Thermodynamics I

Spring 1432/1433H (2011/2012H)
Saturday, Wednesday 8:00am - 10:00am & Monday 8:00am - 9:00am
MEP 261 Class ZA

Dr. Walid A. Aissa
Associate Professor, Mech. Engg. Dept.
Faculty of Engineering at Rabigh, KAU, KSA

Chapter #5
October XX, 2011

<u>Announcements:</u>

Dr. Walid's e-mail and Office Hours

walid_aniss@yahoo.com

Office hours for Thermo 01 will be every Sunday and Tuesday from **9:00 – 12:00 am** in *Dr. Walid's* office (Room 5-213).

<u>Text book:</u>

Thermodynamics An Engineering Approach

Yunus A. Cengel & Michael A. Boles

7th Edition, McGraw-Hill Companies,

ISBN-978-0-07-352932-5, 2008

Chapter 5

MASS AND ENERGY ANALYSIS OF CONTROL VOLUMES

Objectives of CH5: To

- Develop the conservation of mass principle.
- Apply the conservation of mass principle to various systems.
- Apply the first law of thermodynamics to control volumes.
- Identify the energy carried by a fluid stream crossing a control surface as the sum of I.E., flow work, I.E., & P.E. of the fluid and to relate the combination of the I.E. 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.

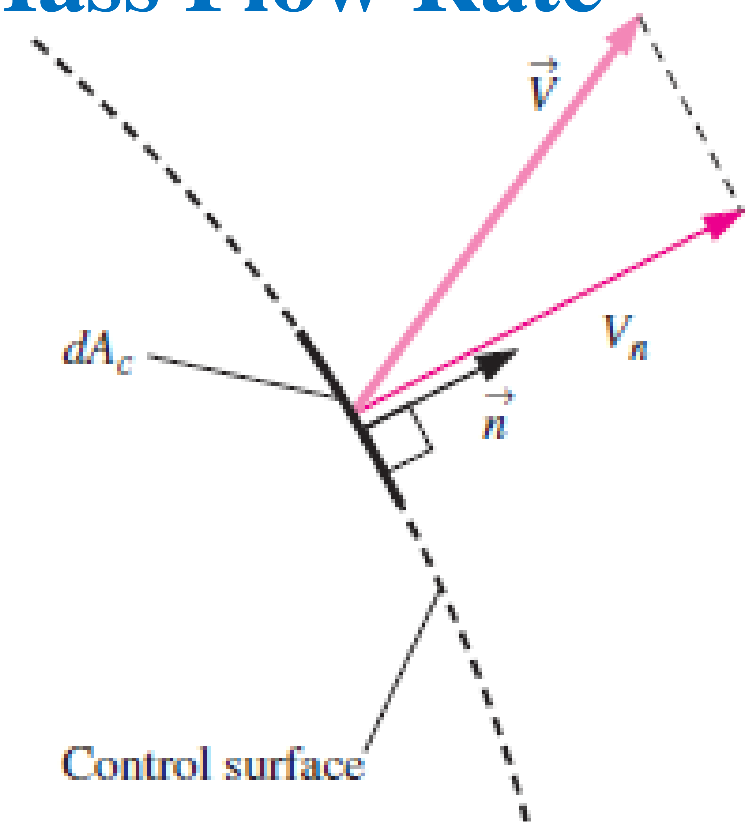
Chapter 5

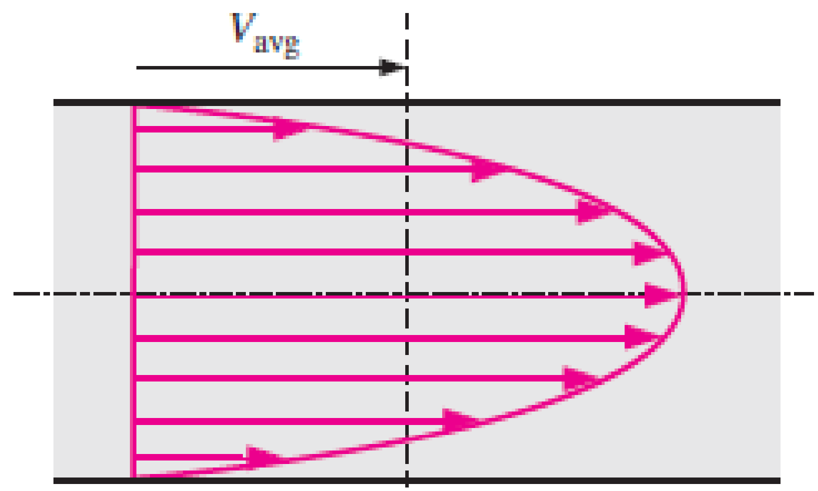
MASS AND ENERGY ANALYSIS OF CONTROL VOLUMES

5–1 ■ CONSERVATION OF MASS

Mass and Volume Flow Rates

Mass Flow Rate





$$\dot{m} = \int_{A_c} \delta \dot{m} = \int_{A_c} \rho V_n \, dA_c \qquad \text{(kg/s)} \qquad \textbf{(5-3)}$$

Average Velocity

$$V_{avg} = \frac{1}{A_c} \int_{A_c} V_n \, dA_c \qquad \textbf{(5-4)}$$

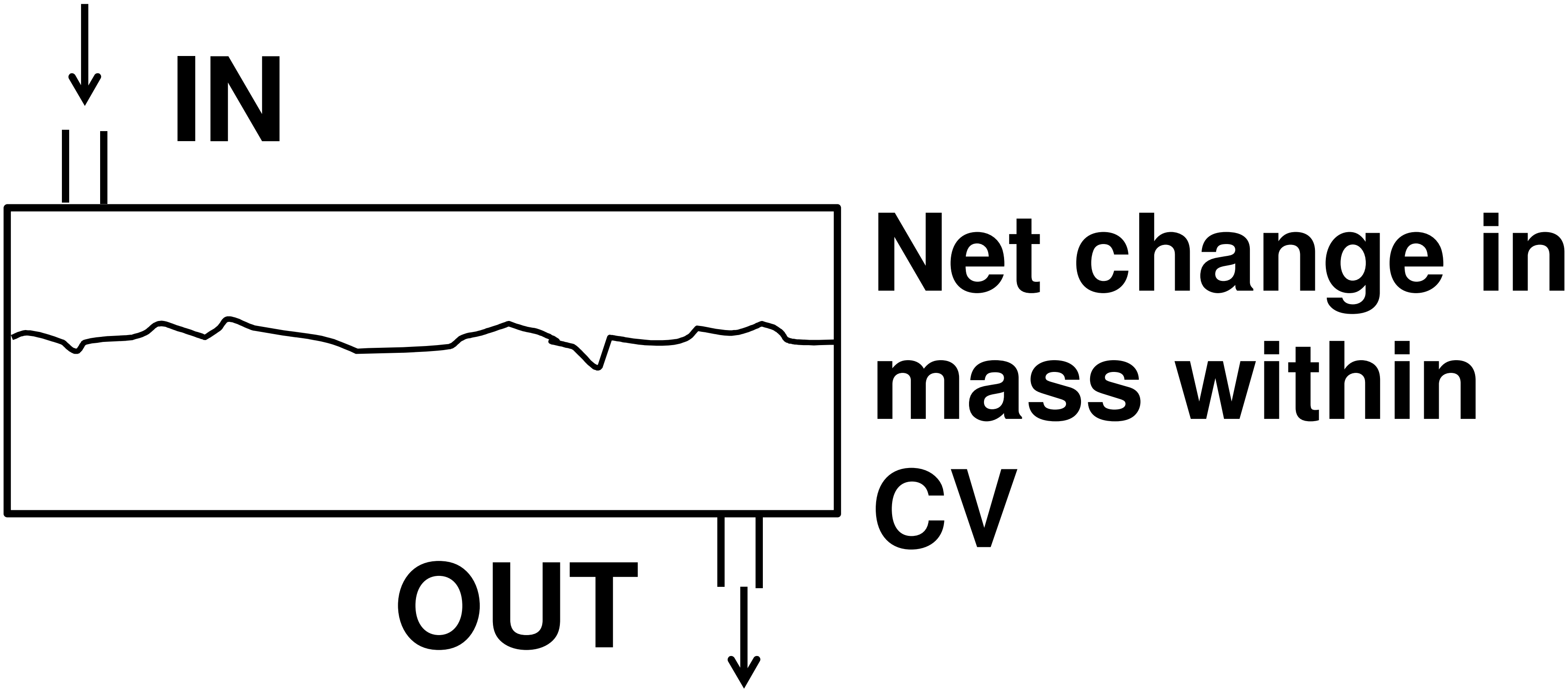
$$\dot{m} = \rho V_{\text{avg}} A_c \qquad (\text{kg/s}) \qquad \textbf{(5-5)}$$

Volume Flow Rate (Discharge)

$$\dot{V} = \int_{A_c} V_n \, dA_c = V_{\text{avg}} A_c = V A_c \qquad (\text{m}^3/\text{s}) \qquad \textbf{(5-6)}$$

$$\dot{m} = \rho \dot{V} \qquad \textbf{(5-7)}$$

Conservation of Mass Principle



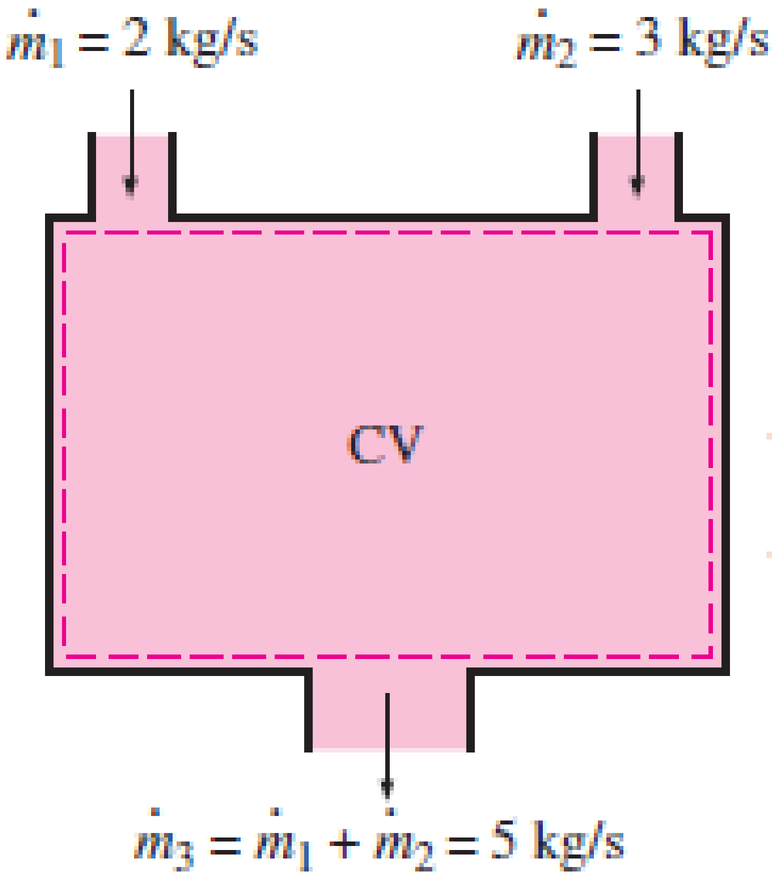
$$\begin{pmatrix}\text{Total mass entering}\\ \text{the CV during } \Delta t\end{pmatrix} - \begin{pmatrix}\text{Total mass leaving}\\ \text{the CV during } \Delta t\end{pmatrix} = \begin{pmatrix}\text{Net change in mass}\\ \text{within the CV during } \Delta t\end{pmatrix}$$

$$\frac{dm_{CV}}{dt} = \sum_{\text{in}} \dot{m} - \sum_{\text{out}} \dot{m} \qquad \textbf{(5-17)}$$

Mass Balance for Steady-Flow Processes

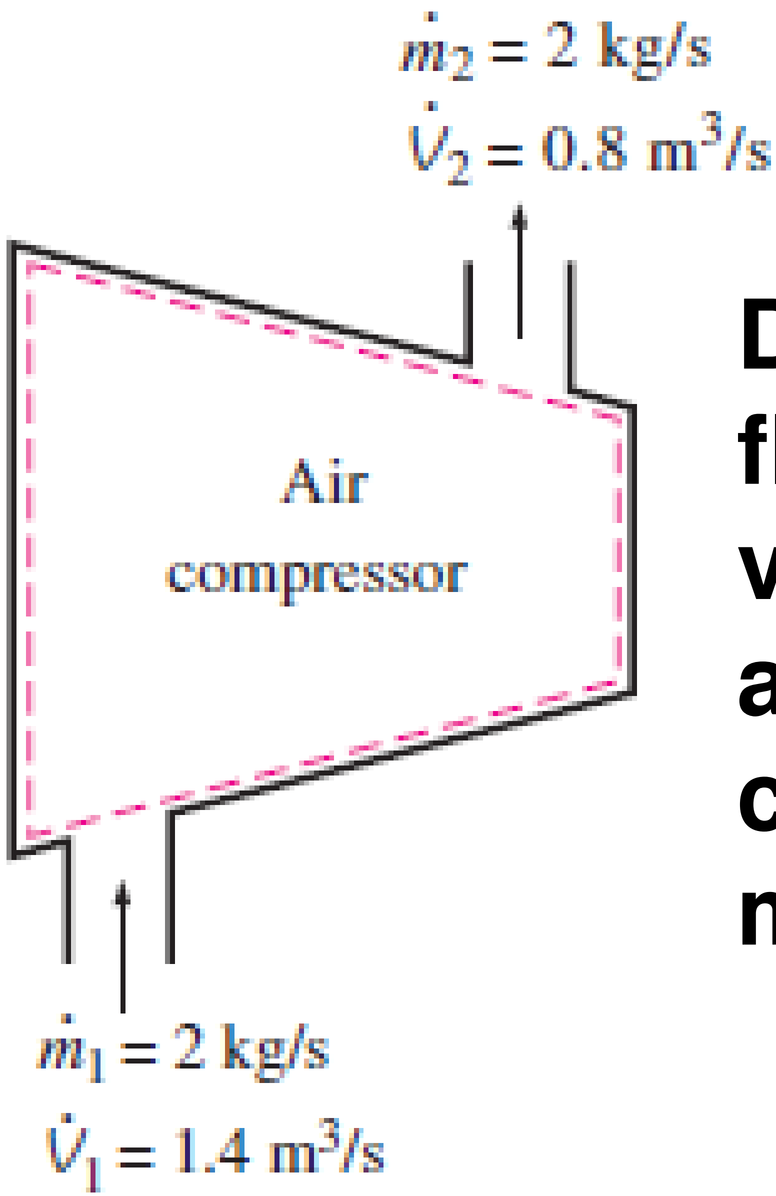
Mass Balance for Steady-Flow Processes

For steady flow process: *d*(any property)/*dt* = 0



$$\sum_{\text{in}} \dot{m} = \sum_{\text{out}} \dot{m} \qquad (\text{kg/s})$$

(5-18)



During a steady-flow process, volume flow rates are not necessarily conserved although mass flow rates are.

Special Case: Incompressible Flow

Steady, incompressible flow:

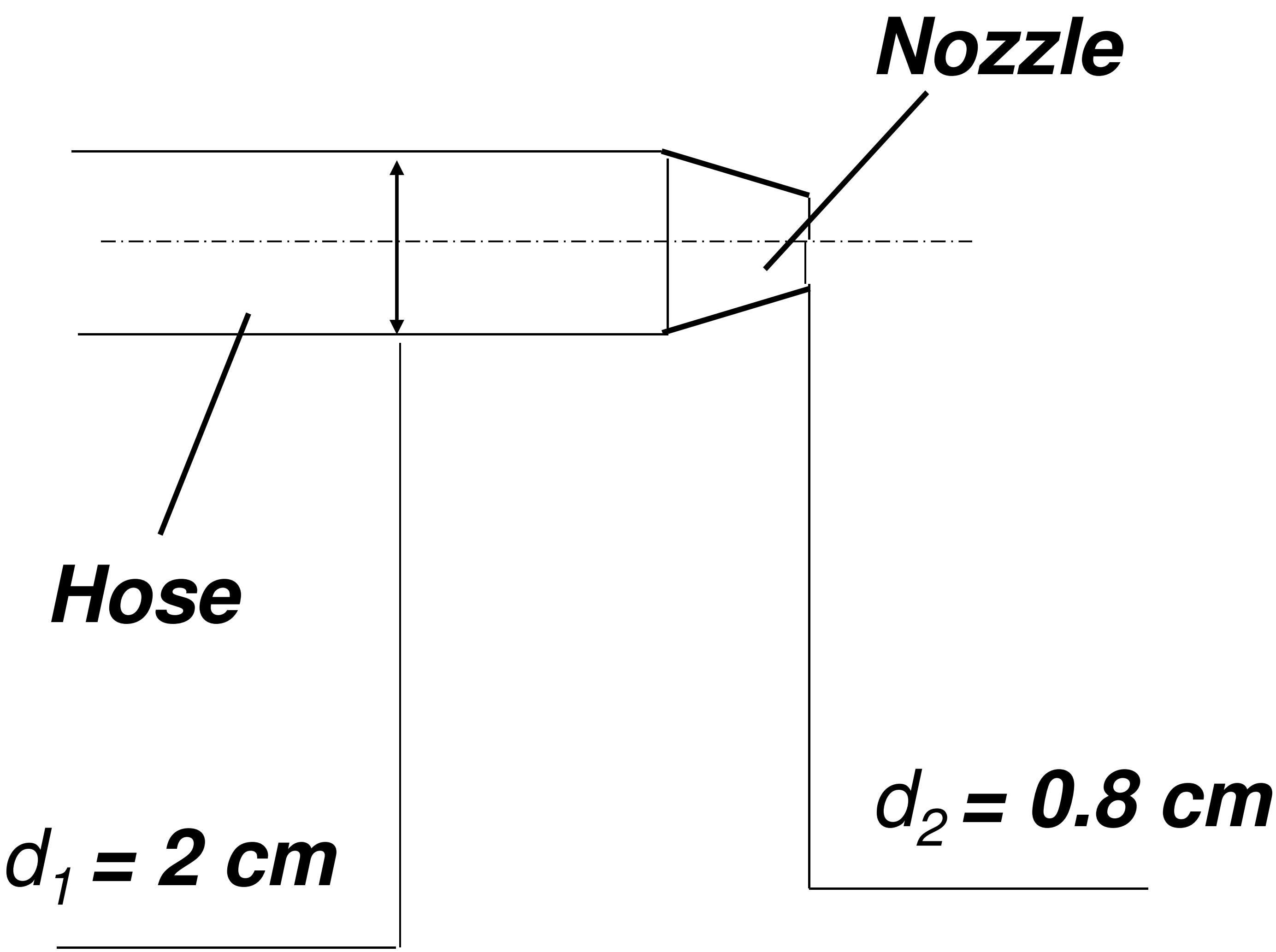
$$\sum_{\text{in}} \dot{V} = \sum_{\text{out}} \dot{V} \qquad (\mathrm{m^3/s}) \tag{5-20}$$

Steady, incompressible flow (single stream):

$$\dot{V}_1 = \dot{V}_2 \rightarrow V_1 A_1 = V_2 A_2 \tag{5-21}$$

EXAMPLE 5–1 Water Flow through a Garden Hose Nozzle.

A garden hose attached with a nozzle is used to fill a 10-gal bucket (1-gal = 3.7854 *l*). The inner diameter of the hose is 2 cm, and it reduces to 0.8 cm at the nozzle Exit . If it takes 50 s to fill the bucket with water, determine:


Nozzle
Hose
d_1 = 2 cm
d_2 = 0.8 cm

(*a*) *the volume and mass flow rates of water through the hose, and*
(b) the average velocity of water at the nozzle exit.

Solution:

$$\dot{V} = [(10 \times 3.7854/1000)/50]\,\mathrm{m^3/s}$$

$$\dot{V} = 0.757\frac{l}{s} = 0.757 \times 10^{-3}\,\mathrm{m^3/s}$$

$$\dot{m} = \rho \times \dot{V} = 1000 \times 0.757 \times 10^{-3} \, kg/s$$

$$\dot{m} = 0.757 \, kg/s$$

$$\dot{V} = V_{avg,2} \times A_2$$

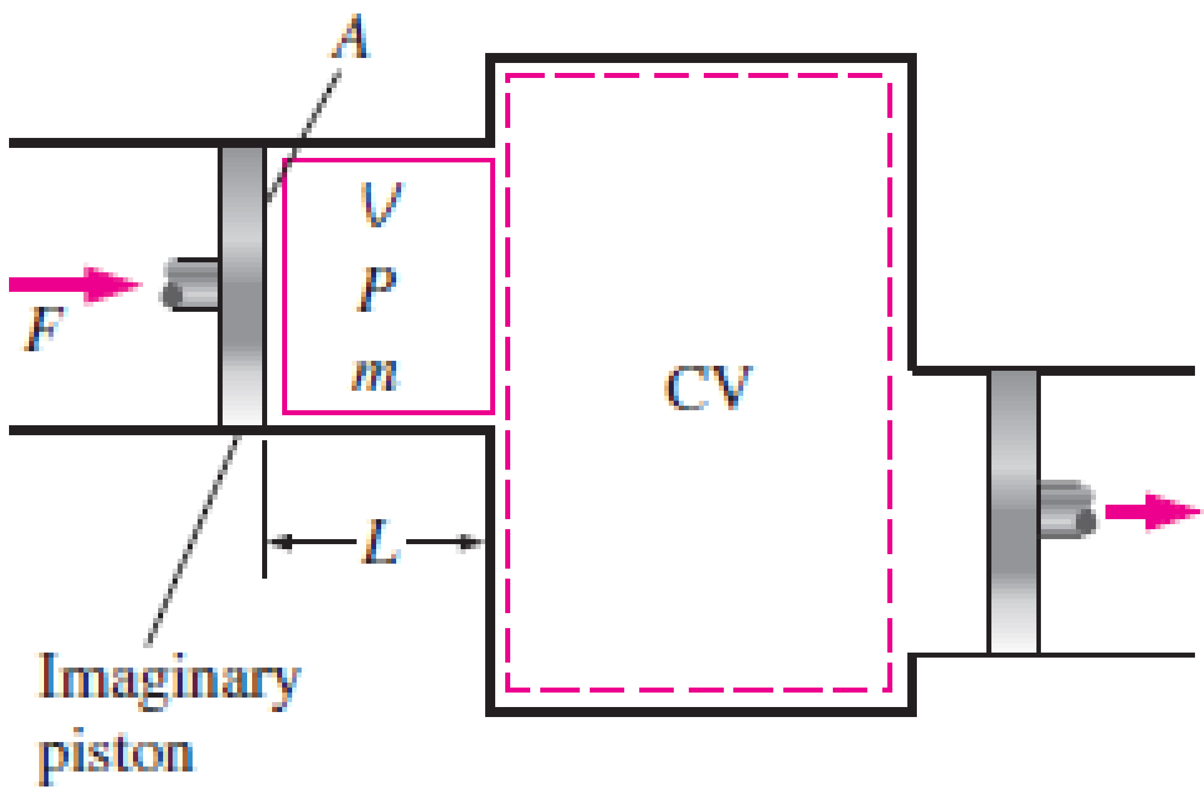
$$0.757 \times 10^{-3} \, m^3/s =$$

$$= V_{avg,2} \times \left[(\pi/4) \times (0.8 \times 10^{-2})^2 \right] m^2$$

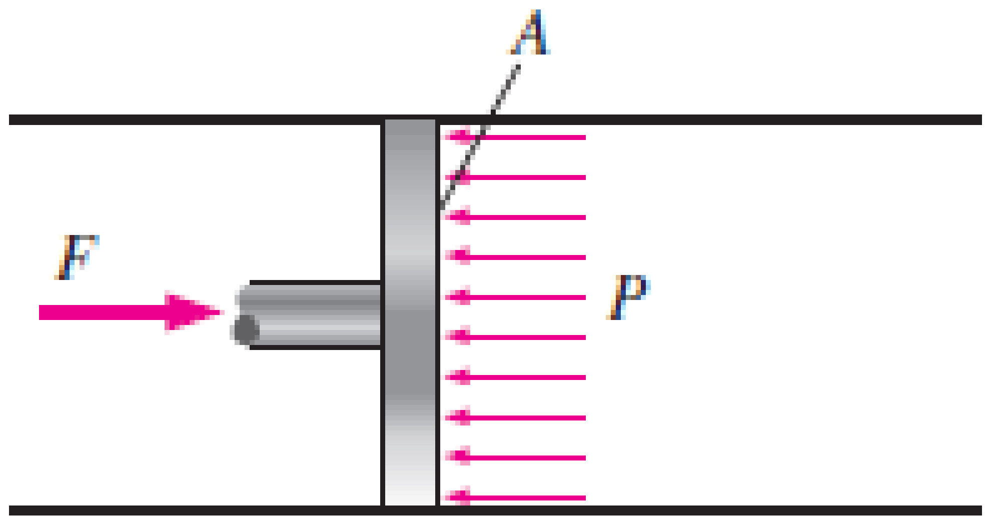
$$V_{avg,2} = 15.056 \, m/s$$

5–2 ■ FLOW WORK AND THE ENERGY OF A FLOWING FLUID

$F = pA$ **(5-22)**



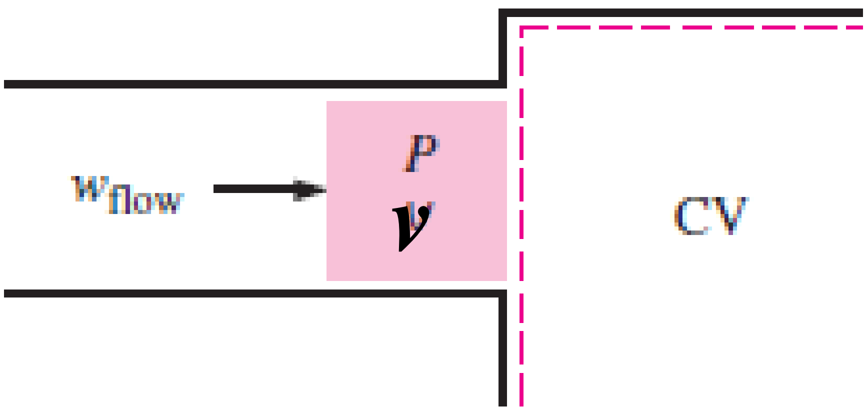
Schematic for flow work



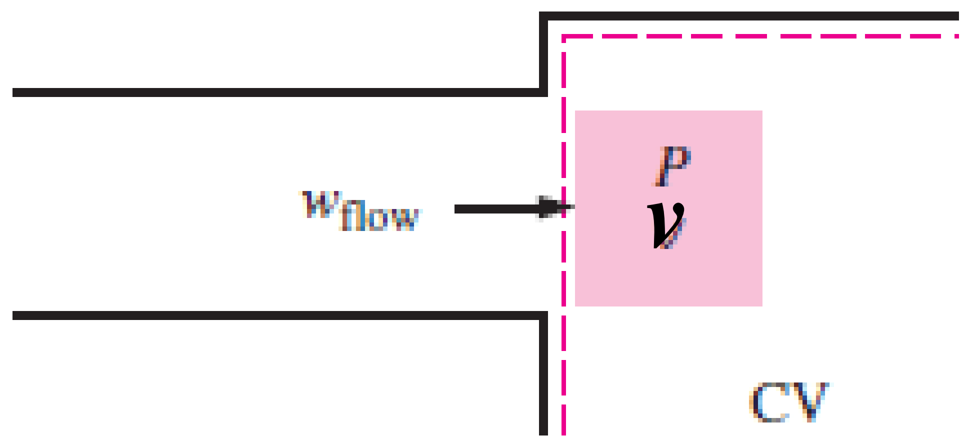
In the absence of acceleration, the force applied on a fluid by a piston is equal to the force applied on the piston by the fluid.

$$W_{flow} = FL = pA\,L = pV \qquad \text{(kJ)} \qquad (5\text{-}23)$$

$$w_{flow} = pv \qquad \text{(kJ/kg)} \qquad (5\text{-}24)$$

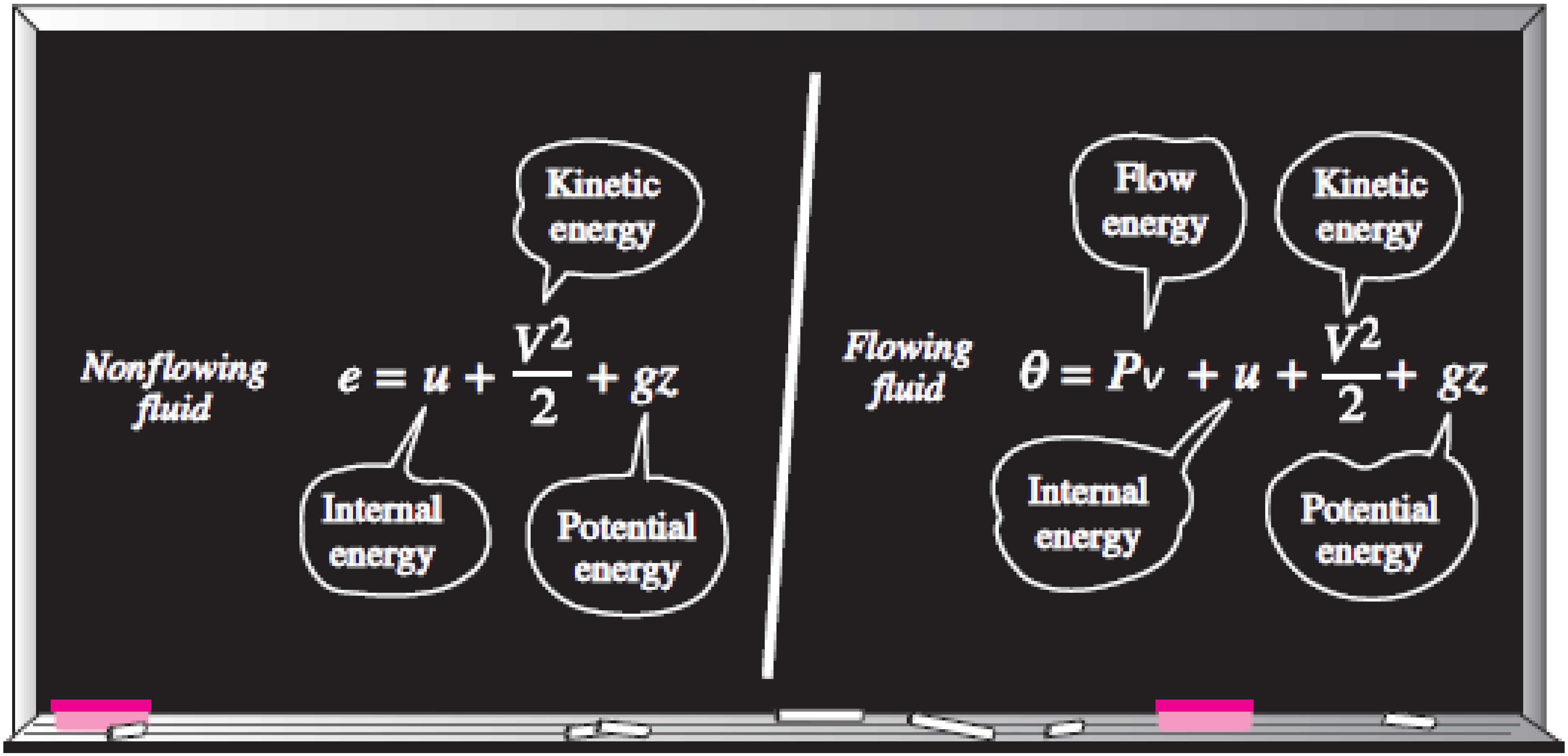


(a) Before entering



(b) After entering

Total Energy of a Flowing Fluid



Total energy consists of three parts for a non-flowing fluid and four parts for a flowing fluid.

- **For non-flowing fluid**

$$e = u + \mathrm{ke} + \mathrm{pe} = u + \frac{V^2}{2} + gz \quad \text{(kJ/kg)} \qquad (5\text{-}25)$$

- **For flowing fluid**

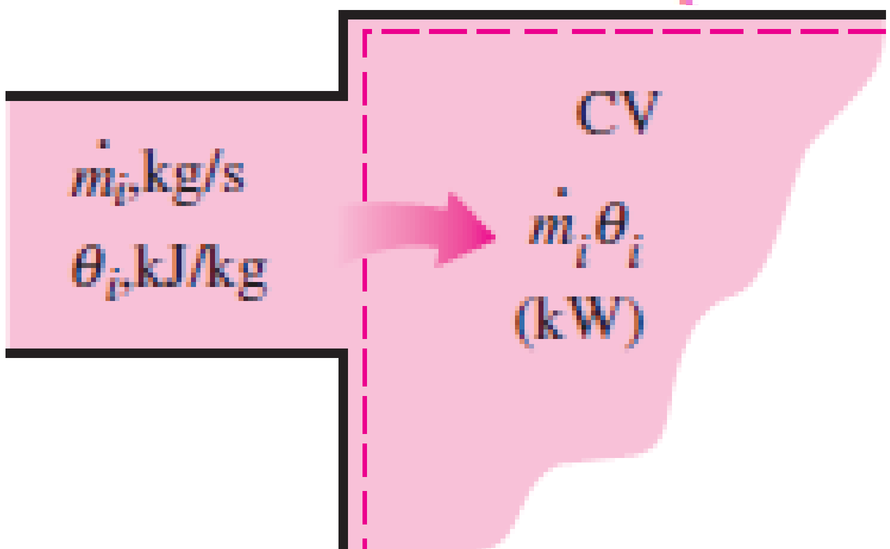
$$\theta = Pv + e = Pv + (u + \mathrm{ke} + \mathrm{pe}) \qquad (5\text{-}26)$$

$$\theta = h + \mathrm{ke} + \mathrm{pe} = h + \frac{V^2}{2} + gz \quad \text{(kJ/kg)} \qquad (5\text{-}27)$$

Energy Transport by Mass

Amount of energy transport:

$$E_{mass} = m\theta = m\left(h + \frac{V^2}{2} + gz\right) \quad \textbf{(kJ)} \quad \textbf{(5-28)}$$



The product $\dot{m}_i\theta_i$ is the energy transported into control volume by mass per unit time.

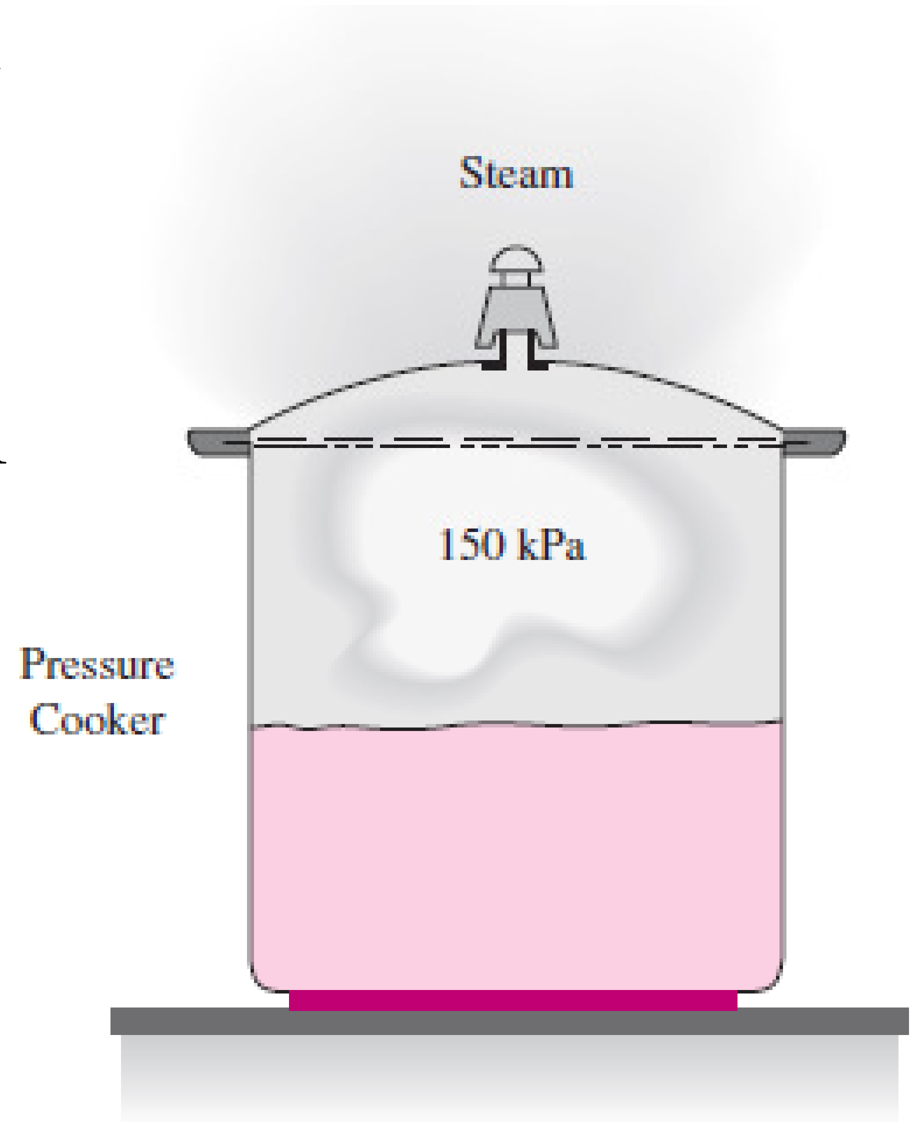
Rate of energy transport:

$$\dot{E}_{mass} = \dot{m}\theta = \dot{m}\left(h + \frac{V^2}{2} + gz\right) \quad \textbf{(kW)} \quad \textbf{(5-29)}$$

EXAMPLE 5–3 Energy Transport by Mass.

Steam is leaving a 4-*l* pressure cooker whose operating pressure is 150 kPa. It is observed that the amount of liquid in the cooker has decreased by 0.6 *l* in 40 min after the steady operating conditions are established, and the cross-sectional area of the exit opening is 8 mm^2.

Determine (*a*) *the mass flow rate of the steam and the exit velocity,* (*b*) *the* total and flow energies of the steam per unit mass, and (*c*) *the rate at which* energy leaves the cooker by steam



Solution:

I/P data: **Flow is steady,** p = 150 kPa, ΔV_{liquid} **= - 0.6** l **,** Δt **= 40 min,** $A_{Exit\ opening}$ = 8 mm^2

Assumptions: 1) *KE* and *PE* are negligible, and thus they are not considered.

2) Saturation conditions exist within the cooker at all times, so liquid has the properties of saturated liquid and the exiting steam has the properties of saturated vapor at the operating pressure

As liquid in the cooker exists as saturated liquid. Hence,

$$v_{liquid} = v_{f)\ at\ p\ =150\ kPa} = 0.001053\ \mathrm{m^3/kg}$$

From table (A-5)

Amount of liquid evaporated = $\boldsymbol{\Delta V_{liquid}} / v_{liquid}$ = **0.6 *l*** / 0.001053 m^3/kg= (0.6×10^{-3} m^3) / (0.001053 m^3/kg) = 0.57 kg

Mass flow rate of exiting steam = Mass flow rate of liquid evaporated = $\dot{m}$ = $\boldsymbol{m_{liquid}} \,/\, \boldsymbol{\Delta t}$ = 0.57 kg / **40 min** = 0.57 kg / (40×60 s) = 2.37×10^{-4} kg/s

As steam exiting from the cooker exit opening is saturated vapor, hence,

$$v_{steam} = v_{g)\ at\ p\ =150\ kPa} = 1.1594\ \mathrm{m}^3/\mathrm{kg}$$

From table (A-5)

But,

$$\dot{m} = \rho_{steam} V_{exit} A_{exit}$$

&

$$v_{steam} = 1/\rho_{steam}$$

Hence, $$\dot{m}\, v_{steam} = V_{exit} A_{exit}$$

$$\boldsymbol{A_{exit}} = 8 \text{ mm}^2 = \mathbf{8\times10^{-6}} \text{ m}^2$$

$$V_{exit} = (\dot{m} \times v_{steam})/A_{exit} = (2.37\times10^{-4} \text{ kg/s} \times 1.1594 \text{ m}^3/\text{kg})/\ (\mathbf{8\times10^{-6}} \text{ m}^2) = 34.3 \text{ m/s}$$

(b) Flow energy of the steam per unit mass; pv

But, $h = u + pv$

Hence, $pv = h - u$

As existing stream is saturated vapor, Hence, flow energy of the steam per unit mass; $pv)_{saturated\ vapor} = h_g - u_g =$ $2693.1\ kJ/kg - 2519.2\ kJ/kg = 173.9\ kJ/kg$

(I) If KE is neglected

(b) the total energy of the steam per unit mass, is

$$\theta = h + \text{ke} + \text{pe} = h + \underbrace{\frac{V^2}{2}}_{\approx 0} + \underbrace{gz}_{0} \quad \textbf{(kJ/kg)}$$

(5-27)

Hence, $\theta = h =$

Hence, $\theta = h_g = 2693.1\ kJ/kg$

(c) *Rate at which* energy leaves the cooker as steam

$\dot{E}_{\text{mass}} = \dot{m}\theta$ **(kW) (5-29)**

$\dot{E}_{\text{mass}} = \dot{m}\theta$ **=** [2.37×10 $^{-4}$ kg/s]×[*2693.1 kJ/kg*] = **0.638 kW**

(II) If KE is not neglected

(b) the total energy of the steam per unit mass, is

$$\theta = h + \text{ke} + \text{pe} = h + \frac{V^2}{2} + \cancelto{0}{gz} \quad \textbf{(kJ/kg)}$$

(5-27)

Hence, $\theta = h_g + (V^2/2) = 2693.1$ *kJ/kg* +

i.e., $\theta = 2693.1$ *kJ/kg* + [($34.3^2/2$)]/1000 *kJ/kg* = *2693.1*+ 0.588 *kJ/kg* = *2693.688 kJ/kg {note the difference is small}*

% Error (neglecting KE) = [$(2693.1-2693.688)/2693.688$ **]×100%=-0.0218 %**

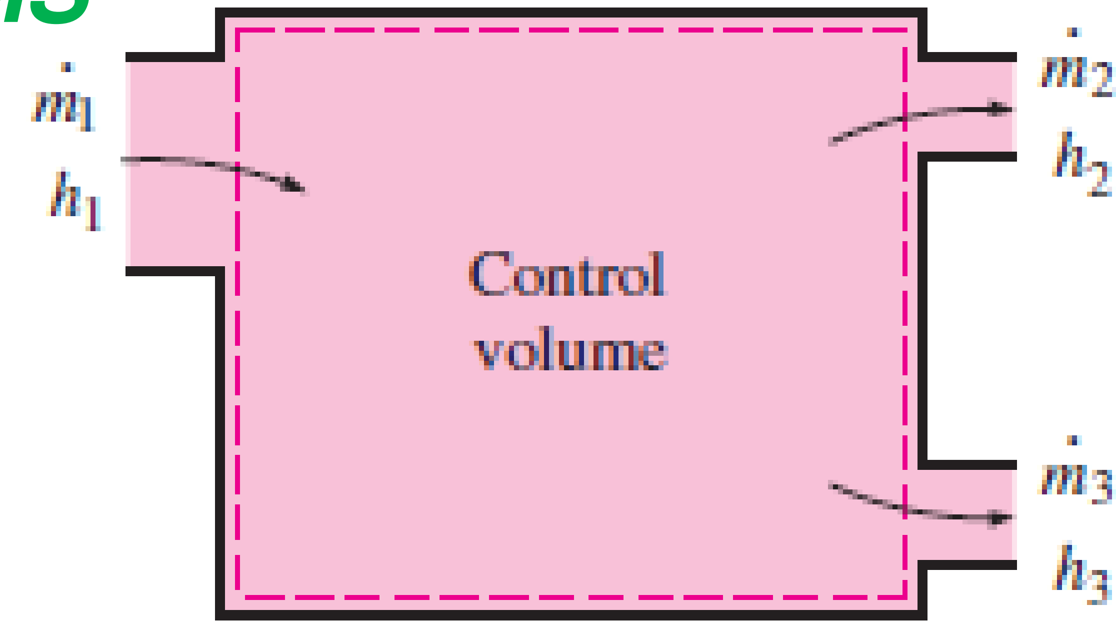
(*c*) *Rate at which* energy leaves the cooker as steam

$\dot{E}_{\text{mass}} = \dot{m}\theta$ **(kW) (5-29)**

$\dot{E}_{\text{mass}} = \dot{m}\theta$ **=** [2.37×10 $^{-4}$ kg/s]×[$2693.688\ kJ/kg$] = **0.6384 kW** *{note the difference is small}*

% Error (neglecting KE) = [(0.638 - 0.6384)/ 0.6384]×100% =-0.0627 %

5–3 ■ ENERGY ANALYSIS OF STEADY-FLOW SYSTEMS



I) Mass balance for a general steady-flow system is

$$\sum_{\text{in}} \dot{m} = \sum_{\text{out}} \dot{m} \qquad (\text{kg/s}) \qquad \textbf{(5-31)}$$

For a single-stream (one-inlet and one-outlet) steady-flow system

$$\dot{m}_1 = \dot{m}_2 \quad \rightarrow \quad \rho_1 V_1 A_1 = \rho_2 V_2 A_2 \qquad \textbf{(5-32)}$$

II) Energy balance for a general steady-flow system is

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

(5-33)

or

$$\underbrace{\dot{E}_{in}}_{\substack{\text{Rate of net energy transfer in} \\ \text{by heat, work, and mass}}} = \underbrace{\dot{E}_{out}}_{\substack{\text{Rate of net energy transfer out} \\ \text{by heat, work, and mass}}} \quad (\text{kW})$$

(5-34)

Hence, Eq. (5-34) can be re-written as:

$$\dot{Q}_{in} + \dot{W}_{in} + \sum_{in} \dot{m}\theta = \dot{Q}_{out} + \dot{W}_{out} + \sum_{out} \dot{m}\theta$$

(5-35)

or

$$\dot{Q}_{\text{in}} + \dot{W}_{\text{in}} + \sum_{\text{in}} \underbrace{\dot{m}\left(h + \frac{V^2}{2} + gz\right)}_{\text{for each inlet}} =$$

$$\dot{Q}_{\text{out}} + \dot{W}_{\text{out}} + \sum_{\text{out}} \underbrace{\dot{m}\left(h + \frac{V^2}{2} + gz\right)}_{\text{for each exit}}$$

(5-36)

When performing a general analytical study or **solving** a **problem** that **involves** an **unknown heat or work interaction**, it is common practice to <u>assume</u> heat to be transferred *into the system* (*<u>heat input</u>*) *at a* rate of $\dot{Q}$, and work produced *by the system* (*<u>work output</u>*) *at a rate of* $\dot{W}$, and then solve the problem. The first-law or energy balance relation in that case for a general steady-flow system becomes

$$\dot{Q} - \dot{W} = \sum_{\text{out}} \underbrace{\dot{m}\left(h + \frac{V^2}{2} + gz\right)}_{\text{for each exit}} - \sum_{\text{in}} \underbrace{\dot{m}\left(h + \frac{V^2}{2} + gz\right)}_{\text{for each inlet}}$$

(5-37)

For a single-stream (one-inlet and one-outlet) steady-flow system

$$\dot{Q} - \dot{W} = \dot{m}\left[h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)\right]$$

(5-38)

Dividing Eq. 5–38 by $\dot{m}$ gives the energy balance on a unit-mass basis as

$$q - w = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

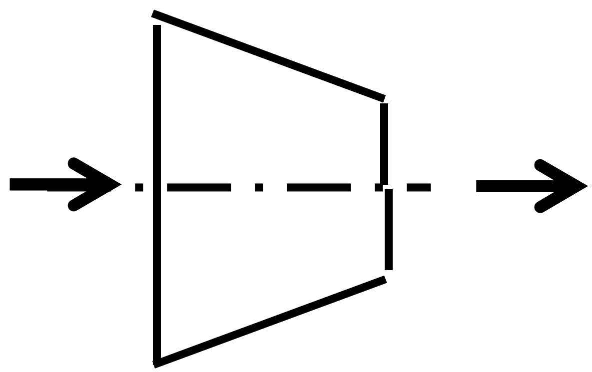
(5-39)

Neglecting *ΔKE* and *ΔPE* yields

$$q - w = h_2 - h_1 \qquad \textbf{(5-40)}$$

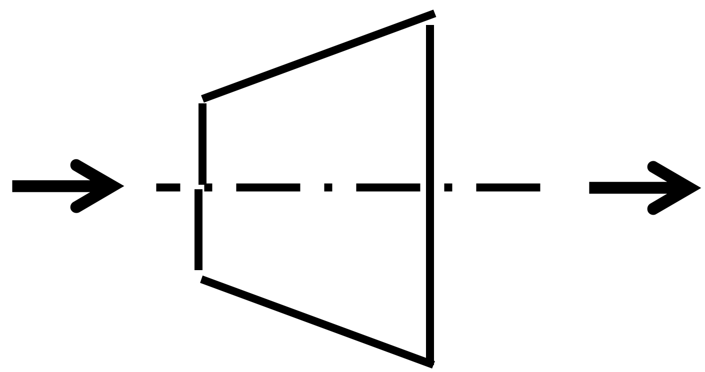
5–4 ■ SOME STEADY-FLOW ENGINEERING DEVICES

1 Nozzles and Diffusers



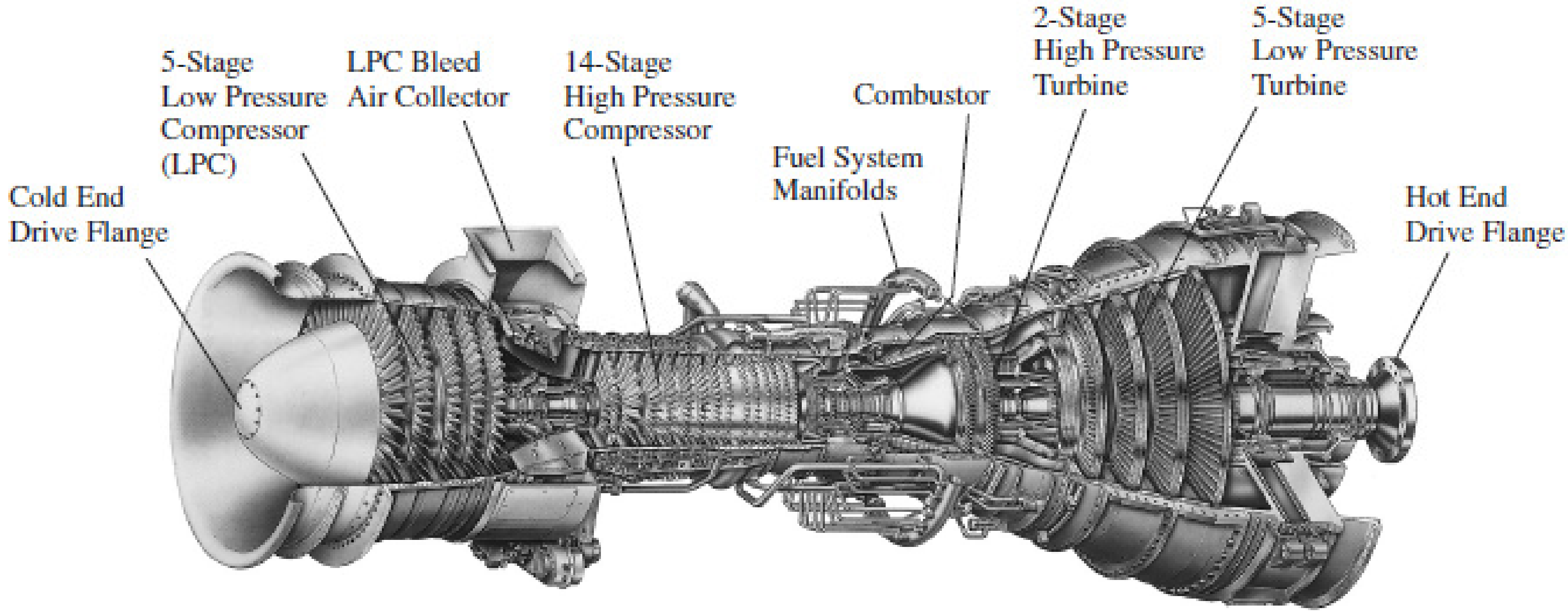
Nozzle

$V \uparrow$ & $p \downarrow$



Diffuser

$V \downarrow$ & $p \uparrow$



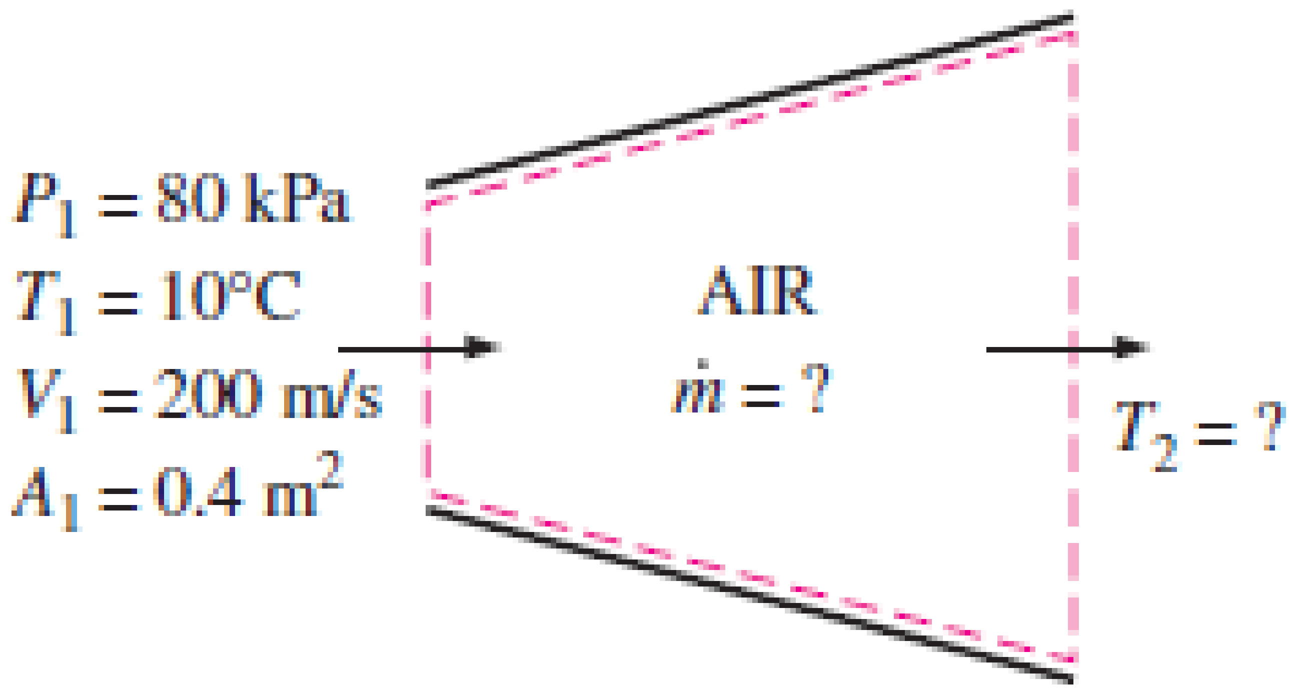
A modern GT (LM5000 model). It produces 55.2 MW at 3600 rpm with steam injection

EXAMPLE 5–4 Deceleration of Air in a Diffuser

Air at 10℃ 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 with the inlet velocity.

Determine (*a) the mass flow rate of the air and (b) the temperature of* the air leaving the diffuser.

Solution



From Equation of state:

$P_1/\rho_1 = R\,T_1$

R = 0.287 kJ/kg.K

$\rho_1 = P_1 / R\,T_1 = 80\ kPa / [$0.287 kJ/kg.K $*(10+273\ K)] = 0.985$ kg/m²

$$\dot{m} = \rho_1 V_1 A_1 = (0.985\,\frac{kg}{m^3}) \times 200\,\frac{m}{s} \times 0.4\,m^2$$

$$\dot{m} = 78.8\,kg/s$$

From Eq. (5-38) for a single-stream (one-inlet and one-outlet) steady-flow system

$$\cancel{\dot{Q}}_{0} - \cancel{\dot{W}}_{0} = \dot{m}\left[h_2 - h_1 + \underbrace{\cancel{\frac{V_2^2 - V_1^2}{2}}}_{\approx 0} + \underbrace{\cancel{g(z_2 - z_1)}}_{0}\right] \qquad \textbf{(5-38)}$$

Hence,

$$0 = (h_2 - h_1) - (V_1^2/2)$$

Hence,

$$h_2 = h_1 + (V_1^2/2) \qquad \textbf{(\#-1)}$$

$T_1 = 10+273\ K = 283\ K$

(1) First method for evaluation of T_2

From table (A-17; ideal gas properties of air)

$h(\text{Air at } T = 280K) = 280.13\ kJ/Kg$

$h(\text{Air at } T = 285K) = 285.14\ kJ/Kg$

$$\frac{h_1 - h_{T=280K}}{h_{T=285K} - h_{T=280K}} = \frac{283-280}{285-280}$$

$$\frac{h_1 - 280.13}{285.14 - 280.13} = \frac{3}{5}$$

h_1=h(Air at T= 283K)= 283.136 kJ/Kg

But, $h_2 = h_1 + (V_1^2/2)$

h_2= 283.136 kJ/Kg + [(200^2 /2)/1000] kJ/Kg = 303.136 kJ/Kg

Hence, From table (A-17), T_2 @ $h_2 =$ *303.136 kJ/Kg is obtained by interpolation between* T=300 @ h= *300.19 kJ/Kg &* T=305 @ h= *305.22 kJ/Kg*

$$\frac{T_2 - 300K}{305\ K - 300K} = \frac{303.136 - 300.19}{305.22 - 300.19}$$

$$T_2 = 302.928\ K \approx 303\ K$$

(2) Another method for evaluation of T_2

From Eq. (#-1)

$$h_2 = h_1 + (V_1^2/2) \qquad \textbf{(\#-1)}$$

$$c_p T_2 = c_p T_1 + (V_1^2/2)$$

$$T_2 = T_1 + (V_1^2/2)/c_p$$

Hence, From table (A-2), $c_{p,\ Air} = 1.005\ kJ/Kg\ .K$

$$T_2 = 283 + [(200^2\ /2)/1000]/1.005$$

$T_2 = 303\ K$

EXAMPLE 5–5 Acceleration of Steam in a Nozzle

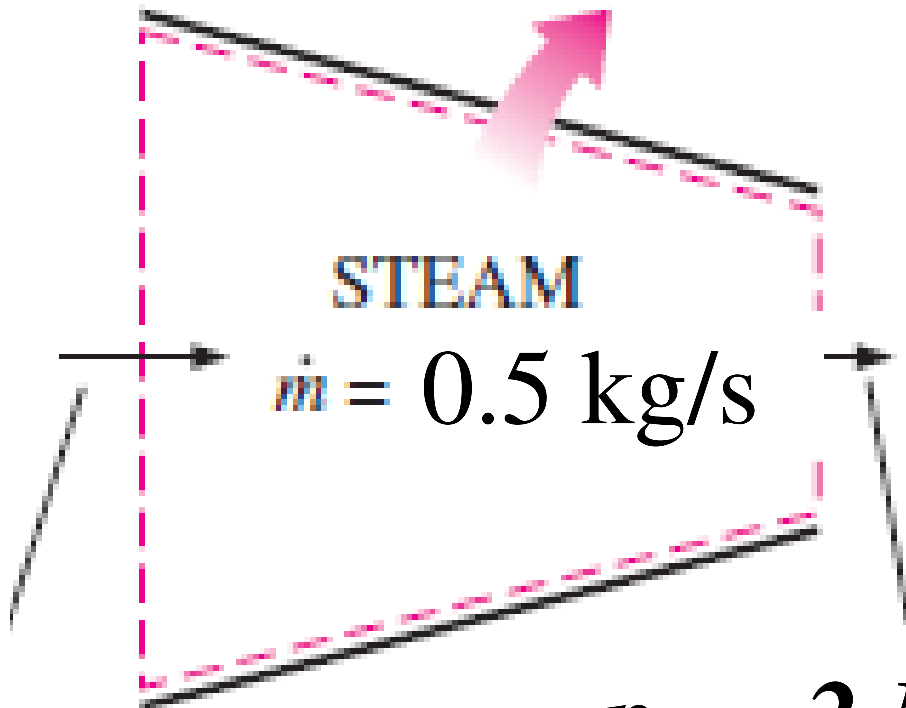
Steam at 2 MPa and 350°C steadily enters a nozzle whose inlet area is 2 cm^2. The mass flow rate of steam through the nozzle is 5 kg/s. Steam leaves the nozzle at 2 MPa with a velocity of 300 m/s. Heat losses from the nozzle per unit mass of the steam are estimated to be 1.2 kJ/kg. Determine

(a) the inlet velocity and

(b) the exit temperature of the steam.

Solution:

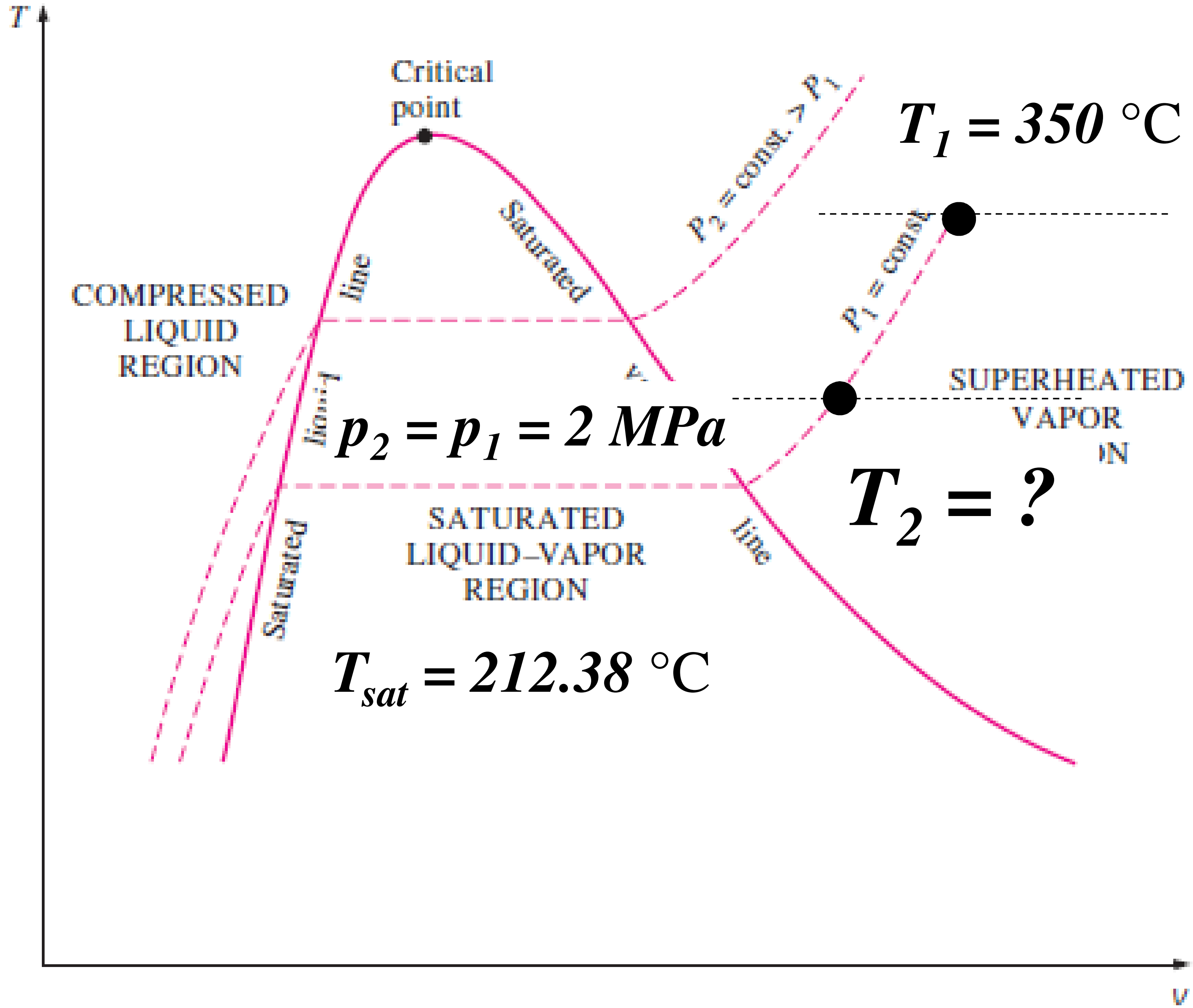
$q_{out} = 1.2$ kJ/kg



$p_1 = 2\ MPa$
$T_1 = 350\ °C$
$A_1 = 20\ cm^2$

$p_2 = 2\ MPa$
$V_2 = 300\ m/s$

$V_1 = ?,\ T_2 = ?$


T
Critical point
$T_1 = 350$ °C
$P_2 = \text{const.} > P_1$
$P_1 = \text{const.}$
Saturated
line
COMPRESSED LIQUID REGION
SUPERHEATED VAPOR
$p_2 = p_1 = 2$ MPa
$T_2 = ?$
SATURATED LIQUID–VAPOR REGION
Saturated
line
$T_{sat} = 212.38$ °C
v

From table (A-5), T_{Sat} (p = 2 *Mpa)* = ***212.38*** °C

As T_1 (= ***350*** °C) > T_{Sat} (p = 2 *Mpa)* = ***212.38*** °C

Hence, state 1 is superheated steam

From table (A-6), v_1 = ***0.1386*** m^3 /kg, h_1 = ***3137.7*** kJ /kg

$$\dot{m} = \rho_1 V_1 A_1 :$$

$$A_1 = 20\ \mathrm{cm}^2 = 20 \times 10^{-4}\ \mathrm{m}^2$$

$$(0.5\ \mathrm{kg/s} \times \mathbf{0.1386}\ \mathrm{m}^3/\mathrm{kg}) = V_1 \times 20 \times 10^{-4}\ \mathrm{m}^2$$

$$V_1 = 34.65\ m/s$$

From Eq. (5-38) for a single-stream (one-inlet and one-outlet) steady-flow system

$$\dot{Q} - \cancelto{0}{\dot{W}} = \dot{m}\left[h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + \cancelto{0}{g(z_2 - z_1)}\right] \tag{5-38}$$

$$\dot{Q}/\dot{m} = (h_2 - h_1) + \frac{V_2^2 - V_1^2}{2}$$

Hence,

$$-1.2\,\frac{kJ}{kg} = \left(h_2 - 3137.7\,\frac{kJ}{kg}\right) + \frac{300^2 - 34.65^2}{2 \times 10^3}\,\frac{kJ}{kg}$$

Hence,

$$h_2 = 3092.1\ kJ/kg$$

Hence, From table (A-6), T_2 @ h_2 = ***3092.1*** *is obtained by interpolation between* T=300 °C @ h= *3024.2 kJ/Kg &*
T=350°C @ h= *3137.7 kJ/Kg*

$$\frac{T_2 - 300\ ℃}{350\ ℃ - 300\ ℃} = \frac{3092.1 - 3024.2}{3137.7 - 3024.2}$$

Hence, T_2 = 329.9 °C≈ 330 °C

2 Turbines and Compressors

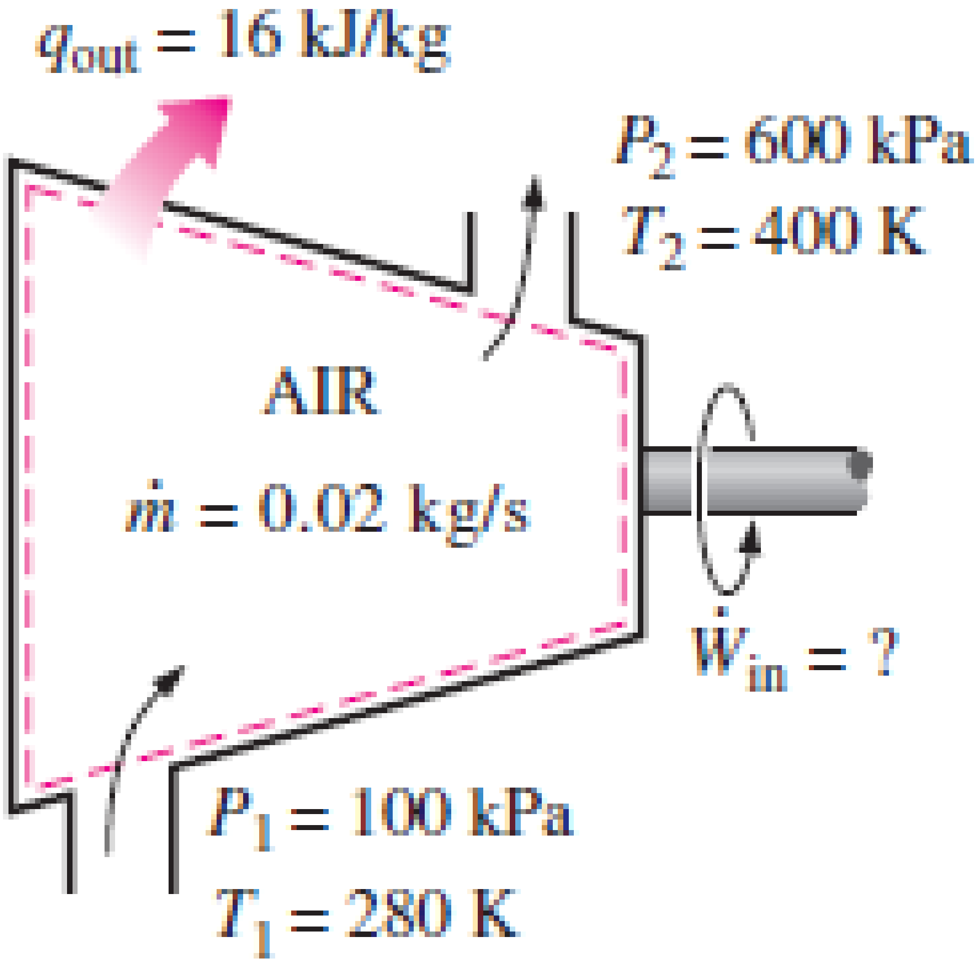
In steam, gas, or hydroelectric power plants, the **device that drives the electric generator is** the turbine. As the fluid passes through the turbine, work is done against the blades, which are attached to the shaft. As a result, the shaft rotates, and the **turbine produces work**.

Compressors, as well as pumps and fans, are **devices used to increase the pressure of a fluid**. Work is supplied to these devices from an external source through a rotating shaft. Therefore, **compressors involve work inputs**.

EXAMPLE 5–6 Compressing Air by a Compressor

Air at 100 kPa and 280 K is compressed steadily to 600 kPa and 400 K. The mass flow rate of the air is 0.02 kg/s, and a heat loss of 16 kJ/kg occurs during the process. Assuming the changes in kinetic and potential energies are negligible, determine the necessary power input to the compressor.

Solution:



Schematic for Example 5–6.

From Eq. (5-38) for a single-stream (one-inlet and one-outlet) steady-flow system

$$\dot{Q} - \dot{W} = \dot{m}\left[h_2 - h_1 + \cancelto{0}{\frac{V_2^2 - V_1^2}{2}} + \cancelto{0}{g(z_2 - z_1)}\right] \qquad \textbf{(5-38)}$$

Hence, (5-38) can be re-written as:

$$\left(\dot{Q} - \dot{W}\right)/\dot{m} = \left(h_2 - h_1\right) \qquad \textbf{(\#-2)}$$

But,

$$\dot{Q}/\dot{m} = -q_{out} \qquad \textbf{(\#-3)}$$

&

$$\dot{W} = -\dot{W}_{in} \qquad \text{(\#-4)}$$

By substituting in Eq. (#-2) to get

$$-q_{out} - \left(-\dot{W}_{in}/\dot{m}\right) = (h_2 - h_1)$$

i.e.

$$-q_{out} + \left(\dot{W}_{in}/\dot{m}\right) = (h_2 - h_1)$$

(#-5)

$$q_{out} = 16\, kJ/kg$$

From table (A-17), h_2 @ T_2 = **400 K** *is 400.98 kJ/kg* & h_1 @ T_1 = **280 K** *is 280.13 kJ/kg*

Hence, Eq. (#-5) can be re-written as:

$- 16\, kJ/kg + (\dot{W}_{in} / 0.02) = 400.98 - 280.13$

Hence,

$$\dot{W}_{in} = 2.737\, KW \approx 2.74\, KW$$

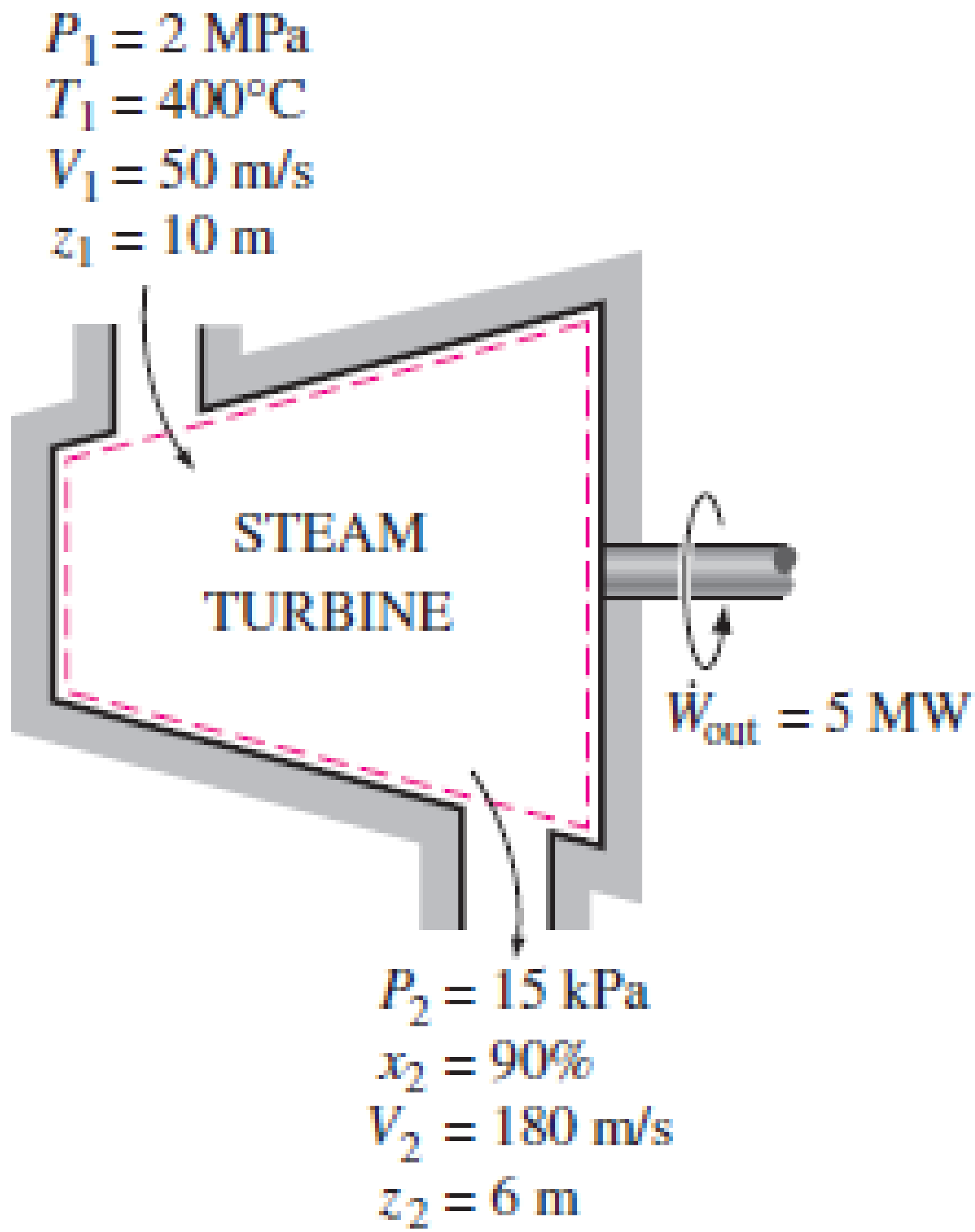
EXAMPLE 5–7 Power Generation by a Steam Turbine

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 attached figure.

(*a*) *Compare the magnitudes 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.

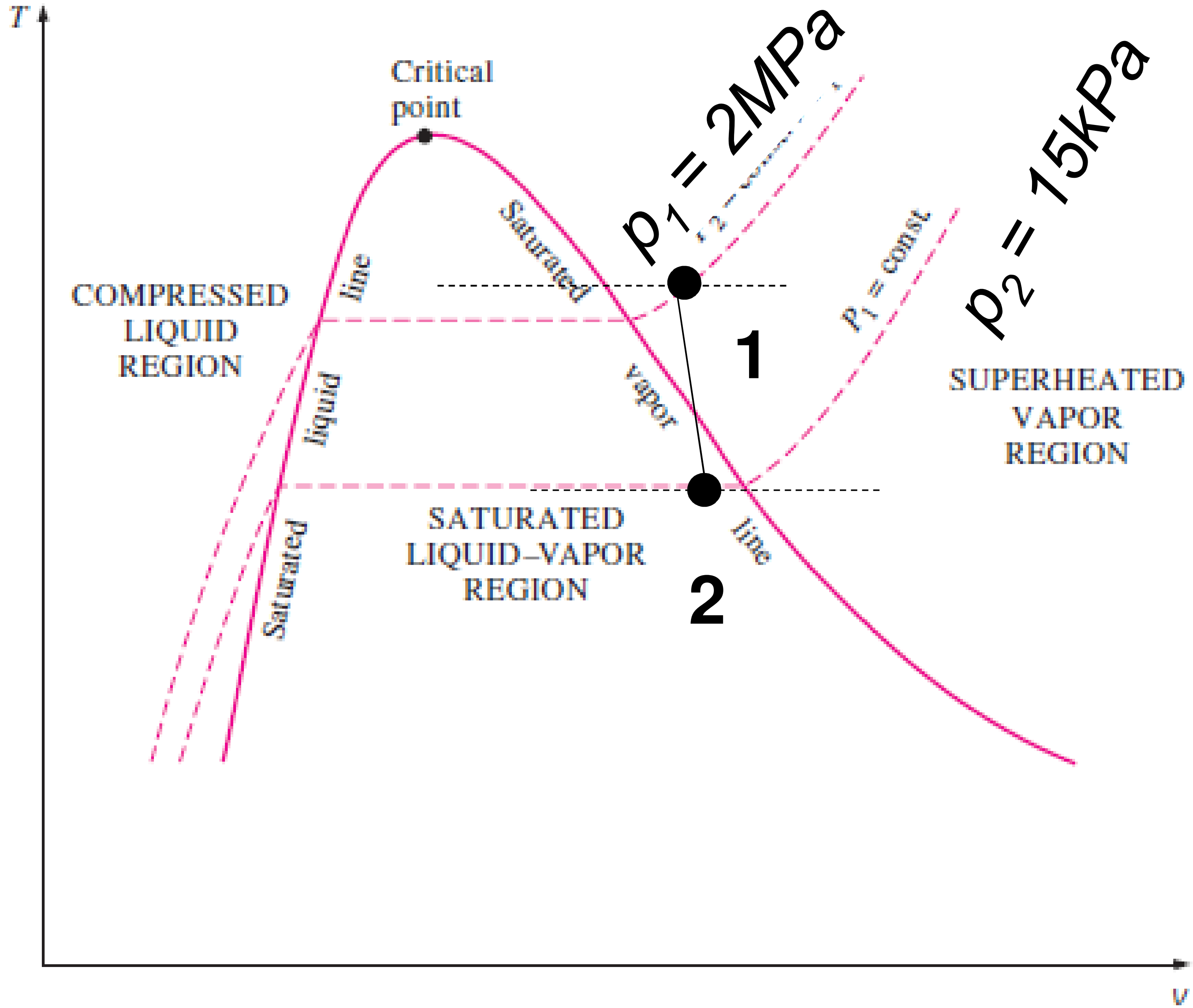


Solution:

Evaluation of conditions at state 1:

From table A-5, T_{sat} (corresponding to p = 2MPa) = 212.38 °C

As T_1 = 400 °C > [T_{sat} (corresponding to p = 2MPa) = 212.38 °C]. Hence, state 1 is superheated steam.


T
Critical point
COMPRESSED LIQUID REGION
Saturated liquid line
SATURATED LIQUID–VAPOR REGION
Saturated vapor line
SUPERHEATED VAPOR REGION
P_1 = const
$p_1 = 2MPa$
$p_2 = 15kPa$
1
2
v

From table A-6 for superheated steam corresponding to , $T_1 = 212.38$ °C & $p_1 =$ 2MPa; $v_1 = 0.15122$ m^3/kg & $u_1 = 2945.9$ kJ/kg, $h_1 = 3248.4$ kJ/kg.

Evaluation of conditions at state 2:

It is clear from I/P data ($x = 0.9$) that condition at state 2 is saturated liquid/vapor mixture.

From table A-5 Saturated water-pressure table corresponding to p_2 = 15 kPa; T_{sat} =53.97 °C , v_f = 0.001014 m^3/kg , v_g = 10.02 m^3/kg ,u_f = 225.93 kJ/kg ,u_{fg}= 2222.1 kJ/kg & u_g = 2448.0 kJ/kg, h_f = 225.94 kJ/kg, h_{fg} = 2372.3 kJ/kg, h_g = 2598.3 kJ/kg.

$v_2 = v_f + x\,(v_g - v_f) = 0.001014 + 0.9$
$(10.02 - 0.001014)\text{m}^3/\text{kg} = 9.0181\text{m}^3/\text{kg}$

$u_2 = u_f + x\,u_{fg} = 225.93 + (0.9 * 2222.1)$
$\text{kJ/kg} = 2448.12\ \text{kJ/kg}$

$h_2 = h_f + x\,h_{fg} = 225.94 + (0.9 * 2372.3)$
$\text{kJ/kg} = 2361.01\ \text{kJ/kg}.$

(*a*) *Magnitudes of Δh, Δke, and Δpe.*

$\Delta h = h_2 - h_1$ = 2361.01 kJ/kg - 3248.4 kJ/kg= -887.39 kJ/kg.

$\Delta ke = (V_2^2 - V_1^2)/2 = (180^2 - 50^2)/(2*1000) = 14.95$ kJ/kg.

$\Delta pe = g(z_2 - z_1) = 9.81*(6-10)/1000 = -0.03924$ kJ/kg.

$v = v_f + x\,(v_g - v_f) = 0.001014 + 0.9\,(10.02 - 0.001014)\text{m}^3/\text{kg} = \text{m}^3/\text{kg}$, $u = u_f + x\,u_{fg} = 225.93 + (0.9 * 2222.1)$ kJ/kg & $h = h_f + x\,h_{fg} = 225.94 + (0.9 * 2372.3)$ kJ/kg, $h_g = 2598.3$ kJ/kg.

(*b*) *Magnitude of specific work* (*w*)

$$\cancelto{0}{\dot{Q}} - \dot{W} = \dot{m}\left[h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)\right]$$

$$-w = \left[(h_2 - h_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)\right]$$

$$-w = \left[(2361.01\ kJ/kg - 3248.4\ kJ/kg) + \frac{180^2 - 50^2}{2 \times 10^3}\frac{kJ}{kg} + \frac{9.81 \times (6-10)}{10^3}\frac{kJ}{kg}\right]$$

$-w = -872.48\ kJ/kg$

$w = 872.48\ kJ/kg$

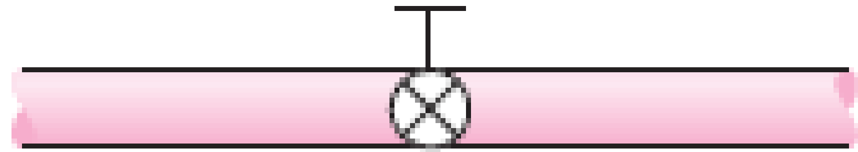
(*c*) *Mass flow rate of the steam.*

$\dot{m} = (5kW)/(872.48\ kJ/kg)$

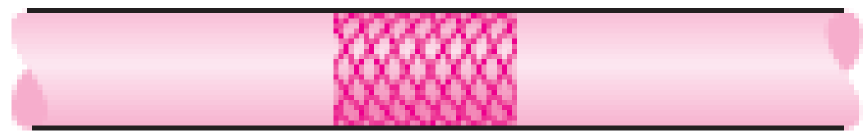
i.e.

$\dot{m} = (5000kJ/s)/(872.48\ kJ/kg) = 5.73\ kg/s$

3 Throttling Valves



(*a*) An adjustable valve



(*b*) A porous plug

(*c*) A capillary tube

Throttling valves are devices that cause large pressure drops in the fluid

Throttling devices are commonly used in refrigeration and air-conditioning applications

Throttling valves are accompanied by:

a) significant pressure drop,
b) Large drop in temperature,
c) Flow through them may be adiabatic ($q \approx 0$),
d) no work done ($w = 0$),
e) $\Delta pe \approx 0$, $\Delta ke \approx 0$.

Hence, conservation of energy equation for this single-stream steady-flow device reduces to

$$h_2 \approx h_1 \qquad kJ/kg \qquad (5\text{-}41)$$

$$But,\ h = u + pv$$

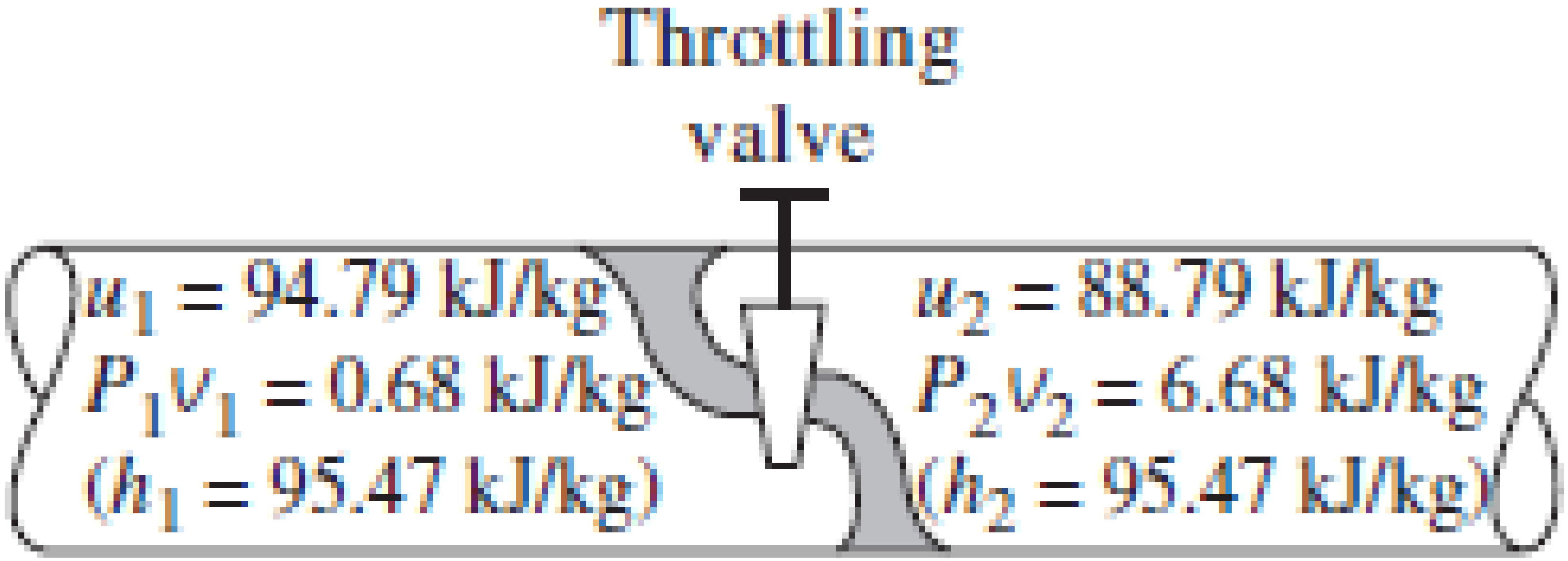
Hence, Eq. (5-41) can be re-written as

$$u_2 + p_2 v_2 \approx u_1 + p_1 v_1 \qquad (\#\text{-}6)$$

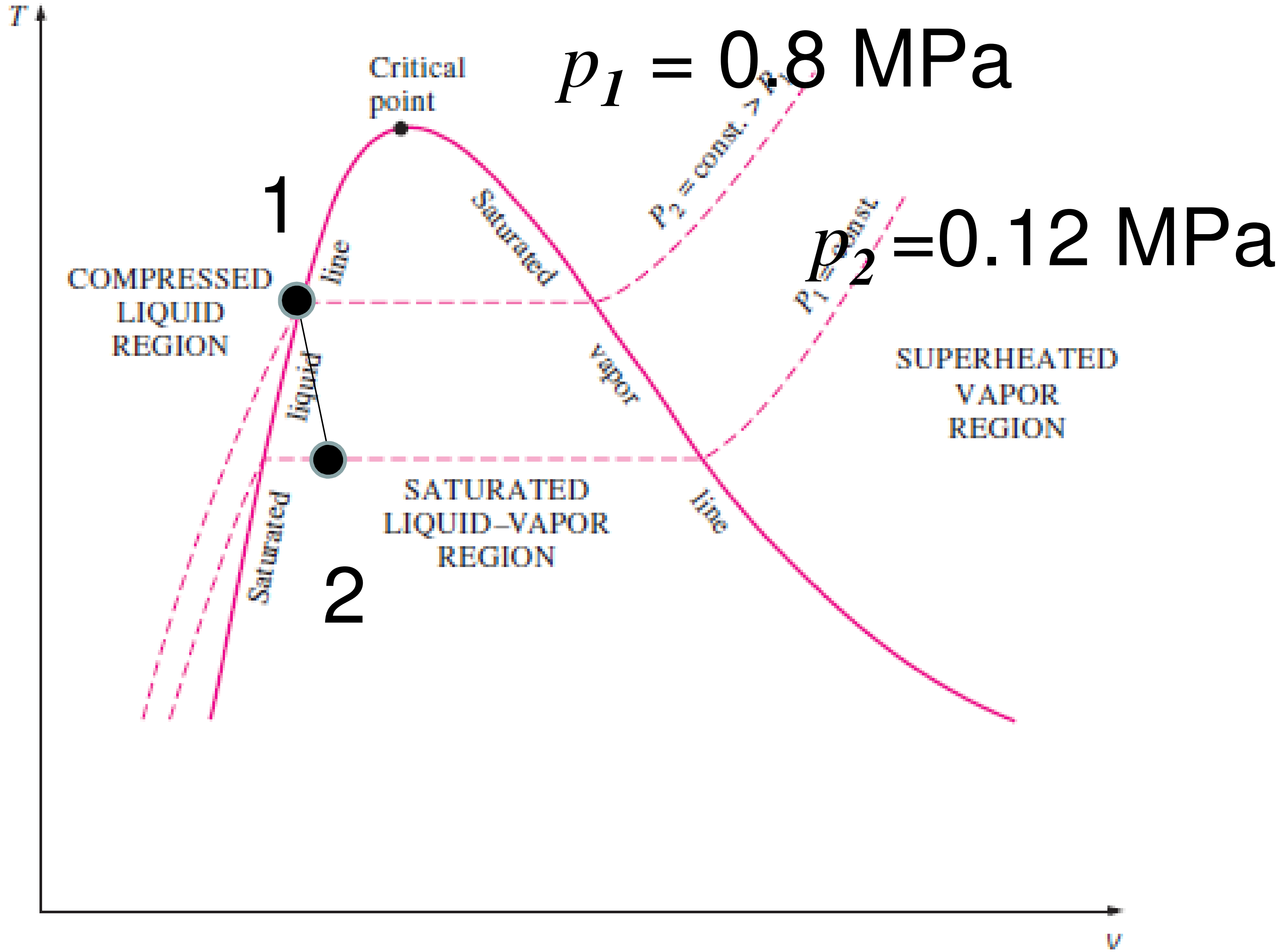
EXAMPLE 5–8 Expansion of Refrigerant-134a in a Refrigerator

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.

Solution:



Throttling valve
$u_1 = 94.79$ kJ/kg
$P_1v_1 = 0.68$ kJ/kg
($h_1 = 95.47$ kJ/kg)
$u_2 = 88.79$ kJ/kg
$P_2v_2 = 6.68$ kJ/kg
($h_2 = 95.47$ kJ/kg)


T
v
Critical point
p_1 = 0.8 MPa
p_2 =0.12 MPa
P_2 = const. > P_1
P_1 = const
1
2
COMPRESSED LIQUID REGION
Saturated liquid line
Saturated vapor line
SATURATED LIQUID–VAPOR REGION
SUPERHEATED VAPOR REGION

From table A-12, for saturated refrigerant-134a – Pressure table (p_1 = 0.8 MPa) , T_1 = (T_{sat} @ p_1 = 0.8 Mpa) = 31.31°C, $v_1 = v_f$ = 0.0008458 m^3/kg, $u_1 = u_f$ = 94.79 kJ /kg& $h_1 = h_f$ = 95.47 kJ /kg. and for (p_2 = 0.12 MPa), T_2 = (T_{sat} @ p_2 = 0.12 Mpa) = -22.32°C, v_f = 0.0007324 m^3/kg, v_g= 0.16212 m^3/kg, u_f = 22.40 kJ /kg , u_g = 217.51kJ /kg , h_f = 22.49 kJ /kg & h_g = 236.97 kJ /kg.

$\Delta T = T_2 - T_1$ = -22.32°C - 31.31°C = -53.63°C

But, as known for throttling process

$$h_2 \approx h_1 \qquad kJ/kg \qquad (5\text{-}41)$$

$h_2 = h_1$ = 95.47 kJ /kg.

$But,\ h_2 = h_f + x_2 (h_g - h_f)$

$Hence,\ x_2 = (h_2 - h_f)/(h_g - h_f)$

$i.e.,\ x_2$ =(95.47- 22.49)/(236.97- 22.49)

$i.e., x_2 = 0.34$

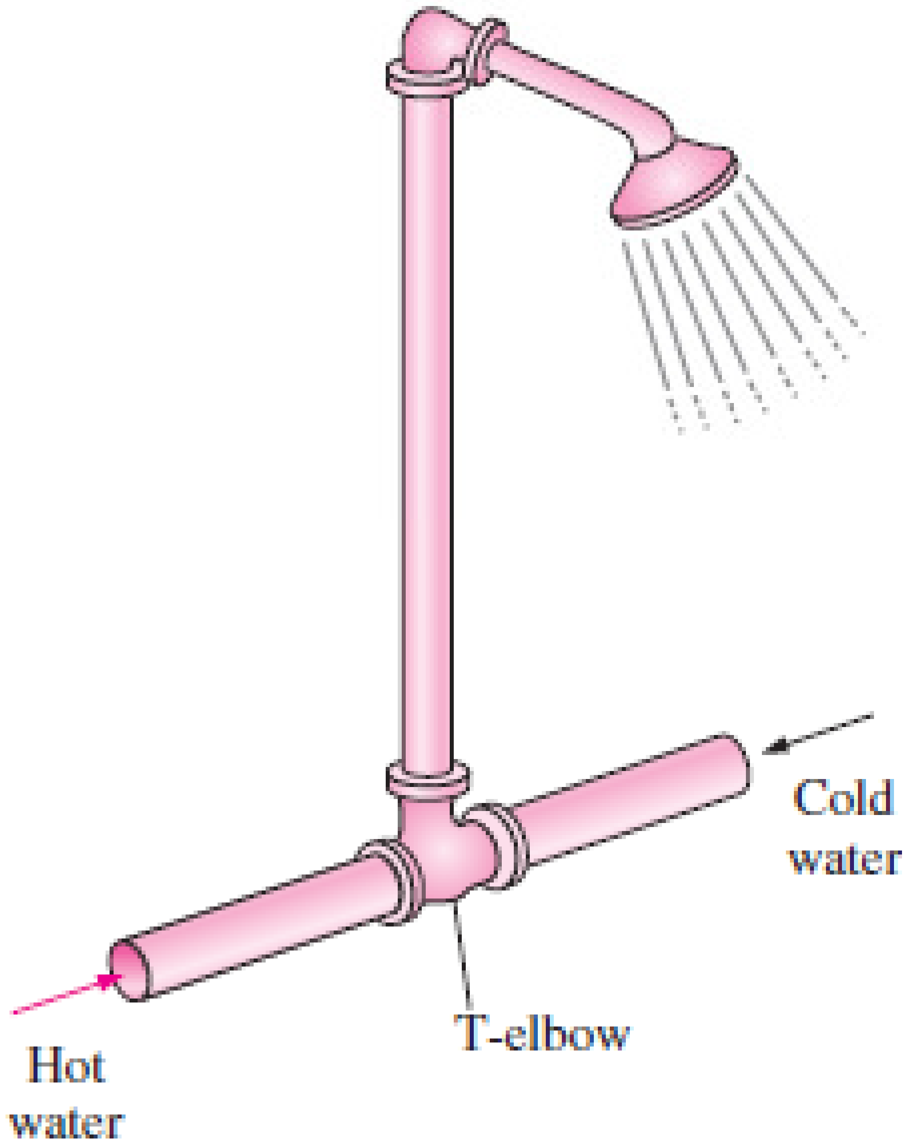
$But, v_2 = v_f + x_2 (v_g - v_f)$

$Hence, v_2 = 0.0008458 + 0.34 (0.16212 - 0.0008458) = 0.05568 \text{ m}^3/\text{kg}$

$But, u_2 = u_f + x_2 (u_g - u_f)$

$Hence, u_2 = 22.40 + 0.34 (217.51 - 22.40) = 88.7374 \text{ kJ/kg}$

4a Mixing Chambers



Example of mixing chambers

The conservation of mass principle for a mixing chamber requires that the sum of the incoming mass flow rates equal the mass flow rate of the outgoing mixture.

Mixing chambers are usually:

a) well insulated ($q \approx 0$)

b) *do not* involve any kind of work ($w = 0$).

c) *kinetic and potential energies* of fluid streams are usually negligible ($ke \approx 0$, $pe \approx 0$).

EXAMPLE 5–9 Mixing of Hot and Cold Waters in a Shower

Consider an ordinary shower where hot water at 60℃ is mixed with cold water at 10℃. If it is desired that a steady stream of warm water at 40℃ be supplied, determine the ratio of the mass flow rates of the hot to cold water. Assume the heat losses from the mixing chamber to be negligible and the mixing to take place at a pressure of 200 kPa.

Solution:

$$\cancel{\dot{Q}}^{0} - \cancel{\dot{W}}^{0} = \underbrace{\sum_{\text{out}} \dot{m}\left(h + \cancel{\frac{V^2}{2}}^{0} + \cancel{gz}^{0}\right)}_{\text{for each exit}} - \underbrace{\sum_{\text{in}} \dot{m}\left(h + \cancel{\frac{V^2}{2}}^{0} + \cancel{gz}^{0}\right)}_{\text{for each inlet}}$$

(5-37)

hot water at 200 kPa & 60℃ is mixed with cold water at 200 kPa & 10℃ to form mixed stream at 200 kPa & 40℃.

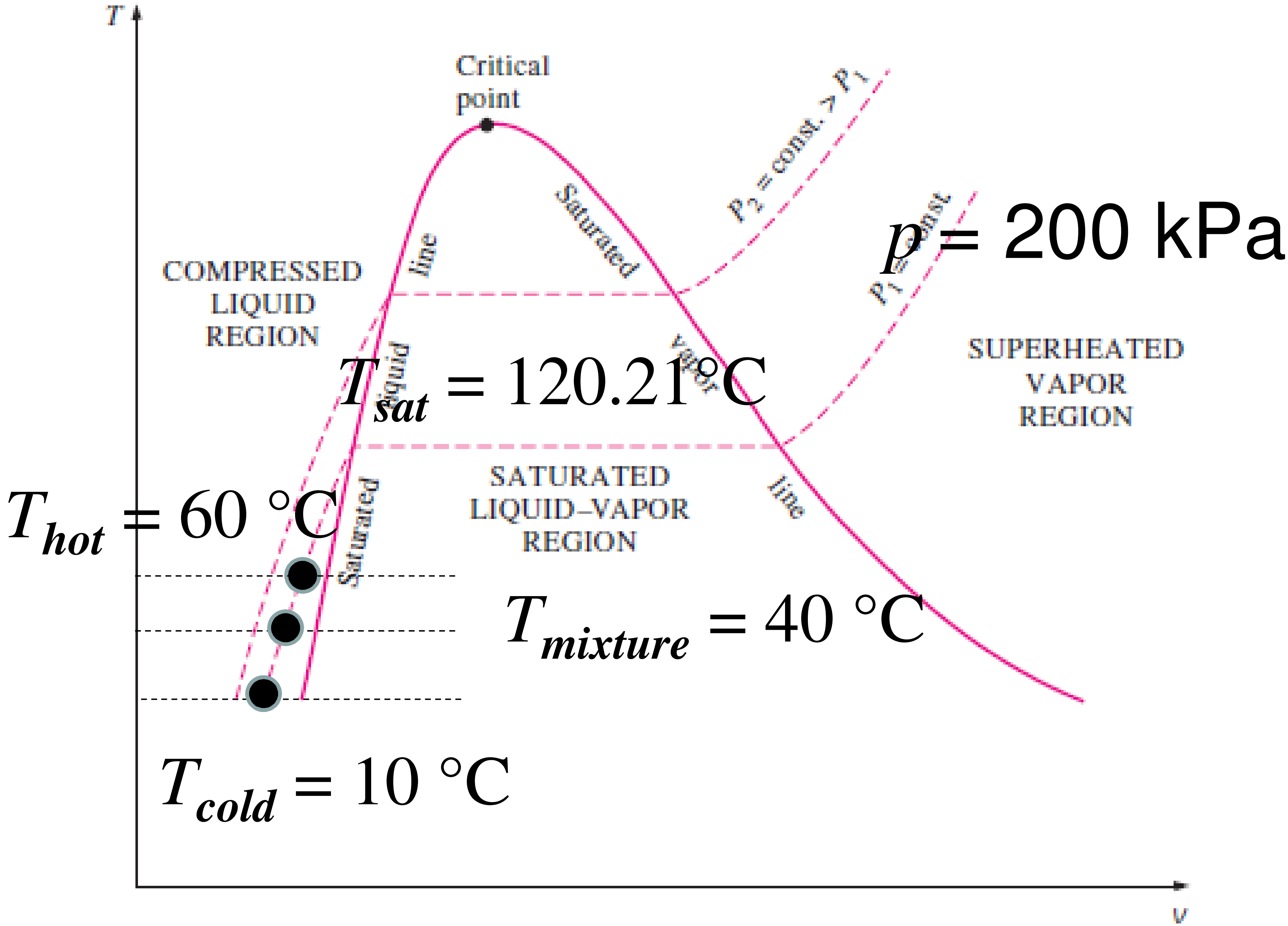
$$Hence,\ (m_{hot} * h_{hot}) + (m_{cold} * h_{cold}) = (m_{mixture} * h_{mixture}) \qquad \textbf{(\#-7)}$$

$$But,\ m_{\boldsymbol{mixture}} = m_{\boldsymbol{hot}} + m_{\boldsymbol{cold}} \qquad \textbf{(\#-8)}$$

Dividing both sides of Eq. (#-7) by $m_{\boldsymbol{cold}}$ to get:

$$Hence,\ [(m_{\boldsymbol{hot}} / m_{\boldsymbol{cold}})^* h_{\boldsymbol{hot}}] + h_{\boldsymbol{cold}} = [(m_{\boldsymbol{hot}} / m_{\boldsymbol{cold}}) + 1]^* h_{\boldsymbol{mixture}} \qquad \textbf{(\#-9)}$$

From table A-5, at 200 kPa , $T_{\boldsymbol{sat}} =$ 120.21°C.


T
Critical point
P_2 = const. > P_1
P_1 = const
p = 200 kPa
Saturated liquid line
Saturated vapor line
COMPRESSED LIQUID REGION
SATURATED LIQUID–VAPOR REGION
SUPERHEATED VAPOR REGION
T_{sat} = 120.21°C
T_{hot} = 60 °C
$T_{mixture}$ = 40 °C
T_{cold} = 10 °C
v

It is clear that (T_{cold} , $T_{mixture}$ & T_{hot}) < T_{sat} (@ 200 kPa) = 120.21°C.

Hence, cold, hot and mixture straems are all compressed (sub-cooled) liquid.

It is clear from table (A-7) that there is are all compressed (sub-cooled) liquid for water at p = 200 kPa.

Hence, (h_{hot} (@ 200 kPa & 60℃) ≈ (h_f (@ T_{sat} = 60℃) = 251.18 kJ/kg.

Hence, (h_{cold} (@ 200 kPa & 10℃) ≈ (h_f (@ T_{sat} = 10℃) = 42.022 kJ/kg.

Hence, ($h_{mixture}$ (@ 200 kPa & 40℃) ≈ (h_f (@ T_{sat} = 40℃) = 167.53 kJ/kg.

From Eq. (#-9)

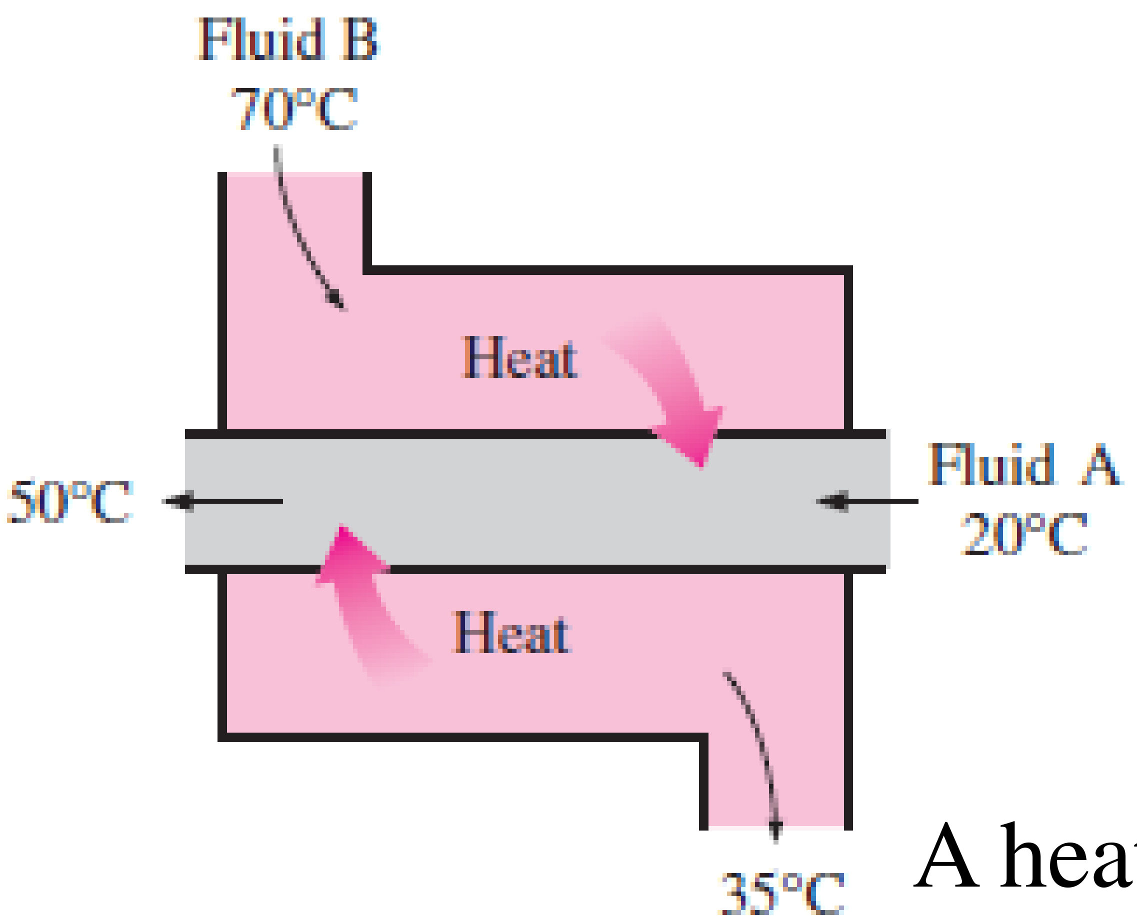
$$[(m_{hot} / m_{cold})^* h_{hot}] + h_{cold} = [(m_{hot} / m_{cold}) + 1]^* h_{mixture}$$

$$Hence,\ [(m_{hot}/m_{cold})*251.18]+42.022 = [(m_{hot}/m_{cold})+1]*167.53$$

$$Hence,\ (m_{hot}/m_{cold})*[251.18-167.53] = [167.53-42.022]$$

$$Hence,\ m_{hot}/m_{cold} = [251.18-167.53] = [167.53-42.022] = 1.5$$

4b Heat Exchangers



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

The simplest form of a heat exchanger is a *double-tube (also called tube and-shell) heat exchanger,* shown in the above figure.

Under steady operation, the <u>mass flow rate of each</u> fluid <u>stream</u> flowing through a heat exchanger <u>remains constant</u>.

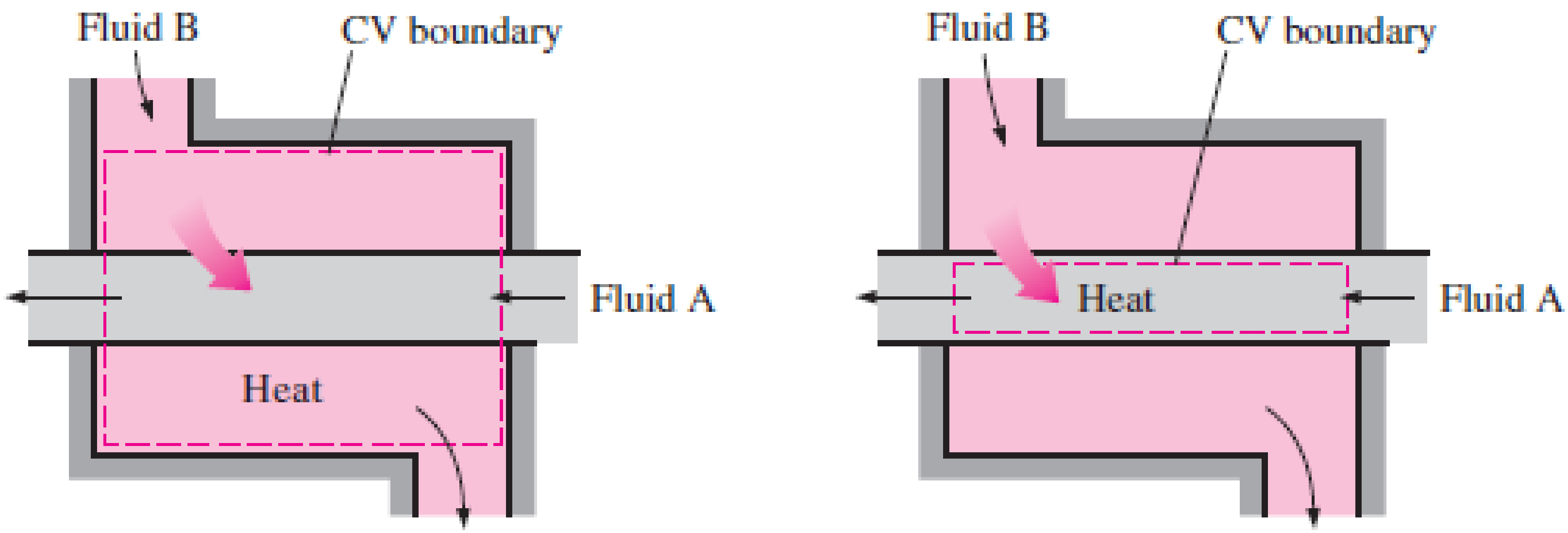
Heat exchangers typically:

a) involve no work interactions ($w=0$) *and*

b) *Negligible* kinetic and potential energy changes ($\Delta ke \approx 0$, $\Delta pe \approx 0$) for each fluid stream.

When the **entire** heat exchanger is **selected as** the **control volume** ; $\dot{Q}$,becomes zero.

If, however, only **one of** the **fluids** is **selected as** the **control volume**, then heat will cross this boundary as it flows from one fluid to the other and; $\dot{Q}$, will not be zero.



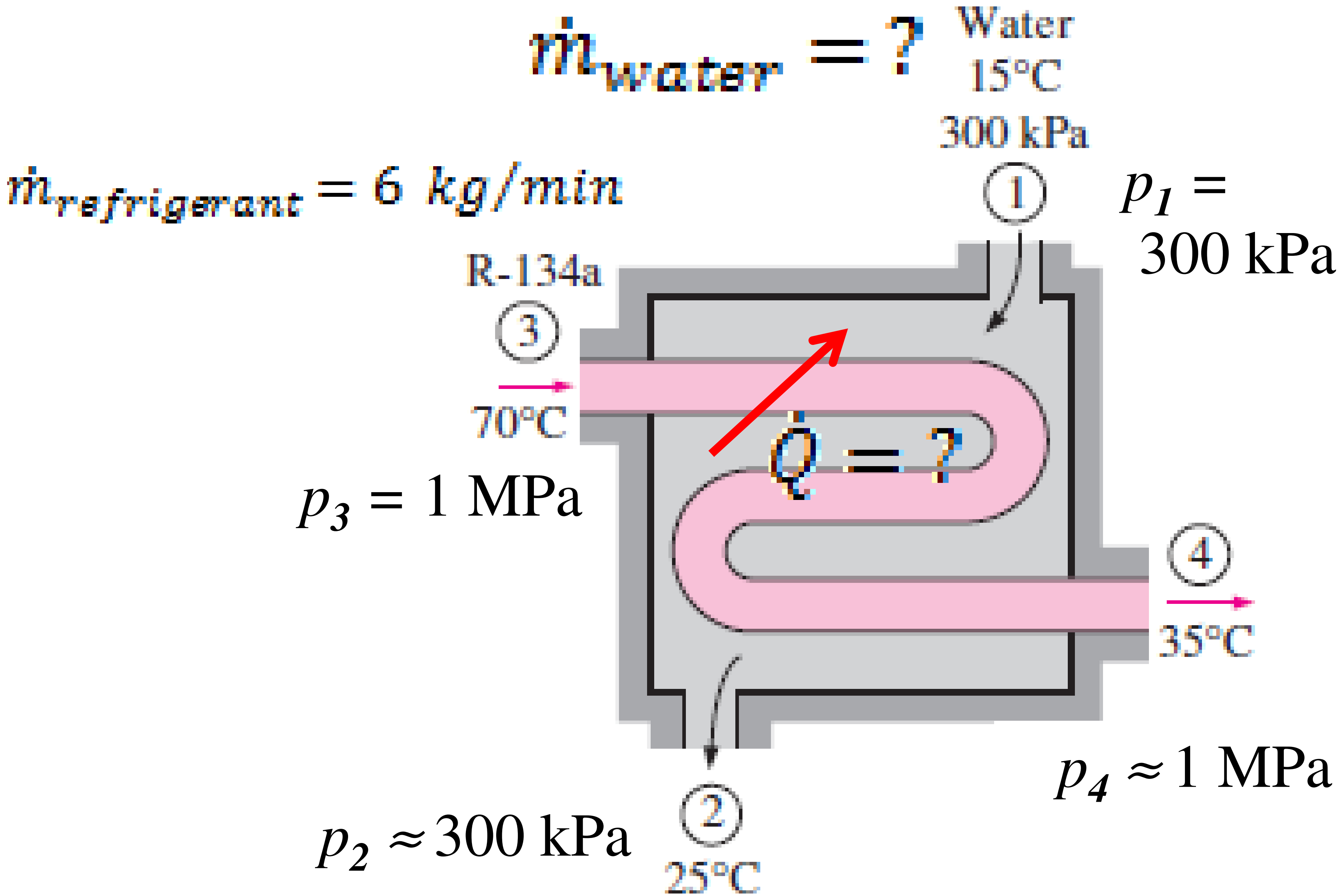
(a) System: Entire heat exchanger ($Q_{CV} = 0$)

(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

EXAMPLE 5–10 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 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 and (b) the heat transfer rate from the refrigerant to water.



$\dot{m}_{water} = ?$
Water
15°C
300 kPa
$\dot{m}_{refrigerant} = 6\ kg/min$
1
$p_1 =$ 300 kPa
R-134a
3
70°C
$\dot{Q} = ?$
$p_3 = 1$ MPa
4
35°C
$p_4 \approx 1$ MPa
2
$p_2 \approx 300$ kPa
25°C

Solution:

$$\dot{m}_{refrigerant} = 6/60\ kg/s = 0.1\,kg/s$$

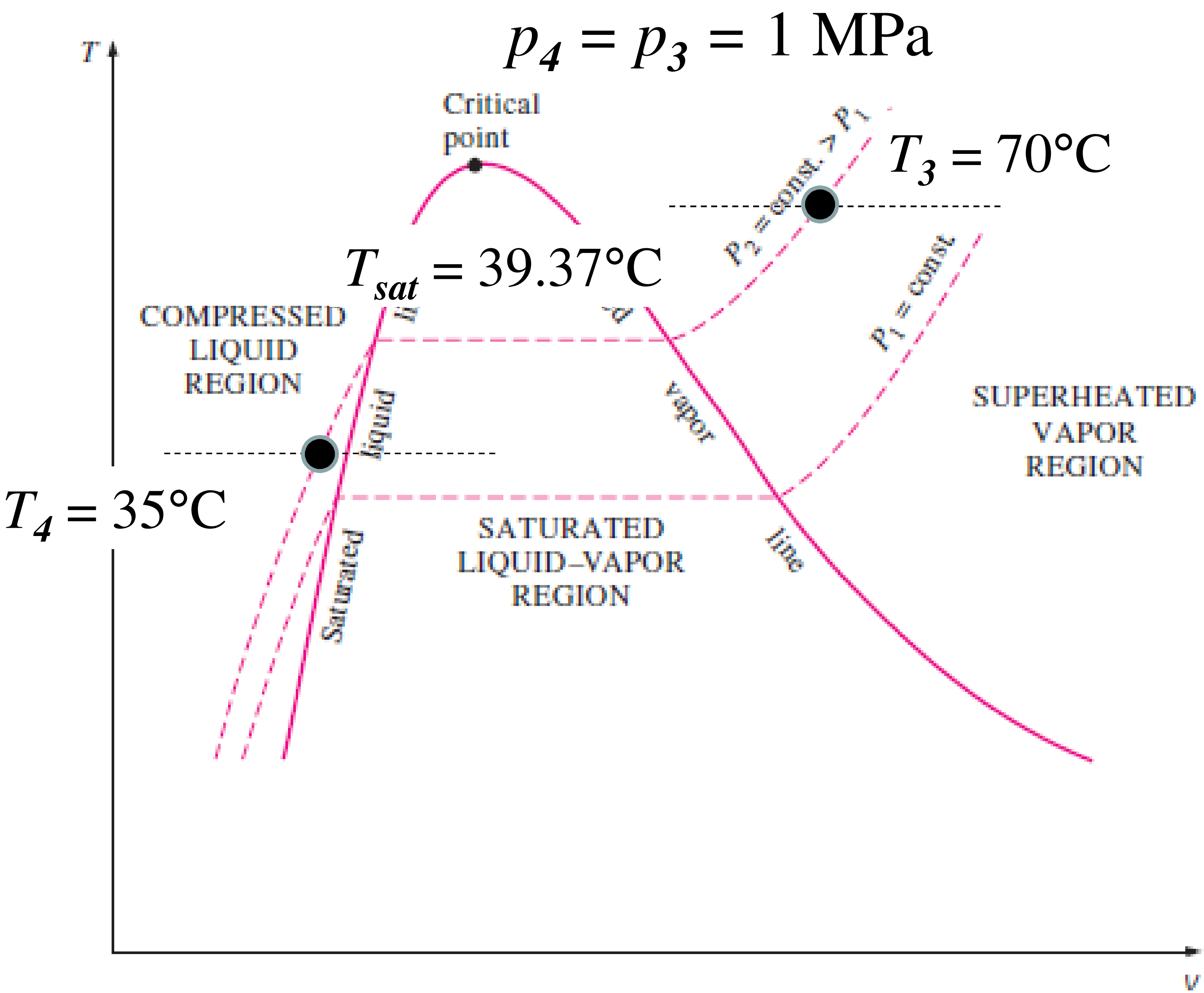
- Refrigerant path:-

From Eq. (5-38):

$$\dot{Q} - \cancel{\dot{W}}^{\,0} = \dot{m}_{refrigerant}\left[h_4 - h_3 + \cancel{\frac{V_4^2 - V_3^2}{2}}^{\,0} + \cancel{g(z_4 - z_3)}^{\,0}\right] \quad \textbf{(5-38)}$$

Hence,

$$\dot{Q} = \dot{m}_{refrigerant}\left[h_4 - h_3\right] \quad \textbf{(\#-10)}$$



$p_4 = p_3 = 1$ MPa
Critical point
$T_3 = 70°C$
$T_{sat} = 39.37°C$
P_2 = const. > P_1
P_1 = const
COMPRESSED LIQUID REGION
Saturated liquid
Saturated vapor line
SUPERHEATED VAPOR REGION
SATURATED LIQUID–VAPOR REGION
$T_4 = 35°C$
T
v

From table (A-12), Saturated Refrigerant-134a-Pressure table

T_{sat} (@ p = 1 MPa) = 39.37°C

As T_3 (= 70°C) > T_{sat} [@ p= 1 MPa) = 39.37°C]

Hence, state 3 is superheated vapor

From table (A-13), Superheated Refrigerant-134a, at p = 1 MPa & T = 70°C

h_3 = 303.85 kJ/kg

As T_4 (= 35°C) < T_{sat} [@ p= 1 MPa) = 39.37°C]

Hence, state 4 is sub-cooled (compressed) liquid

From table (A-11), Saturated Refrigerant-134a-Temperature table, $h_4 = h_f$ (at T_{sat} = 35°C) ≈ (99.40+102.33)/2 = 100.865 kJ/kg

$$\dot{Q} = 0.1\left(\frac{kg}{s}\right)[100.865 - 303.85]\frac{kJ}{kg}$$

Hence, $\dot{Q} = -20.3\ kW$

$$\dot{Q} = 20.3 \times 60 \frac{kJ}{min} = 1218 \frac{kJ}{min}$$

- Water path:-

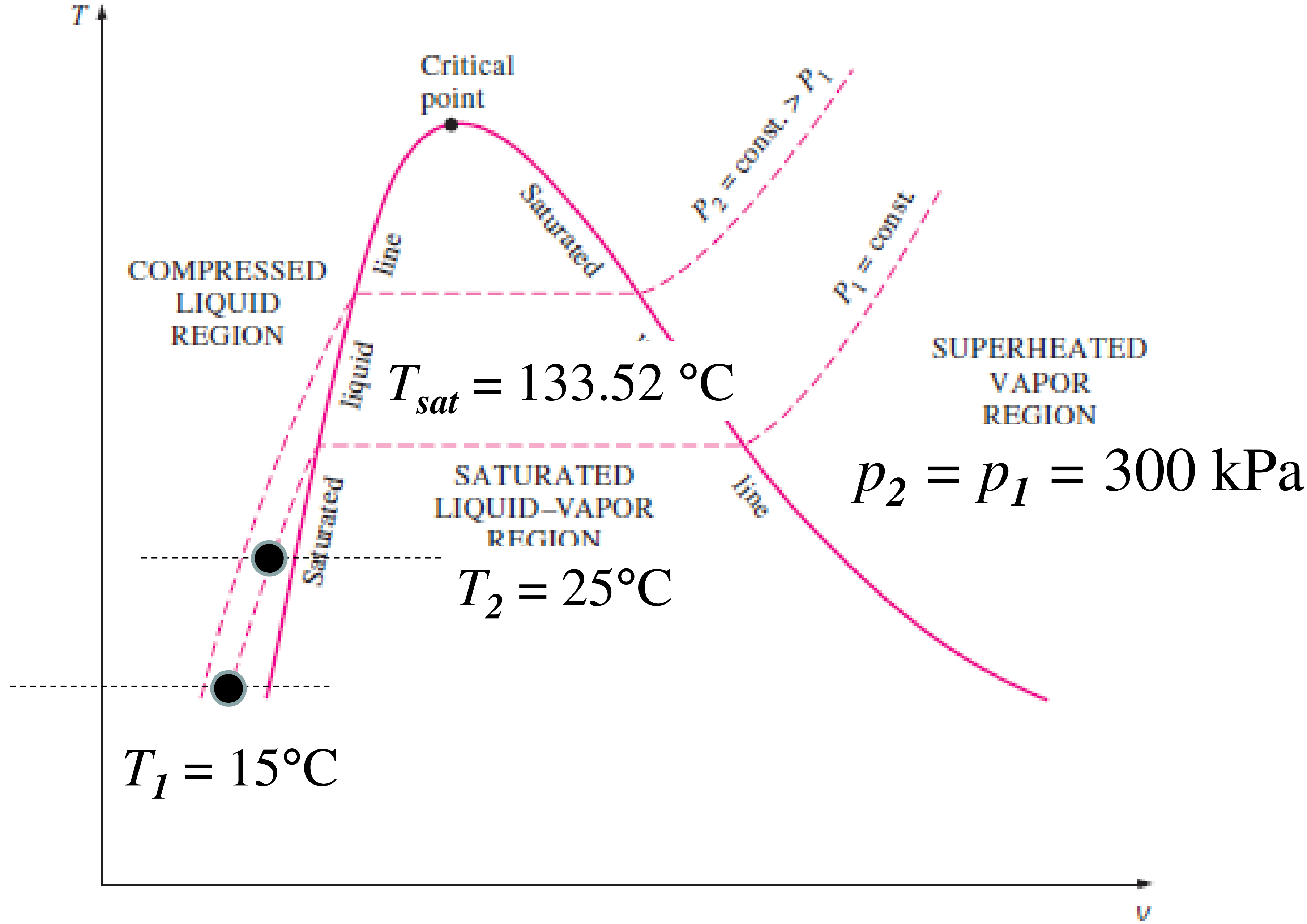
From Eq. (5-38):

$$\dot{Q} - \cancel{\dot{W}} = \dot{m}_{water}\left[h_2 - h_1 + \cancel{\frac{V_2^2 - V_1^2}{2}} + \cancel{g(z_2 - z_1)}\right] \qquad \textbf{(5-38)}$$

0 0 0

Hence,

$$\dot{Q} = \dot{m}_{water}[h_2 - h_1] \qquad \textbf{(\#-11)}$$


T
v
Critical point
COMPRESSED LIQUID REGION
Saturated liquid line
Saturated vapor line
SATURATED LIQUID–VAPOR REGION
SUPERHEATED VAPOR REGION
P2 = const. > P1
P1 = const
Tsat = 133.52 °C
p2 = p1 = 300 kPa
T2 = 25°C
T1 = 15°C

From table (A-5), Saturated water-Pressure table

T_{sat} (@ p = 300 KPa) = 133.52°C

$As\ T_1$ (= 15°C)< T_{sat} [@ p=300 kPa) =133.52°C]

Hence, state 1 is sub-cooled (compressed) liquid

From table (A-4), Saturated water-temperature table , at T = 15°C

$h_1 = h_f$ $(at\ \ T = 15°C)$ = 62.982 kJ/kg

$As\ T_2$ (= 25°C)< T_{sat} [@ p=300 kPa) =133.52°C]

Hence, state 2 is sub-cooled (compressed) liquid

From table (A-4), Saturated water-temperature table , at T = 25°C

$$h_2 = h_f\ \ (at\ \ T = 25°\text{C}) = 104.83\ \text{kJ/kg}$$

But, from Eq. (#-11)

$$\dot{Q} = \dot{m}_{water}[h_2 - h_1] \qquad \textbf{(\#-11)}$$

$$h_2 = h_f \ (at \ \ T = 25°\text{C}) = 104.83 \text{ kJ/kg}$$

Hence,

$$20.3\ kW = \dot{m}_{water}\left[104.83\,\frac{kJ}{kg} - 62.982\,\frac{kJ}{kg}\right]$$

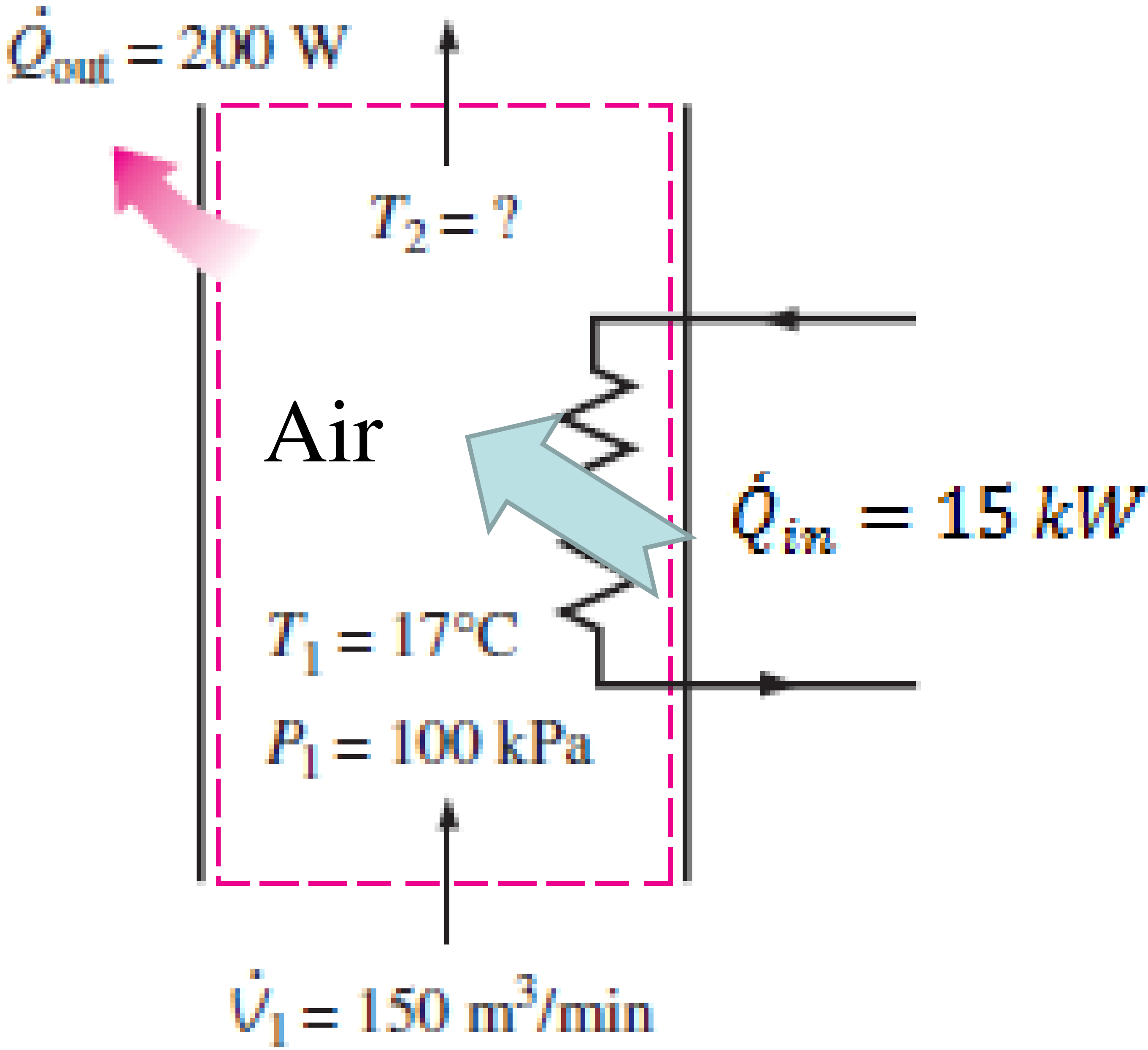
$$\dot{m}_{water} = 0.485\ kg/s$$

$$\dot{m}_{water} = 0.485 \times 60\,\frac{kg}{min} = 29.1\,\frac{kg}{min}$$

5 Pipe and Duct Flow

EXAMPLE 5–11 Electric Heating of Air in a House

The electric heating systems used in many houses consist of a simple duct with resistance heaters. Air is heated as it flows over resistance wires. Consider a 15-kW electric heating system. Air enters the heating section at 100 kPa and 17°C with a volume flow rate of 150 m^3/min. If heat is lost from the air in the duct to the surroundings at a rate of 200 W, determine the exit temperature of air.



$\dot{Q}_{out} = 200$ W
$T_2 = ?$
Air
$\dot{Q}_{in} = 15\ kW$
$T_1 = 17°C$
$P_1 = 100$ kPa
$\dot{V}_1 = 150\ m^3/min$

Solution:

From Eq. (5-38) for a single-stream (one-inlet and one-outlet) steady-flow system

$$\dot{Q} - \dot{W} = \dot{m}\left[h_2 - h_1 + \underbrace{\cancel{\frac{V_2^2 - V_1^2}{2}}}_{0} + \underbrace{\cancel{g(z_2 - z_1)}}_{0}\right] \qquad \textbf{(5-38)}$$

Hence, (5-38) can be re-written as:

$$(\dot{Q} - \underbrace{\cancel{\dot{W}}}_{0})/\dot{m} = (h_2 - h_1) \qquad \textbf{(\#-12)}$$

From equation of state, $h_2 = h_f$ *(at T* $= 25°C) = 104.83$ kJ/kg

From equation of state (at state 1), $p_1/\rho_1 = RT_1$

From Table (A-2), $R = 0.2870\ kJ/kg.K$

$T_1 = (17 + 273)\ K = 290\ K$

$\rho_1 = p_1/RT_1 = 100\ kPa/(0.2870\ kJ/kg.K * 290\ K) = 1.2\ kg/m^3$

Hence,

$$\dot{m} = \rho_1 \dot{V}_1 = 1.2\frac{kg}{m^3} \times (150\frac{m^3}{\mathrm{min}})$$

i.e.,

$$\dot{m} = \rho_1 \dot{V}_1 = 180\frac{kg}{\mathrm{min}} = 3.0\frac{kg}{s}$$

$$\dot{Q} = \dot{Q}_{in} - \dot{Q}_{out} = 15kW - 200W$$

i.e., $$\dot{Q} = 14.8\,kW$$

$$h_2 - h_1 = c_p(T_2 - T_1)$$

(#-12)

From Table (A-12), c_p *= 1.005 kJ/kg.K*

By substituting from Eq. (#-12) in Eq. (#-2) to get

$[14.8\ kW/(3\ kg/s)] = (1.005\ kJ/kg.K)*(T_2 - 17\,°C)$

Hence, $T_2 = 21.9\,°C$

Homework

5–1C, 5–7, 5–8, 5–10, 5–11, 5–12, 5–13, 5–15, 5–16, 5–17, 5–22, 5–24, 5–30, 5–32, 5–34, 5–35, 5–39, 5–34, 5–41, 5–43, 5–44, 5–49, 5–51, 5–53, 5–54, 5–56, 5–57, 5–66, 5–68, 5–71.