DETAILS EXPLANATIONS

ME: Paper-1 (Paper-5) [Full Syllabus]

[PART : A]

1. Temperature at which all molecular motion ceases, according to the kinetic theory of gases. A point which has been determined on the thermodynamic scale (by theoretical considerations) beyond which a further decrease in temperature is inconceivable. This is equal to – 459.6° on the fahrenheit scale and –273.15° on the centigrade scale.
2. Equal volumes of different gases at the same temperature and pressure contain the same number of molecules.
3. A body which absorbs all the radiation falling on it i.e., has a non-reflecting surface. A black body emits the maximum amount of radiation possible at a given temperature, and the amount is proportional to the fourth power of the absolute temperature.
4. The temperature at which a liquid boils for any given surrounding atmospheric pressure. Now the saturation pressure of the vapour equals that of the atmosphere.
5. Is that state at which liquid and vapour coexist in equilibrium. At critical state, latent heat of evaporation becomes zero.
6. At a common temperature, a mixture of gases will exert on the sides of the vessel a total pressure equal to the sum of the pressures which each constituent would exert separately if it alone occupied the vessel.
7. Is a process wherein a fluid from a pressure chamber expands into a vacuum chamber through an orifice of large dimensions.
8. The maximum possible temperature attained by the products of reaction, when the reaction goes to completion and all the heat released is used to heat up the Product.
9. An apparatus used for determining the calorific values of fuels. The bomb consists of a thick walled steel vessel in which a weighed quantity of fuel is ignited in an atmosphere of compressed oxygen. The bomb is immersed in a known volume of water, from the rise of temperature of water the calorific value is calculated.
10. A device that makes the final heat recovery from boiler flue gases and uses the same to preheat the incoming furnace air for its reaction with fuel.
11. The valve which empties the boiler for cleaning, inspection, or repair. It blows out mud, scale, or sediment when the boiler is in operation and prevents excessive concentration of soluble impurities in the boiler. Also used for rapid lowering of boiler water level if it is too high.
12. The upward movement of warm air or gas, compared with ambient air or gas, due to the lesser density of the warm air or gas. Chimney effect may be a cause of uneven heating in buildings two or more stories high.
13. A method of improving the thermal efficiency of steam plant by withdrawing a small part of the steam from the higher pressure stages of a turbine to heat the boiler feed water.
14. A steam turbine in which the steam is expanded both in the fixed blade and the moving blade continuously as the steam passes over them. The pressure drops gradually and continuously over both moving and fixed blades. Often called as REACTION TURBINE.
15. In a multistage steam turbine it is the ratio of the sum of the individual heat drops (cumulative drop) in the different stages to the direct or adiabatic drop in a single step for the whole pressure drop that occurs.
16. Ratio of the rate of doing work per kg of steam (diagram work) to the energy supplied to the stage per kg of steam, in a steam turbine. Also called GROSS STAGE EFFICIENCY. This is product of nozzle efficiency and blade efficiency.

17. Actual quantity of water vapour in the air, usually expressed as so many grains of moisture in a cubic foot of air.

18. Injection of liquid fuel into the cylinder of an oil engine by a high pressure fuel pump, so dispensing with the compressed air necessary in the early diesel engines. Also called SOLID INJECTION or MECHANICAL INJECTION.

19. A fuel metering system in a diesel engine, with a helical groove in the plunger which covers and uncovers ports in the pump barrel and thereby varies the effective stroke of the fuel pump.

20. The time interval between the start of injection and start of ignition. Also called DELAY PERIOD

[PART : B]

21. Liquid water is to be heated in an electric teapot. The heating time is to be determined.

Assumptions :

- Heat loss from the teapot is negligible.
- Constant properties can be used for both the teapot and the water

Properties :

The average specific heats are given to be 0.7 kJ/kg .°C for the teapot and 4.18 kJ/kg .°C for water.

Analysis :

We take the teapot and the water in it as the system, which is a closed system (fixed mass}. The energy balance in this case can be expressed as

$$E_{in} - E_{out} = \Delta E_{System}$$

$$E_{in} = \Delta U_{system} = \Delta U_{water} + \Delta U_{teapot}$$

Then the amount of energy needed to raise the temperature of water and the teapot from 15°C to 95°C is

$$E_{in} = (mC\Delta T)_{water} + (mC\Delta T)_{teapot}$$

$$= (1.2\text{ kg})(4.18\text{ kJ/kg.°C})(95 - 15)\text{°C} + (0.5\text{ kg})(0.7\text{ kJ/kg.°C})(95 - 15)\text{°C}$$

$$= 429.3\text{ kJ}$$

The 1200-W electric heating unit will supply energy at a rate of 1.2 kW or 1.2 kJ per second. Therefore, the time needed for this heater to supply 429.3 kJ of heat is determined from

$$\Delta t = \frac{\text{Total energy transferred}}{\text{Rate of energy iransfer}} = \frac{E_{in}}{\dot{E}_{transfer}}$$

$$= \frac{429.3\text{kJ}}{1.2\text{kJ/s}} = 358\text{ s} = 6.0\text{ min}$$

22. The rates of heat transfer to and from a heat engine are given.

The net power output and the thermal efficiency are to be determined.

Assumptions Heat losses through the pipes and other components are negligible.

Analysis : The furnace serves as the high-temperature reservoir for this heat engine and the river as the low-temperature reservoir. The given quantities can be expressed as

$$\dot{Q}_H = 80\text{ MW and }\dot{Q}_L$$

The net power output of this heat engine is

$$\dot{W}_{Net,out} = \dot{Q}_H - \dot{Q}_L = (80 - 50)\text{ MW} = 30\text{ MW}$$

The the thermal efficiency is easily determined to be

$$\eta_{th} = \frac{\dot{W}_{net,\,out}}{\dot{Q}_H} = \frac{30\text{ MW}}{80\text{ MW}} = 0.375 \text{ (or 37.5\%)}$$

23. Rate of exergy flow is to be determined.

Analysis : The furnace in this example can be modeled as a heat reservoir that supplies heat indefinitely at a constant temperature. The exergy of this heat energy is its useful work potential, that is, the maximum possible amount of work that can be extracted from it. This corresponds to the amount of work that a reversible heat engine operating between the furnace and the environment can produce.

The thermal efficiency of this reversible heat engine is

$$\eta_{th,max} = \eta_{th,rev} = 1 - \frac{T_L}{T_H} = 1 - \frac{T_0}{T_H}$$

$$= 1 - \frac{537R}{2000R} = 0.732 \text{ (or 73.2\%)}$$

That is, a heat engine can convert, at best 73.2 percent of the heat received from this furance to work. Thus the exergy of this furnace is equivalent ot the power produced by the reversible heat engine :

$$\dot{W}_{max} = \dot{W}_{rev} = \eta_{th,rev}\dot{Q}_{in} = (0.732)(3000\text{ Btu/s}) = 2196\text{ Btu/s}$$

24.
- The refrigerant should have low boiling point and low freezing point.
- It must have low specific heat and high latent heat. Because high specific heat decreases the refrigerating effect per kg of refrigerant and high latent heat at low temperature increases the refrigerating effect per kg of refrigerant.
- The pressures required to be maintained in the evaporator and condenser should be low enough to reduce the material cost and must be positive to avoid leakage of air into the system.
- It must have high critical pressure and temperature to avoid large power requirements.
- It should have low specific volume to reduce the size of the compressor.
- It must have high thermal conductivity to reduce the area of heat transfer in evaporator and condenser.
- It should be non-flammable, non-explosive, non-toxic and non-corrosive.
- It should not have any bad effects on the stored material or food, when any leak develops in the system.
- It must have high miscibility with lubricating oil and it should not have reacting properly with lubricating oil in the temperature range of the system.
- It should give high COP in the working temperature range. This is necessary to reduce the running cost of the system.
- It must be readily available and it must be cheap also.

25. Actual height of mercury column= Mecrcury column height + Capillary depression

Specific weight of mercury = $\rho g = 13600 \times 9.8\text{ N/m}^3$

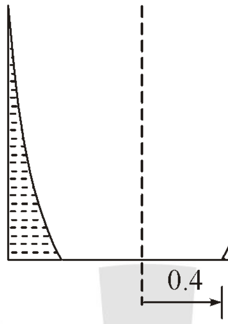
Capillary depression, $$h = \frac{4\sigma \times \cos\beta}{\gamma D} = \frac{4 \times 0.48 \times \cos 135}{0.002 \times 13600 \times 9.81}$$

$$= 5.09 \times 10^{-3}\text{ m} = -5.09\text{ mm (depression)}$$

Corrected height of mercury column = 757 + 5.09 = 762.09 mm

26. $$\left(P_{r_2} - P_{r_1}\right) = \rho \times \left(\frac{\omega^2}{2}\right) \times \left(r_2^2 - r_1^2\right)$$

$$P_{r_1} = 0 \text{ (gauge) at } r_1 = 0.4,\ r_2 = 0.5 \text{ m}$$



$$\omega = \frac{2\pi N}{60} = 2 \times \pi \times \frac{150}{60} = 15.71 \text{ rad/s}$$

$$P_{r_2} - 0 = 1000 \times \left(\frac{15.71^2}{2}\right) \times (0.5^2 - 0.4^2) \text{ N/m}^2$$

$$P_{r_2} = 11106.184 \text{ N/m}^2 \text{ (gauge)}$$

Using equation, $$y_2 - y_1 = \frac{\omega^2}{2g}[r_2^2 - r_1^2]$$

27. In the Lagrangian method a single particle is followed over the flow field, the co-ordinate system following the particle. The flow description is particle based and not space based. A moving coordinate system has to be used. This is equivalent to the observer moving with the particle to study the flow of the particle. This method is more involved mathematically and is used mainly in special cases.

In the Eularian method, the description of flow is on fixed coordinate system based and the description of the velocity etc. are with reference to location and time i.e., V = V (x, y, z, t) and not with reference to a particular particle. Such an analysis provides a picture of various parameters at all locations in the flow field at different instants of time. This method provides an easier visualisation of the flow field and is popularly used in fluid flow studies. However the final description of a given flow will be the same by both the methods.

28. In a flow field if a continuous line can be drawn such that the tangent at every point on the line gives the direction of the velocity of flow at that point, such a line is defined as a stream line. In steady flow any particle entering the flow on the line will travel only along this line. This leads to visualisation of a stream line in laminar flow as the path of a dye injected into the flow.

There can be no flow across the stream line, as the velocity perpendicular to the stream line is zero at all points. The flow along the stream line can be considered as one dimensional flow, though the stream line may be curved as there is no component of velocity in the other directions. Stream lines define the flow paths of streams in the flow. The flow entering between two stream lines will always flow between the lines. The lines serve as boundaries for the stream.

In the cartesian co-ordinate system, along the stream line in two dimensional flow it can be shown that

$$\frac{dx}{u} = \frac{dy}{v}$$

or $$v\,dx - u\,dy = 0$$

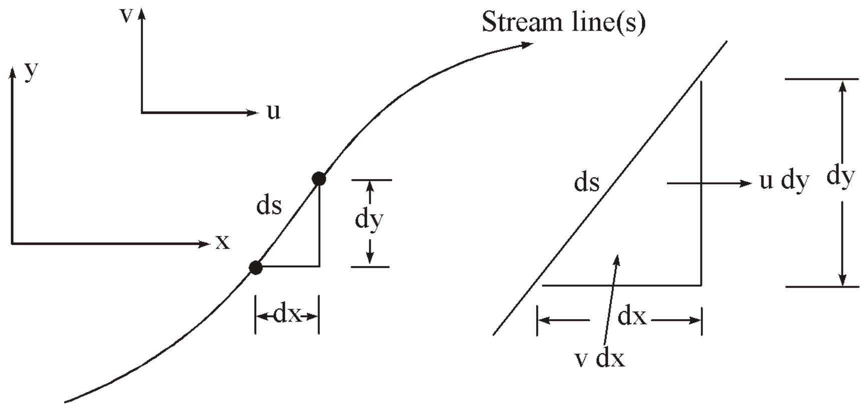


Figure : Velocity components along a stream line

Referring to figure considering the velocity at a point and taking the distance ds and considering its x and y components as dx and dy, and noting that the net flow across ds is zero,

the flow along y direction = dx v

the flow along x direction = dy u

These two quantities should be equal for the condition that the flow across ds is zero, thus proving the equation.

In the next para, it is shown that stream lines in a flow can be described by a stream function having distinct values along each stream line.

9. ***Dynamic pump and Positive displacement pump are following :***

S.No.	Dynamic Pumps	Positive Displacement Pumps
1.	Simple in construction.	More complex, consists of several moving parts.
2.	Can operate at high speed and hence compact.	Speed is limited by the higher inertia of the moving parts and the fluid.
3.	Suitable for large volumes of discharge at moderate pressures in a single stage.	Suitable for fairly low volumes of flow at high pressures.
4.	Lower maintenance requirements.	Higher maintenance cost.
5.	Delivery is smooth and continuous.	Fluctuating flow.

30. These are used more often for oil pumping. Gear pumps consist of two identical mating gears in a casing as shown in figure.

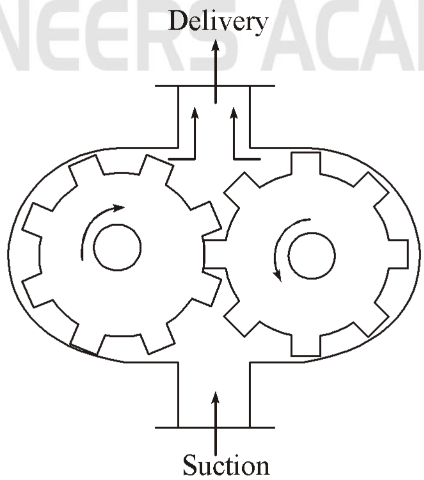


Figure : Gear Pump

The gears rotate as indicated in the sketch. Oil is trapped in the space between the gear teeth and the casing. The oil is then carried from the lower pressure or atmospheric pressure and is delivered at the pressure side. The two sides are sealed by the meshing teeth in the middle. The maximum pressure that can be developed depends on the clearance and viscosity of the oil. The operation is fairly simple. One of the gear is the driving gear directly coupled to an electric motor or other type of drives.

These pumps should be filled with oil before starting.

The sketch shows an external gear pump. There is also another type of gear pump called internal gear pump. This is a little more compact but the construction is more complex and involved and hence used in special cases where space is a premium

31. This type is also popularly used with oil. The diagramatic sketch of a lobe pump is shown in figure.

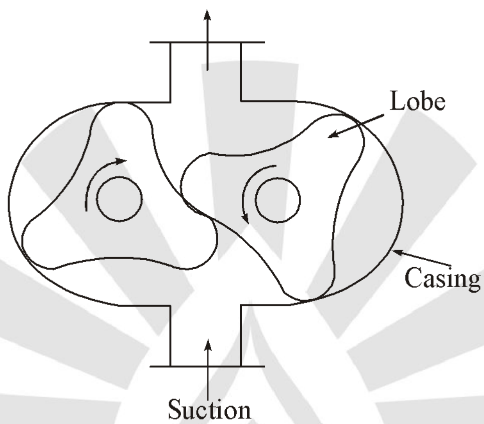


Figure : Lobe Pump

This is a three lobed pump. Two lobe pump is also possible. The gear teeth are replaced by lobes. Two lobes are arranged in a casing. As the rotor rotates, oil is trapped in the space between the lobe and the casing and is carried to the pressure side. Helical lobes along the axis are used for smooth operation. Oil has to be filled before starting the pump. Lobe type of compressors are also in use. The constant contact between the lobes makes a leak tight joint preventing oil leakage from the pressure side.

The maximum pressure of operation is controlled by the back leakage through the clearance. This type of pump has a higher capacity compared to the gear pump.

32. ***Rotameter (Float Meter) :***

The rotameter is a device whose indication is essentially linear with flow rate. This device is also called as variable area meter or float meter. In this device a flot moves freely inside a tapered tube as shown in figure.

The flow takes place upward through the tube. The following forces act on the float

(i) Downward gravity force

(ii) Upward buoyant force

(iii) Pressure

(iv) Viscous drag force

For a given flow rate, the float assumes a position inside the tube where the forces acting on it are in equilibrium.

Through careful design, the effects of changes in viscosity or density may be minimized, leaving only the pressure forces as the main variable. Pressure force depends on flow rate and area available for flow. Hence the position of the float indicates the flow rate.

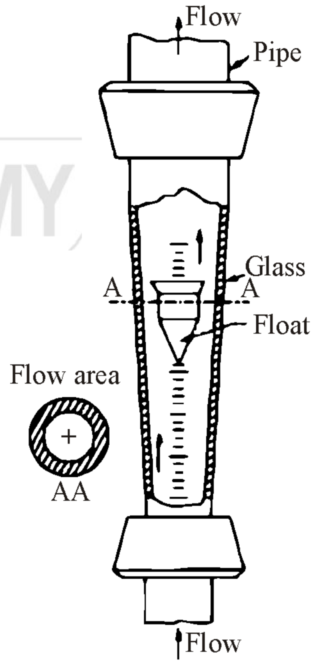


Figure : Rotameter

A major limitation in using rotameters is that these have to be installed in vertical position only. Also it cannot be used with liquids containing large number of solid particles and at high pressure conditions. It is also expensive. The advantage is that its capacity to measure the flow rate can be easily changed by changing the float or the tube.

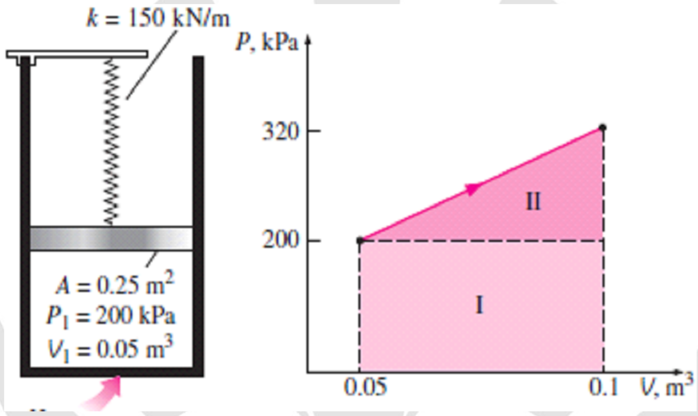
[PART : C]

33. A gas in a piston–cylinder device equipped with a linear spring expands as a result of heating. The final gas pressure, the total work done, and the fraction of the work done to compress the spring are to be determined.

Assumptions :

- The expansion process is quasi-equilibrium.
- The spring is linear in the range of interest.

Analysis : A sketch of the system and ttie P-V diagram of the process are shown in figure.



(i) Trie enclosed volume at the final state is

$$V_2 = 2V_1 = (2)(0.05\ m^3) = 0.1\ m^3$$

Then the displacement of the piston (and of the spfing) becomes

$$x = \frac{\Delta V}{A} = \frac{(0.1-0.05)m^3}{0.25m^2} = 0.2\ m$$

The force applied by the linear spring at the final state is

$$F = kx = (150\ kN/m)(0.2\ m) = 30\ kN$$

The additional pressure applied By the spring on the gas at this state is

$$P = \frac{F}{A} = \frac{30\,kN}{0.25\,m^2} = 120\ kPa$$

Without the spring, the pressure of the gas would remain constant at 200 kPa while the piston is rising. But under the effect of the spring, the pressure rises linearly from 200 kPa to

200 + 120 = 3211 kPa at the final state.

(ii) An easy way of finding the work done is to plot the process on a P-V diagram and find the area under the process curve. From Fig. 4-10 the area under the process curve (a trapezoid) is determined to be

$$W = area = \frac{(200+320)kPa}{2}\left[(0.1-0.05)m^3\right]\left(\frac{1kJ}{1kPa.m^3}\right) = 13\ kJ$$

34. Brayton Cycle (Simple Gas Turbine Cycle) :

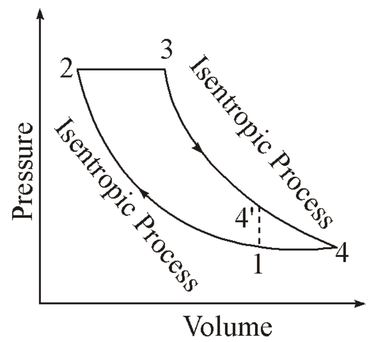


Figure (a)

The Brayton cycle is a theoretical cycle for simple gas turbine. This cycle consists of two isentropic and two constant pressure processes. Figure (a) shows the Brayton cycle on p-v and T-s coordinates. The cycle is similar to the Diesel cycle in compression and heat addition. The isentropic expansion of the Diesel cycle is further extended followed by constant pressure heat rejection. The thermai efficiency is given by,

$$\eta_{th} = \frac{\text{Heat added} - \text{Heat rejected}}{\text{Heat added}}$$

$$\eta_{th} = \frac{mC_p(T_3 - T_2) - mC_p(T_4 - T_1)}{mC_p(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

For isentropic processes, we have,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\gamma-1/\gamma} \text{ and } \frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\gamma-1/\gamma}$$

But, $p_2 = p_3$ and $p_1 = p_4$, thus. $\frac{T_2}{T_1} = \frac{T_3}{T_4}$

and we can write, $\eta_{th} = 1 - \frac{T_4}{T_3} = 1 - \frac{T_1}{T_2}$

$$\frac{T_3}{T_4} = \frac{T_2}{T_1} = \frac{v_2}{v_1} = \frac{1}{r^{\gamma-1}}$$

$$\frac{1}{r^{\gamma-1}} = \left(\frac{v_2}{v_1}\right)^{\gamma-1} \left\{\left(\frac{p_2}{p_1}\right)^{\gamma-1}\right\} = (r_p)^{\gamma-1/\gamma}$$

$$\eta_{th} = 1 - \frac{1}{r_p^{\gamma-1/\gamma}}$$

35. Process 1-2 represents the admission of high pressure steam into the engine cylinder, process 2-3 is the reversible adiabatic expansion of steam in the cylinder and process 3-4 is the exhaust of steam into condenser. Net work done is represented by the area 1-2-3-4-1.

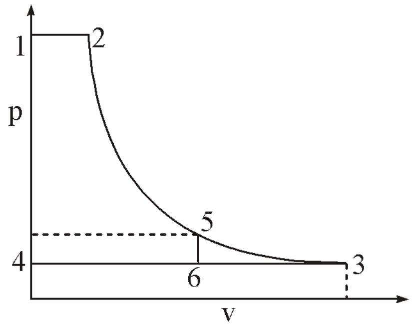


Figure : (a) p-v diagram of modified Rankine cycle

Observe that the area 3-6-5 is very small and in order to obtain this small work, the cylinder volume must be increased from v_6 to v_3.This makes cylinder very bulky. For this reason, the expansion process is terminated at point 5. So that indicator diagram becomes 1-2-5-6-4. The work lost is small but there is large saving in cylinder volume.

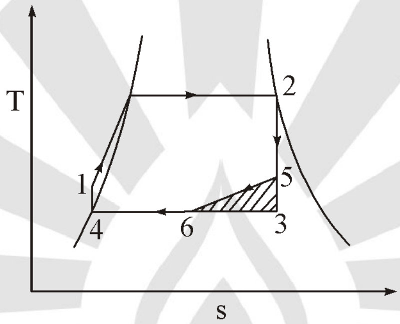


Figure : (b) T-s diagram of modified Rankine cycle

Process 5-6 represents the release of steam into the condenser, thus causing the cylinder pressure to drop from P_5 to P_6. Process 6-4 is the exhaust of steam at constant pressure. Cycle 1-2-5-6-4 is called as the "modified Rankine cycle".

Thermal Efficiency :

Considering the unit mass of working fluid.

$$\text{Heat supplied} = h_2 - h_1$$

$$\text{Net workdone} = \{w_{2-5} + w_{5-6} + w_{4-1}\}$$

$$= (h_2 - h_5) - \int_5^6 v\,dp + (h_4 - h_1)$$

$$= (h_2 - h_5) + v_5(p_5 - p_6) + (h_4 - h_1)$$

v_5 = Specific volume of steam at state 5.

$$\eta_{th} = \frac{\text{Net Workdone}}{\text{Heat Supplied}} = \frac{(h_2 - h_5) + v_5(p_5 - p_6) + (h_4 - h_1)}{(h_2 - h_1)}$$

If pump work is neglected, then $h_4 \approx h_1$

$$\eta_{th} = \frac{(h_2 - h_5) + v_5(p_5 - p_6)}{(h_2 - h_4)}$$

36. Some industrial applications require moderately low temperatures, and the temperature range they involve may be too large for a single vaporcompression refrigeration cycle to be practical. A large temperature range also means a large pressure range in the cycle and a poor performance for a reciprocating compressor. One way of dealing with such situations is to perform the refrigeration process in stages, that is, to have two or more refrigeration cycles that operate in series. Such refrigeration cycles are called cascade refrigeration cycles. A two-stage cascade refrigeration cycle is shown in figure.

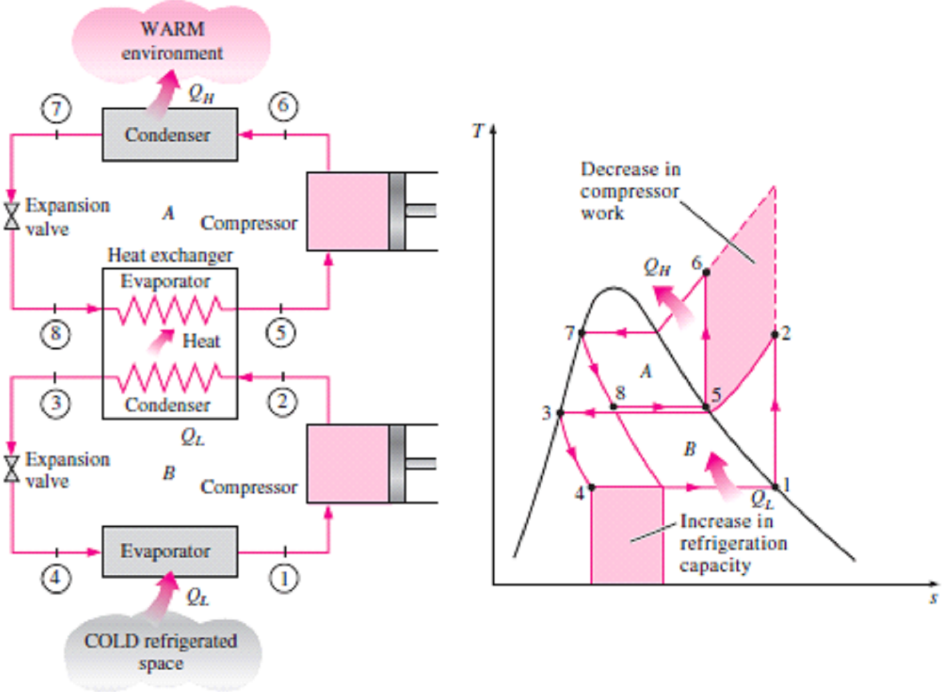


Figure : A two stage cascade refrigeration system with the same refrigerant in both stages

The two cycles are connected through the heat exchanger in the middle, which serves as the evaporator for the topping cycle (cycle A) and the condenser for the bottoming cycle (cycle B). Assuming the heat exchanger is well insulated and the kinetic and potential energies are negligible, the heat transfer from the fluid in the bottoming cycle should be equal to the heat transfer to the fluid in the topping cycle. Thus, the ratio of mass flow rates through each cycle should be

$$\dot{m}_A(h_5 - h_g) = \dot{m}_B(h_1 - h_3) \rightarrow \frac{\dot{m}_A}{\dot{m}_B} = \frac{h_2 - h_3}{h_5 - h_g} \quad ...(1)$$

Also,

$$COP_{R,cascade} = \frac{\dot{Q}_L}{\dot{W}_{net,in}} = \frac{\dot{m}_B(h_1 - h_4)}{\dot{m}_A(h_6 - h_5) + \dot{m}_g(h_2 - h_1)} \quad ...(2)$$

In the cascade system shown in the figure, the refrigerants in both cycles are assumed to be the same. This is not necessary, however, since there is no mixing taking place in the heat exchanger. Therefore, refrigerants with more desirable characteristics can be used in each cycle. In this case, there would be a separate saturation dome for each fluid, and the T-s diagram for one of the cycles would be different. Also, in actual cascade refrigeration systems, the two cycles would overlap somewhat since a temperature difference between the two fluids is needed for any heat transfer to take place.

It is evident from the T-s diagram in figure that the compressor work decreases and the amount of heat absorbed from the refrigerated space increases as a result of cascading. Therefore, cascading improves the COP of a refrigeration system. Some refrigeration systems use three or four stages of cascading.

37.

$$bp = \frac{2\pi NT}{60000} = \frac{2 \times \pi \times 4100 \times 160}{60000} = 68.66$$

$$P_{bm} = \frac{bp \times 60000}{LAnK} = \frac{68.66 \times 60000}{0.1 \times \frac{\pi}{4} \times (0.08)^2 \times \frac{4100}{2} \times 6} = 6.66 \times 10^5 \ P_a$$

$$pb_m = 6.66 \text{ bar}$$

$$\eta_{bth} = \frac{bp}{m_f \times C_v} = \frac{68.66 \times 3600}{19.8 \times 43000} \times 100 = 29.03\%$$

Compression ratio, $$r = \frac{V_s + V_d}{V_d}$$

$$V_s = \frac{\pi}{4} D^2 L = \frac{\pi}{4} \times 8^2 \times 10 = 502.65 \text{ cc}$$

$$r = \frac{502.65 + 70}{70} = 8.18$$

Air-standard efficiency, $$\eta_{otto} = 1 - \frac{1}{(8.18)^{0.4}} = 1 - \frac{1}{2.3179} = 0.56858$$

Relative efficiency, $$\eta_{rel} = \frac{0.2903}{0.568} \times 100 = 51.109\%$$

$$\eta_{bth} = \frac{bp}{m_f \times C_v} = \frac{119.82 \times 60}{(4.4/10) \times 44000}$$

$$\eta_{bth} = 37.134\%$$

Volume flow rate of air at intake condition.

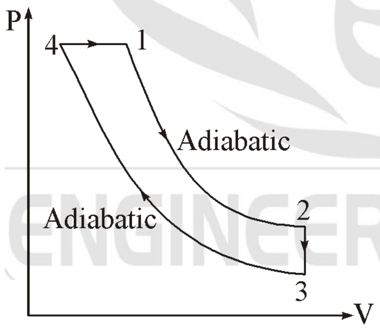
$$a = \frac{6 \times 287 \times 300}{1 \times 10^5} = 5.17 \text{ m}^3/\text{min}$$

Swept volume per minute, $$V_s = \frac{\pi}{4} D^2 LnK = \frac{\pi}{4} \times (0.1)^2 \times 0.9 \times \frac{4500}{2} \times 9 = 127.17 \text{ m}^3/\text{min}$$

Volumetric efficiency, $$\eta_v = \frac{5.17}{127.17} \times 100 = 4.654\%$$

Air-fuel ratio, $$\frac{A}{F} = \frac{6.0}{0.44} = 13.64$$

38. This cycle is shown in figure. Diesel engines using diesel fuel work on this cycle. The main difference lies in the fact that at the end of compression process sufficiently high temperature is obtained and fuel which is injected at this point ignites without any aid. In case of O_{tto} cycle, a spark is needed to cause ignition of the fuel which is present during process of compression.



1-2 : Adiabatic expansion
2-3 : Constant volume heat rejection
3-4 : Adiabatic compression
4-1 : Constant pressure heat addition

Figure : Diesel Cycle

In this cycle, the heat is transferred to fuel during constant pressure process when fuel is injected. The fuel burns during constant pressure process only. The gas (air) then expands adiabatically followed by heat rejection which occurs at constant volume. The air is then compressed adiabatically.

$$\eta = \frac{\text{Work done}}{\text{Heat added}} = \frac{\text{Work added} - \text{Heat rejected}}{\text{Heat added}}$$

$$\eta = \frac{Q_{41} - Q_{23}}{Q_{41}} = \frac{C_p(T_1 - T_4) - C_v(T_2 - T_3)}{C_p(T_1 - T_4)}$$

$$= 1 - \frac{C_v(T_2 - T_3)}{C_p(T_1 - T_4)} = \frac{T_3(T_2/T_3 - 1)}{\gamma T_4(T_1/T_4 - 1)}$$

For adiabatic compression, $\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1}$

For adiabatic expansion, $\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{\gamma-1}$

Calling $V_4 = 1$, $V_1 = \rho$, the cutt off ratio

and $V_2 = V_3 = r$ = compression ratio

$$\frac{V_3}{V_4} = r,\ \frac{V_2}{V_1} = \left(\frac{r}{\rho}\right)$$

For constant pressure process 4 -1

$$p_4 = p_1$$

$$\therefore \quad \frac{p_1V_1}{T_1} = \frac{p_4V_4}{T_4}$$

or $$\frac{\rho}{T_1} = \frac{1}{T_4} \text{ or } \frac{T_1}{T_4} = \rho$$

For constant volume process, 2-3

$$\frac{p_2V_2}{T_2} = \frac{p_3V_3}{T_3}$$

or $$\frac{p_2}{p_3} = \frac{T_2}{T_3}$$

For adiabatic process 1-2

For adiabatic process 3-4 $\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{\gamma-1} = (r)^{\gamma-1}$

Substituting for $\frac{T_2}{T_3}$, $\frac{T_1}{T_4}$ and $\frac{T_3}{T_4}$ in expression for η

$$\eta = 1-\frac{1}{\gamma(r)^{\gamma-1}}\times\frac{\rho^{\gamma}-1}{\rho-1}$$

Diesel cycle normally has much higher compression ratio. For same compression ratio the efficiency decreases for increasing cut off ratio.

39. A typical indicator card obtained from a reciprocating compressor is shown in figure.

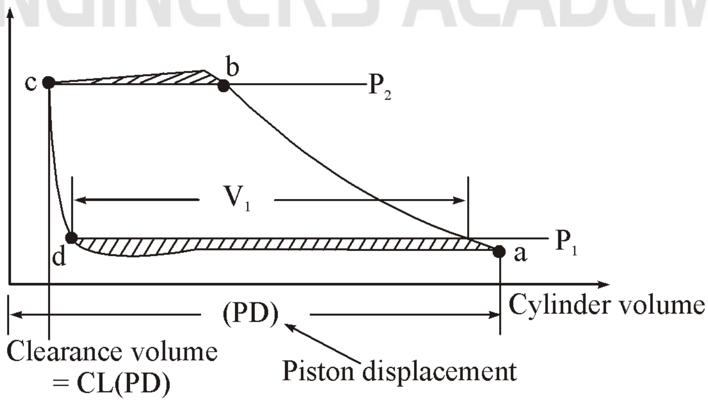


Figure : Compressor Indicator Diagram

The sequence of operation in the cylinder is as follows :

- ***Compression :***

  Starting at maximum cylinder volume, point a, slighlly below the inlet pressure p, as the volume decreases the pressure rises until it reaches p_2 at b; the discharge valve does not open until the pressure in the cylinder exceeds p_2 by enough to overcome the valve spring force.

- ***Discharge :***

  Between b and c gas flows out a pressure higher than p_2 by the amount of the pressure loss through tlie valves; at C, the point of minimum volume, the discharge valve is closed by its spring.

- ***Expansion :***

  From c to d, as the volume increases, the gas remaining in the clearance volume expands and its pressure falls; the suction valve does not open until the pressure falls sufficiently below p_1 to overcome the sprig force.

- ***Intake :***

  Between d and a gas flows into the cylinder at a pressure lower than p_1 by the amount of pressure loss through die valve.

  The total area of the diagram represents the actual work of the compressor on the gas. The cross-hatched areas of the diagram above p_2 and below p_1 represent work done solely because of pressure drop through the valves and port passages. This work is called the valve loss.

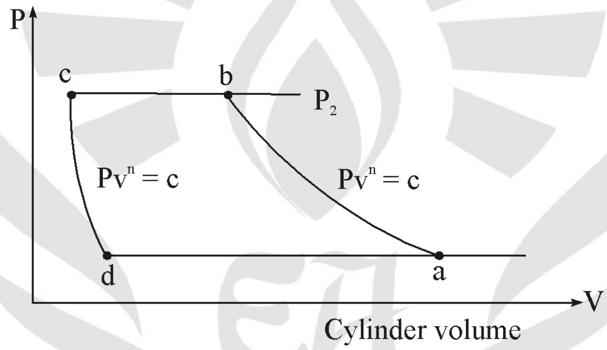


Figure : Ideal Indicator Diagram

The idealized machine to which an actual machine is compared has an indicator diagram like figure in which there are no pressure loss effects, and the processes a - b and c - d are reversible polytropic processes. Assuming no state change in the intake d - a and discharge b - c processes, and assuming equal values of the exponent n in tlie compression a - b and expansion processes c - d. the ideal work of compression can be found by taking the integral of pdv around the diagram. In m_f is the mass of fluid taken in and discharged per machine cycle, then the total work interaction per cycle is

$$w = w_{a-b} + w_{b-c} + w_{c-d} + w_{d-a}$$

$$= \frac{p_b v_b - p_a v_a}{1-n} + p_2(v_c - v_b) + \frac{p_d v_d - p_c v_c}{1-n} + p_1(v_a - v_d)$$

$$= \frac{p_2(v_b - v_c)}{1-n} - p_2(v_b - v_c) + \frac{p_1(v_d - v_a)}{1-n} - p_1(v_d - v_a)$$

$$= \frac{n}{1-n}[p_2(v_b - v_c) + p_1(v_d - v_a)]$$

$$= \frac{n}{1-n}[p_2 m_f v_2 - p_1 m_f v_1] = m_f \frac{n}{1-n}[p_2 v_2 - p_1 v_1]$$

$$= m_f \frac{n}{1-n} p_1 v_1 \left[\frac{p_2 v_2}{p_1 v_1} - 1 \right]$$

Since $\quad pv^n = \text{Constant}$

$$\frac{p_2 v_2}{p_1 v_1} = \left(\frac{p_2}{p_1} \right)^{n-1/n}$$

Substituting this in the above expression

$$w = m_f \cdot \frac{n}{1-n} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{n-1/n} - 1 \right]$$

Thus, we see that the work per kg of fluid flow is the same as obtained from the steady flow analysis (equation). It is therefore unnecessary to make any further analysis of the work of the idealized reciprocating comprressor since all desired results have already been obtained by the steady flow analysis.

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