Separate streams of air and water flow through the compressor and heat exchanger arrangement shown in Fig. P4.100. Steady-state operating data are provided on the figure. Heat transfer with the surroundings can be neglected, as can all kinetic and potential energy effects. The air is modeled as an ideal gas. Determine (a) the total power required by both compressors, in kW. (b) the mass flow rate of the water, in kg/s.

**KNOWN:** Separate streams of air and water flow through a compressor and heat exchanger arrangement.

**FIND:** (a) The total power required by both compressors, in kW, and (b) the mass flow rate of the water, in kg/s.

**SCHEMATIC AND GIVEN DATA:**

**ENGINEERING MODEL:**
1. Control volumes at steady state enclose the compressors and heat exchanger.
2. For each control volume, heat transfer with the surroundings is negligible and kinetic and potential effects can be ignored.
3. The air is modeled as an ideal gas.

**ANALYSIS:**
(a) A mass balance for the air flowing through compressor A, the heat exchanger, and compressor B gives

\[ \dot{m}_1 = \dot{m}_2 = \dot{m}_3 = \dot{m}_4 = 0.6 \text{ kg/s} \]

The energy rate balance for compressor A

\[ 0 = \dot{Q}_{\text{inA}} - \dot{W}_{\text{cvA}} + \dot{m} [(h_1 - h_2) + \frac{1}{2} (V_1^2 - V_2^2) + g(z_1 - z_2)] \]

simplifies to

\[ \dot{W}_{\text{cvA}} = \dot{m}(h_1 - h_2) \]

The specific enthalpies for air at state 1 and state 2 are obtained from Table A-22: \( h_1 = 300.19 \) kJ/kg and \( h_2 = 607.02 \) kJ/kg. Substituting values and solving yield

\[ \dot{W}_{\text{cvA}} = \left(0.6 \frac{\text{kg}}{\text{s}}\right) \left(300.19 \frac{\text{kJ}}{\text{kg}} - 607.02 \frac{\text{kJ}}{\text{kg}} \right) \left(1 \frac{\text{kW}}{\text{kJ}} \frac{1}{\text{s}} \right) = -184.1 \text{ kW} \]

Similarly for compressor B,

\[ \dot{W}_{\text{cvB}} = \dot{m}(h_3 - h_4) \]

The specific enthalpies for air at state 3 and state 4 are obtained from Table A-22: \( h_3 = 451.80 \) kJ/kg and \( h_4 = 821.95 \) kJ/kg. Substituting values and solving yield

\[ \dot{W}_{\text{cvB}} = \left(0.6 \frac{\text{kg}}{\text{s}}\right) \left(451.80 \frac{\text{kJ}}{\text{kg}} - 821.95 \frac{\text{kJ}}{\text{kg}} \right) \left(1 \frac{\text{kW}}{\text{kJ}} \frac{1}{\text{s}} \right) = -222.1 \text{ kW} \]

The total power required by both compressors is

\[ \dot{W}_{\text{total}} = \dot{W}_{\text{cvA}} + \dot{W}_{\text{cvB}} = (-184.1 \text{ kW}) + (-222.1 \text{ kW}) = -406.2 \text{ kW} \]

The negative sign indicates power is to the compressors.

(b) Since the air and water do not mix in the heat exchanger, the steady state mass balance reduces to

\[ \dot{m}_2 = \dot{m}_3 = 0.6 \text{ kg/s} \]

\[ \dot{m}_5 = \dot{m}_6 \]

The steady state form of the energy rate balance
\[ 0 = \dot{q}_{cv} - \dot{w}_{cv} + \sum_i \dot{m}_i \left( h_i + \frac{V_i^2}{2} + \phi_i \right) - \sum_e \dot{m}_e \left( h_e + \frac{V_e^2}{2} + \phi_e \right) \]

simplifies to

\[ 0 = \dot{m}_2 h_2 + \dot{m}_5 h_5 - \dot{m}_3 h_3 - \dot{m}_6 h_6 \]

or substituting results from the mass balance

\[ 0 = \dot{m}_2 (h_2 - h_3) + \dot{m}_5 (h_5 - h_6) \]

Solving for the mass flow rate of water gives

\[ \dot{m}_5 = \frac{\dot{m}_2 (h_2 - h_3)}{h_6 - h_5} \]

The specific enthalpies of water at state 5 and state 6 are obtained from Table A-2:

\[ h_5 \approx h_{i5} = 83.96 \text{ kJ/kg} \quad \text{and} \quad h_6 \approx h_{i6} = 125.79 \text{ kJ/kg} \]

Substituting values and solving yield

\[ \dot{m}_5 = \frac{\left( 0.6 \frac{\text{kg}}{\text{s}} \right) \left( 607.02 \frac{\text{kJ}}{\text{kg}} - 451.80 \frac{\text{kJ}}{\text{kg}} \right)}{125.79 \frac{\text{kJ}}{\text{kg}} - 83.96 \frac{\text{kJ}}{\text{kg}}} \]

\[ \dot{m}_5 = 2.23 \text{ kg/s} \]
Figure P4.99 shows a turbine-driven pump that provides water to a mixing chamber located 25 m higher than the pump. Steady-state operating data for the turbine and pump are labeled on the figure. Heat transfer from the water to its surroundings occurs at a rate of 2 kW. For the turbine, heat transfer with the surroundings and potential energy effects are negligible. Kinetic energy effects at all numbered states can be ignored. Determine
(a) The power required by the pump, in kW, to supply water to the inlet of the mixing chamber.
(b) The mass flow rate of steam, in kg/s, that flows through the turbine.

**KNOWN:** A steam turbine drives a pump through which water flows to a mixing chamber located 25 m higher than the pump.

**FIND:** (a) The power required by the pump, in kW, to supply water to the inlet of the mixing chamber and (b) the mass flow rate of steam, in kg/s, that flows through the turbine.

**SCHEMATIC AND GIVEN DATA:**

**ENGINEERING MODEL:**
1. All components operate at steady state.
2. Define control volume A to encompass the pump and the line to the mixing chamber.
3. Define control volume B to encompass the turbine.
4. For both control volumes A and B, kinetic energy effects can be ignored.
5. No stray heat transfer occurs between control volume B and its surroundings.
6. For control volume B, potential energy effects can be ignored.
ANALYSIS:
(a) The energy rate balance for control volume A

\[0 = \dot{Q}_{cv} - \dot{W}_{cv} + \dot{m} \left[ (h_1 - h_2) + \frac{1}{2} (V_{1}^{2} - V_{2}^{2}) + g(z_1 - z_2) \right]\]

simplifies to

\[\dot{W}_{cv} = \dot{Q}_{cv} + \dot{m} \left[ (h_1 - h_2) + g(z_1 - z_2) \right]\]

since \( \dot{m}_1 = \dot{m}_2 \). The specific enthalpy at state 1 is obtained from Table A-3: \( h_1 = h_{f1} = 417.46 \) kJ/kg. Since the elevation at state 2 is higher than the elevation at state 1, \( (z_1 - z_2) = -25 \) m. Substituting values and solving yield

\[\dot{W}_{cv} = -2 \text{ kW} + \left( 50 \frac{\text{kg}}{\text{s}} \right) \left[ (417.46 \frac{\text{kJ}}{\text{kg}} - 417.69 \frac{\text{kJ}}{\text{kg}}) + \left( 9.81 \frac{\text{m}}{\text{s}^2} \right) (-25 \text{ m}) \right]\]

\[\dot{W}_{cv} = -25.8 \text{ kW}\]

Since the value for power is negative, work is to the pump.

(b) The energy rate balance for control volume B

\[0 = \dot{Q}_{cv} - \dot{W}_{cv} + \dot{m} \left[ (h_3 - h_4) + \frac{1}{2} (V_{3}^{2} - V_{4}^{2}) + g(z_3 - z_4) \right]\]

simplifies to

\[0 = -\dot{W}_{cv} + \dot{m}_3 (h_3 - h_4)\]

since \( \dot{m}_3 = \dot{m}_4 \). Solving for mass flow rate of steam yields

\[\dot{m}_3 = \frac{\dot{W}_{cv}}{h_3 - h_4}\]

Since both states 3 and 4 are superheated vapor, their specific enthalpies are obtained from Table A-4: \( h_3 = 3230.9 \) kJ/kg and \( h_4 = 2812.0 \) kJ/kg. Since the turbine produces the power required by the pump, \( \dot{W}_{cv} \) (turbine) = \( -\dot{W}_{cv} \) (pump) = \( -(25.8 \text{ kW}) = 25.8 \) kW. Substituting values and solving yield

\[\dot{m}_3 = \frac{25.8 \text{ kW}}{3230.9 \frac{\text{kJ}}{\text{kg}} - 2812.0 \frac{\text{kJ}}{\text{kg}}} = \frac{1 \text{ kW}}{1 \frac{\text{kJ}}{\text{s}}} - \frac{1 \text{ kW}}{1 \frac{\text{kJ}}{\text{s}}}

\[\dot{m}_3 = 0.0616 \text{ kg/s}\]