ENGINEERING TRIPOS PART IB

Tuesday 5 June 2001 2 to 4

Paper 4

THERMOFLUID MECHANICS

*Answer not more than **four** questions.*

*Answer **two** questions from each section*

Answers to questions in each section should be tied together and handed in separately.

All questions carry the same number of marks.

*The **approximate** number of marks allocated to each part of a question is indicated in the right margin.*

(TURN OVER

SECTION A

1 (a) Wet steam at 0.1 MPa and unknown dryness fraction flows steadily along a pipe and through an insulated throttle valve. With the aid of a sketch on an *h-s* diagram, explain how the throttling process may be used to determine the dryness fraction upstream of the valve. Consider first the case where the kinetic energy of the fluid may be neglected and then indicate how this argument must be modified if kinetic energy changes have to be taken into account. [6]

(b) The pressure and temperature downstream of the valve are 0.01 MPa and 55°C respectively. The mass flow rate of steam is 0.3 kg/s and the diameter of the pipe is 150 mm.

(i) Calculate the velocity downstream of the valve.

(ii) It is found that the kinetic energy upstream of the valve is negligible by comparison with the downstream value. Give a physical explanation for why this is plausible.

(iii) Find the upstream dryness fraction.

(iv) Find the specific entropy upstream and downstream of the valve and explain why the values are different.

[11]

(c) Suggest how the throttling process might be modified so that the upstream dryness fraction can be determined without knowing the mass flow rate of the steam. [3]

2 (a) A counter-flow heat exchanger is used to generate steam for a combined cycle power plant. Temperature variations for the hot and cold flows within the heat exchanger are shown schematically in Fig. 1 as a function of the percentage of heat delivered to the steam. Explain briefly why the gas temperature distribution varies linearly from Q to P while the steam temperature is constant in the same range. [3]

(b) The heat exchanger is required to generate steam at 25 bar and 450°C. Exhaust gas from the turbine enters the heat exchanger with a temperature of 650°C and at a mass flow rate of 50 kg/s. The temperature difference at the pinch point, P, is to be 20°C. Assume that the pressure in the steam generator is constant and that the turbine exhaust gas has the properties of air.

(i) Calculate the temperature of the exhaust gas at point P.

(ii) By considering the transfer of energy from the gas to the steam between the gas inlet and the pinch point, find the mass flow rate of steam.

(iii) Calculate the temperature of the exhaust gas at point Q.

(iv) Neglecting any pressure losses, find the rate of change of entropy between P and Q in both the gas and the steam and hence find the overall rate in increase in entropy. Explain the origin of the increase.

[12]

(c) The overall heat transfer coefficient between the exhaust gas and the steam flow is constant and equal 0.2 kW/m^{2}K. By considering the variation of temperature of the exhaust gas along the heat exchanger between P and Q, determine the required surface area for heat transfer in the boiling section only. [5]

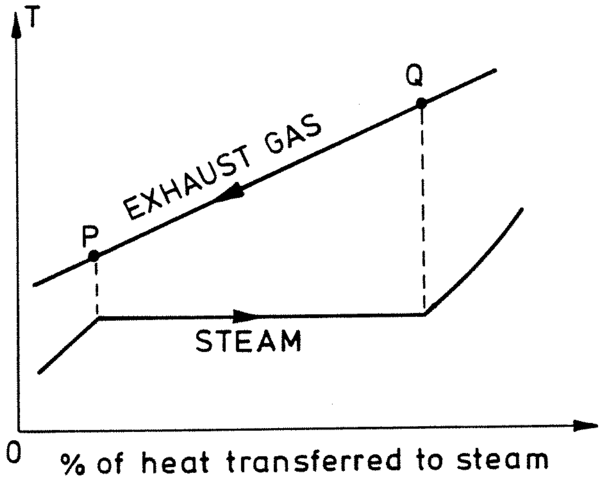

Fig.1

3 The heat pump shown in Fig. 2 uses Refrigerant 12 as the working fluid. The compressor has an isentropic efficiency of 70%, and the refrigerant is in a saturated state at exit from both the evaporator and condenser. The pressures in the evaporator and condenser are 0.261 MN/m^2 and 0.745 MN/m^2 respectively.

(a) Sketch the cycle on *T-s* and *p-h* diagrams, indicating the thermodynamic states 1 – 4 in both cases. [3]

(b) If the heat pump is required to supply heat at a rate of 50kW, calculate the mass flow rate of the refrigerant and the power input to the compressor. [9]

(c) Compare the performance energy ratio of the cycle with that of a reversible heat pump operating between the same evaporator and condenser exit temperatures. Explain the difference. [3]

(d) In order to improve the cycle performance it is suggested that the throttle be replaced by a turbine having an isentropic efficiency of 85%. Indicate the change on the *T-s* and *p-h* diagrams and determine the new performance energy ratio, assuming the work output from the turbine is used by the compressor. Would you make the suggested modification? Give your reasons. [5]

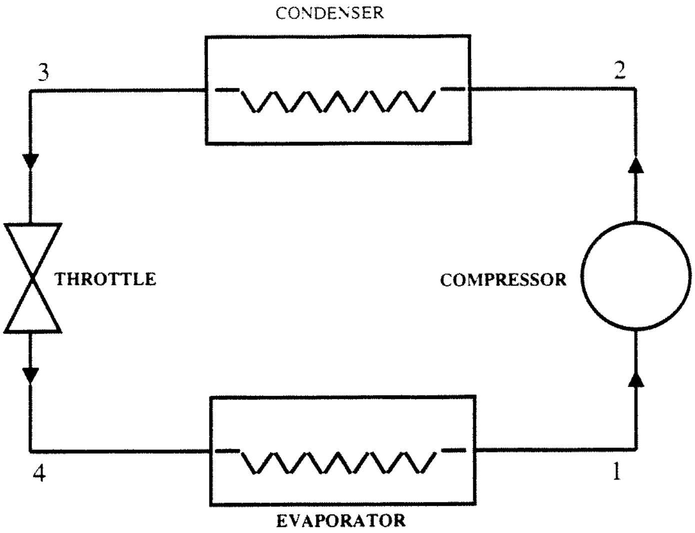

Fig. 2

SECTION B

4 An incompressible fluid of uniform viscosity μ is held between two long, vertical parallel plates as shown in Fig. 3. The surface of the plates are a distance $4W$ apart and a third plate, of thickness $2W$, is used as a plunger to pump fluid vertically upward. The plunger has a speed V and the flow is laminar. At points well removed from the end of the plunger, such as M and N, conditions are independent of the height. The effects of gravity are negligible.

(a) Show, from first principles, that the velocity in the gap at MN is governed by the equation

$$\mu \frac{d^2u}{dx^2} = \frac{dP}{dz}$$

where P is the fluid pressure and u is the vertical component of velocity. [6]

(b) Determine the velocity profile in the gap in terms of V, dP/dz, μ, W and x. [7]

(c) Now use the principle of mass conservation to derive an expression for the velocity profile in terms of V, W and x only. [7]

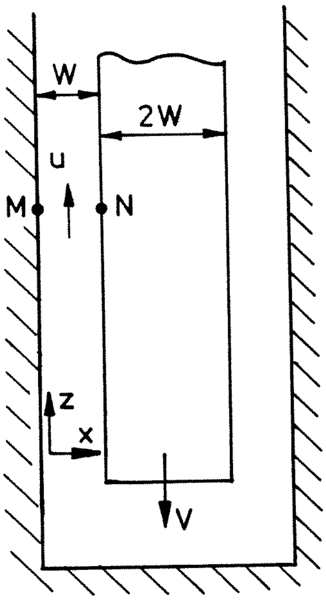

Fig. 3

5 An incompressible fluid of density ρ flows through a pipe expansion and then through a filter as shown in Fig. 4. The flow may be taken as uniform upstream of the expansion and downstream of the filter (sections *A* and *B* in Fig.4). The flow is steady on average and the Reynolds number is high. The drag coefficient for the filter, based on speed V_B, is C_D, and the pressure on the shoulder of the expansion is uniform and equal to the upstream pressure P_A.

(a) Use the control volume shown in Fig. 4 and the force-momentum equation to show that the pressure drop from *A* to *B* is:

$$P_A - P_B = C_D\left(\frac{1}{2}\rho V_B^2\right) + \rho V_B\left(V_B - V_A\right)$$

where V_A and V_B are the speeds at *A* and *B*. [8]

(b) Show that the loss of stagnation pressure in the fluid is:

$$\frac{1}{2}C_D\rho V_B^2 + \frac{1}{2}\rho\left(V_A - V_B\right)^2$$

[8]

(c) The area ratio in the expansion is 2:1 and the upstream speed is V_A=20 m/s. At this speed the drag coefficient for the filter is found to be C_D=1.0. Calculate the temperature rise in the fluid between *A* and *B* for the case where $\rho = 10^3$ kg/m^3 and the specific heat is c = 10^3 J/kg/K. (You may neglect heat loss from the pipe.) [4]

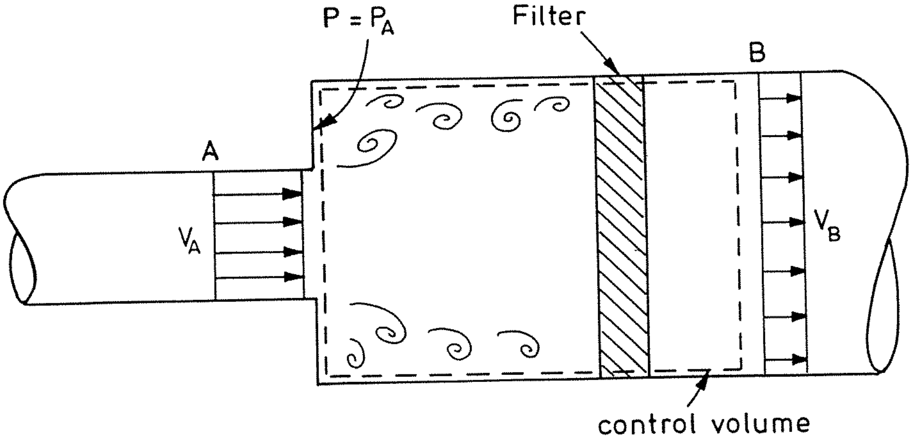

Fig. 4

6 An electrically heated cylinder of diameter d and length L is cooled by a cross flow as shown in Fig. 5. The speed and temperature of the coolant well upstream of the cylinder are uniform and equal to V and T_∞ respectively. The kinematic viscocity, ν, thermal diffusivity, α, and thermal conductivity, λ, of the coolant may be treated as uniform. The rate of heating of the cylinder per unit length is $\dot{Q}$, and the difference in temperature between the coolant and the surface of the cylinder is ΔT. The cylinder is long, $L >> d$, and it is found that $\dot{Q}$ depends only on ΔT, V, d, λ, α and ν.

(a) Use dimensional analysis to show that $\dot{Q}$ is directly proportional to ΔT. [8]

(b) For gases it is found that $\dot{Q}/\Delta T$ is proportional to $\nu^{-0.2}$. Give a physical explanation for why, when $\dot{Q}$ is fixed, the temperature difference increases if ν increases. [4]

(c) For liquid-metal coolants it is found that ν ceases to be a significant parameter, and that $\dot{Q}/\Delta T$ is proportional to $V^{0.4}$. Show that $\dot{Q}/\Delta T$ is also proportional to $d^{0.4}$. What is the dependence of $\dot{Q}/\Delta T$ on α? [4]

(d) When the Reynolds number of the flow is around 100 it is found that regular oscillations in temperature occur in the wake of the cylinder. Sketch the flow and explain the origin of the oscillations. [4]

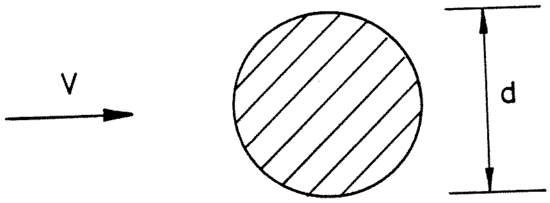

Fig. 5

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