ENGINEERING TRIPOS PART IB

Tuesday 5 June 2012 2 to 4

Paper 4

THERMOFLUID MECHANICS

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

Answer not more than ***two*** *questions from each section.*

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.*

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

There are no attachments.

STATIONERY REQUIREMENTS
Single-sided script paper

SPECIAL REQUIREMENTS
Engineering Data Book
CUED approved calculator allowed

You may not start to read the questions printed on the subsequent pages of this question paper until instructed that you may do so by the Invigilator

nra04

SECTION A

*Answer not more than **two** questions from this section.*

1 A stream of fluid undergoes a reaction which gives uniform heat generation. The fluid must be kept at a constant temperature, T_f = 60 °C. The fluid has density $\rho = 1000\ \mathrm{kg\,m^{-3}}$, dynamic viscosity $\mu = 5 \times 10^{-3}\ \mathrm{Nm^{-2}\,s}$, thermal conductivity $\lambda_i = 0.5\ \mathrm{Wm^{-1}K^{-1}}$ and Prandtl number $Pr = 1$. It is passed through a thick walled tube of circular cross section, with inner radius $r_i = 25$ mm, outer radius $r_o = 30$ mm, and thermal conductivity $\lambda = 0.25\ \mathrm{Wm^{-1}K^{-1}}$.

(a) Assuming that the flow within the tube is fully developed and that the Reynolds Analogy holds, obtain an estimate for the mean internal heat transfer coefficient, h_i, in terms of the pressure drop in the tube, Δp, between two points at a distance L apart. Give your answer in terms of the mean fluid velocity U, the specific heat capacity of the fluid c_p, and the tube inner radius r_i. [5]

(b) Heat is removed from the fluid by forced convection across the outside of the tube. It can be assumed that the cooling fluid has temperature $T_o = 10$ °C, and the heat transfer coefficient is $h_o = 50\ \mathrm{Wm^{-2}K^{-1}}$. Constant temperature conditions are achieved with a bulk mean fluid velocity inside the tube of $U = 5\ \mathrm{m\,s^{-1}}$.

(i) Find the Reynolds number of the flow within the tube. This time, use the appropriate correlation from the Thermofluids Data Book to find the inner surface heat transfer coefficient, h_i. [3]

(ii) Show that the overall thermal resistance of the system in this situation is $0.227\ \mathrm{K\,W^{-1}}$. [5]

(iii) Calculate the heat transferred out of the fluid per unit length of tubing and hence the volumetric heat release in $\mathrm{W\,m^{-3}}$. [2]

(c) Cooling by natural convection alone is being considered instead. The tube is arranged such that its axis is horizontal. For this situation, you may assume that the coolant flow has a Grashof number $Gr = 1 \times 10^6$, thermal conductivity $\lambda_o = 0.05$ W m^{-1}K^{-1}, and Prandtl number $Pr = 1$.

(i) Estimate the heat transfer coefficient for the outer surface of the tube using the correlation

$$Nu = 0.5\,(Gr\,Pr)^{0.25}$$

[3]

(ii) Suggest, with a brief justification, how the tube could be arranged to increase the Grashof number. [2]

2 (a) The specific steady flow availability function, b, is defined by the expression

$$b = h - T_0 s$$

where T_0 is the temperature of the environment, h is the specific enthalpy and s is the specific entropy of the flow.

(i) Using this definition together with the First and Second Laws of Thermodynamics for a steady flow, show that the maximum power potential between two states can be given by the following expression

$$b_2 - b_1 = -\dot{w}_x + \int \left(1 - \frac{T_0}{T}\right) \mathrm{d}\dot{q} - T_0 \Delta s_{irrev}$$

[5]

(ii) Give a physical explanation for each of the terms on the right hand side of this expression. [2]

(b) A stream of air at a pressure of 18 bar and a temperature of 1800 K is used for power generation. The ambient conditions are 1 bar and 300 K. You may assume that the air may be treated as an ideal gas with ratio of specific heats $\gamma = 1.4$, specific heat capacity at constant pressure $c_p = 1.005\,\mathrm{kJ\,kg^{-1}K^{-1}}$ and specific gas constant $R = 0.287\,\mathrm{kJ\,kg^{-1}K^{-1}}$.

(i) The steady flow is expanded to ambient pressure through an adiabatic turbine with isentropic efficiency 85%. Calculate the exit temperature and the power produced per unit mass flow rate. [2]

(ii) Calculate the specific maximum power potential between the inlet and the outlet of the turbine. Comment on the physical origin of the difference between this value and your answer to part (b)(i). [5]

(c) After the turbine, the flow enters a counter-flow heat-exchanger where it transfers heat to a second stream of gas. The turbine exit flow leaves the heat exchanger at ambient temperature. The second stream of gas enters the heat exchanger at ambient conditions and leaves at a temperature which is 20 °C lower than the turbine exit temperature calculated in part (b)(i). The second stream also has the properties given in part (b) and the heat exchanger has negligible pressure drop.

(i) Calculate the ratio of mass flow rates of the two streams. [1]

(ii) Calculate the change in steady flow availability of each stream, per unit mass of air on the turbine exit side. [3]

(iii) Comment on the physical origin of the difference between these two values. [2]

3 (a) A refrigeration cycle for an air conditioning plant uses R-134A as its working fluid. The condenser operates at a pressure of 12 bar, and the evaporator operates at a pressure of 3.5 bar. The fluid leaves the evaporator as saturated vapour. The compressor has an isentropic efficiency of 70%. The fluid leaves the condenser as saturated liquid and is then expanded through a turbine which also has an isentropic efficiency of 70%.

(i) Sketch the cycle on $T-s$ and $p-h$ diagrams. [4]

(ii) Find the temperatures at exit from the condenser and inlet to the compressor. [2]

(iii) Calculate the coefficient of performance of the cycle. You should assume that all the work produced by the turbine is used to help drive the compressor. [8]

(iv) Compare this to the Carnot coefficient of performance. [2]

(b) Calculate the reduction in the coefficient of performance of the cycle if the turbine were to be replaced with a throttle valve. [2]

(c) It is desired to use the cycle as a heat pump in colder weather. In this case, the evaporator will be required to function with an ambient temperature as low as $-20\ °C$. If the evaporator needs a $5\ °C$ minimum temperature difference to operate, what must its working pressure be reduced to? [2]

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SECTION B

*Answer not more than **two** questions from this section.*

4 A cylindrical tube with inner radius R_2 moves at velocity V over a stationary shaft of radius R_1, Fig. 1. The gap between R_1 and R_2 is filled by an incompressible, viscous Newtonian fluid and the flow is laminar. The ends of the tube are open to the atmosphere. Away from the ends, the flow is fully developed in the sense that the streamwise velocity, u, and shear stress, τ, do not change in the streamwise direction.

(a) By considering an appropriate element of fluid, or otherwise, show that

$$\frac{d(r\tau)}{dr} = 0$$

where r is the radial co-ordinate. [4]

(b) Find an expression for the streamwise fluid velocity u in terms of R_1, R_2, r, and the dynamic viscosity μ. [6]

(c) In terms of the same variables, find an expression for the force per unit length required to move the tube over the shaft. [5]

(d) A scale model of the rod and tube is constructed. The dimensions of the model are one tenth those of the original case and the same fluid is used. For dynamic similarity, what ratio of the tube velocity in the model to that of the original case is required? By what factor will the force per unit length calculated above differ between the model and the original case? State any assumptions made. [5]

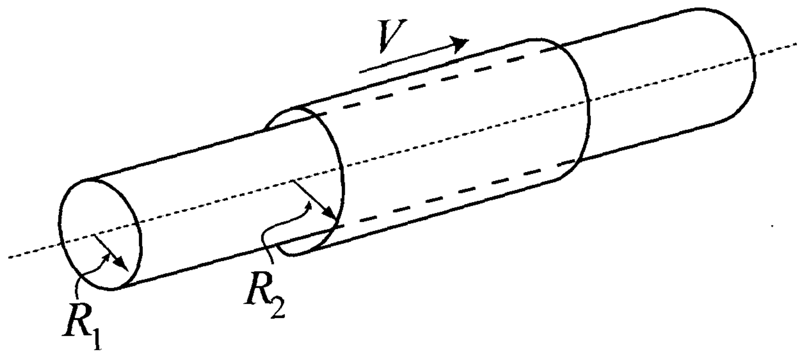

Fig. 1

(TURN OVER

5 A manufacturing process requires a nozzle to provide a jet of fluid. The nozzle is connected to a pump which is supplied by two large tanks, each containing the same fluid, as shown in Fig. 2. The diameter of all pipes is D and the nozzle exit diameter is d. The length of the pipes from the tanks to the pump are L_1 and L_2 for tanks 1 and 2 respectively. The length of the pipe from the pump to the nozzle is L_3. The height of the fluid surface in both tanks is z above the nozzle exit height. The fluid is incompressible and has density ρ. The coefficient of friction for all of the pipes is c_f.

(a) Identify and explain the causes of stagnation pressure loss in the pipe network shown in Fig. 2. Use sketches where appropriate. [5]

(b) The nozzle loss coefficient, based on nozzle exit velocity, is K_N. Show that the stagnation pressure immediately downstream of the pump $p_{0pump,2}$ is given by

$$p_{0pump,2} = p_a + \frac{1}{2}\rho V_3^2 \left[4c_f \frac{L_3}{D} + \frac{D^4}{d^4}\left(1 + K_N\right)\right]$$

where p_a is atmospheric pressure and V_3 is the velocity in the pipe between the pump and the nozzle. [4]

(c) The stagnation pressure rise provided by the pump is the same for the flow from both tanks. The loss coefficient for both 90 degree bends is K_B and the inlet loss from the tanks to the pipes may be neglected. Show that the ratio α between the velocity in pipe 1, V_1, and the velocity in pipe 2, V_2, is given by

$$\alpha = \frac{V_1}{V_2} = \sqrt{\frac{K_B + 4c_f L_2/D}{K_B + 4c_f L_1/D}}$$

[5]

(d) For the particular case when $V_1 = V_2$, find an expression for the stagnation pressure immediately upstream of the pump, $p_{0pump,1}$. Hence find the work input to the pump. State any assumptions made. [6]

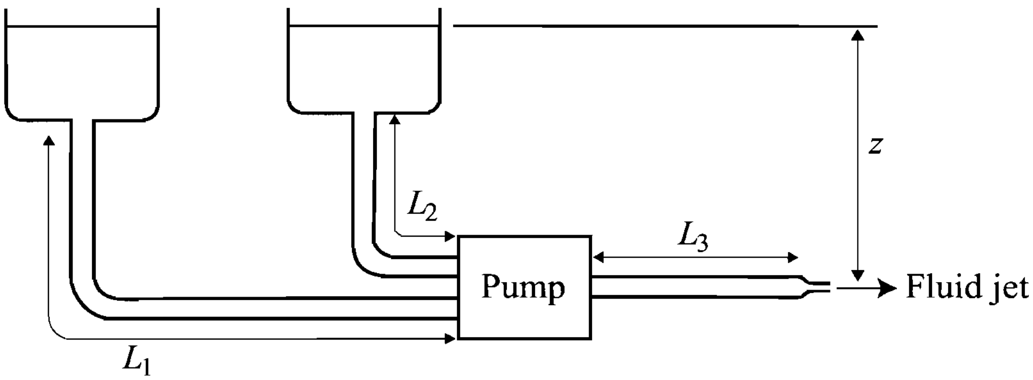

Fig. 2

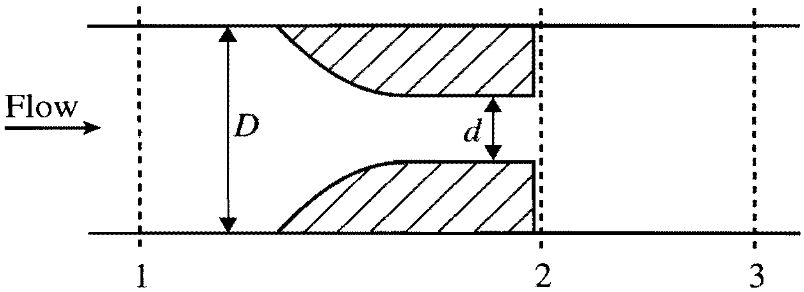

Fig. 3

6 A pipe of circular cross-section is fitted with the device shown in Fig. 3. Flow is from left to right and is incompressible with density ρ. The diameter of the pipe is D. At the narrowest point of the device, the diameter is reduced to d. The flow is uniform far upstream of the device at plane 1 and far downstream of the device at plane 3. Plane 2 is immediately downstream of the device. At plane 2, there is a uniform jet of radius d in the centre of the pipe.

(a) Using the assumption of inviscid flow between planes 1 and 2 show that the velocity at plane 1, V, is given by

$$V = \sqrt{\frac{2(p_1 - p_2)}{\rho\left[\left(\frac{D}{d}\right)^4 - 1\right]}}$$

where p_1 and p_2 are the pressure at plane 1 and plane 2 respectively. [4]

(b) Explain why the pressure can be taken as uniform at plane 2. Hence find an expression for the change in pressure between planes 2 and 3, $p_3 - p_2$, in terms of V, ρ, D and d. State any assumptions made. [6]

(c) Find the change in stagnation pressure along the centre-line between planes 2 and 3, $p_{03} - p_{02}$, in terms of the same variables. What is the cause of this difference?

[6]

(d) To minimise the change in stagnation pressure from planes 2 to 3, an insert is added to the device so that the diameter gradually increases from d to D over a distance L. With reference to boundary layers, discuss the likely effectiveness of this insert as L is varied. [4]

END OF PAPER