DETAILED SOLUTIONS

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ESE 2020 : Prelims Exam
CLASSROOM TEST SERIES

MECHANICAL ENGINEERING

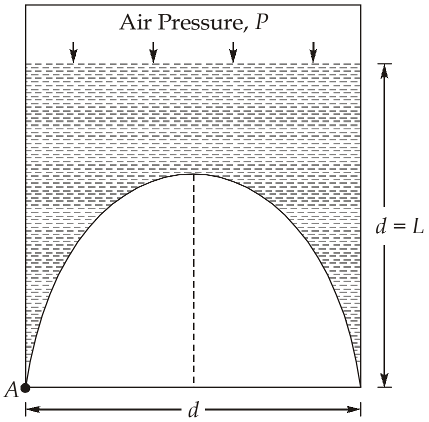
Test 6

Section A : Fluid Mechanics and Turbo Machinery[All Topics]
Section B : Production Engg. & Material Science-1 [Part Syllabus]
Section C : Thermodynamics-2 + RAC-2 [Part Syllabus]

1.	**(c)**	**16.**	**(c)**	**31.**	**(b)**	**46.**	**(b)**	**61.**	**(a)**
2.	**(d)**	**17.**	**(d)**	**32.**	**(b)**	**47.**	**(c)**	**62.**	**(a)**
3.	**(b)**	**18.**	**(d)**	**33.**	**(c)**	**48.**	**(b)**	**63.**	**(b)**
4.	**(b)**	**19.**	**(c)**	**34.**	**(b)**	**49.**	**(b)**	**64.**	**(a)**
5.	**(d)**	**20.**	**(c)**	**35.**	**(a)**	**50.**	**(c)**	**65.**	**(a)**
6.	**(b)**	**21.**	**(b)**	**36.**	**(d)**	**51.**	**(d)**	**66.**	**(b)**
7.	**(c)**	**22.**	**(c)**	**37.**	**(b)**	**52.**	**(b)**	**67.**	**(a)**
8.	**(b)**	**23.**	**(a)**	**38.**	**(a)**	**53.**	**(a)**	**68.**	**(d)**
9.	**(d)**	**24.**	**(d)**	**39.**	**(c)**	**54.**	**(b)**	**69.**	**(b)**
10.	**(b)**	**25.**	**(c)**	**40.**	**(a)**	**55.**	**(d)**	**70.**	**(d)**
11.	**(d)**	**26.**	**(a)**	**41.**	**(b)**	**56.**	**(b)**	**71.**	**(c)**
12.	**(d)**	**27.**	**(c)**	**42.**	**(c)**	**57.**	**(c)**	**72.**	**(d)**
13.	**(c)**	**28.**	**(c)**	**43.**	**(b)**	**58.**	**(b)**	**73.**	**(b)**
14.	**(d)**	**29.**	**(c)**	**44.**	**(b)**	**59.**	**(c)**	**74.**	**(a)**
15.	**(c)**	**30.**	**(b)**	**45.**	**(d)**	**60.**	**(a)**	**75.**	**(b)**

DETAILED EXPLANATIONS :

1. (c)



Let ρ = density of water.

Due to symmetry, horizontal force will be cancelled on two halves of hemisphere.

Now, we need to convert the air pressure above the water level into head of water, i.e.,

$$P = \rho g h; \; h = \frac{P}{\rho g}$$

Total height of water level in tank, $H = (h + d)$

Given: $$P_A = P + \rho g d$$

$$\Rightarrow \quad d = \frac{P_A - P}{\rho g}$$

Volume of water above hemisphere,

$$V = \left(\frac{\pi}{4} d^2 \cdot H - \frac{\pi d^3}{12}\right) \quad (\text{Volume of hemisphere} = \frac{\pi d^3}{12})$$

$$V = \frac{\pi}{4} d^2 \left[H - \frac{d}{3}\right]$$

Vertical force, F_V = Weight of water above hemispherical bulge.

$$F_V = \rho g V = \rho g \frac{\pi}{4} d^2 \left[H - \frac{d}{3}\right]$$

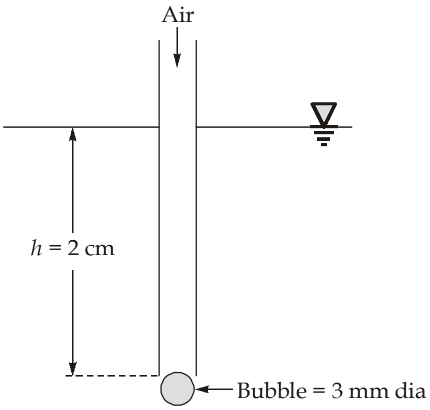
$$= \rho g A\left[h + d - \frac{d}{3}\right] = \rho g A\left[h + \frac{2d}{3}\right] = \rho g A\left[\frac{P}{\rho g} + \frac{2}{3}\left(\frac{P_A - P}{\rho g}\right)\right]$$

$$= A\left[\frac{2P_A}{3} + \frac{P}{3}\right]$$

2. (d)

1. No slip condition is applicable to flow of all real fluids.
2. The Mohr circle for fluid element inside the fluid body at rest is point on normal stress axis.

3. (b)



Pressure inside the bubble, $P_i = 210\ \text{N/m}^2$

Internal diameter of tube, $d_{\text{tube}} = 3\ \text{mm} = 3 \times 10^{-3}\ \text{m}$

Diameter of bubble should be same as diameter of tube.

$\therefore\ d_{\text{bubble}} = d_{\text{tube}} = 3 \times 10^{-3}\ \text{m}$

$$\text{Density of fluid} = (\text{Specific gravity}) \times 1000 = 0.80 \times 1000 = 800\ \text{kg/m}^3$$

Pressure outside the bubble,
$$P_o = \rho g h = 800 \times 10 \times \frac{2}{100} = 160\ \text{N/m}^2$$

Difference in pressure, $\Delta P = P_i - P_o = (210 - 160)\ \text{N/m}^2 = 50\ \text{N/m}^2$

We know that for a bubble,
$$\Delta P = \frac{4\sigma}{d_{\text{bubble}}} = \frac{4\sigma}{3 \times 10^{-3}}$$

$$50 = \frac{4\sigma}{3 \times 10^{-3}}$$

$$\sigma = \frac{0.150}{4} = 0.0375\ \text{N/m}$$

4. (b)

$$\text{Volume of balloon, } V = \frac{\pi d^3}{6} = \frac{\pi (14)^3}{6}$$

The forces acting on the balloon in vertical direction are :

1. Weight (combined load) of basket and goods (downward)
2. Weight of hot air inside the balloon (downward)
3. Buoyancy force (upward)

Now $\Sigma F_y = 0$

$$F_{\text{buoyancy}} - W_{\text{hot air}} - W_{\text{basket}} = 0$$

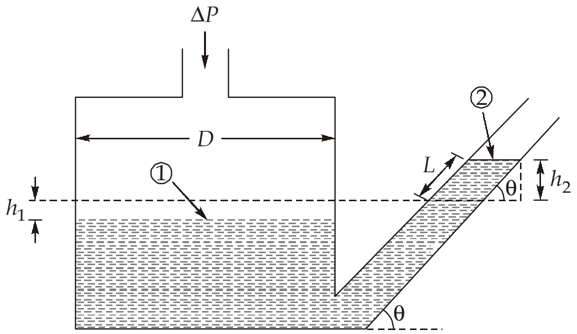
$$\rho_{\text{atm}} g V - \rho_{\text{hot air}} g V - 4312 = 0$$

$$\rho_{\text{hot air}} = \rho_{\text{atm}} - \frac{4312}{gV} = 1.3 - \frac{6\times 4312}{10\times\pi(14)^3}$$

$$= 1.3 - \frac{6\times 4312\times 7}{10\times 22\times 14\times 14^2} = 1.3 - \frac{6\times 14^2\times 22}{10\times 22\times 2\times 14^2}$$

$$= 1.3 - 0.3 = 1 \text{ kg/m}^3$$

5. (d)



Assume static and incompressible fluid.

By hydrostatic law, between (1) and (2),

$$P_1 - P_2 = \Delta P = \rho_l g(h_1 + h_2) \quad \text{...(1)}$$

Since volume of manometer liquid is constant.

∴ Volume displaced from reservoir = Volume that rises in the tube

$$\frac{\pi}{4}D^2 h_1 = \frac{\pi d^2}{4}L$$

$$h_1 = L\left(\frac{d}{D}\right)^2 \quad \text{...(2)}$$

and $h_2 = L\sin\theta$ ...(3) (from geometry of manometer)

From (1), (2) and (3)

$$\Delta P = \rho_l g\left[L\sin\theta + L\left(\frac{d}{D}\right)^2\right]$$

$$L = \frac{\Delta P}{\rho_l g\left[\sin\theta + \left(\frac{d}{D}\right)^2\right]}$$

6. (b)

Given, $V_r = 0,\ V_\theta = 3.0r$

$$\text{Vorticity} = 2 \times \text{rotation } (\omega) = \nabla\times\vec{V}$$

For motion in r–θ plane, the component of rotation and vorticity will be in in z-direction only.

$$\text{Vorticity} = 2\omega_z = \frac{1}{r}\frac{\partial(rV_\theta)}{\partial r} - \frac{1}{r}\frac{\partial(V_r)}{\partial\theta}$$

as $V_r = 0$,

$$\text{Vorticity} = 2\omega_z = \frac{1}{r}\frac{\partial(rV_\theta)}{\partial r}$$

$$\text{Vorticity} = \frac{1}{r}\frac{\partial(r\cdot 3r)}{\partial r} = \frac{1}{r}(3)\times\frac{\partial(r^2)}{\partial r} = \frac{3\times 2r}{r}$$

$$\text{Vorticity} = 6\ \text{rad/s}$$

$$\text{Circulation} = \oint_C V\cdot dS = \text{Vorticity}\times A$$

$$= 6\times\pi r^2 = 6\times\frac{22}{7}\times 1.4\times 1.4$$

$$= 6\times 2\times 11\times 0.28 = 12\times 0.28\times 11$$

$$= 3.36\times 11 = 36.96 \approx 37\ \text{m}^2/\text{s}$$

7. **(c)**

Given,

$$\phi = 2(x^2 + 2y - y^2)$$

$$u = \frac{\partial\phi}{\partial x} = 4x$$

$$v = \frac{\partial\phi}{\partial y} = 4 - 4y$$

Equation of streamline,

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

$$\frac{dy}{4-4y} = \frac{dx}{4x}$$

$$\frac{dy}{(1-y)} = \frac{dx}{x}$$

Integrating, we get,

$$\ln(1-y) = \ln x + \ln c$$

$$\ln(1-y) = \ln(xc)$$

$$1 - y = (x)(c)$$

At (1, 2),

$$1 - 2 = c \Rightarrow \quad c = -1$$

$\therefore$

$$1 - y = -x$$

$$y - x = 1$$

8. **(b)**

Given : Stagnation pressure, $P_{\text{stag}} = 210$ kPa

Static pressure, $P = 200$ kPa

Density, $\rho = 550\ \text{kg/m}^3$

$$P_{\text{stag}} = P + \frac{\rho V^2}{2000}$$

$$210 = 200 + \frac{550V^2}{2000}$$

$$\frac{10\times 2000}{550} = V^2$$

$$V = \sqrt{\frac{400}{11}} = \sqrt{36.36}$$
$$V = 6.03\ \text{m/s}$$

9. (d)

A pathline is a time exposed flow path of an individual particle over some time period.

10. (b)

$$\Delta h = y\left(\frac{S_m}{S_p} - 1\right)$$

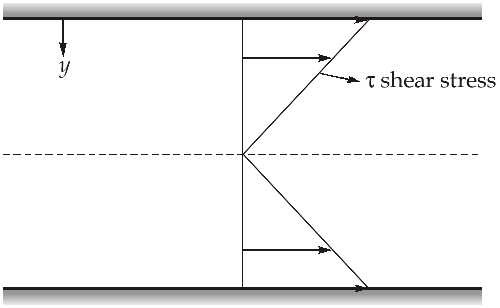
S_m = relative density of manometric fluid
S_p = relative density of fluid flowing in pipe

From above equation we can say that the Δh depends on gauge reading y regardless of the orientation of the venturimeter.

11. (d)

1. Shear stress variation for flow between two parallel plate is linear with maximum at boundary and zero at centre

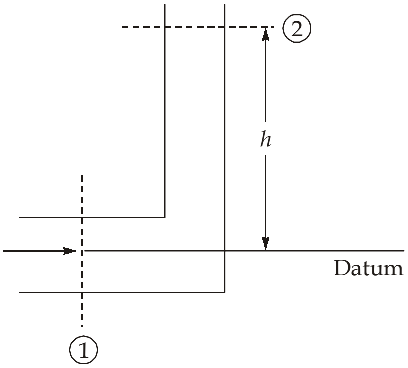
$$\tau = \tau_0\left(\frac{y}{B/2} - 1\right)$$



At $y = B/2$, $\tau = 0$, i.e., at centre.

2. The separation of boundary layer takes place when pressure gradient is positive and shear stress is zero as velocity gradient is also zero.
3. The intense mixing of fluid in turbulent flow as a result of rapid fluctuation enhances momentum transfer between fluid particles, which increases the friction force on pipewall. The friction factor reaches a maximum when flow becomes turbulent.

12. (d)



$$\begin{aligned} P_2 &= P_{atm} \\ (P_{gauge})_1 &= 120 \text{ kPa (atmospheric pressure is already included in gauge)} \\ \therefore \quad (P_1 - P_2) &= 120 \text{ kPa} \\ V_1 &= 5 \text{ m/s} \end{aligned}$$

For maximum height at 2, $V_2 = 0$

Applying equation between section (1) and (2)

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + Z_2 + h_f + h_{bend}$$

$$\left(\frac{P_1 - P_2}{\rho g}\right) + \frac{V_1^2}{2g} - \frac{V_2^2}{2g} - h_f - h_{\text{bend}} = (Z_2 - Z_1) = h$$

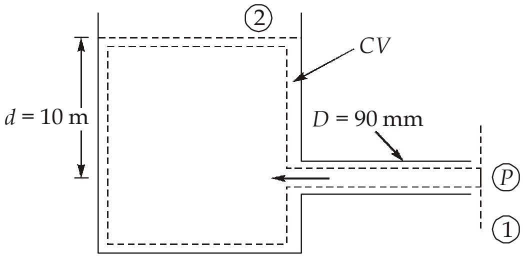
$$\frac{120 \times 10^3}{1000 \times 10} + \frac{5^2}{2 \times 10} - 2 - 0.8 = h$$

$$12 + 1.25 - 2 - 0.8 = h$$

$$\therefore \quad h = 10.45 \text{ m}$$

13. (c)

Given : $l = 100$ m, $D = 90$ mm $= 0.090$ m, $f = 0.01$, $V = 3$ m/s



Applying energy equation between (1) and (2)

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + Z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + Z_2 + h_f + h_e$$

$$\left(\frac{P_1 - P_2}{\rho g}\right) + \frac{V^2}{2g} = 0 + (Z_2 - Z_1) + \frac{flV^2}{2gD} + \frac{V^2}{2g}$$

($h_f = \dfrac{flV^2}{2gD}$, h_e = exit loss = $\dfrac{V^2}{2g}$ and $V_2 = 0$)

$$\frac{P_{\text{gauge}}}{\rho g} = d + \frac{flV^2}{2gD} \Rightarrow \Delta P = \rho\left[gd + f\frac{l}{D}\frac{V^2}{2}\right]$$

$$\Delta P = 1000\left[9.81\times 10 + \frac{0.01\times 100\times 3^2}{2\times 0.090}\right]$$

$$P_a = 1000\left[98.1 + \frac{9}{2\times 0.09}\right]$$

$$= 1000(98.1 + 50) = 148.1 \times 1000 \text{ Pa}$$

$$P_{(\text{gauge})} = 148.1 \text{ kPa}$$

14. (d)

Drag force, $F_D = \dfrac{1}{2}C_D\rho AV^2$

where C_D = Drag coefficient, ρ = Density, A = Projected area, V = Velocity

Power, $P = F_DV = \dfrac{1}{2}C_D\rho AV^2\cdot V$

$$P = \frac{C_D\rho AV^3}{2}$$

As given in question, P = Constant, and we can assume ρ and A to be constant.

$\therefore$ C_DV^3 = Constant

$$C_{D1}V_1^3 = C_{D2}V_2^3$$

Now, $C_{D_2} = 0.512C_{D_1}$ (as 48.8% reduction)

$$C_{D_1}V_1^3 = 0.512C_{D_1}V_2^3$$

$$\left(\frac{V_2}{V_1}\right) = \left(\frac{1}{0.512}\right)^{1/3} = \frac{1}{0.8} = \frac{5}{4} = 1.25$$

$$V_2 = 1.25V_1$$

$\therefore$ 25% increase in velocity.

15. (c)

For fully developed rough turbulent pipe flow, f is independent of Reynolds number. So, statement 2 is incorrect.

16. (c)

Option (c) is incorrect one as gravity effects are neglected and r-component of velocity u_r is zero as it is parallel flow.

17. (d)

For the laminar flow between two parallel stationary plate the velocity distribution is given by :

$$V = V_m\left[2\left(\frac{y}{B/2}\right)-\left(\frac{y}{B/2}\right)^2\right]$$

$$B = \text{Distance between the plates} = 8 \text{ mm}$$

$$V_m = \text{Centre line or maximum velocity} = 2.5 \text{ m/s}$$

$$y = \text{Distance from the boundary} = 2 \text{ mm}$$

$$\therefore \quad V = 2.5\left[2\left(\frac{2}{8/2}\right)-\left(\frac{2}{8/2}\right)^2\right]$$

$$= 2.5\left[\frac{4}{4}-\left(\frac{2}{4}\right)^2\right] = 2.5(1-0.5^2)$$

$$= 2.5 \times (1-0.25) = 2.5(0.75)$$

$$V = 1.875 \text{ m/s}$$

18. (d)

Both statements are incorrect. The correct statements are as :

1. The repeating variables must include among them all the fundamental dimensions, not necessarily in each one but collectively.
2. The dependent variable or the output parameter of the physical phenomenon should not be included in the repeating variables.

19. (c)

As per given information

$$Q = 0.2 \text{ m}^3/\text{s}$$

$$H_s = 4 \text{ m}$$

$$H_d = 16 \text{ m}$$

$$\text{Power required, } P = \rho g Q\,(H_s + H_d)$$

$$= 10^3 \times 10 \times 0.2(4+16)$$

$$= 40 \text{ kW}$$

20. (c)

The specific speed of a Pelton wheel depends on the ratio of jet diameter d and the wheel pitch diameter, D (the diameter at the centre of the bucket). If the hydraulic efficiency of a Pelton wheel is defined as the ratio of the power delivered (P) to the wheel to the head available H at the nozzle entrance, then we can write :

$$\text{Hydraulic power, } P = \rho Q g H \eta_h = \frac{\pi \rho d^2 V_1^3 \eta_h}{4 \times 2C_v^2} \qquad \ldots(1)$$

Since $$Q = \frac{\pi d^2}{4}V_1 \text{ and } V_1 = C_v(2gH)^{1/2}$$

The specific speed, $N_{ST} = \dfrac{NP^{1/2}}{H^{5/4}}$

- The optimum value of the overall efficiency of a Pelton turbine depends both on the values of the specific speed and the speed ratio.
- The Pelton wheels with a single jet operate in the specific speed range of 4-16, and therefore the ratio D/d lies between 6 to 26.
- A large value of D/d reduces the rpm as well as the mechanical efficiency of the wheel. It is possible to increase the specific speed by choosing a lower value of D/d, but the efficiency will decrease because of the close spacing of buckets.
- The value of D/d is normally kept between 14 and 16 to maintain high efficiency.

21. **(b)**

As per given data :

$$P = 6600 \text{ kW} \qquad H = 25 \text{ m}, \; N = 100 \text{ rpm}$$

$$N_{ST} = \frac{N\sqrt{P}}{H^{5/4}} = \frac{100\sqrt{6600}}{25^{5/4}} = \frac{100\times 10\sqrt{66}}{25^{5/4}} = 145.33$$

In this case, since $50 < N_S < 400$, it can be a Francis turbine.

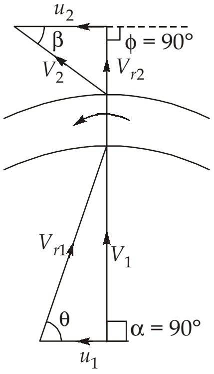
22. **(c)**

As we know:

$$V_{w_2} = u_2 - V_{f2}\cot\phi$$

$$\text{Discharge, } Q = \pi D_2 B_2 V_{f2}$$

$$V_{f2} = \frac{Q}{\pi D_2 B_2}$$



So,

$$V_{w2} = u_2 - \frac{Q}{\pi D_2 B_2}\cot\phi$$

$$\text{Head, } H = \frac{V_{w2}u_2}{g} = \left(u_2 - \frac{Q}{\pi D_2 B_2}\cot\phi\right)\frac{u_2}{g}$$

$$= \frac{u_2^2}{g} - \frac{u_2 \times Q}{\pi D_2 B_2}\cot\phi$$

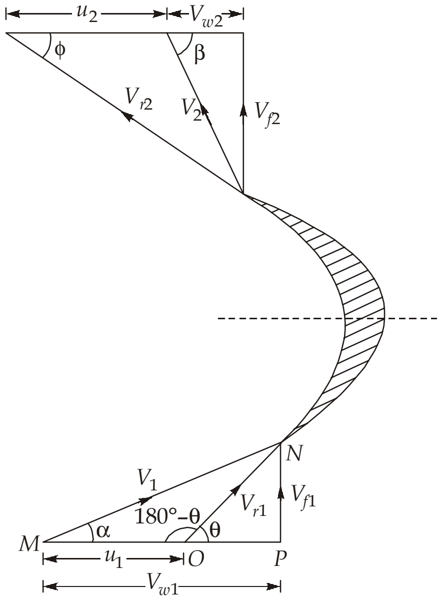
For radial vane,

$$\phi = 90°$$

$$\text{Then, } H = \frac{u_2^2}{g}$$

So, theoretical head = $\dfrac{u_2^2}{g}$ which is independent from discharge.

24. (d)



As per the given information :

$$V_1 = 20 \text{ m/s}$$

$$\theta = 30°$$

$$u_1 = u_2 = u = 5 \text{ m/s}$$

From the velocity triangle at inlet ∠MON

$$\frac{V_1}{\sin(180° - \theta)} = \frac{u_1}{\sin(\theta - \alpha)}$$

$$\frac{20}{\sin(180° - \theta)} = \frac{5}{\sin(30 - \alpha)}$$

$$\Rightarrow \quad \frac{20}{\sin\theta} = \frac{5}{\sin(30 - \alpha)}$$

$$\sin(30° - \alpha) = \frac{1}{2} \times \frac{1}{4} = 0.125$$

$$30° - \alpha = \sin^{-1}(0.125) = 7.18°$$

$$\alpha = 30° - 7.18°$$

$$\alpha = 22.82°$$

25. (c)

According to the given data :

$$A = \frac{\pi}{4} \times 0.25^2$$

$$L = 0.40 \text{ m}$$

$$N = 30 \text{ rpm}$$

$$Q_{\text{act}} = 0.009 \text{ m}^3/\text{s}$$

$$H_s = 7 \text{ m}$$

$$H_d = 15 \text{ m}$$

% of Slip,

$$\text{Slip} = \frac{(Q_{\text{th}} - Q_{\text{act}})\times 100}{Q_{\text{th}}}$$

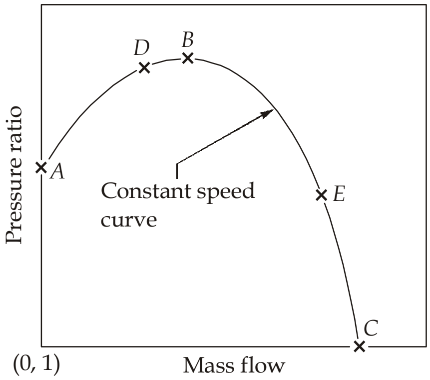
$$Q_{\text{th}} = \frac{ALN}{60} = \frac{\frac{\pi}{4}\times 0.25^2 \times 0.4 \times 30}{60} \ \text{m}^3/\text{s}$$

$$Q_{\text{th}} = 0.009817 \text{ m}^3/\text{s}$$

$$\text{Slip} = \left|\frac{0.009 - 0.009817}{0.009817}\right| \times 100$$

$$= 8.322\%$$

26. (a)



- Surging in compressor can be understood by following way: If we suppose that the compressor is operating at a point '*D*' on the part of characteristics curve having a positive slope, then a decrease in mass flow will be accompanied by a fall in delivery pressure.
- If the pressure of the air downstream of the compressor does not fall quickly enough, the air will tend to reverse its direction and will flow back in the direction of the resulting pressure gradient.
- When this occurs, the pressure ratio drops rapidly causing a further drop in mass flow until the point '*A*' is reached, where the mass flow is zero. When the pressure downstream of the compressor has reduced sufficiently due to reduced mass flow rate, the positive flow becomes established again and the compressor picks up to repeat the cycle of events which occurs at high frequency.
- This surging of air may not happen immediately when the operating point moves to the left of '*B*' because the pressure downstream of the compressor may at first fall at a greater rate than the delivery pressure.
- As the mass flow is reduced further, the flow reversal may occur and the conditions are unstable between '*A*' and '*B*'.
- As long as the operating point is on the part of the characteristics having a negative slope, however, decrease in mass flow is accompanied by a rise in delivery pressure and the operation is stable.
- Choking represents a point 'E' on the characteristics curve having limit of maximum flow rate at the particular rotational speed for which the curve is drawn.

27. **(c)**

As per given information :

Prototype	Model
D_P	$D_m = \frac{D_P}{10}$
$H_P = 40$ m,	$H_m = 4$ m
	$P_m = 8$ kW
	$N_m = 500$ rpm

Model to prototype similarity relationship

$$\left(\frac{H}{N^2D^2}\right)_P = \left(\frac{H}{N^2D^2}\right)_m$$

$$\left(\frac{H_P}{H_m}\right)\left(\frac{D_m}{D_P}\right)^2 \times N_m^{\ 2} = N_p^{\ 2}$$

$$\left(\frac{40}{4}\right)\left(\frac{1}{10}\right)^2 \times 500^2 = N_p^{\ 2}$$

$$N_P = (10)^{1/2}\left(\frac{1}{10}\right)(500)$$

$$N_P = 3.16 \times 50 = 158 \text{ rpm}$$

28. **(c)**

- The first term of equation is readily seen to be the change in absolute kinetic energy or dynamic head of the fluid while flowing through the rotor.
- The second term of equation represents a change in fluid energy due to the movement of the rotating fluid from one radius of rotation to another.
- As the work is done on or by the fluid element due to its displacement from radius r_1 to radius r_2 and hence becomes equal to the energy held or lost by it. Since the centrifugal force field is responsible for this energy transfer, the corresponding head (energy per unit weight) $U^2/2g$ is termed as centrifugal head. The transfer of energy due to a change in centrifugal head $\left[\frac{(U_2^2 - U_1^2)}{2g}\right]$ causing a change in the static head of the fluid.
- The third term represents a change in the static head due to a change in fluid velocity relative to the rotor.
- Regarding the effect of flow area on fluid velocity V_r relative to the rotor, a converging passage in the direction of flow through the rotor increases the relative velocity ($V_{r2} > V_{r1}$) and hence decreases the static pressure. This usually happens in case of turbines.

  Similarly, a diverging passage in the direction of flow through the rotor decreases the relative velocity ($V_{r2} < V_{r1}$) and increases the static pressure as occurs in case of pumps and compressors.

29. **(c)**

As given :

$$\begin{aligned}\text{The velocity of blade tip,} \quad u_2 &= 450\ \text{m/s}\\ \text{Slip factor, } \sigma &= 0.8\\ V_{w1} &= 0,\ V_{w2} = \sigma u_2\\ \text{Work done per kg of flow} &= V_{w2}u_2\\ &= \sigma u_2^2\\ &= 0.8 \times (450)^2 = 162000 = 162\ \text{kJ/kg}\end{aligned}$$

30. **(b)**

As per given data

$$\eta_h = 1 - 0.22 = 0.78$$

From radial discharge,

$$\eta_h = \frac{V_{w1}u_1 \times \rho Q}{\rho Q H \times g}$$

$$\eta_h = \frac{V_{w1}u_1}{g \cdot H} \quad \ldots(1)$$

As we know,

$$H = 8\ \text{m}$$

$$u_1 = 12.03\ \text{m/s}$$

$$V_{f1} = 4.51\ \text{m/s}$$

From equation (1)

$$0.78 = \frac{V_{w1} \times 12.03}{9.81 \times 8}$$

$$V_{w1} = \frac{0.78 \times 9.81 \times 8}{12.03}$$

$$= \frac{0.78 \times 78.48}{12.03}$$

$$= \frac{61.21}{12.03} = 5.08\ \text{m/s}$$

31. **(b)**

In aircraft gas turbine cycles, the useful power developed in order to run compressor and other accessories but not used for thrust. In case of turbojet and turbofan, the whole thrust is generated in the propelling nozzle, whereas with the turboprop most is produced by a propeller with only a small contribution from the exhaust nozzle.

33. **(c)**

The device in which the kinetic, potential or intermolecular energy held by the fluid is converted in the form of mechanical energy by a rotating member is known as a turbine.

34. **(b)**

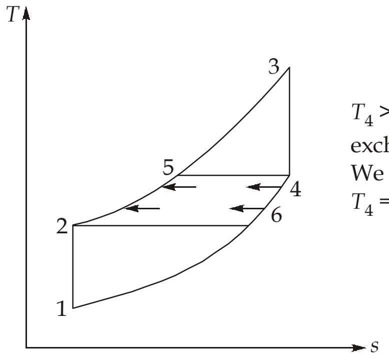
The friction head due to friction in suction and delivery pipes

$$h_{fs} = \frac{fL_s}{2gd_s}\left(\frac{A_p}{A_s} r\omega \sin\theta\right)^2$$

$$(h_{fs})_{\max} \Rightarrow (\sin\theta)_{\max} \Rightarrow \text{at } \theta = 90^\circ$$

$$(h_{fs})_{\max} = \frac{fL_s}{2gd_s}\left(\frac{A_p}{A_s} r\omega\right)^2$$

35. **(a)**



$T_4 > T_2$ for heat exchange to take place
We assume ideal exchanger
$T_4 = T_5$ and $T_6 = T_2$

Simple gas turbine cycle with heat exchange

With ideal heat exchanger, the cycle efficiency can be expressed as

$$\eta_{\text{with}} = \frac{c_p(T_3 - T_4) - c_p(T_2 - T_1)}{c_p(T_3 - T_5)}$$

$$\eta_{\text{without}} = \frac{c_p(T_3 - T_4) - c_p(T_2 - T_1)}{c_p(T_3 - T_2)}$$

$$c_p(T_3 - T_2) > c_p(T_3 - T_5)$$

$$\eta_{\text{with}} > \eta_{\text{without}}$$

Hence, efficiency is more than that of simple cycle, with heat exchanger (ideal) the specific output does not change but the efficiency is increased.

36. **(d)**

- The machine for which the change in static head in the rotor is zero is known as impulse machine. In these machines, the energy transfer in the rotor takes place only by the change in dynamic head of the fluid.
- For an impulse machine $R = 0$, because there is no change in static pressure in the rotor. It is difficult to obtain a radial flow impulse machine, since the change in centrifugal head is obvious there. Nevertheless, an impulse machine of radial flow type can be conceived by having a change in static head in one direction contributed by the centrifugal effect and an equal change in the other direction contributed by the change in relative velocity.
- However, this has not been established in practice, thus for an axial flow impulse machine $U_1 = U_2$, $V_{r1} = V_{r2}$. For an impulse machine, the rotor can be made open, that is, the velocity V_1 can represent an open jet of fluid flowing through the rotor, which needs no casing. A very simple example of an impulse machine is a paddle wheel rotated by the impingement of water from a stationary nozzle.

37. (b)

Statement I is correct as for submarines, by locating engines and cabins for the crew at lower half, most of its weight will get shifted to bottom and an immersed body is stable if the body is bottom heavy and thus point *G* is directly below *B*.

Statement II is also correct because the centroid of displaced volume shifts to the side to a point *B*′ during rotational disturbance while *G* remains unchanged. If these two points are sufficiently far then there will be restoring moment and it will return the body to its original position. The measure of stability for floating bodies is metacentric height.

38. (a)

As the critical Reynolds number of dimped golf balls is 4×10^4 and generally at this value of Reynolds number flow is laminar and drag coefficient is high. Due to this reason golf balls are intentionally roughened to induce turbulance at lower Reynolds number to take the advantage of sharp drop in drag coefficient at the onset of turbulence in boundary layer.

39. (c)

Infiltration is the finishing operation in which powder metallurgy product is dipped into a low melting temperature alloy liquid, such that the liquid would flow into the voids simply by capillary action, thereby decreasing the porosity and improving the strength of the components. The process is used quite extensively with ferrous parts using copper as infiltrant but to avoid erosion, an alloy of copper containing iron and manganese, is often used.

40. (a)

Methods to reduce roll force in rolling:

1. Reduce friction.
2. Using smaller diameter rolls to reduce contact area.
3. Taking smaller reduction per pass.
4. Rolling at elevated temperature to reduce strength.
5. To apply longitudinal tension to strip during rolling, as a result compressive stresses required to deform the material plastically becomes smaller.

41. (b)

$$\text{Forward slip} = \frac{V_f - V}{V} = \frac{15 - 12.5}{12.5} = 0.2 = 20\%$$

43. (b)

From conversation of mass

$$\begin{aligned} A_i V_i &= A_f V_f \\ b_i\, t_i\, V_i &= b_f\, t_f\, V_f \\ 300 \times 8 \times 10 &= b_f \times 0.75 \times 8 \times 12.5 \\ b_f &= 320 \text{ mm} \end{aligned}$$

$$\%\text{ increase in width of plate} = \frac{320 - 300}{300} \times 100 = 6.67\%$$

44. (b)

If more heat is required at the workpiece side, such as for thicker sheets or for the work materials which have higher thermal conductivity such as aluminium and copper, the workpiece can be made as anode, liberating large heat near it. This is termed as straight polarity or DCEN (Direct Current Electrode Negative). This gives rise to higher peneration. However for thinner materials, where less heat is required in the weld zone or large deposition rate for wide gaps are rquired, the polarity could be reversed by making the workpiece as negative. This is termed as reversed polarity or DCEP (Direct Current Electrode Positive). In reversed polarity, the penetration is small.

46. (b)

Chaplets are used to support the core inside the mould cavity to take care of its own weight and overcome the metallostatic force.

47. (c)

In mass production, quality control is typically defined in terms of acceptable quality level or AQL, which means that a certain minimum level of fraction defects is tolerated. In lean production, by contrast, perfect quality is required. The just-in-time delivery discipline used in lean production necessitates a zero defects level in parts quality, because if the part delivered to the downstream workstation is defective, production is forced to stop. There is little or no inventory in a lean system to act as a buffer.

48. (b)

Screw rotation angle 'A_s' corresponding to a distance, x = 300 mm is

$$A_s = \frac{360° \times x}{P} = \frac{360° \times 300°}{6} = 18000°$$

Number of pulse to move table 300 mm is:

$$n_p = \frac{\text{Motor rotation angle}}{\text{Step angle}} = \frac{18000° \times 5 \times 50}{360°}$$

$$= 12500 \text{ pulses}$$

49. (b)

To qualify as being flexible, an automated manufacturing system should satisfy the following four tests of flexibility.

1. Part-variety test: Can the system process different part or product styles in a mixed-model (non-batch mode)?
2. Schedule-change test: Can the system readily accept the changes in production schedules, that is the changes in part mix and/or production quantities?
3. Error-recovery test: Can the system recover gracefully from equipment malfunctions and breakdowns, so that production is not completely disrupted?

4. New-part test: Can new part designs be introduced into the existing part mix with relative ease if their features qualify them as being members of the part family for which the system was designed? Also, can design changes be made in existing parts without undue challenge to the system?

These four tests, as they are called here, are sometimes referred to as types or dimensions of flexibility. The part-variety tests is called machine flexibility or production flexibility. The schedule change tests is called mix flexibility or volume flexibility. The error-recovery test is called routing flexibility, and the new-part test is called product flexibility.

50. (c)

For tetragonal crystal system, $a = b \neq c$

Here, $a = 0.4$ nm

$b = 0.4$ nm

$c = 0.55$ nm

So, $a = b \neq c$

Hence, it is tetragonal crystal system.

In (110) plane one atom is at centre, so it will be body centred.

Therefore, it will be body centred tetragonal.

51. (d)

	x	y	z
Intercept -	1	1/2	1/2
Reciprocals -	1	2	2
Enclosure -		(1 2 2)	

52. (b)

For BCC, highest planar density is along {1 1 0} slip plane and slip occurs along $<\bar{1}11>$-type direction as it has highest linear density.

54. (b)

Since the eutectic composition in silica-alumina phase diagram is very near to silica extremity (7.7 wt% Al_2O_3), even small additions of Al_2O_3 lowers the liquidus temperature significantly, which means the substantial amounts of liquid may be present at temperature in excess of 1600°C. Thus the alumina content should be held to a minimum normally in between 0.2 and 1.0 wt%.

56. (b)

The microstructural product of full annealing is coarse pearlite (in addition to any proeutectoid phase) that is relatively soft and ductile.

58. (b)

$$\text{Exergy, } \phi = U - U_o + p_o(V - V_o) - T_o(S - S_o) \qquad \text{... (i)}$$

Since vacuum has zero mass,

$$U = 0, \qquad S = 0$$

At dead state, $U_o = 0, \quad S_o = 0, \quad V_o = 0$

From (i),

$$\begin{aligned}\phi &= 0 - 0 + 101.3 \times (0.5 - 0) - T_o(0 - 0)\\ &= 50.65 \text{ kJ}\end{aligned}$$

59. (c)

$$\begin{aligned}\text{Number of moles of N}_2,\ n_1 &= 2 \text{ kmol}\\ \text{Number of moles of CO}_2,\ n_2 &= 1 \text{ kmol}\\ \text{Molar mass of N}_2,\ M_1 &= 28 \text{ kg/kmol}\\ \text{Molar mass of CO}_2,\ M_2 &= 44 \text{ kg/kmol}\end{aligned}$$

$$\text{Mass fraction of CO}_2,\ x_2 = \frac{n_2M_2}{n_1M_1 + n_2M_2} = \frac{1\times 44}{2\times 28 + 1\times 44} = \frac{44}{100} = 0.44$$

60. (a)

$$\begin{aligned}\text{Mass of water, } m &= 100 \times 1000 = 10^5 \text{ kg}\\ \text{Elevation, } h &= 60 \text{ m}\end{aligned}$$

Maximum power that can be generated,

$$\begin{aligned}W_{\max} &= mgh = 10^5 \times 9.81 \times 60/10^3 \text{ kJ}\\ &= 588.6 \times 10^2 \text{ kJ}\\ &= \frac{588.6\times 10^2}{3600} = \frac{588.6}{36} = 16.35 \text{ kWh}\end{aligned}$$

61. (a)

Given: $C = 0.04$, $r = 1.5$, $v = 0.176$ m^3/kg, $RE = 6$ kW

As,
$$\text{RE} = \dot{m}(h_1 - h_4)$$

$$6 = \dot{m}(173 - 65)$$

$$\dot{m} = \frac{6}{108}\text{ kg/ s}$$

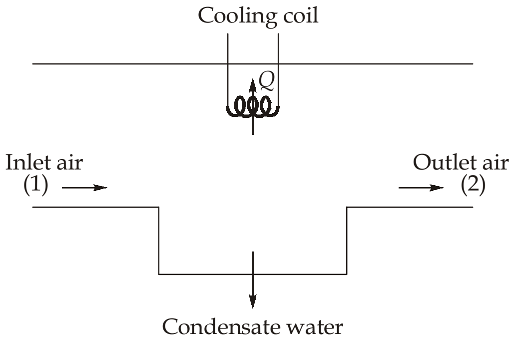
$$\begin{aligned}\eta_v &= 1 + C - C\left(\frac{P_2}{P_1}\right)^{1/\gamma} = 1 + 0.04 - 0.04\left(\frac{6.4}{0.8}\right)^{1/1.5}\\ &= 1 + 0.04 - 0.04\,(8)^{2/3}\\ &= 1 + 0.04 - 0.16 = 0.88\end{aligned}$$

As,
$$\eta_v = \frac{\text{Actual suction volume}}{\text{Displacement volume}}$$

$$0.88 = \frac{0.176\times\frac{6}{108}}{\dot{V}_s}$$

$\Rightarrow$
$$\dot{V}_s = 0.2\times\frac{1}{18} = \frac{0.1}{9} = 0.011\text{ m}^3/\text{ s}$$

62. (a)



Given: $\dot{m}_i = 3\text{ kg/s} = \dot{m}_e = \dot{m}, h_i = 80\text{ kJ/kg}, \omega_i = 0.020\text{ kg w.v/kg d.a.}$

$$h_o = 40\text{ kJ/kg},$$
$$\omega_o = 0.008\text{ kg w.v./kg d.a.}$$
$$h_w = 70\text{ kJ/kg}$$
$$\dot{m}_w = \dot{m}(\omega_i - \omega_o) = 3(0.020 - 0.008) = 0.036\text{ kg}$$
$$\dot{m}_i h_i = \dot{m}_o h_o + Q + \dot{m}_w h_w$$

As,
$$3 \times 80 = 3 \times 40 + Q + 0.036 \times 70$$
$$Q = 120 - 2.52$$
$$= 117.48\text{ kW}$$

63. (b)

As the system is closed, so there is no transfer of mass, the humidity ratio won't vary. As the temperature decreases, the water vapour holding capacity decreases and hence relative humidity increases.

64. (a)

We will assume that the relative humidity and temperature is almost constant in the atmosphere. For the initial condition, the water temperature is also 25°C.

Now,
$$\phi = \frac{P_v}{P_{vs}}$$
$$\Rightarrow \quad 0.6 = \frac{P_v}{3.17}$$
$$\Rightarrow \quad P_v = 1.902\text{ kPa}$$

As at 25°C, the vapour pressure of water in swimming pool, i.e., 3.17 kPa is more than the vapour pressure of moist air, i.e., 1.902 kPa. Water will evaporate extracting heat from remaining water which will decrease temperature of water in swimming pool. This process will continue till vapour pressure of both air and water will be equal. As ambient air will remain at vapour pressure of 1.902 kPa, the temperature of water will decrease till due point temperature.

By considering linear variation we will get

$$\frac{t_{dpt}-15}{1.902-1.71} = \frac{20-15}{2.34-1.71}$$

$$\Rightarrow \quad t_{dpt} - 15 \approx 5 \times 0.3$$

$$\Rightarrow \quad t_{dpt} - 15 \approx 1.5$$

$$\Rightarrow \quad t_{dpt} \approx 16.5°\text{C}$$

So, final temperature of water will be 16.5°C.

65. **(a)**

Given : $m_{wv} = 0.5$ kg, $m_{da} = 40$ kg, $T = 24°$C, $P = 100$ kPa, $P_{vs} = 3$ kPa

$$\text{Humidity ratio, } \omega = \frac{m_{wv}}{m_{da}} = \frac{0.622P_v}{P-P_v}$$

$$\Rightarrow \quad \frac{0.5}{40} = \frac{0.622P_v}{100-P_v}$$

$$\Rightarrow \quad 100 - P_v = 80 \times 0.622P_v$$

$$\Rightarrow \quad 100 = (1 + 49.76)P_v$$

$$\Rightarrow \quad P_v = \left(\frac{100}{50.76}\right)\text{kPa}$$

Now,

$$\text{Relative humidity, } \phi = \frac{P_v}{P_{vs}}$$

$$= \frac{\left(100/50.76\right)}{3} \approx \frac{2}{3} = 0.66$$

ϕ is almost equal to 0.66 or 66%.

66. **(b)**

In a thermoelectric refrigerator, the cooling capacity is equal to electric power supplied or even less than that because of some losses. While in VCRS, generally the COP is more than one.

67. **(a)**

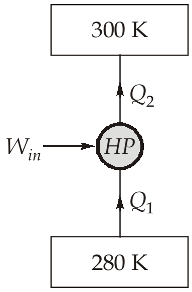
Given : $P_2 = 1.84$ bar, $P_1 = 0.23$ bar, $T_1 = 210$ K, $\gamma = 1.4$

$$\text{COP} = \frac{1}{r_p^{\frac{\gamma-1}{\gamma}}-1} = \frac{1}{\left(\frac{1.84}{0.23}\right)^{\frac{1.4-1}{1.4}}-1} = \frac{1}{8^{0.2857}-1} = 1.23$$

68. **(d)**

COP of electric resistance heater,

$$\text{COP} = \frac{\text{Heat supplied to building }(Q)}{\text{Electric work }(W)} = 1 \qquad [\because Q = W]$$



COP of reversible heat pump,

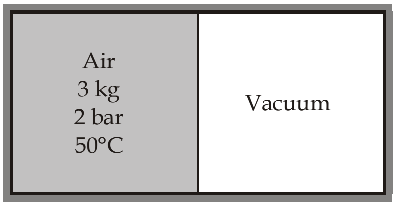
$$\text{COP}_{\text{rev}} = \frac{T_1}{T_1 - T_2} = \frac{300}{300-280} = \frac{300}{20} = 15$$

Second law efficiency,

$$\eta_{\text{II}} = \frac{\text{COP}}{\text{COP}_{\text{rev}}} = \frac{1}{15} \times 100 = 6.67\%$$

69. (b)

Given: $m = 3$ kg, $p_1 = 2$ bar, $T_1 = 50 + 273 = 323$ K, $T_o = 300$ K



From I$^{\text{st}}$ law of thermodynamics:

$$Q = \Delta U + W$$

$$0 = \Delta U + 0 \qquad [\because Q = 0 \text{ (insulated)}, W = 0 \text{ (Rigid)}]$$

$$0 = mc_v \,\Delta T$$

$$\Delta T = 0$$

$$T_2 - T_1 = 0$$

$$\Rightarrow \quad T_2 = T_1$$

Entropy change of air,

$$\Delta S = m\left[c_v \ln\frac{T_2}{T_1} + R\ln\frac{V_2}{V_1}\right]$$

$$= m\left[0 + R\ln\frac{2V_1}{V_1}\right] \qquad [\because V_2 = 2V_1, T_2 = T_1]$$

$$= 3 \times 0.287 \times \ln 2$$

$$= 3 \times 0.287 \times 0.693$$

$$\Delta S = 0.597 \text{ kJ/K}$$

Also,

$$\Delta S = \oint \frac{đQ}{T} + S_{gen} = 0 \qquad [\because Q = 0]$$

$$S_{gen} = \Delta S = 0.597 \text{ kJ/K}$$

Loss of available energy, $I = T_o\, S_{gen} = 300 \times 0.597$

$$I = 179.1 \text{ kJ}$$

71. **(c)**

Condenser is provided with fan in a split air conditioning unit which requires electric energy. There are auxillary electronic devices which also use miniscule amount of electric energy. So, the Statement II is not correct.

72. **(d)**

Flurocarbons are odourless. The halide torch method is used with fluorocarbons, in this methyl alcohol or hydrocarbon flame is used which is light blue in colour, but turns bluish green in the presence of halocarbon vapours.

73. **(b)**

Exergy change of a system can be positive or negative during a process but exergy destroyed can't be negative.

$$X_{\text{destroyed}} \begin{cases} > 0, \text{Irreversible} \\ = 0, \text{Reversible} \\ < 0, \text{Impossible} \end{cases}$$

74. **(a)**

Otto cycle is internally reversible cycle because it involves heat transfer through a finite temperature difference during heat addition and heat rejection process, which are irreversible. Therefore, thermal efficiency of Otto cycle is less than that of Carnot cycle operating between the same temperature limits.

75. **(b)**

Thermal comfort depends on mean radiant temperature. The mean radiant temperature is defined as the uniform temperature of an imaginary enclosure in which the radiant heat transfer from the human body is equal to radiant heat transfer in the actual non-uniform enclosure.

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