UNIVERSITY OF CAMBRIDGE

ENGINEERING TRIPOS PART IB

Tuesday 1 June 2004 2 to 4

Paper 4

THERMOFLUID MECHANICS

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

*Answer **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 to this paper.

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

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SECTION A

Answer two questions from this section.

1 A vertical cylinder contains 1 kg of dry saturated steam at a pressure of 10 bar. The steam is contained in the cylinder by a vertical piston. The pressure is held at a constant 10 bar by the weighted piston. Heat is removed reversibly from the gas until the contents of the cylinder become wet saturated.

(a) Sketch a temperature-entropy *(T-s)* diagram of the process. Show clearly the saturation line of the water. How can the heat flow be represented on the *T-s* diagram? Show that the total heat extraction Q is 2014 kJ. [5]

(b) The total heat flow Q from the cylinder is used to power a reversible heat engine, as shown in Fig. 1. The heat engine rejects heat to a 70 kg iron block. The block starts at a temperature of 17 °C. What is the final temperature of the block? How much work does the engine produce? The specific heat capacity of iron is 437 $Jkg^{-1}K^{-1}$. It may be assumed that no heat is lost to the environment. [8]

(c) Heat flow from the cylinder continues reversibly until the water and iron reach the same temperature. Show that this final equilibrium temperature is 75 °C. Calculate the total work produced by the engine. Explain how the total entropy of the system and the final equilibrium temperature of the iron block would change if, instead of connecting the two to a reversible heat engine, the cylinder and iron block were simply placed in thermal contact. [7]

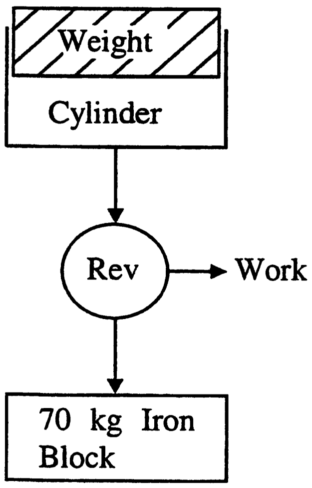

Fig. 1

2 The jet engine shown in Fig. 2 has a compressor pressure ratio of 20. At the exit of the turbine the air passes through a propulsion nozzle. The engine is to be tested in a test cell with an inlet pressure of 1 bar and a temperature of 15 °C. The heat addition per kg of air flow q into the combustion chamber is 450 $\mathrm{kJkg^{-1}}$. The exhaust gas may be treated as air with $c_p = 1.01\ \mathrm{kJkg^{-1}K^{-1}}$, $R = 0.287\ \mathrm{kJkg^{-1}K^{-1}}$ and $\gamma = 1.4$. The compressor, turbine and propulsion nozzle may be assumed to be adiabatic. The work produced by the turbine is equal to the work absorbed by the compressor. Neglect the mass flow rate of fuel and pressure losses in the combustion chamber.

(a) Sketch the cycle on a temperature entropy *(T-s)* diagram. If the compressor has an isentropic efficiency of 85%, show that the turbine inlet temperature is 919.1 °C. [8]

(b) Calculate the exit temperature of the turbine. The turbine has an isentropic efficiency of 87%. Calculate the exit velocity from the propulsion nozzle, assuming that this is large compared to other velocities. Determine the thermal efficiency of the jet engine. It may be assumed that the useful work produced by the engine is equal to the kinetic energy produced by the propulsion nozzle. The propulsion nozzle expands to a pressure of 1 bar and may be assumed to be isentropic. [7]

(c) If the propulsion nozzle is removed from the jet engine what is the maximum work, per kg of air mass flow, that could be theoretically done by the exit flow from the turbine in an environment at 1 bar and 15 °C? Calculate the efficiency of this theoretical cycle. Is this theoretical efficiency achievable in a practical aero-jet engine? Explain your reasoning. [5]

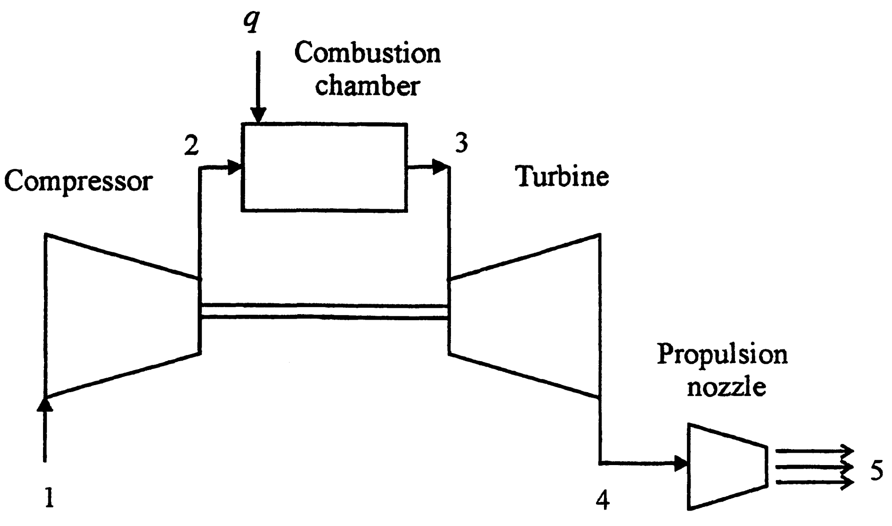

Fig. 2

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3 (a) A high pressure supply of steam is used to heat carbon dioxide in the counter-flow heat exchanger shown in Fig. 3. The carbon dioxide enters the heat-exchanger at 15 °C and leaves at 230 °C. The steam passes through the heat-exchanger at a constant pressure of 40 bar and leaves as water at a temperature of 25 °C. The ratio of mass flow rate of carbon dioxide to steam is 15. The minimum temperature difference between the two streams is located at the carbon dioxide inlet. Sketch a diagram showing the variation of temperature along the flow passages for the two streams. Show that the inlet temperature of the steam is 356 °C.

The specific heat capacity of carbon dioxide should be taken as $c_p = 0.93\ \mathrm{kJkg^{-1}K^{-1}}$ throughout this question. [6]

(b) A second heat exchanger is designed to heat a different mass flow rate of carbon dioxide. The mass flow rate, temperatures and enthalpies of the flow of steam and the inlet temperature of the carbon dioxide are the same as for the heat exchanger in part (a). The pinch point between the dry saturated steam and the carbon dioxide is 10 K. Calculate the ratio of mass flow rate of carbon dioxide to steam. Calculate the temperature of the carbon dioxide leaving the heat exchanger for this case. Sketch the variation of temperatures for the new heat exchanger on the diagram of part (a). [7]

(c) Calculate the difference in the total entropy change, per kg of steam flow, between the heat-exchanger described in part (a) and the heat-exchanger described in part (b). What is the cause of this difference? [5]

(d) Suggest a scheme to replace the heat-exchanger in part (b) with a device that is reversible and which allows the same heat flow to leave the steam side of the heat-exchanger. [2]

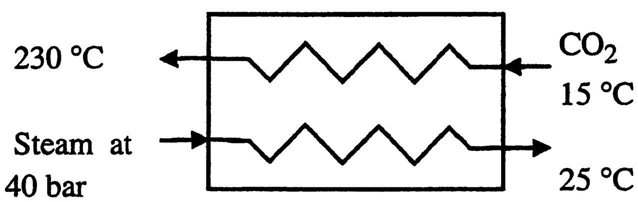

Fig. 3

SECTION B

Answer ***two*** *questions from this section.*

4 An incompressible fluid of density ρ flows through a fan in an annular duct then through a flow straightener into a horizontal duct of diameter d_o as shown in Fig. 4. The central body of the fan and flow straightener has a diameter d_i. The velocity after the flow straightener is axial and uniform and is equal to V_2. The pressure across station (2) is uniform and is equal to p_2.

(a) What is the relationship between the velocity at the far downstream station, V_3, and the velocity in the annulus V_2? [4]

(b) If the effect of friction on the walls is assumed to be negligible, what is the pressure difference between the stations (2) and (3) in Fig. 4 in terms of V_2, ρ, d_i and d_o? [7]

(c) What is the difference in the flux of mechanical energy crossing stations (2) and (3)? What happens to the energy associated with this difference? [5]

(d) The pressure is constant across the annular duct at station (1), between the fan and the flow straightener. The component of velocity in the circumferential direction at station (1) is constant and equal to V_θ. The flow straightener removes the component of circumferential velocity so that at station (2) the flow is axial. If the plates forming the flow straightener are thin, so that there is no axial force and the effects of friction in the axial direction may be neglected, determine the difference in the flux of mechanical energy crossing stations (1) and (2). [4]

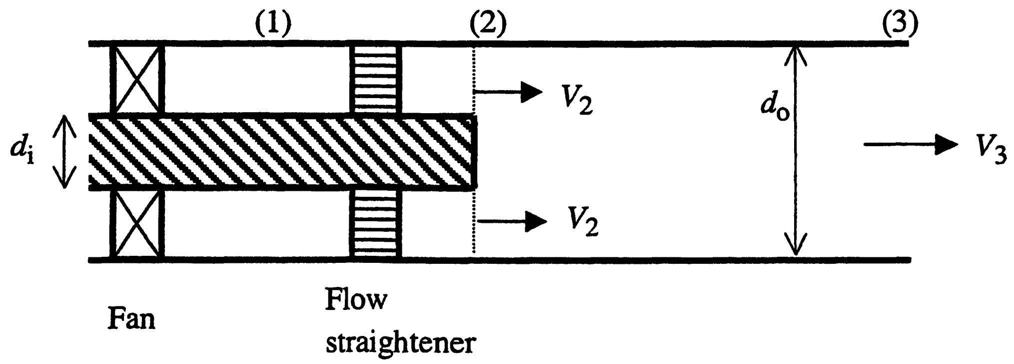

Fig. 4

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5 The performance of a racing yacht of length 15 m and speed 8 ms^{-1} is to be modelled using a 1/10 scale model. A water towing tank is to be used to assess the drag on the hull. It is assumed that the wave drag and the viscous drag are independent and sum to give the total drag. The tests are conducted at the Froude number corresponding to the full size yacht. The density and viscosity of water are 1000 kgm^{-3} and 0.001 $kgs^{-1}m^{-1}$ respectively.

(a) Determine the speed of the test. [2]

(b) Explain why it is not possible to hold the Reynolds number Re constant in these tests, and the likely effect of this on the resulting drag comparison. [3]

(c) To assess the Reynolds number dependence, a model of the wetted hull shape of the yacht is made to the same 1/10 scale. The model is placed in a wind tunnel to measure the viscous drag. The air has a viscosity of 1.757×10^{-5} $kgs^{-1}m^{-1}$ and a density of 1.2 kgm^{-3}. Two test speeds of 30 ms^{-1} and 50 ms^{-1} are used to measure the drag on the model. The resulting wind tunnel tests give the drag on the hull as 0.845 N and 2.12 N respectively. It is postulated that the non-dimensionalised drag force is proportional to Re^{-n}. Determine n and the constant of proportionality. [6]

(d) The drag measured on the model in the towing tank is 12 N. What proportion of this is due to the wave drag? What would be the equivalent wave drag on the full size yacht? [6]

(e) Determine the total drag on the full-size yacht. [3]

6 A vertical pipe of radius r_2 carries oil of density ρ_o and viscosity μ_o upwards as shown in Fig. 5. A thin layer of laminar flow water moves in the region from radius r_1 to r_2. Oil flows in the central part of the pipe.

(a) Explain why the pressure varies only in the vertical direction and the velocity of both fluids varies only in the radial direction. [2]

(b) The shear stress at the interface of the water and oil is τ_i. By considering the forces acting on the oil show that:

$$\tau_i = \left(\frac{dp}{dz} + \rho_o g\right)\frac{r_1}{2}$$

where τ_i acting on the oil is defined as positive upwards, $\frac{dp}{dz}$ is the pressure gradient and g is the acceleration due to gravity. [6]

(c) If the density and the viscosity of the water are ρ and μ respectively, show that the velocity of the flow in the water layer is given by the expression

$$u = \left(\frac{dp}{dz} + \rho g\right)\left(\frac{r^2 - r_2^2}{4\mu}\right) + (\rho_o - \rho)\frac{g r_1^2}{2\mu}\ln\left(\frac{r}{r_2}\right)$$ [10]

(d) Explain why the velocity of the water does not depend on the viscosity of the oil or the flow rate of the oil. [2]

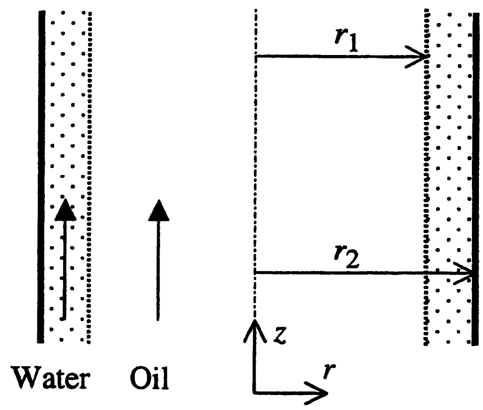

Fig. 5

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