Chapter 2

Thermodynamics Review

2.1 Conservation of Energy for a Stationary Control Volume

Figure 2.1 illustrates a control volume with one entry (subscript 1) and one exit (subscript 2). Conditions are steady, thus the net rate at which the total energy of the fluid is increasing is equal to the rate of heat addition $\dot{Q}$, including internal heat generation, less the rate at which the fluid does work on the environment, namely, shaft work $\dot{W}_{shaft}$ plus shear work along the boundary $\dot{W}_{shear}$, plus expansion work $\dot{W}_{norm}$ resulting from forces normal to the boundary. Conservation of energy requires that

$$\dot{m}\left(u_2 - u_1 + \frac{1}{2}V_2^2 - \frac{1}{2}V_1^2 + gz_2 - gz_1\right) = \dot{Q} - \dot{W}_{shaft} - \dot{W}_{shear} - \dot{W}_{norm}. \qquad (2.1)$$

The expansion work term may be expressed in terms of the more familiar flow work p/ρ. For example, the rate at which work is done on the surroundings as fluid leaves the control volume is the dot product of the velocity and the force acting on the surroundings, namely $[\dot{m}/\rho_2 A_2][p_2 A_2] = \dot{m}p_2/\rho_2$. The shear work will be neglected for the time being and the shaft work simply denoted as $\dot{W}$. The specific enthalpy is $i = u + p/\rho = u + pv$ and the stagnation internal energy and enthalpy are $u^o = u + V^2/2$ and $i^o = i + V^2/2$. Thus, the conservation of energy may be expressed as

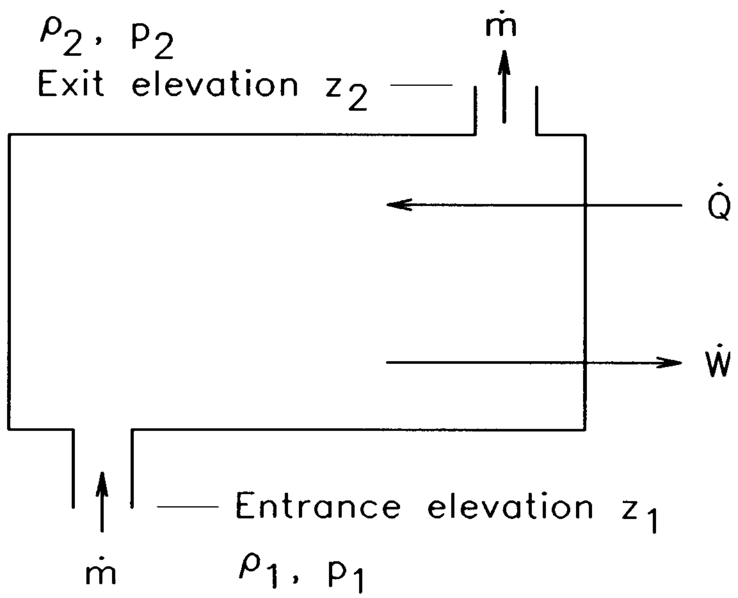

Fig. 2.1. Steady flow through a stationary control volume.

$$\dot{m}\left(\Delta u^{o} + \Delta\frac{p}{\rho} + g\Delta z\right) = \dot{m}\left(\Delta i^{o} + g\Delta z\right) = \dot{Q} - \dot{W}. \tag{2.2}$$

2.2 Pumping Power and Efficiency

For a pump, one may ordinarily neglect Δz, ΔV^2, Δu and $\dot{Q}$. Suppose that flow through the pump is isentropic.[1] Under these approximations, the shaft work requirement, under ideal conditions, is given by

$$-\dot{W}_{ideal} = \dot{m}\Delta i = \dot{m}\Delta\frac{p}{\rho}. \tag{2.3}$$

Since $du = Tds - pdv$, if the fluid is incompressible then the change in internal energy is zero under isentropic (ideal) conditions, and

$$-\dot{W}_{ideal} = \frac{\dot{m}}{\rho}\Delta p. \tag{2.4}$$

If η_p is the pump efficiency, then, by definition,

[1]Setting $\dot{Q}$ to zero in isentropic flow implies zero viscous energy dissipation.

$$(\Delta i)_{actual} = \frac{(\Delta i)_{ideal}}{\eta_p}. \tag{2.5}$$

2.3 Turbine Efficiency

Under isentropic conditions, with negligible kinetic energy effects, the output shaft work is given by

$$-\dot{W}_{ideal} = \dot{m}\Delta i. \tag{2.6}$$

If η_t is the turbine efficiency, then

$$(\Delta i)_{actual} = \eta_t(\Delta i)_{ideal}. \tag{2.7}$$

2.4 Thermal Efficiency

The thermal efficiency for a cycle is ratio of the net actual useful work to the heat addition. For a power plant thermodynamic cycle, such as that illustrated in Fig. 2.2, with the control volume including the reactor, the heat addition is just the heat added in the boiler or nuclear reactor. The net useful work is the actual turbine shaft work less the actual shaft work required for pumps.

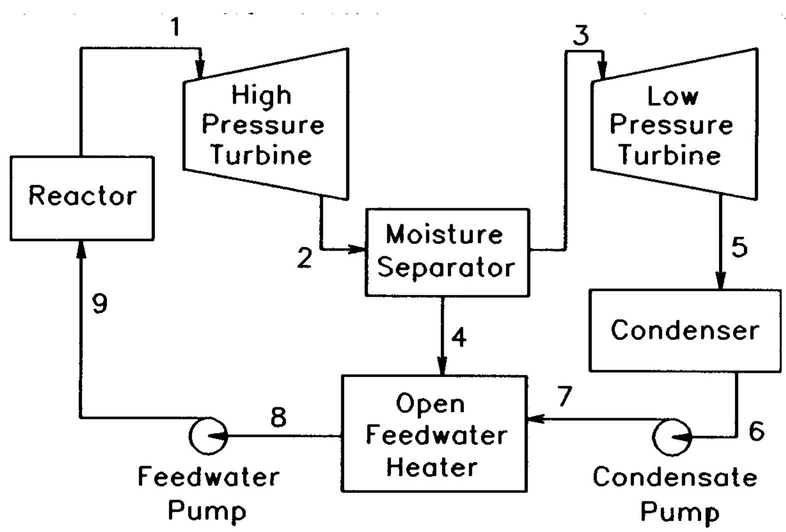

Fig. 2.2. Flow diagram for BWR system.

2.5 Problems

1. Consider the BWR system depicted in Fig. 2.2. Both turbines and both pumps, respectively, have isentropic efficiencies of 90% and 85%. Conditions at each point and saturated steam properties are given below. [Todreas and Kazimi, Vol. 1, problems 6-3 and 6-5].

Point	Condition)	Pressure (kPa)
1	Saturated vapor	6890
2		1380
3	Saturated vapor	1380
4	Saturated liquid	1380
5		6.89
6	Saturated liquid	6.89
7		1380
8		1380
9		6890

	Pressure (kPa)		
	6890	1380	6.89
T_{sat} (°C)	284.8	194.4	38.7
i_g (kJ/kg)	2773.2	2789.0	2571.1
i_f	1261.3	827.28	162.12
i_{fg}	1511.9	1961.7	2408.9
s_g (kJ/kg K)	5.8210	6.4734	8.2794
s_f	3.1112	2.2778	0.5551
ρ_f (kg/m^3)	741.88	871.18	992.66

(a) Calculate thermal efficiency without reheat and feedwater heat, with 100% pump and turbine isentropic efficiencies.

(b) Calculate thermal efficiency without reheat and feedwater heat, with 85% pump efficiency and 90% turbine isentropic efficiencies.

(c) Calculate thermal efficiency with 100% pump and turbine isentropic efficiencies.

(d) Calculate thermal efficiency with 85% pump efficiency and 90% turbine isentropic efficiencies.

2. It is desired to design a vapor pressurizer for a PWR that operates normally at 2200 psia. The primary loop has a volume of 8000 ft^3 and a hot-leg temperature of 610°F. The pressurizer is to be normally 60% full of water and designed to prevent a system pressure rise of more than 1% for a 10°F hot-leg temperature rise. Ignoring the spray and heat added or lost to the ambient, calculate the necessary pressurizer volume. [El-Wakil, Powerplant Technology, problem 10-6.]

3. A PWR powerplant loop is 8000 ft^3 in volume and operates at an average temperature of 580°F. It has a 1000 ft^3 vapor pressurizer which normally contains 60% water by volume at 2200 psia. An accident occurred in which the relief valve suddenly stuck in an open position and fluid discharged to a relief tank. The system pressure steadily dropped to 1600 psia, during which time the electric heaters were fully activated to help slow down the rate of pressure drop. At 1600 psia, the primary loop average temperature was 550°F, the pressurizer was 95% full of steam, the heaters were turned off to protect them from overheating, and the emergency core cooling system (ECCS) was activated. This replenished the primary loop with water to prevent uncovering and damage to the fuel elements. The relief tank is assumed to remain at nearly atmospheric pressure, but there is a 15.3 psi pressure drop in the line connecting it to the pressurizer relief valve. Ignoring the effects of spray and heat losses to ambient, calculate (a) the initial mass composition of the pressurizer [contents], (b) the condition of the fluid leaving the relief valve at the instant it opened (pressure, temperature, and quality or degrees superheat), (c) the total loss of fluid from the primary loop (before ECCS) assuming for simplicity that its temperatures remained the same, and (d) the condition of the fluid leaving the relief valve at the instant the ECCS came on the line. [El-Wakil, Powerplant Technology, problem 10-11].

4. A PWR powerplant producing 10^7 lb_m/h of 1100 psia saturated steam uses reheat from live steam. The high and low pressure turbines exhaust at 250 and 1 psia respectively. For simplicity, assume that both turbines and pump are adiabatic reversible, that there are no feedwater heaters, and that the reheater drain is returned to the steam generator. Calculate (a) the fraction of live steam that is diverted to the reheater, if steam is reheated to 550°F, (b) the cycle net power, in megawatts, and (c) the cycle efficiency. [El-Wakil, Powerplant Technology, problem 10-12].

5. A boiling water reactor operating at a pressure of 70 bar produces 1200 kg/s of saturated steam from feedwater at 200°C. The average core exit quality is 10 percent. Calculate (a) the recirculation ratio,[2] (b) the core inlet enthalpy (J/kg), temperature (°C), and degrees subcooling (°C), and (c) the heat generated in the reactor, in megawatts. [El-Wakil, Powerplant Technology, problem 10-16].

6. Consider a prototype pressurized-water reactor. Assume that flow is uniform across the core of the reactor and that fluid properties may be evaluated at "core average" condi-

[2] The recirculation ratio is the ratio of the mass flow rate of recirculated water to the mass flow rate of steam produced.

tions, namely, 15.51 MPa pressure and 312°C temperature. For that reactor, determine the following additional characteristics. Use SI units unless otherwise indicated.

(a) The hydraulic diameter D_h for flow parallel to fuel rods, defined as 4 times the flow area divided by the wetted perimeter (16×16 arrays of fuel rods).

(b) Coolant mass flux G (kg/m^{2}s) and average velocity V (m/s).

(c) Average friction factor.

(d) Pressure drop across the core due to gravity Δp_g.

(e) Pressure drop across the core due to friction Δp_f, neglecting effects of grid spacers.

(f) Upward force on the core due to frictional shear stresses on fuel rods.

(g) Pumping power requirements (hP and kW) for primary coolant, based on Δp_g and Δp_f, as computed above.

(h) Verification of the 3800 MW(t) thermal output in terms of coolant enthalpy increase.