ESO 201A: Thermodynamics

2016-2017-I semester

Mass-Energy Analysis: part 4

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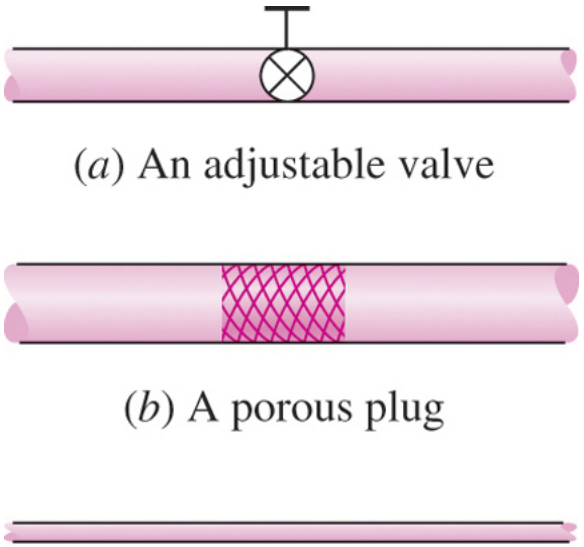
Learning objectives

- Develop the conservation of mass principle.
- Apply the conservation of mass principle to various systems including steady- and unsteady-flow control volumes.
- Apply the first law of thermodynamics as the statement of the conservation of energy principle to control volumes.
- Identify the energy carried by a fluid stream crossing a control surface as the sum of internal energy, flow work, kinetic energy, and potential energy of the fluid and to relate the combination of the internal energy and the flow work to the property enthalpy.
- Solve energy balance problems for common steady-flow devices such as nozzles, compressors, turbines, throttling valves, mixers, heaters, and heat exchangers.
- Apply the energy balance to general unsteady-flow processes with particular emphasis on the uniform-flow process as the model for commonly encountered charging and discharging processes.

Throttling valves



(*a*) An adjustable valve

(*b*) A porous plug

(*c*) A capillary tube

Throttling valves are *any kind of flow-restricting devices* that cause a significant pressure drop in the fluid.

What is the difference between a turbine and a throttling valve?

The pressure drop in the fluid is often accompanied by a *large drop in temperature*, and for that reason throttling devices are commonly used in refrigeration and air-conditioning applications.

Throttling valves

$$q \approx 0$$

$$w = 0$$

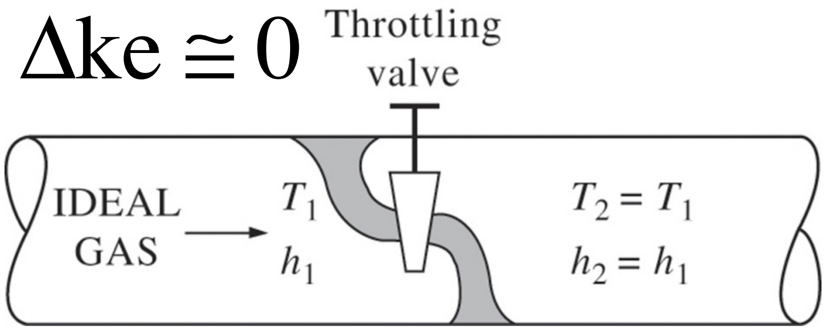
$$\Delta \text{pe} \cong 0$$

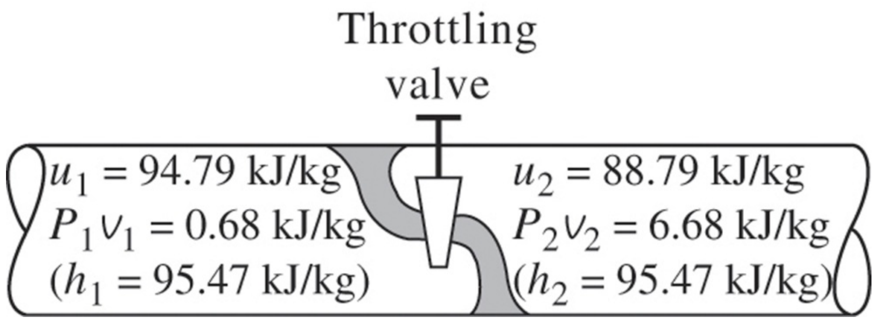
$$\Delta \text{ke} \cong 0$$

$$h_2 \cong h_1 \quad \text{(kJ/kg)}$$

$$u_1 + P_1 v_1 = u_2 + P_2 v_2$$

Internal energy + Flow energy = Constant



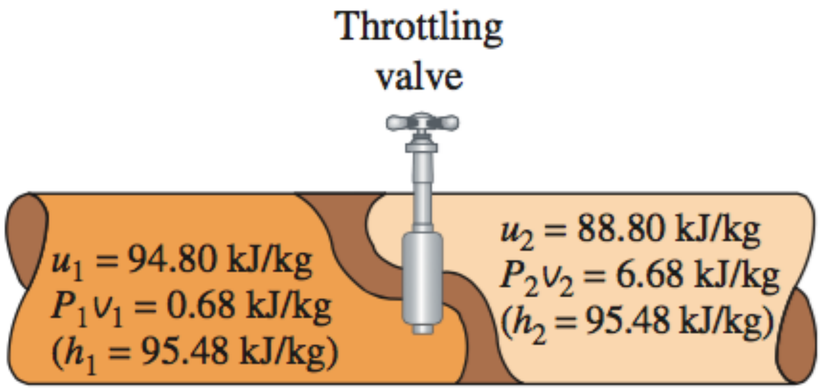


The temperature of an ideal gas does not change during a throttling (h = constant) process since $h = h(T)$.

During a throttling process, the enthalpy of a fluid remains constant. But internal and flow energies may be converted to each other.

Example

Refrigerant-134a enters the capillary tube of a refrigerator as saturated liquid at 0.8 MPa and is throttled to a pressure of 0.12 MPa. Determine the quality of the refrigerant at the final state and the temperature drop during this process.



Example

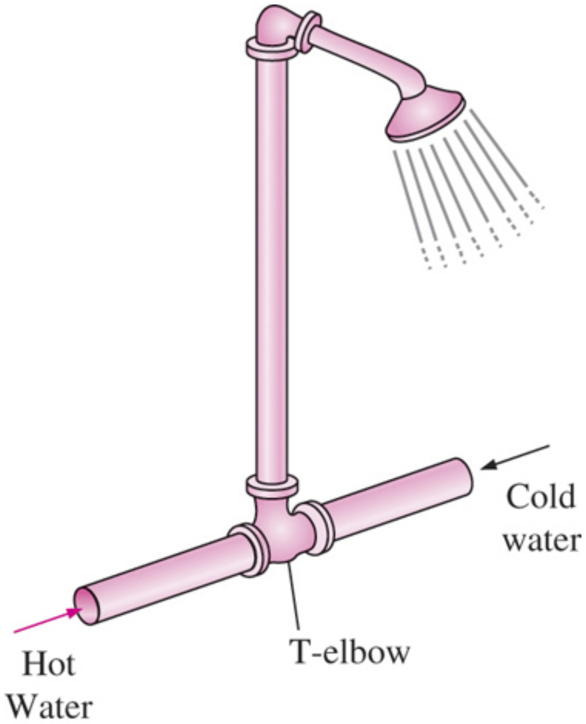
TABLE A–12

Saturated refrigerant-134a—Pressure table

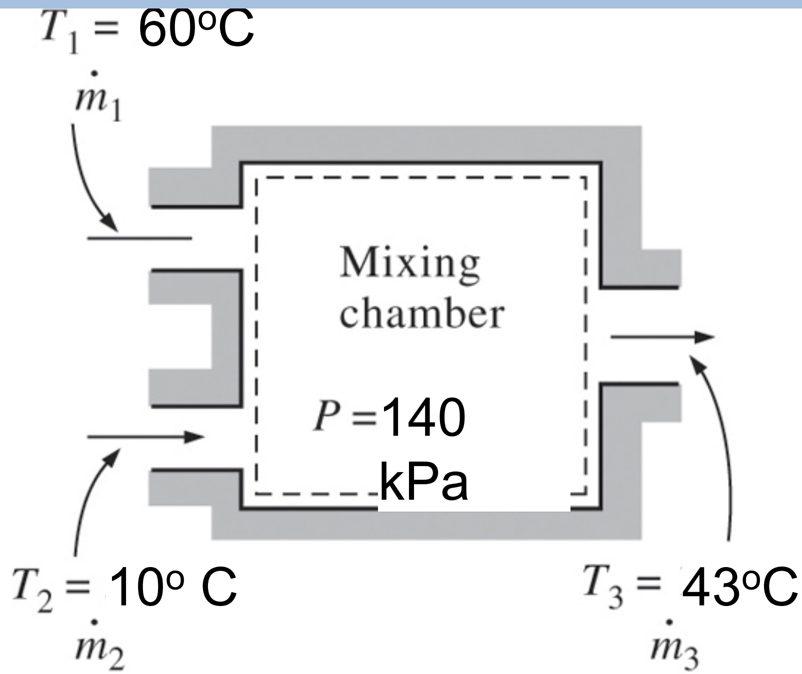
		Specific volume, m^3/kg		Internal energy, kJ/kg			Enthalpy, kJ/kg		
Press., P kPa	Sat. temp., T_{sat} °C	Sat. liquid, v_f	Sat. vapor, v_g	Sat. liquid, u_f	Evap., u_{fg}	Sat. vapor, u_g	Sat. liquid, h_f	Evap., h_{fg}	Sat. vapor, h_g
60	−36.95	0.0007097	0.31108	3.795	205.34	209.13	3.837	223.96	227.80
70	−33.87	0.0007143	0.26921	7.672	203.23	210.90	7.722	222.02	229.74
80	−31.13	0.0007184	0.23749	11.14	201.33	212.48	11.20	220.27	231.47
90	−28.65	0.0007222	0.21261	14.30	199.60	213.90	14.36	218.67	233.04
100	−26.37	0.0007258	0.19255	17.19	198.01	215.21	17.27	217.19	234.46
120	−22.32	0.0007323	0.16216	22.38	195.15	217.53	22.47	214.52	236.99
140	−18.77	0.0007381	0.14020	26.96	192.60	219.56	27.06	212.13	239.19
160	−15.60	0.0007435	0.12355	31.06	190.31	221.37	31.18	209.96	241.14
180	−12.73	0.0007485	0.11049	34.81	188.20	223.01	34.94	207.95	242.90
200	−10.09	0.0007532	0.099951	38.26	186.25	224.51	38.41	206.09	244.50
240	−5.38	0.0007618	0.083983	44.46	182.71	227.17	44.64	202.68	247.32
280	−1.25	0.0007697	0.072434	49.95	179.54	229.49	50.16	199.61	249.77
320	2.46	0.0007771	0.063681	54.90	176.65	231.55	55.14	196.78	251.93
360	5.82	0.0007840	0.056809	59.42	173.99	233.41	59.70	194.15	253.86
400	8.91	0.0007905	0.051266	63.61	171.49	235.10	63.92	191.68	255.61
450	12.46	0.0007983	0.045677	68.44	168.58	237.03	68.80	188.78	257.58
500	15.71	0.0008058	0.041168	72.92	165.86	238.77	73.32	186.04	259.36
550	18.73	0.0008129	0.037452	77.09	163.29	240.38	77.54	183.44	260.98
600	21.55	0.0008198	0.034335	81.01	160.84	241.86	81.50	180.95	262.46
650	24.20	0.0008265	0.031680	84.72	158.51	243.23	85.26	178.56	263.82
700	26.69	0.0008331	0.029392	88.24	156.27	244.51	88.82	176.26	265.08
750	29.06	0.0008395	0.027398	91.59	154.11	245.70	92.22	174.03	266.25
800	31.31	0.0008457	0.025645	94.80	152.02	246.82	95.48	171.86	267.34

Mixing chamber

In engineering applications, the section where the mixing process takes place is commonly referred to as a **mixing chamber**.



The T-elbow of an ordinary shower serves as the mixing chamber for the hot- and the cold-water streams.



$$\dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + \dot{m}_2 = \dot{m}_3$$

Energy balance for the adiabatic mixing chamber in the figure is:

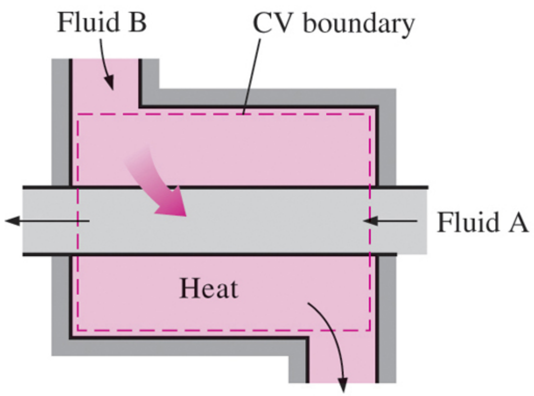
$$\dot{E}_{in} = \dot{E}_{out}$$

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

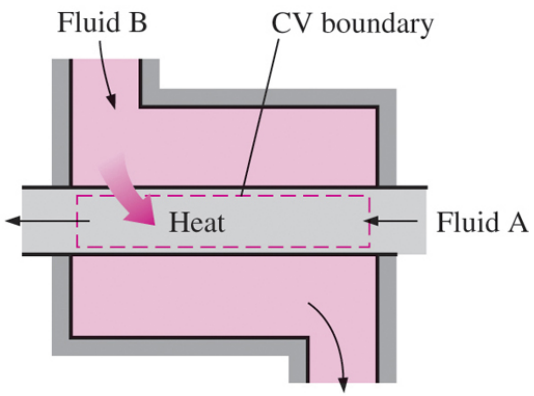
$$(\text{since } \dot{Q} \cong 0,\ \dot{W} = 0,\ \text{ke} \cong \text{pe} \cong 0)$$

Heat exchanger

Heat exchangers are devices where two moving fluid streams exchange heat without mixing. Heat exchangers are widely used in various industries, and they come in various designs.

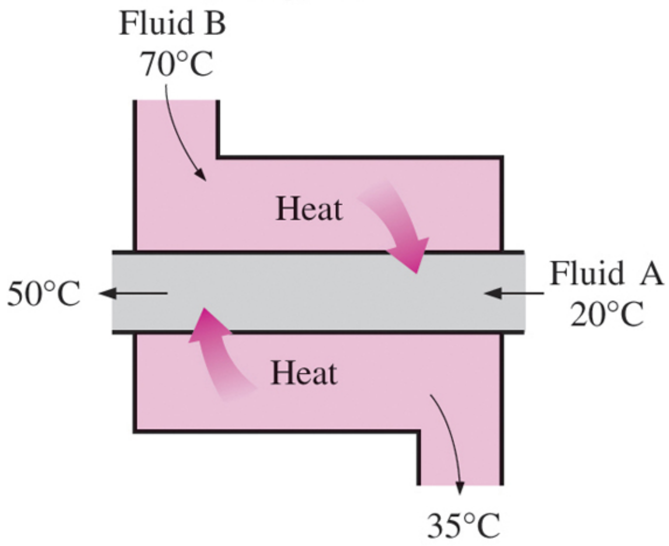


(*a*) System: Entire heat exchanger ($Q_{CV} = 0$)



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

The heat transfer associated with a heat exchanger may be zero or nonzero depending on how the control volume is selected.



A heat exchanger can be as simple as two concentric pipes.

Example

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 35°C. The cooling water enters at 300 kPa and 15°C and leaves at 25°C. Neglecting any pressure drops, determine (a) the mass flow rate of the cooling water required

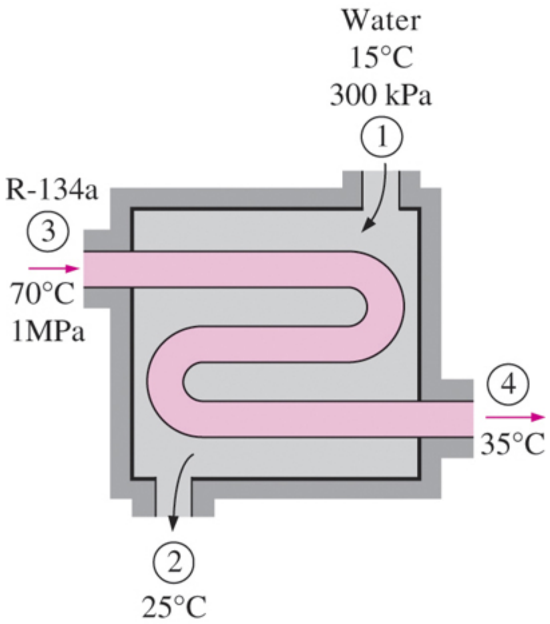
Mass and energy balances for the adiabatic heat exchanger in the figure is:

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

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_R$$

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

$$\dot{m}_1 h_1 + \dot{m}_3 h_3 = \dot{m}_2 h_2 + \dot{m}_4 h_4$$



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

Example

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

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

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

$$\left.\begin{aligned} P_3 &= 1\ \mathrm{MPa} \\ T_3 &= 70^\circ\mathrm{C} \end{aligned}\right\} \quad h_3 = 303.87\ \mathrm{kJ/kg} \qquad \text{(Table A–13)}$$

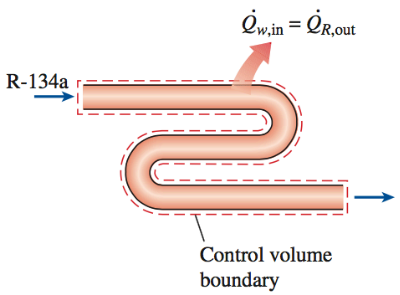
$$\left.\begin{aligned} P_4 &= 1\ \mathrm{MPa} \\ T_4 &= 35^\circ\mathrm{C} \end{aligned}\right\} \quad h_4 \cong h_{f\,@\,35^\circ\mathrm{C}} = 100.88\ \mathrm{kJ/kg} \qquad \text{(Table A–11)}$$

Substituting, we find

$$\dot{m}_w(62.982 - 104.83)\ \mathrm{kJ/kg} = (6\ \mathrm{kg/min})[(100.88 - 303.87)\ \mathrm{kJ/kg}]$$

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

(b) the heat transfer rate from the refrigerant to water.



(*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

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

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

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

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. 5–41), 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.

Next lecture

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