TURBO MACHINES

BME IV/I

Chapter one: Introduction

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Chapter overview

- Introduction and types of Turbo machine
- Components and working of a Turbo machine
- General Classification of Turbines
- Application of First and Second Laws of Thermodynamics on turbo machine
- Efficiencies related to turbo machines
- Dimensionless Parameters and Their Physical Significance
- Effect of Reynolds Number and Specific Speed

Turbo machine

- A turbo machine is a device in which energy transfer occurs between a flowing fluid and rotating element due to dynamic action.
- This results in change of pressure and momentum of the fluid.

- If the fluid transfers energy for the rotation of the impeller, fixed on the shaft, it is known as **power generating turbo machine** (any turbine).

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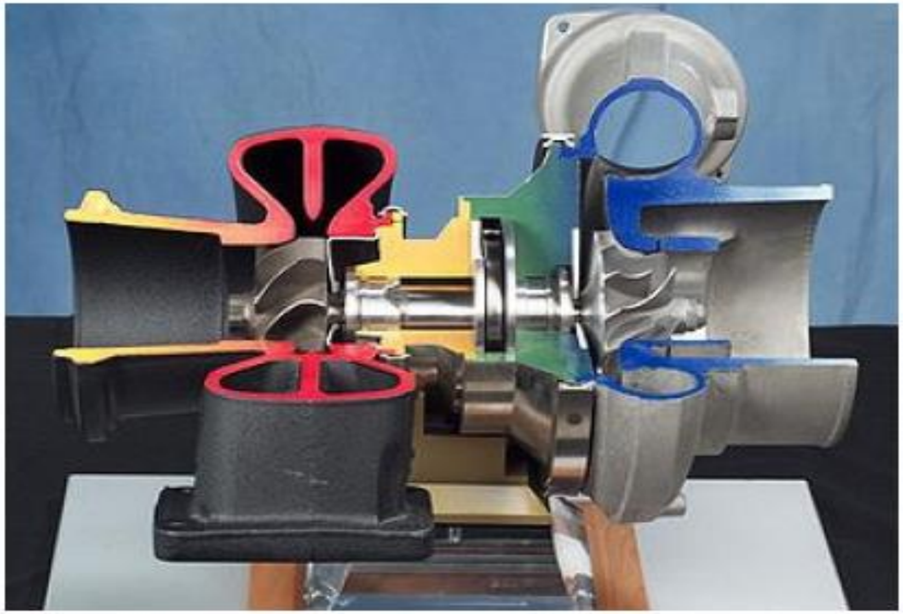
- If the machine transfers energy in the form of angular momentum fed to the fluid from the rotating impeller, fixed on the shaft, it is known as **power absorbing turbo machine.**
- Eg. Pump, compressor

- The figures show a typical turbo charger used in diesel engines to improve its thermal efficiency by increasing the pressure of air pumped into engine combustion chamber.



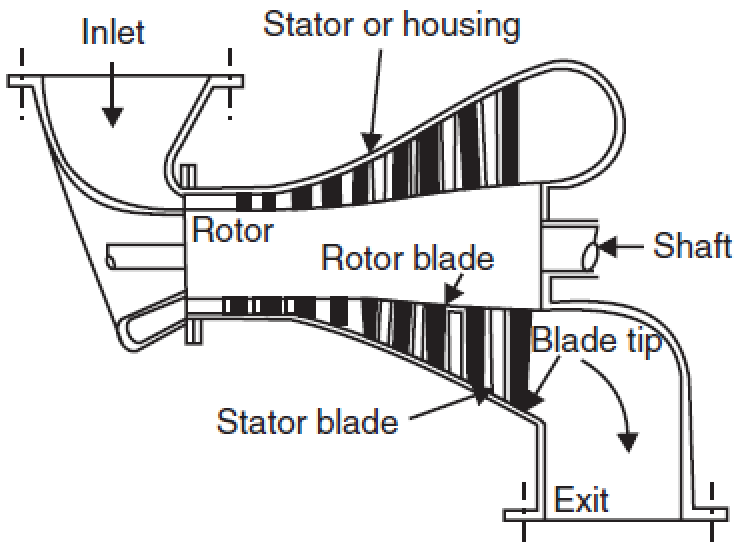
Components of Turbo machine

1**. Rotating element** (runner, vane, impeller or blades)

2. **Stationary elements** – which usually guide the fluid in proper direction for efficient energy conversion process.

3. **Shaft** – which either gives input power or takes output power from fluid under dynamic conditions and runs at required speed.

4. **Housing** – to keep various rotating, stationery and other passages safely under dynamic conditions of the flowing fluid.



Classification of Turbo machine

1. Based on energy transfer

a) Energy is given by fluid to the rotor - Power generating turbo machine E.g. Turbines

b) Energy given by the rotor to the fluid – Power absorbing turbo machine E.g. Pumps, blowers and compressors

2. Based on fluid flowing in turbo machine

a) Water

b) Air

c) Steam

d) Hot gases

e) Liquids like petrol etc.

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3. Based on direction of flow through the impeller or vanes or blades, with reference to the axis of shaft rotation

a) Axial flow – Axial pump, compressor or turbine

b) Mixed flow – Mixed flow pump, Francis turbine

c) Radial flow – Centrifugal pump or compressor

d) Tangential flow – Pelton water turbine

4. Based on condition of fluid in turbo machine

a) Impulse type (constant pressure) E.g Pelton water turbine

b) Reaction type (variable pressure) E.g. Francis reaction turbine

5. Based on position of rotating shaft

a) Horizontal shaft – Pelton, Steam turbines

b) Vertical shaft – Francis, Kaplan water turbines

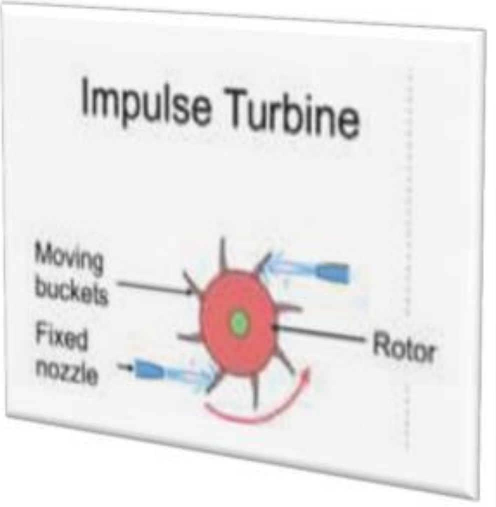
c) Inclined shaft – Bulb turbines, Archemedies screw

Classification of Turbines

A. Based on Operating Principal

A.1. Impulse Turbine

- The impulsive force imparted by high velocity jet of water on runner bucket produce a mechanical power on the turbine shaft
- Pelton, Turgo, Cross flow



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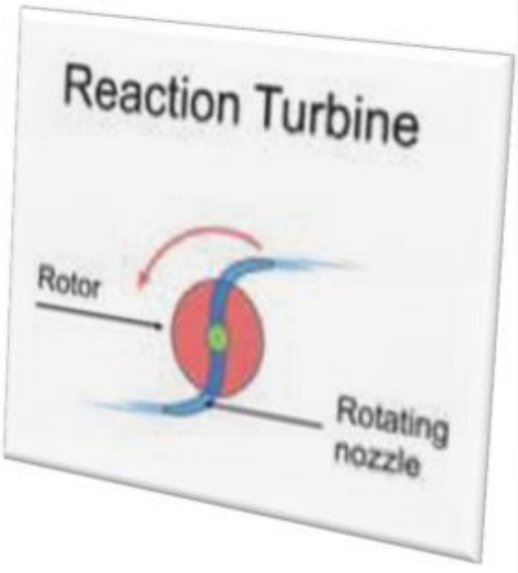
A.2. Reaction Turbine

- Two effects (pressure and impact) cause the energy transfer from the flow to mechanical energy on turbine shaft
- Francis, Propeller/ Kaplan



B. Based on Operating Direction of Flow

C. Based on Specific speed

Thermodynamic Laws on turbo machine

- Turbo machines are concerned with fluid flow either in compressible or incompressible nature.
- So, two state properties are essential: static and total
- Static properties:
  - Refers to the fluid properties taken in an assumption to a static fluid. Eg. P,v,t etc
  - The state of the particle fixed by a set of static properties is known as static state.

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Stagnation or total properties

- It is defined as the terminal state of a fictitious, isentropic, work free and steady flow process during which the macroscopic kinematic and potential energies of the fluid particle are reduced to zero.
- Then enthalpy is created stagnation.
- It is possible to obtain expressions for stagnation properties in terms of static properties following by laws of thermodynamics.

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- Considering any steady-flow process, the First Law of Thermodynamics gives the equation:

$$q - w = \Delta h + \Delta ke + \Delta pe$$

  - where, q and w are respectively the energy transfers as heat and work per unit mass flow,
  - h is the static enthalpy
  - ke and pe are respectively the macroscopic kinetic and potential energies per unit mass.
- The difference in enthalpies between the stagnation and static states is obtained by setting
  - $q = w = 0, \Delta h = ho - hi,\ keo = 0$ and $peo = 0$
  - $ho - (hi + kei + pei) = 0$
  - $ho = (h + ke + pe)$

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- According to the Second Law of Thermodynamics, since

$$T.ds = dh - v.dp,$$

- The entropy remains constant in the change from static to stagnation state,

$ds = 0$ and $dh = v.dp$, $v = 1/\rho$, being the specific volume and ρ, the density.

- Integration yields for the change from static to stagnation state:

$$ho - h = \int dh = \int v.dp$$

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- The steady flow equation of the First Law of Thermodynamics in the form:

$$Q + \dot{m}\ (h_1 + V_1^2/2 + gz_1) = P + \dot{m}\ (h_2 + V_2^2/2 + gz_2)$$

Where Q = Rate of energy transfer as heat across the boundary of the control volume,

P = Power output due to the turbo machine, and

m = Mass flow rate.

- *q – w* = Δ*ho*,
- where, Δ*ho* = *ho*2 – *ho*1, represents the *change in stagnation enthalpy between the inlet and the outlet of the turbo machine.*
- Also, *q* = *Q*/ *m* dot and, *w* = *P*/ *m* dot represent respectively, the heat and mass transfer per unit mass flow through the control volume.

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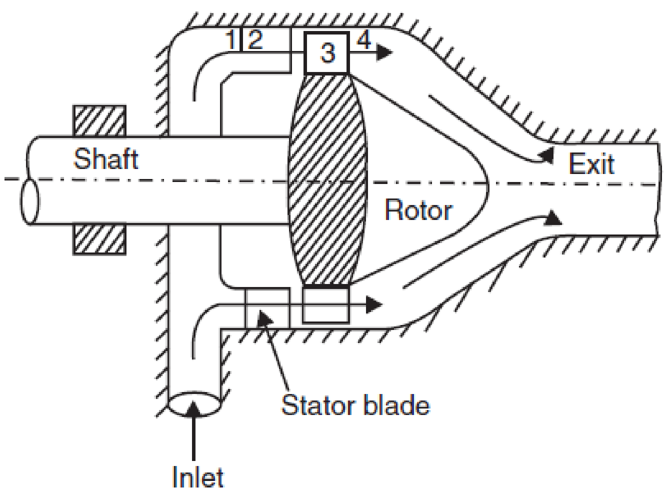

- By neglecting q, turbo machine can treated like a perfectly insulated device.

  $\Delta ho = -w = -P/\text{m } dot$

  $dho = -\delta w,$

- In a power-generating turbo machine, w is positive as defined so that Δho is negative, i.e., the stagnation enthalpy at the exit of the machine is less than that at the inlet.
- In a power-absorbing turbo machine, Δho is positive.

- If $ho3 > ho4$, the machines develops power
- If $ho3 < ho4$, the machine needs a driver and absorbs power.

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- The corresponding work input is higher in a *power-absorbing* machine as compared with that in an ideal process.

  We've, $T_o ds_o = dh_o - v_o dp_o$ and $dh_o = -\delta w$

  So, $-\delta w = v_o dp_o + T_o ds_o$

- In a *power-generating* machine, dp_o is negative since the flowing fluid undergoes a pressure drop when mechanical energy output is obtained.
- In a real machine, $T_o ds_o > 0$,
- So that $\delta w_i - \delta w = T_o ds_o > 0$

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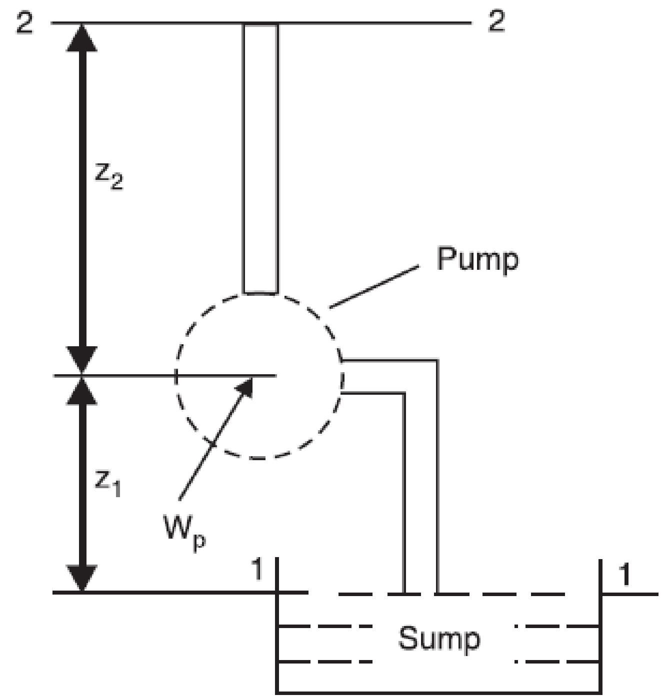
- **Example 1.** A turbo machine handling liquid water is located 8 m above the sump level and delivers the liquid to a tank located 15 m above the pump. The water velocities in the inlet and the outlet pipes are respectively 2 m/s and 4 m/s. Find the power required to drive the pump if it delivers 100 kg/min of water.



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$$w = q - \Delta h_o = -\Delta p_o/\rho,$$
$$= -[(p_2 - p_1)/\rho + (V_2^2 - V_1^2)/2 + g(z_2 - z_1)]$$
$$= [0 + (4^2 - 2^2)/2 + 9.81(15 + 8)] = 231.6 \text{ J·kg}^{-1}.$$

- The minimum power required is

$$\boldsymbol{P} = \dot{m}w = (100/60)(231.6) = \mathbf{386\ W}$$

- The actual power needed to drive the pump will be larger than that calculated above due to losses in friction in the pipes, entry and exit losses, leakage, etc.

Efficiencies related to turbo machines

$$\eta_{pg} = \frac{\text{Actual Shaft Work Output}}{\text{Ideal Work Output}} = \frac{w_s}{w_i},$$

$$\eta_{pa} = \frac{\text{Ideal Work Input}}{\text{Actual Shaft Work Input}} = \frac{w_i}{w_s}.$$

- Losses occur in turbo machines due to:

(a) bearing friction, windage, etc., all of which may be classified as *mechanical* losses,

(b) unsteady flow, friction between the blade and the fluid, etc., which are internal to the system and may be classified as *fluid-rotor losses*.

- There are other losses like leakage across blades, hole leakage, etc. which are covered under fluid-rotor losses.

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- If the mechanical and fluid-rotor losses are separated,

$$\eta_{pg} = \frac{\text{Fluid-Rotor Work}}{\text{Ideal Work Output}} \times \frac{\text{Shaft Work Output}}{\text{Fluid-Rotor Work}} = \frac{w_r}{w_i} \times \frac{w_s}{w_r},$$

$$\eta_{pa} = \frac{\text{Ideal Work Input}}{\text{Rotor-Fluid Work}} \times \frac{\text{Rotor-Fluid Work}}{\text{Shaft-Work Input}} = \frac{w_i}{w_r} \times \frac{w_r}{w_s}.$$

- The quantity *wr* /*wi*, is called the *adiabatic*, *isentropic* or *hydraulic* efficiency of the power generating system, since *wi*, is always calculated on the basis of a loss-free isentropic flow.

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- Adiabatic efficiency for power generating m/c:

$$\eta_a = \frac{\text{Mechanical Energy Supplied by the Rotor}}{\text{Hydrodynamic Energy Available from the Fluid}}$$

- Adiabatic efficiency for power absorbing m/c:

$$\eta_a = \frac{\text{Hydrodynamic Energy Supplied to the Fluid}}{\text{Mechanical Energy Supplied to the Rotor}}$$

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- The difference between *ws* and *wr* is expressed in terms of mechanical efficiency,
- For power generating,

$$\eta_m = \frac{w_s}{w_r} = \frac{\text{Shaft Work Output}}{\text{Fluid-Rotor Work}}$$

- For power absorbing,

$$\eta_m = \frac{w_s}{w_r} = \frac{\text{Rotor-Fluid Work}}{\text{Actual Work Input to Shaft}}$$

- Overall efficiency,

$$\eta_{pg} = \eta_a.\eta_m \text{ (}pg\text{ and }pa\text{ machines, both)}$$

- Moreover, mechanical losses are not strong functions of load and fluid states, since most turbines are governed to run at a constant speed.
- It is usual to assume the mechanical efficiency to be unity in many cases, the overall efficiency (of all large turbo machines, power-generating or power-absorbing), equals its adiabatic efficiency, i.e.,

$$\eta_{pg} = \eta_a \text{ and } \eta_{pa} = \eta_a$$

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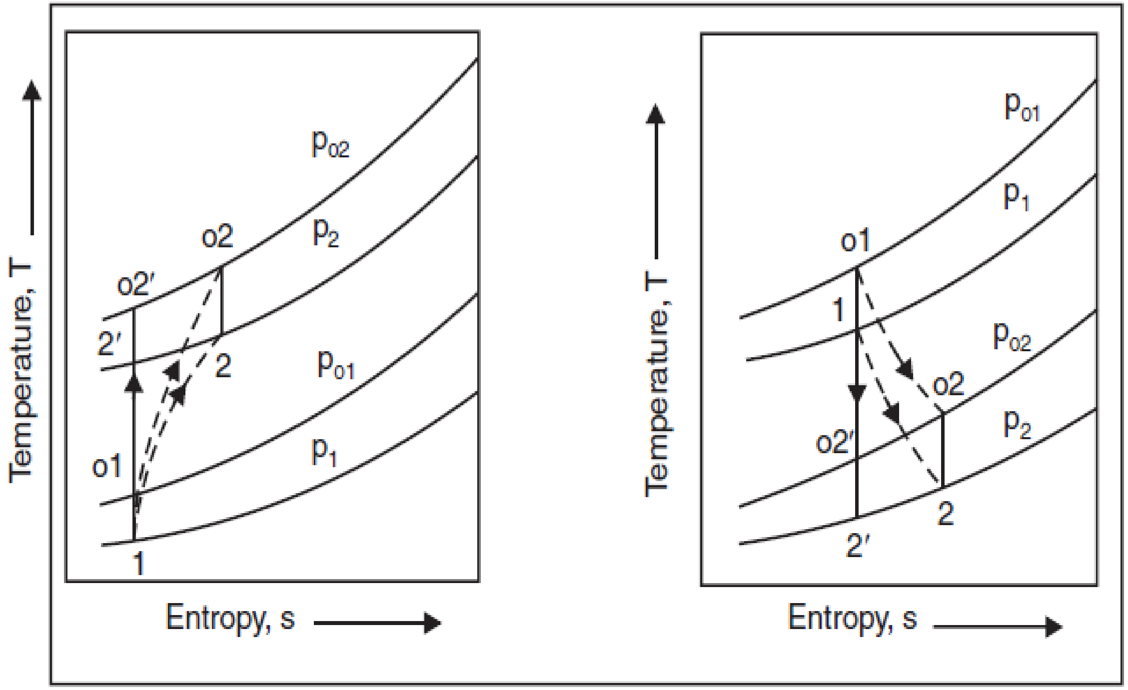

(a) Compression (b) Expansion

- The fluid has initially the static pressure and temperature at state 1
- State o1, is the corresponding stagnation state.
- After passing through the turbo machine, the final static properties of the fluid are at state 2.
- State o2 is the corresponding stagnation state.
- If the process were reversible, the final fluid static state would be 2′, and the stagnation state would be o2′.

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- The dashed-lines 1–2 in static coordinates and *o*1–*o*2 in stagnation coordinates represent the real process in each of the two figures.
- The actual work input or output *w*, is the quantity *ho*1–*ho*2 whereas the ideal work *wi*, can be calculated by any one of the following four equations:

(*i*) $w_{t-t} = h'_{o2} - h_{o1}$, (initial and final states both total),

(*ii*) $w_{t-s} = h'_2 - h_{o1}$, (initial state total, final state static),

(*iii*) $w_{s-t} = h'_{o2} - h_1$, (initial state static, final state total),

(*iv*) $w_{s-s} = h'_2 - h_1$. (initial and final states static).

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For power-absorbing machines, the applicable definitions of efficiency are:

$$(i)\quad \eta_{t-t} = (h_{o1} - h_{o2})/(h_{o1} - h_{o2}').$$

$$(ii)\quad \eta_{t-s} = (h_{o1} - h_{o2})/(h_{o1} - h_{2}').$$

$$(iii)\quad \eta_{s-t} = (h_{o1} - h_{o2})/(h_{1} - h_{o2}').$$

$$(iv)\quad \eta_{s-s} = (h_{o1} - h_{o2})/(h_{1} - h_{2}').$$

For power-absorbing machines, the applicable definitions of efficiency are:

$$(i)\quad \eta_{t-t} = (h_{o2}' - h_{o1})/(h_{o2} - h_{o1}).$$

$$(ii)\quad \eta_{t-s} = (h_{2}' - h_{o1})/(h_{o2} - h_{o1}).$$

$$(iii)\quad \eta_{s-t} = (h_{o2}' - h_{1})/(h_{o2} - h_{o1}).$$

$$(iv)\quad \eta_{s-s} = (h_{2}' - h_{1})/(h_{o2} - h_{o1}).$$

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Example 2: Air flows through an air turbine where its stagnation pressure is decreased in the ratio 5:1. The total-to-total efficiency is 0.8 and the air flow rate is 5 kg/s. If the total power output is 400 kW, find:

(a) the inlet total temperature, T_{o1}

(b) the actual exit total temperature, T_{o2}

(c) the actual exit static temperature, T_2 if the exit flow velocity is 100 m.s–1

(d) the total-to-static efficiency η_{t-s} of the device.

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Data: Air as a perfect gas, inlet-to-exit total pressure ratio $p_{o1}/p_{o2} = 5$, total-to-total efficiency $\eta_{t\text{-}t} = 0.8$, $m = 5$ kg/s, $P = 400$ kW and $V_2 = 100$ m/s.

Solution:

(a) Work output/unit mass flow of air, ($c_p = 1.004$ kJ.kg^{-1}K^{-1}).

$w = -\Delta h_o = -c_p(T_{o2} - T_{o1}) = P/\dot{m} = 400/5 = 80$ kJ.kg^{-1}

Thus, $T_{o2} - T_{o1} = -80.0/1.004 = -79.7$ K.

However, since the stagnation pressure ratio is 5 and the total-to-total efficiency is 0.8,

$T_{o2}'/T_{o1} = (p_{o2}/p_{o1})^{(\gamma-1)/\gamma} = (0.2)^{(0.4/1.4)} = 0.631.$

$T_{o1} - T_{o2} = \eta_{t\text{-}t}(T_{o1} - T_{o2}') = 0.8(1.0 - 0.631)T_{o1} = 79.7.$

$\boldsymbol{T_{o1} = 270}$ **K.**

(b) $\boldsymbol{T_{o2}} = T_{o1} - 79.7 = 270 - 79.7 = 190.3$ K.

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(c) For the static temperature, we have: $T = T_o - V^2/(2c_p)$,

so $\boldsymbol{T_2} = T_{o2} - V_2{}^2/(2c_p)$ = 190.3 – 1002/[(2)(1004)] = **185.3 K.**

(d) From *a*) and (*b*) above,

$T_{o2}' = 0.631 T_{o1}$ = (0.631)(270) = 170.4 K.

Hence, $T_2' = T_{o2}' - V_2{}^2/(2c_p)$ = 170.4 – 1002/[(2)(1004)] = 165.4 K

$\eta_{t\text{-}s} = (T_{o1} - T_{o2})/(T_{o1} - T_2')$ = (270 – 190.4)/(270 – 165.4) = **0.76**

Dimensionless Parameters

- Similarities law: geometric, kinematic and dynamic
- Dimensionless numbers: Re, Fr, Eu, We, M.
- Buckingham's π theorem
- Unit quantities: unit discharge/ power/ speed

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Example 3: An axial-flow pump with a rotor diameter of 300 mm handles liquid water at the rate of 162 $m3/h$ while operating at 1500 RPM. The corresponding energy input is 125 J/kg, the total-to-total efficiency being 75%. If a second geometrically similar pump with a diameter of 200 mm operates at 3000 RPM, what are its:

(*a*) flow rate

(*b*) change in total pressure

(*c*) input power

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Data: Pump rotor $D1 = 0.3$ m, $Q1 = 162$ m3/h, $N1 = 1500$ RPM. $\eta t\text{-}t = 0.75$. Pump output $w1 = 125$ J/kg. Second pump, $D2 = 0.2$ m, $N2 = 3000$ RPM.

(*a*) For dynamic similarity between the two pumps,

$\pi 2 = Q1/(N1D1^3) = Q2/(N2D2^3)$,

$Q2 = Q1\, N2D2^3/(N1D1^3) = (162)(3000)(0.2)^3/[(1500)(0.3^3)] = 96$ m3/h

(*b*) Since the head-coefficient is constant,

$E2 = E1(N2D2)^2/(N1D1)^2 = (125)(3000 \times 0.2)^2/(1500 \times 0.3)^2 = 222$ J/kg

Change in total pressure:

$\Delta \boldsymbol{po} = \rho\eta \boldsymbol{t\text{-}tE2} = (1000)(0.75)(222) = 1.66 \times 105$ N/m2 = **1.67 bar**

(*c*) Input power**,**

$P = \rho Q2E2 = (1000)(96/3600)(0.222) = 5920$ W = 5.92 kW. (E=gH)

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Effect of Reynolds Number

- Reynold's number: In pipe flow, it is an important parameter that represents either the flow is turbulent or laminar.
- It is also important for small pump, compressors, fans and blowers on which performance improves with increase in Re.
- Most of the turbo machines use relatively low viscosity fluids like air, water and light oil.
- Viscous action of the fluid has very little effect on the power output of the machine.
- Machine handling light fluids undergo efficiency changes under varying load conditions and sizes.

Effect of Specific Speed

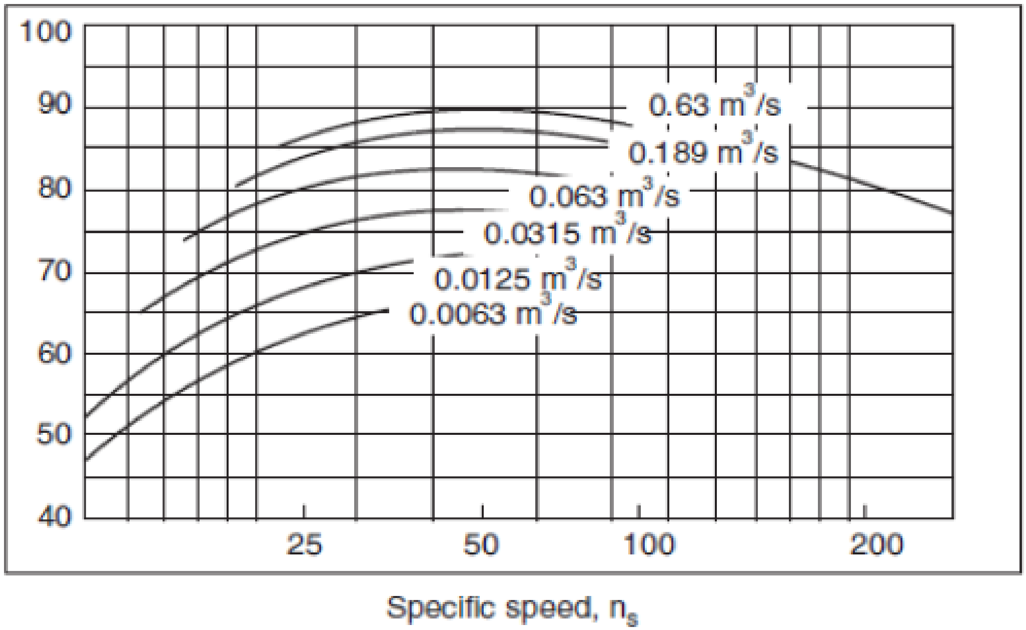

Fig. 1.7(a): Efficiency as a function of specific-speed in pumps. (After [17])

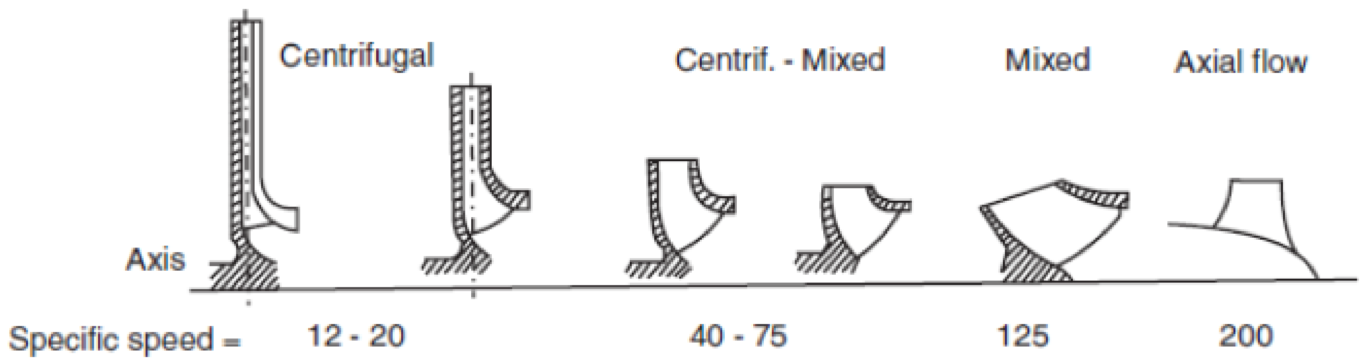

Fig. 1.7(b): Impeller shape variation with specific-speed in pumps. (After Troskolanski [17])

THANK YOU !!!