GATEFLIX

THERMODYNAMICS

For
MECHANICAL ENGINEERING

THERMODYNAMICS

SYLLABUS

Thermodynamic systems and processes; properties of pure substances, behaviour of ideal and real gases; zeroth and first laws of thermodynamics, calculation of work and heat in various processes; second law of thermodynamics; thermodynamic property charts and tables, availability and irreversibility; thermodynamic relations.
Power Engineering: Air and gas compressors; vapour and gas power cycles, concepts of regeneration and reheat.
I.C. Engines: Air-standard Otto, Diesel and dual cycles.
Refrigeration and air-conditioning: Vapour and gas refrigeration and heat pump cycles; properties of moist air, psychrometric chart, basic psychrometric processes.

ANALYSIS OF GATE PAPERS

Exam Year	1 Mark Ques.	2 Mark Ques.	Total
2003	5	6	17
2004	3	6	15
2005	2	7	16
2006	-	7	14
2007	2	6	14
2008	2	9	20
2009	3	4	11
2010	2	4	10
2011	2	7	16
2012	1	2	5
2013	1	4	9
2014 Set-1	2	4	10
2014 Set-2	2	2	6
2014 Set-3	1	3	7
2014 Set-4	-	4	8
2015 Set-1	4	3	10
2015 Set-2	3	4	11
2015 Set-3	2	3	8
2016 Set-1	3	3	9
2016 Set-2	3	2	7
2016 Set-3	1	2	5
2017 Set-1	1	4	9
2017 Set-2	2	4	10
2018 Set-1	2	3	8
2018 Set-2	2	4	10

CONTENTS

1 BASIC CONCEPTS OF THERMODYNAMICS

1.1 THERMODYNAMICS

Thermodynamics is the science of energy transfer and its effect on the physical properties of the substances.

Some students have difficulties with thermodynamics because of global nature of its applicability. Most students are used to courses that focus on a few specific topics like statics, dynamics, fluid flows etc. all deals with the limited range of topics. Thermodynamics, on the other hands, deals with many issues that are inherent in every engineering system. A thermodynamic analysis can span from analyzing a complete power plant to analyzing the smallest component in power plant.

We begin by introducing some basic thermodynamic terms and definitions. Some of these terms are already in our everyday vocabulary as a result of the broad result of thermodynamics concepts in non engineering concepts (for example cooling process of tea in the open environment is a thermodynamic process).

1.2 MICROSCOPIC APPROACH AND MACROSCOPIC APPROACH

In microscopic approach, a certain quantity of matter is considered with considering the event occurring at molecular level. It is called as **statistical Thermodynamics**. In Macroscopic approach, a certain quantity of matter is considered without considering the event occurring at molecular level. In this, average behaviour of molecules is considered. It is called as **Dynamic Thermodynamic**.

At the higher altitude or where the density of the system is low, the microscopic approach is used for checking the behaviour of the system.

1.3 SYSTEM

A quantity of matter or region in space upon which attention is concentrated is known as thermodynamic System. The system and the boundary are specified by the analyst, these are not specified in a problem statement.

1.3.1 SURROUNDING

Anything external to the system is called as surrounding or Environment. The combination of the system and surrounding makes universe. It means in the universe if anything is specified by the analyst as a system the things except that system are considered as the surrounding.

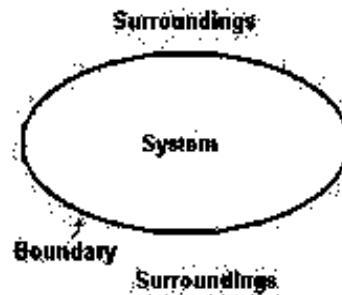**Fig.1.1system, surrounding and boundary**

1.3.2 BOUNDARY

The separation line which separates the system from surrounding is called as Boundary.

➢ Boundary may be fixed or rigid, may be real or imaginary.

1.3.3 UNIVERSE

The combination of system and surrounding is called as Universe.
Universe = system + Surrounding

1.3.4 TYPE OF SYSTEM

There are three types of thermodynamic systems.

1) Closed System:

The system in which only energy can transfer without transferring the mass across the system boundary is called as closed system. An example of closed system is mass of gas in the piston cylinder without valve.

Let us consider the piston cylinder arrangement as shown in fig 1.2. When heat is supplied to system, gas inside the cylinder expands and thermal energy is converted into mechanical work. But mass of the gas will be the same after the process in cylinder it means there is no mass which crosses the boundary of the system so it is considered as closed system.
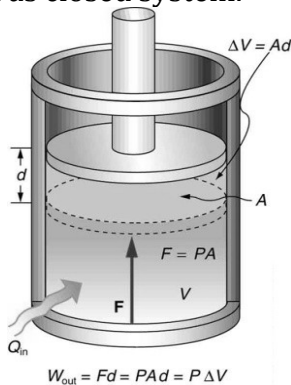
fig.:1.2 closed system

2) Open System:

The system in which energy and the mass can transfer across the system boundary is called as open system. Most of engineering devices are open systems. Examples: Turbine, Pump, Boiler, condenser Etc.
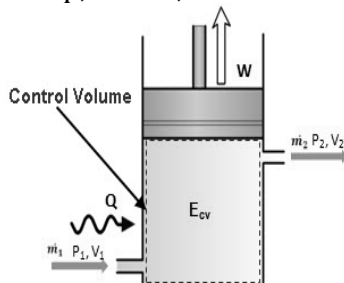
Fig. 1.3 open system

Let us consider the piston cylinder arrangement with inlet and outlet valve. When mass of a fluid enters in control volume of system, the mass and energy of the system change and then this mass of

fluid id transmitted from the outlet valve (for example the process of compression in compressor),the mass with energy leaves from the system it means in this system mass and energy both crosses the boundary so this is considered as the Open System.

3) Isolated System:

The system in which, neither energy nor mass can transfer across the system boundary is called as isolated system. Example: Universe, Insulated thermal flask. Let consider a proper insulated thermal flask and hot water is filled in the thermal flask after passing of the time mass and temperature are same which shows neither energy nor mass crosses the thermal boundary of the flask, so it is considered as an isolated system.

Fig. 1.4 isolated system

1.4 PROPERTY OF SYSTEM

A property of the system is a characteristic of system that depends upon the state of the system .as long as state is fixed, property of system is fixed. It does not depend how the state is reached. So properties depend upon end points only. These are point functions and exact differential.

Examples: Pressure, temperature, entropy etc. There are two types of properties:

(1) Extensive property:

These properties depend upon the mass of the system. If the system mass is changed then these types of properties will change.

For example, volume, internal energy, enthalpy, entropy etc.

2) Intensive Property:

These properties do not depend upon the mass of the system. For example, specific volume, specific internal energy, specific enthalpy, specific entropy, pressure, Temperature etc.

Most of the extensive properties can be converted in to intensive properties by dividing the extensive property by system mass or number of moles in the system. Intensive properties created in this way are called as specific property. It means all the specific properties are intensive property.

Let us we an example: we know that volume of the system is considered as the extensive property and its unit is m^3.

But Specific volume (v) = $\frac{V}{m}$. The specific volume is given by m^3/kg. It is given as intensive property.

1.5 STATE

It is the condition of a system at the instant of time. Each property has its single value at every state. As long state is fixed the property at that state is fixed. It means that the property does not depend the parameter other than the state or the specific point.

Let us consider a state 1 on P- V diagram as shown in fig 1.5. the properties at point 1 are P_1, V_1 . The state change from 1 to 2 and the properties at state 2 are P_2, V_2. then we follow the same points but change the path between the state 1 and state 2 .but the properties remains same. It means properties do not depends upon the path. it only depends upon the point, So result of this discussion is

➢ All properties are the point function.
➢ Differentiation of all the properties is exact differential (dT, dP etc.)
➢ Properties do not depend upon the past history. These only depends the present condition of the system

➢ Because in a cycle initial and the final point are same. So for the thermodynamic cycle, a thermodynamic property remains same.
➢ Difference of properties for reversible and irreversible process is same.

1.5 Cycle and path
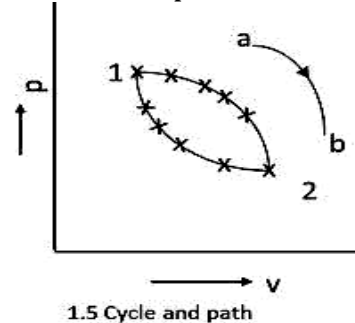
1.5.1 PROCESS

A process is said to be done by the system when it change its state. Process may be non flow process in mass does not transfer or it may be flow process in which mass and energy both will transfer. Type of Process: A Process may be reversible process or irreversible process.

1.5.1.1 REVERSIBLE PROCESS

The process is said to be reversible process if when it is reversed in direction it follows the same path as former path without leaving any effect on system and surrounding. Reversible process is most efficient process.

1.5.1.2 IRREVERSIBLE PROCESS

A process which does not follow the condition of the reversible process is called as the irreversible process. The friction is main reason for the process being the irreversible process.

1.5.1.3 QUASISTATIC PROCESS

Quasi means 'almost' static means 'slowness'. When process is carried out in very slow manner, it is called as Quasistatic process. Infinite slowness is characteristics of a Quasistatic process. Friction less

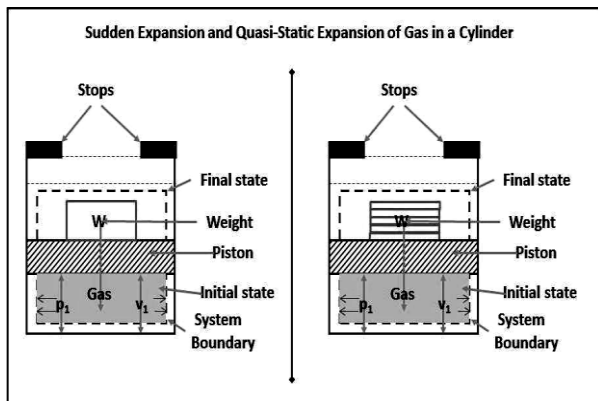quasistatic process is considered as reversible process.

Fig.1.6 quasistatic process

Let us consider the expansion of a gas from initial state 1 to final state 2 as shown in fig 1.6. In first case the weight W is release from the system and the gas expands quickly from state 1 to 2. But when weight W is divided into number of parts and then allow to release from the gas in second case. the gas passes through the number of equilibrium state and then reach the final state 2. So second process is very slow process it is known as quasistatic process. The P-V diagram for quasistatic process is shown in fig 1.7.
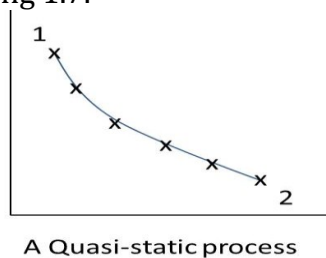
Fig. 1.7

1.5.2 CYCLE

A thermodynamic cycle is defined as a series of state changes such that final state is identical with initial state. for a cycle difference of all the properties is zero.

1.6 THERMODYNAMIC EQUILIBRIUM

A system is said to be in thermodynamic equilibrium when there is no change in the macroscopic property of the system.

A system is said to be in thermodynamic equilibrium when it follows:
i) Mechanical Equilibrium
ii) Thermal Equilibrium
iii) Chemical Equilibrium

i) Mechanical Equilibrium:

If there is no unbalanced force within the system and also between the system and surrounding, the system is said to be in mechanical equilibrium.

ii) Thermal Equilibrium:

if temperature of the system is same in all the parts of system, the system is said to be in Thermal equilibrium.

iii) Chemical Equilibrium:

if there is chemical reaction or transfer of matter from one part of system to another, the system is said to be in Chemical equilibrium.

1.7 ZEROTH LAW OF THERMODYNAMICS

It states that when body A is in thermal equilibrium with body B is in thermal equilibrium with body C , then body A,B and C will be in thermal equilibrium. It is the basic of temperature measurement.

fig. 1.8 Zeroth law of thermodynamics

If $T_A = T_B$ & $T_B = T_C$, then $T_A = T_B = T_C$.

1.7.1 APPLICATION OF ZEROTH LAW

The main application of Zeroth law of thermodynamics is to measure the temperature. First thermometric property is measure which change with respect to change of temperature.

Let us consider Mercury in glass thermometer (generally which is used to measure the temperature of human body) as shown in fig 1.9.

The length of the Hg change with respected to temperature. So the thermometric property in this type of thermometer is length.
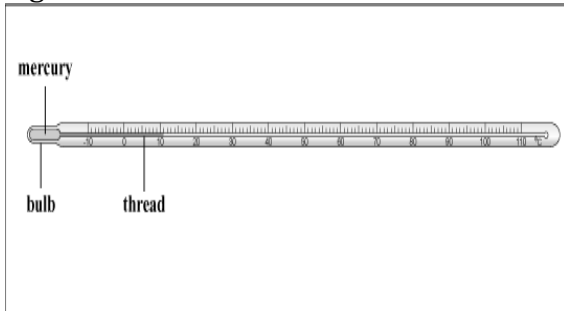
Fig. 1.9 Hg in glass thermometer

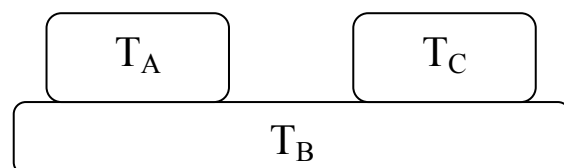It is more important to find the change of length so that we can calculate the temperature. So first length of the Hg which is change is found then temperature is calculated. In next topic we explain the process of measurement of the temperature.

1.7.2 TEMPERATURE MEASUREMENT

In the thermometer, Zeroth law of thermometer is used to measure the temperature of the body. Thermometric Property: The property which changes with change in temperature is called as thermometric property.

1.7.2.1 THERMOMETRIC PRINCIPLE

The thermometric property which change with the change of temperature is found first and with help of this thermometric property temperature of that specified state is calculated.

1.7.2.2 TYPE OF THERMOMETER

There are five types of thermometers.

a) Resistance thermometer

It works on Wheatstone bridge circuit. As temperature hang with respect to change of temperature resistance change .so thermometric Property in resistance thermometer is resistance. So to find the temperature of any state first resistance of that state is found and then temperature is measured.
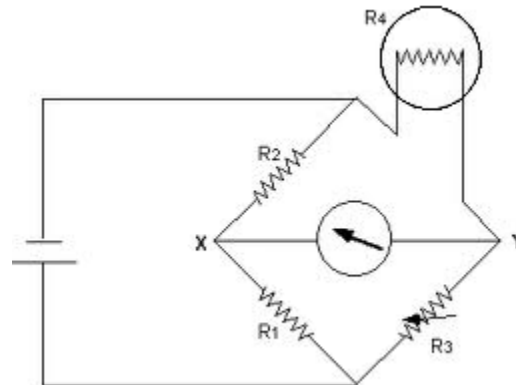
Fig. 1.10 Resistance Thermometer

(b) Thermocouple

This works on See beck Effect. When two dissimilar metal at different temperature is joint with each other, an electro magnetive force is generated the induced e.m.f. depends upon the temperature difference of two ends of the dissimilar metal. So thermometric property in this thermocouple is induced e.m.f.
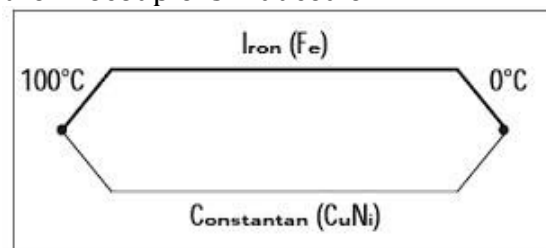
Fig. 1.11 thermocouple

c) Constant Volume gas thermometer

When the volume in constant volume gas thermometer is constant, the pressure changes with respect to temperature. In this type of thermometers, pressure is considered as the thermometric property.

d) Constant pressure gas thermometer

When the pressure in constant Pressure gas thermometer is constant, the volume

changes with respect to temperature. in this type of thermometers, volume is considered as the thermometric property.

e) Mercury –in- glass thermometer

In this type of thermometers, length change with change of the temperature so in mercury in glass thermometer, length is considered as the thermometric property.

1.7.3 METHOD FOR TEMPERATURE MEASUREMENT

1.7.3.1 METHOD FOR TEMPERATURE MEASUREMENT: BEFORE 1954

Before 1954 temperature measurement method, the measurement of temperature is based on two reference point freezing point and boiling point. Let temperature at freezing point is T_1 and thermometric property at that temperature is X_1 and Temperature at boiling point is T_2 and thermometric property at that temperature is X_2. Let thermometric Property at temperature T is X .then,

$$\frac{T_1}{T} = \frac{X_1}{X}$$ ------------- (1)

and Temperature at boiling point is T_2 and thermometric property at that temperature is X_2. Let thermometric Property at temperature T is X.then,

$$\frac{T_2}{T} = \frac{X_2}{X}$$ ------------- (2)

From equation (1) and (2)

$$\frac{T_1 - T_2}{T} = \frac{X_1 - X_2}{X}$$

$$T = \frac{T_1 - T_2}{X_1 - X_2}.X$$ --------- (3)

1.7.3.2 METHOD FOR TEMPERATURE MEASUREMENT: AFTER 1954

The measurement of temperature is based on single reference point triple point of water. Let temperature at triple point is T_i and thermometric property at that temperature is X_i. Let thermometric Property at temperature T is X. then

$T_i = a.X_i$ --------- (1)
and $T = a.X$ ---------(2)
Then,

$$T = 372.15\frac{X}{X_i}$$ ----------- (3)

1.7.3.3 TEMPERATURE SCALES

There are following temperature scales such as Rankin, Celsius, Kelvin and Fahrenheit. There are two common absolute temperature scales Rankin (R), and Kelvin scale (K). They are related as follows:

$$T(R) = \frac{9}{5}T(K)$$

Fig. 1.12 different temperature scale
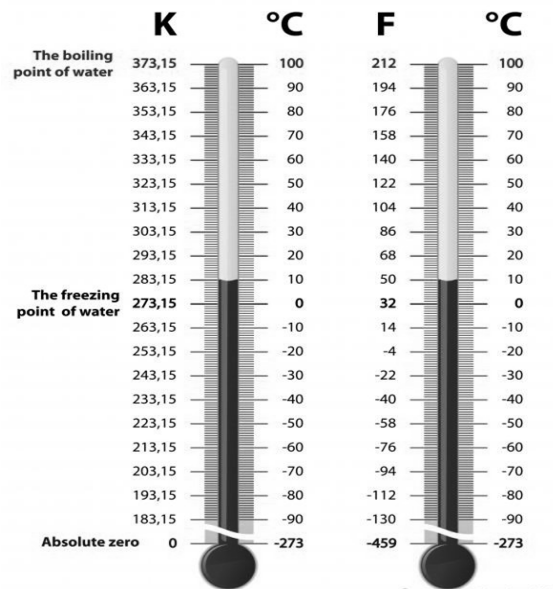
The relationship between the Celsius scale and Fahrenheit scale can be given as

$$T(^0F) = \frac{9}{5}T(^0C) + 32$$ --------- (2)

The relationship between the Celsius scale and Kelvin scale can be given as

$$T(K) = T(^0C) + 273.15$$ ------------- (3)

1.8 IDEAL GAS EQUATION AND PROCESS

The **ideal gas law** is the equation of state of a hypothetical ideal gas. It is a good

approximation to the behaviour of many gases under many conditions, although it has several limitations. The ideal gas law is often introduced in its common form:

$$PV = n\overline{R}T$$

Where P is the absolute pressure of the gas, V is the volume of the gas, n is number of moles of gas, $\overline{R}$ is universal gas constant, & T is the absolute temperature of the gas.
For air $\overline{R}$ = 8.314 KJ/mol.K
n (in moles) is equal to the mass m (in gm) divided by the molar mass M (in gm/mole)

$$n = \frac{m}{M}$$

By replacing n with m / M, we get:

$$PV = \frac{m}{M}\overline{R}T$$

Defining the specific gas constant R as the ratio $\overline{R}$/M

$$PV = mRT$$

Where R is specific Gas Constant.
For air R =0.287 KJ/Kg.K

Different process:

1) Constant volume process
An isochoric process, also called a constant-volume process, or an isometric process, is a thermodynamic process during which the volume of the system undergoing such a process remains constant.
For isocoric process, V = Constant

$$\frac{P_1}{T_1} = \frac{P_2}{T_2}$$
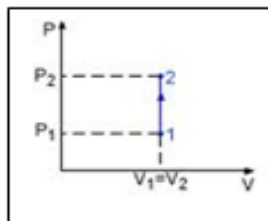
2) Constant pressure process
An isobaric process is a Thermodynamic process in which the pressure stays constant.
For isobaric process, P = Constant.

$$\frac{V_1}{T_1} = \frac{V_2}{T_2}$$
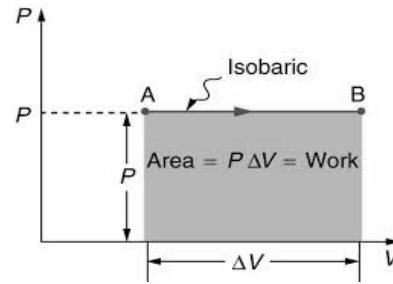
3) Isothermal Process:
An is other mal process is a change of a system, in which the temperature remains constant. It is also called as hyperbolic process. For Isothermal process, T = Constant or PV = Constant.
For the isothermal process,

$$P_1V_1 = P_2V_2$$
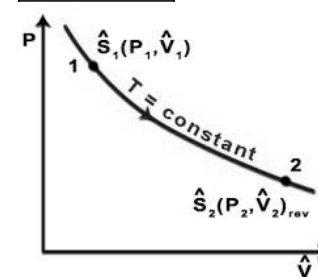
(4) Adiabatic Process
An adiabatic process is a process that occurs without the transfer of heat or matter between a system and its surroundings. Such processes are usually followed or preceded by events that do involve heat transfer.
For adiabatic process, PV^γ = Constant.
$P_1V_1{}^\gamma = P_2V_2{}^\gamma$

OR $\quad \dfrac{T_2}{T_1} = \left(\dfrac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$
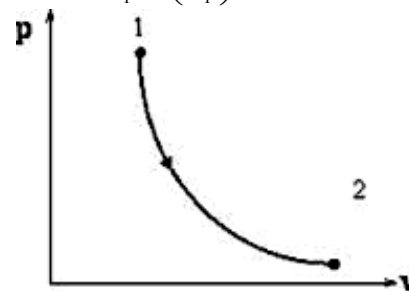
Adiabatic Process

OR, $\dfrac{T_2}{T_1} = \left(\dfrac{V_2}{V_1}\right)^{\gamma-1}$

5) Polytropic Process

A polytropic process is a thermodynamic process that obeys the relation:

$$pv^n = C$$

Where n is polytropic index.

$$P_1 V_1^n = P_2 V_2^n$$

OR, $\dfrac{T_2}{T_1} = \left(\dfrac{P_2}{P_1}\right)^{\frac{n-1}{n}}$
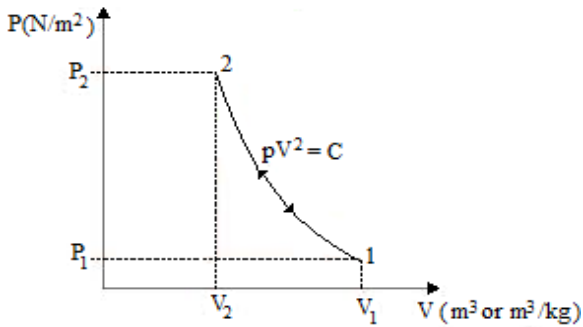
Polytropic Process

OR, $\dfrac{T_2}{T_1} = \left(\dfrac{V_2}{V_1}\right)^{n-1}$

Representation of different process on P-V diagram:

$$\boxed{PV^k = C}$$

If k=0, P= C, process is isobaric process.
If k = ∞, V =C, process is isocoric process.
If k=1 PV=C, process is isothermal process.
If k=n PV^n =C, process is polytropic process.
If k=γ PV^γ =C, process is adiabatic process.
Different processes on P-V diagram are shown in fig.
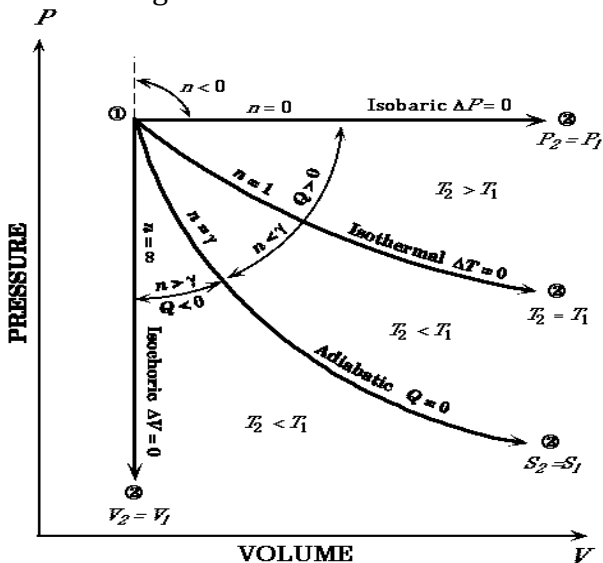
Slop of isothermal and adiabatic process on P-V diagram

➢ For isothermal process
PV = C
Taking log in both the sides
logP + logV = log C
Differentiating the equation

$$\frac{dP}{P} + \frac{dV}{V} = 0$$

$$\frac{dP}{P} = -\frac{dV}{V}$$

$$\boxed{\left(\frac{dP}{dV}\right) = -\frac{P}{V}}$$

➢ For adiabatic process
PV^γ = C
Taking log in both the sides
log P + γ logV = log C
Differentiating the equation

$$\frac{dP}{P} + \gamma \frac{dV}{V} = 0$$

$$\frac{dP}{P} = -\gamma \frac{dV}{V}$$

$$\boxed{\frac{\text{Slop of adiabatic process}}{\text{Slop of isothermal process}} = \gamma}$$

➢ The slop of adiabatic process is higher than the slop of isothermal process.

WORK AND HEAT TRANSFER

1.9 WORK TRANSFER
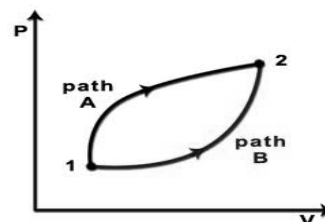
Work is said to be done by a system if the Sole effect on things external to system can

be reduced to rising of a Weight. The Weight may not be raised, but the net effect external to system would be the raising of a Weight.

Let consider the Battery and Motor.

fig. Electric work   fig. Mechanical Work
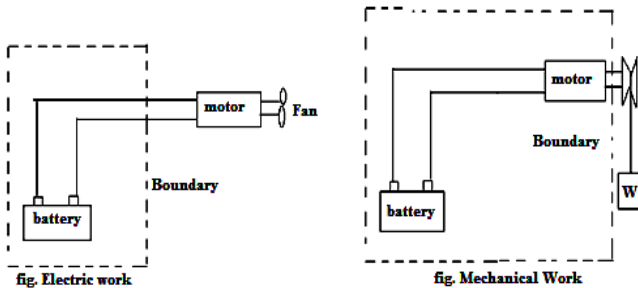
The nature of work changes when boundary between system and Surrounding changes. So work done is boundary phenomenon. It is transient form of energy.

W_{out}

Surrounding   System
W_{in}
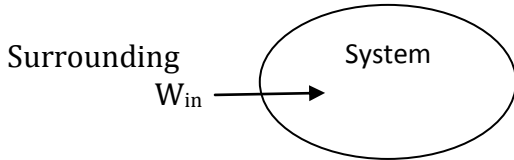
→ When Work is done on the system it is considered as negative.
→ When work is done by the system, it is considered as positive.

Let Consider a piston cylinder arrangement (i.e. closed).
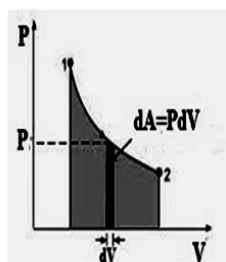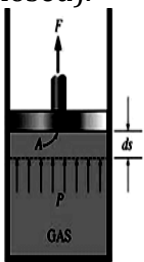
Work done by the system
$dW = F.\,dl = P.A.ds = P.dV$

$$W_{1-2} = \int_{1}^{2} P.dV \qquad \text{-------- (1)}$$

It is valid for closed system undergoing a reversible process.

So work done by closed system in reversible process is area under p−v diagram about V−axis.

If Path A and Path B are two paths but initial and final point is same. Area is different so work done for different path is different. So work transfer is path function. It is inexact differential.

1.9.1 WORK TRANSFER FOR CLOSED SYSTEM IN VARIOUS REVERSIBLE PROCESSES

A) Constant Pressure Process

In the isobaric process pressure remains constant.

So work transfer

$$W_{1-2} = \int_{1}^{2} p.dv = P(V_2 - V_1)$$
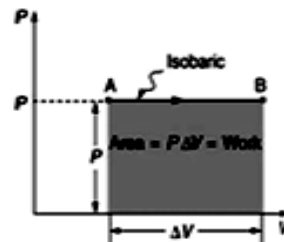
B) Constant Volume Process

In the isochoric process volume remains constant.

So work transfer in this process is zero.

V= constant

Work transfer $W_{1-2} = \int_{1}^{2} p.dv = 0$
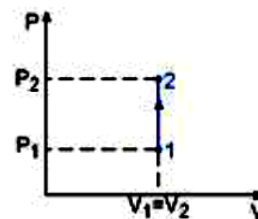
C) Isothermal Process

In isothermal process, T= C or

$$P.V = P_1 V_1 = P_2 V_2 = C$$

$$P = \frac{P_1 V_1}{V} = \frac{P_2 V_2}{V} \quad \text{---- (1)}$$

$$W_{1-2} = \int_{1}^{2} p.dv = P_1 V_1 = \int_{1}^{2} \frac{dv}{V}$$

$$= P_1 V_1 \ln\left(\frac{V_2}{V_1}\right)$$

$$\boxed{W_{1-2} == P_1 V_1 \ln\left(\frac{V_2}{V_1}\right) = m.RT \ln\left(\frac{V_2}{V_1}\right)}$$
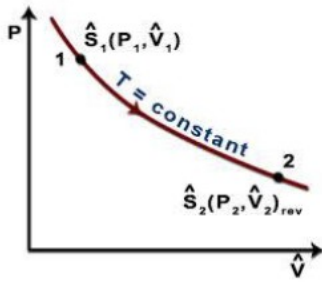
D) Adiabatic Process:

In adiabatic Process,
$$PV^{\gamma} = P_1 V_1^{\gamma} = P_2 V_2^{\gamma} = C$$

$$P = \frac{\left(P_1 V_1^{\gamma}\right)}{V^{\gamma}} = \frac{P_2 V_2^{\gamma}}{V^{\gamma}}$$

$$W_{1-2} = \int_1^2 p\,dv = \int_1^2 \frac{P_1 V_2^{\gamma}}{V^{\gamma}}$$

$$W_{1-2} = P_1 V_1^{\gamma} \left[\frac{V_2^{1-\gamma} - V_1^{1-\gamma}}{1-\gamma}\right]$$

$$= \frac{P_2 V_2^{\gamma}.V_2^{1-\gamma} - P_1 V_1^{\gamma}.V_1^{1-\gamma}}{1-\gamma}$$

$$\boxed{W_{1-2} = \frac{P_1 V_1 - P_2 V_2}{\gamma - 1} = \frac{MR(T_1 - T_2)}{\gamma - 1}}$$
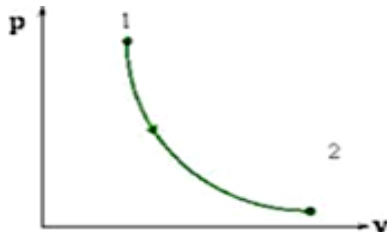
Adiabatic Process

(E) Polytrophic Process

$$\boxed{W_{1-2} = \frac{P_1 V_1 - P_2 V_2}{n-1} = \frac{MR(T_1 - T_2)}{n-1}}$$

Where n = Polytrophic index
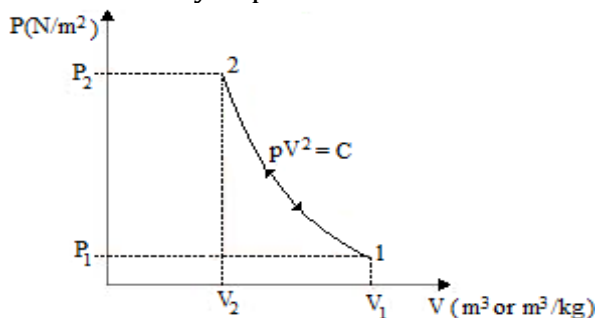
Polytropic Process

1.10 HEAT TRANSFER

Heat is defined as a form of energy that is transferred across the boundary by virtue of a temperature difference. The temperature difference is the potential and heat transfer is the flux. Heat flow into the system is taken as positive and heat flow out of system is taken as negative. The process in which that does not cross the boundary is adiabatic process. Heat transfer is a boundary phenomenon. Heat transfer is transient form of energy. Heat transfer is Path function.

For the m mass of substance and ΔT temperature difference, Heat transfer
$$Q \propto m.\Delta T$$
$$Q = m.C.\Delta T$$
Where m= mass of substance,
ΔT = temperature difference
C = Specific heat and its unit is KJ/Kg.K.
If m = 1Kg,
ΔT = 1 ^{0} C, then C = Q

1.10.1 SPECIFIC HEAT

The specific heat of a substance is defined as the amount of heat required to raise the unit mass of substance through a unit rise of temperature.

For the gases, in constant volume process it is taken as C_V and in constant pressure Process it is taken as C_p.

Examples

Q.1 If a gas of Volume 6000 cm³ and at a pressure of 100 KPa is compressed in reversible Process according to PV² = C until volume becomes 2000 cm³, determine the final pressure and work done ?

Solution:
Given V_1 = 6000 cm³ P_1 = 100 KPa, V_2 = 2000 cm³
P_2 =? , W_{1-2} = ?

$$P_1 V_1^2 = P_2 V_2^2$$
$$100 \times (6000)^2 = P_2 \times (2000)^2$$

$$P_2 = \frac{100 \times 6000^2}{2000^2}$$

$$P_2 = 900 \text{KPa}$$
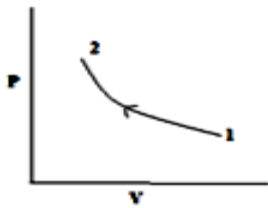
Work Transfer

$$W_{1-2} = \int_1^2 p.dv \qquad \text{----- (i)}$$

In the process $PV^2 = P_1V_1^2 = P_2V_2^2 = C$

$$P = \frac{C}{V^2} \qquad \text{----- (ii)}$$

Substituting the Value a/p in equation (1)

$$W_{1-2} = \int_1^2 \frac{C}{V^2} dv$$

$$= -C\left[\frac{1}{V_2} - \frac{1}{V_1}\right]$$

$$= -\left[\frac{P_2V_2^2}{V_2} - \frac{P_1V_1^2}{V_1}\right]$$

$$W_{1-2} = P_1V_1 - P_2V_2$$

$= 100 \times (6000) \times 10^{-6} - 900 \times (2000) \times 10^{-6}$

$$\boxed{W_{1-2} = -1.2\text{KJ}}$$

It means 1.2 KJ work is done on the system.

Q.2 Determine the total work done by a gas system, which follows the expansion Process as shown in Figure.
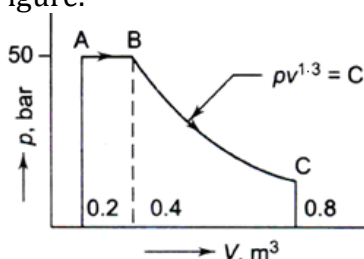
Solution:
Given $P_A = P_B = 50$ bar $= 50 \times 10^5$ Pa,
$V_A = 0.2 \text{ m}^3$, $V_B = 0.4 \text{ m}^3$, $V_C = 0.8 \text{ m}^3$,
$W_{A-C} = ?$

Work done in A-B process = Area under AB
$W_{A-B} = 50 \times 10^5 \times (0.4 - 0.2)$ J
$= 10^6$ J $= +1$ MJ
For the polytrophic process,
$$P_B V_B^n = P_B V_C^n$$
$50 \times 0.4^{1.3} = P_B 0.8^{1.3}$
$P_B = 8.25$ bar $= 20.3 \times 10^5$ Pa
Work done in B-C process = Area under BC on P-V diagram

$$W_{B-C} = \frac{P_B - V_B - P_B V_C}{n-1}$$

$$= \frac{(50 \times 10^5 \times 0.4) - (20.3 \times 10^5 \times 0.8)}{n-1}$$

$W_{B-C} = 1.25 \times 10^6$ J $= 1.25$MJ
So the total work done
$W = W_{A-B} + W_{B-C} = 1 + 1.25$ MJ
$= 2.25$ MJ

Q.3 The limiting value of the ratio of the pressure of gas at the steam point and at the triple point of water when the gas is kept at constant volume is found to be 1.36605. What is the ideal gas temperature of the steam point?

Solution:

$$\text{Given}: \frac{P}{P_t} = 1.36605$$

$$T = 273.16 \left(\frac{P}{P_t}\right) = 273.16 \times 1.36605$$

$= 373.15$ K
$T = 100^0$C

Q.4 A platinum wire is used as a resistance thermometer. The wire resistance was found to be 10 ohm and 16 ohm at ice point and steam point respectively, and 30 ohm at sulphur boiling point of 444.6°C. Find the resistance of the wire at 500°C, if the resistance varies with temperature by the relation.
$$R = R_0 (1 + \alpha t + \beta t^2)$$

Solution:

Given: $t_i = 0°C$, $R_i = 10$ ohm, $t_s = 100°C$, $R_s = 16$ ohm
For sulphur, $t_b = 444.6°C$, $R_b = 30$ ohm, $t = 500°C$, $R = ?$
$10 = R_0 (1 + 0 \times \alpha + \beta \times 0^2)$
$16 = R_0 (1 + 100 \times \alpha + \beta \times 100^2)$
$30 = R_0 (1 + 444.6 \times \alpha + \beta \times 444.6^2)$
Solving the equation, we get
$R = 10 (1 + 6.45 \times 10^{-3}t + 4.48 \times 10^{-6}t^2)$
Substituting the value of $t = 500°C$
$R = 10 (1 + 6.45 \times 10^{-3} \times 500 - 4.48 \times 10^{-6} \times 500^2)$
$R = 31.05$ ohm.

Q.5 The piston of an oil engine, of area 0.0045 m², moves downwards 75 mm, drawing. In 0.00028 m³ of fresh air from the atmosphere. The pressure in the cylinder is uniform during the process at 80 KPa, while the atmospheric pressure is 101.325 KPa, the difference being due to the flow resistance in the induction pipe and the inlet valve. Estimate the displacement work done by the air finally in the cylinder.

Solution:
Given: $A = 0.0045$ m², length of stroke = 75 mm = 0.075 m
Volume of piston stroke = 0.0045×0.075 m³
= 0.0003375 m³
$\therefore V_2 - V_1 = 0.0003375$ m³
As pressure is constant $P = 80$ KPa
So work done = $P \cdot (V_2 - V_1)$
= 80×0.0003375 KJ
= 0.027 KJ
= 27 J

Q.6 A mass of 1.5 kg of air is compressed in a quasi-static process from 0.1 MPa to 0.7 MPa for which PV = constant. The initial density of air is 1.16 kg/m³. Find the work done by the piston to compress the air.

Solution:
Given: m=1.5 kg, P_1=0.1 MPa, P_2 =0.7 MPa, ρ_1= 1.16 kg/m³, W=?
For quasi-static process, PV = C

$\therefore P_1 V_1 = P_2 V_2 = C$

$$V_1 = \frac{m_1}{\rho_1} = \frac{1.5}{1.16} = 1.29 \text{ m}^3$$

Work done

$$W_{1-2} = P_1 V_1 \ln\left(\frac{V_2}{V_1}\right) = P_1 V_1 \ln\left(\frac{P_1}{P_2}\right)$$

$$= 0.1 \times 1.29 \ln\left(\frac{0.1}{0.7}\right) = -2.51 \text{MJ}$$

Q.7 680 kg of fish at 5°C are to be frozen and stored at – 12°C. The specific heat of fish Above freezing point is 3.182 KJ/Kg.K, and below freezing point is 1.717 KJ/Kg K. The freezing point is – 2°C, and the latent heat of fusion is 234.5 KJ/Kg. How much heat must be removed to cool the fish, and what percent of this is latent heat?

Solution:
Given: m=680 kg, T_1=5°C, T_2 = -12°C, C_{p1} = 3.182 KJ/Kg.K,
C_{p2} = 1.717 KJ /Kg.K, T_f = -2°C LH = 234.5 KJ/Kg,
Heat to be removed above freezing point
= $680 \times 3.182 \times \{5 - (-2)\}$ kJ
= 15.146 MJ
Heat to be removed latent heat
= 680×234.5 kJ
= 159.460 MJ
Heat to be removed below freezing point
= $680 \times 1.717 \times \{-2 - (-12)\}$ kJ
= 11.676 MJ
$\therefore$ Total Heat = 186.2816 MJ
Percentage of Latent heat = (Latent heat/total heat) × 100
= (159.460/186.2816) × 100
= 85.6 %

2 FIRST LAW OF THERMODYNAMICS

2.1 FIRST LAW OF THERMODYNAMIC FOR A CYCLE

Heat and work are different forms of energy. Both form of energy can be Conserved. As the result of Joule's experiment, for a closed system undergoing a Cycle

$(\Sigma Q)_{cycle} = J (\Sigma W)_{cycle}$

J is Joule's equivalent. This is first law of thermodynamics for a closed system undergoing a cycle. Both Heat and work are measured in the desired unit of energy Joule (J). So constant of proportionality 'J' is unity.

2.2 APPLICATION OF FIRST LAW OF THERMODYNAMICS

2.2.1 HEAT TRANSFER IS A PATH FUNCTION

Let consider a cycle 1-A-2-B-1 and apply first law of thermodynamics for a cycle

$(\Sigma Q)_{cycle} = J (\Sigma W)_{cycle}$

$(dQ)_A + (dQ)_B = (\delta w)_A + (\delta w)_B$ ---- (I)

Let consider another cycle 1-A-2-B-1 and apply the first law of Thermodynamics again

$(dQ)_A + (dQ)_C = (\delta w)_A + (\delta w)_C$ ------ (II)

Solving the equation (I) or equation (II)

$(dQ)_B - (dQ)_C = (\delta w)_B + (\delta w)_C$ ------ (III)

Difference $(\delta w)_B - (\delta w)_C$ is not Zero because work transfer is a Path function as it was proved with help of chapter 1, so $(dQ)_B - (dQ)_C$ is not equal to Zero. So heat transfer for the different path has the different value. It depends upon the path. So results of this discussion are:

➤ Heat is a path function.
➤ The differentiation of the heat transfer is inexact differential (δQ).
➤ It is boundary phenomenon.
➤ it is transient form of energy.

2.2.2 ENERGY IS A PROPERTY OF SYSTEM

From the equation (III), We get

$(dQ)_B - (dQ)_C = (\delta w)_B + (\delta w)_C$ ------ (III)

Rearranging the equation (III)

$(\delta Q)_B + (\delta W)_B = (\delta Q)_C + (\delta w)_C$

$[(\delta Q) - (\delta W)]_B = [(\delta Q) - (\delta W)]_C$

For path B and C, the difference of heat transfer and work transfer is constant. So this different must be a point function and property of the system. This property is known as energy.

$\delta Q - \delta W = dE$ --------- (IV)

This is known as first law of thermodynamic for closed system undergoing a process. It is valid for the reversible and irreversible process.

Total Energy E = Kinetic Energy + Potential Energy + Internal Energy.

In absence of motion and gravity, kinetic & potential energy is Zero. In that case E = U

$\delta Q - \delta W = dU$ ----- (V)
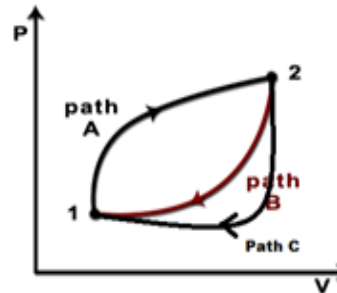
Where,

➤ U = Internal energy
➤ Unit of internal energy = KJ
➤ Internal energy is an extensive property.
➤ This is first Law of thermodynamics for reversible and irreversible process.
But we know that $\delta W = P.dV$ for a reversible process, so

$\delta Q = P.dV + dU$ ---- (VI)

➤ This is first Law of thermodynamics for a closed System undergoing a reversible Process.

2.2.3 ENERGY OF AN ISOLATED SYSTEM IS CONSTANT

According to I Law of thermodynamics
$\delta Q = \delta w + dE$
For isolated System $\delta Q = 0$ $\delta w = 0$
$dE = 0$
$E_1 = E_2$

➤ Energy of an isolated System is always constant.
➤ Universe is an isolated system so energy of the Universe remains Constant.
➤ It is known as law of conservation of energy.

2.2.4 PERPETUAL MOTION MACHINE (PMM-I) IS NOT POSSIBLE

Perpetual machine is the machine which delivers the work without taking any input. There is no such machine which would continuously supply mechanical work without some other form of energy dissipating. Such a machine is called as PMM-I. The efficiency of PMM-1 is given by

$$n_{PMM-I} = \frac{W}{Q} = \frac{W}{O}$$

$$n_{PMM-I} = \infty$$

It means efficiency of PMM-1 is Infinite.
But according to first law of thermodynamics
$(\Sigma Q)_{cycle} = (\Sigma W)_{cycle}$
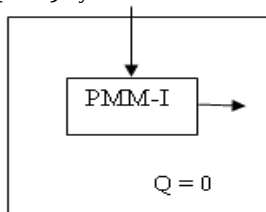
If it is possible it will violate of first law of thermodynamics but it is law of nature. So
➤ PMM-1 is not possible.
➤ The machine having the infinite efficiency is not possible.

2.2.5 ENTHALPY

Enthalpy of a Substance is defined as

$H = P.V + U$
Enthalpy is an extensive property and its unit is KJ.
Enthalpy of an ideal gas is given by
$H = m.C_p.T$

2.2.5.1 SPECIFIC ENTHALPY

It is amount of enthalpy per unit mass. Specific enthalpy is an intensive property. Its unit is KJ/Kg.

2.2.6 INTERNAL ENERGY

All fluids store energy. The store of energy within any fluid can be increased or decreased as the result of various processes carried out on or by the fluid. The energy stored within the fluid which results from the internal motion of its atom and molecules is called as Internal Energy.
The Internal energy of any ideal gas is given:
$U = m.C_v.T$
Internal Energy is an extensive property.
It is a point function.
Unit of internal energy is KJ.

2.2.6.1 SPECIFIC INTERNAL ENERGY

It is amount of internal energy per unit mass. Specific internal energy is an intensive property. It is denoted by u. Its unit is KJ/Kg.

2.3 MEYER'S FORMULA

We know that Enthalpy can be given as
$H = U + PV$
For an ideal gas, $H = m.C_p.T$, $U = m.C_v.T$ and $PV = mRT$
Substituting the values in equation
$H = m.C_v.T + mRT$
$m.C_p.T = mT (C_v+R)$

$$C_P - C \frac{CP}{CV} = \gamma, C_p = \frac{\gamma R}{\gamma - 1} v = R$$

This formula is known as Meyer's formula.

We know that $\frac{CP}{CV} = \gamma, C_p = \frac{\gamma R}{\gamma - 1}$

And $C_v = \dfrac{R}{\gamma - 1}$

2.4 HEAT TRANSFER IN DIFFERENT PROCESS

1) Constant Volume Process:
According to first law of thermodynamics
$\delta Q = P.dV + dU$
$\delta Q = 0 + dU$
$\delta Q = dU = mC_V dT$
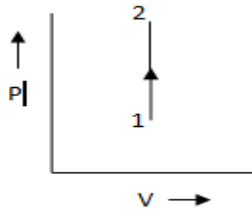
2) Constant Pressure Process:
According to first law of thermodynamics
$\delta Q = P.dV + dU$
We know that for isobaric Process
$dH = P.dV + dU$
$\delta Q = dH = m.C_p.dT$
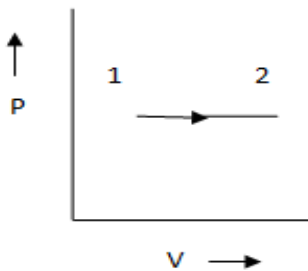
3) Isothermal Process:
According to first law of thermodynamics
$\delta Q = P.dV + dU$
For isothermal Process
$dU = 0$
$\delta Q = P.dV = \delta W$

$\delta Q = \delta W = P_1 V_1 l_n \left(\dfrac{V_2}{V_1}\right)$
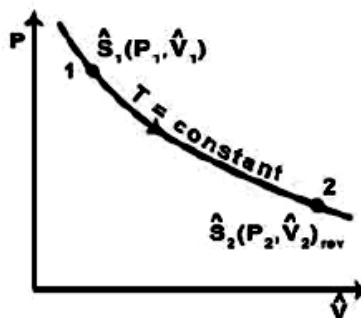
4) Adiabatic Process:

There is no heat transfer in adiabatic Process.

5) Polytrophic Process:
According to first law of thermodynamics
$\delta Q = \delta W + dU$

$= mC_v dT + \dfrac{mr(T_1 - T_2)}{n-1}$

$= -\dfrac{mR(T_1 - T_2)}{\gamma - 1} + \dfrac{mR(T_1 - T_2)}{n-1}$

$= mR(T_1 - T_2)\left\{\dfrac{1}{n-1} + \dfrac{1}{\gamma - 1}\right\}$

$= \dfrac{mR(T_1 - T_2)}{n-1}\left\{\dfrac{\gamma - n}{\gamma - 1}\right\}$

$$\boxed{\delta Q = \left\{\dfrac{\gamma - n}{\gamma - 1}\right\} \times \text{work done}}$$

Polytropic Process
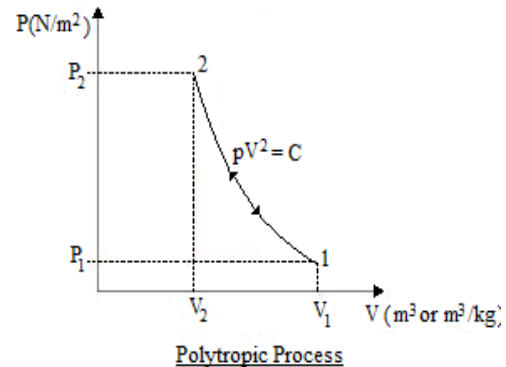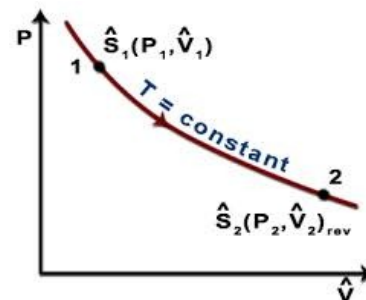
2.5 FREE EXPANSION

The expansion of gas against vacuum is known as free Expansion. There is no work done in free Expansion.

$\int\limits_{1}^{2} \delta W = 0$ although $\int\limits_{1}^{2} P.dv \neq 0$

Since work transfer $= \int_1^2 P.dv$ is considered for reversible Process but free expansion is highly irreversible Process. Vacuum does not offer any resistance, so there is no work transfer involved in free expansion.

2.6 FIRST LAW OF THERMODYNAMICS FOR FLOW PROCESS

Steady Flow Process

Steady flow means the rate of flow of mass and energy across the control volume are constant. At steady state of a System, any thermodynamic property will have a fixed value at the same state and does not change with respect to time.

Flow Work

The Flow work is the work done by Fluid of Mass m either at inlet or at the outlet of control Volume. Flow work at inlet is taken as $(-P_1V_1)$ and at outlet of control volume is $(+P_1V_1)$. where P_1, V_1 is pressure and volume at the inlet of control volume and P_2,V_2 is the pressure and volume at the outlet of control volume.

so total work $W_{total} = W_{CV} + P_2V_2 - P_1V_1$

2.6.1 STEADY FLOW ENERGY EQUATION

By Appling the energy balance equation at section 1-1 and section 2-2
(Energy at Section 1-1)= (Energy at Section 2-2)

Reference plane
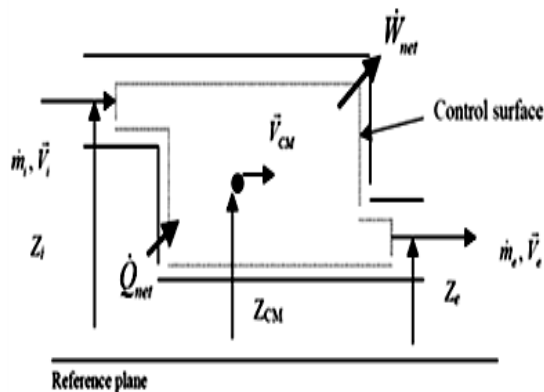
Steady state energy flow
[Internal Energy + Kinetic energy + Potential energy + Heat] $_{1-1}$
= [Internal Energy + Kinetic Energy + Potential Energy + Work] $_{2-2}$

$$U_1 + \frac{1}{2}mC_1^2 + mgZ_1 + \delta Q = U_2 + \frac{1}{2}mC_2^2 + mgZ_2 + \delta w_{Total}$$

$$U_1 + \frac{1}{2}mC_1^2 + mgZ_1 + \delta Q = U_2 +$$

$$\frac{1}{2}mC_2^2 + mgZ_2 + \delta w_{cv} - P_1V_1 + P_2V_2$$

$$U_1 + P_1V_1 + \frac{1}{2}mC_1^2 + mgZ_1 + \delta Q$$

$$= U_2 + P_2V_2 + \frac{1}{2}mC_2^2 + mgZ_2 + \delta w_{cv}$$

$$H_1 + \frac{1}{2}mC_1^2 + mgZ_1 + \delta Q$$

$$= H_2 + \frac{1}{2}mC_2^2 + mgZ_2 + \delta w_{cv} \text{ Or}$$

$$\boxed{\begin{aligned} H_1 + \frac{C_1^2}{2} + gZ_1 + \frac{\delta Q}{m} = \\ = H_2 + \frac{1}{2}C_2^2 + gZ_2 + \frac{\delta w_{cv}}{m} \end{aligned}}$$

It is known as steady Flow energy Equation (SFEE)
Where
h_1,h_2 = Specific enthalpy at inlet and outlet KJ/Kg
C_1,C_2 = Velocity at inlet and outlet m/Sec.
δw_{cv} = Work done by System KJ
δQ = Heat transfer KJ
Z_1, Z_2 = Elevation above on arbitrary datum , m

2.6.2 APPLICATION OF S.F.E.E.

1) Nozzle

For a turbine which is well insulated, and $Z_1 = Z_2$
SFEE equation:

$$h1 + \frac{C_1^2}{2} + gZ_1 + \frac{\delta Q}{m^o} = h_2 + \frac{C_2^2}{2} + gZ_2 + \frac{\delta W}{m^o}$$

Here $\dfrac{\delta Q}{m^o} = \dfrac{\delta W}{m^o} = 0$

$Z_1 = Z_2$

$h_1 + \dfrac{C_1^2}{2} = h_2 + \dfrac{C_2^2}{2}$

$C_2^2 - C_1^2 = 2(h_1 - h_2)$

If C_1 is very less as Compared to C_2, then

$C_2 = \sqrt{2(h_1 - h_2)}$ m/s.
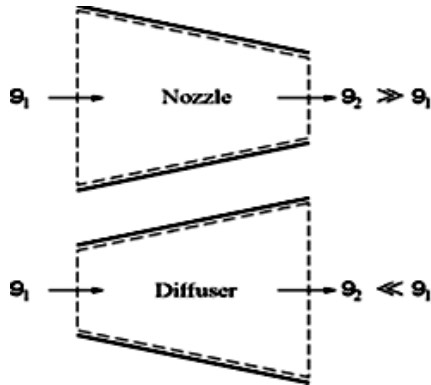
2) Turbine

For a turbine which is well insulated, the flow Velocities are often small, so the K.E can be neglected and $Z_1 = Z_2$

SFEE equation:

$h_1 + \dfrac{C_1^2}{2} + gZ_1 + \dfrac{\delta Q}{m^o} = h_2 + \dfrac{C_2^2}{2} + gZ_2 + \dfrac{\delta W}{m^o}$

$h_1 = h_2 + \dfrac{\delta W_{cv}}{m^o}$
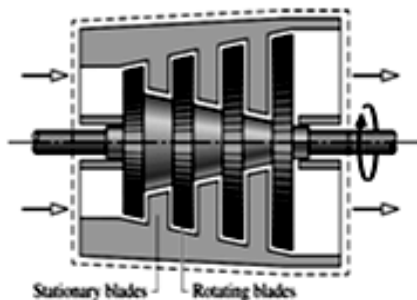
$\delta W = m^0(h_1 - h_2) KJ$

3) Compressor

For a turbine which is well insulated, the flow Velocities are often small, so the K.E can be neglected and $Z_1 = Z_2$

SFEE equation:

$h_1 + \dfrac{C_1^2}{2} + gZ_1 + \dfrac{\delta Q}{m^o} = h_2 + \dfrac{C_2^2}{2} + gZ_2 + \dfrac{\delta W}{m^o}$

$h_1 = h_2 + \dfrac{\delta W_{cv}}{m^o}$
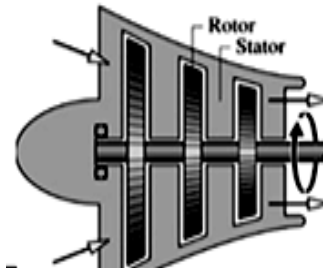
$\delta W_{cv} = m^0(h_1 - h_2) KJ$

But in Case of turbine h_1 is greater than h_2 and in Case of Compressor h_2 is greater than h_1.

4) Throttling Process

When the fluid flows through a constricted passage, like an opened valve or an orifice it is called as throttling Process. Throttling process is irreversible process.
In throttling Process,

$\dfrac{\delta Q}{m} = \dfrac{\delta W}{m} = 0$

And change of P.E. P.K.E. is very small and ignored, then

$h_1 + \dfrac{C_1^2}{2} + gZ_1 + \dfrac{\delta Q}{m} = h_2 + \dfrac{C_2^2}{2} + gZ_2 + \dfrac{\delta W}{m}$

$h_1 = h_2$

So Enthalpy of fluid before throttling and after throttling remains Constant.
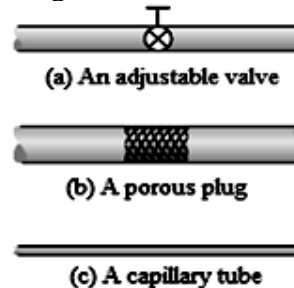
2.7 FIRST LAW OF THERMODYNAMICS FOR VARIABLE FLOW PROCESS

Variable Flow

If there is a change of property at the same state, Process is called as Variable flow or unsteady Flow Process. In the unsteady flow then there is an accumulation of mass and energy.

The mass Accumulated in control volume is given by

$$\left(\frac{dm}{dt}\right)_{cv} = \frac{dm_1}{dt} - \frac{dm_2}{dt}$$

Applying the law of conservation of energy
Rate of energy accumulation = Rate of energy inlet – Rate of energy outlet

$$\left(\frac{dE}{dt}\right)cv = \frac{d}{dt}\left[m_1^0 h_1 + \frac{1}{2}m_1 c_1^2 + m_1 g Z_1 + \delta Q\right]$$

$$-\frac{d}{dt}\left[m_2 h_2 + \frac{1}{2}m_2 c_2^2 + m_2 Z_2 + \delta W\right]$$

Supply line

Control volume

CV boundary
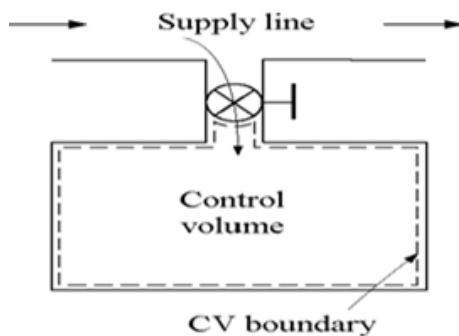
Unsteady flow

$$\left(\frac{dE}{dt}\right)cv = \left[m_1^0 h_1 + \frac{1}{2}m_1^0 c_1^2 + m_1^0 g 2_1 + \frac{\delta Q}{\delta t}\right] -$$

$$\left[m_e^0 h_2 + \frac{1}{2}m_e^0 c_2^2 + m_e^0 g 2_2 + \frac{\delta w}{\delta t}\right]$$

Neglecting the Kinetic energy and potential energy

$$\left(\frac{dU}{dt}\right)cv = m_1^0 h_1 - m_e^0 h_2 + \delta Q^0 - \delta W^0$$

Where
m_1^0 = Mass Flow rate at inlet Kg/Sec

m_e^0 = Mass Flow tare at outlet Kg/Sec

δQ^0 = Heat transfer Rate KJ/Sec

δW^0 = Work done KJ/Sec

h_1, h_2 = Specific Enthalpy at Inlet or Outlet.

Examples

Q.1 A slow chemical reaction takes place in a fluid at the constant pressure of 0.1 MPa. The fluid is surrounded by a perfect heat insulator during the reaction which begins at state 1 and ends at state 2. The insulation is then removed and 105 kJ of heat flow to the surroundings as the fluid goes to state 3. The following data are observed for the fluid at states 1, 2 and 3.

State	v (m3)	t (°C)
1	0.003	20
2	0.3	370
3	0.06	20

For the fluid system, calculate E2 and E3, if E1 = 0

Solution:
From first law of thermodynamics
$\delta Q = dE + PdV$
$Q_{1\text{-}2} = (E_2 - E_1) + \int_1^2 PdV$
$= (E_2 - E_1) + 0.1 \times 10^3 (0.3 - 0.003)$
Because Process 1-2 is insulated so
$Q_{1\text{-}2} = 0$ and E1 = 0
So $E_2 = -29.7$ KJ
$Q_{2\text{-}3} = (E_3 - E_2) + \int_2^3 PdV$
$-105 = (E_3 + 29.7) + 0.1 \times 10^3 (0.06 - 0.3)$
$E_3 = -110.7$ KJ

Q.2 A domestic refrigerator is loaded with food and the door closed. During a certain period the machine consumes 1 kWh of energy and the internal energy of the system drops by 5000 kJ. Find the net heat transfer for the system.

Solution:
According to first law of thermodynamics
$\delta Q - \delta W = dE$
$\delta Q - \delta W = dE$
$\delta W = -1 \times 3600 = -3600$ KJ.
$dE = -5000$ KJ
$\delta Q + 3600 = -5000$
$\delta Q = -8600$ KJ $= -8.6$ MJ

Q.3 The properties of a certain fluid are related as follows:
$u = 196 + 0.718t$
$pv = 0.287 (t + 273)$
Where u is the specific internal energy (kJ/kg), t is in °C, p is pressure (KN/m^2), and v is specific volume (m^3/kg). For this fluid, find Cv and Cp.

Solution:

we know that $C_p = \left(\dfrac{dh}{dT}\right)_P$

$= \left(\dfrac{d(u+pu)}{dT}\right)_P$

$= \left(\dfrac{d(196+0.718t+0.287(t+273))}{dT}\right)_P$

$= \left(\dfrac{0+0.718dt+0.287(dt+0)}{dT}\right)_P$

$= \left(\dfrac{0.718dt+0.287dt}{dT}\right)_P$

$= \left(\dfrac{1.005dt}{dT}\right)_P$

but $T = t + 273$ so $dT = dt$

$C_p = 1.005 KJ/Kg.K$

again $C_v = \left(\dfrac{du}{dT}\right)_V$

$= \left(\dfrac{d(196+0.718t)}{dT}\right)_V$

$= \left(\dfrac{0.718dt}{dT}\right)_V$

but $T = t + 273$ so $dT = dt$

$C_v = 0.718 KJ/Kg.K$

Q.4 A mass of 8 kg gas expands within a flexible container so that the P-V relationship is $PV^{1.2}$ = constant. The initial pressure is 1000 KPa and the initial volume is 1 m³. The final pressure is 5 KPa. If specific internal energy of the gas decreases by 40 KJ/kg, find the heat transfer in magnitude and direction.

Solution:

For the polytrophic process

$P_1V_1^n = P_2V_2^n$

$1000 \times 1^{1.2} = 5 \times V_2^{1.2}$

$V_2 = 82.7$ m³

work done $W_{1-2} = \dfrac{P_1V_1 - P_2V_2}{n-1}$

$= \dfrac{1000 \times 1 - 5 \times 82.7}{1.2 - 1}$

$W_{1-2} = 2932.5 KJ$

$dE = -8 \times 40 = -320$ KJ

$Q = dE + W_{1-2} = -320 + 2932.5$

$Q = 2612.5 KJ$

2612.5 KJ heat is supplied to the system.

Q.5 The heat capacity at constant pressure of a certain system is a function of temperature only and may be expressed as

$C_p = 2.093 + \dfrac{41.87}{t+100} J/^0C$

Where t is the temperature of the system in °C. The system is heated while it is maintained at a pressure of 1 atmosphere until its volume increases from 2000 cm³ to 2400 cm3 and its temperature increases from 0°C to 100°C.

a) Find the magnitude of heat interaction.
b) How much does the internal energy of the system increase?

Solution:

If T is temperature in K, then t = T - 273 and t + 100 = T − 173

$Q_{1-2} = \displaystyle\int_{273}^{373} \left(2.093 + \dfrac{41.87}{T-173}\right)dT$

$= \left[2.093T + \ln\dfrac{418.7}{T-173}\right]_{273}^{373}$

$= 2.093(373 - 273) + \ln\dfrac{200}{100}$

$= 209.3 + 41.87\ln2 = 238.32$ J

$Q_{1-2} = (E_2 - E_1) + \displaystyle\int_1^2 PdV$

$E_2 - E_1 = Q_{1-2} - P.(V_2 - V_1)$

$= 238.32 - 101.325(0.0024\ 0.0020) \times 1000J$

$= 197.79$ J

Q.6 a gas undergoes a thermodynamic cycle consisting of three processes beginning at an initial state where P_1 = 1 bar, V_1 = 1.5 m³ and U_1 = 512 kJ. The processes are as follows:

i) Process 1–2: Compression with PV = constant to P_2 = 2 bar, U_2 = 690 kJ

ii) Process 2–3: W_{23} = 0, Q_{23} = –150 kJ, and

iii) Process 3–1: W_{31} = +50 kJ. Neglecting KE and PE changes, determine the heat interactions Q_{12} and Q_{31}.

Solution:

Process (1-2): process is isothermal process. for isothermal process,

Q_{1-2} = $(U_2-U_1) + P_1 V_1 l_n \left(\frac{V_2}{V_1}\right)$

= $(690 – 512) + 1 \times 10^5 \times 1.5 l_n \left(\frac{P_1}{P_2}\right)$

= $(690 – 512) + 1 \times 10^2 \times 1.5 \, l_n \left(\frac{1}{2}\right)$

= 178 – 103.972

= 74.03 KJ

As W_{2-3} is ZERO so it is constant volume process. As W_{31} is positive so expansion is done.

For the process (2-3)

Q_{2-3} = $W_{2-3} + (U_3-U_2)$

–150 = 0 + $(U_3 - 690)$

U_3 = 540 KJ

Process (3-1)

Q_{3-1}=$W_{3-1}+ (U_1-U_3)$ = 50+(512-540)

= 50 -28 = 22 KJ

Q.7 A blower handles 1 kg/s of air at 20°C and consumes a power of 15 kW. The inlet and outlet velocities of air are 100 m/s and 150 m/s respectively. Find the exit air temperature, assuming adiabatic conditions. Take Cp of air is 1.005 KJ/Kg-K.

Solution:

From S.F.E.E.

$mh_1 + m\dfrac{c_1^2}{2} + mgZ_1 + \delta Q / dt$

$= mh_2 + \dfrac{1}{2}mC_2^2 + mgZ_2 + \delta W / dt$

$1 \times h_1 + 1 \times (100)^2/2000 + 1 \times 9.8 \times Z + 0 = 1 \times h_2 + 1 \times (200)^2/2000 + 1 \times 9.8 \times Z + 15$

$h_2 – h_1 = 8.75$

$C_p (T_2-T_1) = 8.75$

$T_2 = 20 + 8.7$

= 28.7 °C

Q.8 A turbine operates under steady flow conditions, receiving steam at the following state: Pressure 1.2 MPa, temperature 188°C, enthalpy 2785 kJ/kg, velocity 33.3 m/s and elevation 3 m. The steam leaves the turbine at the following state: Pressure 20 KPa, enthalpy 2512 kJ/kg, velocity 100 m/s, and elevation 0 m. Heat is lost to the surroundings at the rate of 0.29 KJ/s. If the rate of steam flow through the turbine is 0.42 kg/s, what is the power output of the turbine in kW?

Solution:

P_1 = 1.2 MPa t_1 = 188°C h_1 = 2785 kJ/kg C_1 = 33.3 m/s Z_1= 3 m P_2 = 20 kPa

h_2 = 2512 kJ/kg C_2 = 100 m/s

dQ/dt = – 0.29 kJ/s dW/dt =?
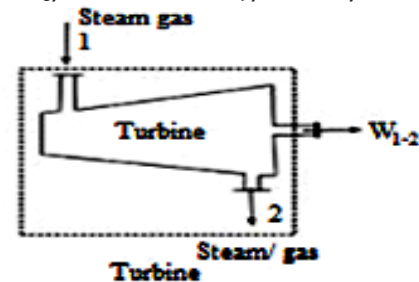
By SFEE

$m(h_1 + \dfrac{C_1^2}{2000} + \dfrac{gZ_1}{1000}) + \dfrac{\delta Q}{dt} =$

$m(h_2 + \dfrac{C_2^2}{2000} + \dfrac{gZ_2}{1000}) + \dfrac{\delta W}{dt}$

$0.42(2785 + \dfrac{33.3^2}{2000} + \dfrac{9.8 \times 3}{1000}) - 0.29 =$

$0.42(2512 + \dfrac{100^2}{2000} + \dfrac{9.8 \times 0}{1000}) + \dfrac{\delta W}{dt}$

$1169.655 = 1057.14 + \dfrac{\delta W}{dt}$

$\dfrac{\delta W}{dt} = 112.515 kW$

Q.9 A nozzle is a device for increasing the velocity of a steadily flowing stream. At the inlet to a certain nozzle, the enthalpy of the fluid passing is 3000 kJ/kg and the

velocity is 60 m/s. At the discharge end, the enthalpy is 2762 KJ/kg. The nozzle is horizontal and there is negligible heat loss from it.

a) Find the velocity at exists from the nozzle.
b) If the inlet area is 0.1 m2 and the specific volume at inlet is 0.187 m3/Kg, find the mass flow rate.
c) If the specific volume at the nozzle exit is 0.498 m3/Kg, find the exit area of the nozzle.

Solution:

Velocity at exit from S.F.E.E.

$$m(h_1 + \frac{C_1^2}{1000} + \frac{gZ_1}{1000}) + \frac{\delta Q}{dt} =$$

$$m(h_2 + \frac{C_2^2}{2000} + \frac{gZ_2}{1000}) + \frac{\delta W}{dt}$$
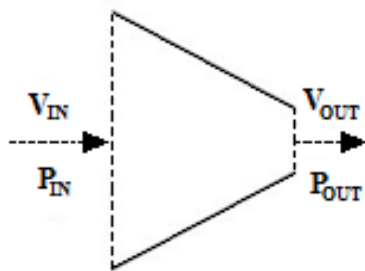
h1 = 3000 kJ/kg C1 = 60 m/s h2 = 2762 kJ/kg

$$\frac{\delta W}{dt} = \frac{\delta Q}{dt} = 0 \quad Z_1 = Z_2$$

$$\left(h_1 + \frac{c_1^2}{1000}\right) + 0 = \left(h_2 + \frac{c_1^2}{1000}\right) + 0$$

$$(3000 + \frac{60^2}{1000}) + 0 = (2762 + \frac{c_2^2}{1000}) + 0$$

C2² = 60²/ 2000 +3000 - 2762
C2 = 692.532 m/ s

Mass flow rate (m) $= \frac{A_1 C_1}{v_1} = \frac{0.1 \times 60}{0.187}$

= 32.1 Kg/sec

Mass flow rate (m) $= \frac{A_2 C_2}{v_2}$

$$32.1 = \frac{A_2 \times 692.532}{0.498}$$

A2 = 8.02 m²

Q.10 A gas flows steadily through a rotary compressor. The gas enters the compressor at a temperature of 16°C, a pressure of 100 kPa, and an enthalpy of 391.2kJ/ kg. The gas leaves the compressor at a temperature of 245°C, a pressure of 0.6 MPa, and an enthalpy of 534.5kJ/ kg. There is no heat transfer to or from the gas as it flows through the compressor.

a) Evaluate the external work done per unit mass of gas assuming the gas velocities at entry and exit to be negligible.
b) Evaluate the external work done per unit mass of gas when the gas velocity at entry is 80 m/s and that at exit is 160 m/s.

Solution :

t₁ = 16°C, P₁ = 100 kPa, h₁ = 391.2 kJ/kg,t2=245°C P2=0.6 MPa=600 kPa h2 = 534.5 kJ/kg

For V1 and V2 negligible and Z1= Z2
So SFEE

$$m(h_1 + 0) + 0 = m(h_2 + 0) + \delta W$$

$$\delta W = 1 \times (h_1 - h_2) = (391.2 - 5345)kJ / kg$$

= –143.3 kJ/kg

C₁ = 80 m/s; C₂ = 160 m/s
Then SFEE

$$m(h_1 + \frac{c_1^2}{1000} + 0) + 0 =$$

$$m(h_2 + \frac{c_2^2}{1000} + 0) + \delta W$$

$$1 \times \left(391.2 + \frac{80^2}{1000} + 0\right)$$

$$= 1 \times \left(534.5 + \frac{160^2}{1000} + 0\right) + \delta W$$

δW= (−143.3 − 9.6) kJ/kg
= −152.9 kJ/kg

Q.11 Air flows steadily at the rate of 0.4 kg/s through an air compressor, entering at 6 m/s with a pressure of 1 bar and a specific volume of 0.85 m³/kg, and leaving at 4.5 m/s with a pressure of 6.9 bar and a specific volume of 0.16 m³/kg. The internal energy of the air leaving is

88 KJ/kg greater than that of the air entering. Cooling water in a jacket surrounding the cylinder absorbs heat from the air at the rate of 59 W. Calculate the power required to drive the compressor and the inlet and outlet cross-sectional areas.

Solution :

$m_1 = 0.4$ kg/s, $C_1 = 6$ m/s $P_1 = 1$ bar = 100 KPa $v_1 = 0.85$ m³ /kg $m_2 = 0.4$ kg/s $C_2 = 4.5$ m/s

$P_2 = 6.9$ bar = 690 kPa $v_2 = 0.16$ m³ /kg $u_2 = u_1 + 88$ kJ/kg

$$A_2 = \frac{m.V_2}{V_2} = \frac{0.4 \times 0.16}{4.5} = 0.01422 \text{ m}^2$$
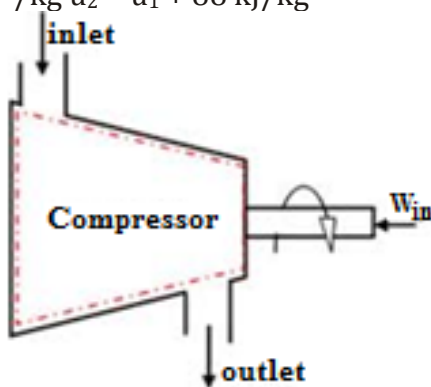
by SFEE

$$m(h_1 + \frac{c_1^2}{1000} + 0) + \frac{\delta Q}{dt}$$

$$= m(h_2 + \frac{c_2^2}{2000} + 0) + \frac{\delta W}{dt}$$

$$m(u_1 + P_1 v_1 + \frac{c_1^2}{1000}) + \frac{\delta Q}{dt}$$

$$= m(u_2 + P_2 v_2 + \frac{c_2^2}{2000}) + \frac{\delta W}{dt}$$

$$\frac{\delta W}{dt} = m$$

$$u_1 - u_2 + P_1 v_1 - P_2 v_2 + \frac{c_1^2 - c_2^2}{2000}) - \frac{\delta Q}{dt}$$

$$= 0.4(-88 + 85 + 110.4 - 0.0076) - 0.059$$

$$= -45.357 - 0.059$$

$$= -45.416 \text{ kJ}$$

Mass flow rate $(m) = \dfrac{A_1 C_1}{V_1}$

$$A_1 = \frac{m.v_1}{c_1} = \frac{0.4 \times 0.85}{6} = 0.0567 \text{m}^2$$

Mass flow rate $(m) = \dfrac{A_2 C_2}{V_2}$

3 SECOND LAW OF THERMODYNAMICS, ENTROPY

The first law of thermodynamics states that certain energy is balanced when system undergoes a thermodynamic process. But it does not give any idea whether a particular process is in feasible condition or not. It does not predict the direction of process. It is second law of thermodynamics which predict whether process is in feasible or not with the concept of entropy.

3.1 THERMAL RESERVOIR

A thermal energy reservoir is defined as a large body of infinite heat capacity which is capable absorbing or rejecting the heat without change of temperature.
Source: The thermal energy reservoir from which heat can be absorbed without change of temperature is known as Source.
Sink: The thermal energy reservoir in which heat can be rejected without change of temperature is known as Sink.
There are two statements of second law of thermodynamics

3.1.1 KELVIN – PLANK STATEMENT OF SECOND LAW

It is impossible for any device that operates on a cycle to receive heat from a single reservoir and produce a net amount of work. In other words, no heat engine can have a thermal efficiency of 100%. The machine having 100% efficiency is known as PMM-II. So PMM-II is not possible.
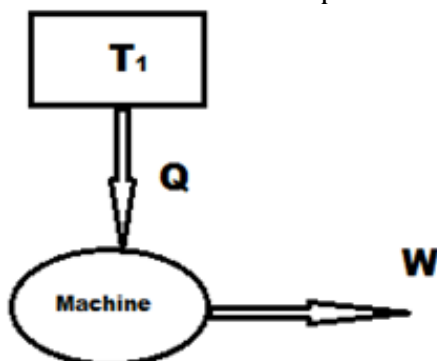
A heat engine that violates the Kelvin-Planck statement of the second law cannot be built.

3.1.2 CLAUSIUS STATEMENT OF SECOND LAW

It is impossible to construct a device that operates in a cycle and produces transfer of heat from a lower-temperature body to higher-temperature body without any external work. If such a device is possible its C.O.P. will be infinite. So Refrigerator having COP infinite is not possible.
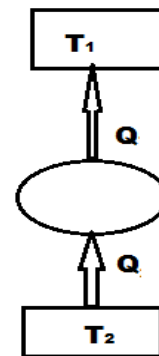
A refrigerator that violates the Clausius statement of the second law cannot be built.

3.1.3 HEAT ENGINE

Heat engines convert heat to work. Heat engine is a thermodynamic system operating in a thermodynamic cycle to which heat is transfer and it is converted in to work.
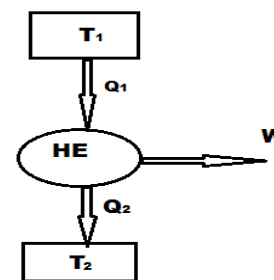
Fig.: Heat Engine

3.1.4 THERMAL EFFICIENCY

Thermal efficiency is the index of performance of a heat engine. It is the fraction of the heat input that is converted to the network.

Work output $W_{out} = Q_1 - Q_2$

Efficiency

$$\eta = \frac{W}{Q_1} = \frac{Q_1 - Q_2}{Q_1} = 1 - \frac{Q_2}{Q_1} \quad \text{-------(3.1)}$$

3.1.5 REFRIGERATOR AND HEAT PUMP

The transfer of heat from a low temperature region to a high-temperature one requires special devices called refrigerators. Refrigerators are cyclic devices, and the working fluids used in the cycles are called refrigerant.
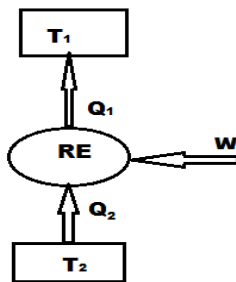
Fig: Refrigerator

3.1.6 CO-EFFICIENT OF PERFORMANCE

The index of performance of a refrigerator is given in the term of co-efficient of performance (COP).It is defined as the ratio of desired effect to given input.

For the refrigerator

Desired effect = Heat rejected from low temperature body (Q_2)

Work done on the refrigerator $W = Q_1 - Q_2$

So

$$(COP)_R = \frac{\text{Desised Effect}}{\text{work input}} = \frac{Q_2}{W}$$

$$(COP)_R = \frac{Q_2}{Q_1 - Q_2}$$

3.1.7 HEAT PUMP

Heat pump is a thermodynamic system operating in a cycle that removes heat from low temperature body and delivers heat to high temperature body. To accomplish this energy transfer Heat pumps receive the external energy in the form of work or heat.
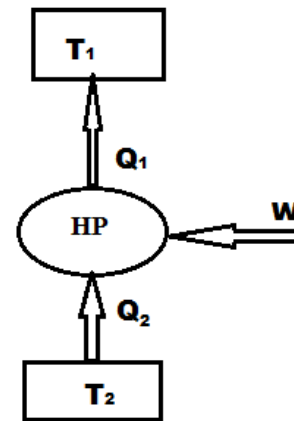
Fig: Heat Pump

3.1.8 CO-EFFICIENT OF PERFORMANCE

The index of performance of a heat pump is given in the term of co-efficient of performance (COP).It is defined as the ratio of desired effect to given input. For the Heat pump desired effect = Heat supplied to high temperature body (Q_1)

Work done on the refrigerator $W = Q_1 - Q_2$

So

$$(COP)_R = \frac{\text{Desired Effect}}{\text{workinput}} = \frac{Q_1}{W}$$

$$(COP)_{HP} = \frac{Q_1}{Q_1 - Q_2} \quad \text{-------(3.3)}$$

Subtracting the equation (3.2) from equation (3.3)

$$(COP)_{HP} - (COP)_R = \frac{Q_1 - Q_2}{Q_1 - Q_2} = 1$$

$$(COP)_{HP} = (COP)_R + 1 \quad \text{------(3.4)}$$

From the equation (3.1) and equation (3.3)

$$(COP)_{HP} = \frac{1}{\eta} \quad \text{---------(3.5)}$$

Refrigerators and heat pumps are essentially the same devices; they differ in their Objectives only. Refrigerator is to maintain the refrigerated space at a low temperature. On the other hand, a heat pump absorbs heat from a low-temperature source and supplies the heat to a warmer medium.

3.2 EQUIVALENCE OF KELVIN PLANK AND CLAUSIUS STATEMENT

The two statements of the second law are equivalent. In other words, any device violates the Kelvin-Planck statement also violates the Clausius statement and vice versa.
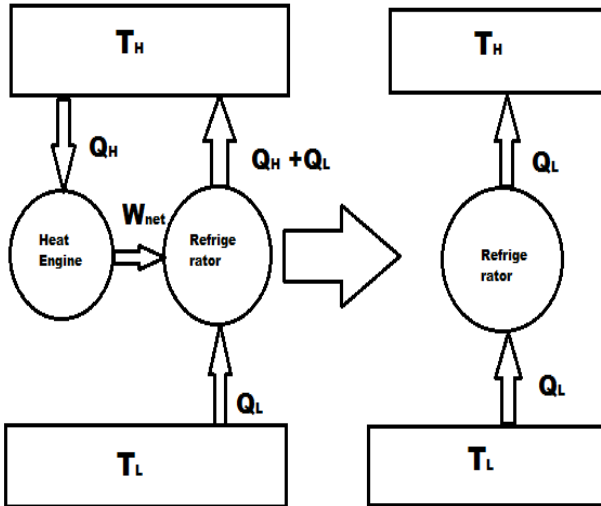
Fig:-The violation of the Kelvin-Planck statement leads to violation of Clausius.

3.3 CARNOT CYCLE

The efficiency of a heat-engine cycle greatly depends on how the individual processes that make up the cycle are executed. The net work (or efficiency) can be maximized by using reversible processes. The best known reversible cycle is the Carnot cycle.
Consider a gas in a cylinder-piston (closed system). The Carnot cycle has four processes:

Process (1-2) Reversible isothermal expansion:
The gas expands slowly, doing work on the surroundings. The heat transfers in the Reversible process from the heat source at T_H to the gas.

Process (2-3) Reversible adiabatic expansion:
The cylinder-piston is now insulated (adiabatic) and gas continues to expand reversibly. So, the gas is doing work on the

surrounding and as a result of expansion the gas temperature reduces from TH to TL.

1-2 : isothermal expansion   2-2 : adiabatic expansion
3-4 : isothermal compression   4-1 : adiabatic compression
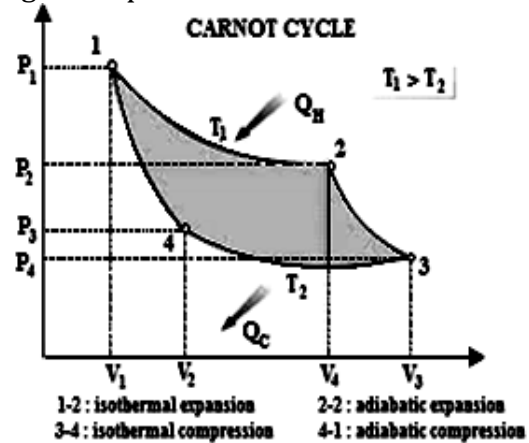
Process (3-4):
Reversible isothermal compression: The gas is allowed to exchange heat with a sink at temperature TL as the gas is being slowly compressed. The heat is transferred from the system to the surroundings such that the gas temperature remains constant at TL.

Process (4-1):
Reversible adiabatic compression: The gas temperature is increasing from T_L to T_H as a result of reversible adiabatic compression. Carnot cycle is the most efficient cycle operating between two specified temperature limits. The efficiency of all reversible heat engines operating between the two same reservoirs are the same.
The thermal efficiency of a heat engine (reversible or irreversible) is:

$$\eta_{th} = \frac{W}{Q_H} = 1 - \frac{Q_L}{Q_H}$$

For the Carnot cycle, it can be shown:

$$\eta_{th} = 1 - \frac{T_L}{T_H} \qquad \text{-------(3.6)}$$

3.3.1 CARNOT THEOREM

It states that all heat engines operating between same temperature limit none has a higher efficiency than a reversible engine. The efficiency of an irreversible (real) cycle is always less than the efficiency of the Carnot cycle operating between the same two reservoirs.

➤ If $\eta_{th} \leq \eta_{th}$ Carnot, it may be reversible or irreversible heat engine

➤ If $\eta_{th} > \eta_{th}$ Carnot, Heat engine is not possible.

3.4 REFRIGERATION AND HEAT PUMP WORKING ON REVERSED CARNOT CYCLE

A refrigerator or heat pump that operates on the reverse Carnot cycle is called a Carnot Refrigerator, or a Carnot heat pump. The Coefficient of performance of any refrigerator or heat pump (reversible or irreversible) is given by:

$$(COP)_R = \frac{Q_L}{Q_H - Q_L}, \quad (COP)_{HP} = \frac{Q_H}{Q_H - Q_L}$$

COP of all reversible refrigerators or heat pumps can be given by:

$$(COP)_R = \frac{T_L}{T_H - T_L} \text{ and } (COP)_{HP} = \frac{T_H}{T_H - T_L}$$

If $(COP)_R < (COP)_{R,Rev}$ The refrigerator is irreversible Refrigerator

If $(COP)_R = (COP)_{R,Rev}$ The refrigerator is reversible Refrigerator

If $(COP)_R > (COP)_{R,Rev}$ Refrigerator is not possible.

3.5 CLAUSIUS THEOREM

it states that for a cycle (reversible or irreversible cycle) the cyclic integral of $\left(\frac{\delta Q}{T}\right)$ is equal to zero or less than Zero. It means

$$\oint\left(\frac{\delta Q}{T}\right) \leq 0 \qquad \text{-------(3.8)}$$

if $\oint\left(\frac{\delta Q}{T}\right) < 0$, the cycle is irreversible cycle.

if $\oint\left(\frac{\delta Q}{T}\right) = 0$ the cycle is reversible cycle.

if $\oint\left(\frac{\delta Q}{T}\right) > 0$ the cycle is not possible.

3.6 ENTROPY

3.6.1 PHYSICAL SIGNIFICANCE OF ENTROPY

Entropy of a system is defined as the measure of degree of molecular disorder or random existing in the system. If higher is the randomness higher will be entropy. When heat is supplied to the system, randomness of the system increases, so the entropy of the system increases. Reversed effect can be measured when heat is removed from the system.

3.6.2 CLAUSIUS INEQUALITY LAW FOR REVERSIBLE CYCLE

Consider the cycle shown below composed of two reversible processes A and B. Apply the Clausius inequality for this cycle.
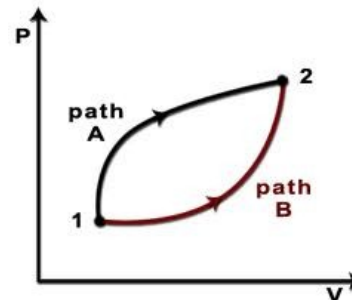
Apply the Clausius inequality for the cycle made of two internally reversible processes:

$$\oint\left(\frac{\delta Q_{net}}{T}\right)_{int\,rev} = 0$$

$$\underbrace{\int_1^2\left(\frac{\delta Q_{net}}{T}\right)_{int\,rev}}_{\text{along path A}} = \underbrace{\int_1^2\left(\frac{\delta Q_{net}}{T}\right)_{int\,rev}}_{\text{along path B}}$$

Since the quantity $(\delta Q_{net}/T)_{rev}$ is independent of the path and must be a property, this property is known as entropy S.

The entropy change during a process is related to the heat transfer and the temperature of the system. The entropy is given the symbol S (kJ/K), and the specific entropy is s (kJ/kg·K).

The entropy change during a reversible process is defined as

$$dS = \left(\frac{\delta Q}{T}\right)_{Rev}$$

$$S_2 - S_1 = \int_1^2\left(\frac{\delta Q}{T}\right)_{Rev} \qquad \text{--------(3.9)}$$

3.6.3 CLAUSIUS INEQUALITY LAW FOR IRREVERSIBLE CYCLE

Consider the cycle 1-A-2-B-1, shown below, where process A is arbitrary that is, it can be either reversible or irreversible, and process B is internally reversible.
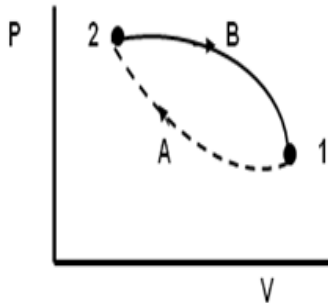
A cycle composed of reversible and irreversible processes.

$$\oint \left(\frac{\delta Q_{net}}{T} \right)_{int\,rev} \leq 0$$

$$\underbrace{\int_1^2 \left(\frac{\delta Q_{net}}{T} \right)_{int\,rev}}_{\text{along path A}} + \underbrace{\int_1^2 \left(\frac{\delta Q_{net}}{T} \right)_{int\,rev}}_{\text{along path B}} \leq 0$$

The integral along the reversible path, process B, is the entropy change $S_1 - S_2$. Therefore

$$\int_1^2 \left(\frac{\delta Q_{net}}{T} \right) + S_1 - S_2 \leq 0$$

$$S_2 - S_1 \geq \int_1^2 \frac{\delta Q_{net}}{T}$$

In general the entropy change during a process is defined as

$$dS \geq \frac{\delta Q_{net}}{T} \qquad \text{-------(3.10)}$$

➢ **=** holds for the internally reversible process
➢ **>** holds for the irreversible process.

3.6.4 ENTROPY CHANGE OF SYSTEM IN REVERSIBLE PROCESS

i) Heat is added to the system
if heat is added to the system, for a reversible process,

$$ds = \frac{dQ}{T}$$

the net heat transfer is positive so the entropy change will be positive. It means entropy of the system will increase.

ii) Heat is rejected from the system

if heat is rejected from the system, for reversible process

$$ds = \frac{dQ}{T}$$ the net heat transfer is negative so the entropy change will be negative. It means entropy of the system will decrease.

iii) No heat Transfer or adiabatic Process

If there is no heat transfer between the system and surrounding the net heat transfer will be zero. It means entropy of the system will be constant.

$$ds = \frac{dQ}{T}$$

$dS = 0$

$S_2 = S_1$

So reversible adiabatic process is known as isentropic process.

3.6.5 ENTROPY CHANGE FOR IRREVERSIBLE PROCESS

$$\Delta S_{sys} = S_2 - S_1 \geq \int_1^2 \frac{\delta Q_{net}}{T} \left(\frac{kJ}{K} \right)$$

Here, the inequality is to remind us that the entropy change of a system during an irreversible process is always greater than $\left(\frac{\delta Q}{T} \right)$, called the entropy transfer. That is, some entropy is generated or created during an irreversible process, and this generation is due entirely to the presence of irreversibility. The entropy generated during a process is called **entropy generation** and is denoted as S_{gen}. We can remove the inequality by noting the following

$$\Delta S_{sys} = S_2 - S_1 \geq \int_1^2 \frac{\delta Q_{net}}{T} + S_{gen} \left(\frac{kJ}{K} \right)$$

S_{gen} is always a positive quantity or zero. Its value depends upon the process and thus it is not a property. S_{gen} is zero for an internally reversible process. And its value increases with the irreversibility. So entropy generation is a path function.

Entropy Change of an Isolated System:- For the isolated system the heat transfer is zero.

So, $dS \geq \dfrac{\delta Q}{T}$ for the isolated system, $\delta Q = 0$
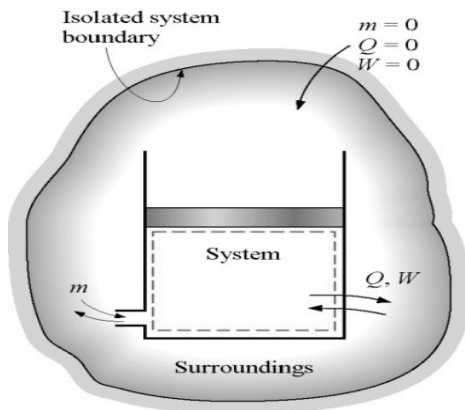
So
$dS \geq 0$

The total entropy change of an isolated system during a process always increases or, in the limiting case of a reversible process, remains constant. Universe is an isolated system so entropy of universe during a process always increases or, in the limiting case of a reversible process, remains constant.

The increase in entropy principle can be summarized as follows:

$$S_{gen} = \Delta S_{total} \begin{cases} > 0 & \text{Irreversible processes} \\ = 0 & \text{Reversible processes} \\ < 0 & \text{Impossible processes} \end{cases}$$

Heat Transfer as the Area under a T-S Curve:

For the reversible process, the equation for dS implies that

$$dS = \frac{\delta Q_{net}}{T}$$

$$\delta Q_{net} = TdS$$

The heat transfer in a reversible process is the differential area under the process curve plotted on the T-S diagram.

$$Q_{net} \int_1^2 TdS \quad --------(3.11)$$

This is the second law of thermodynamics for a reversible process.
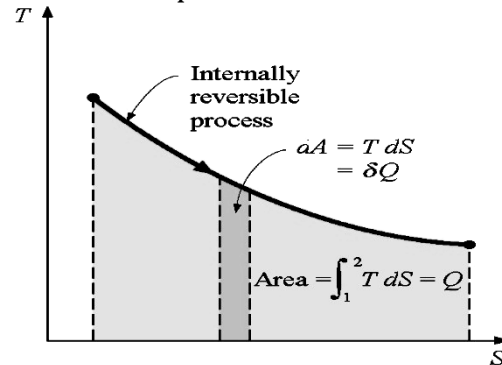
In the above figure, the heat transfer in an internally reversible process is shown as the area under the process curve plotted on the T-S diagram.

3.7 COMBINED FIRST LAW AND SECOND LAW OF THERMODYNAMICS

According to first law of thermodynamics
$\delta Q = P.dV + dU$ -----------------(3.12)
for the reversible process
According to first law of thermodynamics
$\delta Q = T.dS$ ---------------(3.13)
for the reversible process
From the equation (3.12) and (3.13)
$T.dS = P.dV + dU$ ----------- (3.14)
it is valid for reversible as well as irreversible process.
Equation (3.14) can be written as
$T.dS = dH - VdP$ ------(3.15)

3.8 ENTROPY CHANGE FOR AN IDEAL GAS

From the Equation (3.14) entropy change is given:

$$dS = \frac{P.dV}{T} + \frac{dU}{T}$$

For an ideal gas,

$$dU = m.C_V dT \text{ and } \frac{P}{T} = \frac{m.R}{V}$$

then

$$dS = \frac{m.R.dV}{V} + \frac{m.CVdT}{T}$$

Integrating the equation from state 1 to state 2

$$\int_1^2 dS = \int_1^2 \frac{m.R.dV}{V} + \int_1^2 \frac{m.C_v dT}{T}$$

$$S_2 - S_1 = m.R \ln \frac{V_2}{V_1} + m.C_v .\ln \frac{T_2}{T_1} ------(3.16)$$

From the equation (3.15)

$$dS = \frac{dH}{T} - \frac{VdP}{T}$$

For an ideal gas, dH = m.C_P dT and

$$\frac{V}{T} = \frac{m.R}{P}$$

Then $$dS = \frac{m.R.dP}{P} - \frac{m.C_p dT}{T}$$

$$\int_1^2 dS = \int_1^2 \frac{m.R.dP}{P} - \int_1^2 \frac{m.C_p dT}{T}$$

$$S_2 - S_1 = m.R \ln \frac{P_2}{P_1} - m.C_p .\ln \frac{T_2}{T_1} ------(3.17)$$

Solving the equation (3.16) and (3.17), We get

$$S_2 - S_1 = m.C_p \ln \frac{V_2}{V_1} + m.C_v .\ln \frac{P_2}{P_1} -------(3.18)$$

3.9 ENTROPY CHANGE FOR FINITE BODY

For the finite body entropy change may be written as

$$\int_1^2 dS = \int_1^2 \frac{\delta Q}{T} = \int_1^2 \frac{m.c.dT}{T}$$

$$S_2 - S_1 = m.C.\ln \frac{\tau_2}{\tau_1} ------(3.19)$$

3.10 SLOP OF ISOBARIC PROCESS AND ISOCHORIC PROCESS ON T-S DIAGRAM
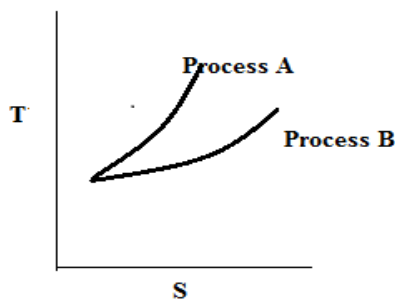
Isobaric and isocoric process

From the Equation (3.14)

T.dS = P.dV + dU

For the constant volume process, dV = 0

So T.dS = dU = m.C_V dT

$$\left(\frac{dT}{dS} \right)_v = \frac{T}{m.C_v} -----(3.20)$$

From the Equation (3.1)

T.dS = dH - V.dP

For the constant pressure process, dP = 0

So

T.dS = dH = m.C_P dT

$$\left(\frac{dT}{dS} \right)_p = \frac{T}{m.C_p} -----(3.21)$$

From the equation 3.20 and 3.21 it is cleared that slop of isobaric process is lesser than slop of isocoric process on T – S diagram.

Examples

Q.1 An ideal gas cycle is represented by a rectangle on a P-V diagram. If P_1 and P_2 are the lower and higher pressures and V_1 and V_2, the smaller and larger volume, respectively. Then
a) Calculate the work done per cycle.
b) Indicate which parts of the cycle involve heat flow in the gas
c) Show that

$$\eta = \frac{\gamma - 1}{\frac{\gamma P_2}{P_2 - P_1} + \frac{V_1}{V_2 - V_1}}$$

If heat capacities are constant.
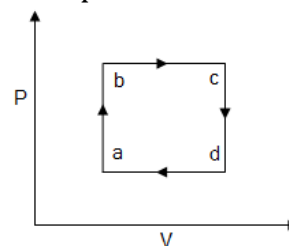
Solution:
a) w=area of the cycle
$$= (P_2 - P_1)(V_2 - V_1)$$
b) Process a-b and b-c
Heat absorbed by 1 Mole of gas in one cycle

$$Q = Q_{ab} + Q_{bc} = C_V(T_b - T_a) + C_P(T_C - T_b)$$

Now, $T_a = T_b \dfrac{P_1}{P_2}$ and $P_2 V_1 = RT_b$

$$T_c = T_b \dfrac{V_2}{V_1}; T_b = \dfrac{P_2 V_1}{R}$$

$$Q = C_V T_b\left(1 - \dfrac{P_1}{P_2}\right) + C_P T_P\left(\dfrac{V_2}{V_1} - 1\right) =$$

$$\dfrac{P_2 V_1}{R}\left(C_P\left(\dfrac{P_2 - P_1}{P_2}\right) + C_P\left(\dfrac{V_2 - V_1}{V_1}\right)\right)$$

Ans....c)

$$\eta = \dfrac{W}{Q} = \dfrac{(P_2 - P_1)(V_2 - V_1)}{\dfrac{P_2 V_1}{R}\left[C_P\left(\dfrac{P_2 - P_1}{P_2}\right) + C_V\left(\dfrac{V_2 - V_1}{V_1}\right)\right]}$$

$$= \dfrac{R(P_2 - P_1)(V_2 - V_1)}{C_V V_1(P_2 - P_1) + C_P P_2(V_2 - V_1)}$$

$$= \dfrac{C_P - C_V}{C_V \dfrac{V_1}{V_2 - V_1} + C_P \dfrac{P_2}{P_2 - P_1}}$$

$$= \dfrac{\gamma - 1}{\dfrac{V_1}{V_2 - V_1} + \dfrac{\gamma P_2}{P_2 - P_1}}$$

Q.2 A reversible heat engine operates between two reservoirs at temperatures of 600°C and 40°C. The engine drives a reversible refrigerator which operates between reservoirs at temperatures at 40°C and -20°C. The heat transfer to the heat engine is 2000 KJ and the net work output of the combined engine refrigerator plant is 360 KJ. (a)Evaluate the heat transfer to the refrigerant and the net heat transfer to the reservoir at 40°C (b) Reconsider (i) given that the efficiency of the heat engine and the cop of the refrigerator are each 40% of the maximum possible values.
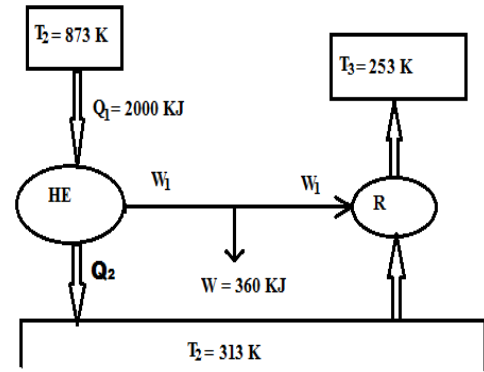
(a) Maximum efficiencies of the heat engine cycle

$$n_{max} = 1 - \dfrac{T_2}{T_1} = 1 - \dfrac{313}{873} = 1 - 0.358 = 0.642$$

= 64.2 %

Again $\dfrac{W_1}{Q_1} = 0.642$

$W_1 = 0.642 \times 2000 = 1284 KJ$

Maximum cop of the refrigerator cycle

$$(cop)_{max} = \dfrac{T_3}{T_2 - T_3}$$

$$= \dfrac{253}{313 - 253} = 4.22$$

Also $cop = \dfrac{Q_4}{W_2} = 4.22$

Since $w_1 - w_2 = w = 360 KJ$

$\therefore w_2 = w_1 - w = 1284 - 360 - 924 KJ$

$\therefore Q_4 = 4.22 \times 924 = 3899$ KJ

$\therefore Q_2 = Q_1 - W_1 = 2000 - 1284 = 716$ KJ

Heat rejection to $t_w = 40$°C reservoir

$= Q_2 + Q_3 = 716 + 4823 = 5539$ KJ

b) Efficiency of the actual heat engine Cycle

N = 0.4 × η_{max} = 0.4 × 0.642

$W_1 = 0.4 \times 0.642 \times 2000 = 513.6$ KJ

$\therefore W_2 = 513.6 - 360 = 153.6$ KJ

Cop of the actual refrigerator as do.

COP = 0.4 × 4.22 = 1.69

Therefore

$Q_4 = 153.6 \times 1.69 = 259.6$ KJ

$Q_3 = 259.6 + 153.6 = 413.2 KJ$

$Q_2 = Q_1 - W_1 = 2000 - 513.6 = 1486.4$ KJ

Heat rejected the 40^0C reservoir

$= Q_2 - Q_3 = 413.2 + 1486.4 = 1899.6$ KJ

Q.3 A System has a heat capacity at constant volume $C_V = AT^2$ Where $A = 0.042$J/K^3. The system is originally at 200 K, and a thermal reservoir at 100 K is available. What is the maximum amount of work that can be recovered as the system is cooled down to the temp of the reservoir?

Solution

Heat removed from the system
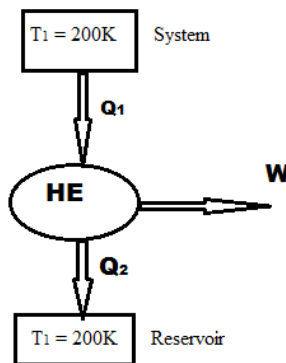
$$Q = \int_{T_1}^{T_2} C_V dT = \int_{T_1=200K}^{T_2=100K} 0.042 T^2 dT$$

$$= 0.042 [T^3 / 3]_{(200K)}^{(100K)}$$

$$= \frac{0.042}{3} J/K^3 \times (100^3 - 200^3) K^3$$

$$= -98 \times 10^3 J$$

$$(\Delta S) \text{System} = \int_{200K}^{100K} C_V \frac{dT}{T} = \int_{200K}^{100K} 0.042 T^2 \frac{dT}{T}$$

$$= \frac{0.042}{2}$$

$$J/K^3 \times (100^2 - 200^2) K^2 = -630 J/K$$

$$(\Delta S) \text{res} = \frac{Q_1 - W}{T_{res}} = \frac{98 \times 10^3 - W}{100}$$

(ΔS) Working flued in HE = 0

$(\Delta S) = (\Delta S)$ system $= (\Delta S)$ ros

$$= -630 + \frac{98 \times 10^3 - W}{100}$$

Since $(\Delta S) \geq 0$

$$\therefore -630 + \frac{98 \times 10^3 - W}{100} \geq 0$$

$$980 - \frac{W}{100} - 630 \geq 0$$

$$\frac{W}{100} \leq 350$$

W (max) = 35000 J = 35 KJ

Q.4 A body of constant heat capacity C_P and initial temperature T_1 is placed in contact with a heat reservoir at temperature T_2 and comes to thermal equilibrium with it. If $T_2 > T_1$, Calculate the entropy change of the universe and show that it is always positive.

Solution:

$$\Delta S \text{body} = m_{CP} \int_{T_1}^{T_2} \frac{dT}{T} = m_{CP} l_n \frac{T_2}{T_1}$$

$$\Delta S \text{ reservoir} = -m_{CP} \frac{(T_2 - T_1)}{T_2}$$

$$\Delta S \text{ universe} = m_{CP} \left[\ln \frac{T_2}{T_1} - \left(1 - \frac{T_1}{T_2} \right) \right]$$

Putting $1 - \frac{T_1}{T_2} = n$, $\frac{T_2}{T_1} = \frac{1}{1-n}$

Since $\frac{T_1}{T_2} > 1$, n is positive

$$\frac{\Delta S \text{universe}}{m_{CP}} = \ln 1 - l_n (1-n) - n = \frac{n^2}{2} + \frac{x^3}{3} + \frac{n^4}{4} +$$

(ds) > 0 It means entropy of universe is always positive.

Q.5 A hypothetical device is supplied with 2 kg/s of air at 4 bar, 300 K. Two separate streams of air leave the device, as shown in figure below. Each stream is at ambient pressures of 1 bar, and the mass flow rate is the same for both streams. One of the exits streams is said to be at 330 K while the other is at 270 K. The ambient temperature is 300 K.

Determine whether such a device is possible.

Solution:

The entropy generation rate for the control Volume
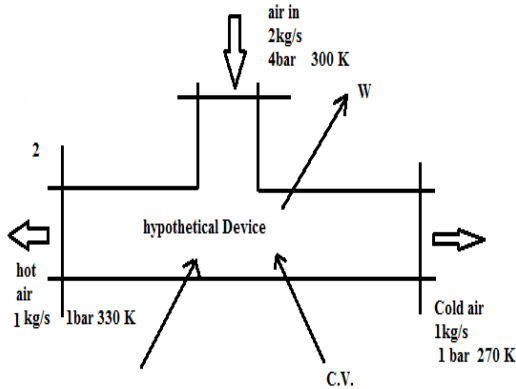
$$\dot{S}\,\text{gen} = \sum \dot{m}_e S_e - \sum \dot{m}_i S_i$$

$$= \dot{m}_2 S_2 + \dot{m}_3 S_3 - \dot{m}_1 S_1$$

$$= \dot{m}_2 S_2 + \dot{m}_3 S_3 - (\dot{m}_2 + \dot{m}_3) S_1$$

$$= \dot{m}_2 (S_2 - S_1) + \dot{m}_3 (S_3 - S_1)$$

Now $S_2 - S_1 = C_p \ln \dfrac{T_2}{T_1} - R \ln \dfrac{P_2}{P_1}$

$$= 1.005 \ln \frac{330}{300} - 0.287 \ln \frac{1}{4} = 0.494 \, KJ/kg.K$$

$$S_3 - S_1 = C_p \ln \frac{T_3}{T_1} - R \ln \frac{P_3}{P_1}$$

$$= 1.005 \ln \frac{270}{300} - 0.287 \ln \frac{1}{4}$$

$$= 0.294 \, KJ/Kg.K$$

$$\dot{s}gen = 1 \times 0.484 + 1 \times 0.292$$

$$= 0.786 \, KJ/K$$

$$= 0.786 \, K$$

Since

$\dot{S}$gen >0, the device is possible.

Such device actually exist and

called Vortex tube.

Q.6 A mass of m kg of fluid at temperate me T_1 is mixed with an equal mass of game fluid at temperature T_2 . Prove that the resultant change of entropy of universe is

$$= 2 \, ME l_n \frac{\left(\dfrac{T_1 + T_2}{2}\right)}{\sqrt{T_1 . T_2}}$$

Where C = Specific heat of fluid.

Solution:

Heat given by part-1

= heat taken by part-2

$$mc(T_1 - T) = mc(T - T_2)$$

$$T = \left(\frac{T_1 + T_2}{2}\right)$$

Change of entropy

$$ds = (dS)_1 + (dS)_2$$

$$= mc \int_{T_1}^{T} \frac{dT}{T} + mc \int_{T_2}^{T} \frac{dT}{T}$$

$$= mc \ln\left(\frac{T}{T_1}\right) + mc \ln\left(\frac{T}{T_2}\right)$$

$$= mc \ln\left(\frac{T^2}{T_1 . T_2}\right) = mc \ln\left(\frac{T}{T_1 . T_2}\right)^2$$

But $T = \dfrac{T_1 + T_2}{2}$

Substituting the value of T in the Expression

$$dS = 2mc \ln \left\{ \frac{\left(\dfrac{T_1 + T_2}{2}\right)}{\sqrt{T_1 . T_2}} \right\}$$

Stopcock opened
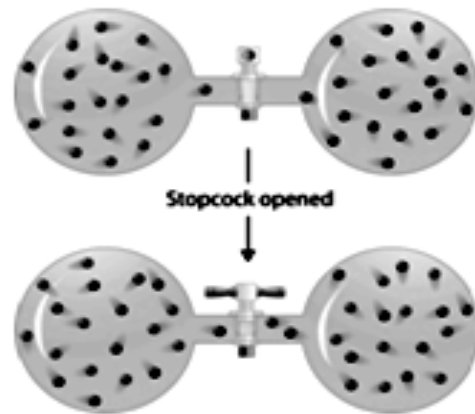
Q.7 A heat pump operates between two identical bodies. In the beginning , both the bodies are at same temp T_1 but operation of heat pump cools down one of the body to temperature T_2. Show that for the operation of heat pump the minimum work input needed by the heat pump for unit mass if given

$W_{min} = \left[\dfrac{T_1^2}{T_2} + T_2 - 2T_1\right]$. Where C is specific heat of bodies.

Solution:

Let the final temperature attained by body to which heat is rejected is T_f

$W = Q_B - Q_A$

Heat transfer from body A,

$Q_A = C(T_1 - T_2)$

Heat absorbed by body B,

$Q_B = C(T_f - T_1)$

Work input =

$Q_B - Q_A = C(T_f + T_2 - 2T_1)$

Now entropy change of body A

$(dS)_A = C \ln\left(\dfrac{T_2}{T_1}\right)$

Entropy change of body B

$(dS)_B = C \ln\left(\dfrac{T_f}{T_1}\right)$

For a reversible cycle entropy change of universe = 0.

$(dS)_{universe} = 0$

$(dS)_A + (dS)_B = 0$

substituting the values, we get C

$\ln\ln\left(\dfrac{T_2}{T_1}\right) + C\ln\left(\dfrac{T_f}{T_1}\right) = 0$

$C\ln\left(\dfrac{T_2 \cdot T_f}{T_1^2}\right) = 0$

$T_1 = \sqrt{T_2 \cdot T_f}$

$T_f = \dfrac{T_1^2}{T_2}$

Substituting the value of T_f in equation (i)

$W_{min} = C\left[\dfrac{T_1^2}{T_2} + T_2 - 2T_1\right]$

Q.8 A finite thermal system having heat capacity $C = 0.04T^2$ J/K initially at 600 K. Estimate the maximum work obtained from the thermal system if surrounding temperature is 300 K.

Solution:

$Q = C\, dT$
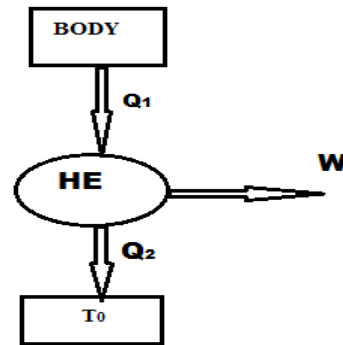
$Q = \int_{600}^{300} \left(0.04T^2\right) dT$

$= \left[\dfrac{0.04T^3}{3}\right]_{600}^{300}$

$= \dfrac{0.04}{3}\left[300^3 - 600^3\right]$

= -2520 kJ

Now the change of entropy of the body

$(ds)_{body} = \int_{T_1}^{T_2} \dfrac{dQ}{T} = \int_{T_1}^{T_2} \dfrac{C dT}{T} = \int_{T_1}^{T_2} \dfrac{0.04T^2 dT}{T}$

$\dfrac{0.04}{2}\left[300^2 - 600^2\right] = -5400\,J/k$

Entropy change of sink

$(dS)_2 = \dfrac{Q_2}{T_0} = \dfrac{Q - W}{T_0}$

For maximum work considering (dS) Universe = 0

$(dS)_1 + (dS)_2 = 0$

or $-5400 + \dfrac{Q - W}{T_0} = 0$

or $-5400 + \dfrac{2520 - W}{300} = 0$

so W = 900 kJ

The second law of thermodynamics tells us that it is not possible to convert all the heat absorbed by the system in to the work. Only some fraction of heat can be converted on the work. This work will be more valuable as compare to the heat. So Work is high grade energy and heat is the low grade energy.

Example: High grade energy:

1) Mechanical work
2) electrical energy
3) water power
4) wind power
5) kinetic energy of a jet
6) tidal power.

Example: Low grade energy:
1) Heat or thermal energy
2) heat derived from nuclear fission or fusion.
3) Heat derived from combustion of fossil fuels.
4) Solar energy.

4.1 AVAILABLE ENERGY

The maximum amount of work that can be obtained by a system in a cycle is called as Available Energy (AE).

Suppose a certain quantity of energy Q_1 as heat can be received from a reservoir, at temperature T_1. The sink temperature is T_2. It rejects the heat to the surrounding at temperature of T_2. The maximum possible work done by the system in the cycle is known as Available Energy. The maximum possible work can be achieved when the cycle is reversible cycle.
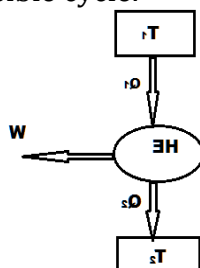
Fig: heat engine

The maximum work done obtained in the cycle will be the work done in the reversible cycle.

So for the reversible cycle,

$$\eta_{th} \frac{W}{Q_1} = 1 - \frac{T_2}{T_1}$$

$$W = Q_1 \cdot \eta_{th} = Q_1 \cdot \left(1 - \frac{T_2}{T_1} \right)$$

But the ambient temperature is T_0. The sink temperature will be T_0.

$$W\max = Q_1 \cdot \left(1 - \frac{T_0}{T_1} \right) = Q_1 - T_0 \, (dS)$$

Where dS is the entropy change of the system. The Carnot cycle and the available energy are shown in figure. The area 1-2-3-4 represents the available energy. Suppose a finite body is used as a source. Let a large number of differential Carnot engines be used with the given body as the source.
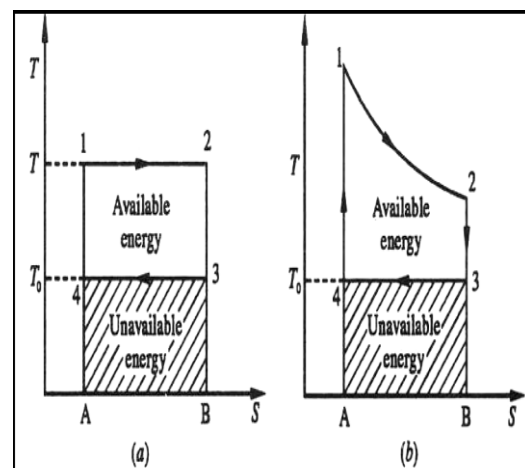
fig.: Available and unavailable Energy

Suppose a finite body is used as a source. Let a large number of differential Carnot engines be used with the given body as the source.

$$dW = dQ_\eta = dQ \left(1 - \frac{T_0}{T} \right)$$

If the initial and final temperatures of the source are T_1 and T_2 respectively, the total

work done or the available energy is given by

$$w = \int dQ_\eta = \int_{T_1}^{T_2} dQ\left(1 - \frac{T_0}{T}\right) = Q - T_o\int_{T_1}^{T_2} \frac{dQ}{ET}$$

or $w = Q - T_0 |\Delta S|$

So

Available Energy $(AE) = W_{max} = Q_1\cdot\left(1 - \frac{T_0}{T_1}\right)$

$= Q_1 - T_0\cdot (dS)$ ------(4.1)

4.2 UNAVAILABLE ENERGY

The shaded area 4-3-B-A represents the energy, which is discarded to the ambient atmosphere and this quantity of energy, cannot be converted into work and is called **Unavailable energy.**

$$UAE = T_{0.} (dS) = Q_1\left(\frac{T_o}{T_1}\right). \quad \text{----- (4.2)}$$

4.3 LOSS IN AVAILABLE ENERGY

Suppose a certain quantity of energy Q is transferred from a body at constant temperature T1 to another body at constant temperature T_2 (T_2<T_1).
Initial available energy, with the body at T_1,
Available   Energy

$$(AE)_1 = W_{max} = Q_1\cdot\left(1 - \frac{T_0}{T_1}\right) = Q_1 - T_0\cdot (dS)$$

Available energy, with the body at T_2,
Available   Energy

$$(AE)_2 = W_{max} = Q_1\cdot\left(1 - \frac{T_0}{T_2}\right) = Q_1 - T_0\cdot(dS')$$
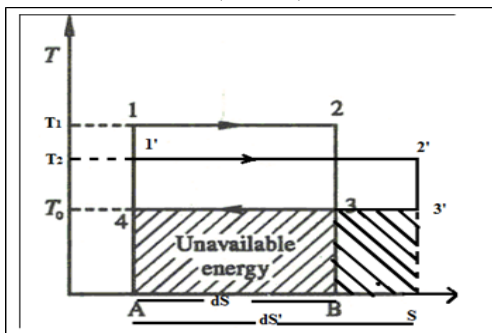
fig.: loss of available energy
Decrease in available Energy

$$= (AE)_1 - (AE)_2 = Q_1\cdot\left(\frac{T_0}{T_2} - \frac{T_0}{T_1}\right)$$

$$= Q_1\cdot T_0\left(\frac{T_1 - T_2}{T_1\cdot T_2}\right)$$

$= T_0\cdot (dS) - T_0\cdot (dS')$

Decrease in available energy or increase in unavailable energy

$$= Q_1\cdot T_0\left(\frac{T_1 - T_2}{T_1\cdot T_2}\right) = T_0\cdot(dS) - T_0\cdot(dS^1) \text{ ----- (4.3)}$$

From equation 3 it clears that when same amount of heat is supplied at the lower temperature its value is low, because it produces the less amount of work as compare to work done produced by the heat when it is supplied by higher temperature.

So according to first law of thermodynamics heat at higher and lower temperature has the same significance. But according to second law of thermodynamics heat at the higher temperature is more valuable as compare to heat at lower temperature.
So,
➢ First law of thermodynamic is called as quantitative law.
➢ Second law of thermodynamic is called as qualitative law.

4.4 AVAILABILITY FUNCTION

The availability of a given system is defined as the maximum useful work that can be obtained in a process in which the system comes to equilibrium with the surroundings or attains the dead state. It is given by maximum work done – atmospheric work.

A) AVAILABILITY FUNCTION FOR NON-FLOW PROCESS

Let P_0 be the ambient pressure, V_1 and V_0 be the initial and final volumes of the system respectively. If in a process, the system comes into equilibrium with the

surroundings, the work done in pushing back the ambient atmosphere is $P_0(V_0-V_1)$.

Availability $= W_{useful} = W_{max} - P_0(V_0 - V_1)$.

Consider a system which interacts with the ambient at T_0. Then from the first law of thermodynamics

$\delta Q - \delta W = Du$

Max work can be obtained when process is reversible process.

Let S_1 and S_2 be the entropies of the systems at the entrance and exit of the device then $(dS)_{system} = S_2 - S_1$. and (dS) surr $= - (Q / To)$

From the entropy principle

$(S_2 - S_1) - (Q / To) \geq 0$.

For the reversible process,

$(S_2 - S_1) - (Q / To) = 0$

Therefore $Q = To(S_2 - S_1)$

$W_{max} = (U_1 - U_0) - T_0(S_1 - S_0)$

$= (U_1 - T_0 S_1) - (U_0 - T_0 S_0)$

So Availability $= W_{useful} = (U_1 - T_0 S_1) - (U_0 - T_0 S_0) - P_0(V_0 - V_1)$

$= (U_1 + P_0 V_1 - T_0 S_1) - (U_0 + P_0 V_0 - T_0 S_0)$

$= \phi_1 - \phi_0$

Availability Function $(\phi) = U + P_0 V - T_0 S$ ---- (4.4)

Where $\phi = U + P_0 V - T_0 S$ is called the availability function for the non flow process. Thus, the availability on any given state: $\phi_1 - \phi_0$

If a system undergoes a change of state from the initial state 1 (where the availability is $(\phi_1 - \phi_0)$ to the final state 2 (where the availability is $(\phi_2 - \phi_0)$, the change in the availability or the change in maximum useful work associated with the process, is $\phi_1 - \phi_2$.

So the work done by the system when system undergoes from state 1 to state 2 is given by

$W_{1-2} = f_1 - f_2 = (U_1 + P_0 V_1 - T_0 S_1) - (U_2 + P_0 V_2 - T_0 S_2)$

-------- (4.5)

B) AVAILABILITY FUNCTION FOR FLOW PROCESS

The maximum power that can be obtained in a steady flow process while the control volume exchanges energy as heat with the ambient at T_0:

The steady stet steady flow equation is given by

$H_1 + \frac{1}{2}mC_1^2 + mgZ_1 + \delta Q = H_2 + \frac{1}{2}mC_2^2 + mgZ_2 + \delta w_{cv}$

The subscript 1 and 2 refer to the entrance and exit respectively. At the exit let the system be in equilibrium with the environment at pressure P_o and temperature To. Let symbols without subscripts refer to the entrance condition of the system and changes in KE and PE are negligible. Then useful work is given by

$W = (H_2 - H_1) + Q$

The greater the value of Q larger will be the useful work. Thus W will be maximum when Q is a maximum.

Let S_1 and S_2 be the entropies of the systems at the entrance and exit of the device then $(dS)_{system} = S_2 - S_1$. and (dS)surr $= - (Q / To)$

From the entropy principle

$(S_2 - S_1) - (Q / To) \geq 0$

For the reversible process,

$(S_2 - S_1) - (Q / To) = 0$

Therefore $Q = To (S_2 - S_1)$.

The useful work $W_{useful} = W = (H_2 - H_1) + To (So - S)$

$= (H_1 - ToS_1) - (H_2 - ToS_1)$

So the maximum possible work done

$W_{max} = \psi_1 - \psi_2$

Where ψ is known as the availability function for steady flow.

$\psi = H - ToS$ --------- (4.6)

4.5 IRREVERSIBILITY

The actual work done by a system is always less than the idealized reversible work, and the difference between the ideal work and actual work is called the irreversibility of the process.

So irreversibility can be given as

$I = W_{max} - W_{act}$

This is also some time referred to as degradation or dissipation. For a non flow process between the equilibrium states,

when the system exchanges heat only with the environment

$I = \{(U1 - U2) - T_o(S1 - S2)\}_{system} - \{(U1 - U2) + Q\}_{surr}$

= To (S2-S1) – Q

= To (dS) system + To (dS)surr

= To [(dS)system + (dS)surr] Hence I> 0.

Similarly for the steady flow process

I = Wmax – W$_{act}$

= To (S2-S1) – Q

= To (dS) system + To (dS)surr

= To[(dS)system + (dS)surr]

Thus same expression for irreversibility applies to both flow and non-flow process. The quantity To[(dS)system + (dS)surr] represents irreversibility.

I = To(dS)$_{universe}$ = To[(dS)system + (dS)surr]

-------- (4.7)

THERMODYNAMIC RELATIONS

4.6 EXACT DIFFERENTIAL EQUATIONS

Theorem No. 1: If a variable Z is a function of the two variable x and y i.e.

If   Z = f (x , y)

dZ = Mdx + Ndy

M and N are the function of x and y.

The differential is exact differential if

$\left(\dfrac{\delta M}{\delta y}\right)_x = \left(\dfrac{\delta N}{\delta x}\right)_y$ ---------- (4.8)

Theorem No. 2: if a variable f is the function of the x, y, z and a relation exists i.e. If f = f (x, y,z)

$\left(\dfrac{\delta x}{\delta y}\right)_f \left(\dfrac{\delta y}{\delta z}\right)_f \left(\dfrac{\delta z}{\delta x}\right)_f = 1$ ----------- (4.9)

Theorem No. 3: If a variable z is the function of x, y i.e.

If   x = f (y, z)   ------- (4.10)

4.7 MAXWELL'S EQUATIONS

A pure substance in a single phase has only two independent variables.

According to combined first Law & Second Law of thermodynamics

i)   dU = Tds – PdV

ii)   dH = Tds + VdP

Gibbs function (G) and the Helmholtz function (F) can be given as

dG = dH – TdS

Gibbs function

dF = dU –TdS

Helmholtz function

Or these functions can be written as

iii) dF=-PdV - SdT   – Helmholtz function

iv) dG = VdP - SdT - Gibbs function

U, H, F and G are thermodynamic properties and exact differential ,so

Appling the Exact differential rule equation, we get Maxwell equations:

$\left(\dfrac{\delta T}{\delta V}\right)_s = -\left(\dfrac{\delta P}{\delta S}\right)_v$ ------------ (4.11)

$\left(\dfrac{\delta T}{\delta P}\right)_s = -\left(\dfrac{\delta V}{\delta S}\right)_P$ ------------ (4.12)

$\left(\dfrac{\delta P}{\delta T}\right)_V = \left(\dfrac{\delta S}{\delta V}\right)_T$ ---------- (4.13)

$\left(\dfrac{\delta V}{\delta T}\right)_P = -\left(\dfrac{\delta S}{\delta P}\right)_T$ ------------ (4.14)

These four equations are known as Maxwell's equations.

4.8 T – DS EQUATIONS

Let Entropy is function of temperature and volume S = f (T , V)

Differentiating the equation

$dS = \left(\dfrac{\partial S}{\partial T}\right)_V dT + \left(\dfrac{\partial S}{\partial V}\right)_T dV$

multiplying the equation with temperature T

$Tds = T\left(\dfrac{\partial S}{\partial T}\right)_V dT + T\left(\dfrac{\partial S}{\partial V}\right)_T dV$

We know that

$T\left(\dfrac{\partial S}{\partial T}\right)_V = C_V$ and $\left(\dfrac{\partial S}{\partial V}\right)_T = \left(\dfrac{\partial P}{\partial T}\right)_V$

substituting the values in TdS equation, we get

$Tds = C_v dT + T\left(\dfrac{\partial p}{\partial T}\right)_v dv$ ------------(4.15)

This is known as first T.ds Equation.

Let the entropy is the function of temperature T and Pressure P

S = f (T , P)

Differentiating the equation

$$ds = \left(\frac{\partial S}{\partial T}\right)_P dT + \left(\frac{\partial S}{\partial P}\right)_T dP$$

multiplying the equation with temperature

$$Tds = T\left(\frac{\partial S}{\partial T}\right)_P dT + T\left(\frac{\partial S}{\partial P}\right)_T dP$$

We know that,

$$T\left(\frac{\partial S}{\partial T}\right)_P = C_p \text{ and } \left(\frac{\partial S}{\partial P}\right)_T = -\left(\frac{\partial V}{\partial T}\right)_P$$

substituting the values in TdS equation, we get

$$TdS = C_p dT - T\left(\frac{\partial v}{\partial T}\right)_P dp \text{ --------(4.16)}$$

This is known as Second T -dS Equation

4.9 ENERGY EQUATION

From combined first Law & second Law of thermodynamics

dU = TdS – P.Dv

Substituting the first TdS equation in the internal energy equation,

$$dU = C_V dT + T\left(\frac{\partial P}{\partial T}\right)_V dV - P.dV$$

or

$$dU = C_v dT + \left[T\left(\frac{\partial p}{\partial T}\right)_v - p\right] dv \text{ ----(4.17)}$$

For an ideal gas

PV = mRT

$$\text{or } T\left(\frac{\partial P}{\partial T}\right)_V = \frac{mRT}{V} = P$$

substituting the value in equation 4.17, we get

$$dU = C_V dT + [P - P] dV \text{ ----(4.18)}$$

dU = C_V.dT

4.10 ENTHALPY EQUATION

From combined first and second Law of thermodynamics,

dH = TdS + VdP

$$Tds = \left[C_p dT - T\left(\frac{\partial V}{\partial T}\right)_P\right] dP$$

$$d = \left[C_p dT - T\left(\frac{\partial V}{\partial T}\right)_P\right] dP + Vdp$$

$$dH = C_p dT + \left[V - T\left(\frac{\partial v}{\partial T}\right)_p\right] dP \text{ ------- (4.19)}$$

For an ideal gas

PV = mRT

$$T\left(\frac{\partial V}{\partial T}\right)_P = \frac{mRT}{P} = V$$

Substituting the Value in equation 4.19, we get

$$dH = C_p dT + [V - V] dP$$

$$dH = C_V/dT \text{ ---- (4.20)}$$

4.11 DIFFERENCE IN HEAT CAPACITIES

Equating the first and second T- dS Equation from equation 4.15 and 4.16

$$TdS = C_p dT - T\left(\frac{\partial V}{\partial T}\right)_P dP = C_V dT + T\left(\frac{\partial P}{\partial T}\right)_V dV$$

$$\text{Or } (C_P - C_V) dT = T\left(\frac{\partial P}{\partial T}\right)_V dV + T\left(\frac{\partial V}{\partial T}\right)_P dP$$

Or

$$dT = \frac{T\left(\frac{\partial P}{\partial T}\right)_V dV}{C_P - C_V} + \frac{T\left(\frac{\partial V}{\partial T}\right)_P dP}{C_P - C_V}$$

Or

$$dT = \left(\frac{\partial T}{\partial V}\right)_P dV + \left(\frac{\partial T}{\partial P}\right)_V dP$$

$$\therefore \frac{T\left(\frac{\partial P}{\partial T}\right)_V}{C_P - C_V} = \left(\frac{\partial T}{\partial V}\right)_P \text{ and } \frac{T\left(\frac{\partial V}{\partial T}\right)_P}{C_P - C_V} = \left(\frac{\partial T}{\partial P}\right)_V$$

----------- (4.21)

From the equation 4.21

$$C_P - C_V = T\left(\frac{\partial P}{\partial T}\right)_V \left(\frac{\partial V}{\partial T}\right)_P$$

But $\left(\frac{\partial P}{\partial T}\right)_V \left(\frac{\partial T}{\partial V}\right)_P \left(\frac{\partial V}{\partial P}\right)_T = -1$

$$C_P - C_V = -T\left(\frac{\partial V}{\partial T}\right)_T^2 \left(\frac{\partial P}{\partial V}\right)_T \text{ ------ (4.22)}$$

It indicates the following facts

(a) Since $\left(\dfrac{\partial V}{\partial T}\right)^2$ is always positive and $\left(\dfrac{\partial P}{\partial V}\right)_T$ is negative so $C_P - C_V$ is always positive.

(b) at absolute temperature (T=0K); $C_P - C_V$

(C) for an ideal gas

$P_V = mRT$

$\left(\dfrac{\partial V}{\partial T}\right)_P = \dfrac{MR}{P} = \dfrac{V}{T}$

$\left(\dfrac{\partial P}{\partial V}\right)_T = \dfrac{-MRT}{V^2}$

$\therefore C_P - C_V = -T \times \dfrac{V^2}{T^2} \times \left(\dfrac{-MRT}{V^2}\right)$

$C_P - C_V = R$ ------- (4.23)

4.11.1 VOLUME EXPANSIBILITY (β)

$\beta = \dfrac{1}{V}\left(\dfrac{\partial V}{\partial T}\right)_P$ -----(4.24)

4.11.2 ISOTHERMAL COMPRESSIBILITY

$K = -\dfrac{1}{V}\left(\dfrac{\partial U}{\partial P}\right)T$ ------(4.25)

substituting the values from equation 4.24 and 4.25 in equation 4.22,

$C_P - C_V = \dfrac{TV\left[\dfrac{1}{V}\left(\dfrac{\partial V}{\partial T}\right)_P\right]^2}{-\dfrac{1}{V}\left(\dfrac{\partial V}{\partial P}\right)_T}$

$C_P - C_V = \dfrac{TVB^2}{K}$ -----(4.26)

4.12 JOULE KELVIN COEFFICIENT

The slop of an isenthalpic Curve (throttling process) on T-P diagram is known as Joule Kelvin Co-efficient (μ).

$\mu = \left(\dfrac{\delta T}{\delta P}\right)_h$ ------------ (4.27)

A gas is made to undergo continuous throttling process by a valve as shown in fig. Let P_1, T_1 be the arbitrarily chosen pressure and temperature before throttling and P_2, T_2 are the pressure and temperature after the throttling. These are plotted on T-P diagram.
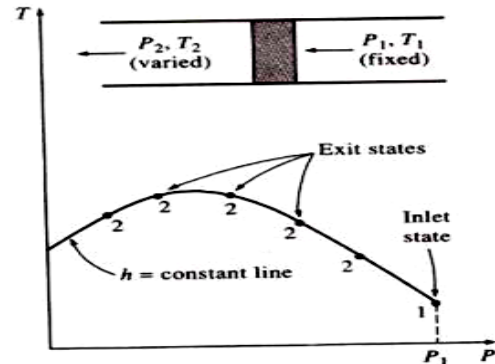
Then the initial temperature and pressure of the gas are set to new values, a family of isenthalpic is obtained for the gas as shown in fig. the locus of the points where μ is zero is known as Inversion curve. The region inside the inversion curve when μ is negative is called as heating region. The region inside the inversion curve when μ is positive is called as cooling region.
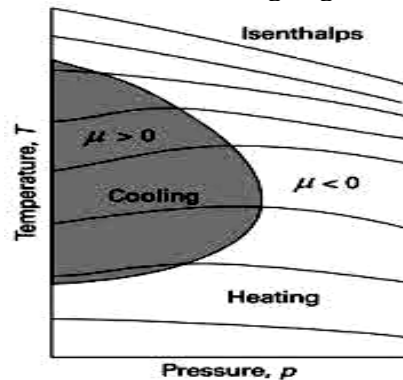
The enthalpy change of a gas is given by

$dh = C_p dT - \left\{T\left(\dfrac{\partial V}{\partial T}\right)_p - V\right\}dP$

$dh = 0$

$\mu = \left(\dfrac{\partial T}{\partial P}\right)_h = \dfrac{1}{Cp}\left\{T\left(\dfrac{\partial V}{\partial T}\right)_p - V\right\}$ ---(4.28)

For an ideal gas,

$T\left(\dfrac{\partial V}{\partial T}\right)_p - V = 0$

So $\mu = 0$

So there are some important points:

- For an ideal gas Joule Kelvin Co-efficient is Zero.
- In the cooling region the value of Joule Kelvin Co efficient is positive.
- In the heating region the value of Joule Kelvin Co efficient is negative.
- At the inversion curve, value of Joule Kelvin Co efficient is zero.

Examples

Q.1 In a steam generator, water is evaporated at 26.0^0c while the combustion gas $(C_P = 1.08 \frac{KJ}{kg}.k)$ is cooled from 1300^0C to 320^0c. The surrounding are at 30.0^0c.
Determine the loss of available energy due to the above heat transfer per kg of water evaporated (Latent heat of vaporization at water at $260^0c = 1662.5 KJ/kg$).

Solution:
Availability decrease of gas
$$AE_1 = H_1 - T_0S_1 - (H_2 - T_0S_2)$$
$$= H_1 - H_2 - T_0(S_1 - S_2)$$
$$= m_g c_p (T_1 - T_2) - T_0\left[m_g c_p \ln\left(\frac{T_2}{T_1}\right)\right]$$
$$= m_g c_p \left[(T_1 - T_2)\right] - T_0 \ln\left(\frac{T_2}{T_1}\right)$$

Given
$$T_1 = 1573k, T_2 = 320 + 273$$
$$= 593k \quad T_0 = 30 + 273 = 303k$$
$$AE_1 = m_g X1.08\left[(1573 - 593) - 303\ln\left(\frac{593}{1573}\right)\right]$$
$$= 739.16 m_g KJ$$

Availability increase of water
$$AE_2 = Q_1 - T_0 ds$$
$$= T_1 ds - T_0 ds$$
$$= (T_1 - T_0)ds$$
$$= (T_1 - T_0)m\left(\frac{LH}{T_1}\right)$$
$$= 1 \times 1662.5\left[1 - \frac{303}{533}\right]$$

$$= 717.4 KJ$$
For m mass flow rate of gas
Heat given by gas=heat gain by water
$$m_g c_{pg}(T_2 - T_1) = m_w(LH)$$
$$m_g X1.08(1300 - 320) = 1 X1662.5$$
$$m_g = 1.57 kg$$
$$AE_1 = 1.57 \times 739.16 = 1161.1 KJ$$
Loss of Availability = Agas - Aw
= (1161.1-717.4)
= 443.67 KJ

Q.2 Air expands through a turbine from 500 Kpa, 520^0C to 100 KPa, 300^0C. During expansion 10 KJ/kg of heat is lost to surrounding which is at 98 KPa, 20^0C. neglecting K.E. and P.E. changes, determine per kg of air (a) the decrease in availability (b) maximum work (c) the Irreversibility. For air, take $c_p = 1.005 KJ/kg.k, h = c_p T,$ where Cp is constant.

Solution:
The change of entropy of the air is
$$S_2 - S_1 = mc_p \ln\left(\frac{T_2}{T_1}\right) - mR \ln\left(\frac{P_2}{P_1}\right)$$
For unit mass of air
$$S_2 - S_1 = 1 \times 1.005 l_n(\frac{513}{793}) - 1 \times 0.287 \ln\left(\frac{1}{5}\right)$$
$$= 0.4619 - 0.3267$$
$$= 0.1352 KJ/kg\ k$$
Change in availability
$$\varnothing_1 - \varnothing_2 = H_1 - T_0S_1 - (H_2 - T_0S_2) = (H_1 - H_2) - T_0(S_1 - S_2)$$
$$= mC_P(T_1 - T_2) + T_0(S_2 - S_1)$$
$$= 1 \times 1.005(520 - 300) + 293(0.1352) = 260.7\ KJ/kg$$
Maximum work done
$$w_{max} = \varnothing_1 - \varnothing_2 = 260.7 KJ/kg$$
But from S.F.E.E
$$Q + h_1 = w + h_2$$
[Neglecting K.E. and P.E.]
$$w_{actual} = Q + (h_1 - h_2)$$
$$= -10 + 1 X1.005(520 - 300)$$

$= Q + (h_1 - h_2)$

$= 211.1 \text{ KJ/kg}$

So,

Irreversibility $I = w_{max} - w_{actual}$

$= 260.7 - 211.1$

$= 49.6 \text{KJ/kg}$

Q.3 0.2 kg of air at $300^0 c$ is heated reversible at constant pressure to 2066 k. Find the available and unavailable energies of the heat added. Take $T_O = 30^0 c$ and

$c_p = 1.0017 \text{ KJ/kg.k}$.

Solution:

Let the 0.2 kg of air is heated at cons leant pressure. So entropy change

$$S_2 - S_1 = mC_P l_n \left(\frac{T_2}{T_1}\right) - mRl_n \left(\frac{P_2}{P_1}\right)$$

$$= 0.2 \times 1.0047 l_n \left(\frac{2066}{573}\right)$$

$$= 0.2577 \text{KJ/K}$$

Increase in available energy of air

$AE = (H_2 - H_1) - T_0 (S_2 - S_1)$

$= mC_P (T_2 - T_1) - 303 \times 0.2577$

$= 0.2 X 1.0047 (2066 - 573) - 78.084$

$= 1250.14 - 78.08$

$= 1172.2 \text{KJ}$

Heat $\quad\quad$ Input $= mC_P (T_2 - T_1)$

$= 0.2 \times 1.0047(2066 - 573)$

$= 1250.24$

Unavailable Energy = Heat Supplied −

Available energy

$= 1250.24 - 1172 = 78.08 \text{KJ}$

Q.4 A pressure vessel has a volume of 1m^3 and contains air at 1.4mPa, $175^0 c$. The air is coaled to $25^0 c$ by heat transfer to surrounding at $25^0 c$. Calculate the availability in the initial and final states and the irreversibility of this process. Take $P_0 = 100 \text{KP}_a$.

Solution:

Let mass of air in the vessel is m kg

Mass of air $(m) = \dfrac{P_1 V_1}{RT_1}$

$$= \frac{1400 \times 1}{0.287 \times 998}$$

$$= 10.89 \text{kg}$$

Volume of gas after reaching the dead state

$V_0 = \dfrac{mRT_0}{P_0}$

$$= \frac{10.89 \times 287 \times 298}{100}$$

$V_0 = 9.31 \text{m}^3$

Availability at initial state

$AE_1 = \varnothing_1 - \varnothing_2$

$= U_1 - U_0 - T_0 (S_1 - S_0) + P_0(V_1 - V_0)$

$= mC_V (T_1 - T_2) - T_0$

$$\left[mC_P \ln\left(\frac{V_1}{V_0}\right) + mC_V \ln\left(\frac{P_1}{P_0}\right) \right] + P_0 (V_1 - V_0)$$

$= m0.718 X (448 - 298) - 298$

$$\left[1.005 \ln \frac{1}{931} \right] + 0.718 \ln \frac{1400}{100}$$

$+ 100(1-9.33)$

$= 9.31(150 + 103.52 - 831)$

Q.5 A heat source at $627^0 c$ transfer heat at rate of 3000 KJ/min. To a system maintained at $287^0 c$. A heat link is available at $27^0 c$. Assuming these temperatures to remain constant Find:

i)   Change in entropy of source

ii) Entropy production accompanying heat transfer

iii) The original available energy

iv) The Available energy after heat transfer

Solution:

Temperature $\quad\quad$ of $\quad\quad$ source

$T_1 = 627^0 c = 627 + 273 = 900 \text{K}$

Temperature of a system

$T_2 = 287^0 c = 287 + 273 = 560 \text{K}$

Sink temperature

$T_0 = 27^0 c = 27 + 273 = 300K$

i) Change of entropy of the source

$ds = -\dfrac{Q_1}{T_1} = \dfrac{-3000}{900 \times 60} = 0.056 KJ/Ksec$

ii) Entropy production accompanying heat transfer heat transfer by source
= Heat absorbed by System

$T_1 ds = T_2 . ds'$

$ds' = \dfrac{T_1 . ds}{T_2}$

$= \dfrac{3000}{560 \times 60} = 0.089 \dfrac{KJ}{Ksec}$

iii) The original available energy

$= Q_1 \left(1 - \dfrac{T_0}{T_1}\right) = 3000 \left(1 - \dfrac{300}{900}\right)$

$= 2000 KJ/min$

iv) Available energy after heat transfer

$= Q_1 \left(1 - \dfrac{T_0}{T_2}\right) = 3000 \left(1 - \dfrac{300}{560}\right)$

$= 1392.8 KJ/min$

Q.6 4 kg of water at $40^0 c$ mixed with 6 kg of water at $100^0 c$ in steady flow process.
Calculate:
(i) Temperature a resulting mixture
(ii) Change in entropy
(iii) Unavailable energy with respect to energy receiving water at $10^0 c$.

Solution:
(i) The final temperature at the mixture is T_f, then
Heat transfer by 4 kg at water =
Heat transfer by 6 kg at water

$m_1 c (T_f - T_1) = m_2 c (T_2 - T_f)$

$4 \times 4.18 (T_f - 40) = 6 \times 9.18 (100 - T_f)$

$T_f - 40 = 1.5 (100 - T_f)$

$T_f - 1.5 T_f = 150 + 10$

$T_f = T6^0 c$

(ii) Entropy change of mixture
Ds=Entropy change of 4 kg of water+Entropy change of b kg at water

$= m_1 c_p \ln\left(\dfrac{T_f}{T_1}\right) + m_2 c_p \ln\left(\dfrac{T_f}{T_2}\right)$

$= 4 \times 4.18 \ln\left(\dfrac{349}{313}\right) + 6 \times 4.18 \ln\left(\dfrac{349}{373}\right)$

$= 1.82 - 1.66 = 0.15 KJ/K$

(iii) Unavailable energy with respect to water at $40^0 c$

$UAE = T_0 (ds) system$

$= 29891.82) = 542.36 KJ$

Q.7 Using Maxwell's relations, show that for a pure substance

$TdS = C_p dT - TV_\beta dV$

Where β is co-efficient of thermal expansion, K is co-efficient of compressibility and C_P, C_V are specific heat at constant pressure and constant volume respectively.

Solution:
Let entropy of the pure substance
$S = f(P, T)$

$TdS = T \left(\dfrac{\partial T}{\partial P}\right)_T d\beta + T \left(\dfrac{\partial S}{\partial T}\right)_P dT$ _(1)

But

$\left(\dfrac{\partial T}{\partial S}\right)_P = Slop\ of\ Isobaric\ process = \dfrac{T}{C_P}$

So $T \left(\dfrac{\partial S}{\partial T}\right)_P = C_P$

And from Maxwell's equation

$\left(\dfrac{\partial S}{\partial P}\right)_T = -\left(\dfrac{\partial V}{\partial T}\right)_P$

Substituting the values in equation(1)

$TdS = C_P dT - T \left(\dfrac{\partial V}{\partial T}\right)_P dP$

Coefficient of thermal expansion

$B = \dfrac{1}{V}\left(\dfrac{\partial V}{\partial T}\right)_P$

$TdS = C_P dT - TV\beta dP$

And entropy of pure substance again
$$S = f(T, V)$$

$$dS = \left(\frac{\partial S}{\partial T}\right)_V dT + \left(\frac{\partial S}{\partial V}\right)_T dV$$

$$TdS = T\left(\frac{\partial S}{\partial T}\right)_V dT + \left(\frac{\partial S}{\partial V}\right)_T dV --- (2)$$

$$\left(\frac{\partial T}{\partial S}\right)_V = \text{Slop of isocoric process} = \frac{T}{C_V}$$

$$T\left(\frac{\partial S}{\partial T}\right)_V = C_V$$

And from Maxwell's Relation

$$\left(\frac{\partial S}{\partial T}\right)_V = \left(\frac{\partial P}{\partial T}\right)_V$$

Substituting the values in equation(2)

$$Tds = C_V dT + T\left(\frac{\partial P}{\partial T}\right)_V dV ----(B)$$

And

$$K = -\frac{1}{V}\left(\frac{\partial V}{\partial P}\right)_T \text{ and } \beta = \frac{1}{V}\left(\frac{\partial V}{\partial T}\right)_P$$

$$\frac{B}{K} = -\left(\frac{\partial P}{\partial V}\right)_T \cdot \left(\frac{\partial V}{\partial T}\right)_P$$

From cyclic property

$$\left(\frac{\partial P}{\partial V}\right)_T \cdot \left(\frac{\partial V}{\partial T}\right)_P \left(\frac{\partial T}{\partial P}\right)_V = -1$$

$$-\left(\frac{\partial P}{\partial V}\right)_T \cdot \left(\frac{\partial V}{\partial T}\right)_P = \left(\frac{\partial P}{\partial T}\right)_V$$

So $$\frac{B}{K} = \left(\frac{\partial P}{\partial T}\right)_V$$

Substituting the values in equation(3)

$$TdS = C_V dT + T\frac{B}{K} dV$$

Q.8 Derive the expression for $(dh)_T$ for a substance that obeys the equation of state given by
$$P = \frac{RT}{V} - \frac{q}{V^2}$$

Solution:

We know that internal energy of pure substance

$$dU = C_V dT + \left[T\left(\frac{\partial P}{\partial T}\right)_V - P\right]dV$$

If $U = f(T, V)$

$$dU = \left(\frac{\partial U}{\partial T}\right)_V dT + \left(\frac{\partial U}{\partial V}\right)_T dV$$

$$\left(\frac{\partial U}{\partial T}\right)_V = C_V, \left(\frac{\partial U}{\partial T}\right)_T = T\left(\frac{\partial P}{\partial T}\right)_V - P$$

For isothermal process
$$(dU)_T = 0$$

$$\left(\frac{\partial U}{\partial V}\right)_T = T\left(\frac{\partial P}{\partial T}\right)_V - P$$

But from the relation

$$P = \frac{RT}{V} - \frac{q}{V^2}$$

$$\left(\frac{\partial P}{\partial T}\right)_V = \frac{R}{V} - 0$$

$$T\left(\frac{\partial P}{\partial T}\right)_V = \frac{TR}{V}$$

$$T\left(\frac{\partial P}{\partial T}\right)_V - P = \frac{RT}{V} - P = \frac{q}{V^2}$$

So $$\left(\frac{\partial U}{\partial V}\right)_T = \frac{q}{V^2}$$

$$(dU)_T = \frac{q}{V^2} dV$$

Integrating the equation
$$\int_1^2 dU = \int_1^2 \frac{q}{V^2} dV$$

$$U_2 - U_1 = q\left[\frac{1}{V_1} - \frac{1}{V_2}\right]$$

$$(dh)_T = (U_2 - U_1) + (P_2 V_2 - P_1 V_1) h_2 - h_1 =$$

$$(P_2 V_2 - P_1 V_1) + q\left[\frac{1}{V_1} - \frac{1}{V_2}\right]$$

$$(h2 - h1)_T = (P_2 V_2 - P_1 V_1) + q$$

$$\left[\frac{1}{V_1} - \frac{1}{V_2}\right]$$

5 PROPERTIES OF PURE SUBSTANCE & GAS MIXTURE

5.1 PURE SUBSTANCE

A substance that has a fixed chemical composition throughout is called pure substance. Examples-Water, helium carbon dioxide, nitrogen Etc.

It does not have to be a single chemical element just as long as it is homogeneous throughout, like air. A mixture of phases of two or more substance is can still a pure substance if it is homogeneous, like ice and water or water and steam.

```
┌─────────────────────┐
│      Vapour          │
│                      │
│ ------------------   │
│ ------------------   │
│ -----Water---------  │
│ ------------------   │
│ ------------------   │
└─────────────────────┘
```
Pure Substance
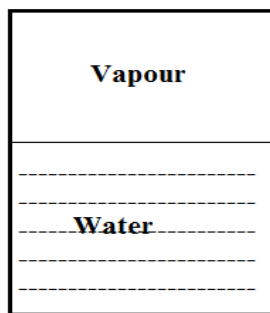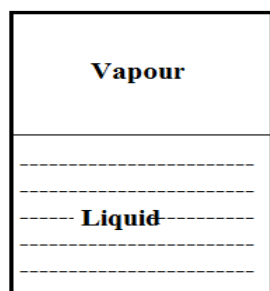
```
┌─────────────────────┐
│      Vapour          │
│                      │
│ ------------------   │
│ ------------------   │
│ ----- Liquid-------  │
│ ------------------   │
│ ------------------   │
└─────────────────────┘
```
Air

(Not a pure substance because the composition of liquid air is different)

5.1.1 PHASE OF PURE SUBSTANCES

There are three principle phases – solid, liquid and gas, but a substance can have several other phases within the principle phase. Nevertheless, thermodynamics deals with the primary phases only.

In general

➤ Solids have strongest molecular bonds. Their molecules do not move relative to each other.

➤ Liquid molecular spacing is comparable to solids but their molecules can float about in groups.

➤ Gas molecules have weakest molecular bond strength. Molecules in the gas phases are far apart, they have no ordered structure. The molecules move randomly and collide with each other.

5.2 P- V DIAGRAM OF PURE SUBSTANCE

Let the ice be heated slowly so that its temperature is always uniform.fig show the state change of a pure substance (other than water which contract due to conversion from ice to water).line passing through all the saturated solid states is called the saturated solid line.

The line passing all the saturated liquid points is called as the saturated liquid line.

The line passing all the saturated Vapour points is called as the saturated Vapour line.

Where the two lines (saturated liquid line and saturated vapour line) meets with each other is called as critical point.

The triple point on P-V diagram is a line where all the phases, i.e. solid liquid and gas phases exist in equilibrium.
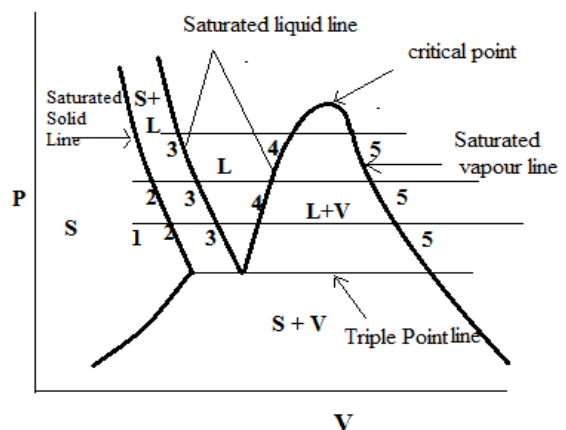
Fig.: P-V diagram of pure substance

At the pressure below than triple point substance can exist in liquid form. The

region below the triple point line is solid vapour mixture region.

For Water

Critical pressure (P_c) = 221.2 bar

Critical Temperature (T_c) = 374.15^{0}C

5.3 TRIPLE POINT

Triple point is state at which solid, liquid and gas phases exist in equilibrium. On a P-T diagram, this condition is a point. But on P-V Diagram it is a line. The curve which shows the phase change of solid into liquid or liquid into solid is called as fusion curve. The curve which shows the phase change of liquid into vapour or vapour into liquid is called as vaporization curve.
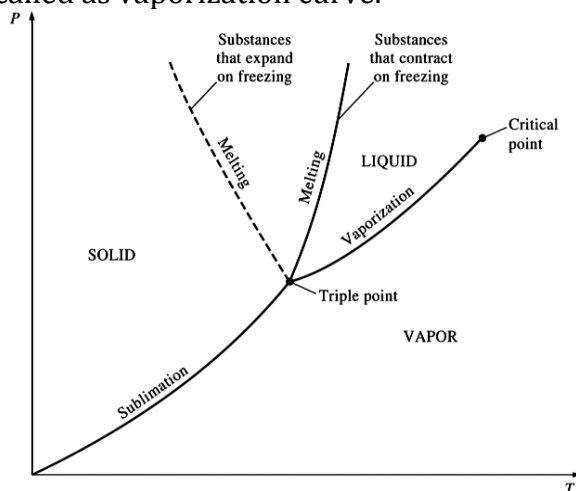
Fig.: Triple point on P- T diagram

The curve which shows the phase change of solid into vapour directly or vapour into solid is called as sublimation curve. The slop of vaporization and sublimation curve is positive. The slop of fusion curve for most of the substance is positive but for water is negative.

For water,

Triple point temperature (T_t) = 273.16 K

Triple point pressure (P_t) = 0.6113 KPa

5.4 GIBBS PHASE RULE

Gibbs phase rule describes the number of degrees of freedom, or the number of variables that must be fixed to specify the composition of a phase.

According to Gibbs phase rule,

2+C=F+P

Where

C = Number of chemically independent components

F = Number of degrees of freedom of intensive properties

P = Number of phases presented

For a pure substance existing in a phase C = 1, P = 1 so,

F = C + 2 – P = 1+2-1 = 2

It means there are two properties required to b3 known to fix up the state of the system at equilibrium.

For a single component and two phase system (i.e. critical point),

C =1, P = 2

F = F = C + 2 – P = 1+2-2 = 1

It means only one property is required to fix up the state of the system at equilibrium.

if all three exists in equilibrium (i.e. triple point) then C = 1, P = 3

F = C + 2 – P = 1+2-3 = 0

This state is unique for a substance. It means at triple point all the intensive properties are fixed.

5.5 PHASE CHANGE OF PURE SUBSTANCE

Thermodynamics deals only with liquid to gases to generate power. So it is important to understand the conversion of liquid into vapour or vice versa.

Consider water at room temperature (20°C) and normal atmospheric pressure as shown in fig.
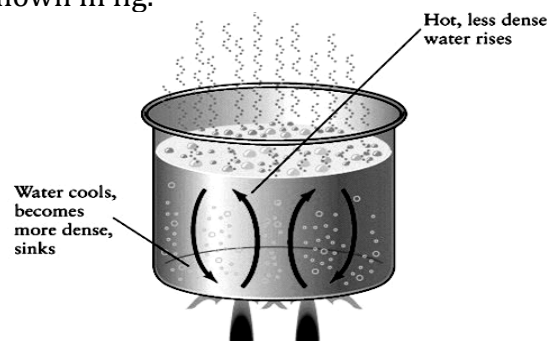
Fig. formation of steam

The water is in liquid phase and it is called **compressed liquid** or sub cooled liquid.

Now heat is supplied to water, its temperature will increase. Due to the increase in temperature, the specific volume v, specific enthalpy h and specific entropy s will increase. As a consequence, the piston will move slightly upward therefore maintaining constant pressure. Now if continue heat is added to the water, the temperature will increase further until 100°C. At this point, any additional addition of heat will not increase the temperature of water; it will utilise to vaporize some water. This specific point where water starts to vaporize is called **saturated liquid point**. And the condition of the liquid is called as saturated liquid.

If we continue to add heat to water, more and more vapour will be created, while the temperature and the pressure remain constant (T=100°C and P=1 atm). These conditions will remain the same until the last drop of liquid is vaporized. At this point, the entire cylinder is filled with vapour at 100°C. This state is called **saturated vapour**.

The state between saturated liquid and saturated vapour where two phases exist is called saturated liquid-vapour mixture or wet vapour.

After the saturated vapour phase, any addition of heat will increase the temperature of the vapour; this state is called **superheated vapour**.

This concept can be applied to pure substance other than water**.**

5.6 T-S DIAGRAM OF PURE SUBSTANCE

During the phase change of a pure substance as discussed in the previous topic first addition of heat is utilized to increase the temperature end entropy of the system. And then further addition of heat is to change the phase of liquid in which only entropy change temperature remains constant. But further addition of heat is to superheat the steam in which temperature and entropy both change. At a

given pressure process of phase change is shown T-S diagram.
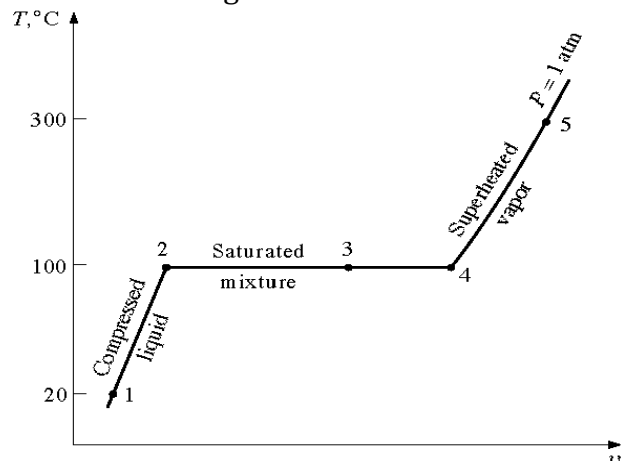
Fig.:T-S diagram representing phase change for water at constant pressure

Let us consider the different pressures and repeat process of phase change. On the different pressure we get the different saturated liquid point and saturated vapour point. The locus of all saturated liquid points is called as saturated liquid curve. And the locus of all the saturated vapour points at different pressure is called as saturated vapour curve.

Where the saturated liquid and vapour curve meet with each other is called as **Critical Point.** The same process is shown on T-s and P-s diagram.
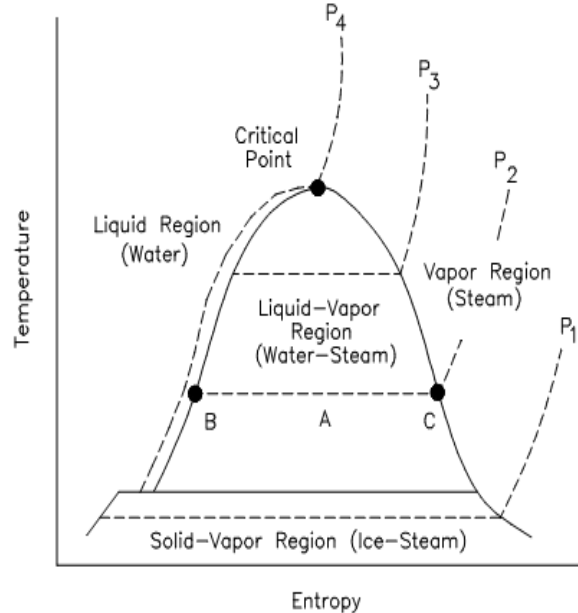
Fig. T-s diagram of pure substance
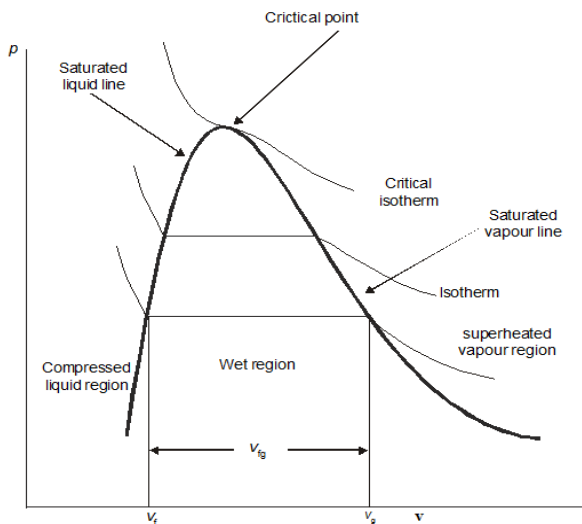
Fig. P-v diagram of pure substance

5.7. PROPERTIES OF PURE SUBSTANCE

Saturation Temperature

The temperature at which water starts boiling depends on the pressure. In other words, water starts boiling at 100 ºC but only at 1 atm. At different pressures, water boils at different temperatures.

Saturation Pressure

At a given pressure, the temperature at which a pure substance changes phase is called the saturation temperature (Tsat). Likewise, at a given temperature, the pressure at which a pure substance changes phase is called the saturation pressure (Psat).

Sensible Heating
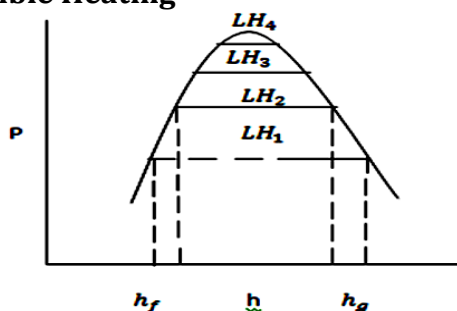
The addition of heat which is utilised to increase the temperature is called as sensible heating. In T- S diagram, Process

1-2 represents the sensible heat of water at 1 atmospheric pressure. At 100ºC saturation point is attained.

Latent Heating

The addition of heat which is utilized to phase change is called as latent heating. In process 2-4, heat is added to water that is utilised to convert the phase of liquid in to vapour and it is called as latent heating. the latent heat decreases as pressure increases and at the critical point latent heat of vaporization is zero.

Critical Point

It is point at which liquid directly converts into vapour phase. At the critical point, latent heat is zero. For water at the critical point,
P_{cr} = 220.8 bar
T_{cr} = 374.14ºC

Degree of Sub cooling

If the actual temperature of liquid is less than the saturated temperature of liquid at given pressure the liquid is called as sub cooled liquid. The difference of saturated temperature and actual temperature of liquid is called as degree of sub cooling. Let the saturated temperature is $T_{sat.}$ and actual temperature is $T_{sub.,}$ then
Degree of sub cooling = T_{sat} -T_{sub}

Degree of Sub cooling

If the actual temperature of vapour is greater than the saturated temperature of vapour at given pressure the vapour is called as superheated vapour. The difference of actual temperature of vapour and saturated temperature is called as degree of superheating. Let the saturated temperature is $T_{sat.}$ and actual temperature is $T_{sup.,}$ then
Degree of superheating =T_{sup} - T_{sat}

Wet Vapour

Wet vapour is mixture of liquid and vapour phase. in wet vapour region liquid and vapour phase exists in equilibrium.

Dryness Fraction or quality of mixture (x)

If vapour is made wet by liquid droplets in suspension, it is important to know the degree of wetness. Any mass of wet consists of dry saturated liquid and dry saturated vapour.

The ratio of mass of dry saturated vapour to total mass of water vapour is known as dryness fraction (x). It is also known as quality of steam.

Let the mass of dry saturated vapour is m_v and the mass of dry saturated liquid in wet vapour is m_l, then dryness fraction

$$x = \frac{m_v}{m_l + m_v}$$
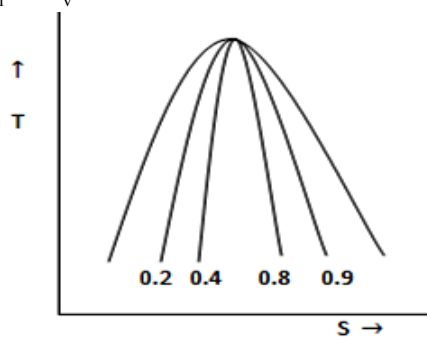
Important points

➤ At saturated liquid line mass of vapour is zero so value of x is zero along saturated liquid line.
➤ At saturated vapour line mass of liquid is zero, so dryness fraction is 1. So dryness fraction varies from 0 to 1.
➤ All dryness fraction lines converge at original point.

5.8 SPECIFIC VOLUME, ENTHALPY AND ENTROPY OF DIFFERENT PHASES

The specific enthalpy entropy and specific volume are very important parameters to define the performance of any power producing system such as steam power plant working on Rankin cycle and power absorbing system such as vapour compression refrigeration system. In most of the thermodynamic problem, these important phases can be existed. So it is important to determine the parameters in different region of pure substance.

1) Specific Volume, enthalpy and entropy of saturated liquid

In the T-S and P-h diagram it is shown by point 1. Enthalpy of this point is the heat supplied to increase the temperature from 0^0 C to saturated temperature at given constant pressure. these all parameters can be found Corresponding to saturated temp. at given pressure with help of stem table.

Consider specific enthalpy, entropy and volume of saturated liquid at a given pressure and temperature are h_f, s_f and v_f. specific enthalpy, entropy and volume of saturated vapour at a given pressure and temperature are h_g, s_g and v_g.

Then latent heat of vaporization
$h_{fg} = h_g - h_f$
specific volume $V_{fg} = V_g - v_f$, Specific entropy $S_{fg} = S_g - S_f$
From steam table at given pressure
Enthalpy of saturated liquid
$h_1 = h_f$
Entropy of saturated liquid
$s_1 = s_f$
Volume of the saturated liquid
$v_1 = v_f$

Fig. T- S diagram
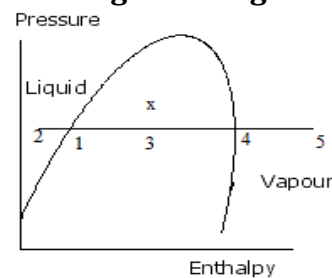
Fig. P- h diagram

2) Specific Volume, enthalpy & entropy of sub cooled liquid

In the T-S and P-h diagram it is shown by point 2.
Enthalpy of subcooled liquid

$h_2 = h_f - c_p(T_1 - T_1)$,

Entropy of subcooled liquid

$S_5 = S_f - c_p l_n\left(\dfrac{T_1}{T_2}\right)$

Volume of subcooled liquid can be found at particular temperature at the given pressure from the steam table.
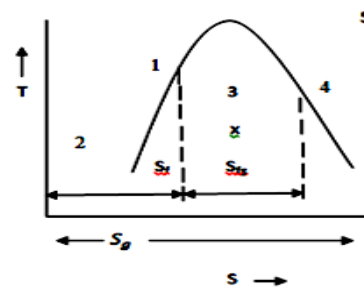
3) Specific Volume, enthalpy and entropy of saturated vapour

In the T-S and P-h diagram it is shown by point 4.
From steam table at given pressure
Enthalpy of saturated vapour

$h_4 = h_g$,

Entropy of saturated vapour

$s_4 = s_g$

Volume of saturated vapour

$v_4 = v_g$

4) Specific Volume, enthalpy and entropy of wet vapour

In the T-S and P-h diagram it is shown by point 3.
Enthalpy of wet vapour

$h_3 = h_f + x.h_{fg}$,

Entropy of wet vapour

$s_3 = s_f + xs_{fg} = s_f + x\dfrac{LH}{T}$

Volume of wet vapour

$v_3 = x.v_g$

5) Specific Volume, enthalpy and entropy of superheated vapour

In the T-S and P-h diagram it is shown by point 5.
Enthalpy of superheated vapour

$h_5 = h_g + c_p(T_5 - T_4)$

Entropy of superheated vapour

$S_5 = S_g + c_p l_n\left(\dfrac{T_5}{T_4}\right)$

Volume of superheated vapour

$v_3 = v_g.\left(\dfrac{T_5}{T_4}\right)$

PROPERTIES OF GAS MIXTURE

5.9 EQUATION OF STATE

The ideal gas equation of state is P.v = RT can be established for the real gas with two important assumptions of the ideal gas. The ideal gas assumptions are:
a) There is no attraction between the molecules of the gas.
b) The volume occupied by the molecules is negligible.

When the pressure is very low and temperature very large, the intermolecular attraction and volume of molecules has no more significance. As the pressure of gas increases, the intermolecular force forces of attraction and repulsion increases and also volume of the molecules becomes appreciable compare to total volume of the gas.

5.9.1 VANDER WAALS EQUATION

Vander Waals, by applying the law of mechanism of individual molecule introduced two correction terms in ideal gas equation. This is valid for the real gases.

$\left(P + \dfrac{\alpha}{v^2}\right)(v - b) = RT$

Where a and b are Vander Waals constant.
a is introduced to account the existence of mutual attraction between molecules. The term a/v^2 is called as force due to cohesion. The term b is introduced to account the volume of molecules is known as co – volume.

5.9.2 PROPERTIES OF CRITICAL POINT

According to Vander Waals equation,

$\left(P + \dfrac{a}{v^2}\right)(v-b) = RT$ or

$$P = \frac{RT}{(v-b)} - \frac{a}{v^2} \qquad \text{-------- (1)}$$
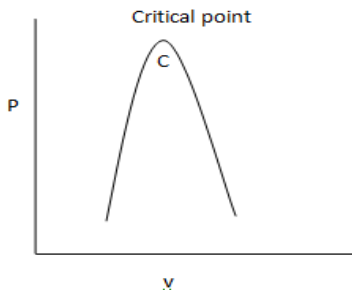
Slop of P-v diagram at critical point is zero, so differentiating the equation with respect to v and putting equal to zero.

$$0 = \left(\frac{dp}{dv}\right)_c = \frac{-RT}{(v-b)^2} + \frac{2a}{v^3}$$

$$\frac{RT}{(v_c-b)^2} = \frac{2a}{v_c^3} \qquad \text{-------------- (II)}$$

critical point is not only zero slop curve but at this point slop change so second derivative of P with respect to v is also zero. Differentiating the equation II and putting equal to Zero,

$$\left(\frac{d^2p}{dv^2}\right)_c = 0 = \frac{2RT}{(v_c-b)^3} - \frac{6a}{v_c^4}$$

$$\frac{RT}{(v_c-b)^3} = \frac{3a}{v_c^4} \qquad \text{----------------- (III)}$$

Now dividing the equation II by equation III, we get

$$(v_c - b) = \frac{2}{3}v_c$$

Vc = 3b

Substituting the value of v_c in equation III, we get

$$RT = \frac{2a(2b)^2}{27b^3} = \frac{8a}{27b}$$

$$Tc = \frac{8a}{27bR}$$

Substituting the value of T_c and v_c in equation I, we get

$$P = \frac{R X \dfrac{8a}{27bR}}{2b} - \frac{a}{9b^2}$$

$$p_c = \frac{a}{27b^2}$$

5.9.3 COMPRESSIBILITY FACTOR

The ideal gas equation can be given as
Pv = RT
If the gas is compressible (real gas), then

$$Z = \frac{pv}{RT}$$

Z > 1   or   Z < 1

5.9.4 COMPRESSIBILITY FACTOR AT CRITICAL POINT

At the critical point the compressibility factor is given by

$$Z_c = \frac{P_c V_c}{RT_c}$$

Substituting the value of P, V and T at critical point, we get

$$Z_c = \frac{\dfrac{a}{27b^2} X 3b}{R X \dfrac{8a}{27b R}} = \frac{3}{8}$$

$$Z_c = \frac{3}{8} = .375$$

5.10 PROPERTIES OF GAS MIXTURE

5.10.1 DALTON'S LAW OF PARTIAL PRESSURES

Let consider a homogeneous mixture of inert ideal gas at pressure P, Volume V and temperature T. Let us suppose there are n_1 numbers of molecules of gas A, n_2 number of molecules of gas B and n_3 number of molecules of gas C.

fig. gas mixture

Then the equation of state for mixture of the gas is

$PV = (n_1 + n_2 + n_3)\bar{R}T$

$\bar{R}$ = universal gas constant = 8.314 KJ/ kg mol K

Partial pressure of gas A, B and C can be written as

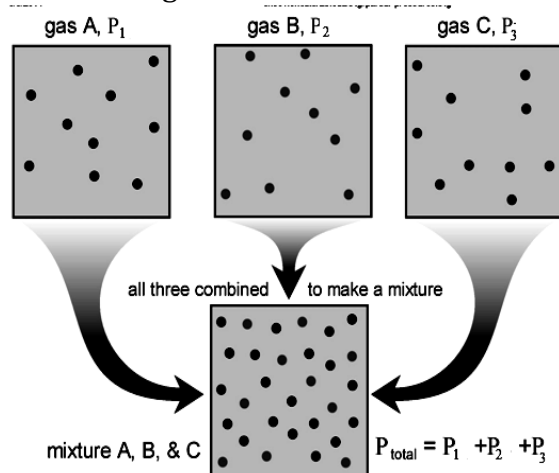$$P_1 = \frac{n_1 \bar{R}.T}{V}, P_2 = \frac{n_2 \bar{R}.T}{V}, P_3 = \frac{n_3 \bar{R}.T}{V}$$

According to Dalton law of partial pressure

$P = P_1 + P_2 + P_3$

$$P = \frac{n_1 \bar{R}.T}{V} + \frac{n_2 \bar{R}.T}{V} + \frac{n_3 \bar{R}.T}{V}$$

or $P = \dfrac{n_K \bar{R}.T}{V}$

where,

$$n_k = \sum n = n_1 + n_2 + n_3$$

5.10.2 MOLE FRACTION

The ratio of number of mole of a given gas to the total number of moles in the given mixture of ideal gas is known as mole fraction (x).

$$x_1 = \frac{n_1}{\sum n}, \quad x_2 = \frac{n_2}{\sum n} \text{ and }, \quad x_3 = \frac{n_3}{\sum n}$$

Equivalent Gas constant (R_e)

Ideal gas equation for the gas mixture in the term of masses can be written as
For gas A in the gas mixture
$P_1 V = m_1 R_1 T$
For gas B in the gas mixture
$P_2 V = m_2 R_2 T$
For gas C in the gas mixture
$P_3 V = m_3 R_3 T$
Using Dalton law of partial pressure,
$PV = P_1 V + P_2 V + P_3 V$
or $PV = m_1 R_1 T + m_2 R_2 T + m_3 R_3 T$
$PV = (m_1 R_1 + m_2 R_2 + m_3 R_3)T$ -----------(I)
Ideal gas equation for a gas mixture,
$PV = (m_1 + m_2 + m_3).R_e.T$ ------------(II)
Comparing the equation I and II, we get

$$R_e = \frac{m_1 R_1 + m_2 R_2 + m_3 R_3}{m_1 + m_2 + m_3}$$

In the same way internal energy, enthalpy and specific heats are given as

Specific Internal Energy of the mixture

$$U_e = \frac{m_1 u_1 + m_2 u_2 + m_3 u_3}{m_1 + m_2 + m_3}$$

Specific enthalpy of the mixture

$$H_e = \frac{m_1 h_1 + m_2 h_2 + m_3 h_3}{m_1 + m_2 + m_3}$$

Specific heat at constant pressure of the mixture

$$C_p = \frac{m_1 C p_1 + m_2 C p_2 + m_3 C p_3}{m_1 + m_2 + m_3}$$

Specific heat at constant volume of the mixture

$$C_{ve} = \frac{m_1 C_{v1} + m_2 C_{v2} + m_3 C_{v3}}{m_1 + m_2 + m_3}$$

Examples

Q.1 Determine the specific enthalpy of steam at 2 MPa and with a temperature of $250^0 C$.

Solution:

From the steam table

At 2 MPa, $T_{sat} = 212.4^0 C$

It must be the condition of super heating because the temperature of steam is higher than saturated steam.

So Degree of superheating

$T_{sup} - T_{sat} = 250 - 212.4 = 37.6K$

Specific enthalpy of steam at 2MPa is given

$h_g = 2797 KJ/kg$

So enthalpy

$h = h_g + c_p(T_{sup} - T_{sat})$

$= 2797.2 + 2.09 \times 37.5$

$= 2875.9 \ KJ/kg$

Q.2 Steam 0.95 dry at a pressure of 0.7 MPa is supplied to a heater through a pipe of 25mm internal diameter. The velocity in the pipe is 12 m/sec. water enters the heater at 19^0C, the steam is blow into it and the mixture of water and condensate leaves the heater at 90^0C. Calculate:

a) mass of steam entering the heater

b) mass of water entering the heater

Properties of steam are:

Pressure Mpa	Sat. temp. T_{sat} (^0C)	Specific enthalpy (KJ/kg)			Specific volume v_g m^3/kg
		h_f	h_{fg}	h_g	
0.7	165	699.1	2064.9	2762.00	0.273

Solution:

Specific volume of dry (0.95) at pr. 0.7 MPa is

$V_1 = xv_g = 0.95 \times 0.273$

$= 0.259 \ m^3/kg$

a) Steam volume passing/sec

$= \dfrac{\pi}{4}d^2 l$

$= \dfrac{\pi}{4} \times \left(25 \times 10^{-3}\right)^2 \times 12$

Steam volume passing /

$h_r = \dfrac{\pi}{4} \times \left(0.025\right)^2 \times 12 \times 3600$

∴ mass of steam entering / hr

$= \dfrac{\dfrac{\pi}{4} \times \left(0.025\right)^2 \times 12 \times 3600}{0.259}$

$= 81.9 \ kg/hr$

b) Specific enthalpy of steam entering heater is

$h_2 = h_f + xh_{fg} = 697.1 + 0.95 \times 2064.9$

$= 697.1 + 1961.7$

$= 2658.8 \ KJ/kg$

From the steam table at 90^0C

$h_f = 376.8 KJ/kg$

And at 19^0C

$h_f = 79.8 KJ/kg$

So for the heater applying energy balance equation

Enthalpy gained by water = Enthalpy lost by steam

$\dot{m}\left(376.8 - 79.8\right) = 81.9(2658.8 - 376.8)$

$\dot{m} = \dfrac{81.9 \times 2282}{297}$

$\dot{m} = 629.28 \ kg/hr.$

Q.3 1.5 kg of steam originally at pressure of 1 MPa and temperature 225^0C is expended until the pressure becomes 0.28 MPa. The dryness fraction at the steam is 0.9. Determine the change of internal energy.

Solution:

At 1 MPa and 225^0C, from steam table

$h_1 = 2886 KJ/kg, v_1 = 0.2198 m^3/kg$

So $u_1 = h_1 - p_1 v_1 = 2886 - \dfrac{1 \times 10^6}{10^3} \times 0.2198$

$= \ 2886 - 219.8$

$= \ 2566.2 \ KJ/kg$

At 0.28 MPa and dryness fraction 0.9

$h_2 = h_{f2} + xh_{fg2}$

$= \ 551.4 + 0.9 \times 2170.1$

$= \ 551.4 + 1953.09$

$= \ 2504.49 \ KJ/kg$

Volume

$v_2 = xv_{g2} = 0.9 \times 0.646$

$= 0.581 \ m^3/kg$

∴ Internal energy $u_2 = h_2 - p_2 v_2$

$= 2504.49 - \left(0.28 \times 10^3 \times 0.5814\right)$

$= \ 2504.49 - 162.8$

$= \ 2341.69 \ KJ/kg$

Hence change of internal energy

$U_2 - U_1 = m\left(u_2 - u_1\right)$

$= 1.5 \times \left(2341.69 - 2566.2\right)$

$= \ -489.77 \ KJ$

Q.4 The dryness fraction of steam at a pressure of 2.2 MPa is measured using throttling calorimeter. After throttling, the pressure in the calorimeter is 0.13 MPa and the temperature is 112^0C. Determine

the dryness fraction of steam at 2.2 MPa.

Solution: -

From the steam table

Pr MPa	Sat.temp (°C)	Specific enthalpy kJ/kg		
		h_f	H_{fg}	hg
2.2	217.2	931	1870	2801

Property of Superheated steam

P_r MPa	Sat.temp (°C)	Specific enthalpy kJ/kg	
		hg	At 150°C
0.1	99.6	2675	2777
0.5	111.4	2693	2773

Enthalpy of this stem after throttling can be found with help of linear interpolation

	P_r	Sat. temp	hg	At 150°C
Difference	0.15	114.4	2693	2773
Require	0.10	96.6	2675	2777
	0.05	11.8	18	-4
	0.03	11.8 X	18 X	-4 X
		0.03	0.03	0.03
		0.05	0.05	0.05
		= −2.4	= −2.4	= −2.4
Adding	0.13	106.68	2685.8	2779.6

h_2 at the temp. of 112°C can be found by linear interpolation

$$h_2 = h_g + \left(h_{150} - h_g\right)\frac{T_{sup} - T_{sat}}{T_{sup} - T_{sat}}$$

=2685.2+(2774.6-2685.8)

$$\frac{(112-106.68)}{(150-106.68)}$$

= 2685.2 + 10.9 = 2696.1 KJ/kg

Now for the wet steam before steam throttling

$h_1 = h_{f1} + x_1 h_{fg1}$

= 931 + x_1 X 1870 KJ/kg

For the throttling process

$h_1 = h_2$

931+(x_1×1870) =2696.1

$$x_1 = \frac{1765.1}{1870}$$

$x_1 = 0.94$

Q.5 Steams at 1.4 MPa and dryness fraction 0.7 is throttled to 0.11 MPa.

Determine the dryness fraction of steam after throttling.

Solution:

For the throttling process

$h_1 = h_2$

$h_{f1} + x_1 h_{fg1} = h_{f2} + x_2 h_{fg2}$

830.1+0.7×1957.7=928.8+X₂+2250.8

$$x_2 = \frac{1771.7}{2250.8}$$

$x_2 = 0.787$

➤ The steam becomes drier in this case.

Q.6 a) Determine the volume occupied by 1 kg of steam at a pr. Of 0.85 MPa and having a dryness fraction of 0.97 .

b) This volume is expanded adiabatically to a pressure of 0.17 MPa, law of expansion is PV^1.3 = C. Determine

i) Final dryness fraction of the steam

ii) Change of internal energy of system during the expansion.

Solution:

Properties of system from steam table

At 0.85 MPa , v_g = 0.2268 m³/kg

a) So

$v_1 = x_1 v_{g1} = 0.97 \times 0.2268 = 0.22 m^3/kg$

b) (i) Final dryness fraction

$$P_1 V_1^{1.3} = P_2 V_2^{1.3}$$

$$V_2 = 0.22\left(\frac{0.85}{0.17}\right)^{\frac{1}{1.13}}$$

$$V_2 = 0.22 \times 4.15$$

= 0.913 m³/kg

At 0.17 MPa ,

v_{g2} = 1.031 m³/kg

The steam is wet, so dryness fraction is

$$x_2 = \frac{u_2}{u_{g2}} = \frac{0.913}{1.031} = 0.886$$

(ii) For the adiabatic expansion

δQ = 0

So change of internal energy

$$U_2 - U_1 = -W$$

$$U_2 - U_1 = -\frac{(P_1V_1 - P_2V_2)}{n-1}$$

$$= -\frac{(0.85 \times 0.22 - 0.17 \times 0.913) \times 10^3}{1.13 - 1}$$

$$= -246 \text{ KJ/kg}$$

There is a loss of internal energy.

Q.7 A mixture of ideal gases consisting of 3 kg of nitrogen and 5 kg of carbon monoxide at a pressure of 300 KPa and temperature of 20^0C. Find (a) the mole fraction of constituent (b) Equivalent molecular weight of the mixture (c) Equivalent gas constant of the mixture, (d) the specific heat at constant pressure and volume of the mixture. Take $\gamma = 1.286 \text{ for CO}_2$ and $\gamma = 1.4 \text{ for N}_2$.

Solution:

let the gas CO_2 as the gas 1 and gas N_2 as the gas 2 is mixture.

The mole fraction for CO_2

$$x_1 = \frac{n_1}{\sum n} \text{ and } n_1 = \frac{m_1}{M_1}$$

$$x_1 = \frac{\dfrac{m_1}{M_1}}{\dfrac{m_1}{M_1} + \dfrac{m_2}{M_2}} = \frac{\dfrac{5}{44}}{\dfrac{5}{44} + \dfrac{3}{28}} = 0.52$$

Similarly the mole fraction for N_2

$$x_1 = \frac{\dfrac{m_2}{M_2}}{\dfrac{m_1}{M_1} + \dfrac{m_2}{M_2}} = \frac{\dfrac{3}{28}}{\dfrac{5}{44} + \dfrac{3}{28}} = 0.48$$

(b) Equivalent molar weight of the mixture

$M_e = x_1M_1 + x_2M_2$

$= 0.52 \times 44 + 0.48 \times 28 = 36.32$ Kg/Kg mole

(c).Equivalent gas constant of the mixture gas constant of gas 1

$$R_1 = \frac{R}{M_1} = \frac{8.314}{44} = 0.19 \text{ KJ/Kg.K}$$

$$R_1 = \frac{\overline{R}}{M_2} = \frac{8.314}{28} = 0.30 \text{ KJ/Kg.K}$$

$$Re = \frac{m_1R_1 + m_2R_2}{m_1 + m_2} = \frac{44 \times 0.19 + 28 \times 0.30}{44 + 28} = 0.22 \text{ KJ/Kg.K}$$

(d) The Equivalent specific heat at constant pressure

$$C_{v1} = \frac{R_1}{\gamma - 1} = \frac{0.19}{1.286 - 1} = 0.66 \text{ KJ/Kg.K}$$

$$C_{P1} = \gamma C_{V1} = 1.286 \times 0.66$$

$$= 0.85 \text{ KJ/Kg.K}$$

$$C_{V2} = \frac{R_2}{\gamma - 1} = \frac{0.30}{1.4 - 1} = 0.75 \text{ KJ/Kg.K}$$

$$C_{P2} = \gamma C_{V2} = 1.4 \times 0.75$$

$$= 1.05 \text{ KJ/Kg.K}$$

The equivalent specific heat of mixture at constant pressure

$$C_{pe} = \frac{m_1C_{p1} + m_2C_{p2}}{m_1 + m_2}$$

$$= \frac{44 \times 0.85 + 28 \times 1.05}{44 + 28} = 0.928 \text{ KJ/Kg.K}$$

The equivalent specific heat of mixture at constant volume

$$C_{ve} = \frac{m_1C_{V1} + m_2C_{V2}}{m_1 + m_2}$$

$$= \frac{44 \times 0.66 + 28 \times 0.75}{44 + 28}$$

$$= 0.695 \text{ KJ/Kg.K}$$

Q.8 from an experimental determination the specific heat ratio for acetylene (C_2H_2) is found to 1.26. Find the two specific heats. If heat is supplied to increase the temperature from 30^0 C to 60^0 C at constant pressure then find the increase in the enthalpy and entropy of the system.

Solution:

Gas constant of acetylene

$$(C_2H_2)(R) = \frac{\overline{R}}{M} = \frac{8.3143}{26} \text{ kJ/kgK}$$

= 0.3198 kJ/kgK

As adiabatic index (γ) = 1.26, then

We know that $c_p = \dfrac{\gamma}{\gamma-1}R$

$= \dfrac{1.26}{1.26-1} \times 0.3198 = 1.55 \dfrac{KJ}{Kg}K$

And $\quad c_v = \dfrac{R}{\gamma-1}$

$= \dfrac{0.3196}{1.26-1} = 1.23 KJ/kgK$

Change in enthalpy is given by

$h_2 - h_1 = c_p(T_2 - T_1)$

$= 1.55 \times 30 = 4.65 \dfrac{KJ}{Kg}$

Change in enthalpy is given by

$S_2 - S_1 = c_p \ln\left(\dfrac{T2}{T1}\right) - R \ln\left(\dfrac{P2}{P1}\right)$

$= c_p \ln\left(\dfrac{T2}{T1}\right) - 0$

$= C_p \ln\left(\dfrac{T2}{T1}\right)$

$= 1.55 \times \ln\left(\dfrac{333}{303}\right)$

= 0.146 KJ/Kg.K

$$\left(W_{12}\right) = \int_1^2 pdV = p\left(V_2 - V_1\right) = 27.266 kJ$$

(b) Heat transferred

$Q_{1-2} = u_2 - u_1 + W_{12}$

$= mc_v\left(T_2 - T_1\right) + W_{1-2} = 95.476 kJ$

(c) Entropy change of the gas

$S_2 - S_1 = mc_p \ln\dfrac{V_2}{V_1} + mc_v \ln\dfrac{p_2}{p_1}$

$= mc_p \ln\dfrac{V_2}{V_1} = 0.29543\,\dfrac{kJ}{kg}.K$

Q.9 One kg of air in a closed system, initially at 5⁰C and occupying 0.3 m³ volume, undergoes a constant pressure heating process to 100⁰C. There is no work other than Pdv work. Find (a) the work done during the process, (b) the heat transferred, and (c) the entropy change of the gas.

Solutions:

$T_1 = 278$ K

$V_1 = 0.3$ m³

$M = 1$ kg

$\therefore P_1 = 265.95$ kP$_a$

$T_2 = 100^0C = 373$ K

$P_2 = 265.95$ kPa

$\therefore V_2 = \dfrac{mRT_2}{P_2} = 0.40252 m^3$

(a) Work during the process

6 REFRIGERATION

6.1 INTRODUCTION

In generally, refrigeration is defined as any process of heat removal. Refrigeration is defined as branch of Science that deals with the process of reducing and maintaining temperature of a space or material below the temperature of the surrounding.

We see in nature that the heat spontaneously flows of heat from a high temperature body to a low temperature body. The reverse process to complete the thermodynamic cycle, in which heat Q will flow back from low temperature body to high temperature body, is not possible. So the logical conduction is that there must be a process in which some work is done.

6.1.1 NEED OF THERMAL INSULATION

Since heat well always flow from a region of high temperature to a region of lower temperature. There is always a continuous flow of heat into the refrigerated region from the warmer surroundings. To limit the heat flow into refrigerated region, it is usually necessary to isolate the region from its surrounding with a good insulation.

6.1.2 REFRIGERATION LOAD

The rate at which heat must be removed from refrigerated space or material in order to produce and maintain the desired temperature condition is called as refrigeration load or cooling load.

The total refrigeration load is the sum of heat gains from different sources:-
(1) Heat transmitted by conduction through the insulated walls.
(2) Heat that must be removed from warm air that enters the space through openings and closing doors.
(3) Heat that must be removed from refrigerated product to reduce the temperature of products to the storage temperature.

6.1.3 REFRIGERANT

In the refrigeration process, the substance employed as the heat absorber or coaling agent is called the refrigerant. When the absorbed heat causes an increase in temperature of refrigerant, cooling process is said to be sensible, whereas when the absorbed heat causes a change in physical state of refrigerant, cooling effect is said to be latent. Ice has been used successfully for many years as a refrigerant. Not to many years ago ice was the only cooling agent available for use in domestic and small commercial refrigerator.

6.1.4 UNIT OF REFRIGERATION

It is given by Tonnes of Refrigeration (TR). 1TR is defined as refrigeration effect produced by uniform melting of 1000 kg of ice from and at 0^0C in 24 hours.

1 TR = 1000×334 KJ refrigeration effect in

$$24 \text{ hours} = \frac{1000 \times 334}{24 \times 3600} = 3.86 \text{ KJ / Sec.}$$

➢ For practical problem it is taken as 3.5 KJ/Sec or 210 KJ/Minute.

6.1.5 CO EFFICIENT OF PERFORMANCE

The co efficient of performance (COP) of refrigerator is ratio of heat extracted in the refrigerator to the work done on the refrigerant working for the refrigeration cycle.

$$COP = \frac{\text{Heat extracting from refrigerated space}}{\text{Work Done}} = \frac{Q}{W}$$

6.2 AIR STANDARD REFRIGERATION CYCLE

Air cycle refrigeration systems belong to the general class of gas cycle refrigeration systems, in which a gas is used as the working fluid. The gas does not undergo

any phase change during the cycle; consequently, all the internal heat transfer processes are sensible heat transfer processes.

➢ Gas cycle refrigeration systems find applications in air craft cabin cooling and
➢ It also in the liquefaction of various gases.

6.2.1 AIR STANDARD CYCLE ANALYSIS

Air cycle refrigeration system analysis is considerably simplified if one makes the following assumptions:

i. The working fluid is a fixed mass of air that behaves as an ideal gas
ii. All the processes within the cycle are reversible, i.e., the cycle is internally reversible.
iv. The specific heat of air remains constant throughout the cycle.

First of all we study the ideal refrigeration cycle i.e. Reversed Carnot Cycle.

6.2.2 A REFRIGERATOR WORKING ON REVERSED CARNOT CYCLE:

Reversing the Carnot cycle does reverse the directions of heat and work interactions. A refrigerator or heat pump that operates on the reversed Carnot cycle is called a Carnot refrigerator or a Carnot heat pump. Reversed Carnot cycle is shown in Fig. It consists of the following processes.
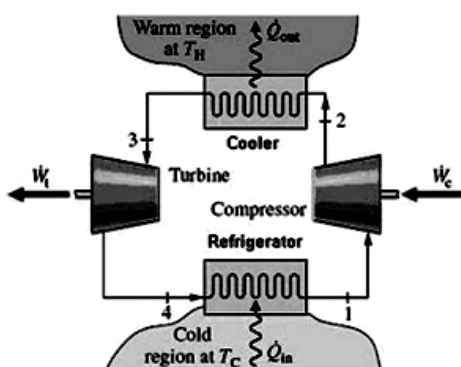

Fig. a refrigerator working on reversed Carnot Cycle

Process 4-1: Absorption of heat by the working fluid from refrigerator at constant

low temperature T_L during isothermal expansion.

Process 1-2: Isentropic compression of the working fluid with the aid of external work. The temperature of the fluid rises from T_L to T_H.

Process 2-3: Isothermal compression of the working fluid during which heat is rejected at constant high temperature T_H.

Process 3-4: Isentropic expansion of the working fluid. The temperature of the working fluid falls from T_H to T_L.
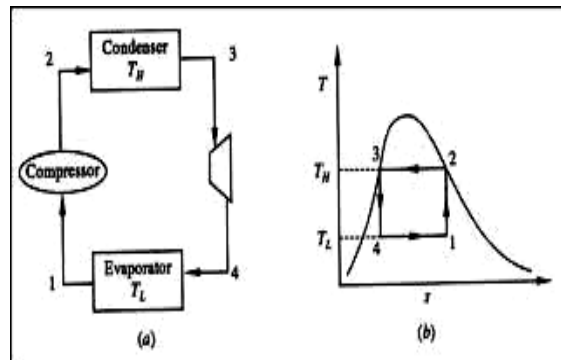

Fig. P-V and T-S Diagram of Reversed Carnot Cycle

➢ The reversed Carnot cycle is the most efficient refrigeration cycle operating between two specified temperature levels. It sets the highest theoretical COP. The coefficient of performance for Carnot refrigerators and heat pumps are:

COP of Refrigerator

$$COP_R = \frac{\text{Heat absorbed}}{\text{Work supplied}} =$$

$$\frac{\text{Heat absorbed}}{\text{Heat rejected} - \text{Heat absorbed}}$$

$$= \frac{Q_L}{Q_{H-Q_L}} = \frac{T_L(dS)}{T_{H(dS)-T_L(dS)}}$$

So,

$$COP_R = \frac{Q_L}{Q_H - Q_L} = \frac{T_L}{T_H - T_L} \text{ -----(7.1)}$$

COP of Heat Pump

$$COP_{HP} = \frac{\text{Heat Rejected}}{\text{Work supplied}}$$

$$= \frac{\text{Heat Rejected}}{\text{Heat rejected} - \text{Heat absorbed}}$$

$$= \frac{Q_H}{Q_{H-Q_L}} = \frac{T_H(dS)}{T_{H(dS)-T_L(dS)}}$$

So $COP_{HP} = \dfrac{Q_H}{Q_H - Q_L} = \dfrac{T_H}{T_H - T_L}$ --(7.2)

From the equation 7.1 and 7.2

$COP_{HP} = COP_R + 1$ ------------ (7.3)

➢ Notice that turbine is used for expansion process between higher and lower temperature limit. Work done by the turbine helps supply some of the work required by the compressor from the external source.

➢ Practically, the reversed Carnot cycle cannot be used for refrigeration purpose as the isentropic process requires very high speed operation, whereas the isothermal process requires very low speed operation.

6.3 REFRIGERATOR WORKING ON REVERSED BRAY TON CYCLE

This is an important cycle frequently employed in gas cycle refrigeration systems. This may be thought of as a modification of reversed Carnot cycle, as the two isothermal processes of Carnot cycle are replaced by two isobaric heat transfer processes. This cycle is also called as Joule or Bell-Coleman cycle. Figure shows the schematic of a closed reverse Bray ton cycle and also the cycle on T-s diagram.
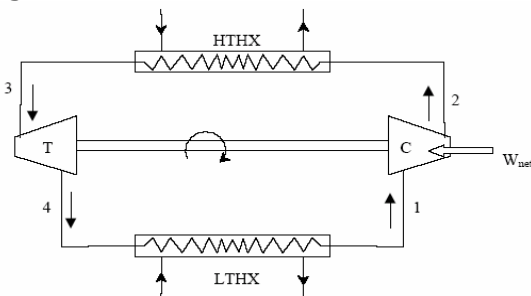

Fig. Reversed Bray ton Refrigeration System

There are four important processes in reversed Bray ton Cycle:

Process 1-2: Reversible adiabatic compression in a compressor

Process 2-3: Reversible isobaric heat rejection in a heat exchanger

Process 3-4: Reversible adiabatic expansion in a turbine

Process 4-1: Reversible isobaric heat absorption in a heat exchanger

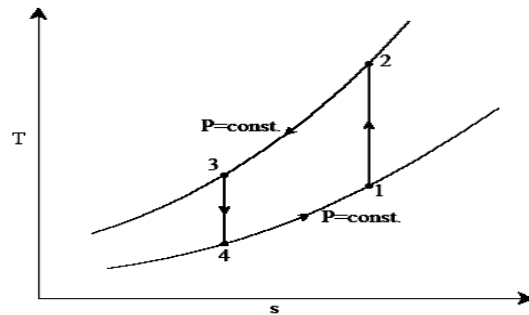All the Processes are on P-V and T-S diagram.

Fig. Reversed Bray ton cycle on T-S diagram

Process 1-2: Gas at low pressure is compressed isentropic ally from state 1 to state 2.

Applying steady flow energy equation and neglecting changes in kinetic and potential energy,

W_{1-2} = m (h_2-h_1) = m. Cp. (T_2-T_1)

But process 1-2 is reversible adiabatic Process or isentropic process. So,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(r_p\right)^{\frac{\gamma-1}{\gamma}} ----(7.4)$$

Where r_p = (P_2/P_1) = pressure ratio

Process 2-3: Hot and high pressure gas flows through a heat exchanger and rejects heat sensibly and isobaric to a heat sink. The enthalpy and temperature of the gas drop during the process due to heat exchange, no work transfer takes place and the entropy of the gas decreases.

Q_{2-3} = m (h_2-h_3) = m. Cp. (T_2-T_3)

Process 3-4: High pressure gas from the heat exchanger flows through a turbine undergoes isentropic expansion and

delivers net work output. The temperature of the gas drops during the process from T_3 to T_4. From steady flow energy equation:

$W_{3-4} = m (h_3-h_4) = m. Cp. (T_3-T_4)$

But process 3-4 is reversible adiabatic Process or isentropic process. So,

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(r_p\right)^{\frac{\gamma-1}{\gamma}} \text{------------ (7.5)}$$

Where $r_p = (P_3/P_4)$ = pressure ratio

So, $\dfrac{T_3}{T_4} = \dfrac{T_2}{T_1}$

Process 4-1: Cold and low pressure gas from turbine flows through the low temperature heat exchanger and extracts heat sensibly and isobaric ally from a heat source, providing a useful refrigeration effect. The enthalpy and temperature of the gas rise during the process due to heat exchange, no work transfer takes place and the entropy of the gas increases. The refrigerating effect by the refrigerant can be given as:

6.3.1 REFRIGERATING EFFECT

$Q_{4-1} = m (h_1-h_4) = m. Cp. (T_1-T_4)$

COP of reversed Bray ton cycle can be given:

$\text{COP} = \left(\text{Refrigerating Effect}\right)/\left(\text{NetWork Done}\right)$

$$= \frac{Q_{4-1}}{W_{1-2} - W_{3-4}}$$

$$= \frac{m.c_{p.}(T_{1-}T_4)}{m.c_{p.}(T_{2-}T_1) - m.c_{p.}(T_{3-}T_4)}$$

$$= \frac{(T_{1-}T_4)}{(T_{2-}T_3) - (T_{1-}T_4)}$$

Solving the equation

$$= \frac{T_1\left(1 - \dfrac{T_4}{T_1}\right)}{T_2\left(1 - \dfrac{T_3}{T_2}\right) - T_1\left(1 - \dfrac{T_4}{T_1}\right)}$$

$$= \frac{T_1}{T_2 - T_1} = \frac{1}{\dfrac{T_2}{T_1} - 1} = \frac{1}{\left(r_p\right)^{\frac{\gamma-1}{\gamma}} - 1}$$

So, $\text{COP} = \dfrac{1}{\left(r_p\right)^{\frac{\gamma-1}{\gamma}} - 1}$

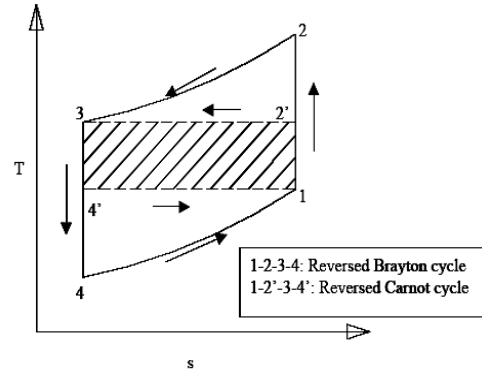From the above expression for COP, the following observations can be made:

Fig. Comparison between Bray ton Cycle and Carnot Cycle.

➢ For fixed heat rejection temperature (T_3) and fixed refrigeration temperature (T_1), the COP of reverse Bray ton cycle is always lower than the COP of reverse Carnot cycle as shown in fig.

➢ COP of Bray ton cycle approaches COP of Carnot cycle as T_1 approaches T_4 (thin cycle), however, the specific refrigeration effect $[c_p(T_1-T_4)]$ also reduces simultaneously.

➢ COP of reverse Bray ton cycle decreases as the pressure ratio r_p increases.

6.3.2 ACTUAL REVERSE BRAY TON CYCLE

The actual reverse Bray ton cycle differs from the ideal cycle due to:

➢ Non-isentropic compression and expansion processes.
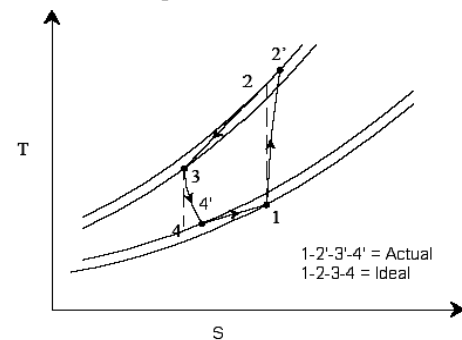
➢ Pressure drops in cold & hot heat exchangers.

Fig. Actual Reversed Bray ton Cycle

Figure shows the ideal and actual cycles on T-s diagram. Due to this irreversibility, the compressor work input increases and turbine work output reduces.

If $\eta_{compressor\,,isen}$ and $\eta_{turbine\,,isen}$ are the isentropic efficiencies of compressor and turbine, respectively, these are defined as:

$$\eta_{Copm.} = \frac{h_2 - h_1}{h_2' - h_1} = \frac{T_2 - T_1}{T_2' - T_1}$$

$$\eta_{Turbine} = \frac{h_3 - h_4}{h_3 - h_4'} = \frac{T_3 - T_4}{T_3 - T_4'}$$

Thus the net work input increases due to increase in compressor work input and reduction in turbine work output. The refrigeration effect also reduces due to the irreversibility's. As a result, the COP of actual reverse Bray ton cycles will be considerably lower than the ideal cycles. Design of efficient compressors and turbines plays a major role in improving the COP of the system.

In practice, reverse Bray ton cycles can be open or closed.

6.3.3 AIRCRAFT COOLING SYSTEMS

An aircraft, cooling systems are required to keep the cabin temperatures at a comfortable level. Even though the outside temperatures are very low at high altitudes, still cooling of cabin is required due to:

➤ Large internal heat generation due to occupants, equipment etc.
➤ Heat generation due to skin friction caused by the fast moving aircraft
➤ At high altitudes, the outside pressure will be sub-atmospheric. When air at this low pressure is compressed and supplied to the cabin at pressures close to atmospheric, the temperature increases significantly.
➤ Solar radiation

6.3.4 SIMPLE AIRCRAFT REFRIGERATION CYCLE

Figure shows the schematic of a simple aircraft refrigeration system and the operating cycle on T-S diagram. This is an open system. As shown in the T-S diagram, the outside low pressure and low temperature air is compressed due to ram effect to ram pressure in the process 1-2. During this process its temperature increases from T_1 to T_2. This air is compressed in the main compressor in process 2-3, and is cooled to temperature T_4 in the air cooler. Its pressure is reduced to cabin pressure in the turbine, as a result its temperature drops from T_4 to T_5. The cold air at state T_5 is supplied to the cabin. It picks up heat as it flows through the cabin providing useful cooling effect. The power output of the turbine is used to drive the fan, which maintains the required air flow over the air cooler.

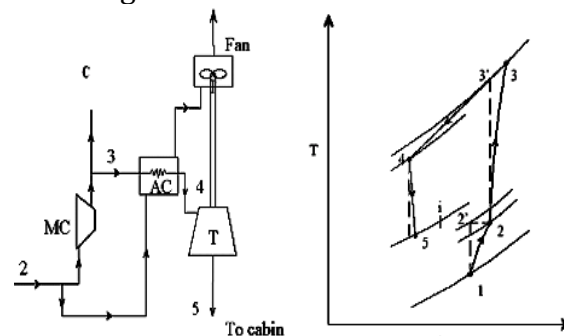➤ This simple system is good for ground cooling.

fig: Simple air craft refrigeration system

$$\frac{T_2'}{T_1} = 1 + \frac{\gamma - 1}{2}M^2$$

Where M is the Mach number, which is the ratio of velocity of the aircraft (V) to the sonic velocity C

Mach number: $M = \dfrac{V}{C} = \left(\dfrac{V}{\sqrt{\gamma R T_1}}\right)$

6.4 VAPOUR COMPRESSION REFRIGERATION CYCLE

Vapour compression cycle is an improved type of air refrigeration cycle in which a suitable working substance, termed as refrigerant, is used. The refrigerant used, does not leave the system, but is circulated throughout the system alternately condensing and evaporating.

In evaporating, the refrigerant absorbs its latent heat from the solution which is used for circulating it around the cold chamber. In condensing, it gives out its latent heat to the circulating water of the cooler.

Vapour compression refrigeration system is now-a-days used for all purpose refrigeration. It is used for all industrial purposes from a small domestic refrigerator to a big air conditioning plant.

6.4.1 SIMPLE VAPOUR COMPRESSION REFRIGERATION SYSTEM
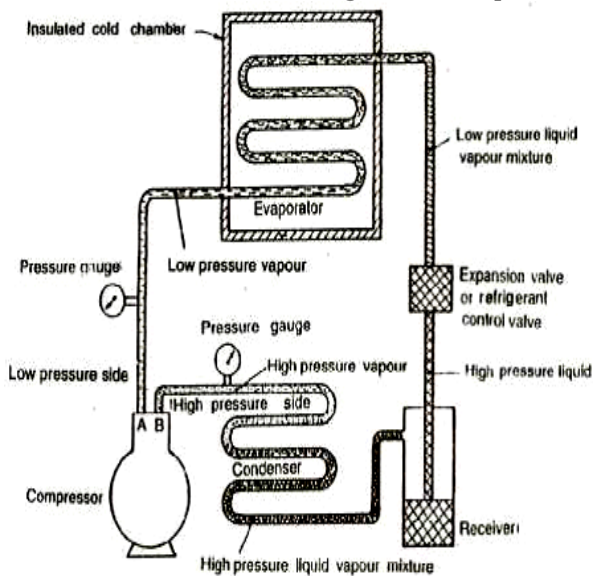
It consists of the following essential parts:

Fig. Simple vapour compression refrigeration system

Compressor
The low pressure and temperature vapour refrigerant from evaporator is drawn into the compressor through the suction valve A, where it is compressed to a high pressure and temperature. This high pressure & temperature vapour refrigerant is discharged into the condenser through the discharge valve B.

Condenser
The condenser consists of coils of pipe in which the high pressure and temperature vapour refrigerant is cooled and condensed. The refrigerant, while passing through the condenser, gives up its latent heat to the surrounding condensing medium which is normally air or water.

Receiver
The condensed liquid refrigerant from the condenser is stored in a vessel known as receiver from where it is supplied to the evaporator through the expansion valve or refrigerant control valve.

Expansion Valve

It is also called throttle valve or refrigerant control valve. The function of the expansion valve is to allow the liquid refrigerant under high pressure and temperature to pass at a controlled rate after reducing its pressure and temperature. Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporized in the evaporator at the low pressure & temperature

Evaporator
An evaporator consists of coils of pipe in which the liquid-vapour. refrigerant at low pressure and temperature is evaporated and changed into vapour refrigerant. In evaporating, the liquid vapour refrigerant absorbs its latent heat of vaporization from the medium which is to be cooled.

6.4.2 THEORETICAL VAPOUR COMPRESSION CYCLE

A vapour compression cycle with dry saturated vapour after compression is shown on T-S and P-h diagrams as shown in Figures. At point 1, let the temperature, pressure and entropy of the vapour refrigerant is T1, p1 and S1 respectively. The four processes of the cycle are as follows:
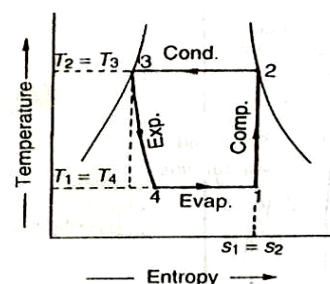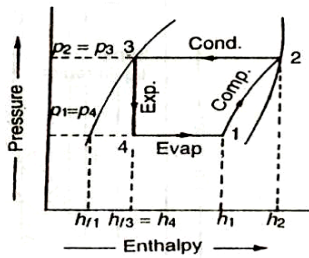
Fig: VCR cycle on T-s and P-h diagram

Process 1-2-Compression Process

The vapour refrigerant at low pressure P1 and temperature T1 is compressed isentropic ally to dry saturated vapour as shown by the vertical line 1-2 on the T-s diagram and by the curve 1-2 on p-h diagram. The pressure and temperature rise from P1 to P2 and T1 to T2 respectively. The work done during isentropic compression per kg of refrigerant is given by
w = h2 – h1
Where h1 = Enthalpy of vapour refrigerant at temperature T1, i.e. at suction of the compressor, and h2=Enthalpy of the vapour refrigerant at temperature T2. i.e. at discharge of compressor.

Process 2-3- Condensing Process

The high pressure and temperature vapour refrigerant from the compressor is passed through the condenser where it is completely condensed at constant pressure P_2 and temperature T2 as shown by the horizontal line 2-3 on T-s and p-h diagrams. The vapour refrigerant is changed into liquid refrigerant. The refrigerant, while passing through the condenser, gives its latent heat to the surrounding.

Process 3-4 - Expansion Process

The liquid refrigerant at pressure $P_3 = P_2$ and temperature T3 = T2, is expanded by throttling process (Isenthalpic Process) through the expansion valve to a low pressure P4 = P_1 and Temperature T4 = T1 as shown by the curve 3-4 on T-s diagram and by the vertical line 3-4 on P-h diagram.

Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporized in the evaporator. We know that during the throttling process, no heat is absorbed or rejected by the liquid refrigerant.

Process 4-1-Vaporizing Process

The liquid-vapour mixture of the refrigerant at pressure $P_4=P_1$ and temperature T4=T1 is evaporated and changed into vapour refrigerant at constant pressure and temperature, as shown by the horizontal line 4-1 on T-s and p-h diagrams. During evaporation, the liquid-vapour refrigerant absorbs its latent heat of vaporization. The heat which is absorbed by the refrigerant is called refrigerating effect and it is briefly written as RE.
The refrigerating effect or the heat absorbed or extracted by the refrigerant during evaporation per kg of refrigerant is given by RE = $h_1 – h_4 = h_1 – h_{f3}$
Where h_{f3} = Sensible heat at temperature T3, i.e. enthalpy of liquid refrigerant leaving the condenser.

CO Efficient of Performance

Coefficient of performance, C.O.P. = (Refrigerating effect)/ (Work done)

$$\text{O.P} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

6.5 EFFECT OF PARAMETERS ON COP OF VAPOUR COMPRESSION REFRIGERATION CYCLE

1. Effect of Suction Pressure
The suction or evaporator pressure decreases due to the frictional resistance of flow of the refrigerant. Let us consider a theoretical vapour compression cycle 1-2-3-4 when the suction pressure decreases from P_s to P_s' as shown on p-h diagram in Figure.
It may be noted that the decrease in suction pressure:

a) Decreases the refrigerating effect from $(h_1 - h_4)$ to $(h_1^1 - h_4)$.

 b) Increases the work required for compression from $(h_2 - h_1)$ to $(h_2^1 - h_1^1)$.
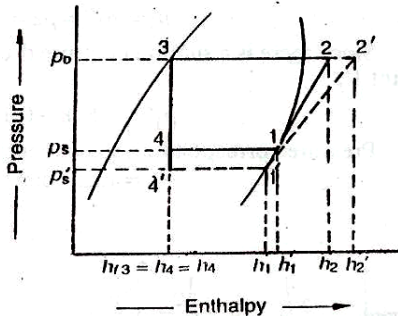

fig. Effect of suction pressure

Since the C.O.P, of the system is the ratio of refrigerating effect to the work done, therefore with the decrease in suction pressure, the net effect is to decrease the C.O.P. of the refrigerating system for the same refrigerant flow. Hence with the decrease in suction pressure

➤ The refrigerating capacity of the system decreases.
➤ The refrigeration cost increases.

2. Effect of Discharge Pressure

In actual practice, the discharge or condenser pressure increases due to frictional resistance of flow of the refrigerant. Let us consider a theoretical vapour compression cycle l-2-3-4 when the discharge pressure increases from P_D to P_D^1 as shown on p-h diagram in Figure resulting in increased compressor work from $(h_2 - h_1)$ to $(h_2^1 - h_1)$ and decreasing refrigeration effect from $(h_1 - h_4)$ to $(h_1 - h_4^1)$. So COP is ratio of refrigeration effect to work done. So COP of the cycle decreases.
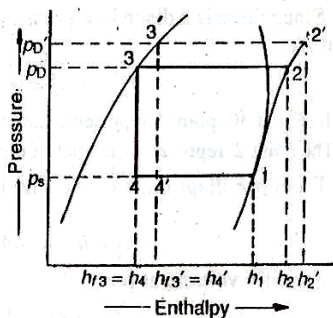

fig. Effect of suction pressure

6.5.1 EFFECT OF LIQUID SUB COOLING

It is possible to reduce the temperature of the liquid refrigerant to within a few degrees of the temperature of the water entering the condenser. In some condenser designs it is achieved by installing a sub-cooler between the condenser and the expansion valve. The effect of sub-cooling of the liquid from $t_3 = t_k$ to t_3^1 is shown in Figure. It will be seen that sub-cooling reduces flashing of the liquid during expansion and increases the refrigerating effect. But there is no effect on the work done by the compressor. And the COP of the cycle is ratio of refrigeration effect to work done by the compressor. So the COP of the cycle increases in sub-cooling of the liquid.
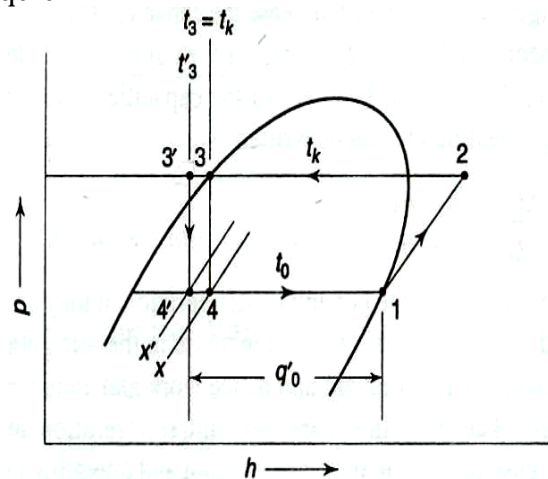

fig. Effect of suction pressure

6.6 VAPOUR ABSORPTION REFRIGERATION SYSTEMS

The Refrigeration effect by Vapour Compression Refrigeration System is an efficient method. But the input energy given in the VCR system is work i.e. high grade energy and therefore very expensive. So In Vapour Absorption Refrigeration Systems, the mechanical work is replaced by the heat. Hence these systems are also called as heat operated or thermal energy driven systems. Since these systems run on low-grade thermal energy, they are preferred when low-grade energy such as waste heat or solar energy is available.

Since conventional absorption systems use natural refrigerants such as water or ammonia they are environment friendly.

6.6.1 WORKING OF SIMPLE VAPOUR ABSORPTION REFRIGERATION CYCLE

The systematic diagram of VAR system is shown. In this, compressor is replaced by absorber system (i.e. Absorber, Pump, Heat Exchanger Heat generator and rectifier). Working of the system is almost same as the working of VCR system excepting the functionality of compressor. The working can be explained with help of sketch diagram.

The low temperature refrigerant enters the evaporator and absorbs the required heat from the evaporator and leaves the evaporator as saturated vapour. Slightly low pressure NH_3 vapour is absorbed by the weak solution of NH_3 which is sprayed in the absorber as shown in fig.
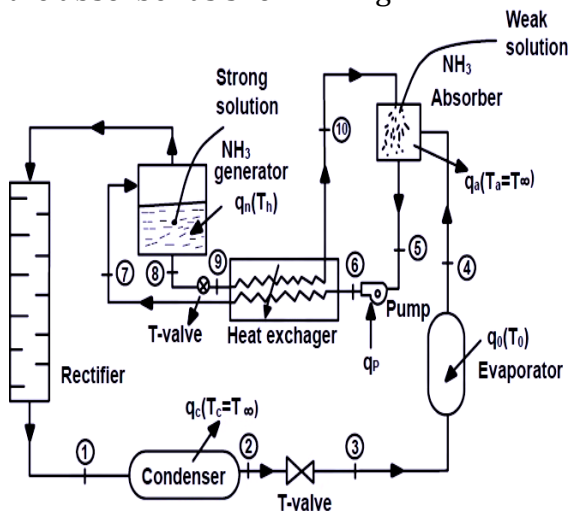

Fig. Vapour Absorption Refrigeration System

Weak NH_3 solution (i.e. know as aqua–ammonia) enters in to absorber and becomes strong solution after absorbing NH_3 vapour. Then it is pumped to the generator through the heat exchanger. The pump increases the pressure of the strong solution.

The strong NH_3 solution absorbs heat form high temperature weak NH_3 solution in the heat exchanger. The weak high temperature ammonia solution from the generator is passed to the heat exchanger through the pressure reducing valve. The pressure of the liquid is reduced to the absorber pressure by the throttle valve.

Ammonia vapour is produced in the generator at high pressure by a heating generator. The water vapour carried with ammonia is removed in the rectifier and only the dehydrated ammonia gas enters into the condenser. High pressure NH_3 vapour is condensed in the condenser. The cooled NH_3 solution is passed through a throttle valve and the pressure and temperature of the refrigerant are reduced below the temperature to be maintained in the evaporator. The refrigerant absorbs the heat from the medium and converts in to the vapour phase and again it goes to generator.

6.6.2 COP OF IDEAL VAR CYCLE

The working of VAR system is shown in fig. Let us consider the Heat Q_E is absorbed by the refrigerant in absorber at temperature of T_E. In the heating generator, Q_G amount of heat is added at temperature T_G to the refrigerant. The Q_C amount heat is rejected in condenser at temperature of T_C. Pump work is W_P.

The COP of system is given as

$$(COP)_{VAR} = \frac{\text{Refrigeration Effect}}{\text{Given input}}$$

$$= \frac{Q_E}{Q_G + W_P} \approx \frac{Q_E}{Q_G}$$

Considering a reversible refrigeration cycle,

$$Q_C = Q_G + Q_E \quad \text{-----------(1)}$$

for a reversible cycle Clausius Inequality Law,

$$\oint \frac{\delta Q}{T} = 0, \frac{Q_C}{T_C} = \frac{Q_E}{T_E} + \frac{Q_G}{T_G}$$

Substituting the value from equation 1

$$\frac{Q_G + QE}{T_C} = \frac{Q_E}{T_E} + \frac{Q_G}{T_G}$$

Or, $\dfrac{Q_E}{T_C} - \dfrac{Q_E}{T_E} = \dfrac{Q_G}{T_G} - \dfrac{Q_G}{T_C}$

Or, $\dfrac{Q_E(T_C - T_E)}{T_C} = \dfrac{Q_G(T_G - T_C)}{T_G}$

Or, $\dfrac{Q_E}{Q_G} = \dfrac{(T_G - T_C)}{T_G} \times \dfrac{T_C}{(T_C - T_E)}$

$(COP)_{VAR} = \eta_{Carnot}\,(COP)_{Carnot}$

6.7 REFRIGERANT

Refrigerant is the fluid used for heat transfer in a refrigerating system that absorbs heat during evaporation from the region of low temperature, and releases heat during condensation at a region of higher temperature.

Due to several environmental issues such as ozone layer depletion and global warming and their relation to the various refrigerants used, the selection of suitable refrigerant has become one of the most important issues in recent times.

For Example, Air used in an air cycle refrigeration system can also be considered as a refrigerant.

6.7.1 CLASSIFICATION OF REFRIGERANT

There are two types of refrigerants
1. Primary Refrigerant
2. Secondary Refrigerant

1. Primary Refrigerant

Primary refrigerants are those fluids, which are used directly as working fluids, for example in vapour compression and vapour absorption refrigeration systems.

2. Secondary Refrigerant

Secondary refrigerants are those liquids, which are used for transporting thermal energy from one location to other. Secondary refrigerants are also known under the name brines or antifreezes. Generally, the freezing point of brine will be lower than the freezing point of its constituents. So these are used as Secondary refrigerants.

➢ If the operating temperatures are above 0^oC, then pure water can also be used as secondary refrigerant, The commonly used secondary refrigerants are:
➢ The solutions of water and ethylene glycol,
➢ The solutions of propylene glycol or calcium chloride.

6.7.2 PROPERTIES OF REFRIGERANT

There should be the following properties of the refrigerants:
a) Latent heat of vaporization:
it should be as large as possible so that the required mass flow rate per unit cooling capacity will be small.

b) Isentropic index of compression:
It should be as small as possible so that the temperature rise during compression will be small.

c) Vapour specific heat:
It should be large so that the degree of superheating will be small.

d) Thermal conductivity:
Thermal conductivity in both liquid as well as vapour phase should be high for higher heat transfer coefficients.

e) Viscosity:
Viscosity should be small in both liquid and vapour phases for smaller frictional pressure drops.

f) Ozone Depletion Potential (ODP):
According to the **Montreal protocol**, the ODP of refrigerants should be zero, i.e., they should be non-ozone depleting substances. Refrigerants having non-zero ODP have either already been phased-out (e.g. R11, R 12).

g) Global Warming Potential (GWP):
Refrigerants should have as low GWP value as possible to minimize the problem of global warming.

h) Toxicity

The refrigerants used in a refrigeration system should be non-toxic.

6.8 DESIGNATION OF REFRIGERANTS

There are following types of refrigerants. The designation of these refrigerants can be done by following manner:

1) Halocarbon compounds:
These refrigerants are derivatives of alkenes (C_nH_{2n+2}) such as methane (CH_4), ethane (C_2H_6). These refrigerants are designated by R-ABC
Where:
A+1 indicates the number of Carbon (C) atoms
B-1 indicates number of Hydrogen (H) atoms, and Z indicates number of Fluorine (F) atoms

Example: R- 22
A = 0 ⇒ No. of Carbon atoms = 0+1 = 1
B = 2 ⇒ No. of Hydrogen atoms = 2-1 = 1
C = 2 ⇒ No. of Fluorine atoms = 2
The balance = 4 – no. of (H+F) atoms
= 4-1-2 = 1, So No. of Chlorine atoms = 1

The chemical formula of R 22 = $CHClF_2$

The commonly used refrigerant
i.e. **R-134a = $C_2H_2F_4$**

2) Inorganic refrigerants:
These are designated by number 7 followed by the molecular weight of the refrigerant.
Examples:
(a) Ammonia: Molecular weight is 17, the designation is **R -717**
(b) Carbon dioxide: Molecular weight is 44, the designation is **R -744**
(c) Water: Molecular weight is 18, the designation is **R -718**

3) Azeotropic Refrigerant:
Azeotropic mixtures are designated by 500 series, these are the mixture of two or more halocarbon compound.
Some Azeotropic mixtures:

R -500: Mixture of R 12 and R 152a
R -502: Mixture of R 22 and R 115
R-503: Mixture of R 13 and R 23

(4) Hydrocarbons:
Propane (C_3H_8) : R 290
n-butane (C_4H_{10}) : R 600
Iso-butane (C_4H_{10}) : R 600 a

Examples
Q.1 In reversed Bray ton cycle, the temperature at the end of heat absorption and heat rejection are 0^0C and 30^0C. The pressure ratio is 4 and the pressure in the cooler is 4 bar, determine the temperature at all the states and volume flow rates at inlet of compressor and at exit of turbine for 1 TR cooling capacity.

Solution:
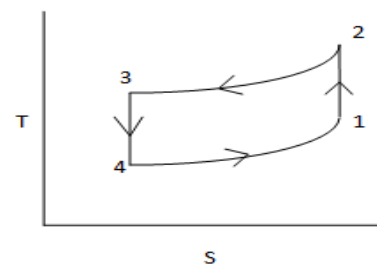T_1 = 273 K
T_3 = 303 K
Temperature at all points
$$\frac{T_2}{T_1} = (r)^{\frac{\gamma-1}{\gamma}}$$
$$T_2 = 273(4)^{\frac{0.4}{1.4}} = 405.67K$$
$$\frac{T_3}{T_4} = (r)^{\frac{\gamma-1}{\gamma}}$$
$$T_4 = \frac{303}{(4)^{\frac{0.4}{1.4}}} = 203.9K$$

Volume flow rate of air for 1 TR or 3.5KJ/sec
Refrigeration effect $= m_a c_p (T_1 - T_4)$
$3.53.5 = 3.5 = m_p X1.005(273 - 203.9)$
$m_p = 0.05kg/sec$
At the inlet of compressor density,
$$P_1 = \frac{P_1}{RT_1} = \frac{30^{-5}}{287X273} = 1.27kg/sec.$$

Volume flow rate
V = P₁ X mₐ = 1.27 X 0.05
= 0.064 m³/sec.

Q.2 The compressed air from main compressor of an air craft cooling system is bled off at 4.5 bar and 200⁰C. It is then passed through the heat exchange in which the ram air is forced through for cooling purpose by fan driver by cooling turbine. The condition of the inlet of cooling turbine is 4 bar an 30⁰C. The air expands in cooling turbine is 4 bar an 30⁰C. The air expands in cooling turbine up to 0.7 bar. The isentropic efficiency of cooling turbine is 80% and flow rate through cooling turbine is 30 kg/min. Find

a) Actual exit temperature from cooling turbine
b) The power delivered to ram air blow fan
c) Tons of refrigeration, if temperature of cabin cockpit area is 25⁰C.

Solution:-

1) Exit temp. Of turbine

$$\frac{T_2}{T_3} = \left(\frac{P_2}{P_3}\right)^{\frac{\gamma-1}{\gamma}} \quad T_3 = 300\left(\frac{0.7}{4}\right)^{\frac{0.4}{1.4}} = 184.15K$$
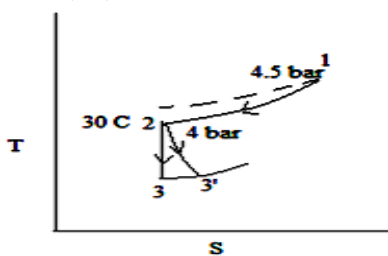
T_3 = 184.15 K

Efficiency of the turbine

$$0.8 = \frac{T_2 - T_3^1}{T_2 - T_3}$$

$$T_3^1 = -(T_2 - T_3)0.8 + T_2$$

$$T_3^1 = 303 - 0.8(303 - 184.15)$$

$$T_3^1 = 207.92k$$

b) Power delivered to fan = work output from turbine

$$= \frac{30}{60} \times 1.005\left(T_2 - T_3^1\right)$$

= 0.5 × 1.005 (303 – 207.92)

= 47.78 KW

c) Refrigeration effect

$$RE = m_a c_p \left(T_{cabin} - T_3^1\right)$$

= 0.5 × 1.005 (298 – 207.92)

=45.27 KJ/s =45.27/3.5 = 12.9 TR

Q.3 A dense air refrigeration machine operating on Bell-Coleman cycle operates between 3.4 bar and 17 bar. The temperature of air after the cooler is 15⁰C and after the refrigerator is 6⁰C. The refrigeration capacity is 6 Tonnes, calculate:

1) temperature after the compression and expansion
2) air circulation per minute
3) work of compressor and expander
4) Theoretical COP.

Solution:

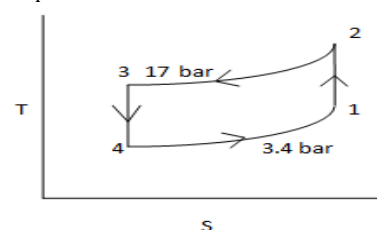$$T_3 = 15^0C = 288K$$

$$T_1 = 6^0C = 279K$$

RE = 6 tonnes

(I) temp. After comp. & expansion

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{17}{3.4}\right)^{\frac{0.4}{1.4}}$$

T₂=279×1.5=445.69 k

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{V-1}{V}} = \left(\frac{3.4}{17}\right)^{\frac{0.4}{1.4}} = .62$$

T₄ = 288 × 0.62 =180.2 K

(II) Air circulation per minute
Refrigeration effect= $m_a c_p (T_1 - T_4)$

6 ×210 = mₐ ×1.005(279-180.2)

Mₐ=12.7 kg/min.

(III) Work of compressor:-

$$w_{comp} = \frac{n}{n-1} m_a R (T_2 - T_1)$$

$$= \frac{1.4}{1.4-1} \times \frac{12.7}{60} \times 287(445.29 - 279)$$

= 35.44 KW

Work of expander

$$w_{turbine} = \frac{n}{n-1} m_a R (T_3 - T_4)$$

$$= \frac{1.4}{1.4-1} \times \frac{12.7}{60} \times 287(288 - 180.2)$$

= 22.89 KW

(IV) Cop of cycle

$$(Cop) = \frac{Heat\,absorbed}{work\,done} = \frac{6 \times 3.5}{35.44 - 22.89}$$

COP = 1.67

Q.4 A simple saturation cycle using R-12 is desired taking a load of 10 Tons. The refrigerating and ambient temperature are 0⁰C and 30⁰C. A minimum temperature difference of 5⁰C is required in evaporator and condenser for heat transfer. Find
a) Mass flow rate through system
b)Power required in kw. The properties of R-12 are following:-

Sat. Temp. ⁰C	Sat. Pr. bar	Specific volume m³/kg	
		Sat (10⁻³)	Sat. vapour
-5⁰C	2.61	0.71	0.0650
0⁰C	3.08	0.72	0.0554
30⁰C	7.45	0.77	0.0235
35⁰C	8.47	0.79	0.0206

Enthalpy kJ/kg		Entropy kJ/kg. k	
Sat. Liquid	Sat. vapour	Sat. Liquid	Sat. vapour
31.4	185.4	0.1251	0.6991
36.1	187.5	0.1420	0.6966
64.4	199.6	0.2399	0.6854
69.5	201.5	0.2559	0.6839

Specific heat of vapour R-12 is 0.95 KJ/KgK

Solution:-

Temp. Of refrigerant in evaporator is -5⁰C and in the condenser is 35⁰C. At low pressure
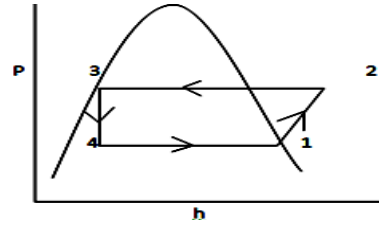
$h_1 = h_g$ at -5⁰C

$h_1 = 185.4 kJ/kg$

Enthalpy of point 4

$h_4 = h_3 = h_{f3} = 69.5kJ$

Refrigerant effect

$= \dot{m}(h_1 - h_4) = \dot{m}(185.4 - 69.5)$

$= \dot{m} \times 115.9 kJ/sec$

$10 \times 3.5 = 115.9 \times \dot{m}$

Mass flow rate

$\dot{m} = \frac{35}{115.9} = 0.3 kg/sec.$

In process(1-2)i.e.isentropic process

$s_1 = s_2$

$$s_{g1} = s_{g2} + c_p l_n \left(\frac{T_{sap}}{T_{sat}} \right)$$

$$0.6991 = 0.6839 + c_p l_n \left(\frac{T_{sap}}{308} \right)$$

T_{sup} = 312.9 k

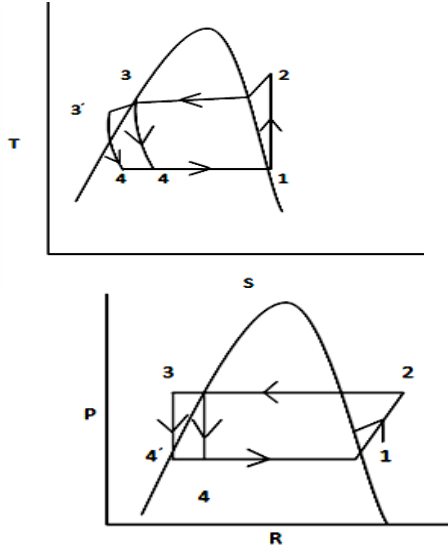$h_2 = h_{g2} + c_p (T_{sap} - T_{sat})$

= 201.5 + 0.95 (312.9-308)

= 206.25 KJ/Kg

(II) Power required by compressor

$w = \dot{m}(h_2 - h_1) =$

$0.3 \times (206.25 - 185.4) = 6.3 KW$

Q.5 An ideal vapour compression refrigerator using R-12 operates between temperature limits of -10⁰C and 40⁰C. The refrigerator leaves the condenser dry saturated. The rate of flow of refrigerant through the unit is 150 Kg/hr.
Calculate the refrigerating effect per kg of refrigerant , COP If
1) refrigerant leaves the condenser as dry saturated
2) refrigerant is sub cooled to 20⁰C before throttling.

Assume that enthalpy of refrigerant before throttling is approximately equal to enthalpy of refrigerant at under cooled temperature of 20⁰C and it as $h_f = 50.59$ kJ/kg.

Solution:

(1) if the refrigerant leaves dry saturated $h_1 = 183.19$ kJ/kg

$s_1 = 7019$ kJ/kg.k

$s_2 = s_1$

$s_1 = s_g + c_p l_n \left(\dfrac{T_{sap}}{T_{sat}} \right)$

At 40⁰C $S_g = 0.6825$
$C_p = 0.609$ kJ/kgk

$s_1 = 0.6825 + 0.609 l_n \left(\dfrac{T_{sap}}{313} \right)$

$0.7019 = 0.6825 + 0.609 l_n \left(\dfrac{T_{sap}}{313} \right)$

$T_{sap} = 323.1$ K

$h_2 = h_g + c_p \left(T_{sap} - T_{sat} \right)$

$= 203.20 + 0.609 (323.1-313)$

$h_2 = 209.37$ kJ/kg.k

$h_3 = h_{f3} = h_4 = 74.59$ kJ/kg

(i) Refrigerating effect $= h_1 - h_4 = 183.1 - 74.59$

$= 108.6$ KJ/kg

$(Cop) = \dfrac{h_1 - h_4}{h_2 - h_1} = \dfrac{108.6}{209.6 - 183.1} = 4.15$

(iii) off refrigerant is sub cooled

$h_3^1 = 50.59 = h_4$

Refrigerating effect $= h_1 - h_4$

$= 183.1 - 50.59$

$= 132.51$ KJ/kg

$(Cop) = \dfrac{h_1 - h_4}{h_2 - h_1} = \dfrac{132.51}{209.6 - 183.19} = 5.06$

Q.6 A two cylinder R-12 compressor has bore and stroke equal to 5.65 cm and 5 cm. Its rpm is 1450 rpm. If liquid refrigerant is throttled by throttle valve at 40⁰C. Determine the mass of refrigerant circulated per minute and theoretical capacity when suction temperature is −10⁰C. Assume γ = 1.13 , properties of R-12 are:

Temp	Pr.(bar)	Volume vapour m³/kg	Enthalpy liquid kJ/kg
-10⁰C	2.1928	0.07702	190.72
40⁰C	9.5944	0.01863	239.29

Enthalpy vapour kJ/kg	Entropy liquid kJ/kgk	Entropy vapour kJ/kgk
347.96	0.9656	1.5632
368.67	1.1324	1.5456

Solution:

Theoretical piston displaced volume

$V_P = \dfrac{\pi}{4} d^2 L N k$

$= \dfrac{\pi}{4} \times \left(5.65 \text{X} 10^{-2} \right)^2 \times 5 \times 1450 \times 2 \times 10^{-2}$

$= 0.335$ m³/min.

$V_g = 0.07702$ m³/kg at -10⁰C

So refrigerant mass circulated/min at -10⁰C suction temperature

$= 0.335/0.07702 = 4.6$ kg/min

$h_1 = 347.96$ kJ/kg

$h_4 = 239.29$ kJ/kg

Refrigerating effect

$RE = m \left(h_1 - h_4 \right)$

$= 4.6$ X $(347.96 - 239.29)$

$= 500.8$ kJ/min

Q.7 A R-12 refrigeration machine has saturated suction temperature of

5^0C and saturated discharge temperature of 40^0C. Determine

i) COP

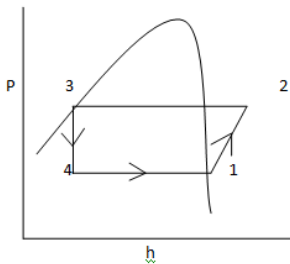ii) Theoretical power. Per ton of refrigeration when the compression is dry properties are:

Temp.	Pr. bar	Volume m³/kg	h_f
5^0C	3.6375	0.048	204.68
40^0C	9.5909	0.0184	239.03

Enthalpy vapour kJ/kg	Entropy liquid kJ/kgk	Entropy vapour kJ/kgk
347.96	0.9656	1.5632
368.67	1.1324	1.5456

Solution:

Process 1-2 is isentropic process.

$S_1 = S_2$

$$1.5569 = S_{g2} + c_p l_n \left(\frac{T_{sap}}{T_{sat}} \right)$$

$$1.5569 = 1.5569 + 0.762 l_n \left(\frac{T_{sap}}{313} \right)$$

$$T_{sup} = 317.5k$$

$$h_2 = h_{g2} + c_p \left(T_{sap} - T_{sat} \right)$$

$$= 368.87 + 0.762 (317.5 - 313)$$

$$= 372.3 \text{ kJ/kg}$$

$h_3 = h_f$ at $313K = 239.03$ kJ/kg $= h_4$

Refrigerating effect $= \dot{m}(h_1 - h_4)$

$$\frac{210}{60} = \dot{m}(h_1 - h_4)$$

$$\dot{m} = 0.03 \text{kg/sec}$$

Theoretical power per ton of refrigeration

$$P = \dot{m}(h_2 - h_1)$$

$$= 0.03(372.3 - 354.885) = 0.53 \text{KW/TR}$$

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{354.885 - 239.03}{372.3 - 354.885} = 6.65$$

Q.8 Assume a simple saturated cycle with suction saturation temp. -10^0C and condensing saturation temperature of 30^0C. If clearance volume 15% of stroke volume. How will the volumetric efficiency vary for different refrigerant namely

(1) R-12   (2) R-22   (3)NH₃?

Solution:

Volumetric efficiency can be given

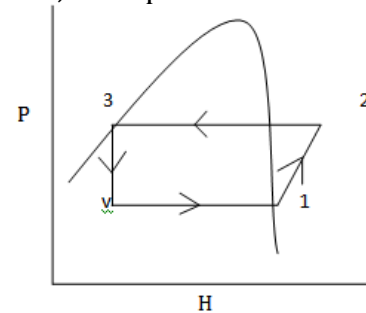$$n_{vol.} = 1 + C - C \left[\frac{v_{suction}}{v_{discharge}} \right]$$

1) For R-12, from p-h chart

$$v_{suction} = 0.07813 \frac{m^3}{kg}, v_{discharge} = 0.025 m^3/kg$$

So $n_{vol.} = 1 + 0.05 - 0.05 \left[\frac{0.07813}{0.025} \right]$

$= 0.894 = 89.4\%$

For R-22, from p-h chart

$$v_{suction} = 0.07 m^3/kg \quad v_{discharge} = 0.0225 m^3/kg$$

So $n_{vol.} = 1 + 0.05 - 0.05 \left[\frac{0.07}{0.0225} \right]$

$= 1.05 - 0.0156 = 0.894 = 89.4\%$

(3) For NH₃, from p-h chart,

$$v_{suction} = 0.4184 m^3/kg, v_{discharge} = 0.135 m^3/kg$$
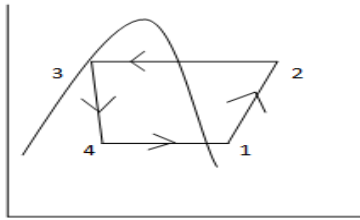
So $n_{vol.} = 1 + 0.05 - 0.05 \left[\frac{0.4184}{0.135} \right]$

$= 0.895 = 89.5\%$

Q.9 Catalogue data from R-12 compressor shows that compressor delivers 15 TR at saturation suction temperature of -5^0C and saturated condensing temperature of 40^0C and actual suction temperature is

15^0C. Liquid leaving the condenser is saturated. Calculate the mass flow rate of this compressor.

Solution:

from the p-h diagram

$h_{g1} = 353 KJ/kg, v_g = 0.66 m^3$,

$h_1 = 363 KJ/kg, v_1 = 0.072 m^3/kgP$

The refrigeration effect

$= \dot{m}(h_1 - h_4)$

$15 \times 210 = \dot{m}(363 - 239)$

$\dot{m} = \dfrac{15 \times 210}{124} = 25.40$

$\dot{m} = 25.40 kg/min$.

And volume flow rate

V = 25.4 X 0.072

V = 1.828 m³/min.

Q.10 A geothermal wall at 130^0C supplies heat at a rate of a 100500 kJ/hr to an absorption refrigeration system. The environment is at 30^0C and the refrigerated space is maintained at -22^0C. Determine the maximum possible heat removed from refrigerated space.

Solution:

$T_C = 303 k T_E = 251 k T_G = 403 k$

$Q_G = 100500 kJ/hr$

For a reversible absorption cycle

$COP = \dfrac{Q_E}{Q_G} = \dfrac{T_E}{T_C - T_E} X \dfrac{T_G - T_C}{T_G}$

$= \dfrac{251}{303 - 251} \times \dfrac{403 - 303}{403}$

$Q_E = Q_G X 1.19 = \dfrac{100500}{3600} X 1.19$

$Q_E = 32.22 KW$

7 PSYCHOMETRY

Psychrometry is the branch of science which deals with study of the properties of the moist air i.e. mixtures of air and water vapour. Atmospheric air is a mixture of many gases plus water vapour and a number of pollutants.

The amount of water vapour and pollutants vary from place to place. A mixture of various gases that constitute air and water vapour is known as moist air.

Constituent	Molecular weight	Mol fraction
Oxygen	32	0.2095
Nitrogen	28.016	0.7809
Carbon dioxide	44.01	0.0003

7.1 PROPERTIES OF MOIST AIR

7.1.1 MOIST AIR

The moist air can be thought of as a mixture of dry air and moisture. Based on the above composition the molecular weight of dry air is found to be 28.966 and the gas constant R is 0.287035 KJ/Kg.K.

If the partial pressure of dry air and water vapour is P_a and P_V, then according to Dalton's law of partial pressure

$$P_b = P_a + P_v \quad \text{------- (7.1)}$$

Where P_b is barometer reading of atmospheric pressure. In general, atmospheric pressure is taken as 760 mm of Hg.

7.1.2 SATURATED AIR

At a given temperature and pressure, the dry air can only hold a certain maximum amount of moisture. When the moisture content in the moist air is maximum, the air is known as saturated air.
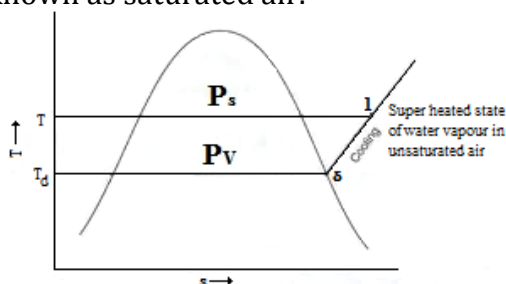

Fig. Water vapour in air mixture

7.1.3 HUMIDITY RATIO

The humidity ratio or specific humidity (W) is the mass of water per kg of dry air in a given mixture of dry air and water vapour. If m_v is the mass of water vapour and m_a is the mass of dry air. Then humidity ratio

$$W = \frac{m_v}{m_a}$$

We know that

$$P_V V = m_V R_V T \text{ for the water vapour}$$

And $P_a V = m_a R_{aV} T$ for the dry air

Substituting the values

$$W = \frac{\dfrac{P_V V}{R_V T}}{\dfrac{P_a V}{R_a T}} = \frac{P_V \times Ra}{P_a \times R_V}$$

$$= \frac{P_V \times 8.314/28}{P_a \times 8.314/18} = 0.622 \frac{P_V}{P_b - P_V}$$

$$W = 0.622 \frac{P_V}{P_b - P_V} \quad ----(7.2)$$

If air is saturated at temperature T, then Specific humidity of saturated air

$$Ws = 0.622 \frac{P_s}{P_b - P_s} \quad ----(7.3)$$

7.1.4 DEGREE OF SATURATION

The degree of saturation is the ratio of the humidity ratio (W) of the mixture to the humidity ratio of a saturated mixture (W_S) at the same temperature and pressure. it is denoted by μ.

Degree of Saturation

$$\mu = \frac{W}{W_s} = \frac{\dfrac{0.622 P_V}{P_b - P_V}}{\dfrac{0.622 P_s}{P_b - P_s}} = \frac{P_V(P_b - P_s)}{P_s \cdot (P_b - P_V)}$$

$$\mu = \frac{P_V(P_b - P_s)}{P_s \cdot (P_b - P_V)} \quad ----(7.4)$$

7.1.5 RELATIVE HUMIDITY

Relative humidity is defined as the ratio of the mass of water vapour present in a given volume of air at temperature T to the mass of water vapour when the same volume of air is saturated at the same temperature T and pressure. Relative humidity is normally expressed as a percentage. When Φ is 100 percent, the air is saturated.

Relative humidity

$$\phi = \frac{\text{mass of water vapour in a given volume of air at Temperature T}}{\text{mass of water vapour when same volume of air is saturated at Temperature T}}$$

$$= \frac{m_V}{m_S}$$

Assuming the water vapour is an ideal gas,

$$P_V V = m_V R_V T$$

And for the saturated air condition

$$P_s V = m_s R_V T$$

So from the equation () and Equation (), we get

$$\frac{m_V}{m_S} = \frac{P_V}{P_S}$$

So $\phi = \dfrac{m_V}{m_S} = \dfrac{P_V}{P_S}$ --------- (7.5)

for the dry air m_V and P_V are zero, so

$$\Phi = 0$$

for the dry air m_V and P_V are equal to m_s and P_s , so

$$\Phi = 100\%$$

So relative humidity varies from 0% to 100%.

7.1.6 DRY BULB TEMPERATURE

DBT is the temperature of the air as measured by a standard thermometer when it is unaffected by thermometer.

7.1.7 DEW-POINT TEMPERATURE (DPT)

If moist air is cooled at constant pressure, then the temperature at which the moisture presents in the air begins to condense is known as dew-point temperature (DPT) of air.

7.1.8 WET BULB TEMPERATURE (WBT)

It is temperature of air measured by thermometer when wet cloth is covered over the thermometer.

7.1.9 DENSITY OF DRY AIR

The density of air or air density is the mass of air per unit volume Air density decreases with increasing altitude. It also changes with variation in temperature or humidity. At sea level and at 15 °C, air has a density of approximately 1.225 kg/m³.density of dry air can be calculated using the ideal gas law, expressed as a function of temperature and pressure

$$\rho_a = \frac{P_a}{R_a T_D} \ ---- \ (7.6)$$

Where ρ = Air density

P_a = Absolute pressure
T_D = Dry Bulb temperature
R_a = Specific gas constant for dry air
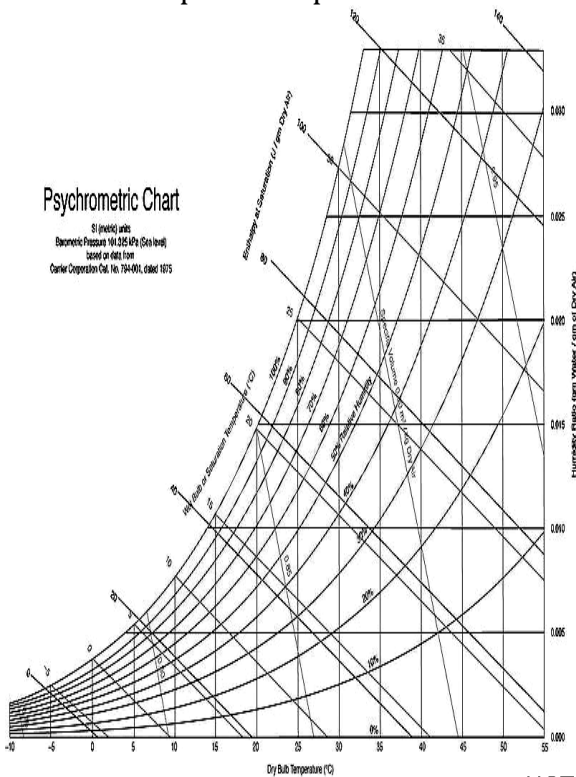
7.1.10 ENTHALPY OF THE AIR

The enthalpy of moist air is the sum of the enthalpy of the dry air and the enthalpy of the water vapour. Enthalpy values are always based on some reference value. The enthalpy of the mixture is given by

$$h = h_a + W h_V = C_p T_a + W \left[h_g + 1.88 \left(t_W - t_{dp} \right) \right]$$

------ (7.7)

7.2 PSYCHROMETRIC CHART

A Psychrometric chart graphically represents the thermodynamic properties of moist air. Standard Psychrometric charts are bounded by the dry-bulb temperature line (abscissa) and the vapour pressure or humidity ratio (ordinate). The Left Hand Side of the Psychrometric chart is bounded by the saturation line. Figure shows the schematic of a Psychrometric chart.

- Psychrometric charts are readily available for standard barometric pressure of 101.325 KPa at sea level and for normal temperatures (0-50°C).
- The Relative humidity at the saturation curve is 100 %.
- At the saturation curve, Dry bulb Temperature, Wet bulb Temperature and Dew point temperature are same.

Psychrometric Chart

SI (metric) units
Barometric Pressure 101.325 KPa (Sea level)
based on data from
Carrier Corporation Cat. No. 794-001, dated 1975

COLD ⟶ HOT

fig. Psychrometric Chart

The parameters of Psychrometric chart are relative humidity, dry bulb temperature, wet bulb temperature, dew point temperature, specific humidity, specific volume, specific enthalpy and vapour pressure.

The Psychrometric chart allows all the parameters of some moist air to be determined from any three independent parameters, one of which must be the pressure.

7.3 SIGNIFICANCE OF DIFFERENT LINES ON PSYCHROMETRY CHART

7.3.1 DRY-BULB TEMPERATURE LINES

Dry bulb temperature is temperature of an air sample, as determined by an ordinary thermometer. It is typically plotted as the abscissa (horizontal axis) of the graph. The SI unit for temperature is degrees Celsius. These lines are drawn straight, parallel to each other. Each line represents a constant temperature.
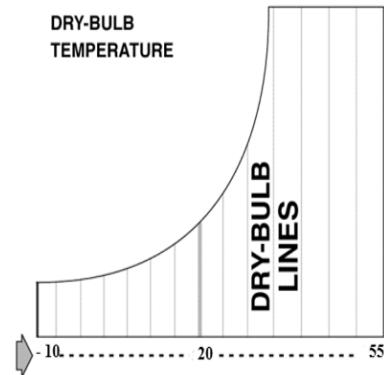

fig. Dry Bulb Temperature lines

7.3.2 WET-BULB TEMPERATURE (WBT) LINE

Wet-bulb temperature is temperature of an air sample after the air has passed over a large surface of liquid water in an insulated channel. When the air sample is saturated with water, the WBT will read the same as the DBT. These lines are oblique lines that differ slightly from the enthalpy lines. They are identically straight but are not exactly parallel to each other. These intersect the saturation curve at DBT point.
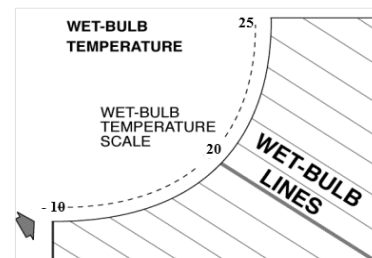

fig. Wet Bulb Temperature line

7.3.3 DEW POINT TEMPERATURE (DPT) LINE

Dew point temperature (DPT) is the temperature at which a moist air at the same pressure would reach water vapour saturation. At this point further removal of

heat would result in water vapour condensing into liquid water fog. From the state point follow the horizontal line of constant humidity ratio to the intercept of 100% RH, also known as the saturation curve. The dew point temperature is equal to the fully saturated dry bulb or wet bulb temperatures.
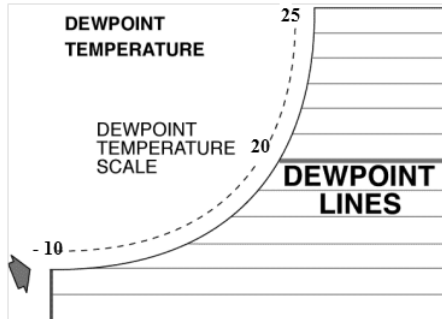

fig. Dew Point temperature line

7.3.4 Relative humidity (RH) Line

Relative humidity (RH) is the ratio of the mole fraction of water vapor to the mole fraction of saturated moist air at the same temperature and pressure. RH is dimensionless, and is usually expressed as a percentage. Lines of constant RH reflect the physics of air and water: they are determined via experimental measurement. **Relative humidity:** These hyperbolic lines are shown in intervals of 10%. The saturation curve is at 100% RH, while dry air is at 0% RH.
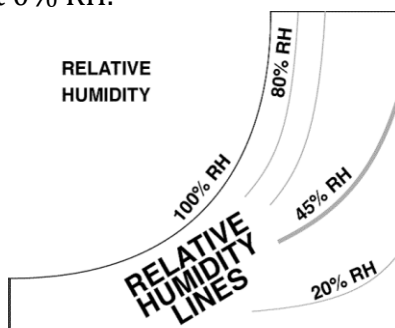

fig. relative humidity lines

7.3.5 HUMIDITY RATIO LINE

Humidity ratio is the proportion of mass of water vapour per unit mass of dry air at the given conditions. It is also known as the moisture content or mixing ratio. It is

typically plotted as the ordinate (vertical axis) of the graph. For a given DBT there will be a particular humidity ratio for which the air sample is at 100% relative humidity. These are the horizontal lines on the chart.
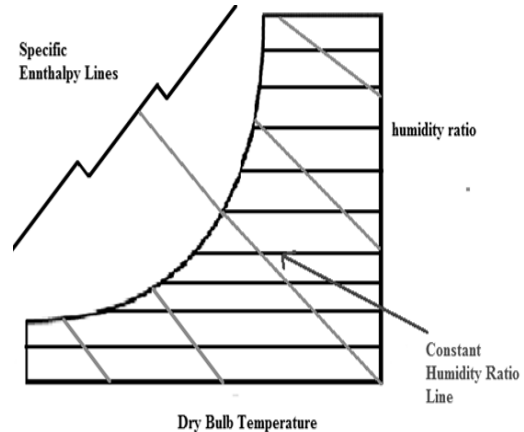

Fig. Humidity ratio lines

7.3.6 SPECIFIC ENTHALPY LINE

These are oblique lines drawn diagonally downward from left to right across the chart that are parallel to each other. In the approximation of ideal gases, lines of constant enthalpy are parallel to lines of constant WBT.
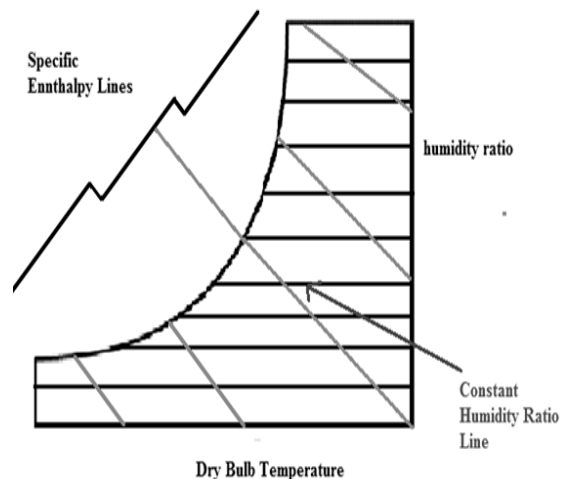

Fig. Specific Enthalpy Lines

7.3.7 SPECIFIC VOLUME LINE

Specific volume is the volume of the mixture containing one unit of mass of dry air. The SI units are cubic meters per kilogram of dry air. These lines are a family of equally spaced straight inclined lines that are nearly parallel.
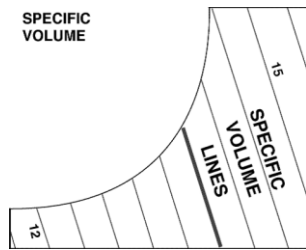
fig. Specific Volume lines

7.4 DIFFERENT PROCESS ON PSYCHROMETRIC CHART

7.4.1 SENSIBLE COOLING PROCESS

During this process, the moisture content of air remains constant but its temperature decreases as it flows over a cooling coil. For moisture content to remain constant the surface of the cooling coil should be dry and its surface temperature should be greater than the dew point temperature of air. If the cooling coil is 100% effective, then the exit temperature of air will be equal to the coil temperature. However, in practice, the exit air temperature will be higher than the cooling coil temperature. The heat transfer during the cooling process is given as

$$Q_C = m_a \left(h_O - h_A \right) = m_a C_{pm} \left(T_O - T_A \right) ----(7.8)$$

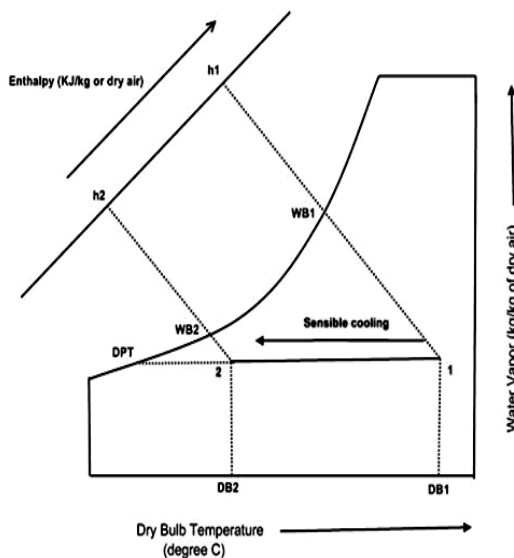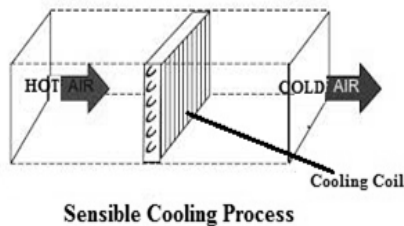**Sensible Cooling Process**

Fig. Sensible cooling Process

7.4.2 SENSIBLE HEATING PROCESS

During this process, the moisture content of air remains constant and its temperature increases as it flows over a heating coil. The heat transfer rate during this process is given by
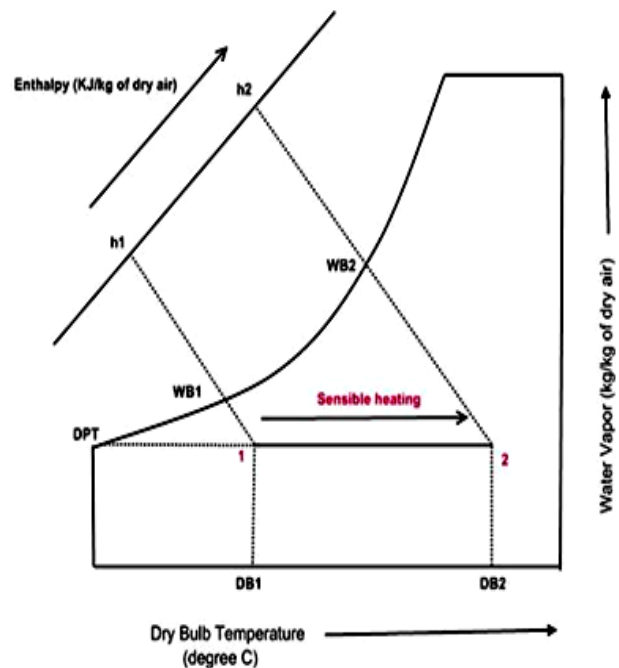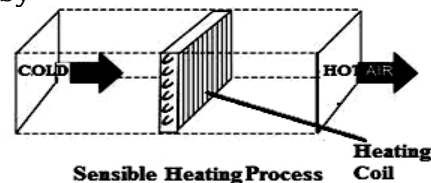

Sensible Heating Process

fig. Sensible Heating Process

$$Q_h = m_a \left(h_b - h_o \right) = m_a C_{pm} \left(T_b - T_O \right) ----(7.9)$$

Where C_{pm} is the humid specific heat. ma is the mass flow rate of dry air (kg/s).

7.4.2.1 BY PASS FACTOR

In Fig., the temperature T_{d3} is the effective surface temperature of the heating coil, and is known as apparatus dew-point (ADP) temperature. T_{d1} and T_{d2} are the temperature of air at inlet and outlet of the coil. In an ideal situation, when all the air comes in perfect contact with the heating coil surface, then the exit temperature of air will be same as ADP of the coil.

Sensible Heating Process Coil
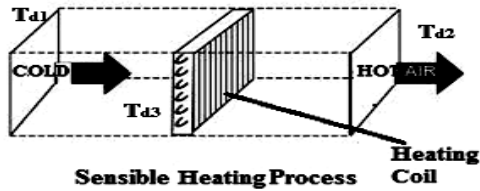
But in actual case the exit temperature of air will always be lesser than the apparatus dew-point temperature. So By -pass factor (BPF) is defined as the ratio of heat loss to maximum possible heat transfer.

$$BPF = \frac{\text{Heat Loss}}{\text{Maximum Possible heat transfer}} = \frac{T_{d3} - T_{d2}}{T_{d3} - T_{d1}}$$
-------- (7.10)

In the same way, Bypass factor for cooling coil can be given as

$$BPF = \frac{\text{Heat Loss}}{\text{Maximum Possible heat transfer}} = \frac{T_{d2} - T_{d3}}{T_{d1} - T_{d3}}$$
-------- (7.11)

7.4.2.2 EFFICIENCY OF COIL

The efficiency of heating Coil is defined as the ratio of actual heat transfer to the cold air to maximum possible heat transfer to the cold air by heating coil.
Efficiency

$$\eta = \frac{\text{Actual Heat transfer}}{\text{Maximum Possible heat transfer}}$$
$$= \frac{T_{d2} - T_{d1}}{T_{d3} - T_{d1}}$$
------- (7.12)

In the same way efficiency for cooling coil can be given as
The efficiency of cooling Coil is defined as the ratio of actual heat transfer to the cold air to maximum possible heat transfer to the cold air by heating coil.
Efficiency

$$\eta = \frac{\text{Actual Heat transfer}}{\text{Maximum Possible heat transfer}}$$
$$= \frac{T_{d1} - T_{d2}}{T_{d1} - T_{d3}} - - - -(7.13)$$

By comparing the equation (7.11) and equation (7.13) of Bypass factor and efficiency of the heating and cooling coil, we get

$$\mu = 1 - BPF$$
-------- (7.14)

7.4.3 COOLING AND DEHUMIDIFICATION PROCESS

When moist air is cooled below its dew-point by bringing it in contact with a cold surface as shown in Fig., some of the water vapour in the air condenses and leaves the air stream as liquid, as a result both the temperature and humidity ratio of air decreases as shown. The heat and mass transfer rates can be expressed in terms of the initial and final conditions by applying the conservation of mass and conservation of energy equations.
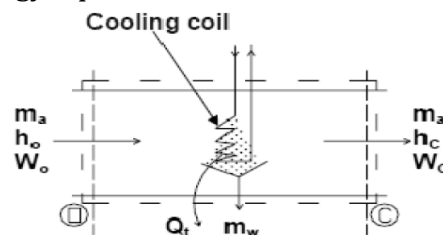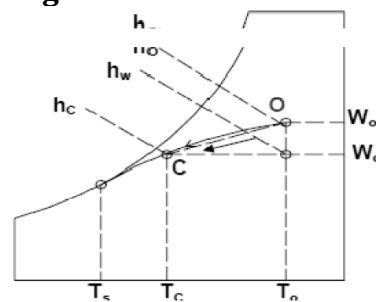

Fig. Cooling

Fig. dehumidification

By applying mass balance for the water
$m_a.W_o = m_a.W_C + m_w$
By applying energy balance
$m_a.h_o = Q_t + m_a.h_C + m_w.h_w$
From the above two equations, the load on the cooling coil, Qt is given by:

$Q_t = m_a(h_o.-h_C) + m_a.(W_o-W_C).h_w$ ------- (7.15)
Where W_o and W_c are the humidity ration before and after the cooling process. m_w is mass of water vapour separating from air.

It can be observed that the cooling and de-humidification process involves both latent and sensible heat transfer processes, hence, the total, latent and sensible heat transfer rates (Qt, Ql and Qs) can be written as
$Q_t = Q_L + Q_S$
$Q_L = m_a.(W_o - W_C).h_w$
$Q_S = m_a (h_o.- h_C) = m_a.C_{pm}. (T_o.- T_C)$

7.4.3.1 SENSIBLE HEAT FACTOR

By separating the total heat transfer rate from the cooling coil into sensible and latent heat transfer rates, a useful parameter called Sensible Heat Factor (SHF) is defined.

So SHF is defined as the ratio of sensible to total heat transfer rate.

$$SHF = \frac{\text{Sensible Heat Transfer}}{\text{Total Heat Transfer}}$$

$$= \frac{\text{Sensible Heat(SH)}}{\text{Sensible Heat (SH)} + \text{Latent heat(LH)}}$$

7.4.4 HEATING AND HUMIDIFICATION PROCESS

During winter it is essential to heat and humidifies the room air for comfort. This is normally done by first sensible heating the air and then adding water vapour to the air stream through steam nozzles as shown in the fig. By this process, dry bulb temperature and humidity ratio increase. Appling Mass balance of water vapour for the control volume

$$m_w = m_a (W_D - W_o)$$
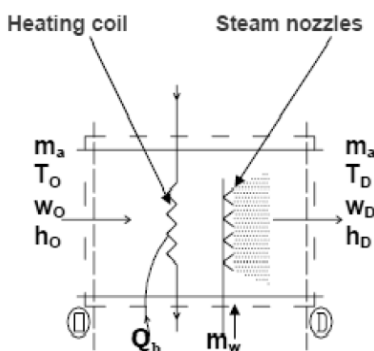
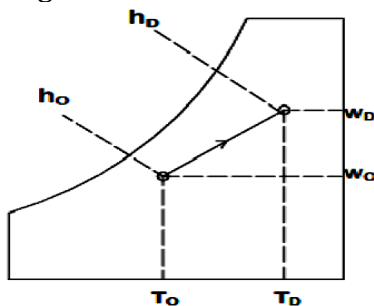Where m_a is the mass flow rate of dry air.

Fig. Heating Process

Fig. Humidification Process

From energy balance:

$$Q_h = m_a (h_D - h_o) - m_w h_w \quad \text{--------- (7.16)}$$

Where Q_h is the heat supplied through the heating coil. m_w and h_w is the mass and the enthalpy of steam. Since this process also involves simultaneous heat and mass transfer, we can define a sensible heat factor for the process in a way similar to that of a cooling and dehumidification process.

7.4.5 COOLING & HUMIDIFICATION PROCESS

During this process, the air temperature drops and its humidity increases. This process is shown in Fig. This can be achieved by spraying cool water in the air stream. The temperature of water should be lower than the dry-bulb temperature of air but higher than its dew point temperature to avoid condensation ($T_{DPT} < T_w < T_O$).

It can be seen that during this process there is sensible heat transfer from air to water and latent heat transfer from water to air. Hence, the total heat transfer depends upon the water temperature.
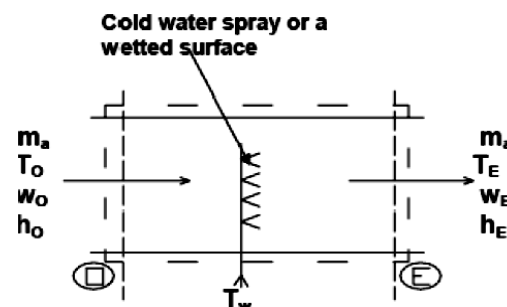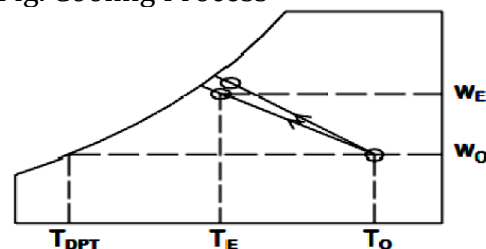

Fig. Cooling Process

Fig. Humidification process

➢ If the temperature of the water sprayed is equal to the wet-bulb temperature of air, then the net transfer rate will be

zero as the sensible heat transfer from air to water will be equal to latent heat transfer from water to air. The process is called as adiabatic dehumidification process.

➢ If the water temperature is greater than WBT, then there will be a net heat transfer from water to air.

➢ If the water temperature is less than WBT, then the net heat transfer will be from air to water.

7.4.6 CHEMICAL DE-HUMIDIFICATION PROCESS

This process can be achieved by using a hygroscopic material, which absorbs or adsorbs the water vapour from the moisture.
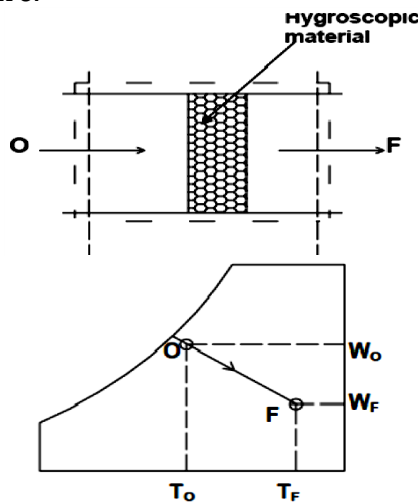

Fig. Chemical De-Humidification Process

If this process is thermally isolated, then the enthalpy of air remains constant and the process is called as adiabatic dehumidification process. As a result the temperature of air increases as its moisture content decreases as shown in Fig. In general, the absorption of water by the hygroscopic material is an exothermic reaction, as a result heat is released during this process, which is transferred to air and the enthalpy of air increases.

7.4.7 MIXING OF AIR STREAMS

Mixing of air streams at different states is commonly encountered in many processes, including in air conditioning. Depending upon the state of the individual streams, the mixing process can take place with or without condensation of moisture. Fig shows an adiabatic mixing of two moist air streams. As shown in the figure, when two air streams at state points 1 and 2 mix, the resulting mixture condition 3 can be obtained from mass and energy balance.
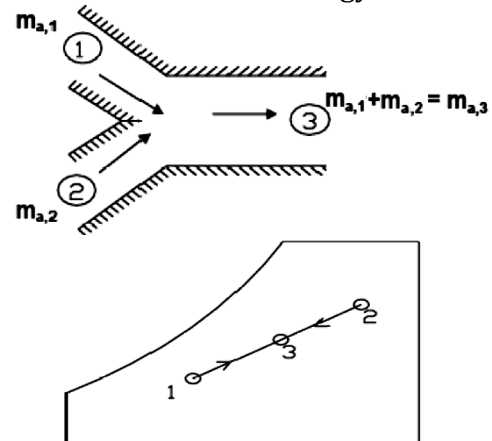

fig. Mixing of two air streams

From the mass balance of dry air and water vapour:

$$m_{a1} + m_{a2} = m_{a3}$$

From energy balance:

$$m_{a1}h_1 + m_{a2}h_2 = m_{a3}h_3 = (m_{a1} + m_{a2})h_3$$

$$h_3 = \frac{m_{a1}h_1 + m_{a2}h_2}{(m_{a1} + m_{a2})} ----(7.17)$$

From the above equation, it can be observed that the final enthalpy of mixture is weighted average of inlet enthalpies.

Example:

Q.1 Atmospheric air at 760mm Hg pressure and dry bulb and wet bulb temperature15^0 C and 12^0 C enter the heating coil whose temperature is 41^0 C. The By pas factor of heating coil is 0.5. Determine the dry bulb temperature, wet bulb temperature, relative humidity of air leaving the heating coil and also the sensible heat supplied per kg of dry air. Solve the problem with help of psychrometric chart.
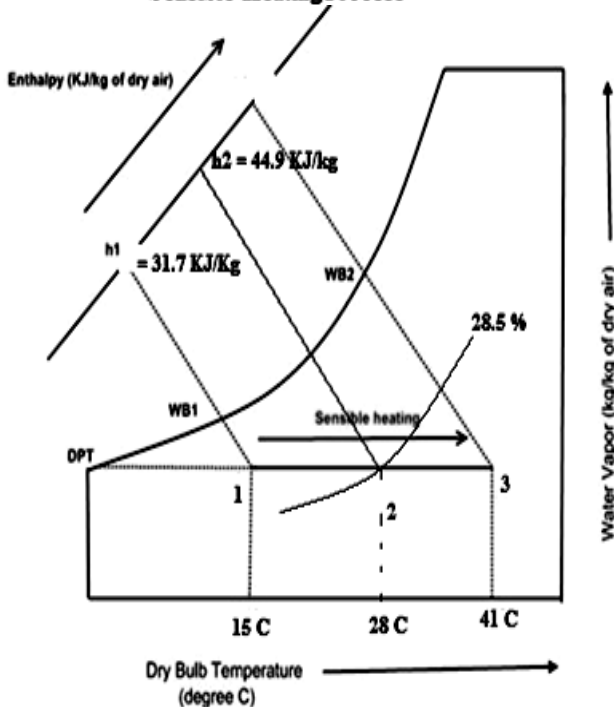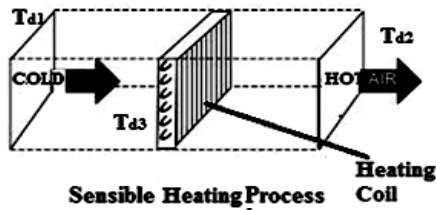
Solution:

By pass factor of heating coil is given as

$$BPF = \frac{T_{d3} - T_{d2}}{T_{d3} - T_{d1}}$$

$$0.5 = \frac{41 - T_{d2}}{41 - 15}$$

$$T_{d2} = 41 - 0.5(41 - 15) = 28^0C$$

Sensible Heating Process

Thus the actual condition at which air comes out of the heating coil is 28⁰C.With the help of the psychrometric chart relative humidity at this point is 28.5% Wet bulb temperature at this point is 16⁰C.

From psychrometric chart,
Enthalpy at point 1 h₁ = 31.7 KJ/Kg
Enthalpy at point 2 h₂ = 44.9 KJ/Kg
So the sensible heat supplied to the air Q =h₂- h₁
= 44.9-31.7
= 13.2 KJ/Kg of the dry air.

Q.2 The humidity ratio of an atmospheric air at 27.5⁰C is 0.016 kg/ kg of dry air. determine partial pressure of vapour, relative humidity and dew point temperature. Assume standard barometric pressure is 760 mm of Hg.

Solution:
Specific humidity or humidity ratio is given by

$$W = 0.622 \frac{P_V}{P_b - P_V}$$

$$0.016 = 0.622 \frac{P_V}{760 - P_V}$$
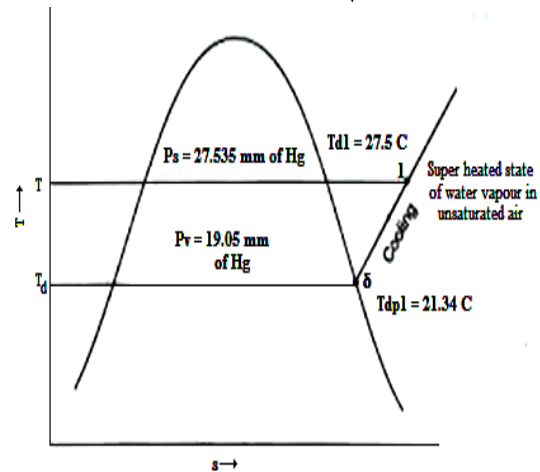
$$Pv = \frac{0.016 \times 760}{0.622 \times 1.02575}$$

= 19.05 mm of Hg.
Relative humidity

$$RH = \frac{P_{V1}}{P_{S1}}$$

Where P_{s1} is saturated pressure corresponding to the saturation temperature equal to dry bulb temperature of 27.5⁰ C. From the table it can be read as 27.535 mm of Hg.
So,

$$RH = \frac{P_{V1}}{P_{S1}} = \frac{19.05}{27.535} = 0.694$$

= 69.4 %

Dew point temperature is the saturation temperature corresponding to the vapour pressure of the 19.05 mm of Hg as read from table. So, T_{dp1} = 21.34⁰ C

Q.3 Air enters a chamber at 5°C dry bulb temperature (DBT) And 2.5° C wet

bulb temperature at the rate of $100 m^3/min$ and the pressure of 1 bar. While passing through the chamber the air absorbs 50 KW heat and pick's up 35 kg/hr of saturated steam at 110°C. Show the process on a psychrometric chart and find the dry and wet bulb temp of the leaving air at 110°C, enthalpy of saturated steam is 2691.3 kj/kg.
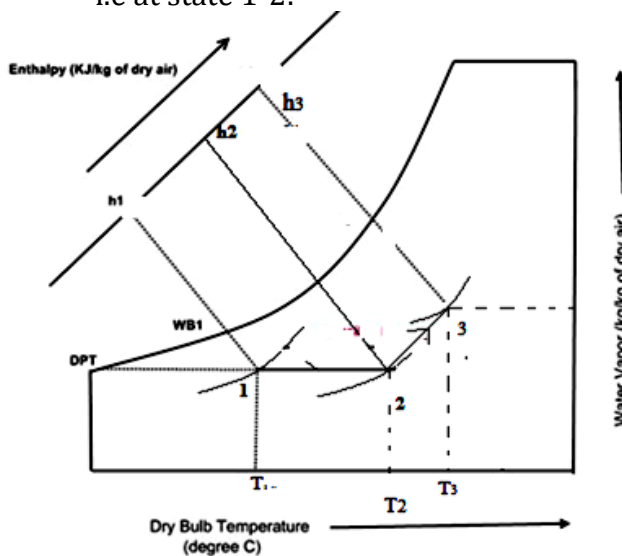
Solution:
For process 1-2 this is sensible heating process
Heat absorbed = 50 kw
$= m(h_2 - h_1)$
i.e at state 1-2:

Enthalpy (KJ/kg of dry air)
Dry Bulb Temperature (degree C)
Water Vapor (kg/kg of dry air)

$W_1 = 0.0035$ (from psychometric chart)
Specific volume $= 0.792 m^3/kg$
$h_1 = 14$ kj/kg
Volume flow rate $= 100 \ m^3/min$
$= 1.67 \ m^3/s$

Mass flow rate $= \dfrac{1.67}{0.792}$

$= 2.104$ kg/s
Now heat transferred:-
$Q = m(h_2 - h_1) = 2.104 \ m(h_2 - 14)$
$50 = 2.104 \ m(h_2 - 14)$
$h_2 = 37.76$ kj/kg
Now at state 2, from psychometric chart
$W_2 = 0.0035, h_2 = 37.76 \ kj/kg$
After humidification by steam injection

$w_3 = w_{2+} \dfrac{ms}{mq}$

$= 0.0035 + \dfrac{+35}{3600 \times 2.1} = 0.0081$

Enthalpy of living air :

$h_3 = h_2 + \dfrac{ms}{mq}(h_v)$

$h_v = 2691.3$ kj/kg at 110 °C

$h_3 = 37.76 + \dfrac{+35}{3600 \times 2.104} \times 2691.3$

$= 50.19$ ks/kg
At state (3):
$t_3 = 31^0C, \quad t_{w2} = 18.3 \ ^0C$

Q.4 In an air conditioned space, 50 kg dry air/sec of fresh air at 45 °C DBT and 30% RH is introduced. The room air at 25 °C DBT and 50% RH is recalculated at 450 kg dry air/sec. The mixed air flows over a cooling coil which has apparatus dew point of 12 °C and by pass factor of 0.15. Determine the conditions at outlet of cooling coil, RSH, RLH, the cooling load of coil and the condensate rate .The saturation pressure of water of required temperature are

Temperature (°C)	P_{ws} (bar)
12	0.14016
25	0.03166
45	0.09584

Enthalpy at temperature, h=1.005 T +W (2500+1.885) kg/kg dry air.

Solution:
Temperature of air after mixing
$T_x = 0.9 \ t_2 + 0.1 t_1$
$= 0.9 \times 25 + 0.4 \times 45$
Specific humidity:
a) At outlet condition:

$W_1 = 0.622 \times \dfrac{Pvl}{(Po - Pvl)}$

$Pv_1 = 0.3 \times 0.09584$
$= 0.02875$ bar

$W_1 = 0.622 \times \dfrac{0.02875}{(1.0132 - 0.02875)}$

$= 0.01816$ kg vapour/kg dry air

b) In the Room

$$w_2 = 0.622 \times \frac{P_{v_2}}{P_v - P_{v_2}}$$

P_{v2} = 0.5 ×0.03166

= 0.01583 bar

$$W_2 = \frac{0.622 \times 0.01583}{(1.0132 - 0.01583)}$$

W_2 = 0.00987 kg vapour /kg dry air

Specific humidity at **A**

$$W_A = \frac{0.622 \times 0.014016}{0.0132 - 0.014016}$$

= 0.008725 kg vapor /kg dry air

Specific humidity at inlet to cooling coil

W_x = 0.9 W_2+0.1 W_1

= 0.9×0.009870 + 0.1×0.01816

=0.0107 kg vapour /kg dry air

Enthalpy at X

h_x=1.00×27+0.00107(25001.88+27)

= 54.43 kj/kg dry air

Enthalpy at 2

H_2=1.005×25

+0.00987(25001.88×25)

= 50.3 KJ/kg dry air

B.P.F. of cooling coil

$$BPF = \frac{ts - ta}{(tx - ta)}$$

$$0.15 = \frac{ts - ta}{(tx - ta)} = \frac{ts - 12}{27 - 12}$$

$T_s = 14.25^0 C$

$$0.15 = \frac{ws - wa}{(wx - wa)} = \frac{ws - 0.008725}{0.0107 - 0.008725}$$

W_s = 0.009 kg vapour/kg dry air

Temp (0C)	Sat Pressure (bar)	SP. Volume of vapour (m³/Kg)
25	0.03166	43.4
15	0.01703	77.98
10	0.01078	-
0	0.01002	-

Enthalpy at supply point

hs=1.005×14.25+0.009

(2500+1.88×14.25)

=37.06 KJ/kg dry air.

Condition at outlet

T_s=14.25 ^{0}C

hs =37.06 KJ/kg dry air

W_s=0.009 kg vapour/kg dry air

Room sensible heat

RSH = M_a L_p (T_2-T_S)

=500×1.0216 (25-14.25

= 5491.1 K_W

Room latent heat

RLH = Ma (w_2-w_s)+2500

=500 (0.00987-0.009) ×2500

= 1087.5 kw

Load on cooling coil = ma (h_x-h_s)

=500(54.43-37.06)

=8685 kw = $\frac{8685}{3.5}$

= 2481.43 tones

Condensate rate = (w_x-w_s) M_a

= (0.0107-0.009) ×500

=0.85 kg vapour /sec

=3060 kg vapour/hr

Q.5 Calculate all the psychometric properties of air at 1 bar, 30^{0}C DBT and 25^{0}C WBT. The following properties of water may be assumed :-

Temp (0C)	Sat Pressure (bar)	SP. Volume of vapour (m³/Kg)
25	0.03166	43.4
15	0.01703	77.98
10	0.01078	-
0	0.01002	-

Solution:

The following expression may be used, if necessary

$$P_v = P_v - \frac{(P - Pw)(DBT - WBT) + 1.8}{2824 - 1.325(1.8 \times DBT + 32)}$$

DBT=25^{0}C, WBT=15^{0}C

$$P = (Pv) - \frac{(P - Pw)(DBT - WBT) + 1.8}{2854 - 1.325(1.8tDBT + 32)}$$

$$= 0.017307 - \frac{(1 - 0.03166)(25 - 15) + 1.8}{2854 - 1.325(1.8 + 25 + 32)}$$

$$v = 0.017307 - \frac{17.43}{2751.975}$$

Pv = 0.01097 bar

Pyschrometric properties:-

1. specific humidity

$$W = 0.622 - \frac{pv}{p - pv}$$

$$= 0.622 - \frac{0.01097}{1 - 0.01097}$$

W= 0.006899 kg vapor/kg of dry air

2. Relative humidity (ϕ)

$$\phi = \frac{Pv}{P - P_V} \times 100$$

$$= \frac{0.01097}{0.03160} \times 100$$

ϕ = 34.65%

3. Degree of saturation (μ)

$$\mu = \phi(\frac{P - P_S}{P - P_V})$$

$$= 0.3465 \frac{(1 - 0.03166)}{(1 - 0.00109)}$$

$$\mu = 0.3465 \left(\frac{0.96834}{0.9890} \right)$$

μ = 0.339 or 33.9%

4) Enthalpy of moist air (h)

h = C_{pa} + W [2500 + 1.88]

where C_{pa} =1.005 kj/kg.k

$$h = (1.005 \times 25) + 0.006899[2500 + 1.88 \times 25]$$

h = 25.125+0.006899(2547)

$h = 42.696 KJ / Kg$ of dry air.

8 POWER PLANT ENGINEERING

Power plant engineering is the branch of thermal science which deals with study of chemical energy of fuel is converted into thermal energy and then utilises into the mechanical work done in the efficient manner. When the thermal energy is converted into the mechanical energy it is impossible to convert whole amount of heat into work. So efficiency play an important role in power plant to check the performance of the power plant. The main important power cycles are gas power cycle and steam power cycle. Gas power cycle works on Brayton cycle and steam power cycle works on Rankine cycle.

8.1 STIRLING CYCLE

Stirling cycle has two reversible isothermal heat additions & heat rejection process and two reversible isocoric processes. It differs from cannot cycle in that two isentropic processes of Carnot cycle are replaced by two constant volume process.
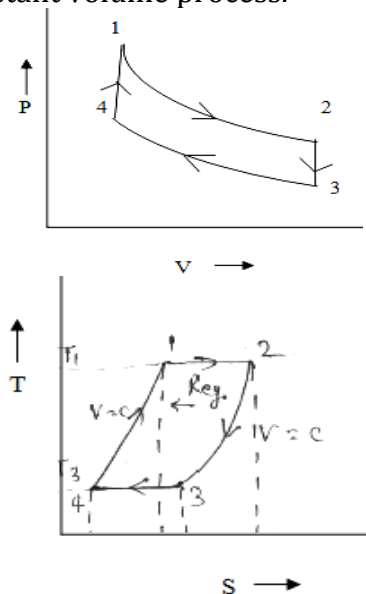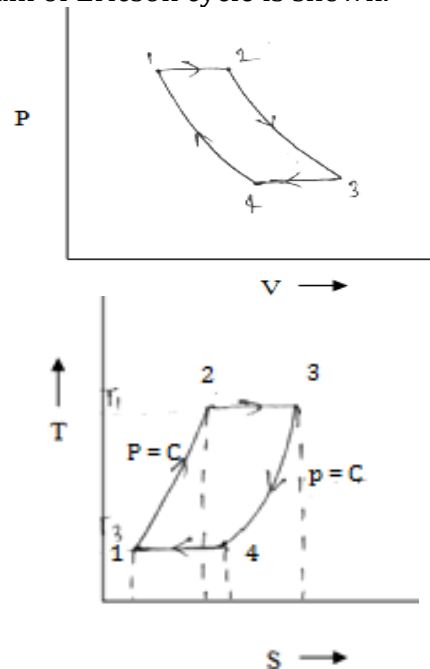

The efficiency of Stirling cycle is given as

$$\eta = 1 - \frac{T_3}{T_1} = 1 - \frac{T_{min}}{T_{max}}$$

If cycle is regenerative cycle then it has same efficiency as that of cannot cycle i.e.

area covered by 2 – 3 process on T-S diagram is equal to the area covered by 4 – 1 process.

8.2 ERICSON CYCLE

Ericson cycle has two reversible isothermal heat additions & heat rejection process and two reversible isobaric processes. It differs from cannot cycle in that two isentropic processes of Carnot cycle are replaced by two constant pressure process. P-V and T-s diagram of Ericson cycle is shown.

The efficiency of Stirling cycle is given as

$$\eta = 1 - \frac{T_2}{T_1}$$

Regenerative Ericson cycle has same efficiency as that at cannot cycle. And both Stirling and Ericson cycle has same efficiency it temperature limits are same.

Gas turbine cycle with inter cooling, reheating & regeneration closely resemble the Ericson cycle. Both cycles are difficult in practice because heat transfer at constant temperature requires either infinite long surface area or infinite long time.

8.3 GAS POWER PLANT

In gas power cycle, the gas is a working fluid. It does not go under the phase change during the cycle. Gas power cycle works on Brayton cycle. There are two type of cycle:

Open Cycle

In this cycle, the working fluid renew after completion of the cycle.

Closed cycle

In closed cycle, chemical composition of working fluid remains same as heat is supplied through the heat transfer and not by direct combustion.

Advantages of closed cycle over open cycle

1) Use of high Pr. Throughout cycle reduces size of plant
2) Close cycle has higher thermal efficiency.
3) Elimination of possible of turbine blades erosion
4) Closed cycle can use any working fluid or gas of higher value of γ to increases the power output and efficiency.

8.3.1 BRAYTON CYCLE

Brayton cycle consists of four processes:

Process 1–2: Isentropic Compression Process: In this process the heat transfer is zero.

dQ = 0

Process 2–3 : Constant pressure Heat addition Process: Heat addition in this process is given by

$Q_S = m. C_P (T_3 - T_2)$

Process 3–4 : Isentropic Expansion Process: In this process the heat transfer is zero.

dQ = 0

Process 4–1 : Constant pressure Heat rejection Process: The heat is reject in constant pressure process in heat exchanger. Heat rejection in this process is given by

$Q_R = m. C_P (T_4 - T_1)$

Heat Exchanger
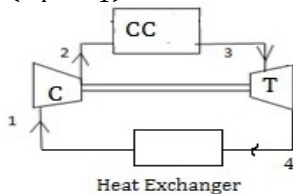
The P-V and T-S diagram of Brayton cycle has been shown.
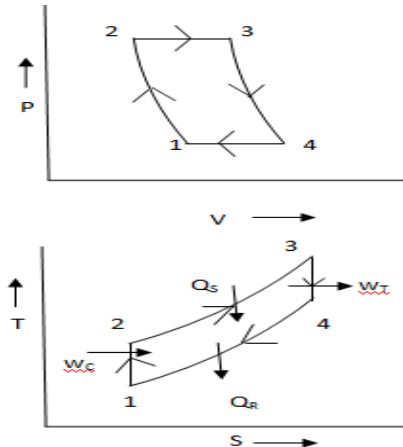
The efficiency of Brayton cycle is given by

$$\eta = 1 - \frac{Q_R}{Q_S} = 1 - \frac{C_P (T_4 - T_1)}{C_P (T_3 - T_2)}$$

$$= 1 - \frac{T_1 \left[\frac{T_4}{T_1} - 1 \right]}{T_2 \left[\frac{T_3}{T_2} - 1 \right]} \qquad \text{------ (8.1)}$$

process 1-2 and process 3-4 are isentropic process, so

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} = (r)^{\frac{\gamma-1}{\gamma}} = \frac{T_3}{T_4}$$

Where r = compression ratio

$$\frac{T_3}{T_2} = \frac{T_4}{T_1} \qquad \text{------ (8.2)}$$

substituting the values in equation 8.1, we get

$$\eta = 1 - \frac{T_1}{T_2}$$

$$\boxed{\eta = 1 - \frac{1}{(r)^{\frac{\gamma-1}{\gamma}}}}$$

➢ Thus efficiency of Bray ton cycle depends upon compression ration and nature of gas.

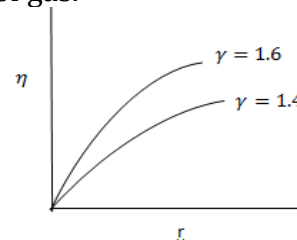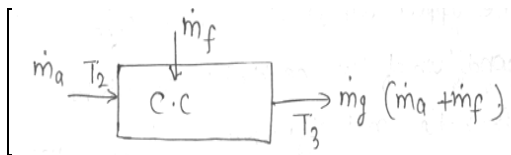## 8.3.2 ENERGY BALANCE EQUATION ON COMBUSTION CHAMBER

Heat released due to combustion of fuel is converted into enthalpy increases of gas in combustion chamber.

Heat input = Enthalpy change of gas

$$\dot{m}_f \times CV \times \eta_{c.c} = \dot{m}_g \dot{c}_{pg}[T_3 - T_2]$$

$$\dot{m}_f \times CV \times \eta_{c.c} = (\dot{m}_a + \dot{m}_f)c_{pg}[T_3 - T_2]$$

8.3.3 ANALYSIS OF GAS TURBINE CYCLE

The upper limit of temperature is fixed due to metallurgical condition of turbine blades. Though by increasing the Pressure Ratio, net efficiency $\left(1 - \dfrac{1}{(R_p)^{\frac{\gamma-1}{\gamma}}}\right)$ can be increased but net work will decrease. So there is a optimum limit of the compression ratio at which optimum efficiency and work output can be obtained.
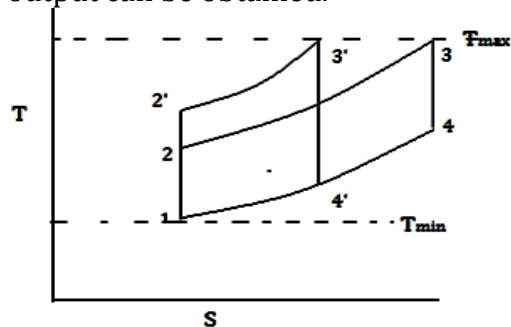
Net work done in gas power plant is given by

$$w_{net} = w_T - w_C$$

$$= C_P[(T_3 - T_4) - (T_2 - T_1)]$$

$$= C_P[(T_3 - T_4) - (T_2 - T_1)] \quad \text{----- (8.3)}$$

form the process 1-2 and process 3-4

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \Rightarrow T_4 = \frac{T_1 T_3}{T_2}$$

substituting the value in the equation 8.3

$$w_{net} = C_P\left[T_3 - \frac{T_3 T_1}{T_2} - T_2 + T_1\right]$$

for optimum condition differentiating the equation with respect to T_2

$$\frac{dw_{net}}{dt} = 0 = C_p\left[0 - \frac{T_3 T_1}{-T_2^2} - 1\right] = 0$$

$$T_2 = \sqrt{T_1 T_3}$$

$$T_4 = \frac{T_1 T_3}{\sqrt{T_1 T_3}} = \sqrt{T_1 . T_3}$$

$$T_4 = \sqrt{T_1 . T_3}$$

Compression ratio for optimum work done

$$r = \frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$$

$$r = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}}$$

$$w_{net} = C_P\left[T_3 - \sqrt{T_1 T_3} - \sqrt{T_1 T_3} + T_1\right]$$

$$w_{net\,max} = C_P\left[\sqrt{T_3} - \sqrt{T_1}\right]^2$$

8.3.4 REGENERATION IN GAS TURBINE

Efficiency of gas can be increased by utilizing the energy of exhaust gases from the turbine exit in heating air leaving the comp. In heat exchanger, and this process is called as regeneration.
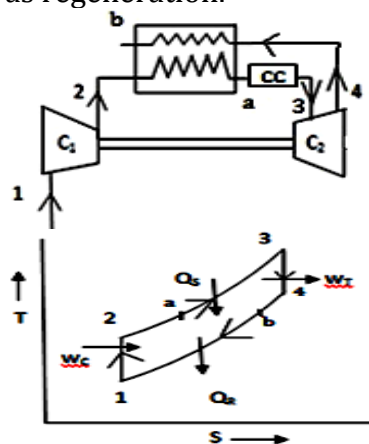
8.3.5 EFFECTIVENESS OF HEART TRANSFER

It is ratio of actual temperature rise of air to maximum possible rise of the

temperature. It is also known as degree of regeneration.

$$\in = \frac{\dot{m}_a C_{pa}(T_a - T_2)}{(\dot{m}_a + \dot{m}_f)C_{pa}(T_4 - T_2)}$$

If the mass of air is higher as compare to mass of the fuel, so the effectiveness of heat exchanger is

$$\in = \frac{Actual\,temp.Rise}{max^m\,possibleRise} = \frac{T_a - T_2}{T_4 - T_2}$$

8.3.6 EFFECT OF REGENERATION

1) There is no change in turbine work.
2) There is no change in compressor work.
3) Net work done remains same
4) Heat Supplied is decreased so the efficiency of the plant increase

Efficiency of ideal regeneration cycle

$$\eta = 1 - \frac{Q_R}{Q_S} = 1 - \frac{(T_b - T_1)}{(T_3 - T_a)}$$

for the perfect regeneration,

$$T_a = T_4, T_b = T_2$$

substituting the values in equation, we get

$$\eta = 1 - \frac{T_1\left(\frac{T_b}{T_1} - 1\right)}{T_3\left(1 - \frac{T_a}{T_3}\right)} \Rightarrow 1 - \frac{T_1\left(\frac{T_2}{T_1} - 1\right)}{T_3\left(1 - \frac{T_4}{T_3}\right)}$$

$$\eta = 1 - \frac{T_1}{T_3}\left[\frac{(r)^{\frac{\gamma-1}{\gamma}} - 1}{1 - (r)^{\frac{\gamma-1}{\gamma}}}\right]$$

$$\eta_{reg} \Rightarrow 1 - \frac{T_1}{T_3}(r)^{\frac{\gamma-1}{\gamma}}$$

So as compression ration increases the efficiency of simple Brayton cycle increases but the efficiency of regenerative Brayton cycle will decreases.
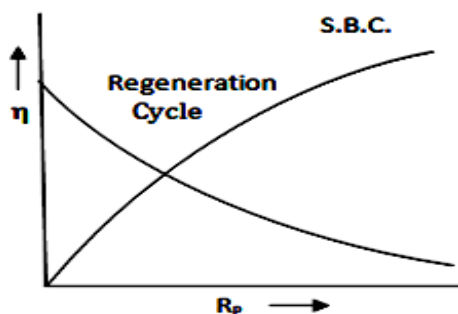
8.3.7 BRAY TON CYCLE WITH INTER COOLING

The efficiency of Brayton cycle can be improved by use of stages compression with inter cooling.

8.3.7.1 EFFECT OF INTER COOLING

1) Decrease in compressor work
2) No change in turbine work
3) Network increases
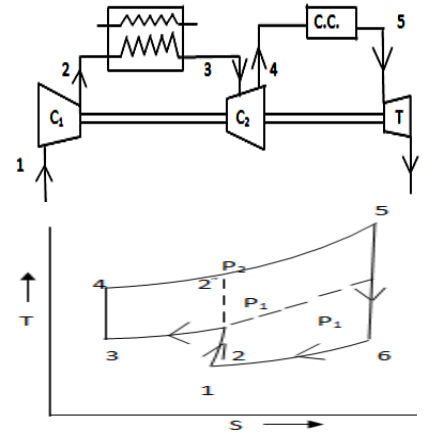
Optimum Pressure Ratio for minimum compressor work

$$w_C = w_{C1} + w_{C2}$$

$$= C_P(T_2 - T_1) + C_P(T_1 - T_3)$$

For perfect inter cooling $T_1 = T_3$

$$= C_P(T_2 - T_1) + C_P(T_4 - T_1)$$

$$= C_P T_1\left[\frac{T_2}{T_1} + \frac{T_4}{T_1} - 2\right]$$

$$\frac{T_2}{T_1} = \left(\frac{P_i}{P_1}\right)^{\frac{\gamma-1}{\gamma}}, \frac{T_4}{T_1} = \left(\frac{P_2}{P_i}\right)^{\frac{\gamma-1}{\gamma}}$$

$$w_C = C_P T_1\left[\left(\frac{P_i}{P_1}\right)^{\eta} + \left(\frac{P_2}{P_i}\right)^{\eta} - 2\right]$$

Differentiating the equation with respect to P_i, we get

$$p_i = \sqrt{p_1 p_2}$$

substituting the value of p_i in compressor work for perfect inter cooling

$$w_{C1} = C_P T_1\left[\left(\sqrt{\frac{P_2}{P_1}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$$w_{C2} = C_P(T_4 - T_3) = C_P T_3\left(\frac{T_4}{T_3} - 1\right)$$

$$w_{C2} = C_P T_1\left[\left(\sqrt{\frac{P_2}{P_1}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

8.3.8 BRAY TON CYCLE WITH REHEATING

The efficiency of Brayton cycle can be increased by reheating the gas and passing it through the number of the turbines.

Effect of reheating
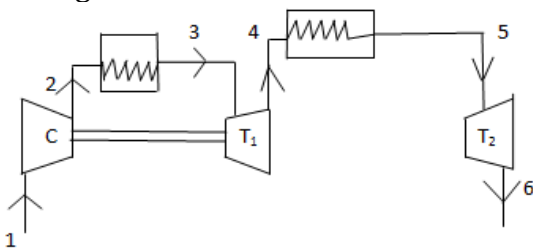1) Turbine work increases because const Pressure Line diverge
2) No change in compressor work
3) Network done increases

Conditions for perfect reheating
1) $T_3 = T_5$
2) $P_i = \sqrt{P_1 P_2}$
3) $w_{T1} = w_{T2}$
4) $w_T = 2w_{T1}$

8.3.9 ISENTROPIC EFFICIENCY OF COMPRESSOR AND TURBINE

Due to irreversibility turbine produce less work as compare to ideal work and compressor requires more work as compare to ideal work required for compressor.

efficiency of the compressor is given by

$$\eta_C = \frac{\text{Isentropic work}}{\text{actual work}} = \frac{h_2 - h_1}{h_2^1 - h_1} = \frac{T_2 - T_1}{T_2^1 - T_1}$$

efficiency of the turbine is given by

$$\eta_T = \frac{\text{actual work}}{\text{Isentropic work}} = \frac{h_3 - h_4}{h_3 - h_4^1} = \frac{T_3 - T_4}{T_3 - T_4^1}$$

8.4 STEAM POWER PLANT

In steam power plant, the working substance recalculates in two phases liquid and vapour. The components of simple vapour plant are shown in figure.
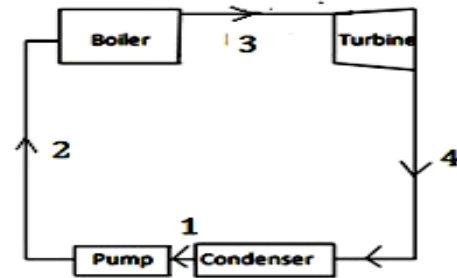
Steam power plant

Simple vapour plant works on Rankine cycle. Heat is transferred to water in the boiler from external source. On P-h and T-S diagram, Rankine cycle can be shown.
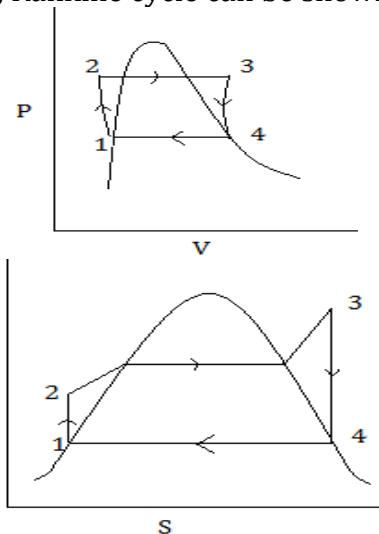
Rankine cycle consists of four processes:
Process1-2:IsentropicCompressionProcess: In this process woter is compressed isentropically in feed pump. Pump work can be given:
$W_p = h_2 - h_1$
Process 2 – 3 : Constant pressure Heat addition Process in the boiler: The heat is added to water in the boiler. It is given as
$Q_S = h_3 - h_2$
Process 3 – 4 : Isentropic Expansion Process in the turbine: In this process the heat transfer is zero and work done by the turbine is given as
$W_T = h_3 - h_4$

Process 4 – 1 : Constant pressure Heat rejection Process in the condenser: The heat is rejected at constant pressure process in condenser. Heat rejection in this process is given by

$$Q_R = h_4 - h_1$$

The efficiency of Rankine cycle is given by

$$\eta = 1 - \frac{Q_R}{Q_S} = 1 - \frac{h4 - h1}{h3 - h2}$$

8.4.1 REHEAT RANKINE CYCLE

In reheating cycle, the expansion of steam from the initial state at turbine is carried out in two or more no. of re heater. The network out increases and hence steam rate decreases.

$$P = m \times w_{net}$$

1) Work done by the turbine
$$w_T = (h_3 - h_4) + (h_5 - h_6)$$

2) Work done by the turbine
$$w_p = h_2 - h_1 = vdp$$

3) Heat supplied in the boiler
$$Q_S = (h_3 - h_2) + (h_5 - h_4)$$

$$\eta = \frac{w_T - w_p}{Q_S}$$

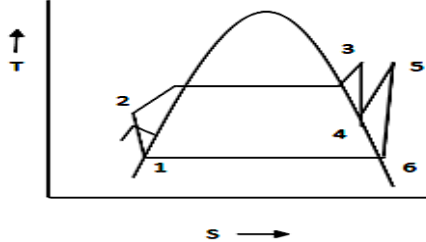➢ The main advantage of reheat factor is to increase dryness fraction at the turbine exit.

8.4.2 REGENERATION IN RANKINE CYCLE

In order to increase the mean temperature of the heat addition so that heat can be transferred at higher temperature. So the efficiency of the cycle can be increased. In this cycle, it is possible by transferring the heat from vapour as it flows through the turbine to the liquid flowing around the turbine. With the help of open feed water heater (OFWH) heat is transferred between

the steam from turbine and water just before entering the boiler. Regenerative cycle can be shown on T-S diagram.
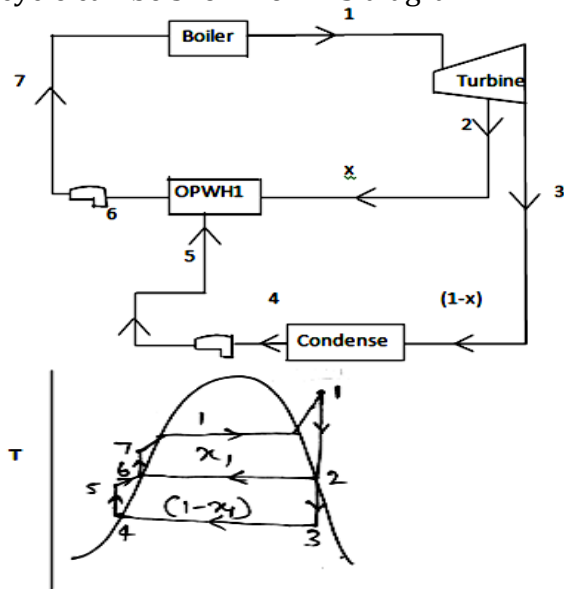
Turbine work in regeneration is given by
$$w_T = (h_1 - h_2) + (1 - x_1)(h_2 - h_3)$$
Energy balance equation on OFWH
$$xh_2 + (1 - x)h_5 = h_6$$

8.4.3 EFFECT OF PARAMETERS ON PERFORMANCE OF STEAM POWER PLANT

1) Effect of Boiler pressure If the boiler pressure is increased, the effect on the performance can be checked with help of T-S diagram
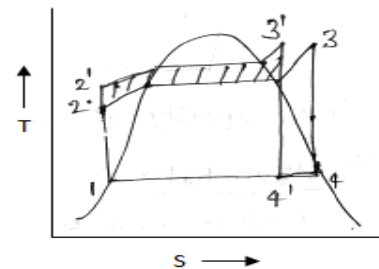
1) w_p increases
2) w_T – same
3) Q_S Decreases
4) η increases
5) Dryness fraction at outlet of turbine decreases

2) Effect of Superheating
If steam is superheated in the boiler, the effect on the performance can be checked with help of T-S diagram
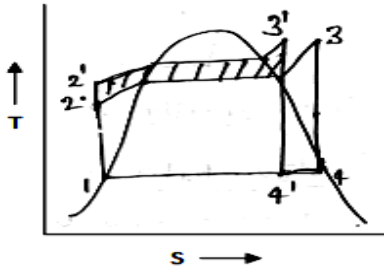
1) w_p same

2) w_T increases

3) w_{net} increases

4) Q_S increases

5) η iincreses

6) Dryness at the outlet of turbine increases

8.5 NOZZLE AND DIFFUSER

8.5.1 NOZZLE

It is a device of varying cross section used to convert the Pressure Energy of working fluid into kinematic energy. It is mainly used to produce the jet of steam or gas of high velocity to produce throat for Jet propulsion to derive steam turbine or gas turbine.

8.5.2 DIFFUSER

It is a device of varying cross section causing rise in Pressure Energy atthe cost of kinematic energy. These are used in centrifugal and axial compressor.

➢ The smallest section of nozzle is known as throat.

8.5.3 SPEED OF SOUND

It may be defined as velocity of Pressure Wave in fluid medium.

$$C = \sqrt{\left(\frac{dp}{dp}\right)_s}$$

8.5.4 VELOCITY OF SOUND IN IDEAL GAS

For an ideal gas, flow is isentropic

$$Pv^\gamma = C$$

$$\frac{P}{\rho^\gamma} = C$$

Taking log in both sides

$$l_g p - \gamma l_g \rho = l_g C$$

differentiating the equation, we get

$$\frac{dp}{p} - \gamma \frac{d\rho}{\rho} = 0$$

$$\left(\frac{dp}{d\rho}\right)_S = \frac{\gamma p}{\rho}$$

$$C^2 = \frac{\gamma \rho RT}{\rho}$$

So the velocity of sound in a medium is given by

$$C = \sqrt{\gamma RT}$$

Where R is the specific Gas constant.

8.5.5 MACH NUMBER

Mach number is defined as the ratio of actual velocity to the velocity of the sound or sonic velocity.

$$M = \frac{V}{C}$$

#M>1, then V>C flow is supersonic flow.

M = 1 , then V = C flow is Sonic flow.

M < 1 then V < C flow is Subsonic flow.

M>5 then V>5C flow is hypersonic flow.

8.5.6 STAGNATION POINT AND STAGNATION PROPERTIES

Stagnation point is state in the fluid flow where velocity of flow is become Zero and kinetic energy converts into pressure energy.

The Pressure, Density, temperature etc. of stagnation point are known as stagnation Properties.
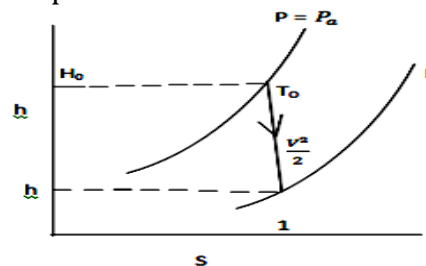
for the ideal gas at stagnation point,

$$h_0 = h + \frac{V^2}{2}$$

$$C_P T_O = C_P T + \frac{V^2}{2}$$

$$T_O = T + \frac{V^2}{2C_P}$$

We know that $C_P = \frac{\gamma R}{\gamma - 1}$

$$T_0 = T + \frac{V^2(\gamma - 1)}{2(\gamma R)}$$

$$\frac{T_0}{T} = 1 + \frac{V^2}{2\gamma RT}(\gamma - 1)$$

$$\frac{T_0}{T} = 1 + \frac{(\gamma - 1)}{2} \cdot \frac{V^2}{C^2}$$

$$\frac{T_0}{T} = 1 + \frac{(\gamma - 1)}{2} M^2$$

$$\frac{P_0}{P} = \left(\frac{T_0}{T}\right)^{\frac{\gamma}{\gamma - 1}}$$

8.5.7 FLOW THROUGH NOZZLE & DIFFUSER

Assumption for flow
1) flow is the isentropic flow.
2) Stagnation enthalpy is constant $h_0 = c$

$$h_0 = h + \frac{V^2}{2} ----(8.4)$$

differentiating the equation
dh $= -VdV$ ------ (8.5)
From combined first and second law of thermodynamic
$TdS = dh - VdP$
differentiating the equation, we get

$$dh = VdP = \frac{dP}{\rho} -------- (8.6)$$

or $\frac{dP}{\rho} = -Vd$

$$\frac{d\rho}{dV} = -\rho V$$

$$dV = -\frac{dP}{\rho V} -------- (8.7)$$

Continuity equation for the mass flow
$\dot{m} = \rho AV$
taking log in both the sides

$\log m = \log \rho + \log A + \log V$
differentiating the equation

$$0 = \frac{d\rho}{\rho} + \frac{dA}{A} + \frac{dV}{V}$$

$$\frac{dA}{A} = -\frac{d\rho}{\rho} + \frac{dP}{PV^2}$$

$$\frac{dA}{A} = \frac{dV}{V}\left[1 - \frac{d\rho}{\rho} X \frac{V}{dV}\right]$$

Substituting the value of dV from equation (8.7), we get

$$\frac{dA}{A} = \frac{dP}{\rho V^2}\left[1 - \frac{V^2}{C^2}\right]$$

$$\frac{dA}{A} = \frac{dV}{V}\left[M^2 - 1\right]$$

1) When M < 1, flow is Subsonic flow

a) Converging passage $\left(\frac{dA}{A} = -Ve\right)$
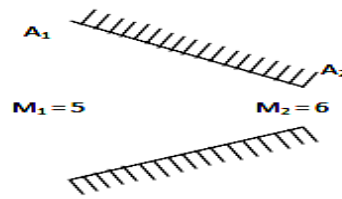
$A\downarrow \Rightarrow V\uparrow \Rightarrow P\downarrow \Rightarrow M\uparrow (\rho\downarrow)$

b) Diverging passage $\left(\frac{dA}{A} = +Ve\right)$
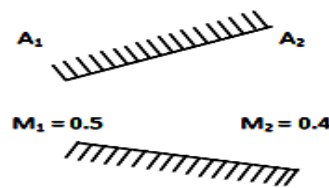
$A\uparrow \Rightarrow V\downarrow \Rightarrow P\uparrow \Rightarrow M\downarrow (\rho\uparrow)$

2) When M > 1 flow is supersonic flow

a) $\left(\frac{dA}{A} = -Ve\right)$
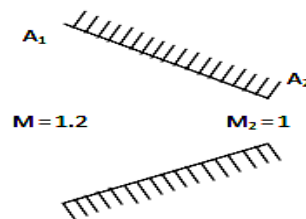
$A\downarrow \Rightarrow V\downarrow \Rightarrow P\uparrow \Rightarrow M\downarrow (\rho\uparrow)$

b) Diverging Passage $\left(\frac{dA}{A} = +Ve\right)$
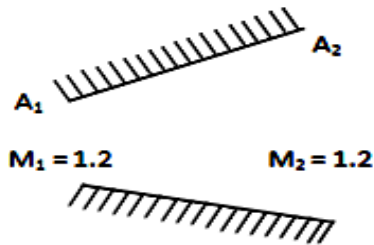
$$A \uparrow \Rightarrow V \uparrow \Rightarrow P \downarrow \Rightarrow M \uparrow (\rho \downarrow)$$

Conclusion

a) For subsonic flow and compressible fluid convergent passage acts as a nozzle and divergent passage acts as a diffuser.

b) For supersonic flow and compressible fluid convergent passage acts as diffuser & divergent passage acts as a nozzle.

c) For an incompressible fluid converging passage always aits as nozzle and divergent as a diffuser.

$$\frac{\rho}{\rho} + \frac{dA}{A} + \frac{dV}{V} = 0$$

$$\frac{dA}{A} = -\frac{dV}{V}$$

8.5.8 MASS FLOW RATE PER UNIT AREA

mass flow rate per unit area is given by

$$\frac{\dot{m}}{A_2} = \rho_1 \sqrt{2 C_P T_1 \left[(r)^{\frac{2}{n}} - (r)^{\frac{n+1}{n}} \right]}$$

For maximum flow rate

$$\frac{d}{dR}\left(\frac{\dot{m}}{A_2}\right) = 0$$

$$r = \frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

$\dfrac{\dot{m}}{A}$ will be max when Area is minimum and it occurs at throat. So the critical condition occurs at throat. So Critical pressure is given by

$$\frac{P_c}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

8.6 COMPRESSOR

A compressor is a device in which work is done on the gas to raise the pressure of the gas. The work done by the compressor is negative in the compression process because the work is done on the system to raise the pressure of the gas.

8.6.1 WORK DONE IN COMPRESSOR

Process of compression in compressor is shown in the fig.

1) Work done in adiabatic process

$$W = \frac{\gamma}{\gamma - 1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$
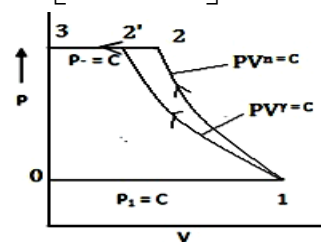
2) Work done in polytropic process

$$W = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right]$$

8.6.2 ACTUAL DIAGRAM OF COMPRESSOR

The actual work done will be more due to resistance offered by suction value and discharge value. The shaded portion represents the extra work to counter value resistance. The actual diagram with the clearance volume is shown in fig.
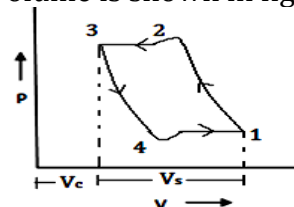
Fig.: Actual P-V diagram of compressor

8.6.3 CONDITION FOR MINIMUM WORK REQUIRED IN MULTISTAGE COMPRESSOR WITH PERFECT INTER COOLING

For perfect inter cooling

$$T_1 = T_3$$

$$P_2 = \sqrt{P_1 P_3}$$

For multi stage compressor

$$\frac{P_2}{P_1} = \frac{P_3}{P_2} = \frac{P_4}{P_3} = \frac{P_{n+1}}{P_n} = C$$

$$\frac{P_{N+1}}{P_1} = C^N$$
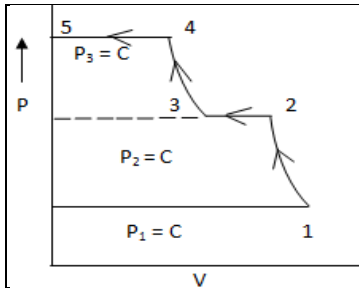
8.7 STEAM TURBINE

Steam turbine is a prime mover which is used to convert the potential energy into the kinetic energy and then utilize it into the mechanical work. Main use of steam turbine is to produce the mechanical work in steam power plant.

8.7.1 TYPES OF TURBINE

There are two types of turbines:
1) **Impulse Turbine:** In impulse turbine pressure drops in nozzles but remains same in moving blade.
2) **Reaction Turbine:** In reaction turbine pressure drops in fixed blade and it also changes during the expansion of the steam in moving blades.

8.7.2 COMPOUNDING IN TURBINE

In this type of compounding two shafts are used to drive separate generator at different RPM. There are two type of compound turbine:

1) **Pressure compounding-** Rateua turbine
2) **Velocity compounding -** Curtis turbine

8.7.3 VELOCITY TRIANGLE OF TURBINE

The velocity triangle is shown in the fig. From the outlet of nozzle, steam flow at absolute velocity V_1. But due to blade velocity u, steam strikes at the blade with relative velocity V_{r1}.

Following are the parameters of the turbine.
α=Angle at which steam enters or guide blade angle
θ= Vane angle at inlet
ϕ= Vane angle at outlet
β= Angle at exit of moving blade
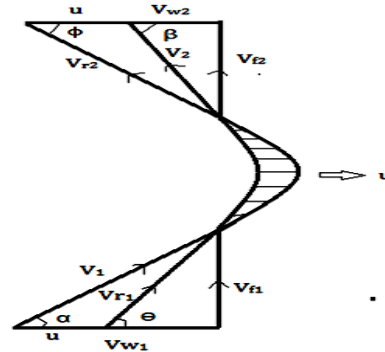
fig.: velocity triangle

$V_1 \& V_2$ = Absolute velocity at entrance and exit

$V_{w1} \& V_{w2}$ = Velocity of whirl at entry and at exit or tan agent of absolute velocity

$V_{r1} \& V_{r2}$ = Relative velocity at entry and exital blade

$V_{f1} \& V_{f2}$ = axial component of absolute velocity

u = blade velocity

8.7.4 PERFORMANCE PARAMETERS OF TURBINE

1) Velocity Ratio
It is the ratio of blade velocity to the flow velocity at the inlet of the blade. It is given by ρ.

$$\rho = \frac{u}{V_1}$$

2)Blade or Diagram efficiency
It is ratio of net work done by turbine to kinetic energy at the inlet of the blade.

$$\eta_D = \frac{2u(V_{w1} + V_{w2})}{V_1^2}$$

If $\dot{m}_a$ $T_2 \theta = \varnothing, V_{r1} = V_{R2}$ then $V_{w1} = V_1 \cos \propto$

$$V_{w2} = V_{r2}\cos\varnothing - u = V_{r2}\cos\varnothing - u$$
$$= (V_1\cos\propto -u) - u = V_1\cos\alpha - 2u$$

$$\eta_D = \frac{2u(V_{w1} + V_{w2})}{V_1^2} = \frac{2u(2V_1\cos\cos\alpha - u)}{V_1^2}$$

$$\eta_D = \frac{4u\cos\propto}{V_1} - 4\frac{u^2}{V_1^2} = 4\rho\cos\alpha - 4\rho^2$$

$$\eta_D = 4\rho(\cos\alpha - \rho) \quad \text{--- (8.8)}$$

For maximum diagram efficiency, differentiating the equation with respect to ρ.

$$\frac{d\eta_D}{d\rho} = 0 = 4\cos\alpha - 8\rho$$

$$\rho = \frac{\cos\alpha}{2}$$

Substituting the value of ρ in equation 8.8, get

$$\eta_{Dmax} = 4\frac{\cos\propto}{2}\left[as\propto -\frac{as\propto}{2}\right] = \cos^2\propto$$

$$\eta_{Dmax} = \cos^2\propto$$

For n number of stages (Curtis stage)

$$\rho_{opt} = \frac{\cos\alpha}{2\eta}$$

$$\eta_{Dmax} = \cos^2\alpha$$

3)Blade or diagram efficiency of reaction turbine

$$\eta_D = \frac{2(2\rho\cos\alpha - \rho^2)}{1 + 2\rho\cos\alpha - \rho^2}$$

For maximum diagram efficiency, differentiating the equation with respect to ρ, we get

$$\frac{d\eta_D}{d\rho} = 0$$

$$\rho = \cos\alpha$$

Substituting the value of in equation of diagram efficiency of reaction turbine, we get

$$\eta_{Dmax} = \frac{2[2\cos^2\alpha]}{1 + \cos^2\alpha}$$

4) Stage efficiency
It is the ratio of the net work done to change of enthalpy at the inlet of the turbine.

$$\eta_{stage} = \frac{u(V_{w1} + V_{w2})}{(h_1 - h_2)}$$

Examples

Q.1 Determine the minimum no. of stage required in an air comp. which takes air at 1 bar 27^0C and delivers 180 bar. The max discharge temperature is 150^0C and considers the index of polytrophic as 1.25 and perfect inter cooling b/w stages.

Solution:

$$\frac{P_{N+1}}{P_1} = C^N$$

$$\frac{P_{n+1}}{P_1} = \left(\frac{T_2}{T_1}\right)^{\frac{1.25}{1.25}} = 5.57^N$$

$$\frac{180}{1} = 5.57^N$$

N= 3

Q.2 A gas turbine plant operates on Brayton cycle, between T_{min} =300K and T_{max}=1073K. Find the maximum work done per kg of air and cycle efficiency. Compare the efficiency with the efficiency of Carnot cycle with same temperature limit.

Solution:

$$W_{netmax} = Cp\left(\sqrt{T_{Max}} - \sqrt{T_{Min}}\right)^2$$

$$= 1.005\times\left(\sqrt{1073} - \sqrt{300}\right)^2$$

$$= 239.28 \text{ KJ/Kg}$$

Efficiency of the Brayton cycle

$$\eta = 1 - \frac{1}{r^{\frac{\gamma-1}{\gamma}}}$$

$$= 1 - \frac{\sqrt{T_{Min}}}{\sqrt{T_{Max}}} = 1 - \frac{\sqrt{300}}{\sqrt{1073}} = 47\%$$

Efficiency of the Carnot cycle

$$\eta_C = 1 - \frac{T_{min}}{T_{max}} = 1 - \frac{300}{1073} = 72.1\%$$

$$\frac{\eta}{\eta_C} = \frac{0.47}{0.721} = 0.65$$

Q.3 A gas turbine plant draws in air at 1.013 bar, 10°C and has a pressure ratio of 5.5. the maximum temperature in the cycle is limited to 750°C. Compression is conducted in rotary compressor having an isentropic efficiency of 82%, and expansion takes place in a turbine with an isentropic efficiency of 85%. A heat exchanger with an efficiency of 70% is fitted between the compressor outlet and combustion chamber. For an air flow of 40 kg/s, find:

a) The overall cycle efficiency,
b) The turbine output, and

Solution:

$P_1 = 101.3$ KPa, $T_1 = 283$ K, $r = 5.5$,

$T_4 = 750°C = 1023$ K

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = (r)^{\frac{\gamma-1}{\gamma}} = (5.5)^{\frac{1.4-1}{1.4}}$$

$T_2 = 460.6$ K

Isentropic efficiency of compressor

$$\eta_C = \frac{\text{Isentropic work}}{\text{actual work}} = \frac{h_2 - h_1}{h_2^1 - h_1} = \frac{T_2 - T_1}{T_2^1 - T_1}$$

$$0.82 = \frac{T_2 - T_1}{T_2^1 - T_1} = \frac{460.6 - 283}{T_2^1 - 283}$$

$T_2' = 499.6$ K

$$\frac{T_4}{T_5} = (r)^{\frac{\gamma-1}{\gamma}} = (5.5)^{\frac{1.4-1}{1.4}}$$

$T_5 = 628.6$ K

Efficiency of turbine

$$\eta_T = \frac{\text{actual work}}{\text{Isentropic work}} = \frac{T_4 - T_5}{T_4 - T_5^1}$$

$$0.85 = \frac{1023 - 638.6}{1023 - T_5^1}$$
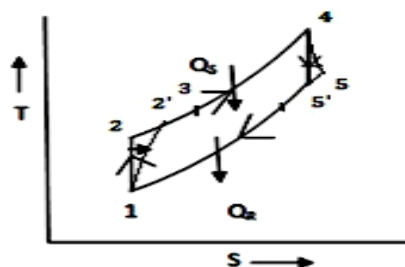
$T_5' = 687.7$ K

Maximum possible heat from heat exchanger $= C_p(T_5 - T_2)$

Actual Heat transfer $= 0.7 C_p(T_5 - T_2)$

$= 132$ KJ/kg of air

$C_p(T_3 - T_2) = (1 + m) \times 132$

$C_p T_3 = 634.43 + 132.33$ m

$m = 8.68 \times 10^{-3}$ kJ/kg of air

Heat addition $(Q_1) = C_p(T_4 - T_3) = C_p T_4 - C_p T_3$

$= 393.7 - 132.33$m

$= 392.6$ kJ/kg of air

$W_T = (1 + m)(h_4 - h_5)$

$= (1 + m) Cp (T_4 - T_5')$

$= 1.00868 \times 1.005 \times (1023 - 687.7)$ kJ/kg of air

$= 340$ kJ/kg

$W_c = (h_2 - h_1) = Cp(T_2' - T_1) = 1.005 \times (499.6 - 283)$

$= 217.7$ kJ/kg of air

$W_{net} = W_T - W_c = 122.32$ KJ/kg

a) $\eta = \frac{\text{Networkdone}}{\text{Heatsupplied}} = \frac{122.32}{392.6} = 31.16\%$

b) Turbine output $= (W_T) = 122.32$ kJ/kg of air $= 4893$ kW

Q.4 A simple steam power cycle uses solar energy for the heat input. Water in the cycle enters the pump as a saturated liquid at 40°C, and is pumped to 2 bar. It then evaporates in the boiler at this pressure, and enters the turbine as saturated vapour. At the turbine exhaust the conditions are 40°C and 10% moisture. The flow rate is 150 kg/h. determine:

a) the turbine isentropic efficiency,
b) The net work output
c) The cycle efficiency

Solution

From Steam Table

$T_1 = 120.23°C = 393.23$ K, $h_3 = 2706.7$ kJ/kg, $s_3 = 7.1271$ kJ/kg – K

At 40°C saturated pressure 7.384 kPa

$h_f = 167.57 KJ/Kg$, $h_{fg} = 2406.7 KJ/Kg$

$s_f = 0.5725$, $s_g = 8.2570$

$h_4' = h_f + 0.9 \times 2406.7$

$= 2333.6$ kJ/kg
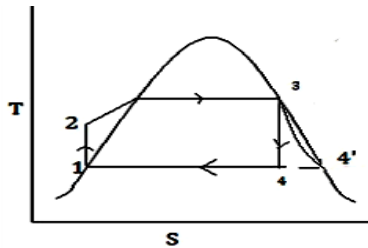
For h4s if there is dryness fraction x

$7.1271 = 0.5725 + x \times (8.2570 - 0.5725)$

$x = 0.853$

$h_4 = 167.57 + 0.853 \times 2406.7$

$= 2220.4$ kJ/kg

Isentropic efficiency,

$$\eta_{isentropic} = \frac{h3 - h4}{h3 - h4'} = \frac{2706.7 - 2333.6}{2706.7 - 2220.7}$$

$= 76.72\%$

Net Turbine work output $W_T = h_3 - h_4$

$= 373.1$ kJ/ kg

Pump work, $WP = v(P_1 - P_2)$

$= 1.008 \times 10^{-3}(200 - 7.384)$ kJ/kg

$= 0.1942$ kJ/kg

Power $= m \times (W_T - W_p) = 15.55$ kW

$h_3 = 2706.7$ kJ/kg, $h_2 = 167.76$ kJ/kg

$Q_1 = (h_3 - h_2) = (2706.7 - 167.76)$ kJ/kg

$= 2539$ kJ/kg

$$\eta_{cycle} = \frac{W_T - W_P}{Q_1} = \frac{373.1 - 0.1942}{2539} = 14.69\ \%$$

Q.5 A regeneration Rankin cycle as steam entering turbine at 200 bar and 250°C superheated state and leaving at 0.05 bar. Considering feed water to be open type. Determine the efficiency of plant when feed water is operating at 8 bar.

Solution:

At 200 bar and 250°C

$h = 3675.2 KJ/Kg$, $s = 6.6502 KJ/Kg.K$,

$v_f = 0.001005$ m³/Kg

At 0.005 bar $h_f = 137.82$ KJ/Kg,

$h_{fg} = 2423.7 KJ/Kg$,

$s_f = 0.4764 \dfrac{KJ}{Kg}.K$ $s_{fg} = 7.9187 KJ/Kg.K$

$v_f = 0.001005$ m³/ Kg

At 8 bar

$h_f = 721.11 KJ/Kg$ $h_{fg} = 2048 KJ/Kg$,

$s_f = 2.0462 \dfrac{KJ}{Kg}.K$ $s_{fg} = 4.6166$

KJ/Kg.K

$v_f = 0.001115$ m³/ Kg

Regeneration cycle can be shown on T-s diagram
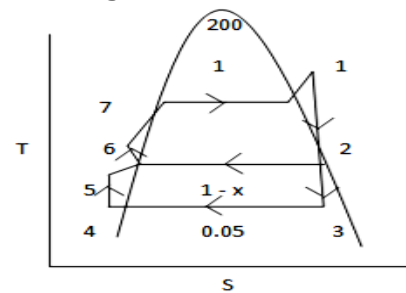
Process 1-2 is isentropic process

$s_1 = s_2$, $h_1 = 3675.2$

$6.6502 = 2.0462 + x_2\ 4.6166$

$\boxed{x_2 = 0.997}$

Enthalpy of point 2

$h_2 = h_{f2} + x_2 h_{fg2} = 721.11 + 0.997 \times 2048$

$h_2 = 2763.54$ kJ/kg

Process 1-3 is isentropic process

$s_1 = s_3$

$6.6502 = 0.4764 + x_3\ X\ 7.9187$

$\boxed{x_3 = 7796}$

Enthalpy of Point 3

$h_3 = h_{f3} + x_3 h_{fg3}$

$= 137.82 + 0.7796 X 2423.7$

$= 2027.45 kJ/kg$

Work done by pump 1

$w_{p1} = h_5 - h_4 = vdp = 0.001005(8 - 0.05) \times 10^2$

$= 0.798$

$h_5 = 0.798 + 137.82 = 138.618$ kJ/kg

Work done by pump 1

$w_{p2} = h_7 - h_6 = 0.001115(200 - 8) X 10^2$

$= 22.08\ kJ/kg$

$h_7 = h_6 + 22.08 = 723.19$ kJ/kg

Energy Balance equation at open feed water heater

$x \times h_2 + (1-x)h_5 = h_6$

$x \times 2763.54 + (1-x) \times 138.618 = 721.11$

x = 0.2919

$w_{net} = (h_1 - h_2) + (1-x)(h_2 - h_3)$

$-(w_{p1} + w_{p2})$

$= (3675.2 - 2763.54) + (1 - 0.2919)$

$(2763.54 - 2027.45) - (798 + 22.08)$

$= 1462.4 KJ/kg$

$Q_s = h_1 - h_7 = 3675.2 - 723.19$

$= 2932.78$ kJ/kg

Efficiency of the plant

$\eta = \dfrac{w_{net}}{Q_s}$

$= \dfrac{1462.4}{2932.78} = 49.86\%$

Q.6 A stream of air flow in a duct of 100mm diameter at the rate of 1 kg/sec. The stagnation temperature is 37⁰C. At one section of the duct the static pressure is 40 KPa. Calculate the Mach number, velocity and stagnation pressure at that section.

Solution:

T_0 = 37+ 273 =310 K, P= 40 KPa, and γ = 1.4

The mass flow rate per unit area is

$\dfrac{m}{A} = \rho \times V = \dfrac{P}{RT} \times M\sqrt{\gamma RT}$

$= \sqrt{\dfrac{\gamma}{R}} \times \dfrac{PM}{\sqrt{T_0}}\sqrt{\dfrac{T_0}{T}} = \sqrt{\dfrac{\gamma}{R}} \times \dfrac{PM}{\sqrt{T_0}}\sqrt{1+\dfrac{(\gamma-1)M^2}{2}}$

$\dfrac{1}{\dfrac{\pi 0.1^2}{4}} = \sqrt{\dfrac{1.4}{0.287}} \times \dfrac{40M}{\sqrt{310}}\sqrt{1+\dfrac{(1.4-1)M^2}{2}}$

$127.39 = \sqrt{\dfrac{1.4}{0.287 \times 10^3 \times 310}} \times 40 \times 10^3 M\sqrt{1+0.2M^2}$

$0.803 = M\sqrt{1+0.2M^2}$

Squaring both the sides

$0.645 = M^2(1+0.2M^2)$

$0.2 M^4 + M^2 - 0.645 = 0$

Solving the equation, we get

M = 0.76

$\dfrac{T_0}{T} = 1 + \dfrac{(\gamma-1)M^2}{2} = 1 + \dfrac{(1.4-1) \times 0.76^2}{2}$

T = 277.78K

$C = \sqrt{\gamma RT}$

$= \sqrt{1.4 \times 0.287 \times 277.78 \times 10^3}$

= 334.08 m/sec

V = c × M = 334.08×0.76 = 253.9 m/s

$\dfrac{P_0}{P} = \left(\dfrac{T_0}{T}\right)^{\frac{\gamma-1}{\gamma}} = \left(\dfrac{310}{277.78}\right)^{\frac{1.4-1}{1.4}}$

$P_0 = 40 \times 1.468 = 58.72$ KPa

Q.7 A simple open cycle gas turbine has a compressor turbine and a free power turbine. It develops electrical power output of 250 MW .the cycle takes in air at 1 bar and 288 K. The total compressor pressure ratio is 14. The turbine inlet of compressor is 1500 K. The isentropic efficiency of compressor and turbine is 0.86 and 0.89 respectively. The mechanical efficiency of each shaft is 0.98. Combustion effect is 0.98 while combustion pressure loss is 3% of compressor delivery pressure. Alternator efficiency is 0.98. Take calorific value of fuel is 42000 KJ/kg. Cpa = 1.005 KJ/Kg K and Cpg = 1.15 KJ/Kg. K. Calculate

i) Air –fuel ratio

ii) Specific work output

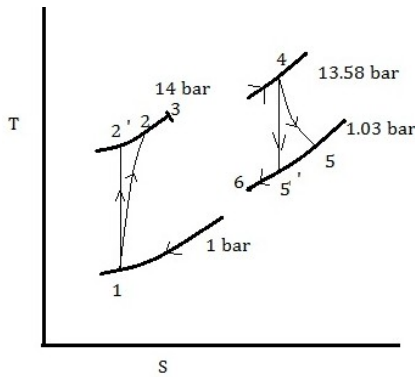**Solution:**

Electrical power =250mw

Power output from power turbine

$= \dfrac{250}{\text{alternator efficiency} \times \text{mechanical efficiency}}$

$= \dfrac{250}{0.98 \times 0.98} = 260.3082$ MW

= 260308.2 KW

$= 288(14)\dfrac{0.4}{1.4} = 612.15$ K

$T_2' = T_1(14)\dfrac{\gamma-1}{\gamma} \quad T_2 = T_2 + \dfrac{T21-T1)}{\eta c}$

$$= 288 + \frac{612.15 - 288}{0.86} = 664.92K$$

Compressor is run by compressor turbine

MaCpa(T_2 - T_1) = ma (1 + f) (T_4 – T_5) cpg × 0.98

1.005 × (664.92-288) = (1+t) ($t_u t_5$) × 1.15 × 0.98

$$(1+f)(T_4 - T_5)\frac{250}{0.98 xo.98} = \frac{329.4}{0.98}$$

(1+F)(1500-T_5) = 336.12 --------- (1)

Total efficiency of turbine = 0.89

$$\frac{T_4^{1.33}}{P_4^{1.33-1}} = \frac{T_6^{1.33}}{P_6^{1.33-1}}$$

$$T_6^1 = T_4\left(\frac{P_6^1}{P_4}\right)^{\frac{0.33}{1.33}} = 791k$$

$$0.89 = \frac{T_4 - T_6}{(T_6 - T_6^1)} = \frac{1500 - T_6}{1500 - 791} =$$

T_6 = 1500 – 0.89 (1500 – 791)

T_6 = 869K

Power output from power turbine,

M_a(1+F) Cpg (T_5 − T_6) = 260308.2

M_a(1+F)×1.15(T_5 −869) = 260308.2

M_a(1+F) (T_5 − 869) = 226355

Heat balance equation in turbine,

M_a(1+F)CpgT_4 − M_aCpaT_2 =

M_aCV × combustion effiency(1+F)

×1.15×1500 − 1.005×664.92 = 41160 f

1725 – 668.25 = (41160 – 1725) F

F = 0.0268

i) Air fuel ratio = $\frac{1}{F}$ = 37.32

ii) Specific work output

Substituting the value of F in equation 1

(1+0.0268) (1500 - T_5) = 336.12

T_5 = 1500 – 327.35

T_5 = 1172.65 K

M_a (1+0.0268) (T_5 − 869) = 226355

M_a = 725.987

M_a = 726 kg/sec

Special output =(1+F) Cpg(T_5 − T_6)

= 1.0268 × 1.15 × (1172.65 – 869)

= 358.56 KW/Kg

Q.8 In a multi-stage parson's reaction turbine at one of the stages the rotor diameter is 125 cm and speed ratio 0.72. The speed of the rotor is 3000rpm. Determine
1) The blade inlet angel if the blade outlet angle is 22°
2) Diagram efficiency

Solution:

Parson's reaction turbine

D_1=1.25m, N=300rpm

ρ = u/v_1=0.72m, $β_1$=?, $β_2$=22

For parson's reaction turbine

$α_1$ =$β_2$ and $α_2$= $β_1$

$$u = \frac{π × D × N}{60} = \frac{π × 1.25 × 3000}{60}$$

= 196.34 m/s

V_1 = u/V1 = u/0.72 = 272.69 m/s

V_{r2} = V_1 = 272.69 m/s

Inlet velocity triangle and Outlet velocity triangle
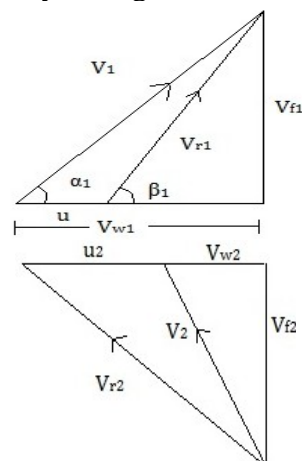
Inlet velocity triangle outlet velocity triangle

V_{w1} = 272.69 cos20 = 252.83 m/s

V_{f1}= 272.69 sin20 = 102.15 m/s

$$\tan \beta_1 = \frac{Vf1}{vw1 - v1} = \frac{102.15}{252.83 - 196.34}$$

So, $\beta_1 = 61.05$ and $\alpha_2 = \beta_1 - 61.05$

diagram efficiency

$$\eta_{d=} \frac{power}{\frac{1}{2}m[V1^2 + V2^2 - Vf1^2]}$$

power = mu ($v_{w1} + v_{w2}$)

u = 196.34 m/s

v_{w1} = 252.83 m/s

From outlet velocity triangle

$v_{r2} \cos \beta_2 = u + V_{w2}$

272.69 cos 22 = 196.34 + V_{w2}

V_{w2} = 56.49 m/s

Power = 1 × 196.34 × (25 + 56.49)

= 60.73KW

$$\eta_d = \frac{60 \times 10^3}{\frac{1}{2}m[272.69^2 + 272.69^2 - 102.15^2]}$$

= 0.90 or 90 %

Q.9 A de level turbine has a mean blade speed of 180 mps. The nozzles are inclined at 17^0 to the tangent. The steam flow velocity through the nozzle is 550 m/sec. For a mass flow of 3300 kg/hour and for axial exit condition: Find
1) The inlet and outlet angles of the blade system.
2) The power output.
3) Diagram efficiency.

Solution:

u= 180 m/s

Nozzle angle $\overset{\circ}{\alpha} = 17^0$, Mass flow rate = 3300 kg/hr = 0.9167 kg/sec

V_1 = 550 m/s

Inlet velocity triangle and outlet velocity triangle
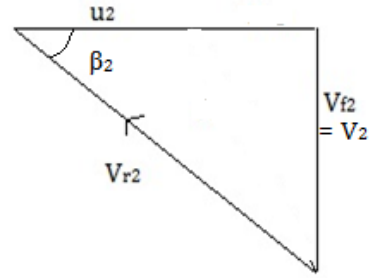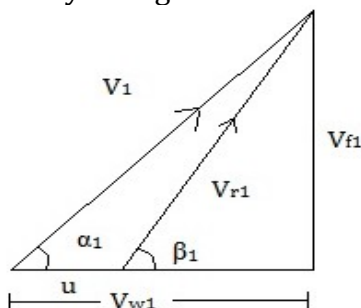
Inlet velocity triangle Outlet velocity triangle from velocity triangle

V_{w1} = v_1 cosα = 550 cos17 = 525.96 m/s

V_{f1} = v_1 sinα = ssosinl7 = 160.80 m/s

$$\tan\beta_1 = \frac{vf1}{vw1 - v1}$$

$$\tan\beta_1 = \frac{160.80}{525.96 - 180}$$

$\beta_1 = 24.92$

$$Tan\beta_2 = \frac{vf2}{v2}$$

$$\tan\beta_2 = \frac{vf1}{u} = \frac{160.80}{180}$$

$\beta_2 = 41.27°$

Power output = m ($V_{w1} + V_{w2}$) u

v_{w1} = 525.96 m/s

v_{w2} = 0 (due to axial outlet condition)

u = 180 m/s

$$P = \frac{3300}{3600}(525.96 + 0) \times 180$$

P = 86.78 KW

iii) Diagram efficiency

$$\eta = \frac{2(vw1 + vw2) \times u}{V_1^2} \times 100$$

$$\eta = \frac{2 \times (525.96 + 0) \times 180}{(550)2} \times 100$$

= 62.59 %

Q.10 A single acting two-stage air compressor deals with 4 m³/min of air at 1.013 bar and 15°C with a speed of 250 RPM. The delivery pressure is 80 bar. Assuming complete inter -cooling, find the power required by compressor and bore and stroke of the compressor. Assume apiston speed of 3 m/s, mechanical efficiency 75% and volumetric efficiency of 80% per

stage. Assume the polytropic index of compression in both stage to be n = 1.25 and neglect clearance.

Solution:

Intermediate pressure for perfect intercooling

$P_2 = \sqrt{P_1 P_3} = \sqrt{1.013 \times 90} = 9$ bar

Minimum power required by compressor

$$W = \frac{2n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \times \frac{1}{\eta_{mech}}$$

$$= \frac{2 \times 1.25}{1.25-1} \times \frac{1.013 \times 100}{0.75} \times \frac{4}{60} \left[\left(\frac{9}{1.013} \right)^{\frac{1.25-1}{1.25}} - 1 \right]$$

$$= \frac{1013 \times 4}{45} \times 0.548 = 49.34 KW$$

L is the length of stroke of the piston

$2L \dfrac{N}{60} = 3 \text{ m/s}$

$L = \dfrac{90 \times 100}{250} = 36 \text{ cm}$

Effective swept volume $V_s = 4/250$ = 0.016 m³

$\dfrac{\pi}{4} \times D_{LP}^2 \times L \times \eta_{Vol} = 0.016$

$D_{LP} = \sqrt{\dfrac{0.016 \times 4}{3.14 \times 0.36 \times 0.8}}$

= 0.266 m = 26.6 cm

Ideal gas equation

$\dfrac{P_1 V_1}{T_1} = \dfrac{P_3 V_3}{T_3}$

for perfect intercooling, $T_1 = T_3$ So.

$\dfrac{V_1}{V_3} = \dfrac{P_3}{P_1}$

$\dfrac{\dfrac{\pi}{4} \times D_{LP}^2 \times L}{\dfrac{\pi}{4} \times D_{HP}^2 \times L} = \dfrac{9}{1.013}$

$D_{HP} = 0.266 \sqrt{\dfrac{1.013}{9}}$

= 0.89 m = 89 cm

9 INTERNAL COMBUSTION ENGINE

9.1 HEAT ENGINE

Heat engine is a device which transforms the chemical energy into heat energy and the utilized it into mechanical work in an efficient manner. Thus the thermal energy is transforms into mechanical energy into heat engine. There are two types of heat engine:
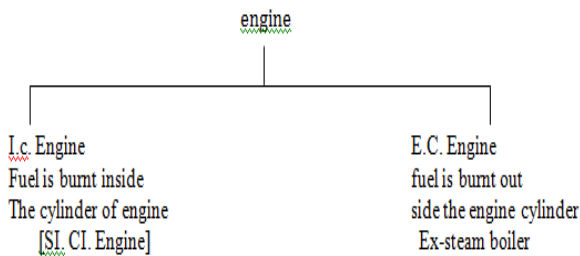
```
                 engine
                    |
        ┌───────────┴───────────┐
   I.c. Engine              E.C. Engine
   Fuel is burnt inside     fuel is burnt out
   The cylinder of engine   side the engine cylinder
      [SI. CI. Engine]         Ex-steam boiler
```

The internal combustion engine is a heat engine that converts heat energy (Chemical energy of a fuel) into mechanical energy inside the cylinder of the engine (usually made available on a rotating output shaft).

The External combustion engine is a heat engine that converts heat energy (Chemical energy of a fuel) into mechanical energy outside the cylinder of the engine. The internal combustion engine can be categorized into two types:

1) Spark Ignition (SI) Engine

An SI engine starts the combustion process in each cycle by use of a spark plug. The spark plug gives a high voltage electrical discharge between two electrodes, which ignites the air fuel mixture in the combustion chamber surrounding the plug. In early engine development, before the inventor of electric spark plug, many forms of torch holes were used to initiate combustion from an external flame.

2) Compression Ignition (CI) Engine

The combustion process in a CI engine starts when the air-fuel mixture self-ignites due to high temperature in the combustion chamber caused by high compression.

9.2 APPLICATIONS OF IC ENGINES

Mainly used as 'prime movers', e.g. for be the propulsion of a vehicle car, bus, truck, locomotive, marine vessel, or airplane. Other applications include stationary saws, lawn mowers, bull-dozers, cranes, electric generators, etc.

9.3 CLASSIFICATIONS OF IC ENGINES

IC engines can be classified according to:
1. Number of cylinders – 1, 2, 3, 4, 5, 6 to 16 cylinder engines.
2. Arrangement of cylinders – Inline, V-type, Flat type, etc.
3. Type of cooling – Air-cooled, Water-cooled, etc.
4. Number of strokes per cycle – 2-stroke, 4-stroke engines.
5. Method of ignition – Spark Ignition (SI), Compression Ignition (CI).
6. Primary mechanical motion – Reciprocating, rotary.

9.4 BASIC COMPONENTS OF I.C. ENGINE

Fig: Internal Combustion Engine

Engine components:

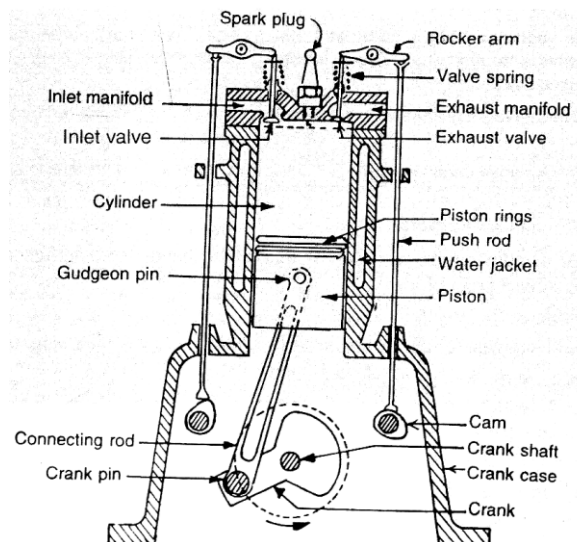The following is the list of major components found in most reciprocating internal combustion engines

Block: Body of engine containing the cylinders made of cast iron or aluminium. In many older engines the valves and the valve ports were contained in the block. On air cooled engines the exterior surface of the block has cooling fins.

Camshaft: Rotating shaft used to push open valves at the proper time in the engine cycle either directly or through mechanical or hydraulic linkage. Most modern automobile engines have one or more camshafts mounted in the engine head.

Carburettor: Venturi flow device that meters the proper amount of fuel into the air flow by means of pressure differential.

Catalytic converter: Chamber mounted in exhaust flow containing catalytical material that promotes reduction of emission by chemical reaction.

Choke: Butterfly valve at carburettor intake, used to create rich fuel-air mixture in intake system for cold weather starting.

Combustion chamber: The end of the cylinder between the head and the piston face where the combustion occurs.

Piston: The cylindrically shaped mass that reciprocates back and forth in the cylinder transmitting the pressure forces in the combustion chamber rotating the crank shaft. The top of the piston is called crown and the sides are called skirt. Pistons are made of cast iron Aluminium or steel.

Piston rings: Metal rings that fit into circumferential groups around the piston and form a sliding surface against the cylinder walls. The purpose of the rings is to form a seal between the piston and cylinder walls.

Flywheel: Rotating mass with large moment of inertia connected to the crankshaft of the engine. The purpose of the flywheel is to store energy and furnish a large angular momentum that that keeps the engine rotating between power strokes and smoothes out engine operation.

Fuel pump: Electrically or mechanically driven pump to supply fuel from the fuel tank to the engine.

9.5   TERM   SUSEDIN   INTERNAL COMBUSTION ENGINE

i) **Top-Dead-Centre (TDC):** Position of the piston when it stops at the furthest point away from the crankshaft.

ii) **Bottom-Dead-Centre (BDC):** Position of the piston when it stops at the point closest to the crankshaft.

iii) **Bore**: Diameter of the cylinder or diameter of the piston face, which is the same minus a very small clearance.

iv) **Stroke**: Movement distance of the piston from one extreme position to the other: TDC to BDC or BDC to TDC. It is denoted by L.

v) **Clearance volume**: Minimum Volume in the combustion chamber with piston at TDC. It is given by V_c.

vi) **Displacement volume**: Volume displaced by the piston as it travels through one stroke. It is also known as swept volume (V_s).

vii) **Air Fuel Ratio:** It is the ratio of mass air to mass of fuel input into engine.

viii) **Stroke to Bore Ratio:** It is the ratio of the stroke length to the diameter of the cylinder.

Stroke to bore ratio $= \dfrac{L}{d}$

➤ If $\dfrac{L}{d} = 1$ It is known as Square Engine

➤ $\dfrac{L}{d} < 1$ It is known as Under Square engine

> $\dfrac{L}{d} > 1$ It is known as Over Square engine

ix) Compression ratio: It is the ratio of the total volume to clearance volume.

$$r_P = \frac{V_T}{V_C} = \frac{V_C + V_S}{V_C} = 1 + \frac{V_S}{V_C}$$

9.6 DIFFERENCE BETWEEN FOUR STROKE AND TWO STROKE ENGINE

Four stroke	Two stroke
(1) One power stroke in two Revolution of crank shaft	(1) power stroke in each revolution of crank shaft
(2) flywheel - heavier	(2) flywheel – Lighter
(3) value	(3) Ports
(4) more volumetric eff.	(4) less volumetric efficiency
(5) part load eff. Is high	(5) part Load efficiency is Low

9.7 PERFORMANCE PARAMETERS

1) **Indicated thermal efficiency:** it is the ratio of the energy in the indicated power to input fuel energy in the appropriate unit.

$$\eta_{ith} = \frac{I.P.(KJ/Sec)}{energy\ of\ fuel/sec}$$

2) **Brake thermal efficiency:** it is the ratio of the energy in the brake power to input fuel energy in the appropriate unit.

$$\eta_{ith} = \frac{B.P.(KJ/Sec)}{energy\ of\ fuel/sec}$$

3) **Mechanical efficiency:** it is the ratio of the energy in the brake power to indicated power in the appropriate unit.

$$\eta_{m} = \frac{B.P..(KJ/Sec)}{I.P..(KJ/Sec)}$$

4) **Friction Power**: friction power is the difference between indicated thermal power and mechanical power or brake power.
   F.P. = I.P.-B.P.

5) **Relative efficiency:** it is the ratio of the actual thermal efficiency to air standard efficiency.

$$\eta_{r} = \frac{actually\ thermal\ eff.}{Air-standard\ efficiency}$$

6) **Mean effective pressure:** M.E.P. is average pr. inside the cylinder of an internal combustion engine on power output

$$I.P. = \frac{P_{im} X\ A\ X\ L\ X\ NXK}{60} watt$$

Where $N \rightarrow \dfrac{N}{2}$ for 4 – stroke engine and $N \rightarrow N$ for 2 – stroke engine the mean effective pressure can be given as

$$P_{im} = \frac{area\ of\ indicator\ diagram}{Length\ of\ indicator\ diagram}$$

9.8 AIR STANDARD CYCLE AND EFFICIENCY

The analysis of actual cycle is very complicated. So the analysis of cycle is done on the basis of air standard cycle. The air standard cycle is based on such assumptions:
I.   The working medium is perfect gas.
II. There is no change of the mass of the working fluid.
III. All the processes in the cycle are reversible processes.

9.8.1 OTTO CYCLE

Nicolaus Otto proposed a constant volume heat addition and rejection process in place of isothermal process of Carnot cycle. Otto cycle is used now a day in spark ignition engine.
4-stroke petrol engine operates on air standard Otto cycle. It completes the Otto cycle in 4 strokes (4 TDC to BDC movements of the piston). In 4 stroke crank shaft rotates 720^0.
The four stroke of four stroke engine are:
1) Suction Stroke,
2) Compression Stroke,
3) Power Stroke,
4) Exhaust Stroke.
Otto cycle shown below consists of four processes:

Process 1–2: Isentropic Compression Process: In this process the heat transfer is zero. dQ = 0

Process 2 – 3 : Constant Volume Heat addition Process: The fuel and air mixture is burnt at constant volume process. Heat addition in this process is given by

$$Q_S = m.C_v(T_3 - T_2)$$

Process 3–4 : Isentropic Expansion Process: In this process the heat transfer is zero. dQ = 0

Process 4–1 : Constant Volume Heat rejection Process: The heat is reject in constant volume process. Heat rejection in this process is given by

$$Q_R = m.C_v(T_4 - T_1)$$
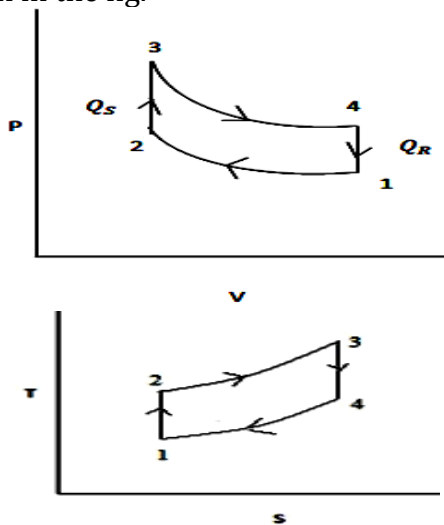
P-V and T-S diagram for Otto engine is shown in the fig.

fig: Otto cycle: P-V and T- S diagram

Efficiency of the cycle is given by

$$\eta = 1 - \frac{Q_R}{Q_S}$$

$$= 1 - \frac{m.C_V(T_4 - T_1)}{m.C_V(T_3 - T_2)}$$

$$= 1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}$$

$$= 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)} \quad \text{----- (9.1)}$$

process 1-2 is isentropic process so for the isentropic process,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = (r)^{\gamma-1}$$

Process 3-4 is the isentropic process so for the isentropic process,

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_3}\right)^{\gamma-1} = (r)^{\gamma-1}$$

$$\frac{V_1}{V_2} = \frac{V_4}{V_3} = r$$

$$\frac{T_2}{T_1} = \frac{T_3}{T_4}$$

Substituting the value in equation 9.1

$$\eta = 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)} = 1 - \frac{T_1}{T_2}$$

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \quad \text{------------ (9.2)}$$

The thermal efficiency is the function of the compression ratio and the ratio of specific heat γ. if the **γ** is considered as constant the efficiency depends upon the compression ratio. The effect on efficiency of the Otto cycle can be understood with the help of the efficiency vs. compression ratio.
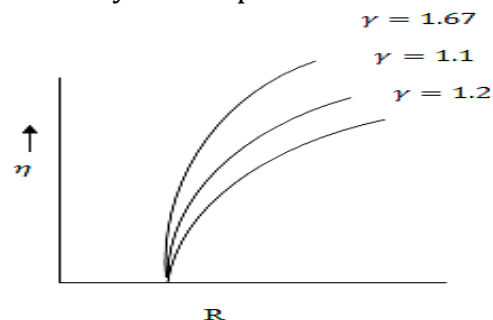

fig: efficiency vs. compression curve

9.8.2 DIESEL CYCLE

In actual spark ignition engine, upper limit of compression ratio is limited due to self-ignition temperature of the petrol. This can be overcome by compressing the air separately. In diesel cycle fuel is burnt at constant pressure. Other three processes are the same processes as in Otto cycle.

Diesel cycle shown below consists of four processes:

Process 1–2: Isentropic Compression Process: In this process the heat transfer is zero.

dQ = 0

Process 2 – 3 : Constant pressure Heat addition Process: The fuel is burnt at constant pressure process. Heat addition in this process is given by

$$Q_S = m.C_P(T_3 - T_2)$$

Process 3–4 : Isentropic Expansion Process: In this process the heat transfer is zero.

dQ=0

Process 4–1 : Constant Volume Heat rejection Process: The heat is rejected in constant volume process. Heat rejection in this process is given by

$$Q_R = m.C_v(T_4 - T_1)$$

P-V and T-S diagram for diesel engine is shown in the fig.
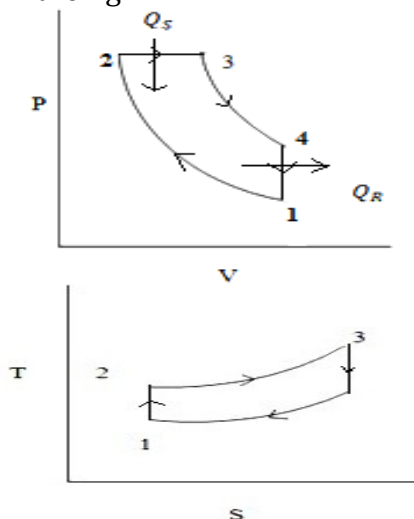
Efficiency of the cycle is given by

$$\eta = 1 - \frac{Q_R}{Q_S}$$

$$\eta_D = 1 - \frac{m.C_V(T_4 - T_1)}{m.C_P(T_3 - T_2)} =$$

$$1 - \frac{1(T_4 - T_1)}{\gamma(T_3 - T_2)} -----(9.3)$$

Compression Ratio $r = \dfrac{V_1}{V_2}$

Cut off ratio $\rho_C = \dfrac{V_3}{V_2}$

Expansion ratio $r_e = \dfrac{V_4}{V_3}$

$$= \frac{V_4}{V_3} \times \frac{V_2}{V_2}$$

$$= \frac{V_1}{V_2} \times \frac{V_2}{V_3} = \frac{r}{\rho_C}$$

$$r = \rho_C \times r_e$$

Substituting the values of the temperature in term of the compression ratio and cut off ratio, we get efficiency of diesel engine

$$\eta_D = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{\rho_c^\gamma - 1}{\gamma(\rho_c - 1)}\right] \quad ----------- (9.4)$$

The bracket term is always greater than one, so efficiency of diesel cycle is always less than efficiency Of Otto cycle for given compression ratio. The normal range of compression ratio in petrol engine is from 6–10 and in diesel 15–20. So efficiency of diesel engine is more than efficiency of gasoline engine.

9.8.3 DUAL CYCLE

Dual cycle is also called as mixed cycle or limited pressure cycle. It is the compromise between the Otto cycle and Diesel cycle. Because for chemical reaction some time is required to so combustion does not occurs at constant volume and due to rapid uncontrolled combustion in diesel engines, combustion does not occur at constant pressure.

Dual cycle shown below consists of five processes:

Process1-2: Isentropic Compression Process: In this process the heat transfer is zero.

dQ = 0

Process 2–3 : Constant volume Heat addition Process: The fuel is burnt at constant volume process. Heat addition in this process is given by

$$Q_{S1} = m.C_V(T_3 - T_2)$$

Process 3–4: Constant pressure Heat addition Process: The fuel is burnt at constant pressure process. Heat addition in this process is given by
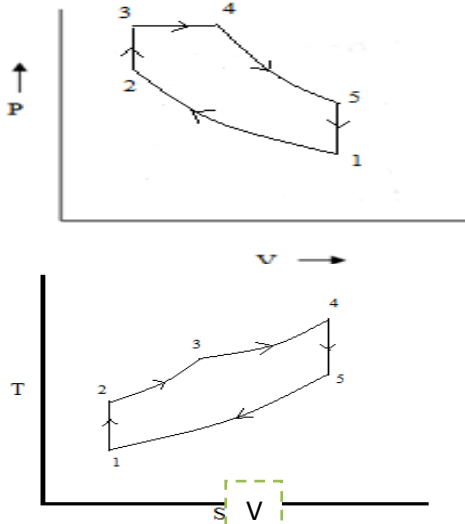
$Q_{S2} = m.C_P(T_4 - T_3)$

Process 4–5 : Isentropic Expansion Process: In this process the heat transfer is zero.

dQ = 0

Process 5–1: Constant Volume Heat rejection Process: The heat is rejected in constant volume process. Heat rejection in this process is given by

$Q_R = m.C_v(T_5 - T_1)$

P-V and T-S diagram of Dual cycle is shown in the fig.

Efficiency of the Dual cycle is given by

$\eta = 1 - \dfrac{Q_R}{Q_S}$

$= 1 - \dfrac{m.c_v(T_5 - T_1)}{m.c_p(T_4 - T_3) + m.c_v(T_3 - T_2)}$

$= 1 - \dfrac{(T_5 - T_1)}{\gamma(T_4 - T_3) + (T_3 - T_2)}$

Compression Ratio $r = \dfrac{V_1}{V_2}$

Pressure ratio $rp = \dfrac{P_3}{P_2}$

Cut off ratio $\rho_C = \dfrac{V_4}{V_3}$

Expansion ratio $r_e = \dfrac{V_5}{V_4} = \dfrac{V_5}{V_4} \times \dfrac{V_3}{V_3}$

$= \dfrac{V_1}{V_2} \times \dfrac{V_3}{V_4} = \dfrac{r}{\rho_C}$

$r = \rho c \times r_e$

Substituting the values of temperatures in the term of compression ratio, pressure ratio and cut off ratio, we get

Efficiency of Dual cycle

$$\eta_{DUAL} = 1 - \dfrac{1}{r^{\gamma-1}}\left[\dfrac{r_p \rho_c^\gamma - 1}{(r_p - 1) + \gamma r_p (\rho_c - 1)}\right] \text{--- (9.5)}$$

9.9 COMPARISON OF OTTO, DIESEL, DUAL CYCLE

1) **Same comp. ratio and heat addition**

Otto cycle 1-2-3-4, Diesel cycle 1-2-3'-4' and Dual cycle 1-2-2'-3"-4" are shown on P-V and T-S diagram. From the T-S diagram it is clear that heat input i.e. area of T-S curve along S- axis is same.
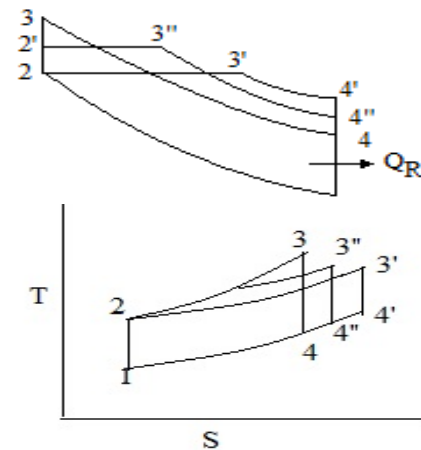
But the heat rejected in Otto cycle is minimum and in Diesel cycle is maximum So

$\eta = 1 - \dfrac{Q_R}{Q_S}$

So the efficiency of Otto cycle is maximum and the efficiency of diesel cycle will be minimum in this case.

$\eta_{otto} > \eta_{Dual} > \eta_{Deisel}$

2) **For Same compression Ratio and heat Rejection**

Otto cycle 1-2-3-4, Diesel cycle 1-2-3'-4 and Dual cycle 1-2-3"-4"-4 are shown on P-V and T-S diagram. From the T-S diagram it is clear that heat rejection i.e. area of T-S curve in process 4-1 along S-axis is same.
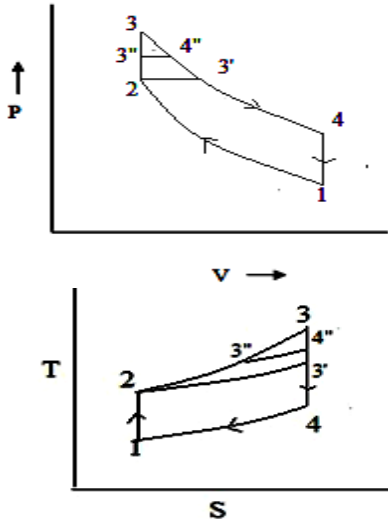
But the heat supplied in Otto cycle is maximum and in Diesel cycle is minimum So

$$Q_{S\,Deisel} < Q_s dual < Q_{S\,otto}$$

and $\eta = 1 - \dfrac{Q_R}{Q_S}$

So the efficiency of Otto cycle is maximum and the efficiency of diesel cycle will be minimum in this case.

$$\eta_{otto} > \eta_{Dual} > \eta_{Deisel}$$

Examples

Q.1 A Diesel engine has a compression ratio of 20 and cut off takes place at 5% of stroke find air standard efficiency $\gamma = 1.4$

Solution:
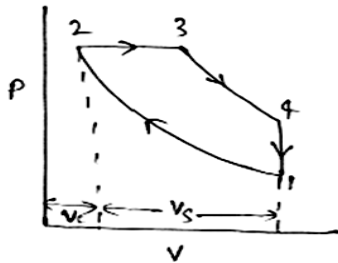
$$r = \frac{V_1}{V_2} = 20, V_1 = 20V_2 = 20V_2$$

$$V_S = 19V_2 = 19V_C$$

$$V_3 = 0.05 X V_S + V_C$$

$$= 0.05 X 19V_C + V_C$$

$$V_3 = 1.95V_C$$

$$\rho = \frac{V_3}{V_2} = \frac{1.95V_C}{V_C} = 1.95$$

$$\eta = 1 - \frac{1}{(r)^{\gamma-1}} \left[\frac{\rho_C^\gamma - 1}{\gamma(\rho_C - 1)} \right]$$

$$= 1 - \frac{1}{(20)^4} \left[\frac{1.95^{1.4} - 1}{1.4(1.95 - 1)} \right] = 64.9\%$$

Q.2 4-Cylinder, 2-stroke IC engine has the following particulars: engine speed = 3000 rpm, bore = 120 mm, crank radius = 60 mm, mechanical efficiency = 90% and the engine develops 75 BHP. Calculate the swept volume and mean effective pressure (MEP).

Solution:

Mechanical efficiency,

$$\eta = \frac{BrakePower\ bhp}{Indicated\ power\ Ihp}$$

$0.9 = \dfrac{75}{P}$,   So, P = 83.33 hp

We know that

Engine Power, P = T ω

Here, P = 83.33 HP = 83.33 x 746 W

= 62166.67 W

$$\omega = \frac{2\pi N}{60} = \frac{2 \times \pi \times 3000}{60}$$

= 314.16 rad/s

T = P/ ω = 62166.67/314.16

= 197.88 N.m

Mean   Effective   Pressure (MEPorPmep)

$$MEP = \frac{2\pi N_C T}{V_d}$$

$V_s = K.(\pi/4).d^2. L$

K = No. of the cylinder

Stroke L = 2 x crank radius

= 2 x 0.06 m

= 0.12 m

$$Vd = 4 \times \frac{\pi}{4} \times 0.12^2 \times 0.12 = 5.43 \times 10^{-3} m^3$$

Nc = No. of cycle per power stroke = 1 for a 2-stroke engine

Therefore,

$$\text{MEP} = \frac{2\pi \times 1 \times 197.88}{0.00543}$$
$$= 228971.77 \text{ N/m}^2$$
$$= 228.97 \text{ KPa}$$

Q.3 Two engines are to operate on Otto and Diesel cycles with the following data: Maximum temperature 1400 K, exhaust temperature 700 K. State of air at the beginning of compression 0.1 MPa, 300 K. Estimate the compression ratios, the maximum pressures, efficiencies, and rate of work outputs (for 1 kg/min of air) of the respective cycles.

Solution:

$T_3 = 1400$ K

$T_4 = 700$ K

$P_1 = 100$ KPa

$T_1 = 300$ K

$$v_1 = \frac{RT_1}{P_1} = \frac{0.287 \times 300}{100}$$

$$= 0.861 \text{ m}^3/\text{kg}$$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{V_4}{V_3}\right)^{\gamma-1}$$

$$\frac{1400}{700} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = 2$$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = 2$$

$$T_2 = 2 \times 300 = 600 \text{K}$$

$$rc = \frac{V_1}{V_2} = (2)^{\frac{1}{\gamma-1}} = 5.657$$

Work done W= $Q_1 - Q_2 = C_v (T_3 - T_2) - C_v (T_4 - T_1)$

$= 0.718[(1400-600)-(700-300)] \text{KJ/Kg}$

$= 287.2 \text{ KJ/Kg}.$

Q.4 A six cylinder ,four stroke spark ignition engine of 10cm×12cm (bore stroke) with a compression ratio of 6 is tested at 4800 rpm-on a dynamometer of arm 55 cm. during a 10 minutes test, the dynamometer reads 45kg and engine consumed 5 kg of petrol of CV 45 MJ/kg. The carburettor receives air at 29 ^{0}C and 1 bar at the rate of 10kg/min. Calculate:-

1) Brake power
2) Brake mean effective pressure
3) Brake specific fuel consumption
4) Brake specific air consumption
5) Brake thermal efficiency
6) Air full ratio

Solution:

6 cylinders, 4 strokes

Bore =10cm =0.1m

Stroke=0.12m

n=6, N=4800rpm

Arm of dynamometer=55cm= 0.55m

Dynamometer reads 45 Kg in 10min

1. Brake power

   B.P. = T×ω = 2πNT/60

   T= F×r = 45× 9.81 × 0.55

   = 242.79 Nm.

   B.P. = 2×3.14×4800×242.79/60

   = 122.039KW

2. Brake mean effective pressure

   Brake power = $\dfrac{pm \times l \times a}{60}\left(\dfrac{N}{2}\right) \times n$

   $$P_m = \frac{122.039 \times 10 \times 10 \times 10 \times 60 \times 2}{0.12 \times \left(\dfrac{\pi}{4} \times 0.1 \times 0.1\right) \times 4800 \times 6}$$

   $= 5.39 \text{bar}$

3. Brake specific fuel consumption.

   $$\text{Bsfc} = \frac{mf}{\text{B.P.}}$$

   $$mf = \frac{5}{10} \times 60$$

   $= 30 \text{kg/hr}$

   $$= \frac{30}{122.039}$$

   $= 0.245 \text{kg/KWh}$

4. Brake specific air consumption

   $$\text{bsac} = \frac{ma}{bp}$$

$$m_a = 10 \times 60 \frac{kg}{h}$$

$$= \frac{600}{122.039} = 4.92 kg / KWh$$

5. Brake thermal efficiency

$$\eta_{bth} = \frac{\text{brake power}}{m \times C.V.}$$

$$= \frac{122.039}{\frac{5 \times 45 \times 10 \times 10 \times 10}{10 \times 60}} = 32.54\%$$

6. Air fuel ratio

$$\left(\frac{A}{F}\right) = \left(\frac{10}{0.5}\right) = 20\%$$

Q.5 A sharp edged circular orifice of diameter 3.8 cm and co-efficient of discharge as 0.6 is used to measure air consumption of a four stroke petrol engine. The pressure drop through the orifice is 145 cm of water and barometer reads 75.5cm of Hg. The compression ratio of the engine is 6 and the piston displacement volume is 2000cm³. The temp of air is taken to be 26⁰C at 2600 rpm, the engine brake power recorded is 29.5 KW. The fuel consumption is 0.14 kg/min and the calorific value of fuel used is 43960 KJ/kg. Calculate following:
(1) Volumetric efficiency
(2) Air – fuel radio
(3) Brake mean effective pressure
(4) Brake thermal efficiency

Solution:
Diameter of orifice d =3.8 cm
Coefficient of discharge=0.6
Pressure drop =145 mm of water
= 1,422.45 Pa
Barometer reading =75.5 cm of hg
= 1,00,729.08 Pa
r = 6, V_s =2000 cm³
Temperature of air =299k
B.P.=29.5 kw at 2600 rpm
F.C.=0.14 kg/min
CV=43960 kj/kg

1) volumetric efficiency

$$\eta v = \left(\frac{\text{altual volume innaled / cycle}}{\text{sweet volume.}}\right)$$

$$Pair = \frac{1.00729 \times 10^5}{287 \times 299} = 1.1738 kg / m^3$$

Now in orifice air enter per second,

$$= Cd\sqrt{2\Delta Psair}$$

$$= 0.6 \times \frac{\pi}{4} \times 0.038^2 \times \sqrt{(2 \times 1422 \times 1.173)}$$

$$= 0.03932 \text{ kg/s}$$

Sweet volume

$$= 2000^{-6} \times 10^{-6} \times \frac{2600}{2 \times 60}$$

= 0.0433 m³/rev
Actual mass enter/sec
= 0.03932 kg/s
Actual volume enter/sec

$$= \frac{0.03932}{1.1738}$$

= 0.3349 m³/s

$$\eta_v = \frac{0.3349}{0.0433} = 77.36\%$$

2) air fuel ratio
m_a= 0.03932 kg/s

$$Sfc = 0.14 \text{ kg/min} = \frac{0.14}{60} kg / s$$

$$= 2.33 \times 10^{-3} \text{ kg/s}$$

$$\left(\frac{A}{F}\right) = \frac{0.03932}{2.33 \times 163} = 16.85$$

3) Brake mean effective Pressure

$$B.P. = \frac{Pm \times l \times a}{60}\left(\frac{N}{2}\right)$$

$$= \frac{60 \times 2 \times 29.5 \times 10^3}{2600 \times 2000 \times 10^{-6}} = 6.8 bar$$

4) brake thermal efficiently

$$\eta_{bth} = \frac{BP}{mf \times CV}$$

$$= \frac{29.5 \times 10^3}{2.33 \times 10^3 \times 43960 \times 10^3}$$

$$= 28.86\%$$

Q.6 The following data are know for a four cylinder stroke petrol engine: cylinder dimension = 11 cm bore, 13

cm stroke engine speed = 2250 rpm, brake power=50kw, friction power =15 KW

fuel consumption rate =10.5 kg/h, calorific value of fuel= 50,000 KJ/kg, air inhalation rate = 300 kg/h, ambient condition =15^{0}C, 1.03 bar.

Estimate:

1. Brake mean effective pressure
2. Brake thermal efficiency
3. Mechanical efficiency

Solution:

1. Brake mean effective pressure

$$B.P = \frac{PLANK}{60 \times 2}$$

$$P = \frac{B.P. \times 60 \times 2}{LANK}$$

$$= \frac{50 \times 60 \times 2}{2250 \times 4 \times \pi / 4 \times 0.11^2 \times 0.13}$$

$$= 5.396 \text{ bar}$$

2. Brake thermal efficiency

$$\eta_{BTh} = \frac{brakepower}{mf \times CV}$$

$$= \frac{50}{(10.5/3600) X 50000} = 34.29\%$$

3. Mechanical Efficiency

BP = 50 KW

FP = 15 KW

IP = 50 + 15 = 65KW

$$\text{Mechanical efficiency} = \frac{BP}{IP} = \frac{50}{65}$$

$$= 76.92\%$$

Q.7 A liquid fuel $C_7 h_{16}$ is burned with 10% more air than the stoichiometric air assuming complete combustion calculate:

1. The mass of air supplied per kg of fuel and
2. The volumetric analysis of the dry products of combustion. Assume air contains 21 % O_2 by volume.

Solution:

Equation for stoichiometric composition:

$$C_7h_{16} + 11O_2 \rightarrow 7CO_2 + 8H_2O$$
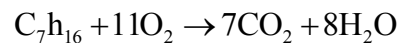
If 10% more air is supplied

$$C_7H_{16} + 12.1O_2 + [12.1x(\frac{79}{21})]N_2 \rightarrow$$

$$7CO_2 + 8H_2O + 1.1O_2 + 45.52N_2$$
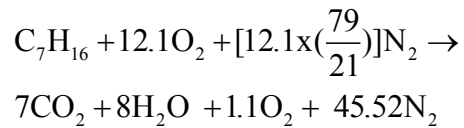
Mass of O_2 for 100 kg of fuel = 12.1 x 32 = 387.3 kg of O_2

Mass of air for 100 kg of fuel = 1683.4 kg of air

Mass of air required for 1 kg of fuel =16.82 kg

Dry exhaust gas Analysis on volume basis

	Moles	%
Co_2	7	11.36 %
O_2	1.1	1.78 %
H_2O	8	12.9 %
N_2	45.52	73.87 %

Q.8 Find the percentage increase in the efficiency of a Diesel engine having a compression ratio of 16 and cut-off ratio (r) is 10% of the swept volume if C_V by 2% take C_v = o.717 KJ/Kg.K and $\gamma = 1,4$

Solution:

The efficiency of Diesel engine

$$\eta = 1 - \frac{1}{r^{\gamma-1}} \frac{[\rho^\gamma - 1]}{[\gamma(\rho - 1)]}$$

Where

$$r = \frac{vl}{v2} = 16$$

$$\rho = \text{cut of ratio} = \frac{v3}{v2}, \gamma = \frac{cp}{cv}$$
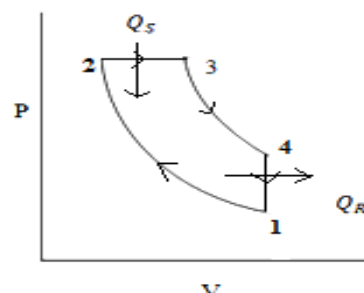
C_v = 0.717 KJ/kg k, γ = 1.4

C_p = $\gamma \times C_v$ = 1.0038 kj/kg k

R = C_p-C_v = 1.0038 – 0.717

= 0.2868 kj/kg k

$V_3 - V_2 = 0.1(V_1 - V_2)$

dividing the equation by V_2, we get

$$\left(\frac{V3}{V2}-1\right)=0.10\left(\frac{V1}{V2}-1\right)$$

$(\rho-1)=0.10(r-1)$

$(\rho-1)=0.10(16-1)$

$\rho=2.5$

$$\eta=1-\frac{1}{r^{\gamma-1}}\frac{[\rho^{\gamma}-1]}{[\gamma(\rho-1)]}$$

$$=1-\frac{1}{16^{1.4-1}}\frac{[2.5^{1.4}-1]}{[1.4\times(2.5-1)]}$$

$=0.5905=59.05\%$

When c_v is decreases by 2%

$C_V' = 0.70266$

$R = 0.2868 = c_p' - c_v'$

$C_p' = 0.98946$

$$\gamma'=\frac{C_p'}{C_v'}=1.4081$$

$$\eta'=1-\frac{1}{r^{\gamma'-1}}\frac{[\rho^{\gamma}-1]}{[\gamma'(\rho-1)]}$$

$$=1-\frac{1}{16^{1.408-1}}\frac{[2.5^{1.408}-1]}{[1.408\times(2.5-1)]}$$

$=0.5975=59.75\%$

Increase in efficiency of diesel cycle

$$=\frac{\eta'-\eta}{\eta}\times100=\frac{0.5975-0.5905}{0.5905}$$

$=1.185\%$

GATE QUESTIONS

Topics

1. THERMODYNAMIC SYSTEM AND PROCESSES

2. FIRST LAW, HEAT, WORK AND ENERGY

3. SECOND LAW, CRANOT CYCLE AND ENROPY

4. AVALILABILITY AND IRREVERSIBILITY

5. PURE SUBSTANCES

6. POWER SYSTEM (RANKINE, BRAYTON, ETC.)

7. IC ENGINE

8. REFRIGERATION AND AIR CONDITIONING

Q.1 The following four figures have been drawn to represent a fictitious thermodynamic cycle, on the p -v and T -s planes.
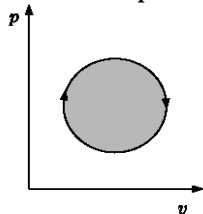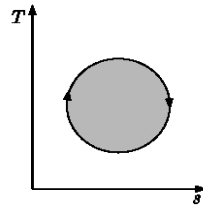
fig. 1   fig. 2
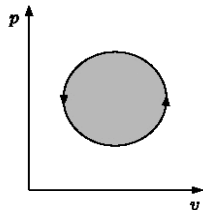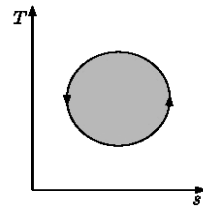
fig. 3   fig.4

According to the first law of thermodynamics, equal areas are enclosed by
a) Figures 1 and 2
b) Figures 1 and 3
c) Figures 1 and 4
d) Figures 2 and 3

[GATE–2005]

Q.2 A reversible thermodynamic cycle containing only three processes and producing work is to be constructed. The constraints are
i) There must be one isothermal process,
ii) There must be one isentropic process,
iii) The maximum and minimum cycle pressures and the clearance volume are fixed, and
iv) Polytropic processes are not allowed.
Then the number of possible cycles are
a) 1   b) 2
c) 3   d) 4

[GATE–2005]

Q.3 Match items from groups I, II, III, IV and V.

Group I	Group II	Group III
	When added to the system is	Differential
E Heat	G Positive	I Exact
F Work	H Negative	J Inexact

Group IV	Group V
Function	Phenomenon
K Path	M Transient
L Point	N Boundary

a) F-G-J-K-M   b) E-G-I-K-M
   E-G-I-K-N   F-H-I-K-N
c) F-H-J-L-N   d) E-G-J-K-N
   E-H-I-L-M   F-H-J-K-M

[GATE–2006]

Common Data for Q.4 and Q.5
A thermodynamic cycle with an ideal gas as working fluid is shown below.
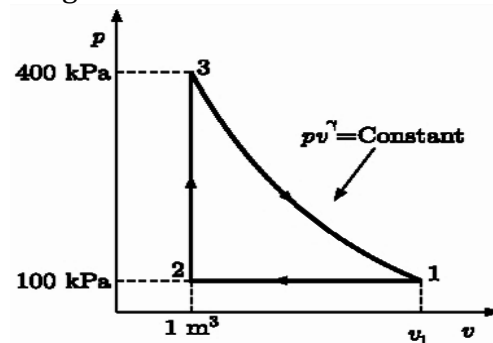
Q.4 The above cycle is represented on T -s plane by

a)

b)

c)

d)
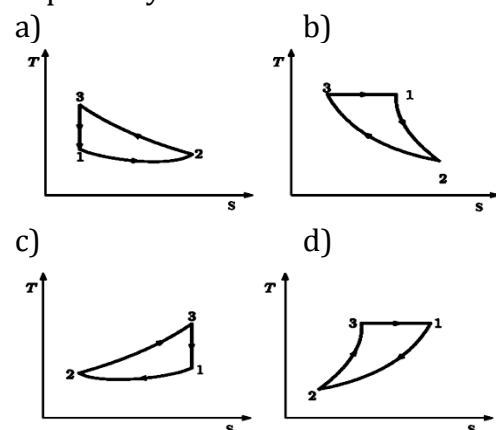
[GATE–2007]

Q.5 If the specific heats of the working fluid are constant and the value of specific heat ratio is 1.4, the thermal efficiency (%) of the cycle is
a) 21 b) 40.9
c) 42.6 d) 59.7
[GATE–2007]

Q.6 If a closed system is undergoing an irreversible process, the entropy of the system
a) must increase
b) always remains constant
c) Must decrease
d) can increase, decrease of remain constant
[ME-GATE –2009]

Q.7 Heat and work are
a) intensive properties
b) extensive properties
c) point functions
d) path functions
[ME-GATE –2012]

Q.8 A certain amount of an ideal gas is initially at a pressure p_1 and temperature T_1. First, it undergoes a constant pressure process 1-2 such that $T_2 = 3T_1/4$. Then, it undergoes a constant volume process 2-3 such that $T_3 = T_1/2$. The ratio of the final volume to the initial volume of the ideal gas is
a) 0.25 b) 0.75
c) 1.0 d) 1.5
[ME-GATE -2014(3)]

Q.9 Two identical metal blocks L and M (specific heat = 0.4kJ/ kg. K), each having a mass of 5kg, are initially at 313K. A reversible refrigerator extracts heat from block L and rejects heat to block M until the temperature of block L reaches 293K. The final temperature (in K) of block M is ____.
[ME-GATE -2014(4)]

Q.10 Which of the following statements are TRUE with respect to heat and work?
i) They are boundary phenomena
ii) They are exact differentials
iii) They are path functions
a) both (i) and (ii)
b) both (i) and (iii)
c) both (ii) and (iii)
d) only (iii)
[GATE -2016(1)]

Q.11 A mass m of a perfect gas at pressure p_1 and volume V_1 undergoes an isothermal process. The final pressure is p_2 and volume is V_2. The work done on the system is considered positive. If R is the gas constant and T is the temperature, then the work done in the process is

a) $p_1 V_1 \ln \dfrac{V_2}{V_1}$ b) $-p_1 V_1 \ln \dfrac{p_1}{p_2}$

c) $RT \ln \dfrac{V_2}{V_1}$ d) $-mRT \ln \dfrac{p_2}{p_1}$

[ME-GATE -2017(1)]

Q.12. The volume and temperature of air (assumed to be an ideal gas) in a closed vessel is 2.87 m3 and 300K, respectively. The gauge pressure indicated by a manometer fitted to the wall of the vessel is 0.5bar. If the gas constant of air is R = 287 J/kg. K and the atmospheric pressure is 1 bar, the mass of air (in kg) in the vessel is
a) 1.67 b) 3.33
c) 5. d) 6.66
[ME-GATE -2018(2)]

Q.13 A n engine operates on the reversible cycle as shown in the figure. The work output from the engine (in kJ/cycle) is ____ (correct to two decimal places).
[ME-GATE -2018(2)]
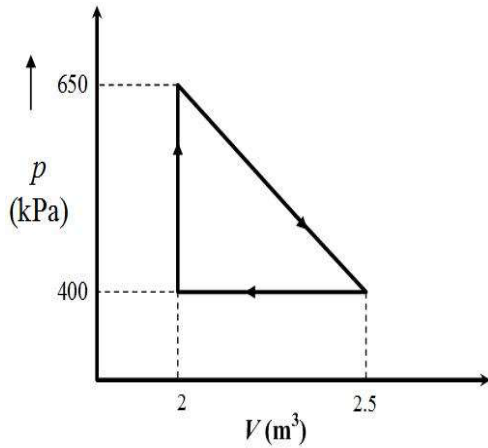
Q.14 Air is held inside a non-insulated cylinder using a piston (mass M=25 kg and area A=100 cm²) and stoppers (of negligible area), as shown in the figure. The initial pressure P_i and temperature T_i of air inside the cylinder are 200 kPa and $400\,^0C$, respectively. The ambient pressure P_∞ and temperature T_∞ are 100 kPa and $27\,^0C$, respectively. The temperature of the air inside the cylinder (^0C) at which the piston will begin to move is _____ (correct to two decimal places).

[ME-GATE-2018 (2)]

ANSWER KEY:

1	2	3	4	5	6	7	8	9	10	11	12	13
(a)	(d)	(d)	(c)	(a)	(d)	(d)	(b)		(b)	(b)	(c)	62.5
14												
146												

EXPLANATIONS

Q.1 **(a)**

From the first law of thermodynamics for a cyclic process

$\Delta U = 0$

And $\oint \delta Q = \oint \delta W$

The symbol $\oint \delta Q$, which is called the cyclic integral of the heat transfer represents the heat transfer during the cycle and $\oint \delta W$ the cyclic integral of the work represents the work during the cycle.

We easily see that figure 1 and 2 satisfy the first law of thermo-dynamics, both the figure are in same direction (clockwise) and satisfies the relation.

$\oint \delta Q = \oint \delta W$

Q.2 **(d)**

Two cycles having constant volume process and two cycles having constant pressure process can be formed.

Therefore a total of four cycles can be formed.

Q.3 **(d)**

Q.4 **(c)**

In the given p - v diagram, three processes are occurred.

i) Constant pressure (Process 1 – 2)
ii) Constant Volume (Process 2 – 3)
iii) Adiabatic (Process 3 – 1)

We know that, Constant pressure and constant volume lines are inclined curves in the T-s curve, and adiabatic process is drawn by a vertical line on a T-s curve.
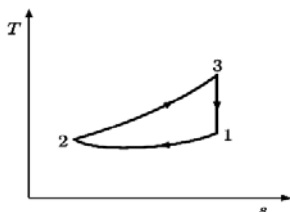
Given P-v diagram is clockwise. So T-s diagram must be clockwise.

Q.5 **(a)**
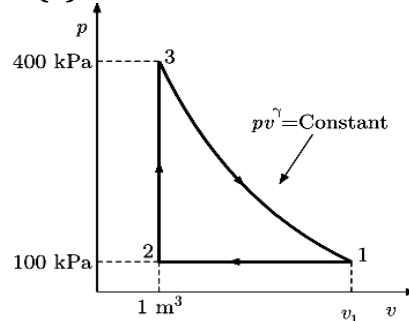
This cycle shows the Lenoir cycle. For Lenoir cycle efficiency is given by

$$\eta_L = 1 - \gamma \left(\frac{r_p^{\frac{1}{\gamma}} - 1}{r_p - 1} \right)$$

Where, $r_p = \dfrac{p_2}{p_1} = \dfrac{400}{100} = 4$

And $\gamma = \dfrac{C_p}{C_v} = 1.4\,(\text{Given})$

So,

$$\eta_L = 1 - 1.4 \left[\frac{(4)^{\frac{1}{1.4}} - 1}{4 - 1} \right] = 1 - 0.789$$

= 0.211

$\eta_L = 21.1\% ;\ 21\%$

Q.6 **(d)**

If a closed system is undergoing an irreversible process, the entropy of the system can increase, decrease or remain constant.

Q.7 **(d)**

Q.8 **(b)**

For constant pressure process (1to 2): $\dfrac{V_1}{T_1} = \dfrac{V_2}{T_2}$

$$\Rightarrow V_2 = V_1\left(\frac{T_2}{T_1}\right) = \frac{3V_1}{4}$$

For constant volume process (2 to 3)

$\therefore$

$$\frac{\text{Final volume}}{\text{Initial volume}} = \frac{V_3}{V_1} = \frac{V_2}{V_1} = \frac{3}{4} = 0.75$$

Q.9 (333 to 335)

Q.10 (b)

Q.11 Sol: (b)

$$\because w = \int Pdv$$

For ideal gas $PV = mRT = \text{const (c)}$

$$PV = C \Rightarrow P = \frac{C}{V}$$

$$W = \int_{V_1}^{V_2} \frac{C}{V}\, dv$$

$$= c[\ln v]_{V_1}^{V_2}$$

$$w = c\ln\frac{V_2}{V_1}$$
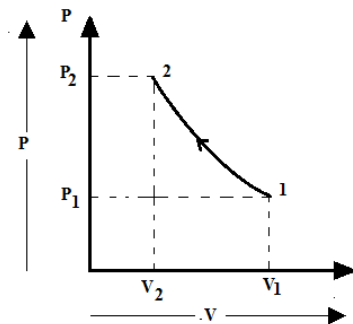
$$\Rightarrow w = P_1 V_1 \ln\frac{V_2}{V_1}$$

When work is done on the system (compression), $V_2 < V_1$ and $P_2 > P_1$

For isothermal process

$$P_1 V_1 = P_2 V_2 = c$$

$$\therefore \frac{V_2}{V_1} = \frac{P_1}{P_2}$$

$$\therefore w = P_1 V_1 \ln\frac{P_1}{P_2}$$

If the work done on the system is + ive, then

$$w = -P_1 V_1 \ln\frac{P_1}{P_2}$$

Q.12 (c)

$$P_{abs} = P_{gauge} + P_{atm}$$

$$= 50 + 100 = 150\text{kPa}$$

By ideal gas eqn

$$P_{abs} V = mRT$$

$$m = \frac{150 \times 2.87}{.287 \times 300}$$

$$m = 5\text{kg}$$

Q.13 (62.5)

For a reversible process,

$$\text{work output} = \int PdV$$

Hence, for the given reversible cycle, Work Output = Area enclosed by the triangle

$$= \frac{1}{2} \times 0.5 \times 250 = 62.5 \text{ kJ/cycle}$$

Q.14 (146.03)
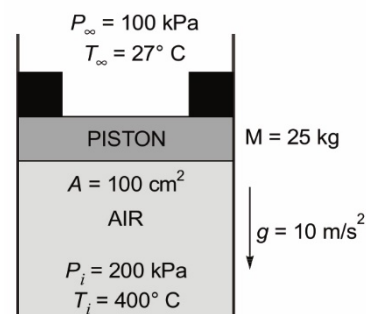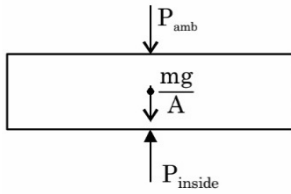
$$P_{inside} = P_{amb} + \frac{mg}{A}$$

$$= 100 + \frac{25 \times 9.81}{100 \times 10^{-4}} \times 10^{-3}$$

$$P_2 = 124.525 \text{ kN/m}^2$$

As mass and volume remains constant

$$\frac{P_1}{T_1} = \frac{P_2}{T_2}$$

$$\frac{200}{673} = \frac{124.525}{T_2}$$

$$T_2 = 419.02 \text{ K} = 146.03 \,^{\circ}\text{C}$$

Q.1 A small steam whistle (perfectly insulated and doing no shaft work) causes a drop of 0.8 kJ/kg in the enthalpy of steam from entry to exit. If the kinetic energy of the steam at entry is negligible, the velocity of the steam at exit is

a) 4 m/s   b) 40 m/s
c) 80 m/s   d) 120 m/s

[GATE–2001]

Q.2 A 2 kW, 40 liters water heater is switched on for 20 minutes. The heat capacity C_p for water is 4.2 kJ/kgK. Assuming all the electrical energy has gone into heating the water, increase of the water temperature in degree centigrade is

a) 2.7   b) 4.0
c) 14.3   d) 25.25

[GATE–2003]

Common Data for Q.3 and 4
Nitrogen gas (molecular weight 28) is enclosed in a cylinder by a piston, at the initial condition of 2 bar, 298 K and 1 m³. In a particular process, the gas slowly expands under isothermal condition, until the volume becomes 2m³. Heat exchange occurs with the atmosphere at 298 K during this process.

Q.3 The work interaction for the Nitrogen gas is

a) 200 kJ   b) 138.6 kJ
c) 2 kJ   d) -200 kJ

[GATE–2003]

Q.4) The entropy changes for the Universe during the process in kJ/K is

a) 0.4652   b) 0.0067
c) 0   d) -0.6711

[GATE–2003]

Q.5 A gas contained in a cylinder is compressed, the work required for

compression being 5000 kJ. During the process, heat interaction of 2000 kJ causes the surroundings to be heated. The changes in internal energy of the gas during the process is

a) -7000 kJ   b) -3000 kJ
c) +3000 kJ   d) +7000 kJ

[GATE–2004]

Common Data For Q.6 and Q.7
A football was inflated to a gauge pressure of 1 bar when the ambient temperature was 15°C. When the game started next day, the air temperature at the stadium was 5°C. Assume that the volume of the football remains constant at 2500 cm³.

Q.6 The amount of heat lost by the air in the football and the gauge pressure of air in the football at the stadium respectively equal

a) 30.6 J, 1.94 bar   b) 21.8 J, 0.93 bar
c) 61.1 J, 1.94 bar   d) 43.7 J, 0.93 bar

[GATE–2006]

Q.7 Gauge pressure of air to which the ball must have been originally inflated so that it would be equal 1 bar gauge at the stadium is

a) 2.23 bar   b) 1.94 bar
c) 1.07 bar   d) 1.00 bars

[GATE–2006]

Q.8 Which of the following relationships is valid only for reversible processes undergone by a closed system of simple compressible substance? (neglect changes in kinetic and potential energy)

a) $\delta Q = dU + \delta W$ b) $Tds = dU + pdv$
c) $Tds = dU + \delta W$ d) $\delta Q = dU + pdv$

[GATE–2007]

Q.9 A gas expands in a frictionless piston-cylinder arrangement. The expansion process is very slow, and is resisted by an ambient pressure

of 100 kPa. During the expansion process, the pressure of the system (gas) remains constant at 300 kPa. The change in volume of the gas is $0.01m^3$. The maximum amount of work that could be utilized from the above process is

a) 0 kJ   b) 1 kJ
c) 2 kJ   d) 3 kJ.

[GATE–2008]

Q.10 A balloon containing an ideal gas is initially kept in an evacuated and insulated room. The balloon ruptures and the gas fills up the entire room. Which one of the following statements is TRUE at the end of above process?

a) The internal energy of the gas decreases from its initial value, but the enthalpy remains constant
b) The internal energy of the gas increases from its initial value, but the enthalpy remains constant
c) Both internal energy & enthalpy of the gas remain constant
d) Both internal energy and enthalpy of the gas increase

[GATE–2008]

Q.11 A rigid, insulated tank is initially evacuated. The tank is connected with a supply line through which air (assumed to be ideal gas with constant specific heats) passes at 1MPa, 350⁰C. A valve connected with the supply line is opened and the tank is charged with air until the final pressure inside the tank reaches 1MPa. The final temperature inside the tank.

Air Supply Line

Valve⊗
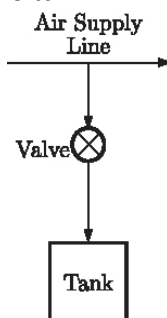
Tank

a) is greater than 350⁰C
b) is less than 350⁰C
c) is equal to 350⁰C
d) may be greater than, less than, or equal to, 350⁰C depending on the volume of the tank

[GATE–2008]

Q.12 In a steady state flow process taking place in a device with a single inlet and a single outlet, the work done per unit mass flow rate is given by $W= -\int_{inlet}^{outlet} vdp$ where v is the specific volume and p is the pressure. The expression for W given above

a) is valid only if the process is both reversible and adiabatic
b) is valid only if the process is both reversible and isothermal
c) is valid for any reversible process
d) is incorrect; it must be
$$W = \int_{inlet}^{outlet} pdv$$

[GATE–2008]

Q.13 A frictionless piston-cylinder device contains a gas initially at 0.8MPa and 0.015 m^3. It expands quasi-statically at constant temperature to a final volume of 0.030 m^3. The work output (in kJ) during this process will be

a) 8.32   b) 12.00
c) 554.67   d) 8320.00

[GATE–2009]

Q.14 A compressor undergoes a reversible, steady flow process. The gas at inlet and outlet of the compressor is designated as state 1 and state 2 respectively. Potential and kinetic energy changes are to be ignored. The following notations are used:

v = Specific volume and p = pressure of the gas. The specific work required to be supplied to the
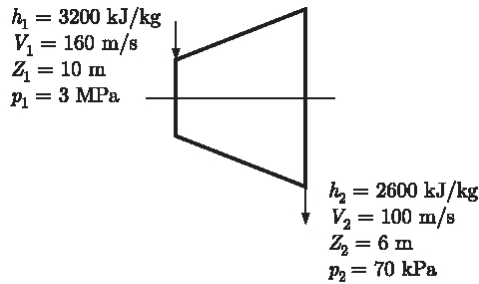
compressor for this gas compression process is

a) $\int\limits_1^2 pdv$

b) $\int\limits_1^2 vdp$

c) $v_1(p_2\text{-}p_1)$

d) $-p_2(v_1 - v_2)$

[GATE–2009]

Common Data for Q.15 and Q.16

The inlet and the outlet conditions of steam for an adiabatic steam turbine are as indicated in the figure. The notations are as usually followed.


$h_1 = 3200$ kJ/kg
$V_1 = 160$ m/s
$Z_1 = 10$ m
$p_1 = 3$ MPa

$h_2 = 2600$ kJ/kg
$V_2 = 100$ m/s
$Z_2 = 6$ m
$p_2 = 70$ kPa

Q.15 If mass rate of steam through the turbine is 20 kg/s, the power output of the turbine (in MW) is

a) 12.157

b) 12.941

c) 168.001

d) 168.785

[GATE–2009]

Q.16 Assume the above turbine to be part of a simple Rankine cycle. The density of water at the inlet to the pump is 1000 kg/m^3. Ignoring kinetic and potential energy effects, the specific work (in kJ/kg) supplied to the pump is

a) 0.293

b) 0.351

c) 2.930

d) 3.510

[GATE–2009]

Common Data For Q.17 and Q.18

The temperature and pressure of air in a large reservoir are 400 K and 3 bar respectively. A converging-diverging nozzle of exit area 0.005 m^2 is fitted to the wall of the reservoir as shown in the figure. The static pressure of air at the exit section for isentropic flow through the nozzle is 50 kPa. The characteristic gas constant and the ratio of specific heats of air are 0.287kJ/kgK and 1.4 respectively

Flow from the Reservoir
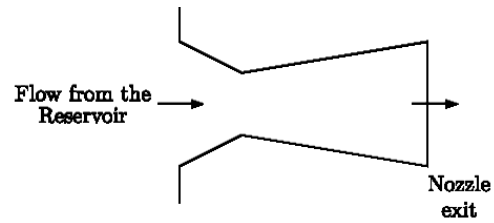
Nozzle exit

Q.17 The density of air in kg/m^3 at the nozzle exit is

a) 0.560

b) 0.600

c) 0.727

d) 0.800

[GATE–2011]

Q.18 The mass flow rate of air through the nozzle in kg/s is

a) 1.30

b) 1.77

c) 1.85

d) 2.06

[GATE–2011]

Q.19 The contents of a well-insulated tank are heated by a resistor of 23 Ω in which 10 A current is flowing. Consider the tank along with its contents as a thermodynamic system. The work done by the system and the heat transfer to the system are positive. The rates of heat (Q), work (W) and change in internal energy (ΔU) during the process in kW are

a) Q = 0, W = -2.3, ΔU = +2.3

b) Q = +2.3, W = 0, ΔU = +2.3

c) Q = -2.3, W = 0, ΔU = -2.3

d) Q = 0, W = +2.3, ΔU = -2.3

[GATE–2011]

Common Data for Q.20 and Q.21

Air enters an adiabatic nozzle at 300 kPa, 500 K with the velocity of 10 m/s. It leaves the nozzle at 100 kPa with a velocity of 180 m/s. The inlet area is 80 cm². The specific heat of air C_p is 1008 J/kgK.

Q.20 The exit temperature of the air is

a) 516 K

b) 532 K

c) 484 K

d) 468 K

[GATE–2012]

Q.21 The exit area of the nozzle in cm² is

a) 90.1

b) 56.3

c) 4.4

d) 12.9

[GATE–2012]

Q.22 For an ideal gas with constant values of specific heats, for calculation of the specific enthalpy,
a) it is sufficient to know only the temperature
b) both temperature and pressure are required to be known
c) both temperature and volume are required to be known
d) both temperature and mass are required to be known

[GATE-2015 Set-1]

Q.23 A well insulated rigid container of volume 1m^3 contains 1.0 kg of an ideal gas [C_p = 1000J/kgK) and C_v = 800J/kgK] at a pressure of 10^5 Pa. A stirrer is rotated at constant rpm in the container for 1000 rotations and the applied torque is 100 N-m. The final temperature of the gas (in K) is _____.

[GATE-2015 Set-1]

Q.24 Work is done on adiabatic system due to which its velocity changes from 10 m/s to 20 m/s, elevation increases by 20m and temperature increases by 1 K. The mass of the system is 10 kg, C_v=100 J/kgK and gravitational acceleration is 10 m/s^2. If there is no change in any other component of the energy of the system, the magnitude of total work done (in kJ) on the system is ____.

[GATE-2015 Set-2]

Q.25 Steam enters a turbine at 30 bar, 300°C (u = 2750 kJ/kg, h = 2993 kJ/kg) and exits the turbine as saturated liquid at 15 kPa (u = 225 kJ/kg, h = 226 kJ/kg). Heat loss to the surrounding is 50 kJ/kg of steam flowing through the turbine. Neglecting changes in kinetic energy and potential energy, the work output of the turbine (in kJ/kg of steam) is_____

[GATE-2015 Set-3]

Q.26 A mixture of ideal gases has the following composition by mass

N$_2$	O$_2$	CO$_2$
60%	30%	10%

If the Universal gas constant is 8314 J/mol-K, the characteristic gas constant of the mixture (in J/kg.K) is _____.

[GATE-2015 Set-3]

Q.27 An ideal gas undergoes a reversible process in which the pressure varies linearly with volume. The conditions at the start (subscript 1) and at the end (subscript 2) of the process with usual notation are: p_1 = 100 kPa, V_1 = 0.2 m^3 and p_2 = 200 kPa, V_2 = 0.1 m^3 and the gas constant, R = 0.275 kJ/kg-K. The magnitude of the work required for the process (in kJ) is _____

Q.28 The internal energy of an ideal gas is a function of
a) temperature and pressure
b) volume and pressure
c) entropy and pressure
d) temperature only

[GATE-2016 Set-2]

Q.29 A piston-cylinder device initially contains 0.4 m^3 of air (to be treated as an ideal gas) at 100 kPa and 80°C. The air is now isothermally compressed to 0.1 m^3. The work done during this process is _____ kJ. (Take the sign convention such that work done on the system is negative)

[GATE-2016 Set-2]

Q.30 Steam at an initial enthalpy of 100 kJ/kg and inlet velocity of 100 m/s, enters an insulated horizontal nozzle. It leaves the nozzle at 200 m/s. The exit enthalpy (in kJ/kg) is _____

[GATE-2016 Set-3]

Q.31 The molar specific heat at constant volume of an ideal gas is equal to 2.5 times the universal gas constant (8.314 J/mol.K). When the temperature increases by 100K, the change in molar specific enthalpy is _____ J/mol

[GATE-2017 Set-1]

Q.32 A frictionless circular piston of area 10^{-2} m² and mass 100 kg sinks into a cylindrical container of the same area filled with water of density 1000 kg/m³ as shown in the figure. The container has a hole of area 10^{-3} m² at the bottom that is open to the atmosphere. Assuming there is no leakage from the edges of the piston and considering water to be incompressible, the magnitude of the piston velocity (in m/s) at the instant shown is ____ (correct to three decimal places).

[GATE-2018 Set-2]
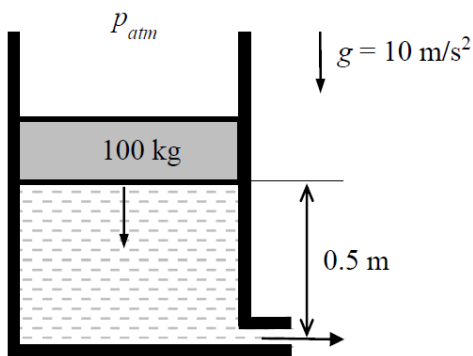
ANSWER KEY:

1	2	3	4	5	6	7	8	9	10	11	12	13	14
(b)	(c)	(b)	(a)	(c)	(d)	(c)	(d)	(c)	(c)	(a)	(c)	(a)	(b)
15	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**	**27**	**28**
(a)	(c)	(c)	(d)	(a)	(c)	(d)	(a)		4.5	2717			(d)
29	**30**	**31**	**32**										
-	-	29.09	1.456										

EXPLANATIONS

Q.1 **(b)**

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2}$$

$$C_1 = 0$$

$$h_1 - h_2 = \frac{C_2^2}{2}$$

$$0.8 \times 10^3 = \frac{C_2^2}{2}$$

$$C_2 = 40 \text{m/s}$$

Q.2 **(c)**

Heat Supplied

H = Power × Time

=2 × 1000 × 20 × 60

=2400 × 1000 J

But heat required in increasing the water temperature

H= $mC_p\Delta T$

2400×1000=1000×0.04

×4.2×1000×ΔT

ΔT=14.3°C

Q.3 **(b)**

For isothermal process,

$$\text{Work} = P_1 V_1 \ln \frac{V_2}{V_1}$$

$$= (2 \times 100) \times (1) \times \ln \frac{2}{1}$$

= 138.63 kJ

Q.4 **(a)**

Q.5 **(c)**

W = -5000 kJ (Negative sign shows that work is done on the system)

Q = - 2000 kJ (Negative sign shows that heat is rejected by the system)

From the first law of thermodynamics,

Δ Q = ΔW + Δ U

So

ΔU = ΔQ − Δ W = −2000 − (−5000) = 3000 kJ

Q.6 **(d)**

Given

P_1 = 1 bar (Gauge)

(P_1) absolute = 1+1.013= 2.013 bar.

T_1 =15°C=288K

T_2=5°C=278K

Vol. = Constant = 2500 cm³

at constant vol. (15°C→5°C)

Q= $mC_v dT$

Q= m×718×(278-288)

We need to calculate the mass first,

Using PV=mRT

$$2.013 \times 10^5 \times 2500 \times 10^{-6} = m \times 287 \times 288$$

m = 6.088 × 10^{-3} kg

∴ Q = 6.088 × 10^{-3}×718 × (278-288)

Q = −43.7 J (-ve sign indicates that heat lost by the air in the ball)

T_1 = 288

P_2 = ?

T_2 = 278

Vol. = C

$$\frac{T_2}{T_1} = \frac{P_2}{P_1}$$

$$\Rightarrow P_2 = 2.013 \times \frac{278}{288}$$

=1.93 bar. (absolute)

$$\Rightarrow P_{absol} = P_{atm} + P_{gauge}$$

$$\Rightarrow P_{gauge} = 0.93 bar$$

Q.7   (c)

We know that

Since volume is constant

$$\therefore \quad \frac{P_1}{P_2} = \frac{T_1}{T_2}$$

$$\therefore \quad P_2 = 1 \text{ bar (gauge)}$$

= 2.013 bars (absolute)

Substituting the values, we get

P_1= 2.085 bars (absolute)

= 1.07 bar (gauge)

Q.8   (d)

$\delta q = dU + pdv$ This equation holds good for a closed system when only pdv work is present. This is true only for a reversible (quasi-static) process.
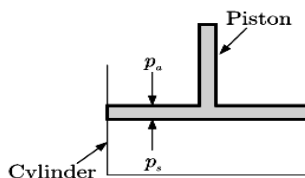
Q.9   (c)

Given: p_a = 100 kPa, p_s = 300 kPa, $\Delta v = 0.01$ m³

Net pressure across the piston

$p = p_s - p_a = 300 - 100 = 200$ kPa

∴ Maximum work that can be utilized from the system is given by:

W = p∆v = 200 × 0.01 = 2 kJ

Q.10   (c)

We know that enthalpy,

h = u + pv

Given that room is insulated. So there is no interaction of Energy

(Heat) between system (room) and surrounding (atmosphere).

It means internal Energy dU = 0 and U = constant.

Now flow work pv must also remain constant thus we may conclude that during free expansion process pv i.e. product of pressure and specific volume change in such a way that their product remains constant.

So, it is a constant internal energy and constant enthalpy process.

Q.11   (a)

Given: $p_1 = 1$ MPa, $T_1 = 350^oC$= 623 K

For air γ = 1.4

We know that final temperature (T_2) inside the tank is given by,

$T_2 = \gamma T_1 = 1.4 \times 623 = 872.2$ K = 599.2^oC

T_2 is greater than 350^oC

Q.12   (c)

Q.13   (a)

Content Temperature Process
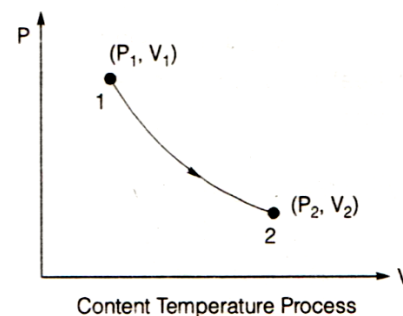
$P_1 = 0.8$ MPa

$V_1 = 0.015$ m³

$V_2 = 0.030$ m³

Isothermal process

Work output

$$W = P_1 V_1 \ln\left(\frac{V_2}{V_1}\right)$$

$$= (0.8 \times 10^3)(0.015) \ln\left(\frac{0.030}{0.015}\right)$$

$$= 0.8 \times 10^3 \times 0.015 \times \ln(2)$$

= 8.32 kJ

Q.14   (b)

Reversible steady flow process.

State 1 : Inlet
State 2: Outlet
Potential and Kinetic energy change are to be ignored.
v= Specific volume of the gas
P= Pressure of the gas
Specific work required to be supplied to the compressor for this gas compression process is
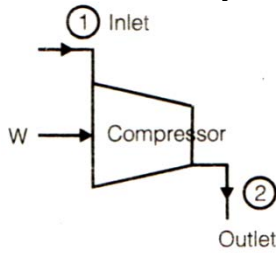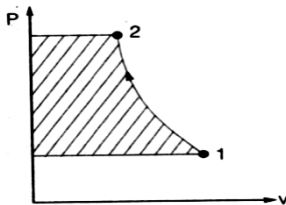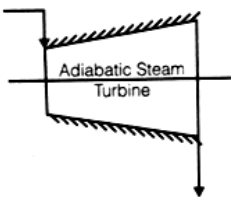$w = \int_1^2 vdp$
(Compressor is an open flow system)

Q.15 (a)

Apply steady flow energy equation [SFEE]

$$h_1 + \frac{V_1^2}{2} + gz_1 + q = h_2 + \frac{V_2^2}{2} + gz_2 + w$$

q = 0; because it is an adiabatic steam turbine

$$\Rightarrow 3200 \times 10^3 + \frac{(100)^2}{2} + 9.81 \times 10 + 0$$

$$= 2600 \times 10^3 + \frac{(100)^2}{2} + 9.81 \times 6 + W$$

$\Rightarrow$ W=607839.24 J/kg
= 607.84kJ/kg
Mass flow rate of steam through turbine is 20kg/s.
The power output of the turbine is
=20×607.84

=12156.78 kW=12.157 MW

Q.16 (c)

Neglecting Kinetic and potential energy effects
SFEE :
$h_1 + q = h_2 + w_T$ (For Turbine)
$\Rightarrow$ 3200=2600+w_T
$\Rightarrow w_T = 600$kJ/kg
Specific work supplied to the pump is
(Since the cycle works between the pressure limits of 3 MPa and 70 kPa)

$$W_p = \int vdP = v(P_1 - P_2)$$

$$= \frac{P_1 - P_2}{\rho}$$

$$= \frac{(3000 - 70)\,kN/m^2}{1000kg/m^3} = 2.930kJ/kg$$

Q.17 (c)

Density of air in (kg/m³) at nozzle exit is

$$\frac{P_1}{\rho_1^\gamma} = \frac{P_2}{\rho_2^\gamma}$$

Where ρ=density

$$\rho_2 = \left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}} \cdot \rho_1$$

$$\rho_2 = \left(\frac{50}{300}\right)^{1/1.4} \times \frac{3\times10^5}{0.287\times1000\times400}$$

$$\rho_2 = 0.727kg/m^3$$

Q.18 (d)

Given:
ρ_2 = 0.787 kg/m³, A_2 = 0.005m²,
V_2 =?
For isentropic expansion,

$$V_2 = \sqrt{2C_p(T_1 - T_2)}$$

$$= \sqrt{2\times1.005\times10^3 \times(400 - 239.73)}$$

= 567.58 m/s
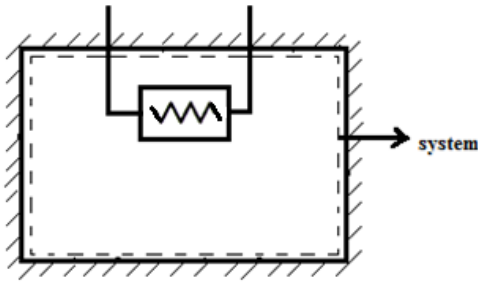Mass flow rate at exit
m = $\rho_2 A_2 V_2$ =0.727×0.005×567.78
= 2.06 kg/s

Q.19 (a)

W=−I²R [∵work is done on the system]
$= -(10)^2 \times 23 = -2300W = -2.3kW$
Q = 0 [∵ system is insulated]
From 1st law: - ΔQ = ΔU + ΔW
ΔU + ΔW = 0
$\Rightarrow \Delta U = -\Delta W = +2.3$ kW

Q.20 (c)
From energy balance for steady flow system
$E_{in} = E_{out}$

$$m\left(h_1 + \frac{V_1^2}{2}\right) = m\left(h_2 + \frac{V_2^2}{2}\right) \quad ...(i)$$

As Equation (1) become

$$C_p T_1 + \frac{V_1^2}{2} = C_p T_2 + \frac{V_2^2}{2}$$

$$T_2 = \left(\frac{V_1^2 - V_2^2}{2 \times C_p}\right) + T_1 = \frac{10^2 - 180^2}{2 \times 1008} + 500$$

$$= -16.02 + 500$$
$$= 483.98 \simeq 484 \text{ K}$$

Q.21 (d)
From Mass conservation
$m_{in} = m_{out}$

$$\frac{V_1 A_1}{v_1} = \frac{V_2 A_2}{v_2} \quad ...(i)$$

where v = specific volume of air = $\frac{RT}{P}$

Therefore Eq. (1) becomes

$$\frac{P_1 V_1 A_1}{RT_1} = \frac{P_2 V_2 A_2}{RT_2}$$

$$A_2 = \frac{P_1 \times V_1 \times A_1 \times T_2}{P_2 \times V_2 \times T_1}$$

$$= \frac{300 \times 10 \times 80 \times 484}{100 \times 180 \times 500}$$

$A_2 = 12.9 \text{ cm}^2$

Q.22 (a)

Q.23 (1283.4 to 1287.4)

Q.24 (4.5)
Using SFEE

$$W = m\left[\left(\frac{V_1^2}{2} - \frac{V_2^2}{2}\right) + (z_1 - z_2)g + (h_1 - h_2)\right]$$

$$W = 10\left[\frac{10^2}{2} - \frac{20^2}{2} + (-20) \times 10 + (100 \times -1)\right]$$

W = − 4.5 kJ
∴ Work done on the system is 4.5 kJ.

Q.25 (2717)
By Steady flow energy equation:
W = (h₁ − h₂) − q
W= (2993−226)− 50 = 2717 kJ/kg
(Neglecting kinetic and potential energy changes)

Q.26 (274 to 276)

Q.27 (14.75 to 15.25)
Given: Pressure varies linearly with volume
∴ p = aV+b   --------- (A)
at $V_1 = 0.2m^3, p_1 = 100kPa$
∴100 = 0.2a + b   ----- (1)
at $V_2 = 0.1m^3, p_2 = 200kPa$
200 = 0.1 a + b   --------- (2)
From equation (1) & (2), we get
a = −1000 and b = 300
∴ equation (A) ⇒ p = 300 − 1000V

$$\text{Work} = \int_{V_1}^{V_2} p\,dV$$

$$= \int_{V_1}^{V_2} [300 - 1000V]\,dV$$

$$= 300V - 1000\left[\frac{V^2}{2}\right]_{V_1}^{V_2}$$
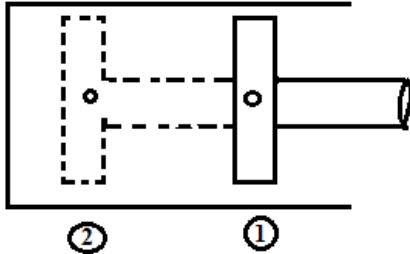
= 300[0.1 − 0.2] − 500 [0.1² − 0.2²]
Work = −15kJ (-ve sign implies work is done on the system)

∵ in the question only magnitude is asked.

∴ Work = 15kJ

Q.28 (d)

Q.29 (-55.6 to -55.4)

Given:-

$P_1 = 100$ kPa

$V_1 = 0.4$ m^3

$V_2 = 0.1$ m^3

Work done in an isothermal process is given by

$$= P_1 V_1 \ln\left[\frac{V_2}{V_1}\right]$$

$$= 100 \times 0.4 \ln\left[\frac{0.1}{0.4}\right]$$

$$W = -55.45\text{kJ}$$

Q.30 (84 to 86)
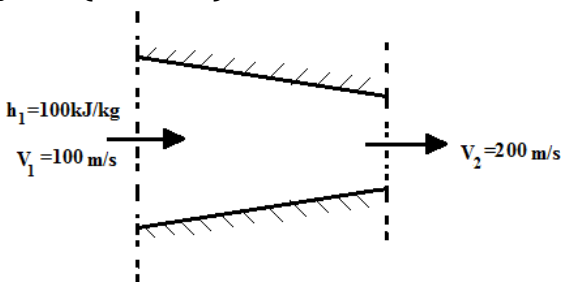
By SFEE:

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2} \qquad \text{[Neglecting}$$

potential head, q = 0 & $W_{c.v} = 0$]

$$h_2 = 100 + \frac{1}{2}\left[\frac{100^2 - 200^2}{10^3}\right]$$

$$h_2 = 85\text{kJ/kg}$$

Q.31 (29.09)

Given that $\overline{C_v} = 2.5\overline{R}$

$$\overline{C_p} - \overline{C_v} = \overline{R}$$

$$\overline{C_p} = \overline{C_v} + \overline{R} = 2.5\overline{R} + \overline{R} = 3.5\overline{R}$$

For ideal gas, Molar specific enthalpy change (($\Delta\overline{H}$)

$$\Delta H = n\overline{C_p}dT$$

$$\Delta\overline{H} = \frac{\Delta H}{n} = \overline{C_p}dT = 3.5\overline{R}dT$$

$$= 3.5 \times 8.314 \times 100$$

$$= 2909.9\text{J / mole}$$

Q.32 (1.456)

Given. Piston mass = 100 kg

Pressure at point (1)

$$P_1 = \frac{100 \times 10}{10^{-2}} = 10^5\,\text{N/m}^2$$
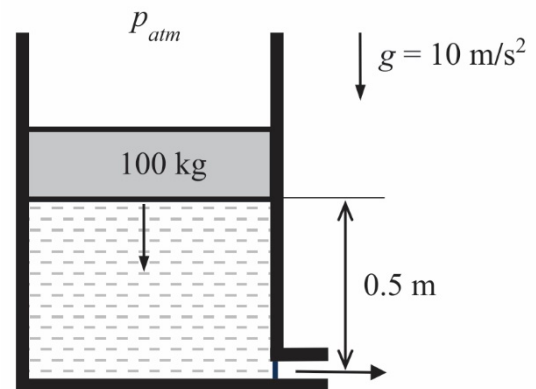
Applying Bernoulli's equation for points (1) and (2):

$$\frac{P_1}{\gamma_w} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\gamma_w} + \frac{V_2^2}{2g} + Z_2 \ldots\ldots(1)$$

where $P_2 = 0$ & $Z_2 = 0$ (datum line)

From continuity equation:

$A_1 V_1 = A_2 V_2$ (discharge from hole= volume swept by the piston per unit time)

$10^{-2} V_1 = 10^{-3} V_2$

$\Rightarrow V_2 = 10 V_1$ .....(2)

From eq. (1) and (2)

$$\frac{10^5}{10^3 \times 10} + \frac{V_1^2}{2 \times 10} + 0.5 = \frac{\left(10V_1\right)^2}{2 \times 10}$$

or $10.5 = \dfrac{99 V_1^2}{20}$

$\Rightarrow V_1 = 1.456$ m/s

Q.1 A cyclic heat engine does 50 kJ of work per cycle. If the efficiency of the heat engine is 75%, the heat rejected per cycle is

a) $16\frac{2}{3}$ kJ

b) $33\frac{1}{3}$ kJ

c) $37\frac{1}{2}$ kJ

d) $66\frac{2}{3}$ kJ

[GATE–2001]

Q.2 A Carnot cycle is having an efficiency of 0.75. If the temperature of the high temperature reservoir is 727^0C, what is the temperature of low temperature reservoir?

a) 23^0C

b) -23^0C

c) 0^0C

d) 250^0C

[GATE–2002]

Q.3 A solar collector receiving solar radiation at the rate of 0.6 kW/m^2 transforms it to the internal energy of a fluid at an overall efficiency of 50%. The fluid heated to 350 K is used to run a heat engine which rejects heat at 315 K. If the heat engine is to deliver 2.5 kW power, the minimum area of the solar collector required would be

a) $83.33 \, m^2$

b) $16.66 \, m^2$

c) $39.68 \, m^2$

d) $79.36 \, m^2$

[GATE–2004]

Q.4 A heat transformer is device that transfers a part of the heat, supplied to it at an intermediate temperature, to a high temperature reservoir while rejecting the remaining part to a low temperature heat sink. In such a heat transformer, 100 kJ of heat is supplied at 350 K. The maximum amount of heat in kJ that can be transferred to 400 K, when the rest is rejected to a heat sink at 300 K is

a) 12.50

b) 14.29

c) 33.33

d) 57.14

[GATE–2007]

Q.5 A cyclic device operates between three thermal reservoirs, as shown in the figure. Heat is transferred to/from the cycle device. It is assumed that heat transfer between each thermal reservoir and the cyclic device takes place across negligible temperature difference. Interactions between the cyclic device and the respective thermal reservoirs that are shown in the figure are all in the form of heat transfer.
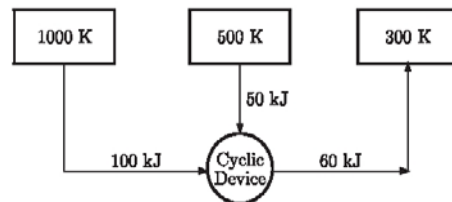
The cyclic device can be
a) a reversible heat engine
b) a reversible heat pump or a reversible refrigerator
c) an irreversible heat engine
d) an irreversible heat pump or an irreversible refrigerator

[GATE–2008]

Q.6 An irreversible heat engine extracts heat from a high temperature source at a rate of 100 kW and rejects heat to a sink at a rate of 50 kW. The entire work output of the heat engine is used to drive a reversible heat pump operating between a set of independent isothermal heat reservoirs at 17°C and 75°C. The rate (in kW) at which the heat pump delivers heat to its high temperature sink is

a) 50

b) 250

c) 300   d) 360
[GATE–2009]

Q.7 One kilogram of water at room temperature is brought into contact with a high temperature thermal reservoir. The entropy change of the universe is
a) equal to entropy change of the reservoir
b) equal to entropy change of water
c) equal to zero
d) always positive
[GATE–2010]

Q.8 Consider the following two processes ;
a) A heat source at 1200 K loses 2500 kJ of heat to a sink at 800 K
b) A heat source at 800 K loses 2000 kJ of heat to a sink at 500 K
Which of the following statements is true ?
a) Process I is more irreversible than Process II
b) Process II is more irreversible than Process I
c) Irreversibility associated in both the processes are equal
d) Both the processes are reversible
[GATE–2010]

Common Data for Q.9 and Q.10
In an experimental set up, air flows between two stations P and Q adiabatically. The direction of flow depends on the pressure and temperature conditions maintained at P and Q. The conditions at station P are 150 kPa and 350 K. The temperature at station Q is 300 K. The following are the properties and relations pertaining to air:
Specific heat at constant pressure, C_P = 1.005 kJ/kgK;
Specific heat at constant volume, C_V = 0.718 kJ/kgK;
Characteristic gas constant, R=0.287 kJ/kgK
Enthalpy, $h = C_P T$

Internal energy, $u = C_V T$

Q.9 If the air has to flow from station P to station Q, the maximum possible value of pressure in kPa at station Q is close to
a) 50   b) 87
c) 128   d) 150
[GATE–2011]

Q.10 If the pressure at station Q is 50 kPa, the change in entropy $(S_Q - S_P)$ in kJ/kgK is
a) -0.155   b) 0
c) 0.160   d) 0.355
[GATE–2011]

Q.11 An ideal gas of mass m and temperature T_1 undergoes a reversible isothermal process from an initial pressure P_1 to final pressure P_2. The heat loss during the process is Q. The entropy change Δs of the gas is
a) $mR\ln\left(\dfrac{P_2}{P_1}\right)$ b) $mR\ln\left(\dfrac{P_1}{P_2}\right)$
c) $mR\ln\left(\dfrac{P_2}{P_1}\right) - \dfrac{Q}{T_1}$ d) zero
[GATE–2013]

Q.12 Which one of the following pairs of equations describes an irreversible heat engine?
a) $\oint \delta Q > 0 \text{ and } \oint \dfrac{\delta Q}{T} < 0$
b) $\oint \delta Q < 0 \text{ and } \oint \dfrac{\delta Q}{T} < 0$
c) $\oint \delta Q > 0 \text{ and } \oint \dfrac{\delta Q}{T} > 0$
d) $\oint \delta Q < 0 \text{ and } \oint \dfrac{\delta Q}{T} > 0$
[GATE-2014 (3)]

Q.13 A source at a temperature of 500 K provides 1000 kJ of heat. The of temperature environment is 27°C. The maximum useful work (in kJ)

that can be obtained from the heat source is _____.

[GATE-2014 (3)]

Q.14 A reversible heat engine receives 2 kJ of heat from a reservoir at 1000 K and a certain amount of heat from a reservoir at 800 K. It rejects 1 kJ of heat to a reservoir at 400 K. The net work output (in kJ) of the cycle is
a) 0.8   b) 1.0
c) 1.4   d) 2.0

[GATE-2014 (1)]

Q.15 An amount of 100 kW of heat is transferred through a wall in steady state. One side of the wall is maintained at 127°C and the other side at 27°C. The entropy generated (in W/K) due to the heat transfer through the wall is ___.

[GATE-2014 (3)]

Q.16 A closed system contains 10 kg of saturated liquid ammonia at 10°C. Heat addition required to convert the entire liquid into saturated vapour at a constant pressure is 16.2 MJ. If the entropy of the saturated liquid is 0.88 kJ/kg.K, the entropy (in kJ/kg. K) of saturated vapor is _____

[GATE-2014 Set-4]

Q.17 A Carnot engine (CE-1) works between two temperature reservoirs A and B, where T_A = 900 K and T_B = 500 K. A second Carnot engine (CE-2) works between temperature reservoirs B and C, where T_C = 300 K. In each cycle of CE-1 and CE-2, all the heat rejected by CE-1 to reservoir B is used by CE-2. For one cycle of operation, if the net Q absorbed by CE-1 from reservoir A is 150 MJ, the net heat rejected to reservoir C by CE-2(in MJ) is _____.

[GATE-2015 (1)]

Q.18 One kg of air (R=287 J/kgK) undergoes an irreversible process between equilibrium state1 (20°C, 0.9 m³) and equilibrium state2 (20°C, 0.6 m³). The change in entropy $s_2 - s_1$ (in J/kg.K) is _____.

[GATE-2015 (2)]

Q.19 The heat removal rate from a refrigerated space and the power input to the compressor are 7.2 kW and 1.8 kW, respectively. The coefficient of performance (COP) of the refrigerator is _____

[GATE-2016 (2)]

Q.20 A reversible cycle receives 40 kJ of heat from one heat source at a temperature of 127°C and 37 kJ from another heat source at 97°C. The heat rejected (in kJ) to the heat sink at 47°C is _____

[GATE-2016 (2)]

Q.21 A heat pump absorbs 10 kW of heat from outside environment at 250 K while absorbing 15 kW of work. It delivers the heat to a room that must be kept warm at 300K. The Coefficient of Performance (COP) of the heat pump is _____.

[GATE-2017 (1)]

Q.22 One kg of an ideal gas (gas constant, R = 400 J/kg.K; specific heat at constant volume, $c_v = 1000 \text{J/kg.K}$ at 1 bar, and 300 K is contained in a sealed rigid cylinder. During an adiabatic process, 100kJ of work is done on the system by a stirrer. The increase in entropy of the system is _____ J/K.
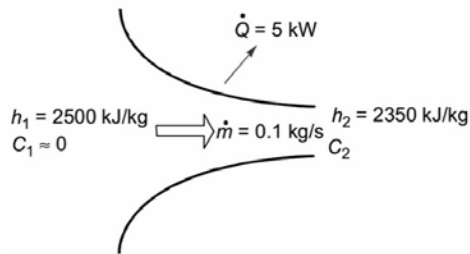
[GATE-2017 (1)]

Q.23 A n ideal gas undergoes a process from state 1 (T_1 = 300 K, p_1 = 100 kPa) to state 2 (T_2 = 600 K, p_2 = 500 kPa). The

specific heats of the ideal gas are : cp = 1 kJ/kg-K and cv = 0.7 kJ/kg-K. The change in specific entropy of the ideal gas from state 1 to state 2 (in kJ/kg-K) is _____ (correct to two decimal places).

[GATE-2018 Set-1]

Q.24 Steam flows through a nozzle at a mass flow rate of $\dot{m} = 0.1 \, kg/s$. with a heat loss of 5 kW. The enthalpies at inlet and exit are 2500kJ/kg and 2350 kJ/kg, respectively. Assuming negligible velocity at inlet ($C_1 \approx 0$), the velocity (C_2) of steam (in m/s) at the nozzle exit is _____ (correct to two decimal places).

[GATE-2018 Set-1]

Q.25 For an ideal gas with constant properties undergoing a quasi-static process, which one of the following represents the change of entropy (Δs) from state 1 to 2?

a) $\Delta s = C_P \ln\left(\dfrac{T_2}{T_1}\right) - R \ln\left(\dfrac{P_2}{P_1}\right)$

b) $\Delta s = C_V \ln\left(\dfrac{T_2}{T_1}\right) - C_P \ln\left(\dfrac{V_2}{V_1}\right)$

c) $\Delta s = C_P \ln\left(\dfrac{T_2}{T_1}\right) - C_V \ln\left(\dfrac{P_2}{P_1}\right)$

d) $\Delta s = C_V \ln\left(\dfrac{T_2}{T_1}\right) + R \ln\left(\dfrac{V_1}{V_2}\right)$

ANSWER KEY:

1	2	3	4	5	6	7	8	9	10	11	12	13	14
(a)	(b)	(a)	(d)	(a)	(c)	(d)	(b)	(b)	(c)	(b)	(a)	400	(c)
15	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**			
83.33	6.6	50	-16.36	4	64	1.67	287	0.21	447	(a)			

EXPLANATIONS

Q.1 (a)

$$\eta = \frac{W}{Q_S}$$

$$0.75 = \frac{50}{Q_S}$$

$$Q_S = 66.67$$

$$\therefore Q_R = 66.67 - 50$$

$$= 16.67 \text{kJ}$$

$$= 16\frac{2}{3}\text{kJ}$$

Q.2 (b)

$$\eta = 1 - \frac{T_L}{T_H}$$

$$0.75 = 1 - \frac{T_L}{1000}$$

$$\frac{T_L}{1000} = 0.25$$

$$T_L = 250K$$

$$T_L = -23°C$$

Q.3 (a)

Internal energy of fluid after absorbing solar radiation
=0.5×600=300W/m²

$$\eta_{engine} = \left(1 - \frac{315}{350}\right)$$

$$= \frac{W}{Q}$$

$$0.1 = \frac{2500}{Q_1}$$

$$\therefore Q_1 = 25000W$$

Let A be the minimum area of collector

$$\therefore Q_1 = A \times 300$$

$$\therefore A = \frac{25000}{300}$$

A= 83.33m²

Q.4 (d)

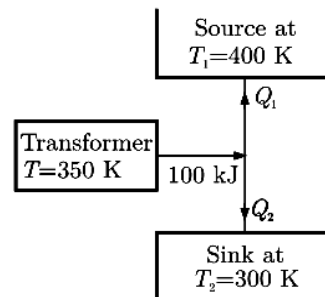Given: T_1 = 400K, T_2 = 300 K,
T = 350 K, Q = 100kJ
$Q_1 \rightarrow$ Heat transferred to the source by the transformer
$Q_2 \rightarrow$ Heat transferred to the sink by the transformer

Applying energy balance on the system

$$Q = Q_1 + Q_2$$

$$Q_2 = Q - Q_1 = 100 - Q_1 \quad \text{...(i)}$$

Apply Claudius inequality on the system.

$$\frac{Q}{T} = \frac{Q_1}{T_1} + \frac{Q_2}{T_2} = \frac{100}{350} = \frac{Q_1}{400} + \frac{Q_2}{300}$$

Substitute the value of Q_2 from equation (i)

$$\frac{100}{350} = \frac{Q_1}{400} + \left(\frac{100 - Q_1}{300}\right) = \frac{Q_1}{400} + \frac{100}{300} - \frac{Q_1}{300}$$

$$\frac{100}{350} = \frac{100}{300} + Q_1\left[\frac{1}{400} - \frac{1}{300}\right]$$

Solving the above equation, we get
Q_1=57.14 kJ
Therefore the maximum amount of heat that can be transferred at 400 K is 57.14 kJ

Q.5 (a)

A heat engine cycle is a thermodynamic cycle in which there is a net heat transfer from higher temperature to a lower temperature device. So it is a Heat Engine.

Applying Clausius inequality on the system for checking the reversibility of the cyclic device
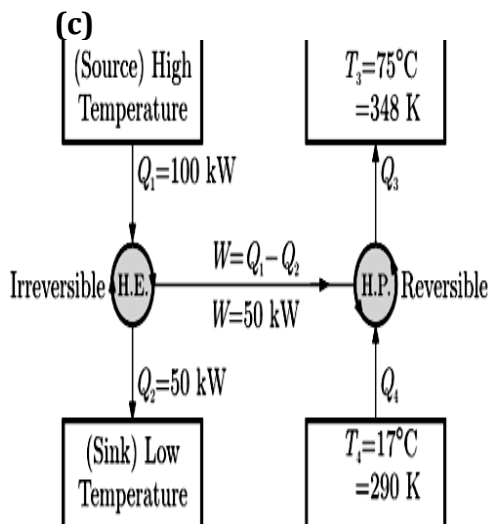
$$\oint \frac{dQ}{T} = 0$$

$$\frac{Q_1}{T_1} + \frac{Q_2}{T_2} - \frac{Q_3}{T_3} = 0$$

$$\frac{100 \times 10^3}{1000} + \frac{50 \times 10^3}{500} - \frac{60 \times 10^3}{300} = 0$$

$100 + 100 - 200 = 0$

Here, the cyclic integral of is zero. This implies, it is a reversible heat engine.

Q.6 **(c)**

We know that coefficient of performance of a Heat pump for the given system is,

$$(COP)_{H.P.} = \frac{Q_3}{W}$$

Since it is a reversible heat pump

$$(COP)_{H.P.} = \frac{T_H}{T_H - T_L}$$

$$\frac{348}{348 - 290} = \frac{Q_3}{50}$$

$$\therefore Q_3 = \frac{348 \times 50}{58} = 300K$$

Q.7 **(d)**
We know that,
Entropy of universe is always increases.
$$\Delta S_{universe} > 0$$

$$(\Delta S)_{system} + (\Delta S)_{surrounding} > 0$$

Q.8 **(b)**
We know from the clauses Inequality,

If $\oint \frac{dQ}{T} = 0$, the cycle is reversible

$\oint \frac{dQ}{T} < 0$ the cycle is irreversible and possible

For case (I)

$$\oint \frac{dQ}{T} = \frac{2500}{1200} - \frac{2500}{800}$$

$$= \frac{25}{12} - \frac{25}{8} = -1.041 kJ/kg$$

For case (II)

$$\oint \frac{dQ}{T} = \frac{2000}{800} - \frac{2000}{500}$$

$$= \frac{20}{8} - \frac{20}{5} = -1.5 kJ/kg$$

Since, $\oint \frac{dQ}{T}$ for the process II is

more in magnitude than process I.
Therefore, process (II) is more irreversible than process (I)

Q.9 **(b)**
Maximum possible value of pressure has been asked in the question, so we take it as a case of reversible flow.
Tds = dh – vdp

$$ds = C_p \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right)$$

The maximum possible value of pressure at station Q can be found as follows:

$$C_p \ln\left(\frac{T_Q}{T_P}\right) - R \ln\left(\frac{P_Q}{P_P}\right) = 0$$

$$1.005 \ln\left(\frac{300}{350}\right) = 0.287 \ln\left(\frac{P_Q}{150}\right)$$

$$P_Q = 87.43 kPa$$

Q.10 **(c)**

By using Tds=dh–vdP, after simplifying we get:

$$(S_Q - S_p) = C_P \ln\left(\frac{T_Q}{T_P}\right) - R \ln\left(\frac{P_Q}{P_P}\right)$$

$$= 1.005 \ln\left(\frac{300}{350}\right) - 0.287 \ln\left(\frac{50}{150}\right)$$

$$= 0.160$$

Q.11 (b)

Q.12 (a)

Clausius inequality for irreversible heat engine, $\oint \frac{dQ}{T} < 0$. Heat content of irreversible heat engine dQ > 0

Q.13 (400)

Maximum work can be derived by using a reversible engine between the two temperature limits.

$$W_{max} = Q_s \eta_{max}$$

$$= Q_s\left(1 - \frac{T_L}{T_H}\right)$$
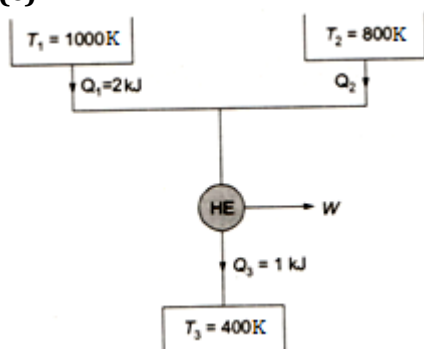
$$W_{max} = Q_s - T_L \frac{Q_s}{T_H}$$

Given, $Q_s = 1000kJ$

$T_L = 273 + 27 = 300K$

$T_H = 500K$

$$\therefore W_{max} = 1000 - 300 \times \frac{1000}{500}$$

$$= 400 \text{ kJ}$$

Q.14 (c)

From Clausius inequality

$$\oint \frac{dQ}{T} = 0, \text{ for a reversible cycle}$$

Or $\quad \frac{Q_1}{T_1} + \frac{Q_2}{T_2} - \frac{Q_3}{T_3} = 0$

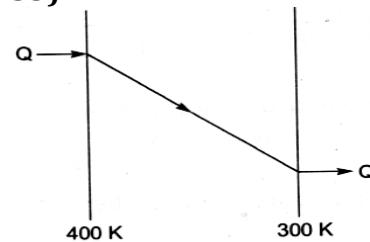Or $\quad \frac{2}{1000} + \frac{Q_2}{800} - \frac{1}{400} = 0$

Or $Q_2 = 0.4kJ$

From 1st law of thermodynamics for a cycle:

$Q_1 + Q_2 = W + Q_3$

Or 2+0.4 = W+1

Or W = 1.4 kJ

Q.15 (83.33)

$$S_2 - S_1 = \int_1^2 \frac{dQ}{T} + S_{gen}$$

$$\therefore 0 = \frac{100}{400} - \frac{00}{300} + S_{gen}$$

Or $\quad S_{gen} = \frac{1}{3} - \frac{1}{4} = \frac{1}{12} kW/K$

$$\therefore S_{gen} = \frac{1000}{12} = \frac{W}{K} = 83.33W/K$$

Q.16 (6.6)

$$S_{fg} = \frac{h_{fg}}{T} = \frac{16.2 \times 10^3}{283} = 57.24 \text{ kJ}/K$$
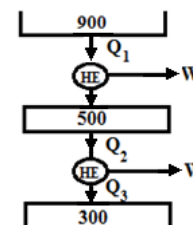
$$s_{fg} = \frac{57.24}{10} = 5.724 \text{ kJ}/kgK$$

Since specific entropy = $\frac{\text{entropy}}{\text{mass}}$)

$s_g = s_{fg} + s_f$

$= 5.724 + 0.88$

$= 6.604 \text{ kJ}/ \text{ kg K}$

Q.17 (50)

$$\eta_l = 1 - \frac{T_{L1}}{T_{H1}} = \frac{900 - 500}{900} = 0.444$$

Also, $\eta_l = 1 - \frac{Q_2}{Q_1}$

$\therefore Q_2 = 83.33\,MJ$

Also, $\eta_2 = 1 - \frac{T_{L2}}{T_2} = 1 - \frac{300}{500} = 0.4$

$\eta_2 = 1 - \frac{Q_3}{Q_2}$

$\therefore Q_2 = 50\ MJ$

Q.18 (-116.36)

$$s_2 - s_1 = R\ In \frac{V_2}{V_1} = 287 \ln \frac{0.6}{0.9}$$

$= -116.368\ J/kg/K$

Q.19 (4)

$$COP_{ref} = \frac{Q_L}{W_{i/p}}$$

$$COP_{ref} = \frac{7.2}{1.8} = 4$$

Q.20 (64)
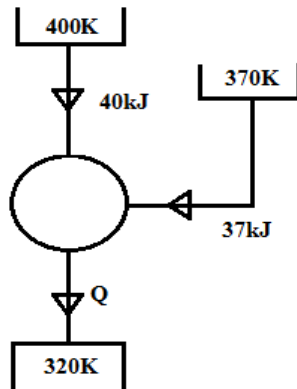
By Clausius inequality

$$\oint \frac{\delta Q}{T} = 0$$

[For reversible cycle]

$$\Rightarrow \frac{40}{400} + \frac{37}{370} - \frac{Q}{320} = 0$$

Q = 64kJ

Q.21.
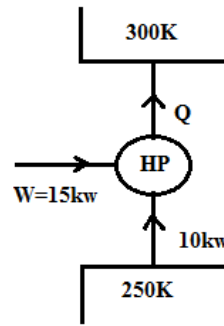
Q = 15 + 10 = 25 kw

$$Cop = \frac{desired\ effect}{work\ input}$$

$$= \frac{25}{15} = \frac{5}{3}$$

Cop = 1.67

Q.22. By first law

$$\delta Q = du + \delta w$$

$$0 = C_v[T_f - T_i] - 100$$

$$1 \times [T_f - 300] = 100$$

$$T_f = 400k$$

Entropy change of an ideal gas is given by

$$S_2 - S_1 = C_v \ln \frac{T_f}{T_i} + R \ln \frac{V_f}{V_i}$$

$$Q\ V_f = V_i$$

$$\therefore S_2 - S_1 = C_v \ln \frac{T_f}{T_i}$$

$$= 1000 \ln \left(\frac{400}{300} \right)$$

$$S_2 - S_1 = 287.6 J/k$$

Q.23 (0.21)

Ideal gas

State -1 : $T_1 = 300\,K$, $P_1 = 100\,kPa$

State -2 : $T_2 = 600\,K$, $P_2 = 500\,kPa$

$c_p = 1\,kJ/kg\text{-}K$, $c_p - c_v = R$, $c_v = 0.7\,kJ/kg\text{-}K$,

$\Rightarrow \quad c_p - c_v = 1 - 0.7 = R$

$R = 0.3\ kJ/kg\text{-}K$

Change in specific entropy

$$s_2 - s_1 = c_p \ln\frac{T_2}{T_1} - R \ln\frac{P_2}{P_1}$$

$$= 1 \times \ln\frac{600}{300} - 0.3\ln\frac{500}{100} = 0.21\ kj/kg\text{-}K$$

Q.24 **(447.213)**

$\dot{m} = 0.1\ kg/s$, $\dot{Q} = 5\,kW\,(\text{heat loss})$

Applying SFEE

$$\dot{m}\left(h_1 + \frac{1}{2}c_1^2 + gz_1\right) + \dot{Q} =$$
$$\dot{m}\left(h_2 + \frac{1}{2}c_2^2 + gz_2\right) + \dot{w}_{cv}$$

$c_1 = 0$ and $\dot{w}_{cv} = 0$
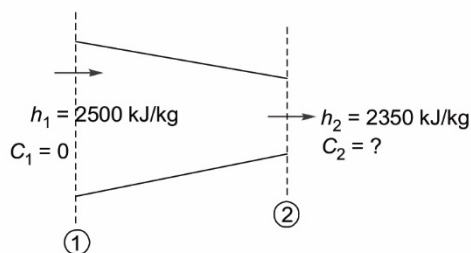
$z_1 = z_2\,(\text{assume})$

$$\dot{m}h_1 + \dot{Q} = \dot{m}h_2 + \dot{m}\frac{1}{2}c_2^2$$

$$\Rightarrow \dot{m} \times \frac{1}{2}c_2^2 = \dot{m}\left(h_1 - h_2\right) + \dot{Q}$$

$$\Rightarrow 0.1 \times \frac{1}{2}c_2^2 \times 10^{-3} = 0.1\left(2500 - 2350\right) - 5$$

$c_2 = 447.213\ m/s$

$h_1 = 2500\ kJ/kg$ $h_2 = 2350\ kJ/kg$
$C_1 = 0$ $C_2 = ?$

Q.25 **(a)**

4 AVALILABILITY AND IRREVERSIBILITY

Q.1 Considering the relationship Tds = dU + Pdv between the entropy (s), internal energy (u), pressure (P), temperature (T) and volume (v), which of the following statements is correct ?
a) It is applicable only for a reversible process
b) For an irreversible process, Tds > du + Pdv
c) It is valid only for an ideal gas
d) It is equivalent to Ist law, for a reversible process
[GATE–2003(ME)]

Q.2 A steel billet of 2000 kg mass is to be cooled from 1250 K to 450 K. The heat released during this process is to be used as a source of energy. The ambient temperature is 303 K and specific heat of steel is 0.5 kJ/ kgK. The available energy of this billet is
a) 490.44 MJ   b) 30.95 MJ
c) 10.35 MJ   d) 0.10 MJ
[GATE–2004(ME)]

Q.3 The pressure, temperature and velocity of air flowing in a pipe are 5 bars, 500 K and 50 m/s, respectively. The specific heats of air at constant pressure and at constant volume are 1.005 kJ/ kg K and 0.718 kJ/ kg K, respectively. Neglect potential energy. If the pressure and temperature of the surrounding are 1 bar and 300 K, respectively, the available energy in kJ/kg of the air stream is
a) 170   b) 187
c) 191   d) 213
[GATE–2013(ME)]

Q.4 The maximum theoretical work obtainable, when a system interacts to equilibrium with a reference environment, is called
a) Entropy   b) Enthalpy
c) Exergy   d) Rothalpy
[GATE-2014 (ME) Set-1]

Q.5 One side of a wall is maintained at 400 K and the other at 300 K. The rate of heat transfer through the wall is 1000 W and the surrounding temperature is 25°C. Assuming no generation of heat within the wall, the irreversibility (in W) due to heat transfer through the wall is _____
[GATE-2015 (ME) Set-3]

Q.6 One kg of an ideal gas (gas constant R = 287 J/kg.K) undergoes an irreversible process from state-1 (1 bar, 300 K) to state -2 (2 bar, 300 K). The change in specific entropy ($s_2 – s_1$) of the gas (in J/kg. K) in the process is _____
[GATE-2017 Set-2]

Q.7` A calorically perfect gas (specific heat at constant pressure 1000 J/kg.K) enters and leaves a gas turbine with the same velocity. The temperatures of the gas at turbine entry and exit are 1100 K and 400 K. respectively. The power produced is 4.6 MW and heat escapes at the rate of 300 kJ/s through the turbine casing. The mass flow rate of the gas (in kg/s) through the turbine is.

a) 6.14   b) 7.00

c) 7.50   d) 8.00

[GATE-2017 Set-2]

ANSWER KEY:

1	2	3	4	5	6	7
(d)	(a)	(b)	(c)	248.23	-198.9	(b)

EXPLANATIONS

Q.1 **(d)**
TdS=du+Pdv
This equation holds good for any process reversible or irreversible, undergone by a closed system, since it is a relation among properties which are independent of both.

Q.2 **(a)**
$Q = mC_p \Delta T$
Q=2000×(0.5kJ/kg K)×(1250-450)
= 800000 kJ
$\Delta S = mC \ln \frac{T_1}{T_2} = 2000 \times 0.5 \times \ln \frac{1250}{450}$
=1021.165
$A.E = Q - T_0(\Delta s)$
=490439.67 k
=490.44 MJ

Q.3 **(b)**
Available energy
$= (h_1 - h_0) - T_0(s_1 - s_0) + \left(\frac{C_1^2}{2} - \frac{C_0^2}{2} \right)$
..... (i)
$s_1 - s_0 = C_p \ln \frac{T_1}{T_0} - R \ln \frac{P_1}{P_0}$
$= 1.005 \ln \left(\frac{500}{300} \right) - 0.287 \ln \left(\frac{5}{1} \right)$
=0.05147
∴ eq.(i)
Available energy
=1.005 (500 – 300)−300(0.05147)
$+ \frac{(50)^2}{2} \times 10^{-3}$
=186.8 kJ/ kg
= 187 kJ/ kg

Q.4 **(c)**
Exergy is also known as available energy and it is maximum theoretical work obtainable, when a system interacts to equilibrium with dead state or reference environment.

Q.5 **(248.23)**
By Gauy-Studola theorem:
$I = T_0(\Delta s)_{uni}$
$(\Delta s)_{uni} = \frac{1000}{300} - \frac{1000}{400}$
= 0.8333 W/K
∴ Irreversibility:-I= (25 + 273) 0.833
=248.23 W

Q.6 **(-198.9)**

Change in specific entropy of ideal gas

$S_f - S_i = C_p \ln \frac{T_f}{T_i} - R \ln \frac{P_f}{P_i}$

$S_f - S_i = -287 \ln \left(\frac{2}{1} \right)$

$S_f - S_i = -198.93 J / kg.k$

Q.7 **(b)**
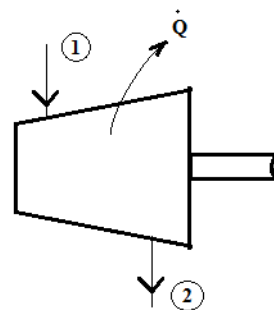
By SFEE

$h_1 + \frac{C_1^2}{2} + z_1 g + q = h_2 + \frac{C_2^2}{2} + z_2 g + w$

$C_1 = C_2$ (given)

$z_1 = z_2$ (assume)

$$\dot{m}C_p T_1 - \dot{Q} = \dot{m}C_p T_2 + \dot{W}$$

$$\dot{m} = \frac{\dot{W} + \dot{Q}}{C_p[T_2 - T_1]}$$

$$= \frac{4600 + 300}{1 \times [1100 - 400]}$$

$$\dot{m} = \frac{4900}{700} = 7\,kg\,/\,sec$$

Q.1 Which combination of the following statements is correct?

P: A gas cools upon expansion only when its Joule-Thomson coefficient is positive in the temperature range of expansion.

Q: For a system undergoing a process, its entropy remains constant only when the process is reversible.

R: The work done by closed system in an adiabatic is a point function.

S: A liquid expands upon freezing when the slope of its fusion curve on pressure- Temperature diagram is negative.

a) R and S   b) P and Q
c) Q, R and S   d) P, Q and R

[GATE–2007]

Q.2 Water has a critical specific volume of 0.003155m³/kg. A closed and rigid steel tank of volume 0.025m³ contains a mixture of water and steam at 0.1MPa. The mass of the mixture is 10 kg. The tank is now slowly heated. The liquid level inside the tank

a) will rise
b) will fall
c) will remain constant
d) may rise or fall depending on the amount of heat transferred

[GATE–2007]

Q.3 2 moles of oxygen are mixed adiabatically with another 2 moles of oxygen in mixing chamber, so that the final total pressure and temperature of the mixture become same as those of the individual constituents at their initial states. The universal gas constant is given as R. The change in entropy due to mixing, per mole of oxygen, is given by

a) -Rln2   b) 0
c) Rln2   d) Rln4

[GATE–2008]

Common Data For Q.4, 5 and 6

In the figure shown, the system is a pure substance kept in a piston-cylinder arrangement. The system is initially a two-phase mixture containing 1 kg of liquid and 0.03 kg of vapour at a pressure of 100 kPa. Initially, the piston rests on a set of stops, as shown in the figure. A pressure of 200 kPa is required to exactly balance the weight of the piston and the outside atmospheric pressure. Heat transfer takes place into the system until its volume increases by 50%. Heat transfer to the system occurs in such a manner that the piston, when allowed to move, does so in a very slow (quasi-static/quasi-equilibrium) process. The thermal reservoir from which heat is transferred to the system has a temperature of 400°C. Average temperature of the system boundary can be taken as 175°C. The heat transfer to the system is 1 kJ, during which its entropy increases by 10 J/K.
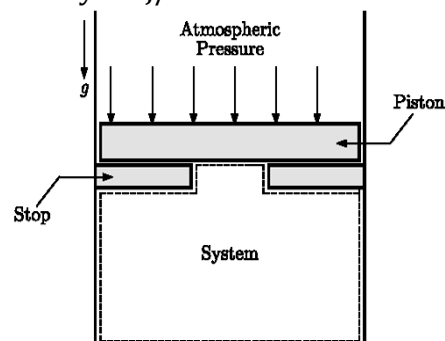
Specific volume of liquid (v_f) and vapour (v_g) phases, as well as values of saturation temperatures, are given in the table below.

Pressure (kPa)	Saturation temperature, T_{sat}(°C)	V_f (m³/kg)	v_g (m³/kg)
100	100	0.001	0.1
200	200	0.0015	0.002

Q.4 At the end of the process, which one of the following situations will be true?
a) superheated vapour will be left in the system
b) no vapour will be left in the system
c) a liquid + vapour mixture will be left in the system
d) the mixture will exist at a dry saturated vapour state

[GATE–2008]

Q.5 The work done by the system during the process is
a) 0.1 kJ   b) 0.2 kJ
c) 0.3 kJ   d) 0.4 kJ

[GATE–2008]

Q.6 The net entropy generation (considering the system and the thermal reservoir together) during the process is closest to
a) 7.5 J/K   b) 7.7 J/K
c) 8.5 J/K   d) 10 J/K

[GATE–2008]

Q.7 A pure substance at 8 MPa and 400°C is having a specific internal energy of 2864 kJ/kg and a specific volume of 0.03432 m³/kg. Its specific enthalpy (in kJ/ kg) is ___.

[GATE-2014 (2)]

Q.8 1.5 kg of water is in saturated liquid state at 2bar (V_f=0.001061 m³/kg, u_f=504.0 kJ/kg, h_f = 505kJ/kg). Heat is added in a constant pressure process till the temperature of water reaches 400°C (v=1.5493m³/kg, u=2967.0kJ/kg, h=3277.0 kJ/kg). The heat added (in kJ) in the process is _____.

[GATE-2014 (1)]

Q.9 The Vander Waals equation of state is $\left(p+\dfrac{a}{v^2}\right)(v-b)=RT$, where p is pressure, v is specific volume, T is temperature and R is characteristic gas constant. The SI unit of a is

a) $\dfrac{J}{kg.K}$ b) $\dfrac{m^3}{kg}$

c) $\dfrac{m^5}{kgs^2}$ d) $\dfrac{Pa}{kg}$

[GATE-2015 (2)]

Q.10 A rigid container of volume 0.5 m³ contains 1.0 kg of water at 120°C ($v_f = 0.00106 \ m^3/kg$, $v_g = 0.8908 \ m^3/kg$)
The state of water is
a) Compressed liquid
b) Saturated liquid
c) A mixture of saturated liquid and saturated vapour
d) Superheated vapor

[GATE-2015 (3)]

Q.11 For water at 25°C, $\dfrac{dp_s}{dT_s} = 0.189$ kPa/K (p_s is the saturation pressure in kPa and T_s is the saturation temperature in K) and the specific volume of dry saturated vapour is 43.38 m³/kg. Assume that the specific volume of liquid is negligible in comparison with that of vapour. Using the Clausius-Clapeyron equation, an estimate of the enthalpy of evaporation of water at 25°C (in kJ/kg) is _____

[GATE-2016 (1)]

Q.12 The INCORRECT statement about the characteristics of critical point of a pure substance is that
a) there is no constant temperature vaporization process
b) it has point of inflection with zero slope
c) the ice directly converts from solid phase to vapor phase
d) saturated liquid and saturated vapor states are identical

[GATE-2016 (3)]

Q.13 Which one of the following statements is correct for a superheated vapour?

(A) Its pressure is less than the saturation pressure at a given temperature.

(B) Its temperature is less than the saturation temperature at a given pressure.

(C) Its volume is less than the volume of the saturated vapour at a given temperature.

(D) Its enthalpy is less than the enthalpy of the saturated vapour at a given pressure.

[GATE-2018 Set-1]

Q.14 A vehicle powered by a spark ignition engine follows air standard Otto cycle (γ=1.4). The engine generates 70 kW while consuming 10.3 kg/hr of fuel. The calorific value of fuel is 44,000 kJ/kg. The compression ratio is _____ (correct to two decimal places).

[GATE-2018 Set-1]

Q.15 A tank of volume 0.05 m3 contains a mixture of saturated water and saturated steam at. The mass of the liquid present is 8 kg. The entropy (in kJ/kg K) of the mixture is _____ (correct to two decimal places).

Property data for saturated steam and water are:

At $200\,^0C, p_{sat} = 1.5538$ MPa

$v_f = 0.001157\ m^3/kg, v_g = 0.12736\ m^3/kg$

$s_{fg} = 4.1014 kJ/kg\ K,\ s_f = 2.3309 kJ/kg\ K$

[GATE-2018 Set-1]

ANSWER KEY:

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
(a)	(a)	(b)	(a)	(d)	(c)	3138	4158	(c)	(c)	2443	(c)	(a)	7.61	2.5

EXPLANATIONS

Q.1 **(a)**

Q.2 **(a)**
Given
$v_{cri} = 0.003155 \, m^3/kg$,
$V = 0.025 m^3, P = 0.1MPa$ and
M=10 kg.
We know, Rigid means volume is constant.
Specific volume,
$$v_s = \frac{V}{m} = \frac{0.025}{10} = 0.0025 m^3/kg$$
We see that the critical specific volume is more than the specific volume and during the heating process, both the temperature and the pressure remain constant, but the specific volume increases to the critical volume (i.e. critical point). The critical point is defined as the point at which the saturated liquid and saturated vapour states are identical.
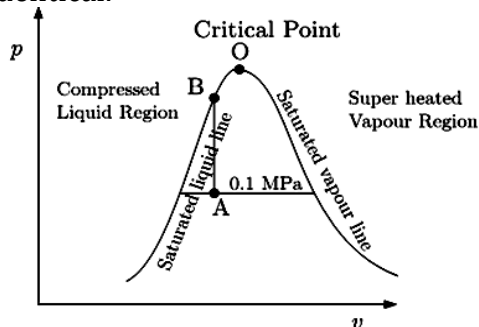
So, point (B) will touch the saturated liquid line and the liquid line will rise the Point O.

Q.3 **(b)**

Given: $T_1 = T_2, p_1 = p_2$
Universal Gas constant = R
Here given oxygen are mixed adiabatically
So, $dQ = 0$
We know $ds = \dfrac{dQ}{T} = 0$ (since dQ=0)

Q.4 **(a)**
When the vapour is at a temperature greater than the saturation temperature, it is said to exist as superheated vapour. The pressure and temperature of super heated vapour are independent properties, since the temperature may increase while the pressure remains constant.
Here vapour is at 400°C and saturation temperature is 200°C
So, at 200 kPa pressure, superheated vapour will be left in the system.

Q.5 **(d)**
Given $p_1 = 100 KPa, p_2 = 200 kPa$
$m_l = 1 kg$ and $m_v = 0.03$ kg
$$x_1 = \frac{0.03}{1.03} = 0.029$$
$$v_1 = v_f + x_1 v_{fg}$$
$$= 0.001 + 0.0029(0.1 - 0.001)$$
$$= 0.0038$$
$$V_1 = mv_1 = 1.03v_1 = 0.0039$$
Now, given that heat transfer takes place into the system until its volume increases by % 50

Therefore, $V_2 = 1.5V_1 = 0.00585$

Now Work done $= P_2(V_2 - V_1)$

$= 200(0.00585 - 0.0039)$

$= 0.4\,kJ$

Q.6 (c)

$$S_{gen} = \frac{-1000}{673} + 10$$

$= 8.514\,J/K$

Q.7 (3138.56)

$h = u + pv$

$h = 2864 + 8 \times 10^3 \times 0.03432$

$h = 3138.56\,kJ/kg$

Q.8 (4158)

$Q = m\,(h_2 - h_1)$

$= 1.5(3277 - 505)$

$= 4158\,kJ$

Q.9 (c)

Q.10 (c)

$$v = \frac{0.5}{1}\,m^3/kg = 0.5 m^3/kg$$

Since $v_f < v < v_g$ the state of water is mixture of saturated water and saturated vapour.
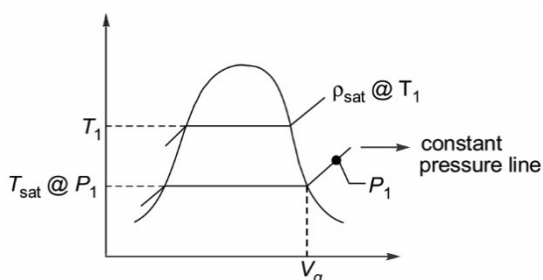
Q.11 (2443.24)

By Causius-Clapeyron equation

$$\frac{dP_s}{dT_s} = \frac{s_g - s_f}{v_g} = \frac{h_{fg}}{T_{sat} \times v_g}$$

$\therefore h_{fg} = 0.189 \times 298 \times 43.38$

$h_{fg} = 2443.24 kJ/kg$

Q.12 (c)

Q.13 (a)

$P_{sat}\,@\,T_1 \rightarrow$ saturation pressure at T_1 temperature

$P_1 \rightarrow$ pressure of superheated vapour at state 1.

$P_1 < P_{sat}\,@\,T_1$

Q.14 (7.61)

$\gamma = 1.4$

$B.P. = 70\,kN$

$\dot{m}_f = 10.3\,kg/hr$

$CV = 44000\,kJ/kg$

$$\eta = \frac{B.P.}{\dot{m}_f \times CV} = \frac{70}{\dfrac{10.3}{3600} \times 44000} \times 100\,\%$$

$\eta = 0.556$

$$\eta_{otto} = 1 - \frac{1}{(r)^{\gamma-1}} = 0.556$$

$$\Rightarrow \quad 1 - \frac{1}{(r)^{1.4-1}} = 0.556$$

$$\frac{1}{(r)^{0.4}} = 0.4439$$

$r = 7.61$

Q.15 (2.49)

Volume of liquid $= m_f \times V_f$

$= 8 \times 0.001157 = 9.256 \times 10^{-3} m^3$

$= 0.009256 \, m^3$

So volume of steam $= 0.05 - 0.009256$

$= 0.040744 \, m^3$

Mass of steam $= \dfrac{\text{Volume of steam}}{v_g}$

$= \dfrac{0.040744}{0.12736}$

$= 0.319912 \text{ kg} = m_s$

Total mass of mixture $= m_f + m_s$

$= 8 + 0.319912$

$= 8.319912 \, kg$

dryness fraction $= x = \dfrac{m_s}{m_f + m_s}$

$= \dfrac{0.319912}{8.319912} = 0.03845$

So, the entropy of the mixture is given by

$s = s_f + x \, s_{fg}$

$= 2.3309 + 0.03845 \times 4.1014$

$= 2.488 \, kg/kgK$

Q.1 The Rateau turbine belongs to the category of
a) pressure compounded turbine
b) reaction turbine
c) velocity compounded turbine
d) radial flow turbine
[GATE–2001]

Q.2 The efficiency of superheat Rankine cycle is higher than that of simple Rankine cycle because
a) the enthalpy of main stream is higher for superheat cycle
b) the mean temperature of heat addition is higher for superheat cycle
c) the temperature of steam in the condenser is high
d) the quality of steam in the condenser is low.
[GATE–2002]

Q.3 In Rankine cycle, regeneration results in higher efficiency because
a) pressure inside the boiler increases
b) heat is added before steam enters the low pressure turbine
c) average temperature of heat addition in the boiler increases
d) total work delivered by the turbine increases
[GATE–2003]

Q.4 Considering the variation of static pressure and absolute velocity in an impulse steam turbine, across one row of moving blades
a) both pressure and velocity decreases
b) pressure decreases but velocity increases
c) pressure remains constant, while velocity increases
d) pressure remains constant, while velocity decreases
[GATE–2003]

Q.5 In a gas turbine, hot combustion products with the specific heats Cp = 0.98 kJ/kgK, and Cv = 0.7538 kJ/kgK enters the turbine at 20 bar, 1500 K and exits at 1 bar. The isentropic efficiency of the turbine is 0.94. The work developed by the turbine per kg of gas flow is
a) 689.64 kJ/kg   b) 794.66 kJ/kg
c) 1009.72 kJ/kg   d) 1312.00 kJ/kg
[GATE–2003]

Q.6 The compression ratio of a gas power plant cycle corresponding to maximum work output for the given temperature limits of T_{min} and T_{max} will be

a) $\left(\dfrac{T_{max}}{T_{min}}\right)^{\frac{\gamma}{2(\gamma-1)}}$

b) $\left(\dfrac{T_{min}}{T_{max}}\right)^{\frac{\gamma}{2(\gamma-1)}}$

c) $\left(\dfrac{T_{max}}{T_{min}}\right)^{\frac{\gamma-1}{\gamma}}$

d) $\left(\dfrac{T_{min}}{T_{max}}\right)^{\frac{\gamma-1}{\gamma}}$

[GATE–2004]

Common Data for Q.7 and Q.8:
Consider a steam power plant using a reheat cycle as shown. Steam leaves the boiler and enters the turbine at 4 MPa , 350^0C (h_3 = 3095 kJ/kg). After expansion in the turbine to 400 kPa (h_4 = 2609 kJ/kg), the steam is reheated to 350°C (h_5 = 3170 kJ/kg) and then expanded in a low pressure turbine to 10 kPa (h_6 = 2165 kJ/kg). The specific volume of liquid handled by the pump can be assumed to be
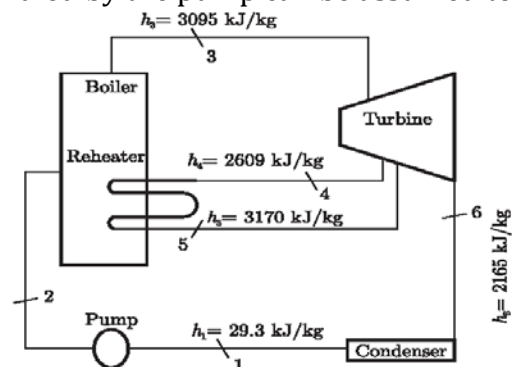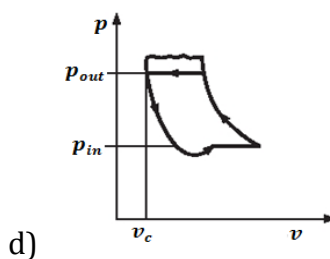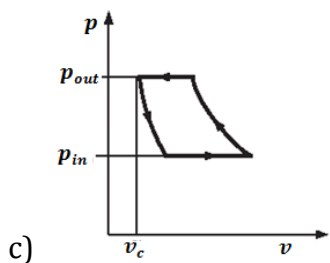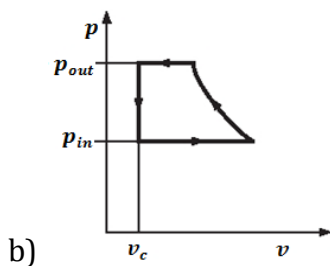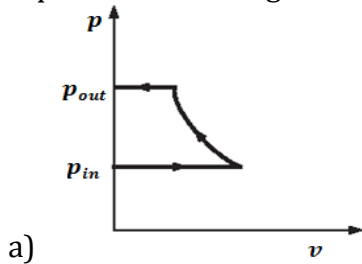
Q.7 The thermal efficiency of the plant neglecting pump work is
a) 15.8% b) 41.1%
c) 48.5% d) 58.6%
[GATE–2004]

Q.8 The enthalpy at the pump discharge (h_2) is
a) 0.33 kJ/kg b) 3.33 kJ/kg
c) 4.0 kJ/k d) 33.3 kJ/kg
[GATE–2004]

Q.9 A p -v diagram has been obtained from a test on a reciprocating compressor. Which of the following represents that diagram?

a)

b)

c)

d)

[GATE–2005]

Q.10 In the velocity diagram shown below, u = blade velocity, C = absolute fluid velocity and W = relative velocity of fluid and the subscripts 1 and 2 refer to inlet and outlet. This diagram is for
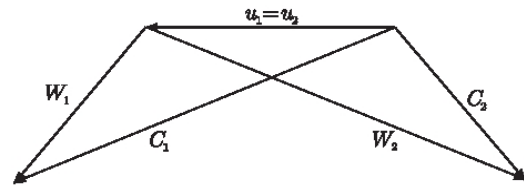
a) an impulse turbine
b) a reaction turbine
c) a centrifugal compressor
d) an axial flow compressor
[GATE–2005]

Common data for Q.11 and Q.12

In two air standard cycles-one operating in the Otto and the other on the Brayton cycle air is isentropically compressed from 300 to 450K. Heat is added to raise the temperature to 600K in the Otto cycle and to 550K in the Brayton cycle.

Q.11 In η_o and η_B are the efficiencies of the Otto and Bray ton cycles, then
a) $\eta_o = 0.25, \eta_B = 0.18$
b) $\eta_o = \eta_B = 0.33$
c) $\eta_o = 0.5, \eta_B = 0.45$
d) it is not possible to calculate the efficiencies unless the temperature after the expansion is given
[GATE–2005]

Q.12 If Wo and W_B are work outputs per unit mass, then
a) Wo > W_B
b) Wo < W_B
c) Wo = WB
d) it is not possible to calculate the work outputs unless the temperature after the expansion is given
[GATE–2005]

Common Data for Q.13 and Q.14:

The following table of properties was printed out for saturated liquid and saturated vapour of ammonia. The titles for only the first two columns are available. All that we know that the other columns (column 3 to 8) contain data on specific properties, namely, internal energy (kJ/kg), enthalpy (kJ/kg) and entropy (kJ/kgK)

t (°C)	p(kPa)						
-20	190.2	88.76	0.3657	89.05	5.6155	1299.5	1418.0
0	429.6	179.69	0.7114	180.36	5.3309	1318.0	1442.2
20	587.5	272.89	1.0408	274.30	5.0860	1332.2	1460.2
40	1554.9	368.74	1.3574	371.43	4.8662	1341.0	1470.2

Q.13 The specific enthalpy data are in columns
   a) 3 and 7   b) 3 and 8
   c) 5 and 7   d) 5 and 8
                          [GATE–2005]

Q.14 When saturated liquid at 40°C is throttled to – 20°C, the quality at exit will be
   a) 0.189   b) 0.212
   c) 0.231   d) 0.788
                          [GATE–2005]

Q.15 Given below is an extract from steam tables.

Temperature in	P_sat (Bar)	Specific volume m³/kg		Enthalpy (kJ/kg)	
		Saturated Liquid	Saturated Vapour	Saturated liquid	Saturated vapou
45	0.095	0.001010	15.26	188.4	2394.
342.24	150	0.001658	0.01033	1610.	2610.

Specific enthalpy of water in kJ/kg at 150 bar and 45°C is
   a) 203.60   b) 200.53
   c) 196.38   d) 188.45
                          [GATE–2006]

Q.16 Determine the correctness or otherwise **Assertion (A)** and the **Reason (R) Assertion (A):** In a power plant working on a Rankin cycle, the regenerative feed water heating improves the efficiency of the steam turbine.

Reason (R): The regenerative feed water heating raises the average temperature of heat addition in the Rankin cycle.
  a) Both (A) and (R) are true and (R) is the correct reason for (A)
  b) Both (A) and (R) are true but (R) is NOT the correct reason for (A)
  c) Both (A) and (R) are false
  d) (A) is false but (R) is true
                      [GATE–2006]

Q.17 Determine the correctness or otherwise of the following **Assertion (A)** and the **Reason (R)**.
Assertion (A): Condenser is an essential equipment in a steam power plant.
Reason (R): For the same mass flow rate and the same pressure rise, a water pump requires substantially less power than a steam compressor.
  a) Both (A) and (R) are true and (R) is the correct reason for (A)
  b) Both (A) and (R) are true and (R) is NOT the correct reason for (A)
  c) Both (A) and (R) are false
  d) (A) is false but (R) is true
                      [GATE–2006]

Q.18 Group I shows different heat addition process in power cycles. Likewise, Group II shows different heat removal processes. Group III lists power cycles. Match items from Groups I, II and III.

Group I	Group II	Group III
P. Pressure Constant	S. Pressure Constant	1.Rankine Cycle
Q. Volume Constant	T. Volume Constant	2. Otto cycle
R. Temperature Constant	U. Temperature Constant	3. Carnot cycle
		4. Diesel cycle
		5.Brayton cycle

  a) P-S-5, R-U-3, P-S-1, Q-T-2
  b) P-S-1, R-U-3, P-S-4, P-T-2
  c) R-T-3, P-S-1, P-T-4, Q-S-5
  d) P-T-4, R-S-3, P-S-1, P-S-5
                      [GATE–2006]

Q.19 Which combination of the following statements is correct? The incorporation of reheater in a steam power plant:
P: always increases the thermal efficiency of the plant.
Q: always increases the dryness fraction of steam at condenser inlet
R: always increases the mean temperature of heat addition.
S: always increases the specific work output.
a) P and S
b) Q and S
c) P, R and S
d) P, Q, R and S
[GATE–2007]

Q.20 A thermal power plant operates on a regenerative cycle with a single open feed water heater, as shown in the figure. For the state points shown, the specific enthalpies are: h_1 = 2800 kJ/kg and h_2 = 200 kJ/kg. The bleed to the feed water heater is 20% of the boiler steam generation rate. The specific enthalpy at state 3 is
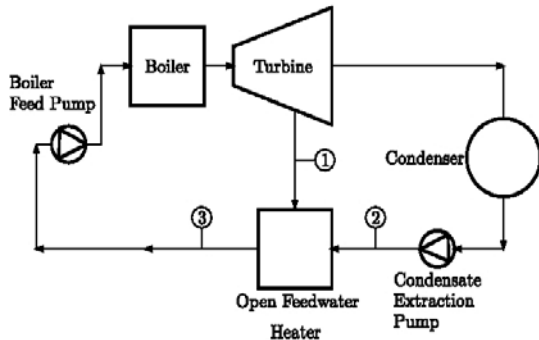
a) 720 kJ/kg
b) 2280 kJ/kg
c) 1500 kJ/kg
d) 3000 kJ/kg
[GATE–2008]

Common Data For Q.21 and Q.22
In a steam power plant operating on the Rankine cycle, steam enters the turbine at 4MPa, 350° C and exists at a pressure of 15 kPa. Then it enters the condenser and exits as saturated water. Next, a pump feeds back the water to the boiler. The adiabatic efficiency of the turbine is 90%. The thermodynamic states of water and steam are given in table h is specific enthalpy, s is specific entropy and v the specific volume;

subscripts f and g denote saturated liquid state and saturated vapour state.

State	h(kJ/kg)		s(kJ/kgK)		v(m³/kg)	
Steam: 4 MPa, 350°C	3092.5		6.5821		0.06645	
	h_f(kJ/kg)	h_g(kJ/kg)	s_f(kJ/kgK)	s_g(kJ/kgK)	v_f(m³/kg)	v_g(m³/kg)
Water: 15kPa	225.94	2599.1	0.7549	8.0085	0.001014	10.02

Q.21 The network output (kJ/kg) of the cycle is
a) 498
b) 775
c) 860
d) 957
[GATE–2010]

Q.22 Heat supplied (kJ/kg) to the cycle is
a) 2372
b) 2576
c) 2863
d) 3092
[GATE–2010]

Q.23 The values of enthalpy of steam at the inlet and outlet of a steam turbine in a Rankine cycle are 2800 kJ/kg and 1800 kJ/kg respectively. Neglecting pump work, the specific steam consumption in kg/kWh is
a) 3.60
b) 0.36
c) 0.06
d) 0.01
[GATE–2011]

Q.24 An ideal Brayton cycle, operating between the pressure limits of 1 bar and 6 bar, has minimum and maximum temperature of 300 K and 1500 K. The ratio of specific heats of the working fluid is 1.4. The approximate final temperatures in Kelvin at the end of compression & expansion processes are respectively
a) 500 and 900
b) 900 and 500
c) 500 and 500
d) 900 and 900
[GATE–2011]

Q.25 Specific enthalpy and velocity of steam at inlet and exit of a steam turbine, running under steady state, are as given below:

	Specific Enthalpy (kJ/kg)	Velocity (m/s)
Inlet steam condition	3250	180
Exit steam condition	2360	5

The rate of heat loss from the turbine per kg of steam flow rate is 5 kW. Neglecting changes in potential energy of steam, the power developed in kW by the steam turbine per kg of steam flow rate is

a) 901.2   b) 911.2
c) 17072.5   d) 17082.5

[GATE–2013]

Common Data For Q.26 and 27

In a simple Brayton cycle, the pressure ratio is 8 and temperatures at the entrance of compressor and turbine are 300 K and 1400 K, respectively. Both compressor and gas turbine have isentropic efficiencies equal to 0.8. For the gas, assume a constant value of C_p (specific heat at constant pressure) equal to 1 kJ/kg-K and ratio of specific heats as 1.4. Neglect changes in kinetic and potential energies.

Q.26 The power required by the compressor in kJ/kg of gas is

a) 194.7   b) 243.4
c) 304.3   d) 378.5

[GATE–2013]

Q.27 The thermal efficiency of the cycle in percentage (%) is

a) 24.8   b) 38.6
c) 44.8   d) 53.1

[GATE–2013]

Q.28 In a power plant, water (density =1000 kg/ m³) is pumped from 80 kPa to 3 MPa. The pump has an isentropic efficiency of 0.85. Assuming that the temperature of the water remains the same, the specific work (in kJ/ kg) supplied to the pump is

a) 0.34   b) 2.48
c) 2.92   d) 3.43

[GATE-2014 (1)]

Q.29 The thermal efficiency of an air-standard Brayton cycle in terms of pressure ratio r_p and γ (= C_p/C_v) is given by

a) $1 - \dfrac{1}{r_p^{\gamma-1}}$ b) $1 - \dfrac{1}{r_p^{\gamma}}$

c) $1 - \dfrac{1}{r_p^{1/\gamma}}$ d) $1 - \dfrac{1}{r_p^{(\gamma-1)/\gamma}}$

[GATE-2014 (2)]

Q.30 For a gas turbine power plant, identify the correct pair of statements.

P. Smaller in size compared to steam power plant for same power output
Q. Starts quickly compared to steam power plant
R. Works on the principle of Rankine cycle
S. Good compatibility with solid fuel

a) P and Q   b) R and S
c) Q and R   d) P and S

[GATE-2014 (3)]

Q.31 An ideal reheat Rankine cycle operates between the pressure limits of 10 kPa and 8 MPa, with reheat being done at 4 MPa. The temperature of steam at the inlets of both turbines is 500°C and the enthalpy of steam is 3185 kJ/kg at the exit of the high pressure turbine and 2247 kJ/kg at the exit of low pressure turbine. The enthalpy of water at the exit from the pump is 191 kJ/ kg. Use the following table for relevant data

Super heated steam temperature (°C)	Pressure (MPa)	v (m³/kg)	h (kJ/kg)	s (kJ/kg. K)
500	4	0.08644	3446	7.0922
500	8	0.04177	3399	6.7266

Disregarding the pump work, the cycle efficiency (in percentage) is ___

[GATE-2014 (1)]

Q.32 In an ideal Brayton cycle, atmospheric air (ratio of specific heats, C_p/C_v = 1.4, specific heat at constant pressure = 1.005 kJ/kgK) at 1 bar and 300 K is compressed to

8 bar. The maximum temperature in the cycle is limited to 1280 K. If the heat is supplied at the rate of 80 MW, the mass flow rate (in kg/s) of air required in the cycle is

[GATE-2014 (2)]

Q.33 Steam at a velocity of 10m/s enters the impulse turbine stage with symmetrical blading having blade angle 30°. The enthalpy drop in the stage is 100 kJ. The nozzle angle is 20°.The maximum blade efficiency (in percent) is _____.

[GATE-2014 (2)]

Q.34 Steam with specific enthalpy (h) 3214 kJ/kg enters an adiabatic turbine operating at steady state with a flow rate 10 kg/s. As it expands, at a point where h is 2920 kJ/kg, 1.5 kg/s is extracted for heating purposes. The remaining 8.5 kg/s further expands to the turbine exit, where h = 2374 kJ/kg. Neglecting changes in kinetic and potential energies, the net power output (in kW) of the turbine is ____.

[GATE-2014 Set-4]

Q.35 Which of the following statements regarding a Rankine cycle with reheating are TRUE?
  i)   increase in average temperature of heat addition
  ii) reduction in thermal efficiency
  iii) drier steam at the turbine exit
  a) only (i) and (ii) are correct
  b) only (ii) and (iii) are correct
  c) only (i) and (iii) are correct
  d) (i),(ii) and (iii) are correct

[GATE-2015 (2)]

Q.36 Steam enters a well insulated turbine and expands isentropically throughout. At an intermediate pressure, 20 percent of the mass is extracted for process heating and the remaining steam expands isentropically to 9 kPa

Inlet to turbine: P = 14MPa, T = 560°C, h= 3486 kJ/kg, s = 6.6 kJ/kgK
Intermediate stage: h = 2776 kJ/kg
Exit of turbine: P = 9kPa,
h_f =174 kJ/kg, h_g =2574 kJ/kg,
 s_f = 0.6kJ/kgK, s_g = 8.1kJ/kgK
If the flow rate of steam entering the turbine is 100 kg/s, then the work output (in MW) is ___.

[GATE-2015 (1)]

Q.37 In a Rankine cycle, the enthalpies at turbine entry and outlet are 3159 kJ/kg. and 2187 kJ/kg respectively. If the specific pump work is 2 kJ/kg, the specific steam consumption (in kg/kW-h) of the cycle based on the net output is ____.

[GATE-2015 (2)]

Q.39 The INCORRECT statement about regeneration in vapor power cycle is that
  a) it increases the irreversibility by adding the liquid with higher energy content to the steam generator
  b) heat is exchanged between the expanding fluid in the turbine and the compressed fluid before heat addition
  c) the principle is similar to the principle of Stirling gas cycle
  d) it is practically implemented by providing feed water heaters

[GATE-2016(1)]

Q.40 In a steam power plant operating on an ideal Rankine cycle, superheated steam enters the turbine at 3 MPa and 350°C. The condenser pressure is 75 kPa. The thermal efficiency of the cycle is _____ percent.
Given data:
For saturated liquid,

At P=75kPa, h_f =384.39kJ/kg, v_f =0.001037m³/kg, s_f =1.213kJ/kg-K
At 75 kPa, h_{fg} = 2278.6 kJ/kg, s_{fg}= 6.2434 kJ/kg-K
At P=3MPa & T=350°C (superheated steam), h = 3115.3 kJ/kg, s = 6.7428 kJ/kg-K

[GATE-2016(1)]

Q.41 Consider a simple gas turbine (Brayton) cycle and a gas turbine cycle with perfect regeneration. In both the cycles, the pressure ratio is 6 and the ratio of the specific heats of the working medium is 1.4. The ratio of minimum to maximum temperatures is 0.3 (with temperatures expressed in K) in the regenerative cycle. The ratio of the thermal efficiency of the simple cycle to that of the regenerative cycle is _____

[GATE-2016(2)]

Q.42 In a 3-stage air compressor, the inlet pressure is p_1, discharge pressure is p_4 and the intermediate pressures are p_2 and p_3 (p₂ < p₃). The total pressure ratio of the compressor is 10 and the pressure ratios of the stages are equal. If p_1 = 100 kPa, the value of the pressure p_3 (in kPa) is _____

[GATE-2016(3)]

Q.43 The Pressure ratio across a gas turbine (for air, specific heat at constant pressure,

c_p = 1040J/kg.K and ratio of specific heats, $\gamma = 1.4$) is 10. If the inlet temperature to the turbine is 1200K and the isentropic efficiency is 0.9, the gas temperature at turbine exit is _____ K.

[GATE-2017 Set-1]

Q.44 In the Rankine cycle for a steam power plant the turbine entry and exit enthalpies are 2803 kJ/kg and 1800 kJ/kg, respectively. The enthalpies of water at pump entry and exit are 121 kJ/kg and 124 kJ/kg, respectively. The specific steam consumption (in kg/k W.h) of the cycle is _____
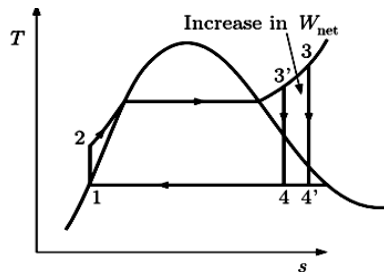
[GATE-2017 Set-2]

ANSWER KEY:

1	2	3	4	5	6	7	8	9	10	11	12	13	14
(a)	(b)	(c)	(d)	(a)	(a)	(b)	(d)	(d)	(b)	(b)	(a)	(d)	(b)
15	**16**	**17**	**18**	**19**	**20**	**21**	**22**	**23**	**24**	**25**	**26**	**27**	**28**
(d)	(a)	(b)	(a)	(b)	(a)	(c)	(c)	(a)	(a)	(a)	(c)	(a)	(d)
29	**30**	**31**	**32**	**33**	**34**	**35**	**36**	**37**	**38**	**39**	**40**	**41**	**42**
(d)	(a)	40.73	108.07	88.3	7581	(c)	125.56		(a)	3.7	26.01	0.8	464.14
43	**44**												
679	3.6												

EXPLANATIONS

Q.1 (a)

The Rateau turbine is a pressure compounded turbine.

Q.2 (b)

The average temperature at which heat is transferred to steam can be increased without increasing the boiler pressure by superheating the steam to high temperatures. The effect of superheating on the performance of vapour power cycle is shown on a T - s diagram. Thus both the net work and heat input increase as a result of superheating the steam to a higher temperature. The overall effect is an increase in thermal efficiency.

Since, the average temperature at which heat is added increases.

Q.3 (c)
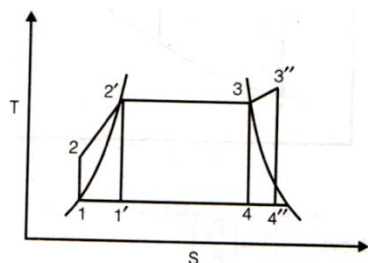
The object of the regenerative feed heating cycle is to supply the working fluid to the boiler at same state between 2 and 2' (rather than at state 2) there by increasing the average temperature of heat addition to the cycle.

Q.4 (d)

In impulse turbine, the steam pressure remains constant while it flows through the moving blades, where only the kinetic energy converts into mechanical energy.
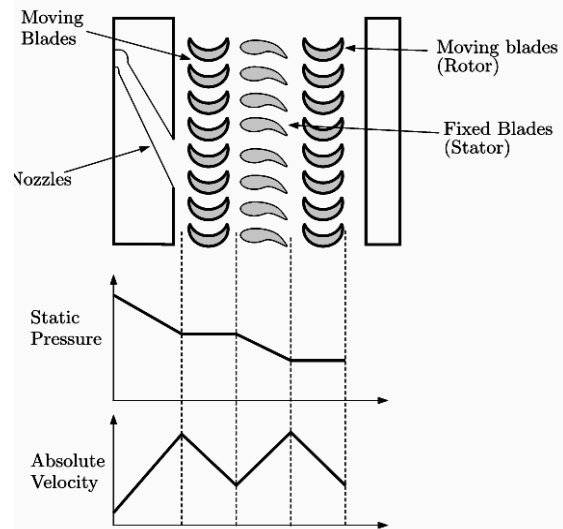
Q.5 (a)

$\therefore$ Given, $\quad T_3 = 1500 \text{ K}$

$C_p = 0.98 \text{ kJ/kg K}$

$C_V = 0.7538 \text{ kJ/ kg K.}$

$\therefore \gamma = \dfrac{C_p}{C_V} = 1.3$
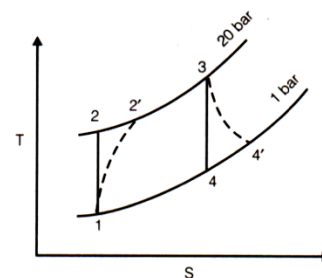
Now, $\quad \dfrac{T_4}{T_3} = \left(\dfrac{P_4}{P_3}\right)^{\frac{\gamma-1}{r}}$

$\Rightarrow \quad \dfrac{T_4}{1500} = \left(\dfrac{1}{20}\right)^{\frac{0.3}{1.3}}$

$T_4 = 751.37\text{K}$

Now turbine efficiency;

$$\eta = \frac{T_3 - T_4'}{T_3 - T_4}$$

$$\Rightarrow 0.94 = \frac{1500 - T_4'}{1500 - 751.34}$$

$$\therefore \quad T_4' = 796.25 \text{ K}$$

Turbine work;

$$W_t = c_p(T_3 - T_4')$$
$$= 0.98 \, (1500 - 796.25)$$
$$= 689.67 \text{ kJ/kg K}$$

Q.6 (a)
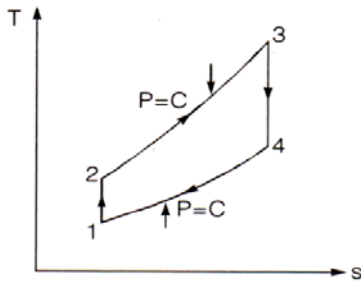
Work output during the cycle

$$= mC_p(T_3 - T_2) - mC_p(T_4 - T_1)$$

$$= mC_p(T_3 - T_4) - mC_p(T_2 - T_1)$$

$$= mC_p\left[T_3\left(1 - \frac{T_4}{T_3}\right) - T_1\left(\frac{T_2}{T_1} - 1\right)\right]$$

Since .

$$= \frac{T_3}{T_4} = \frac{T_2}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

$$\left(\text{Let } \frac{\gamma-1}{\gamma} = x, mC_p = K\right)$$

$$\therefore \text{ Work/cycle}$$

$$= K\left[T_3\left(1 - \frac{1}{r_p^x}\right) - T_1\left(\frac{T_4}{r_p^x} - 1\right)\right]$$

For maximum power, differentiate with respect to r_p

$$\Rightarrow \quad \frac{dW}{dr_p} = K\left[T_3 \times \frac{x}{r_p^{(x-1)}} - T_1 x r_p^{(x-1)}\right] = 0$$

(For max.)

$$\Rightarrow \quad \frac{xT_3}{r_p^{(x+1)}} = T_1 x (r_p)^{(x-1)}$$

$$\Rightarrow \quad r_p^{2x} = \frac{T_3}{T_1}$$

$$\therefore \quad r_p = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2x}}$$

$$\Rightarrow \quad r_p = \left(\frac{T_{max}}{T_{min}}\right)^{\frac{\gamma}{2(\gamma-1)}}$$

Q.7 (b)

Power obtained from plant

$$= (h_3 - h_4) + (h_5 - h_6)$$

Heat supplied to the plant.

$$= (h_3 - h_1) - (h_5 - h_4)$$

Thermal efficiency

$$= \frac{\text{Power obtained}}{\text{Heat supplied}}$$

$$= \frac{(3095 - 2609) + (3170 - 2165)}{(3170 - 2609) + (3095 - 29.3)}$$

$$= 0.4111 = 41.11\%$$

Q.8 (d)

Enthalpy at exit of pump must be greater than enthalpy at inlet of pump i.e. h_2 must be greater, then h_1 = 29.3 kJ/kg. Among the given four options only one option is greater than h_1 = 29.3 kJ/kg, which is 33.3 kJ/kg. Hence option (D) is correct.
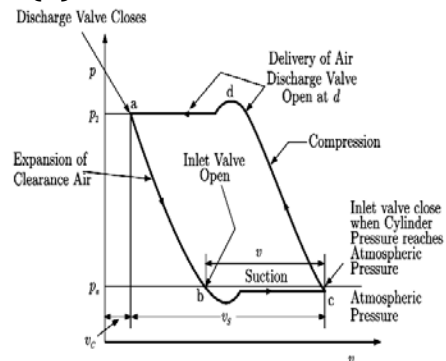
Or Pump work = vdP

$$= \frac{1}{\rho}(\text{Boiler pressure } - \text{Condenser pressure})$$

$$= \frac{1}{10^3} \, (4000 - 10) = 3.99 \text{ kJ/kg}$$

$$\therefore h_2 = 29.3 + 3.99 = 33.3 \text{ kJ/kg}$$

Q.9 (d)

From above figure, we can easily see that option (d) is same.

Q.10 (b)

Velocity of flow is constant throughout the stage and the diagram is symmetrical hence it is a diagram of reaction turbine.
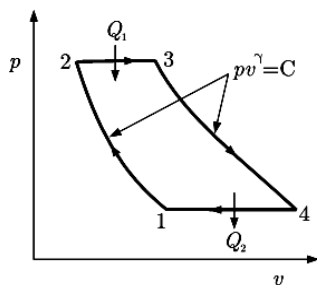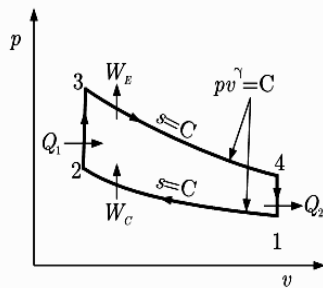
Q.11 (b)

We know that efficiency.

$$\eta_{Otto} = \eta_{Brayton} = 1 - \frac{T_1}{T_2}$$

$$\eta_{Otto} = \eta_{Brayton} = 1 - \frac{300}{450}$$

$$= 1 - \frac{6}{9} = 0.33$$

So, $\eta_{Otto} = \eta_{Brayton} = 33\%$

Q.12 (a)

From the previous part of the question

$T_{3(Otto)} = 600$ K, $T_{3(Brayton)} = 550$ K,

From the $P - v$ diagram of Otto cycle, we have

$$W_0 = Q_1 - Q_2 = C_V(T_3 - T_2) - C_V(T_4 - T_1) \quad ...(i)$$

For process 3 – 4,

$$\frac{T_3}{T_4} = \left(\frac{V_4}{V_5}\right)^{\gamma-1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} \quad V_4 = V_1, V_3 = V_2$$

For process 1 – 2,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1}$$

So, $\dfrac{T_3}{T_4} = \left(\dfrac{T_2}{T_1}\right)$

$$T_4 = \frac{T_3}{T_2} \times T_1 = \frac{600}{450} \times 300 = 400K$$

And

$W_0 = C_V (600 - 450) - C_V (400 - 300)$
$= 150\, C_V - 100\, C_V = 50\, C_V \quad ...(ii)$

From $P - v$ diagram of Brayton cycle, work done, is,

$W_B = Q_1 - Q_2 = C_p(T_3 - T_2) - C_p (T_4 - T_1)$

And

$$T_4 = \frac{T_1}{T_2} \times T_3 = \frac{300}{450} \times 550 = 366.67K$$

$$W_s = C_p (550 - 450) - (366.67 - 300)$$

$$= 33.33\, c_p \quad ... (iii)$$

Dividing equation (ii) by (iii), we get

$$\frac{W_O}{W_B} = \frac{50 C_v}{33.33 C_p} = \frac{50}{33.33\gamma}$$

$$\frac{C_p}{C_v} = \gamma \text{ and } \gamma = 1.4$$

$$= \frac{50}{33.33 \times 1.4} = \frac{50}{46.662} > 1$$

From this, we see that,

$W_O > W_B$

Q.13 (d)

From saturated ammonia table column 5 and 8 are the specific enthalpy date column.

Q.14 (b)

The enthalpy of the fluid before throttling is equal to the enthalpy of fluid after throttling because in throttling process enthalpy remains constant.

$h_1 = h_2$

$371.43 = 89.05 + x (1418 - 89.05)$

$h = h_f + x\left(h_g - h_f\right)$

$= 89.05 + x (1328.95)$

$$x = \frac{282.38}{1328.95} = 0.212$$

Q.15 (d)

When the temperature of a liquid is less than the saturation temperature at the given pressure, the liquid is called compressed liquid (state 2 in figure).The pressure and temperature of compressed liquid may vary independently and a table of properties like the superheated vapour table could be arranged, to give the properties at any p and T. The properties of liquids vary little with pressure. Hence, the properties are taken from the saturation table at the temperature of the compressed liquid. So, from the given table at T = 45^0 C, Specific enthalpy of water =188.45 kJ/kg.
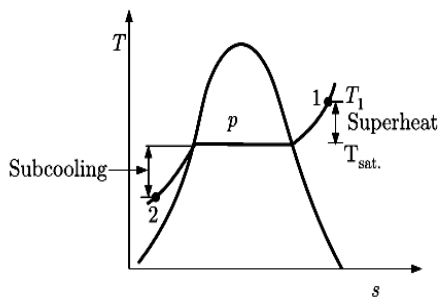
Q.16 (a)

Q.17 (b)

(a) Condenser is essential equipment in a steam power plant because when steam expands in the turbine and leaves the turbine in the form of super saturated steam. It is not economical to feed this steam directly to the boiler. So, condenser is used to condense the steam into water and it is an essential part (equipment) in steam power plant. Assertion (a) is correct.

(b) The compressor and pumps require power input. The compressor is capable of compressing the gas to very high pressures. Pump work very much like compressor except that they handle liquid instead of gases. Now for same mass flow rate and the same pressure rise, water pump require very less power because the specific volume of liquid is very less as compare to specific volume of vapour.

Q.18 (a)

Q.19 (b)

Whether the cycle efficiency increases or not depends upon the mean temperature of heat addition. In practice the use of reheat only gives a small increase in cycle efficiency, but it increases the net work output by making possible the use of higher pressures, keeping the quality of steam at the outlet of the turbine within permissible limits.

Q.20 (a)

Given: $h_1 = 2800$kJ/kg h_2=200kJ/ kg

From the given diagram of thermal power plant, point 1 is directed by the Boiler to the open feed water heater and point 2 is directed by the pump to the open feed water heater. The bleed to the feed water heater is 20% of the boiler steam generation i.e. 20% of h_1

Open Feed Water Heater
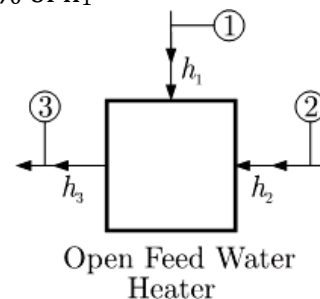
So,h_2 = 20% of h_1+80% of h_2
= 0.2×2800 + 0.8× 200 = 720 kJ/kg

Q.21 (c)

$$h_1 = 3092.5 \text{kJ}/\text{kg}$$

$$h_2 = h_f + x\left(h_g - h_f\right)$$

Now, $\quad s_1 = s_2 = s_{f2} + x\left(s_{g2} - s_{f2}\right)$

$6.5821 = 0.7549 + x(8.0085 - 0.7549)$

x = 0.8033

Dryness fraction at exit of turbine
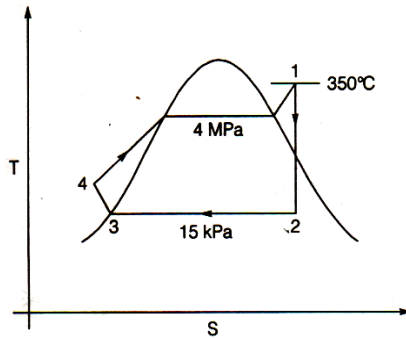
Hence,

$h_2 = 225.94 + 0.8033(2599.1 - 225.94)$

$h_2 = 2132.3 \, kj/kg$

Turbine work = $h_1 - h_2$

$= 3092.5 - 2132.3$

$= 960.2 \, kJ/kg$

$W_{act} = W_{th} \times \eta_{isen}$

$= 0.9 \times 960.2$

$= 864.2 \, kJ/kg$

Pump work $= vdP$

$= 0.001014 \, (4000 - 15)$

$= 4.04079 \, kJ/kg$

Net work= Turbine work – Pump work

$= 864.2 - 4.04$

$= 860.16 \, kJ/kg$

Q.22 (c)

Heat supplied $= h_1 - h_4$

$h_4 = h_3 + $ Pump work

$= 225.94 + 4.04078$

$= 229.98 \, kJ/kg$

Heat supplied = $h_1 - h_4$

$= 3092.5 - 229.98$

$= 2862.519 \, kJ/kg$

Q.23 (a)

Specific steam consumption (kg/kWh)

$= \dfrac{3600}{\text{Network Done}} = \dfrac{3600}{(2900 - 1900)}$

$= 3.6 \, kg/kWh$

Q.24 (a)

$P_1 = 1 \, bar$ $T_{minimum} = 300 \, K$

$P_2 = 6 \, bar$ $T_{maximum} = 1500 \, K$

$\gamma = 1.4$
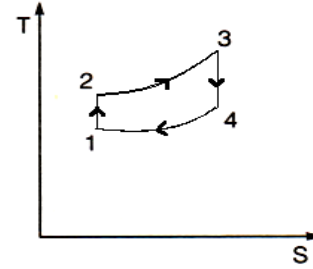
$\dfrac{T_2}{T_1} = \left(\dfrac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$

$\dfrac{T_2}{300} = \left(\dfrac{6}{1}\right)^{\frac{0.4}{1.4}}$

$T_2 = 500.5 K$

$\dfrac{T_3}{T_4} = \left(\dfrac{6}{1}\right)^{\frac{0.4}{1.4}}$

$\Rightarrow T_4 = 900 K$

Q.25 (a)

Applying SFEE
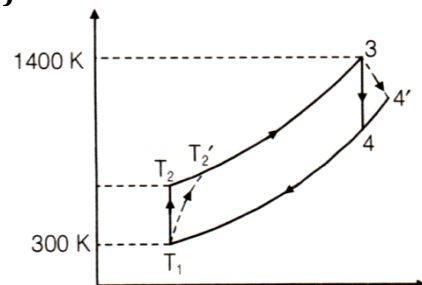
$mh_1 + \dfrac{mV_1^2}{2} + Q = mh_2 + \dfrac{mV_2^2}{2} + W$

$\Rightarrow 3250 + \dfrac{180^2}{2000} - 5 = 2360 + \dfrac{5^2}{2000} + W$

(Since m = 1 kg/s)

$\Rightarrow W = 901.2 \, kW$

Q.26 (c)

$W_C = C_p(T_2' - T_1)$

$\dfrac{T_2}{T_1} = (r_p)^{\frac{\gamma-1}{\gamma}}$

$\Rightarrow T_2 = 300 \times (8)^{\frac{0.4}{1.4}}$

$T_2 = 543.43 K$

Efficiency of compressor

$\eta_c = \dfrac{T_2 - T_1}{T_2' - T_1}$

$$0.8 = \frac{543.43 - 300}{T_2^{'} - T_1}$$

$T_2^{'} = 604.28k$

$W_c = 1(604.28 - 300)$

$W_c = 304.28 \text{ kJ/kg}$

Power required by the compressor

Q.27 (a)

$$\frac{T_3}{T_4} = (\gamma_p)^{\frac{\gamma-1}{\gamma}}$$

$$\Rightarrow T_4 = \frac{1400}{(8)^{\frac{0.4}{1.4}}} = 772.86K$$

Efficiency of turbine

$$\eta_T = \frac{T_3 - T_4^{'}}{T_3 - T_4}$$

$$0.8 = \frac{1400 - T_4^{'}}{T_3 - T_4}$$

$$0.8 = \frac{1400 - T_4^{'}}{1400 - 772.86}$$

$T_4^{'} = 898.288 \text{ K}$

Work of turbine,

$W_T = C_p (T_3 - T_4^{'})$

$= 1 (1400 - 898.288)$

$= 501.712 \text{ kW/ kg}$

Thermal efficiency of the cycle

$$\eta_T = \frac{W_T - W_C}{Q}$$

$$= \frac{501.712 - 304.28}{1(1400 - 604.28)} = 24.8\%$$

Q. 28 (d)

$work_{isen} = vdp$

$$= \frac{1}{\rho water}(P_2 - P_1)$$

$$= \frac{(3000 - 80)}{1000}$$

$$= \frac{2920}{1000} = 2.92 \text{kJ / kg}$$

$$work_{act} = \frac{w}{\eta} = \frac{2.92}{0.85} = 3.43$$

Q.29 (d)

Q.30 (a)

Q.31 (40.73)
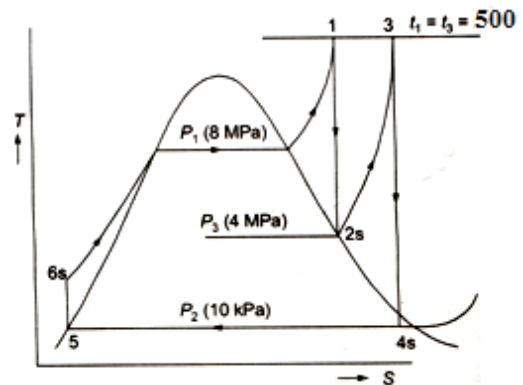
Given,

$P_2 = 10 \text{kPa}$

$P_1 = 8 \text{kPa}$

$P_3 = 4 \text{kPa}$

$h_{2s} = 3185 \text{ kJ/kg}$

$h_{4s} = 2247 \text{ kJ/kg}$

$h_1 = 3399 \text{ kJ/kg}$

$h_3 = 3446 \text{ kJ/kg}$

$h_{6s} = 191 \text{ kJ/kg}$

$W_T = (h_1 - h_{2s}) + (h_3 - h_{4s})$

$Q_S = (h_1 - h_{6s}) + (h_3 - h_{2s})$

$\therefore W_T = (3399 - 3185) + (3446 - 2247)$

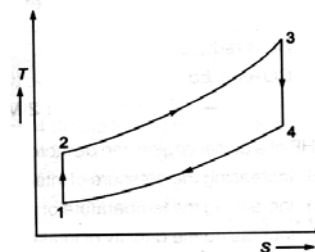$= 1413 \text{ kJ/kg}$

$Q_S = (3399 - 191) + (3446 - 3185)$

$= 3469 \text{ kJ/kg}$

$$\eta = \frac{W_T}{Q_s} = \frac{1413}{3469}$$

$= 0.4073$

Or $= 40.73\%$

Q.32 (108.071)

Given, $\dfrac{C_p}{C_v} = \gamma = 1.4$

$C_p = 1.005 \text{ kJ/ kgK}$

$P_1 = 1 \text{bar}$

$T_1 = 300K$

$P_2 = 8 \text{bar}$

$T_3 = T_{max} = 1280$ K

For process 1-2,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = 8^{\frac{0.4}{1.4}}$$

$= 1.8114$

$\therefore T_2 = 300 \times 1.8114$

$= 543.43$K

We know that $Q_s = \dot{m}C_p (T_3 - T_2)$

$= 80$MW (given)

$$\dot{m} = \frac{80 \times 10^6}{1.005 \times 10^3 (1280 - 543.43)}$$

$= 108.071$ kg/s

Q.33 (88.3)

We know that for maximum blade efficiency,
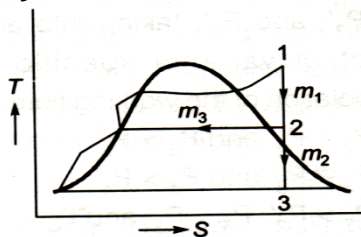
$$\rho = \frac{\cos\alpha}{2}$$

where $\rho = \dfrac{u}{V_1}$

And maximum balde efficiency is given by:

$(\eta_b)_{max} = \cos^2\alpha$

$= (\cos 20°)^2 = 0.883$

$= 88.3\%$

Q.34 (7581)

$m_1 = 10$kg/s

$m_3 = 1.5$kg/s

$m_2 = 8.5$kg/s
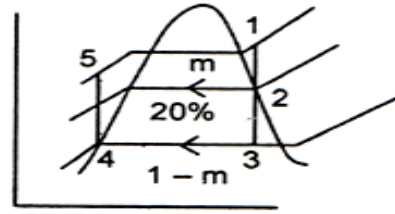
$h_1 = 3214$kJ/kg

$h_2 = 2920$kJ/kg

$h_3 = 2374$ kJ/kg

$W_T = m_1 (h_1 - h_2) + m_2 (h_2 - h_3)$

$W_T = 10(3214 - 2920) + 8.5(2920 - 2374)$

$W_T = 7581$kW

Q.35 (c)

Q.36 (125.56)

$s_1 = s_3$

$\Rightarrow 6.6 = 0.6 + x_3 [7.5]$

$x_3 = 0.8$

$h_3 = 174 + 0.8 [2574 - 174]$

$= 2094$ kJ/kg

$\therefore W = 100[h_1 - h_2] + 80[h_2 - h_3]$

$= 100[3486-2776]+80[2776-2094]$

$= 125560$ kW $= 125.56$ MW

Q.37 *

Q.38 (3.7)

Specific steam consumption $= \dfrac{3600}{W_{net}}$

Now, $W_T = h_2 - h_1$

$= 3159 - 2187$

$\therefore W_T = 972$kJ / kg

And $W_p = 2$kJ / kg

Thus specific steam consumption

$= \dfrac{3600}{972 - 2}$ kg/ kWh= 3.7815 kg/ kWh

Q.39 (a)

Q.40 (26.01)
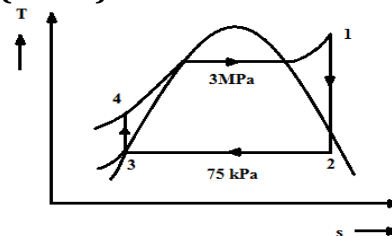
For process $1 \rightarrow 2$:-

$s_1 = s_2$

$6.7428 = 1.213 + x_2 (6.2434)$

$x_2 = 0.8857$

$\therefore h_2 = 384.39 + 0.8857[2278.6]$

$h_2 = 2408.54$kJ / kg

$W_T = h_1 - h_2$

$= 3115.3 - 2402.54$

$W_T = 712.76 \text{kJ} / \text{kg}$

$W_P = v_f \left[P_2 - P_1 \right]$

$= 0.001037 \left[3000 - 75 \right]$

$W_P = 3.033 \text{kJ} / \text{kg}$

$W_{net} = W_T - W_P = 709.727 \text{kJ} / \text{kg}$

$Q_3 = h_1 - h_4$

Also, $W_p = h_4 - h_2$

$\therefore \quad h_4 = W_p + h_2$

$= 3.033 + 384.3$

$h_4 = 387.423 \text{kJ} / \text{kg}$

$\therefore Q_s = 3115.3 - 387.423$

$= 2727.87 \text{ kJ} / \text{ kg}$

$\eta_{th} = \dfrac{W_{net}}{Q_s}$

$\therefore \eta_{th} = \dfrac{709.727}{2727.87} = 0.2601$

$\eta_{th} = 26.01\%$

Q.41 (0.8)

Efficiency   without   regeneration:-

$\eta_{th} = 1 - \dfrac{1}{\left(r_p \right)^{\gamma - 1 / \gamma}}$

$\eta_{th} = 1 - \dfrac{1}{\left(6 \right)^{0.4 / 1.4}} = 0.4006$

Efficiency with regeneration:-

$\left(\eta_{th} \right)_{reg} = 1 - \left(r_p \right)^{\frac{\gamma - 1}{\gamma}} \dfrac{T_{min}}{T_{max}}$

$= 1 - \left(6 \right)^{0.4 / 1.4} \times 0.3$

$= 0.4994 \approx 0.5$

$\text{Ratio} = \dfrac{\eta_{th}}{\left(\eta_{th} \right)_{reg}} = \dfrac{0.4}{0.5}$

$= 0.8$

Q.42 (464.14)

We   know   that   in   multi-stage compression

$\left(r_p \right)_0 = \left(r_p \right)_s^N$

$\left(r_p \right)_s = \left(rp \right)_0^{1/N}$

Where N-Number of stages

$\therefore \left(r_p \right)_s = \sqrt[3]{10}$

$\left(r_p \right)_s = 2.1544$

$\dfrac{p_2}{p_1} = 2.1544$

$p_2 = 2.1544 \times 100$

$= 215.44 \text{ kPa}$

Also $\quad \dfrac{p_s}{p_2} = 2.1544$

$p_3 = 2.1544 \times 215.44$

$\therefore p_3 = 2.1544 \times 215.44$

$p_3 = 464.14 \text{kPa}$

Q.43 (679.38)
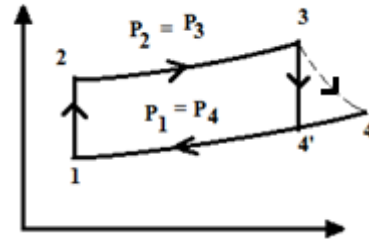
$\eta_{isen} = \dfrac{T_3 - T_4}{T_3 - T_4{}'}$

$Q \dfrac{P_2}{P_1} = \dfrac{P_3}{P_4} = 10$

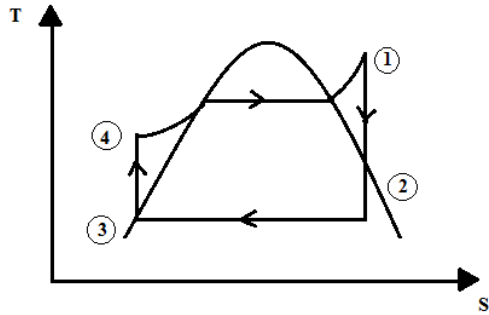$\dfrac{T_4{}'}{T_3} = \left(\dfrac{P_4}{P_3} \right)^{\frac{r-1}{r}}$

$T_4{}' = 1200 \left(\dfrac{1}{10} \right)^{\frac{1.4 - 1}{1.4}}$

$T_4{}' = 621.53 \text{k}$

$0.9 [1200 - 621.53] = T_3 - T_4$

$T_4 = 679.38 \text{k}$

Q.44 (3.6)

Given

$h_1 = 2803 \text{kJ} / \text{kg}$

$h_2 = 1800 \text{kJ} / \text{kg}$

$h_3 = 121 \text{kJ} / \text{kg}$

$h_4 = 124 \text{kJ} / \text{kg}$

$W_T = h_1 - h_2 = 2803 - 1800$

$W_T = 1003 \text{kJ} / \text{kg}$

$W_p = h_4 - h_3$

$= 124 - 121$

$W_p = 3 \text{kJ} / \text{kg}$

$W_{net} = W_T - W_p$

$= 1003 - 3$

$W_{net} = 1000 \text{kJ} / \text{kg}$

$Q \; SSC = \dfrac{3600}{W_{net}} \dfrac{\text{kg}}{\text{kwh}}$

$SSC = \dfrac{3600}{1000} = 3.6 \text{kg} / \text{kwh}$

Q.1 A single-acting two-stage compressor with complete inter cooling delivers air at 16 bar. Assuming an intake state of 1 bar at 15°C, the pressure ratio per stage is
a) 16   b) 8
c) 4   d) 2
[GATE–2001]

Q.2 In a spark ignition engine working on the ideal Otto cycle, the compression ratio is 5.5. The work output per cycle (i.e., area of the P-v diagram) is equal to $23.625 \times 10^5 \times V_c$ (in Joules), where V_c is the clearance volume in m^3. The indicated mean effective pressure is
a) 4.295 bar   b) 5.250 bar
c) 86.870 bar   d) 106.300 bar
[GATE–2001]

Q.3 An ideal air standard Otto cycle has a compression ratio of 8.5. If the ratio of the specific heats of air (γ) is 1.4, what is the thermal efficiency in percentage of the Otto cycle ?
a) 57.5   b) 45.7
c) 52.5   d) 95
[GATE–2002]

Q.4 For a spark ignition engine, the equivalence ratio ($\emptyset$) of mixture entering the combustion chamber has values
a) $\emptyset < 1$ for idling and $\emptyset > 1$ for peak power conditions
b) $\emptyset > 1$ for both idling and peak power conditions
c) $\emptyset > 1$ for idling and $\emptyset < 1$ for peak power conditions
d) $\emptyset < 1$ for both idling and peak power conditions
[GATE–2003]

Q.5 A diesel engine is usually more efficient than a spark ignition engine because
a) diesel being a heavier hydrocarbon, releases more heat per kg than gasoline
b) the air standard efficiency of diesel cycle is higher than the Otto cycle, at a fixed compression ratio
c) the compression ratio of a diesel engine is higher than that of an SI engine
d) self ignition temperature of diesel is higher than that of gasoline
[GATE–2003]

Q.6 An automobile engine operates at a fuel air ratio of 0.05, volumetric efficiency of 90% and indicated thermal efficiency of 30%. Given that the calorific value of the fuel is 45 MJ/kg and the density of air at intake is 1 kg/m^3, the indicated mean effective pressure for the engine is
a) 6.075 bar   b) 6.75 bar
c) 67.5 bar   d) 243 bar
[GATE–2003]

Q.7 For an engine operating on air standard Otto cycle, the clearance volume is 10% of the swept volume. The specific heat ratio of air is 1.4. The air standard cycle efficiency is
a) 38.3%   b) 39.8%
c) 60.2%   d) 61.7%
[GATE–2003]

Q.8 At the time of starting, idling and low speed operation, the carburetor supplies a mixture which can be termed as

a) Lean
b) slightly leaner than stoichiometric
c) stoichiometric
d) rich

Q.9 During a Morse test on a 4 cylinder engine, the following measurements of brake power were taken at constant speed.

All cylinders firing 3037 kW

Number 1 cylinder not firing 2102 kW

Number 2 cylinder not firing 2102 kW

Number 3 cylinder not firing 2100 kW

Number 4 cylinder not firing 2098 kW

The mechanical efficiency of the engine is

a) 91.53%   b) 85.07%
c) 81.07%   d) 61.22%

Q.10 An engine working on air standard Otto cycle has a cylinder diameter of 10 cm and stroke length of 15 cm. The ratio of specific heats for air is 1.4. If the clearance volume is 196.3 cc and the heat supplied per kg of air per cycle is 1800 kJ/kg, the work output per cycle per kg of air is

a) 879.1 kJ   b) 890.2 kJ
c) 895.3 kJ   d) 973.5 kJ

Q.11 The stroke and bore of a four stroke spark ignition engine are 250 mm and 200 mm respectively. The clearance volume is 0.001m³. If the specific heat ratio ⊠ = 1.4, the air-standard cycle efficiency of the engine is

a) 46.40%   b) 56.10%
c) 58.20%   d) 62.80%

Q.12 Which one of the following is NOT a necessary assumption for the air-standard Otto cycle?
a) All processes are both internally as well as externally reversible.
b) Intake and exhaust processes are constant volume heat rejection processes.
c) The combustion process is a constant volume heat addition process.
d) The working fluid is an ideal gas with constant specific heats

Q.13 In an air-standard Otto-cycle, the compression ratio is 10. The condition at the beginning of the compression process is 100 kPa and 27°C. Heat added at constant volume is 1500 kJ/kg, while 700 kJ/kg of heat is rejected during the other constant volume process in the cycle. Specific gas constant for air = 0.287 kJ/kg K. The mean effective pressure (in kPa) of the cycle is

a) 103   b) 310
c) 515   d) 1032

Q.14 A turbo-charged four-stroke direct injection diesel engine has a displacement volume of 0.0259 m³ (25.9 litres). The engine has an output of 950 kW at 2200 rpm. The mean effective pressure (in MPa) is closest to

a) 2   b) 1
c) 0.2   d) 0.1

Q.15 The crank radius of a single-cylinder I.C. engine is 60 mm and the diameter of the cylinder is 80 mm. The swept volume of the cylinder in cm³ is

a) 48   b) 96
c) 302   d) 603

Q.16 In an air-standard Otto cycle, air is supplied at 0.1 MPa and 308 K. The ratio of the specific heats (γ) and the specific gas constant (R) of air are 1.4 and 288.8 J/kgK respectively. If the compression ratio is 8 and the maximum temperature in the cycle is 2660 K, the heat (in kJ/kg) supplied to the engine is _____

[GATE-2014 (1)]

Q.17 A diesel engine has a compression ratio of 17 and cut-off takes place at 10% of the stroke. Assuming ratio of specific heats (γ) as 1.4, the air–standard efficiency (in percent) is _____.

[GATE-2014 (3)]

Q.18 In a compression ignition engine, the inlet air pressure is 1 bar and the pressure at the end of isentropic compression is 32.42 bars. The expansion ratio is 8. Assuming ratio of specific heats (γ) as 1.4, the air standard efficiency (in percent) is _____.

[GATE-2014 Set-4]

Q.19 Air enters a diesel engine with a density of 1.0 kg/m^3. The compression ratio is 21. At steady state, the air intake is 30×10^{-3}kg/s and the net work output is 15 kW. The mean effective pressure (kPa) is _____.

[GATE-2015 (1)]

Q.20 An air-standard Diesel cycle consists of the following processes:
1-2 : Air is compressed isentropically.
2-3 :Heat is added at constant pressure.
3-4 :Air expands isentropically to the original volume.
4-1:Heat is rejected at constant volume.

If γ and T denote the specific heat ratio and temperature, respectively, the efficiency of the cycle is

a) $1-\dfrac{T_4-T_1}{T_3-T_2}$ 　 b) $1-\dfrac{T_4-T_1}{\gamma[T_3-T_2]}$

c) $1-\dfrac{\gamma[T_4-T_1]}{T_3-T_2}$ 　 d) $1-\dfrac{(T_4-T_1)}{(\gamma-1)(T_3-T_2)}$

[GATE-2015 (3)]

Q.21 For the same values of peak pressure, peak temperature and heat rejection, the correct order of efficiency for Otto, Dual and Diesel cycles is
a) $\eta_{Otto} > \eta_{Dual} > \eta_{Diesel}$
b) $\eta_{Diesel} > \eta_{Dual} > \eta_{Otto}$
c) $\eta_{Dual} > \eta_{Diesel} > \eta_{Otto}$
d) $\eta_{Diesel} > \eta_{Otto} > \eta_{Dual}$

[GATE-2015 (2)]

Q22 Air contains 79% N_2 and 21% O_2 on a molar basis. Methane (CH_4) is burned with 50% excess air than required stoichiometrically. Assuming complete combustion of methane, the molar percentage of N_2 in the products is _____

[GATE-2017 Set-1]

Q.23 An engine working on air standard Otto cycle is supplied with air at 0.1 MPa and $35^0 C$. The compression ratio is 8. The heat supplied is 500 kJ/kg. Property data for air: c_p = 1.005 kJ/kg K, c_v = 0.718 kJ/kg K, R = 0.287 kJ/kg K. The maximum temperature (in K) of the cycle is _____ (correct to one decimal place).

[GATE-2018 Set-1]

GATEFLIX

Q.1 **(c)**

Given that the intercooling is perfect

$\therefore P_2 = \sqrt{P_1 P_3}$

$P_2 = \sqrt{1 \times 16}$

$P_2 = 4$ bar

Pressure ratio per stage $= \dfrac{P_2}{P_1} = \dfrac{P_3}{P_2} = 4$

Q.2 **(b)**

mep $= \dfrac{W}{V_s} = \dfrac{W}{V_1 - V_C}$

$= \dfrac{23.625 \times 10^5 \times V_C}{5.5 V_C - V_C}$

$= \dfrac{23.625 \times 10^5 V_C}{4.5 V_C}$

$= 5.25 \times 10^5 \, Pa$

$= 5.250$ bar

Q.3 **(a)**

$\eta = 1 - \left(\dfrac{1}{r} \right)^{\gamma - 1}$

$= 1 - \left(\dfrac{1}{8.5} \right)^{0.4}$

$= 0.5751$ or 57.51%

Q.4 **(b)**

Equivalence ratio is defined as the actual fuel air ratio to stoichiometric fuel air ratio

$\phi = \dfrac{\left(\dfrac{F}{A} \right)_{actual}}{\left(\dfrac{F}{A} \right)_{Stoi}}$

For both idling and peak power conditions the actual fuel-air ratio required is more than the stoichiometric fuel-air ratio.

Q.5 **(c)**

For same compression ratio and the same heat supplied, Otto cycle is most efficient and diesel cycle is least efficient.

In practice, however, the compression ratio of the Diesel engine ranges between 14 and 25 whereas that of the Otto engine between 6 and 12. Because of it the efficiency of Diesel cycle is higher efficiency than that of Otto engine.

Q.6 **(a)**

Volumetric efficiency $= \dfrac{\text{actual volume}}{\text{swept volume}}$

$= \dfrac{V_a}{V_s} = 0.9$

$\therefore V_a = 0.9 V_s$

Mass of air,

$m_a = \rho_{air} V_s = 0.9 V_s$

$m_f = 0.05 \times 0.9 V_s = 0.045 V_s$

$\eta_{thermal} = \dfrac{P_{mep} \times LAN}{m_f \times C.V}$

$\Rightarrow 0.3 = \dfrac{P_{mep} \times V_s}{0.045 v_s \times 45 \times 10^6}$

$\therefore P_{mep} = 6.075$ bar

Q.7 **(d)**

$\eta_{Otto} = 1 - \left(\dfrac{v_c}{v_1} \right)^{\gamma - 1}$

$= 1 - \left(\dfrac{V_c}{V_c + V_s} \right)^{\gamma - 1}$

Where V_c = clearance vol

$= 10\%$ of $V_s = 0.1 V_s$

$\therefore \quad \eta_{Otto} = 1 - \left(\dfrac{0.1 V_s}{0.1 V_s + V_s} \right)^{\gamma - 1}$

$= 1 - \left(\dfrac{0.1}{1.1} \right)^{1.4 - 1}$

$= 0.6167 = 61.67\%$

Q.8 **(d)**

Q.9 **(c)**

When cylinder 1 is not firing then power is 2102 kW and when all cylinders are firing then power is 3037 kW.

Hence power supplied by cylinder No.1 = 3037 − 2102 = 935 kW

Similarly power supplied by cylinder No. 2 = 3037 − 2102 = 935 kW

Similarly power supplied by cylinder No. 3 = 3037 − 2100 = 937 kW

Similarly power supplied by cylinder No.4 = 3037 − 2098 = 939 kW

$IP_{total} = 935 + 935 + 937 + 939 = 3746$

Hence $\eta_{mech} = \dfrac{3037}{935 + 935 + 937 + 939}$

$\eta_{mech} = \dfrac{BP}{IP}$

$= \dfrac{3037}{3746} = 0.8107$

$\therefore \quad \eta_{mech} = 81.07\%$

Q.10 **(d)**

$\eta_{Otto} = 1 - \left(\dfrac{V_c}{V_c + V_s}\right)^{r-1}$

$V_C = 196.3 cc$

$V_s = \dfrac{\pi}{4} \times D^2 L = \dfrac{\pi}{4} \times 10^2 \times 15$

$= 1178.097 cc$

$\gamma = 1.4$

$\therefore \quad \eta_{Otto} = 1 - \left(\dfrac{196.3}{196.3 + 1178.097}\right)^{1.4-1}$

$= 0.5408$

$\therefore \quad \eta_{Otto} = 54.08\%$

$\therefore$ Work output

$= \eta_{Otto} \times$ (Heat supplied)

$= 0.5408 \times 1800 = 973.58 kJ$

Q.11 **(c)**

Given L = 250 mm = 0.25 m, D = 220 mm = 0.2 m.

$v_c = 0.001 m^3, \gamma = \dfrac{C_p}{C_v} = 1.4$

Swept Volume

$V_s = A \times L = \dfrac{\pi}{4}(D)^2 \times L$

$= \dfrac{\pi}{4}(0.2)^2 \times 0.25 = 0.00785 m^3$

Compression ratio

$r = \dfrac{v_r}{v_c} = \dfrac{v_c + v_s}{v_c}$
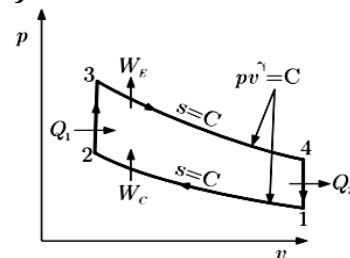
$= \dfrac{0.001 + 0.00785}{0.001}$

$= 8.85$

Air standard efficiency

$\eta = 1 - \dfrac{1}{(r)^{\gamma-1}} = 1 - \dfrac{1}{(8.85)^{1.4-1}}$

$= 1 - \dfrac{1}{2.39} = 1 - 0.418 = 0.582$

or $\quad 58.2\%$

Q.12 **(b)**

Assumptions of air standard otto cycle

a) All processes are internally reversible.

b) Air behaves as ideal gas

c) Specific heats remains constant (C_p & C_v)

d) Intake process is constant volume heat addition process and exhaust pro-cess is constant volume heat rejection process. Intake process is a constant volume heat addition process, from the given options; option (2) is incorrect.

Q.13 **(d)**
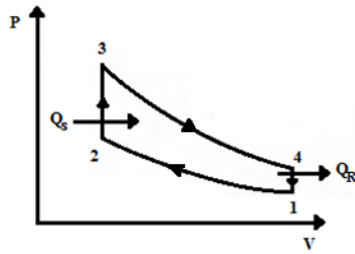
$P_1 = 100$ kPa
Compression ratio,
r = 10
$T_1 = 27 + 273 = 300K$
Heat added,
$Q_S = 1500$kJ/kg
Heat rejected,
$Q_R = 700$kJ/kg
Mean effective pressure

$$= \frac{\text{Work done per cycle}}{\text{Swept volume}}$$

Compression ratio,
$r = V_1/V_2 = 10$
$V_1 = 10V_2$
Swept volume
$= V_1 - V_2$
$= V_1 - \dfrac{V_1}{10} = 0.9V_1$
For initial air
$P_1 V_1 = RT_1$
$\Rightarrow V_1 = \dfrac{RT_1}{P_1}$

$$= \frac{0.287\text{kJ}/\text{kgK} \times 300\text{K}}{100\text{kPa}}$$

$= 0.861 \text{m}^3/\text{kg}$
Swept volume $= 0.9\,V_1$
$= 0.9(0.861)$
$= 0.7749 \text{m}^3/\text{kg}$
Work done in cycle,
$W_{\text{net out}} = Q_{\text{supply}} - Q_{\text{rej}}$
$= 1500 - 700 = 800$ kJ/kg
Mean effective pressure

$$= \frac{W_{\text{net}}}{\text{Swept volume}}$$

$$P_{\text{mep.}} = \frac{800}{0.7749}$$

$P_{\text{mep.}} = 1032.39$ kPa

Q.14 (a)

$$\text{Power} = \frac{P_m L A n}{60,000} \text{kw}$$

As four stroke
$$n = \frac{N}{2} = 1100\text{rpm}$$

$$P_m = \frac{950 \times 60000}{0.0259 \times 1100}$$

$= 2 \times 10^6 \text{N}/\text{m}^2 = 2$ MPa

Q.15 (d)
Given : r = 60 mm, D = 80 mm= 8 cm
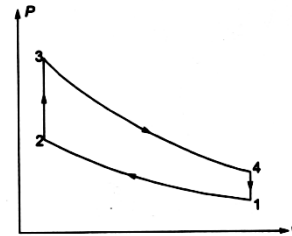Stroke length,
L = 2r = 2 × 60 = 120 mm =12 cm
Swept Volume, $V_s = A \times L$

$$= \frac{\pi}{4}D^2 \times L = \frac{\pi}{4}(8)^2 \times 12$$

$= 603.18 \simeq 603 \text{ cm}^3$

Q.16 (1400 to 1420)

Given,
$P_1 = 0.1$ MPa
$T_1 = 308$ K
$\gamma = 1.4$
R = 288.8 J/kg K
r = 8
$T_3 = 2660$ K

$$C_v = \frac{R}{\gamma - 1} = \frac{288.8}{0.4}$$

= 722 J/kgK

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{\gamma-1} = (8)^{0.4}$$

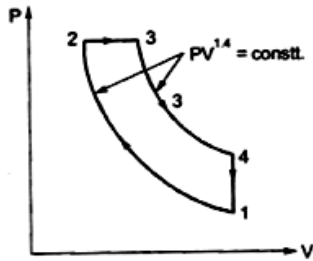$T_2 = 308 \times 8^{0.4} = 707.6$K
$Q_s = C_v\,\Delta T$
$= 722(2660 - 707.6)$
$= 1409.6$ kJ/kg

Q.17 (58 to 62)

$$\frac{V_1}{V_2} = 17$$

$$\frac{C_p}{C_v} = \gamma = 1.4$$

or $\quad V_3 - V_2 = 0.1(V_1 - V_2)$

or $\quad \dfrac{V_3}{V_2} - 1 = 0.1\left(\dfrac{V_1}{V_2} - 1\right)$
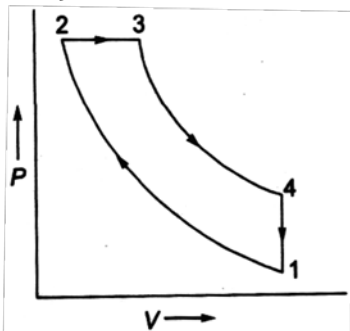
or $\quad \dfrac{V_3}{V_2} = 0.1 \times 16 + 1 = 2.6$

$$\eta_{Diesel} = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{r_c^\gamma - 1}{\gamma(r_c - 1)}\right]$$

$$= 1 - \frac{1}{17^{0.4}}\left[\frac{2.6^{1.4} - 1}{1.4(2.6 - 1)}\right]$$

$$= 1 - \frac{1}{17^{0.4}}\left(\frac{3.81 - 1}{1.4 \times 1.6}\right)$$

$$= 0.596 \text{ or } 59.6\%$$

Q.18 (59 to 61)

Given

P_1= 1 bar, P_2=32.42bar

$\gamma = \dfrac{C_p}{C_v} = 1.4$

$\dfrac{V_4}{V_3} = \dfrac{V_1}{V_3} = 8$

for process $1 \rightarrow 2$,

$P_1 V_1^\gamma = P_2 V_2^\gamma$

$$\therefore \left(\frac{V_1}{V_2}\right)^\gamma = \frac{P_2}{P_1} = 32.42$$

Or $\quad \dfrac{V_1}{V_2} = (32.42)^{1/1.4} = 11.999 \approx 12$

$$r_c = \frac{V_3}{V_2} = \frac{V_1}{8} \times \frac{12}{V_1} = 1.5$$

$$\eta_d = 1 - \frac{1}{r^{\gamma-1}}\left[\frac{r_c^\gamma - 1}{\gamma(r_c - 1)}\right]$$

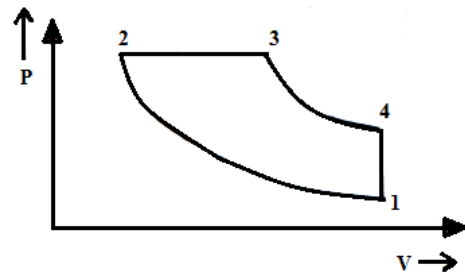$$= 1 - \frac{1}{12^{0.4}}\left[\frac{1.5^{1.4} - 1}{1.4 \times 0.5}\right]$$

$$= 0.596 \text{ or } 59.6\%$$

Q.19 (525)

Q.20 (b)

Heat applied, $Q_s = C_p(T_3 - T_2)$
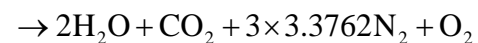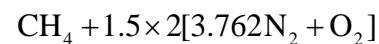
Heat rejected, $Q_r = C_v(T_4 - T_1)$

$$\eta = 1 - \frac{Q_r}{Q_s} = 1 - \frac{1}{\gamma}\frac{(T_4 - T_1)}{(T_3 - T_2)}$$

Q.21 (b)

Q.22 (73.83)

100 Ltr air contains 79 ltr of N_2 and 21 ltr of O_2

The combustion equation with 50% excess air is

$$CH_4 + 1.5 \times 2[3.762N_2 + O_2]$$

$$\rightarrow 2H_2O + CO_2 + 3 \times 3.3762N_2 + O_2$$

It is assumed that nitrogen is insert and does not participate in the reaction. Whatever nitrogen is there in reactants, same will come out in the products.

On the basis of Molar or volume

$$N_2 = \frac{3 \times 3.762}{2 + 1 + 3 \times 3.762 + 1} \times 100$$

$$N_2 = 73.83\%$$

$$0.718\left(T_3 - 707.6\right) = 500$$

$$T_3 = 1403.97 \, K$$

Q.23 (1403.9)

$$P_1 = 0.1 \, MPa, \ T_1 = 35^\circ C = 308 \, K$$

$$\frac{V_1}{V_2} = n = 8$$
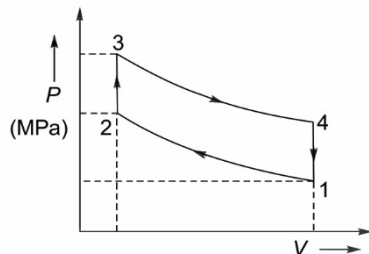
$$Q_s = 500 \, kJ/kg$$

$$c_p = 1.005 \, kJ/kgK$$

$$c_v = 0.718 \, kJ/kgK$$

$$R = 0.287 \, kJ/kgK$$

$$\gamma = \frac{c_p}{c_v} = 1.399 \simeq 1.40$$

$$T_3 = T_{max} = \ ?$$

For process $1-2$

For process $1-2$

$$P_1 V_1^\gamma = P_2 V_2^\gamma$$

$$P_2 = P_1 \left(\frac{V_1}{V_2}\right)^\gamma = 0.1 \times (8)^{1.4}$$

$$P_2 = 1.8379 \, MPa$$

and $\quad \dfrac{P_1 V_1}{T_1} = \dfrac{P_2 V_2}{T_2} \Rightarrow T_2 = \dfrac{P_2 V_2}{P_1 V_1} T_1$

$$T_2 = \frac{1.8379 \times 1}{0.1 \times 8} \times 308$$

$$T_2 = 707.6 \, K$$

For process $2 \rightarrow 3$

$$Q_s = C_v \left(T_3 - T_2\right) = 500 \, kJ/kg$$

Q.1 An industrial heat pump operates between the temperatures of 27^0C and -13^0C.
The rates of heat addition and heat rejection are 750 W and 1000 W, respectively.
The COP for the heat pump is
a) 7.5   b) 6.5
c) 4.0   d) 3.0

[GATE-03]

Q.2 For air with a relative humidity of 80%
a) the dry bulb temperature is less than the wet bulb temperature
b) the dew point temperature is less than wet bulb temperature
c) the dew point and wet bulb temperature are equal
d) the dry bulb and dew point temperature are equal

Common Data For Q.3 and Q.4
A refrigerator based on ideal vapour compression cycle operates between the temperature limits of -20^0C and 40^0C. The refrigerant enters the condenser as saturated vapour and leaves as saturated liquid. The enthalpy and entropy
values for saturated liquid and vapour at these temperatures are given in the table below.

T($^\circ$C)	h_f(kJ/kg)	h_g(kJ/kg)	s_f(kJ/kg K)	s_g(kJ/kg K)
-20	20	180	0.07	0.7366
40	80	200	0.3	0.67

Q.3 If refrigerant circulation rate is 0.025 kg/s, the refrigeration effect is equal to
a) 2.1 kW   b) 2.5 kW
c) 3.0 kW   d) 4.0 kW

[GATE-03]

Q.4 The COP of the refrigerator is
a) 2.0   b) 2.33
c) 5.0   d) 6.0

[GATE-03]

Q.5 In the window air conditioner, the expansion device used is
a) capillary tube
b) thermostatic expansion valve
c) automatic expansion valve
d) float valve

[GATE-04]

Q.6 During the chemical dehumidification process of air
a) dry bulb temperature and specific humidity decreases
b) dry bulb temperature increases and specific humidity decreases
c) dry bulb temperature decreases and specific humidity increases
d) dry bulb temperature and specific humidity increases

[GATE-04]

Q.7 Environment friendly refrigerant R134 is used in the new generation domestic
refrigerators. Its chemical formula is
a) $CHClF_2$ b) $C_2Cl_3F_3$
c) $C_2Cl_2F_4$ d) $C_2H_2F_4$

[GATE-04]

Q.8 A heat engine having an efficiency of 70% is used to drive a refrigerator having
a coefficient of performance of 5. The energy absorbed from low temperature reservoir by the refrigerator for each kJ of energy absorbed from high temperature
source by the engine is
a) 0.14 kJ   b) 0.71 kJ

c) 3.5 kJ   d) 7.1 kJ

Q.9 Dew point temperature of air at one atmospheric pressure (1.013 bar) is 18^0C.

The air dry bulb temperature is 30^0C. The saturation pressure of water at 18^0C

and 30^0C are 0.02062 bar and 0.04241 bar respectively. The specific heat of air and water vapour respectively are 1.005 and 1.88 kJ/kg K and the latent heat of vaporization of water at 0^0C is 2500 kJ/kg. The specific humidity (kg/kg of dry

air) and enthalpy (kJ/kg or dry air) of this moist air respectively, are

a) 0.01051, 52.64   b) 0.01291, 63.15

c) 0.01481, 78.60   d) 0.01532, 81.40

Q.10 A R-12 refrigerant reciprocating compressor operates between the condensing

temperature of 30^0C and evaporator

temperature of -20^0C. The clearance volume ratio of the compressor is 0.03. Specific heat ratio of the vapour is 1.15 and the specific volume at the suction is 0.1089 m3/kg. Other properties at various states are given in the figure. To realize 2 tons of refrigeration, the actual volume displacement rate considering the effect of clearance is
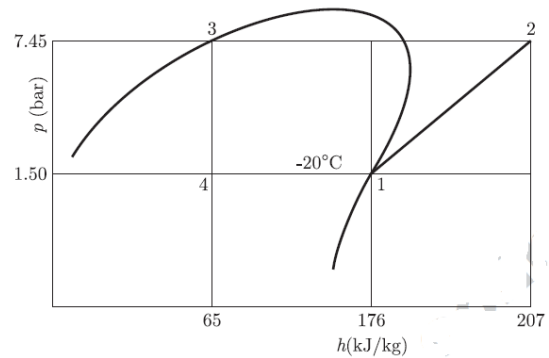
a) $6.35 \times 10^{-3} m^3/s$ b) $63.5 \times 10^{-3} m^3/s$
c) $635 \times 10^{-3} m^3/s$ d) $4.88 \times 10^{-3} m^3/s$

Q.11 For a typical sample of ambient air (at 35^0C, 75% relative humidity and standard

atmosphere pressure), the amount of moisture in kg per kg of dry air will be approximately

a) 0.002   b) 0.027
c) 0.25   d) 0.75

Q.12 Water at 42^0C is sprayed into a stream of air at atmospheric pressure, dry bulb

temperature of 40^0C and a wet bulb

temperature of 20^0C. The air leaving the

spray humidifier is not saturated. Which of the following statements is true ?
a) Air gets cooled and humidified
b) Air gets heated and humidified
c) Air gets heated and dehumidified
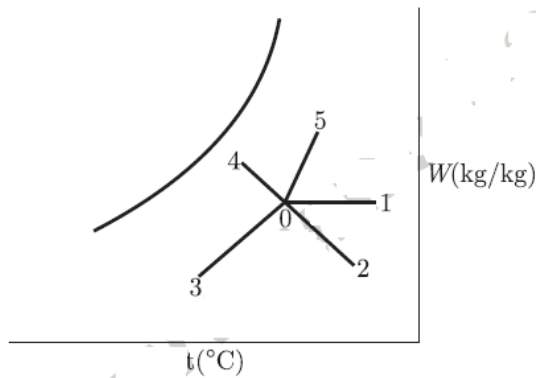d) Air gets cooled and dehumidified

Q.13 The vapour compression refrigeration cycle is represented as shown in the figure

below, with state 1 being the exit of the evaporator. The coordinate system used in this figure is
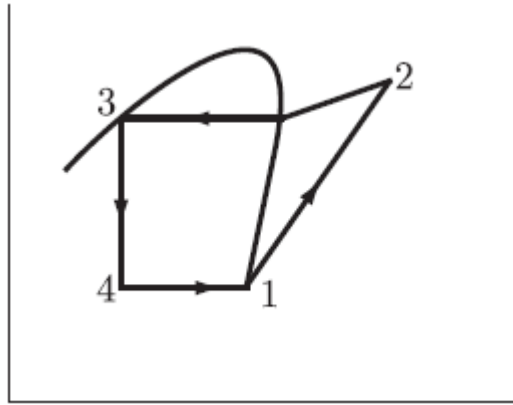
a) *p-h* b) *T -s*
c) *p-s* d) *T –h*

Q.14 Various psychometric processes are shown in the figure below

	Process in Figure		Name of the process
P.	0 - 1	(i).	Chemical dehumidification
Q.	0 - 2	(ii).	Sensible heating
R.	0 - 3	(iii).	Cooling and dehumidification
S.	0 - 4	(iv).	Humidification with steam injection
T.	0 - 5	(v).	Humidification with water injection

The matching pairs are
a) P-(i), Q-(ii), R-(iii), S-(iv), T-(v)
b) P-(ii), Q-(i), R-(iii), S-(v), T-(iv)
c) P-(ii), Q-(i), R-(iii), S-(iv), T-(v)
d) P-(iii), Q-(iv), R-(v), S-(i), T-(ii)

Q.15 A vapour absorption refrigeration system is a heat pump with three thermal
reservoirs as shown in the figure. A refrigeration effect of 100 W is required at
250 K when the heat source available is at 400 K. Heat rejection occurs at 300 K.

The minimum value of heat required (in W) is
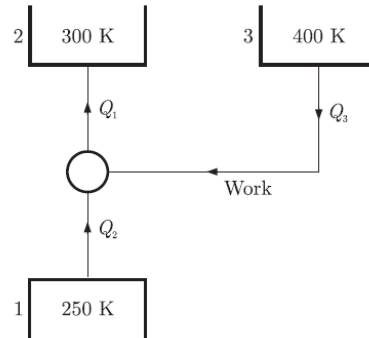
a) 167   b) 100
c) 80   d) 20

Q.16 Dew point temperature is the temperature at which condensation begins when
the air is cooled at constant
a) volume   b) entropy
c) pressure d) enthalpy

Q.17 The statements concern psychometric chart.
1. Constant relative humidity lines are uphill straight lines to the right
2. Constant wet bulb temperature lines are downhill straight lines to the right
3. Constant specific volume lines are downhill straight lines to the right
4. Constant enthalpy lines are coincident with constant wet bulb temperature
Lines Which of the statements are correct ?
a) 2 and 3   b) 1 and 2
c) 1 and 3   d) 2 and 4

Q.18 A building has to be maintained at 21^0C (dry bulb) and 14.5^0C (wet bulb). The dew point temperature under these conditions is 10.17^0C. The outside

temperature is -23^0C (dry bulb) and the internal and external surface heat transfer coefficients are 8 W/m2 K and 23 W/m2 K respectively. If the building wall has a thermal conductivity of 1.2 W/m K, the minimum thickness (in m) of
the wall required to prevent condensation is
a) 0.471   b) 0.407
c) 0.321   d) 0.125

[GATE-07]

Q .19 Atmospheric air at a flow rate of 3 kg/s (on dry basis) enters a cooling and dehumidifying coil with an enthalpy of 85 kJ/ kg of dry air and a humidity ratio of 19 grams/kg of dry air. The air leaves the coil with an enthalpy of 43 kJ/kg of dry air and a humidity ratio of 8 grams/kg of dry air. If the condensate water
leaves the coil with an enthalpy of 67 kJ/kg, the required cooling capacity of the
coil in kW is
a) 75.0   b) 123.8
c) 128.2   d) 159.0

[GATE-07]

Q.20 Moist air at a pressure of 100 kPa is compressed to 500 kPa and then cooled to 35^0C in an aftercooler. The air at the entry to the aftercooler is unsaturated and becomes just saturated at the exit of the aftercooler. The saturation pressure of
water at 35^0C is 5.628 kPa. The partial pressure of water vapour (in kPa) in the moist air entering the compressor is closest to
a) 0.57   b) 1.13
c) 2.26   d) 4.52

[GATE-08]

Q.21 Air (at atmospheric pressure) at a dry bulb temperature of 40^0C and wet

bulb temperature of 20^0C is humidified in an air washer operating with continuous water recirculation. The wet bulb depression (i.e. the difference between the dry and wet bulb temperature) at the exit is 25% of that at the inlet. The dry bulb temperature at the exit of the air washer is closest to
a) 10^0C b) 20^0C
c) 25^0C d) 30^0C

[GATE-08]

Q.22 In an ideal vapour compression refrigeration cycle, the specific enthalpy of refrigerant (in kJ/kg) at the following states is given as:
Inlet of condenser :283
Exit of condenser :116
Exit of evaporator :232
The COP of this cycle is
a) 2.27   b) 2.75
c) 3.27   d)3.75

[GATE-09]

Q.23 A moist air sample has dry bulb temperature of 30^0C and specific humidity of
11.5 g water vapour per kg dry air. Assume molecular weight of air as 28.93. If the
saturation vapour pressure of water at 30^0C is 4.24 kPa and the total pressure is
90 kPa, then the relative humidity (in %) of air sample is
a) 50.5   b) 38.5
c) 56.5   d) 68.5

[GATE-10]

Q.24 If a mass of moist air in an airtight vessel is heated to a higher temperature, then
a) specific humidity of the air increases
b) specific humidity of the air decreases
c) relative humidity of the air increases
d) relative humidity of the air decreases

[GATE-11]

Q.25 The rate at which heat is extracted, in kJ/s from the refrigerated space is
a) 28.3 b) 42.9
c) 34.4   d) 14.6

[GATE-12]

Q.26 The power required for the compressor in kW is
a) 5.94   b) 1.83
c) 7.9   d) 39.5

[GATE-12]

Q.27 The pressure, dry bulb temperature and relative humidity of air in a room are
1 bar, 30^0C and 70%, respectively. If the saturated pressure at 30^0C is 4.25 kPa, the specify humidity of the room air in kg water vapour/kg dry air is
a) 0.0083   b) 0.0101
c) 0.0191   d) 0.0232

[GATE-13]

Q.28 which one of the following is a CFC refrigerant ?

(a) R744   (b) R290

(C) R502   (d) R718

[GATE-2014 (1)]

Q.29 A sample of moist air at a total pressure of 85 kPa has a dry bulb temperature of 30^0C (saturation vapour pressure of water = 4.24 kPa). If the air sample has a relative humidity of 65%, the absolute humidity (in gram) to water vapour per kg of dry air is_____.

[GATE-2014 (3)]

Q.30 A heat pump with refrigerant R22 is used for space heating between temperature limits of $-20°$C and 25°C. The heat required is 200 MJ/h. Assume specific heat of vapour at the time of discharge as 0.98 kJ/kg-K. Other

relevant properties are given below. The enthalpy (in kJ/kg) of the refrigerant at isentropic compressor discharge is
_____

Saturation temperature	Pressure	Specific enthalpy		Specific entropy	
$T_{sat}(°C)$	$P(MN/m^2)$	$h_f(kJ/kg)$	$h_g(kJ/kg)$	$s_f(kJ/kgK)$	$s_g(kJ/kgK)$
-20	0.2448	177.21	397.53	0.9139	1.7841
25	1.048	230.27	413.02	1.1047	1.7183

[GATE-2014 (2)]

Q. 31 A reversed Carnot cycle refrigerator maintains a temperature of $-5°$C. The ambient air temperature is 35°C. The heat gained by the refrigerator at a continuous rate is 2.5 kJ/s. The power (in watt) required to pump this heat out continuously is ___ .

[GATE-2014 (4)]

Q.32 Air in a room is at 35°C and 60% relative humidity (RH). The pressure in the room is 0.1 MPa. The saturation pressure of water at 35°C is 5.63 kPa.
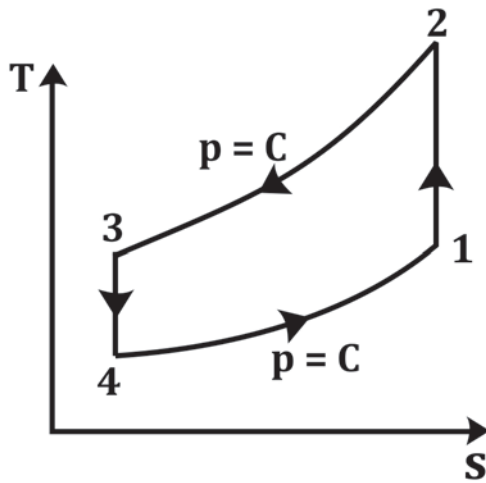
The humidity ratio of the air (in gram/kg of dry air)is_____.

[GATE-2015(3)]

Q.33 Refrigerant vapor enters into the Compressor of a standard vapor compression cycle at - 10ºC (h = 402 kJ/kg) and leaves the compression at 50°C (h = 432 kJ/kg). It leaves the condenser at 30°C (h = 237 kJ/kg). The COP of the Cycle is _____

[GATE-2015(3)]

Q.34 The thermodynamic cycle shown figure *(T- diagram)* indicates

(a) reversed Carnot cycle

(b) reversed Brayton cycle

(c) vapor compression cycle

(d) vapor absorption cycle

[GATE-2015(3)]

Q.35 The COP of a Carnot heat pump operating between 6°C and 37°C is _____

[GATE-2015(2)]

Q.36 A stream of moist air (mass flow rate 10.1 kg/s) with humidity ratio of 0.01 kg/kg dry air mixes with a second stream of superheated water vapour flowing at 0.1 kg/s. Assuming proper and Uniform mixing with no condensation, the humidity ratio of the final stream in (kg/kg dry air) is _____.

[GATE-2015(1)]

Q.37 In a mixture of dry air and water vapor at a total pressure of 750 mm of Hg, the partial pressure of water vapor is 20 mm of Hg. The humidity ratio of the air in grams of water vapor per kg of dry air (g_w/kg_{da}) is _____.

[GATE-2016 (3)]

Q.38 A refrigerator uses R-134 a as its refrigerant and operates on a ideal vapour-compression refrigeration cycle between 0.14 MPa and 0.8 MPa. If the mass flow rate of the refrigerant is 0.05 kg/s the rate of heat rejection to the environment is ___kW.

Given data :

At P = 0.14 MPa, h 236.04 kJ/kg,

s = 0.9322 kJ/kgK

At P = 0.8 MPa, h = 272.05 kJ/kg (superheated vapour)

At P = 0.8 MPa, h = 93.42 kJ/kg(saturated liquid)

[GATE-2016 (2)]

Q.39 In the vapour compression cycle shown in the figure, the evaporating and condensing temperatures are 260 K and 310K, respectively. The compressor takes in liquid-vapour mixture (state 1) and isentropically compresses it to a dry saturated vapour condition (state 2). The specific heat of the liquid refrigerant is 4.8 kJ/kgK and may be treated as constant. The enthalpy of evaporation for the refrigerant at 310 K is 1054 kJ/kg.
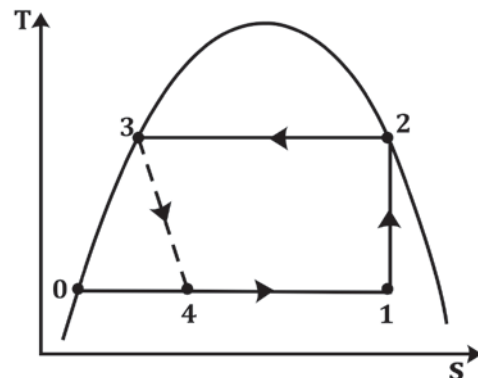
The difference between the enthalpies at state point 1 and 0 (in kJ/kg) is_____.

[GATE-2016 (3)]

Q.40 The partial pressure of water vapour in a moist air sample of relative humidity

70% is 1.6 kPa, the total pressure being 101.325 kPa. Moist air may be treated as an ideal gas mixture of water vapour and dry air. The relationbetween saturation temperature (T_s in K) and saturation pressure (P_s in kPa) for water is given

by $\ln\left(\dfrac{p_s}{p_0}\right) = \dfrac{14.317 - 5304}{T_s}$

where p_0 = 101.325 kPa. The dry bulb temperature of the moist air sample (in $^{\circ}$C) is_____

[GATE-2016 (2)]

Q.41 Moist air is treated as an ideal gas mixture of water vapor and dry air (molecular weight of air = 28.84 and molecular weight of water = 18). At a location, the total pressure is 100 kPa, the temperature is 30°C and the relative humidity is 55%. Given that the saturation pressure of water at 30°C is 4246 Pa, the mass of water vapor per kg of dry air is _____ grams.

[GATE-2017 (1)]

Q.42 If a mass of moist air contained in a closed metallic vessel is heated, then its

a) relative humidity decreases

b) relative humidity increases

c) specific humidity increases

d) specific humidity decreases

[GATE-2017 (2)]

Q.43 Ambient air is at a pressure of 100 kPa, dry bulb temperature of 30 $^{\circ}$C and 60% relative humidity. The saturation pressure of water at 30 $^{\circ}$C is 4.24 kPa. The specific humidity of air (in g/kg of dry air) is _____ (correct to two decimal places).

[GATE-2018 (1)]

Q.44 A standard vapor compression refrigeration cycle operating with a condensing temperature of 35 $^{\circ}$C and an evaporating temperature of -10 $^{\circ}$C develops 15 kW of cooling.

The *p-h* diagram shows the enthalpies at various states. If the isentropic efficiency of the compressor is 0.75, the magnitude of compressor power (in kW) is _____ (correct to two decimal places).
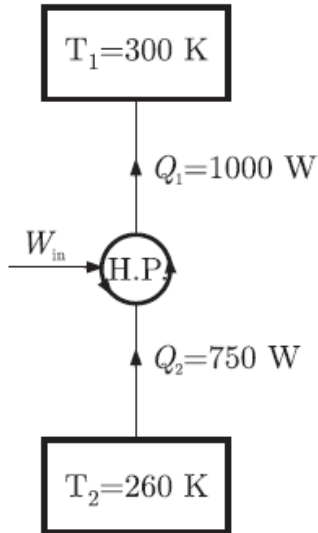
[GATE-2018 (2)]

ANSWER KEY:

1	2	3	4	5	6	7	8	9	10	11	12	13
(c)	(b)	(a)	(b)	(a)	(b)	(d)	(c)	(b)	(a)	(b)	(b)	(a)
14	15	16	17	18	19	20	21	22	23	24	25	26
(b)	(c)	(c)	(a)	(b)	(c)	(b)	(c)	(a)	(b)	(d)	(a)	(c)
27	28	29	30	31	32	33	34	35	36	37	38	39
(c)	(b)	20.84	(b)	433.3	373.13	21.74	5.5	(b)	10	0.02	17.04	8.93
27	28	29	30	31	32							
1103.51	19.89	14.9	(a)	16.236	10							

EXPLANATIONS

Q.1(c)

So, $(COP)_{H.P.} = \dfrac{Q_1}{Q_1 - Q_2} = \dfrac{1000}{1000 - 750} = 4$

Q.2 (b) We know that for saturated air, the relative humidity is 100% and the dry bulb temperature, wet bulb temperature and dew point temperature is same. But when air is not saturated, dew point temperature is always less than the wet bulb temperature.
DPT < WBT

Q.3(a)

(B) p-h Diagram

(A) T-S Diagram
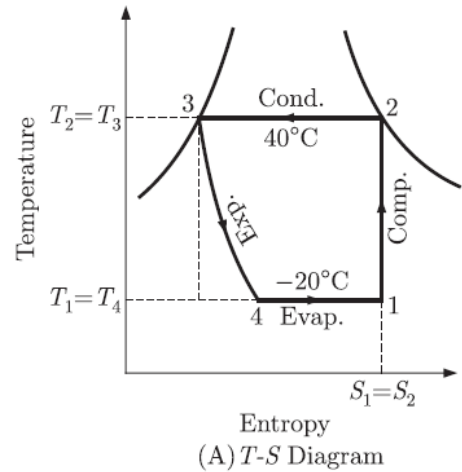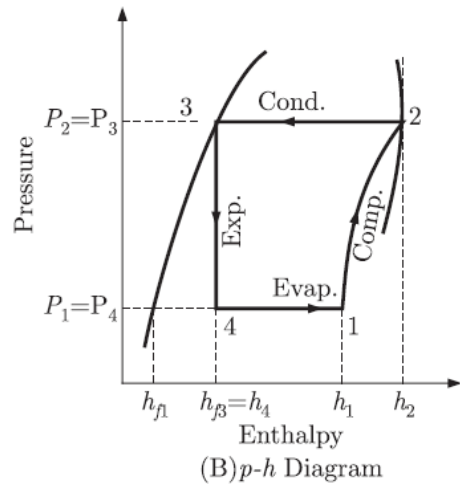
Given : $T_1 = T_4 = -20^0C = (-20 + 273)K = 253\ K$,

$\dot{m} = 0.025\ kg/sec$

$T_2 = T_3 = 40^0C = (40 + 273)K = 313\ K$

From the given table,

At, $T_2 = 40^0C$, $h_2 = 200\ kJ/kg$

And $h_3 = h_4 = 80\ kJ/kg$

From the given T s - curve

$s_1 = s_2$

$s_2 = s_f + xs_{fg}$

{s_2 is taken 0.67 because s_2 at the temperature 40^0C & at 2 high temperature and pressure vapour refrigerant exist.}

$0.67 = 0.07 + x(0.7366 - 0.07)$

$x = 0.90$

And Enthalpy at point 1 is,

$h_1 = h_f + xh_{fg}$

$= 20 + 0.90(180 - 20) = 164\ kJ/kg$

Now refrigeration effect is produce in the evaporator.

Heat extracted from the evaporator or refrigerating effect,

$RE = \dot{m}\ (h_1 - h_4)$

$= 0.025(164 - 80) = 2.1\ kW$

Q.4 (b)

$(COP)\text{refrigerator} = \dfrac{\text{Refrigerating effect}}{\text{Work done}}$

$= \dfrac{h_1 - h_4}{h_2 - h_1} = \dfrac{164 - 80}{200 - 16} = 2.33$

[GATE-03]

Q.5 (a)Air conditioner mounted in a window or through the wall are self-contained units of small capacity of 1 TR to 3 TR. The capillary tube is used as an expansion device in small capacity refrigeration units

Q.6 (b)
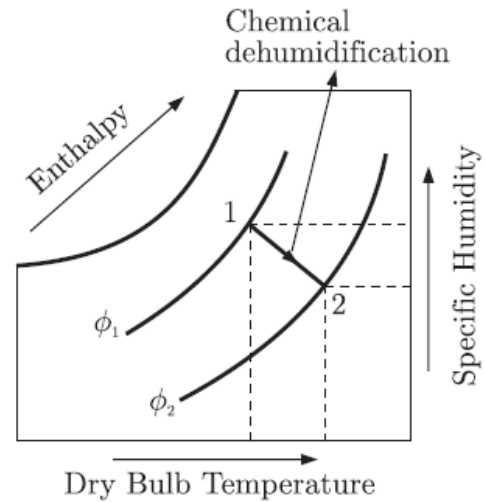
In the process of chemical dehumidification of air , the air is passed over chemicals which have an affinity for moisture and the moisture of air gets condensed out and gives up its latent heat. Due to the condensation, the specific humidity decreases and the heat of condensation supplies sensible heat for heating the air and thus increasing its dry bulb temperature.

So chemical dehumidification increase dry bulb temperature & decreases specific humidity

**Q.7** (d) If a refrigerant is written in the form of R_{abc}.

The first digit on the right (c) is the number of fluorine (F) atoms, the second digit from the right (b) is one more than the number of hydrogen (H) atoms required & third digit from the right (a) is one less than the Number of carbon (C) atoms in the refrigerant. So, For R134

First digit from the Right = 4 = Number of Fluorine atoms

Second digit from the right = 3 - 1 = 2 = Number of hydrogen atoms

Third digit from the right = 1 + 1 = 2 = Number of carbon atoms

Hence, Chemical formula is $C_2H_2F_4$

Q.8 **(c)**
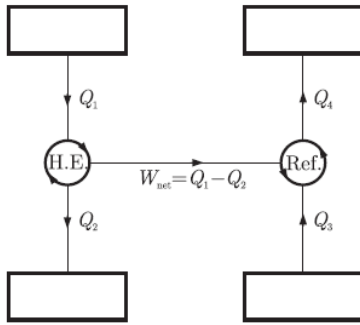
$(COP)_{ref} = 5,\ (\eta)_{H.E} = 70\% = 0.7$

$(COP)_{ref} = \dfrac{Q_3}{W} = 5........(i)$

$(\eta)_{H.E} = \dfrac{W}{Q_1} = 0.7..........(ii)$

Multiplying (i) and (ii)

$\dfrac{Q_3}{W}\dfrac{W}{Q_1} = 5 \times 0.7$

$\dfrac{Q_3}{Q_1} = 3.5$

Hence, Energy absorbed (Q_3) from low temperature reservoir by the refrigerator for each kJ of energy absorbed (Q_1) from high temperature source by the engine= 3.5 kJ

Q.9(b)

Given : t_{dp} = 18⁰C
= (273 + 18)K
= 291K,
$p = p_{atm}$ = 1.013 bar
t_{db} = 30⁰C = (273 + 30)K = 303 K
p_v = 0.02062 bar (for water vapour at dew point).
c_{air} = 1.005 kJ/kg K,
c_{water} = 1.88 kJ/kg K
Latent heat of vaporization of water at 0⁰C.
h_{fgdp} = 2500 kJ/kg

Specific humidity, $W = 0.622 \times \dfrac{p_v}{p - p_v}$

$W = 0.01291 \dfrac{kg}{kg\ of\ dry\ air}$

Enthalpy,

$h = 1.022\ T_{db} + \omega\left(h_{fgdp} + 2.3\ t_{dp}\right)$

$h = 63.15\ \dfrac{KJ}{kg}$

Q.10 (a)

Q.11 (b)
From steam table, saturated air pressure corresponding to dry bulb temperature of 35⁰C is p_s = 0.05628 bar.
Relative humidity

Relative humidity $\phi = \dfrac{P_v}{P_{vs}}$

$0.75 = \dfrac{P_v}{P_{vs}} \Rightarrow P_v = 0.04221$ bar

Specific humidity $\omega = 0.622\ \dfrac{P_v}{P_b - P_v}$

$\omega = 0.622\left(\dfrac{0.04221}{1.01 - 0.04221}\right)$

$= 0.0271$ kg/kg of dry air

Q.12 (b)Given :

Hence air gets heated, Also water is added to it, so it gets humidified.
$t_{sp} = 42^0 C, t_{db} = 40^0 C, t_{wb} = 20^0 C$
Here we see that $t_{sp} > t_{db}$

Q.13 (a)Given curve is the theoretical *p-h* curve for vapour compression refrigeration cycle.

Q.14(b)

Q.15(c)

Q.16(c)
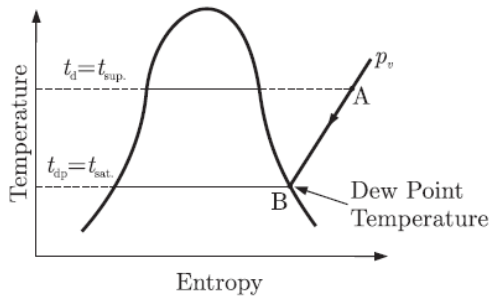
It is the temperature of air recorded by a thermometer, when the moisture (water vapour) present in it begins to condense. If a sample of unsaturated air, containing superheated water vapour, is cooled at constant pressure, the partial pressure (p_v) of each constituent remains constant until the water vapour reaches the saturated state as shown by point B. At this
point B the first drop of dew will be formed and hence the temperature at point
B is called dew point temperature

Q.17(a)

Q.18   (b)
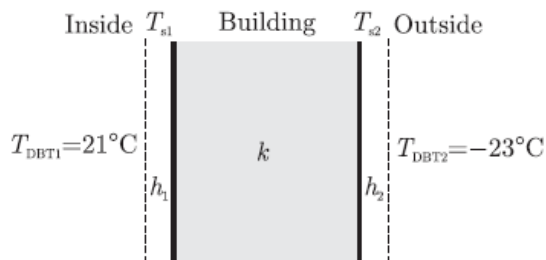
Let h_1 & h_2 be the internal and external surface heat transfer coefficients respectively and building wall has thermal conductivity k.
Given : h_1 = 8 W/m K
h_2 =23W/m² K,
k=1.2 W/m K,
T_{DPT}=10.17 ⁰C
Now to prevent condensation, temperature of inner wall should be more than or equal to the dew point temperature. It is the

limiting condition to prevent condensation
T_{s1}=10.17 ⁰C
Here T_{s1} & T_{s2} are internal & external wall surface temperature of building.
Hence, heat flux per unit area inside the building,

$$q_i = \frac{Q}{A} = h_1\left(T_{DBT1} - T_{s1}\right) \quad \text{.......(i)}$$

$$q_i = 86.64 \text{ W/m}^2$$

Heat flux per unit area outside the building is
$q_0 = h_2(T_{s2} - T_{DBT2})$
    $= 23(T_{s2} + 23)$ ....(ii)
Heat flow will be same at inside & outside the building. So from equation (i) & (ii)
$q_i = q_0$
T_{s2} = -19.23 ⁰C
For minimum thickness of the wall, use the Fourier's law of conduction for the building. Heat flux through wall,

$$q = \frac{k\left(T_{s1} - T_{s2}\right)}{x}$$

$x = 0.407$ m

Q.19   (c)
Given : m_a = 3 kg/sec,
Using subscript 1 and 2 for the inlet and outlet of the coil respectively.
h_1 = 85 kJ/kg of dry air,
W_1 = 19 grams/kg of dry air
=19 ×10⁻³ kg/kg of dry air
h_2 = 43 kJ/kg of dry air,
W_2 = 8 grams/kg of dry air
    = 8 ×10⁻³ kg/kg of dry air
h_3 = 67 kJ/kg
Mass flow rate of water vapour at the inlet of the coil is,
m_{v1} = W_1 ×m_1= 57 ×10⁻⁸ kg/sec
And mass flow rate of water vapour at the outlet of coil is,
m_{v2} = W_2 ×m_a

$m_{v2} = 24 \times 10^{-8}$ kg/sec
So, mass of water vapour condensed in the coil is,
Therefore, required cooling capacity of the coil = change in enthalpy of dry air +
change in enthalpy of condensed water
$= (85 - 43) \times 3 + 67 \times 33 \times 10^{-3}$
$= 128.211$ kW

Q.20 (b) Given : $p_1 = 100$ kPa,
$p_2 = 500$ kPa,
$p_{v1} = ?$
$p_{v2} = 5.628$ Pa
(Saturated pressure at 35^0C)
We know that,

Specific humidity, $W = 0.622 \times \dfrac{p_v}{p - p_v}$

For CASE-II

$W = 0.622 \left(\dfrac{0.628}{500 - 5.628} \right)$

$W = 7.08 \times 10^{-8} kg / kg \ of \ dry \ air$
For saturated air specific humidity remains same. So, for case (I) :

$W = 0.622 \times \dfrac{p_{v1}}{p_1 - p_{v1}}$

On substituting the values, we get
$p_{v1} = 1.13 \ kPa$

Q.21 (C)
Given: At inlet $t_{DBT} = 40 \ ^0$C
$t_{WBT} = 20 \ ^0$C
We know that,
Wet bulb depression $= t_{DBT} - t_{WBT}$
$\qquad\qquad = 40 - 20 = 20 \ ^0$C
And given wet bulb depression at the exit = 25% of wet bulb depression at inlet
This process becomes adiabatic saturation and for this process,

t_{WBT}(inlet) = t_{WBT}(outlet)
So, t_{DBT}(exit)- 20 = 0.25*20
t_{DBT}(exit) = 20 + 5 = 25 ^{0}C

Q.22 (a)
p-h curve for vapour compression refrigeration cycle is as follows
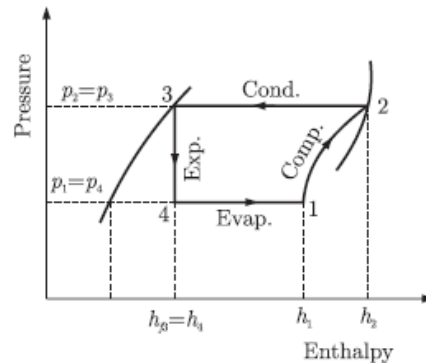
The given specific enthalpies are
Inlet of condenser $h2 = 283$ kJ/kg

Exit of condenser $h3 = 116 \dfrac{kJ}{kg}$

$= h4$
(from p-h curve)
Exit of evaporator $h1 = 232$ kJ/kg
Now,

$COP = \dfrac{\text{Refrigeration Effect}}{\text{Work Done}}$

$= \dfrac{h_1 - h_4}{h_2 - h_1} = 2.27$

Q.23 (b)

Given : $t_{DBT} = 30 \ ^0$C,
$W = 11.5$ g water vapour/kg dry air

$p_s = 4.24$ kPa, $p = 90$ kPa

$$\text{Specific humidity}, W = 0.622 \times \frac{p_v}{p - p_v}$$

$$11.5 \times 10^{-8} = 0.622\left(\frac{p_v}{90 - p_v}\right)$$
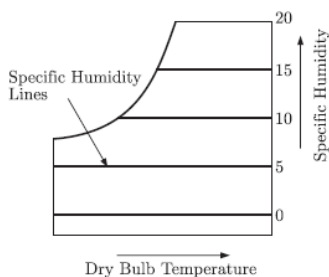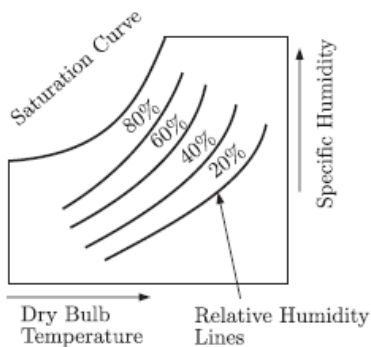
$$p_v = 1.634 \text{ kPa}$$

$$\text{Relative humidity}, (\phi) = \frac{p_v}{p_s}$$

$$= \frac{1.634}{4.24}$$

$$\phi = 0.385 = 38.5\%$$

Q.4 (d)

From the given curve, we easily see that relative humidity of air decreases, when temperature of moist air in an airtight vessel increases.
So, option (d) is correct.
Specific humidity remain constant with temperature increase, so option *a & b* are incorrect

A refrigerator operates between 120 kPa and 800 kPa in an ideal vapour compression cycle with R-134a as the refrigerant. The refrigerant enters the compressor as saturated vapour and leaves the condenser as saturated liquid. The

mass flow rate of the refrigerant is 0.2 kg/s. Properties for R134a are as follows :

Saturated R-134a					
P (kPa)	T (°C)	h_f(kJ/kg)	h_g(kJ/kg)	s_f(kJ/kg.K)	s_g(kJ/kg.K)
120	−22.32	22.5	237	0.093	0.95
800	31.31	95.5	267.3	0.354	0.918
Superheated R-134a					
P (kPa)		T (°C)	h (kJ/kg)		s (kJ/kg.K)
800		40	276.45		0.95

Q.25 (a)

T-S
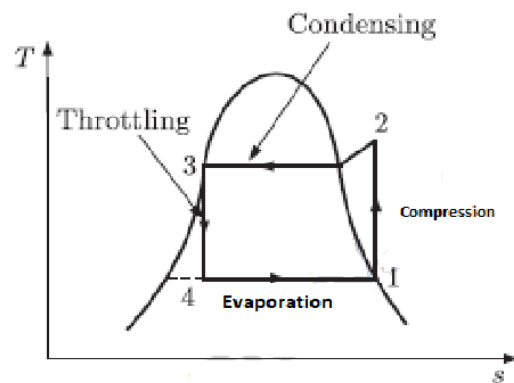
T-S Diagram for given Refrigeration cycle is given above
Since Heat is Extracted in Evaporation process.
So rate of heat extracted $= m(h_1 - h_4)$
From above diagram ($h_3 = h_4$) for throttling process, so
Heat extracted = $m (h_1 - h_3)$
From given table
$h_1 = h_g$ at 120 kPa, h_g = 237 kJ/kg
$h_3 = h_f$ at 120 kPa, h_f = 95.5 kJ/kg
Hence Heat extracted = $m (h_g - h_f)$
= 0.2(237 - 95.5)
= 28.3 kJ/s

Q.26 (c)

Since power is required for compressor in refrigeration is in compression cycle (1-2)
Hence, Power required = h_2-h_1
= h_2-h_f
Since for isentropic compression process.

$s_1 = s_2$ from figure. $= 0.95$

For entropy $s = 0.95$ the enthalpy

$h = 276.45$ kJ/kg

$h = h_2 = 276.45$ (From table)

Hence Power $= 0.2(276.45 - 237)$

$\qquad = 7.89 = 7.9$ kW

Q.27 (c)

Specific Humidity is given by

$$\omega = 0.622 \times \frac{p_v}{p_a - p_v} \quad \ldots\ldots(i)$$

Where,

$$p_v = \begin{pmatrix} \text{Relative} \\ \text{humidity} \end{pmatrix} \times \begin{pmatrix} \text{Saturated} \\ \text{steam pressure} \end{pmatrix}$$

$$= \phi \times p_s = 0.7 \times 0.0425$$

$$= 0.02975 \text{ bar}$$

So that from equation (i), we have

$$\omega = 0.622 \times \frac{0.02975}{1 - 0.02975} \left[\because P_a = 1 \text{ bar} \right]$$

$$= 0.0191 \text{ kg/kg of dry air}$$

Q.28

R744-CO_2

R290-C_3H_3 (Propane)

R502-$CHClF_3 + CClF_2CF_3$

R718-Water

Q.29

Total pressure : $p = 85$ kPa

Dry bulb temperature,

$T_{db} = 30^\circ C$

$p_s = 4.24$ kPa

$\phi = 65 \% = 0.65$

Relative humidity,

$$\phi = \frac{p_v}{p_s}$$

$$0.65 = \frac{p_v}{4.24}$$

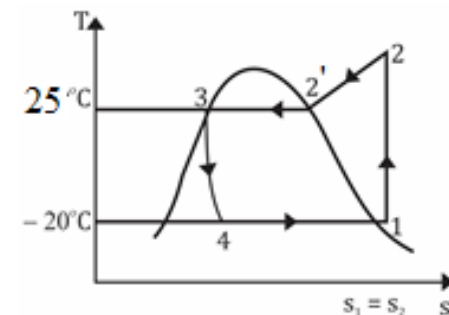$or \quad p_v = 0.65 \times 4.24 = 2.756$ kPa

Absolute humidity,

$$\omega = \frac{0.622 \; p_v}{p - p_v}$$

$= 0.02084$ kg of w.v/kg of dry air

$= 20.84$ gram of w.v./kg of dry air

$\omega_2 < \omega_1 \qquad$ Q Dehumidifier

Q.30

$$s_2 - s_{2'} = c_p \; \log_e \frac{T_2}{T_{2'}}$$

$s_1 = s_2$ (for isentropic process)

$s_2 = 1.7841$ kJ/kgK

$s_2 = 1.7183$ kJ/kgK

$c_p = 0.98$ kJ/kgK

$$\therefore \; \log_e \frac{T_2}{T_{2'}} = \frac{s_2 - s_{2'}}{c_p}$$

$$\log_e \frac{T_2}{T_{2'}} = \frac{1.7841 - 1.7183}{0.98} = 0.0671$$

or $T_2 = 1.0694 \times T_{2'} = 1.0694 \times 298$

$= 318.695 \, K$

$h_2 - h_{2'} = c_p \left(T_2 - T_{2'} \right)$

$h_2 = 413.02 + 0.98 \left(318.695 - 298 \right)$

$h_2 = 433.3 \, kJ/kg$

Q.31

Given data :

$T_2 = -5°C = \left(-5 + 273\right) K = 268 \, K$
$T_1 = 35°C = \left(35 + 273\right) K = 308 \, K$
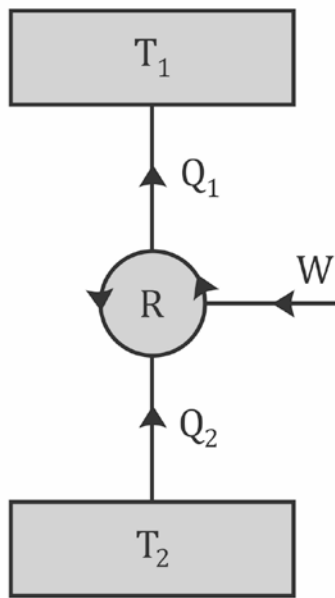
Refrigerator

$Q_2 = 2.5 \, kJ/s = 2.5 \, kW = 2500 \, W$

$\left(COP\right)_R = \dfrac{T_2}{T_1 - T_2}$

$\left(COP\right)_R = \dfrac{268}{308 - 268} = 6.7$

also $\left(COP\right) = \dfrac{Q_2}{W}$

$\therefore \quad 6.7 = \dfrac{2500}{W}$

or $\quad W = \dfrac{2500}{6.7} = 373.13 \, watt$

Q.32

Given data :

T_{db} = 35°C

$\phi = 60 \% = 0.60$

p = 0.1 MPa = 100 kPa

p_s = 5.63 kPa at 35°C

Relative humidity,

$\phi = \dfrac{p_v}{p_s}$

$0.60 \dfrac{p_v}{5.63}$

$p_v = 0.60 \times 5.63 = 3.378 \, kPa$

Humidiy ratio,

$\omega = \dfrac{0.622 \, p_v}{p - p_v} = \dfrac{0.622 \times 3.378}{100 - 3.378}$
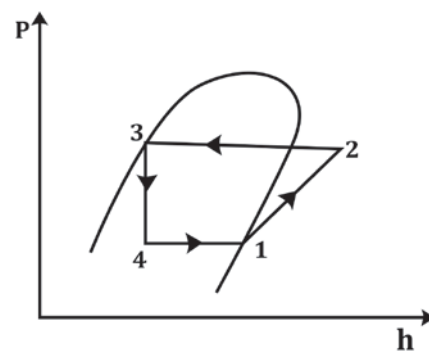
= 0.02174 kg of w.v./kg of dry air

= 21.74 gram of w. v./kg of dry air

Q.33

Given data :

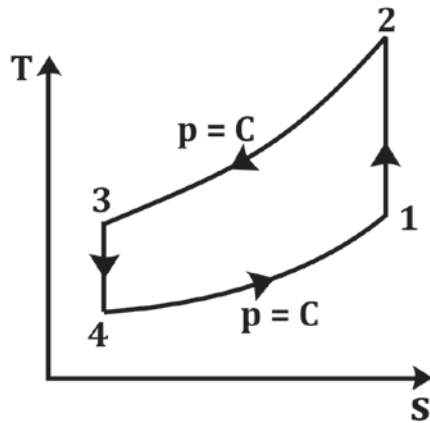$h_1 = 402 \, kJ/kg, \quad h_2 = 432 \, kJ/kg$
$h_3 = h_4 = 237 \, kJ/kg$

$COP = \dfrac{h_1 - h_4}{h_2 - h_1} = \dfrac{402 - 237}{432 - 402} = \dfrac{165}{30}$

$= 5.5$

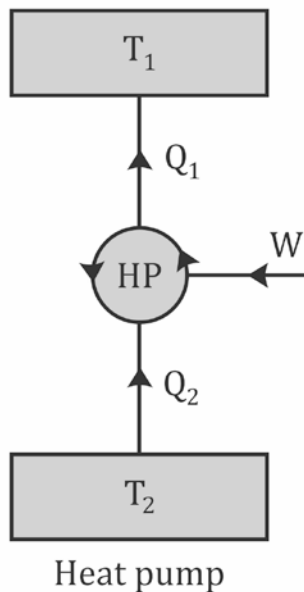**Q.34 (b)**

Reversed Brayton cycle is shown in the figure

Q.35

Given data :

$T_2 = 6°C = (6+273)K = 279\,K$

$T_1 = 37°C = (37+273)K = 310\,K$

Heat pump

$(COP)_{HP} = \dfrac{T_1}{T_1 - T_2} = \dfrac{310}{310 - 279}$

$= 10$

Q.36

Mass flow rate of moist air = 10.1 kg/s

Humidity ratio : $\omega = \dfrac{m_v}{m_a} = 0.01\,kg/s$

Mass of moist air = Mass of dry air + Mass of water vapour

$10.1 = m_a + 0.01 \times m_a$

$10.1\,m_a = 10.1$

Mass of dry air $m_a = \dfrac{10.1}{1.01} = 10\,kg/s$

Mass of water vapour,

$m_{v1} = 10.1 - 10 = 0.1\,kg/s$

$m_{v2} = 0.1\,kg/s$

$(m_v)_{total} = m_{v1} + m_{v2} = 0.2\,kg/s$

Humidity ratio $\omega_{final} = \dfrac{m_v}{m_a} = \dfrac{0.2}{10}$

= 0.02 kg/kg of dry air

Q.37

Given data :

Total pressure,

p = 750mm of Hg

Partial pressure, V

P_v = 20mm of Hg

We know that humidity ratio,

$\omega = \dfrac{0.622\,p_v}{p - p_v}$ kg. w.v./kg d.a.

$= \dfrac{0.622 \times 20}{750 - 20}$

$= 0.01704\,kg\ w.v./kg\ of\ d.a.$

$= 17.04\,g\ of\ w.v./kg\ of\ d.a.$

Q.38

Given data:

m = 0.05kg/s

P_e = 0.14 MPa

h_1 = 236.04 kJ/kg

$S_1 = 0.9322$ kJ/kgK

p = 0.8 MPa

$h_3 = h_4 = h_f = 93.42$ kJ/kg
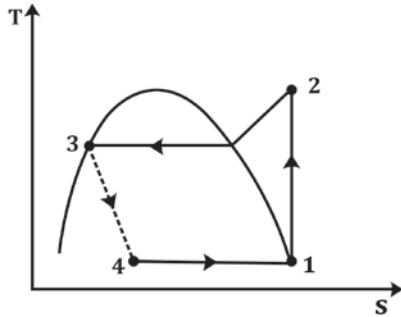
$h_2 = 272.02$ kJ/kg

Rate of heat rejection to the environment,

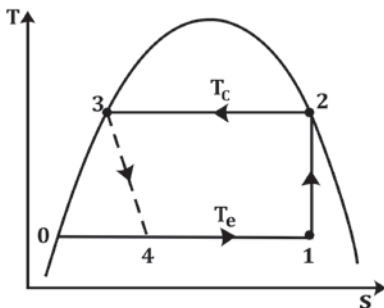$$Q_{2-3} = m[h_2 - h_3] = 0.05 \times [275.02 - 93.42]$$
$$= 8.93 \text{ kW}$$

Q.39

Given data :

$T_e = 260$K

$T_C = 310$K

$c_{pl} = 4.8$ kJ/kgK

$h_{fg} = 1054$ kJ/kg at $T_C = 310$ K

Taking reference temperature T, at which entropy

is S_T.

$$\therefore \quad s_3 - s_T = c \ln \frac{T_3}{T}$$

$$s_3 = s_T + c \ln \frac{T_3}{T}$$

$$= s_T + 4.8 \ln \frac{310}{T}$$

Similarly,

$$s_0 = s_T + c \ln \frac{260}{T}$$

$$= s_T + 4.8 \ln \frac{260}{T}$$

$$\therefore \quad s_2 - s_3 = \frac{h_{fg}}{T} = \frac{1054}{310}$$

$$s_2 = s_3 + \frac{1054}{310}$$

Substitute the value of s_3 in above equation, we get

$$s_2 = s_1$$

$$s_1 = s_T + 4.8 \ln \frac{310}{T} + \frac{1054}{310}$$

$$\therefore \quad h_1 - h_0 = T_e [s_1 - s_0]$$

$$= 260 \times \left[s_T + 4.8 \ln \frac{310}{T} + \frac{1054}{310} - s_T - 4.8 \ln \frac{260}{T} \right]$$

$$= 260 \left[4.8 \ln \left(\frac{310}{T} \times \frac{T}{260} \right) + \frac{1054}{310} \right]$$

$$= 1103.51 \text{ kJ/kg}$$

Q.40

Given data :

Relative humidity,

ϕ = 70 % = 0.70

Partial pressure,

p_v = 1.6 kPa

Total pressure,

p_0 = 101.325 kPa

We know that,

$$\phi = \frac{p_v}{p_s}$$

$$0.70 = \frac{1.6}{p_s}$$

$$or \quad p_s = \frac{1.6}{0.70} = 2.2857 \text{ kPa}$$

Temperature corresponding to saturation pressure

is dry bulb temperature,

$$\ln\left\{\frac{p_s}{p_o}\right\} = 14.317 - \frac{5304}{T_s}$$

$$\ln\left\{\frac{2.2857}{101.325}\right\} = 14.317 - \frac{5304}{T_s}$$

$$-3.79166 = 14.317 - \frac{5304}{T_s}$$

$$\frac{5304}{T_s} = 14.317 + 3.79166$$

$$\frac{5304}{T_s} = 18.10866$$

$$or \quad T_s = 292.89 \text{ K}$$
$$= (292.89 - 273)\,^\circ\text{C}$$
$$= 19.89\,^\circ\text{C}$$

Q.41

Relative humidity $\phi = \dfrac{P_v}{P_{vs}}$

$$0.55 = \frac{P_v}{4.246}$$

$$P_v = 2.335$$

$$\omega = \left(\frac{M_{H_2O}}{M_{air}}\right)\left(\frac{P_v}{P_t - P_v}\right)$$

$$\omega = \frac{18}{28.84}\left(\frac{2.335}{100 - 2.335}\right)$$

$$\omega = 0.0149\,\frac{\text{Kg of water vapour}}{\text{Kg of dry air}}$$

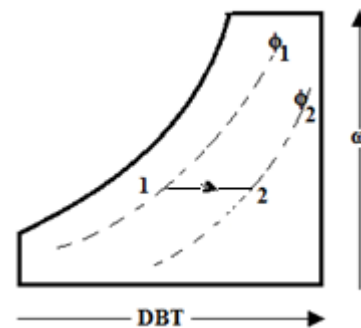$$Q \; \omega = \frac{m_v}{m_a}$$

$$m_v = \omega m_a$$

$$= 0.0149 \times 10^3 \times 1$$

$$m_v = 14.9 \text{grm}$$

Q.42   (a)

For closed metallic vessel sp. Humidity (ω) remain const

From the figure

$$\phi_2 < \phi_1$$

Q.43

Relative humidity $\phi = \dfrac{P_v}{P_s}$

But, $\phi = 60\% = 0.6$, $P_s = 4.24$ kPa

$\Rightarrow p_a = \phi \times p_s = 0.6 \times 4.24 = 2.544$ kPa

Specific humidity $\omega = 0.622\,\dfrac{p_v}{p_b - p_v}$

where p_b = ambient air pressure = 100 kPa

$$\Rightarrow \omega = 0.622 \times \left(\frac{2.544}{100 - 2.544}\right)$$

$\Rightarrow \omega = 0.016236$ kg/kg of dry air

$\Rightarrow \omega = 16.236$ g/kg of dry air

Q.44 (10)

RC = 15 kW

$$RC = \dot{m} \times \left(\left(h_1 - h_4 \right) \right)$$

$$RC = \dot{m} \times \left(400 - 250 \right)$$

$$\dot{m} = 0.1 \text{ kg/sec}$$

$$W_{\text{isentropic}} = \left(h_2 - h_1 \right) = \left(475 - 400 \right) = 75 \, \text{kJ/kg}$$

$$\eta_C = \frac{W_{\text{isentropic}}}{W_{\text{actual}}}$$

$$W_{\text{actual}} = \dot{m} \times W_{\text{actual}} = 0.1 \times 100$$

$$P_{\dot{m}} = 10 \text{ kW}$$