# SJE (PAPER-II) - 2018 MAINS
# OFFLINE TEST SERIES

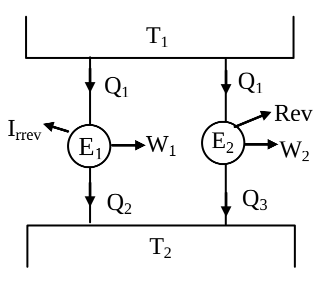
**MECHANICAL ENGINEERING** **Full Length MOCK TEST-2**

# SOLUTIONS

**01(a)(i).**

**Sol:** To show that no heat engine working between two fixed temperature can have an efficiency greater than that of a reversible engine.

Let us consider two engines one reversible and other irreversible operating between same temperature limits. Receiving same heat $Q_1$ .




Let us assume $\eta_{irrev} > \eta_{rev}$

$$\eta_{irrev} = \frac{W_1}{Q_1}$$

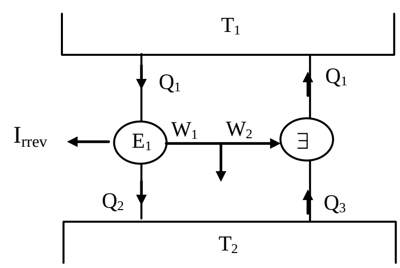
$$\eta_{rev} = \frac{W_2}{Q_1}$$

$$\frac{W_1}{Q_1} > \frac{W_2}{Q_1}$$

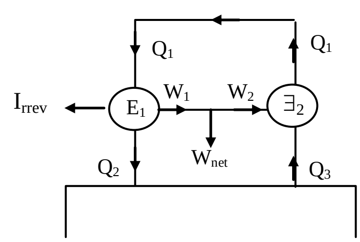
$\therefore W_1 > W_2$

As engine $E_2$ is a reversible engine,

Now reversing the direction it will operate as refrigerator




So work produced by engine 1 is supplied to refrigerator and heat $Q_1$ is supplied directly to the irreversible engine.

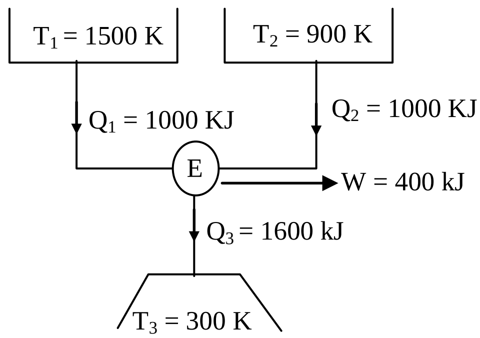


This is the violation of Kelvin Planck statement and hence our assumption is wrong.

Therefore efficiency of irreversible engine can never be greater than that of reversible engine.

**01(a)(ii).**

**Sol:**




(i) Applying first law

$\Sigma Q = \Sigma W$

$Q_1 + Q_2 - Q_3 = W$

$1000 + 1000 - 1600 = 400 \text{ kJ} = W$

$$\oint \frac{\delta Q}{T} \le 0$$

$$\oint \frac{\delta Q}{T} = \frac{Q_1}{T_1} + \frac{Q_2}{T_2} - \frac{Q_3}{T_3}$$

$$= \frac{1000}{1500} + \frac{1000}{900} - \frac{1600}{300}$$

$= 0.67 + 1.11 - 5.33$

$= -3.55$ kJ/K

$\oint \frac{\delta Q}{T} < 0$ Hence it is a irreversible cycle

**01(b).**

Spherical balloon

Initial diameter , $D_1 = 0.3$ m

Pressure , $P_1 = 1.5$ kgf/cm$^2$

$P_1 = 147.15$ kPa

Final diameter , $D_2 = 0.4$ m

Pressure $\propto$ Diameter

***To find:***

*(i) Work done during this process*

**Assumption:**

(i) Air is treated as a system

(ii) The system is a closed system

(iii) Process is assumed to be reversible process

For a closed system undergoing reversible process

Work done, $W = \int P dv$

Pressure $\alpha$ Diameter

$$P = KD$$

Volume of sphere , $V = \frac{4}{3}\pi R^3$

$$V = \frac{\pi}{6} D^3$$

Change in volume,

$$dV = \frac{\pi}{6}\left(3D^2 dD\right)$$

$$dV = \frac{\pi}{2} D^2 dD$$

$$k = \frac{P}{D}$$

$$K = \frac{P_1}{D_1} = \frac{147.15}{0.3} \Rightarrow K = 490.5\left(\frac{kN}{m^3}\right)$$

$$W = \int P dv$$

$$= \int kD \times \frac{\pi}{2} D^2 dD$$

$$= \frac{K\pi}{2}\int_{D_1}^{D_2} D^3 dD$$

$$= \frac{490.5}{2} \times \pi\left[\frac{0.4^4 - 0.3^4}{4}\right]$$

W = 3.37 kJ

As the work done is positive, therefore work is done by the system.

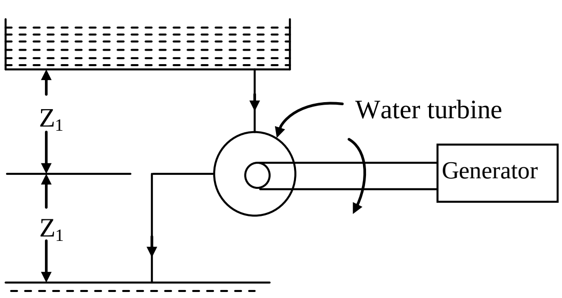
**01(c).**

**Sol: First law of thermodynamics for a Steady Flow Energy Equation on time basis :**

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

***(i) Water turbine:***

The turbine uses the potential and kinetic energy of water and converts into a useful mechanical energy.




The flow conditions imply the following:

Steady state: $\dot{m}_1 = \dot{m}_2 = \dot{m}$

Insulated: $\dot{Q} = 0$

No change in temperature of water: du = 0

$$\dot{W} = \left(\frac{p_1}{v_1} + \frac{V_1^2}{2} + gZ_1\right) - \left(\frac{p_2}{v_2} + \frac{V_2^2}{2} + gZ_2\right)$$

W is positive because work is done by the system.

***(ii) Centrifugal pump:***

These are used for pumping of liquids from lower level to higher level.

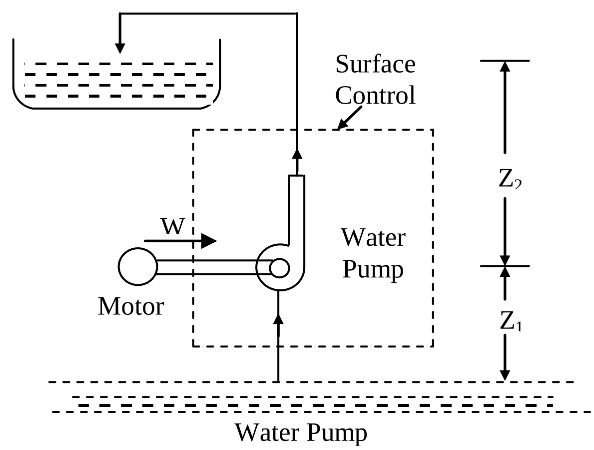



**Fig.: Centrifugal Water Pump**

The flow conditions imply the following:

Steady state: $\dot{m}_1 = \dot{m}_2 = \dot{m}$

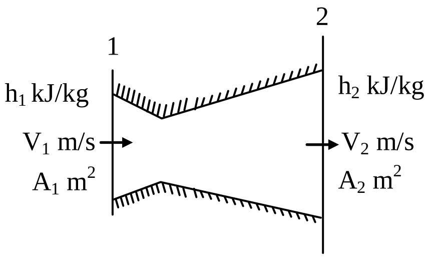
Insulated: $\dot{Q} = 0$

No change in temperature of water: du = 0

$$\dot{W} = \left(\frac{p_2}{v_2} + \frac{V_2^2}{2} + gZ_2\right) - \left(\frac{p_1}{v_1} + \frac{V_1^2}{2} + gZ_1\right)$$

***(iii) Steam Nozzle:***

- A schematic diagram of a nozzle is shown in fig. Since the primary purpose of a nozzle is to increase the flow velocity of the fluid the change in the kinetic energy cannot be ignored.




The flow conditions imply the following:

Steady state: $\dot{m}_1 = \dot{m}_2 = \dot{m}$

Horizontal: $Z_2 = Z_1$

Insulated: $\dot{Q} = 0$

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

The control volume expression gives

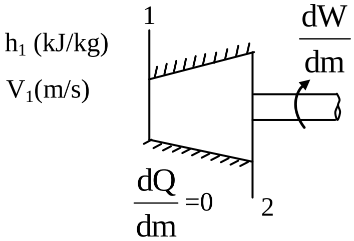
$$V_2 = \sqrt{V_1^2 + 2000(h_1 - h_2)} \quad (\because V_1 <<< V_2)$$

$$\Rightarrow V_2 = \sqrt{2000(h_1 - h_2)}$$

$$\therefore V_2 = 44.72\sqrt{\Delta h}$$

***(iv) Gas Turbines***

In turbines heat energy is converted into work energy




The flow conditions imply the following:

Steady state: $\dot{m}_1 = \dot{m}_2 = \dot{m}$

Horizontal : $Z_2 = Z_1$

Insulated: $\dot{Q} = 0$

$$h_1 + \frac{V_1^2}{2000} + \frac{dQ}{dm} = h_2 + \frac{V_2^2}{2000} + \frac{dW}{dm}$$

$$\frac{dW}{dm} = (h_1 - h_2) + \frac{(V_1^2 - V_2^2)}{2000} \quad \left(\because \frac{dW}{dm} = +ve\right)$$

$\therefore$ Work is done by the system.

**01(d).**

**Sol: Refrigerator:**

Refrigerator is a cyclically operating device which absorbs energy as heat from a low temperature body and rejects energy as heat to a high temperature body when work is performed on the device.

*The objective of this device is to maintain a body at low temperature than the surrounding temperature.*

Usually, it uses atmosphere as the high temperature reservoir.

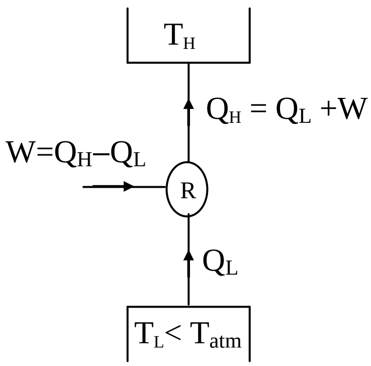
***Heat Pump:***

Heat Pump is a cyclically operating device which absorbs energy from a low temperature body and rejects energy as heat to a high temperature body.

*The objective of this device is to reject energy as heat to a high temperature reservoir.*

**COP of Refrigerator:**

The COP of a refrigerator is defined as the ratio of the rejected to the work done.




**COP (Coefficient of performance)**

$$(COP)_R = \frac{Q_L}{W} = \frac{Q_L}{Q_H - Q_L} = \frac{T_L}{T_H - T_L}$$

**COP of Heat pump:**

The COP of a heat pump is defined as the ratio of the heat supplied to the work done.

$$COP_{HP} = \frac{Q_H}{W} = \frac{Q_H}{Q_H - Q_L} \frac{T_H}{T_H - T_L}$$

**02(a).**

**Sol :**

| Description | SI Engine | CI Engine |
|---|---|---|
| **Basic cycle** | Works on Otto cycle or constant volume heat addition cycle. | Works on Diesel cycle or constant pressure heat addition cycle. |
| **Fuel** | Gasoline, a highly volatile fuel. Self-ignition temperature is high. | Diesel oil, a non-volatile fuel. Self-ignition temperature is comparatively low. |
| **Introduction of fuel** | A gaseous mixture of fuel-air is introduced during the suction stroke. A carburetor and an ignition system are necessary. Modern engines have gasoline injection. | Fuel is injected directly into the combustion chamber at high pressure at the end of the compression stroke. A fuel pump and injector the necessary. |
| **Load control** | Throttle controls the quantity of fuel-air mixture introduced. | The quantity of fuel is regulated. Air quantity is not controlled. |
| **Ignition** | Requires an ignition system with spark plug in the combustion chamber. Primary voltage is provided by either a battery or a magneto. | Self-ignition occurs due to high temperature of air because of the high compression. Ignition system and spark plug are not necessary. |
| **Compression Ratio [CR]** | 6 to 10. Upper limit is fixed by antiknock quality of the fuel. | 16 to 20. Upper limit is limited by weight increase of the engine. |
| **Speed** | Due to light weight and also due to homogeneous combustion, they are high speed engines. | Due to heavy weight and also due to heterogeneous combustion, they are low speed engines. |
| **Thermal efficiency** | Because of the lower CR the maximum value of thermal efficiency that can be obtained is lower. | Because of higher CR, the maximum value of thermal efficiency that can be obtained is higher. |

| | | |
|---|---|---|
| **Weight** | Lighter due to lower peak pressures. | Heavier due to higher peak pressures. |

**02(b).**

**Sol:**

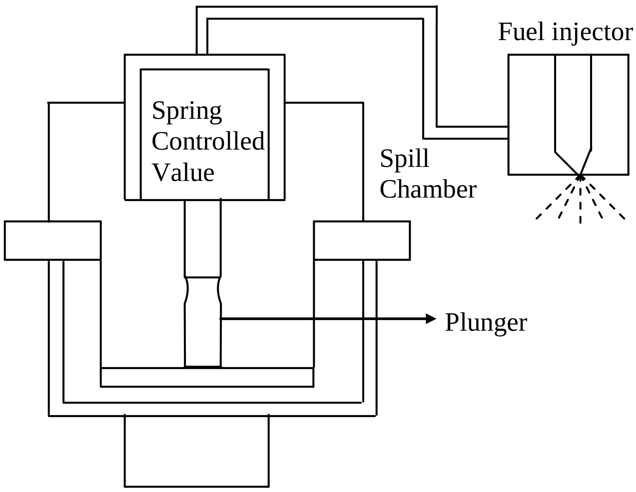



***Fig: Fuel Injection System***

**Features of a Fuel Injection System:**

1. The fuel should be introduced into the combustion chamber within a precisely defined period of the cycle.
2. The amount of the fuel injected per cycle should be metered very accurately. The clearances between the working parts of a fuel pump as well as the size of the orifice are very small. The working clearance is as small as 0.001 mm and the nozzle orifice size of even a big engine is as small as 0.625 mm in diameter. If it is enlarged by even 0.075 mm, the output would vary by about 35 percent. This increased output may result in imbalance, overheating or smoky exhaust.
3. The rate of injection should be such that it results in the desired heat release pattern.
4. The quantities of the fuel metered should vary to meet changing speed and load requirements.
5. The injected fuel must be broken into very fine droplets, i.e., good at atomization should be obtained.
6. The spray-pattern must be such that it results in rapid mixing of fuel and air.
7. The beginning and the end of injection should be sharp, i.e., there should not be any dribbling or after-injection.
8. The injection timing, if desired, should change to suit the engine speed and load requirements.
9. In the case of multi cylinder engines, the distribution of the metered fuel among various cylinders should be uniform.
10. In addition to the above requirements, the weight and the size of the fuel injection system must be minimum. It should not be costly to manufacture and expensive to attend to, adjust or repair.

**02(c).**

**Sol:** Given :

Q = 10 TR,

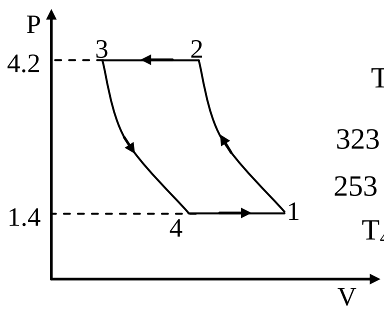
$p_2 = p_3 = 4.2$ bar,

$p_1 = p_4 = 1.4$ bar $= 1.4 \times 10^5$ N/m$^2$ ,

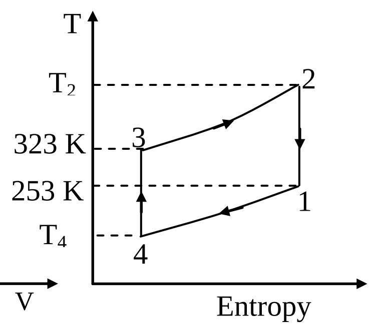
$T_3 = 50°C = 50 + 273 = 323$ K ,

$T_1 = -20°C = 20 + 273 = 253$ K

The cycle on p-v and T-s planes is shown in figure (a) and (b) respectively.




*(a) p-v diagram*




*(b) T-s diagram*

**(i). Coefficient of performance (COP)**

Let $T_2$ and $T_4$ = temperatures at the end of compression and expansion respectively.

Let us assume the compression and expansion to be isentropic and $\gamma$ for air as 1.4.

We know that

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4.2}{1.4}\right)^{\frac{1.4-1}{1.4}} = (3)^{0.286} = 1.369$$

$T_2 = T_1 \times 1.369 = 253 \times 1.369 = 346$ K

similarly $$\frac{T_3}{T_4} = \left(\frac{p_3}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4.2}{1.4}\right)^{\frac{1.4-1}{1.4}}$$

$$= (3)^{0.286} = 1.369$$

$$T_4 = \frac{T_3}{1.369} = \frac{323}{1.369} = 236\ \text{K}$$

we know that,

$$\text{COP} = \frac{T_1 - T_4}{(T_2 - T_3) - (T_1 - T_4)}$$

$$= \frac{253 - 236}{(346 - 323) - (253 - 236)} = \frac{17}{6} = 2.83$$

**(ii) Mass of air circulated per minute,**

Since the capacity of the machine is 10 TR, therefore heat extracted per min

$= 10 \times 210 = 2100$ kJ/min

($\because$ 1 TR = 210 kJ/min)

We know that heat extracted from the refrigerator per kg of air

$= c_p (T_1 - T_4)$

$= 1 \times (253 - 236) = 17$ kJ/kg

Mass of air circulated per minute,

$$m_a = \frac{\text{Heat extracted / min}}{\text{Heat extracted / kg}} = \frac{2100}{17}$$

$= 123.5$ kg/min

**(iii) Theoretical piston displacement of compressor,**

Let, $v_1$ = theoretical piston displacement of compressor per min.

we know that,

$$v_1 = \frac{m_a R_a T_1}{p_1}$$

$$= \frac{123.5 \times 287 \times 253}{1.4 \times 10^5} = 64\ \text{m}^3$$

**(iv) Theoretical displacement of expander,**

Let, $v_4$ = theoretical displacement of expander per minute.

We know that for constant pressure process 4-1 :

$$\frac{v_4}{T_4} = \frac{v_1}{T_1}$$

$$v_4 = \frac{v_1 \times T_4}{T_1} = 64 \times \frac{236}{253} = 60 \text{ m}^3$$

**(v) Net power per tonne of refrigeration,**

We know that net work done on the refrigerating machine per minute.

$= m_a$ (Heat rejected – Heat extracted)

$= m_a c_p [(T_2 - T_3) - (T_1 - T_4)]$

$= 123.5 \times 1 [(346 - 323) - (253 - 236)]$

= 741 kJ/min

Net power of the refrigerating machine

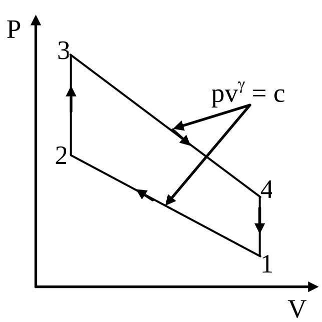
$$= \frac{741}{60} = 12.35 \text{ kW}$$

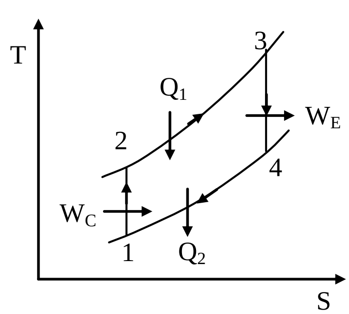
Net power per tonne of refrigeration

$$= \frac{12.35}{10} = 1.235 \text{ kW/TR}$$

**02(d).**

**Sol:**




$T_1 = 273 + 35 = 308$ K,

$p_1 = 0.1$ MPa $= 100$ kN/m$^2$

$Q_1 = 2100$ kJ/kg, $r_k = 8$, $\gamma = 1.4$

$$\eta_{cycle} = 1 - \frac{1}{r_k^{\gamma-1}} = 1 - \frac{1}{(8)^{0.4}} = 1 - \frac{1}{2.3}$$

$= 0.565$ or $56.5\ \%$

$$\frac{v_1}{v_2} = 8$$

$$v_1 = \frac{RT_1}{p_1} = \frac{0.287 \times 308}{100} = 0.844 \text{ m}^3\text{/kg}$$

$$v_2 = \frac{0.884}{8} = 0.11 \text{ m}^3/\text{kg}$$

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = (8)^{0.4} = 2.3$$

$T_2 = 2.3 \times 308 = 708.4$ K

$Q_1 = c_v (T_3 - T_2) = 2100$ kJ/kg

$$T_3 - 708.4 = \frac{2100}{0.718} = 2925 \text{ K}$$

$T_3 = T_{max} = 3633$ K

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = (8)^{1.4} = 18.37$$

$\therefore\ p_2 = 1.873$ MPa

$$\frac{p_3 v_3}{T_3} = \frac{p_2 v_2}{T_2}$$

$$p_3 = p_{max} = 1.837 \times \frac{363}{708} = 9.426 \text{ MPa}$$

$W_{net} = Q_1 \times \eta_{cycle}$

$= 2100 \times 0.565 = 1186.5$ kJ/kg

$= p_m (v_1 - v_2)$

$= p_m (0.884 - 0.11)$

$$p_m = \text{m.e.p} = \frac{1186.5}{0.774}$$

$= 1533$ kPa $= 1.533$ MPa

**03(a).**

**Sol:** At any radius $r \le r_o$,

$$u = \omega r$$

Shear stress on the inclined wall

$$\tau = \mu \frac{du}{dy} = \mu \frac{u}{h} = \mu \frac{\omega r}{h}$$

Considering an elemental area ( $2\pi r\, ds$ ),

$$dA = 2\pi r \,.\, \frac{dr}{\sin\theta}$$

d(torque) = dT = r d(force)

$$= r\,(\tau \times dA)$$

$$= r\,\tau\, 2\,\pi\, r \frac{dr}{\sin\theta}$$

$$= r\left(\mu \frac{\omega r}{h}\right)\left(2\pi r \times \frac{dr}{\sin\theta}\right)$$

$$= \mu \frac{2\pi\omega}{h} \frac{1}{\sin\theta} .r^3 dr$$

Torque $T = \int_0^{r_o} dT = \frac{2\pi\omega\mu}{h\sin\theta}\int_0^{r_0} r^3 dr$

$$T = \frac{\pi\omega\mu}{2h\sin\theta} r_0^4$$

**03(b).**

**Sol:** Speed, N = 1500 rpm ; Head, H = 15 m

Overall efficiency $\eta_o = 0.68$

Discharge Q = 0.2 $m^3/s$

Specific speed of centrifugal pump is given by

$$N_s = \frac{N\sqrt{Q}}{(H)^{3/4}} = \frac{1500 \times \sqrt{0.2}}{(15)^{3/4}} = 88.011$$

Non-dimensional specific speed is

$$N_s = \frac{\omega\sqrt{Q}}{(gH)^{3/4}} = \frac{\frac{2\pi \times 1500}{60}\sqrt{0.2}}{(9.81 \times 15)^{3/4}}$$

$$= \frac{157.08 \times \sqrt{0.2}}{42.25} \cong 5.26$$

Unit Speed is given by

$$N_u = \frac{1500}{\sqrt{15}} = \frac{900}{\sqrt{H_1}} \Rightarrow H_1 = 5.4m$$

Unit Discharge is given by

$$Q_u = \frac{0.2}{\sqrt{15}} = \frac{Q_1}{\sqrt{5.4}} \Rightarrow Q_1 = 0.12 m^3/s$$

Overall efficiency, $\eta_o = \frac{\rho g Q H_m}{P}$

$$P = \frac{9810 \times 0.2 \times 15}{0.68} = 43.279 kW$$

$$P_u = \frac{43.279}{(15)^{3/2}} = \frac{P_1}{(5.4)^{3/2}}$$

$$\Rightarrow P_1 = 9.35\, kW$$

Assuming overall efficiency unchanged at new rpm

Power required = $\frac{P_1}{\eta_0}$ = 13.75 kW

**03(c).**

**Sol: Assumptions are as follows :**

**(i) Flow is steady:**

The term steady implies no change of properties at a point with time. During steady flow, the fluid properties can change from point to point within a device, but at any fixed point they remain constant.

**(ii) Flow is irrotational :**

In the case of flow of an ideal fluid for which no tangential stresses occur and flow is assumed to be irrotational.

**(iii) Fluid is incompressible :**

Incompressibility is an approximation in which flow is said to be incompressible if the density remains nearly constant throughout. Appropriate if the flow mach number is less than 0.3.

**(iv) Inviscid regions of flow :**

Inviscid regions of flow implies that the regions where net viscous or frictional forces are small compared to other forces acting on fluid particles.

**(v) Flow along a single streamline**

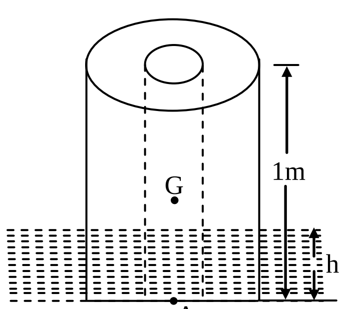
The Bernoulli equation results from a force balance along a streamline.

Hence $\frac{\partial}{\partial s}\left(\frac{V_s^2}{2}+\frac{p}{\rho}+gz\right)=0$

(or) $\frac{p}{\rho}+\frac{V^2}{2}+gz = \text{constant}$

**03(d).**

**Sol:**




Volume displaced by hollow cylinder

= Weight of cylinder

$$\Rightarrow \frac{\pi}{4}\left(D_1^2 - D_2^2\right)h \times g \times \rho_w = 700\,N$$

$$\frac{\pi}{4}\left((0.6)^2 - (0.4)^2\right)9.81 \times 1000 \times h = 700$$

$\therefore$ h = 0.454 m.

Hence, only 0.454 m of cylinder is inside water.

The centre of buoyancy (B) from A,

$$AB = \frac{h}{2} = \frac{0.454}{2} = 0.227\ m$$

Centre of Gravity (G) from A, AG = 0.5m

$\therefore$ BG = 0.5 – 0.227 = 0.272 m.

Now, metacentric height GM is given by

$$GM = \frac{I}{\forall} - BG$$

where, $I = \frac{\pi}{64}\times\left(D_1^4 - D_2^4\right) = 0.0051\ m^4$

And $\forall$ = volume of cylinder in water

$$= \frac{\pi}{4}\times\left(D_1^2 - D_2^2\right)\times h$$

$$= \frac{\pi}{4}\times\left((0.6)^2 - (0.4)^2\right)\times 0.454$$

$$= 0.0713\ m^3$$

$$\therefore GM = \frac{0.0051}{0.0713} - 0.272 = -0.2m$$

As metacentric height is negative, the point M lies below centre of gravity. Hence, the cylinder will be in unstable equilibrium and will not float vertically.

**04(a).**

**Sol: Alloying elements and their effects:**

1. **Chromium**
   - Increase Corrosion resistance
   - Increase Wear resistance
   - Increase tensile strength
   - Increase resistance to heat
   - Reduces machinability
2. **Tungsten**
   - Promotes heat resistance
   - Carbide stabilizer
   - Increases strength and hardness
   - Improves wear resistance
   - Reduce graphitization
   - Increases the basic mechanical properties when added in small percentage
3. **Manganese**
   - Stabilizes austenite
   - Acts as deoxidizer
   - Refines the graphite and pearlite
   - Refines the grains.
   - Increases the basic mechanical properties when added in low percentage.
4. **Molybdenum**
   - Improve tensile strength, fatigue strength and hardness
   - Refines the graphite and pearlite
   - Improves heat resistance
   - Act as austenitizer
   - Increases hardness and promotes uniform microstructure
5. **Nickel**
   - Austenitizer (Stabilize austenite)
   - Refine pearlite and graphite
   - Improves the toughness of castings
   - Refine the grains
   - Resistance to corrosion

**04(b).**

**Sol:** Based on the amount of oxygen taken from the oxygen cylinder, the flame produced in the oxy-acetylene welding is divided into three types, they are neutral flame, oxidizing flame and carburizing or reducing flame.

**Neutral flame:** In this ratio of oxygen to the acetylene is equal to 1. In this flame two cones are produced in the flame, one is inner cone which is in yellow or reddish in color and outer cone which is light blue color . yellow color indicates the incomplete combustion and light blue color indicates complete combustion.

Neutral flame

The length of the inner cone is about N = 10 to 15mm and the maximum temperature occurs at the intersection of the inner and outer cones which is about 3260°C whereas average temperature of the flame is about 2000 to 2100°C. This flame can be used for joining and cutting of all ferrous and non ferrous metals except brass.

**Oxidizing flame:** In this ratio of oxygen to the acetylene is equal to 1.15 to 1.5. In this flame also two cones are produced, one is inner cone which is in yellow or reddish in color and outer cone which is light blue color

Carburizing flame

The length of the inner cone is about N/3 to N/2 and the maximum temperature occurs at the intersection of the inner and outer cones which is about 3380°C whereas average temperature of the flame is about 2100 to 2200°C. This flame can be used for joining and cutting of all ferrous and non ferrous metals including brass. But because of excess supply of oxygen from the oxygen cylinder, there is a possibility of free oxygen may present in the flame so that if this flame is used for joining of highly reactive metals, the chances of oxidation is very high hence it should not be used for joining of highly reactive metals like Al, Mg etc.

**Carburizing flame:** In this ratio of oxygen to the acetylene is equal to 0.85 to 0.95. In this flame two cones are produced in the flame, one is inner cone which is in yellow or reddish in color and outer cone which is light blue color.

oxidizing flame

The length of the inner cone is about 2N to 3N mm and the maximum temperature occurs at the intersection of the inner and outer cones which is about 3040°C whereas average temperature of the flame is about 1800 to 1900°C. because of the lower average temperature of the flame this flame cannot be used for joining of high Mp materials. Also due to short supply of oxygen, there is a possibility of free carbon may present in the flame hence it should not be used for joining of ferrous materials, because iron may absorb the free carbon and brittleness of the weld bead will increase. This flame is most commonly used for joining and cutting of high carbon steels, because it already has saturated with carbon hence no carbon absorption will takes place and also in case of steels with increase of carbon content the MP of steels will reduces.

**04(c).**

**Sol:** Deforming the metal at a temperature less than the recrystallization temperature is called cold working. Deforming the metal at a temperature greater than or equal to the recrystallization temperature is called hot working.

The features of the hot and cold working are

i) For same amount of deformation to be produced, the force and energy required for hot working is less than the cold working.

ii) Because the hot working is done at a temperature higher than the recrystallization temperature, the scale formation will takes place. Due to this
- Surface finish produced on the component is poor
- Poor dimensional accuracy
- Friction produced during the process is high, i.e 0.5 to 0.6

Because the cold working is done at a temperature less than the recrystallization temperature, no scales will be formed. Due to this
- Surface finish produced on the component is very good
- Close dimensional accuracy is possible
- Friction produced during the process is low, i.e 0.1 to 0.2

iii) Because of higher temperature the handling of component in hot working is difficult and it is easier in the cold working.

iv) The mechanical properties of the cold worked component are such that it always has the higher strength & hardness and lower ductility & toughness than the original material. Where in the hot worked component, the mechanical properties will purely depend on the temperature at which the hot working has been completed.

**04(d).**

**Sol:**

**(i) Counter sinking and counter boring :** The method of enlarging the end of the hole as a tapered hole is called counter sinking operation. This will done by using large size drill bit than the existing hole. The method of enlarging the end of the hole for providing space to the bolt head or nut etc. using internal turning operation is called counter boring operation.

**(ii) Mandrel and arbor:** mandrel is a solid cylinder used for during machining of thin hallow components without bending due to cutting forces. This will be kept inside the thin hallow component during machining. Whereas the arbor is an accurately machined shaft for holding and driving the milling the peripheral milling cutters on the horizontal milling machines and it is tapered at one end to fit the spindle nose and has two slots to fit the nose keys for locating and driving it.

**(iii) Tap and die :** tapping and die threading are machining operations in which internal and external screw threads are produced by the helical (cutting motion of multi point tools called taps and dies respectively. Taps and dies can be visualized as helical broaches.

**(iv) Gear shaping and gear hobbing :** In gear hobbing, the tool is like cylinder with slots and gashes provided so that each edge will act like cutting tool and hob resembles like worm. Whereas in gear shaping the tool is similar to the pinion or a rack type. The gear blank and hobbing tool are rotating continuously but in gear shaping the tool is reciprocating and after each teeth is cut the blank is indexed by using indexing mechanism.

**05(a).**

**Sol: Kinematic Pair:**

Interconnection between two links that allows the relative motion between them is known as kinematic pair.

**Classification of Kinematic pairs:**

**(a) Based on Degrees of Freedom:**

- A kinematic pair allows few degrees of freedom and constraints some of them.
- A pair that constrains all the degrees of freedom of the second link relative to the first link is not considered as a kinematic pair, it is a rigid joint.
- On the basis of degrees of freedom there are following 6 pairs.

| Pair | Symbol | Pair Variable | Degrees of Freedom | Relative motion |
|---|---|---|---|---|
| Revolute | R | $\Delta\theta$ | 1 | Circular |
| Prismatic | P | $\Delta s$ | 1 | Rectilinear |
| Helical | H | $\Delta\theta$ or $\Delta s$ | 1 | Helical |
| Cylinder | C | $\Delta\theta$ and $\Delta s$ | 2 | Cylindric |
| Sphere | S | $\Delta\theta$, $\Delta\phi$, $\Delta\psi$ | 3 | Spheric |
| Flat | F | $\Delta x$, $\Delta y$, $\Delta\theta$ | 3 | Planar |

**(b) Based on Nature of contact:**

Based on the nature of contact between the two links the kinematic pairs are classified as *lower pairs and higher pairs*.

**Lower pairs:**

- When the two bodies have surface or area contact between them they are referred as lower pairs.
- The relative motion in a lower pair is either purely turning or sliding.

**Ex:** Nut turning on screw, universal joint etc.

**Higher pairs:**

- When the contact between the bodies is a point or line contact they are referred as higher pairs.
- The relative motion in a higher pair is a combination of sliding and turning.

**Ex:** Cam and Follower pair, ball and roller bearings and gears etc.

**(c)** Kinematic pair according to Mechanical construction.

(i) Closed pair

(ii) Open pair

**05(b).**

**Sol:** Given that

$$T = 25 - 7.5 \sin 3\theta \text{ kNm}$$

Resisting torque, $T_r = 25 + 3.6 \sin\theta$ kN-m

$I = 360$ kgm$^2$, $N = 450$ rpm

(i) Power = $T_{mean} \times \omega$

$$T_{mean} = \frac{1}{\pi}\int_0^{\pi} T d\theta = 25 \text{ kNm}$$

$$\therefore \text{Power} = 25 \times \frac{2\pi \times 450}{60}$$

$$P = 1178.097 \text{ kN}$$

(ii) $\Delta T = T - T_r = -7.5 \sin 3\theta - 3.6 \sin\theta$

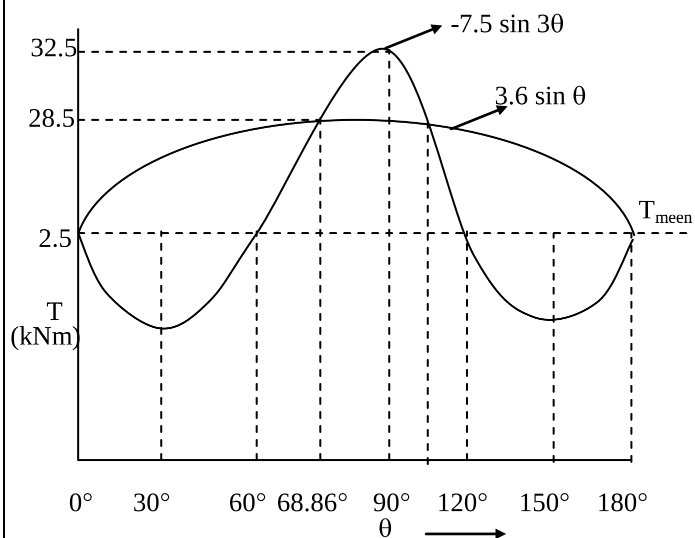
$\Delta T = 0$ when $7.5 \sin 3\theta + 3.6 \sin\theta = 0$

$\Rightarrow 7.5\,(3\sin\theta - 4\sin^3\theta) + 3.6 \sin\theta = 0$

$\Rightarrow \sin\theta\,(26.1 - 30\sin^2\theta) = 0$

$\Rightarrow \sin\theta = \pm 0.9327$

$\Rightarrow \theta = 68.859°$ or $\theta = 111.141$ or $\theta = 0°$




From the figure maximum fluctuation occurs between 0° to 68.859°

$$\therefore (\Delta KE)_{max} = \Delta E = \int_{0°}^{68.859°} \Delta T d\theta$$

$$\Delta E = \int_{0°}^{68.859°} -7.5 \sin 3\theta - 3.6 \sin\theta$$

$$= \left[\frac{7.5\cos 3\theta}{3} + 3.6\cos\theta\right]_{0^\circ}^{68.859^\circ}$$

$$= -4.7358 - 2.3016$$
$$= 7.0374 \text{ kNm}$$

$\therefore$ Maximum fluctuation of energy per cycle is 7.0374 kN-m

(iii) $$C_s = \frac{\Delta E}{I\omega^2} = \frac{7.0374\times 10^3}{360\times\left(\frac{2\times\pi\times 450}{60}\right)^2}$$

$$C_s = 0.88\%$$

**05(c).**

**Sol:**

| Uniform Pressure Theory | Uniform Wear Theory |
|---|---|
| 1. This theory is applicable only when the friction lining is new. | 1. This theory is applicable when the friction lining gets worn out. |
| 2. The friction radius for new clutches is more. | 2. The friction or effective radius is less. |
| 3. Torque transmission capacity is more. | 3. Torque transmission capacity is less. |
| 4. Wear is directly proportional to radius. | 4. Wear is directly proportional to product of pressure and radius. |

*(i)* Outer radius, $r_1 = 100$ mm,

Inner radius, $r_2 = 90$ mm

*Uniform pressure theory:*

Friction radius, $$r_{fp} = \frac{2}{3}\left(\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right)$$

$$= \frac{2}{3}\left(\frac{100^3 - 90^3}{100^2 - 90^2}\right) = 95.1\,\text{mm}$$

*Uniform wear theory:*

Friction radius,

$$r_{fw} = \frac{r_1 + r_2}{2} = \frac{100+90}{2} = 95\,\text{mm}$$

Since, $r_{fp} \approx r_{fw}$, torque transmission is approximately same in uniform pressure and uniform wear theories.

*(ii)* Outer radius, $r_1 = 100$ mm

Inner radius, $r_2 = 25$ mm

*Uniform pressure theory:*

$$r_{fp} = \frac{2}{3}\left(\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right)$$

$$= \frac{2}{3}\left(\frac{100^3 - 25^3}{100^2 - 25^2}\right) = 70\,\text{mm}$$

*Uniform wear theory:*

$$r_{fw} = \frac{r_1 + r_2}{2} = \frac{100+25}{2} = 62.5\,\text{mm}$$

Since, $r_{fp} > r_{fw}$

Torque transmitted under uniform pressure is much more than that of uniform wear theory.

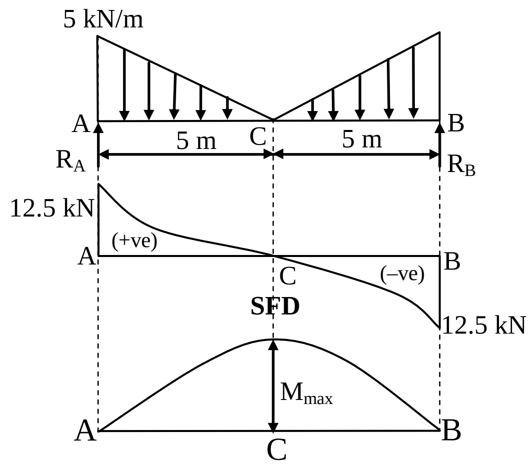
**05(d).**

**Sol: Comparison between Involute and Cycloidal tooth profiles:**

| Involute Teeth | Cycloidal Teeth |
|---|---|
| 1. Being the angle made by common tangent of base circles with common tangent to pitch circles at pitch point, the pressure angle remains constant throughout the engagement. This ensures smooth running of the gears. | Pressure angle varies continuously; being zero at the pitch point and maximum at the point of engagement and disengagement. This causes continuous variation in power component and also in bearing load. The running is less smooth. |
| 2. Tooth profile consists of a single (involute) curve and the rack cutter used for generating the profile has straight teeth. The rack cutter is cheaper and the method of manufacture is simpler. This leads to reduction in the cost of manufacture of involute teeth | The tooth profile consists of two curves- epicycloid and hypocycloid. The method of manufacture, therefore, is more involved and leads to costlier gear tooth. |
| 3. Perhaps the most desirable feature of involute teeth is that a small variation in centre distance does not change the velocity ratio. Thus distance between shafts need not necessarily be maintained exact as per design specifications. This gives great flexibility during assembly and larger tolerances may be permitted. | Exact centre distance is necessary to transmit constant velocity ratio. |
| 4. Since involute curve does not exist within base circle, interference is always possible if base circle radius is larger than dedendum circle radius. | Since outside the pitch (Directing) circle epicycloidal curve exists and inside it the hypocycloidal curve exists, cycloidal curve can exist everywhere on tooth profile and no interference exists. |
| 5. The radius of curvature of involute curve, near the base circle, is quite small and contact stresses are likely to be very high. The tooth profile in flank portion is almost radial. Both the factors together produce a tooth weaker in flank region compared to cycloidal tooth. | Cycloidal curve (hypocycloidal in particular) produces a spreading flank and, for this reason, cycloidal tooth is stronger compared to involute tooth. |
| 6. Convex surface of pinion tooth comes in contact with convex portion of gear tooth and this leads to more wear. | In cycloidal tooth profile epicycloidal shape of face of gear tooth comes in contact with hypocycloidal flank portion of pinion tooth. Thus a convex flank has a contact with concave face which results in lesser wear. |

**06(a).**

**Sol:**




**Reaction:**

Due to symmetry of the beam,

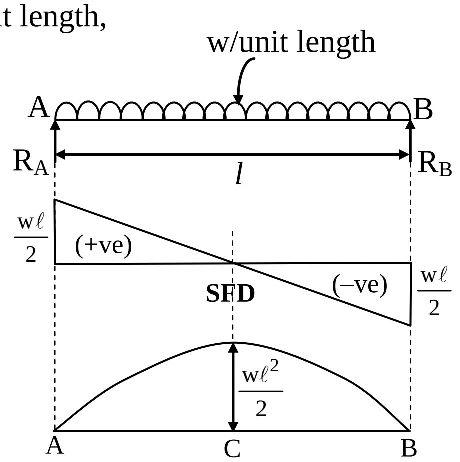
$$R_A = R_B = \frac{1}{2}\times 5\times 5 = 12.5\ \text{kN}$$

Thus, maximum shear force = 12.5 kN

Bending moment is maximum at the point of zero shear force, (i.e., centre of beam)

$$M_C = M_{max} = R_B(5) - \frac{1}{2}\times 5\times 5\times\frac{2}{3}\times 5$$
$$= 20.83\ \text{kN.m}$$

By replacing the given beam with the simply supported beam subjected to uniformly distributed load of intensity 'w' per unit length,




Maximum shear force = Reaction at the support

$$= \frac{w\ell}{2} = 12.5\ \text{kN} \quad \text{------(1)}$$

Maximum bending moment,

$$M_{max} = \frac{w\ell^2}{8} = 20.83\ \text{kN.m} \quad \text{------(2)}$$

By taking rato of equation (1) and (2),

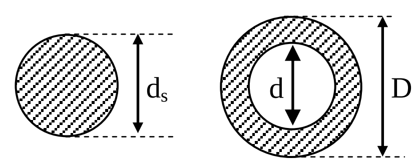
$$\frac{\frac{w\ell}{2}}{\frac{w\ell^2}{8}} = \frac{12.5}{20.8}$$

$$\therefore\ \ell = \frac{20.8\times 8}{12.5\times 2} = 6.67\ \text{m}$$

$$\therefore\ w = \frac{12.5\times 2}{6.67} = 3.74\ \text{kN/m}$$

**06(b).**

**Sol:**




$$K = \frac{d}{D} = \frac{d}{2d} = \frac{1}{2}$$

Given condition for a same material,

$$\tau_S = \tau_h$$

$$\therefore\ \frac{T_S}{(Z_p)_S} = \frac{T_h}{(Z_p)_h}$$

Also given that, $T_S = T_h = T$

$$\therefore (Z_p)_S = (Z_p)_h$$

$$\therefore\ \frac{\pi}{16}d_S^3 = \frac{\pi}{16D}\left(D^4 - d^4\right)$$

$$\therefore\ 100^3 = \frac{D^4}{D}\left(1-0.5^4\right)$$

$\therefore$ D = 102.2 mm and d = 51.1 mm

Now, torsional rigidity = GJ

$$\therefore\ \frac{(GJ)_S}{(GJ)_h} = \frac{GJ_S}{GJ_h} \quad (\because G_S = G_h = G)$$

$$= \frac{J_S}{J_R} = \frac{\frac{\pi}{32}\left(d_S^4\right)}{\frac{\pi}{32}\left(D^4-d^4\right)} = \frac{100^4}{\left(102.2^4-51.1^4\right)}$$

= 0.98

**06(c).**

**Sol:** Length of cylinder, L = 3000 mm

Diameter of cylinder, D = 1000 mm

Thickness, t = 10 mm

Internal pressure, p = 1.5 MPa

E = 200 GPa,

$\mu$ = 0.3

$$\frac{D}{20} = \frac{1000}{20} = 50\text{mm}$$

As t < D/20 so it is a thin pressure vessel

Hoop stress,

$$\sigma_h = \frac{PD}{2t} = \frac{1.5\times1000}{2\times10} = 75\,\text{MPa}$$

Longitudinal stress,

$$\sigma_\ell = \frac{PD}{4t} = \frac{1}{2}\sigma_h = \frac{75}{2} = 37.5\,\text{MPa}$$

Maximum shear stress,

$$\tau_{max} = \sigma_\ell = 37.5\,\text{MPa}$$

Strains

$$\varepsilon_h = \frac{\delta D}{D} = \frac{\sigma_h}{E} - \mu\frac{\sigma_\ell}{E}$$

$$= \frac{75}{2\times10^5} - 0.3\times\frac{37.5}{2\times10^5}$$

$= 3.1875 \times 10^{-4}$

Change in dia of thin cylinder

$$\delta D = 3.1875\times10^{-4}\times1000 = 0.319\text{ mm}$$

$$\varepsilon_\ell = \frac{\delta\ell}{\ell} = \frac{\sigma_\ell}{E} - \mu\frac{\sigma_h}{E}$$

$$= \frac{37.5}{2\times10^5} - 0.3\times\frac{75}{2\times10^5}$$

$$\frac{\delta\ell}{3000} = 7.5\times10^{-5} \Rightarrow \delta\ell = 0.225\,\text{mm}$$

Change in volume $\varepsilon_v = \varepsilon_\ell + 2\varepsilon_h$

$$\varepsilon_V = \frac{\delta v}{v} = 3.1875\times10^{-4} + 2\times7.5\times10^{-5}$$

$$= 4.6875\times10^{-4}$$

$$\delta V = 4.6875\times10^{-4}\times\frac{\pi}{4}\times1000^2\times3000$$

$= 1.104\times10^6$ mm$^3$ = 11.4 cc

**06(d).**

**Sol:** $\Sigma F_H = 0$

$N_B = F_A = \mu N_A$

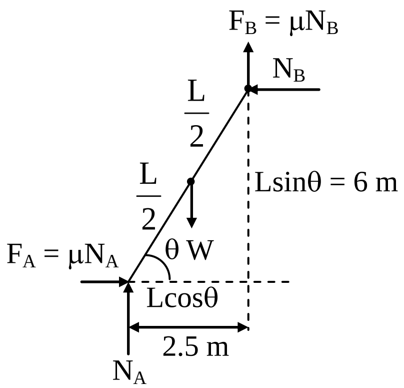
$\Sigma F_V = 0$

$N_A + F_B = W$

$N_A + \mu N_B = W$

$N_A + \mu^2 N_A = W$

$$N_A = \frac{W}{1+\mu^2} \quad \& \quad N_B = \frac{\mu W}{1+\mu^2}$$




$\Sigma M_A = 0$

$$\Rightarrow N_B\, L \sin\theta + F_B\, L\cos\theta = \frac{WL\cos\theta}{2}$$

$$\frac{\mu W}{1+\mu^2}\sin\theta + \frac{\mu^2 W}{1+\mu^2}\cos\theta = \frac{W}{2}\cos\theta$$

$$2\mu\tan\theta + 2\mu^2 = 1 + \mu^2$$

$$\mu^2 + 2\mu\tan\theta - 1 = 0$$

$$\mu = \frac{-2\tan\theta \pm \sqrt{4\tan^2\theta + 4}}{2}$$

$$= \pm\sec\theta - \tan\theta$$

$$= \pm\frac{6.5}{2.5} - \frac{6}{2.5} = 0.2 \text{ or } -5$$

$\mu = 0.2$ (as $\mu$ cannot be negative)