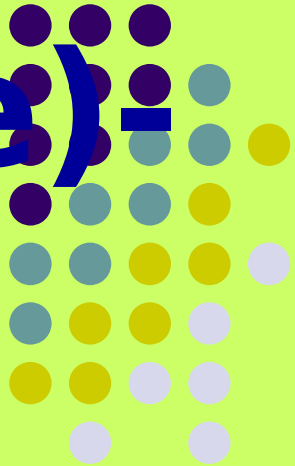
Open System (Control Volume) – EXAMPLES



Steam at 0.4 MPa, 300 ^{o}C, .#1 enters an adiabatic nozzle with a low velocity and leaves at 0.2MPa with a quality of 90%. .Find the exit velocity, in m/s

Q#1 Given

- Steam
- $P_1 = 0.4\,MPa$
- $T_1 = 300^\circ C$
- Adiabatic nozzle
- $\vec{V}_1 = 0$
- $P_2 = 0.2\,MPa$
- $x_2 = 90\% = 0.9$

Required

$\vec{V}_2 = ?$ in m/s

Solution

Assumptions

- Steady state steady flow device (SSSF).
- Single inlet & single exit
- No heat interaction with the surrounding since it is adiabatic nozzle. ($\dot{Q}_i = \dot{Q}_e = 0$)
- No work interactions with the surrounding since nozzle is neither a work producing nor a work consuming device. ($\dot{W}_i = \dot{W}_e = 0$)
- Change in elevation (Potential) energy is negligible. ($\Delta PE = 0$)
- Since the initial velocity is slow, we can ignore it. ($\vec{V}_1 = 0$)

Analysis

① mass balance

$$\sum \dot{m}_i - \sum \dot{m}_e = \frac{dm_{cv}}{dt}^{0} \quad \text{SSSF}$$

$\Rightarrow \sum \dot{m}_i = \sum \dot{m}_e$

$\Rightarrow \dot{m}_1 = \dot{m}_e = \dot{m}$ (single inlet & exit)

② Energy balance

$$\dot{E}_{in} - \dot{E}_{out} = \frac{dE_{cv}}{dt} \to 0 \text{ (SSSF)}$$

$\Rightarrow \dot{E}_{in} = \dot{E}_{out}$

$\Rightarrow \dot{Q}_i^{0} + \dot{W}_i^{0} + \sum \dot{m}_i \left(h + \frac{\vec{V}^2}{2} + gz\right)_i = \dot{Q}_e^{0} + \dot{W}_e^{0} + \sum \dot{m}_e \left(h + \frac{\vec{V}^2}{2} + gz\right)_e$

$\Rightarrow \sum \dot{m}_i \left(h + \frac{\vec{V}^2}{2}\right)_i = \sum \dot{m}_e \left(h + \frac{\vec{V}^2}{2}\right)_e$

$\Rightarrow \dot{m}_i \left(h + \frac{\vec{V}^2}{2}\right)_i = \dot{m}_e \left(h + \frac{\vec{V}^2}{2}\right)_e$

$\Rightarrow h_i + \frac{\vec{V}_i^{2\,0}}{2} = h_e + \frac{\vec{V}_e^2}{2}$

$\Rightarrow h_i = h_e + \frac{\vec{V}_e^2}{2}$

$\Rightarrow h_1 = h_2 + \frac{\vec{V}_2^2}{2} \Rightarrow \vec{V}_2 = \sqrt{2(h_1 - h_2)}$

State 1

$P_1 = 0.4\,MPa$, $T_1 = 300^\circ C$ } Table A-6 $\Rightarrow h_1 = 3069.1\ kJ/kg$

State 2

$P_2 = 0.2\,MPa$, $x_2 = 0.9$ } Table A-5 $\Rightarrow h_{f2} = 504.7\ kJ/kg$, $h_{fg2} = 2201.6\ kJ/kg$

$h_2 = h_{f2} + x_2 h_{fg2} = 2486.15\ kJ/kg$

Therefore,

$$\vec{V}_2 = \sqrt{2\left[3069.1 - 2486.15\right] kJ/kg}$$

$$\Rightarrow \vec{V}_2 = 1077.9\ m/s$$

Example 2: Deceleration of Air in a Diffuser

Air at 10^{o}C and 80 kpa enters the diffuser of a jet engine steadily with a velocity of 200 m/s. The inlet area of the diffuser is 0.4 m^{2}. The air leaves the diffuser with a velocity that is very small compared to the inlet velocity.

Determine

(1) The mass flow rate of the air and
(2) The temperature of the air leaving the diffuser.

Solution Air enters the diffuser of a jet engine steadily at a specified velocity. The mass flow rate of air and the temperature at the diffuser exit are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** Air is an ideal gas since it is at a high temperature and low pressure relative to its critical-point values. **3** The potential energy change is zero, $\Delta pe = 0$. **4** Heat transfer is negligible. **5** Kinetic energy at the diffuser exit is negligible. **6** There are no work interactions.

Analysis We take the *diffuser* as the system (Fig. 5–26). This is a *control volume* since mass crosses the system boundary during the process. We observe that there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

(*a*) To determine the mass flow rate, we need to find the specific volume of the air first. This is determined from the ideal-gas relation at the inlet conditions:

$$v_1 = \frac{RT_1}{P_1} = \frac{0.287\ \text{kPa} \cdot \text{m}^3/\text{kg} \cdot \text{K})(283\ \text{K})}{80\ \text{kPa}} = 1.015\ \text{m}^3/\text{kg}$$

Then,

$$\dot{m} = \frac{1}{v_1} V_1 A_1 = \frac{1}{1.015\ \text{m}^3/\text{kg}} (200\ \text{m/s})(0.4\ \text{m}^2) = \mathbf{78.8\ kg/s}$$

Since the flow is steady, the mass flow rate through the entire diffuser remains constant at this value.

(*b*) Under stated assumptions and observations, the energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\text{Rate of net energy transfer by heat, work, and mass}} = \underbrace{dE_{\text{system}}/dt}_{\text{Rate of change in internal, kinetic, potential, etc., energies}} \overset{0 \text{ (steady)}}{\rightarrow} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) \qquad (\text{since } \dot{Q} \cong 0,\ \dot{W} = 0, \text{ and } \Delta\text{pe} \cong 0)$$

$$h_2 = h_1 - \frac{V_2^2 - V_1^2}{2}$$

The exit velocity of a diffuser is usually small compared with the inlet velocity ($V_2 \ll V_1$); thus, the kinetic energy at the exit can be neglected. The enthalpy of air at the diffuser inlet is determined from the air table (Table A–17) to be

$$h_1 = h_{@\ 283\ \text{K}} = 283.14\ \text{kJ/kg}$$

Substituting, we get

$$h_2 = 283.14\ \text{kJ/kg} - \frac{0 - (200\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

$$= 303.14\ \text{kJ/kg}$$

From Table A–17, the temperature corresponding to this enthalpy value is

$$T_2 = \mathbf{303\ K}$$

Discussion This result shows that the temperature of the air increases by about 20°C as it is slowed down in the diffuser. The temperature rise of the air is mainly due to the conversion of kinetic energy to internal energy.

Example 3: Acceleration of Steam in a Nozzle

Steam at 1.8Mpa and 350℃ steadily enters a nozzle whose inlet area is 0.02 m^2. The mass flow rate of the steam through the nozzle is 5.2 kg/s. Steam leaves the nozzle at 1.2Mpa with a velocity of 300m/s. The heat losses from the nozzle per unit mass of the steam are estimated to be 2.8 KJ/kg.

Determine:

(a) the inlet velocity and

(b) the exit temperature of the steam.

Answers: ***a) 40 m/s,***
b) ____℃

8. Given

- Steam
- $P_1 = 1.8\,MPa$
- $T_1 = 350^\circ C$
- Nozzle
- $A_1 = 0.02\,m^2$
- $\vec{V}_1 = 5.2\,m/s$
- $P_2 = 1.2\,MPa$
- $\vec{V}_2 = 300\,m/s$
- $\theta_{out} = 2.8\,kJ/kg$

Required

(a) $\vec{V}_1 = ?$

(b) $T_2 = ?$

Solution

Assumptions

- Nozzle is steady state steady flow device
- Single inlet & single exit
- No work interactions with the surrounding ($\dot{W}_{in} = \dot{W}_{out} = 0$)
- No heat input ($\dot{Q}_{in} = 0$)
- No change in potential energy since the elevation difference is negligible ($\Delta PE \cong 0$)

Analysis

(1) mass balance

$$\sum \dot{m}_i - \sum \dot{m}_e = \frac{dm_{cv}}{dt}^{0 \text{ SSSF}}$$

$$\Rightarrow \sum \dot{m}_i = \sum \dot{m}_e$$

$$\Rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}$$

(2) energy balance

$$\dot{E}_{in} - \dot{E}_{out} = \frac{dE_{cv}}{dt}^{0}$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\Rightarrow \dot{Q}_{in} + \dot{W}_{in} + \sum \dot{m}_i \left(h + \frac{V^2}{2} + gz\right)_i = \dot{Q}_{out} + \dot{W}_{out} + \sum \dot{m}_e \left(h + \frac{V^2}{2} + gz\right)_e$$

$$\Rightarrow \sum \dot{m}_i \left(h + \frac{V^2}{2}\right)_i = \dot{Q}_{out} + \sum \dot{m}_e \left(h + \frac{V^2}{2}\right)_e$$

$$\Rightarrow \dot{m}_i \left(h_1 + \frac{V_1^2}{2}\right) = \dot{Q}_{out} + \dot{m}_e \left(h_2 + \frac{V_2^2}{2}\right)_e \quad (1)$$

(a) From the mass balance

$$\dot{m}_1 = \dot{m}_2 = \dot{m} = \rho_1 A_1 \vec{V}_1 = \rho_2 A_2 \vec{V}_2$$

$$\Rightarrow \dot{m} = \rho_1 A_1 \vec{V}_1 = \frac{A_1 \vec{V}_1}{v_1}$$

$$\Rightarrow \vec{V}_2 = \frac{\dot{m} v_2}{A_2}$$

State 1 — Table A-6

$P_1 = 1.8\,MPa$, $T_1 = 350^\circ C$ $\Rightarrow$ $h_1 = 3141.9\,kJ/kg$, $v_1 = 0.1546\,m^3/kg$

$$\Rightarrow \vec{V}_2 = \frac{5.2\,m/s \times 0.1546\,m^3/kg}{0.02\,m^2}$$

$$\Rightarrow \vec{V}_2 = 40.2\,m/s$$

(b) From energy balance

$$\dot{m} h_1 + \dot{m} \frac{\vec{V}_1^2}{2} = \dot{Q}_{out} + \dot{m} h_2 + \dot{m} \frac{\vec{V}_2^2}{2}$$

$$\Rightarrow h_2 = \frac{\left(\dot{m} h_1 + \frac{\dot{m} \vec{V}_1^2}{2(1000)} - \dot{Q}_{out} - \frac{\dot{m} \vec{V}_2^2}{2(1000)}\right) kJ/s}{(\dot{m})\,kg/s}$$

$$\Rightarrow h_2 = \frac{\left[5.2(3141.9) + \frac{(40.2)^2}{2000} - 2.8 - \frac{(5.2)(300)^2}{2000}\right] \cdot \frac{kJ}{kg}}{5.2}$$

$$\Rightarrow h_2 = 3035.55\,kJ/kg$$

State 2 — Table A-6

$P_2 = 1.2\,MPa$, $h_2 = 3035.55\,kJ/kg$ $\Rightarrow$ $T_2 = 322.42^\circ C$

After interpolation

Example (4-12) Power Generation by a Steam Turbine

The power output of an adiabatic steam turbine is 5 MW, and the inlet and the exit conditions of the steam are as indicated in the figure on the right.

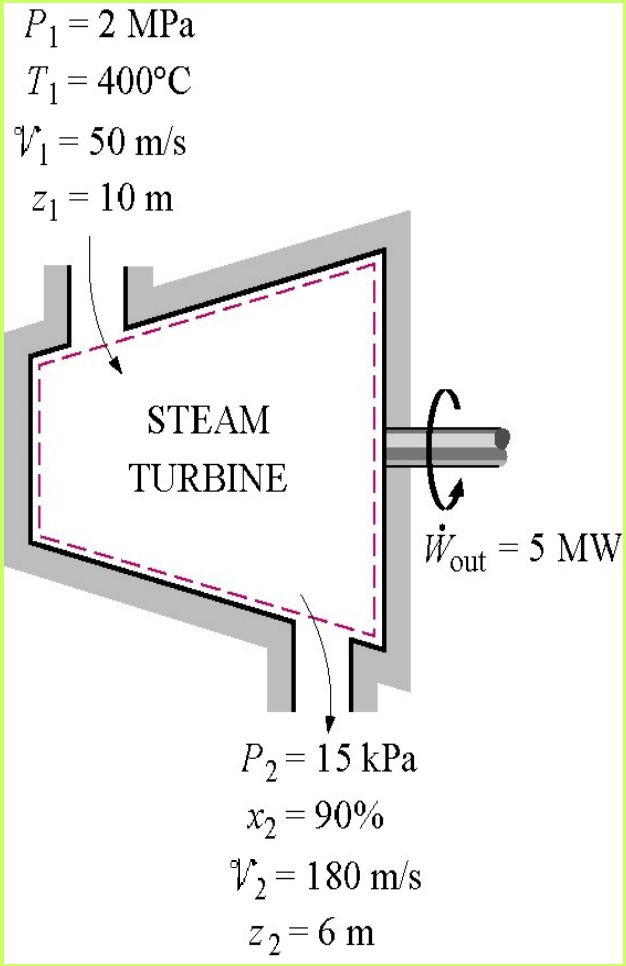
a) Compare the magnitude of Δh, Δke, and Δpe.

b) Determine the work done per unit mass of the steam flowing through the turbine.

c) Calculate the mass flow rate of the steam.

Answers: a) Δh = -885.9 kJ/kg,

Δke = 14.95 kJ/kg, Δpe = -0.04 kJ/kg,



Solution The inlet and exit conditions of a steam turbine and its power output are given. The changes in kinetic energy, potential energy, and enthalpy of steam, as well as the work done per unit mass and the mass flow rate of steam are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The system is adiabatic and thus there is no heat transfer.

Analysis We take the *turbine* as the system. This is a *control volume* since mass crosses the system boundary during the process. We observe that there is only one inlet and one exit and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Also, work is done by the system. The inlet and exit velocities and elevations are given, and thus the kinetic and potential energies are to be considered.

(*a*) At the inlet, steam is in a superheated vapor state, and its enthalpy is

$$\left.\begin{aligned} P_1 &= 2\ \text{MPa} \\ T_1 &= 400^\circ\text{C} \end{aligned}\right\} \quad h_1 = 3248.4\ \text{kJ/kg} \qquad \text{(Table A–6)}$$

At the turbine exit, we obviously have a saturated liquid–vapor mixture at 15-kPa pressure. The enthalpy at this state is

$$h_2 = h_f + x_2 h_{fg} = [225.94 + (0.9)(2372.3)]\ \text{kJ/kg} = 2361.01\ \text{kJ/kg}$$

Then

$$\Delta h = h_2 - h_1 = (2361.01 - 3248.4)\ \text{kJ/kg} = \mathbf{-887.39\ kJ/kg}$$

$$\Delta ke = \frac{V_2^2 - V_1^2}{2} = \frac{(180\ \text{m/s})^2 - (50\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{14.95\ kJ/kg}$$

$$\Delta pe = g(z_2 - z_1) = (9.81\ \text{m/s}^2)[(6 - 10)\ \text{m}]\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{-0.04\ kJ/kg}$$

(*b*) The energy balance for this steady-flow system can be expressed in the rate form as

$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{dE_{system}/dt \overset{0\ \text{(steady)}}{\nearrow}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc., energies}}} = 0$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2} + gz_1\right) = \dot{W}_{out} + \dot{m}\left(h_2 + \frac{V_2^2}{2} + gz_2\right) \qquad (\text{since } \dot{Q} = 0)$$

Dividing by the mass flow rate $\dot{m}$ and substituting, the work done by the turbine per unit mass of the steam is determined to be

$$w_{out} = -\left[(h_2 - h_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)\right] = -(\Delta h + \Delta ke + \Delta pe)$$

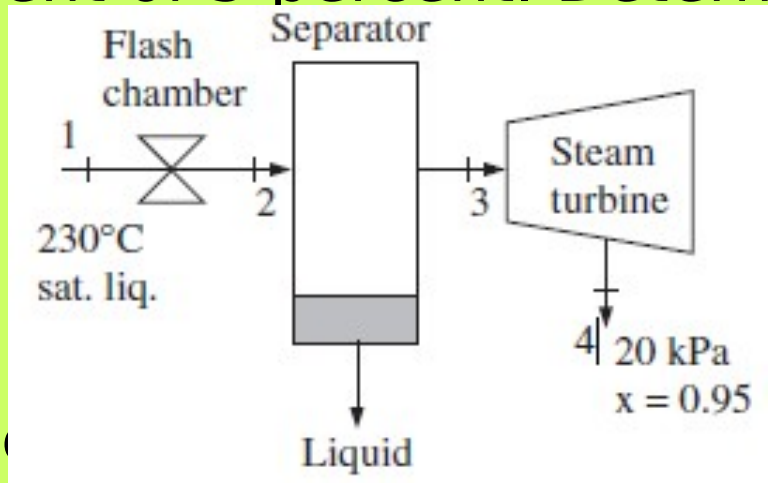
$$= -[-887.39 + 14.95 - 0.04]\ \text{kJ/kg} = \mathbf{872.48\ kJ/kg}$$

(*c*) The required mass flow rate for a 5-MW power output is

$$\dot{m} = \frac{\dot{W}_{out}}{w_{out}} = \frac{5000 \text{ kJ/s}}{872.48 \text{ kJ/kg}} = \mathbf{5.73 \text{ kg/s}}$$

Discussion Two observations can be made from these results. First, the change in potential energy is insignificant in comparison to the changes in enthalpy and kinetic energy. This is typical for most engineering devices. Second, as a result of low pressure and thus high specific volume, the steam velocity at the turbine exit can be very high. Yet the change in kinetic energy is a small fraction of the change in enthalpy (less than 2 percent in our case) and is therefore often neglected.

In a single-flash geothermal power plant, geothermal **.#5** water enters the flash chamber (a throttling valve) at 230°C as a saturated liquid at a rate of 50 kg/s. The steam resulting from the flashing process enters a turbine and leaves at 20kPa with a moisture content of 5 percent. Determine i) the temperature of the steam after d ii) the power output from the turbine if th at the exit of the



5.4 Given

- $T_1 = 230°C$
- Sat. liquid
- $\dot{m}_1 = 50 kg/s$
- $P_4 = 20 kPa$
- $x_4 = 0.95$

Required

1) $T_2 = ?$
2) $\dot{W}_{out} = ?$ for
   a) $P_2 = 2 MPa$
   b) $P_2 = 500 kPa$
   c) $P_2 = 100 kPa$
   d) $P_2 = 50 kPa$

Solution

Assumptions

- Flash chamber, separator & steam turbine are steady state, steady flow devices
- All devices lose no heat interactions with the surroundings ($\dot{E}_{in} - \dot{E}_{out} = 0$)
- NO change in kinetic & potential energies in the three devices ($\Delta KE = \Delta PE = 0$)

Flash chamber

- Single inlet & single exit device
- NO work interactions with the surroundings ($\dot{W}_{in} = \dot{W}_{out} = 0$)

Separator (opposite of mixing chamber)

- Single inlet & two exits
- NO work interactions with the surroundings ($\dot{W}_{in} = \dot{W}_{out} = 0$)

Steam Turbine

- Single inlet & single exit
- NO work- Irreversible ($\dot{W}_{in} = 0$) but energy is leave out but since it is the work produced device

Analysis

① Flash chamber

$\dot{m}_3 = \dot{m}_2$

- With the assumptions above, the energy balance reduces to

$h_1 = h_2$

Solved

$T_1 = 230°C$, sat. liquid $\Rightarrow h_1 = h_{@T_1} = 990.14 kJ/kg = h_2$ (Table A-4)

Therefore, we have two properties for state 2.

(a) For $P_2 = 1 MPa = 1000 kPa$

State 2: $P_2 = 1 MPa$, $h_2 = h_1 = 990.14 kJ/kg$ (Table A-5)

$h_{f2} = 762.51 kJ/kg$
$h_{g2} = 2777.1 kJ/kg$
$h_{fg2} = 2014.6 kJ/kg$

Since $h_{f2} < h_2 < h_{g2}$ it is Saturated liquid-vapor mixture.

Hence, $T_2 = T_{sat@P_2} = 179.88°C$

② Separator

① mass balance

$\dot{m}_2 = \dot{m}_3 + \dot{m}_4$; where $\dot{m}_3$ is mass of sat. vapor, $\dot{m}_4$ is mass of sat. liquid

$\Rightarrow \dot{m}_4 = \dot{m}_2 - \dot{m}_3$

② Energy balance

- Using the assumptions above for separator we have:

$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e$

$\Rightarrow \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_4 h_4$

$\Rightarrow \boxed{\dot{m}_2 h_2 = \dot{m}_3 h_3 + (\dot{m}_2 - \dot{m}_3) h_4}$ - eqn(1)

$\dot{m}_1 = \dot{m}_2 = 50 kg/s$

(b) For $P_2 = 1 MPa = 1000 kPa$ & $P_3 = P_2$

$h_2 = h_1 = 990.14 kJ/kg$ (Isenthalpic process)

$h_3 = h_{@1MPa} = 2777.1 kJ/kg$ = sat. vapor at $P = 1 MPa$

$h_4 = h_{@1MPa} = 762.51 kJ/kg$ = sat. liquid at $P = 1 MPa$

Substituting values in eqn (2), we will get:

$$50\frac{kg}{s} \times 990.14\frac{kJ}{kg} = \dot{m}_3(h_3 - h_4) + \dot{m}_2 h_4$$

$$\Rightarrow 50\frac{kg}{s} \times 990.14\frac{kJ}{kg} = \dot{m}_3(2777.1 - 762.5)\frac{kJ}{kg} + 50\frac{kg}{s} \times 762.5\frac{kJ}{kg}$$

$$\Rightarrow \dot{m}_3 = 5.65\frac{kg}{s}$$

③ Steam Turbine

① mass balance

$\dot{m}_3 = \dot{m}_4 = 5.65\frac{kg}{s}$ if $P_2 = P_3 = 1MPa$

② Energy balance

- using the assumptions above for steam turbine:

$$\dot{W}_{out} = \dot{m}_3(h_3 - h_4)$$

If $P_4 = P_3 = 1MPa = 1000kPa$, then

$h_3 = h_{@1MPa} = 2777.1\ kJ/kg$

State 4 — Table A-5

$P_4 = 20kPa$ } $h_{f4} = 251.42\ kJ/kg$

$x_4 = 0.95$ } $h_{fg4} = 2357.5\ kJ/kg$

$\Rightarrow h_4 = h_{f4} + x_4 h_{fg4}$

$\Rightarrow h_4 = 251.42\frac{kJ}{kg} + 0.95 \times 2357.5\frac{kJ}{kg}$

$\Rightarrow h_4 = 2491\ kJ/kg$

Hence, the power output is:

$\dot{W}_{out} = 5.65\frac{kg}{s} \times (2777.1 - 2491)\frac{kJ}{kg}$

$\Rightarrow \dot{W}_{out} = 1616\ kW$ Ans

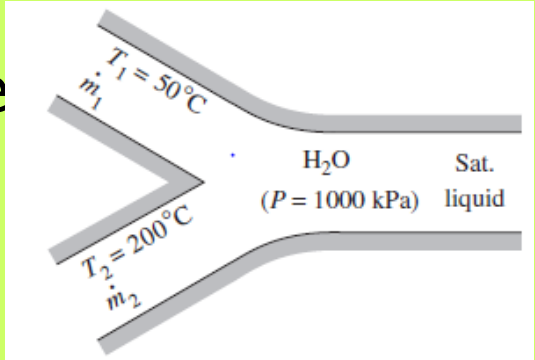
Repeating the similar calculations for other pressures, we obtain:

(b) For $P_2 = 300kPa$, $T_2 = 133.52^\circ C$, $\dot{W}_{out} = 2184kW$

(c) For $P_2 = 100kPa$, $T_2 = 99.6^\circ C$, $\dot{W}_{out} = 2333kW$

(d) For $P_2 = 50kPa$, $T_2 = 81.3^\circ C$, $\dot{W}_{out} = 2495kW$

In steam power plants, open feed-water heaters are .#6 frequently utilized to heat the feed-water by mixing it with steam bled off the turbine at some intermediate stage. Consider an open feed-water heater that operates at a pressure of 1000kPa. Feed-water at 50°C and 1000kPa is to be heated with superheated steam at 200°C and 1000kPa. In an ideal feed-water heater, the mixture leaves the heater as saturated liquid at the feed-water pressure. Determine the ratio of the mass flow rates of the fee superheated vapor .for this case



BHC

Given

- Steam is the working fluid
- $P_1 = P_2 = P_3 = P = 1000 kPa$
- $T_1 = 50^\circ C$
- $T_2 = 200^\circ C$
- State 3 is sat. liquid

Required

$\frac{\dot{m}_1}{\dot{m}_2} = ?$

Solution

Assumptions

- Open - feed-water heater is the mixing chamber & steady state steady flow device.
- Two inlets & single exit
- No heat interactions with the surroundings ($\dot{Q}_{in} = \dot{Q}_{out} = 0$)
- No work interactions with the surroundings ($\dot{W}_{in} = \dot{W}_{out} = 0$)
- Change in kinetic & potential energies are negligible ($\Delta KE = \Delta PE = 0$)

Analysis

(1) mass balance

$\sum \dot{m}_i - \sum \dot{m}_e = \frac{dm_{cv}}{dt}^{\nearrow 0 \text{ SSSF}}$

$\Rightarrow \sum \dot{m}_i = \sum \dot{m}_e$

$\Rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3 \quad \cdots (1)$

(2) Energy balance

$\dot{E}_{in} - \dot{E}_{out} = \frac{dE_{cv}}{dt}^{\nearrow 0 \text{ SSSF}}$

$\Rightarrow \dot{E}_{in} = \dot{E}_{out}$

$\dot{m}_1 h_1^{\nearrow 0} + \dot{m}_2 h_2^{\nearrow 0} + \sum \dot{m}_i \left(h + \frac{V^2}{2}^{\nearrow 0} + gz^{\nearrow 0}\right)_i = \dot{m}_3 h_3^{\nearrow 0} + \sum \dot{m}_e \left(h + \frac{V^2}{2}^{\nearrow 0} + gz^{\nearrow 0}\right)_e$

$\Rightarrow \sum \dot{m}_i h_i = \sum \dot{m}_e h_e$

$\Rightarrow \dot{m}_1 h_1 + \dot{m}_2 h_2 = \dot{m}_3 h_3$

$\Rightarrow \dot{m}_1 h_1 + \dot{m}_2 h_2 = (\dot{m}_1 + \dot{m}_2) h_3$

$\Rightarrow \dot{m}_1 (h_1 - h_3) = \dot{m}_2 (h_3 - h_2)$

$\Rightarrow \boxed{\frac{\dot{m}_1}{\dot{m}_2} = \frac{h_3 - h_2}{h_1 - h_3}}$

State 1

$P_1 = 1000 kPa$, $T_1 = 50^\circ C$ } since $T_1 < T_{sat}$ at 1 MPa, $T_{sat} = 179.9^\circ C$, i.e. it is compressed liquid

$\Rightarrow h_1 = h_{@T=50^\circ C} = 209.34 \text{ kJ/kg}$

Inlet State 2

$P_2 = 1000 kPa$, $T_2 = 200^\circ C$ } $T_2 > T_{sat}$ = superheated vapor

$h_2 = 2828.3 \text{ kJ/kg}$

Exit State 3

$P_3 = 1000 kPa$, sat. liquid } $h_3 = h_{@P_3} = 762.51 \text{ kJ/kg}$

Therefore,

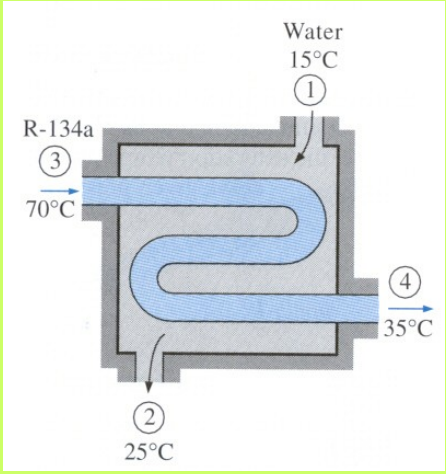
$\frac{\dot{m}_1}{\dot{m}_2} = \frac{762.51 \text{ kJ/kg} - 2828.3 \text{ kJ/kg}}{209.34 \text{ kJ/kg} - 762.51 \text{ kJ/kg}}$

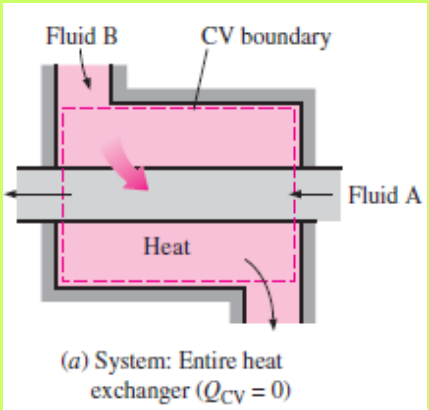
$\Rightarrow \frac{\dot{m}_1}{\dot{m}_2} = 3.734$

Example: Cooling of Refrigerant-134a by Water

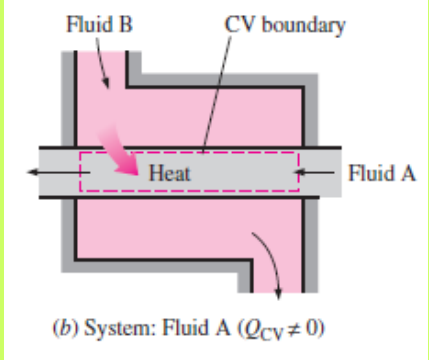
Refrigerant-134a is to be cooled by water in a condenser. The refrigerant enters the condenser with a mass flow rate of 6 kg/min at 1 MPa and 70°C and leaves at 34°C. The cooling water enters at 300 kPa and 15°C and leaves at 25°C. Neglecting any pressure drop, determine

(a) the mass flow rate of the cooling water required and (b) the heat transfer rate from the refrigerant to water.





(a) System: Entire heat exchanger ($Q_{CV} = 0$)



(b) System: Fluid A ($Q_{CV} \neq 0$)

Solution Refrigerant-134a is cooled by water in a condenser. The mass flow rate of the cooling water and the rate of heat transfer from the refrigerant to the water are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time at any point and thus $\Delta m_{CV} = 0$ and $\Delta E_{CV} = 0$. **2** The kinetic and potential energies are negligible, ke $\cong$ pe $\cong$ 0. **3** Heat losses from the system are negligible and thus $\dot{Q} \cong 0$. **4** There is no work interaction.

Analysis We take the *entire heat exchanger* as the system (Fig. 5–37). This is a *control volume* since mass crosses the system boundary during the process. In general, there are several possibilities for selecting the control volume for multiple-stream steady-flow devices, and the proper choice depends on the situation at hand. We observe that there are two fluid streams (and thus two inlets and two exits) but no mixing.

(*a*) Under the stated assumptions and observations, the mass and energy balances for this steady-flow system can be expressed in the rate form as follows:

Mass balance:
$$\dot{m}_{in} = \dot{m}_{out}$$

for each fluid stream since there is no mixing. Thus,

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_w$$
$$\dot{m}_3 = \dot{m}_4 = \dot{m}_R$$

Energy balance:
$$\underbrace{\dot{E}_{in} - \dot{E}_{out}}_{\substack{\text{Rate of net energy transfer} \\ \text{by heat, work, and mass}}} = \underbrace{dE_{system}/dt \nearrow^{0\ (\text{steady})} = 0}_{\substack{\text{Rate of change in internal, kinetic,} \\ \text{potential, etc., energies}}}$$

$$\dot{E}_{in} = \dot{E}_{out}$$

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \qquad (\text{since } \dot{Q} \cong 0, \dot{W} = 0, \text{ke} \cong \text{pe} \cong 0)$$

Combining the mass and energy balances and rearranging give

$$\dot{m}_w(h_1 - h_2) = \dot{m}_R(h_4 - h_3)$$

Now we need to determine the enthalpies at all four states. Water exists as a compressed liquid at both the inlet and the exit since the temperatures at both locations are below the saturation temperature of water at 300 kPa (133.52°C). Approximating the compressed liquid as a saturated liquid at the given temperatures, we have

$$\begin{aligned} h_1 &\cong h_{f\,@\,15^\circ\text{C}} = 62.982 \text{ kJ/kg} \\ h_2 &\cong h_{f\,@\,25^\circ\text{C}} = 104.83 \text{ kJ/kg} \end{aligned} \qquad \text{(Table A–4)}$$

The refrigerant enters the condenser as a superheated vapor and leaves as a compressed liquid at 35°C. From refrigerant-134a tables,

$$\left.\begin{aligned} P_3 &= 1 \text{ MPa} \\ T_3 &= 70^\circ\text{C} \end{aligned}\right\} \quad h_3 = 303.85 \text{ kJ/kg} \qquad \text{(Table A–13)}$$

$$\left.\begin{aligned} P_4 &= 1 \text{ MPa} \\ T_4 &= 35^\circ\text{C} \end{aligned}\right\} \quad h_4 \cong h_{f\,@\,35^\circ\text{C}} = 100.87 \text{ kJ/kg} \qquad \text{(Table A–11)}$$

Substituting, we find

$$\dot{m}_w(62.982 - 104.83)\ \text{kJ/kg} = (6\ \text{kg/min})[(100.87 - 303.85)\ \text{kJ/kg}]$$

$$\dot{m}_w = \mathbf{29.1\ kg/min}$$

(*b*) To determine the heat transfer from the refrigerant to the water, we have to choose a control volume whose boundary lies on the path of heat transfer. We can choose the volume occupied by either fluid as our control volume. For no particular reason, we choose the volume occupied by the water. All the assumptions stated earlier apply, except that the heat transfer is no longer zero. Then assuming heat to be transferred to water, the energy balance for this single-stream steady-flow system reduces to

$$\underbrace{\dot{E}_{\text{in}} - \dot{E}_{\text{out}}}_{\substack{\text{Rate of net energy transfer}\\ \text{by heat, work, and mass}}} = \underbrace{dE_{\text{system}}/dt \overset{0\ (\text{steady})}{\nearrow}}_{\substack{\text{Rate of change in internal, kinetic,}\\ \text{potential, etc., energies}}} = 0$$

$$\dot{E}_{\text{in}} = \dot{E}_{\text{out}}$$

$$\dot{Q}_{w,\text{in}} + \dot{m}_w h_1 = \dot{m}_w h_2$$

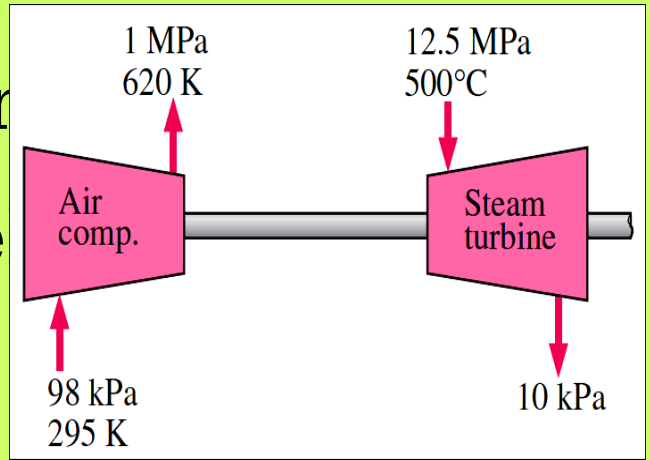
Rearranging and substituting,

$$\dot{Q}_{w,\text{in}} = \dot{m}_w(h_2 - h_1) = (29.1\ \text{kg/min})[(104.83 - 62.982)\ \text{kJ/kg}]$$

$$= \mathbf{1218\ kJ/min}$$

Discussion Had we chosen the volume occupied by the refrigerant as the control volume (Fig. 5–38), we would have obtained the same result for $\dot{Q}_{R,\text{out}}$ since the heat gained by the water is equal to the heat lost by the refrigerant.

An adiabatic air compressor is to be .#8 powered by a direct-coupled adiabatic steam turbine that is also driving a generator. Steam enters the turbine at 12.5MPa and 500°C at a rate of 25 kg/s and exits at 10kPa and a quality of 0.92. Air enters the compressor at 98kPa and 295 K at a rate of 10 kg/s and exits at 1MPa ar Determine the net power de .generator by the turbine



Q#2

Given

- Adiabatic air compressor
- Adiabatic steam turbine

Required

$\dot{m}_2 = ?$

* Steam Turbine

$P_1 = 12.5\,MPa$

$T_1 = 500^\circ C$

$P_2 = 10\,kPa$

$x_2 = 0.92$

$\dot{m}_s = 25\,kg/s$

* Air compressor

$P_1 = 92\,kPa$

$T_1 = 295\,K$

$\dot{m}_a = 10\,kg/s$

$P_2 = 1\,MPa$

$T_2 = 620\,K$

Solution

Assumptions

- Both compressors & Turbines are steady state steady flow devices. ($\frac{dm_{cv}}{dt} = \frac{dE_{cv}}{dt} = 0$)
- Both have 1 inlet & 1 exit
- The change in kinetic & potential energies for both is negligible. ($\Delta KE = \Delta PE \approx 0$)
- Both are adiabatic & no heat interactions with the surroundings ($\dot{Q}_{in} = \dot{Q}_{out} = 0$)
- NO work output for the Turbine ($\dot{W}_{in} = 0$)
- NO work input for the compressor ($\dot{W}_{out} = 0$)
- Air is an ideal gas.

Analysis

(1) mass balance

$\Sigma \dot{m}_{in} - \Sigma \dot{m}_{out} = \frac{dm_{cv}}{dt} = 0$ (SSSF)

$\Rightarrow \Sigma \dot{m}_{in} = \Sigma \dot{m}_{out}$

$\Rightarrow \dot{m}_1 = \dot{m}_2 = \dot{m}$ for both

(2) Energy balance

- Taking the above assumptions

(a) For compressor

$\dot{W}_{in} = \dot{m}_a (h_2 - h_1)_c$

State 1 — Table A-17

$T_1 = 295\,K \Rightarrow h_1 = 295.17\,kJ/kg$

State 2

$T_2 = 620\,mPa \Rightarrow h_2 = 628.07\,kJ/kg$

$\Rightarrow \dot{W}_{in} = 10\,kg/s \times (628.07 - 295.17)\,kJ/kg$

$\Rightarrow \dot{W}_{in} = 3329\,kW$

(b) For Turbine

$\dot{W}_{out} = \dot{m}_s (h_1 - h_2)_T$

State 1 — Table A-6

$P_1 = 12.5\,MPa$, $T_1 = 500^\circ C \Rightarrow h_1 = 3343.6\,kJ/kg$

State 2 — Table A-5

$P_2 = 10\,kPa \Rightarrow h_{f2} = 191.81\,kJ/kg$, $h_{fg2} = 2392.1\,kJ/kg$

$x_2 = 0.92$

$\Rightarrow h_2 = h_{f2} + x_2 h_{fg2} = 2392.54\,kJ/kg$

$\Rightarrow \dot{W}_{out} = 25\,kg/s \times (3343.6 - 2392.54)\,kJ/kg$

$\Rightarrow \dot{W}_{out} = 23,796.45\,kW$

Therefore,

$\dot{W}_c = (\dot{W}_{out})_{Turbine} - (\dot{W}_{in})_{compressor}$

$\Rightarrow \dot{W}_c = 20,467.45\,kW$ or $20.45\,MW$