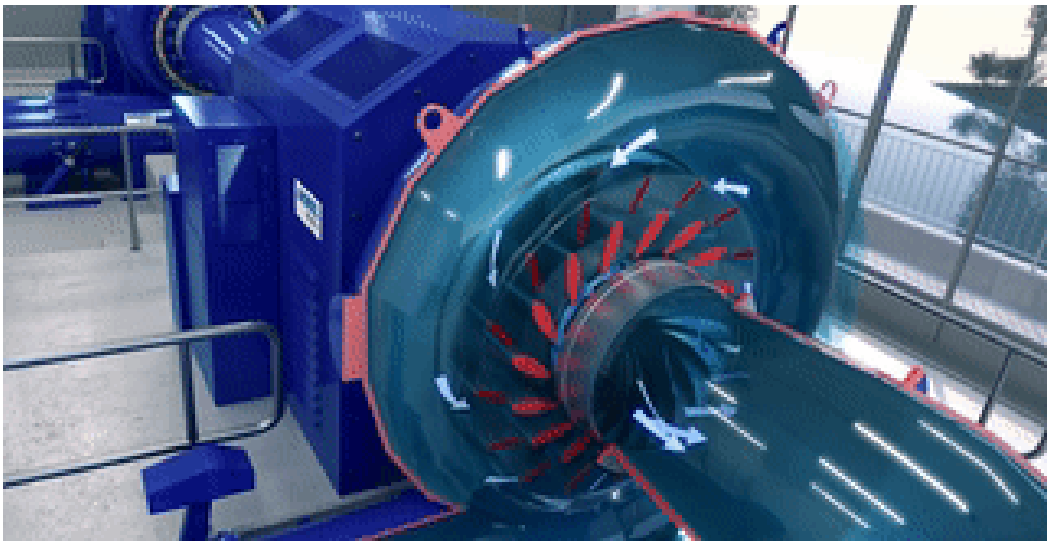
The Francis turbine is designed as a radial flow machine in the range of specific speeds of 60–120. As the specific speed corresponding to the data of the project site keeps on increasing, the design shifts to a mixed flow machine and then to almost an axial flow machine.

An installation of the Francis turbine is shown in Fig. Water from the penstock pipe enters an outer spiral casing that may be fabricated out of steel plates or cast in concrete with a lining of steel plates.



Schematic layout of a Francis turbine.

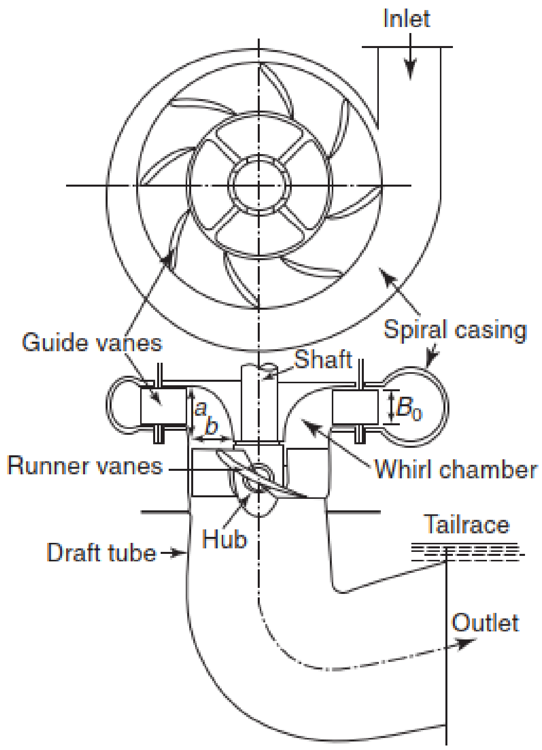
The guide vanes have airfoil shapes so that the passages between them act like nozzles that convert a part of the pressure energy of water into kinetic energy. The water coming from the casing is directed on to the rotor vanes. Each guide vane has its own axis about which it can swing, so as to vary the area of flow of water. The swinging of all the guide vanes (about their individual axes) is controlled by a governor-actuated regulator ring, so that the flow of water can be controlled.



Francis Turbine

3. Kaplan Turbine

The Kaplan turbine or propeller turbine is a low head (up to 75 m), large flow rate, high specific speed (300–1000), axial flow, reaction turbine. A schematic view of the Kaplan turbine is shown in Fig.

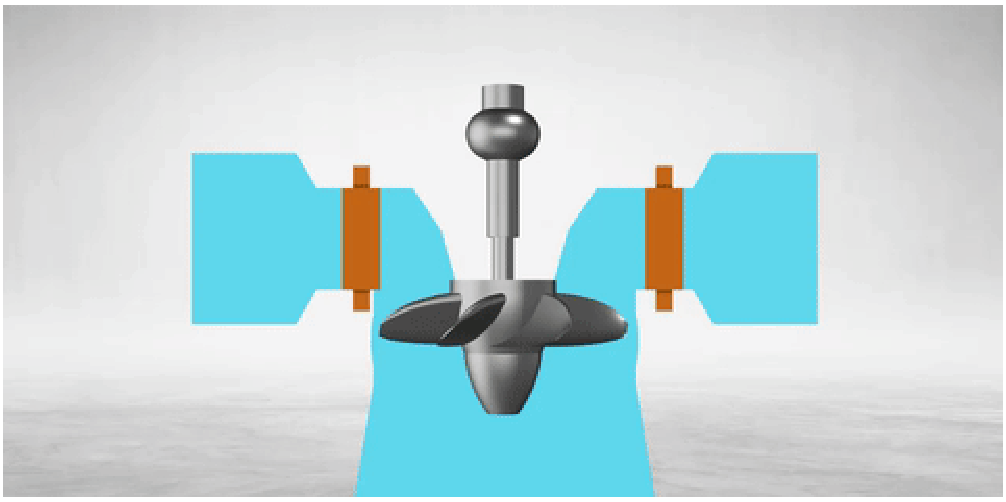


Kaplan turbine with an ellipsoidal whirl chamber.

A schematic view of the Kaplan turbine is shown in Fig. Water enters a spiral casing and passes on to the ring of guide vanes, just like in the Francis turbine. The guide vanes have their individual axes, about which they can swing, to control the quantity of flow of water. As the water passes through the guide vanes, its head energy partly gets converted into kinetic energy, and it acquires a whirl component of its velocity.

The water then enters an ellipsoidal whirl chamber, where it gets into a spiral vortex flow pattern before meeting the leading edge of the runner blades. As the water flows through the passages between the runner blades, the runner blades absorb the energy from the water. The water is then discharged with an axial flow

pattern into a draft tube and then to the tailrace.



Kaplan turbine

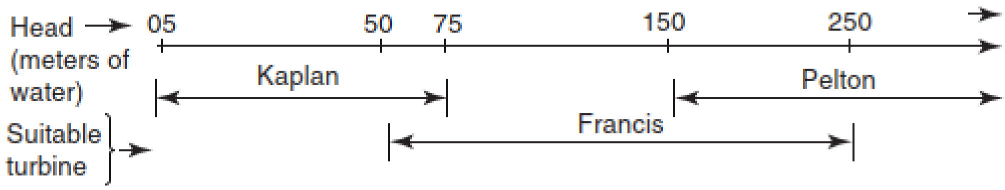
4. Selection of Hydraulic Turbines

The power projects, where hydraulic turbines are to be installed, are generally huge projects, involving very high investments on head works and machinery. Because of the wide variation of the two basic data, namely, the head and the flow rate, each project requires some unique design. Therefore, the selection and design of a particular type of turbine must be undertaken with some discretion, so as to have the highest possible efficiency of the turbine.

There are two approaches to decide the type of turbine suitable for a given project site, with specified head and flow rate:

4.1. The head

One criterion is the head (meters of water) available. Shown in Fig. is a scale that indicates the head and the corresponding suitable turbine.



Selection of turbines on the basis of head.

It may be observed that there are some ranges of overlaps, such as the 50-75 m or 150-250 m stretches. In these ranges, the turbine can be selected by the criterion of the specific speed, mentioned in the next paragraph, to include the effect of the available flow rates also.

4.2. The specific speed

Another criterion is the specific speed corresponding to the site data. The head, H (m of water), and the flow rate, Qm^3/s, are taken as data. An overall efficiency η of the order of 0.85 or 0.88 can be assumed. Then, the power P is given by

$$P = \frac{wQH\eta}{1000}\,\mathrm{kW} \tag{7.2}$$

Now, the speed of the turbine N rpm has to be selected that must be one of the synchronous speeds (N = 3000/p, p =1, 2, 3, …, to have a frequency of 50 Hz of electrical power supply) because turbines drive the alternators. With this, the specific speed can be calculated by

$$N_s = \frac{N\sqrt{P}}{H^{5/4}} \tag{7.3}$$

Figure indicates a scale of specific speeds and the corresponding suitable turbines.

Conclusion

Hydraulic turbines are the prime movers that drive alternators in hydroelectric power stations. The energy flow to the turbines is in the form of flow rate of water $Qm^3/s$and head of water (H_m of water).

XVI Did you understand the basics of Hydraulic Turbines?

Quiz 1

What type of turbine is suitable for medium heads (50–250 m)

- Pelton turbine
- Propeller turbine
- Francis turbine
- Kaplan turbine

Quiz 2

Which factors are critical for selecting a hydraulic turbine?

- **A** Available head
- **B** Flow rate
- **C** Turbine material
- **D** Specific speed

Quiz 3

What are the characteristics of a Pelton turbine?

- **A** Tangential flow
- **B** High specific speed range
- **C** Used for low-head applications
- **D** Impulse turbine

Quiz 4

Which of the following describe Kaplan turbines?

- **A** Axial flow
- **B** High head applications
- **C** Reaction turbine
- **D** Low specific speed

Quiz 5

What is the specific speed range for Kaplan turbines?

- 5–70
- 60–300
- 300–1000
- 1000–2000

XVII Exit exercises

Quiz 1

What type of turbine is suitable for medium heads (50–250 m)

- Pelton turbine
- Propeller turbine
- Francis turbine
- Kaplan turbine

Quiz 2

Which of the following is a characteristic of axial flow machines?

- Fluid flows parallel to the axis of rotation
- Fluid flows perpendicular to the axis of rotation
- Fluid is compressed in a diffuser
- Fluid is accelerated in a nozzle

Quiz 3

Which similarity law ensures that the flow patterns in two turbomachines are the same?

- Geometric Similarity
- Kinematic Similarity
- Dynamic Similarity
- Thermodynamic Similarity

Quiz 4

Which of the following is NOT a type of turbomachine?

- Turbine
- Compressor
- Pump
- Heat exchanger

Quiz 5

Dynamic Similarity in turbomachinery is achieved when:

- The machines are operating at the same temperature
- The machines are of the same size
- The machines have the same efficiency
- The forces in the machines are in the same ratio

Quiz 6

What is the primary function of a turbomachine?

- To convert fluid energy into mechanical work
- To convert mechanical energy into electrical energy
- To convert mechanical energy into thermal energy
- To convert thermal energy into mechanical work

Quiz 7

An air compressor is designed such that the speed at the inlet V1 is equal to the speed at the outlet V2. This simply results in different sections in 1 and 2.

Data:

input: P1 =10 N/cm2, T1=20°C

Output: P2=30N/cm2

Mass flow Qm=20 kg/s.

What is the power absorbed by this compressor?

Quiz 8

A test is to be carried out on a proposed design for a large pump which is to deliver 1.5 m^3/s of water from an impeller of 40 cm diameter with a pressure rise of 400 kPa. A model with an 8 cm diameter wheel should be used to perform the experiment.

What flow rate should you use?

Conclusion

As we conclude our journey through the fascinating world of turbomachinery, we reflect on the extensive knowledge and skills we have acquired. From understanding the fundamental principles of thermodynamics and fluid mechanics to applying these concepts to analyze and design efficient turbomachines, we have developed a solid foundation in this critical field of engineering.

Turbomachinery plays a pivotal role in modern society, powering everything from household appliances to large-scale power plants. The expertise we have gained here not only prepares us for the technical challenges of the industry but also empowers us to contribute to sustainable energy solutions and innovative engineering designs.

Moving forward, I encourage you to continue exploring this dynamic field. Stay curious, keep learning, and apply your knowledge to solve real-world problems. Remember, the principles of turbomachinery are not confined to the classroom; they are the driving force behind many technologies that shape our daily lives.

Let this course be the starting point of your professional journey in turbomachinery. Embrace the challenges, seek out opportunities for further learning, and strive to make a positive impact on the world through your work.

BIBLIOGRAPHY

1. *Dixon (1998), Fluid mechanics and thermodynamics of turbomachinery, 4^{th}ed Butterworth-Heinemann*
2. *Albin BOLCS (1990), Turbomachines thermiques, EPFL Lausanne, VOL 1 & 2.*
3. *R. Comolet, Mécanique expérimentale des fluides, Tome II, dynamique des fluides réels, turbomachines, Editions Masson, 1982.*
4. *B. Lakshminarayana, Fluid Dynamics and Heat Transfer of Turbomachinery, Wiley, New York, 1996.*
5. *J.H. Horlock, Axial Flow Turbines, London Butterworth Scientific Publications*
6. *O.E. Balje, Turbomachines, A Guide to Design and Theory, John Wiley*
7. *W. Traupel, Thermische Turbomaschinen, Springer Verlag.*
8. *S. Korpela, Principles of Turbomachinery 2nd Edition, John Wiley*
9. *S. Dixon, C. Hall, Fluid Mechanics and Thermodynamics of Turbomachinery, Sixth Edition, Butterworth-Heinemann*