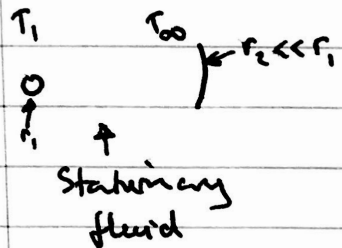
Q1) a) i) $\dot{Q} = \frac{T_1 - T_2}{R_{th}}$ $\quad \dot{Q} = -\lambda_s 4\pi r^2 \frac{dT}{dr} \Rightarrow \int_{r_1}^{r_2} \frac{\dot{Q}}{r^2} dr = -\int_{T_1}^{T_2} 4\pi\lambda \, dT \Rightarrow \dot{Q} = \frac{T_1 - T_2}{\frac{1}{4\pi\lambda}\left[\frac{1}{r_1} - \frac{1}{r_2}\right]}$ //

ii) $R_{overall} = R_{th} + \frac{1}{hA}$. $\frac{\partial R_{overall}}{\partial r_2} = \frac{\partial}{\partial r_2}\left\{\frac{1}{4\pi\lambda_s}\left[\frac{1}{r_1} - \frac{1}{r_2}\right] + \frac{1}{h 4\pi r_2^2}\right\} = 0$

$$\frac{1}{\lambda_s r_2^2} - \frac{2}{h r_2^3} = 0$$

$$\Rightarrow r_2 = 2\lambda_s / h \text{ //}$$

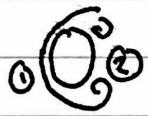
b) 

i) $Nu_d = \frac{h(2r_1)}{\lambda}$ $\quad \dot{Q} = \frac{T_1 - T_\infty}{R_{th}} = h(4\pi r_1^2)(T_1 - T_\infty)$

$$\Rightarrow \frac{1}{h} = 4\pi r_1^2 R_{th} = 4\pi r_1^2 \cdot \frac{1}{4\pi\lambda}\left[\frac{1}{r_1} - \cancel{\frac{1}{r_2}}\right] = r_1/\lambda$$

$$\Rightarrow Nu_d = \left(\frac{\lambda}{r_1}\right) \cdot \frac{2 r_1}{\lambda} = 2 \text{ //}$$

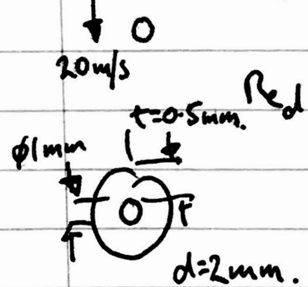
ii)

other terms are  laminar and separated boundary layers.

$$Nu_d = 2 + \left[0.4 Re_d^{1/2} + 0.06 Re_d^{2/3}\right] Pr^{0.4}$$

① laminar. ② turbulent/separated. Prandtl number

c) $Nu_d = \frac{h(d)}{\lambda_{air}} = 2 + \left[0.4 Re_d^{1/2} + 0.06 Re_d^{2/3}\right] Pr^{0.4}$



$$Re_d = \frac{\rho_{air} U_{air} d}{\mu_{air}} = \frac{1.25 \times 20 \times 2\times10^{-3}}{1.8\times10^{-5}} = 2778 \text{ //}$$

$$\Rightarrow Nu_d = 2 + 18.4 + 10.3 = 30.7$$

$$\text{so } h = \frac{Nu_d \times \lambda_{air}}{d} = \frac{30.7 \times 0.026}{2\times10^{-3}} = 400$$

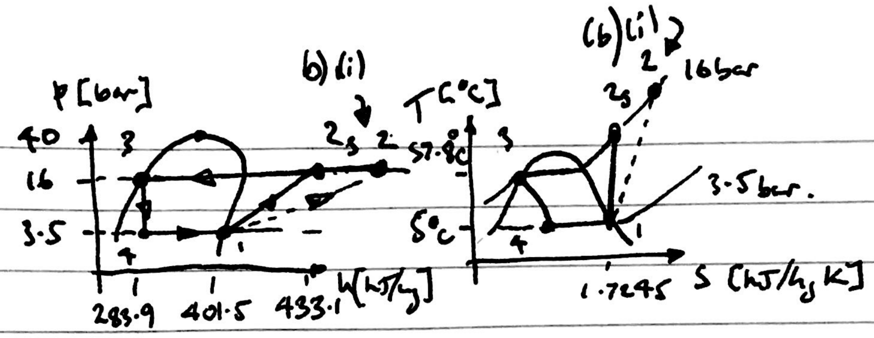
$$Bi = \frac{ht}{\lambda_s} = \frac{400 \times 0.5\times10^{-3}}{0.26} = 0.77 \sim 1 \text{ so unsteady.}$$

$$Fo \sim 1 \Rightarrow \tau = \frac{t^2}{\alpha} = \frac{(0.5\times10^{-3})^2}{1.84\times10^{-7}} = 1.4 \text{ seconds}$$

Question 1. Attempts 62, mean mark 14.9/25, maximum 25, minimum 2.
This question covered conduction and convection from a sphere. It was the least popular question, but in line with the popularity of heat transfer questions in previous years. Part (a)(i) asked the candidates to derive the expression for the thermal resistance of a spherical shell. This was answered well by almost all candidates. The only issues were sign errors in the working. Part (a)(ii) asked the candidates to find the minimum thermal resistance of a spherical shell with convection on the outer surface, this was also done very well by all candidates, with the only issues relating to poor differentiation. Part (b) of the question gave the students a convection correlation for the flow around a sphere at subcritical conditions. The first part asked them to explain the first term, which is not a function of Reynolds number. About 25% of the candidates spotted that it was related to conduction alone, but only a few were able to use the thermal resistance derived in part (a)(i) to justify the numerical value. The next part asked the candidates to explain the other terms. Only about 10% spotted that the two terms related to the two distinct regions of flow past a sphere for the Reynolds number given – attached flow on the front and separated on the rear. Many answers were just regurgitated hopefully from the notes. The final part (c) asked the candidates to evaluate the given correlation and estimate the characteristic time constant of cooling using a Fourier number of ~1. There were many good attempts. The candidates were told to use the thickness of the shell for the characteristic dimension for the Biot number, but many also used it instead of the diameter to evaluate the Nusselt number. Another very common mistake was to use the value of thermal diffusivity given in the question for convective heat transfer coefficient, instead of calculating it from the Nusselt number correlation.

Q2)

a) i)



ii)

① : dry sat @ 3.5 bar $\quad h_1 = 401.5$ kJ/kg (table)

② : 16 bar $s_2 = s_1$, $s_1 = 1.7245$ kJ/kgK (table) $\Rightarrow h_2 = 432$ kJ/kg (chart)

③ : 16 bar wet sat. $\quad h_3 = 282$ kJ/kg (chart).

Comp. $\cancel{\dot{q}} - \dot{w}_x = \Delta h = \overset{h_2 - h_1}{432 - 401.5} = 30.5$ kJ/kg //

Cond. $\dot{q} - \cancel{\dot{w}_x} = \Delta h = h_2 - h_3 = 432 - 282 = 150$ kJ/kg //

$$CoP_{HP} = \frac{\dot{Q}_{out}}{\dot{W}_{x in}} = \frac{\dot{q}_{out}}{\dot{w}_{x in}} = \frac{150}{30.5} \simeq 4.92 //$$

b) $\eta_i = 0.6 = \dfrac{h_{2s} - h_1}{h_2 - h_1} \Rightarrow h_2 - h_1 = (h_{2s} - h_1)/\eta_i + h_1 = \dfrac{432 - 401.5}{0.6} + 401.5 = 452.3$ kJ/kg.

$$\Rightarrow CoP_{HP\,real.} = \frac{h_2 - h_3}{h_2 - h_1} = \frac{452.3 - 282}{452.3 - 401.5} = 3.35 //$$

$$\Delta = \frac{4.92 - 3.35}{4.92} = 31.9\% \text{ drop.}$$

c) i) $\dot{Q}_{out} = 2 \times 10^6$, $\dot{m} = \dfrac{\dot{Q}_{out}}{\dot{q}_{out}} = \dfrac{2 \times 10^6}{(452.3 - 282) \times 10^3} = 11.74$ kg s^{-1}

$$\dot{W}_x = \frac{\dot{Q}_{out}}{CoP_{HP real}} = \frac{2 \times 10^6}{3.35} = 597 \text{ kW} //$$

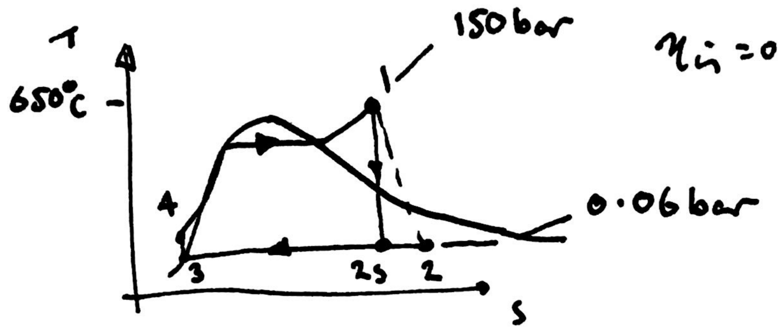
ii) $\dot{Q}_{out\,HP} = \dot{Q}_{in\,GAS\,PS} \times 0.62 \times 0.9 \times 0.98 \times 3.35 \approx$

$\dot{Q}_{out\,BOILER} = \dot{Q}_{in\,GAS} \times 0.9 \quad \Rightarrow \dot{Q}_{in\,GAS}\Big|_{BOILER} = 2 \times \dot{Q}_{in\,GAS}\Big|_{HP}$ //

Question 2. Attempts 271, average mark 15.4/25, maximum 24, minimum 1.
This question, dealing with a heat pump cycle, was very popular and was well done by most of those who attempted it. Most candidates were able to gain the majority of the marks for parts (a) and (b) as routine cycle calculations are involved, showing robust learnings on basic and fundamental parts of the cycles. However, a small number of candidates tried to apply perfect gas relationship to the real gas, leading to errors. Part (c) of the problem is to compare the overall (fuel-to-heat) efficiency of the heat pump to that of a gas fired boiler. Most of candidates were able to calculate the cumulative losses from the power generation to the domestic mains supply using the data provided, but some forgot to multiply this with the COP of the heat pump thus arriving at the wrong conclusion of the heat pump being less energy efficient than a gas fired boiler. A small number even managed to lump the inefficiency due to the combustion of the gas fired boiler into the electric power inefficiency.

Q3) a)

i)



ii) @ 1 $T = 650°C$, $p = 150$ bar. $h_1 = 3712.1$ (table p.23) kJ/kg

$s_1 = 6.8233$ (table p.24) kJ/kg/K.

@ 2s $s_1 = s_2 = 6.8233 \Rightarrow h_2 = 2100$ (chart) kJ/kg

$$\eta_{is} = \frac{h_1 - h_2}{h_1 - h_{2s}} \Rightarrow h_2 = h_1 - \eta_{is}(h_1 - h_{2s})$$

$$= 3712.1 - 0.85(3712 - 2100) = 2341.5 \text{ kJ/kg}$$

$\Rightarrow x_{2s} = 0.805$ (chart)

$x_2 = \cancel{0.895}$ (chart) $\cancel{\ldots}$ $\underline{\underline{0.907}}$ (chart).

$$x_2 = \frac{h_2 - h_f}{h_{fg}} = \frac{2341.5 - 151.5}{2415.2}$$

$$= 0.91$$

$$\eta_{cycle} = \frac{\dot{W}_{x\,out}}{\dot{Q}_{in}} = \frac{h_1 - h_2}{h_1 - h_4} = \frac{h_1 - h_2}{h_1 - h_3}$$

h_3 - saturated @ 0.06 bar. $\Rightarrow h_3 = 151.5$ kJ/kg.

$$\Rightarrow \eta_{cycle} = \frac{3712.1 - \cancel{151.5}\ 2341.5}{3712.1 - \cancel{2341.5}\ 151.5} = 38.5\%.$$

$$\dot{m}_{H_2O} = \frac{600 \times 10^3}{(3712.1 - 2341.5)} = 438 \text{ kg s}^{-1}$$

b)

25% reduction in $\dot{Q}_{in}$

so $h_1' - h_3 = 0.75(3712.1 - 151.5)$

$= 2670.5$

so $h_1' = 2822$ kJ/kg so still superheated.

$s_1' =$

interpolate between 350 and 375°C

$$T_1' = 350 + (375-350) \times \frac{2822 - 2693.1}{2888.9 - 2693.1} = 369.4^\circ C$$

$$\eta_{ci} = \frac{h_1 - h_2}{h_1 - h_{2s}} \qquad s_1' = 5.4437 + (5.7050 - 5.4437) \times \frac{369.4 - 350}{375 - 350}$$

$s_{2s} = s_1' = 5.626$ kJ/kgK. $h_{2s} = 1740$ kJ/kg

↑ chart.

so $h_2 = h_1 - \eta_i(h_1 - h_{2s}) = 2822 - 0.85(2822 - 1740)$

$= 1902$ kJ/kg.

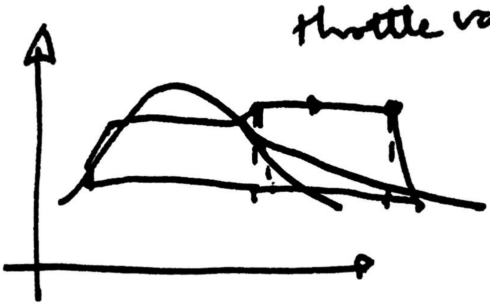
$x_2' = 0.725$ not viable!

So $\dot{W}_x = (2822 - 1902) \times 438$

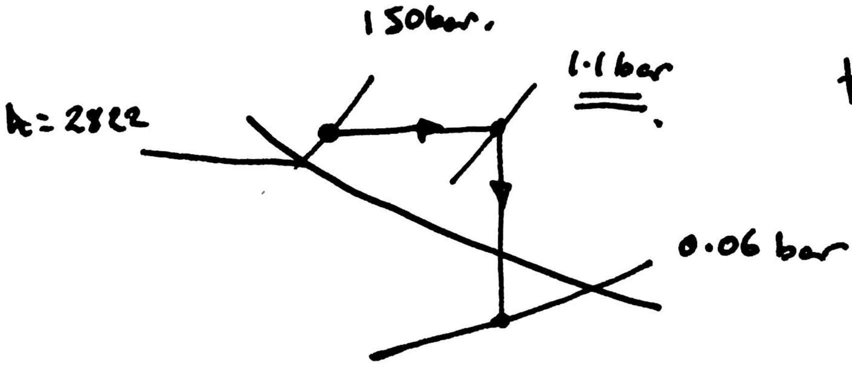
$= 402.96$ MW

$$\eta = \frac{2822 - 1902}{2822 - 151.5} = 34.5\%$$

c)



throttle valve will increase dryness @ turbine exhaust.



target point is

$x = 0.91$

$h \approx 2380$

so only 48% of work output

so $\Delta p \approx 149$ bar.

$\eta = 16.6\%$

very low

Question 3. Attempts 286, average mark 15.4/25, maximum 24, minimum 3.
This was the most popular question, attempted by most of candidates. Part (a)(i) (superheated Rankine cycle T–s diagram) was best answered; nearly everyone obtained full marks, apart from a few who did not realise that the pressure and temperature would hold constant inside the water-vapour dom. Part (a)(ii) asked for the cycle efficiency, mass flow rate and the wetness of the cycle. Most candidates calculated the cycle efficiency and mass flow rate correctly (some were surprised by the large number of the mass flow rate!) but a significant number did not calculate the actual wetness with turbine irreversibility. Part (b) asked the candidates to calculate the cycle with reduced heat input, thus lower turbine entry temperature. It was in general well done and most of candidates were able to realise that main problem of the cycle is the excessive wetness. In Part (c) it was proposed to use a throttle to mitigate the wetness problem. Most candidates could work out how this would reduce the wetness and a large number could calculate the required the pressure drop correctly. Many guessed correctly that the cycle efficiency would drop, but none actually tried to calculate/estimate the new cycle efficiency.

2P4 SECTION B

4. (a) Mass Continuity : $\rho_1 V_1 (1-\alpha) + \beta \phi \rho_1 V_1 \alpha = \rho_2 V_2$ [2]

Volumetric flow continuity : $V_1 (1-\alpha) + \alpha \phi V_1 = V_2$ [2]

$$\rho_2 = \frac{\rho_2 V_2}{V_2} = \frac{\rho_1 V_1}{V_1} \frac{[(1-\alpha) + \beta\phi\alpha]}{[(1-\alpha) + \alpha\phi]}$$

$$\rho_2 = \rho_1 \left(\frac{1-\alpha + \beta\phi\alpha}{1-\alpha + \phi\alpha} \right)$$ [1]

5

(b) Momentum:

$$(P_1 + \rho_1 V_1^2)(1-\alpha) + (P_2 + \phi^2 \beta \rho_1 V_1^2)\alpha = P_2 + \rho_2 V_2^2$$ [3]

$$P_1 - P_2 = \rho_2 V_2^2 - \rho_1 V_1^2 [(1-\alpha) + \beta\phi^2\alpha]$$

from part (a) $\rho_2 V_2^2 = \rho_1 V_1^2 [1-\alpha + \alpha\beta\phi][1-\alpha + \alpha\phi]$ [3]

$$\frac{P_1 - P_2}{\rho_1 V_1^2} = [1-\alpha + \alpha\beta\phi][1-\alpha+\alpha\phi] - [1-\alpha + \beta\phi^2\alpha]$$ [2]

$$= (1-\alpha)^2 + \alpha\phi(1-\alpha) + \alpha\beta\phi(1-\alpha) + \alpha^2\phi^2\beta - (1-\alpha) - \beta\phi^2\alpha$$

$$\frac{P_1 - P_2}{\rho_1 V_1^2} = (1-\alpha)\left[1-\alpha+\phi\alpha+\phi\alpha\beta-\alpha\beta\phi^2-1\right]$$

$$\Rightarrow \quad \frac{P_1 - P_2}{\rho_1 V_1^2} = -\alpha(1-\alpha)\left[\phi^2\beta-\phi(\beta+1)+1\right] \qquad [2]$$

10

(c) Reduction in pressure $P_1 > P_2$

$$-\alpha(1-\alpha)\left(\phi^2-\phi(\beta+1)+1\right) > 0 \qquad [1]$$

α must be positive and <1

hence $\phi^2\beta - \phi(\beta+1)+1 < 0$ [1]

for $\Delta p = 0$, $\phi = \dfrac{(\beta+1) \pm \sqrt{(\beta+1)^2-4\beta}}{2\beta}$

if $\beta > 1$ $\quad \frac{1}{\beta} < \phi < 1$ [1]

if $\beta < 1$ $\quad 1 < \phi < 1/\beta$ [1]

5

(d) Rate of change of mechanical energy $= \Delta(QP_0)$ [2]

$$\Delta(QP_0) = \left(P_2+\tfrac{1}{2}\rho_2 V_2^2\right)V_2 A - \left(P_1+\tfrac{1}{2}\rho_1 V_1^2\right)V_1(1-\alpha)A - \left(P_1+\tfrac{1}{2}\rho_1 V_1^2\phi^2\beta\right)V_1\phi\alpha A$$

$\phi = 2$, $\beta = 1$, $\alpha = 0.5 \Rightarrow V_2 = 3/2\, V_1$, $\rho_2 = \rho_1$

$$\frac{P_2 - P_1}{\rho_1 V_1^2} = \frac{1}{4} \quad \text{(pressure rise)}$$

$$\Delta(Q P_0) = -\left[-\frac{3}{8} + \frac{9}{16}\right] \rho_1 V_1^3 A$$

$$= -\frac{3}{16} \rho_1 V_1^3 A \qquad [2]$$

Mechanical energy lost through viscous dissipation [1]

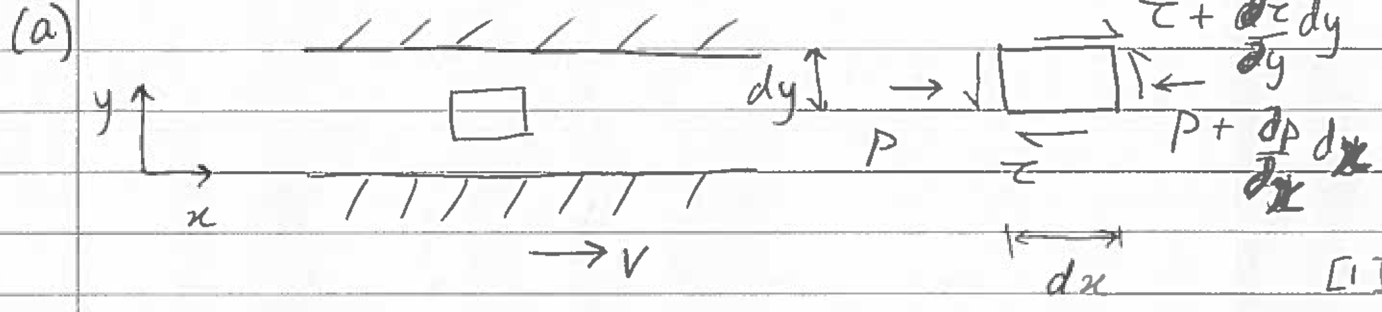
5

Question 4. Attempts 298, mean mark 17.4/25.

This question concerned the mixing of two streams of different density. It was the most popular question, attempted by virtually all candidates. On the whole, this question was completed very well. One common mistake was to attempt to use Bernoulli to derive the expression for pressure change across the mixing process. This gave an answer similar to that given in the question but with a factor of a half which many students ignored (or simply crossed-out). The candidates on the whole were unable to determine the mechanical energy change across the mixing process. Many assumed that this was either the change in pressure or the change in kinetic energy. Only a very small number of students (<5%) computed this correctly.

5

(a)



[1]

$$\rho v \frac{\partial u}{\partial y} + \rho u \frac{\partial u}{\partial x} = -\frac{\partial p}{\partial x} + \frac{\partial \tau}{\partial y} \quad [1]$$

$u = u(y)$, $\tau = \tau(y)$ streamlines straight and parrallel $\rightarrow v = 0$ [1]

$$\frac{\partial u}{\partial x} = 0$$

$$\frac{dp}{dx} = \frac{d\tau}{dy} \qquad \frac{\partial p}{\partial y} = 0$$

$$\frac{dp}{dz} = \frac{P_2 - P_1}{l} \Rightarrow \frac{d\tau}{dy} = \frac{P_2 - P_1}{l} \quad [2]$$

5

(b) $\tau = \mu \frac{du}{dy} \Rightarrow \mu \frac{d^2 u}{dy^2} = \frac{(P_2 - P_1)}{l}$ [1]

$$\frac{du}{dy} = \frac{(P_2 - P_1)}{l\mu} y + \left(\frac{du}{dy}\right)_{y=0} \quad [1]$$

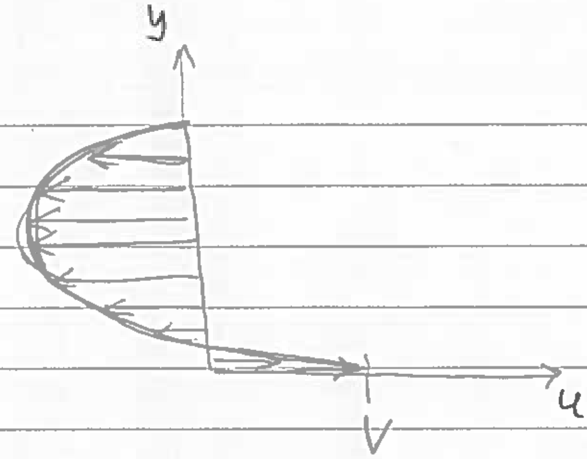
$$\left(\frac{du}{dy}\right)_{y=0} = \frac{\tau_w}{\mu} \qquad \frac{du}{dy} = \left(\frac{P_2 - P_1}{l\mu}\right) y + \left(\frac{\tau_w}{\mu}\right)$$

$y = 0$, $u = V$

$$\Rightarrow u = \left[\frac{P_2 - P_1}{l\mu}\right] \frac{y^2}{2} + \left[\frac{\tau_w}{\mu}\right] y + V \quad [2]$$

5

(c)



[3]

Continuity:

$$Q = \frac{-Vh}{2} \quad \text{(through one clearance)}$$ [2]

(d)

$$\frac{-Vh}{2} = \int_0^c u\,dy = \frac{(P_2 - P_1)}{6\mu l}c^3 + \left[\frac{\tau_w}{\mu}\right]\frac{c^2}{2} + Vc$$

[2]

from velocity profile, when $y = c$, $u = 0$

$$0 = \frac{(P_2 - P_1)}{2\mu l}c^2 + \left[\frac{\tau_w}{\mu}\right]c + V$$ [1]

Combining equations:

$$\frac{-Vh}{2} = \left(\frac{\tau_w}{\mu}\right)\left[\frac{c^2}{2} - \frac{c^2}{3}\right] + V\left[c - c/_3\right]$$

$$\frac{\tau_w}{\mu} = V\left[-\frac{h}{2} - \frac{2c}{3}\right]\frac{6}{c^2}$$ [2]

$$\frac{-Vh}{2} = \frac{(P_2 - P_1)}{\mu l}\left[\frac{c^3}{6} - \frac{c^3}{4}\right] + \frac{Vc}{2}$$

$$\frac{P_2 - P_1}{\mu l} = V\left[-\frac{h}{2} - \frac{c}{2}\right]\cdot\left(\frac{-12}{c^3}\right)$$ [2]

Force per unit width
applied by fluid on block

$$F = [P_1 - P_2]h + 2\tau_w l \qquad [2]$$

$$F = \left[\frac{-h}{2} - \frac{c}{2}\right] \times \left(\frac{12}{c^3}\right) V\mu l h$$

$$+ \left[\frac{-h}{2} - \frac{2}{3}\right] \times \left(\frac{12}{c^2}\right) V\mu l$$

$$F = -\frac{6V\mu l}{c^3}\left[h^2 + 2hc + \frac{4}{3}c^2\right] \qquad [1]$$

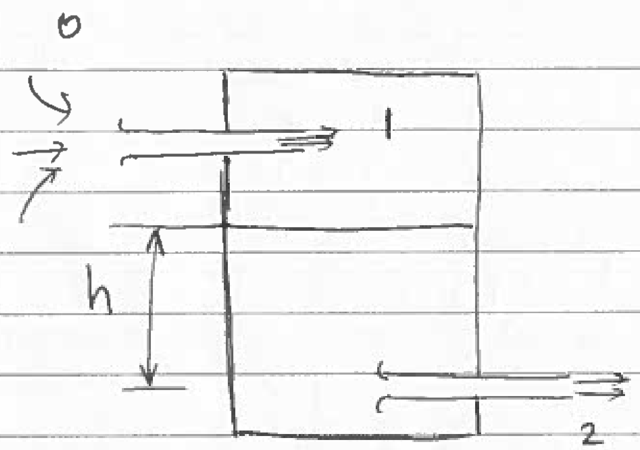
Neglecting momentum of flow outside clearance
$(c \ll h)$

10

Question 5. Attempts 285, mean mark 13.6/25.

This question was about viscous flow in a damper. It was also a popular question, attempted by most candidates. Most students were able to derive the relationship between the gradient of shear stress and streamwise pressure gradient within the damper clearance, as well as to derive the expression for the velocity profile. A few candidates were confused about the direction of shear stress on the elemental control volume within the clearance. Around 20% of the candidates were able to correctly sketch the velocity profile within the clearance. Most candidates struggled to relate both the pressure and wall shear to the force exerted by the fluid. Some chose to ignore either the pressure term or the shear stress term. Around 30% of candidates were able to relate the motion of the piston to the volumetric flow rate within the clearance. Overall most candidates were not able to find an expression for the force as required.

6 (a)



Assuming incompressible flow

$$V_a \pi d^2 = V_w \pi d^2$$
$$V_a = V_w \qquad [2]$$

$$Re_a = \rho_a V_a d / \mu_a \qquad Re_w = \rho_w V_w d / \mu_w$$

$$Re_a / Re_w = \frac{\rho_a V_a / \mu_a}{\rho_w V_w / \mu_w} = \frac{\rho_a}{\mu_a} \cdot \frac{\mu_w}{\rho_w}$$

$\rho_a = 1.225$ kg/m³ $\qquad \rho_w = 1000$ kg/m³ (or use ν)

$\mu_a = 17.9 \times 10^{-6}$ Pas $\qquad \mu_w = 1.14 \times 10^{-3}$ Pas

$$Re_a / Re_w = 0.078 \qquad [3]$$

5

(b) Assuming parrallel flow leaving each pipe

frictionless flow (Bernoulli)

$$P_0 = P_{atm} = P_1 + \frac{1}{2}\rho_a V_a^2$$

$$P_1 + \rho_w g h = P_{atm} + \frac{1}{2}\rho_w V_w^2$$

$V_a = V_w \Rightarrow V = \sqrt{\dfrac{2\rho_w g h}{\rho_a + \rho_w}} \approx \sqrt{2gh}$ [3]

$h = 0.4m$, $V = 2.8 m/s$

$Re_a = 1917$

$Re_w = 24579$ [2]

5

(c) $k/d = 0.01$

Inlet pipe $C_{fa} \simeq 0.008$ (laminar)

Exit pipe $C_{fw} \simeq 0.01$ (turbulent)

[2]

Below $Re \sim 2000$ the flow is laminar and so skin friction not dependent on roughness [1]

For $Re > 4000$ the flow is turbulent and skin friction depends on roughness. [1]

Dependency on Reynolds number as either roughness or Reynolds number is increased, reduces as turbulent momentum transfer dominates (or similar) [1]

5

(d) $\Delta P_o = C_f \times \frac{1}{2}\rho V^2 \times \left(\frac{l}{d}\right) \times 4$

Inlet pipe $\Delta P_o = 3.8$ Pa [2]

Exit pipe $\Delta P_o = 3920$ Pa [2]

Friction effect in exit pipe approximately equal to $\rho_w g h$.

→ Friction can't be neglected. [1]

5

(e) $V_a = V_w$ continuity still holds

[2]

$$P_{atm} - C_{fa} \cdot \frac{1}{2}\rho_a V_a^2 \cdot \left(\frac{l}{d}\right) \cdot \cancel{4} = P_1 + \frac{1}{2}\rho_a V_a^2$$

$$P_1 + \rho_w g h - C_{fw} \frac{1}{2}\rho_w V_w^2 \left(\frac{l}{d}\right) \cdot 4 = P_{atm} + \frac{1}{2}\rho_w V_w^2$$

$$P_1 + \rho_w g h - C_{fw} \cdot \frac{1}{2}\rho_w V_w^2 \cdot \left(\frac{l}{d}\right) \cdot 4 =$$

$$P_1 + \underbrace{\frac{1}{2}\rho_a V_a^2 + C_{fa}\frac{1}{2}\rho_a V_a^2 (l/d) 4}_{\text{Neglect (small compared to } \rho_w g h)} + \frac{1}{2}\rho_w V_w^2$$

$$\rho_w g h = \frac{1}{2}\rho_w V^2 \left[1 + C_{fw} \cdot 4 \cdot (l/d)\right]$$

$$V_w = V_a = \sqrt{\frac{2gh}{1 + 4(l/d) C_{fw}}} = 1.98 \text{ m/s}$$ [2]

Check new $Re_w = 17380$ [1]

C_{fw} not significantly altered

5

Question 6. Attempts: 34, mean mark 15.9/25.

This question concerned flow in the inlet and outlet pipes of a water tank. This was the least popular question, only attempted by 34 candidates. A common mistake was to apply mass conservation, as opposed to volumetric flow conservation to this problem. This led to unrealistically high flow speeds within the inlet pipe of the water tank. As a consequence, students who made this mistake, incorrectly determined that the flow in the inlet would be at a very high Reynolds number. Nearly all students were able to correctly determine the pressure losses due to friction, making use of the Moody chart given. Only a small number were able to then determine the bulk velocity in the pipe with friction included.