$$\tan \alpha_1 = \frac{132}{377}$$

$$\alpha_1 = 19{\cdot}36^\circ$$

$$\text{Nozzle angle} = 19{\cdot}36^\circ \quad \text{Ans. (a)(i)}$$

$$v_1 = \frac{377}{\cos \alpha_1}$$

$$= 399{\cdot}6 \text{ m/s} \quad \text{Ans. (a)(ii)}$$

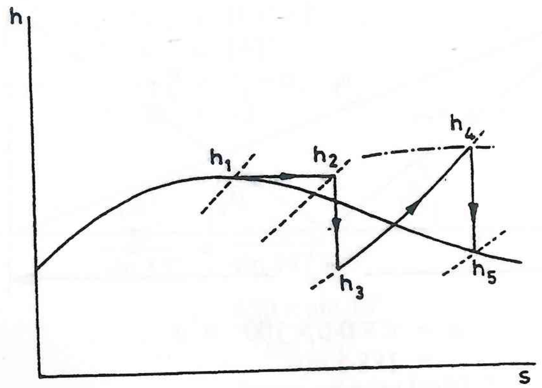
$$\text{Power} = \dot{m} v_w u$$

$$= 1 \times 377 \times 188{\cdot}5 \times 10^{-3}$$

$$= 71{\cdot}06 \text{ kW} \quad \text{Ans. (a)(iii)}$$

Assumed – no frictional losses across the blades Ans. (b)

5. From *h-s* chart $h_1 = 2795$ kJ/kg, $s_1 = 6{\cdot}27$ kJ/kg K
$h_2 = 2795$ kJ/kg
$h_3 = 2455$ kJ/kg
$h_4 = 2870$ kJ/kg
$h_5 = 2615$ kJ/kg, $s_5 = 7{\cdot}63$ kJ/kg K



Changes in enthalpy during each stage, final – initial
Stage 1 $h_2 - h_1 = 0$
Stage 2 $h_3 - h_2 = -340$ kJ/kg
Stage 3 $h_4 - h_3 = 415$ kJ/kg
Stage 4 $h_5 - h_4 = -255$ kJ/kg Ans.

Overall changes in entropy = $s_5 - s_1 = 1{\cdot}36$ kJ/kg K Ans.

Condition of steam at end of final expansion is *dry saturated*. Ans.

6.
$$p_T = p_1 \left[\frac{2}{\gamma + 1}\right]^{\frac{\gamma}{\gamma - 1}}$$

$$= 7 \left[\frac{2}{1{\cdot}67 + 1}\right]^{\frac{1{\cdot}67}{1{\cdot}67 - 1}}$$

$$= 3{\cdot}407 \text{ bar}$$

$$T_T = T_1 \left[\frac{p_T}{p_1}\right]^{\frac{\gamma - 1}{\gamma}}$$

$$T_T = 423 \left[\frac{3{\cdot}407}{7}\right]^{\frac{1{\cdot}67 - 1}{1{\cdot}67}}$$

$$= 316{\cdot}9 \text{ K or } 43{\cdot}87^\circ\text{C}$$

$$\text{Enthalpy drop to throat} = c_p[T_1 - T_T]$$

$$\text{i.e. } h = 833{\cdot}9[423 - 316{\cdot}87]$$

$$h = 88501{\cdot}8 \text{ J/kg}$$

$$\text{Velocity of gas at throat} = \sqrt{2h}$$

$$v_T = \sqrt{2 \times 88501{\cdot}8}$$

$$v_T = 420{\cdot}7 \text{ m/s} \quad \text{Ans.}$$

$$p_T \dot{V}_T = \dot{m} R T_T$$

$$\dot{V}_T = \frac{\dot{m} R T_T}{p_T}$$

$$= \frac{0{\cdot}25 \times 2078{\cdot}5 \times 316{\cdot}9}{3{\cdot}407 \times 10^5}$$

$$= 0{\cdot}4833 \text{ m}^3/\text{s}$$

$$\text{Throat area} = \frac{\dot{V}_T}{v_T} = \frac{0{\cdot}4833}{420{\cdot}7}$$

$$= 0{\cdot}001149 \text{ m}^2 \text{ or } 11{\cdot}49 \text{ cm}^2 \quad \text{Ans.}$$

7. Velocity at nozzle exit v_1

$$= \sqrt{2 \times 312{\cdot}5 \times 10^3 \times 0{\cdot}9}$$
$$= 750 \text{ m/s}$$

Ref. Fig. 57:

$$v_{a1} = v_1 \sin \alpha_1 = 750 \times \sin 20^\circ = 256{\cdot}5 \text{ m/s}$$
$$v_{w1} = v_1 \cos \alpha_1 = 750 \times \cos 20^\circ = 704{\cdot}9 \text{ m/s}$$
$$x = \frac{v_{a1}}{\tan \beta_1} = \frac{256{\cdot}5}{\tan 35^\circ} = 366{\cdot}4 \text{ m/s}$$

$$\text{Blade velocity} = u = v_{w1} - x$$
$$= 704{\cdot}9 - 366{\cdot}4$$
$$= 388{\cdot}5 \text{ m/s} \quad \text{Ans. (i)}$$

Absolute velocity of exit steam is in the direction of the turbine axis, therefore $\alpha_2 = 90^\circ$

$$\tan \beta_2 = \frac{v_2}{u} = \frac{204}{338{\cdot}5} = 0{\cdot}6027$$

Exit angle of blades = 31° 5′ Ans. (ii)

$$v_{r1} = \frac{v_{a1}}{\sin \beta_1} = \frac{256{\cdot}5}{\sin 35^\circ} = 447{\cdot}2 \text{ m/s}$$

$$v_{r2} = \frac{v_2}{\sin \beta_2} = \frac{204}{\sin 31^\circ 5'} = 395{\cdot}2 \text{ m/s}$$

Loss of kinetic energy of steam across the blades

$$= \tfrac{1}{2} \dot{m}(v_{r1}^2 - v_{r2}^2)$$
$$= \tfrac{1}{2} \times 1 \times (447{\cdot}2^2 - 395{\cdot}2^2)$$
$$= 21\,900 \text{ J} = 21{\cdot}9 \text{ kJ/kg steam} \quad \text{Ans. (iii)}$$

$$\text{Axial thrust} = \dot{m}(v_{a1} - v_{a2})$$
$$= 1 \times (256{\cdot}5 - 204)$$
$$= 52{\cdot}5 \text{ N/kg of steam} \quad \text{Ans. (iv)}$$

$$\text{Power} = \dot{m} v_w u$$

Since the steam leaves the turbine axially, that is, at 90° to the blade movement, there is no velocity of whirl at exit, $v_{w2} = 0$ $\therefore$ $v_w = v_{w1}$

$$\text{Power} = 1 \times 704{\cdot}9 \times 338{\cdot}5$$
$$= 2{\cdot}386 \times 10^5 \text{ W}$$
$$= 238{\cdot}6 \text{ kW/kg of steam} \quad \text{Ans. (v)}$$

$$\text{Diagram efficiency} = \frac{\text{work done on blades}}{\text{work supplied}}$$
$$= \frac{\dot{m} v_w u \text{ [J/s]}}{\tfrac{1}{2}\dot{m} v_1^2 \text{ [J/s]}} = \frac{2uv_w}{v_1^2}$$
$$= \frac{2 \times 338{\cdot}5 \times 704{\cdot}9}{750^2}$$
$$= 0{\cdot}8484 \text{ or } 84{\cdot}84\% \quad \text{Ans. (vi)}$$

8. Tables page 7, 15 bar 250°C, $h = 2925$ $s = 6{\cdot}711$
page 3, 0·16 bar, $h_f = 232$ $s_f = 0{\cdot}772$
$h_{fg} = 2369$ $s_{fg} = 7{\cdot}213$

$$\text{Entropy after expansion} = \text{Entropy before}$$
$$0{\cdot}772 + x \times 7{\cdot}213 = 6{\cdot}711$$
$$x \times 7{\cdot}213 = 5{\cdot}939$$
$$\therefore \text{ dryness fraction } x = 0{\cdot}8235 \quad \text{Ans. (i)}$$

$$\text{At } 0{\cdot}16 \text{ bar, } h = h_f + xh_{fg}$$
$$= 232 + 0{\cdot}8235 \times 2369 = 2183$$

$$\text{Rankine efficiency} = \frac{h_1 - h_2}{h_1 - h_{f2}}$$
$$= \frac{2925 - 2183}{2925 - 232} = \frac{742}{2693}$$
$$= 0{\cdot}2755 \text{ or } 27{\cdot}55\% \quad \text{Ans. (ii)}$$

9. Tables page 4, 14 bar, $h_g = 2790$ $v_g = 0{\cdot}1408$
10 bar, $h_f = 763$ $h_{fg} = 2015$
$v_g = 0{\cdot}1944$

$$p_1 v_1^{1{\cdot}135} = p_2 v_2^{1{\cdot}135}$$
$$14 \times 0{\cdot}1408^{1{\cdot}135} = 10 \times v_2^{1{\cdot}135}$$
$$v_2 = 0{\cdot}1408 \times {}^{1{\cdot}35}\sqrt{1{\cdot}4} = 0{\cdot}1893 \text{ m}^3\text{/kg}$$

Spec. vol. of dry steam at 10 bar is 0·1944 m³/kg therefore,

$$\text{dryness after expansion} = \frac{0{\cdot}1893}{0{\cdot}1944} = 0{\cdot}974 \quad \text{Ans. (i)}$$

$$\text{Spec. enthalpy drop} = 2790 - (763 + 0{\cdot}974 \times 2015)$$
$$= 64 \text{ kJ/kg} \quad \text{Ans. (ii)}$$

$$\text{Velocity} = \sqrt{2 \times 64 \times 10^3} = 357{\cdot}8 \text{ m/s} \quad \text{Ans. (iii)}$$

Volume flow [m³/s] = area [m²] × velocity [m/s]

For mass flow of 1 kg/s:

$$\text{Area [m}^2\text{]} = \frac{0{\cdot}1893}{357{\cdot}8} = 5{\cdot}293 \times 10^{-4}\ \text{m}^2$$

$$5{\cdot}293 \times 10^{-4} \times 10^{6} = 529{\cdot}3\ \text{mm}^2 \quad \text{Ans. (iv)}$$

10. Referring to Fig. 65:

$$\gamma = \frac{c_P}{c_V} = \frac{1{\cdot}005}{0{\cdot}718} = 1{\cdot}4$$

$$r_p^{(\gamma-1)/\gamma} = 5{\cdot}7^{0{\cdot}4/1{\cdot}4} = 1{\cdot}644$$

$$\frac{T_2}{T_1} = \left\{\frac{p_2}{p_1}\right\}^{\frac{\gamma-1}{\gamma}}$$

$$T_2 = 294 \times 1{\cdot}644 = 483{\cdot}3\ \text{K}$$

Temperature at end of compression = 210·3°C Ans. (i)

$$\frac{T_4}{T_3} = \left\{\frac{p_4}{p_3}\right\}^{\frac{\gamma-1}{\gamma}}$$

$$T_4 = \frac{953}{1{\cdot}644} = 579{\cdot}7\ \text{K}$$

Alternatively, $\frac{T_4}{T_3} = \frac{T_1}{T_2}$ because pressure ratios are equal

$$\therefore\quad T_4 = \frac{953 \times 294}{483{\cdot}3} = 579{\cdot}7\ \text{K}$$

Temperature at end of expansion ≐ 306·7°C Ans. (ii)

Heat energy supplied per kg

$$\begin{aligned} &= m \times c_P \times (T_3 - T_2) \\ &= 1 \times 1{\cdot}005 \times (953 - 483{\cdot}3) \\ &= 472{\cdot}1\ \text{kJ/kg} \quad \text{Ans. (iii)} \end{aligned}$$

Increase in internal energy per kg from inlet to exhaust

$$\begin{aligned} &= m \times c_V \times (T_4 - T_1) \\ &= 1 \times 0{\cdot}718 \times (579{\cdot}7 - 294) \\ &= 205{\cdot}1\ \text{kJ/kg} \quad \text{Ans. (iv)} \end{aligned}$$

$$\text{Ideal thermal effic.} = 1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}$$

$$= 1 - \frac{1}{1{\cdot}644} = 0{\cdot}3918 \quad \text{Ans. (v)}$$

Alternatively,

$$\text{Thermal effic.} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

$$= 1 - \frac{579{\cdot}7 - 294}{953 - 483{\cdot}3} = 0{\cdot}3919$$

SOLUTIONS TO TEST EXAMPLES 13

1. Tables page 4, water 130°C, $h = 546$

   ... ... 7, steam 30 bar 375°C, by interpolation,

$$\begin{aligned} h \text{ at 30 bar } 400°\text{C} &= 3231 \\ h \text{ at 30 bar } 350°\text{C} &= 3117 \\ \text{difference for } 50°\text{C} &= 114 \\ \text{difference for } 25°\text{C} &= 57 \end{aligned}$$

$\therefore$ h at 30 bar 375°C $= 3117 + 57 = 3174$

Heat energy transferred to steam per hour

$= 30\,000 \times (3174 - 546) = 30\,000 \times 2628$ kJ/h

Hourly fuel consumption $= \dfrac{53 \times 10^3}{24} = 2209$ kg/h

Heat energy supplied by fuel per hour

$= 2209 \times 42 \times 10^3$ kJ/h

$$\text{Efficiency} = \frac{30\,000 \times 2628}{2209 \times 42 \times 10^3}$$

$= 0{\cdot}85$ or 85% Ans. (i)

Heat energy supplied to plant by fuel [kJ/s = kW]

$$= \frac{2209 \times 42 \times 10^3}{3600}$$

Energy converted into engine power

$= 0{\cdot}13 \times$ energy supplied

$$= \frac{0{\cdot}13 \times 2209 \times 42 \times 10^3}{3600} = 3349 \text{ kW} \quad \text{Ans. (ii)}$$

Tables page 2, h_{fg} at 100°C = 2256·7

Evaporative capacity from and at 100°C

$$= \frac{30\,000 \times 2628}{2256{\cdot}7} = 34920 \text{ kg/h} \quad \text{Ans. (iii)}$$

Equivalent evaporation, per kg fuel, from and at 100°C

$$= \frac{34920}{2209} = 15{\cdot}81 \text{ kg steam/kg fuel} \quad \text{Ans. (iv)}$$

2. Solids in initially + solids put in = solids in finally

$$\text{water in boiler} \times \text{initial p.p.m.} + \text{amount of feed} \times \text{feed p.p.m.} = \text{water in boiler} \times \text{final p.p.m.}$$

$$\begin{aligned} 3{\cdot}5 \times 40 + 0{\cdot}875 \times 24 \times \text{feed p.p.m.} &= 3{\cdot}5 \times 2500 \\ 21 \times \text{feed p.p.m.} &= 8610 \\ \text{feed p.p.m.} &= 410 \quad \text{Ans.} \end{aligned}$$

3. mass of solids put in = mass of solids blown out + mass of solids in evaporated output

mass of feed × feed p.p.m. = mass of blow out × blow out p.p.m. + mass evaporated × evaporated p.p.m.

Let x be mass flow per day of sea water feed
then $(x - 10)$ is mass flow per day of brine discharge

$$\begin{aligned} x \times 31\,250 &= (x - 10) \times 78125 + 10 \times 250 \\ x &= 16{\cdot}613 \\ (x - 10) &= 6{\cdot}613 \end{aligned}$$

Mass flow of sea water feed = 16·613 tonne/day Ans. (a)
Mass flow of brine discharge = 6·613 tonne/day Ans. (b)

4. Per kg of fuel:

$$\text{Available hydrogen} = H_2 - \frac{O_2}{8}$$

$$= 0{\cdot}13 - \frac{0{\cdot}02}{8} = 0{\cdot}1275 \text{ kg}$$

$$\begin{aligned} \text{Cal. value} &= 33{\cdot}7\,C + 144\left(H_2 - \frac{O_2}{8}\right) \\ &= 33{\cdot}7 \times 0{\cdot}85 + 144 \times 0{\cdot}1275 \\ &= 28{\cdot}64 + 18{\cdot}36 = 47 \text{ MJ/kg} \quad \text{Ans. (i)} \end{aligned}$$

$$\begin{aligned} \text{Stoichiometric air} &= \frac{100}{23}\left\{2\tfrac{2}{3}\,C + 8\left(H_2 - \frac{O_2}{8}\right)\right\} \\ &= \frac{100}{23}\{2\tfrac{2}{3} \times 0{\cdot}85 + 8 \times 0{\cdot}1275\} \\ &= \frac{100}{23} \times 3{\cdot}287 = 14{\cdot}29 \text{ kg air/kg fuel} \end{aligned}$$

Actual air $= 1{\cdot}5 \times 14{\cdot}29 = 21{\cdot}44$ kg air/kg fuel Ans. (ii)

Products of combustion per kg of fuel burned

$= 21{\cdot}44 + 1$ kg fuel $= 22{\cdot}44$ kg

Heat energy carried away

$=$ mass × spec. heat × temp. rise

$= 22{\cdot}44 \times 1{\cdot}005 \times (553 - 304) = 5614$ kJ/kg fuel

as a percentage of the heat energy supplied

$$= \frac{5614}{47 \times 10^3} \times 100 = 11{\cdot}95\ \% \quad \text{Ans. (iii)}$$

5. Available hydrogen $= H_2 - \dfrac{O_2}{8}$

$$= 0{\cdot}13 - \frac{0{\cdot}02}{8} = 0{\cdot}1275 \text{ kg}$$

$$\text{Cal. value} = 33{\cdot}7 \times 0{\cdot}84 + 144 \times 0{\cdot}1275$$
$$= 28{\cdot}31 + 18{\cdot}36 = 46{\cdot}67 \text{ MJ/kg} \quad \text{Ans. (i)}$$

$$\text{Stoichiometric air} = \frac{100}{23} \times \text{oxygen required}$$

$$= \frac{100}{23}\{2\tfrac{2}{3} \times 0{\cdot}84 + 8 \times 0{\cdot}1275\}$$

$$= \frac{100}{23} \times 3{\cdot}26 = 14{\cdot}18 \text{ kg air/kg fuel} \quad \text{Ans. (ii)}$$

Per kg fuel burned:
Mass of gases in the products from
22 kg air + (1 kg fuel – 0·01 kg incombustibles)
= 22·99 kg
Mass of oxygen in 22 kg of air
$= 0{\cdot}23 \times 22 = 5{\cdot}06$ kg
Surplus oxygen $= 5{\cdot}06 - 3{\cdot}26 = 1{\cdot}8$ kg
Mass of nitrogen in 22 kg of air
$= 0{\cdot}77 \times 22 = 16{\cdot}94$ kg
CO_2 formed $= 3\tfrac{2}{3} \times 0{\cdot}84 = 3{\cdot}08$ kg
H_2O formed $= 9 \times 0{\cdot}13 = 1{\cdot}17$ kg
% composition of gases by mass Ans. (iii):

$$CO_2 = \frac{3{\cdot}08}{22{\cdot}99} \times 100 = 13{\cdot}4\%$$

$$H_2O = \frac{1{\cdot}17}{22{\cdot}99} \times 100 = 5{\cdot}09\%$$

$$O_2 = \frac{1{\cdot}8}{22{\cdot}99} \times 100 = 7{\cdot}83\%$$

$$N_2 = \frac{16{\cdot}94}{22{\cdot}99} \times 100 = 73{\cdot}68\%$$

6. Per kg of fuel burned:

$$\text{Stoichiometric air} = \frac{100}{23}\left\{2\tfrac{2}{3}\,C + 8\left(H_2 - \frac{O_2}{8}\right)\right\}$$
$$= \frac{100}{23}\{2\tfrac{2}{3} \times 0{\cdot}85 + 8 \times 0{\cdot}1275\}$$
$$= \frac{100}{23} \times 3{\cdot}216 = 13{\cdot}98 \text{ kg air/kg fuel}$$
$$CO_2 = 3\tfrac{2}{3} \times 0{\cdot}855 = 3{\cdot}135 \text{ kg}$$

When air supply is stoichiometric:
Mass of products of combustion
= 13·98 kg air + (1 kg fuel – 0·01 kg impurities)
= 13·98 + 0·99 = 14·97 kg

$$\%\ CO_2 = \frac{3{\cdot}135}{14{\cdot}97} \times 100 = 20{\cdot}94\% \quad \text{Ans. (i)}$$

When air supply is 25% excess:
Mass of products of combustion
$= 1{\cdot}25 \times 13{\cdot}98$ kg air $+ 0{\cdot}99 = 18{\cdot}47$ kg

$$\%\ CO_2 = \frac{3{\cdot}135}{18{\cdot}47} \times 100 = 16{\cdot}97\% \quad \text{Ans. (ii)}$$

When air supply is 50% excess:
Mass of products of combustion
$= 1{\cdot}5 \times 13{\cdot}98 + 0{\cdot}99 = 21{\cdot}96$ kg

$$\%CO_2 = \frac{3{\cdot}135}{21{\cdot}96} \times 100 = 14{\cdot}28\% \quad \text{Ans. (iii)}$$

When air supply is 75% excess:
Mass of products of combustion
$= 1{\cdot}75 \times 13{\cdot}98 + 0{\cdot}99 = 25{\cdot}45$ kg

$$\%CO_2 = \frac{3{\cdot}135}{25{\cdot}45} \times 100 = 12{\cdot}31\% \quad \text{Ans. (iv)}$$

7. Stoichiometric air $= \dfrac{100}{23}[\,2\tfrac{2}{3} \times 0{\cdot}84 + 0{\cdot}14 \times 8]$

$= 14{\cdot}6$ kg/kg of fuel
Actual air supplied $= 14{\cdot}6 \times 1{\cdot}2$
$= 17{\cdot}52$ kg/kg of fuel
i.e. $17{\cdot}52 \times 0{\cdot}23 = 4{\cdot}03$ kg of O_2
$17{\cdot}52 - 4{\cdot}03 = 13{\cdot}49$ kg of N_2
CO_2, $3{\cdot}667 \times 0{\cdot}84 = 3{\cdot}08$ kg/kg of fuel

$$i.e.\ 2{\cdot}667 \times 0{\cdot}84 = 2{\cdot}24\ O_2$$

$$H_2O,\ 0{\cdot}14 \times 9 = 1{\cdot}26 \text{ kg/kg of fuel}$$

$$i.e.\ 0{\cdot}14 \times 8 = \underline{1{\cdot}12\ O_2}$$

$$\underline{3{\cdot}36\ O_2}$$

O_2, $4{\cdot}03 - 3{\cdot}36 = 0{\cdot}67$ kg/kg of fuel. Ans.

DFG	m	M	N	N%
CO_2	3·08	44	$3{\cdot}08 \div 44 = 0{\cdot}07$	12·22
O_2	0·67	32	$0{\cdot}67 \div 32 = 0{\cdot}021$	3·65
N_2	13·49	28	$13{\cdot}49 \div 28 = 0{\cdot}482$	84·13
			Total $= 0{\cdot}573$	100
H_2O	1·26	18	$1{\cdot}26 \div 18 = 0{\cdot}07$	

Total mass of gases (wet and dry) $= 3{\cdot}08 + 0{\cdot}67 + 13{\cdot}49 + 1{\cdot}26$
$= 18{\cdot}5$ kg/kg of fuel

Alternatively $17{\cdot}52 + 0{\cdot}98 = 18{\cdot}5$ kg/kg of fuel

Mass flow rate of gases $= \dot{m} = 100 \times 18{\cdot}5 = 1850$ kg/h

$$\text{From } pV = mRT,\ R = \frac{R_0}{M}$$

$$pV = \frac{m}{M} R_0 T$$

$$\frac{m}{M} = 0{\cdot}573 + 0{\cdot}07 = 0{\cdot}643$$

$$\therefore \quad V = \frac{0{\cdot}643 \times 8{\cdot}3143 \times 523}{1 \times 10^2}$$

$$= 27{\cdot}95 \text{ m}^3\text{/kg of fuel}$$

$$\dot{V} = 27{\cdot}95 \times 100$$

$$= 2795 \text{ m}^3\text{/h} \quad \text{Ans.}$$

8. Stoichiometric air $= \dfrac{100}{23}(2\tfrac{2}{3}C + 8H + S)$

$$= 4{\cdot}348(2{\cdot}667 \times 0{\cdot}86 + 8 \times 0{\cdot}12 + 0{\cdot}02)$$

$$= 14{\cdot}23 \text{ kg/kg fuel}$$

Actual air $= 20{\cdot}00$ kg/kg fuel

Excess air $= 5{\cdot}77$ kg/kg fuel

Mass products of combustion per kg fuel:

$$CO_2 = 3\tfrac{2}{3} \times 0{\cdot}86 = 3{\cdot}154 \text{ kg}$$

$$H_2O = 9 \times 0{\cdot}12 = 1{\cdot}08$$

$$SO_2 = 2 \times 0{\cdot}02 = 0{\cdot}04$$

$$\text{Excess } O_2 = 0{\cdot}23 \times 5{\cdot}77 = 1{\cdot}329$$

$$\text{All } N_2 = 0{\cdot}77 \times 20 = \underline{15{\cdot}40}$$

$$\text{Total} = 21{\cdot}003 \text{ kg}$$

% mass analysis of the wet flue gases:

$$CO_2 = \frac{315{\cdot}4}{21} = 15{\cdot}01$$

$$H_2O = \frac{108}{21} = 5{\cdot}14 \qquad \text{Ans. (a)}$$

$$SO_2 = \frac{4}{21} = 0{\cdot}19$$

$$O_2 = \frac{132{\cdot}9}{21} = 6{\cdot}32$$

$$N_2 = \frac{1540}{21} = 73{\cdot}33$$

% volume analysis of the wet flue gases:

DFG	m%	M	N	N%
CO_2	15·01	44	0·341	9·89
H_2O	5·14	18	0·286	8·30
SO_2	0·19	64	0·003	0·09
O_2	6·32	32	0·198	5·73
N_2	73·33	28	2·619	76·00
			3·447	

Ans. (b)

9. Mass of 1 mol of the fuel $= 12 \times 6 + 1 \times 6$
$= 78$ kg

$$H_2 \text{ fraction by mass} = \frac{6}{78} = 0{\cdot}0769$$

$$C \text{ fraction by mass} = \frac{72}{78} = \underline{0{\cdot}9231}$$

$$1{\cdot}0000$$

$$\text{Stoichiometric air} = \frac{100}{23}(2{\cdot}667 \times 0{\cdot}9231 + 0{\cdot}0769 \times 8)$$

$$= 13{\cdot}38 \text{ kg/kg fuel} \quad \text{Ans. (a)}$$

$$\text{Mass of gas} = 14{\cdot}38 \text{ kg/kg fuel}$$

$$\begin{aligned}
CO_2 &= 3{\cdot}667 \times 0{\cdot}9231 = 3{\cdot}385 \\
H_2O &= 0{\cdot}0769 \times 9 = 0{\cdot}692 \\
N_2 &= 0{\cdot}77 \times 13{\cdot}38 = 10{\cdot}303 \\
& \qquad\qquad\qquad\quad \overline{14{\cdot}380}
\end{aligned}$$

$$CO_2 = \frac{3{\cdot}385}{14{\cdot}38} \times 100 = 23{\cdot}54\%$$

$$H_2O = \frac{0{\cdot}692}{14{\cdot}38} \times 100 = 4{\cdot}81\% \qquad \text{Ans. (b)}$$

$$N_2 = \frac{10{\cdot}303}{14{\cdot}38} \times 100 = 71{\cdot}65\%$$

$$DFG = 14{\cdot}38 - 0{\cdot}692 = 13{\cdot}688 \text{ kg/kg fuel}$$

$$CO_2 = \frac{3{\cdot}385}{13{\cdot}688} \times 100 = 24{\cdot}73\%$$

$$N_2 = \frac{10{\cdot}303}{13{\cdot}688} \times 100 = 75{\cdot}27\%$$

DFG	*m*%	*M*	N	N%
CO_2	24·73	44	0·562	17·29
N_2	75·27	28	2·688	82·71
			Total 3·250	

Ans. (c)

10.

DFG	N	*M*	*m*	*m*%
CO_2	10·8	44	475·2	15·83
CO	0·8	28	22·4	0·75
O_2	7·2	32	230·4	7·68
N_2	81·2	28	2273·6	75·75
			Total 3001·6	

Let x be C mass
then $(1 - x)$ is H_2 mass

$$\begin{aligned}
\text{Relative gas mass} &= 3001{\cdot}6 \\
\text{Relative C mass} &= 12\,(10{\cdot}8 + 0{\cdot}8) \\
&= 139{\cdot}2 \\
\text{Dry flue gas mass} &= \frac{3001{\cdot}6 \times x}{139{\cdot}2} \\
&= 21{\cdot}56x \text{ kg} \\
\text{Water vapour} &= 9 - 9x \text{ kg} \\
\text{Total gases} &= 9 - 12{\cdot}56x \text{ kg} \\
\text{Mass of air supplied} &= 8 - 12{\cdot}56 \text{ kg/kg fuel} \quad (1) \\
\text{Mass of } N_2 \text{ supplied} &= \frac{2273 \times x}{139{\cdot}2} \\
&= 16{\cdot}33 \text{ kg} \\
\text{Mass of air supplied} &= \frac{16{\cdot}33 \times 100}{77} \\
&= 21{\cdot}21x \text{ kg/kg fuel} \quad (2)
\end{aligned}$$

From (1) and (2):

$$\begin{aligned}
8 - 12{\cdot}56x &= 21{\cdot}21x \\
x &= 0{\cdot}925 \\
(1 - x) &= 0{\cdot}075 \\
\text{Mass percentage of C} &= 92{\cdot}5\% \text{ in the fuel} \\
\text{Mass percentage of } H_2 &= 7{\cdot}5\% \text{ in the fuel} \qquad \text{Ans.}
\end{aligned}$$

SOLUTIONS TO TEST EXAMPLES 14

Refer to Fig. 69 for all solutions

1. Specific enthalpy gain of refrigerant through evaporator

$$= h_1 - h_4 = 320 - 135 = 185 \text{ kJ/kg}$$

(h_4 being equal to h_3 because there is no change of enthalpy in the throttling process through the expansion valve)

$$\begin{aligned} \text{Refrig. effect [kJ/h]} &= \text{mass flow [kg/h]} \times (h_1 - h_4) \text{ [kJ/kg]} \\ &= 5 \times 60 \times 185 \\ &= 5{\cdot}55 \times 10^4 \text{ kJ/h or } 55{\cdot}5 \text{ MJ/h} \quad \text{Ans.} \end{aligned}$$

2. From tables page 13, Freon-12,

$$\begin{array}{llll} 5{\cdot}673 \text{ bar} & h_f = 54{\cdot}87 & & \\ 1{\cdot}509 \text{ bar} & t_s = -20^\circ\text{C} & h_f = 17{\cdot}82 & h_g = 178{\cdot}73 \end{array}$$

$$h_{fg} = h_g - h_f = 178{\cdot}73 - 17{\cdot}82 = 160{\cdot}91$$

Since the saturation temperature at 1·509 bar is –20°C and the refrigerant at this pressure leaves the evaporator at –5°C, it is superheated by 15°.

$$h \text{ at } 1{\cdot}509 \text{ bar superheated } 15^\circ = 187{\cdot}75$$

Throttling between condenser exit and evaporator inlet:

$$\begin{aligned} \text{Enthalpy after } (h_4) &= \text{Enthalpy before } (h_3) \\ 17{\cdot}82 + x_4 \times 160{\cdot}91 &= 54{\cdot}87 \\ x_4 \times 160{\cdot}91 &= 37{\cdot}05 \\ x_4 &= 0{\cdot}2303 \quad \text{Ans. (i)} \end{aligned}$$

$$\begin{aligned} \text{Refrig. effect/kg} &= h_1 - h_4 = h_1 - h_3 \\ &= 187{\cdot}75 - 54{\cdot}87 \\ &= 132{\cdot}88 \text{ kJ/kg} \quad \text{Ans. (ii)} \end{aligned}$$

3. From tables page 12, NH_3,

$$\begin{array}{llll} 8{\cdot}57 \text{ bar} & h_f = 275{\cdot}1 & h_g = 1462{\cdot}6 & \\ 1{\cdot}902 \text{ bar} & h_f = 89{\cdot}8 & h_g = 1420{\cdot}0 & v_g = 0{\cdot}6237 \\ & h_{fg} = 1420 - 89{\cdot}8 = 1330{\cdot}2 & & \end{array}$$

Specific enthalpy drop through condenser

$$= h_2 - h_3 = 1462{\cdot}6 - 275{\cdot}1 = 1187{\cdot}5 \text{ kJ/kg}$$

Heat rejected in condenser (by 2 kg)

$$= 2 \times 1187{\cdot}5 = 2375 \text{ kJ/min} \quad \text{Ans. (i)}$$

Specific enthalpy gain in evaporator

$$\begin{aligned} & h_1 - h_4 = h_1 - h_3 \\ &= (89{\cdot}8 + 0{\cdot}96 \times 1330{\cdot}2) - 275{\cdot}1 \\ &= 1366{\cdot}8 - 275{\cdot}1 = 1091{\cdot}7 \text{ kJ/kg} \end{aligned}$$

Refrigerating effect

$$= 2 \times 1091{\cdot}7 = 2183{\cdot}4 \text{ kJ/min} \quad \text{Ans. (ii)}$$

Specific volume of refrigerant leaving evaporator and entering compressor $= 0{\cdot}96 \times 0{\cdot}6237 \text{ m}^3/\text{kg}$

Volume taken into compressor per minute

$$= 2 \times 0{\cdot}96 \times 0{\cdot}6237 = 1{\cdot}198 \text{ m}^3/\text{min} \quad \text{Ans. (iii)}$$

4. Since there is a change only in the dryness fraction of the refrigerant through the evaporator, the enthalpy of saturated liquid (h_f) and of evaporation (h_{fg}) being the same for h_1 as for h_4 then:

Specific enthalpy gain of the CO_2 through evaporator

$$\begin{aligned} &= h_1 - h_4 \\ &= (h_f + 0{\cdot}92h_{fg}) - (h_f + 0{\cdot}28h_{fg}) \\ &= (0{\cdot}92 - 0{\cdot}28) \times 290{\cdot}7 \\ &= 0{\cdot}64 \times 290{\cdot}7 = 186{\cdot}1 \text{ kJ/kg} \quad \text{Ans. (i)} \end{aligned}$$

Heat to be extracted from water to make one kg of ice

$$\begin{aligned} &= 4{\cdot}2 \times 14 + 335 + 2{\cdot}04 \times 5 \\ &= 58{\cdot}8 + 335 + 10{\cdot}2 = 404 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} \text{Let } m \text{ [kg]} &= \text{mass of ice made per second} \\ \text{when } 0{\cdot}5 \text{ kg/s} &= \text{mass flow of } CO_2 \end{aligned}$$

Heat transfer:

$$\begin{aligned} \text{from water} &= \text{to } CO_2 \\ m \times 404 &= 0{\cdot}5 \times 186{\cdot}1 \\ m &= \frac{0{\cdot}5 \times 186{\cdot}1}{404} = 0{\cdot}2303 \text{ kg/s} \end{aligned}$$

Mass of ice in tonnes per 24 hours

$$= \frac{0{\cdot}2303 \times 3600 \times 24}{10^3} = 19{\cdot}9 \text{ tonne/day} \quad \text{Ans. (ii)}$$

5. Quantity of heat extracted per kg of water

$$\begin{aligned} &= 4{\cdot}2 \times 18 + 335 + 2{\cdot}04 \times 7 \\ &= 75{\cdot}6 + 335 + 14{\cdot}28 = 424{\cdot}88 \text{ kJ/kg} \end{aligned}$$

Heat energy extracted per second (refrigerating effect)

$$= \frac{2 \cdot 5 \times 10^3}{24 \times 3600} \times 424 \cdot 88 \text{ kJ/s}$$

Energy supplied per second, kJ/s = kW = 2·25

$$\text{Coeff. of performance} = \frac{\text{heat energy extracted}}{\text{heat energy supplied}}$$

$$= \frac{2 \cdot 5 \times 10^3 \times 424 \cdot 88}{24 \times 3600 \times 2 \cdot 25}$$

$$= 5 \cdot 464 \quad \text{Ans. (i)}$$

If ice at 0°C was made from water at 0°C, heat extracted would be equal to enthalpy of fusion only = 335 kJ/kg

∴ Capacity from and at 0°C

$$= \frac{2 \cdot 5 \times 424 \cdot 88}{335} = 3 \cdot 171 \text{ tonne/day} \quad \text{Ans. (ii)}$$

6. From tables page 13, Freon-12,
5·673 bar, sat. temp. = 20°C
∴ at 50°C refrigerant is superheated by 30°
h_2 (compressor discharge) = 216·75
1·826 bar, sat. temp. = −15°C
∴ at 0°C refrigerant is superheated by 15°
h_1 (compressor suction and evaporator exit) = 190·15
h_3 (condenser outlet) = h_f at 5·673 bar = 54·87
h_4 (evaporator inlet) = h_3 = 54·87

$$\text{Coeff. of performance} = \frac{\text{refrigerating effect [kJ/kg]}}{\text{work transfer [kJ/kg]}}$$

$$= \frac{h_1 - h_4}{h_2 - h_1}$$

$$= \frac{190 \cdot 15 - 54 \cdot 87}{216 \cdot 75 - 190 \cdot 15} = \frac{135 \cdot 28}{26 \cdot 6}$$

$$= 5 \cdot 087 \quad \text{Ans.}$$

7. Tables page 12, NH_3:

14·7 bar, sat. temp. = 38°C
∴ vapour is superheated (63 − 38) = 25°

By interpolation,
14·7 bar 50° supht, h = 1620·1

14·7 bar no supht, h = 1472·6
increase for 50° = 147·5
increase for 25° = ½ × 147·5 = 73·75
∴ 14·7 bar 25° supht, h = 1472·6 + 73·75 = 1546·35 = h_2
14·7 bar 50°C supht, s = 5·340
14·7 bar no supht, s = 4·898
increase for 50° = 0·442
increase for 25° = ½ × 0·442 = 0·221
∴ 14·7 bar 25° supht, s = 4·898 + 0·221 = 5·119 = s_2
2·077 bar, h_f = 98·8 h_g = 1422·7
h_{fg} = 1422·7 − 98·8 = 1323·9
s_f = 0·404 s_g = 5·593
s_{fg} = 5·593 − 0·404 = 5·189

Isentropic compression in compressor:

$$s_1 = s_2$$
$$0 \cdot 404 + x_1 \times 5 \cdot 189 = 5 \cdot 119$$
$$x_1 = 0 \cdot 9086$$
$$h_1 = 98 \cdot 8 + 0 \cdot 9086 \times 1323 \cdot 9 = 1301 \cdot 8$$
$$h_4 = h_3 = h_f \text{ at } 14 \cdot 7 \text{ bar} = 362 \cdot 1$$

Refrig. effect [kJ/s] = $(h_1 - h_4)$ [kJ/kg] × mass flow [kg/s]
= (1301·8 − 362·1) × 0·15
= 140·9 kJ/s Ans. (i)

Work transfer [kJ/s] = $(h_2 - h_1)$ [kJ/kg] × mass flow [kg/s]
= (1546·35 − 1301·8) × 0·15
= 36·68 kJ/s Ans. (ii)

$$\text{Coeff. of performance} = \frac{\text{refrigerating effect}}{\text{work transfer}}$$

$$= \frac{140 \cdot 9}{36 \cdot 68} = 3 \cdot 842 \quad \text{Ans. (iii)}$$

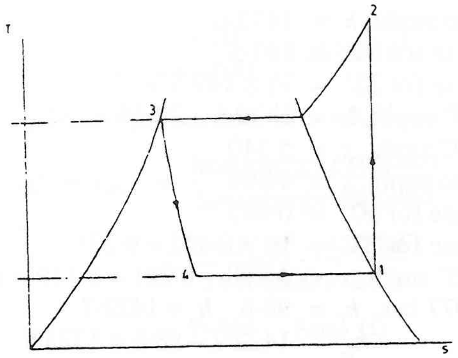
8.

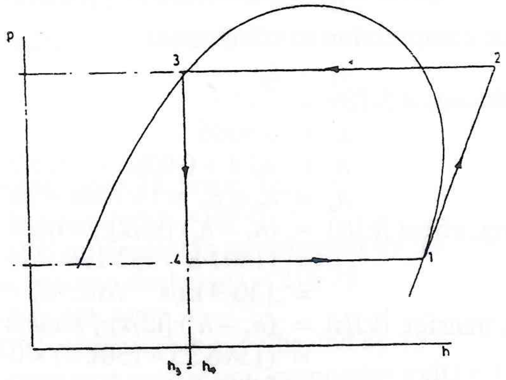
At 1·004 bar $h_1 = h_g$ = 174·2 kJ/kg
$s_1 = s_g$ = 0·7170 kJ/kg K
$v_1 = v_g$ = 0·1594 m³/kg

$$\text{Mass flow of refrigerant} = \dot{m} = \frac{0 \cdot 15}{0 \cdot 1594}$$

$$= 0 \cdot 941 \text{ kg/s}$$

Isentropic compression $s_1 = s_2$ = 0·717 kJ/kg K





By interpolation

$$0{\cdot}717 = 0{\cdot}6901 + \frac{\theta}{15}(0{\cdot}7251 - 0{\cdot}6901)$$

$$\theta = 11{\cdot}53°C$$

$$h_2 = 193{\cdot}78 + \frac{11{\cdot}53}{15}(204{\cdot}1 - 193{\cdot}78)$$

$$= 201{\cdot}7 \text{ kJ/kg}$$

At 4·914 bar $\quad h_4 = h_3 = h_f = 50{\cdot}1$ kJ/kg

$$\text{c.o.p.} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$= \frac{174{\cdot}2 - 50{\cdot}1}{201{\cdot}7 - 174{\cdot}2}$$

$$= 4{\cdot}513 \quad \text{Ans. (a)}$$

$$\text{power input} = \dot{m}\,[h_2 - h_1]$$

$$= 0{\cdot}941 \times [201{\cdot}7 - 174{\cdot}2]$$

$$= 25{\cdot}88 \text{ kW} \quad \text{Ans. (b)}$$

9. At 40°C specific enthalpy

$$\text{leaving compressor} = h_2$$

$$h_2 = h_g = 1473{\cdot}3 \text{ kJ/kg}$$

$$s_2 = 4{\cdot}877 \text{ kJ/kg K}$$

With isentropic compression $s_1 = s_2 = 4{\cdot}877$ kJ/kg K

$$\therefore 4{\cdot}877 = 0{\cdot}44 + x\,(5{\cdot}563 - 0{\cdot}44)$$

$$x = 0{\cdot}8661 \text{ dry}$$

$$\text{at } -16°C \therefore h_1 = h_f + 0{\cdot}8661\,(h_g - h_f)$$

$$h_1 = 107{\cdot}9 + 0{\cdot}8661\,(1425{\cdot}3 - 107{\cdot}9)$$

$$= 1248{\cdot}9 \text{ kJ/kg}$$

If the ammonia leaves the heat exchanger as saturated liquid then at 40°C $h_3 = h_f = 371{\cdot}9$ kJ/kg

$$\text{Work done in compressor} = h_2 - h_1$$

$$= 1473{\cdot}3 - 1248{\cdot}9$$

$$= 224{\cdot}4 \text{ kJ/kg}$$

$$\text{Heat transfer in condenser} = h_2 - h_3$$

$$= 1473{\cdot}3 - 371{\cdot}9$$

$$= 1101{\cdot}4 \text{ kJ/kg}$$

$$\text{c.o.p.} = \frac{h_2 - h_3}{h_2 - h_1}$$

$$= \frac{1101{\cdot}4}{224{\cdot}4} = 4{\cdot}908. \quad \text{Ans.}$$

$$\text{Mass of air changed } \dot{m}_1 = \frac{p\dot{V}}{RT}$$

$$= \frac{1{\cdot}013 \times 1200}{287 \times 293} \times 10^5$$

$$= 1445{\cdot}6 \text{ kg/h}$$

$$\dot{m}_1 \times c_p \times (T_2 - T_1) = (h_2 - h_3) \times \dot{m}$$

$$1445{\cdot}6 \times \frac{1005}{10^3}(293 - 286) = 1101{\cdot}4 \times \dot{m}$$

$$\dot{m} = 9{\cdot}233 \text{ kg/h} \quad \text{Ans.}$$

10. Tables Freon - 12 page 13:

$$25°C, \quad h_f = 59{\cdot}7 \quad h_g = 197{\cdot}33$$

$$-15°C, \quad h_f = 22{\cdot}33 \quad h_g = 180{\cdot}97 \quad (h_{fg} = 158{\cdot}64)$$

$$\text{Refrigerating effect} = h_1 - h_4$$

$$= 180{\cdot}97 - 59{\cdot}7$$

$$= 121{\cdot}27 \text{ kJ/kg}$$

$$\text{Cooling load} = \text{refrigerating effect} \times \text{mass flow}$$
$$h_3 = h_4 \text{ as throttling, no undercool}$$
$$73{\cdot}3 = 121{\cdot}27 \times \dot{m}$$
$$\dot{m} = 0{\cdot}6044 \text{ kg/s} \quad \text{Ans. (a)}$$
$$h_2 = 208{\cdot}5$$

i.e. 6·516 bar, 40°C (15° superheat), from tables

$$\text{Compressor work done} = h_2 - h_1$$
$$= 208{\cdot}5 - 180{\cdot}97$$
$$= 27{\cdot}53 \text{ kJ/kg}$$
$$\text{Compressor power} = 27{\cdot}53 \times 0{\cdot}6044$$
$$= 16{\cdot}64 \text{ kW} \quad \text{Ans. (b)}$$
$$h_4 = h_f + x h_{fg}$$
$$59{\cdot}7 = 22{\cdot}33 + 158{\cdot}64 \times x$$
$$\text{Condition after expansion valve} = 0{\cdot}2356 \text{ dry} \quad \text{Ans. (c)}$$

SELECTION OF EXAMINATION QUESTIONS CLASS TWO

1. The efficiency of an auxiliary boiler is 70% when dry saturated steam at 8 bar is generated from feed water at 43°C. If the calorific value of the fuel is 42·5 MJ/kg and it is burned at the rate of 6 tonne per day, calculate (i) the hourly steam production, (ii) the equivalent evaporation per kg of fuel from and at 100°C.

2. An electric motor was tested by coupling it to a dynamometer and the brake load was 112 N at 40 rev/s. A steady flow of water passed through the brake and was raised in temperature from 15°C to 48°C. If the power absorbed by this brake is given by $Wn/330$ where W is the brake load in newtons and n is the rotational speed in rev/s, and assuming 98% of the heat generated at the brake is carried away by the cooling water, find the quantity of water passing through the dynamometer in litres per minute.

3. (a) A refrigerated hold of 580m² surface area is lined with a 100 mm thick layer of insulation of thermal conductivity 0·17 W/m K. The interior and exterior surface temperatures of the insulation are 0°C and 17°C respectively. Calculate the heat flow rate from the hold.
   (b) The exterior surface temperature is increased to 50°C. Calculate the additional thickness of insulating material of thermal conductivity 0·07 W/m K required to be placed on the inside surfaces to keep the heat flow rate at the same level.

4. The bore of an I.C. engine is exactly 300 mm at 20°C. The diameters of the piston at 20°C are 298 mm at the crown and 299 mm at mid-depth of the body. Under working conditions the mean temperatures are: piston crown 250°C, piston body 100°C, cylinder 180°C. Take the coefficients of linear expansion of the piston and cylinder materials as $1{\cdot}2 \times 10^{-5}$ and $1{\cdot}1 \times 10^{-5}$/°C respectively and calculate the diametrical clearances at the crown and mid-depth of the piston under working conditions.

5. An indicator card taken off one cylinder of a six-cylinder, two-stroke, single-acting engine, has an area of 2850 mm² and length 75 mm when running at 1·75 rev/s. One millimetre height on the card represents 0·4 bar. If the cylinder bore is 550 mm and stroke

850 mm, calculate the indicated power of the engine assuming equal powers are developed in all cylinders.

6. A vessel of volume 0·65 m^3 contains air at 27·6 bar and 18°C. Calculate the final pressure after 3·5 kg of air is added if the final temperature is 20·5°C. Take R for air = 0·287 kJ/kg K.

7. One kilogramme of steam at 7·0 bar, 0·95 dry, is expanded according to the law $pV^{1\cdot 3}$ = a constant until the pressure is 3·5 bar. Calculate the final dryness fraction of the steam.

8. A turbine plant consisting of H.P. and L.P. units is supplied with steam at 15 bar 300°C. The steam is expanded in the H.P. and leaves at 2·5 bar 0·97 dry. At this point some steam is bled off to the feed heater, the remaining steam passes to the L.P. where it is expanded to 0·14 bar 0·84 dry. If the same quantity of work transfer takes place in each unit, calculate the amount of steam bled off expressed as a percentage of the steam supplied.

9. The mass analysis of a fuel is 87% carbon, 11% hydrogen, and 2% oxygen. Calculate the volume of air in cubic metres at 1·0 bar and 25°C required for stoichiometric combustion per kg of fuel. Take the values: R for air = 0·287 kJ/kg K. Mass analysis of air = 23% oxygen, 77% nitrogen. Atomic weights, hydrogen 1, carbon 12, oxygen 16.

10. Gas at pressure 0·95 bar, volume 0·2 m^3 and temperature 17°C, is compressed until the pressure is 2·75 bar and volume 0·085 m^3, calculate the compression index and the final temperature.

11. A ship's cold room has dimensions of 9 m × 4 m × 2·5 m and is lined on the inside with 15 mm thick boarding. The deck, deckhead and bulkheads are of 10 mm thick steel and a 70 mm thick layer of cork insulation is sandwiched between the steel plate and the boarding. The thermal conductivities of steel, cork and boarding are 45 W/m K, 0·06 W/m K and 0·11 W/m K respectively. The surface heat transfer coefficients at the exposed board inner and steel outer surfaces are 1·62 W/m^{2}K and 13 W/m^{2}K respectively. Calculate the cooling load required to maintain the cold room at –6°C when the ambient temperature is 27°C.

12. A boiler working at 16 bar produces 9000 kg of steam per hour from feed water at 95°C, the dryness fraction of the steam being

0·98. If the boiler efficiency is 87% and the calorific value of the fuel 42 MJ/kg, calculate the daily fuel consumption.

13. The scavenge ports of a two-stroke diesel engine are just covered when the piston is 800 mm from the top of its stroke. The contents of the cylinder at this instant are at a pressure of 1·21 bar and temperature 40°C. The piston diameter is 700 mm and the clearance is equivalent to 70 mm. Find the mass of air taken in per cycle if the scavenge efficiency is 0·95. R for air = 0·287 kJ/kg K.

14. The diameter of an air compressor cylinder is 130 mm, the stroke is 180 mm, and the clearance volume is 73 cm^3. The pressure in the cylinder at the beginning of the stroke is 1·0 bar and the pressure during delivery is constant at 4·6 bar. Taking the law of compression as $pV^{1\cdot 2}$ = constant, calculate the distance moved by the piston during the delivery period and express this as a fraction of the stroke.

15. In a single-stage impulse turbine the steam leaves the nozzles at a velocity of 500 m/s at 18° to the plane of rotation of the blades, and the linear velocity of the blades is 230 m/s. Neglecting friction across the blades and assuming the steam leaves the blade wheel in an axial direction, calculate (i) the inlet angle of the blades so that the steam enters without shock, (ii) the outlet blade angle.

16. The clearance volume of a reciprocating compressor of 100 mm bore and 100 mm stroke is 5% of its swept volume. At the end of the suction stroke the air in the cylinder is at 1 bar 25°C.

(a) Show the cycle on a pressure volume diagram.
(b) Calculate the mass of air in the cylinder at the beginning of the delivery stroke.
(c) Explain why the compressor takes in less than the swept volume of air during each suction stroke.

Note: for air R = 287 J/kg K

17. In an NH_3 refrigerating plant the ammonia leaves the condenser as a saturated liquid at 10·34 bar. The evaporator pressure is 2·265 bar and the refrigerant leaves the evaporator as a vapour 0·95 dry. If the circulation of the refrigerant through the plant is 4 kg/min, calculate (i) the dryness fraction at inlet to the evaporator, (ii) the refrigerating effect per minute, and (iii) the volume of refrigerant taken into the compressor per minute.

18. (a) Write down the combustion equations for the complete combustion of carbon to carbon dioxide, hydrogen to water and sulphur to sulphur dioxide.
(b) A fuel oil of mass analysis 86·3% carbon, 12·8% hydrogen and 0·9% sulphur is burned with 25% excess air. The flue gases are at 1·5 bar 370°C. Calculate the mass and volume of flue gases per kg of fuel burned.

Note: air contains 23% oxygen by mass
relative atomic masses: hydrogen 1, carbon 12, oxygen 16, sulphur 32
for flue gases R = 276 J/kg K

19. Water at 100°C flows through a steel pipe of 150 mm inside diameter and 160 mm outside diameter. The surface heat transfer coefficients at the inside and outside surfaces are 240 $W/m^2 K$ and 12 $W/m^2 K$ respectively. The air surrounding the pipe is at 15°C. The thermal resistance of the steel may be neglected and the diameter of the pipe is large compared to its wall thickness.

Calculate the rate of heat loss from the water per metre length of pipe.

20. A volume of 0·8 m^3 of steam at 17 bar 0·95 dry is passed through a reducing valve and throttled to 6 bar. Calculate the dryness fraction and the volume after throttling.

21. At the beginning of a voyage a boiler contains 6 tonne of water having 120 p.p.m. dissolved solids. The feed rate is 1250 kg/h and after 24 hours the boiler water contains 1080 p.p.m. dissolved solids. Calculate the average dissolved solids in the feed water.

22. 1·5 m^3 of wet steam at 2·8 bar are blown into 36 kg of water at 16°C and the resulting temperature of the mixture is 55°C. Calculate the dryness fraction of the steam.

23. A single stage impulse turbine has a mean blade diameter of 600 mm and a blade velocity of 120 m/s. The nozzle angle is 18° and the enthalpy drop across the nozzles is 465 kJ/kg.

Determine:
(a) the turbine rotational speed;
(b) the blade inlet angle;
(c) the axial component of the steam at the blade inlet.

24. The air in a ship's saloon is maintained at 19°C and is changed twice every hour from the outside atmosphere which is at 7°C. The saloon is 27 m by 15 m by 3 m high. Calculate the kilowatt loading to heat this air, taking the saloon to be at atmospheric pressure = 1·013 bar, R for air = 0·287 kJ/kg K, c_p = 1·005 kJ/kg K.

25. A six-cylinder, single-acting, four-stroke oil engine, of 200 mm stroke and 225 mm bore runs at 5 rev/s when the mean effective pressure is 17 bar. If the mechanical efficiency is 85% calculate the indicated and brake powers.

26. A single cylinder single acting compressor of 200 mm stroke and 100 mm bore runs at 5 rev/s and takes in air at 1 bar 17°C. It is used to charge a 3 m^3 capacity receiver from 1 bar 25°C to 7 bar 25°C. The compressor has negligible piston clearance.
(a) Sketch the cycle on a pressure volume diagram.
(b) Calculate the time required to charge the receiver.

27. An ammonia refrigerator has a cooling load of 3·5 kW. The ammonia leaves the condenser as a liquid at 24°C, is throttled to 2·68 bar and leaves the evaporator as a dry saturated vapour.
(a) Sketch component and pressure enthalpy diagrams.
(b) Calculate the mass flow rate of the ammonia.

28. A boiler working at 15 bar generates 7000 kg of steam per hour. The steam leaves the boiler steam drum dry and saturated and then passes through the superheater tubes at constant pressure. The flue gases enter the nests of superheaters at 822°C and leaves at 690°C. The fuel consumption is 750 kg per hour and 24 kg of air are supplied per kg of fuel burned. Find (i) the temperature of the superheated steam, (ii) the mass of injection water to the de-superheater, at 21°C, required to desuperheat each kg of steam. Take the specific heat of the flue gases as 1·007 kJ/kg K.

29. A gas initially at 12 bar, 216°C and volume 9900 cm^3 is expanded in a cylinder. The volumetric expansion is 6 and the index of expansion is 1·33. Calculate the final volume, pressure, and temperature.

30. 1 kg of steam initially occupying a volume of 0·08 m^3 at 40 bar is expanded to 2 bar according to the law $pV^{1·3}$ = const.

Calculate:
(a) the work transfer;
(b) the heat transfer.

31. Steam leaves the nozzles and enters the blade wheel of a single-stage impulse turbine at a velocity of 840 m/s and at an angle of 20° to the plane of rotation. The blade velocity is 350 m/s and the exit angle of the blades is 25° 12'. Due to friction, the steam loses 20% of its relative velocity across the blades. Calculate (i) the blade inlet angle, (ii) the magnitude and direction of the absolute velocity of the steam at exit.

32. The mass composition of a fuel oil is 84·6% carbon, 11·4% hydrogen, 0·4% sulphur, 2·4% oxygen, and 1·2% impurities. Calculate the calorific value of the fuel and the theoretical mass of air required for stoichiometric combustion of one kg of fuel. Take the values:

	CALORIFIC VALUE MJ/kg	ATOMIC WEIGHT
Hydrogen	144	1
Carbon	33·7	12
Sulphur	9.3	32
Oxygen	—	16

Mass analysis of air = 23% oxygen, 77% nitrogen.

33. In a Freon-12 refrigerating plant, the refrigerant leaves the condenser with a specific enthalpy of 50 kJ/kg. The pressure in the evaporator is 1·826 bar and the refrigerant leaves the evaporator at this pressure and at a temperature of 0°C. Calculate (i) the dryness fraction of the freon at inlet to the evaporator, and (ii) the refrigerating effect per minute if the flow rate of the refrigerant is 0·4 kg/s.

34. A single stage double-acting air compressor deals with 18·2 m³ of air per minute measured at conditions of 1·01325 bar 15°C. The condition at the beginning of compression is 0·965 bar 27°C and the discharge pressure is 4·82 bar. The compression is according to the law $pV^{1\cdot32}$ = constant. If the mechanical efficiency of the compressor is 0·9 calculate the input power required to drive the compressor.

35. The mean area of indicator cards taken from a cylinder of a double-acting two-stroke engine is 346 mm² and the length is 75 mm. The spring used in the indicator deflects one mm under a force of 60 N and the movement of the stylus is six times that of the indicator piston. The diameter of the indicator piston is 7 mm. Calculate (i) the mean effective pressure. If the diameter of the engine cylinder is 600 mm, stroke 900 mm, and rotational speed 2·1 rev/s, calculate (ii) the indicated power per cylinder.

36. At the entrance of a nozzle of circular cross-section, the velocity of the steam is 457 m/s and the specific volume is 0·2765 m³/kg. At exit, the velocity is 1524 m/s and the specific volume 7·404 m³/kg. If the mass flow of steam through the nozzle is 0·315 kg/s, calculate the entrance and exit diameters in millimetres.

37. A quantity of air of volume 0·2 m³ at 1·1 bar and 15°C is heated at constant pressure until its temperature is 150°C, and then compressed to 7·15 bar according to the law $pV^{1\cdot32}$ = a constant. Calculate (i) the amount of heat energy transferred to the air at constant pressure, and (ii) the temperature at the end of compression. For air, R = 0·287 kJ/kg K, c_p = 1·005 kJ/kg K.

38. A cold store wall 6 m long and 3 m high is constructed of 120 mm thick brick with an inside layer of 80 mm thick cork insulation. The thermal conductivity of the brick is 1·15 W/m K whilst that of cork is 0·043 W/m K. The inner and outer wall surface temperatures are –4°C and 21°C respectively. Calculate:
(a) the amount of heat flow through the wall in 24 hours;
(b) the interface temperature between the cork and the brick.

39. The clearance volume of a reciprocating compressor of 381 mm stroke is 7% of the swept volume. The piston travels 267 mm from bottom dead centre to the point at which the delivery valves open and raises the air pressure from 1·013 bar to 4 bar. The compression process is polytropic.
(a) Sketch the cycle on a pV diagram.
(b) Calculate the value of the polytropic index n.

40. The mass analysis of a hydrocarbon fuel A is 88·5% carbon and 11·5% hydrogen. Another hydrocarbon fuel B requires 6% more air than fuel A for stoichiometric combustion. Calculate the mass analysis of fuel B taking the following values: Atomic weights, carbon 12, hydrogen 1, oxygen 16, mass content of oxygen in air = 23%.

41. The stroke of an internal combustion engine is 90 mm and the clearance volume is equivalent to a linear clearance of 15 mm. If the clearance is reduced by 2·5 mm, find the pressure at the end of compression before and after the alteration, taking the initial pressure in each case as 1·0 bar and the index of compression as 1·33.

42. The kinetic energy of the steam jet leaving the nozzles of a single-stage impulse turbine is equivalent to 250 kJ/kg. The entrance and exit angles of the blades are equal at 35°, and the steam leaves the blade wheel in an axial direction. Neglecting friction across the blades and assuming shockless flow, calculate (i) the nozzle angle, and (ii) the blade velocity.

43. A four-cylinder, single-acting, two-stroke engine develops 600 kW indicated power when the mean effective pressure is 12·56 bar and the speed is 4·5 rev/s.

(a) If the stroke is 25% greater than the cylinder diameter, calculate the diameter of the cylinders and the stroke to the nearest millimetre.
(b) When burning fuel of calorific value 42 MJ/kg the fuel consumption is 0·225 kg/kWh (indicated), find the indicated thermal efficiency.

44. The refrigerating effect of a plant using ammonia as the refrigerant is 800 kJ/min. At the exit points of the components the conditions of the refrigerant are:

Evaporator, dry saturated vapour at 1·902 bar
Compressor, vapour at 7·529 bar and 66°C
Condenser, saturated liquid at 7·529 bar

Calculate (i) the mass flow of the refrigerant through the plant, in kg/min, (ii) the heat rejected in the condenser, in kJ/min, and (iii) the output power of the compressor in kW.

45. Steam enters a desuperheater at 30 bar and 400°C and leaves at the same pressure as dry saturated steam. The temperature of the water injected into the desuperheater is 38°C. Calculate (i) the mass of injection water used per kg of steam desuperheated, and (ii) the percentage change in volume from that occupied by one kg of superheated steam to the volume occupied by the dry saturated steam resulting from the mixture of one kg of superheated steam and its injected water.

46. A gas is compressed in a cylinder from 1 bar and 35°C at the beginning of the stroke to 37 bar at the end of the stroke. If the clearance volume is 850 cm^3 and the compression index 1·32, calculate the stroke volume and the temperature at the end of compression.

47. When 1 kg of a fuel containing only carbon and hydrogen is burned in air, 15·6 kg of exhaust containing only nitrogen, carbon dioxide and water vapour is produced.

Determine the fuel mass analysis.

Air contains 23% oxygen by mass

Atomic mass relationships: hydrogen 1, carbon 12, oxygen 16

48. (a) A ship's ice making machine produces 27·2 kg/h of ice at 0°C from water at 14·4°C with a coefficient of performance of 7·51. Calculate the power input to the machine.
(b) Using the concept of a reversed heat engine, explain why refrigerators require some power input.

Note: *specific heat capacity of water is 4·186 kJ/kg K*
specific enthalpy of fusion of ice is 332·6 kJ/kg

49. The following data relate to a single stage impulse turbine

Blade:	mean diameter	1·32 m
	inlet angle	34°
	outlet angle	30°
Steam:	inlet axial velocity component	550 m/s
	relative outlet velocity	850 m/s
	flow rate	0·0833 kg/s
Turbine:	power developed	67 kW

Determine the turbine rotational speed.

50. Water is kept at a temperature of 66°C in a closed tank 1 m long × 0·75 m wide × 1·6 m high. The heat loss from the water must not exceed 200 W when the ambient temperature is 18°C. The tank is to be lagged with insulating material of thermal conductivity 0·048 W/mK. The exterior surface heat transfer coefficient is 1 W/m^2K. The thermal resistances of the tank walls and lid and the heat loss through the base are negligible.

Calculate:
(a) the thickness of lagging required;
(b) the additional thickness of lagging required to reduce the heat loss to 100 W.

SOLUTIONS TO EXAMINATION QUESTIONS CLASS TWO

1. From steam tables

Steam 8 bar, $h_g = 2769$ kJ/kg

Water 43°C, by interpolation,

$$\begin{aligned} h \text{ at } 44^\circ\text{C} &= 184{\cdot}2 \\ h \text{ at } 42^\circ\text{C} &= 175{\cdot}8 \\ \text{increase for } 2^\circ &= 8{\cdot}4 \\ \text{increase for } 1^\circ &= 4{\cdot}2 \\ \therefore\ h \text{ at } 43^\circ\text{C} &= 175{\cdot}8 + 4{\cdot}2 = 180 \text{ kJ/kg} \end{aligned}$$

$$\text{Fuel consumption} = \frac{6 \times 10^3}{24} = 250 \text{ kg/h}$$

$$\text{Boiler efficiency} = \frac{\text{heat energy transferred to steam [kJ/h]}}{\text{heat energy supplied by fuel [kJ/h]}}$$

$$0{\cdot}7 = \frac{\text{mass of steam [kg/h]} \times (2769 - 180)}{250 \times 42{\cdot}5 \times 10^3}$$

$$\text{Mass of steam}/h = \frac{0{\cdot}7 \times 250 \times 42{\cdot}5 \times 10^3}{2589}$$

$$= 2873 \text{ kg/h} \quad \text{Ans. (i)}$$

From tables, h_{fg} at 100°C = 2256·7 kJ/kg

Equivalent evaporation from and at 100°C, per kg of fuel,

$$= \frac{2873 \times (2769 - 180)}{250 \times 2256{\cdot}7}$$

$$= 13{\cdot}18 \text{ kg steam/kg fuel} \quad \text{Ans. (ii)}$$

2. $$\text{Brake power} = \frac{Wn}{330} = \frac{112 \times 40}{330} = 13{\cdot}58 \text{ kW}$$

Energy at brake = 13·58 kJ/s

Energy carried away by water

$= 0{\cdot}98 \times 13{\cdot}58 = 13{\cdot}3$ kJ/s

From steam tables,

Water 48°C, h = 200·9

... 15°C, h = 62·9

Enthalpy gain of cooling water

$= 200{\cdot}9 - 62{\cdot}9 = 138$ kJ/kg

Heat energy transferred to water [kJ/s]

= mass flow [kg/s] × spec. enthalpy gain [kJ/kg]

$$\therefore \text{ mass flow} = \frac{13{\cdot}3}{138} = 0{\cdot}09638 \text{ kg/s}$$

$$\begin{aligned} 0{\cdot}09638 \times 60 &= 5{\cdot}783 \text{ kg/min} \\ &= 5{\cdot}783 \text{ litre/min} \quad \text{Ans.} \end{aligned}$$

3. $$\begin{aligned} Q &= \frac{kAt\,(T_1 - T_2)}{S} \\ &= \frac{0{\cdot}17 \times 580 \times 1\ (0 - 17)}{0{\cdot}1} \\ &= -16762 \text{ W i.e. } \textit{into} \text{ the hold} \\ &= -16{\cdot}762 \text{ kW} \quad \text{Ans. (a)} \end{aligned}$$

For the two thicknesses:

$$T_1 - T_3 = \frac{Q}{At}\left\{\frac{S_1}{k_1} + \frac{S_2}{k_2}\right\}$$

$$0 - 50 = \frac{-16762}{580 \times 1}\left\{\frac{S_1}{0{\cdot}07} + \frac{0{\cdot}1}{0{\cdot}17}\right\}$$

$$\begin{aligned} S_1 &= 0{\cdot}07\left\{\frac{50 \times 580 \times 1}{16762} - \frac{0{\cdot}1}{0{\cdot}17}\right\} \\ &= 0{\cdot}07993 \text{ m} \\ &= 79{\cdot}93 \text{ mm} \quad \text{Ans. (b)} \end{aligned}$$

4. Linear (diametrical) expansion = $\alpha \times d \times (\theta_2 - \theta_1)$

Increase in cylinder diameter

$= 1{\cdot}1 \times 10^{-5} \times 300 \times (180 - 20) = 0{\cdot}5279$ mm

Working diameter = 300 + 0·5279 = 300·5279 mm

Increase in piston crown diameter

$= 1{\cdot}2 \times 10^{-5} \times 298 \times (250 - 20) = 0{\cdot}8224$ mm

Working diameter = 298 + 0·8224 = 298·8224 mm

Increase in piston body diameter

$1{\cdot}2 \times 10^{-5} \times 299 \times (100 - 20) = 0{\cdot}2871$mm

Working diameter = 299 + 0·2871 = 299·2871 mm

Diametrical clearances at working temperatures:

$$\left.\begin{aligned}\text{Piston crown} &= 300{\cdot}5279 - 298{\cdot}8224 = 1{\cdot}7055\text{ mm}\\ \text{Piston body} &= 300{\cdot}5279 - 299{\cdot}2871 = 1{\cdot}2408\text{ mm}\end{aligned}\right\}\text{ Ans.}$$

5. Mean height of card $= \dfrac{2850}{75} = 38$ mm

$$\text{Mean effective pressure} = 38 \times 0{\cdot}4 \text{ bar}$$
$$38 \times 0{\cdot}4 \times 10^2 = 1520 \text{ kN/m}^2$$

Indicated power of 6 cylinders

$$= p_m ALn \times 6$$
$$= 1520 \times 0{\cdot}7854 \times 0{\cdot}55^2 \times 0{\cdot}85 \times 1{\cdot}75 \times 6$$
$$= 3224 \text{ Ans.}$$

6. $pV = mRT$

Initial mass of air,

$$m = \frac{pV}{RT} = \frac{27{\cdot}6 \times 10^2 \times 0{\cdot}65}{0{\cdot}287 \times 291} = 21{\cdot}48 \text{ kg}$$

$$\text{Final mass} = 21{\cdot}48 + 3{\cdot}5 = 24{\cdot}98 \text{ kg}$$

$$\text{Final pressure, } p = \frac{mRT}{V}$$
$$= \frac{24{\cdot}98 \times 0{\cdot}287 \times 293{\cdot}5}{0{\cdot}65}$$
$$= 3238 \text{ kN/m}^2 = 32{\cdot}38 \text{ bar} \quad \text{Ans.}$$

7. From steam tables

7 bar, $v_g = 0{\cdot}2728$ 3·5 bar, $v_g = 0{\cdot}5241$

Volume of one kg of steam at 7 bar 0·95 dry

$$= 0{\cdot}95 \times 0{\cdot}2728 = 0{\cdot}2592 \text{ m}^3$$

$$p_1 v_1^{1{\cdot}3} = p_2 v_2^{1{\cdot}3}$$
$$7 \times 0{\cdot}2592^{1{\cdot}3} = 3{\cdot}5 \times V_2^{1{\cdot}3}$$
$$v_2 = 0{\cdot}2592 \times \sqrt[1{\cdot}3]{\frac{7}{3{\cdot}5}}$$
$$= 0{\cdot}4417 \text{ m}^3$$

Specific volume of dry sat. steam at 3·5 bar is 0·5241 m³/kg therefore dryness fraction of expanded steam

$$= \frac{0{\cdot}4417}{0{\cdot}5241} = 0{\cdot}8428 \quad \text{Ans.}$$

8. From steam tables,

15 bar 300°C,	$h = 3039$	
2·5 bar	$h_f = 535$	$h_{fg} = 2182$
0·14 bar,	$h_f = 220$	$h_{fg} = 2376$

Specific enthalpy at 2·5 bar 0·97 dry

$$= 535 + 0{\cdot}97 \times 2182 = 2651 \text{ kJ/kg}$$

Specific enthalpy at 0·14 bar 0·84 dry

$$= 220 + 0{\cdot}84 \times 2376 = 2215 \text{ kJ/kg}$$

Specific enthalpy drop through H.P.

$$= 3039 - 2651 = 388 \text{ kJ/kg}$$

Specific enthalpy drop through L.P.

$$= 2651 - 2215 = 436 \text{ kJ/kg}$$

Total enthalpy drop through each unit is to be the same, let 1 kg of steam be supplied and x kg bled off, then 1 kg passes through H.P. and $(1 - x)$ kg passes through L.P.

$$1 \times 388 = (1 - x) \times 436$$
$$436x = 436 - 388$$
$$x = \frac{48}{436} = 0{\cdot}1101$$

Expressed as a percentage

Amount bled off = 11·01 % Ans.

9. Mass of air required per kg of fuel

$$= \frac{100}{23} \times \text{oxygen required}$$
$$= \frac{100}{23}\left\{2\tfrac{2}{3}C + 8\left(H_2 - \frac{O_2}{8}\right)\right\}$$
$$= \frac{100}{23}\{2\tfrac{2}{3}C + 8H_2 - O_2\}$$
$$= \frac{100}{23}\{2\tfrac{2}{3} \times 0{\cdot}87 + 8 \times 0{\cdot}11 - 0{\cdot}02\}$$
$$= \frac{100}{23} \times 3{\cdot}18 = 13{\cdot}82 \text{ kg air/kg fuel}$$

$$pV = mRT$$
$$V = \frac{13{\cdot}82 \times 0{\cdot}287 \times 298}{1 \times 10^2}$$
$$= 11{\cdot}82 \text{ m}^3 \text{ air/kg fuel} \quad \text{Ans.}$$

10.
$$p_1V_1^n = p_2V_2^n$$
$$0{\cdot}95 \times 0{\cdot}2^n = 2{\cdot}75 \times 0{\cdot}085^n$$
$$n = 1{\cdot}243 \quad \text{Ans. (i)}$$

$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2}$$

$$T_2 = \frac{290 \times 2{\cdot}75 \times 0{\cdot}085}{0{\cdot}95 \times 0{\cdot}2} = 356{\cdot}8 \text{ K}$$

Final temp. = 83·8°C Ans. (ii)

11. Area of top and bottom = $9 \times 4 \times 2$
Area of sides = $9 \times 2{\cdot}5 \times 2$
Area of ends = $4 \times 2{\cdot}5 \times 2$
∴ Area exposed to heat source = $72 + 45 + 20 = 137 \text{ m}^2$

Heat passing across inner film:

$$Q = h_i At\,(T_i - T_1) \quad \therefore T_i - T_1 = \frac{Q}{h_iAT}$$

Temperature rise across the three thicknesses:

$$T_1 - T_o = \frac{Q}{At}\left\{\frac{S_1}{k_1} + \frac{S_2}{k_2} + \frac{S_3}{k_3}\right\}$$

Heat passing across outer film:

$$Q = h_oAt\,(T_4 - T_o) \quad \therefore T_4 - T_0 = \frac{Q}{h_oAt}$$

$$T_i - T_4 = \frac{Q}{At}\left\{\frac{1}{h_i} + \frac{S_1}{k_1} + \frac{S_2}{k_2} + \frac{S_3}{k_3} + \frac{1}{h_o}\right\}$$

$$(-6 - 27) = \frac{Q}{137 \times 1}\left\{\frac{1}{1{\cdot}62} + \frac{0{\cdot}015}{0{\cdot}11} + \frac{0{\cdot}07}{0{\cdot}06} + \frac{0{\cdot}01}{45} + \frac{1}{13}\right\}$$

$$-33 \times 137 = Q\,(0{\cdot}6172 + 0{\cdot}1364 + 1{\cdot}167 + 0{\cdot}0002 + 0{\cdot}077)$$
$$Q = -2263{\cdot}3 \text{ W} \quad \text{i.e. into the room}$$
$$= 2{\cdot}2633 \text{ kW} \quad \text{cooling load} \quad \text{Ans.}$$

Note: $\frac{1}{U}$ equals the term above in brackets i.e. $U = 0{\cdot}5006$

$$Q = UAt(T_i - T_4)$$

12. From steam tables,
Steam 16 bar, $h_f = 859$ $h_{fg} = 1935$
Water 95°C, $h = 398$
Boiler steam $h_1 = 859 + 0{\cdot}98 \times 1935 = 2755$
Feed water $h_w = 398$

Heat energy transferred to water to make steam

$$= h_1 - h_w = 2755 - 398 = 2357 \text{ kJ/kg}$$
$$= 2357 \times 9000 \text{ kJ per hour}$$

Heat energy released by m kg of fuel per hour

$$= 42 \times 10^3 \times m \text{ kJ per hour}$$

$$\text{Boiler efficiency} = \frac{\text{heat energy transferred to steam}}{\text{heat energy supplied by fuel}}$$

$$0{\cdot}87 = \frac{2357 \times 9000}{42 \times 10^3 \times m}$$

$$m = \frac{2357 \times 9000}{0{\cdot}87 \times 42 \times 10^3} = 580{\cdot}6 \text{ kg/h}$$

Tonnes of fuel used per day

$$= 580{\cdot}6 \times 24 \times 10^3 = 13{\cdot}93 \text{ tonne/day} \quad \text{Ans.}$$

13. When scavenge ports are just closed, distance from cylinder cover to piston = 800 + 70 = 870 mm
Volume enclosed = $0{\cdot}7854 \times 0{\cdot}7^2 \times 0{\cdot}87 \text{ m}^3$
Volume of air taken in (scavenge effic. being 0·95)

$$= 0{\cdot}95 \times 0{\cdot}7854 \times 0{\cdot}7^2 \times 0{\cdot}87$$
$$= 0{\cdot}3181 \text{ m}^3$$
$$pV = mRT$$
$$m = \frac{pV}{RT} = \frac{1{\cdot}21 \times 10^2 \times 0{\cdot}3181}{0{\cdot}287 \times 313}$$
$$= 0{\cdot}4285 \text{ kg} \quad \text{Ans.}$$

14.
$$\text{Linear clearance [mm]} = \frac{\text{volumetric clearance [mm}^3]}{\text{area of cylinder [mm}^2]}$$
$$= \frac{73 \times 10^3}{0{\cdot}7854 \times 130^2} = 5{\cdot}5 \text{ mm}$$

Representing volumes by linear dimensions:

$$V_1 = 180 + 5{\cdot}5 = 185{\cdot}5 \text{ mm}$$
$$p_1V_1^{1\cdot2} = p_2V_2^{1\cdot2}$$
$$1 \times 185{\cdot}5^{1\cdot2} = 4{\cdot}6 \times V_2^{1\cdot2}$$
$$V_2 = \frac{185{\cdot}5}{\sqrt[1\cdot2]{4{\cdot}6}} = 52 \text{ mm}$$

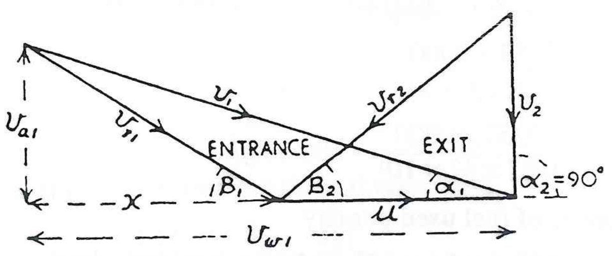
Movement of piston during delivery period

$= 52 - 5{\cdot}5 = 46{\cdot}5$ mm Ans. (i)

as a fraction of the stroke

$= \frac{46{\cdot}5}{180} = 0{\cdot}2583$ Ans. (ii)

15.



$$v_{a1} = v_1 \sin\alpha_1 = 500 \times \sin 18° = 154{\cdot}5 \text{ m/s}$$
$$v_{w1} = v_1 \cos\alpha_1 = 500 \times \cos 18° = 475{\cdot}5 \text{ m/s}$$
$$x = v_{w1} - u = 475{\cdot}5 - 230 = 245.5$$

$$\tan\beta_1 = \frac{v_{a1}}{x} = \frac{154{\cdot}5}{245{\cdot}5} = 0{\cdot}6292$$

$\therefore$ Inlet blade angle = 32° 11′ Ans. (i)

Neglecting friction across the blades

$$v_{r2} = v_{r1} = \frac{v_{a1}}{\sin\beta_1} = \frac{154{\cdot}5}{\sin 32°\ 11'} = 290 \text{ m/s}$$

$$\cos\beta_2 = \frac{u}{v_{r2}} = \frac{230}{290} = 0{\cdot}7930$$

$\therefore$ Outlet blade angle = 37° 32′ Ans. (ii)

16. Refer to Fig. 34 Ans. (a)

Stroke volume (s) $= 0{\cdot}7854 \times 0{\cdot}1^2 \times 0{\cdot}1$

$= 0{\cdot}0007854 \text{ m}^3$

Clearance volume (c) $= 0{\cdot}0007854 \times 0{\cdot}05$

$= 0{\cdot}00003927 \text{ m}^3$

$V_1 = s + c$ i.e. $0{\cdot}0008247$

$$pV = mRT$$
$$m = \frac{1 \times 100 \times 0{\cdot}0008247}{0{\cdot}287 \times 298}$$
$= 0{\cdot}0009642$ Ans. (b) (i)

The air trapped in the clearance space re-expands and reduces the intake volume on each suction stroke Ans. (b) (ii)

17. From tables for NH_3

10·34 bar, $h_f = 303{\cdot}7$

2·265 bar, $h_f = 107{\cdot}9$ $h_g = 1425{\cdot}3$ $v_g = 0{\cdot}5296$

$h_{fg} = 1425{\cdot}3 - 107{\cdot}9 = 1317{\cdot}4$

Ref. Fig. 69:

Throttling process through expansion valve:

Enthalpy after (h_4) = Enthalpy before (h_3)

$$107{\cdot}9 + x_4 \times 1317{\cdot}4 = 303{\cdot}7$$
$$x_4 \times 1317{\cdot}4 = 195{\cdot}8$$
$$x_4 = 0{\cdot}1486$$ Ans. (i)

Enthalpy gain of refrigerant through evaporator

$= h_1 - h_4$

$= (107{\cdot}9 + 0{\cdot}95 \times 1317{\cdot}4) - (107{\cdot}9 + 0{\cdot}1486 \times 1317{\cdot}4)$

$= (0{\cdot}95 - 0{\cdot}1486) \times 1317{\cdot}4$

$= 1055$ kJ/kg

Refrigerating effect per minute

$= 4 \times 1055 = 4220$ kJ/min Ans. (ii)

Specific volume of refrigerant entering compressor

$= 0{\cdot}95 \times 0{\cdot}5296 \text{ m}^3\text{/kg}$

Total volume entering compressor per minute

$= 4 \times 0{\cdot}95 \times 0{\cdot}5296 = 2{\cdot}013 \text{ m}^3\text{/min}$ Ans. (iii)

18.
$$\left.\begin{aligned} C + O_2 &= CO_2 \\ 2H_2 + O_2 &= 2H_2O \\ S + O_2 &= SO_2 \end{aligned}\right\} \text{Ans. (a)}$$

Stoichiometric air $= \frac{100}{23} \times$ oxygen required

$$= \frac{100}{23}\,(2\tfrac{2}{3}C + 8H_2 + S)\ \text{kg air/kg fuel}$$

$$= \frac{100}{23}\,(2{\cdot}667 \times 0{\cdot}863 + 8 \times 0{\cdot}128 + 0{\cdot}009)$$

$$= 14{\cdot}5\ \text{kg}$$

Actual air is $14{\cdot}5 \times 1{\cdot}25$
$= 18{\cdot}125$

Mass of the gases/kg fuel burned = 19·125 kg Ans. (b)(i)

$$pV = mRT$$

$$V = \frac{19{\cdot}125 \times 0{\cdot}276 \times 643}{1{\cdot}5 \times 100}$$

Volume of flue gases/kg fuel burned
$= 22{\cdot}63\ \text{m}^3$ Ans. (b)(ii)

19. Heat passing across water film:

$$Q = h_i At\,(T_i - T) \quad \therefore T_i - T = \frac{Q}{h_i At}$$

Heat passing across air film:

$$Q = h_o At\,(T - T_o) \quad \therefore T - T_o = \frac{Q}{h_o At}$$

$$T_i - T_o = \frac{Q}{At}\left\{\frac{1}{h_i} + \frac{1}{h_o}\right\}$$

$$100 - 85 = \frac{Q}{\pi \times 0{\cdot}155 \times 1}\left\{\frac{1}{240} + \frac{1}{12}\right\}$$

$$Q = \frac{85 \times \pi \times 0{\cdot}155 \times 1 \times 240}{21}$$

$$= 473{\cdot}1\ \text{W}\quad \text{Ans.}$$

Note: area A of large diameter pipe, of small thickness, is mean circumference (per unit length)

20. From steam tables,

17 bar,	$h_f = 872$	$h_{fg} = 1923$	$v_g = 0{\cdot}1167$
6 bar,	$h_f = 670$	$h_{fg} = 2087$	$v_g = 0{\cdot}3156$

Throttling process:

Enthalpy after = Enthalpy before

$$670 + x \times 2087 = 872 + 0{\cdot}95 \times 1923$$

$$x \times 2087 = 872 + 1827 - 670$$

$$x \times 2087 = 2029$$

$$x = 0{\cdot}9721 \quad \text{Ans. (i)}$$

Spec. vol. of steam before throttling
$= 0{\cdot}95 \times 0{\cdot}1167\ \text{m}^3/\text{kg}$

Mass of steam passed through

$$= \frac{0{\cdot}8}{0{\cdot}95 \times 0{\cdot}1167}\ \text{kg}$$

Spec. vol. of steam after throttling
$= 0{\cdot}9721 \times 0{\cdot}3156\ \text{m}^3/\text{kg}$

$$\text{Final volume of steam} = \frac{0{\cdot}8 \times 0{\cdot}9721 \times 0{\cdot}3156}{0{\cdot}95 \times 0{\cdot}1167}$$

$$= 2{\cdot}214\ \text{m}^3 \quad \text{Ans. (ii)}$$

21. Total feed in 24 hours
$= 1250 \times 10^{-3} \times 24 = 30$ tonne

$$\text{Solids in boiler initially} + \text{solids put in} = \text{solids in finally}$$

$$\text{water in boiler} \times \text{initial p.p.m.} + \text{amount of feed} \times \text{feed p.p.m.} = \text{water in boiler} \times \text{final p.p.m.}$$

$$6 \times 120 + 30 \times \text{feed p.p.m.} = 6 \times 1080$$

$$30 \times \text{feed p.p.m.} = 6480 - 720$$

$$\text{feed p.p.m.} = \frac{5760}{30}$$

$$= 192\ \text{p.p.m.} \quad \text{Ans.}$$

22. From steam tables

Steam 2·8 bar,	$= h_f = 551$	$h_{fg} = 2171$	$v_g = 0{\cdot}6462$
Water 16°C,	$= h = 67{\cdot}1$		
Water 55°C,	$= h = 230{\cdot}2$		

Let x = dryness fraction
m = mass of steam [kg]

$$\text{Total enthalpy of steam and water before mixing} = \text{Total enthalpy of water after mixing}$$

$$m\,(551 + x \times 2171) + (36 \times 67{\cdot}1) = (m + 36) \times 230{\cdot}2$$

$$551m + 2171mx + 2415 = 230{\cdot}2m + 8287$$

$$320{\cdot}8m + 2171mx = 5872$$

$$320{\cdot}8 + 2171x = \frac{5872}{m} \quad \ldots \ldots \ldots \ldots \ldots \quad \text{(i)}$$

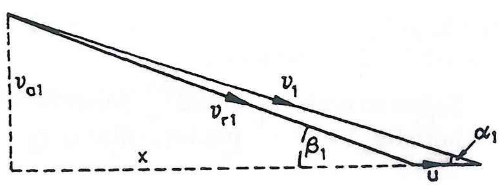
Also, spec. vol. of steam = $x \times 0{\cdot}6462$ m^3/kg
$\therefore$ mass of 1·5 m^3 of wet steam

$$m = \frac{1{\cdot}5}{x \times 0{\cdot}6462} \text{ kg} \quad \ldots \ldots \ldots \quad \text{(ii)}$$

Substituting for m from (ii) into (i):

$$320{\cdot}8 + 2171x = \frac{5872 \times 0{\cdot}6462x}{1{\cdot}5}$$
$$320{\cdot}8 + 2171x = 2528x$$
$$320{\cdot}8 = 357x$$
$$x = 0{\cdot}8984 \quad \text{Ans.}$$

23.



$$u = \pi dn$$
$$120 = \pi \times 0{\cdot}6 \times n$$
$$n = 63{\cdot}65 \text{ rev/s} = 3819{\cdot}0 \text{ rev/min} \quad \text{Ans. (a)}$$
$$v_1 = \sqrt{2 \times 10^3 \times \text{spec. enthalpy drop}}$$
$$= \sqrt{2 \times 10^3 \times 465}$$
$$= 974 \text{ m/s}$$
$$x = v_1 \cos \alpha_1 - u$$
$$= 974 \cos 18° - 120$$
$$= 806{\cdot}4$$
$$v_{a1} = 974 \sin 18°$$
$$= 301 \text{ m/s}$$
$$\tan \beta_1 = \frac{301}{974}$$
$$\beta_1 = 20{\cdot}5°$$

Blade inlet angle = 20·5° Ans. (b)
Axial component of steam at blade inlet = 301 m/s Ans. (c)

24. Volume of air to be heated every hour
$= 27 \times 15 \times 3 \times 2 = 2430$ m^3

Mass of air to be heated every hour:

$$pV = mRT$$
$$m = \frac{pV}{RT}$$
$$= \frac{1{\cdot}013 \times 10^2 \times 2430}{0{\cdot}287 \times 291} = 2937 \text{ kg/h}$$

Heat energy supplied = mass × spec. heat × temp. rise

Energy supplied per second

$$= \frac{2937}{3600} \times 1{\cdot}005 \times (19 - 7)$$
$$= 9{\cdot}837 \text{ kJ/s}$$

Kilowatt loading = 9·837 kW Ans.

25. For a 4-stroke single-acting engine

$$n = \text{rev/s} \div 2 = 2{\cdot}5$$

For six cylinders:

$$\text{ip} = p_m ALn \times 6$$
$$= 17 \times 10^2 \times 0{\cdot}7854 \times 0{\cdot}225^2 \times 0{\cdot}2 \times 2{\cdot}5 \times 6$$
$$= 202{\cdot}8 \text{ kW} \quad \text{Ans. (i)}$$
$$\text{bp} = \text{ip} \times \text{mech. efficiency}$$
$$= 202{\cdot}8 \times 0{\cdot}85$$
$$= 172{\cdot}36 \text{ kW} \quad \text{Ans. (ii)}$$

26. Refer to Fig. 35a Ans. (a)

Volume of receiver = 3 m^3

Volume of free air = $3 \times \frac{7}{1} = 21$ m^3

(at the same temperature)

Volume to compress = $21 - 3 = 18$ m^3 at 25°C

$$\frac{V_1}{T_1} = \frac{V_2}{T_2}$$
$$\frac{V_1}{290} = \frac{18}{298}$$
$$V_1 = 17{\cdot}52 \text{ m}^3$$

Volume to compress = 17·52 m^3 at 17°C
suction volume/s = $0{\cdot}7854 \times 0{\cdot}1^2 \times 0{\cdot}2 \times 5$
$= 0{\cdot}007854$ m^3/s

$$\text{Time to charge receiver} = \frac{17{\cdot}52}{0{\cdot}007854 \times 60}$$

$$= 37{\cdot}18 \text{ min} \quad \text{Ans. (b)}$$

27. Refer to Figs. 69 and 70 Ans. (a)

$$h_3 = h_4 = 294{\cdot}1 \text{ kJ/kg}$$

i.e. from tables, page 12, at 24°C

$$h_4 = h_f + x_4(h_g - h_f)$$
$$294{\cdot}1 = 126{\cdot}2 + x_4\,(1430{\cdot}5 - 126{\cdot}2)$$

i.e. from tables, page 12, at 2·68 bar

$$x_4 = 0{\cdot}1287$$
$$\begin{aligned} \text{Heat extracted} &= h_1 - h_4 \\ &= h_g - x_4\,h_{fg} \\ &= 1430{\cdot}5 - 0{\cdot}1287\,(1430{\cdot}5 - 126{\cdot}2) \\ &= 1262{\cdot}2 \text{ kJ/kg} \end{aligned}$$

$$3{\cdot}5 = \dot{m} \times 1262{\cdot}6$$
$$\dot{m} = 0{\cdot}00308 \text{ kg/s}$$

Mass flows rate of ammonia = 0·00308 kg/s Ans. (b)

28. From steam tables, 15 bar, $h_g = 2792$
Mass of flue gases per kg of fuel

$$= 1 \text{ kg fuel} + 24 \text{ kg air} = 25 \text{ kg}$$

Heat energy transferred from flue gases per hour

$$\begin{aligned} &= \text{mass} \times \text{spec. heat} \times \text{temp. change} \\ &= 25 \times 750 \times 1{\cdot}007 \times (822 - 690) \\ &= 2{\cdot}492 \times 10^6 \text{ kJ/h} \end{aligned}$$

Let h = spec. enthalpy of the superheated steam,

Heat energy transferred to steam per hour to superheat it

$$= 7000 \times (h - 2792)$$

$$\text{Heat gained by steam} = \text{Heat lost by gases}$$
$$7000(h - 2792) = 2{\cdot}492 \times 10^6$$

$$h - 2792 = \frac{2{\cdot}492 \times 10^6}{7000}$$

$$h = 356 + 2792 = 3148 \text{ kJ/kg}$$

From superheated steam tables,
h for 15 bar 350°C reads 3148
∴ Temperature of steam = 350°C Ans. (i)

From steam tables, water 21°C, $h = 88.0$

Enthalpy of 1 kg sup. steam and m kg water entering desuperheater = Enthalpy of $(1 + m)$ kg of dry sat. steam leaving desuperheater

$$\begin{aligned} (1 \times 3148) + m \times 88 &= (1 + m) \times 2792 \\ 3148 + 88m &= 2792 + 2792\,m \\ 356 &= 2704\,m \\ m &= 0{\cdot}1316 \text{ kg} \quad \text{Ans. (ii)} \end{aligned}$$

29. $$\text{Ratio of expansion} = \frac{\text{final volume}}{\text{initial volume}}$$

∴ final volume = $6 \times 9900 = 59400$ cm³

or, 59·4 litre, or, 0·0594 m³ Ans. (i)

$$p_1V_1^{1{\cdot}33} = p_2V_2^{1{\cdot}33}$$

where V_1 and V_2 may be represented by 1 and 6 respectively.

$$12 \times 1^{1{\cdot}33} = p_2 \times 6^{1{\cdot}33}$$

$$p_2 = \frac{12}{6^{1{\cdot}33}} = 1{\cdot}108 \text{ bar} \quad \text{Ans. (ii)}$$

$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2}$$

$$\frac{12 \times 1}{489} = \frac{1{\cdot}108 \times 6}{T_2}$$

$$T_2 = \frac{489 \times 1{\cdot}1018 \times 6}{12} = 270{\cdot}9 \text{ K}$$

$$= -2{\cdot}1\text{°C} \quad \text{Ans. (iii)}$$

30. $$\begin{aligned} p_1v_1^{\,n} &= p_2v_2^{\,n} \\ 40 \times 0{\cdot}08^{1{\cdot}3} &= 2 \times v_2^{1{\cdot}3} \\ v_2 &= 0{\cdot}8013 \\ 0{\cdot}8856x &= 0{\cdot}8013 \\ x &= 0{\cdot}9048 \end{aligned}$$

$$\text{work transfer} = \frac{p_1v_1 - p_2v_2}{n-1}$$

$$= \frac{40 \times 10^2 \times 0{\cdot}08 - 2 \times 10^2 \times 0{\cdot}8013}{0{\cdot}3}$$

$$= 532{\cdot}5 \text{ kJ} \quad \text{Ans. (a)}$$

$$u_1 + q_{in} = u_2 + w_{out}$$

At 40 bar, 0·08 m^3/kg, steam is 450°C (tables)

$$3010 + q_{in} = (505 + 0{\cdot}9048 \times 2025) + 532{\cdot}5$$
$$q_{in} = -3010 + 2337{\cdot}2 + 532{\cdot}5$$

Heat transfer = −140·3 kJ i.e. loss Ans. (b)

Note: this is non-flow work, u not h is used.

$$u_1 = h - pv_g$$
$$= 3330 - 40 \times 10^2 \times 0{\cdot}08$$
$$= 2337{\cdot}2 \text{ (as above)}$$

Similarly for u_2, by calculation

31. Referring to Fig. 57:

$$v_{a1} = v_1 \sin \alpha_1 = 840 \times \sin 20° = 287{\cdot}4 \text{ m/s}$$
$$v_{w1} = v_1 \cos \alpha_1 = 840 \times \cos 20° = 789{\cdot}4 \text{ m/s}$$
$$x = v_{w1} - u = 789{\cdot}4 - 350 = 439{\cdot}4 \text{ m/s}$$

$$\tan \beta_1 = \frac{V_{a1}}{x} = \frac{287{\cdot}4}{439{\cdot}4} = 0{\cdot}6539$$

∴ Blade inlet angle = 33°11′ Ans. (i)

$$v_{r1} = \frac{v_{a1}}{\sin \beta_1} = \frac{287{\cdot}4}{\sin 33° \, 11'} = 525 \text{ m/s}$$

$$v_{r2} = 0{\cdot}8 \times 525 = 420 \text{ m/s}$$
$$v_{a2} = v_{r2} \times \sin \beta_2 = 420 \times \sin 25° \, 12' = 178{\cdot}8 \text{ m/s}$$
$$v_{w2} + u = v_{r2} \times \cos \beta_2 = 420 \times \cos 25° \, 12' = 380 \text{ m/s}$$
$$v_{w2} = 380 - 350 = 30 \text{ m/s}$$

$$\tan \varphi = \frac{v_{w2}}{v_{a2}} = \frac{30}{178{\cdot}8} = 0{\cdot}1678$$

$$\varphi = 9° \, 32'$$

$$v_2 = \frac{v_{a2}}{\cos \varphi} = \frac{178{\cdot}8}{\cos 9° \, 32'} = 181{\cdot}3 \text{ m/s}$$

Absolute velocity of steam at exit:

$$\left.\begin{aligned} \text{Magnitude} &= 181{\cdot}3 \text{ m/s} \\ \text{Direction to axis} &= 9° \, 32' \\ \text{or, direction to plane of wheel} &= 80 \, 28' \end{aligned}\right\} \quad \text{Ans. (ii)}$$

32. Available hydrogen $= H_2 - \dfrac{O_2}{8}$

$$= 0{\cdot}114 - \frac{0{\cdot}024}{8} = 0{\cdot}111 \text{ kg}$$

$$\text{c.v.} = 33{\cdot}7 \text{ C} + 144 \left[H_2 - \frac{O_2}{8}\right] + 9{\cdot}3 \text{ S}$$
$$= 33{\cdot}7 \times 0{\cdot}846 + 144 \times 0{\cdot}111 + 9{\cdot}3 \times 0{\cdot}004$$
$$= 28{\cdot}51 + 15{\cdot}99 + 0{\cdot}0372$$
$$= 44{\cdot}5372 \text{ MJ/kg} \quad \text{Ans. (i)}$$

$$\text{Air required} = \frac{100}{23} \times \text{oxygen required}$$
$$= \frac{100}{23}\left\{2\tfrac{2}{3} \text{ C} + 8\left[H_2 - \frac{O_2}{8}\right] + \text{S}\right\}$$
$$= \frac{100}{23}\{2\tfrac{2}{3} \times 0{\cdot}846 + 8 \times 0{\cdot}111 + 0{\cdot}004\}$$
$$= \frac{100}{23} \times 3{\cdot}148$$
$$= 13{\cdot}69 \text{ kg air/kg fuel} \quad \text{Ans. (ii)}$$

33. From tables, Freon-12, 1·826 bar,

$$h_f = 22{\cdot}33 \qquad h_g = 180{\cdot}97$$
$$h_{fg} = 180{\cdot}97 - 22{\cdot}33 = 158{\cdot}64$$

Sat. Temp. at 1·826 bar = – 15°C therefore freon at evaporator outlet at 0°C is superheated by 15°.

1·826 bar 15° superheat, $h = 190{\cdot}15$

Referring to Fig. 69:

Throttling effect through expansion valve between condenser outlet and evaporator inlet:

$$\text{Enthalpy after throttling } (h_4) = \text{Enthalpy before } (h_3)$$
$$22{\cdot}33 + x_4 \times 158{\cdot}64 = 50$$
$$x_4 \times 158{\cdot}64 = 27{\cdot}67$$
$$x_4 = 0{\cdot}1744 \quad \text{Ans. (i)}$$

$$\text{Refrig. effect/kg} = h_1 - h_4 \text{ (note } h_4 = h_3)$$
$$= 190{\cdot}15 - 50$$
$$= 140{\cdot}15 \text{ kJ/kg}$$

Refrigerating effect per minute

$$= 0{\cdot}4 \times 60 \times 140{\cdot}15$$
$$= 3364 \text{ kJ/min} \quad \text{Ans. (ii)}$$

34. $$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2}$$

$$\frac{1{\cdot}01325 \times 18{\cdot}2}{288} = \frac{0{\cdot}965 \times V_2}{300}$$

$$V_2 = 19{\cdot}91 \text{ m}^3\text{/min}$$

$$p_2V_2^n = p_3V_3^n$$

$$0{\cdot}965 \times 19{\cdot}91^{1\cdot32} = 4{\cdot}82 \times V_3^{1\cdot32}$$

$$V_3 = 5{\cdot}891 \text{ m}^3\text{/min}$$

$$\text{Work/cycle} = \frac{n}{n-1}(p_3V_3 - p_2V_2)$$

$$\text{Work/s} = \frac{1{\cdot}32 \times 10^2}{0{\cdot}32}\left(\frac{4{\cdot}82 \times 5{\cdot}891}{60 \times 2} - \frac{0{\cdot}965 \times 19{\cdot}91}{60 \times 2}\right)$$

$$= 31{\cdot}56 \text{ kW}$$

$$\text{Input power} = \frac{31{\cdot}56}{0{\cdot}9}$$

$$= 35{\cdot}06 \text{ kW} \quad \text{Ans.}$$

35. For one mm movement of the stylus (one mm on height of card) indicator piston deflection = ⅙ mm and this would be under a force of 60 ÷ 6 = 10 N in the indicator cylinder.

$$\text{force [N]} = \text{pressure [N/m}^2] \times \text{area [m}^2]$$

$$10 = \text{pressure [N/m}^2] \times 0{\cdot}7854 \times 7^2 \times 10^{-4}$$

∴ pressure scale on card per mm of height

$$= \frac{10}{0{\cdot}7854 \times 7^2 \times 10^{-6}}$$

$$= 2{\cdot}598 \times 10^5 \text{ N/m}^2 = 2{\cdot}598 \text{ bar}$$

$$\text{Mean height of diagram [mm]} = \frac{\text{area [mm}^2]}{\text{length [mm]}}$$

$$= \frac{346}{75} \text{ mm}$$

$$\therefore \text{Mean effective press.} = \frac{346}{75} \times 2{\cdot}598$$

$$= 11{\cdot}99 \text{ bar} \quad \text{Ans. (i)}$$

$$\text{ip} = p_mALn$$

where n = 2 × rev/s for a double-acting 2-stroke

$$\text{ip} = 11{\cdot}99 \times 10^2 \times 0{\cdot}7854 \times 0{\cdot}6^2 \times 0{\cdot}9 \times 2{\cdot}1 \times 2$$

$$= 1281 \text{ kW} \quad \text{Ans. (ii)}$$

36. $$\text{Cross–sect. area [m}^2] = \frac{\text{volume flow [m}^3\text{/s]}}{\text{velocity [m/s]}}$$

$$= \frac{\text{mass flow [kg/s]} \times \text{spec. vol. [m}^3\text{/kg]}}{\text{velocity [m/s]}}$$

$$\text{Diameter} = \sqrt{\frac{\text{area}}{0{\cdot}7854}}$$

$$\text{Entrance diameter} = \sqrt{\frac{0{\cdot}315 \times 0{\cdot}2765}{457 \times 0{\cdot}7854}}$$

$$= 0{\cdot}01558 \text{ m} = 15{\cdot}58 \text{ mm} \quad \text{Ans. (ii)}$$

37. $$pV = mRT$$

$$m = \frac{pV}{RT} = \frac{1{\cdot}1 \times 10^2 \times 0{\cdot}2}{0{\cdot}287 \times 288} = 0{\cdot}2662 \text{ kg}$$

Heat energy supplied [kJ]

$$= \text{mass [kg]} \times \text{spec. heat [kJ/kg K]} \times \text{temp. rise [K]}$$

$$= 0{\cdot}2662 \times 1{\cdot}005 \times (423 - 288)$$

$$= 36{\cdot}1 \text{ kJ} \quad \text{Ans. (i)}$$

$$p_1V_1^{1\cdot32} = p_2V_2^{1\cdot32}$$

$$1{\cdot}1 \times 0{\cdot}2^{1\cdot32} = 7{\cdot}15 \times V_2^{1\cdot32}$$

$$V_2 = 0{\cdot}2 \times \sqrt[1\cdot32]{\frac{1{\cdot}1}{7{\cdot}15}}$$

$$= 0{\cdot}04844 \text{ m}^3$$

$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2}$$

$$\frac{1{\cdot}1 \times 0{\cdot}2}{288} = \frac{7{\cdot}15 \times 0{\cdot}04844}{T_2}$$

$$T_2 = \frac{288 \times 7{\cdot}15 \times 0{\cdot}04844}{1{\cdot}1 \times 0{\cdot}2} = 453{\cdot}4 \text{ K}$$

$$= 180{\cdot}4\text{°C} \quad \text{Ans. (ii)}$$

38. For the two thicknesses:

$$T_1 - T_3 = \frac{Q}{At}\left\{\frac{S_1}{k_1} + \frac{S_2}{k_2}\right\}$$

$$-4-21 = \frac{Q}{6\times3\times1}\left\{\frac{0{\cdot}12}{1{\cdot}15}+\frac{0{\cdot}08}{0{\cdot}043}\right\}$$

$$Q = \frac{-25\times18}{0{\cdot}1043+1{\cdot}860}$$

$= -229{\cdot}09$ W i.e. *into* the store

$$Q = 229{\cdot}09\times3600\times24 \text{ J/day}$$

$= 19{\cdot}793$ MJ/day Ans. (a)

$$T_1 - T_2 = \frac{-229{\cdot}09}{6\times3\times1}\left\{\frac{0{\cdot}08}{0{\cdot}043}\right\}$$

$$-4 - T_2 = -23{\cdot}67$$

$T_2 = 19{\cdot}67$°C Ans. (b)

39. Refer to Fig. 33 Ans. (a)

$$\text{Clearance volume} = 0{\cdot}07\times381$$
$$= 26{\cdot}67 \text{ mm}$$
$$V_1 = 381+26{\cdot}67$$
$$= 407{\cdot}67 \text{ mm}$$
$$V_2 = 407{\cdot}67-267$$
$$= 140{\cdot}67 \text{ mm}$$

$$p_1V_1^n = p_2V_2^n$$
$$1{\cdot}013\times407{\cdot}67^n = 4\times0{\cdot}1407^n$$
$$2{\cdot}898^n = 3{\cdot}949$$

$n = 1{\cdot}291$ Ans. (b)

40. Stoichiometric air required per kg of fuel A

$$= \frac{100}{23}\{2\tfrac{2}{3}\times0{\cdot}885+8\times0{\cdot}115\}$$
$$= \frac{100}{23}\times3{\cdot}28 \text{ kg}$$

Stoichiometric air required per kg of fuel B

$$= \frac{100}{23}\{2\tfrac{2}{3}\,C+8\,H_2\}$$

and this is 6% more than for fuel A, therefore,

$$\frac{100}{23}\{2\tfrac{2}{3}C+8H_2\} = 1{\cdot}06\times\frac{100}{23}\times3{\cdot}28$$

Cancelling $\frac{100}{23}$ and multiplying throughout by $\frac{3}{8}$:

$$C + 3H_2 = 1{\cdot}304 \quad \ldots \quad \text{(i)}$$

Also, fractional analysis of fuel B:

$$C + H_2 = 1$$
$$\therefore C = 1 - H_2 \quad \ldots \quad \text{(ii)}$$

Substituting value of C from (ii) into (i):

$$C + 3H_2 = 1{\cdot}304$$
$$1 - H_2 + 3H_2 = 1{\cdot}304$$
$$2H_2 = 0{\cdot}304$$
$$H_2 = 0{\cdot}152$$
$$\text{and, } C = 1 - 0{\cdot}152 = 0{\cdot}848$$

$\therefore$ Mass analysis of fuel B =

84·8% carbon, 15·2% hydrogen Ans.

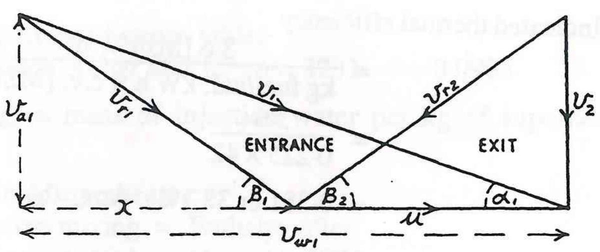
41. Before alteration:

$$V_1 = 90+15 = 105 \qquad V_2 = 15$$
$$p_1V_1^{1\cdot33} = p_2V_2^{1\cdot33}$$
$$1\times105^{1\cdot33} = p_2\times15^{1\cdot33}$$
$$p_2 = \left\{\frac{105}{15}\right\}^{1\cdot33} = 13{\cdot}3 \text{ bar}$$ Ans. (i)

After alteration:

$$V_1 = 90+12{\cdot}5 = 102{\cdot}5 \qquad V_2 = 12{\cdot}5$$
$$1\times102{\cdot}5^{1\cdot33} = p_2\times12{\cdot}5^{1\cdot33}$$
$$p_2 = \left\{\frac{102{\cdot}5}{12{\cdot}5}\right\}^{1\cdot33} = 16{\cdot}43 \text{ bar}$$ Ans. (ii)

42.



Kinetic energy = $\frac{1}{2}mv^2$

Kinetic energy being in joules when m is the mass in kg and v is the velocity in m/s, hence,

$$250\times10^3 = \tfrac{1}{2}\times1\times v^2$$
$$v = \sqrt{2\times250\times10^3} = 707{\cdot}1 \text{ m/s}$$

With no friction, $v_{r2} = v_{r1}$
and, since $\beta_2 = \beta_1$
then $x = u = \frac{1}{2} v_{w1}$

$$\frac{v_{a1}}{x} = \tan\beta_1 \qquad \therefore \frac{v_1 \sin\alpha_1}{\frac{1}{2} v_1 \cos\alpha_1} = \tan 35^\circ$$

v_1 cancels, $\sin\alpha_1 \div \cos\alpha_1 = \tan\alpha_1$, therefore

$$2\tan\alpha_1 = \tan 35^\circ$$

$$\tan\alpha_1 = \frac{0{\cdot}7002}{2} = 0{\cdot}3501$$

$\therefore$ Nozzle angle $\alpha_1 = 19^\circ\ 18'$ Ans. (i)

Blade velocity $u = \frac{1}{2} v_{w1} = \frac{1}{2} v_1 \cos\alpha_1$
$= \frac{1}{2} \times 707{\cdot}1 \times \cos 19^\circ\ 18'$
$= 333{\cdot}7$ m/s Ans. (ii)

43. Let d = diameter of cylinder
then $1{\cdot}25d$ = stroke

For 4 cylinders:

$$\text{ip} = p_m ALn \times 4$$

$$600 = 12{\cdot}56 \times 10^2 \times 0{\cdot}7854 d^2 \times 1{\cdot}25d \times 4{\cdot}5 \times 4$$

$$d = \sqrt[3]{\frac{600}{1256 \times 0{\cdot}7854 \times 1{\cdot}25 \times 4{\cdot}5 \times 4}}$$

$$= 0{\cdot}3 \text{ m} = 300 \text{ mm}$$

Cylinder diameter = 300 mm
Stroke = $1{\cdot}25 \times 300$ = 375 mm } Ans. (a)

Indicated thermal efficiency

$$= \frac{3{\cdot}6\ [\text{MJ/kW h}]}{\text{kg fuel/ind. kW h} \times \text{c.v. [MJ/kg]}}$$

$$= \frac{3{\cdot}6}{0{\cdot}225 \times 42}$$

$= 0{\cdot}381$ or $38{\cdot}1\%$ Ans. (b)

44. From NH_3 tables,
$1{\cdot}902$ bar, $h_g = 1420$
$7{\cdot}529$ bar, $h_f = 256$, sat. temp. = 16°C

$\therefore$ at 66°C vapour is superheated 50°
$7{\cdot}529$ bar 50° superheat, $h = 1591{\cdot}7$

Referring to Fig. 69
Enthalpy gain per kg through evaporator

$= h_1 - h_4$ (note $h_4 = h_3$)
$= 1420 - 256 = 1164$ kJ/kg

Refrigerating effect [kJ/min]

= mass flow [kg/min] × enthalpy gain [kJ/kg]

$\therefore\ m = \frac{800}{1164} = 0{\cdot}6873$ kg/min Ans.

Enthalpy drop per kg through condenser
$= h_2 - h_3$
$= 1591{\cdot}7 - 256 = 1335{\cdot}7$ kJ/kg

Heat rejected in condenser
$= 0{\cdot}6873 \times 1335{\cdot}7 = 918{\cdot}1$ kJ/min Ans. (ii)

Enthalpy gain per kg in compressor
$= h_2 - h_1$
$= 1591{\cdot}7 - 1420 = 171{\cdot}7$ kJ/kg

Energy given to refrigerant in compressor
$= 0{\cdot}6873 \times 171{\cdot}7$ kJ/min

Power [kW = kJ/s]

$$= \frac{0{\cdot}6873 \times 171{\cdot}7}{60}$$

$= 1{\cdot}967$ kW Ans. (iii)

45. From saturated steam tables:

Steam 30 bar, $h_g = 2803$ $v_g = 0{\cdot}06665$
Water 38°C, $h = 159{\cdot}1$

From superheated steam tables
Steam 30 bar 400°C, $h = 3231$ $v = 0{\cdot}0993$

Let m[kg] = mass of injection water per kg of superheated steam.

Mixing in desuperheater:

Enthalpy before mixing = Enthalpy after
$1 \times 3231 + m \times 159{\cdot}1 = (1 + m) \times 2803$
$3231 + 159{\cdot}1\,m = 2803 + 2803m$
$428 = 2643{\cdot}9m$
$m = 0{\cdot}1618$ kg Ans. (i)

Volume of one kg superheated steam = $0{\cdot}0993$ m^3
Volume of $(1 + m)$ kg of dry saturated steam

$$= 1{\cdot}1618 \times 0{\cdot}06665 = 0{\cdot}07743 \text{ m}^3$$

Percentage change in volume

$$= \frac{0{\cdot}0993 - 0{\cdot}07743}{0{\cdot}0993} \times 100$$

$$= 22{\cdot}03\% \quad \text{Ans. (ii)}$$

46. Let V_1 = volume of gas in cylinder at beginning of stroke, this is stroke volume + clearance volume.
V_2 = volume of gas in cylinder at end of stroke, this is the clearance volume.

$$p_1 V_1^{1{\cdot}32} = p_2 V_2^{1{\cdot}32}$$

$$1 \times V_1^{1{\cdot}32} = 37 \times 850^{1{\cdot}32}$$

$$V_1 = 850 \times \sqrt[1{\cdot}32]{37} = 13\,100 \text{cm}^3$$

$$\text{Stroke volume} = 13100 - 850 = 12250 \text{ cm}^3$$

$$= 12{\cdot}25 \text{ litre or } 0{\cdot}01225 \text{ m}^3 \quad \text{Ans. (i)}$$

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$$

$$\frac{1 \times 13100}{308} = \frac{37 \times 850}{T_2}$$

$$T_2 = \frac{308 \times 37 \times 850}{13100} = 739{\cdot}4 \text{ K}$$

$$= 466{\cdot}4°\text{C} \quad \text{Ans.}$$

47. Let x be kg of C/kg fuel
$1 - x$ is kg of H_2/kg fuel

$$\text{Oxygen required for C} = 2{\cdot}667x$$

$$\text{Oxygen required for H}_2 = 8(1 - x)$$

$$\text{Stoichiometric air} = \frac{100}{23}(2{\cdot}667x + 8 - 8x)$$

Exhaust gas includes 1 kg of fuel burned

$$15{\cdot}6 - 1 = \frac{100}{23}(2{\cdot}667x + 8 - 8x)$$

$$3{\cdot}358 - 8 = -5{\cdot}333x$$

$$x = 0{\cdot}8704$$

$$1 - x = 0{\cdot}1296$$

$$\left.\begin{aligned} \text{Mass of carbon} &= 0{\cdot}8704 \text{ kg} \\ \text{Mass of hydrogen} &= 0{\cdot}1296 \text{ kg} \end{aligned}\right\} \text{Ans.}$$

48.

$$\text{Heat extracted} = 27{\cdot}2(4{\cdot}186 \times 14{\cdot}4 + 332{\cdot}6) \text{ kJ/h}$$

$$= \frac{10686{\cdot}3 \times 10^3}{3600} \text{ J/s}$$

$$= 2968{\cdot}4$$

$$\text{c.o.p.} = \frac{\text{heat extracted by refrigerant}}{\text{work done on refrigerant}}$$

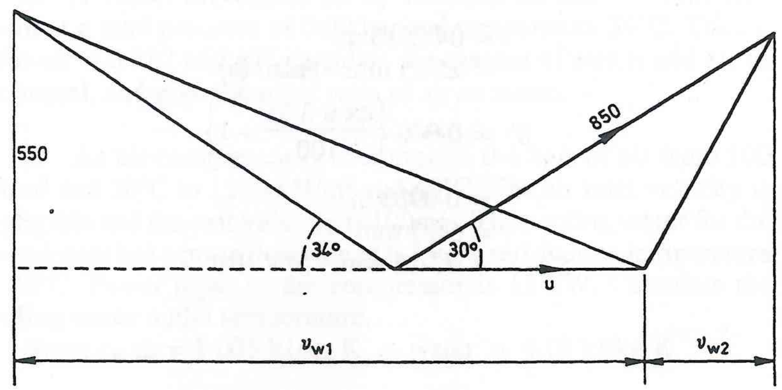
$$\text{Work done on refrigerant} = \frac{2968{\cdot}4}{7{\cdot}51}$$

$$= 395 \text{ W}$$

$$\text{Power input to machine} = 395\text{W} \quad \text{Ans. (a)}$$

For a heat engine; heat is taken in and heat is given out (at a lower temperature) and overall there is a net work output. Theoretically the cycle can be operated in reverse (reversibility), i.e. heat taken in at a lower temperature and given out at a high temperature but it does require a net work input (heat pump or refrigerator) Ans.

49.



$$v_{w1} + v_{w2} = \frac{550}{\tan 34°} + 850 \cos 30°$$

$$= 815{\cdot}4 + 736{\cdot}1$$

$$v_w = 1551{\cdot}5 \text{ m/s}$$

$$\text{Power} = \text{Force} \times \text{velocity}$$

$$= \dot{m} v_w u$$

$$1000 \times 67 = 0{\cdot}0833 \times 1551{\cdot}5 \times \pi \times 1{\cdot}32 \times n$$

$$n = 124{\cdot}9 \text{ rev/s}$$

$$\text{Turbine rotational speed} = 124{\cdot}9 \text{ rev/s} \quad \text{Ans.}$$

50.
$$\text{Area of top} = 1 \times 0{\cdot}75$$
$$\text{Area of sides} = 1 \times 1{\cdot}6 \times 2$$
$$\text{Area of ends} = 1{\cdot}6 \times 0{\cdot}75 \times 2$$
$$\text{Area exposed to heat source} = 0{\cdot}75 + 3{\cdot}2 + 2{\cdot}4 = 6{\cdot}35 \text{ m}^2$$

Temperature fall across the lagging:

$$T_i - T = \frac{QS}{Atk}$$

Heat passing across air film:

$$Q = h_o At\,(T - T_o) \quad \therefore \quad T - T_o = \frac{Q}{h_o At}$$

$$T_i - T_o = \frac{Q}{AT}\left\{\frac{S}{0{\cdot}048} + \frac{1}{1}\right\}$$

$$66 - 18 = \frac{200}{6{\cdot}35 \times 1}\left\{\frac{S}{0{\cdot}048} + 1\right\}$$

$$S = 0{\cdot}048\left\{\frac{48 \times 6{\cdot}35}{200} - 1\right\}$$
$$= 0{\cdot}02515 \text{ m}$$
$$= 25{\cdot}15 \text{ mm} \quad \text{Ans. (a)}$$

$$S' = 0{\cdot}048\left\{\frac{48 \times 6{\cdot}35}{100} - 1\right\}$$
$$= 0{\cdot}0983 \text{ m}$$
$$= 98{\cdot}3 \text{ mm}$$
$$S' - S = 73{\cdot}15 \text{ mm} \quad \text{Ans. (b)}$$

A SELECTION OF EXAMINATION QUESTIONS
CLASS ONE

1. A simple open-cycle gas turbine plant operates at a pressure ratio of 5:1 and at an air/fuel ratio of 50:1. The power output of the plant is 5 MW with maximum and minimum temperatures in the cycle of 925 K and 300 K respectively. The isentropic efficiencies of the compressor and turbine are 0·82 and 0·86 respectively. Calculate:
(a) the temperature of the exhaust gases leaving the turbine;
(b) the mass of fuel used per second

For air: $c_P = 1005$ J/kg K, $\dfrac{c_P}{c_V} = 1{\cdot}4$

For the turbine gases: $c_P = 1150$ J/kg K, $\dfrac{c_P}{c_V} = 1{\cdot}3$

2. A vessel of volume 15 m^3 contains air and dry saturated steam at a total pressure of 0·09 bar and temperature 39°C. Taking R for air = 0·287 kJ/kg K calculate the masses of steam and air in the vessel, and give the mass ratio of air to steam.

3. An air compressor is to compress 0·4 kg/s of air from 100 kN/m^2 and 20°C to 1500 kN/m^2 and 40°C. The air inlet velocity is negligible and the exit velocity is 100m/s. The cooling water for the compressor has a mass flow rate of 0·1 kg/s and an inlet temperature of 20°C. Power input to the compressor is 13 kW. Calculate the cooling water outlet temperature.
Note: c_P air = 1·005 kJ/kg K, c_P water = 4·18 kJ/kg K

4. The mass analysis of the fuel burned in a boiler is 87% carbon, 11% hydrogen, and 2% oxygen, and the fuel is burned at the rate of 1·8 tonne/h. Calculate the mass flow rate [kg/s] of each of the constituents of the flue gases if combustion is stoichiometric. Express the composition of the flue gases as percentages by mass.
Atomic weights: hydrogen 1, carbon 12, nitrogen 14, oxygen 16. Mass composition of air: 23% oxygen, 77% nitrogen.

5. Dry saturated steam enters a convergent-divergent nozzle at 9 bar, and the pressures at the throat and exit are 5 bar and 0·14 bar respectively. The specific enthalpy drop of the steam from entrance to throat is 107 kJ/kg, and from entrance to exit it is 633

kJ/kg. Assuming 8% of the enthalpy drop is lost to friction in the divergent part of the nozzle, calculate the areas in mm^2 of the nozzle at the throat and exit to pass 23 kg of steam per minute.

6. In a Freon-12 refrigerating plant the compressor takes the refrigerant in at 0·8071 bar and discharges it at 12·19 bar and 65°C. At condenser outlet the Freon is saturated liquid at 12·19 bar. If compression is isentropic and the flow of the refrigerant is 15 kg/min, calculate the refrigerating effect and the coefficient of performance.

7(a) A quantity of a perfect gas is compressed at constant temperature from initial pressure p_1 to final pressure p_2. Show that the area under the p–V curve for this process can be written as:

$$RT \ln\left(\frac{p_1}{p_2}\right)$$

Note: Area under pV curve is given by $\int pdV$.

(b) An isothermal compression from 1 bar to 8 bar takes place on a perfect gas. The initial volume of the gas is 0·25 m^3. Calculate the heat energy transfer during the process.

8. The compressor of an open cycle gas turbine unit receives air at 1 bar 18°C and delivers it at 4 bar 200°C to the combustion chamber where the temperature is raised at constant pressure to 650°C. The products of combustion pass through the turbine, which has an isentropic efficiency of 0·85, and exhaust at 1 bar. The power required by the compressor is provided by the turbine. Calculate:

(a) the isentropic efficiency of the compressor;
(b) the net work output of the turbine per kilogram of air.

Note: For air $c_P = 1{\cdot}005$ kJ/kg K, $\gamma = 1{\cdot}4$
For combustion gases $c_P = 1{\cdot}15$ kJ/kg K, $\gamma = 1{\cdot}333$

9. In a steam turbine plant steam expands in the high pressure turbine from 50 bar 500°C to 6 bar with an isentropic efficiency of 0·9. The steam is then reheated at constant pressure to 500°C before entering the low pressure turbine. In the low pressure turbine steam is expanded to 0·05 bar with an isentropic efficiency of 0·85. Neglecting feed pump work,

(a) sketch the expansion and reheat processes on a temperature–entropy diagram;
(b) calculate using chart and tables the thermal efficiency of the plant.

10. The walls of a cold room consist of an outer layer of wood of thickness 30 mm and thermal conductivity 0·18 W/m K, and a cork lining of thickness 70 mm and thermal conductivity 0·05 W/m K. If the surface heat transfer coefficient from and to each exposed surface is 10 W/m^2 K and the heat flow through the wall is 24 W/m^2, calculate, (i) the temperature differences across the thicknesses of the wood and cork, (ii) the total temperature difference between the outside atmosphere and inside of room, and (iii) the temperature of the room when the external ambient temperature is 20°C.

11. A surface type feed heater is supplied with steam at 2 bar and 0·95 dry. The temperature of the feed water entering and leaving the heater are 55°C and 105°C respectively. The mass flow rate of feed water through the heater is 20 000 kg/hour and the overall heat transfer coefficient is 4540 W/m^2 K. Calculate, stating any assumptions made:

(a) the mass flow rate of heating steam in kg/hour;
(b) the effective heating surface area of the feed heater.

For the feed water: Specific heat capacity = 4·18 kJ/kg K.
Note: Logarithmic mean temperature difference

$$\theta_m = \frac{\theta_1 - \theta_2}{\ln\left(\dfrac{\theta_1}{\theta_2}\right)}$$

where θ_1 = temperature difference between hot and cold fluid in inlet.
θ_2 = temperature difference between hot and cold fluid at outlet.

12. In a test on a single-cylinder, two-stroke, diesel engine, the mean effective pressure was 8·9 bar, running speed 2·3 rev/s, brake load 8 kN, brake radius 1·25 m, specific fuel consumption 0·251 kg/ kWh (brake). The diameter of the cylinder is 360 mm, stroke 780 mm, calorific value of the fuel 41·5 MJ/kg. Calculate (i) the indicated power, (ii) brake power, (iii) indicated thermal efficiency, and (iv) the total heat energy loss per second.

13. The volumetric analysis of a mixture of gases shows it to contain 80% hydrogen and 20% oxygen. A vessel holds 0·7 m^3 of this mixture at 38°C and 3·5 bar.
Calculate the masses of hydrogen and oxygen in the vessel.
Universal Gas Constant = 8·3143 kJ/mol K.
Atomic mass relationships: hydrogen = 1, oxygen = 16

14. In a single-stage impulse turbine the steam enters the nozzle at 7 bar 300°C and is discharged at 1·2 bar dry saturated, directed at 20° to the plane of rotation. The blade velocity is 40% of the steam jet velocity and the relative velocity of the steam at exit is 80% of the relative velocity at entrance. The outlet angle of the blades is 35° and the steam flow 0·5 kg/s. Calculate (i) the axial thrust, (ii) the power developed.

15. Air expands isentropically through a convergent nozzle from 6 bar 260°C to 4 bar. The velocity of the air at nozzle inlet is 900 m/s and the nozzle cross-sectional area at exit is 0·025 m². Calculate:
(a) the air velocity at nozzle exit;
(b) the mass flow rate of air;
(c) the nozzle cross-sectional area at inlet.

For air: $c_P = 1005$ J/kg K , $\frac{c_P}{c_V} = 1{\cdot}4$

16. The pressure, volume and temperature of a gas mixture sample in a closed vessel is 1·01 bar, 500 cm³ and 20°C, and is composed of 14% carbon dioxide and 86% nitrogen by volume. Taking R for $CO_2 = 0{\cdot}189$ kJ/kg K, calculate (i) the partial pressure of each gas, (ii) the mass of carbon dioxide in the sample.

17. A perfect gas is compressed, polytropically from 1 bar, 22°C, 0·037 m³ to 35 bar, 420°C. Determine:
(a) the index of compression;
(b) the work done;
(c) the change of internal energy;
(d) the heat transfer.

For the gas $c_v = 718$ J/kg K, $R = 282$ J/kg K.

18. In a Freon-12 refrigeration plant the refrigerant leaves the condenser as a liquid at 25°C. The refrigerant leaves the evaporator as dry saturated vapour at −15°C and leaves the compressor at 6·516 bar and 40°C. The cooling load is 73·3 kW. Calculate:
(a) the mass flow rate of refrigerant;
(b) the compressor power;
(c) the coefficient of performance of the plant.

19. In a compression ignition engine working on the ideal dual-combustion cycle, the volumetric compression ratio is 12·5:1.

The cycle consists of (a) adiabatic compression from 1·013 bar, 35°C, (b) heat received at constant volume to a maximum pressure of 40 bar, (c) heat received at constant pressure to a maximum temperature of 1425°C, (d) adiabatic expansion to the initial volume, (e) heat rejected at constant volume. Make a sketch of the *pV* diagram and calculate the mean effective pressure. Take $\gamma = 1{\cdot}4$ for air and products of combustion.

20. A rigid vessel contains a mixture of superheated steam and air at a total pressure of 0·02 bar and at 30°C. The steam/air mass ratio is 20:1. Assume the superheated steam has the properties of a perfect gas. Calculate:
(a) the partial pressures of steam and air;
(b) the density of the mixture

Note: For air $R = 287$ J/kg K. For superheated steam $R = 462$ J/kg K.

21. The mass analysis of a fuel burned in a boiler is 85·5% carbon, 13·5% hydrogen, and 1% oxygen, and the air supply is 25% in excess of the minimum required for stoichiometric combustion. Calculate (i) the percentage mass analysis of the wet flue gases, (ii) the percentage volumetric analysis of the dry flue gases. Take mass composition of air = 23% oxygen, 77% nitrogen. Atomic weights: hydrogen 1, carbon 12, nitrogen 14, oxygen 16.

22. The wall of a cold room is composed of two materials, an inner material of thickness 100 mm, having a thermal conductivity of 0·115 W/m K and a layer of cork of thermal conductivity 0·06 W/m K. The external air temperature is 24°C and the room air temperature is −23°C. The surface heat transfer coefficient of exposed surfaces is 12 W/m² K. The heat transfer through the wall is 30 W/m². Calculate:
(a) the temperature of the exposed surfaces;
(b) the temperature of the interface;
(c) the thickness of the cork.

23. In a refrigerating plant using Freon as the refrigerating agent, the refrigerant leaves the condenser as saturated liquid at 15°C, leaves the evaporator and enters the compressor at 1·509 bar and −5°C, and is delivered from the compressor into the condenser at 4·914 bar 45°C. Calculate the coefficient of performance.

24. The gravimetric analysis of a liquid fuel is: carbon 82%,

hydrogen 18%. Assume stoichiometric combustion. Calculate:
(a) the air/fuel ratio by mass;
(b) the volumetric analysis of the wet products of combustion.
Air contains 23% oxygen by mass.
Atomic mass relationships: oxygen = 16, nitrogen = 14, carbon = 12, hydrogen = 1.

25. Steam expands in a turbine from 25 bar 320°C to 0·04 bar with an isentropic efficiency of 0·73. The power output of the turbine is 3 MW. The system is to be modified by fitting a new boiler, generating steam at 60 bar 370°C, which supplies a new higher pressure turbine exhausting to the original turbine at 25 bar. The high pressure turbine has an isentropic efficiency of 0·76. Between the turbines the steam is reheated to 320°C at constant pressure. Using tables and chart as required calculate:
(a) the enthalpy change of the steam during reheating;
(b) the condition of the steam at condenser inlet;
(c) the percentage reduction in steam flow due to the plant modification when total power output is unchanged.

26. The power absorbed by a single-acting, single-stage reciprocating air compressor is 13·58 kW when the mean piston speed is 2·8 m/s and rotational speed 3·5 rev/s. The air is compressed from 1·0 bar and delivered at 10 bar, the index of the law of compression being 1·32. Neglecting clearance, calculate (i) the stroke of the compressor piston, (ii) the cylinder diameter, and (iii) the mean effective pressure.

27. In an engine working on the ideal diesel cycle, the temperature of the air at the beginning of compression is 37°C, compression takes place according to the law $pV^{1\cdot4}$ = a constant, and the volumetric compression ratio is 13:1. At the end of compression the air receives heat energy at constant pressure and is then expanded to the original volume. If 1 kg of fuel is burned per 35 kg of air compressed, the calorific value of the fuel being 42 MJ/kg, calculate (i) the temperature at the end of compression, (ii) the temperature at the end of heat reception, (iii) the volumetric expansion ratio. Take $c_p = 1{\cdot}02$ kJ/kg K

28. At a certain stage of a reaction turbine, the steam leaves the guide blades at a velocity of 135 m/s, the exit angle being 20°. The linear velocity of the moving blades is 87 m/s. Assuming the channel section of fixed and moving blades to be identical, and assuming ideal conditions, calculate (i) the entrance angle of the moving blades, (ii) the stage power per kg/s steam flow.

29. In an open cycle gas turbine plant, a heat exchanger is included to heat the air before entering the combustion chamber by the exhaust gases from the engine. The gases enter the heat exchanger at 300°C and 140 m/s and leave at 240°C and 10 m/s. The air enters the exchanger at 200°C and the air/fuel ratio is 84. Calculate the temperature of the air at the exchanger exit, taking c_P as 1·1 kJ/kg K for the gases and 1·005 kJ/kg K for air.

30. A CO_2 refrigerating machine produces 250 kg of ice per hour at −10°C from water at 15°C. The refrigerant enters the evaporator 0·2 dry and leaves 0·95 dry. The compressor is single-acting, runs at 4·15 rev/s, and the stroke/bore ratio is 2:1. Calculate (i) the mass flow of the refrigerant through the circuit, and (ii) the diameter and stroke of the compressor piston. Take the following values:

Specific heat of ice = 2·04 kJ/kg K
Latent heat of fusion = 335 kJ/kg
Specific heat of water = 4·2 kJ/kg K
CO_2 vapour at evaporator pressure,
h_{fg} = 290·2 kJ/kg, v_g = 0·02168 m³/kg

31. Air is taken into a single-stage air compressor at 1 bar and delivered at 5 bar. The piston swept volume is 1440 cm³ and the clearance volume is 40 cm³. Taking the index of compression and expansion as 1·3, calculate (i) the fraction of the stroke when the delivery valves open, (ii) the fraction of the stroke when the suction valves open, (iii) the mean indicated pressure.

32. Steam is supplied to a turbine at 30 bar 350°C and the condenser pressure is 0·045 bar. The power developed is 5 MW when the steam consumption is 22·5 Mg/h. Calculate (i) the ideal efficiency of the Rankine cycle, (ii) the actual efficiency of the engine, (iii) the efficiency ratio.

33. A sample of steam at 10 bar is tested by a combined separating and throttling calorimeter, the data obtained were:
Mass of water collected in separator = 0·113 kg
Mass of condensed water after throttling = 3·03 kg
Pressure of steam in throttling calorimeter = 1·2 bar
Temperature of steam in throttling calorimeter = 109·8°C

Take specific heat of superheated steam at calorimetric pressure as 2·02 kJ/kg K and calculate the dryness fraction of the sample.

34. The piston swept volume of an engine working on the ideal dual combustion cycle is 0·1068 m^3 and the clearance volume is 8900 cm^3. At the beginning of compression the pressure is 1 bar and temperature 42°C. The maximum pressure in the cycle is 45 bar and maximum temperature 1500°C. Taking $\gamma = 1{\cdot}4$ and $c_V = 0{\cdot}715$ kJ/kg K for air and products of combustion, calculate the proportion of heat received at constant volume to that received at constant pressure.

35. A fuel has an analysis by mass of 85% C, 11% H_2 and 3% O_2. This fuel is burnt in a combustion chamber with an air/fuel ratio by mass of 11:1. For every kg of fuel burnt calculate:
(a) the mass of carbon burnt to produce carbon monoxide;
(b) the mass of carbon burnt to produce carbon dioxide.
Atomic mass relationships are: C = 12, O = 16, H = 1. Air contains 23% O_2 by mass.

36. At one stage of a steam turbine the inlet velocity of the steam to the rotating blades is 590 m/s and the relative velocity at outlet is 620 m/s. The blade speed is 320 m/s and the inlet and outlet angles are 37° and 26° respectively. Calculate the force on the blades and the power developed at this stage for a steam flow of 0·075 kg/s.

37. A single-acting air compressor takes in air at 1 bar and delivers it at 4 bar, the cylinder is 300 mm diameter, stroke 450 mm, and it runs at 5 rev/s. Initially the index of compression was 1·15 and after running for some time the index of compression was found to be 1·35. Neglecting clearance, calculate the power absorbed in each case and the percentage increase in power.

38. A steam pipe 140 mm outside diameter and 23 m long is lagged with insulating material of thermal conductivity 0·13 W/m K. Steam passes along the pipe at the rate of 1200 kg/h, entering at 18 bar dry saturated and leaving at the same pressure 0·985 dry. The outside surface temperature of the lagging is 35°C and the inside surface may be taken as equal to the steam temperature. Calculate the thickness of the lagging taking the rate of heat transfer through the insulating material, in J/s per unit length of pipe, as:

$$\frac{2\pi k\,(T_1 - T_2)}{\ln\,(r_2/r_1)}$$

39. Steam is supplied to a turbine at 20 bar 400°C and exhausts at 0·04 bar and 0·85 dry. At the stage in the turbine where the pressure is 1·4 bar, 13·4% of the steam is withdrawn and passed to the feed heater and this heats the feed water to the saturation temperature of the tapped off steam. Compare the thermal efficiencies with and without feed heating.

40. Air is compressed in a cylinder according to the law pV^n = a constant. The initial condition of the air is 0·125 m^3, 1·01 bar and 19°C, and the final condition is 36 bar and 508°C. Taking $c_P = 1{\cdot}005$ kJ/kg K and $c_V = 0{\cdot}718$ kJ/kg K, calculate (i) the index of compression, (ii) the mass of air compressed, (iii) the work done during compression, (iv) the change of internal energy, (v) the transfer of heat to or from the air.

41. The following data were taken during a test on a four-cylinder, four-stroke, compression ignition engine of cylinder diameter 320 mm and stroke 480 mm while running at 4 rev/s. Mean effective pressure 14·9 bar, brake load 12 kN on a radius of 960 mm, fuel consumption 99 kg/h, calorific value of fuel 44·5 MJ/kg, mass flow of engine cooling water 154 kg/min, water inlet and outlet temperatures 14°C and 47°C. Calculate the indicated and brake thermal efficiencies, percentage of heat carried away in the cooling water, and draw up a heat balance. Specific heat of cooling water = 4·2 kJ/kg K.

42. In a simple gas turbine working on the ideal cycle, the pressure ratio of both the compressor and turbine is 4·3:1. The temperatures at inlet to the compressor and at inlet to the turbine are 16°C and 600°C respectively. Calculate (i) the temperature at the outlet from the compressor, (ii) the temperature at the outlet from the turbine, (iii) the heat supplied per kilogramme of working fluid, (iv) the thermal efficiency. Take $\gamma = 1{\cdot}4$ and $c_P = 1{\cdot}005$ kJ/kg K.

43. A simple Freon-12 refrigerator operates with evaporator at 1·509 bar –20°C and condenser at 8·477 bar 35°C. The compression between these two states is isentropic and the refrigerant leaving the compressor is dry saturated vapour. Calculate:
(a) using thermodynamic tables:
    (i) the refrigerating effect per kg of refrigerant;
    (ii) the coefficient of performance.
(b) the coefficient of performance of a reversed Carnot cycle operating between the same temperatures.

44. A two pass oil cooler consists of 350 tubes and is required to cool 4 kg/s of oil from 50°C to 20°C. The overall heat transfer coefficient of the tubes is 70 W/m² K, cooling water temperature 15°C. Determine:
(a) Logarithmic mean temperature difference of the oil in the cooler.
(b) Surface area of the tubes.
(c) Length of the thin tubes if their diameter is 19 mm.

$$\text{Mean temperature difference } \theta_m = \frac{T_1 - T_2}{\ln\left(\dfrac{T_1 - T_C}{T_2 - T_C}\right)}$$

where T_1 = oil inlet temperature
T_2 = oil outlet temperature
T_C = cooling water temperature.

The specific heat capacity of oil is 1395·6 J/kg K

45. Dry saturated steam at 7 bar is throttled to 0·5 bar and then passed through two oil heaters in series. The steam leaves the first heater 0·98 dry and is throttled to 0·16 bar before entering the second heater. If there is no pressure drop in the heaters determine using the enthalpy/entropy chart:
(a) final steam condition if there has been no overall change in entropy;
(b) mass flow of steam if 0·72 kg/s of oil, specific heat capacity 2·1 kJ/kg K is raised in temperature through 72°C.

46. A marine boiler installation is fired with methane (CH_4). For stoichiometric combustion calculate:
(a) the correct air to fuel mass ratio;
(b) the percentage composition of the dry flue gases by volume.
Atomic mass relationships: hydrogen 1, oxygen 16, carbon 12, nitrogen 14. Air contains 23% oxygen and 77% nitrogen by mass.

47. Gas enters a rotary compressor at 15°C with a velocity of 75 m/s and a specific enthalpy of 80 kJ/kg. It is discharged at 200°C with a velocity of 175 m/s and a specific enthalpy of 300 kJ/kg. If the compressor loses 10 kJ/kg of gas flowing through the compressor, calculate:
(a) the external work transfer per kilogram of gas;
(b) the datum temperature on which the specific enthalpies given are based.

48. Saturated steam at 50 bar is supplied through a pipe which has two layers of insulation. Outside diameter of the pipe is 200 mm, inner layer of insulation 100 mm thick thermal conductivity 0·05 W/m K, outer layer 50 mm thick thermal conductivity 0·15 W/m K which has a surface heat transfer coefficient of 8 W/m² K. If the ambient temperature is 20°C, calculate the condensation rate per metre of pipe length.

49. Steam at 65 bar and 500°C is supplied to a three stage turbine. It leaves the first stage at 15 bar, 330°C then passes through a reheater, which it leaves at 500°C with specific enthalpy of 3475 kJ/kg. After leaving the second stage at 330°C and 3·5 bar the steam passes through the third stage and exhausts at 0·05 bar, 0·94 dry.
Determine using the enthalpy/entropy chart:
(a) the pressure drop in the reheater;
(b) the isentropic efficiency of each stage;
(c) the ratio of powers developed in each stage.

50. The following successive processes are performed on one kilogram of air from a complete cycle.
(i) Isentropic compression from 120°C and 1 bar to 10 bar.
(ii) Heating at constant volume to 800°C.
(iii) Isentropic expansion.
(iv) Heat rejection at constant pressure.

Sketch the cycle on p–V and T–s axes and determine (a) cycle efficiency, (b) mean effective pressure.

SOLUTIONS TO EXAMINATION QUESTIONS CLASS ONE

1. Refer to Fig. 67

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{925}{T_4} = 5^{\frac{0{\cdot}4}{1{\cdot}4}}$$

$$T_4 = 637{\cdot}8\text{ K}$$

$$\frac{T_3 - T_4^1}{T_3 - T_4} = 0{\cdot}86$$

$$925 - T_4^1 = (925 - 637{\cdot}8) \times 0{\cdot}86$$

$$T_4^1 = 678\text{ K}$$

Temperature of exhaust gases leaving turbine = 403°C Ans. (a)

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_2 = 300 \times 5^{\frac{0{\cdot}4}{1{\cdot}4}}$$

$$= 475\text{ K}$$

$$\frac{T_2 - T_1}{T_2^1 - T_1} = 0{\cdot}82$$

$$475 - 300 = (T_2^1 - 300) \times 0{\cdot}82$$

$$T_2^1 = 513{\cdot}7\text{ K}$$

$$\text{Net work} = \dot{m}_g\, c_{pg}\,(925 - 678) - \dot{m}_a\, c_{pa}\,(513{\cdot}7 - 300)$$

$$5000 = \frac{51}{50}\dot{m}_a \times 1{\cdot}150 \times 247 - \dot{m}_a \times 1{\cdot}005 \times 213{\cdot}7$$

$$100 = \dot{m}_f(289{\cdot}73 - 214{\cdot}8) \quad \text{i.e. } \dot{m}_f = \frac{\dot{m}_a}{50}$$

$$\dot{m}_f = 1{\cdot}334$$

$$\text{Fuel mass} = 1{\cdot}334\text{ kg/s} \quad \text{Ans. (b)}$$

2. From steam tables, for a saturation temperature of 39°C, pressure is 0·07 bar, and spec. volume 20·53 m³/kg. Therefore mass of steam in volume of 15 m³

$$= \frac{15}{20{\cdot}53} = 0{\cdot}7307\text{ kg} \quad \text{Ans. (i)}$$

Partial pressure due to air

$$= \text{total pressure} - \text{steam pressure}$$

$$= 0{\cdot}09 - 0{\cdot}07 = 0{\cdot}02\text{ bar} = 2\text{ kN/m}^2$$

$$pV = mRT$$

$$\text{mass of air } m = \frac{pV}{RT}$$

$$= \frac{2 \times 15}{0{\cdot}287 \times 312} = 0{\cdot}335\text{ kg} \quad \text{Ans. (ii)}$$

Ratio of air to steam

$$= 0{\cdot}335 : 0{\cdot}7307$$

$$= \frac{0{\cdot}335}{0{\cdot}7307} : \frac{0{\cdot}7307}{0{\cdot}7307}$$

$$= 0{\cdot}4584 : 1 \quad \text{Ans. (iii)}$$

3. $h_1 + \frac{1}{2}c_1^2 + q = h_2 + \frac{1}{2}c_2^2 + w$

is the Steady Flow Energy Equation.

$$H_1 + \tfrac{1}{2}mc_1^2 + Q = H_2 + \tfrac{1}{2}mc_2^2 + W$$

$$Q = H_2 - H_1 + \tfrac{1}{2}m\,(c_2^2 - c_1^2) + W$$

$$= mc_p\,(T_2 - T_1) + \tfrac{1}{2}m\,(c_2^2 - c_1^2) + W$$

$$= 0{\cdot}4\,[1{\cdot}005 \times 10^3(40 - 20) + 0{\cdot}5(100^2 - 0^2)] - 13000$$

$$= 0{\cdot}4\,(20100 + 5000) - 13000$$

$$= 10040 - 13000$$

$$Q = -2960\ W$$

Q negative (heat out) and W negative (work on air)

$$Q = mc_p\,(T_o - T_i)$$

$$2960 = 0{\cdot}1 \times 10^3 \times 4{\cdot}18\,(T_o - 20)$$

$$T_o = 27{\cdot}08°\text{C}$$

Cooling water outlet temperature = 27·08°C Ans.

Note: the pressures given are not required

4. Stoichiometric oxygen required per kg of fuel

$$= 2\tfrac{2}{3}C + 8\left\{H_2 - \frac{O_2}{8}\right\}$$

$$= 2\tfrac{2}{3}C + 8H_2 - O_2$$

$$= 2\tfrac{2}{3} \times 0{\cdot}87 + 8 \times 0{\cdot}11 - 0{\cdot}02$$

$$= 3{\cdot}18\text{ kg}$$

Stoichiometric air $= \frac{100}{23} \times 3{\cdot}18 = 13{\cdot}82$ kg

Mass of nitrogen in 13·82 kg of air

$$= 0{\cdot}77 \times 13{\cdot}82 = 10{\cdot}64 \text{ kg}$$

(or, 13·82 kg air − 3·18 kg O_2 = 10·64 kg N_2)

$$CO_2 \text{ formed} = 3\tfrac{2}{3} \times 0{\cdot}87 = 3{\cdot}19 \text{ kg}$$

$$H_2O \text{ formed} = 9 \times 0{\cdot}11 = 0{\cdot}99 \text{ kg}$$

Total gases/kg fuel

$$= 10{\cdot}64 + 3{\cdot}19 + 0{\cdot}99 = 14{\cdot}82 \text{ kg}$$

also, 13·82 kg air + 1 kg fuel = 14·82 kg

At 1·8 tonne of fuel per hour, fuel rate

$$= \frac{1{\cdot}8 \times 10^3}{3600} = 0{\cdot}5 \text{ kg/s}$$

Mass flow rate of each of the gases, Ans. (i):

$$\text{Nitrogen} = 0{\cdot}5 \times 10{\cdot}64 = 5{\cdot}32 \text{ kg/s}$$

$$CO_2 = 0{\cdot}5 \times 3{\cdot}19 = 1{\cdot}595 \text{ kg/s}$$

$$H_2O = 0{\cdot}5 \times 0{\cdot}99 = 0{\cdot}495 \text{ kg/s}$$

As a percentage analysis, Ans. (ii):

$$\text{Nitrogen} = \frac{10{\cdot}64}{14{\cdot}82} \times 100 = 71{\cdot}79\%$$

$$CO_2 = \frac{3{\cdot}19}{14{\cdot}82} \times 100 = 21{\cdot}53\%$$

$$H_2O = \frac{0{\cdot}99}{14{\cdot}82} \times 100 = 6{\cdot}68\%$$

5. From steam tables,

9 bar, $h_g = 2774$

5 bar, $h_f = 640$ $h_{fg} = 2109$ $v_g = 0{\cdot}3748$

0·14 bar, $h_f = 220$ $h_{fg} = 2376$ $v_g = 10{\cdot}69$

Enthalpy drop from 9 bar to 5 bar =

$$2774 - (640 + x \times 2109) = 107$$

$$2774 - 640 - 107 = x \times 2109$$

$$2027 = x \times 2109$$

$$\text{dryness at throat, } x = 0{\cdot}9614$$

Specific volume of steam at throat

$$= 0{\cdot}9614 \times 0{\cdot}3748 = 0{\cdot}3603 \text{ m}^3\text{/kg}$$

$$\text{Velocity [m/s]} = \sqrt{2 \times \text{spec. enthalpy drop [J/kg]}}$$

Velocity through throat

$$= \sqrt{2 \times 107 \times 10^3} = 462{\cdot}6 \text{ m/s}$$

area [m²] × velocity [m/s] = mass flow [kg/s] × spec. vol. [m³/kg]

∴ Area at throat [mm²]

$$= \frac{23 \times 0{\cdot}3603}{60 \times 462{\cdot}6} \times 10^6 = 298{\cdot}5 \text{ mm}^2 \quad \text{Ans. (i)}$$

Effective enthalpy drop from entrance to exit

$$= 0{\cdot}92 \times 633 = 582{\cdot}4 \text{ kJ/kg}$$

Enthalpy drop from 9 bar to 0·14 bar =

$$2774 - (220 + x \times 2376) = 582{\cdot}4$$

$$2774 - 220 - 582{\cdot}4 = x \times 2376$$

$$1971{\cdot}6 = x \times 2376$$

$$\text{dryness at exit, } x = 0{\cdot}8299$$

Specific volume of steam at exit

$$= 0{\cdot}8299 \times 10{\cdot}69 = 8{\cdot}869 \text{ m}^3\text{/kg}$$

Velocity at exit

$$= \sqrt{2 \times 582{\cdot}4 \times 10^3} = 1079 \text{ m/s}$$

Area at exit [mm²]

$$= \frac{\text{mass flow [kg/s]} \times \text{spec. vol. [m}^3\text{/kg]}}{\text{velocity [m/s]}} \times 10^6$$

$$= \frac{23 \times 8{\cdot}869}{60 \times 1079} \times 10^6 = 3150 \text{ mm}^2 \quad \text{Ans. (ii)}$$

6. From Freon-12 tables,

0·8071 bar, $h_f = 4{\cdot}42$ $h_g = 171{\cdot}9$

$\therefore h_{fg} = 171{\cdot}9 - 4{\cdot}42 = 167{\cdot}48$

$s_f = 0{\cdot}0187$ $s_g = 0{\cdot}7219$

$\therefore s_{fg} = 0{\cdot}7219 - 0{\cdot}0187 = 0{\cdot}7032$

12·19 bar, $h_f = 84{\cdot}94$, sat. temp. = 50°C

∴ at 65°C Freon is superheated by 15°

$h = 218{\cdot}64$ $s = 0{\cdot}7166$

Ref. Fig. 69,

Isentropic compression: $s_1 = s_2$

$$0{\cdot}0187 + x_1 \times 0{\cdot}7032 = 0{\cdot}7166$$

$$x_1 \times 0{\cdot}7032 = 0{\cdot}6979$$
$$x_1 = 0{\cdot}9924$$

$$h_1 = h \text{ leaving evaporator} = h \text{ entering compressor}$$
$$= 4{\cdot}42 + 0{\cdot}9924 \times 167{\cdot}48$$
$$= 4{\cdot}42 + 166{\cdot}3 = 170{\cdot}72$$

$$h_4 = h \text{ entering evaporator} = h \text{ leaving condenser } (h_3)$$

$$\text{Refrigerating effect/kg} = h_1 - h_4$$
$$= 170{\cdot}72 - 84{\cdot}94 = 85{\cdot}78 \text{ kJ/kg}$$

Refrigerating effect for flow of 15 kg/min

$$= 15 \times 85{\cdot}78 = 1286 \text{ kJ/min} \quad \text{Ans. (i)}$$

Work transfer in compressor/kg $= h_2 - h_1$

$$= 218{\cdot}64 - 170{\cdot}72 = 47{\cdot}92 \text{ kJ/kg}$$

$$\text{Coeff. of performance} = \frac{\text{refrigerating effect}}{\text{work transfer}}$$

$$= \frac{85{\cdot}78}{47{\cdot}92} = 1{\cdot}79 \quad \text{Ans. (ii)}$$

7. Area = Work done $= \int p\,dV$

Constant temperature (isothermal) $pV = C$

$$p = \frac{C}{V}$$

$$\text{Area} = C\int_{V_1}^{V_2} \frac{dV}{V}$$

$$= C \ln\left(\frac{V_2}{V_1}\right)$$

$$= C \ln\left(\frac{p_1}{p_2}\right) \quad \text{as } p_1V_1 = p_2V_2$$

$$= pV \ln\left(\frac{p_1}{p_2}\right) \quad \text{as } pV = C$$

$$= RT \ln\left(\frac{p_1}{p_2}\right) \quad \text{as } pV = RT$$

$$\therefore \text{ Area under } p\text{–}V \text{ curve} = RT \ln\left(\frac{p_1}{p_2}\right) \quad \text{Ans. (a)}$$

Isothermal process, no change of internal energy

Heat extracted = Work done on the gas

$$\text{Work done} = \int p\,dV$$

$$= p_1V_1 \ln\left(\frac{p_1}{p_2}\right)$$

$$= 1 \times 100 \times 0{\cdot}25 \ln\left(\frac{1}{8}\right)$$

$$= -51{\cdot}99 \text{ kJ}$$

Negative result confirming work done on the gas

Heat energy transfer $= -51{\cdot}99$ kJ Ans. (b)

8. Refer to Figs. 65 and 67:

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_2 = 291 \times 4^{\frac{0{\cdot}333}{1{\cdot}333}}$$

$$= 432{\cdot}4 \text{ K}$$

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{923}{T_4} = 4^{\frac{0{\cdot}4}{1{\cdot}4}}$$

$$T_4 = 652{\cdot}8 \text{ K}$$

$$\eta_c = \frac{T_2 - T_1}{T_2' - T_1} \times 100$$

$$= \frac{432{\cdot}4 - 291}{473 - 291} \times 100$$

Compressor isentropic efficiency $= 77{\cdot}69\%$ Ans. (a)

$$\eta_T = \frac{T_3 - T_4'}{T_3 - T_4}$$

$$0{\cdot}85 = \frac{923 - T_4'}{923 - 652{\cdot}8}$$

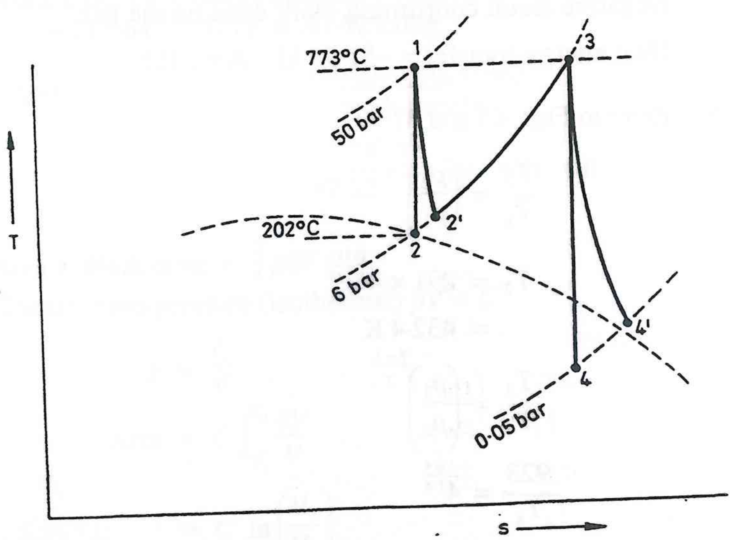
$$T_4' = 693{\cdot}3 \text{ K}$$

$$\begin{aligned} \text{Turbine work} &= c_P\,(T_3 - T_4') \\ &= 1{\cdot}15\,(923 - 693{\cdot}3) \\ &= 264{\cdot}2 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} \text{Compressor work} &= c_P\,(T_2' - T_1) \\ &= 1{\cdot}005\,(473 - 291) \\ &= 182{\cdot}9 \text{ kJ/kg} \end{aligned}$$

$$\text{Net turbine work} = 81{\cdot}3 \text{ kJ/kg} \quad \text{Ans. (b)}$$

9.



See the T–s diagram Ans. (a)

$s_1 = 6{\cdot}975 \quad h_1 = 3433$ (tables or $h - s$ chart)

$s_2 = 6{\cdot}975 \quad h_2 = 2854$ (202°C, just superheated)

$$\begin{aligned} \Delta h &= 3433 - 2854 \\ &= 579 \\ 0{\cdot}9\,\Delta h &= 521 \\ h_2' &= 2912 \quad \text{i.e., } 3433 - 521 \end{aligned}$$

$s_3 = 8{\cdot}001 \quad h_3 = 3483$ (tables or $h - s$ chart)

$s_4 = 8{\cdot}001$ (0·05 bar, steam wet)

$$\begin{aligned} 8{\cdot}001 &= 0{\cdot}476 + 7{\cdot}918\,x \\ x &= 0{\cdot}9504 \text{ (dryness fraction)} \end{aligned}$$

$$\begin{aligned} h_4 &= 138 + 0{\cdot}9504 \times 2423 \\ &= 2441 \quad \text{(or from } h\text{–}s \text{ chart directly)} \end{aligned}$$

$$\begin{aligned} \Delta h &= 3483 - 2441 \\ &= 1042 \\ 0{\cdot}85\,\Delta h &= 885{\cdot}7 \\ h_4' &= 2598{\cdot}7 \text{ i.e., } 3483 - 885{\cdot}7 \end{aligned}$$

$$\begin{aligned} \text{Heat supplied in boiler} &= h_{500} - h_{f\,0{\cdot}05} \\ &= 3433 - 138 = 3295 \\ \text{Heat supplied in reheat} &= h_3 - h_2' \\ &= 3483 - 2912 = 571 \\ \text{Work output} &= 521 + 885{\cdot}7 = 1406{\cdot}7 \\ \text{Thermal efficiency} &= \frac{1406{\cdot}7}{3866} \times 100 = 36{\cdot}39\% \quad \text{Ans. (b)} \end{aligned}$$

Note: The h–s chart is more used; the T–s sketch was specified here.

10. Temperature difference across thickness of wood

$$= \frac{QS_W}{k_W A t}$$

$$= \frac{24 \times 0{\cdot}03}{0{\cdot}18 \times 1 \times 1} = 4 \text{ K} \quad \text{Ans. (i) (a)}$$

Temperature difference across thickness of cork

$$= \frac{QS_C}{k_C A t}$$

$$= \frac{24 \times 0{\cdot}07}{0{\cdot}05 \times 1 \times 1} = 33{\cdot}6 \text{ K} \quad \text{Ans. (i) (b)}$$

Temperature difference between inside and outside atmospheres and their respective exposed surfaces

$$= \frac{Q}{hAt}$$

$$= \frac{24}{10 \times 1 \times 1} = 2{\cdot}4 \text{ K}$$

Total temperature difference between outside atmosphere and inside atmosphere

$= 2{\cdot}4 + 4 + 33{\cdot}6 + 2{\cdot}4 = 42{\cdot}4$ K or 42·4°C Ans. (ii)

Room temperature $= 20 - 42{\cdot}4 = -22{\cdot}4$°C Ans. (iii)

11.
$$\text{Heat lost by steam} = \text{Heat gained by feed water}$$
$$m_s \times x \times h_{fg} = m_w \times c_P \times \text{temp. rise}$$
$$m_s = \frac{20\,000 \times 4{\cdot}18\,(105-55)}{3600 \times 0{\cdot}95 \times 2202}$$
$$= 0{\cdot}555 \text{ kg/s}$$
$$= 1998 \text{ kg/h} \quad \text{Ans. (a)}$$

Assumption: no undercooling of condensed steam

$$\theta_m = \frac{(120{\cdot}2-55) - (120{\cdot}2-105)}{\ln\dfrac{(120{\cdot}2-55)}{(120{\cdot}2-105)}}$$
$$= \frac{50}{1{\cdot}456}$$
$$= 34{\cdot}34^\circ\text{C}$$
$$Q = m_w \times c_P \times \text{temp. rise}$$
$$= \frac{20000 \times 4{\cdot}18 \times 50}{3600}$$
$$= 1161{\cdot}1 \text{ kW}$$
$$Q = UAt\,\theta_m$$
$$1161100 = 4540 \times A \times 1 \times 34{\cdot}34$$
$$A = 7{\cdot}448 \text{ m}^2 \quad \text{Ans. (b)}$$

12.
$$\text{ip} = p_m ALn$$
$$= 8{\cdot}9 \times 10^2 \times 0{\cdot}7854 \times 0{\cdot}36^2 \times 0{\cdot}78 \times 2{\cdot}3$$
$$= 162{\cdot}5 \text{ kW} \quad \text{Ans. (i)}$$
$$\text{bp} = T\omega$$
$$= 8 \times 1{\cdot}25 \times 2\pi \times 2{\cdot}3$$
$$= 144{\cdot}5 \text{ kW} \quad \text{Ans. (ii)}$$

Heat energy supplied per second

$$= \frac{0{\cdot}251 \times 144{\cdot}5 \times 41{\cdot}5 \times 10^3}{3600} = 418{\cdot}1 \text{ kJ/s}$$

Heat energy converted into work in cylinder per second

$$= 162{\cdot}5 \text{ kJ/s}$$

Indicated thermal efficiency

$$= \frac{162{\cdot}5}{418{\cdot}1} = 0{\cdot}3887 \text{ or } 38{\cdot}87\% \quad \text{Ans. (iii)}$$

Total heat energy loss per second

$$= \text{heat supplied} - \text{heat converted into work}$$
$$= 418{\cdot}1 - 162{\cdot}5 = 255{\cdot}6 \text{ kJ/s} \quad \text{Ans. (iv)}$$

13.
$$\text{Hydrogen } R_1 = \frac{8{\cdot}3143}{2} = 4{\cdot}1572$$
$$\text{Oxygen } R_2 = \frac{8{\cdot}3143}{32} = 0{\cdot}2598$$
$$\text{Ratio of partial pressures} = \text{Ratio of volumes}$$
$$\text{Hydrogen, partial pressure} = 0{\cdot}8 \times 3{\cdot}5 = 2{\cdot}8 \text{ bar}$$
$$\text{Oxygen, partial pressure} = 0{\cdot}2 \times 3{\cdot}5 = 0{\cdot}7 \text{ bar}$$
$$pV = mRT$$
$$\text{Mass of hydrogen} = \frac{2{\cdot}8 \times 100 \times 0{\cdot}7}{4{\cdot}1572 \times 311}$$
$$= 0{\cdot}1516 \text{ kg} \quad \text{Ans.}$$
$$\text{Mass of oxygen} = \frac{0{\cdot}7 \times 100 \times 0{\cdot}7}{0{\cdot}2598 \times 311}$$
$$= 0{\cdot}6064 \text{ kg} \quad \text{Ans.}$$

14. From steam tables:

$$1{\cdot}2 \text{ bar}, \; h_g = 2683$$
$$7 \text{ bar } 300^\circ\text{C}, \; h = 3060$$

Enthalpy drop through nozzle

$$= 3060 - 2683 = 377 \text{ kJ/kg}$$

Velocity of steam at nozzle exit [m/s]

$$= \sqrt{2 \times \text{spec. enthalpy drop [J/kg]}}$$
$$= \sqrt{2 \times 377 \times 10^3} = 868{\cdot}4 \text{ m/s}$$

Ref. Fig. 58:

$$v_{w1} = v_1 \cos\alpha_1 = 868{\cdot}4 \times \cos 20^\circ = 816 \text{ m/s}$$
$$u = 40\% \text{ of } v_1 = 0{\cdot}4 \times 868{\cdot}4 = 347{\cdot}4 \text{ m/s}$$
$$x = v_{w1} - u = 816 - 347{\cdot}4 = 468{\cdot}6 \text{ m/s}$$
$$v_{a1} = v_1 \sin\alpha_1 = 868{\cdot}4 \times \sin 20^\circ = 297 \text{ m/s}$$
$$v_{r1} = \sqrt{297^2 + 468{\cdot}6^2} = 554{\cdot}8 \text{ m/s}$$
$$v_{r2} = 0{\cdot}8 \times 554{\cdot}8 = 443{\cdot}8 \text{ m/s}$$
$$v_{a2} = v_{r2} \times \sin\beta_2 = 443{\cdot}8 \times \sin 35^\circ = 254{\cdot}6 \text{ m/s}$$

Axial force on blades [N]

$= $ mass flow [kg/s] $\times$ change of axial velocity [m/s]

$= 0{\cdot}5 \times (297 - 254{\cdot}6)$

$= 21{\cdot}2$ N Ans. (i)

$$v_{w2} = v_{r2}\cos\beta_2 - u$$
$$= 443{\cdot}8 \times \cos 35° - 347{\cdot}4 = 16{\cdot}2 \text{ m/s}$$

Effective change of velocity,

$$v_w = v_{w1} + v_{w2}$$
$$= 816 + 16{\cdot}2 = 832{\cdot}2 \text{ m/s}$$

Tangential force on blades

$$= \dot{m}v_w$$
$$= 0{\cdot}5 \times 832{\cdot}2 = 416{\cdot}1 \text{ N}$$

Power [W] $=$ force [N] $\times$ blade velocity [m/s]

$$= 416{\cdot}1 \times 347{\cdot}4$$
$$= 1{\cdot}445 \times 10^5 \text{ W} = 144{\cdot}5 \text{ kW} \quad \text{Ans. (ii)}$$

15. $$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{533}{T_2} = \left(\frac{6}{4}\right)^{\frac{0{\cdot}4}{1{\cdot}4}}$$

$$T_2 = 474{\cdot}6 \text{ K}$$

$$h_1 - h_2 = c_P(T_1 - T_2) = \tfrac{1}{2}(c_2^2 - c_1^2)$$
$$1005\,(533 - 474{\cdot}6) = \tfrac{1}{2}(c_2^2 - 90^2)$$
$$c_2 = 354{\cdot}2 \text{ m/s}$$

Air velocity at nozzle exit $= 354{\cdot}2$ m/s Ans. (a)

$$pv_s = RT$$
$$4 \times 10^2 \times v_s = 0{\cdot}2871 \times 474{\cdot}6$$
$$v_s = 0{\cdot}3406 \text{ m}^3/\text{kg}$$
$$\dot{m} = \frac{\text{Area} \times \text{velocity}}{\text{Specific volume}}$$
$$= \frac{0{\cdot}025 \times 354{\cdot}2}{0{\cdot}3406}$$

Mass flow rate of air $= 26$ kg/s Ans. (b)

$$pv_s = RT$$

$$4 \times 10^2 \times v_s = 0{\cdot}2871 \times 533$$
$$v_s = 0{\cdot}255$$
$$26 = \frac{\text{Area} \times 90}{0{\cdot}255}$$

Nozzle inlet area $= 0{\cdot}0737 \text{ m}^2$ Ans. (c)

16. The ratio of partial pressures is the same as the ratio of partial volumes, therefore,

$$\left.\begin{aligned}\text{partial pressure of } CO_2 &= 0{\cdot}14 \times 1{\cdot}01 = 0{\cdot}1414 \text{ bar}\\ \text{partial pressure of } N_2 &= 0{\cdot}86 \times 1{\cdot}01 = 0{\cdot}8686 \text{ bar}\end{aligned}\right\} \text{Ans. (i)}$$

For mass of CO_2, $pV = mRT$

$$\text{where } p = 0{\cdot}1414 \times 10^2 \text{ kN/m}^2$$
$$V = 500 \times 10^{-6} \text{ m}^3$$
$$m = \frac{pV}{RT} = \frac{0{\cdot}1414 \times 10^2 \times 500 \times 10^{-6}}{0{\cdot}189 \times 293}$$
$$= 1{\cdot}276 \times 10^{-4} \text{ kg or } 0{\cdot}1276 \text{ g} \quad \text{Ans. (ii)}$$

17. $$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}$$

$$\frac{693}{295} = \left(\frac{35}{1}\right)^{\frac{n-1}{n}}$$

$$n = 1{\cdot}316 \quad \text{Ans. (a)}$$

$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2}$$
$$\frac{1 \times 0{\cdot}037}{295} = \frac{35 \times V_2}{693}$$
$$V_2 = 0{\cdot}02483$$

$$\text{Work done} = \frac{p_1V_1 - p_2V_2}{n-1}$$
$$= \frac{1 \times 100 \times 0{\cdot}037 - 35 \times 100 \times 0{\cdot}02483}{0{\cdot}316}$$
$$= -15{\cdot}794 \text{ kJ} \quad \text{Ans. (b)}$$

Negative (compression) work done on the gas

$$pV = mRT$$

$$1 \times 100 \times 0{\cdot}037 = m \times 282 \times 295$$
$$m = 0{\cdot}0000444 \text{ kg}$$

$$\text{Change of internal energy} = mc_V(T_2 - T_1)$$
$$= 0{\cdot}0000444 \times 718\,(693 - 295)$$
$$= 12{\cdot}69 \text{ kJ} \quad \text{Ans. (c)}$$

$$\text{Heat transfer} = 12{\cdot}69 - 15{\cdot}794$$
$$= -3{\cdot}104 \text{ kJ} \quad \text{Ans. (d)}$$

18. Refer to Fig. 69

$$h_3 = h_4 = 59{\cdot}7 \text{ kJ/kg}$$

i.e. from tables, page 13, at 25°C

$$h_4 = h_f + x_4(h_g - h_f)$$
$$59{\cdot}7 = 22{\cdot}33 + x_4(180{\cdot}97 - 22{\cdot}33)$$

i.e. from tables, page 13, at −15°C

$$x_4 = 0{\cdot}2356$$

$$\text{Heat extracted} = \dot{m}\,(h_1 - h_4)$$
$$= \dot{m}\,\{h_g - h_f - x\,(h_g - h_f)\}$$
$$73{\cdot}3 = \dot{m}\,\{180{\cdot}97 - 23{\cdot}33 - 0{\cdot}2356(180{\cdot}97 - 23{\cdot}33)\}$$
$$= \dot{m}\,(180{\cdot}97 - 59{\cdot}71)$$
$$\dot{m} = 0{\cdot}605 \text{ kg/s}$$

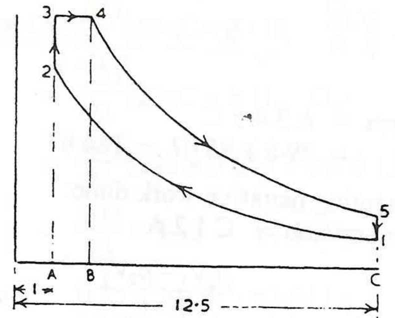
$$\text{Refrigerant mass flow rate} = 0{\cdot}605 \text{ kg/s} \quad \text{Ans. (a)}$$
$$h_2 = 208{\cdot}5 \text{ kJ/kg}$$

i.e. from tables, page 13, at 6·516 bar and 40°C

$$\text{Compressor power} = \dot{m}\,(h_2 - h_1)$$
$$= 0{\cdot}605\,(208{\cdot}5 - 180{\cdot}97)$$
$$= 16{\cdot}66 \text{ kW} \quad \text{Ans. (b)}$$

$$\text{c.o.p.} = \frac{h_1 - h_4}{h_2 - h_1}$$
$$= \frac{180{\cdot}97 - 59{\cdot}71}{208{\cdot}5 - 180{\cdot}97}$$
$$= 4{\cdot}405 \quad \text{Ans. (c)}$$

19.



Representing volumes by ratio:

$$V_1 \text{ and } V_5 = 12{\cdot}5 \qquad V_2 \text{ and } V_3 = 1$$
$$p_1V_1^{\gamma} = p_2V_2^{\gamma}$$
$$1{\cdot}013 \times 12{\cdot}5^{1{\cdot}4} = p_2 \times 1^{1{\cdot}4} \qquad p_2 = 34{\cdot}77 \text{ bar}$$

$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2}$$
$$T_2 = \frac{308 \times 34{\cdot}77 \times 1}{1{\cdot}013 \times 12{\cdot}5} = 845{\cdot}9 \text{ K}$$

$$\frac{T_3}{T_2} = \frac{p_3}{p_2}$$
$$T_3 = \frac{845{\cdot}9 \times 40}{34{\cdot}77} = 973{\cdot}4 \text{ K}$$

$$\frac{V_4}{V_3} = \frac{T_4}{T_3}$$
$$V_4 = \frac{1 \times 1698}{973{\cdot}4} = 1{\cdot}745$$

$$p_4V_4^{\gamma} = p_5V_5^{\gamma}$$
$$40 \times 1{\cdot}745^{1{\cdot}4} = p_5 \times 12{\cdot}5^{1{\cdot}4}$$
$$p_5 = 40 \times \left\{\frac{1{\cdot}745}{12{\cdot}5}\right\}^{1{\cdot}4} = 2{\cdot}541 \text{ bar}$$

Areas representing positive work done:

$$\text{Area during combustion} = \text{A 3 4 B}$$
$$= 40 \times (1{\cdot}745 - 1) = 29{\cdot}8$$
$$\text{Area during expansion} = \text{B 4 5 C}$$

$$= \frac{p_4V_4 - p_5V_5}{\gamma - 1}$$

$$= \frac{40 \times 1{\cdot}745 - 2{\cdot}541 \times 12{\cdot}5}{1{\cdot}4 - 1} = 95{\cdot}07$$

Gross area = A 3 4 5 C
= 29·8 + 95·07 = 124·87

Area representing negative work done:
Area during compression = C 1 2 A

$$= \frac{p_1V_1 - p_2V_2}{\gamma - 1}$$

$$= \frac{1{\cdot}013 \times 12{\cdot}5 - 34{\cdot}77 \times 1}{1{\cdot}4 - 1} = -55{\cdot}27$$

Net area representing useful work
= 124·87 − 55·27 = 69·6

Mean effective pressure = mean height of net area

$$= \frac{\text{area}}{\text{length}} = \frac{69{\cdot}6}{12{\cdot}5 - 1}$$

= 6·052 bar Ans.

Note: m.e.p. and maximum pressure are low by modern standards.

20.
$$p = p_s + p_A$$
$$mR = m_sR_s + m_AR_A$$
$$21 \times R = 20 \times 0{\cdot}462 + 1 \times 0{\cdot}287$$
$$R = 0{\cdot}4537 \text{ kJ/kg K}$$
$$p_s = 0{\cdot}02 \times \frac{20 \times 0{\cdot}462}{21 \times 0{\cdot}4537}$$
$$p_s = 0{\cdot}0194 \text{ bar}$$
$$p_A = 0{\cdot}02 - 0{\cdot}0194 = 0{\cdot}0006 \text{ bar}$$

partial pressure, steam = 0·0194 bar
partial pressure, air = 0·0006 bar } Ans. (a)

$$pv = RT$$
$$100 \times 0{\cdot}02 \times v = 0{\cdot}4537 \times 303$$
$$v = 68{\cdot}735 \text{ m}^3\text{/kg}$$
$$\rho = 0{\cdot}0145 \text{ kg/m}^3$$

Density of mixture = 0·0145 kg/m³ Ans. (b)

21. Working on the basis of 1 kg fuel:

$$\text{Stoichiometric air} = \frac{100}{23}\left\{2\tfrac{2}{3}C + 8\left(H_2 - \frac{O_2}{8}\right)\right\}$$
$$= \frac{100}{23}\{2\tfrac{2}{3}C + 8\,H_2 - O_2\}$$
$$= \frac{100}{23}\{2\tfrac{2}{3} \times 0{\cdot}855 + 8 \times 0{\cdot}135 - 0{\cdot}1\}$$
$$= \frac{100}{23} \times 3{\cdot}35 = 14{\cdot}56 \text{ kg}$$

Excess air = 0·25 × 14·56 = 3·64 kg
Actual air = 14·56 + 3·64 = 18·2 kg

Mass products of combustion per kg fuel:

CO_2 = $3\tfrac{2}{3} \times 0{\cdot}855$ = 3·135 kg
H_2O = 9 H_2 = 9 × 0·135 = 1·215 kg
O_2 = 23% of excess air = 0·23 × 3·64 = 0·8372 kg
N_2 = 77% of all air = 0·77 × 18·2 = 14·014 kg

Total products = 1 kg fuel + 18·2 kg air = 19·2 kg
%mass analysis, Ans. (i):

$$CO_2 = \frac{3{\cdot}135}{19{\cdot}2} \times 100 = 16{\cdot}32\%$$
$$H_2O = \frac{1{\cdot}215}{19{\cdot}2} \times 100 = 6{\cdot}33\%$$
$$O_2 = \frac{0{\cdot}8372}{19{\cdot}2} \times 100 = 4{\cdot}36\%$$
$$N_2 = \frac{14{\cdot}014}{19{\cdot}2} \times 100 = 72{\cdot}99\%$$

Dry flue gases = total gases − H_2O
= 19·2 − 1·215 = 17·985 kg

∴ %Volumetric analysis of dry flue gases, Ans. (ii):

DFG	*m*%	M	*N*	*N*%
CO_2	16·32	44	0·371	11·91%
O_2	4·36	32	0·1362	4·38%
N_2	72·99	28	2·606	83·71%
		Total = 3·1132		

22. Heat passing across inner surface:

$$Q = h_i At\,(T_i - T_1) \;\therefore\; T_i - T_1 = \frac{Q}{h_i At}$$

$$-23 - T_1 = \frac{-30}{12 \times 1 \times 1}$$

Temperature, inner surface (T_1) = $-20{\cdot}5$°C Ans. (a)
i.e. heat flow *into* room is $-Q$

Heat passing across outer surface:

$$Q = h_o At\,(T_3 - T_o) \;\therefore\; T_3 - T_o = \frac{Q}{h_o At}$$

$$T_3 - 24 = \frac{-30}{12 \times 1 \times 1}$$

Temperature, outer surface (T_3) = $21{\cdot}5$°C Ans. (a)

Heat conducted through inner material:

$$Q = \frac{kAt\,(T_1 - T_2)}{S}$$

$$T_1 - T_2 = \frac{QS}{kAt}$$

$$-20{\cdot}5 - T_2 = \frac{-30 \times 0{\cdot}1}{0{\cdot}115 \times 1 \times 1}$$

$$-20{\cdot}5 - T_2 = -26{\cdot}09$$

$$T_2 = 5{\cdot}59\text{°C}$$

Temperature of the interface (T_2) = $5{\cdot}59$°C Ans. (b)

Heat conducted throught cork:

$$T_2 - T_3 = \frac{QS}{kAt}$$

$$5{\cdot}59 - 21{\cdot}5 = \frac{-30 \times S}{0{\cdot}06 \times 1 \times 1}$$

$$S = \frac{15{\cdot}91 \times 0{\cdot}06}{30}$$

$$S = 0{\cdot}0318\text{ m}$$

Thickness of the cork = $31{\cdot}8$ mm Ans. (c)

23. Reference to Freon-12 tables, and Fig. 69,

Compressor suction and evaporator exit:
$1{\cdot}509$ bar, sat. temp. = -20°C,
$\therefore$ at -5°C refrigerant is superheated by 15°
$h_1 = h$ at $1{\cdot}509$ bar, supht. 15° = $187{\cdot}75$
Compressor discharge:
$4{\cdot}914$ bar, sat. temp. = 15°C
$\therefore$ at 45°C refrigerant is superheated by 30°
$h_2 = h$ at $4{\cdot}914$ bar, supht. 30° = $214{\cdot}35$
Condenser outlet: $h_3 = h_f$ at 15°C = $50{\cdot}1$
Evaporator inlet: $h_4 = h_3 = 50{\cdot}1$

Coeff. of performance =

$$\frac{\text{refrigerating effect in evaporator [kJ/kg]}}{\text{work transfer in compressor [kJ/kg]}}$$

$$= \frac{h_1 - h_4}{h_2 - h_1}$$

$$= \frac{187{\cdot}75 - 50{\cdot}1}{214{\cdot}35 - 187{\cdot}75} = 5{\cdot}175 \quad \text{Ans.}$$

24. Stoichiometric air $= \dfrac{100}{23}\,(2\tfrac{2}{3}\text{C} + 8\text{H})$

$$= 4{\cdot}348(2{\cdot}667 \times 0{\cdot}82 + 8 \times 0{\cdot}18)$$

$$= 15{\cdot}77 \text{ kg/kg fuel}$$

Air fuel ratio by mass = $15{\cdot}77$ Ans. (a)

Mass products of combustion per kg fuel:

$$CO_2 = 3\tfrac{2}{3} \times 0{\cdot}8 = 3{\cdot}007 \text{ kg}$$
$$H_2O = 9 \times 0{\cdot}18 = 1{\cdot}62$$
$$N_2 = 0{\cdot}77 \times 15{\cdot}77 = 12{\cdot}143$$
$$\text{Total} = 16{\cdot}77 \text{ kg}$$

% mass analysis of the wet flue gases:

$$CO_2 = \frac{3{\cdot}007}{16{\cdot}77} \times 100 = 17{\cdot}93$$

$$H_2O = \frac{1{\cdot}62}{16{\cdot}77} \times 100 = 9{\cdot}66 \quad \text{Ans. (a)}$$

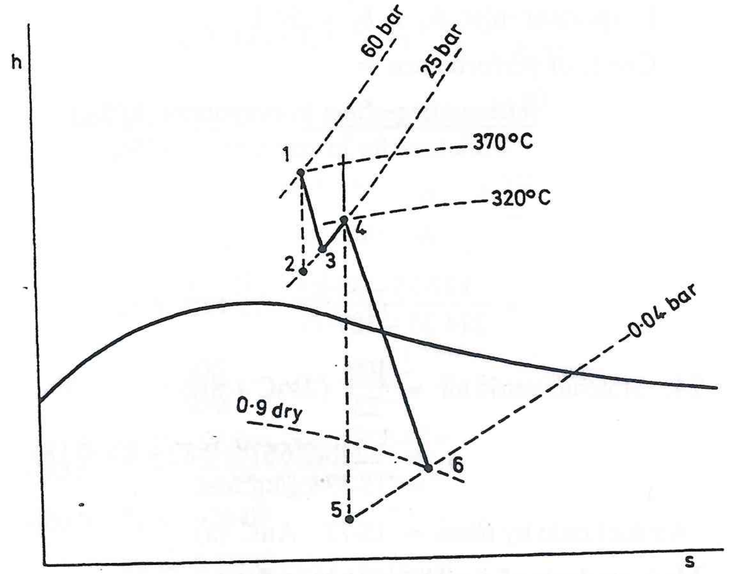
$$N_2 = \frac{12{\cdot}14}{16{\cdot}77} \times 100 = 72{\cdot}39$$

% volume analysis of the wet flue gases:

DFG	$m\%$	M	N	$N\%$
CO_2	17·93	44	0·4075	11·54
H_2O	9·66	18	0·5367	15·20
N_2	72·39	28	2·5854	73·26
			3·5296	

Ans. (b)

25.



From the h–s chart; in kJ/kg

$$\begin{aligned} h_4 &= 3057 \\ h_5 &= 2030 \\ h_4 - h_5 &= 1027 \\ h_4 - h_6 &= 1027 \times 0{\cdot}73 \\ &= 750 \\ h_6 &= 2307 \\ P &= \dot{m}_A(h_4 - h_6) \\ 3 \times 10^3 &= \dot{m}_A \times 750 \\ \dot{m}_A &= 4 \text{ kg/s} \\ h_1 &= 3097 \end{aligned}$$

$$\begin{aligned} h_2 &= 2900 \\ h_1 - h_2 &= 197 \\ h_1 - h_3 &= 197 \times 0{\cdot}76 \\ &= 150 \\ h_3 &= 2947 \\ h_4 - h_3 &= 3057 - 2947 \\ &= 110 \end{aligned}$$

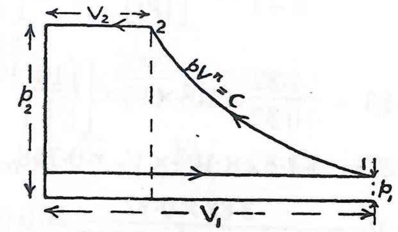
Enthalpy change during reheating is 110 kJ/kg Ans. (a)
Condition of steam at condenser inlet (from chart) 0·9 dry Ans. (b)

$$\begin{aligned} P &= \dot{m}_B\{(h_1 - h_3) + (h_4 - h_6)\} \\ 3 \times 10^3 &= \dot{m}_B\,(150 + 750) \\ \dot{m}_B &= 3{\cdot}33 \text{ kg/s} \end{aligned}$$

$$\frac{100\,(4 - 3{\cdot}33)}{4} = 16{\cdot}8$$

Percentage reduction in steam flow is 16·8 Ans. (c)

26.



$$\begin{aligned} \text{mean piston speed [m/s]} &= \text{distance [m] moved by piston/second} \\ &= 2 \times \text{stroke} \times \text{rev/s} \\ \therefore\ \text{stroke} &= \frac{2{\cdot}8}{2 \times 3{\cdot}5} \\ &= 0{\cdot}4 \text{ m} = 400 \text{ mm} \quad \text{Ans. (i)} \end{aligned}$$

$$\begin{aligned} \text{work per cycle [kJ]} &= \frac{\text{work per second [kJ/s = kW]}}{\text{cycles per second}} \\ &= \frac{13{\cdot}38}{3{\cdot}5} = 3{\cdot}88 \text{ kJ} \end{aligned}$$

Referring to sketch, neglecting clearance:

$$\text{Area under compression curve} = \frac{p_1V_1 - p_2V_2}{n - 1}$$

this is the work done *by* the air,

$$\text{work done } on \text{ the air} = \frac{p_2V_2 - p_1V_1}{n-1}$$

work done on the air per cycle

$$= \text{net area of diagram}$$

$$= \text{compression area} + \text{delivery area} - \text{suction area}$$

$$\frac{p_2V_2 - p_1V_1}{n-1} + p_2V_2 - p_1V_1$$

$$\frac{n}{n-1}(p_2V_2 - p_1V_1)$$

$$= \frac{n}{n-1} mR\,(T_2 - T_1)$$

$$= \frac{n}{n-1} mR\,T_1\left[\frac{T_2}{T_1} - 1\right]$$

$$= \frac{n}{n-1} p_1V_1\left[\left\{\frac{p_2}{p_1}\right\}^{\frac{n-1}{n}} - 1\right]$$

$$\therefore\ 3{\cdot}88 = \frac{1{\cdot}32}{0{\cdot}32} \times 10^2 \times V_1 \times \left[\left\{\frac{10}{1}\right\}^{\frac{0{\cdot}32}{1{\cdot}32}} - 1\right]$$

$$3{\cdot}88 \times 0{\cdot}32 = 1{\cdot}32 \times 10^2 \times V_1 \times 0{\cdot}748$$

$$V_1 = \frac{3{\cdot}88 \times 0{\cdot}32}{1{\cdot}32 \times 10^2 \times 0{\cdot}748} = 0{\cdot}01257\ \text{m}^3$$

$$\text{Diameter} = \sqrt{\frac{\text{volume}}{0{\cdot}7854 \times \text{stroke}}} = \sqrt{\frac{0{\cdot}01257}{0{\cdot}7854 \times 0{\cdot}4}}$$

$$= 0{\cdot}2\ \text{m} = 200\ \text{mm} \quad \text{Ans. (ii)}$$

$$\text{Mean eff. press.} = \frac{\text{net area of diagram}}{\text{length of diagram}}$$

$$= \frac{3{\cdot}88}{0{\cdot}01257} = 308{\cdot}6\ \text{kN/m}^2$$

$$= 3{\cdot}086\ \text{bar} \quad \text{Ans. (iii)}$$

27. Referring to Fig. 28:

$$\frac{T_2}{T_1} = \left\{\frac{V_1}{V_2}\right\}^{n-1}$$

$$T_2 = 310 \times 13^{0{\cdot}4} = 865\ \text{K}$$

$$\text{Temp. at end of compression} = 592^\circ\text{C} \quad \text{Ans. (i)}$$

Assume 35 kg of air is compressed and 1 kg of fuel is burned, mass of gases formed = 35 + 1 = 36 kg

$$\begin{array}{c}\text{Heat energy given}\\ \text{up by 1 kg fuel}\end{array} = \begin{array}{c}\text{Heat energy received}\\ \text{by 36 kg gases}\end{array}$$

$$\text{c.v.} = m \times c_p \times (T_3 - T_2)$$

$$T_3 - T_2 = \frac{42 \times 10^3}{36 \times 1{\cdot}02} = 1144\ \text{K}$$

$$T_3 = 1144 + 865 = 2009\ \text{K}$$

$$\text{Temp. at end of combustion} = 1736^\circ\text{C} \quad \text{Ans. (ii)}$$

$$\frac{V_3}{V_2} = \frac{T_3}{T_2}$$

$$V_3 = 1 \times \frac{2009}{865} = 2{\cdot}322$$

$$\text{Ratio of expansion} = \frac{V_4}{V_3} = \frac{13}{2{\cdot}322} = 5{\cdot}598 \quad \text{Ans. (iii)}$$

28. Referring to Fig 60:

$$\begin{aligned} v_{a1} &= v_1 \sin\alpha_1 = 135 \times 135 \times \sin 20^\circ &&= 46{\cdot}17\ \text{m/s}\\ v_{w1} &= v_1 \cos\alpha_1 = 135 \times \cos 20^\circ &&= 126{\cdot}9\ \text{m/s}\\ x &= v_{w1} - u = 126{\cdot}9 - 87 &&= 39{\cdot}9\ \text{m/s}\end{aligned}$$

$$\tan\beta_1 = \frac{46{\cdot}17}{39{\cdot}9} = 1{\cdot}157$$

$$\therefore\ \text{Entrance angle } \beta_1 = 49^\circ\ 10' \quad \text{Ans. (i)}$$

$$\beta_2 = \alpha_1 \qquad v_{r2} = v_1 \qquad v_{w2} = x$$

$$\text{Effective change of velocity, } v_w = v_{w1} + v_{w2}$$

$$= 126{\cdot}9 + 39{\cdot}9 = 166{\cdot}8\ \text{m/s}$$

Force on blades = change of momentum per second
Force [N] = mass flow [kg/s] × v_w [m/s]
For 1 kg/s, force = $1 \times 166{\cdot}8 = 166{\cdot}8$ N
Power [W = J/s = N m/s] = force [N] × blade velocity [m/s]
$= 166{\cdot}8 \times 87$
$= 1{\cdot}451 \times 10^4$ W
$= 14{\cdot}51$ kW Ans. (ii)

29. Kinetic energy $= \frac{1}{2}mv^2$

Change in kinetic energy per kg of exhaust gases through exchanger due to change of velocity

$= \frac{1}{2}(v_1^2 - v_2^2) = \frac{1}{2}(140^2 - 10^2) = 9750$ J/kg

which is converted into heat energy $= 9{\cdot}75$ kJ/kg

Mass of exhaust gases per kg of fuel
= 84 kg air + 1 kg fuel = 85 kg gases

Heat energy transferred = mass × c_P × temp. change

∴ Transfer of heat in exchanger, from 85 kg of gases, to 84 kg of air:

$85 \times 1{\cdot}1 \times (300 - 240) + 85 \times 9{\cdot}75 = 84 \times 1{\cdot}005 \times$ temp. change

$$\text{Air temp. rise} = \frac{85\,(1{\cdot}1 \times 60 + 9{\cdot}75)}{84 \times 1{\cdot}005} = 76{\cdot}28°$$

Air temp. outlet $= 200 + 76{\cdot}28 = 276{\cdot}28$°C Ans.

30. Heat to be taken from water to make ice [kJ/s]

$$= \frac{250}{3600}(4{\cdot}2 \times 15 + 335 + 2{\cdot}04 \times 10)$$

$= 29{\cdot}05$ kJ/s

Let $\dot{m}$ [kg/s] = mass flow of refrigerant

Heat absorbed by refrigerant in evaporator [kJ/s]
$= \dot{m} \times (0{\cdot}95 - 0{\cdot}2) \times 290{\cdot}2$
$= \dot{m} \times 217{\cdot}7$ kJ/s

Assuming perfect heat transfer,
$\dot{m} \times 217{\cdot}7 = 29{\cdot}05$
$\dot{m} = 0{\cdot}1335$ kg/s Ans. (i)

Volume flow of refrigerant leaving evaporator and entering compressor [m³/s] $= 0{\cdot}1335 \times 0{\cdot}02168 \times 0{\cdot}95$
$= 2{\cdot}749 \times 10^{-3}$ m³/s

Let d = diameter, stroke = $2d$, assuming 100% volumetric efficiency, volume taken into compressor per second [m³/s] = $0{\cdot}7854 \times d^2 \times 4{\cdot}15 = 2{\cdot}749 \times 10^{-3}$

$$d = \sqrt[3]{\frac{2{\cdot}749 \times 10^{-3}}{0{\cdot}7854 \times 2 \times 4{\cdot}15}}$$

$= 0{\cdot}07499$ m say 75 mm

Stroke $= 2 \times 75 = 150$ mm Ans. (ii)

31. Referring to Fig. 38:

V_1 = stroke volume + clearance volume
$= 1440 + 40 = 1480$ cm³
V_3 = clearance volume = 40 cm³
$p_1V_1^n = p_2V_2^n$
$1 \times 1480^{1{\cdot}3} = 5 \times V_2^{1{\cdot}3}$

$$V_2 = \frac{1480}{\sqrt[1{\cdot}3]{5}} = 429{\cdot}1 \text{ cm}^3$$

Volume swept by piston from beginning of compression stroke to point where delivery valves open
$= V_1 - V_2$
$= 1480 - 429{\cdot}1 = 1050{\cdot}9$ cm³

As a fraction of the stroke when delivery valves open

$$= \frac{1050{\cdot}9}{1440} = 0{\cdot}7298 \text{ Ans. (i)}$$

$p_3V_3^n = p_4V_4^n$
$5 \times 40^{1{\cdot}3} = 1 \times V_4^{1{\cdot}3}$
$V_4 = 40 \times \sqrt[1{\cdot}3]{5} = 138$ cm³

Volume swept by piston from beginning of suction stroke to point when suction valves open
$= V_4 - V_3$
$= 138 - 40 = 98$ cm³

As a fraction of the stroke when suction valves open

$$= \frac{98}{1440} = 0{\cdot}06805 \text{ Ans. (ii)}$$

$$\text{Area under compression curve} = \frac{p_2V_2 - p_1V_1}{n - 1}$$

By using $p_2V_2 - p_1V_1$ instead of $p_1V_1 - p_2V_2$ the area represents work done *on* the air instead of work done *by* the air.

Note that the mean indicated pressure will be obtained from net area ÷ length, therefore, since the actual work is not required, it is not necessary to convert pressures into kN/m^2 and volumes into m^3.

$$\text{Area under compression curve} = \frac{p_2V_2 - p_1V_1}{n-1}$$

$$= \frac{5 \times 429{\cdot}1 - 1 \times 1480}{1{\cdot}3 - 1} = 2218{\cdot}3$$

$$\text{Area under delivery line} = p_2 \times (V_2 - V_3)$$
$$= 5 \times (429{\cdot}1 - 40) = 1945{\cdot}5$$

$$\text{Area under expansion curve} = \frac{p_3V_3 - p_4V_4}{n-1}$$

$$= \frac{5 \times 40 - 1 \times 138}{1{\cdot}3 - 1} = 206{\cdot}7$$

$$\text{Area under suction line} = p_1 \times (V_1 - V_4)$$
$$= 1 \times (1480 - 138) = 1342$$

$$\text{Net area of diagram} = 2218{\cdot}3 + 1945{\cdot}5 - 206{\cdot}7 - 1342$$
$$= 4163{\cdot}8 - 1548{\cdot}7 = 2615{\cdot}1$$

$$\text{Mean indicated press.} = \text{mean height of diagram}$$
$$= \frac{\text{net area}}{\text{length}} = \frac{2615{\cdot}1}{1440} = 1{\cdot}816 \text{ bar} \quad \text{Ans. (iii)}$$

Alternatively, the net area of the diagram, representing work per cycle could be obtained from:

$$\frac{n}{n-1} p_1 (V_1 - V_4)\left[\left\{\frac{p_2}{p_1}\right\}^{\frac{n-1}{n}} - 1\right]$$

32. From steam tables,

$$30 \text{ bar } 350^\circ\text{C}, \quad h = 3117 \quad s = 6{\cdot}744$$
$$0{\cdot}045 \text{ bar}, \quad h_f = 130 \quad s_f = 0{\cdot}451$$
$$h_{fg} = 2428 \quad s_{fg} = 7{\cdot}980$$

For the Rankine cycle, isentropic expansion from 30 bar 350°C to 0·045 bar:

Entropy after expansion = Entropy before

$$0{\cdot}451 + x \times 7{\cdot}98 = 6{\cdot}744$$
$$x = 0{\cdot}7886$$
$$h_2 = h \text{ at } 0{\cdot}045 \text{ bar}, 0{\cdot}7886 \text{ dry}$$
$$= 130 + 0{\cdot}7886 \times 2428 = 2044$$

$$\text{Rankine efficiency} = \frac{h_1 - h_2}{h_1 - h_{f2}} = \frac{3117 - 2044}{3117 - 130}$$

$$= \frac{1073}{2987} = 0{\cdot}3592 \text{ or } 35{\cdot}92\% \quad \text{Ans. (i)}$$

$$\text{Actual efficiency} = \frac{\text{heat energy converted into work}}{\text{heat energy supplied by steam}}$$

Heat energy into work [kJ/s = kW] = 5×10^3 kJ/s

Heat energy supplied [kJ/s]

$$= \text{steam consumption [kg/s]} \times (h_1 - h_{f2}) \text{ [kJ/kg]}$$

$$= \frac{22{\cdot}5 \times 10^3}{3600} \times 2987$$

$$\therefore \text{ Engine effic.} = \frac{5 \times 10^3 \times 3600}{22{\cdot}5 \times 10^3 \times 2987}$$

$$= 0{\cdot}2679 \text{ or } 26{\cdot}79\% \quad \text{Ans. (ii)}$$

$$\text{Efficiency ratio} = \frac{0{\cdot}2679}{0{\cdot}3592}$$

$$= 0{\cdot}7457 \text{ or } 74{\cdot}57\% \quad \text{Ans. (iii)}$$

33. From steam tables,

$$10 \text{ bar}, \quad h_f = 763 \quad h_{fg} = 2015$$
$$1{\cdot}2 \text{ bar, sat. temp.} = 104{\cdot}8^\circ\text{C} \quad h_g = 2683$$

$\therefore$ at 1·2 bar 109·8°C, steam is superheated 5°

Dryness fraction by separating calorimeter:

$$x_1 = \frac{3{\cdot}03}{3{\cdot}03 + 0{\cdot}113} = 0{\cdot}964$$

Dryness fraction by throttling calorimeter:

$$\text{Enthalpy before throttling} = \text{Enthalpy after}$$
$$763 + x_2 \times 2015 = 2683 + 2{\cdot}02 \times 5$$
$$x_2 \times 2015 = 1930{\cdot}1$$
$$x_2 = 0{\cdot}9579$$

Dryness fraction of sample

$= 0{\cdot}964 \times 0{\cdot}9579 = 0{\cdot}9234$ Ans.

34. Referring to Fig. 29,

$$T_1 = 315 \text{ K}$$
$$T_4 = 1773 \text{ K}$$
$$V_1 = V_5 = \text{stroke volume} + \text{clearance volume} = 0{\cdot}1068 + 0{\cdot}0089 = 0{\cdot}1157 \text{ m}^3$$
$$V_2 = V_3 = \text{clearance volume} = 0{\cdot}0089 \text{ m}^3$$
$$p_1V_1^{\gamma} = p_2V_2^{\gamma}$$
$$1 \times 0{\cdot}1157^{1{\cdot}4} = p_2 \times 0{\cdot}0089^{1{\cdot}4}$$
$$p_2 = \left\{\frac{0{\cdot}1157}{0{\cdot}0089}\right\}^{1{\cdot}4} = 36{\cdot}26 \text{ bar}$$
$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2}$$
$$T_2 = \frac{315 \times 36{\cdot}26 \times 0{\cdot}0089}{1 \times 0{\cdot}1157} = 878{\cdot}8 \text{ K}$$
$$\frac{T_3}{T_2} = \frac{p_3}{p_2}$$
$$T_3 = \frac{878{\cdot}8 \times 45}{36{\cdot}26} = 1090 \text{ K}$$

Heat received at constant volume

$= m \times c_V \times (T_3 - T_2)$

(per kg of air): $= 1 \times 0{\cdot}715 \times (1090 - 878{\cdot}8)$

$= 151$ kJ/kg

Spec. heat at constant press. $c_P = c_V \times \gamma = 0{\cdot}715 \times 1{\cdot}4 = 1{\cdot}001$

Heat received at constant pressure

$= m \times c_P \times (T_4 - T_3)$

$= 1 \times 1{\cdot}001 \times (1773 - 1090)$

$= 683{\cdot}7$ kJ/kg

Ratio: $\dfrac{\text{heat received at const. vol}}{\text{heat received at const. press}}$

$= \dfrac{151}{683{\cdot}7} = 0{\cdot}2209{:}1$ Ans.

35. Per kg of fuel:

Air supplied $= 11$ kg

Gases $= 11 + 0{\cdot}99$ kg (1)

Let x = C to CO_2

then $(0{\cdot}85 - x)$ = C to CO

$$\text{Air supplied} = \frac{100}{23}\{2\tfrac{2}{3}x + 1\tfrac{1}{3}(0{\cdot}85 - x) + 8 \times 0{\cdot}11 - 0{\cdot}03\}$$
$$= 5{\cdot}796x + 8{\cdot}622 \text{ kg}$$
$$\text{Gases} = 5{\cdot}796x + 8{\cdot}622 + 0{\cdot}99 \text{ kg} \quad (2)$$

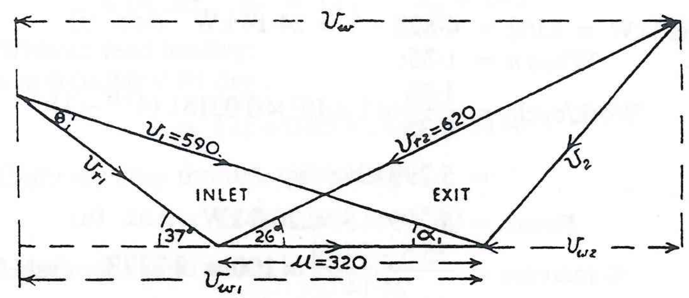
From (1) and (2):

$$11 + 0{\cdot}99 = 5{\cdot}796x + 8{\cdot}622 + 0{\cdot}99$$
$$x = 0{\cdot}41$$
$$(0{\cdot}85 - x) = 0{\cdot}44$$

Carbon to carbon monoxide $= 0{\cdot}44$ kg Ans. (a)

Carbon to carbon dioxide $= 0{\cdot}41$ kg Ans. (b)

36.



By sine rule referring to inlet triangle:

$$\frac{320}{\sin\theta} = \frac{590}{\sin(180° - 37°)}$$
$$\sin\theta = \frac{320 \times 0{\cdot}6018}{590} = 0{\cdot}3263$$
$$\theta = 19°\,3' \quad \alpha_1 = 37° - 19°\,3' = 17°\,57'$$
$$v_{w1} = 590 \times \cos 17°\,57' = 561{\cdot}4 \text{ m/s}$$
$$v_{w2} = 620 \times \cos 26° - 320 = 237{\cdot}3 \text{ m/s}$$

Effective change of velocity $v_w = v_{w1} + v_{w2} = 561{\cdot}4 + 237{\cdot}3 = 798{\cdot}7$ m/s

Force on blades [N] = mass flow [kg/s] × change of velocity [m/s]

$$= 0{\cdot}075 \times 798{\cdot}7 = 59{\cdot}91 \text{ N} \quad \text{Ans. (i)}$$

$$\begin{aligned} \text{Power [W} &= \text{J/s} = \text{N m/s]} = \text{force [N]} \times \text{blade velocity [m/s]} \\ &= 59{\cdot}91 \times 320 \\ &= 1{\cdot}917 \times 10^4 \text{ W} \\ &= 19{\cdot}17 \text{ kW} \quad \text{Ans. (ii)} \end{aligned}$$

37. $V_1 = 0{\cdot}7854 \times 0{\cdot}3^2 \times 0{\cdot}45 = 0{\cdot}03181 \text{ m}^3$

$$\text{Work per cycle} = \frac{n}{n-1} p_1 V_1 \left[\left\{ \frac{p_2}{p_1} \right\}^{\frac{n-1}{n}} - 1 \right]$$

When $n = 1{\cdot}15$:

$$\text{Work/cycle} = \frac{1{\cdot}15}{0{\cdot}15} \times 1 \times 10^2 \times 0{\cdot}03181\,(4^{\frac{0.15}{1.15}} - 1)$$

$$= 4{\cdot}828 \text{ kJ/cycle}$$

At 5 cycles per second:

$$\text{Power [kW} = \text{kJ/s]} = 4{\cdot}828 \times 5 = 24{\cdot}14 \text{ kW} \quad \text{Ans. (i)}$$

When $n = 1{\cdot}35$:

$$\text{Work/cycle} = \frac{1{\cdot}35}{0{\cdot}35} \times 1 \times 10^2 \times 0{\cdot}03181\,(4^{\frac{0.35}{1.35}} - 1)$$

$$= 5{\cdot}299 \text{ kJ/cycle}$$

$$\text{Power} = 5{\cdot}299 \times 5 = 26{\cdot}5 \text{ kW} \quad \text{Ans. (ii)}$$

$$\% \text{ increase} = \frac{26{\cdot}5 - 24{\cdot}14}{24{\cdot}14} \times 100 = 9{\cdot}777\% \quad \text{Ans. (iii)}$$

An alternative solution to the above is to find the value of V_2 in each case, from $p_1 V_1^n = p_2 V_2^n$ then calculate the work per cycle from:

$$\frac{n}{n-1}(p_2 V_2 - p_1 V_1)$$

38. From steam tables,

18 bar, $t_s = 207{\cdot}1°\text{C}$ $h_{fg} = 1912$

Enthalpy drop per kg of steam

$$= (1 - 0{\cdot}985) \times 1912 = 28{\cdot}68 \text{ kJ/kg}$$

Rate of heat loss [J/s] per unit length

$$= \frac{1200 \times 28{\cdot}68 \times 10^3}{3600 \times 25} = 382{\cdot}4 \text{ J/s}$$

and this is to be equal to $\dfrac{2\pi k\,(T_1 - T_2)}{\ln\,(r_2/r_1)}$

$$\frac{2\pi \times 0{\cdot}13 \times (207{\cdot}1 - 35)}{\ln\,(r_2/r_1)} = 382{\cdot}4$$

$$\ln\left(\frac{r_2}{r_1}\right) = \frac{2\pi \times 0{\cdot}13 \times 172{\cdot}1}{382{\cdot}4} = 0{\cdot}3676$$

$$\therefore \frac{r_2}{r_1} = 1{\cdot}444 \qquad r_2 = 1{\cdot}444 \times 70 = 101{\cdot}1 \text{ mm}$$

$$\text{Thickness} = r_2 - r_1 = 101{\cdot}1 - 70 = 31{\cdot}1 \text{ mm} \quad \text{Ans.}$$

39. From steam tables,

20 bar 400°C, $h = 3248$

1·4 bar, $t_s = 109{\cdot}3$ $h_f = 458$ $h_{fg} = 2232$

0·04 bar, $h_f = 121$ $h_{fg} = 2433$

Without feed heating:

h at 0·04 bar 0·85 dry

$$= 121 + 0{\cdot}85 \times 2433 = 2188$$

Enthalpy drop through turbine/kg

$$= 3248 - 2188 = 1060$$

Heat supplied to steam in boiler

$$= 3248 - 121 = 3127$$

$$\text{Thermal eff.} = \frac{\text{heat energy converted into work}}{\text{heat energy supplied}}$$

$$= \frac{1060}{2127} = 0{\cdot}3389 \text{ or } 33{\cdot}89\%$$

With feed heating:

0·134 kg of steam at 1·4 bar is tapped off per kg of supply steam, leaving (1 – 0·134) = 0·866 kg to complete its path through the engine.

Considering heater and hotwell as one system,

$$\text{Enthalpy at entry} = \text{Enthalpy at exit}$$

$$0{\cdot}134\,(458 + x \times 2232) + 0{\cdot}866 \times 121 = 1 \times 458$$
$$61{\cdot}38 + 299x + 104{\cdot}8 = 458$$
$$299x = 291{\cdot}82$$
$$\text{dryness fraction at } 1{\cdot}4 \text{ bar}, x = 0{\cdot}976$$

h at 1·4 bar 0·976 dry

$$= 458 + 0{\cdot}976 \times 2232 = 2636$$

Enthalpy drop through turbine/kg

$$= 1 \times (3248 - 2646) + 0{\cdot}866\,(2636 - 2188)$$
$$= 612 + 388 = 1000$$

Heat supplied to steam in boiler

$$= 3248 - 458 = 2790$$

$$\text{Thermal eff.} = \frac{1000}{2790} = 0{\cdot}3584 \text{ or } 35{\cdot}84\%$$

$$\left\{\begin{array}{ll}\text{Comparison:} & \text{With feed heating, } \eta = 35{\cdot}84\% \\ & \text{Without feed heating, } \eta = 33{\cdot}89\%\end{array}\right\} \text{Ans.}$$

40.
$$T_1 = 292 \text{ K}$$
$$T_2 = 781 \text{ K}$$

$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2}$$

$$V_2 = \frac{1{\cdot}01 \times 0{\cdot}125 \times 781}{36 \times 292} = 0{\cdot}009\,38 \text{ m}^3$$

$$p_1V_1^n = p_2V_2^n$$
$$1{\cdot}01 \times 0{\cdot}125^n = 36 \times 0{\cdot}00938^n$$
$$n = 1{\cdot}38 \quad \text{Ans. (i)}$$

Alternatively, n could be obtained from

$$\frac{T_1}{T_2} = \left\{\frac{p_1}{p_2}\right\}^{\frac{n-1}{n}}$$

$$R = c_P - c_V = 1{\cdot}005 - 0{\cdot}718 = 0{\cdot}287 \text{ kJ/kg K}$$
$$p_1V_1 = mRT_1$$

$$m = \frac{1{\cdot}01 \times 10^2 \times 0{\cdot}125}{0{\cdot}287 \times 292}$$
$$= 0{\cdot}1506 \text{ kg} \quad \text{Ans. (ii)}$$

$$\text{Work done by the air} = \frac{p_1V_1 - p_2V_2}{n-1}$$
$$= \frac{mR\,(T_1 - T_2)}{n-1}$$
$$= \frac{0{\cdot}1506 \times 0{\cdot}287 \times (292 - 781)}{1{\cdot}38 - 1}$$
$$= -55{\cdot}62 \text{ kJ} \quad \text{Ans. (iii)}$$

The minus sign indicates that the work is done *on* the air.
Increase in internal energy

$$U_2 - U_1 = m \times c_V \times (T_2 - T_1)$$
$$= 0{\cdot}1506 \times 0{\cdot}718 \times (781 - 292)$$
$$= 52{\cdot}86 \text{ kJ} \quad \text{Ans. (iv)}$$

$$\text{Heat supplied} = \begin{array}{c}\text{increase in}\\ \text{internal energy}\end{array} + \begin{array}{c}\text{external}\\ \text{work done}\end{array}$$
$$= 52{\cdot}86 + (-55{\cdot}62) = -2{\cdot}76 \text{ kJ} \quad \text{Ans. (v)}$$

The minus sign indicates that the transfer of heat is from the air to its surrounds.

41.
$$\text{ip of 4 cylinders} = p_m ALn \times 4$$
$$\text{where } n = \text{rev/s} \div 2 \text{ for a four-stroke engine}$$
$$\text{ip} = 14{\cdot}9 \times 10^2 \times 0{\cdot}7854 \times 0{\cdot}32^2 \times 0{\cdot}48 \times 2 \times 4$$
$$= 460 \text{ kW}$$
$$\text{bp [kW]} = T\text{ [kN m]} \times \omega\text{[rad/s]}$$
$$= 12 \times 0{\cdot}96 \times 2\pi \times 4 = 289{\cdot}6 \text{ kW}$$

$$\text{Ind. thermal effic.} = \frac{\text{Heat into work in cylinders [kJ/h]}}{\text{Heat energy supplied [kJ/h]}}$$
$$= \frac{460 \times 3600}{99 \times 44{\cdot}5 \times 10^3}$$
$$= 0{\cdot}3758 \text{ or } 37{\cdot}58\% \quad \text{Ans. (i)}$$

$$\text{Brake thermal effic.} = \frac{\text{Heat into work at brake [kJ/h]}}{\text{Heat energy supplied [kJ/h]}}$$
$$= \frac{289{\cdot}6 \times 3600}{99 \times 44{\cdot}5 \times 10^3}$$

$$= 0{\cdot}2367 \text{ or } 23{\cdot}67\% \quad \text{Ans. (ii)}$$

Heat energy carried away by cooling water [kJ/h]

$$= \text{mass} \times \text{spec. ht.} \times \text{temp. rise}$$
$$= 154 \times 60 \times 4{\cdot}2 \times (47 - 14)$$

As a percentage of the heat supplied

$$= \frac{154 \times 60 \times 4{\cdot}2 \times 33}{99 \times 44{\cdot}5 \times 10^3} \times 100$$

$$= 29{\cdot}07\% \quad \text{Ans. (iii)}$$

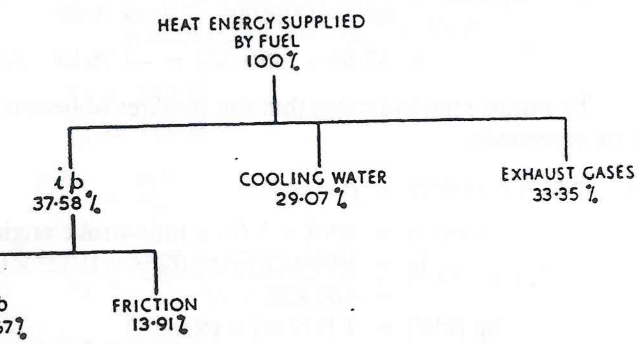
The remainder of the heat losses may be attributed to the heat carried away in the exhaust gases

$$= 100 - (37{\cdot}58 + 29{\cdot}07) = 33{\cdot}35\%$$

Friction and pumping losses

$$= 37{\cdot}58 - 23{\cdot}67 = 13{\cdot}91\%$$

Heat balance diagram:



42. Referring to Fig. 65,

$$\frac{T_2}{T_1} = \left\{\frac{p_2}{p_1}\right\}^{\frac{\gamma-1}{\gamma}} \quad \text{where } \frac{\gamma-1}{\gamma} = \frac{0{\cdot}4}{1{\cdot}4} = \frac{2}{7}$$

$$T_2 = 4{\cdot}3^{\frac{2}{7}} \times 289 = 438{\cdot}4 \text{ K}$$

Temperature at compressor outlet

$$= 165{\cdot}4^\circ\text{C} \quad \text{Ans. (i)}$$

$$\frac{T_4}{T_3} = \frac{T_1}{T_2} \quad \text{because pressure ratios are equal}$$

$$T_4 = \frac{873 \times 289}{438{\cdot}4} = 575{\cdot}4 \text{ K}$$

Alternatively T_4 could be obtained from

$$\frac{T_4}{T_3} = \left\{\frac{p_4}{p_3}\right\}^{\frac{\gamma-1}{\gamma}}$$

Temperature at turbine outlet

$$= 302{\cdot}4^\circ\text{C} \quad \text{Ans. (ii)}$$

Heat supplied per kg of working fluid

$$= \text{mass} \times \text{spec. heat} \times \text{temp. rise}$$
$$= 1 \times 1{\cdot}005 \times (873 - 483{\cdot}4)$$
$$= 436{\cdot}7 \text{ kJ/kg} \quad \text{Ans. (iii)}$$

$$\text{Thermal effic.} = 1 - \frac{T_4 - T_1}{T_3 - T_2} \text{ or } 1 - \frac{T_1}{T_2} \text{ or } 1 - \frac{T_4}{T_3}$$

$$\text{or } 1 - \frac{1}{r_p^{(\gamma-1)/\gamma}}$$

$$1 - \frac{T_1}{T_2} = 1 - \frac{289}{438{\cdot}4} = 1 - 0{\cdot}6592$$

$$= 0{\cdot}3408 \text{ or } 34{\cdot}08\% \quad \text{Ans. (iv)}$$

43. Refer to Fig. 69:
$h_3 = h_4 = 69{\cdot}55$ kJ/kg
i.e. from tables, page 13, at 8·477 bar and 35°C

$$h_4 = h_f + x_4 (h_g - h_f)$$
$$69{\cdot}55 = 17{\cdot}82 + x_4 (178{\cdot}73 - 17{\cdot}82)$$

i.e. from tables, page 13, at 1·509 bar and –20°C

$$x_4 = 0{\cdot}3215$$

$$s_1 = s_2$$

i.e. isentropic compression

$$0{\cdot}0731 + x_1 (0{\cdot}7087 - 0{\cdot}0731) = 0{\cdot}6839$$
$$x_1 = 0{\cdot}961$$

$$h_1 = h_f + x_1(h_g - h_f)$$
$$= 17{\cdot}82 + 0{\cdot}961\,(178{\cdot}73 - 17{\cdot}82)$$
$$= 172{\cdot}45 \text{ kJ/kg}$$

$$\text{Refrigerating effect} = h_1 - h_4$$
$$= 172{\cdot}45 - 69{\cdot}55$$
$$= 102{\cdot}9 \text{ kJ/kg} \quad \text{Ans. (a)(i)}$$

$$\text{c.o.p.} = \frac{h_1 - h_4}{h_2 - h_1}$$
$$= \frac{102{\cdot}9}{201{\cdot}45 - 172{\cdot}45}$$
$$= 3{\cdot}548 \quad \text{Ans. (a)(ii)}$$

$$\text{Reversed Carnot c.o.p.} = \frac{T_L}{T_H - T_L}$$
$$= \frac{253}{308 - 253}$$
$$= 4{\cdot}6 \quad \text{Ans. (b)}$$

44.

$$\theta_m = \frac{50 - 20}{\ln \dfrac{50 - 15}{20 - 15}}$$
$$= 15{\cdot}42\text{°C} \quad \text{Ans. (a)}$$

$$Q = UAt\theta_m \quad \text{also } Q = 4(50 - 20) \times 1395{\cdot}6/10^3$$
$$\therefore\ UA\theta_m = 167{\cdot}5$$
$$A = \frac{167{\cdot}5 \times 10^3}{70 \times 15{\cdot}42}$$
$$= 155 \text{ m}^2 \quad \text{Ans. (b)}$$

$$\pi dLn = A \qquad n = 350 \text{ tubes}$$
$$L = \frac{A}{\pi dn}$$
$$= \frac{155 \times 10^3}{\pi \times 19 \times 350}$$
$$= 7{\cdot}42 \text{ m.} \quad \text{Ans. (c)}$$

45. From h–s chart

$$h_1 = 2760 \text{ kJ/kg}$$
$$h_2 = 2760 \text{ kJ/kg}$$

$$h_3 = 2595 \text{ kJ/kg}$$
$$h_4 = 2595 \text{ kJ/kg}$$
$$h_5 = 2180 \text{ kJ/kg}$$

Final condition of steam = 0·823 dry. Ans.

Enthalpy lost by steam = Enthalpy gained by the oil

$$\dot{m}\,[(h_2 - h_3) + (h_4 - h_5)] = 0{\cdot}72 \times 2{\cdot}1 \times 72$$
$$\dot{m} = 0{\cdot}186 \text{ kg/s} \quad \text{Ans.}$$

46.

$$\text{Mass of 1 mol of fuel} = 12 \times 1 + 1 \times 4$$
$$= 16 \text{ kg}$$
$$H_2 \text{ by mass} = \frac{4}{16}$$
$$= 0{\cdot}25 \text{ kg}$$
$$\text{C by mass} = \frac{12}{16}$$
$$= 0{\cdot}75 \text{ kg}$$
$$\text{Stoichiometric air required} = \frac{100}{23}\,(2\tfrac{2}{3}\text{C} + 8\text{H})$$
$$= 4{\cdot}348\,(2\tfrac{2}{3} \times 0{\cdot}75 + 8 \times 0{\cdot}25)$$
$$= 17{\cdot}39 \text{ kg/kg fuel}$$
$$\text{Correct air-fuel mass ratio} = 17{\cdot}39 \quad \text{Ans. (a)}$$

Mass of dry products of combustion per kg fuel burnt:

$$CO_2 = 3\tfrac{2}{3} \times 0{\cdot}75 = 2{\cdot}75 \text{ kg}$$
$$N_2 = 0{\cdot}77 \times 17{\cdot}39 = 13{\cdot}39$$
$$\text{Total} = 16{\cdot}14 \text{ kg}$$

% mass analysis of the dry flue gases:

$$CO_2 = \frac{275}{16{\cdot}14} = 17{\cdot}04$$
$$N_2 = \frac{1339}{16{\cdot}14} = 82{\cdot}96$$

% volume analysis of the dry flue gases:

DFG	$m\%$	M	N	N%
CO_2	17·04	44	0·3873	11·56
N_2	82·96	28	2·9629	88·44
			3·3502	

Ans. (b)

47. Steady flow energy equation

$$h_1 + \tfrac{1}{2}c_1^2 + q = h_2 + \tfrac{1}{2}c_2^2 + w$$

$$80 + \tfrac{1}{2} \times \frac{75^2}{10^3} - 10 = 300 + \tfrac{1}{2} \times \frac{175^2}{10^3} + w$$

from which $w = -242{\cdot}5$ kJ/kg. Ans. (a)

If t_h = datum temperature on which the specific enthalpies are based

$$\left\{\begin{array}{l}\text{then } h_1 = c_P(t_1 - t_h) \\ \text{and } h_2 = c_P(t_2 - t_h)\end{array}\right\} \text{divide}$$

$$\therefore \quad \frac{h_1}{h_2} = \frac{c_P(t_1 - t_h)}{c_P(t_2 - t_h)}$$

$$\text{i.e.} \quad \frac{80}{300} = \frac{15 - t_h}{200 - t_h}$$

$$\begin{aligned}\text{i.e.}\quad 80(200 - t_h) &= 300\,(15 - t_h) \\ 16\,000 - 80\,t_h &= 4500 - 300\,t_h \\ 16\,000 - 4500 &= 80\,t_h - 300\,t_h \\ \frac{11\,500}{220} &= -t_h \\ \therefore\ t_h &= -52{\cdot}27^\circ\text{C or } 220{\cdot}7\text{ K}\end{aligned}$$

Ans. (b)

48.

For insulation (1) $Q = \dfrac{2\pi k_1(T_1 - T_2)}{\ln\left(\dfrac{D_2}{D_1}\right)}$ W/m

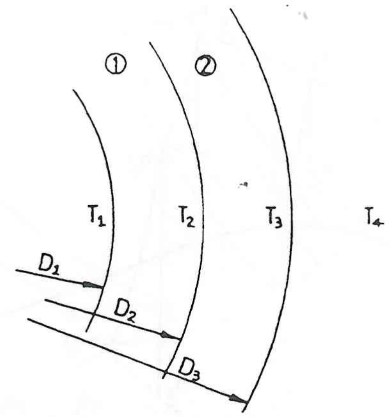
For insulation (2) $Q = \dfrac{2\pi k_2(T_2 - T_3)}{\ln\left(\dfrac{D_3}{D_2}\right)}$ W/m

For surface film $Q = \dfrac{hA\,(T_3 - T_4)}{L}$ W/m

Total temperature drop $T_1 - T_4$

$$= (T_1 - T_2) + (T_2 - T_3) + (T_3 - T_4)$$

$$= \frac{Q\ln(D_2/D_1)}{2\pi k_1} + \frac{Q\ln(D_3/D_2)}{2\pi k_2} + \frac{Q}{h\pi D_3}$$



From steam tables $T_1 = 263{\cdot}9^\circ$C

$$\therefore \quad 263{\cdot}9 - 20 = Q\left[\frac{\ln\left(\frac{400}{200}\right)}{2\pi \times 0{\cdot}5} + \frac{\ln\left(\frac{500}{400}\right)}{2\pi \times 0{\cdot}1} + \frac{10^3}{8\pi \times 500}\right]$$

$$243{\cdot}9 = Q\,[2{\cdot}206 + 0{\cdot}355 + 0{\cdot}0796]$$

$$Q = 92{\cdot}35 \text{ W/m.}$$

Q = mass of steam condensed/metre length of pipe/s × latent heat of steam.

$$Q = \dot{m} \times h_{fg}$$

$$\dot{m} = \frac{92{\cdot}35}{1639}$$

$$= 0{\cdot}0563 \text{ kg/s}$$

Ans.

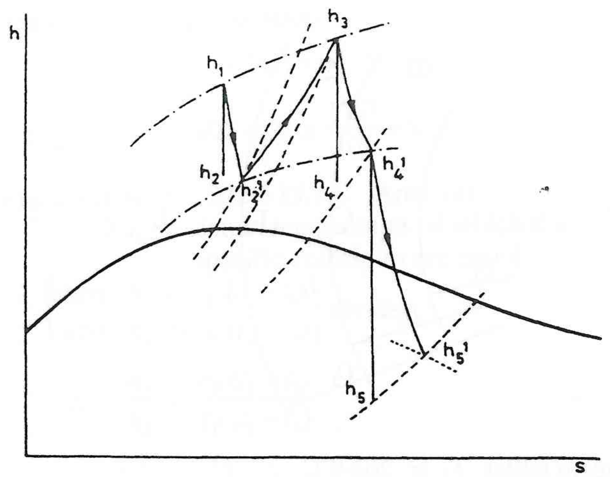
49.

Using the h–s chart

$$\begin{aligned} h &= 3410 \text{ kJ/kg} \\ h_2{}^1 &= 3100 \text{ kJ/kg} \\ h_2 &= 2990 \text{ kJ/kg} \\ h_3 &= 3475 \text{ kJ/kg} \\ h_4{}^1 &= 3130 \text{ kJ/kg} \\ h_4 &= 3085 \text{ kJ/kg} \\ h_5{}^1 &= 2415 \text{ kJ/kg} \\ h_5 &= 2370 \text{ kJ/kg} \end{aligned}$$

Pressure drop in reheater = 15 – 13

= 2 bar. Ans. (a)



Stage 1. Isentropic efficiency $= \dfrac{h_1 - h_2^1}{h_1 - h_2}$

$= 0{\cdot}74$ or 74%

Stage 2. Isentropic efficiency $= \dfrac{h_3 - h_4^1}{h_3 - h_4}$

$= 0{\cdot}88$ or 88%

Stage 3. Isentropic efficiency $= \dfrac{h_4 - h_5^1}{h_4 - h_5}$

$= 0{\cdot}94$ or 94% Ans. (b)

Ratio of powers, $h_1 - h_2^1 : h_3 - h_4^1 : h_4 - h_5^1$

i.e. $310 : 345 : 715$

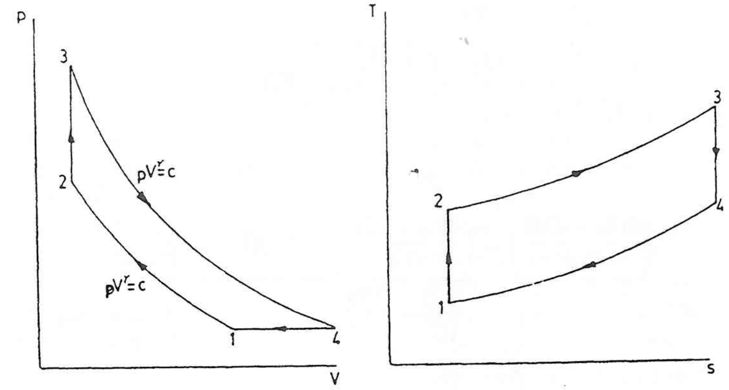
i.e. $1 : 1{\cdot}112 : 2{\cdot}31$ Ans. (c)

50.

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}}$$

2. $T_2 = 393 \times 10^{\frac{0\cdot4}{1\cdot4}} = 759\text{ K} = 486°\text{C}$

3. $T_3 = 800°\text{C}$



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

$$p_3 = 10 \times \frac{1073}{759} = 14{\cdot}14 \text{ bar}$$

4. $$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_4 = 1073\left(\frac{1}{14{\cdot}14}\right)^{\frac{0\cdot4}{1\cdot4}} = 503\text{ K} = 230°\text{C}$$

$$\text{Cycle efficiency} = 1 - \frac{\text{heat rejected}}{\text{heat received}}$$

$$= 1 - \frac{c_P\,(T_4 - T_1)}{c_V\,(T_3 - T_2)}$$

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

$$= 1 - 1{\cdot}4\,\frac{(230 - 120)}{(800 - 486)} \times 100\%$$

$= 50{\cdot}96\%$ Ans. (a)

$$\text{m.e.p.} = \frac{\text{area}}{\text{length}}$$

$$= \frac{\left(\dfrac{p_3V_3 - p_4V_4}{\gamma - 1}\right) - \left(\dfrac{p_2V_2 - p_1V_1}{\gamma - 1}\right) - (p_4V_4 - p_1V_1)}{V_4 - V_2}$$

as $pV = mRT$

$$\text{m.e.p.} = \frac{\dfrac{(T_3 - T_4)}{\gamma - 1} - \dfrac{(T_2 - T_1)}{\gamma - 1} - (T_4 - T_1)}{\dfrac{T_4}{p_4} - \dfrac{T_2}{p_2}}$$

$$= \frac{\left(\dfrac{800 - 230}{0{\cdot}4}\right) - \left(\dfrac{486 - 120}{0{\cdot}4}\right) - (230 - 120)}{\dfrac{503}{10^5} - \dfrac{759}{10^6}}$$

$= 0{\cdot}937$ bar Ans. (b)

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