

# **THERMODYNAMICS II**

## **(ME 365)**

# **TUTORIAL LECTURE**

## **UNIT 1**

**Lead Facilitator**

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# OUTLINE

- HIGHLIGHTS ON THE 1<sup>ST</sup> LAW
- TUTORIAL PROBLEMS ON 1<sup>ST</sup> LAW
- HIGHLIGHTS ON THE 2<sup>ND</sup> LAW
- TUTORIAL PROBLEMS ON 2<sup>ND</sup> LAW

# HIGHLIGHT

## HIGHLIGHTS ON THE 1ST LAW

1. **Internal energy** is the heat energy in a gas. The internal energy of a perfect gas is function of temperature only.
  2. **First Law of thermodynamics states :**
    - Heat and work are mutually convertible but since energy can neither be created nor destroyed, the total energy associated with an energy conversion remains constant.
- OR
- No machine can produce energy without corresponding expenditure of energy, i.e. it impossible construct a perpetual motion machine of the first kind.

# HIGHLIGHTS

## HIGHLIGHTS ON THE 1ST LAW

### First law can be expressed as follows:

$Q \pm W = \Delta E$  (+sign used when European sign convention with work input regarded as positive and – sign used when American sign convention is adapted with work input regarded as negative)

$Q \pm W = \Delta U$  (if changes in electric, magnetic, chemical energies are absent and changes in potential and kinetic energies are neglected)

3. There can be no machine which would continuously supply mechanical work without some form of energy disappearing somewhere. Such a fictitious machine is called perpetual motion machine of the first kind, in brief PMM1. A PMM1 is thus impossible.

4. The energy of an isolated system remains constant.

# HIGHLIGHTS

## HIGHLIGHTS ON THE 1ST LAW

5. In case of

(i) **Reversible** constant volume process or Iso-volumetric process ( $v=\text{constant}$ )

$$\Delta u = c_v (T_2 - T_1); W = 0; Q = c_v (T_2 - T_1)$$

(ii) **Reversible** constant pressure process (isobaric process) ( $p = \text{constant}$ )

$$\Delta u = c_p (T_2 - T_1); W = -p(v_2 - v_1); Q = c_p (T_2 - T_1) \quad h = u + pv$$

(iii) Reversible constant temperature or Isothermal process

$$\Delta u = 0; Q = -W; W = -P_1 V_1 \ln \left( \frac{V_2}{V_1} \right)$$

1 = initial state, 2 = final state

(iv) Reversible adiabatic process (isentropic process) ( $pv^\gamma = \text{constant}$ )

$$Q = 0; W = mR \left( \frac{T_2 - T_1}{\gamma - 1} \right); W = \Delta U; \frac{T_2}{T_1} = \left( \frac{P_2}{P_1} \right)^{(\gamma-1)/\gamma} = \left( \frac{v_1}{v_2} \right)^{\gamma-1}$$

(v) Reversible polytropic process ( $pv^n = \text{constant}$ )

$$Q + W = \Delta U; W = mR \left( \frac{T_2 - T_1}{n - 1} \right); W = \Delta U; \frac{T_2}{T_1} = \left( \frac{P_2}{P_1} \right)^{(n-1)/n} = \left( \frac{v_1}{v_2} \right)^{n-1}$$

$$\Delta U = mc_v (T_2 - T_1); Q = \left( \frac{n - \gamma}{n - 1} \right) W$$

# HIGHLIGHTS

## HIGHLIGHTS ON THE 1ST LAW

6. Steady flow equation can be expressed as

$$Q + W + u_1 + \frac{c_1^2}{2} + gz_1 + p_1 v_1 = u_2 + \frac{c_2^2}{2} + gz_2 + p_2 v_2$$

$$Q + W + h_1 + \frac{c_1^2}{2} + gz_1 = h_2 + \frac{c_2^2}{2} + gz_2$$

Q = heat supplied per kg of fluid    W = work done by 1kg of fluid    u = internal energy per kg of fluid

h = enthalpy per kg of fluid    z = elevation above datum    g = acceleration due to gravity

p = pressure of the fluid    C = velocity of the fluid    pv = energy required per kg of fluid

The equation is applicable in any medium in any steady flow

7. During an adiabatic throttling process, enthalpy remains constant

# TUTORIAL

## TUTORIAL PROBLEMS ON THE 1ST LAW

1. A thin walled and well-insulated cylinder device, as shown in Figure Q1, contains 5 L of saturated liquid water at a constant pressure of 175 kPa. Water is stirred by a paddle wheel while a current of 8 A flows through it for 45 min through a resistor placed in the water. If one-half of the liquid is evaporated during this constant pressure process and the paddle-wheel work amounts to 400 kJ, determine the voltage of the source, in Volts. Also, show the process on a P-v diagram with respect to saturation lines.

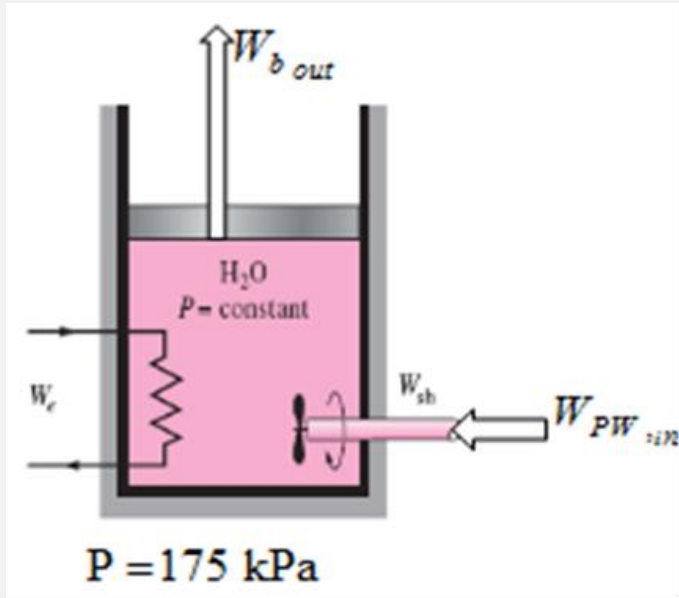


Figure Q1

# TUTORIAL

## TUTORIAL PROBLEMS ON THE 1ST LAW

### QUESTION 1 CLOSED SYSTEM CONSTANT PRESSURE PROCESS

**Assumptions** **1** The cylinder is stationary and thus the kinetic and potential energy changes are zero. **2** The cylinder is well-insulated and thus heat transfer is negligible. **3** The thermal energy stored in the cylinder itself is negligible. **4** The compression or expansion process is quasi-equilibrium.

**Analysis** We take the contents of the cylinder as the system. This is a closed system since no mass enters or leaves. The energy balance for this stationary closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$W_{e,\text{in}} + W_{pw,\text{in}} - W_{b,\text{out}} = \Delta U \quad (\text{since } Q = \text{KE} = \text{PE} = 0)$$

$$W_{e,\text{in}} + W_{pw,\text{in}} = m(h_2 - h_1)$$

$$(\mathbf{VI}\Delta t) + W_{pw,\text{in}} = m(h_2 - h_1)$$

Since  $\Delta U + W_b = \Delta H$  during a constant pressure quasi-equilibrium process. From steam tables, the properties of water are;



# TUTORIAL

## TUTORIAL PROBLEMS ON THE 1ST LAW

$$\left. \begin{array}{l} P_1 = 175 \text{ kPa} \\ \text{sat. liquid} \end{array} \right\} \begin{array}{l} h_1 = h_f @ 175 \text{ kPa} = 487.01 \text{ kJ/kg} \\ v_1 = v_f @ 175 \text{ kPa} = 0.001057 \text{ m}^3/\text{kg} \end{array}$$

$$\left. \begin{array}{l} P_2 = 175 \text{ kPa} \\ x_2 = 0.5 \end{array} \right\} h_2 = h_f + x_2 h_{fg} = 487.01 + (0.5 \times 2213.1) = 1593.6 \text{ kJ/kg}$$

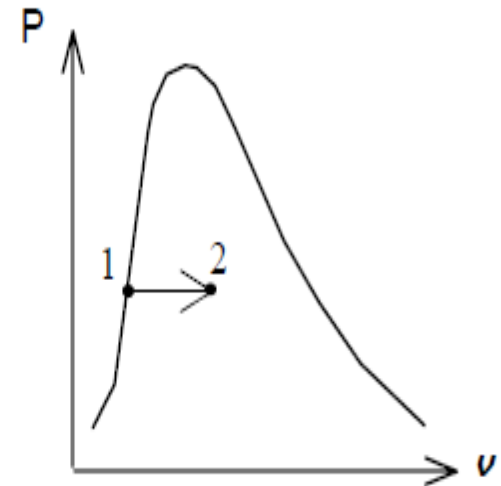
$$m = \frac{V_1}{v_1} = \frac{0.005 \text{ m}^3}{0.001057 \text{ m}^3/\text{kg}} = 4.731 \text{ kg}$$

Substituting,

$$VI\Delta t + (400 \text{ kJ}) = (4.731 \text{ kg})(1593.6 - 487.01) \text{ kJ/kg}$$

$$VI\Delta t = 4835 \text{ kJ}$$

$$V = \frac{4835 \text{ kJ}}{(8 \text{ A})(45 \times 60 \text{ s})} \left( \frac{1000 \text{ VA}}{1 \text{ kJ/s}} \right) = 223.9 \text{ V}$$



# TUTORIAL

## TUTORIAL PROBLEMS ON THE 1ST LAW

2. Consider a 1000-W iron as shown in figure Q2, whose base plate is made of 0.5-cm-thick aluminum alloy 2024-T6 ( $\rho = 2770 \text{ kg/m}^3$  and  $C_p = 875 \text{ J/kg} \cdot ^\circ\text{C}$ ), as shown in Figure Q2. The base plate has a surface area of  $0.03 \text{ m}^2$ . Initially, the iron is in thermal equilibrium with the ambient air at  $22^\circ\text{C}$ . Assuming 85 percent of the heat generated in the resistance wires is transferred to the plate; determine the minimum time, in seconds, needed for the plate temperature to reach  $140^\circ\text{C}$ .



**Figure Q2**

# TUTORIAL

## QUESTION 2 CLOSED SYSTEM WITH HEAT TRANSFER IN AND NO HEAT LOSS

**Assumptions 1** It is given that 85 percent of the heat generated in the resistance wires is transferred to the plate. **2** The thermal properties of the plate are constant. **3** Heat loss from the plate during heating is disregarded since the minimum heating time is to be determined. **4** There are no changes in kinetic and potential energies. **5** The plate is at a uniform temperature at the end of the process.

**Properties:** The density and specific heat of the aluminum alloy plate are given to be

$$\rho = 2770 \text{ kg/m}^3 \text{ and } C_p = 875 \text{ kJ/kg}\cdot^\circ\text{C}.$$

**Analysis** The mass of the iron's base plate is

$$m = \rho V = \rho LA = (2770 \text{ kg/m}^3)(0.005 \text{ m})(0.03 \text{ m}^2) = 0.4155 \text{ kg}$$

Noting that only 85 percent of the heat generated is transferred to the plate, the rate of heat transfer to the iron's base plate is

$$\dot{Q}_{\text{in}} = 0.85 \times 1000 \text{ W} = 850 \text{ W}$$

We take plate to be the system. The energy balance for this closed system can be expressed as

$$\underbrace{E_{\text{in}} - E_{\text{out}}}_{\text{Net energy transfer by heat, work, and mass}} = \underbrace{\Delta E_{\text{system}}}_{\text{Change in internal, kinetic, potential, etc. energies}}$$

$$Q_{\text{in}} = \Delta U_{\text{plate}} = m(u_2 - u_1) \longrightarrow \dot{Q}_{\text{in}} \Delta t = mc(T_2 - T_1)$$

Solving for  $\Delta t$  and substituting,

$$\Delta t = \frac{mc\Delta T_{\text{plate}}}{\dot{Q}_{\text{in}}} = \frac{(0.4155 \text{ kg})(875 \text{ J/kg}\cdot^\circ\text{C})(140 - 22)^\circ\text{C}}{850 \text{ J/s}} = 50.5 \text{ s}$$

Which is the time required for the plate temperature to reach the specified temperature.

# TUTORIAL

## TUTORIAL PROBLEMS ON THE 1ST LAW

3. A hand-held domestic hair-dryer, **such as the one shown in Figure Q4**, is basically a duct in which a few layers of electric resistors are placed. A small fan pulls the air in and forces it through the resistors where it is heated. Air ( $R = 0.287 \text{ kJ/kg K}$ ;  $C_p = 1.005 \text{ kJ/kg K}$ ) enters a 1200-W hair dryer at 100 kPa and  $22^\circ\text{C}$  and leaves at  $47^\circ\text{C}$ . The cross-sectional area of the hair dryer at the exit is  $60 \text{ cm}^2$ . Neglecting the power consumed by the fan and the heat losses through the walls of the hair dryer, determine

- (a) The volume flow rate of air at the inlet to the dryer, in  $\text{m}^3/\text{s}$ , and
- (b) the velocity of the air at the exit of the dryer, in  $\text{m/s}$ .

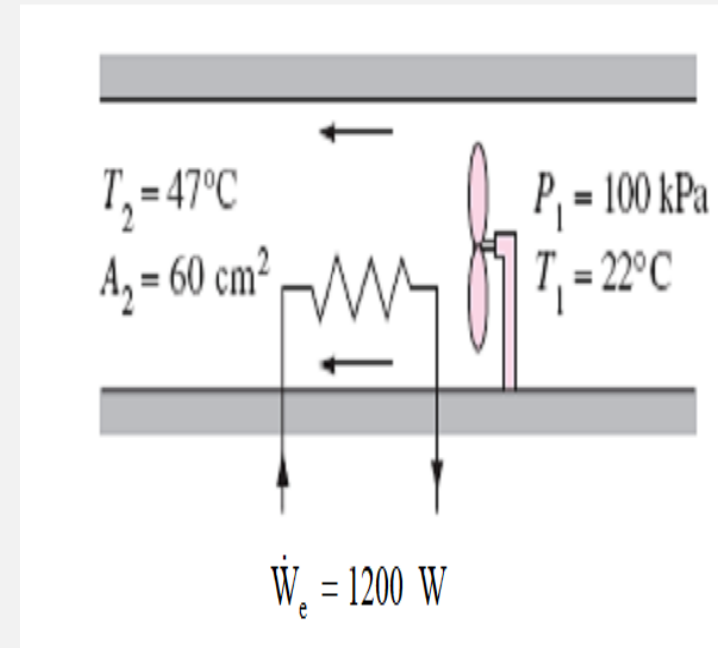


Figure Q4

# TUTORIAL

## QUESTION 3 OPEN SYSTEM WITH IDEAL GAS (AIR) AS THE SYSTEM

**Assumptions 1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats at room temperature. **3** Kinetic and potential energy changes are negligible. **4** The power consumed by the fan and the heat losses are negligible.

**Properties** The gas constant of air is 0.287 kJ/kg.K. The constant pressure specific heat of air at room temperature is  $C_p = 1.005$  kJ/kg.K

**Analysis** We take the hair dryer as the system, which is a control volume (open system) since mass crosses the boundary. There is only one inlet and one exit, and thus  $\dot{m}_1 = \dot{m}_2 = \dot{m}$ . The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{\Delta \dot{E}_{\text{system}}}_{\text{Rate of change in internal, kinetic, potential, etc. energies}} \stackrel{\text{no (steady)}}{=} 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{W}_{e,\text{in}} + \dot{m}h_1 = \dot{m}h_2 \quad (\text{since } \dot{Q}_{\text{out}} \cong \Delta \text{ke} \cong \Delta \text{pe} \cong 0)$$

$$\dot{W}_{e,\text{in}} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Substituting, the mass and volume flow rates of air are determined to be

$$\dot{m} = \frac{\dot{W}_{e,\text{in}}}{c_p(T_2 - T_1)} = \frac{1.2 \text{ kJ/s}}{(1.005 \text{ kJ/kg} \cdot ^\circ\text{C})(47 - 22)^\circ\text{C}} = 0.04776 \text{ kg/s}$$

$$\nu_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(295 \text{ K})}{(100 \text{ kPa})} = 0.8467 \text{ m}^3/\text{kg}$$

$$\dot{V}_1 = \dot{m}\nu_1 = (0.04776 \text{ kg/s})(0.8467 \text{ m}^3/\text{kg}) = 0.0404 \text{ m}^3/\text{s}$$

(b) The exit velocity of air is determined from the mass balance  $\dot{m}_1 = \dot{m}_2 = \dot{m}$  to be

$$\nu_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(320 \text{ K})}{(100 \text{ kPa})} = 0.9184 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{\nu_2} A_2 V_2 \longrightarrow V_2 = \frac{\dot{m}\nu_2}{A_2} = \frac{(0.04776 \text{ kg/s})(0.9184 \text{ m}^3/\text{kg})}{60 \times 10^{-4} \text{ m}^2} = 7.31 \text{ m/s}$$

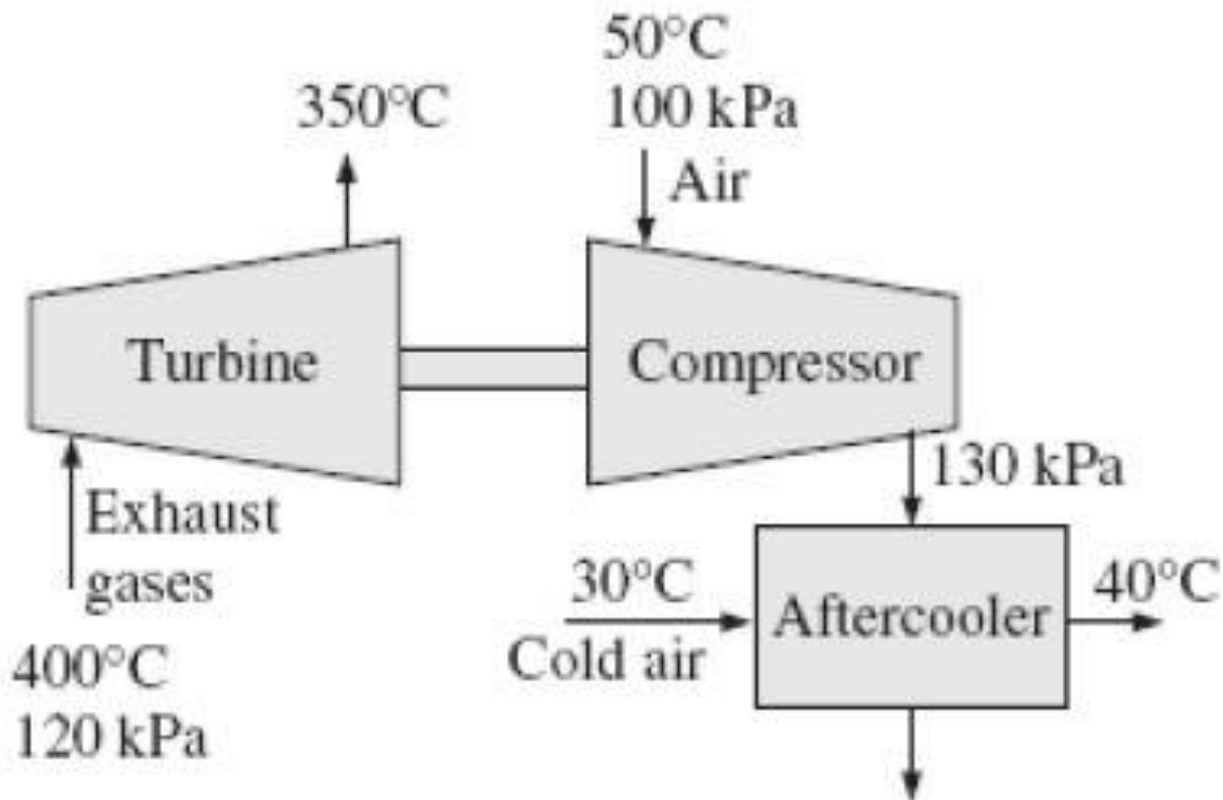
# TUTORIAL

4. The schematic of a turbocharger of an internal combustion engine, as depicted in Figure Q5, consists of a turbine and a compressor. Hot exhaust gases flow through the turbine to produce work and the work output from the turbine is used as the work input to the compressor. The pressure of ambient air is increased as it flows through the compressor before it enters the engine cylinders. Thus, the purpose of a turbocharger is to increase the pressure of air so that more air gets into the cylinder. Consequently, more fuel can be burned and more power can be produced by the engine. In a turbocharger, exhaust gases enter the turbine at 400 °C and 120 kPa at a rate of 0.02 kg/s and leave at 350 °C. Air enters the compressor at 50 °C and 100 kPa and leaves at 130 kPa at a rate of 0.018 kg/s. The compressor increases the air pressure with a side effect: It also increases the air temperature, which increases the possibility of a gasoline engine to experience an engine knock. To avoid this, an after-cooler is placed after the compressor to cool the warm air by cold ambient air before it enters the engine cylinders. It is estimated that the after-cooler must decrease the air temperature below 80 °C if knock is to be avoided. The cold ambient air enters the after-cooler at 30 °C and leaves at 40 °C. Disregarding any frictional losses in the turbine and the compressor and treating the exhaust gases as air, determine (a) the temperature of the air at the compressor outlet, °C, and (b) the minimum volume flow rate of ambient air required to avoid knock, in m<sup>3</sup>/s.

You may assume the following values:

$$C_{p\text{air}} = 1.005 \text{ kJ/(kg air K)} \quad C_{p\text{exhaust gas}} = 1.0063 \text{ kJ/(kg K)}; \quad C_{p\text{warm air}} = 1.008 \text{ kJ/kg K}$$

# TUTORIAL



**Figure Q5**

# TUTORIAL

## QUESTION 4 OPEN SYSTEM (AIR AS THE SYSTEM)

**Assumptions** 1 All processes are steady since there is no change with time. 2 Kinetic and potential energy changes are negligible. 3 Air properties are used for exhaust gases. 4 Air is an ideal gas with constant specific heats. 5 The mechanical efficiency between the turbine and the compressor is 100%. 6 All devices are adiabatic. 7 The local atmospheric pressure is 100 kPa.

**Properties** The constant pressure specific heats of exhaust gases, warm air, and cold ambient air are taken to be  $C_p = 1.063, 1.008, \text{ and } 1.005 \text{ kJ/kg}\cdot\text{K}$ , respectively

**Analysis** (a) An energy balance on turbine (an open system) gives

$$\dot{W}_T = \dot{m}_{\text{exh}} c_{p,\text{exh}} (T_{\text{exh},1} - T_{\text{exh},2}) = (0.02 \text{ kg/s})(1.063 \text{ kJ/kg}\cdot\text{K})(400 - 350)\text{K} = 1.063 \text{ kW}$$

This is also the power input to the compressor since the mechanical efficiency between the turbine and the compressor is assumed to be 100%. An energy balance on the compressor gives the air temperature at the compressor outlet

$$\dot{W}_C = \dot{m}_a c_{p,a} (T_{a,2} - T_{a,1})$$

$$1.063 \text{ kW} = (0.018 \text{ kg/s})(1.008 \text{ kJ/kg}\cdot\text{K})(T_{a,2} - 50)\text{K} \longrightarrow T_{a,2} = 108.6^\circ\text{C}$$

(b) An energy balance on the after cooler gives the mass flow rate of cold ambient air

$$\dot{m}_a c_{p,a} (T_{a,2} - T_{a,3}) = \dot{m}_{\text{ca}} c_{p,\text{ca}} (T_{\text{ca},2} - T_{\text{ca},1})$$

$$(0.018 \text{ kg/s})(1.008 \text{ kJ/kg}\cdot^\circ\text{C})(108.6 - 80)^\circ\text{C} = \dot{m}_{\text{ca}} (1.005 \text{ kJ/kg}\cdot^\circ\text{C})(40 - 30)^\circ\text{C}$$

$$\dot{m}_{\text{ca}} = 0.05161 \text{ kg/s}$$

The volume flow rate may be determined if we first calculate specific volume of cold ambient air at the inlet of after cooler. That is,

$$\nu_{\text{ca}} = \frac{RT}{P} = \frac{(0.287 \text{ kJ/kg}\cdot\text{K})(30 + 273 \text{ K})}{100 \text{ kPa}} = 0.8696 \text{ m}^3/\text{kg}$$

$$\dot{V}_{\text{ca}} = \dot{m} \nu_{\text{ca}} = (0.05161 \text{ kg/s})(0.8696 \text{ m}^3/\text{kg}) = 0.0449 \text{ m}^3/\text{s} = 44.9 \text{ L/s}$$



# HIGHLIGHTS

## HIGHLIGHTS ON THE 2ND LAW

### 1. CLAUSIUS STATEMENT

“It is impossible for a self-acting machine working in a cyclic process, unaided by external agency, to convey heat from a body at a lower temperature to a body at a higher temperature”

### Kelvin-Planck statement:

“It is impossible to construct an engine, which while operating in a cycle produces no other effect except to extract heat from a single- reservoir and do an equivalent amount of work”

Although the above statements of the second law of thermodynamics appear to be different, they are really equivalent in the sense that violation of either statement implies violation of the other.

**2. PERPETUAL MOTION MACHINE OF THE SECOND KIND (PMM2)** is that imaginary machine which would continuously absorb heat from a single thermal reservoir and convert this heat completely into work. The efficiency of such a machine would be 100%.

# HIGHLIGHTS

## HIGHLIGHTS ON THE 2ND LAW

### 3. CLAUSIUS INEQUALITY is given by

$$\sum_{\text{Cycle}} \left( \frac{\delta Q}{T} \right) \leq 0$$

When a system performs a reversible cycle

$$\sum_{\text{Cycle}} \left( \frac{\delta Q}{T} \right) = 0$$

When the cycle is irreversible,

$$\sum_{\text{Cycle}} \left( \frac{\delta Q}{T} \right) < 0$$

**4. ENTROPY** is a function of a quantity of heat which shows the possibility of conversion of heat into work. The increase in entropy is small when heat is added at a high temperature and is greater when heat addition is made at lower temperature. Thus for maximum entropy, there is a minimum availability for conversion into work and for minimum entropy there is maximum availability for conversion into work.

# HIGHLIGHTS

## HIGHLIGHTS ON THE 2ND LAW

### 5. ENTROPY CHANGES FOR A CLOSED SYSTEM PER UNIT MASS OF THE SYSTEM for various processes

#### (i) General case

$$(a) \ s_2 - s_1 = c_v \ln \left( \frac{T_2}{T_1} \right) + R \ln \left( \frac{v_2}{v_1} \right) \text{ (in terms of T and v)}$$

$$(b) \ s_2 - s_1 = c_v \ln \left( \frac{P_2}{P_1} \right) + c_p \ln \left( \frac{v_2}{v_1} \right) \text{ (in terms of P and v)}$$

$$(c) \ s_2 - s_1 = c_p \ln \left( \frac{T_2}{T_1} \right) - R \ln \left( \frac{P_2}{P_1} \right) \text{ (in terms of T and P)}$$

#### (ii) Constant volume process:

$$s_2 - s_1 = c_v \ln \left( \frac{T_2}{T_1} \right)$$

#### (iii) Constant pressure process

$$s_2 - s_1 = c_p \ln \left( \frac{T_2}{T_1} \right)$$

#### (iv) Isothermal process:

$$s_2 - s_1 = R \ln \left( \frac{v_2}{v_1} \right)$$

#### (v) Adiabatic process

$$s_2 - s_1 = 0$$

# HIGHLIGHTS

## HIGHLIGHTS ON THE 2ND LAW

**(vi) Polytropic process:**

$$s_2 - s_1 = c_v \left( \frac{n - \gamma}{n - 1} \right) \ln \left( \frac{T_2}{T_1} \right)$$

### 6. ENTROPY CHANGE FOR AN OPEN SYSTEM

$$dS \geq \left( \frac{dQ}{T_o} \right) + \sum_i s_i \cdot dm_i - \sum_o s_o \cdot dm_o$$

Where,  $T_o$  = temperature of the surroundings

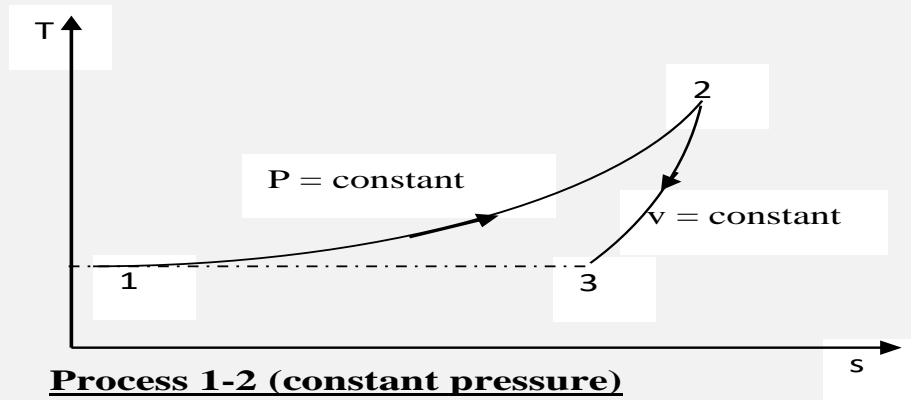
Subscripts i and o refer to inlet and outlet conditions

# TUTORIAL

## TUTORIAL PROBLEMS ON THE 2ND LAW

1.  $1 \text{ m}^3$  of air is heated reversibly at constant pressure from 15 to 300 °C, and is then cooled reversibly at constant volume back to the initial temperature. The initial pressure is 1.03 bar. Calculate the net heat flow and the overall change of entropy, and sketch the processes on a  $T - s$  diagram.

# TUTORIAL



## Process 1-2 (constant pressure)

$$m = \frac{PV}{RT} = \frac{1.03 \times 10^2 \times 1}{0.287 \times 288} = 1.25 \text{ kg}$$

$$Q_{12} = mc_p (T_2 - T_1) = 1.25 \times 1.005 \times (573 - 288) = 358 \text{ kJ}$$

$$\Delta S = m c_p \ln \left( \frac{T_2}{T_1} \right) = 1.25 \times 1.005 \times \ln \left( \frac{573}{288} \right) = 0.864 \text{ kJ/K}$$

## Process 2-3 (constant volume)

$$Q_{23} = mc_v (T_3 - T_2) = 1.25 \times 0.718 \times (288 - 573) = -256 \text{ kJ}$$

$$\Delta S = mc_v \ln \left( \frac{T_3}{T_2} \right) = 1.25 \times 0.718 \times \ln \left( \frac{288}{573} \right) = -0.617 \text{ kJ/K}$$

## Net heat flow

$$Q_{\text{net}} = Q_{12} + Q_{23} = 358 - 256 = 102 \text{ kJ}$$

## Net entropy change

$$\begin{aligned} (\Delta S)_{\text{net}} &= (\Delta S)_{12} + (\Delta S)_{23} \\ &= 0.864 - 0.617 = 0.247 \text{ kJ/K} \end{aligned}$$

# TUTORIAL

## TUTORIAL PROBLEMS ON THE 2ND LAW

2. A system of 100 kg mass undergoes a process in which the specific entropy increases from 0.3 kJ/(kg.K) to 0.4 kJ/(kg. K). At the same time, the entropy of the surroundings of the system decreases from 80 kJ/K to 75 kJ/K. This process is:  
(a) reversible (b) irreversible (c) reversible and adiabatic (d) impossible

**Hint:** Determine the total change in entropy of the system and use the answer to select the correct option

# TUTORIAL

## TUTORIAL PROBLEMS ON THE 2ND LAW

The total entropy of universe never decreases during a process undergone by a system. That is

$$\Delta S_{total} \geq 0$$

$$\Delta S_{total} = \Delta S_{system} + \Delta S_{surroundings}$$

$$\Delta S_{system} = m\Delta s_{system} = m(s_f - s_i) = 100 \text{ kg} \times (0.4 - 0.3) \text{ kJ/K} = 10 \text{ kJ/K}$$

$$\Delta S_{surroundings} = (75 - 80) \text{ kJ/K} = -5 \text{ kJ/K}$$

$$\Delta S_{total} = +10 \text{ kJ/K} - 5 \text{ kJ/K} = +5 \text{ kJ/K} > 0 \rightarrow \text{the process undergone by the system is irreversible}$$



# TUTORIAL

## TUTORIAL PROBLEMS ON THE 2ND LAW

3. Air is flowing through a well-lagged duct. At section A, the pressure and temperature are 1.5 bar and  $187^{\circ}\text{C}$ , respectively, and at some distance along the duct at a section B, the pressure and temperature are 1.3 bar and  $160^{\circ}\text{C}$ . In what direction is the air flowing.

**Hint:** the total change in entropy of the system must be increasing in the direction of flow of the air since there are irreversibility's involved.

# TUTORIAL

$$C_{p, \text{air}} = 1.005 \text{ kJ/kg K}; R_{\text{air}} = 0.287 \text{ kJ/kg K}$$

At section A of the thermally insulated duct,  $P_A = 150 \text{ kN/m}^2$ ,  $T_A = 187 + 273.15 = 460.15 \text{ K}$

At section B of the thermally insulated duct,  $P_B = 130 \text{ kN/m}^2$ ,  $T_B = 160 + 273.15 = 433.15 \text{ K}$

The entropy change of the system for flow of air from i to f is given by  $\Delta S_{\text{system}} = S_f - S_i$

For flow in a thermally insulated duct the change in entropy of the system must increase in the direction of flow to satisfy the entropy of the universe principle.

If the flow of the air is from section A to section B then the entropy change for flow in that direction is given by

$$\begin{aligned} \Delta S_{AB} &= c_{p, \text{air}} \ln(T_B/T_A) - R \ln(P_B/P_A) = 1.005 \ln(433.15/460.15) - 0.287 \ln(130/150) \\ &= -0.061 + 0.041 \text{ kJ/K} = -0.02 \text{ kJ/K} < 0 \text{ kJ/K} \end{aligned}$$

If the flow of air is from B to A then

$$\begin{aligned} \Delta S_{BA} &= c_{p, \text{air}} \ln(T_A/T_B) - R \ln(P_A/P_B) = 1.005 \ln(460.15/433.15) - 0.287 \ln(150/130) \\ &= +0.061 - 0.041 \text{ kJ/K} = +0.02 \text{ kJ/K} > 0 \text{ kJ/K} \end{aligned}$$

$\Delta S_{BA} > 0$  is the desired flow direction. Hence the flow of the air is from B to A

# TUTORIAL

## TUTORIAL PROBLEMS ON THE 2ND LAW

4. 300 kJ/s of heat is supplied at a constant temperature of  $200^{\circ}\text{C}$  to a heat engine. The heat rejection takes place at  $8.5^{\circ}\text{C}$ . The following results were obtained:

- (i) 215 kJ/s are rejected
- (ii) 150 kJ/s are rejected
- (iii) 75 kJ/s are rejected

Classify which of the results report a reversible cycle or irreversible cycle or impossible results.

# TUTORIAL

Heat supplied to the heat engine at  $(290+273.15) \text{ K} = +300 \text{ kJ/s}$

Heat is rejected at  $(8.5 +273.15) \text{ K}$

Applying the Clausius inequality to the cycle or process, we have:

$$(i) \sum(dQ/T) = +300/563.15 + (-215/282.65) = 0.5328 - 0.7637 = -0.2309 < 0$$

Therefore the cycle is **irreversible**

$$(ii) \sum(dQ/T) = +300/563.15 + (-150/282.65) = 0.5328 - 0.5328 = 0$$

Therefore the cycle is **reversible**

$$(iii) \sum(dQ/T) = +300/563.15 + (-75/282.65) = 0.5328 - 0.2664 = +0.2664 > 0$$

Therefore the cycle is **impossible** by the Clausius inequality.

**TUTORIAL ENDS**

**THANK YOU**

**LUKE 19:13**

# **THERMODYNAMICS II**

## **(ME 365)**

# **TUTORIAL LECTURE**

## **UNIT 2**

**Lead Facilitator**  
**Prof. F. K. Forson**

# OUTLINE

- HIGHLIGHTS ON THE VAPOUR POWER CYCLE
- TUTORIAL PROBLEMS ON VAPOUR POWER CYCLE
- HIGHLIGHTS ON THE GAS POWER CYCLE
- TUTORIAL PROBLEMS ON GAS POWER CYCLE

# **HIGHLIGHT**

## **HIGHLIGHTS ON VAPOUR POWER CYCLE**

- Under vapour power cycles, I have considered practical arrangements for vapour power plants, illustrated how vapour power plants are modelled thermodynamically, and considered the principal irreversibility and losses associated with such plants.
- The main components of simple vapour power plants were modeled by the Rankine cycle.



# HIGHLIGHT

## HIGHLIGHTS ON VAPOUR POWER CYCLE

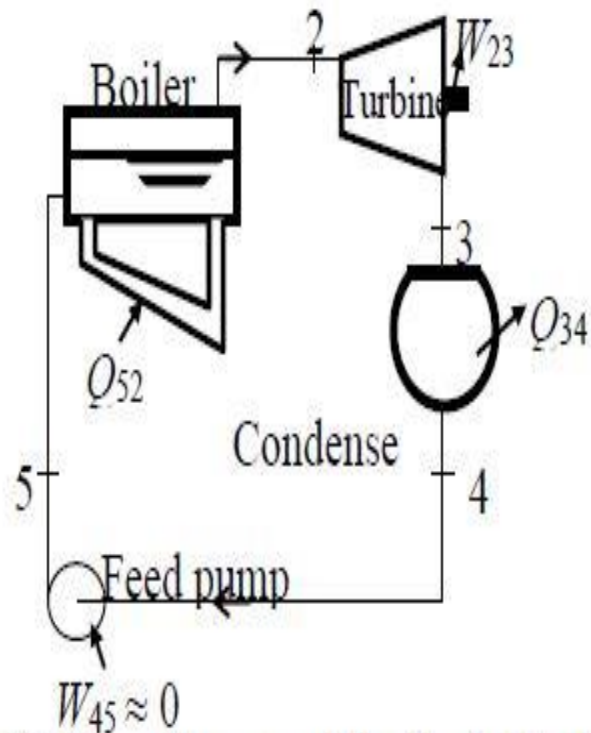


Figure 17: Flow diagram of the Basic Rankine Cycle

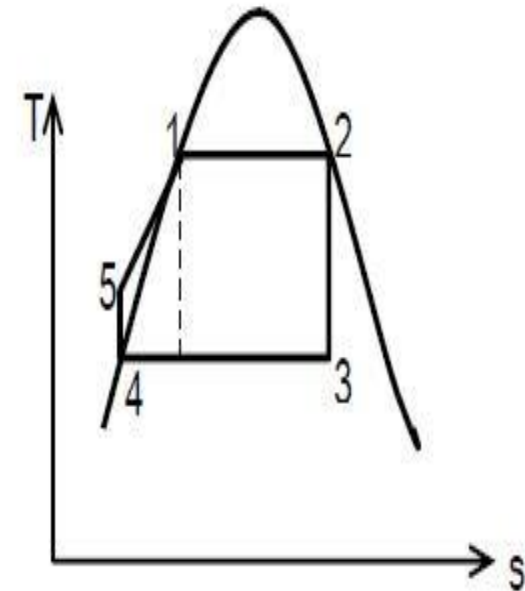


Figure 18: T-s diagram of the Basic Rankine Cycle

# HIGHLIGHT

## HIGHLIGHTS ON VAPOUR POWER CYCLE

- We also introduced modifications to the simple vapour power cycle aimed at improving overall performance.

- These included superheat, reheat, regeneration, and cogeneration cycles.

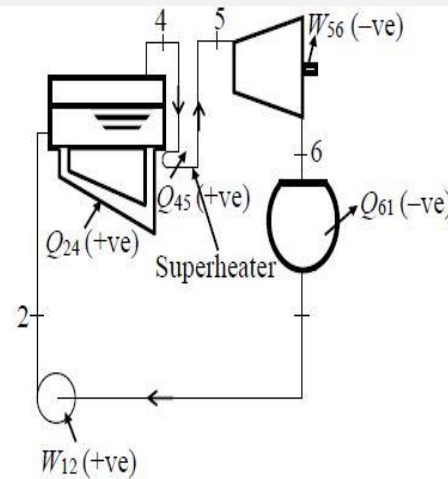


Figure 23: The Basic Rankine cycle with superheat

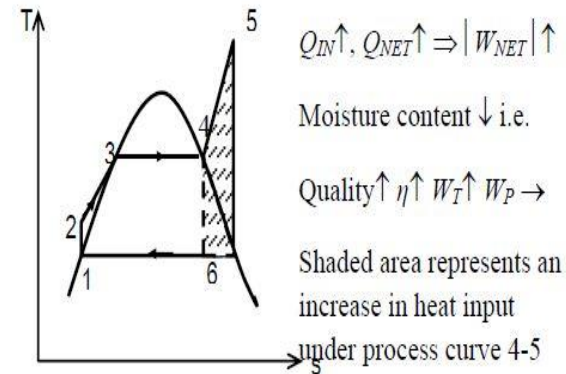


Figure 24: T-s diagram of the Basic Rankine cycle with superheat

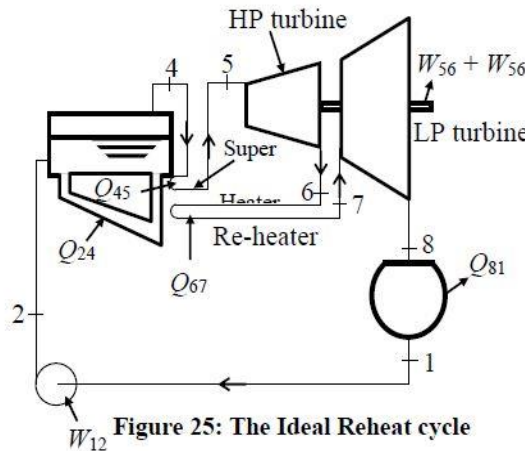


Figure 25: The Ideal Reheat cycle

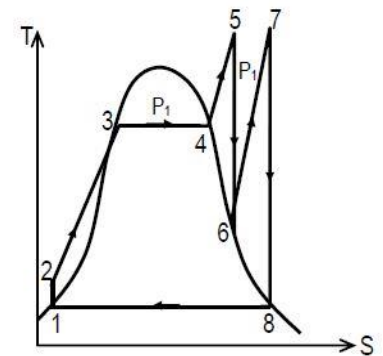
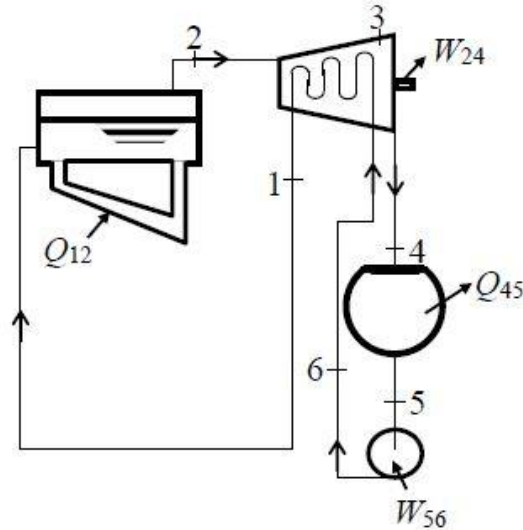
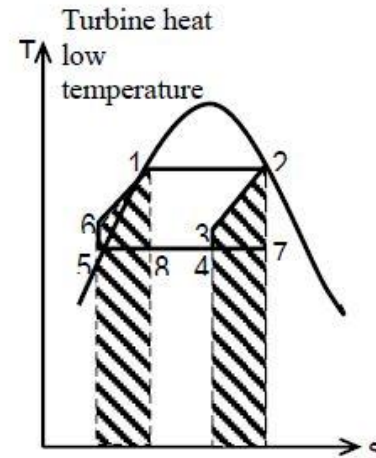


Figure 26: T-s diagram of the Ideal Reheat cycle

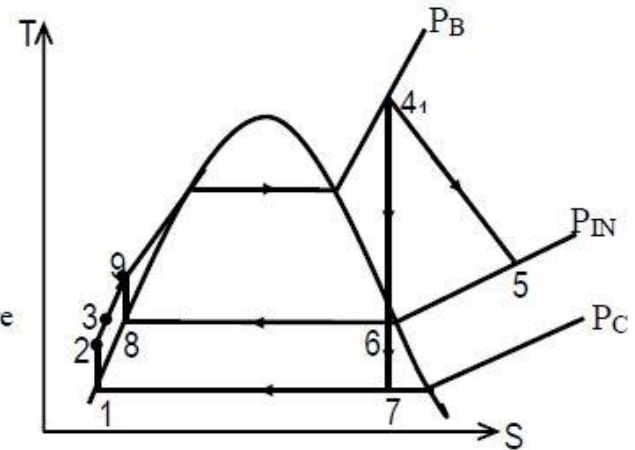
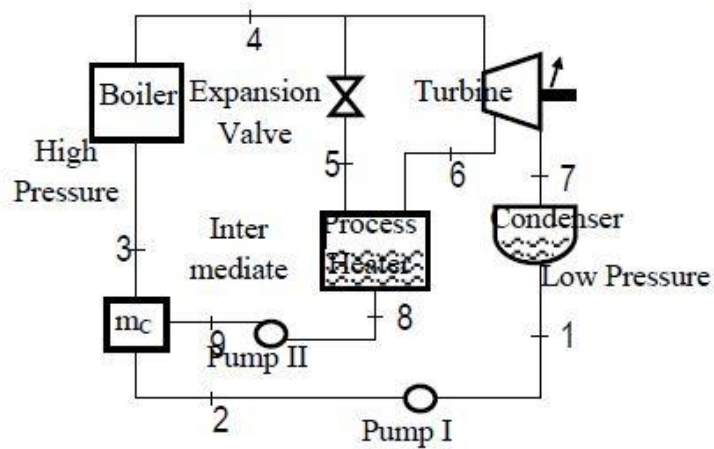
# HIGHLIGHT



**Figure 27: A hypothetical regenerative cycle**



**Figure 28: Corresponding T-s diagram**



**Figure 34: Schematic of a cogeneration plant with T-s diagram**

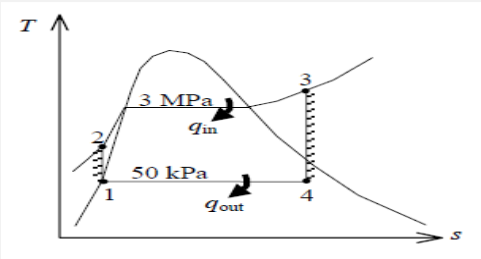
# **TUTORIAL**

## **TUTORIAL PROBLEMS ON VAPOUR POWER CYCLE**

**1. A steam plant operates on a simple ideal Rankine cycle between the pressure limits of 3 MPa and 50 kPa. The temperature of the steam at the turbine inlet is 400 °C, and the mass flow rate of steam through the cycle is 25 kg/s. Show the cycle on a T-s diagram with respect to saturation lines, and determine**

- (a) the thermal efficiency of the cycle and**
- (b) the net power output of the power plant**

# TUTORIAL



**Assumptions:** 1 Steady operating condition exists. 2 Kinetic and potential energy changes are negligible.

Analysis (a) from the steam tables

$$h_1 = h_f @ 50 \text{ kPa} = 340.54 \text{ kJ/kg}$$

$$v_1 = v_f @ 50 \text{ kPa} = 0.001030 \text{ m}^3/\text{kg}$$

$$w_{p,in} = v_1(P_2 - P_1)$$

$$= (0.001030 \text{ m}^3/\text{kg})(3000 - 50) \text{ kPa} = 3.04 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{p,in} = 340.54 + 3.04 = 343.58 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_3 = 30 \text{ bar} \\ T_3 = 400^\circ\text{C} \end{array} \right\} h_3 = 3232 \text{ kJ/kg}, s_3 = 6.923 \text{ kJ/kg}\cdot\text{K}$$

$$\left. \begin{array}{l} P_4 = 0.5 \text{ bar} \\ s_4 = s_3 \end{array} \right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.923 - 1.091}{6.502} = 0.897$$

$$h_4 = h_f + x_4 h_{fg} = 340.54 + 0.897(2305) = 2408.12 \text{ kJ/kg}$$

Thus

$$q_{in} = q_{23} = h_3 - h_2 = 3232 - 343.58 = 2888.42 \text{ kJ/kg}$$

$$q_{out} = q_{41} = h_1 - h_4 = 340.58 - 2408.12 = -2068 \text{ kJ/kg}$$

$$|w_{net}| = |w_T + w_p| = |(h_4 - h_3) + (h_2 - h_1)| = |(2408.12 - 3232) + (3)| = 820.88 \text{ kJ/kg}$$

$$\eta_{th} = 1 + \frac{q_{out}}{q_{in}} = 1 - \frac{2068}{2888.42} = 0.284 = 28.4\%$$

$$\dot{W}_{net} = \dot{m}|w_{net}| = 25 \times 820.88 = 20,552 \text{ kW}$$

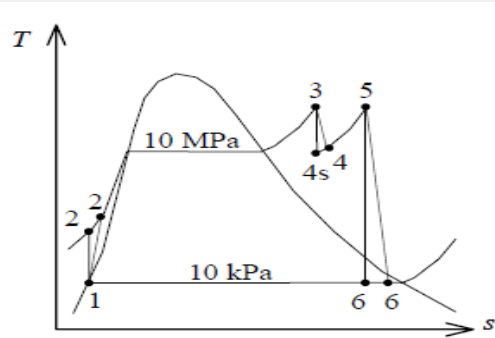
# **TUTORIAL**

## **TUTORIAL PROBLEMS ON VAPOUR POWER CYCLE**

**2. Consider a steam power plant that operates on a reheat Rankine cycle and has a net power output of 150 MW. Steam enters the high-pressure turbine at 10 MPa and 500 °C and the low-pressure turbine at 1 MPa and 500 °C. Steam leaves the condenser as a saturated liquid at a pressure of 10 kPa. The adiabatic efficiency of the turbine is 80 %, and that of the pump is 95%. Show the cycle on a T-s diagram with respect to saturation lines, and determine**

- (a) the quality (or temperature, if superheated) of steam at the turbine exit,**
- (b) the thermal efficiency of the cycle, and**
- (c) the mass flow rate of steam.**

# TUTORIAL



$$h_1 = h_f @ 10 \text{ kPa} = 191.81 \text{ kJ/kg}$$

$$v_1 = v_f @ 10 \text{ kPa} = 0.001010 \text{ m}^3/\text{kg}$$

$$w_{p,\text{in}} = v_1(P_2 - P_1)/\eta_p = \frac{0.001010(10000 - 10)}{0.95} = 10.62 \text{ kJ/kg} \quad h_2 = h_1 + w_{p,\text{in}}$$

$$= 191.81 + 10.62 = 202.43 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_3 = 10 \text{ MPa} \\ T_3 = 500^\circ\text{C} \end{array} \right\} \begin{array}{l} h_3 = 3375.1 \text{ kJ/kg} \\ s_3 = 6.5995 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_{4s} = 1 \text{ MPa} \\ s_{4s} = s_3 \end{array} \right\} h_{4s} = 2783.8 \text{ kJ/kg}$$

$$\eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} \rightarrow h_4 = h_3 - \eta_T(h_3 - h_{4s})$$

$$\begin{aligned} h_4 &= 3375.1 - 0.8(3375.1 - 2783.8) \\ &= 2902.1 \text{ kJ/kg} \end{aligned}$$

$$\left. \begin{array}{l} P_5 = 1 \text{ MPa} \\ T_5 = 500^\circ\text{C} \end{array} \right\} \begin{array}{l} h_5 = 3479.1 \text{ kJ/kg} \\ s_5 = 7.7642 \text{ kJ/kg} \cdot \text{K} \end{array}$$

# TUTORIAL

$$\left. \begin{array}{l} P_4 = 0.1 \text{ bar} \\ s_{6s} = s_5 \end{array} \right\} x_{6s} = \frac{s_{6s} - s_f}{s_{fg}} = \frac{7.7642 - 0.6492}{7.499} = 0.95$$

$$h_{6s} = h_f + x_{6s} h_{fg} = 192 + 0.95(2392) \\ = 2464.4 \text{ kJ/kg}$$

$$\eta_T = \frac{h_5 - h_6}{h_5 - h_{6s}} \rightarrow h_6 = h_5 - \eta_T (h_5 - h_{6s})$$

$$h_6 = 3479.1 - 0.8(3479.1 - 2464.4) \\ = 2667.34 \text{ kJ/kg} > h_g \text{ (superheated vapour)}$$

$$\Rightarrow T_6 = 88.1^\circ\text{C} \text{ (reading from tables @ 10 kPa)}$$

$$w_T = (h_4 - h_3) + (h_6 - h_5) \\ = (2902.1 - 3375.1) + (2667.34 - 3479.1) \\ = -1284.76 \text{ kJ/kg}$$

$$q_{in} = q_{23} + q_{45} = (h_3 - h_2) + (h_5 - h_4) \\ = (3375.1 - 202.43) + (3479.1 - 2902.1) \\ = 3749.7 \text{ kJ/kg}$$

$$|w_{net}| = |w_T + w_p| = |(-1284.76) + 10.62| \\ = 1274.14 \text{ kJ/kg}$$

$$\eta_{th} = \frac{|w_{net}|}{q_{in}} = \frac{1274.14}{3749.7} = 0.34 = 34\%$$

$$\dot{m} = \frac{|\dot{W}_{net}|}{|w_{net}|} = \frac{150,000}{1274.14} = 117.7 \text{ kg/s}$$



# **TUTORIAL**

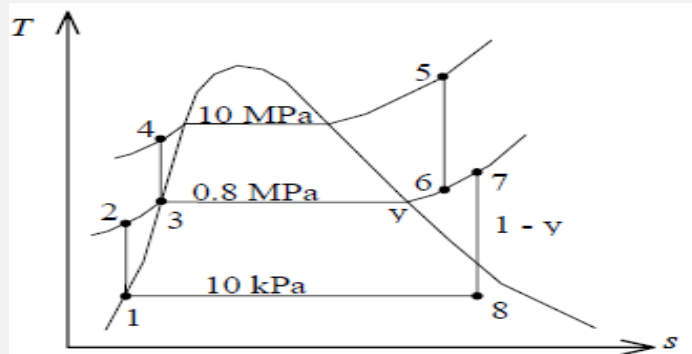
## **TUTORIAL PROBLEMS ON VAPOUR POWER CYCLE**

**3. A steam power plant operates on an ideal reheat-regenerative Rankine cycle and has a net power output of 120 MW. Steam enters the high-pressure turbine at 10 MPa and 550 °C and leaves at 0.8 MPa. Some steam is extracted at this pressure to heat the feed-water in an open feed-water heater. The rest of the steam is reheated to 500 °C and is expanded in the low-pressure turbine to the condenser pressure of 10 kPa. Show the cycle on a T-s diagram with respect to saturation lines, and determine**

**(a) the mass flow rate of steam through the boiler and**

**(b) the thermal efficiency of the cycle.**

# TUTORIAL



$$h_1 = h_f @ 10 \text{ kPa} = 191.81 \text{ kJ/kg}$$

$$v_1 = v_f @ 10 \text{ kPa} = 0.00101 \text{ m}^3/\text{kg}$$

$$w_{pI, \text{in}} = v_1 (P_2 - P_1) = (0.00101 \text{ m}^3/\text{kg}) (800 - 10 \text{ kPa}) = 0.80 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{pI, \text{in}} = 191.81 + 0.80 = 192.61 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_3 = 0.8 \text{ MPa} \\ \text{sat. liquid} \end{array} \right\} \begin{array}{l} h_3 = h_f @ 0.8 \text{ MPa} = 720.87 \text{ kJ/kg} \\ v_3 = v_f @ 0.8 \text{ MPa} = 0.001115 \text{ m}^3/\text{kg} \end{array}$$

$$w_{pII, \text{in}} = v_3 (P_4 - P_3) = (0.001115 \text{ m}^3/\text{kg}) (10,000 - 800 \text{ kPa}) = 10.26 \text{ kJ/kg}$$

$$h_4 = h_3 + w_{pII, \text{in}} = 720.87 + 10.26 = 731.12 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_5 = 10 \text{ MPa} \\ T_5 = 550^\circ\text{C} \end{array} \right\} \begin{array}{l} h_5 = 3502.0 \text{ kJ/kg} \\ s_5 = 6.7585 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_6 = 0.8 \text{ MPa} \\ s_6 = s_5 \end{array} \right\} h_6 = 2812.1 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_7 = 0.8 \text{ MPa} \\ T_7 = 500^\circ\text{C} \end{array} \right\} \begin{array}{l} h_7 = 3481.3 \text{ kJ/kg} \\ s_7 = 7.8692 \text{ kJ/kg} \cdot \text{K} \end{array}$$

# TUTORIAL

$$\left. \begin{array}{l} P_8 = 0.1 \text{ bar} \\ s_8 = s_7 \end{array} \right\} x_8 = \frac{s_8 - s_f}{s_{fg}} = \frac{7.8692 - 0.6492}{7.4996} = 0.9627$$

$$h_8 = h_f + x_8 h_{fg} = 192 + 0.9627(2392.1) = 2494.7 \text{ kJ/kg}$$

The fraction of steam extracted is determined from the steady-flow energy balance equation applied to the feed water heaters.

$$\dot{E}_{in} - \dot{E}_{out} = \Delta \dot{E}_{system} \stackrel{\text{0 (steady)}}{=} 0 \rightarrow \dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_6 h_6 + \dot{m}_2 h_2 = \dot{m}_3 h_3$$

$$\rightarrow y h_6 + (1 - y) h_2 = 1(h_3)$$

Where  $y$  is the fraction of steam extracted from the turbine ( $\dot{m}_6/\dot{m}_3$ ). Solving for  $y$ ,

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{720.87 - 192.61}{2812.1 - 192.61} = 0.2017$$

$$q_{in} = (h_5 - h_4) + (1 - y)(h_7 - h_6) = (3502.0 - 731.12) + (1 - 0.2017)(3481.3 - 2812.1) = 3305.1 \text{ kJ/kg}$$

$$q_{out} = (1 - y)(h_8 - h_1) = (1 - 0.2017)(2494.7 - 191.81) = 1838.5 \text{ kJ/kg}$$

$$w_{net} = q_{in} - q_{out} = 3305.1 - 1838.5 = 1466.6 \text{ kJ/kg}$$

$$\eta_{th} = \frac{|w_{net}|}{q_{in}} = \frac{1466.7}{3305.1} = 0.444 = 44.4\%$$

$$\dot{m} = \frac{|\dot{W}_{net}|}{|w_{net}|} = \frac{120,000}{1466.7} = 81.82 \text{ kg/s}$$

# **TUTORIAL**

## **TUTORIAL PROBLEMS ON VAPOUR POWER CYCLE**

4. How do the following quantities change when a simple ideal Rankine cycle is modified with reheating? Assume the mass flow rate is maintained the same.

Pump work input:	(a) increases	(b) decreases	(c) remains the same
Turbine work output:	(a) increases	(b) decreases	(c) remains the same
Heat supplied:	(a) increases	(b) decreases	(c) remains the same
Heat rejected:	(a) increases	(b) decreases	(c) remains the same
Moisture content at turbine exit:	(a) increases	(b) decreases	(c) remains the same

# TUTORIAL

## TUTORIAL PROBLEMS ON VAPOUR POWER CYCLE

---

Pump work input:	(a) increases	(b) decreases	<input checked="" type="radio"/> (c) remains the same
Turbine work output:	<input checked="" type="radio"/> (a) increases	(b) decreases	(c) remains the same
Heat supplied:	<input checked="" type="radio"/> (a) increases	(b) decreases	(c) remains the same
Heat rejected:	<input checked="" type="radio"/> (a) increases	(b) decreases	(c) remains the same
Moisture content at turbine exit:	(a) increases	<input checked="" type="radio"/> (b) decreases	(c) remains the same

---

# HIGHLIGHT

## HIGHLIGHTS ON GAS POWER CYCLE

- Under Gas Power Cycles, the performance of simple gas turbine power plants is described in terms of the air-standard Brayton cycle.
- For this cycle, we evaluate the principal work and heat transfers along with two important performance parameters: the back work ratio and the thermal efficiency.

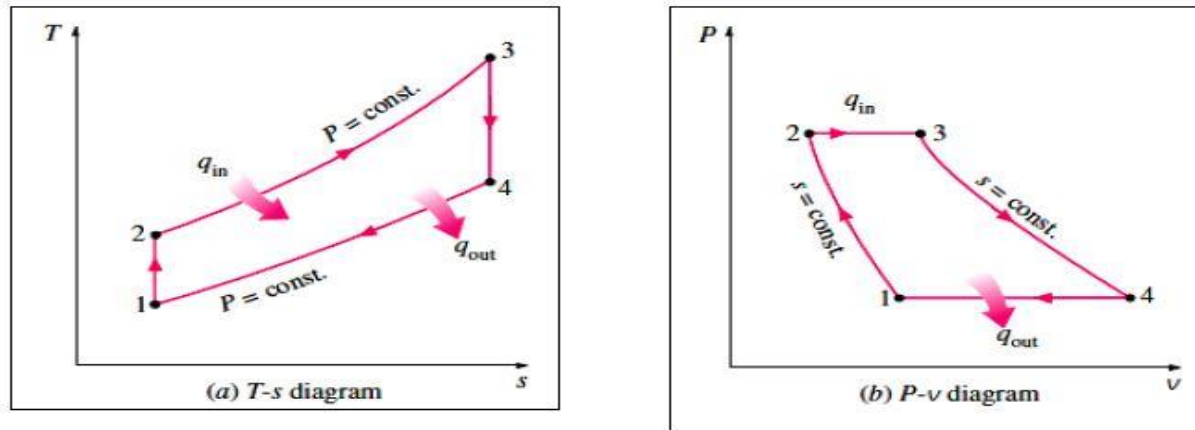
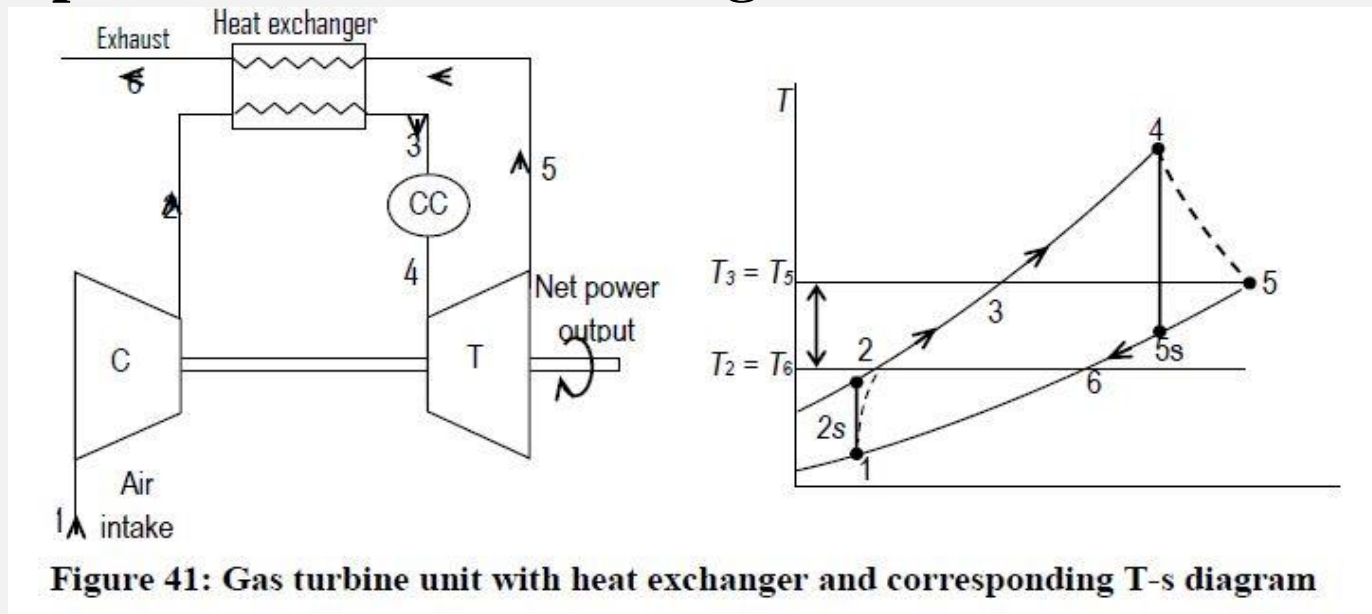


Figure 38: Ideal Brayton cycle with (a) T-s diagram and (b) P-v diagram

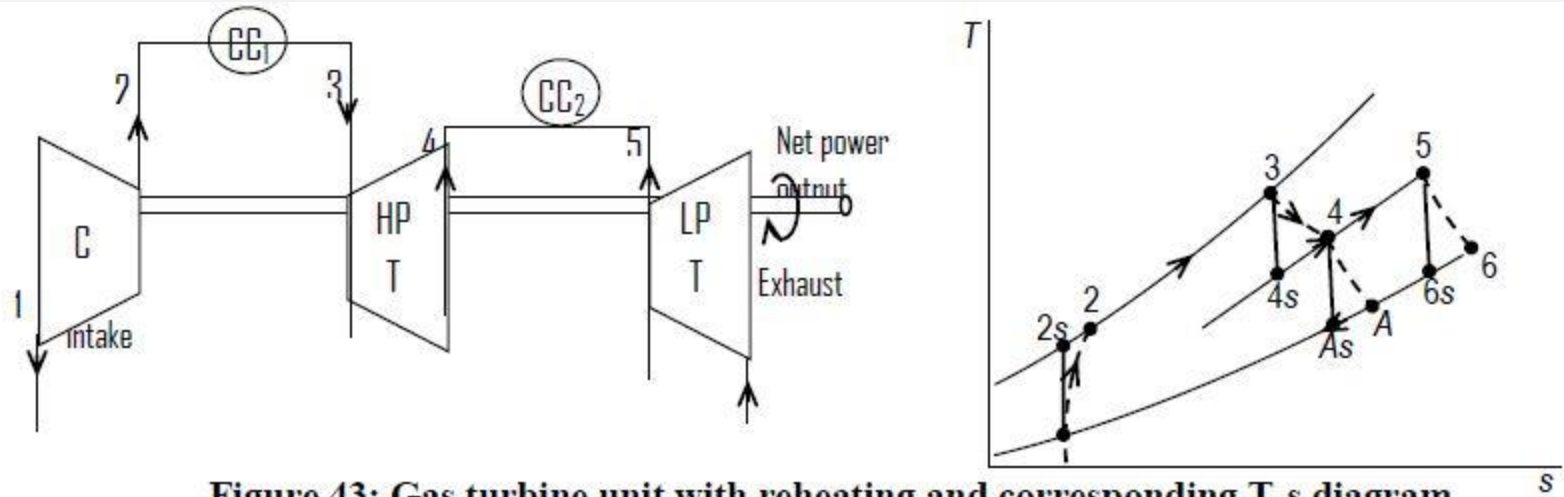
# HIGHLIGHT

## HIGHLIGHTS ON GAS POWER CYCLE

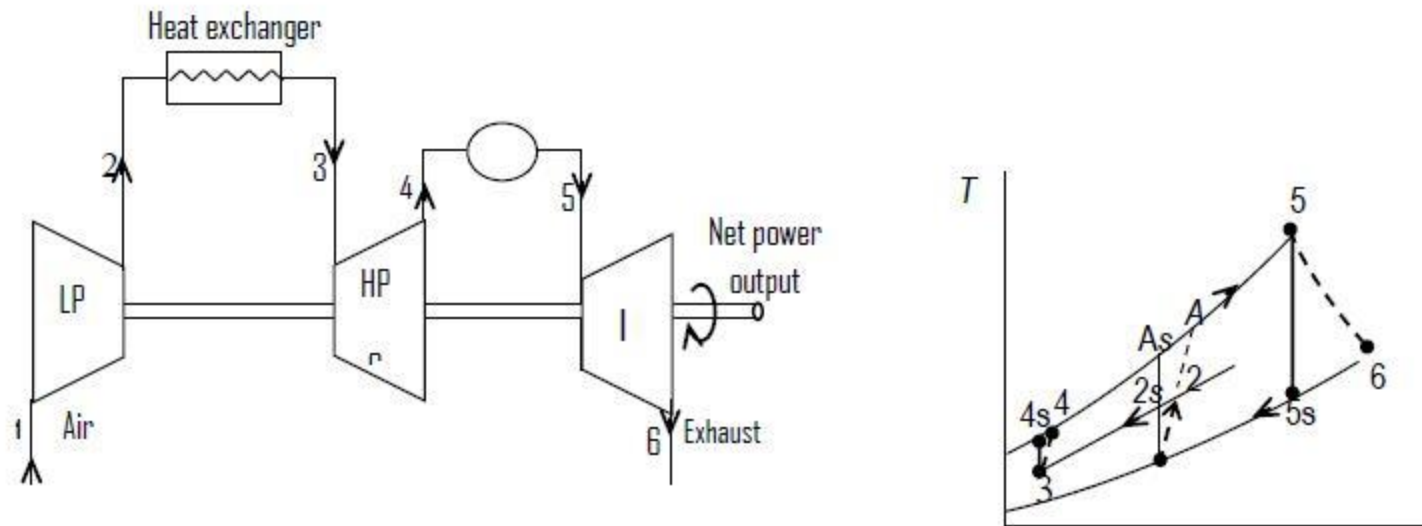
- We also consider the effects on performance of irreversibility's and of varying compressor pressure ratio.
- Three modifications of the simple cycle to improve performance are introduced: regeneration, reheat, and compression with intercooling.



# HIGHLIGHT



**Figure 43: Gas turbine unit with reheating and corresponding T-s diagram**



**Figure 42: Gas turbine unit with inter-cooling and corresponding T-s diagram**



# TUTORIAL

## TUTORIAL PROBLEMS ON GAS POWER CYCLE

1. A gas turbine for an automobile is designed with a regenerator. Air enters the compressor of this engine at 100 kPa and 20 °C. The compressor pressure ratio is 8; the maximum cycle temperature is 800°C; and the cold air stream leaves the regenerator 10 °C cooler than the hot air stream at the inlet of the regenerator. Assuming both the compressor and turbine to be isentropic, determine the rates of heat addition and rejection for this cycle when it produces 150 kW. Use constant specific heats at room temperature.

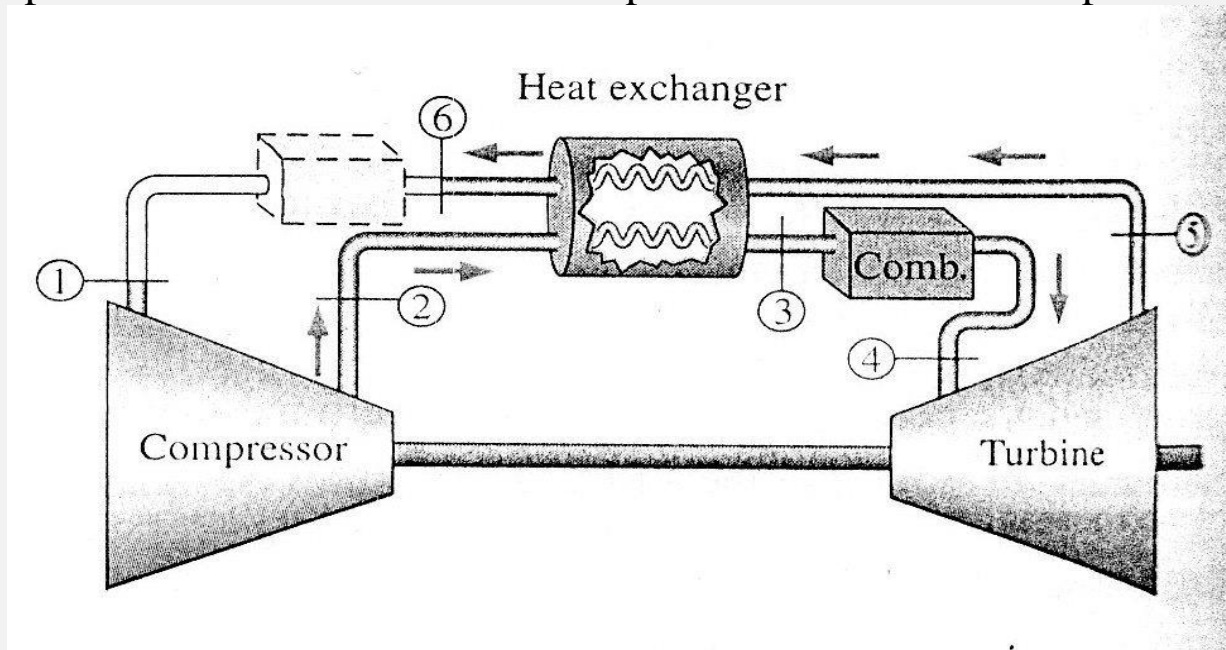
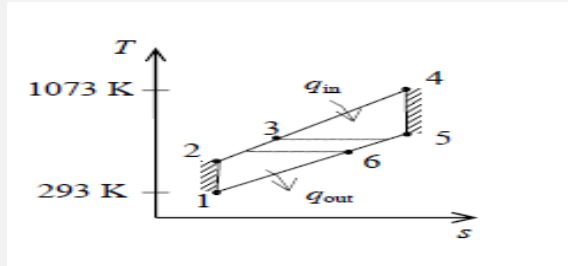


Figure Q1

# TUTORIAL

The T-s diagram for the problem 1 of Gas turbine tutorial problems is indicated below



The properties of air at room temperature are  $c_p = 1.005 \text{ kJ/kg K}$  and  $\gamma = 1.4$

According to the isentropic process expressions for an ideal gas,

$$T_2 = T_1(r_p)^{(\gamma-1)/\gamma} = (293 \text{ K})(8)^{0.4/1.4} = 530.8 \text{ K}$$

$$T_5 = T_4(1/r_p)^{0.4/1.4} = (1073 \text{ K})(1/8)^{0.4/1.4} = 592.3 \text{ K}$$

When first Law is applied to the heat exchanger, the result is

$$T_3 - T_2 = T_5 - T_6 \text{ -----Eqn. (1)}$$

The regenerator temperature specification gives,

$$T_3 = T_5 - 10 = (592.3 - 273 - 10) + 273 = 582.3 \text{ K}$$

$$\text{From Eqn. (1), } T_6 = T_5 - (T_3 - T_2) = 592.3 - (582.3 - 530.8) = 540.8 \text{ K}$$

Application of the first law to the turbine and compressor gives the magnitude of the net work done per unit kg of air as

$$W_{\text{net}} c_p(T_4 - T_5) - c_p(T_2 - T_1) = (1.005 \text{ kJ/kg K})[(1073 - 592.3) - (530.8 - 293)] = 244.1 \text{ kJ/kg}$$

$$\text{Then mass flow rate } \dot{m} = \dot{W} / w_{\text{net}} = [150 \text{ kJ/s}] / 244.1 \text{ kJ/kg} = 0.6145 \text{ kg/s}$$

The rate of heat addition is given by,

$$\dot{Q}_{\text{in}} = \dot{m} c_p (T_4 - T_3) = (0.6145 \text{ kg/s})(1.005 \text{ kJ/kg K})(1073 - 582.3) \text{ K} = \mathbf{303.0 \text{ kW}}$$

The rate of heat rejection is given as

$$\dot{Q}_{\text{out}} = \dot{m} c_p (T_6 - T_1) = (0.6145 \text{ kg/s})(1.005 \text{ kJ/kg K})(540.8 - 293) \text{ K} = \mathbf{153.0 \text{ kW}}$$

# TUTORIAL

## TUTORIAL PROBLEMS ON GAS POWER CYCLE

- 2 Air enters a gas turbine with two stages of compression and two stages of expansion at 100 kPa and 17°C. This system uses a regenerator as well as reheating and intercooling. The pressure ratio across each compressor is 4; 300 kJ/kg of heat is added to the air in each combustion chamber; and the regenerator operates perfectly while increasing the temperature of the cold air by 20°C. Determine this system's thermal efficiency. Assume isentropic operations for the entire compressor and the turbine stages and use constant specific heats at room temperature.

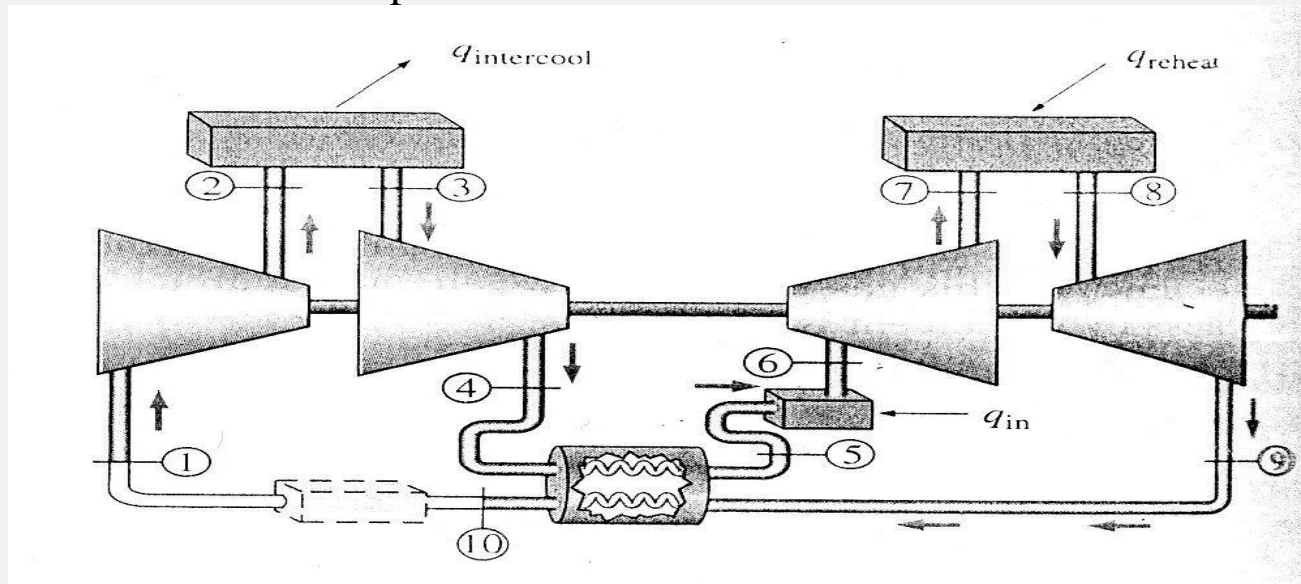
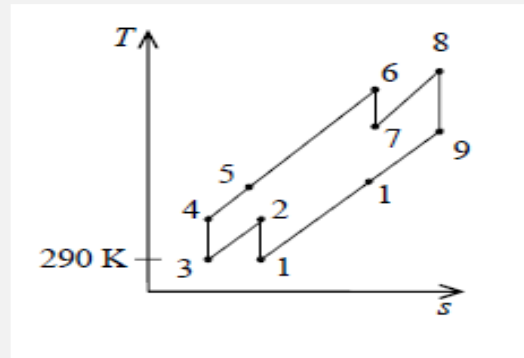


Figure Q2

# TUTORIAL

The T-s diagram for problem 2 of the Gas Turbine tutorial problems is provided below



The temperatures at the various states are obtained as follows:

$$T_3 = T_1 = 290 \text{ K}$$

$$T_2 = T_4 = T_1(r_p)^{0.4/1.4} = (290 \text{ K})(4)^{0.4/1.4} = 430.9 \text{ K}$$

$$T_5 = T_4 - 273 + 20 + 273 = 450.9 \text{ K}$$

$$q_{in} = c_p(T_6 - T_5)$$

$$T_6 = T_5 + (q_{in}/c_p) = 450.9 + (300 \text{ kJ/kg} / 1.005 \text{ kJ/kg K}) = 802.8 \text{ K}$$

$$T_9 = T_8(1/r_p)^{0.4/1.4} = 802.8 \text{ K}(1/4)^{0.4/1.4} = 540.2 \text{ K}$$

$$T_{10} = T_9 - 20 = 520.2 \text{ K}$$

$$\text{The heat input } q_{in} = 300 + 300 = 600 \text{ kJ/kg}$$

The heat rejected is

$$q_{out} = c_p(T_{10} - T_1) + c_p(T_2 - T_3) = (1.005 \text{ kJ/kg K})[(520.2 - 290) + (430.9 - 290) \text{ K}] = 373.0 \text{ kJ/kg}$$

The thermal efficiency of the cycle is then

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{373}{600} = 0.378 \quad \textbf{(37.8 \%)}$$

# **TUTORIAL**

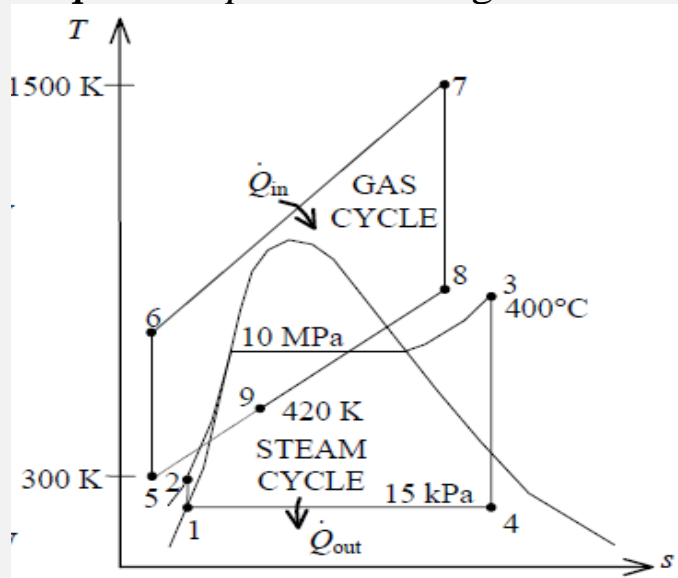
## **TUTORIAL PROBLEMS ON GAS POWER CYCLE**

3. The gas turbine portion of a combined gas-steam power plant has a pressure ratio of 16. Air enters the compressor at 300 K at a rate of 14 kg/s and is heated to 1500 K in the combustion chamber. The combustion gases leaving the turbine are used to heat the steam to 400 °C at 10 MPa in a heat exchanger. The combustion gases leave the heat exchanger at 420 K. The steam leaving the turbine is condensed at 15 kPa. Assuming all the compression and expansion processes to be isentropic, determine:
- (a) The mass flow rate of the steam
  - (b) The net power output
  - (c) The thermal efficiency of the combined cycle.
- For air, assume constant specific heats at room temperature.

# TUTORIAL

**Assumptions** 1 steady operating condition exist. 2 Kinetic and potential energy changes are negligible. 3 Air is an ideal gas with constant specific heats.

**Properties**  $c_p = 1.005 \text{ kJ/kg} \cdot \text{K}$  and  $k = 1.4$



**Analysis** (a) The analysis of gas cycle yields

$$T_6 = T_5 \left( \frac{P_6}{P_5} \right)^{(k-1)/k} = (300 \text{ K})(16)^{0.4/1.4} = 662.5 \text{ K}$$

$$\dot{Q}_{in} = \dot{m}_{air} (h_7 - h_6) = \dot{m}_{air} c_p (T_7 - T_6) = (14 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot \text{K})(1500 - 662.5) \text{ K} = 11,784 \text{ kW}$$

$$\dot{W}_{C, gas} = \dot{m}_{air} (h_6 - h_5) = \dot{m}_{air} c_p (T_6 - T_5) = (14 \text{ kg/s})(1.005 \text{ kJ/kg} \cdot \text{K})(662.5 - 300) \text{ K} = 5100 \text{ kW}$$

# TUTORIAL

$$T_8 = T_7 \left( \frac{P_8}{P_7} \right)^{(k-1)/k} = (1500\text{K}) \left( \frac{1}{16} \right)^{0.4/1.4} = 679.3\text{ K}$$

$$\dot{W}_{T,\text{gas}} = \dot{m}_{\text{air}} (h_7 - h_8) = \dot{m}_{\text{air}} c_p (T_7 - T_8) = (14\text{ kg/s})(1.005\text{ kJ/kg} \cdot \text{K})(1500 - 679.3)\text{ K} \\ = 11,547\text{ kW}$$

$$\dot{W}_{\text{net,gas}} = \dot{W}_{T,\text{gas}} - \dot{W}_{C,\text{gas}} = 11,547 - 5,100 = 6447\text{ kW}$$

From steam tables,

$$h_1 = h_f @ 15\text{ kPa} = 225.94\text{ kJ/kg}$$

$$v_1 = v_f @ 15\text{ kPa} = 0.001014\text{ m}^3/\text{kg}$$

$$w_{pL,\text{in}} = v_1 (P_2 - P_1) = (0.001014\text{ m}^3/\text{kg})(10,000 - 15\text{ kPa}) = 10.12\text{ kJ/kg}$$

$$h_2 = h_1 + w_{pL,\text{in}} = 225.94 + 10.13 = 236.06\text{ kJ/kg}$$

$$\left. \begin{array}{l} P_3 = 10\text{ MPa} \\ T_3 = 400^\circ\text{C} \end{array} \right\} \begin{array}{l} h_3 = 3097.0\text{ kJ/kg} \\ s_3 = 6.2141\text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_4 = 0.15\text{ bar} \\ s_4 = s_3 \end{array} \right\} x_4 = \frac{s_4 - s_f}{s_{fg}} = \frac{6.2141 - 0.7549}{7.2522} = 0.753$$

$$h_4 = h_f + x_4 h_{fg} = 226 + 0.753(2372.3) \\ = 2012.34\text{ kJ/kg}$$

# TUTORIAL

$$\dot{E}_{\text{in}} - \dot{E}_{\text{out}} = \Delta \dot{E}_{\text{system}} \stackrel{\neq 0 \text{ (steady)}}{=} 0 \longrightarrow \dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\begin{aligned} \rightarrow \dot{m}_s (h_3 - h_2) &= \dot{m}_{\text{air}} (h_8 - h_9) \quad \dot{m}_s = \frac{h_8 - h_9}{h_3 - h_2} \dot{m}_{\text{air}} = \frac{c_p (T_8 - T_9)}{h_3 - h_2} \dot{m}_{\text{air}} \\ &= \frac{(1.005 \text{ kJ/kg} \cdot \text{K})(679.3 - 420) \text{ K}}{(3097.0 - 236.06) \text{ kJ/kg}} (14 \text{ kg/s}) = \mathbf{1.275 \text{ kg/s}} \end{aligned}$$

$$\dot{W}_{\text{T,steam}} = \dot{m}_s (h_3 - h_4) = (1.275 \text{ kg/s})(3097.0 - 2011.5) \text{ kJ/kg} = 1384 \text{ kW}$$

$$\dot{W}_{\text{p,steam}} = \dot{m}_s w_p = (1.275 \text{ kg/s})(10.12 \text{ kJ/kg}) = 12.9 \text{ kW}$$

$$\dot{W}_{\text{net,steam}} = \dot{W}_{\text{T,steam}} - \dot{W}_{\text{p,steam}} = 1384 - 12.9 = 1371 \text{ kW}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{net,steam}} + \dot{W}_{\text{net,gas}} = 1371 + 6448 = \mathbf{7819 \text{ kW}}$$

$$\eta_{\text{th}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} = \frac{7819 \text{ kW}}{11,784 \text{ kW}} = \mathbf{66.4\%}$$



# TUTORIAL

## TUTORIAL PROBLEMS ON GAS POWER CYCLE

In question 4 BELOW use the preamble to complete each of the statements that follow with either the *italicised* phrase: *decreases, increases or remains constant* to make the statement complete and true.

4. A simple ideal Brayton cycle with regeneration is modified to incorporate multistage compression with intercooling and multistage expansion with reheating without changing the pressure or temperature limits of the cycle. As a result of these two modifications:
- a) The network output -----
  - b) The back work ratio-----
  - c) The thermal efficiency-----
  - d) The heat rejected-----

### SOLUTION

- a. increases
- b. decreases
- c. decreases
- d. increases

**TUTORIAL ENDS**

**THANK YOU**

**LUKE 19:13**

# **THERMODYNAMICS II**

## **(ME 365)**

# **TUTORIAL LECTURE**

## **UNIT 3**

Lead Facilitator  
**Prof. F. K. Forson**

# OUTLINE

- **HIGHLIGHTS ON REFRIGERATION CYCLES**
- **TUTORIAL PROBLEMS ON**
  - **MECHANICAL VAPOUR COMPRESSION SYSTEMS**
  - **VAPOUR ABSORPTION REFRIGERATION SYSTEM**
  - **GAS REFRIGERATION SYSTEM**

# **HIGHLIGHT**

## **HIGHLIGHTS ON REFRIGERATION CYCLE**

- 1. Refrigeration is the science of producing and maintaining temperatures below that of the surrounding atmosphere.**
- 2. Refrigeration is generally produced in one of the following three ways:**
  - (i) By melting a solid;**
  - (ii) By sublimation of a solid;**
  - (iii) By evaporation of a liquid.**
- 3. Co-efficient of performance (C.O.P.) is defined as the ratio of heat absorbed by the refrigerant while passing through the evaporator to the work input required to compress the refrigerant in the compressor ; in short it is the ratio between heat extracted and work done (in heat units).**

# HIGHLIGHT

## HIGHLIGHTS ON REFRIGERATION CYCLE

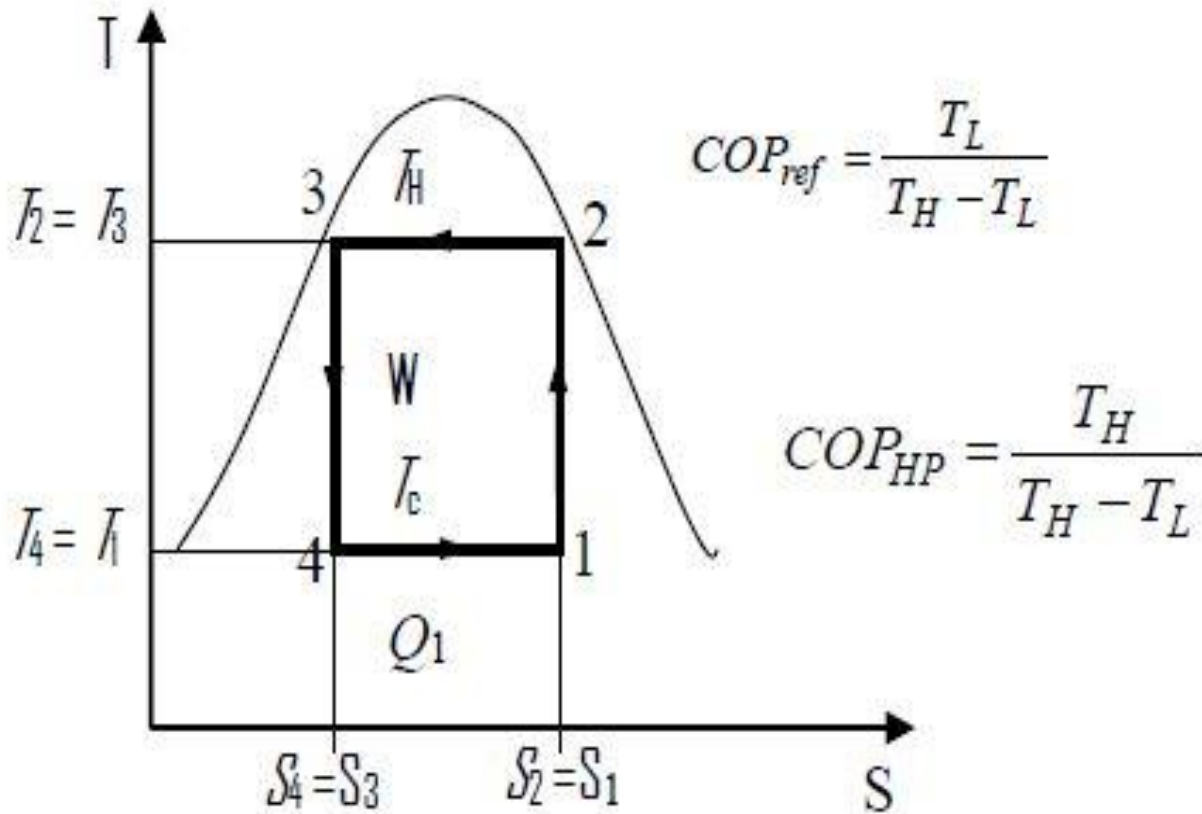


Figure 45: Layout of a reversed Carnot heat engine with T-s diagram

# **HIGHLIGHT**

## **HIGHLIGHTS ON REFRIGERATION CYCLE**

**4. Relative C.O.P. = Actual C.O.P./Theoretical C.O.P**

**5. 1 tonne of refrigeration = 14000 kJ/h.**

**6. The main characteristic feature of air refrigeration system is that throughout the cycle the refrigerant remains in gaseous state.**

**The air refrigeration system may be of two types:**

- (i) Closed system and**
- (ii) Open system.**

# HIGHLIGHT

## HIGHLIGHTS ON REFRIGERATION CYCLE

7. Co-efficient of performance of a 'refrigerator' working on a reversal Carnot cycle

$$C.O.P = \left( \frac{T_L}{T_H - T_L} \right)$$

For a Carnot cycle 'heat pump'  $C.O.P = \left( \frac{T_H}{T_H - T_L} \right)$

8. For air refrigeration system working on reversed Brayton cycle.

$$C.O.P. = \frac{(T_3 - T_2)}{\left( \frac{n}{n-1} \right) \left( \frac{\gamma-1}{\gamma} \right) (T_4 - T_3 + T_2 - 1)} .$$



# **HIGHLIGHT**

## **HIGHLIGHTS ON REFRIGERATION CYCLE**

**9. The following air refrigeration systems are used in aero planes:**

- (i) Simple cooling system**
- (ii) Boot strap system**
- (iii) Regenerative cooling system.**

**10. In a simple vapour compression cycle the following processes are completed:**

- (i) Compression**
- (ii) Condensation**
- (iii) Expansion**
- (iv) Vaporisation**

# HIGHLIGHT

11. The various parts of a simple domestic vapour compression cycle are: Compressor, Discharge line (or hot gas line), Condenser, Receiver tank, Liquid line, Expansion valve (capillary tube), Evaporator and Suction line as illustrated in Figure R1

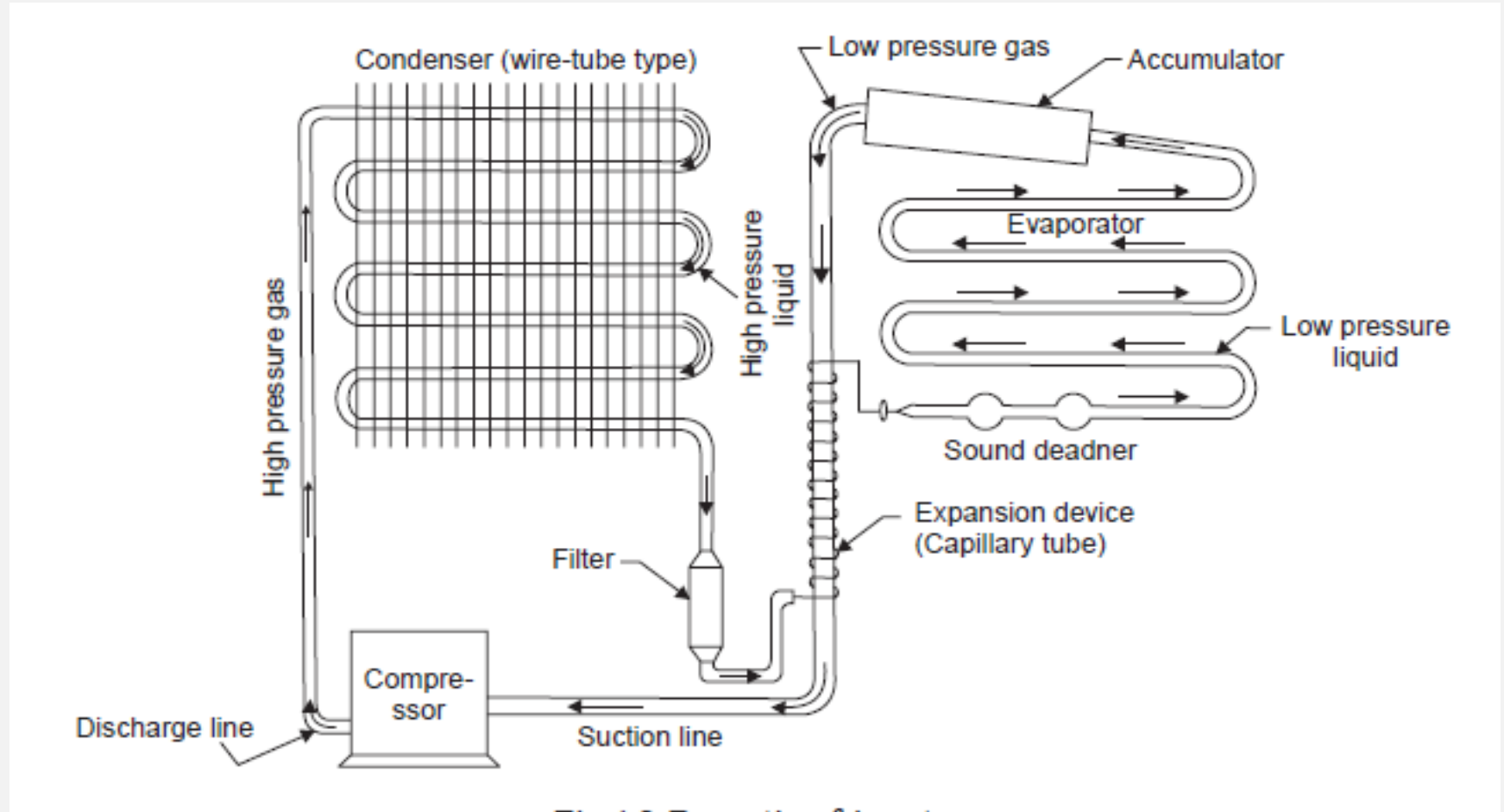


Figure R1

# **HIGHLIGHT**

## **HIGHLIGHTS ON REFRIGERATION CYCLE**

**12. If the vapour is not superheated after compression, the operation is called ‘Wet compression’ and if the vapour is superheated at the end of compression, it is known as ‘Dry compression’. Dry compression, in actual practice is always preferred as it gives higher volumetric efficiency and mechanical efficiency and there are less chances of compressor damage.**

**13. p-h chart gives directly the changes in enthalpy and pressure during a process for thermodynamic analysis.**

# HIGHLIGHT

## HIGHLIGHTS ON REFRIGERATION CYCLE

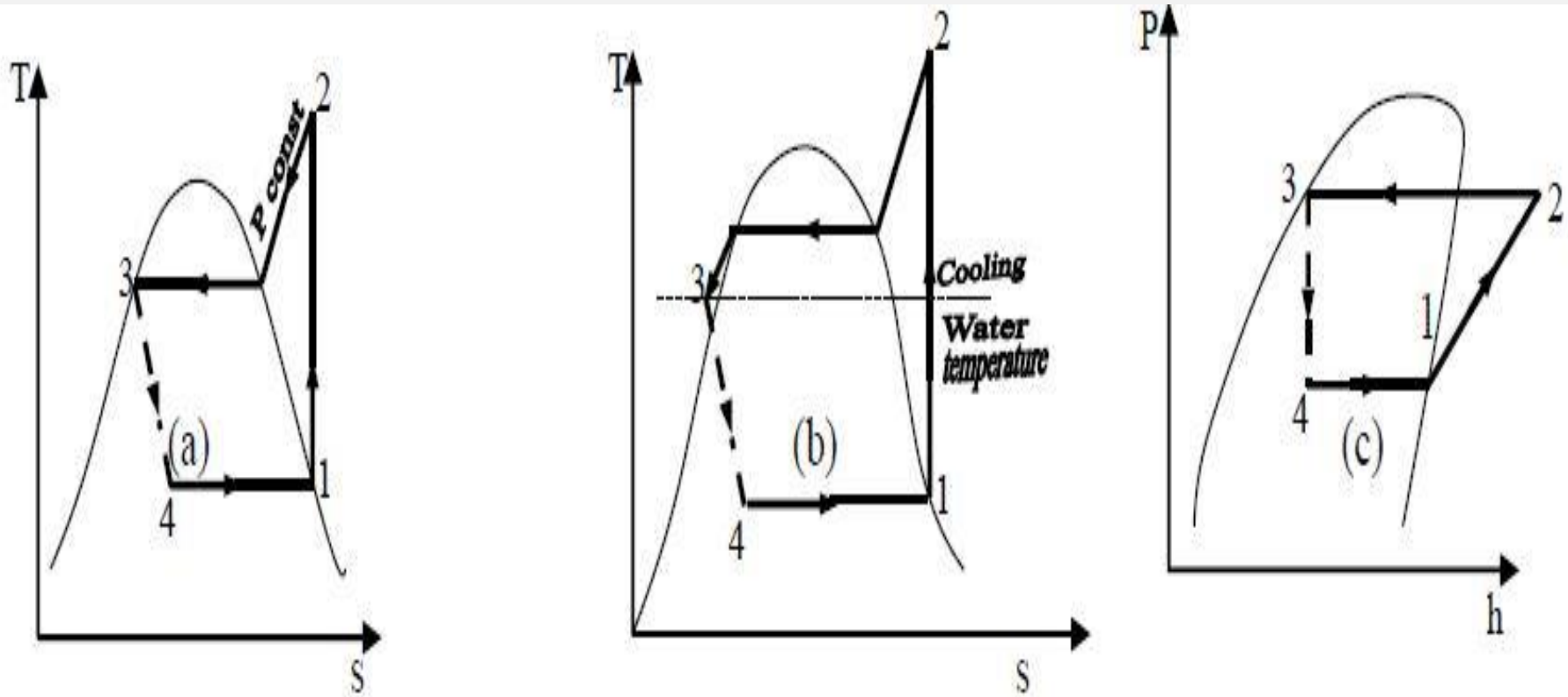


Figure 46: Modifications of a practical ideal refrigeration cycle with  $T-s$  and  $P-h$  diagrams showing superheating and sub cooling.

# **HIGHLIGHT**

## **HIGHLIGHTS ON REFRIGERATION CYCLE**

**14. When suction pressure is decreased, the refrigerating effect is decreased and work required is increased. The net effect is to reduce the refrigerating capacity of the system and the C.O.P.**

**15. The overall effect of superheating is to give a low value of C.O.P.**

**16. ‘Sub-cooling’ results in increase of C.O.P. provided that no further energy has to be spent to obtain the extra cold coolant required.**

# **HIGHLIGHT**

## **HIGHLIGHTS ON REFRIGERATION CYCLE**

**17. The refrigerating system should always be designed to operate at the highest possible vaporizing temperature and lowest possible condensing temperature, of course, keeping in view the requirements of the application.**

**18. ‘Volumetric efficiency’ is defined as the ratio of the actual volume of gas drawn into the compressor (at evaporator temperature and pressure) on each stroke to the piston displacement. If the effect of clearance alone is considered, the resulting expression may be termed ‘clearance volumetric efficiency’. The expression used for grouping into one constant all the factors affecting efficiency may be termed ‘total volumetric efficiency’.**

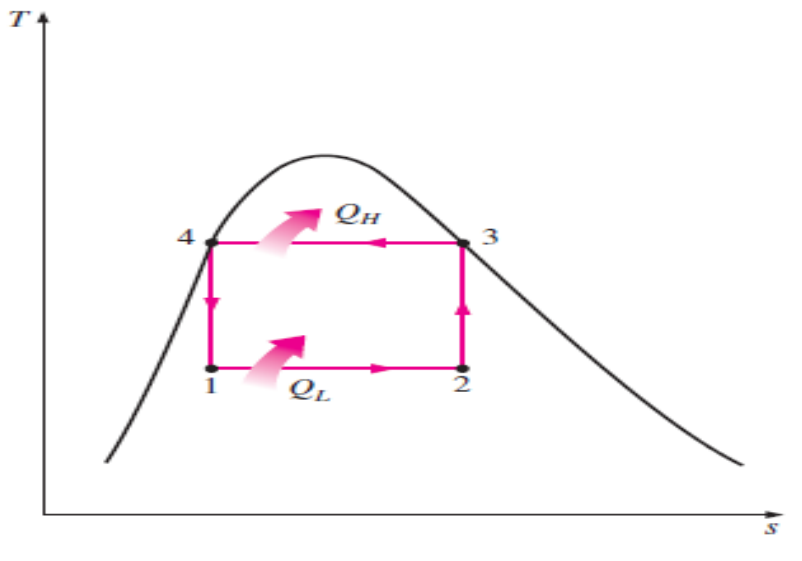
## **TUTORIAL**

### **PROBLEMS ON MECHANICAL VAPOUR COMPRESSION SYSTEMS**

**1. The temperature in a refrigerator evaporator coil is  $-6\text{ }^{\circ}\text{C}$  and that in the condenser coil is  $22\text{ }^{\circ}\text{C}$ . Assuming that the machine operates on the reversed Carnot cycle, calculate the  $\text{COP}_{\text{ref}}$ , the refrigerating effect per kilowatt of input work, and the heat rejected to the condenser.**

# TUTORIAL

## PROBLEMS ON MECHANICAL VAPOUR COMPRESSION SYSTEMS



The COP of the Carnot refrigerator is determined using;

$$\text{COP}_{\text{carnot,ref}} = \frac{T_C}{T_H - T_C} = \frac{\dot{Q}_C}{\dot{W}}$$

$$\Rightarrow \text{COP}_{\text{carnot,ref}} = \frac{-6 + 273}{22 - (-6)} = 9.54$$

The refrigerating effect per kilowatt of input power is;

$$\dot{Q}_C = \text{COP}_{\text{ref}} \times \dot{W} = 9.54 \times 1 \text{ kW} = 9.54 \text{ kW}$$

The heat rejected to the condenser is;

$$\dot{Q}_H = \dot{Q}_C + \dot{W} = 9.54 + 1 = 10.54 \text{ kW}$$



# **TUTORIAL**

## **PROBLEMS ON MECHANICAL VAPOUR COMPRESSION SYSTEMS**

**2. A refrigerator uses refrigerant-134a as the working fluid and operates on an ideal vapor-compression refrigeration cycle between 0.12 and 0.7 MPa. The mass flow rate of the refrigerant is 0.05 kg/s. Show the cycle on a T-s diagram with respect to saturation lines.**

**Determine**

- (a) the rate of heat removal from the refrigerated space and the power input to the compressor,**
- (b) the rate of heat rejection to the environment, and**
- (c) the coefficient of performance.**

# TUTORIAL

## PROBLEMS ON MECHANICAL VAPOUR COMPRESSION SYSTEMS

**Assumptions** 1 steady operating condition exists. 2 Kinetic and potential energy changes are negligible.

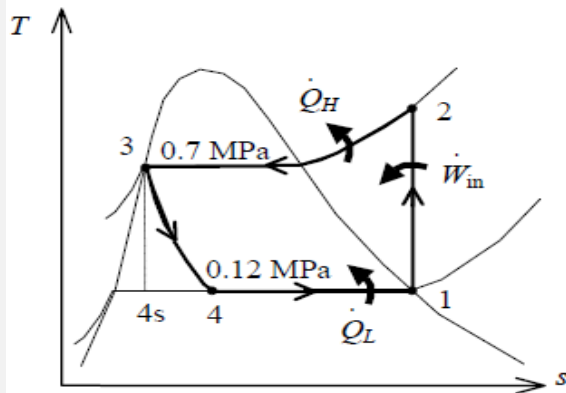
**Analysis** (a) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic; the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables

$$\left. \begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat. vapor} \end{array} \right\} \begin{array}{l} h_1 = h_g @ 120 \text{ kPa} = 236.97 \text{ kJ/kg} \\ s_1 = s_g @ 120 \text{ kPa} = 0.94779 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$\left. \begin{array}{l} P_2 = 0.7 \text{ MPa} \\ s_2 = s_1 \end{array} \right\} \begin{array}{l} h_2 = 273.50 \text{ kJ/kg} \quad (T_2 = 34.95^\circ\text{C}) \end{array}$$

$$\left. \begin{array}{l} P_3 = 0.7 \text{ MPa} \\ \text{sat. liquid} \end{array} \right\} h_3 = h_f @ 0.7 \text{ MPa} = 88.82 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 88.82 \text{ kJ/kg} \text{ (throttling)}$$



# TUTORIAL

## PROBLEMS ON MECHANICAL VAPOUR COMPRESSION SYSTEMS

Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})(236.97 - 88.82) \text{ kJ/kg} = 7.41 \text{ kW}$$

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})(273.50 - 236.97) \text{ kJ/kg} = 1.83 \text{ kW}$$

(b) The rate of heat rejection to the environment is determined from

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 7.41 + 1.83 = 9.23 \text{ kW}$$

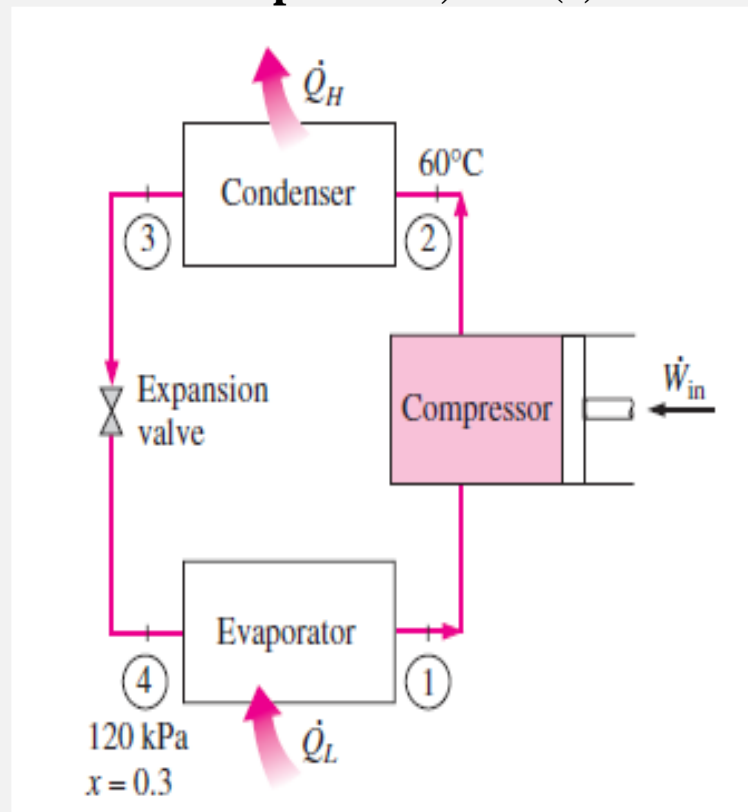
(c) The COP of the refrigerator is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.41 \text{ kW}}{1.83 \text{ kW}} = 4.06$$

# TUTORIAL

## PROBLEMS ON MECHANICAL VAPOUR COMPRESSION SYSTEMS

3. A refrigerator as shown in Figure Q1, uses refrigerant-134a as the working fluid and operates on the ideal vapor-compression refrigeration cycle. The refrigerant enters the evaporator at 120 kPa with a quality of 30 percent and leaves the compressor at 60°C. If the compressor consumes 450 W of power, determine (a) the mass flow rate of the refrigerant, (b) the condenser pressure, and (c) the COP of the refrigerator



# TUTORIAL

## PROBLEMS ON MECHANICAL VAPOUR COMPRESSION SYSTEMS

**Assumptions** 1 steady operating condition exists. 2 Kinetic and potential energy changes are negligible.

**Analysis** From the refrigerant-134a tables

$$\left. \begin{array}{l} P_4 = 120 \text{ kPa} \\ x_4 = 0.30 \end{array} \right\} h_4 = 86.83 \text{ kJ/kg}$$

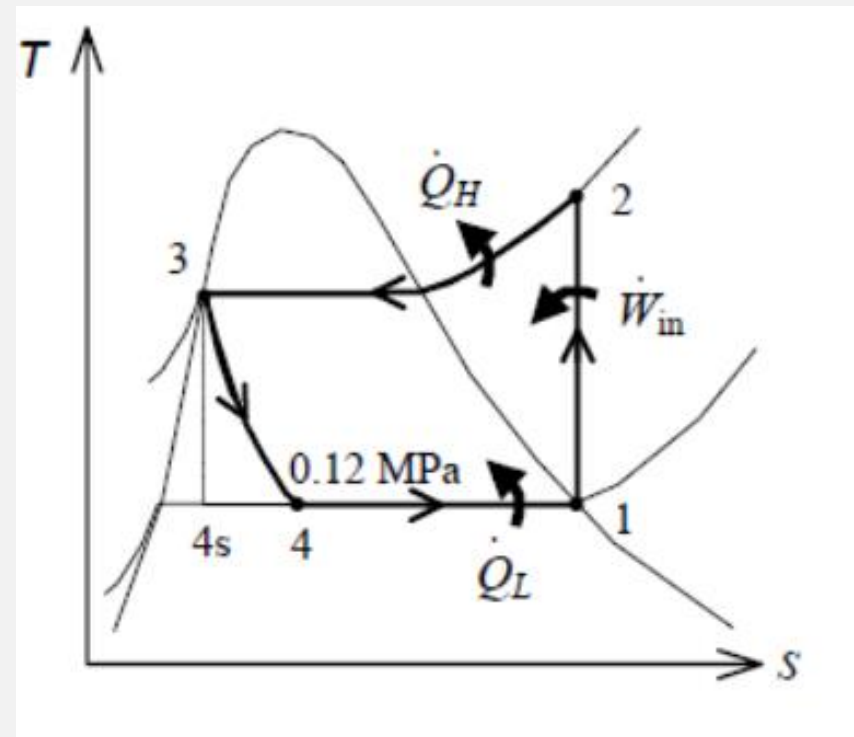
$$h_3 = h_4$$

$$\left. \begin{array}{l} h_3 = 86.83 \text{ kJ/kg} \\ x_3 = 0 \text{ (sat. liq.)} \end{array} \right\} P_3 = 671.8 \text{ kPa}$$

$$P_2 = P_3$$

$$\left. \begin{array}{l} P_2 = 671.8 \text{ kPa} \\ T_2 = 60^\circ\text{C} \end{array} \right\} h_2 = 298.87 \text{ kJ/kg}$$

$$\left. \begin{array}{l} P_1 = P_4 = 120 \text{ kPa} \\ x_1 = 1 \text{ (sat. vap.)} \end{array} \right\} h_1 = 236.97 \text{ kJ/kg}$$



# TUTORIAL

## PROBLEMS ON MECHANICAL VAPOUR COMPRESSION SYSTEMS

The mass flow rate of the refrigerant is determined from

$$\dot{m} = \frac{\dot{W}_{in}}{h_2 - h_1}$$
$$= \frac{0.45 \text{ kW}}{(298.87 - 236.97) \text{ kJ/kg}} = 0.00727 \text{ kg/s}$$

(c) The refrigeration load and the COP are

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.0727 \text{ kg/s})(236.97 - 86.83) \text{ kJ/kg} = 1.091 \text{ kW}$$
$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{1.091 \text{ kW}}{0.45 \text{ kW}} = 2.43$$

## **TUTORIAL**

### **PROBLEMS ON VAPOUR ABSORPTION REFRIGERATION SYSTEM**

**4. Heat is supplied to an absorption refrigeration system from a geothermal well at  $130^{\circ}\text{C}$  at a rate of  $5 \times 10^5 \text{ kJ/h}$ . The environment is at  $25^{\circ}\text{C}$ , and the refrigerated space is maintained at  $-30^{\circ}\text{C}$ .**

**Determine the maximum rate at which this system can remove heat from the refrigerated space.**

# TUTORIAL

## PROBLEMS ON VAPOUR ABSORPTION REFRIGERATION SYSTEM

The conditions at which an absorption refrigeration system operates are specified. The maximum rate at which this system can remove heat from the refrigerated space is to be determined.

*Analysis* The maximum COP that this refrigeration system can have is

$$\text{COP}_{R,\max} = \left(1 - \frac{T_0}{T_s}\right) \left(\frac{T_L}{T_0 - T_L}\right) = \left(1 - \frac{298 \text{ K}}{403 \text{ K}}\right) \left(\frac{243}{298 - 243}\right) = 1.15$$

Thus

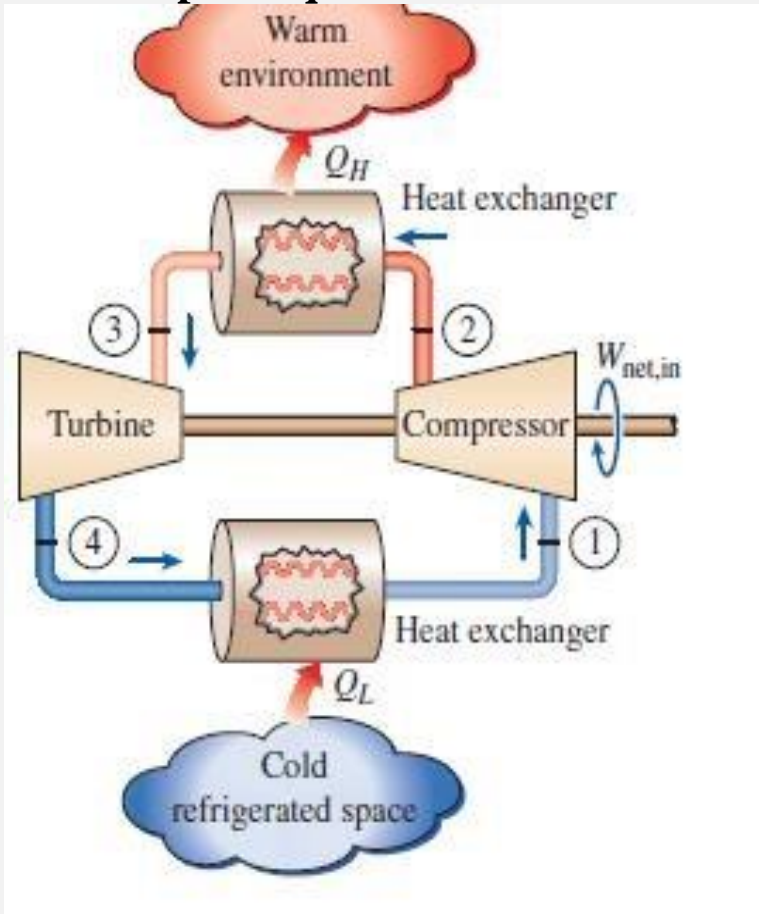
$$\begin{aligned}\dot{Q}_{L,\max} &= \text{COP}_{R,\max} \dot{Q}_{\text{gen}} \\ &= (1.15)(5 \times 10^5 \text{ kJ/h}) = 5.75 \times 10^5 \text{ kJ/h}\end{aligned}$$



# TUTORIAL

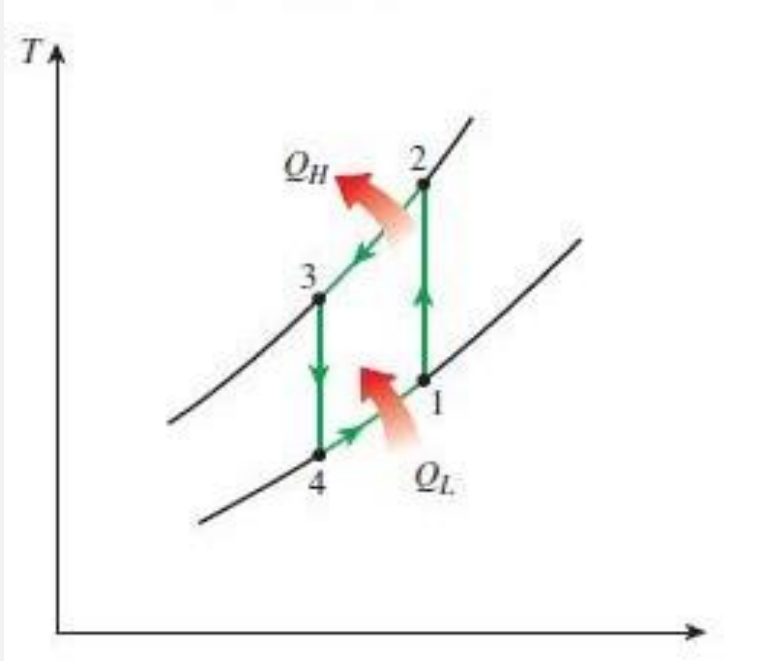
## TUTORIAL PROBLEMS ON GAS REFRIGERATION SYSTEM

6. Consider an ideal gas refrigeration cycle using air as the working fluid. Air enters the compressor at 80 kPa and 25 °C and is compressed to 280 kPa. Air is then cooled to 35 °C before entering the turbine. For a mass flow rate of 0.2 kg/s, the net power input required is:



# TUTORIAL

## TUTORIAL PROBLEMS ON GAS REFRIGERATION SYSTEM



$$T_2 = T_1 \left( \frac{P_2}{P_1} \right)^{(\gamma-1)/\gamma} = 298 \times \left( \frac{280}{80} \right)^{(1.4-1)/1.4} = 426.2K ,$$

$$T_4 = T_3 \left( \frac{P_1}{P_2} \right)^{(\gamma-1)/\gamma} = 308 \times \left( \frac{80}{280} \right)^{(1.4-1)/1.4} = 215K$$

$$\dot{w}_{\text{net,in}} = 0.2 \times 1.005 \times [(426.2 - 298) - (308 - 215)] = 7.08\text{kW}$$

**TUTORIAL ENDS**

**THANK YOU**

**LUKE 19:13**

# **THERMODYNAMICS II**

## **(ME 365)**

# **TUTORIAL LECTURE**

## **UNIT 4**

**Lead Facilitator**

**Prof. F. K. Forson**

# OUTLINE

- HIGHLIGHTS ON IDEAL GAS MIXTURES
- TUTORIAL PROBLEMS ON IDEAL GAS MIXTURES
- HIGHLIGHTS ON AIR-CONDITIONING APPLICATION
- TUTORIAL PROBLEMS ON AIR-CONDITIONING APPLICATION

# HIGHLIGHT

## HIGHLIGHTS ON IDEAL GAS MIXTURES

### 1. According to Dalton's law:

(i) The pressure of a mixture of gases is equal to the sum of the partial pressure of the constituents.

(ii) The partial pressure of each constituent is that pressure which the gas would exert if it occupied alone that volume occupied by the mixture at the same temperature.

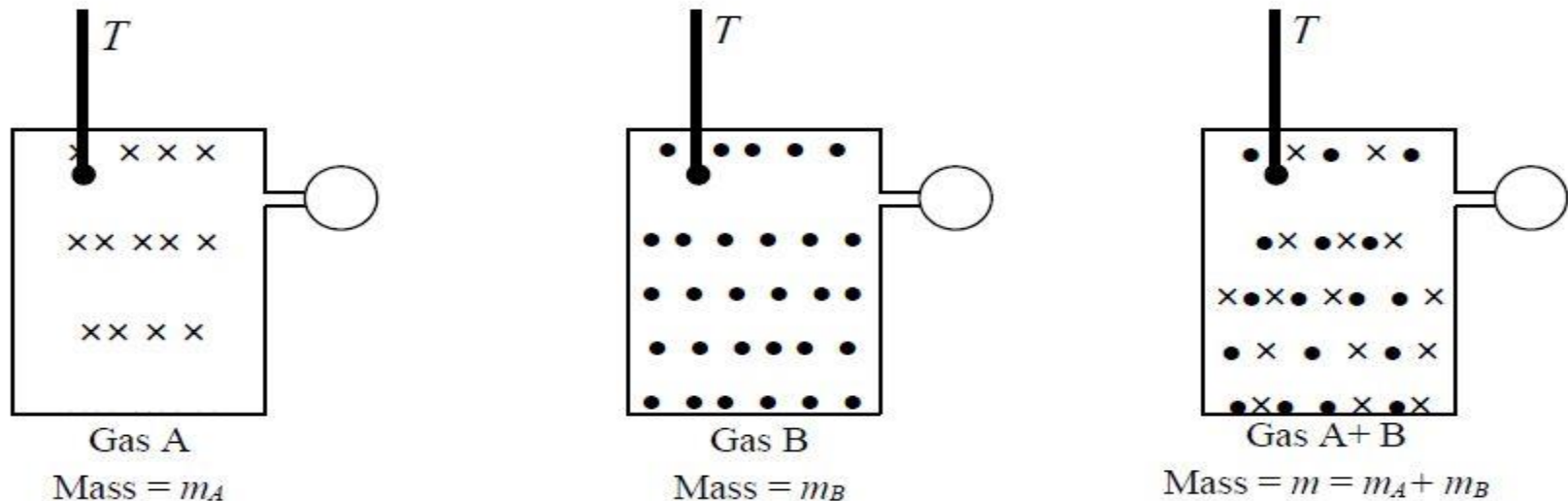


Figure 60: Illustrating the Partial Pressure of a gas

# **HIGHLIGHT**

## **HIGHLIGHTS ON IDEAL GAS MIXTURES**

2. According to Gibbs-Dalton's law:

- (i) The internal energy, Enthalpy and entropy of a gaseous mixture are respectively equal to the sum of the internal energies, enthalpies and entropies of the constituents.
- (ii) Each constituent has that internal energy, enthalpy and entropy, which it would have if it occupied alone that volume occupied by the mixture at the temperature of the mixture.

3. The characteristic equation for mixture is given as:

$$pV = nR_0T$$

Where       $n$  = Number of moles of mixture, and  
               $R_0$  = Universal gas constant

# HIGHLIGHT

## HIGHLIGHTS ON IDEAL GAS MIXTURES

4. Molecular weight (M) may be found out by using the following relations:

$$M = \sum \frac{n_i}{n} M_i \quad \text{and} \quad M = \frac{1}{\sum \frac{m_{fi}}{M_i}}$$

where  $m_f = \frac{m_i}{m}$  = mass fraction of a constituent.

5. The following condition must be satisfied in an adiabatic mixing process of perfect gas in steady flow:

$$T = \frac{\sum m_i c_{pc} T_i}{\sum m_i c_{pi}} = \frac{\sum n_i C_{pi} T_i}{\sum n_i C_{pi}} \quad .$$



**TUTORIAL**  
**PROBLEMS ON IDEAL GAS MIXTURES**

**1. A gas mixture consists of 8 kmol of  $H_2$  and 2 kmol of  $N_2$ . Determine the mass of each gas and the apparent gas constant of the mixture.**

# TUTORIAL

## PROBLEMS ON IDEAL GAS MIXTURES

**Properties** The molar masses of  $H_2$ , and  $N_2$  are 2.0 and 28.0 kg/kmol, respectively

**Analysis** The mass of each component is determined from

$$N_{H_2} = 8 \text{ kmol} \longrightarrow m_{H_2} = N_{H_2} M_{H_2} = (8 \text{ kmol})(2.0 \text{ kg/kmol}) = 16 \text{ kg}$$

$$N_{N_2} = 2 \text{ kmol} \longrightarrow m_{N_2} = N_{N_2} M_{N_2} = (2 \text{ kmol})(28 \text{ kg/kmol}) = 56 \text{ kg}$$

The total mass and the total number of moles are

$$m_m = m_{H_2} + m_{N_2} = 16 \text{ kg} + 56 \text{ kg} = 72 \text{ kg} \quad N_m = N_{H_2} + N_{N_2}$$

$$= 8 \text{ kmol} + 2 \text{ kmol} = 10 \text{ kmol}$$

The molar mass and the gas constant of the mixture are determined from their definitions,

$$M_m = \frac{m_m}{N_m} = \frac{72 \text{ kg}}{10 \text{ kmol}} = 7.2 \text{ kg/kmol}$$

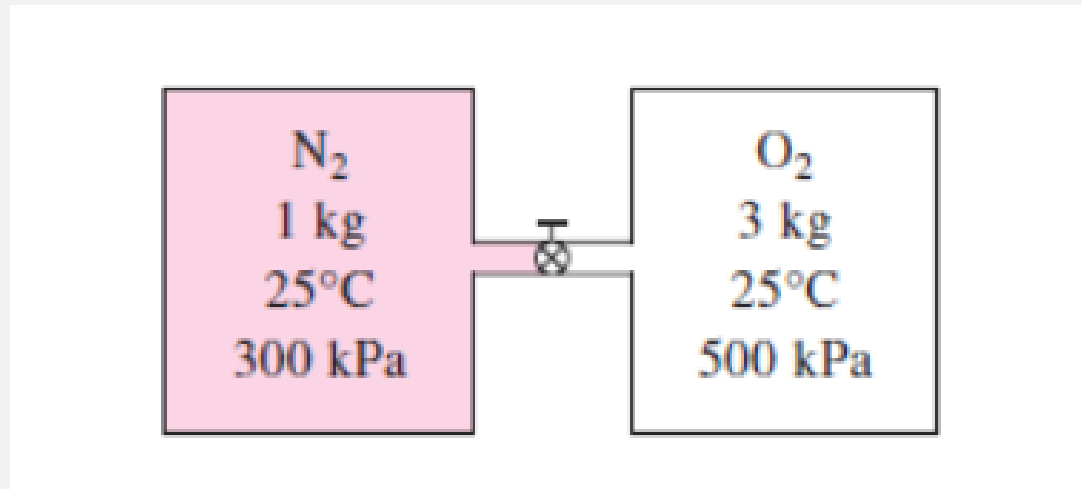
$$= 1.155 \text{ kJ/kg} \cdot \text{K}$$

$$R_m = \frac{R_u}{M_m} = \frac{8.314 \text{ kJ/kmol} \cdot \text{K}}{7.2 \text{ kg/kmol}}$$

# **TUTORIAL**

## **PROBLEMS ON IDEAL GAS MIXTURES**

2. A rigid tank that contains 1 kg of N<sub>2</sub> at 25°C and 300 kPa is connected to another rigid tank that contains 3 kg of O<sub>2</sub> at 25°C and 500 kPa. The valve connecting the two tanks is opened, and the two gases are allowed to mix. If the final mixture temperature is 25°C, determine the volume of each tank and the final mixture pressure.



# TUTORIAL

## PROBLEMS ON IDEAL GAS MIXTURES

**Assumptions** Under specified conditions both  $N_2$  and  $O_2$  can be treated as ideal gases, and the mixture as an ideal gas mixture

**Properties** The molar masses of  $N_2$  and  $O_2$  are 28.0 and 32.0 kg/kmol, respectively

**Analysis** The volumes of the tanks are

$$V_{N_2} = \left( \frac{mRT}{P} \right)_{N_2} = \frac{(1 \text{ kg})(0.2968 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})}{300 \text{ kPa}} = 0.295 \text{ m}^3$$

$$V_{O_2} = \left( \frac{mRT}{P} \right)_{O_2} = \frac{(3 \text{ kg})(0.2598 \text{ kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(298 \text{ K})}{500 \text{ kPa}} = 0.465 \text{ m}^3$$

$$V_{\text{total}} = V_{N_2} + V_{O_2} = 0.295 \text{ m}^3 + 0.465 \text{ m}^3 = 0.76 \text{ m}^3$$

$$m_{N_2} = 1 \text{ kg} \longrightarrow N_{N_2} = \frac{m_{N_2}}{M_{N_2}} = \frac{1 \text{ kg}}{28 \text{ kg/kmol}} = 0.03571 \text{ kmol}$$

Also

$$m_{O_2} = 3 \text{ kg} \longrightarrow N_{O_2} = \frac{m_{O_2}}{M_{O_2}}$$

$$= \frac{3 \text{ kg}}{32 \text{ kg/kmol}} = 0.09375 \text{ kmol}$$

$$N_m = N_{N_2} + N_{O_2}$$

$$= 0.03571 \text{ kmol} + 0.09375 \text{ kmol} = 0.1295 \text{ kmol}$$

$$P_m = \left( \frac{NR_u T}{V} \right)_m = \frac{(0.1295 \text{ kmol})(8.314 \text{ kPa} \cdot \text{m}^3/\text{kmol} \cdot \text{K})(298 \text{ K})}{0.76 \text{ m}^3} = 422.2 \text{ kPa}$$

**TUTORIAL**  
**PROBLEMS ON IDEAL GAS MIXTURES**

**3. Air has the following composition on a mole basis: 21 percent  $O_2$ , 78 percent  $N_2$ , and 1 percent Ar.**

**Determine the gravimetric analysis of air and its molar mass.**

# TUTORIAL

## PROBLEMS ON IDEAL GAS MIXTURES

**Assumptions** All the constituent gases and their mixture are ideal gases.

**Properties** The molar masses of O<sub>2</sub>, N<sub>2</sub>, and Ar are 32.0, 28.0, and 40.0 kg/kmol.

**Analysis** For convenience, consider 100 kmol of air. Then the mass of each component and the total mass are

$$N_{\text{O}_2} = 21 \text{ kmol} \longrightarrow m_{\text{O}_2} = N_{\text{O}_2} M_{\text{O}_2} = (21 \text{ kmol})(32 \text{ kg/kmol}) = 672 \text{ kg}$$

$$N_{\text{N}_2} = 78 \text{ kmol} \longrightarrow m_{\text{N}_2} = N_{\text{N}_2} M_{\text{N}_2} = (78 \text{ kmol})(28 \text{ kg/kmol}) = 2184 \text{ kg}$$

$$N_{\text{Ar}} = 1 \text{ kmol} \longrightarrow m_{\text{Ar}} = N_{\text{Ar}} M_{\text{Ar}} = (1 \text{ kmol})(40 \text{ kg/kmol}) = 40 \text{ kg}$$

$$m_m = m_{\text{O}_2} + m_{\text{N}_2} + m_{\text{Ar}} = 672 \text{ kg} + 2184 \text{ kg} + 40 \text{ kg} = 2896 \text{ kg}$$

Then the mass fraction of each component (gravimetric analysis) becomes

$$\text{mf}_{\text{O}_2} = \frac{m_{\text{O}_2}}{m_m} = \frac{672 \text{ kg}}{2896 \text{ kg}} = 0.232 \text{ or } 23.2\%$$

$$\text{mf}_{\text{N}_2} = \frac{m_{\text{N}_2}}{m_m} = \frac{2184 \text{ kg}}{2896 \text{ kg}} = 0.754 \text{ or } 75.4\%$$

$$\text{mf}_{\text{Ar}} = \frac{m_{\text{Ar}}}{m_m} = \frac{40 \text{ kg}}{2896 \text{ kg}} = 0.014 \text{ or } 1.4\%$$

The molar mass of the mixture is determined from its definitions,

$$M_m = \frac{m_m}{N_m} = \frac{2,896 \text{ kg}}{100 \text{ kmol}} = 28.96 \text{ kg / kmol}$$

## HIGHLIGHT

### HIGHLIGHTS ON AIR-CONDITIONING APPLICATION

1. 'Air-conditioning' is the simultaneous control of temperature, humidity, motion and purity of the atmosphere in a confined space,
2. 'Psychrometry' is an art of measuring moisture content of air.

The science which investigate the thermal properties of moist air, considers the measurement and control of the moisture content of air, and studies the effects of atmospheric moisture on material and human comfort may properly be termed 'Psychrometrics'.

3. When air is saturated *DBT*, *WBT*, *DPT* are equal.

# HIGHLIGHT

## HIGHLIGHTS ON AIR-CONDITIONING APPLICATION

4. Dalton's law of partial pressure states that;

*“the total pressure of a mixture of gases is equal to the sum of partial pressure which the component gases would exert if each existed alone in the mixture volume at the mixture temperature”*

5. Specific humidity,  $W = \frac{0.622 P_v}{P_t - P_v}$

6. Degree of saturation  
( $\mu$ )

$$= \frac{\text{Mass of water vapour associated with unit mass of dry air (W)}}{\text{Mass of water vapour associated with saturated unit mass of dry air (W}_s\text{)}}$$



# HIGHLIGHT

## HIGHLIGHTS ON AIR-CONDITIONING APPLICATION

7. Relative humidity,  $\phi = \frac{P_v}{P_{vs}}$

8. A 'Psychrometer' is a device which is used for measuring dry bulb and wet-bulb temperature simultaneously.

9. The processes which affect the psychrometric properties of air are called '*psychrometric process*'.

Important psychrometric processes are:

(i) Mixing of air streams

(ii) Sensible heating

(iii) Sensible cooling

(iv) Cooling and dehumidification

(v) Cooling and humidification

(vi) Heating and dehumidification

(vii) Heating and humidification

# **TUTORIAL**

## **PROBLEMS ON AIR-CONDITIONING APPLICATION**

**1. An air-water vapour mixture enters a device with a pressure of 0.15 MPa, a temperature of 40 °C, and an unknown relative humidity. The mass of dry air entering per minute is 200 g. The mixture leaves the device at 30 °C, 0.15 MPa, and 80 percent relative humidity. Water is sprayed into the air and evaporated to do this cooling. If the water temperature is 30 °C and the device is adiabatic, how much water is required to run for 1 h?**

# TUTORIAL

## PROBLEMS ON AIR-CONDITIONING APPLICATION

At the entry to the device

$$\left. \begin{aligned} T_1 &= 40^\circ\text{C} \\ \dot{m}_{a_1} &= \dot{m}_{a_2} = \dot{m}_a = 0.2 \text{ kg/min} \end{aligned} \right\} \omega_1 = ?$$

$$h_{s_1} \approx h_{g @ 40^\circ\text{C}} = 2574.3 \text{ kJ/kg}$$

$$h_{a_1} \approx c_p t_1 = 1.005 \times 40 = 40.2 \text{ kJ/kg}$$

$$h_w \approx h_f @ 30^\circ\text{C} = 125.79 \text{ kJ/kg}$$

At the exit from the device

$$\left. \begin{aligned} t_2 &= 30^\circ\text{C} \\ \phi_2 &= 0.8 \end{aligned} \right\} \omega_2 = ?$$

$$P_{s_2} = \phi_2 \times P_{g_2 @ 40^\circ\text{C}} = 0.8 \times 4.246 \text{ kPa}$$

$$= 3.3968 \text{ kPa}$$

$$P_{a_2} = P - p_{s_2} = 150 \text{ kPa} - 3.3968 \text{ kPa}$$

$$= 146.6 \text{ kPa}$$

$$\omega_2 = \frac{0.622 P_{s_2}}{P_a} = \frac{0.622 \times 3.3968}{146.6}$$

$$= 0.014412 \frac{\text{kg of H}_2\text{O}}{\text{kg of dry air}}$$

$$h_{s_2} \approx h_{g @ 30^\circ\text{C}} = 2556.3 \text{ kJ/kg}$$

$$h_{a_2} \approx c_p t_2 = 1.005 \times 30 = 30.15 \text{ kJ/kg}$$

Mass balance of water vapour

$$\dot{m}_{s_1} + \dot{m}_w = \dot{m}_{s_2} \rightarrow \dot{m}_a \omega_1 + \dot{m}_w = \dot{m}_a \omega_2$$

$$\Rightarrow \dot{m}_w = \dot{m}_a (\omega_2 - \omega_1) \dots \dots \dots (1)$$

Energy balance on the device

$$\dot{m}_a h_{a_1} + \dot{m}_a \omega_1 h_{s_1} + \dot{m}_w h_w = \dot{m}_a h_{a_2} + \dot{m}_a \omega_2 h_{s_2} \dots \dots (2)$$

Substituting equation (1) into (2) and solving for  $\omega_1$

$$\dot{m}_a h_{a_1} + \dot{m}_a \omega_1 h_{s_1} + \dot{m}_a (\omega_2 - \omega_1) h_w = \dot{m}_a h_{a_2}$$

$$+ \dot{m}_a \omega_2 h_{s_2} \dots \dots \dots (3)$$

Diving through equation (3) by  $\dot{m}_a$

$$h_{a_1} + \omega_1 h_{s_1} + (\omega_2 - \omega_1) h_w = h_{a_2} + \omega_2 h_{s_2} \dots \dots (4)$$

$$\Rightarrow 40.2 + \omega_1 (2574.3) + (0.014412 - \omega_1) \times 125.79 = 30.15$$

$$+ (0.014412 \times 2556.3)$$

$$\Rightarrow \omega_1 = 0.010202 \frac{\text{kg of H}_2\text{O}}{\text{kg of dry air}}$$

Substituting  $\omega_1$  into (2)

$$\Rightarrow \dot{m}_w = 60.5 (0.014412 - 0.010202)$$

$$= 8.42 \times 10^{-4} \text{ kg/min} = 50.5 \text{ g/h}$$

# TUTORIAL

## PROBLEMS ON AIR-CONDITIONING APPLICATION

2. If the dew point temperature is 24 °C and the dry-bulb temperature is 30 °C, determine the wet-bulb temperature  $T_{wet\ bulb}$ , the humidity ratio  $\omega$ , the partial pressure of the vapour  $P$ , and the relative humidity  $\phi$ .

# TUTORIAL

## PROBLEMS ON AIR-CONDITIONING APPLICATION

Since the pressure of the air-water vapour mixture is not given, it is assumed to be the local atmospheric pressure thus 101.325 kPa.

From the psychrometric chart attached

Given  $T_{db} = 30^{\circ}\text{C}$   
 $T_{dewp} = 24^{\circ}\text{C}$

The point can be located on the chart as shown as point Q2.

$$\omega = 0.019 \pm 0.0002 \frac{\text{kg of H}_2\text{O}}{\text{kg of dry air}}$$

$$P_v = p_s = 22.1 \text{ mmHg}$$
$$= \frac{22.1}{760} \times 101.325 = 2.94 \pm 0.02 \text{ kPa}$$

$$P_g = P_{\text{sat}@30^{\circ}\text{C}} = 4.246 \text{ kPa}$$

$$\phi = \frac{P_v}{P_g} = \frac{2.94}{4.246} = 70\%$$

$$T_{\text{wetbulb}} = 25.5 \pm 0.2^{\circ}\text{C}$$

# TUTORIAL

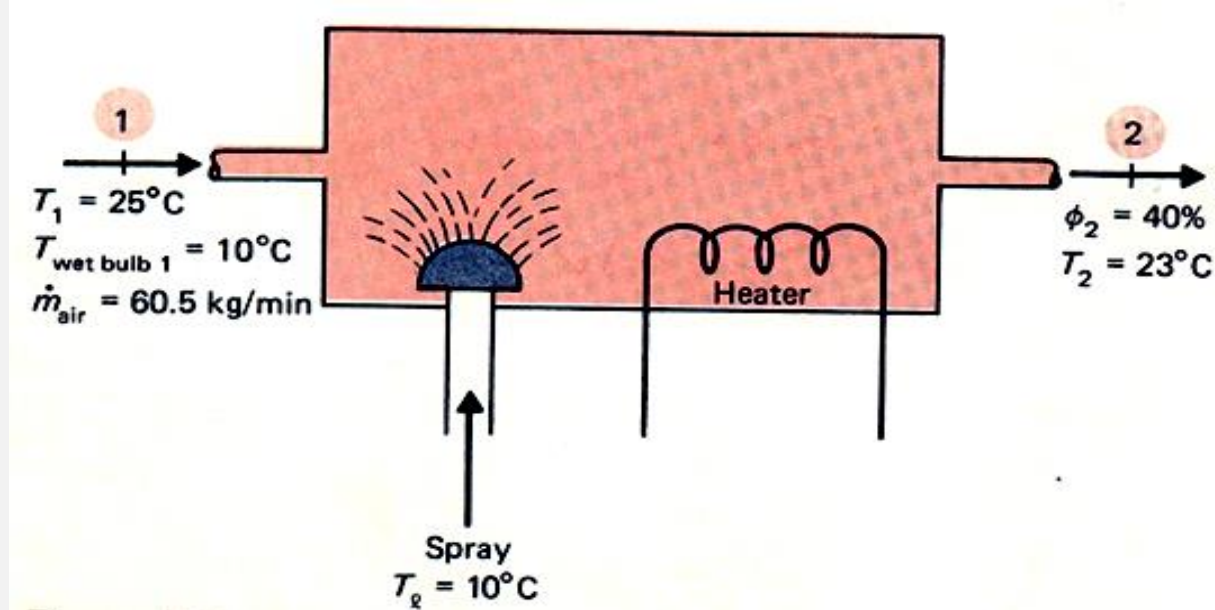
## PROBLEMS ON AIR-CONDITIONING APPLICATION

3. The air conditioner shown in Figure Q.3 operates at a pressure of 101.3 kPa.

(a) Draw the entire process on a psychrometric chart.

(b) Determine the mass flow rate of water required by the spray.

(c) Determine the rate of heat transfer.



# TUTORIAL

## PROBLEMS ON AIR-CONDITIONING APPLICATION

### State 1

$$\left. \begin{array}{l} T_1 = 25^\circ\text{C} \\ T_{wb} = 10^\circ\text{C} \end{array} \right\} \omega_1 = 0.0015 \frac{\text{kg of H}_2\text{O}}{\text{kg of dry air}}$$

$$h_1 = 29 \text{ kJ/kg of dry air}$$

### State 2

$$\left. \begin{array}{l} T_2 = 23^\circ\text{C} \\ \phi_2 = 40\% \end{array} \right\} \omega_2 = 0.007 \frac{\text{kg of H}_2\text{O}}{\text{kg of dry air}}$$

$$h_2 = 41 \text{ kJ/kg of dry air}$$

$$h_w = h_{f @ 10^\circ\text{C}} = 29 \text{ kJ/kg of dry air}$$

### Mass balance of water vapour

$$\dot{m}_{s_1} + \dot{m}_w = \dot{m}_{s_2} \rightarrow \dot{m}_a \omega_1 + \dot{m}_w = \dot{m}_a \omega_2$$

$$\Rightarrow \dot{m}_w = \dot{m}_a (\omega_2 - \omega_1) \dots \dots \dots (1)$$

$$\Rightarrow \dot{m}_w = 60.5(0.007 - 0.0015)$$

$$= 0.3328 \text{ kg / min}$$

### Energy balance on the device

$$\dot{m}_a h_1 + \dot{m}_w h_w + \dot{Q} = \dot{m}_a h_2 \text{ --- (2)}$$

$$\Rightarrow \dot{Q} = \dot{m}_a h_2 - \dot{m}_a h_1 - \dot{m}_w h_w = 60.5(41) -$$

$$0.3328(42) - 60.5(29)$$

$$= 712 \text{ kJ / min} = 11.87 \text{ kW}$$

# **TUTORIAL**

## **PROBLEMS ON AIR-CONDITIONING APPLICATION**

**5. Air enters a heating section at 95 kPa, 15 °C, and 30 % relative humidity at a rate of 6 m<sup>3</sup>/min, and it leaves at 25 °C. Determine:**

- a) The rate of heat transfer in the heating section, and**
- b) The relative humidity of the air at the exit**



The properties of the air at both states are determined to be

$$P_{s_1} = \phi_1 P_{\text{sat @ } 15^\circ\text{C}} = 0.3 \times 1.7050 \text{ kPa} = 0.51 \text{ kPa} \quad P_{a_1} = P_1 - P_{s_1} = 95 - 0.51 = 94.49 \text{ kPa}$$

$$v_1 = \frac{R_a T_1}{P_{a_1}} = \frac{0.287 \times 288}{94.49} = 0.875 \text{ m}^3/\text{kg}$$

$$\omega_1 = \omega_2 = \frac{0.622 P_{s_1}}{P_{a_1}} = \frac{0.622 \times 0.51}{94.49}$$

$$= 0.00336 \frac{\text{kg H}_2\text{O}}{\text{kg dry air}}$$

$$h_1 = c_p t_1 + \omega_1 h_{s_1}$$

$$= 1.005 \times 15 + 0.00336 \times [2500.9 + (1.82 \times 15)] \quad h_1 = 15.08 + 8.49 = 23.57 \text{ kJ/kg dry air}$$

$$h_2 = c_p t_2 + \omega_2 h_{s_2}$$

$$= 1.005 \times 25 + 0.00336 \times [2500.9 + (1.82 \times 25)]$$

$$h_2 = 25.13 + 8.56 = 33.69 \text{ kJ/kg dry air}$$

$$\dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a = \frac{\dot{V}_1}{v_1} = \frac{6 \text{ m}^3/\text{min}}{0.875 \text{ m}^3/\text{kg}}$$

$$= 6.875 \frac{\text{kg}}{\text{min}}$$

From energy balance on the balance, we obtain:

$$(a). Q_{12} = \dot{m}_a (h_2 - h_1) = 6.857(33.69 - 23.57)$$

$$= 69.39 \text{ kJ}/\text{min}$$

Note that the vapour pressure of air remains constant during a simple heating process

$$(b). \phi_2 = \frac{P_{v_2}}{P_{g_2}} \times 100\% = \frac{0.51}{P_{\text{sat @ } 25^\circ\text{C}}} \times 100\%$$

$$= \frac{0.51}{3.169} \times 100\% = 16.1 \%$$

# **TUTORIAL**

## **PROBLEMS ON AIR-CONDITIONING APPLICATION**

**6. Air enters a 40-cm-diameter cooling section at 1 atm, 32 °C, and 30 % relative humidity at 18 m/s. Heat is removed from the air at a rate of 1200 kJ/min. Determine:**

- a) the exit temperature**
- b) the exit relative humidity of the air, and**
- c) the exit velocity**

The properties of the air at inlet are determined from the psychrometric chart to be

$$\left. \begin{array}{l} T_1 = 32 \text{ }^\circ\text{C} \\ \phi_1 = 30 \% \\ T_2 = ? \\ \omega_1 = \omega_2 \end{array} \right\} \begin{array}{l} \omega_1 = 0.0089 \frac{\text{kg H}_2\text{O}}{\text{kg dry air}} \\ v_1 = 0.877 \text{ m}^3/\text{kg dry air} \\ h_1 = 55.0 \text{ kJ/kg dry air} \\ h_2 = ? \\ \omega_2 = 0.0089 \frac{\text{kg H}_2\text{O}}{\text{kg dry air}} \end{array}$$

$$\dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a = \frac{A_1 C_1}{v_1} \quad \dot{m}_a = \frac{1}{0.877} \times 18 \left[ \frac{\pi}{4} \times 0.4^2 \right] = 2.58 \text{ kg/s}$$

From energy balance in the cooling section, we write  $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$

$$\text{i.e. } \dot{m}_{a_1} h_1 = \dot{m}_{a_2} h_2 + |\dot{Q}_{\text{out}}|$$

$$\text{But } \dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a$$

$$|\dot{Q}_{\text{out}}| = \dot{m}_a (h_1 - h_2)$$

$$1200 \text{ kJ} / 60\text{s} = 2.58 (55 - h_2)$$

$$h_2 = 47.24 \text{ kJ/kg dry air}$$

With

$$\left. \begin{array}{l} \omega_2 = 0.0089 \frac{\text{kg H}_2\text{O}}{\text{kg dry air}} \\ h_2 = 47.24 \frac{\text{kJ}}{\text{kg dry air}} \end{array} \right\} \begin{array}{l} T_2 = 24.4 \text{ }^\circ\text{C} \\ \phi_2 = 46.6 \% \\ v_2 = 0.856 \frac{\text{m}^3}{\text{kg dry air}} \end{array}$$

For the exit velocity  $\dot{m}_{a_1} = \dot{m}_{a_2} \Rightarrow$

$$\frac{A_1 C_1}{v_1} = \frac{A_2 C_2}{v_2} \text{ but } A_1 = A_2$$

$$C_2 = \frac{v_2}{v_1} C_1 = \frac{0.856}{0.877} \times 18 \text{ m/s} = 17.57 \text{ m/s}$$

# **TUTORIAL**

## **PROBLEMS ON AIR-CONDITIONING APPLICATION**

**7. Air enters a window air-conditioner at 1 atm, 32 °C and 30 % relative humidity at 8 m<sup>3</sup>/min, and it leaves as saturated air 12 °C. Part of the moisture in the air that condenses during the process is also removed at 12 °C.**

**Determine:**

- a) the rate of heat removal from the air**
- b) the rate of moisture removal from the air**

# TUTORIAL

## PROBLEMS ON AIR-CONDITIONING APPLICATION

Cooling with dehumidification

From the psychrometric chart the properties of air at the various states are determined to be

$$\left. \begin{array}{l} T_1 = 32^\circ\text{C} \\ \phi_1 = 70\% \end{array} \right\} \begin{array}{l} \omega_1 = 0.0212 \frac{\text{kg H}_2\text{O}}{\text{kg dry air}} \\ v_1 = 0.894 \text{ m}^3/\text{kg dry air} \\ h_1 = 86.4 \text{ kJ/kg dry air} \end{array}$$

$$\text{With } v_1 = 8 \frac{\text{m}^3}{\text{min}} = \dot{m}_{a_1} = \frac{\dot{V}_1}{v_1} = \frac{8}{0.894} = 8.95 \frac{\text{kg}}{\text{min}}$$

$$\left. \begin{array}{l} T_2 = 12^\circ\text{C} \\ \phi_2 = 100\% \end{array} \right\} \begin{array}{l} h_2 = 34.1 \text{ kJ/kg dry air} \\ \omega_2 = 0.0087 \frac{\text{kg H}_2\text{O}}{\text{kg dry air}} \end{array}$$

From steam tables

$$h_w = h_{f@T=12^\circ\text{C}} = 50.4 \text{ kJ/kg}$$

Note  $\omega_2 < \omega_1$  so from mass balance, we write

$$\dot{m}_{s_1} = \dot{m}_{s_2} + \dot{m}_w$$

$$\dot{m}_{a_1} \omega_1 = \dot{m}_{a_2} \omega_2 + \dot{m}_w$$

$$\dot{m}_{a_1} = \dot{m}_{a_2} = \dot{m}_a = 8.95 \frac{\text{kg}}{\text{min}}$$

$$\text{Hence, } \dot{m}_w = \dot{m}_a (\omega_1 - \omega_2)$$

$$\dot{m}_w = 8.95(0.0212 - 0.0087) = 0.112 \frac{\text{kg}}{\text{min}}$$

From energy balance, we write  $\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$  as follows

$$\dot{m}_w h_w + |\dot{Q}_{\text{out}}| + \dot{m}_{a_2} h_2 = \dot{m}_{a_1} h_1$$

Therefore

$$|\dot{Q}_{\text{out}}| = \dot{m}_{a_1} (h_1 - h_2) - \dot{m}_w h_w$$

$$|\dot{Q}_{\text{out}}| = 8.95(86.4 - 34.1) - 0.112(50.4)$$

$$|\dot{Q}_{\text{out}}| = 462.4 \frac{\text{kJ}}{\text{min}}$$

# **TUTORIAL**

## **PROBLEMS ON AIR-CONDITIONING APPLICATION**

**8. Air at 1 atm, 15 °C and 60 % relative humidity is first heated to 20 °C in a heating section and then humidified by introducing water vapour. The air leaves the humidifying section at 25 °C and 65 % relative humidity. Determine:**

- a) The amount of water added to the air, and**
- b) The amount of heat transfer to the air in the heat section.**

The properties of the air at various sections are determined from the psychrometric chart to

$$\left. \begin{array}{l} T_1 = 15^\circ\text{C} \\ \phi_1 = 60\% \end{array} \right\} \begin{array}{l} h_1 = 21.1 \text{ kJ/kg dry air} \\ \omega_1 = 0.0064 \frac{\text{kg H}_2\text{O}}{\text{kg dry air}} \end{array}$$

Note that

$$\omega_1 = \omega_2 = 0.0064 \frac{\text{kg H}_2\text{O}}{\text{kg dry air}}$$

$$\left. \begin{array}{l} T_3 = 25^\circ\text{C} \\ \phi_3 = 65\% \end{array} \right\} \begin{array}{l} h_3 = 58.1 \text{ kJ/kg dry air} \\ \omega_3 = 0.0129 \frac{\text{kg H}_2\text{O}}{\text{kg dry air}} \end{array}$$

Note also that there is humidification between states 2 and state 3 and therefore  $\omega_2 < \omega_3$  indicating water has been added per unit mass of dry air.

$$(a) \Delta\omega = \omega_3 - \omega_2 = 0.0129 - 0.0064 = 0.0065 \frac{\text{kg H}_2\text{O}}{\text{kg dry air}}$$

$$(b). \frac{\dot{Q}_{12}}{\dot{m}_a} = h_2 - h_1 = q_{12}$$

$$\left. \begin{array}{l} \omega_3 = 0.0064 \frac{\text{kg H}_2\text{O}}{\text{kg dry air}} \\ T_2 = 20^\circ\text{C} \end{array} \right\} h_3 = 26.2 \frac{\text{kJ}}{\text{kg dry air}}$$

$$q_{12} = 26.2 - 21.1 = 5.1 \text{ kJ/kg dry air}$$

**TUTORIAL ENDS**

**THANK YOU**

**LUKE 19:13**



# **THERMODYNAMICS II**

## **(ME 365)**

# **TUTORIAL LECTURE**

## **UNIT 5**

Lead Facilitator  
**Prof. F. K. Forson**

# OUTLINE

- HIGHLIGHTS ON REACTING MIXTURES AND COMBUSTIONS
- TUTORIAL PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS

# **HIGHLIGHT**

## **HIGHLIGHTS ON REACTING MIXTURES AND COMBUSTIONS**

- 1. A chemical reaction may be defined as the rearrangement of atoms due to redistribution of electrons. ‘Reactants’ comprise of initial constituents which start the reaction while ‘products’ comprise of final constituents which are formed by the chemical reaction.**
- 2. A chemical fuel is a substance which releases heat energy on combustion.**
- 3. The total number of atoms of each element concerned in the combustion process remains constant, but the atoms are rearranged into groups having different chemical properties.**

# HIGHLIGHT

## HIGHLIGHTS ON REACTING MIXTURES AND COMBUSTIONS

4. The amount of excess air supplied varies with the type of the fuel and the firing conditions. It may approach a value of 100 per cent but modern practice is to use 25% to 50% excess air.

5. Stoichiometric (or chemically correct) mixture of air and fuel is one that contains just sufficient oxygen for complete combustion of the fuel. For stoichiometric complete combustion of a hydrocarbon fuel with air the products of combustion are  $\text{CO}_2$ ,  $\text{H}_2\text{O}$ , and  $\text{N}_2$

# HIGHLIGHT

## HIGHLIGHTS ON REACTING MIXTURES AND COMBUSTIONS

**6. Mixture strength = (Stoichiometric A/F ratio)/  
(Actual A/F ratio)**

**7. When analysis of combustion products is known  
air fuel ratio can be calculated by the following  
methods:**

**(a) Fuel composition known**

**(i) Carbon balance method**

**(ii) Hydrogen balance method**

**(iii) Carbon hydrogen balance method**

**(b) Fuel composition unknown**

**(i) Carbon hydrogen balance method.**

# HIGHLIGHT

## HIGHLIGHTS ON REACTING MIXTURES AND COMBUSTIONS

8. The most common means of DRY analysis of the combustion products is Orsat apparatus.

9. The enthalpy of formation ( $\Delta H_f$ ) is the increase in enthalpy when a compound is formed from its constituent elements in their natural form and in a standard state. The standard state is 25°C and 1 atm. pressure (but it must be borne in mind that not all substances can exist in natural form, e.g.  $H_2O$  cannot be a vapour at 1 atm. and 25°C).

# HIGHLIGHT

## HIGHLIGHTS ON REACTING MIXTURES AND COMBUSTIONS

10.

$$\begin{aligned}(\text{HHV})_p &= (\text{LHV})_p + m h_{fg} \\ (\text{HHV})_v &= (\text{LHV})_v + m(u_g - u_f)\end{aligned}$$

where HHV = Higher heating value,  
LHV = Lower heating value,  
 $m$  = Mass of water formed by combustion,  
 $h_{fg}$  = Enthalpy of vapourisation of water, kJ/kg,  
 $u_g$  = Specific internal energy of vapour, kJ/kg, and  
 $u_f$  = Specific internal energy of liquid, kJ/kg.

**HHV<sub>v</sub> = Higher heating value at constant volume**

**HHHp = Higher heating value at constant pressure**

**hfg @ 25 °C = 2443 kJ/kg**

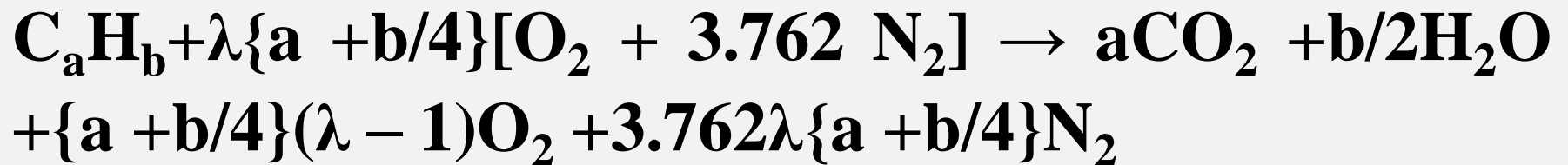
**11. In a given combustion process, that takes place adiabatically and with no work or changes in kinetic or potential energy involved, the temperature of the products is referred to as the ‘*adiabatic flame temperature*’.**

# HIGHLIGHT

## HIGHLIGHTS ON REACTING MIXTURES AND COMBUSTIONS

12. For a given fuel and given pressure and temperature of the reactants, the maximum adiabatic flame temperature that can be achieved is with a 'stoichiometric' mixture.

13. General chemically balanced complete combustion equation of a known hydrocarbon fuel with  $\Phi \leq 1$  or  $\lambda \geq 1$ . Note  $\lambda = 1/\Phi$ .





# HIGHLIGHT

## HIGHLIGHTS ON REACTING MIXTURES AND COMBUSTIONS

14. General complete combustion equation of a known hydrocarbon fuel and known dry volumetric analysis of products of combustion

Fuel  $C_aH_b$

Products  $X\% CO_2$ ;  $Y\% CO$ ,  $Z\% O_2$ ;  $w\% N_2$  only.

Combustion equation:



Determination of the unknown values is by carbon balance, nitrogen balance, and hydrogen-oxygen balance.

Carbon balance:  $a = X + Y$  ----- (4)

Oxygen balance:  $2u = 2X + Y + 2Z + d$  ----- (2)

Nitrogen balance:  $7.524u = 2W$  ----- (3)

Hydrogen balance:  $b = 2d$  ----- (1)

Given the known values  $a$ ,  $b$ ,  $X$ ,  $Y$ ,  $Z$  and  $W$ , the unknown values  $u$ ,  $d$  are solved

# **TUTORIAL**

## **PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS**

**1. Propane ( $\text{C}_3\text{H}_8$ ) is burned with 75 percent excess air during a combustion process. Assuming complete combustion, determine the air–fuel ratio.**

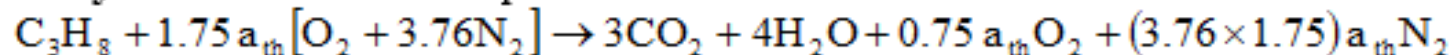
# TUTORIAL

## PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS

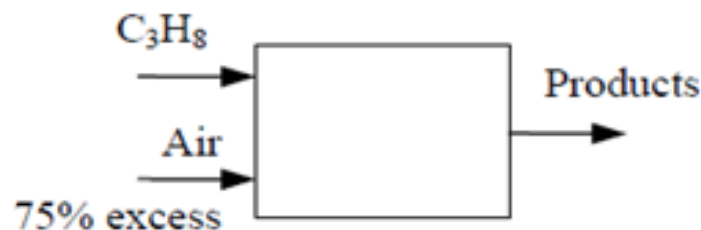
**Assumptions** 1 Combustion is complete. 2 The combustion products contain  $\text{CO}_2$ ,  $\text{H}_2\text{O}$ ,  $\text{O}_2$ , and  $\text{N}_2$  only.

**Properties** The molar masses of C,  $\text{H}_2$ , and air are 12 kg/kmol, 2 kg/kmol, and 29 kg/kmol, respectively.

**Analysis** The combustion equation in this case can be written as

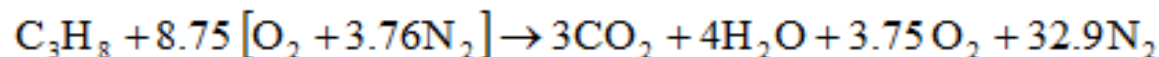


Where  $a_{\text{th}}$  is the stoichiometric coefficient for air. We have automatically accounted for the 75% excess air by using the factor  $1.75 a_{\text{th}}$  instead of  $a_{\text{th}}$  for air. The stoichiometric amount of oxygen ( $a_{\text{th}} \text{O}_2$ ) will be used to oxidize the fuel, and the remaining excess amount ( $0.75 a_{\text{th}} \text{O}_2$ ) will appear in the products as free oxygen. The coefficient  $a_{\text{th}}$  is determined from the  $\text{O}_2$  balance,



Oxygen balance

$$1.75 a_{\text{th}} = 3 + (0.5 \times 4) + 0.75 a_{\text{th}} \Rightarrow a_{\text{th}} = 5$$



The air-fuel ratio is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$A/F = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{(8.75 \times 4.76) \times 29}{(3 \times 12) + (4 \times 2)} = \mathbf{27.5 \text{ kg of air per kg of fuel}}$$

## **TUTORIAL**

### **PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS**

**2. A gaseous fuel with a volumetric analysis of 60 percent  $\text{CH}_4$ , 30 percent  $\text{H}_2$ , and 10 percent  $\text{N}_2$  is burned to completion with 130 percent theoretical air. Determine (a) the air–fuel ratio and (b) the fraction of water vapor that would condense if the product gases were cooled to  $20^\circ\text{C}$  at 1 atm.**

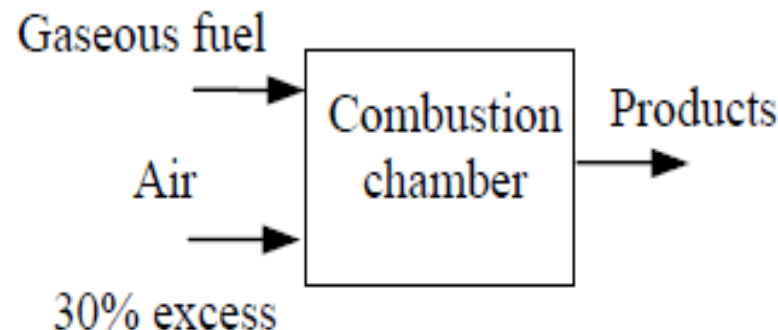
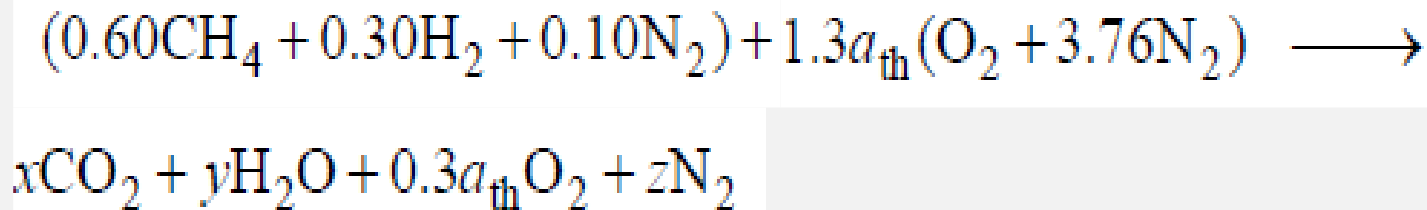
# TUTORIAL

## PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS

**Assumptions** 1 Combustion is complete. 2 The combustion products contain  $\text{CO}_2$ ,  $\text{H}_2\text{O}$ ,  $\text{O}_2$ , and  $\text{N}_2$  only.

**Properties** The molar masses of C,  $\text{H}_2$ ,  $\text{N}_2$ , and air are 12 kg/kmol, 2 kg/kmol, 28 kg/kmol, and 29 kg/kmol, respectively

**Analysis** (a) The fuel is burned completely with excess air, and thus the products will contain  $\text{H}_2\text{O}$ ,  $\text{CO}_2$ ,  $\text{N}_2$ , and some free  $\text{O}_2$ . Considering 1 kmol of fuel, the combustion equation can be written as



# TUTORIAL

## PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS

The unknown coefficients in the above equation are determined from mass balances,

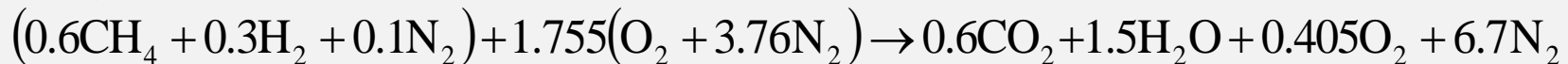
$$\text{C} : 0.60 = x \longrightarrow x = 0.60$$

$$\text{H} : 0.60 \times 4 + 0.30 \times 2 = 2y \longrightarrow y = 1.50$$

$$\text{O}_2 : 1.3a_{\text{th}} = x + y/2 + 0.3a_{\text{th}} \longrightarrow a_{\text{th}} = 1.35$$

$$\text{N}_2 : 0.10 + 3.76 \times 1.3a_{\text{th}} = z \longrightarrow z = 6.70$$

Thus,



The air-fuel ratio for the this reaction is determined by taking the ratio of the mass of the air to the mass of the fuel,

$$m_{\text{air}} = (1.755 \times 4.76 \text{ kmol})(29 \text{ kg/kmol}) = 242.3 \text{ kg}$$

$$m_{\text{fuel}} = (0.6 \times 16 + 0.3 \times 2 + 0.1 \times 28) \text{ kg} = 13.0 \text{ kg}$$

$$A/F = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{242.3}{13} = 18.64 \text{ kg of air per kg of fuel}$$

# TUTORIAL

## PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS

(b) For each kmol of fuel burned,  $0.6 + 1.5 + 0.405 + 6.7 = 9.205$  kmol of products are formed, including 1.5 kmol of  $\text{H}_2\text{O}$ . Assuming that the dew-point temperature of the products is above  $20^\circ\text{C}$ , some of the water vapor will condense as the products are cooled to  $20^\circ\text{C}$ . If  $N_w$  kmol of  $\text{H}_2\text{O}$  condenses, there will be  $1.5 - N_w$  kmol of water vapor left in the products. The mole number of the products in the gas phase will also decrease to  $9.205 - N_w$  as a result. Treating the product gases (including the remaining water vapor) as ideal gases,  $N_w$  is determined by equating the mole fraction of the water vapor to its pressure fraction,

$$\frac{N_v}{N_{\text{prod, gas}}} = \frac{P_v}{P_{\text{prod}}} \quad \frac{1.5 - N_w}{9.205 - N_w} = \frac{2.3392 \text{ kPa}}{101.325 \text{ kPa}} \rightarrow N_w = 1.32 \text{ kmol}$$

Since  $P_v = P_{\text{sat}} @ 20^\circ\text{C} = 2.3392 \text{ kPa}$ . Thus the fraction of water vapor that condenses is  $1.32/1.5 = 0.88$  or **88%**.

# **TUTORIAL**

## **PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS**

**3. Benzene gas ( $\text{C}_6\text{H}_6$ ) at  $25^\circ\text{C}$  is burned during a steady-flow combustion process with 95 percent theoretical air that enters the combustion chamber at  $25^\circ\text{C}$ . All the hydrogen in the fuel burn to  $\text{H}_2\text{O}$ , but part of the carbon burns to  $\text{CO}$ . If the products leave at  $1000\text{ K}$ , determine**

**(a) the mole fraction of the  $\text{CO}$  in the products and**

**(b) the heat transfer from the combustion chamber during this process.**



# TUTORIAL

## PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS

The actual combustion Equation is:



Balancing  $\text{O}_2$  we obtain the value of  $x$  as:

$$7.125 = x + \frac{6 - x}{2} + 1.5 \Rightarrow x = 5.25$$

Hence the actual combustion Equation is



(a) Mole fraction of CO in the products is given by

$$y_{\text{CO}} = \frac{N_{\text{CO}}}{N_{\text{total}}} = \frac{0.75}{(5.25 + 0.75 + 3 + 26.79)} = 0.021 \text{ or } 2.1 \%$$

(b) From Energy balance on the combustion chamber

$$\begin{aligned} \tilde{H}_R &= \tilde{Q}_{\text{out}} + \tilde{H}_P \Rightarrow -\tilde{Q}_{\text{out}} = \tilde{H}_P - \tilde{H}_R \\ -\tilde{Q}_{\text{out}} &= \sum N_P (\tilde{h}_f^\theta + \tilde{h}_T - \tilde{h}_{298\text{K}}^\theta) - \sum N_R (\tilde{h}_f^\theta + \tilde{h}_{298\text{K}} - \tilde{h}_{298\text{K}}^\theta) \end{aligned}$$

# TUTORIAL

## PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS

From tables, we write

$$-\tilde{Q}_{\text{out}} = 5.25(-393,520 + 42,769 - 9,364) + 0.75(-110,530 + 30,355 - 8669) + 3(-241,820 + 35,882 - 9,904) \\ + 26.79(0 + 30,129 - 8,669) - 1(82,930) - 0 - 0$$

$$-\tilde{Q}_{\text{out}} = -1,890,603.75 - 6,663.3 - 647,526 + 574,913.4 - 82,930$$

$$-\tilde{Q}_{\text{out}} = -2,112,779.35 \text{ kJ/kmol of } \text{C}_6\text{H}_6$$

$$\therefore \tilde{Q}_{\text{out}} = 2,112,779.4 \text{ kJ/kmol of } \text{C}_6\text{H}_6$$

Extract of values of table values

Substance	$\tilde{h}_f^\theta$	$\tilde{h}_{1000\text{K}}$	$\tilde{h}_{298\text{K}}$
$\text{C}_6\text{H}_6$	86,930	—	—
$\text{O}_2$	0	31,389	8,682
$\text{N}_2$	0	30,129	8,669
$\text{H}_2\text{O}(\text{g})$	-241,820	35,882	9,904
CO	-110,530	30,355	8,669
$\text{CO}_2$	-393,520	42,769	9,364

All values are in kJ/kmol

## **TUTORIAL**

### **PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS**

**4. Acetylene gas ( $\text{C}_2\text{H}_2$ ) at  $25^\circ\text{C}$  is burned during a steady-flow combustion process with 30 percent excess air at  $27^\circ\text{C}$ . It is observed that 75,000 kJ of heat is being lost from the combustion chamber to the surroundings per kmol of acetylene. Assuming combustion is complete; determine the exit temperature of the product gases.**

# TUTORIAL

## PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS

$$\tilde{H}_R = 75,000 + \tilde{H}_P$$

$$-75,000 = \tilde{H}_P - \tilde{H}_R$$

$$= \sum N_P (\tilde{h}_f^\theta + \tilde{h}_T - \tilde{h}_{298K}^\theta)_P - \sum N_R (\tilde{h}_f^\theta + \tilde{h}_{298K} - \tilde{h}_{298K}^\theta)_R$$

Stoichiometric combustion Equation is:



The actual combustion Equation is:



Thus,

$$\begin{aligned} -75,000 &= 2(-393,520 + \tilde{h}_{\text{CO}_2} - 9,364) + 1(-241,820 + \tilde{h}_{\text{H}_2\text{O}} - 9,904) + 0.75(0 + \tilde{h}_{\text{O}_2} - 8,682) \\ &\quad + 12.22(0 + \tilde{h}_{\text{N}_2} - 8,669) - 1(226,730) - 3.25(0 + 8,736 - 8,682) - 12.22(0 + 8,723 - 8,669) \end{aligned}$$

$$\begin{aligned} &= -805,768 + 2\tilde{h}_{\text{CO}_2} - 251,724 + \tilde{h}_{\text{H}_2\text{O}} - 6,511.5 + 0.75\tilde{h}_{\text{O}_2} - 105,935.18 + 12.22\tilde{h}_{\text{N}_2} \\ &\quad - 226,730 - 175.5 - 659.88 \end{aligned}$$

$$-75,000 = -1,397,504.06 + 2\tilde{h}_{\text{CO}_2} + \tilde{h}_{\text{H}_2\text{O}} + 0.75\tilde{h}_{\text{O}_2} + 12.22\tilde{h}_{\text{N}_2}$$

$$2\tilde{h}_{\text{CO}_2} + \tilde{h}_{\text{H}_2\text{O}} + 0.75\tilde{h}_{\text{O}_2} + 12.22\tilde{h}_{\text{N}_2} = 1,322,504.06 \text{-----Eqn 1}$$

# TUTORIAL

## PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS

Find  $T_p$  by try and error. Assume  $T_p = 2500$  K

L.H.S = 1,352,961 kJ (higher than the R.H.S of Equation 1)

Try  $T_p = 2300$  K

L.H.S = 1,320, 517 kJ (lower than the R.H.S of Equation 1)

$$\text{By interpolating } T_p = -200 \left[ \frac{(1322504.06 - 1352961)}{(1320517 - 1352961)} \right]$$
$$= \mathbf{2312 \text{ K}}$$

Substance	$\tilde{h}_f^\theta$	$\tilde{h}_{300\text{K}}$	$\tilde{h}_{298\text{K}}$
$\text{C}_2\text{H}_2$	226,730	—	—
$\text{O}_2$	0	8,736	8,682
$\text{N}_2$	0	8,723	8,669
$\text{H}_2\text{O}(\text{g})$	-241,820	—	9,904
$\text{CO}_2$	-393,520	—	9,364

All values are in kJ/kmol

# TUTORIAL

## PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS

5. A cigarette lighter burns gaseous butane with 200 percent theoretical air. The fuel and air are at 25 °C, and the products of combustion are at 127 °C and  $P = 100 \text{ kPa}$ .

- (a) How much excess air is used?
- (b) Work out the balanced actual combustion equation.
- (c) Find the air-fuel ratio on a mass basis.
- (d) Find  $H_r$ .
- (e) Find  $H_p$ .
- (f) How much heat transfer is released during the combustion?
- (g) Find the dew point temperature of the  $H_2O$ .

Note:  $\tilde{M}_{\text{butane}} = 53.123 \text{ kJ/k mol}$ ;  $\tilde{M}_{\text{air}} = 28.97 \text{ kJ/k mol}$  are to be used in problem 5 above.

# TUTORIAL

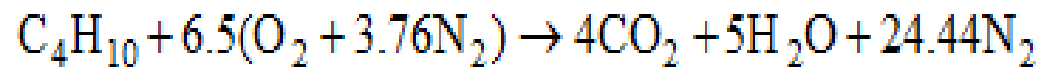
## PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS

Stoichiometric combustion Equation is:



Balancing  $\text{O}_2 \Rightarrow a_{\text{th}} = 4 + 2.5 = 6.5$

Hence, the stoichiometric combustion equation is:



(b) With 200 % theoretical air supplied (i.e.  $\lambda = 2$ ), the actual combustion is,



Balancing  $\text{O}_2 \Rightarrow 13 = 4 + 2.5 + x = 6.5 \Rightarrow x = 6.5$



$$\text{(a) \% excess air} = \frac{n(\text{O}_2)_{\text{actual}} - n(\text{O}_2)_{\text{th}}}{n(\text{O}_2)_{\text{th}}} \times 100\% = \frac{13 - 6.5}{6.5} \times 100\% = 100\% \text{ or } (\lambda - 1) \times 100\% = 100\%$$

$$\text{(c) } \frac{A}{F} = \frac{m_{\text{air}}}{m_{\text{fuel}}} = \frac{N_{\text{air}} \times \tilde{M}_{\text{air}}}{N_{\text{fuel}} \times \tilde{M}_{\text{fuel}}} = \frac{4.76 \times 13 \times 29}{1 \times 58.123} = 30.84$$

$$\text{(d) } \tilde{H}_{\text{R}} = -126,150 \text{ kJ/mol}$$

# TUTORIAL

## PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS

$$(e) \tilde{H}_P = \sum N_P (\tilde{h}_f^\theta + \tilde{h}_{400K} - \tilde{h}_{298K}^\theta)$$

=

$$4(-393,520 + 13,372 - 9,364) + 5(-241,820 + 13,356 - 9,904) + 6.5(0 + 8384 - 8682) + 44.88(0 + 11,640 - 8,669) \\ = -1,558,048 - 1,191,840 - 1937 + 133338.48$$

$$H_P = -2618486.52 \text{ kJ/kmol}$$

$$- \tilde{Q}_{out} = \tilde{H}_P - \tilde{H}_R$$

$$(f) \quad = -126,150 + (2,618486.52)$$

$$\tilde{Q}_{out} = 2,492336.52 \text{ kJ/kmol}$$

(g) Determining the dew point temperature involves finding the mole fraction of water in the products and subsequently determining the partial pressure of water in the products and then evaluating the saturation temperature of water corresponding to the partial pressure of water as follows:

$$y_{H_2O} = \frac{5}{64.88} = 0.0771$$

$$P_{H_2O} = 0.0771 \times 100 \text{ kPa} = 7.71 \text{ kPa}$$

$$t_{dp} = t_{sat} P_{H_2O} = 40.29 + \frac{0.21}{2.5}(5.52) = 40.8^\circ \text{C}$$



# **TUTORIAL**

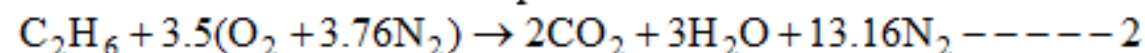
## **PROBLEMS ON REACTING MIXTURES AND COMBUSTIONS**

**6. Ethane gas ( $\text{C}_2\text{H}_6$ ) at  $25^\circ\text{C}$  is burned during a steady-flow combustion chamber at a rate of 5 kg/h with the stoichiometric amount of air, which is preheated to 500 K before entering the combustion chamber. An analysis of the combustion gases reveals that all the hydrogen in the fuels burns to  $\text{H}_2\text{O}$  but 95 % of the carbon in the fuel burns to  $\text{CO}_2$ , the remaining 5 % forming, CO. If the products leave the combustion chamber at 800 K, determine the rate of heat transfer from the combustion chamber, in kJ/h**

From Energy balance on the combustion chamber

$$\tilde{H}_R = \tilde{Q}_{out} + \tilde{H}_P \Rightarrow -\tilde{Q}_{out} = \tilde{H}_P - \tilde{H}_R \text{ ----1}$$

Stoichiometric combustion Equation is:



Granting that 95 % Carbon in the fuel is converted to CO<sub>2</sub> and the remaining 5 % forming CO, the actual combustion equation is written as:

$$\begin{aligned} -\tilde{Q}_{out} &= 1.9[-393,520 + 32,179 - 9,364] + 0.1[-110,530 + 23,844 - 8669] + 3[-241,820 + 27,896 + 9,904] \\ &\quad + 0.05[0 + 24523 - 8,682] + 13.16[0 + 23,714 - 8,669] - 1[-84,680 + \tilde{h}_{298K} - \tilde{h}_{298K}) \\ &\quad - 3.5[0 + 14,770 - 8,682] - 13.16[0 + 14,581 - 8,669) \end{aligned}$$

$$-\tilde{Q}_{out} = -1,201,005 \text{ kJ/kmol of } C_2H_6$$

$$-\dot{Q}_{out} = \dot{N}\tilde{Q}_{out} = \frac{\dot{m}}{\tilde{M}} \tilde{Q}_{out} = \left( \frac{5 \text{ kg/h}}{30 \text{ kg/kmol}} \right) (1,201,005) = 200,167.5 \text{ kJ/h}$$

$$-\dot{Q}_{out} = 200,170 \text{ kJ/h}$$

Substance	$\tilde{h}_f^\theta$	$\tilde{h}_{500K}$	$\tilde{h}_{298K}$	$\tilde{h}_{800K}$
C <sub>2</sub> H <sub>2</sub>	-84,680	---	---	---
O <sub>2</sub>	0	14,770	8,682	24,523
N <sub>2</sub>	0	14,581	8,669	23,714
H <sub>2</sub> O(g)	-241,820	---	9,904	27,896
CO <sub>2</sub>	-393,520	---	9,364	32,179
CO	-110,530	---	8,669	23,844

All values are in kJ/kmol

**TUTORIAL ENDS**

**THANK YOU**

**LUKE 19:13**