DETAILS EXPLANATIONS

Power Plant Engineering + Heat Transfer + I.C. Engine

[PART : A]

1. In convection, heat is transferred to a flowing fluid at the surface over which it flows by combined molecular diffusion and bulk flow. Thus, convection involves conduction and fluid motion.

The rate of convective heat transfer is governed by the Newton's law of cooling.

2. A heat transfer problem can be analogized with electrical circuit problem, in which temperature difference (T) presents the potential difference and rate of heat conducted ($\dot{Q}$) is analogous to current. Thermal resistance (R_{th}) to heat flow is analogous to electrical resistance to current.

Thus $$\dot{Q} = \frac{\Delta T}{R_{th}}$$

Therefore, $$R_{th} = \frac{\Delta T}{\dot{Q}}$$

Using this definition, the expressions for thermal resistance can be derived for different shapes of bodies.

3. Prandtl number (Pr) is the property of a fluid defined as the ratio of kinematic viscosity and thermal diffusivity :

$$Pr = \frac{\nu}{\alpha} = \frac{\mu c}{K}$$

For liquid metals $Pr = 0.003 - 0.01$

Air $Pr \approx 0.7$

and for water $Pr \approx 7$

4. Emissivity of a surface is defined as ratio of emissive power of the surface at given temperature and radiation from a black body at the same given temperature :

$$\in = \frac{E}{E_b}$$

5. Shape factor takes into the account the effects of orientation on radiation heat transfer between two surfaces. It is purely a geometric quantity and is independent of the surface properties and temperature. It is also called the view factor, configuration factor, and angle factor.

6. The boiler efficiency is given by

$$\eta = \frac{215\times 150}{25\times 2000} = 64.5\%$$

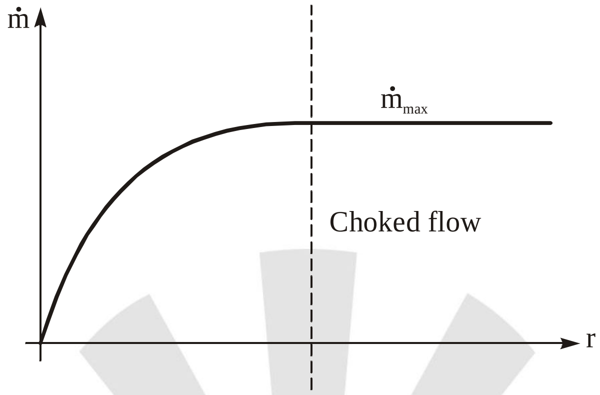
7.
- Open Heaters In an open heater or contact-type, the extracted or bled steam is allowed to mix with feed water and both leave the heater at a common temperature as seen earlier.
- Closed Heaters In a closed heater, the fluid are kept separate, and are not allowed to mix together. The heat transfer takes place through walls and tubes.

8. Steam Rate The capacity of a steam power plant is generally expressed in terms of steam rate (kg/hkW), which is defined as the rate of steam flow required to produce unit shaft output power (1 kW) :

$$\text{Stream rate} = \frac{3600}{W_t - W_p} \text{ kg/h}$$

Where turbine and pump works, W_t and W_p, are in kW.

9. The flow rate through a nozzle can be increased by reducing the back pressure, if the flow is entirely subsonic. A situation reaches when the flow velocity at the minimum cross-sectional area (throat) eventually reaches the speed of sound; Mach number becomes equal to 1. Any further lowering of the back pressure cannot accelerate the flow through the nozzle any more, because that would entail moving the point where M = 1 away from the throat, and so the flow gets stuck.



The flow pattern downstream of the nozzle can still change, but the mass flow rate is fixed to maximum. Such a condition is known as choked flow.

10. To quantify the share of power produced in moving blades, degree of reaction (R) is defined as the ratio of enthalpy drop in moving blades and total enthalpy drop :

$$R = \frac{\Delta h_s \text{in moving blade}}{\Delta h_s \text{in all stages}}$$

Impulse turbines do not involve reaction, therefore, R = 0. Person's turbine is designed for 50% degree of reaction.

11. To produce more power, additional steam can be admitted through a bypass valve to the later stages of the turbine. By-pass governing operates in a turbine that is throttle governed.

12. Idling System With partially closed throttle, the limited air flow causes insufficient depression to draw fuel through the main fuel nozzle. Thus, an idling tube is provided to directly connect the downstream of venturi to the float chamber through which the fuel gets discharged directly into the engine intake due to low pressure on the downstream. The fuel in the idling tube is drawn by the air flow taken from upstream. This constitutes the idling system, which comes into operation during starting, idling range, and low speed operation.

13. Morse Test The Morse test is applicable for multi-cylinder engines. In this method, the cylinders of the engine are made inoperative and the reduction in brake power is noted. It is assumed that pumping and friction losses are the same as when the cylinder is inoperative as well as during firing.

14. Equivalence ratio (ϕ) is defined as the ratio of actual fuel–air ratio to the stoichiometric fuel–air ratio. Quality of fuel–air mixture is assessed in terms of ϕ:

$$\phi > 1 \text{ Rich mixture} = 1$$

$$\text{Stoichiometric mixture} < 1 \text{ Lean mixture}$$

The inverse of equivalence ratio is called relative fuel–air ratio, denoted by λ.

15.
- Dry Analysis The sample of gaseous combustion product is usually cooled down to a temperature, which is below the saturation temperature of the steam present. Therefore, the steam content is not included in the analysis. Such analysis is quoted as dry analysis of the products.
- Wet Analysis Since the products are gaseous, it is usual to quote the analysis by volume. When steam is included in the exhaust, the analysis is called wet analysis.

16. The compression ratio is $r = \dfrac{v_c + v_s}{v_c} = 11$

Efficiency of otto cycle : $\eta_{Otto} = 1 - \dfrac{1}{r^{\gamma-1}} = 61.6785\%$

17. Given, that

$v_s = 0.0259\ m^3$

$P = 950\ kW$

$N = 2200\ rpm$

If p_m is mean effective pressure then for four-stroke engines, then

$$P = \frac{p_m v_s N}{2 \times 60}$$

$p_m = 2.0007\ MPa$

18. Given that

$r = 6\ cm$

$d = 8\ cm$

$L = 2r$

Swept volume is $v_s = \dfrac{\pi d^2}{4} \times 2r = \dfrac{\pi 8^2}{4} \times 2 \times 6 = 603.186\ cm^3$

19. The net heat supplied is given by

$$Q = 4\ (100 - 30) + 2400 \times 0.9 = 2440\ kJ/kg$$

20. Given that brake power of engine is 9 kW. Indicated power of individual cylinders is

$P_{i1} = 9 - 4.25 = 4.75\ kW$

$P_{i2} = 9 - 3.75 = 5.25\ kW$

Hence, mechanical efficiency is $\eta_m = \dfrac{9}{4.75 + 5.25} = 90\%$

[PART : B]

21. ***Counter Flow :***

In counter flow heat exchangers, the two fluids flow in opposite directions. Each of the fluids enter the heat exchanger from opposite ends. Because the cooler fluid exits the counter flow heat exchanger at the end where the hot fluid enters the heat exchanger, the cooler fluid will approach the inlet temperature of the hot fluid.

Cross Flow :

In cross flow heat exchangers, one fluid flows through tubes and the second fluid passes around the tubes at 90^o angle. Cross flow heat exchangers are usually found in applications where one of the fluids change their states (2-phase flow), such as condenser of steam turbine.

22. To quantify the number heat units being transferred between the fluids in a heat exchanger, the number of transfer units (NTU) is defined as

$$NTU = \frac{UA}{C_{min}}$$

This dimensionless group can be viewed as a comparison of the heat capacity of the heat exchanger, expressed in W/K, with the heat capacity of the flow. For convenience sake in the heat exchanger analysis, heat capacity ratio can be defined as

$$r = \frac{C_{min}}{C_{max}}$$

Effectiveness is a function of NTU and r :

$$\varepsilon = f(\text{NTU}, r)$$

The value of heat capacity ratio r can vary from 0 to 1. For the given value of NTU, the effectiveness is maximum when r = 0 (phase-change process) and minimum when r = 1.

The expression for ε are derived for the two types of heat exchangers in the following sections.

23. Maximum heat dissipation occurs when insulation is laid upto critical radius.

For cylindrical (wire) elements :

$$r_c = \frac{\kappa}{h_o} = \frac{0.6}{25} = 24 \text{ mm}$$

Thus, insulation thickness should be $t = 24 - \frac{18}{2} = 15$ mm

24. Given that,

$$\delta = 0.75 \text{ mm}$$
$$\mu = 30 \times 10^{-6} \text{ Pa.s}$$
$$c = 1.8 \times 10^3 \text{ J/kgK}$$
$$\kappa = 0.05 \text{ W/mK}$$

Prandtl number is calculated by $Pr = \frac{\mu c}{K} = 1.08$

Using

$$\frac{\delta}{\delta_t} = Pr^{1/3}$$

$$\delta_t = \frac{\delta}{Pr^{1/3}} = 0.73 \text{ mm}$$

25. Given that

$$r_i = 0.1 \text{ m}$$
$$r_0 = 0.16 \text{ m}$$
$$\kappa = 55 \text{ W/mK}$$
$$T_i = 325^\circ C$$
$$T_o = 95^\circ C$$

Thermal resistance of the sphere is

$$R_{th} = \frac{r_o - r_i}{4\pi r_o r_i \times K} = 4.52^\circ C/kW$$

The rate of heat transfer is given by

$$\dot{Q} = \frac{T_i - T_o}{R_{th}} = 42.39 \text{ kW}$$

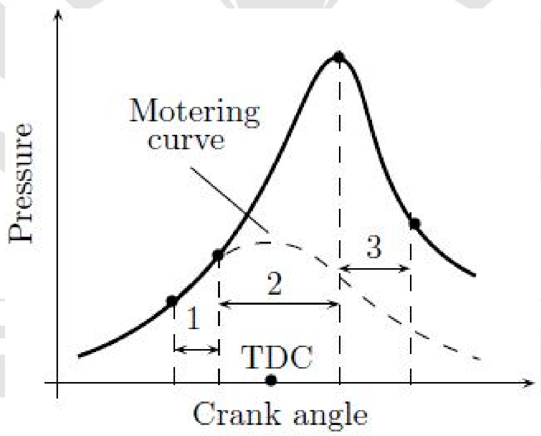
26.
- Noise and Wear To avoid noise and wear of valve surfaces, the clearance between cam, tappet, and valve must be slowly lifted out
- Ram Effect When the piston reaches BDC and starts to move in compression stroke, the inertia of the entering fresh charge tends to cause it to continue to move into cylinder. Therefore, the intake valve is closed after TDC so that maximum air is taken in. This is called ram effect.

- Energy Saving Opening of exhaust valve earlier to BDC reduces the pressure near the end of power stroke and loss of power occurs, but work required to expel the exhaust is reduced.
- Valve timing is often represented in polar diagram, known as valve timing diagram, in which rotation of crankshaft is the angular coordinate with assumption of constant angular speed.

27.
- Idling Range During idling, the engine operates at no load. However, the engine requires very less amount of fresh charge (throttle is nearly closed up to 20% load) to work against friction and operate the accessories. The less amount of fresh charge has two effects :
  (i) Mixing of larger portion of exhaust gases with the fresh charge.
  (ii) Backward flow of exhaust gases into intake manifold at the start of suction stroke.
  Thus, idling range requires richer mixture to avoid possibility of poor combustion.
- Cruising Range: In cruising range, the main interest of design is to achieve maximum fuel economy. This is possible at the optimum fuel– air ratio for any given engine speed which can develop the required torque with smooth and reliable operation.
- Power Range Maximum power range requires richer mixture to
  (i) Produce maximum power from the fuel intake
  (ii) Prevent over heating of exhaust valve by reducing the flame temperature
  (iii) Reduce detonation by reducing the flame temperature

28. Combustion in SI engines occurs in the following three stages

1. Ignition lag (low rate and increasing front speed)
2. Propagation of flame (from the first rise in pressure)
3. After burning (low rate and decreasing front speed)



Spontaneous ignition or auto ignition occurs when the temperature of the unburnt mixture at any point exceeds the self ignition temperature of the fuel during the period of pre-flame reactions (ignition delay). The uneven pressure rise causes noise and disturbs the operation of engine by reducing the mean effective pressure and efficiency. This phenomenon is called knocking or detonation. This can be avoided by selecting fuel of high auto ignition temperature and with long ignition delay period. Any factor which reduces the temperature of unburnt charge should reduce the possibility of detonation. This is possible by reducing compression ratio, mass of inducted charge, inlet temperature and pressure, temperature of combustion chamber, and by retarding the spark timing (having spark closer to TDC). Any factor which increases flame speed will also reduce knock. Flame speed can be increased by increasing turbulence, fuel-air ratio, temperature and pressure at inlet, compression ratio, engine output and speed.

29. Given that

$$d = 2.5 \text{ m}$$
$$N = 2500 \text{ rpm}$$
$$\alpha = 22.5^o$$

The optimum velocity of steam V_1 (at nozzle exit) for maximum efficiency is given by

$$\frac{u}{V_1} = \frac{\cos\alpha}{2}$$

$$V_1 = \frac{2u}{\cos\alpha}$$

$$= \frac{2\pi dN}{60 \times \cos\alpha}$$

$$= 708.42 \text{ m/s}$$

30. Given that

$$\eta_{bth} = 0.35$$
$$CV = 40 \times 10^3 \text{ kJ/kg}$$
$$h_m = 0.75$$

Brake specific fuel consumption is

$$BSFC = \frac{3600}{\eta_{bh} \times CV} = 0.257 \text{ kg/h-kW}$$

Hndicated specific fuel consumption is

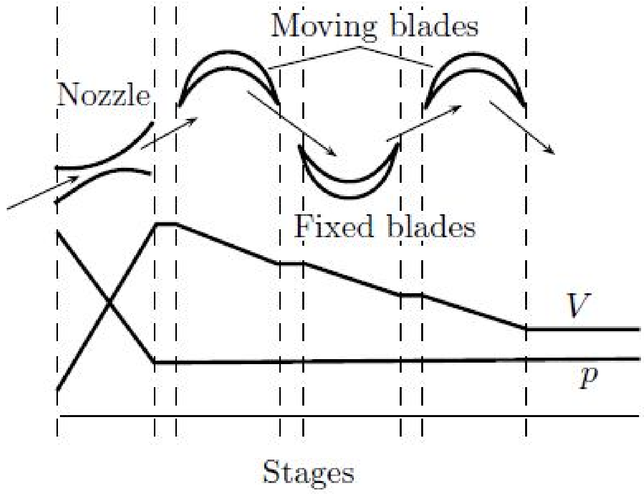
$$ISFC = \frac{\eta_m \times 3600}{\eta_{bth} \times CV} = 0.193 \text{ kg/h-kW}$$

31. Knock rating determines whether or not a fuel will knock in a given engine under the given operating conditions. Hydrocarbons of extended chain-like structure oxidize easily and knock readily and hard, whereas those of clustered structure iodize less easily and do not knock so readily.

Octane Rating Iso-octane (C_8H_{18}) has low boiling point and possesses a very good antiknock quality. On the other hand, n-heptane (n-C_7H_{16}) has very poor antiknock qualities. Octane rating is used for fuels of SI engines. This is equivalent percentage of isooctane with n-heptane. This rating is also represented as octane number (ON), which is scaled from 100 to 0. Since the motor method of determining the octane number uses more severe operating conditions than those in research method, the motor octane number (MON) is found lower than the research octane number (RON). The difference MON and RON is called fuel sensitivity. In this reference, a term, antiknock index is defined as the average of RON and MON.

Cetane Rating Cetane rating, used for fuels of CI engines, is the percentage of cetane ($C_{16}H_{34}$, hexadecane) with isocetane (hepta-methylnonane) to make equivalent fuel with respect to knocking. Lower the cetane number, higher is hydrocarbon emission and noise and leads to increased amount of white smoke. Normal paraffins C_nH_{2n+2}, which are straight-chain compounds, have the highest cetane number and lowest specific gravity. An unnecessary higher octane number in SI engines is a waste, but not harmful in operation; whereas an unnecessary cetane number in CI engines can induce preignition.

32. In velocity compounding, the pressure drop of steam takes place in a single row of nozzles. The resultant kinetic energy of steam is absorbed by the wheel in a number of rows of moving blades with guide blades in between two such rows



The Curtis stage turbine is composed of one stage of nozzles as the single-stage turbine, followed by two rows of moving blades instead of one. These two rows are separated by one row of fixed blades attached to the turbine stator, which has the function of redirecting the steam leaving the first row of moving blades to the second row of moving blades. For maximum efficiency of Curtis turbines, the ratio of work in different stages is as follows.

(i) For two stages, $W_1 : W_2 = 3 : 1$

(ii) For three stages, $W_1 : W_2 : W_3 = 5 : 3 : 1$

[PART : C]

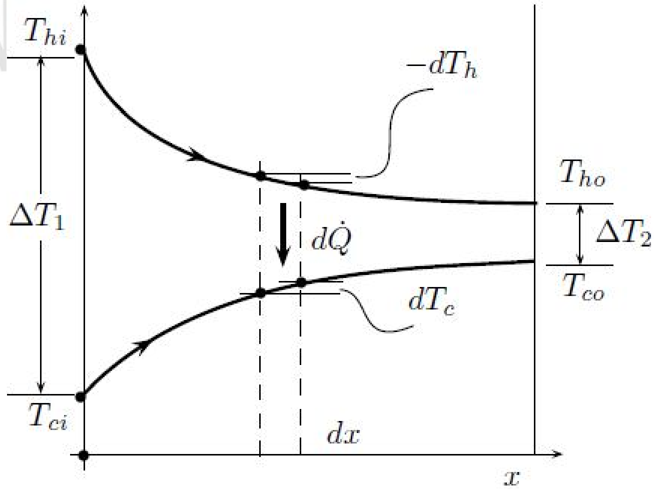
33. ***Parallel Flow :***

In parallel flow heat exchanger, the temperature of hot fluid reduces towards direction of flow, while that of cold fluid increases. Therefore,

(i) Hot Fluid $-C_h dT_h = UdA(T_h - T_c)$

$$dT_h = -\frac{UdA\Delta T}{C_h}$$

$$\dot{Q} = C_h(T_{hi} - T_{ho})$$



(ii) Cold Fluid

$$d\dot{Q} = C_c dT_c$$

$$dT_c = \frac{UdA\Delta T}{C_c}$$

$$\dot{Q} = C_c(T_{co} - T_{ci})$$

The temperature difference along the fluid flow is

$$d(\Delta T) = dT_h - dT_c$$

Using Equation

$$d(\Delta T) = -UdA\Delta T\left(\frac{1}{C_h}+\frac{1}{C_c}\right)$$

Integrating both sides of above equation,

$$\int_1^2 \frac{d(\Delta T)}{\Delta T} = U\left(\frac{1}{C_h}+\frac{1}{C_c}\right)\int_1^2 dA$$

$$\ln\left(\frac{\Delta T_2}{\Delta T_1}\right) = -U\left(\frac{1}{C_h}+\frac{1}{C_c}\right)A$$

where, ΔT_1 and ΔT_2 are the temperature differences at inlet and outlet, respectively.

$$\ln\left(\frac{\Delta T_2}{\Delta T_1}\right) = -UA\left\{\frac{(T_{hi}-T_{ho})}{\dot{Q}}+\frac{-(T_{ci}-T_{co})}{\dot{Q}}\right\}$$

$$= -\frac{UA}{\dot{Q}}\{(T_{hi}-T_{ho})-(T_{ci}-T_{co})\}$$

$$= -\frac{UA}{\dot{Q}}(\Delta T_1 - \Delta T_2)$$

$$\dot{Q} = UA\times\frac{\Delta T_2-\Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} = UA \times LMTD$$

Where,

$$LMTD = \frac{\Delta T_2-\Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$$

This is the expression of LMTD for parallel flow heat exchangers.

34.

$$\theta = C_1e^{-mx} + C_2e^{mx}$$

Infinitely Long Fins :

For a very long fin ($x \to \infty$), the temperature at free end will be equal to the ambient temperature ($\theta \to 0$). Using this feature as the second constraint, is written as

$$0 = \frac{C_1}{e^{m\times\infty}} + C_2e^{m\times\infty}$$

$$C_2 = 0$$

Using $C_1 = \theta_0$

Equation takes the following form :

$$\theta = \theta_0 e^{mx}$$

This is the expression of temperature profile. Using this the rate of heat transfer for long fin can be determined as

$$\dot{Q} = KA\left(\frac{\partial T}{\partial x}\right)_{x=0} = -KA \times \theta_0 m e^{m\times 0}$$

$$= -KAm(T_o - T_\infty) = \sqrt{hPKA}(T_0 - T\infty)$$

35. From Tds Equation for nozzle flow–VdV= $v$dp

$$-VdV = v\,dp$$

$$-\int VdV = -\int v\,dp$$

For isoentorpic expression $pv^n = c$

When n is the expansion index of steam, given by

$$n = \begin{cases} 1.035 + \dfrac{x}{10} & \text{Wet steam} \\ 1.135 & \text{Saturated steam} \\ 1.3 & \text{Superheated steam} \end{cases}$$

Therefore, $v = \left(\frac{C}{p}\right)^{1/n}$

$$-\int_1^2 VdV = \int_1^2 \left(\frac{C}{p}\right)^{1/n} dp$$

$$\frac{V_2^2 - V_1^2}{2} = \frac{n}{n-1}(p_2 v_2 - p_1 v_1)$$

$$V_2 = \sqrt{\frac{2n}{n-1}(p_1 v_1 - p_2 v_2) - V_1^2}$$

The inlet velocity can be assumed to be negligible ($V_1 \approx 0$)

$$V_2 = \sqrt{\frac{2n}{n-1} p_1 v_1 \left(1 - \frac{p_2}{p_1}\right)^{\frac{n-1}{n}}}$$

Specific volume at exit (2) is $v_2 = v_1\left(\frac{p_1}{p_2}\right)$

Denoting pressure ratio $\frac{p_2}{p_1}$ by r, the mass flow rate is given by

$$\dot{m} = \frac{A_2}{v_2}V_2 = A_2\sqrt{\frac{2n}{n-1}\frac{p_1}{v_1}\left\{r^{\frac{2}{n}} - r^{\frac{n+1}{n}}\right\}}$$

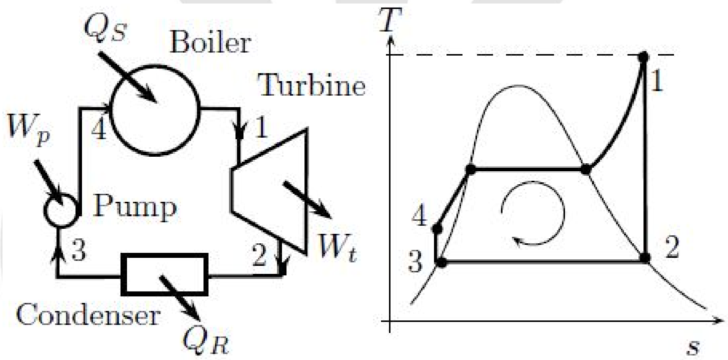
For maximum discharge $\frac{d\dot{m}}{dr} = 0$

Solving the above equation for r,

$$T_c = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

It is called critical pressure ratio.

36. Rankine cycle is the ideal cycle for steam power plants.Simple Rankine cycle consists of the following reversible processes.



1. Isentropic expansion (1 ’& 2) of steam in turbine

2. Isobaric cooling (2 ’& 3) of steam in condenser

3. Isentropic pumping (3 ’& 4) of saturated liquid (water) to boiler,

4. Isobaric heating (4 ’& 1) in boiler.

Cycle Efficiency :

Applying the steady flow energy equation to each of the processes on the basis of unit mass of working substance and neglecting changes in kinetic energy and potential energy, the heat supplied and the heat rejected are found as

$$Q_S = h_1 - h_4$$
$$Q_R = h_2 - h_3$$

Work done by the steam on the turbine, and the work done by pump on the steam, respectively, are

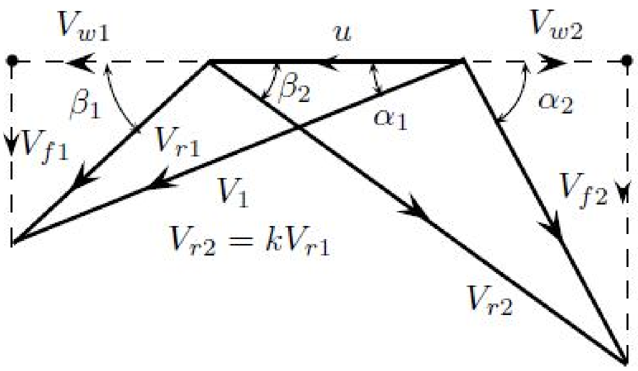
$$W_t = h_2 - h_1$$
$$W_p = h_4 - h_3$$

Therefore, the efficiency of Rankine cycle is

$$\eta_{Rankine} = \frac{W_{net}}{Q_s} = \frac{W_t - W_p}{Q_s}$$

$$= \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_4)} = 1 - \frac{h_2 - h_3}{h_1 - h_4}$$

37.



Referring to figure the velocity change in tangential direction is written as

$$V_{w1} + V_{w2} = V_{r1}\cos\beta_1 + V_{r2}\cos\beta_2$$

$$= V_{r1}\cos\beta_1\left(1 + k\frac{\cos\beta_2}{\cos\beta_1}\right)$$

$$= V_{r1}\cos\beta_1 (1 + kB)$$

$$= (V_1\cos\alpha_1 - u)(1 + kB)$$

Power developed per kg is

$$P = (V_{w1} + V_{w2}) \times u = (V_1\cos\alpha_1 - u)(1 + kB)u$$

Blade efficiency is
$$\eta_b = \frac{(V_1\cos\alpha_1 - u)(1 + kB) \times u}{\frac{V_1^2}{2}} = 2(1 + kB)(\cos\alpha_1 - \rho)\rho$$

Where $\rho = \frac{u}{V_1}$ is the velocity ratio. For maximum or minimum efficiency,

$$\frac{d\eta_b}{dp} = 0$$

Which results in
$$\rho = \frac{\cos\alpha}{2}$$

At this speed ratio, the maximum blade efficiency is found as

$$(\eta_b)_{max} = (1 + kB)\frac{\cos^2\alpha}{2}$$

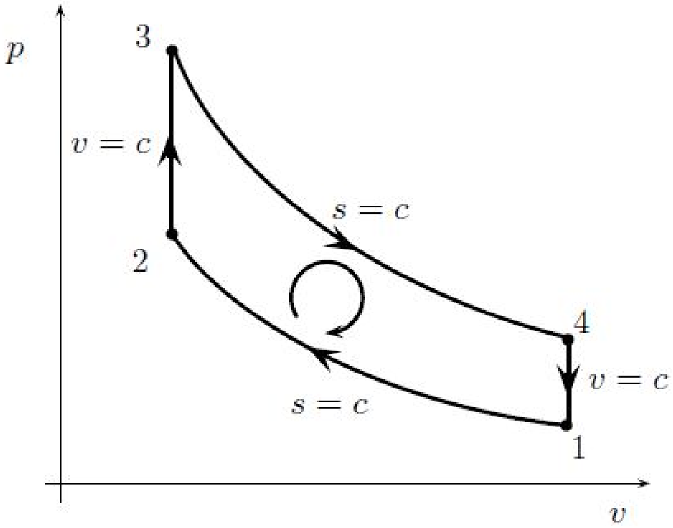
38. Having constant volume heat addition, the Otto cycle forms the basis for the working of today’s spark engines. The Otto cycle is formed by replacing both isothermal processes of Carnot cycle by two isochoric processes. Thus, the cycle consists of following four processes :

1. Adiabatic compression (1 ’& 2)

2. Isochoric heat addition (2 ’& 3)

3. Adiabatic expansion (3 ’& 4)

4. Isochoric heat rejection (4 ’& 1)



In this cycle, for isentropic processes $1 \rightarrow 2$ and $3 \rightarrow 4$,

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{(\gamma-1)}$$

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{(\gamma-1)}$$

But $v_2 = v_3$ and $v_4 = v_1$, therefore

$$\frac{v_1}{v_2} = \frac{v_4}{v_3} = r(\text{say})$$

Where, r is called the compression ratio (in the first insentropic process). Hence,

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} = r^{(\gamma-1)} = x(\text{say})$$

Similarly, in processes $1 \rightarrow 2$ and $3 \rightarrow 4$,

$$\frac{p_2}{p_1} = \frac{p_3}{p_4} = r^{\gamma}$$

and in $2 \rightarrow 3$ and $4 \rightarrow 1$, $\quad \frac{p_3}{p_2} = \frac{p_4}{p_1} = r^{\gamma}$

where, r_p is called pressure ratio. Therefore,

$$p_2 = r^{\gamma} p_1$$

$$p_3 = r_p p_2 = r_p r^{\gamma} p_1$$

$$p_4 = r_p p_1$$

Using the $p - v$ diagram of the cycle, the following quantities can be determined :

1. Heat Supplied : Heat is supplied to the cycle during isochoric process $2 \rightarrow 3$:

$$Q_S = mc_v(T - T_2)$$

2. Heat Rejected : Heat is rejected by the cycle during isochoric process $4 \rightarrow 1$:

$$Q_R = mc_v(T_4 - T_1)$$

3. **Cycle Efficiency :** Efficiency of the Otto cycle is determined as :

$$\eta_{Otto} = 1 - \frac{Q_R}{Q_S} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_4 - T_1}{xT_4 - xT_1} = 1 - \frac{1}{x}$$

$$\eta_{Otto} = 1 - \frac{1}{r^{(\gamma-1)}}$$

This equation can be used to plot η_{Otto} with respect to r and g, as depicted in figure.

39.

$$\text{LMTD} = 20^oC$$

$$T_{hi} = 100^oC$$

$$T_{ci} = 20^oC$$

$$\dot{m}_c = 2\dot{m}_h$$

$$c_h = 2c_c$$

For counter flow heat exchangers,

$$\text{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$$

where,

$$\Delta T_2 = T_{hi} - T_{co}$$

$$\Delta T_1 = Th_o - T_{ci}$$

$$\Delta T_2 > \Delta T_1$$

With LMTD definition, the volume of ΔT_1, ΔT_2 can be determined. Using

$$\dot{m}_c c_c(T_{co} - T_{ci}) = \dot{m}_h c_h(T_{hi} - T_{ho})$$

$$2\dot{m}_h c_c(T_{co} - T_{ci}) = \dot{m}_h 2c_c(T_{hi} - T_{ho})$$

One finds

$$T_{co} - T_{ci} = T_{hi} - T_{ho}$$

$$T_{co} + T_{ho} = T_{hi} + T_{ci} = 120$$

$$T_{ho} = 120 - T_{co}$$

Putting this value into LMTD equation gives

$$T_{co} = 68.36^oC$$

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