Turbomachinery – 1

SOE3211/2 Fluid Mechanics lecture 8

8.1 Definitions (A)

Fluid machine – a device exchanging energy (work) between a fluid and a mechanical system

In particular : a *turbomachine* is a device using a rotating mechanical system

The flow of energy can be in either direction :

1. Turbine $\rightarrow$ the flow does work on the mechanical system (i.e. energy is transmitted from the fluid to the mechanical system)

2. Pumps, fans, compressors, blowers $\rightarrow$ the mechanical system does work on the flow (i.e. energy is transmitted from the rotor to the fluid)

8.2 Turbines (A)

Impulse turbine

- free jet of water impinges on revolving impeller (runner) and is deflected. Change of momentum of water $\equiv$ change of momentum (torque) for impeller.

- Analyse through control volume techniques.

- In impulse turbines, available head converted into kinetic energy head $\rightarrow$ work done on impeller.

- Typical example – Pelton wheel

Alternative : reaction turbine.

- Turbine completely enclosed in water, which can be pressurised (hence, uses available pressure head directly, without conversion to k.e.)

- Shape of blades forces change in momentum of water – torque on blades – power.

- Eg. Kaplan, (axial), centrifugal turbines

Contact between the mechanical and fluid parts is via an *impeller*

Axial flow : fluid approaches and leaves impeller along the axis of rotation

Centrifugal flow : fluid approaches axially, leaves radially (in plane of impeller)

Mixed flow : part axial, part radial

8.3   Dimensional analysis (A)

Start with dimensional analysis for a general impeller :

Power	P	ML^2T^{-3}
Volumetric flow rate	Q	L^3T^{-1}
Head difference	H	L
Rotational speed	N	T^{-1}
Impeller diameter	D	L
Density of fluid	ρ	ML^{-3}
Viscosity	μ	$ML^{-1}T^{-1}$
Bulk modulus	K	$ML^{-1}T^{-2}$
Roughness	ε	L

NB. K expresses compressibility effects
Dimensional analysis gives

$$\left(\frac{P}{N^3D^5\rho}\right) =$$
$$F\left[\left(\frac{Q}{ND^3}\right), \left(\frac{gH}{N^2D^2}\right), \left(\frac{\mu}{ND^2\rho}\right), \left(\frac{K}{N^2D^2\rho}\right), \left(\frac{\varepsilon}{D}\right)\right]$$

$\left(\frac{\mu}{ND^2\rho}\right)$ $\simeq$ Reynolds number

$\left(\frac{K}{N^2D^2\rho}\right)$ relates to Mach number

$\left(\frac{\varepsilon}{D}\right)$ Surface roughness

$\left(\frac{P}{N^3D^5\rho}\right)$ Power coefficient K_P

$\left(\frac{Q}{ND^3}\right)$ Flow coefficient K_Q

$\left(\frac{gH}{N^2D^2}\right)$ Head coefficient K_H

If we plot K_P, K_H vs. K_Q we have performance curves appropriate for geometrically similar machines. Every machine is designed to meet a specific duty, i.e. a design point (normally at the maximum efficiency of the machine).

We can also rate machines using *type number* or *specific speed* :

For pumps K_Q, K_H are the most important parameters, so the ratio would indicate the suitability of a particular pump for large or small volumes relative to the head developed.

If we can also eliminate the impeller diameter we can make the type number independent of the machine size. Write

$$n_s = \frac{(K_Q)^{\frac{1}{2}}}{(K_H)^{\frac{3}{4}}}$$

We could develop this for any point on the characteristic curve, but it is usually quoted for the design point.

For turbines the power developed is the most important. Thus

$$n_s = \frac{(K_P)^{\frac{1}{2}}}{(K_H)^{\frac{5}{4}}}$$

8.4   Losses and Efficiency (A)

Write a generic power loss as
$$P = \rho Q g H$$

For a *pump* : overall efficiency

$$\eta = \frac{\text{Power output of machine}}{\text{Power input to machine}} = \frac{\rho g H Q}{P}$$

Mechanical efficiency
$$\eta_m = \frac{P - P_m}{P}$$

Internal losses of the machine occuring in the casing and impeller are referred to as *hydraulic losses*

$$
\begin{aligned}
P = &\quad P_m + \rho g H_h Q_h + \rho g H Q \\
\text{Power in} = &\text{ Mech} + \quad \text{Hydr} + \text{Useful} \\
&\quad \text{loss} \qquad \text{loss} \quad \text{power}
\end{aligned}
$$

Hydraulic losses can involve :

- Impeller power loss − inefficiencies in the impeller design

- Leakage power loss − some flow evades the impeller

- Casing power loss – friction with casing

Possible to quantify these separately

Hydraulic efficiency

$$\eta_h = \frac{\text{Actual head}}{\text{Theoretical head}} = \eta_i \eta_v \eta_c$$

For a *turbine* : reverse output and input.

$$\rho g H Q = P_m + \rho g H_h Q_h + P$$

8.5   Turbine analysis (A)

In real turbomachines the flow is 3-d : we could write it as in cylindrical polar coordinates

$$\underline{v} = v_r \hat{\underline{r}} + v_\theta \hat{\underline{\theta}} + v_z \hat{\underline{z}}$$

where $\hat{\underline{r}}$, $\hat{\underline{\theta}}$, $\hat{\underline{z}}$ are unit vectors in radial, tangential, axial directions.

However we can make a simplifying assumption ; that $\underline{v}$ is a fn. of r only. This is equivalent to assuming :

1. the blades are infinitely thin – pressure difference across blade produces torque

2. the flow is axisymmetric (number of blades $\to \infty$)

3. no variation axially

Examine flow through C.V. between blades. We need to understand how the flow changes direction.
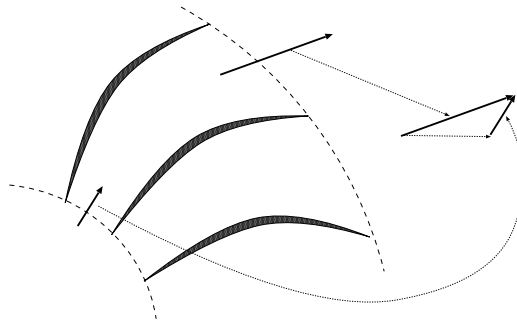
Relate change in flow direction/magnitude to torques on impeller.