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ESE - 2018

MAINS EXAMINATION

Questions with Detailed Solutions

MECHANICAL ENGINEERING

PAPER - I

All Queries related to ESE - 2018 Mains Solutions are to be sent to the following email address
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MECHANICAL ENGINEERING

PAPER – I _ REVIEW

Except one out of scope question from *Fluid Mechanics,* remaining questions in the paper can be easily attempted. Particularly in this paper selection of Questions plays a vital role in securing a good score. For example Section – B is relatively tougher than Section – A, so choosing 3 questions from Section – A will fetch you a big advantage.

SUBJECT WISE REVIEW

Subjects	Level	Marks
Basic thermodynamics	Easy	32
Refrigeration & Air conditioning	Easy	52
IC Engine	Easy to tough	62
Power plant	Easy to moderate	102
Renewable sources of Energy	Easy to tough	84
Heat transfer	Easy to tough	64
Fluid Mechanics & Hydraulic Machines	Easy to tough	84

Getting 180 to 210 marks is a great achievement inview of time constraints and QCAB.

Subjects Experts,
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SECTION – A

01(a). Discuss the sources of minor losses which can take place in circular pipes. (12 M)

01(a).

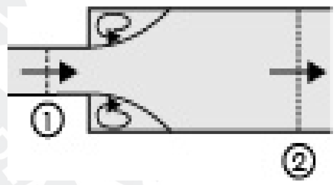
Sol: Sources of minor losses in a circular pipe are:

1. *Loss due to sudden expansion ($h_{L,exp}$)*

   Due to sudden change in cross-sectional area flow separates at the junction which leads to flow separation and subsequent loss.

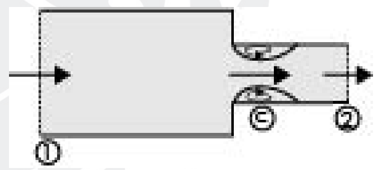
   The head loss due to sudden expansion is given by Borda-Carnot equation.

$$h_{L,exp} = \frac{(V_1 - V_2)^2}{2g}$$



2. *Loss due to sudden contraction ($h_{L,\,cont}$)*

   The sudden contraction causes convergence of the flow. The streamlines keep on converging due to inertia upto a section called vena contracta. The flow upto vena contracta has negligible losses as convergence process is efficient.



   Most of the loss occurs after vena contracta where flow expands. The loss during this process can be calculated by applying formula of sudden expansion.

$$h_{L,cont} = \frac{(V_C - V_2)^2}{2g} = \frac{V_2^2}{2g}\left(\frac{V_C}{V_2} - 1\right)^2 = \frac{V_2^2}{2g}\left(\frac{A_2}{A_C} - 1\right) = \frac{V_2^2}{2g}\left(\frac{1}{C_C} - 1\right)^2$$

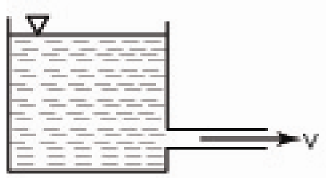
   where, $C_C = \dfrac{A_C}{A_2} = \text{Contraction coefficient.}$

3. *Loss at pipe entrance ($h_{L,ent}$):*

   At sharp entrance of pipe, the flow separates as it cannot change its direction suddenly

$$h_{L,ent} = K_{L,ent}\frac{V^2}{2g}$$

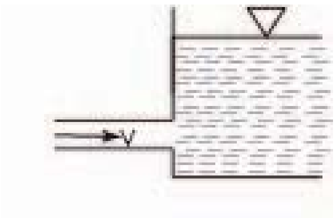
   where $K_{L,ent} = 0.5$ for sharp edged entrance.



4. *Loss at pipe exiting into reservoir ($h_{L,exit}$):*

If pipe is discharging into a reservoir which contains liquid then kinetic energy of the fluid in the pipe is completely lost when it mixes with liquid in the reservoir.

$$\therefore \quad h_{L,exit} = \frac{V^2}{2g}$$

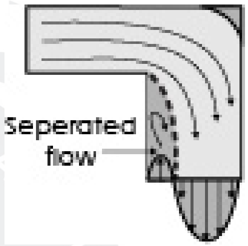


5. *Loss at pipe bend ($h_{L,Bend}$):*

When flow turns suddenly around bend it has tendency to separate at inner radius.

$$h_{L,Bend} = K_{L,Bend}\frac{V^2}{2g}$$

$K_{L,Bend}$ depend upon bend radius.



6. *Losses due to miscellaneous pipe fittings:*

The loss can be present due to pipe fittings such as valves, elbows, tees, and flow measuring devices installed in the pipe.

$$h_L = K_L\frac{V^2}{2g}$$

K_L depends upon type of fitting.

01(b). Calculate the decrease in available energy when 25 kg of water at 95°C mixed with 35 kg of water at 35°C, the pressure being taken as constant and the temperature of the surroundings being 15°C. (Specific heat of water = 4.2 kJ/kg.K) (12 M)

01(b).

Sol: Let the final temperature of mixture = T

By energy balance :

$25 \times c_p \times (95 - T) = 35 \times c_p \times (T - 35)$

$2375 - 25T = 35T - 1225$

$60T = 3600$

$T = 60°C$

Total entropy change

$\Delta S = (\Delta S)_1 + (\Delta S)_2$

$$= 25\times4.2\times\ell n\left(\frac{333}{368}\right)+35\times4.2\times\ell n\left(\frac{333}{308}\right)$$

$= -10.5 + 11.47$

$= 0.972$ kJ/K

So, the decrease in available energy = Increase in irreversibility

$= T_o\,\Delta S$

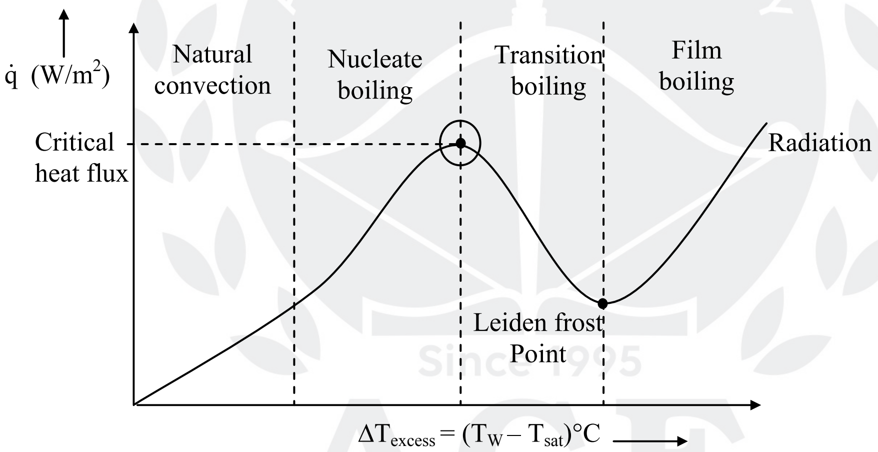
$= (273+15)\times0.972$

$= 280$ kJ

01(c). Draw a typical boiling curve for pool boiling of water at saturation temperature and atmospheric pressure, and mark each boiling regime. (12 M)

01(c).

Sol: Typical boiling curve for water at atmospheric pressure :



01(d). A diesel engine is working with a compression ratio of 18:1 and expansion ratio of 12:1. Calculate the air-standard cycle efficiency. Assume γ = 1.4. If the relative efficiency of the engine is 50% and calorific value of diesel fuel is 45000 kJ/kg, find out the specific fuel consumption of the engine in kg/kWh. If this engine has its application for DG set purpose of 500 kW rating at full-load condition and is expected to operate for two hours every day, work out the inventory requirement of diesel for next 15 days. Also work out fuel cost of diesel for 15 days period if cost of fuel per litre is ₹ 60. Make suitable assumptions if required. Consider diesel density as 0.83 kg/litre. (12 M)

01(d).

Sol: Compression ratio = r_k = 18

Expansion ratio = r_E = 12

Cut off ratio = $r_C = \frac{r_R}{r_E} = \frac{18}{12} = 1.5$

$\gamma = 1.4$

$$\eta_{\text{air std}} = 1 - \frac{1}{\gamma \times r_k^{\gamma-1}}\left[\frac{r_C^{\gamma}-1}{r_C-1}\right]$$

$$= 1 - \frac{1}{(1.4)(18)^{1.4-1}}\left[\frac{1.5^{1.4}-1}{1.5-1}\right] = 0.6564 \text{ or } 65.64\%$$

$$\eta_{\text{relative}} = \frac{\eta_{\text{Br.Th}}}{\eta_{\text{air std}}}$$

$$0.5 = \frac{\eta_{\text{Br.Th}}}{0.6564}$$

$\eta_{\text{Br.Th}} = 0.3282$

$$= \frac{3600}{\text{bsfc}(\text{kg/kWhr}) \times \text{CV}(\text{kJ/kg})}$$

$$0.3282 = \frac{3600}{(\text{bsfc}) \times 45000}$$

$$\text{bsfc} = \frac{3600}{0.3282 \times 45000} = 0.2438\,\text{kg/kWhr}$$

Power rating of DG set at full load = 500 kW

Fuel requirement for its operation = $\text{bsfc}\left(\frac{\text{kg}}{\text{kWhr}}\right) \times \text{BP}(\text{kW})$ = 0.2438 × 500

$$\dot{m}_f = 121.9\,\text{kg/hr}$$

Fuel requirement for 15 days = $\text{No.of days} \times \frac{\text{Hours}}{\text{day}} \times \dot{m}_f\left(\frac{\text{kg}}{\text{hr}}\right)$

$$m_f = 15 \times 2 \times 121.9 = 3657\,\text{kg}$$

Fuel requirement for 15 days in litres, $V_f = \frac{m_f}{\text{diesel density}}$

$$V_f = \frac{3657}{0.83} = 4406.024\,\text{litres}$$

Fuel cost for 15 days = V_f × cost/litre = 4406.024 × 60 = ₹ 2,64,361.45

01(e). A natural draught cooling tower used in a large cold storage plant receives water from the condenser outlet at a flow rate of 35000 kg/s and 40°C temperature. The ratio of flow rate of water to air is 1.2:1 in the cooling tower. Inlet condition of the air entering the cooling tower is dry-bulb temperature (DBT) of 20°C and wet-bulb temperature (WBT) of 10°C. Air leaves the cooling tower at DBT of 35°C with relative humidity of 90%. For this cooling tower

(i) draw the inlet and exit conditions of air in psychrometric chart and name the process;

(ii) determine the rate of evaporation of water in kg/s;

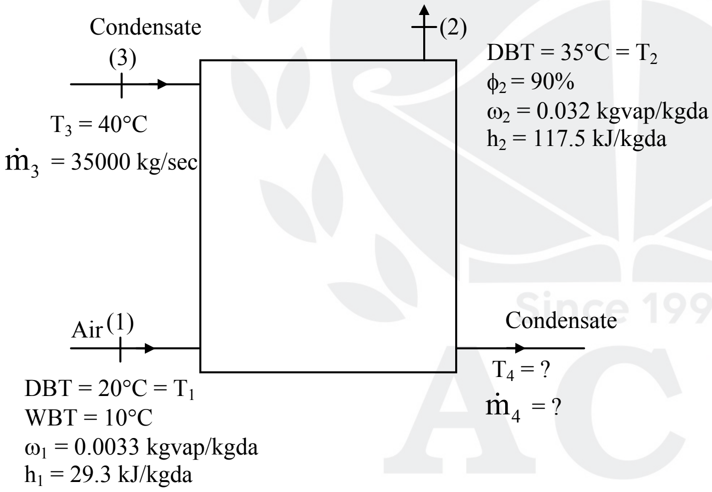
(iii) determine the heat carried away by the air;

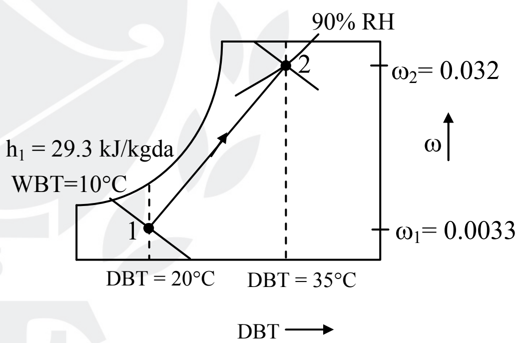
(iv) determine the maximum possible temperature drop of water realizable.

[Psychrometric Chart is placed at the end] (12 M)

01(e).

Sol:





Mass flow rate of water :

$$\frac{\text{mass flow rate of water}}{\text{mass flow rate of air}} = \frac{\dot{m}_w}{\dot{m}_a} = 1.2$$

The process is heating and humidification

$$\dot{m}_a = \text{mass flow rate of air} = \frac{\dot{m}_w}{1.2}$$

$$= \frac{35000}{1.2} = 29166.67 \text{ kg/sec}$$

Heat carried away by air = $\dot{m}_a$ (kg/sec)× ($h_2 - h_1$) (kg/kgda)

$$= 29166.67\ (117.5 - 29.3)$$
$$= 2572500.3\ \text{kJ/sec}$$
$$= 2572.5\ \text{MJ/sec}$$

Rate of evaporation of water = $\dot{m}_a$ (kg/sec) ($\omega_2 - \omega_1$) kg/kgda

$$= 29166.67 \times (0.032 - 0.0033)$$
$$= 837.08\ \text{kg/sec}$$

Maximum possible drop in temperature of water

$$= T_3 - (WBT)_{\text{inlet air}} = 40 - 10 = 30°C$$

02(a). Glycerine is pumped at a constant rate of 20 litres/s through a straight, 100 mm diameter pipe, 45 m long, inclined at 15° to the horizontal. The gauge pressure at the lower inlet end of the pipe is 590 kPa. Verify that the flow is laminar and calculate the pressure at the outlet end of the pipe and the average shear stress at the wall. (Relative density of glycerine = 1.26 and dynamic viscosity of glycerine = 0.9 Pa s) (20 M)

02(a).

Sol: Given: Fluid is Glycerine

Relative density = 1.26

$\mu = 0.9$ Pa.S

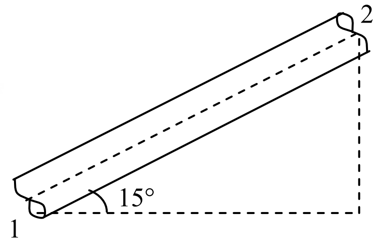
$Q = 20$ litres/s $= 20\times10^{-3}$ m^3/s

Pipe:

Diameter, $D = 100$ mm $= 0.1$ m

Length, $L = 45$ m

Inclined at 15°to horizontal.



$(P_1)_{\text{gauge}}$ (lower inlet) = 590 kPa

Reynolds number,

$$Re = \frac{\rho VD}{\mu}$$

$$= \frac{\rho D}{\mu}\frac{4Q}{\pi D^2}$$

$$= \frac{4\rho Q}{\pi \mu D}$$

$$= \frac{4\times1.26\times10^3\times20\times10^{-3}}{\pi\times0.9\times0.1} = 356$$

$\Rightarrow$ Flow is laminar $\rightarrow$ verified

Applying energy equation between sections (1) and (2), we get

$$\frac{P_1}{\gamma_{gl}} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\gamma_{gl}} + \frac{V_2^2}{2g} + z_2 + h_L$$

$V_1 = V_2$, $z_1 = 0$, $z_2 = 45\sin15° = 11.647$ m

and $h_L = \dfrac{128\mu QL}{\pi\times\gamma_{gl}\times D^4}$

$P_1 = 590$ kPa (gauge) (Given)

Substituting the values given :

$$\frac{590}{1.26\times9.81} + 0 = \frac{P_2}{1.26\times9.81} + 11.647 + \frac{128\times0.9\times20\times10^{-3}\times45}{\pi\times1.26\times10^3\times9.81\times(0.1)^4}$$

$$= \frac{P_2}{1.26\times9.81} + 11.647 + 26.7 \quad \text{(let } P_2 \text{ be in kPa)} \quad \ldots\ldots\ldots\ldots \text{(i)}$$

$$47.732 = \frac{P_2}{1.26\times9.81} + 38.347$$

$\Rightarrow$ $(P_2)_{gauge} = 9.385 \times 1.26 \times 9.81 = 116$ kPa

Average shear stress at the wall,

$$\tau_{wall} = -\frac{dP^*}{ds}\times\frac{R}{2}$$

where dP^* is the difference in pressure due to difference of piezometric head, $\left(\dfrac{P_2}{\gamma_{gl}} + z_2\right) - \left(\dfrac{P_1}{\gamma_{gl}} + z_1\right)$

From the equation (i) we can see that

$$\frac{dP^*}{\gamma_{gl}} = -26.7\,\text{m}$$

Thus, $\tau_{wall} = \dfrac{26.7\times1260\times9.81}{45}\times\dfrac{0.05}{2} = 183.3$ N/m^2

(OR)

$$\tau = -\mu\frac{du}{dr} = -\mu\frac{d}{dr}\left\{u_{max}\left(1-\frac{r^2}{R^2}\right)\right\}$$

$$\tau = \frac{2\mu u_{max} r}{R^2}$$

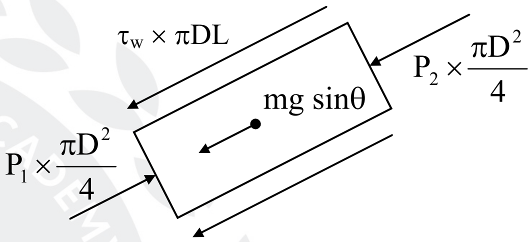
$$\tau_w = \tau(r = R) = \frac{2\mu u_{max}}{R} = \frac{4\mu V}{R} \quad \{u_{max} = 2V\} \text{ for laminar flow}\}$$

$$\therefore \quad \tau_w = \frac{4 \times 0.9 \times 2.546}{0.05} \qquad \left(V = \frac{Q}{\frac{\pi}{4} \times D^2} = \frac{0.02}{\frac{\pi}{4} \times 0.1^2} = 2.546\,\text{m/s} \right)$$

$$= 183.31 \text{ Pa}$$

(OR)

The value of shear stress can also be found from free body diagram of fluid inside the pipe as:



$$\tau_w \times \pi DL + mg \sin\theta = (P_1 - P_2) \times \frac{\pi}{4} D^2$$

$$\tau_w \times \pi DL + \rho g \left(\frac{\pi}{4} D^2 \times L \right) \sin\theta = (P_1 - P_2) \times \frac{\pi}{4} D^2$$

$$\tau_w = \left(\frac{P_1 - P_2}{4L} \right) \times D - \frac{\rho g D}{4} \sin\theta$$

$$= \frac{(590 - 116) \times 10^3 \times 0.1}{4 \times 45} - \frac{1260 \times 9.81 \times 0.1 \times \sin 15}{4}$$

$$= 263.3 - 79.98$$

$$= 183.32 \text{ Pa} \quad \text{(same as above result)}$$

02(b). A very long cylindrical rod of 30 mm diameter is having one of its ends attached to a wall maintained at 500°C. Entire length of the rod is exposed to atmosphere at 25°C with convective heat-transfer coefficient of 25 W/m^2 K. Temperature along the length of the rod at (x_1) distance from the base is 400°C and at (x_1 + 10 mm) distance is 390°C.

(i) Find the thermal conductivity of the rod.

(ii) Find the distance at which temperatures are measured.

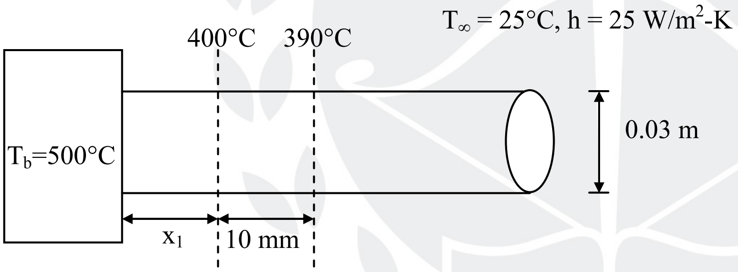
(iii) Plot the graph on a plain paper showing the variation of temperature along the length of the rod.

Consider the following relation:

$$\frac{T - T_\infty}{T_b - T_\infty} = e^{-mx}, \text{ where } m = \sqrt{\frac{hP}{kA}}$$ **(20 M)**

02(b).

Sol:



From the given formula:

$$\frac{T - T_\infty}{T_b - T_\infty} = e^{-mx}$$

$$\frac{400 - 25}{500 - 25} = e^{-mx_1} \quad \text{………….. (1)}$$

$$\frac{390 - 25}{500 - 25} = e^{-m(x_1 + 0.01)} \quad \text{…………….. (2)}$$

from (1) ÷ (2)

$$\frac{375}{365} = e^{-m(x_1 - x_1 - 0.01)}$$

$$1.0273 = e^{0.01\,m}$$

$$m = 2.702/\text{meter} = \sqrt{\frac{Ph}{kA}}$$

$$2.702^2 = \frac{Ph}{kA}$$

$$7.3 = \frac{\pi d \times 25}{k \times \frac{\pi}{4} \times d^2} = \frac{4 \times 25}{k \times 0.03}$$

$$k = 456.57 \text{ W/m-K}$$

From equation (1)

$$\frac{375}{475} = e^{-mx_1}$$

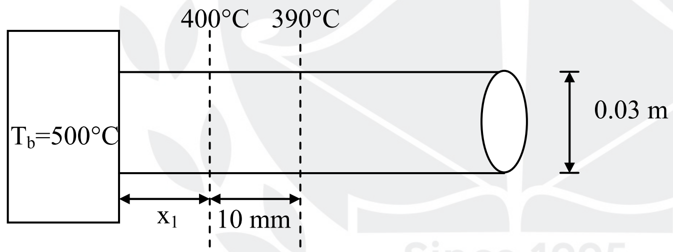
$$\frac{375}{475} = e^{-2.702 \times x_1}$$

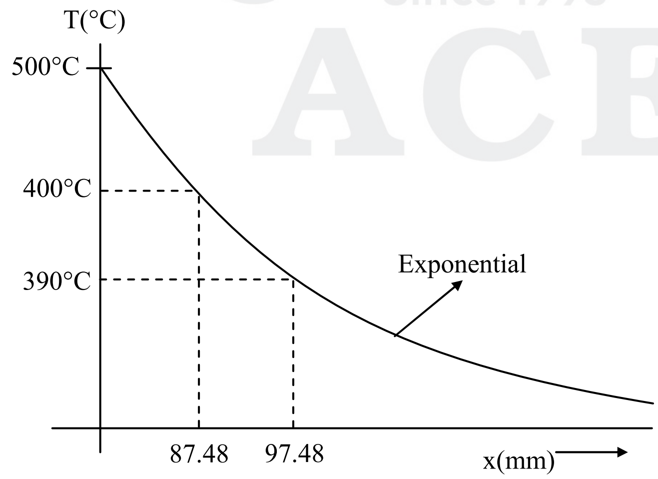
$x_1 = 87.48$ mm,

at $x_1 = 87.48$ mm, T = 400°C

at $x_1 + 10 = 97.48$ mm, T = 390°C

Temperature distribution





02(c). Give general specifications of engine in terms of its power ratings and swept volume for any commonly used two-wheeler and four-wheeler vehicle segment. Why now-a-days multiple inlet and multiple exhaust valves are preferred in engine system of a car over earlier conventional single inlet and single exhaust valve? Also find out for a four-stroke, four-cylinder SI engine operating at 4000 r.p.m., how many number of times the spark will trigger in one minute. (20 M)

02(c).

Sol:

	Two wheeler SI Engine	**Four wheeler SI Engine**
Operating cycle	2/4 strokes	4 strokes
Compression ratio	6 - 10	8 - 10
Bore	0.05 – 0.085 m	0.07 – 0.1 m
Stroke to bore ratio	1.2 – 0.9	1.1 – 0.9
Speed	4500 – 7500 rpm	4500 – 6500 rpm
bmep	4 – 10 atmospheres	7 – 10 atmospheres
Weight to power ratio	5.5 – 2.5 kg/kW	4.2 kg/kW
best bsfc	350 gms/kWhr	270 gms/kWhr

Now a days we use more than one inlet valve and more than one exhaust valve in IC engines. Generally 2 inlet valves and 3 exhaust valves are used so as to improve the volumetric efficiency of the engine and also valve area to piston area increases which permits higher engine speeds. As the valves are small and light weight spring force required is less and inertia of the valves for opening and closing is also less.

Number of cylinders, $n = 4$

Speed, $N = 4000$ rpm

4-stroke

Time, $t = 1$ minute

Number of sparks = Number of cylinders × speed × t

$$= \frac{n \times N \times t}{120}$$

$$= 4 \times \frac{4000}{120} \times 60 \quad = 4 \times 2000 = 8000$$

03(a). A furnace is shaped like a long equilateral triangular duct whose side is 1 m. The base surface has an emissivity 0.7 and is maintained at 600 K. The heated left-side surface of emissivity 1.0 is maintained at 1000 K. The right-side surface is fully insulated. Determine the rate at which the energy must be supplied to the heated side externally per unit length of the duct in order to maintain the given conditions. (Take $\sigma = 5.67 \times 10^{-8}$ W m^{-2} K^{-4}) (20 M)

03(a).

Sol: Using summation rule:

$F_{1-1} + F_{1-2} + F_{1-3} = 1$

$F_{1-1} = 0$

$F_{1-2} = F_{1-3}$ [Due to symmetry]

$F_{1-2} = F_{1-3} = 0.5$

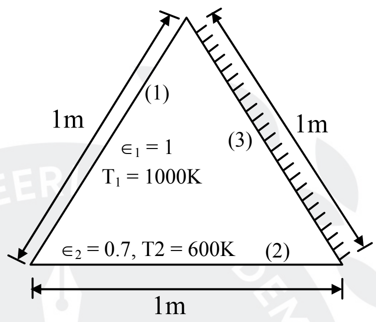
$F_{2-3} = F_{2-1} = 0.5$

$F_{3-1} = F_{2-2} = 0.5$

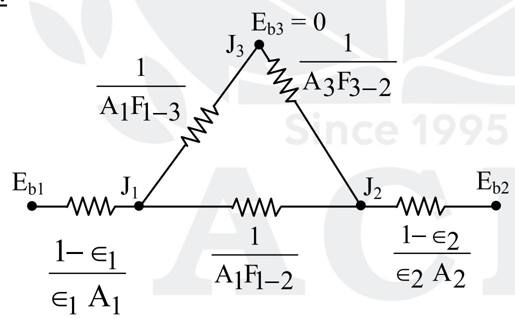
$A_1 = A_2 = A_3 = 1 \times L$ m^2

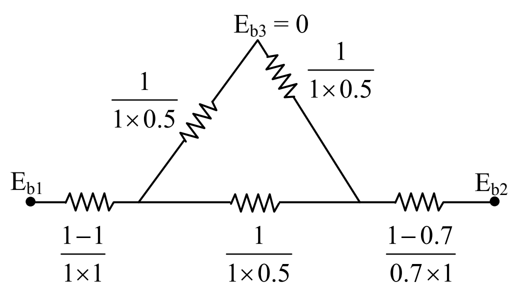
$L = 1$ m

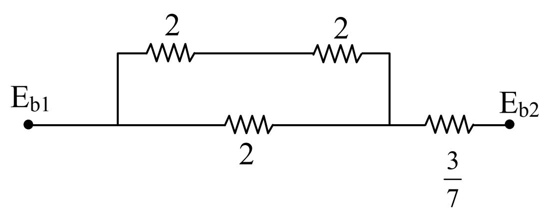
$A_1 = A_2 = A_3 = 1$ m^2



Thermal circuit :







$$\Sigma R_{th} = \frac{4\times 2}{4+2} + \frac{3}{7}$$

$$= \frac{8}{6} + \frac{3}{7} = \frac{4}{3} + \frac{3}{7} = \frac{28+9}{21} = \frac{37}{21}$$

Net heat exchange (Q) between side 1 and side 2

$$Q = \frac{\sigma\left(T_1^4 - T_2^4\right)}{\Sigma R_{th}}$$

$$Q = \frac{5.67\times 10^{-8}\left(1000^4 - 600^4\right)}{\frac{37}{21}}$$

$Q = 28010.412$ W/m

$Q = 28.01$ kW/m

03(b). Comfort condition for a human being is 20°C DBT with relative humidity of 50%. Atmospheric conditions at two places are given below during peak summer:

Place	DBT	Relative Humidity
Jaisalmer	45°C	10%
Chennai	35°C	80%

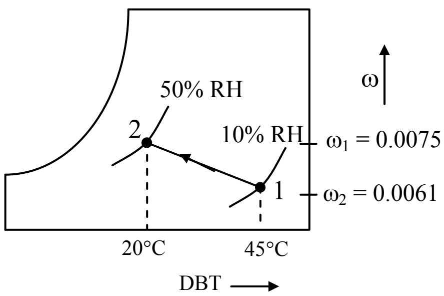
(i) Draw the processes in a plain paper applicable to convert the atmospheric condition air to comfort condition.

(ii) Suggest suitable air-conditioning devices to achieve these processes.

(iii) Determine the quantity of moisture needed to be added/removed per kg of air in these places. [Psychrometric Chart is placed at the end] (20 M)

03(b).

Sol: Jaisalmer Condition :



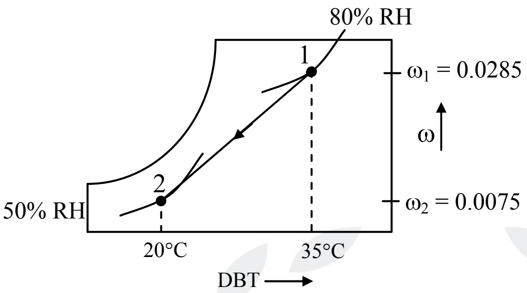
Cooling and humidification is recommended for Jaisalmer climate in air washer.

Amount of moisture added per kg dry air = $\omega_2 - \omega_1$

$= 0.0075 - 0.0061$

$= 0.0014$ kgvap

Chennai condition :



Cooling and dehumidification process is recommended

For Chennai climate in air washer

Amount of moisture removed per kg dry air

$= \omega_1 - \omega_2$

$= 0.0285 - 0.0075$

$= 0.021$ kgvap

03(c). State various losses considered by actual cycle analysis of IC engines. Discuss any one of them in detail. (20 M)

03(c).

Sol: In actual cycles when compared to fuel air cycles and air standard cycles, the actual efficiency is much lower than air standard efficiency. The major losses are :

- variation of specific heat with temperatures
- dissociation of combustion products
- progressive combustion
- incomplete combustion of fuel
- heat transfer into walls of combustion chamber
- blow down at end of exhaust process
- gas exchange process.

FUEL AIR CYCLES AND THEIR ANALYSIS :

Variable specific heats :

All gases, except mono-atomic gases, show an increase in specific heat with temperature. In the temperature range (300K to 2000K) the specific heat curve is nearly a straight line, which is expressed in the form

$$C_p = a_1 + k_1T$$

$$C_v = b_1 + k_1T$$

Where, a_1, b_1 and k_1 are constants. Now,

$$R = C_P - C_V = a_1 - b_1$$

Where, R is the characteristic gas constant.

$$C_p = a_1 + k_1T + k_2T^2$$

$$C_v = b_1 + k_1T + k_2T^2$$

The value of γ decreases with increase in temperature due to variation of specific heats. During the compression stroke, the final temperature and pressure would be lower if constant values of specific heat are used. This is shown in the below figure.

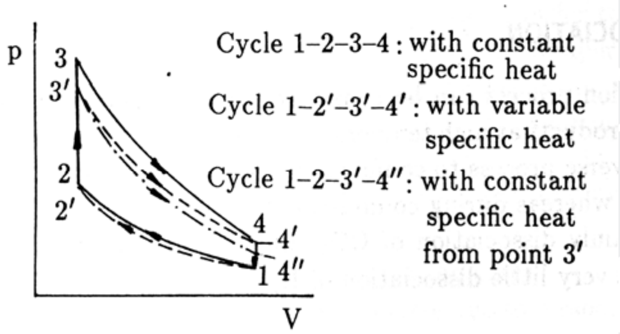


Fig: Loss of power due to variation of specific heat

With variable specific heats, the temperature at the end of compression will be 2′, instead of 2. For the process 1→2, with constant specific heats

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

With variable specific heats,

$$T_2' = T_1\left[\frac{v_1}{v_{2'}}\right]^{k-1}$$

Where, $k = C_P / C_V$.

Note that $v_2' = v_2$ and $v_1/ v_2 = v_1/ v_2' = r$.

For given values of T_1, p_1 and r, the magnitude of T_2' depends on k. Constant volume combustion, from point 2′ will give a temperature T_3' instead of T_3. This is due to increase in the value of C_v.

If expansion takes place at constant specific heats, this would result in the process $3' \to 4''$ where as actual expansion due to variable specific heat will result in $3' \to 4'$ and $4'$ is higher than $4''$. The magnitude in the difference between $4'$ and $4''$ is proportional to the reduction in the value of γ.

Consider the process $3' \to 4''$, $T_4'' = T_3'\left[\dfrac{v_3}{v_4}\right]^{k-1}$

For the process $3' \to 4'$, $T_4' = T_3'\left[\dfrac{v_3}{v_4}\right]^{\gamma-1}$

Reduction in the value of k due to variable specific heat results in increase of temperature from T_4'' to T_4'.

Dissociation:

Dissociation process can be considered as the disintegration of combustion products at high temperature.

The dissociation of CO_2 into CO and O_2 starts commencing around 1000°C and the reaction equation can be written as

$$CO_2 \rightleftharpoons 2CO + O_2 + \text{Heat}$$

Similarly, the dissociation of H_2O occurs at temperature above 1300°C and is written as

$$H_2O \rightleftharpoons 2H_2 + O_2 + \text{Heat}$$

The presence of CO and O_2 in the gases tends to prevent dissociation of CO_2, this is noticeable in a rich fuel mixture, which, by producing more CO, suppresses dissociation of CO_2. On the other hand, there is no dissociation in the fact that the temperature produced is too low for this phenomenon to occur. Hence, the maximum extent of dissociation occurs in the burnt gases of the chemically correct fuel-air mixture when the temperature are expected to be high but decreases with the leaner and richer mixtures.

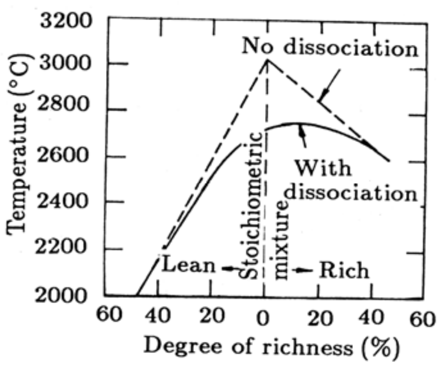


Fig: Effect of dissociation on temperature

Effect of Dissociation on Power:

Dissociation effects are not so pronounced in a CI engine as in an SI engine. This is mainly due to

(i) the presence of a heterogeneous mixture and

(ii) Excess air to ensure complete combustion.

If there was no reassociation due to fall of temperature during expansion the expansion process would be represented by $3'\rightarrow4''$. But due to reassociation the expansion follows the path $3'\rightarrow4'$. By comparing with the ideal expansion $3\rightarrow4$, it is observed that the effect of dissociation is to lower the temperature and consequently the pressure at the beginning of the expansion stroke. This causes a loss of power and also efficiency. Through during recombining the heat is given back it is too late to contribute a convincing positive increase in the output of the engine.

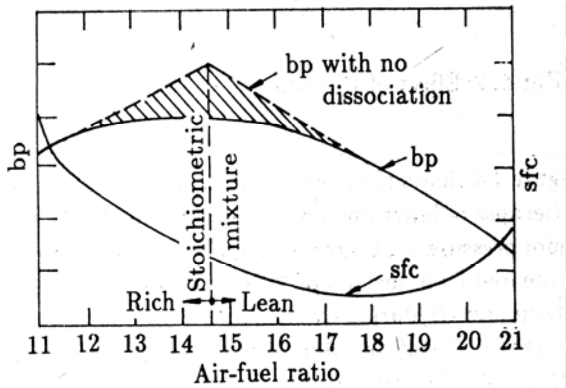


Fig: Effect of dissociation on power

04(a). For the purpose of project calculations, the total cost of moving a fluid over a distance by pipeline, at a steady flow rate Q, can be broken down into two items. First, the manufacture, laying and maintenance of the pipeline are represented by the cost C_1, which is proportional to D^3 (D = diameter of the pipe). The second item C_2 depends solely upon the energy required to pump the fluid. A preliminary design study for a particular project showed that the total cost was a minimum for D = 600 mm. If fuel prices are increased by 150%, and assuming only C_2 is affected, make a revised estimate of the optimum pipe diameter. (20 M)

04(a).

Sol: Power required, $P = \rho g Q\, h_f/\eta$ [where η is the pump efficiency].

$$= \rho g Q \frac{fLQ^2}{12.1D^5} \times \frac{1}{\eta}$$

$$= \frac{\rho g\, f\, L Q^3}{12.1\eta D^5}$$

⇒ From the above relation, we can say that

$$\text{cost} \propto D^{-5}$$

Total cost,

$$C = C_1 + C_2 = mD^3 + nD^{-5}$$

where m and n are two constants.

Now, C is minimum at $\frac{dC}{dD} = 0$

Now, $\frac{dC}{dD} = 3mD^2 - 5nD^{-6} = 0$

or $\quad 3mD^2 = \frac{5n}{D^6}$

or $\quad D^8 = \frac{5n}{3m}$ or $D = \sqrt[8]{\frac{5n}{3m}}$

For initial fuel costs, we write,

$$n = n_1 ,\ D = D_1$$

We may then say that the diameter of the original pipe is

$$D_1 = \sqrt[8]{\frac{5n_1}{3m}} \quad \text{--------(i)}$$

And the diameter of the new pipe, D_2 corresponding to $n = n_2$ is given by

$$D_2 = \sqrt[8]{\frac{5n_2}{3m}} = \sqrt[8]{\frac{5\times(1+1.5)n_1}{3m}}$$

Hence, $$D_2 = \sqrt[8]{\frac{5n_1}{3m}\times 2.5}$$

$$= D_1\sqrt[8]{2.5} \quad \text{[from eq.(i)]}$$

$$= 600\times\sqrt[8]{2.5} = 673 \text{ mm}$$

04(b). A refrigeration unit of 250 TR (1 TR = 3.5 kJ/s) capacity using R-12 as the refrigerant operates between –10°C and 35°C as evaporator and condenser temperatures respectively. Enthalpy of the refrigerant entering the evaporator is same as saturated liquid enthalpy at the condenser outlet. Dry saturated vapour leaves the compressor. Find the following:

(i) Mass flow rate of the refrigerant required.

(ii) Power required to run the compressor assuming isentropic compression

(iii) COP of the unit

(iv) Carnot COP

(v) Heat rejected by the condenser

Refer the following property tables:

T (°C)	P (bar)	v_f (m³/kg)	v_g (m³/kg)	h_f (kJ/kg)	h_{fg} (kJ/kg)	h_g (kJ/kg)	s_f (kJ/kg K)	s_g (kJ/kg K)
–10	2.191	0.000701	0.0821	26.87	156.32	183.19	0.1080	0.7019
35	8.48	0.000786	0.0207	69.56	131.89	201.45	0.2599	0.6839

(20 M)

04(b).

Sol: NRE (kW) = 250 Tonnes

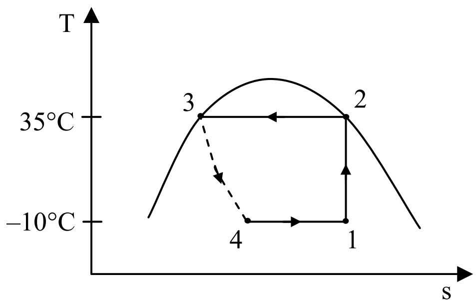
= 250 × 3.5 = 875 kW

$h_2 = (h_g)_{35°C} = 201.45$ kJ/kg

$s_2 = (h_g)_{35°C} = 0.6839$ kJ/kg

$h_3 = (h_4)_{35°C} = 69.56$ kJ/kg

$s_2 = s_1 = 0.6839 < 0.7019\ (s_g)_{-10°C}$



Hence before compression in wet state

$$s_1 = s_{f_1} + x_1(s_{g_1} - s_{f_1})$$

$$0.6839 = 0.1080 + x_1\ (0.7019–0.1080)$$

$$x_1 = \frac{0.6839 - 0.1080}{0.7019 - 0.1080} = \frac{0.5759}{0.5939} = 0.969$$

$$h_1 = h_{f_1} + x_1(h_{f_g})$$

$$= 26.87 + 0.966\ (156.32)$$

$$= 178.34\ \text{kJ/kg}$$

$$\dot{m}_r\left(\frac{kg}{sec}\right)(h_1 - h_4)\frac{kJ}{kg} = NRE(kW)$$

$$\dot{m}_r(178.34 - 69.56) = 875$$

Mass flow rate of refrigerant = $\dot{m}_r = \dfrac{875}{108.78} = 8.04\ \text{kg/sec}$

Power required = $\dot{m}_r\left(\dfrac{kg}{sec}\right)(h_2 - h_1)\dfrac{kJ}{kg}$

$$= 8.04\ (201.45 – 178.34)$$

$$= 185.80\ \text{kW}$$

$$COP = \frac{NRE(kW)}{W_C(kW)} = \frac{875}{185.80} = 4.709$$

$$\text{Carnot COP} = \frac{T_1}{T_2 - T_1} = \frac{263}{308 - 263} = \frac{263}{45} = 5.84$$

Heat rejected in condenser = $\dot{m}_r\left(\dfrac{kg}{sec}\right)(h_2 - h_3)\dfrac{kJ}{kg}$

$$= 8.04(201.45–69.56)$$

$$= 1060.39\ \text{kW}$$

04(c). What are the different types of work in thermodynamics? State whether flow work is path function or point function. Write the steady flow energy equation for a single stream entering and single stream leaving a control volume. Also discuss steady flow energy equation for the following engineering systems.

(i) Throttling device

(ii) Compressor **(20 M)**

04(c).

Sol:

- In thermodynamics work done by a system on its surroundings is defined ***as an interaction whose sole effect, external to the system, could be reduced to the raising of a mass through a distance.***

  Free expansion: Situation in which $\int PdV$ is finite but work done is zero.
- Consider a vessel which is divided into two compartments. One compartment contains a gas at a known pressure while the other compartment is evacuated. If the partition is removed, the gas expands and occupies the entire container. In this case, the expansion of the gas is not restrained by an opposing force (since the other side is vacuum) and the work done by the gas is equal to zero. Such an expansion which is not restrained is called a ***free expansion*** and it is an ***irreversible process***. It is possible to obtain the values of the gas pressure at intermediate stages by allowing the partition to the move in steps. Then one can plot P versus V and obtain the area under this curve. Yes, the work done by the gas is equal to zero and it is not equal to $\int P\,dV$.
- $W = \int P\,dV$ for a reversible process only.
- $W \neq \int P\,dV$ for an irreversible process or for free expansion.

Paddle Wheel Work: Situation in which $\int PdV$ is Zero But Work is Done:

- The paddle wheel work process is a process involving friction in which the volume of the system does not change at all, and still work is done on the system.
- Representation of the process is provided by a system in which a paddle wheel churns a fixed mass of fluid as shown in fig.
- Consider that the system is insulated. Now as the paddle wheel runs. Work enters the system. It increases the stored energy of the system. The temperature of the fluid increases. There is a corresponding change of state of the system from 1 and 2. But there is no movement of the system boundary. Hence, $\int PdV$ is zero, although work has been done on the system. Thus, $\int PdV$ does not represent work for this case. So work may be done on a closed system even though there is no volume change.

Elastic work:

$$\delta W = -\sigma A\,dL \quad \text{or} \quad \frac{\delta W}{V} = -\sigma\, d\epsilon$$

For extension of an elastic rod.

Where, σ = stress, ϵ = strain

A = cross-sectional area of the rod;

V = Volume of solid rod, and

dL = deformation in the rod.

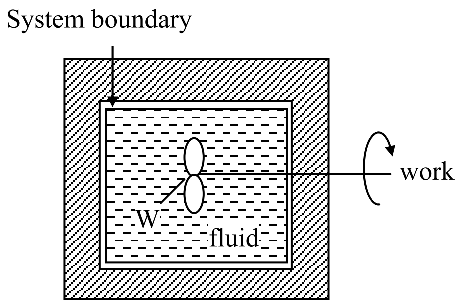


Fig: Paddle-wheel work

Work done in stretching of wire

$\delta W = -\tau\, dL$ for stretching of a wire

Where, τ = tension.

Electrical work

$\delta W = -Ei\,dt$ for a reversible cell

where, E = emf, i = current and t = time.

- Flow work is a **path function**

FIRST LAW OF THERMODYNAMICS FOR A STEADY FLOW ENERGY EQUATION ON TIME BASIS:

$$\dot{Q} - \dot{W} = \sum_{out} \dot{m}\left(h_2 + \frac{V_2^2}{2} + gZ_2\right) - \sum_{in} \dot{m}\left(h_1 + \frac{V_1^2}{2} + gZ_1\right)$$

Where, $\dot{Q}$ = rate of heat transfer in kW.

$\dot{W}$ = rate of shaft work done by the fluid

h = specific enthalpy in kJ/kg ; V = velocity of fluid in m/s

z = elevation above datum in m ; $\dot{m}$ = mass flow rate into and out of the control volume in kg/s

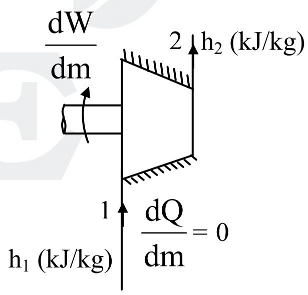
COMPRESSOR :

It is a device which converts low pressure fluid to high pressure fluid by giving external work input.

$$h_1 + \frac{dQ}{dm} = h_2 + \frac{dW}{dm};$$

$$\therefore \frac{dW}{dm} = (h_1 - h_2)$$

$$= -(h_2 - h_1) \text{ kJ/kg}$$

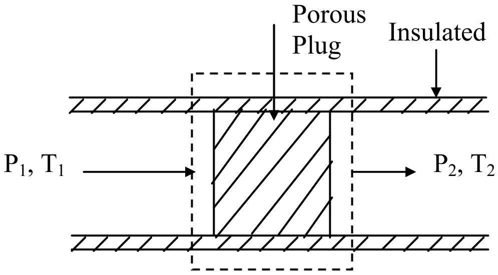


–ve sign indicates work is done on system (compressor)

THROTTLING PROCESS:

- Throttling valves are any type of flow restricting devices that cause a significant pressure drop in the fluid. e.g: capillary tubes and valves
- Unlike turbines they produce a pressure drop without involving any work.
- The pressure drop in the fluid is often accompanied a large drop in temperature.
- Hence throttling devices are commonly used in refrigeration and air conditioning applications.

- Consider the steady –state flow of a fluid through a horizontal, insulated pipe in which a porous plug is inserted as shown in fig. Choose the dotted portion of fig. as the control volume and apply the first law of thermodynamics.



The flow conditions imply the following:

Steady state: $\dot{m}_1 = \dot{m}_2 = \dot{m}$ and (dE /dt) = 0

Horizontal pipe: $Z_2 = Z_1$

Insulated: $\dot{Q} = 0$

No shaft work is involved: $\dot{W}_s = 0$

Ignore the changes in KE i.e, $V_2 = V_1$

Then the first law of thermodynamics reduces to $h_2 = h_1$

SECTION – B

05(a). Discuss why Pelton turbines are unsuitable for low heads. (12 M)

05(a).

Sol: Pelton turbine is classified as a an impulse type water turbine.

In case of impulse type turbines, pressure in and around runner is constant and equal to atmospheric pressure. So, only energy available for turbine is kinetic energy of water. So higher is the K.E, higher is the output of the turbine, subsequently higher overall efficiency (for same frictional losses).

$$K.E = \frac{1}{2}\dot{m}V^2$$

where $V = \sqrt{2gH}$

So higher is the head before runner, higher is the power output.

If low heads permitted, then for same power output, larger flow rates are required, but flow rate for a Pelton wheel is determined by the maximum efficiency condition. Hence the inlet velocity of jet coming from nozzle can not be altered. Therefore, jet diameter needs to be increased for more flow rate. But this leads to larger diameter of runner wheel, which causes bulky and slow rotation speed of turbine runner.

Mathematically,

Output power, $P = \rho\, g\, Q\, H = \gamma\, Q\, H$

For γ = specific weight of water constant

$P \propto Q\,H$

In order to develop required power to match with electric generator, either Q is less for high head available (or) Q is more for low heads.

$P \propto Q \downarrow$ and $H \uparrow$

$P \propto Q \uparrow$ and $H \downarrow$

In Pelton wheel, water jet hits the bucket once at a time to produce the large impulse force in an open atmosphere. It is possible to reduce flow rate by governing method for high heads available. At low heads, discharge is not enough to meet power output demand.

Impulse action is not available at low heads ($V = \sqrt{2gH}$), hence impulse force may not be enough to produce required force.

All impulse turbine are not suitable for low heads for the above mentioned reasons.

05(b). Mention the various advantages and disadvantages of the pulsejet engine and also draw the theoretical and actual pulsejet cycle on a P-V diagram. (12 M)

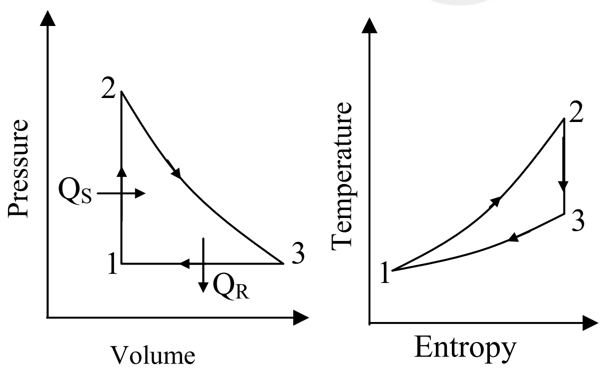
05(b).

Sol: Advantages of pulse jet engine:

- This is a very simple device next to ramjet and is light in weight. It requires very small and occasional maintenance.
- Unlike ramjet, it has static thrust because of the compressed air starting, thus it does not need a device for initial propulsion. The static thrust is even more than the cruise thrust.
- It can run on almost any types of liquid fuels without much effect on the performance. It can also operate on gaseous fuel with a little modifications.
- Pulse jet engine is relatively cheap.

Disadvantages of pulse jet engine:

- The biggest disadvantages is very short life of flapper valves and high rates of fuel consumption. The specific fuel consumption is as high as that of ramjet.
- The speed of the pulse jet is limited to a very narrow range of about 650- 800 km/h because of the limitations in the aerodynamic design of an efficient diffuser suitable for a wide speed range.
- The operational range of the pulse jet is also limited in altitude range.
- The high degree of vibrations due to intermittent nature of the cycle and the buzzing noise has made it suitable for pilotless crafts only.
- It has lower propulsive efficiency than turbojet engines.



P-V and T-s diagrams of Pulse Jet Engine.

05(c). Dry saturated steam at 40°C enters the surface condenser of a 500 MW thermal power plant having specific steam consumption of 3 kg/kWh. This steam is cooled by the water entering at 25°C. Minimum terminal temperature difference in the condenser is 7°C. Water flows through the tubes of internal diameter 3.75 cm and thickness of 3 mm with a velocity of 1 m/s. The overall heat-transfer coefficient of the condenser, U_0 = 1500 W/m^{2}K.

Determine the following for the condenser.

(i) Mass flow rate of water required in kg/s

(ii) Number of tubes required for the given heat-transfer rate

(iii) Length of each tube

Assume correction factor = 1

Density of water = 1000 kg/m^3

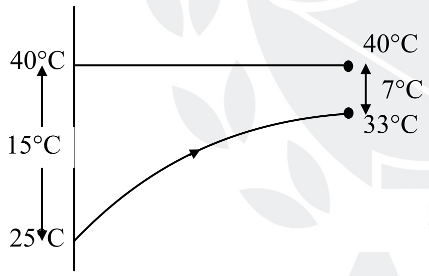
Specific heat of water = 4.2 kJ/kg K

Latent heat of condensation (h_{fg}) = 2407 kJ/kg

Condensed water leaves at saturated condition. (12 M)

5(c).

Sol:



$$\text{LMTD} = \frac{15-7}{\ell n\left(\frac{15}{7}\right)} = 10.49^\circ\text{C}$$

$$\text{Mass flow rate of steam} = \frac{3\,\text{kg}}{\text{kW hr}}$$

$$= \frac{3\,\text{kg} \times 500 \times 1000\,\text{kW}}{\text{kW} \times 3600\,\text{s}}$$

$$\dot{m}_s = 416.7\,\text{kg/s}$$

$$\text{Heat transfer rate} = \dot{m}_s\, h_{fg}$$

$$= 416.7 \times 2407 = 1002.916\ \text{MW}$$

Energy balance:

Heat released by steam = Heat absorbed by water

$$1002.916\times10^{3}=\dot{m}_{w}c_{p}(33-25)$$

$$1002.916\times10^{3}=m_{w}\times4.2\times8$$

Total mass flow rate of water $\dot{m}_{w}=29848.69\,\text{kg/s}$

Mass flow rate of water in each tube = $\rho.\ A_c\ V$ [V = 1m/s (given)]

$$=1000\times\frac{\pi}{4}\times0.0375^{2}\times1$$

$$= 1.1044\ \text{kg/s}$$

Number of tubes (n) = $\dfrac{\text{Total mass flow rate of water}}{\text{Mass flowrate of water in each tube}}$

$$=\frac{29848.69}{1.1044}=27025.44\ \text{tubes}\approx27026\ \text{tubes}$$

Heat transfer rate = 1002.916×10^{6}W

$UA_0\times LMTD = 1002.916\times10^{6}$

$1500\times n\times\pi d_0 L\times10.49 = 1002.916\times10^{6}$

$1500\times27026\times\pi\times0.0435\times L\times10.49 = 1002.916\times10^{6}$ $(d_0 = d+2t = 0.0375+2\times0.003 = 0.0435)$

$\Rightarrow$ L = 17.25 m

05(d). State the factors affecting the performance efficiency of solar PV cell. An inventor claims that his 1 m^2 size PV cell panel is capable of producing 2 kW of instantaneous power for a given location in the Indian context. Is his claim valid? Justify.

Assume suitable data wherever necessary. Consider the normally available PV cell efficiency as 15%. (12 M)

05(d).

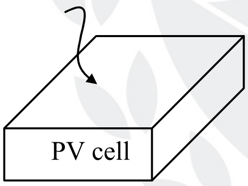
Sol: Factors affecting the performance of solar PV cell are listed below.

(i) Type of solar cell :

- Mono-crystalline silicon cell - highest efficiency. (around 15 - 20%)
- Poly-crystalline silicon cell - moderate efficiency. (around 10 - 15%)
- Amorphous silicon cell - (3 - 5%)

(ii) Effect of ambient condition

A particular range of wave length photons are necessary for PV cell. So if intensity of sun light is more or less than required, its efficiency will drop.



- The solar constant is 1367 W/m^2. This is the energy available outside the atmosphere of earth.
- As India is near the tropical region, let us assume that the intensity of solar radiation coming on surface of PV cell = 700 W/m^2 = I

$\eta_{cell} = 15\%$,

$A_{cell} = 1\ m^2$

$$\eta_{cell} = \frac{\text{output power}}{\text{input power}} = \frac{P_{output}}{I.A}$$

$$0.15 = \frac{P_{output}}{700 \times 1}$$

$P_{output} = 105\ W = 0.105\ kW$

Inventor claims 2 kW

Claim of inventor is totally wrong. 2 kW/m^2 from PV panel is not possible under any circumstances.

05(e). With reference to wind turbine, what is tip speed ratio? State its significance. For a wind turbine meant for generation of electricity, how many number of blades are desirable in general? If the tip of a wind rotor blade is travelling at 45 m/s and wind speed is 32 km/h, obtain the tip speed ratio. (12 M)

05(e).

Sol: Tip-speed ratio = λ

$$\lambda = \frac{\text{tip speed of the blade}}{\text{undisturbed wind velocity}}$$

If the radius of blade = R,

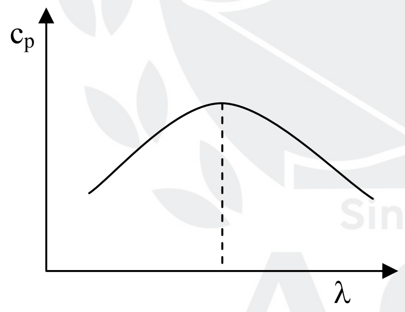
Angular velocity = ω (rad/sec)

Undisturbed wind velocity = u_o (m/s)

$$\lambda = \frac{R\omega}{u_o}$$

Significance of tip-speed ratio :

c_p = power coefficient for wind turbine



For a particular λ , c_p will be maximum.

If λ is more than required

- Higher tip-speed results higher noise level.
- Because of higher λ, Rω will be higher, it means more amount of turbulence will be generated when wind will approach the turbine.

If λ is less than required

Rω is less with respect to the wind velocity. More amount of air will pass from the gap of two without producing any energy.

For optimum power generation,

$\lambda = \dfrac{4\pi}{n}$ (n = number of blades)

$\Rightarrow R\omega = 45$ m/s

$u_o = 32$ km/h,

$u_o = 8.89$ m/s

$$\lambda = \frac{R\omega}{u_o} = \frac{45}{8.89} = 5.06$$

For optimum power generation,

$$\lambda = \frac{4\pi}{n}$$

$$n = \frac{4\pi}{\lambda} \Rightarrow n = \frac{4\pi}{5.06} \approx 2.48 \text{ or 3 blades}$$

In general also two or three blades are required.

06(a). What are surging and stalling in axial flow compressors? Explain briefly how they are developed and their effects. (20 M)

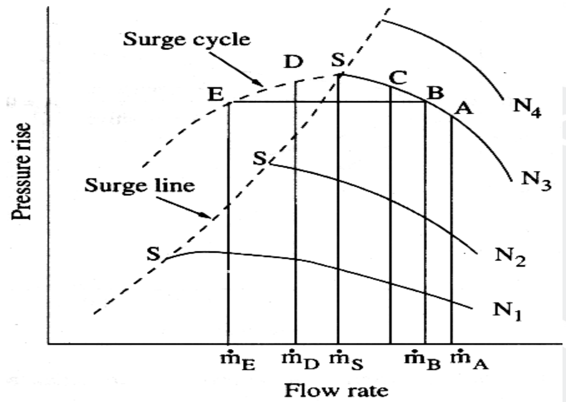
06(a).

Sol: SURGING

Unstable flow in axial compressors can be due to two reasons

(i) the separation of flow from the blade surface called stalling and

(ii) complete breakdown of the steady through flow called surging



- The surge phenomenon is explained with the aid of one of the curves in this figure.
- Let the operation of the compressor at a given instant of time be represented by point A(p_A, m_A) on the characteristic N_3 curve.
- If the flow rate through the machine is reduced to m_B by closing a valve on the delivery pipe, the static pressure upstream of valve is increased.
- This higher pressure, p_B is matched with increased delivery pressure (at B) developed by the compressor.
- With further throttling of the flow (to m_C and m_S), the increased pressures in the delivery pipe are matched by the compressor delivery pressures at C and S on the characteristic curve
- The characteristic curve at flow rates below m_s provides lower pressure as to D and E. However, the pipe pressure due to further closure of the valve (point D) will be higher than these.
- This mismatching between the pipe pressure and compressor delivery pressure can only exist for a very short time.
- This is because the higher pressure in the pipe will blow the air towards the compressor, thus reversing the flow leading to a complete breakdown of the normal steady flow from the compressor to the pipe.

- During this very short period the pressure in the pipe falls and compressor regains its normal stable operation (say at point B) delivering higher flow rate (m_B)
- However, the valve position still corresponding to the flow rate m_D. Therefore, the compressor operating conditions return through point C and S to D. Due to the breakdown of the flow through the compressor, the pressure falls further to p_E and the entire phenomenon, i.e., the surge cycle EBCSDE is repeated again and again.
- The frequency and magnitude of this to and fro motion of the air (surging) depend on the relative volumes of the compressor and delivery pipe, and the flow rate below m_S.
- Surging of the compressor leads to vibration of the entire machine which can ultimately lead to mechanical failure. Therefore, the operation of compressors on the left of the peak of the performance curve is injurious to the machine and must be avoided.

STALLING :

- Stalling is the separation of flow form the blade surface.
- At low flow rates (lower axial velocities) the incidence is increased.
- At large values of the incidence, flow separation occurs on the suction side of the blades which is referred to as positive stalling.
- Negative stall is due to the separation of flow occurring on the pressure side of the blade due to large values of negative incidence.
- In a great majority of cases this is not as significant.
- In high pressure ratio multistage compressor the axial velocity is already relatively small in the higher pressure states on account of higher densities.
- In such stages a small deviation from the design point causes the incidence to exceed its stalling value and stall cells first appear near the hub and tip regions.
- The size and number of these stall cells or patches increase with the decreasing flow rate.
- At very low flow rates they grow larger and cause a significant drop in the delivery pressure which can lead to the reversal of flow or surge.
- The stage efficiency also droops considerably on account of higher losses.

06(b). A coal - based 660 MW capacity thermal power plant is having overall efficiency of 45%. It uses 600 kg/s of steam for running the turbine. Coal used in the power plant is having calorific value of 10000 kJ/kg. Fuel to air ratio is 1:10 for combustion in the boiler. Find the following:

(i) Specific steam consumption in kg/kWh

(ii) Mass flow rate of coal required in Tph (Tonnes per hour)

(iii) Mass flow rate of air required for combustion in kg/s

(iv) Heat required to be supplied to generate one unit of power (in kJ/kWh)

(v) Coal required to be supplied to generate one unit of power (in kg/kWh) (20 M)

06(b).

Sol: Power = 660×10^3 kW

$\eta = 42\ \%$

Mass flow rate of steam, $\dot{m}_s = 600$ kg/sec

$\dot{m}_s$ (kg/sec) $\times$ W_{net} (kJ/kg) = Power (kW)

$600 \times W_{net}\ (\text{kJ/kg}) = 660 \times 10^3$

$$W_{net}\ (\text{kJ/kg}) = \frac{660 \times 10^3}{600} = 1100\ \text{kJ/kg}$$

$$\text{Specific steam consumption} = \frac{3600}{W_T - W_P}\left(\frac{\text{kg}}{\text{kW hr}}\right)$$

$$= \frac{3600}{1100} = 3.273\ \text{kg/kWhr}$$

$$\text{Heat rate} = \frac{3600}{\eta_{th}} = \frac{3600}{0.42} = 8571.42\ \text{kJ/kWhr}$$

Amount of heat to be supplied = Power (kW) $\times$ Heat rate (kJ/kWhr)

$$= 660 \times 10^3 \times 8571.42$$

$$= 5657137.2 \times 10^3$$

$$\dot{Q} = 5657.1372 \times 10^6\ \text{kJ/hr}$$

Calorific value of fuel, CV = 10000 kJ/kg

$$\text{Coal requirement} = \frac{\dot{Q}}{CV}$$

$$= \frac{5657.1372 \times 10^6}{10 \times 10^3}$$

$$= 565713.72 \text{ kg/hr}$$

$$\dot{C} = 565.71372 \text{ Tonnes/hour}$$

Air requirement = $\dot{A} = \dot{C} \times AFR$

$$\dot{A} = 565.71372 \times 10 = 5657.1372 \text{ Tonnes per hour}$$

$$\dot{A}\ (\text{kg/sec}) = \frac{5657.1372 \times 10^3}{3600} = 1571.43 \text{ kg/sec}$$

Coal requirement per unit power = $\dfrac{\dot{C}(\text{kg/hr})}{P(\text{kW})}$

$$= \frac{565.71372 \times 10^3}{660 \times 10^3} = 0.8571 \text{ kg/kWhr}$$

06(c). A hotel industry intends to replace its existing electric water heating system with a solar water heating system. The requirement of hot water is around 5000 litres per day. The proposed solar collector area is around 100 m^2. If the collector efficiency is 60%, estimate the reduction in electric bill of the hotel on a yearly average basis. Consider cost of electricity as Rs. 6/kWh. Make suitable assumptions wherever required. Consider average value of length of the day as 10 hours. Also estimate temperature rise of water for given radiation and collector efficiency data. Assume Indian context. Assume electric geyser efficiency as 95%. (20 M)

06(c).

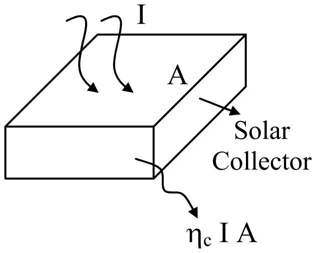
Sol: Given data;

Solar collector area = 100 m^2 = A

Intensity of radiation = 500 W/m^2 = I

Collector efficiency = 60% = η_C

Cost of electricity = ₹ 6/kWh



Average length of day = 10 hours

Electric geyser efficiency = 95% = η_g

$\Rightarrow$ Amount of energy given by solar collector = Q_C

$$Q_C = \eta_C . I . A$$

$$Q_C = 0.6 \times 500\ (\text{W/m}^2) \times 100 \text{ m}^2$$

Q_C = 30,000 W = 30 kW = 30 kJ/sec

Q_C = 30 kJ/sec × length of the day

Q_C = 30 kJ/sec × (10×60×60) sec/day

Q_C = 10,80,0100 kJ/day

$$1\,\text{kJ} = 1\frac{\text{kJ}}{\text{sec}} \times \text{sec} = 1\frac{\text{kJ}}{\text{sec}} \times \left(\frac{1}{3600}\right)\text{h}$$

$$1\,\text{kJ} = \frac{1}{3600}(\text{kWh})$$

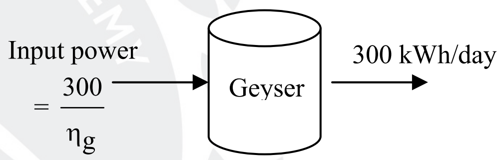
$$Q_C = \frac{10{,}80{,}000}{3600}\,\text{kWh/day}$$

Q_C = 300 kWh/day

Now to calculate the money saving, let us say that same amount of energy is given by electric geyser.

Input electric power needed in geyser

$$= \frac{300}{0.95} = 315.8\,\text{kWh/day}$$



Reduction of electricity bill in one day

= 315.8 kWh/day × ₹ 6/kWh = 1895 ₹/day

Let's assume in a year approx. 300 sunny days are there.

Then electricity bill saving on yearly basis = 1895 ₹/day × 300 days/year

Annual saving = 5,68,500 ₹/year

→ Hot water requirement = 5000 *lt*/day

Let us assume the inlet water temperature is 25°C.

→ Specific heat of water = 4.187 kJ/kg.K

Assume the density of water = 1000 kg/m^3, 1m^3 = 1000 *lt* = 1 kg/*lt*

Hot water requirement = 5000 kg/day

$$Q = mc(T_0 - T_i)$$

10,80,000 kJ/day = 5000 kg/day × 4187kJ/kg.K(T_0 – 25)K

$$T_0 = 76.58°C$$

Temperature rise of water for given condition = $T_0 - T_i$

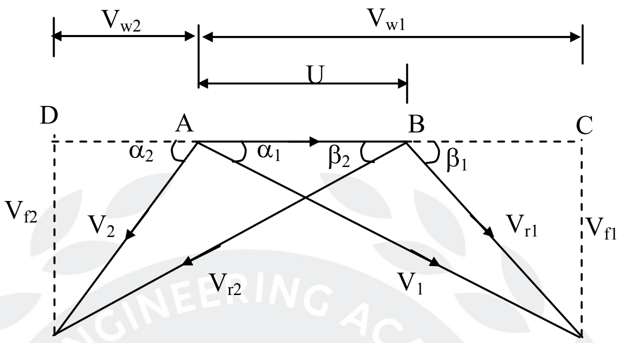
$$= 76.58 - 25$$

$$\Delta T = 51.58°C$$

07(a). A Parsons turbine runs at 400 r.p.m with 50% reaction and it develops 75 kW of power per unit mass of steam flow per second. The exit angle of the blades is 20° and the steam velocity is 1.4 times the blade velocity. Find the blade velocity and inlet angle of the blades. (20 M)

07(a).

Sol:



Power = 75 kW

Speed = N = 400 rpm

$\alpha_1 = \beta_2 = 90°$ (For 50% reaction turbine)

Blade speed = U

Steam speed = $V_1 = 1.4U$

$V_{f1} = V_1 \sin\alpha_1 = 1.4U \sin 20$

$V_{w1} = V_1\cos\alpha_1 = 1.4U \cos 20$

$BC = V_{w1} - U = 1.4U \cos 20 - U$

$= U(1.4 \cos 20 - 1)$

$$\tan\beta_1 = \frac{V_{f1}}{BC} = \frac{1.4U\sin 20}{U(1.4\cos 20 - 1)}$$

$$= \frac{1.4\sin 20}{1.4\cos 20 - 1}$$

$$= \frac{0.4788}{0.3156} = 1.5171$$

$\beta_1 = \tan^{-1}(1.5171) = 56.61° = \alpha_2$

$\beta_1 = \alpha_2 = 56.61°$

Blade velocity ,

$$P = \frac{\dot{m}(2V_1\cos\alpha - U)U}{1000}(kW) = \frac{\dot{m}(2\times 1.4U\cos\alpha - U)U}{1000}(kW)$$

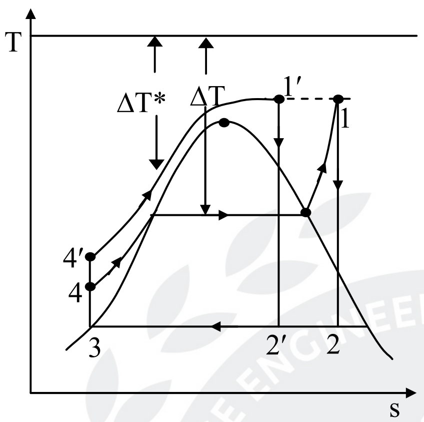
$$75 = \frac{2(1.4\cos 20 - 1)}{1000}U^2$$

$\Rightarrow U = 214.43$ m/s

07(b)(i) Compare the supercritical Rankine cycle and subcritical Rankine cycle used in coal-based thermal power plants. (10 M)

07(b)(i).

Sol:



T-s diagram for subcritical and supercritical Rankine cycle

- Cycle 1-2-3-4-1 → Subcritical Rankine cycle.
- Cycle 1′-2′- 3-4′ - 1′ → Super critical Rankine cycle.

- In super critical plants steam generation occurs at a pressure above critical pressure whereas in subcritical plants steam generation occurs at a pressure below critical pressure.
- Supercritical plants produce steam at a pressure and temperature near 30 MPa and 600°C respectively, permitting thermal efficiency upto 47% where as subcritical plants have efficiency only upto 40%.
- Installation costs of supercritical plants are higher per unit of power generated than subcritical plants.
- Fuel costs of supercritical plants are considerably lower owing to increased thermal efficiency.
- Since less fuel is used for a given power output, super critical plants produce less carbon dioxide, other combustion gases and solid waste than subcritical plants.

07(b).

(ii). How do you estimate the theoretical minimum air required for combustion by knowing the ultimate analysis of the coal? Molecular weight of C, O, H and S can be taken as 12, 16, 1 and 32 units respectively. (10 M)

07(b)(ii).

Sol: $C + O_2 \rightarrow CO_2$

$12 \text{ kg } C + 32 \text{ kg } O_2 \rightarrow 44 \text{ kg } CO_2$

$1 \text{ kg } C + \frac{8}{3} \text{ kg } O_2 \rightarrow$

kg CO_2

$2\,H_2 + O_2 \rightarrow 2\,H_2O$

$4 \text{ kg } H_2 + 32 \text{ kg } O_2 \rightarrow 36 \text{ kg } H_2O$

$1 \text{ kg } H_2 + 8 \text{ kg } O_2 \rightarrow 9 \text{ kg } H_2O$

$S + O_2 \rightarrow SO_2$

$32 \text{ kg } S + 32 \text{ kg } O_2 \rightarrow 64 \text{ kg } SO_2$

$1 \text{ kg } S + 1 \text{ kg } O_2 \rightarrow 2 \text{ kg } SO_2$

O_2 requirement per kg fuel $= \left[\frac{8}{3}C + 8H_2 + 1S - O_2\right]$

Air requirement per kg fuel $= \frac{100}{23}\left[\frac{8}{3}C + 8H_2 + 1S - O_2\right]$

Air contains 23 % O_2 by weight.

07(c). What is the approximate composition of biogas? State any two factors that govern the biogas production. A family living in a village having 5 cows is interested to set up a biogas plant to meet its cooking requirements on daily basis. The family has 5 adult persons. Estimate its biogas requirements on daily basis. Also work out the cow dung requirement on daily basis and also find out whether the number of cows available with family is sufficient to meet its requirement.

The following data may be useful:

Collectable cow dung per cow = 7 kg (approx)

Percent of solid mass in cow dung with balance moisture = 18%

Gas yield per kg of dry matter of cow dung = 0.34 m^3/kg of dry mass

Gas requirement for cooking = 0.227 m^3/person/day **(20 M)**

07(c).

Sol: Approximate composition of biogas

Compound	Formula	%
Methane	CH_4	50-75%
Carbon dioxide	Co_2	25-50%
Nitrogen	N_2	0-10%
Hydrogen	H_2	0-1%
Hydrogen-Sulphide	H_2S	0-3%

Factors affecting the biogas production

1. Temperature inside the digestor

   The choice of temperature is critical for the development of anaerobic digestion process.

   It has a strong influence over quality and quantity of biogas production.

2. Retention time of biomass in the digestor

   It is very important to decide the retention time of biomass according to the type of biomass, as it will affect the anaerobic digestion process.

→ No. of person = 5

Gas requirement for cooking = 0.227 m^3/person/day

Biogas requirement on daily basis = $0.227 \times 5 = 1.135$ m^3/day

→ Collectable cow dung = 7 kg/cow/day

% of solid mass in cow dung = $7 \times 0.18 = 1.26$ kg of dry mass/cow/day

→ Gas yield per kg of dry matter of cow dung = 0.34 m^3/kg of dry mass

→ Gas yield from cow dung = 1.26 × 0.34 = 0.4284 m^3/cow/day

→ Requirement of cows on daily basis = $\dfrac{1.135\,m^3/day}{0.4284\,m^3/cow/day} = 2.65\,cows$

It means that approx 3 cows we need.

So, in the family, number of cows are more than sufficient

Cow dung required per day = 2.65 cows × 7 kg/day/cow

= 18.55 kg/day

08(a). What do you mean by Net Positive Suction Head (NPSH)? Find the height from the water surface at which a centrifugal pump may be installed to avoid cavitation when atmospheric pressure = 1.01bar, vapour pressure = 0.022 bar, losses in suction pipe = 1.42 m, effective head of pump = 49 m and cavitation factor = 0.115. (20 M)

08(a).

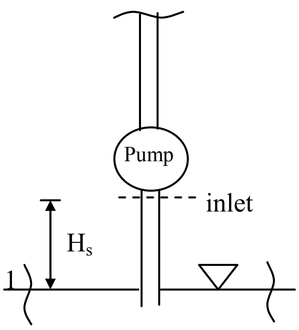
Sol: Net positive suction head. It is a cavitation criteria and defined as the difference between the pump inlet stagnation pressure head and the vapour pressure head.

$$NPSH = \left(\frac{P}{\rho g} + \frac{V^2}{2g}\right)_{pump\,inlet} - \frac{P_V}{\rho g}$$

- Pump manufacturers provide the values of $NPSH_{required}$ which is the minimum NPSH necessary to avoid cavitation in the pump.
- These values are plotted on the same pump performance curve.
- It is to be noted that $NPSH_{required}$ is independent of the type of fluid if expressed properly in units of head of head of liquid being pumped.
- $NPSH_{required}$ is usually much smaller than H over the majority of the performance curve.
- $NPSH_{required}$ increases with volume flow rate (Q) although for some pumps it decreases with Q at low flow rates where the pump is not operating very efficiently.

The actual or available NPSH which depends on the system must be greater than $NPSH_{required}$ for cavitation not to occur.

By Bernoulli's equation,



$$\frac{P_{atm}}{\rho g} = \left(\frac{P}{\rho g} + \frac{V^2}{2g}\right)_{inlet} + H_s + h_{fs}$$

$$\therefore \left(\frac{P}{\rho g} + \frac{V^2}{2g}\right)_{inlet} = \frac{P_{atm}}{\rho g} - H_s - h_{fs}$$

$$\therefore \quad NPSH = \frac{P_{atm}}{\rho g} - H_s - \frac{P_v}{\rho g} - h_{fs}$$

$$NPSH = H_{atm} - H_s - H_v - h_{fs}$$

$$H_{atm} = \frac{1.01 \times 10^5}{9810} = 10.3\,m,$$

$$H_v = \frac{0.022 \times 10^5}{9810} = 0.2243\,m$$

The critical Thoma's cavitation factor is given by

$$\sigma_c = \frac{H_{atm} - H_{s\,max} - H_v - h_{fs}}{H}$$

$$0.115 = \frac{10.3 - H_{s\,max} - 0.2243 - 1.42}{45}$$

$$\therefore \quad H_{s\,max} = 3.02\text{ m}$$

08(b). Economizer of a power boiler operating at 150 bar pressure receives 500 kg/s of water from boiler feed pump with specific enthalpy of 340 kJ/kg. Superheated steam leaves the boiler at 550°C with specific enthalpy of 3448.6 kJ/kg. Efficiency of the boiler is 90% and calorific value of the coal used is 10000 kJ/kg. Find the following:

(i) Heat added in economizer, evaporator and superheater in kJ/s

(ii) Percentage of heat added in economizer, evaporator and superheater out of total heat.

(iii) Rate of coal consumption in kg/s

Also draw T-s plot showing the position of different components and heat added.

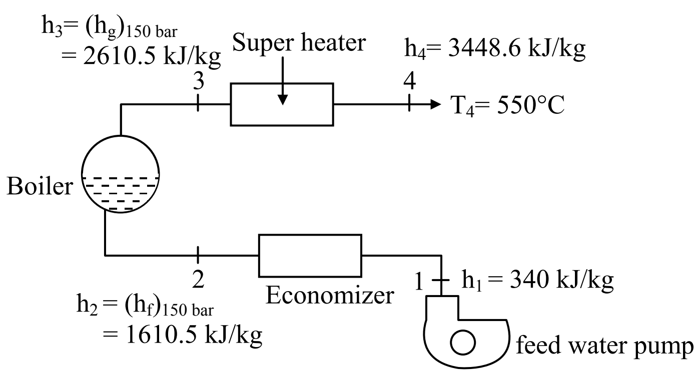
For 150 bar pressure, use the following table:

P_s (bar)	T_s (°C)	h_f (kJ/kg)	h_{fg} (kJ/kg)	h_g (kJ/kg)
150	324.24	1610.5	1000	2610.5

(20 M)

08(b).

Sol:



Boiler efficiency = 90 %

Heat added in economizer, evaporator and super heater = $\dot{m}_s\left(\frac{kg}{sec}\right)(h_4 - h_1)\frac{kJ}{kg}$

$$= 500 \times (3448.6 - 340)$$

$$\dot{Q} = 1554300\,kW$$

Calorific value of coal = 10,000 kJ/kg

Heat added in economizer = $(h_2 - h_1)$ kJ/kg = 1610.5 – 340

A = 1270.5 kJ/kg

Heat added in evaporator = B = $(h_3 - h_2)$

= (2610.5 – 1610.5)

B = 1000 kJ/kg

Heat added in super heater = $(h_4 - h_3)$ = 3448.6 – 2610.5

C = 838.1 kJ/kg

Total heat = A+B+C = 3108.6 kJ/kg

Percentage heat in economizer = $\frac{A}{A+B+C} \times 100$

$$= \frac{1270.5}{3108.6} \times 100 = 40.87\%$$

$$\%\,\text{Heat in evaporator} = \left(\frac{B}{A+B+C}\right) \times 100 = \frac{1000}{3108.6} \times 100 = 32.17\%$$

$$\%\,\text{Heat in super heater} = \left(\frac{C}{A+B+C}\right) \times 100$$

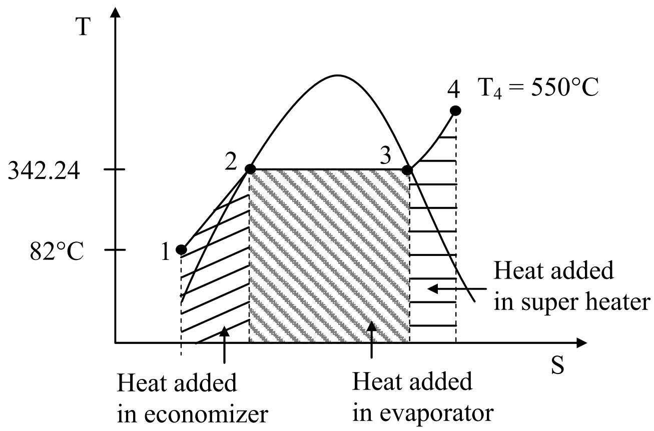
$$= \frac{838.1}{3108.6} \times 100 = 26.96\%$$

Heat supplied by fuel = $\frac{\dot{Q}(kW)}{\eta_{boiler}} = \frac{1554300}{0.9}$

$$\dot{Q}_1 = 1727000\,kW$$

Coal consumption = $\frac{\dot{Q}_1(kW)}{CV(kJ/kg)}$

$$= \frac{1727000}{10{,}000} = 172.7\,kg/sec$$



08(c). Explain the working principle of solar cooker. What are the challenges in making solar cooker more popular? Also describe the thermal energy storage system of solar energy.

(20 M)

08(c).

Sol: Working principle of solar cooker :

Most solar cooker works on the basic principle that sunlight is converted into heat energy and it is retained for cooking.

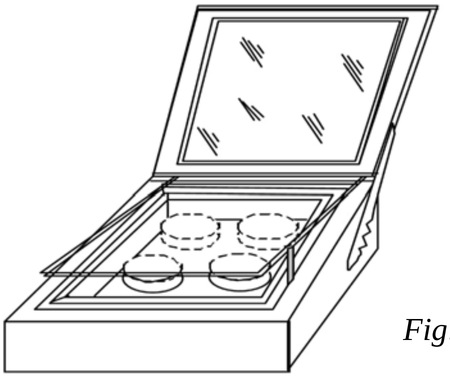


Fig: Solar cooker

- On the sides thermal insulation is there which reduces the heat loss.
- On the top transparent cover is provided which will trap the heat inside the cooker.
- A reflecting mirror will reflect more amount of radiation towards transparent cover
- Inside the transparent cover, place the cook pot.

Challenges for solar cooker :

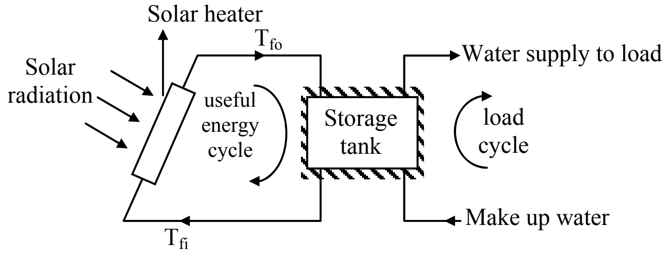
- It needs a location which is sunny for several hours.
- It should be protected from strong winds, otherwise heat loss will be more.
- It can not work at night or on cloudy days, that is why an alternative cooking source is required.
- Some solar cookers need more time for cooking than the conventional stove or oven.
- Cooks may need to learn special cooking techniques.
- The cost is higher than conventional cooker.

Solar thermal energy storage :

In order to run the system smoothly or to match the supply and demand, we need a thermal energy storage device.

Schematic representation of thermal energy storage device

Storage tank should be insulated otherwise heat will be lost to the atmosphere.



Thermal Energy can be stored by two modes:

(a) Sensible heat storage

(b) Latent heat storage

(a) Sensible heat storage:

Heat is stored by virtue of temperature difference.

$Q = m\, c_p\, \Delta T$

or $\quad Q = m \int_{T_1}^{T_2} c_p\, dT \qquad$ [where, c_p = specific heat of liquid/solid]

Heat can be stored in

(i) solid media (Rock, pebbles etc)

(ii) liquid media (Water, refrigerants)

(iii) hybrid (solid + liquid)

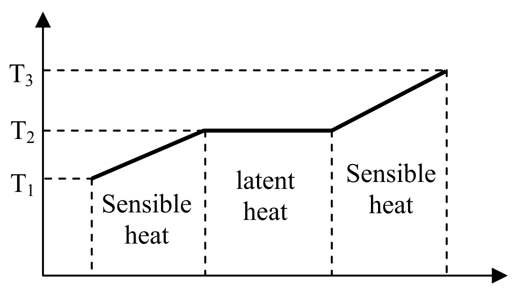
- Heat is not stored in gases due to large volume handling problems and lesser specific heat of gases.

(b) Latent heat storage :

Heat is stored due to phase change.

$Q = mL \qquad$ [where, L = latent heat (kJ/kg)]

- Heat is stored in solid -liquid transformation only.
- Because in solid-gas and liquid-gas transformation we have to handle the large volumes associated with the gases.
- Generally heat is stored in sensible and latent modes both.



$$Q = m \int_{T_1}^{T_2} c_p dT + mL + m \int_{T_2}^{T_3} c_p dT$$

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28	ME	Suhas M
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30	E&T	G.V.V.S.D.E P.K. Yadav
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31	ME	Umakanth
32	CE	Ankur Mani Tripathi
32	E&T	Gnaneswar K
32	EE	Ajita Mohanty
33	EE	ANKIT SHARMA
33	CE	Amarnath Reddy
34	E&T	Satpal
34	EE	Ravindra Singh
35	EE	Vishal Bajaj
36	EE	Prasanth shekar
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40	ME	Ujjwal

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