ESO201A : THERMODYNAMICS
2021-22 Ist semester
IIT Kanpur

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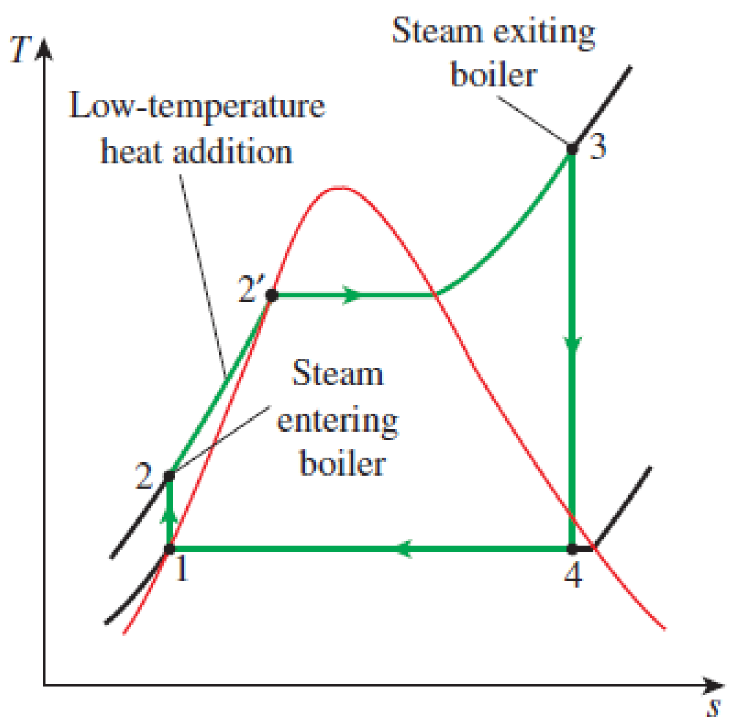
Lecture 30

THE IDEAL REGENERATIVE RANKINE CYCLE

A careful examination of the *T-s* diagram of the Rankine cycle drawn in Fig. reveals that heat is transferred to the working fluid during process 2-2′ at a relatively low temperature. This lowers the average heat-addition temperature and thus the cycle efficiency.

To remedy this shortcoming, we look for ways to raise the temperature of the liquid leaving the pump (called the *feedwater*) before it enters the boiler. One such possibility is to transfer heat to the feedwater from the expanding steam in a counterflow heat exchanger built into the turbine, that is, to use **regeneration**. This solution is also impractical because it is difficult to design such a heat exchanger and because it would increase the moisture content of the steam at the final stages of the turbine.



Ref. Cengel and Boles, 8th Edition (2015)

A practical regeneration process in steam power plants is accomplished by extracting, or "bleeding," steam from the turbine at various points. This steam, which could have produced more work by expanding further in the turbine, is used to heat the feedwater instead. The device where the feedwater is heated by regeneration is called a **regenerator**, or a **feedwater heater (FWH)**.

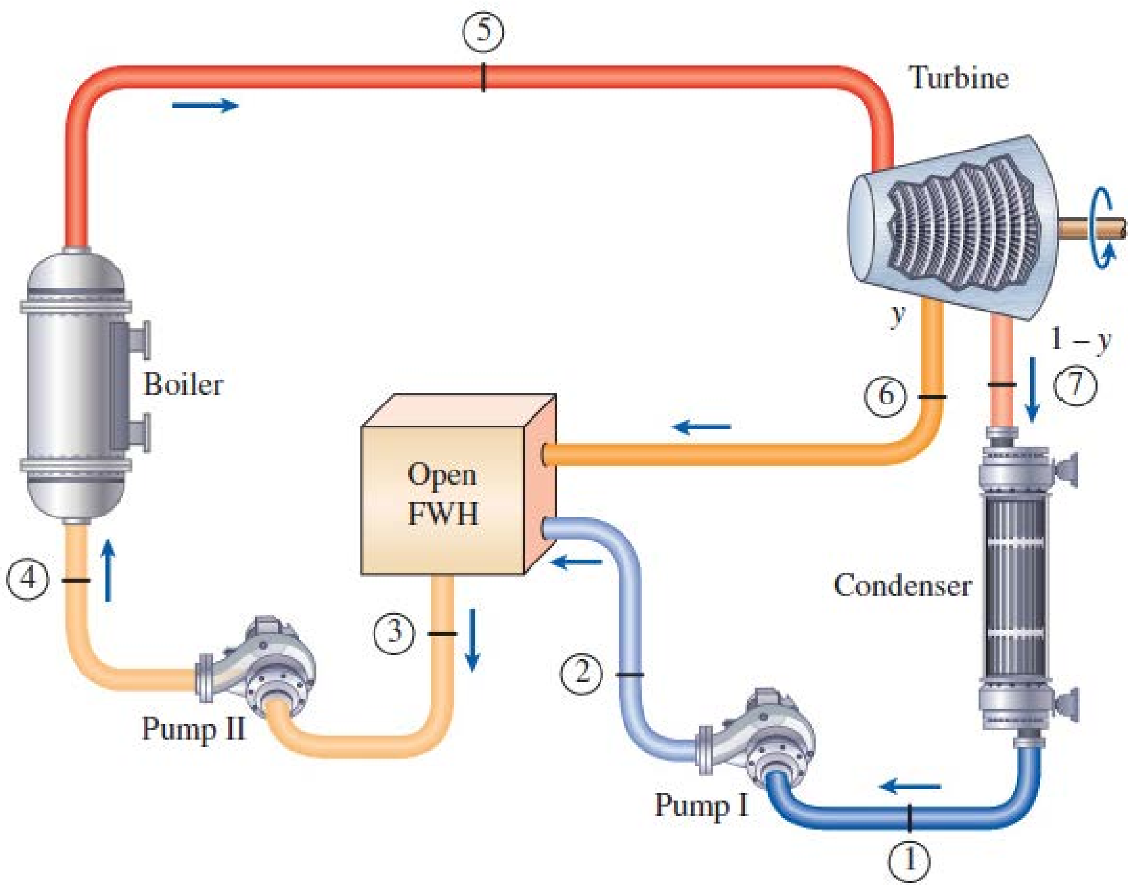
Regeneration not only improves cycle efficiency, but also provides a convenient means of deaerating the feedwater (removing the air that leaks in at the condenser) to prevent corrosion in the boiler. It also helps control the large volume flow rate of the steam at the final stages of the turbine (due to the large specific volumes at low pressures). Therefore, regeneration has been used in all modern steam power plants since its introduction in the early 1920s.

A feedwater heater is basically a heat exchanger where heat is transferred from the steam to the feedwater either by mixing the two fluid streams (open feedwater heaters) or without mixing them (closed feedwater heaters).

Ref. Cengel and Boles, 8th Edition (2015)

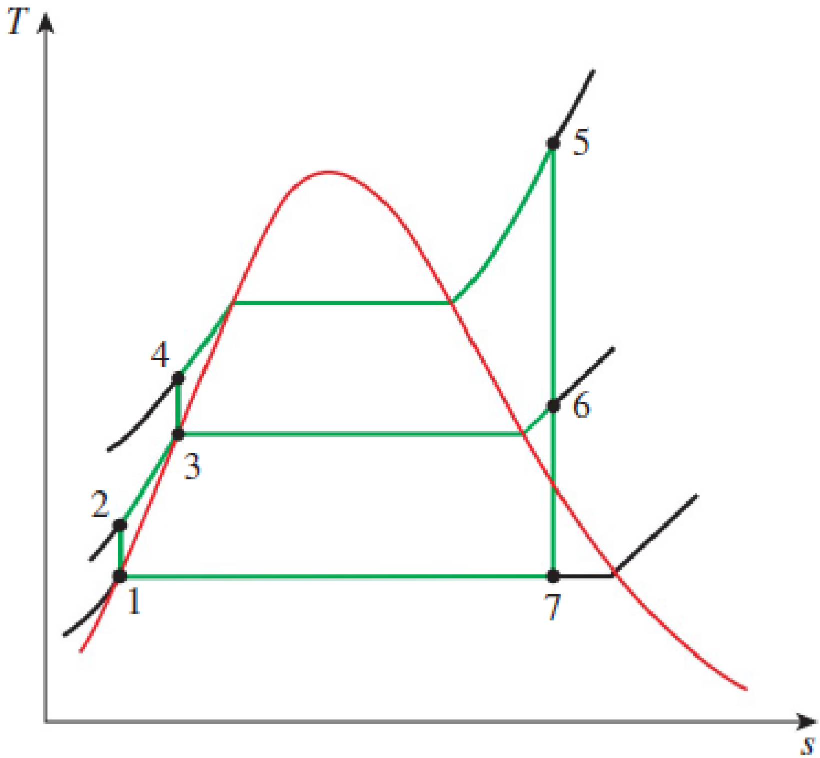
Open Feedwater Heaters

An **open** (or **direct-contact**) **feedwater heater** is basically a *mixing chamber*, where the steam extracted from the turbine mixes with the feed-water exiting the pump. Ideally, the mixture leaves the heater as a saturated liquid at the heater pressure. The schematic of a steam power plant with one open feedwater heater (also called *single-stage regenerative cycle*) and the *T-s* diagram of the cycle are shown in Fig.



Ref. Cengel and Boles, 8th Edition (2015)

For each 1 kg of steam leaving the boiler, y kg expands partially in the turbine and is extracted at state 6. The remaining $(1 - y)$ kg expands completely to the condenser pressure.



$$q_{in} = h_5 - h_4$$

$$q_{out} = (1 - y)(h_7 - h_1)$$

$$w_{turb,out} = (h_5 - h_6) + (1 - y)(h_6 - h_7)$$

$$w_{pump,in} = (1 - y)w_{pump\ I,in} + w_{pump\ II,in}$$

$$y = \dot{m}_6/\dot{m}_5 \qquad \text{(fraction of steam extracted)}$$

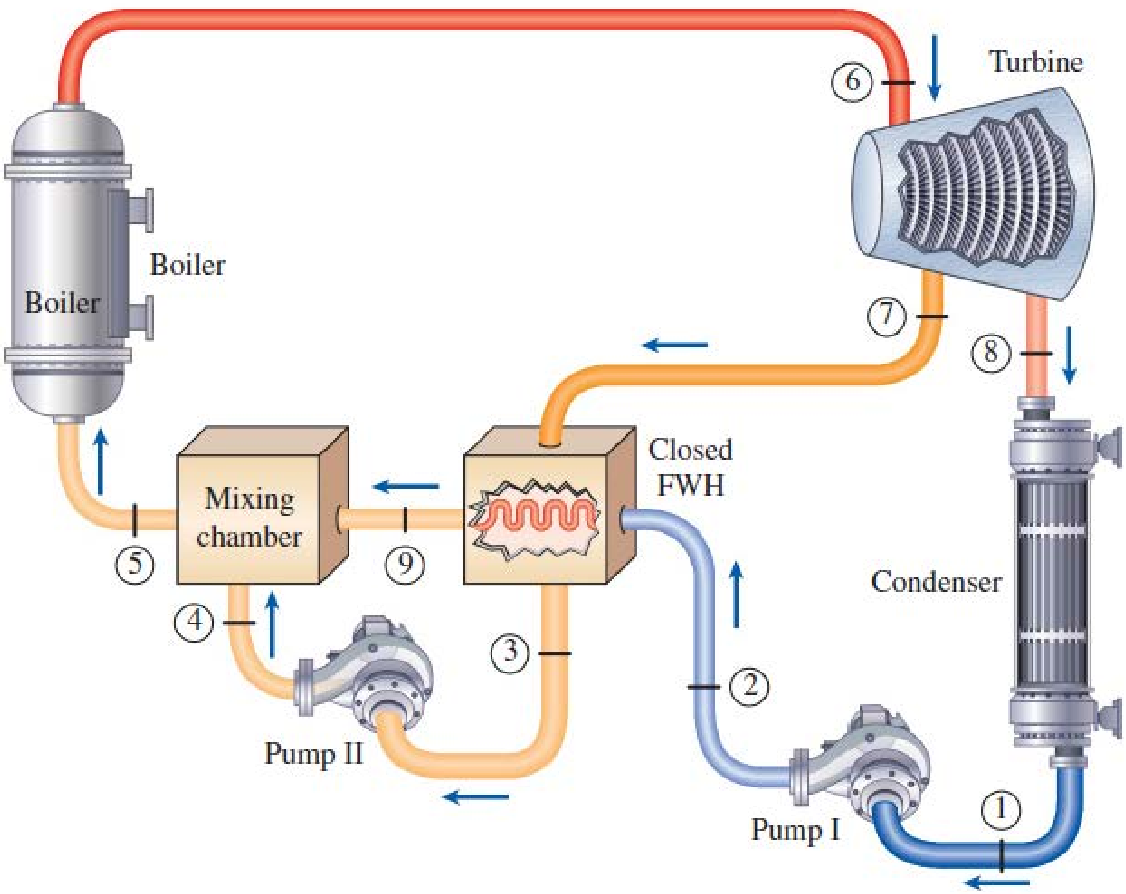
$$w_{pump\ I,in} = v_1(P_2 - P_1)$$

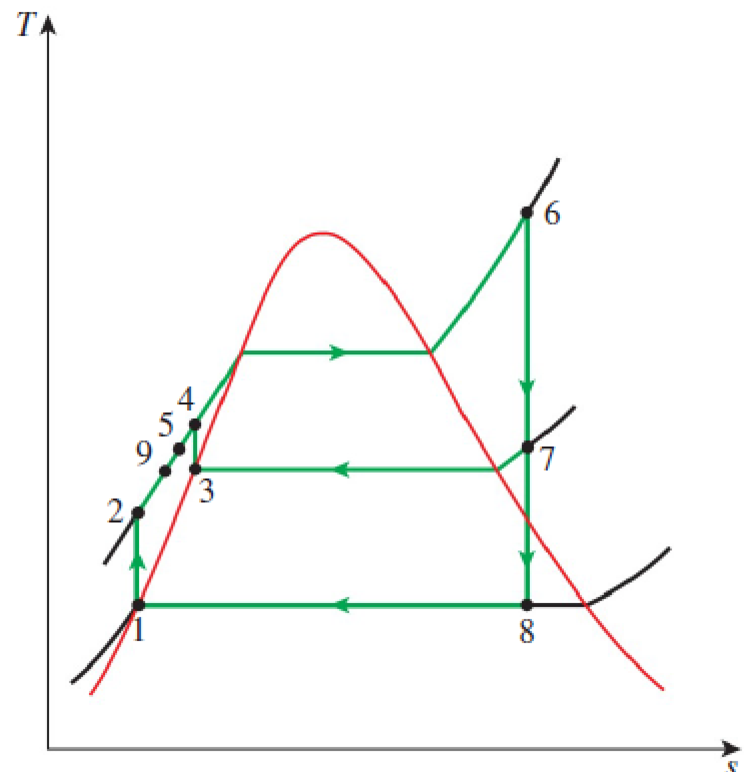
$$w_{pump\ II,in} = v_3(P_4 - P_3)$$

Ref. Cengel and Boles, 8th Edition (2015)

Closed Feedwater Heaters

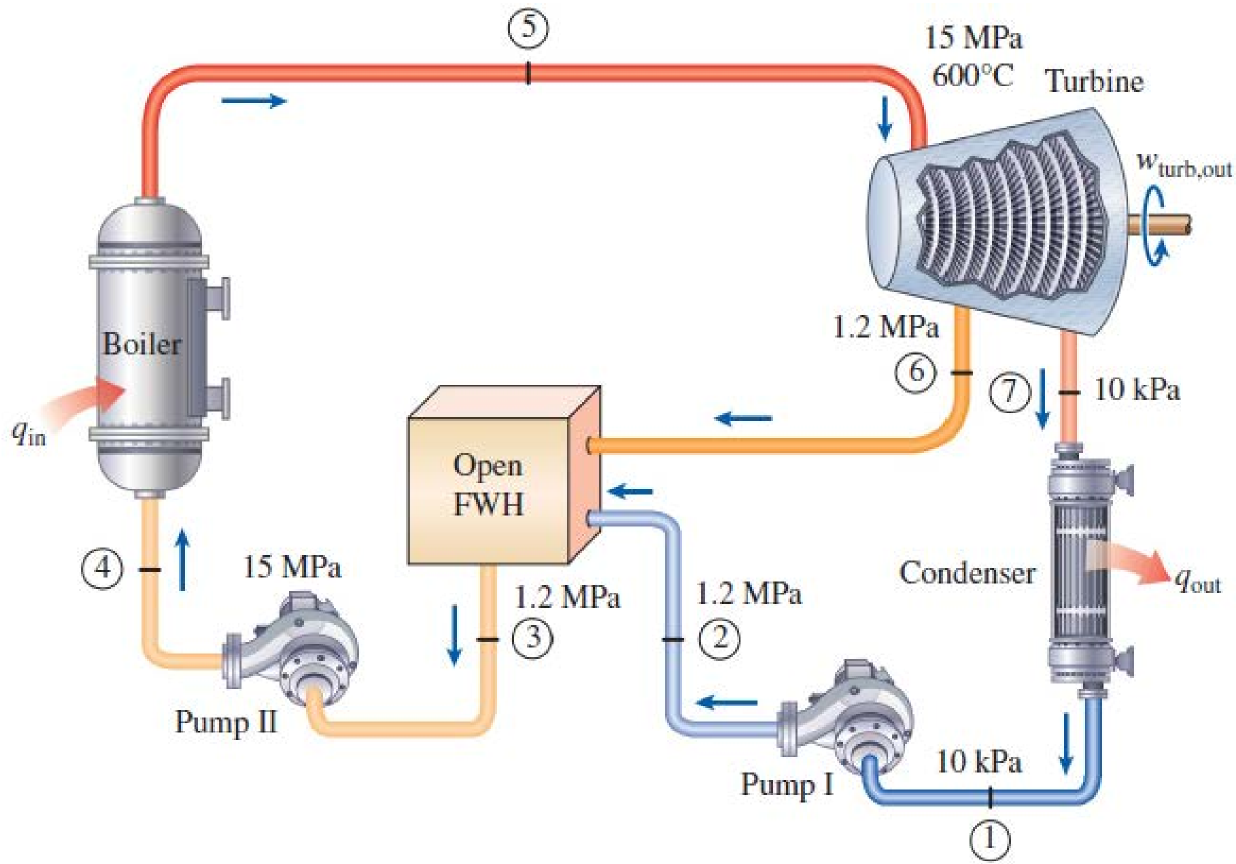
Another type of feedwater heater frequently used in steam power plants is the **closed feedwater heater**, in which heat is transferred from the extracted steam to the feedwater without any mixing taking place. The two streams now can be at different pressures, since they do not mix. The schematic of a steam power plant with one closed feedwater heater and the *T-s* diagram of the cycle are shown in Fig.



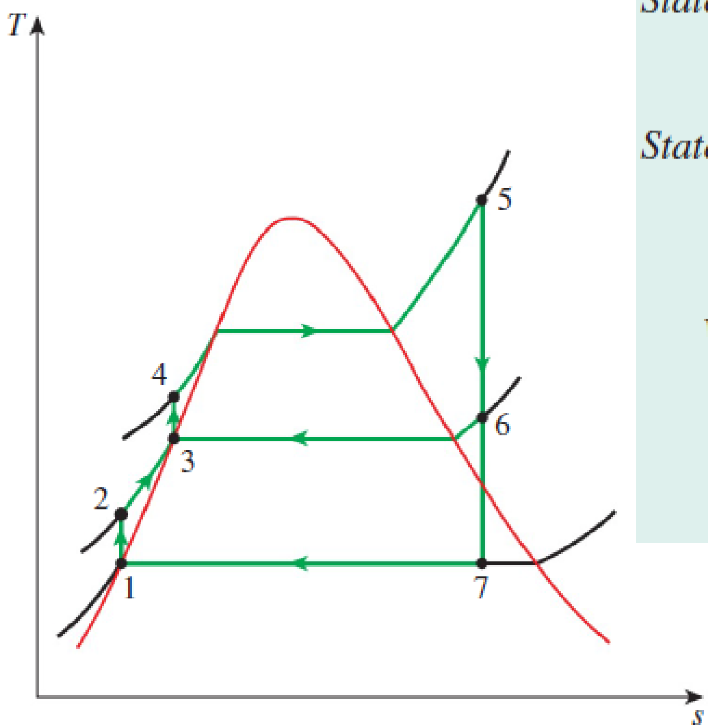


Ref. Cengel and Boles, 8th Edition (2015)

Consider a steam power plant operating on the ideal regenerative Rankine cycle with one open feedwater heater. Steam enters the turbine at 15 MPa and 600°C and is condensed in the condenser at a pressure of 10 kPa. Some steam leaves the turbine at a pressure of 1.2 MPa and enters the open feedwater heater. Determine the fraction of steam extracted from the turbine and the thermal efficiency of the cycle.



Ref. Cengel and Boles, 8th Edition (2015)



Ref. Cengel and Boles, 8th Edition (2015)

State 1: $\left.\begin{array}{l} P_1 = 10 \text{ kPa} \\ \text{Sat. liquid} \end{array}\right\} \begin{array}{l} h_1 = h_{f\,@\,10\,\text{kPa}} = 191.81 \text{ kJ/kg} \\ v_1 = v_{f\,@\,10\,\text{kPa}} = 0.00101 \text{ m}^3\text{/kg} \end{array}$

State 2: $P_2 = 1.2 \text{ MPa}$

$s_2 = s_1$

$$w_{\text{pump I,in}} = v_1(P_2 - P_1) = (0.00101 \text{ m}^3\text{/kg})[(1200 - 10) \text{ kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa·m}^3}\right)$$

$$= 1.20 \text{ kJ/kg}$$

$$h_2 = h_1 + w_{\text{pump I,in}} = (191.81 + 1.20) \text{ kJ/kg} = 193.01 \text{ kJ/kg}$$

State 3: $\left.\begin{array}{l} P_3 = 1.2 \text{ MPa} \\ \text{Sat. liquid} \end{array}\right\} \begin{array}{l} v_3 = v_{f\,@\,1.2\,\text{MPa}} = 0.001138 \text{ m}^3\text{/kg} \\ h_3 = h_{f\,@\,1.2\,\text{MPa}} = 798.33 \text{ kJ/kg} \end{array}$

State 4: $P_4 = 15 \text{ MPa}$

$s_4 = s_3$

$$w_{\text{pump II,in}} = v_3(P_4 - P_3)$$

$$= (0.001138 \text{ m}^3\text{/kg})[(15{,}000 - 1200) \text{ kPa}]\left(\frac{1 \text{ kJ}}{1 \text{ kPa·m}^3}\right)$$

$$= 15.70 \text{ kJ/kg}$$

$$h_4 = h_3 + w_{\text{pump II,in}} = (798.33 + 15.70) \text{ kJ/kg} = 814.03 \text{ kJ/kg}$$

$$\textit{State 5:}\quad \left.\begin{array}{l} P_5 = 15 \text{ MPa} \\ T_5 = 600^\circ\text{C} \end{array}\right\} \quad \begin{array}{l} h_5 = 3583.1 \text{ kJ/kg} \\ s_5 = 6.6796 \text{ kJ/kg·K} \end{array}$$

$$\textit{State 6:}\quad \left.\begin{array}{l} P_6 = 1.2 \text{ MPa} \\ s_6 = s_5 \end{array}\right\} \quad \begin{array}{l} h_6 = 2860.2 \text{ kJ/kg} \\ (T_6 = 218.4^\circ\text{C}) \end{array}$$

$$\textit{State 7:}\quad P_7 = 10 \text{ kPa}$$

$$s_7 = s_5 \quad x_7 = \frac{s_7 - s_f}{s_{fg}} = \frac{6.6796 - 0.6492}{7.4996} = 0.8041$$

$$h_7 = h_f + x_7 h_{fg} = 191.81 + 0.8041(2392.1) = 2115.3 \text{ kJ/kg}$$

The energy analysis of open feedwater heaters is identical to the energy analysis of mixing chambers. The feedwater heaters are generally well insulated ($\dot{Q} = 0$), and they do not involve any work interactions ($\dot{W} = 0$). By neglecting the kinetic and potential energies of the streams, the energy balance reduces for a feedwater heater to

$$yh_6 + (1 - y)h_2 = 1(h_3)$$

where y is the fraction of steam extracted from the turbine ($= \dot{m}_6/\dot{m}_5$). Solving for y and substituting the enthalpy values, we find

$$y = \frac{h_3 - h_2}{h_6 - h_2} = \frac{798.33 - 193.01}{2860.2 - 193.01} = \mathbf{0.2270}$$

Ref. Cengel and Boles, 8th Edition (2015)

$$q_{in} = h_5 - h_4 = (3583.1 - 814.03)\ \text{kJ/kg} = 2769.1\ \text{kJ/kg}$$

$$q_{out} = (1 - y)(h_7 - h_1) = (1 - 0.2270)(2115.3 - 191.81)\ \text{kJ/kg}$$

$$= 1486.9\ \text{kJ/kg}$$

and

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{1486.9\ \text{kJ/kg}}{2769.1\ \text{kJ/kg}} = \mathbf{0.463 \text{ or } 46.3\%}$$

Discussion This example was worked out in part (c) on slide 5 of lecture 30 for the same pressure and temperature limits but without the regeneration process. A comparison of the two results reveals that the thermal efficiency of the cycle has increased from 43.0 to 46.3 percent as a result of regeneration. The net work output decreased by 171 kJ/kg, but the heat input decreased by 607 kJ/kg, which results in a net increase in the thermal efficiency.

Ref. Cengel and Boles, 8th Edition (2015)

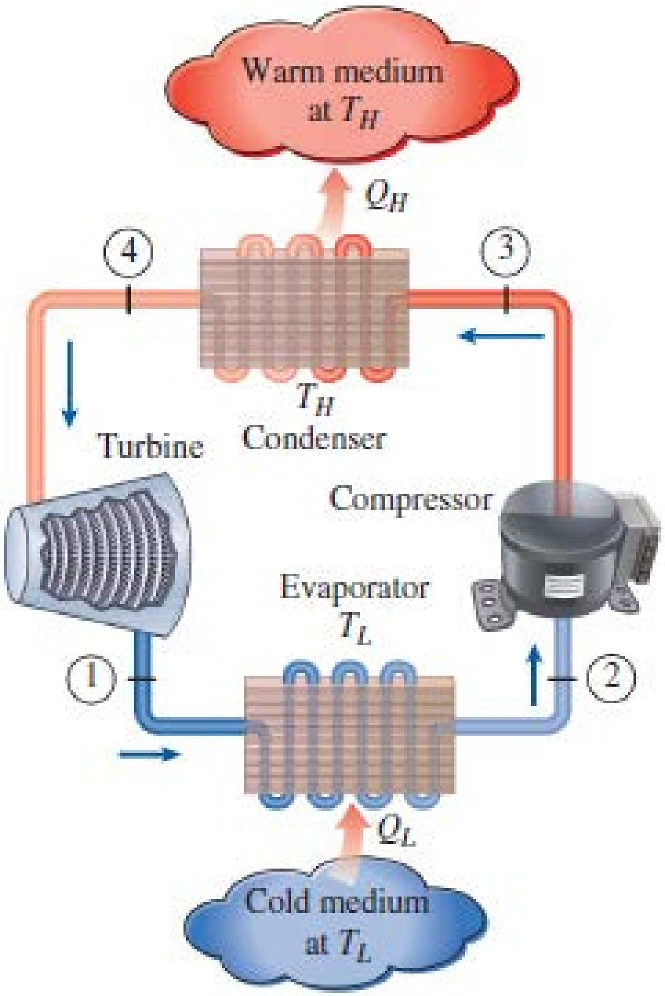
REFRIGERATION CYCLES

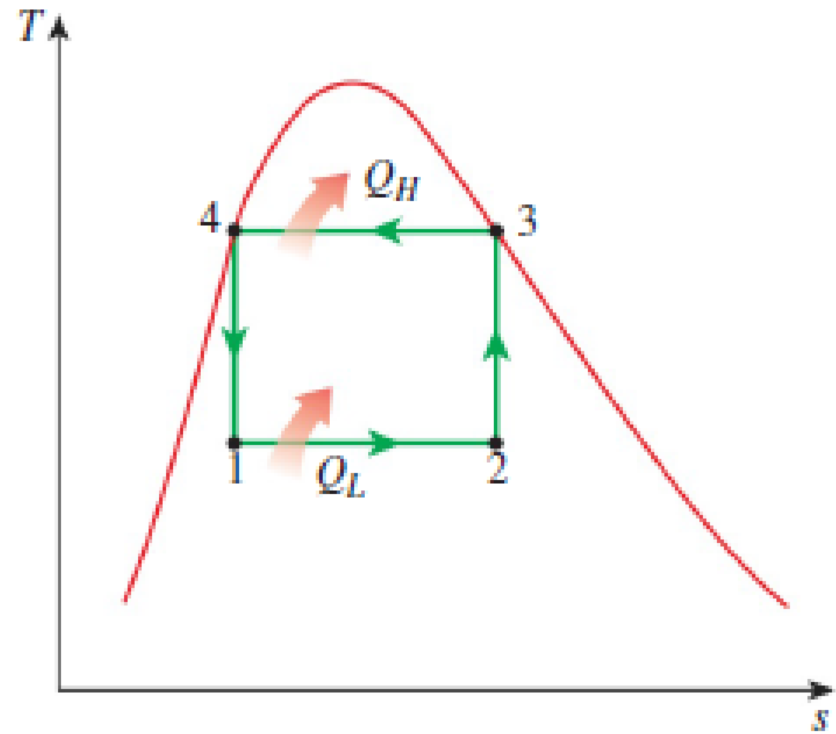
The purpose of refrigeration is the transfer of heat from a lower Temperature region to a higher temperature region (usually the environment).

The cycles on which refrigerators operate are known as refrigeration Cycles.

The most frequently used refrigeration cycle is the vapor-compression refrigeration cycle in which there is a phase Change during the cycle.

Reversed Carnot cycle as ideal cycle for refrigeration ?

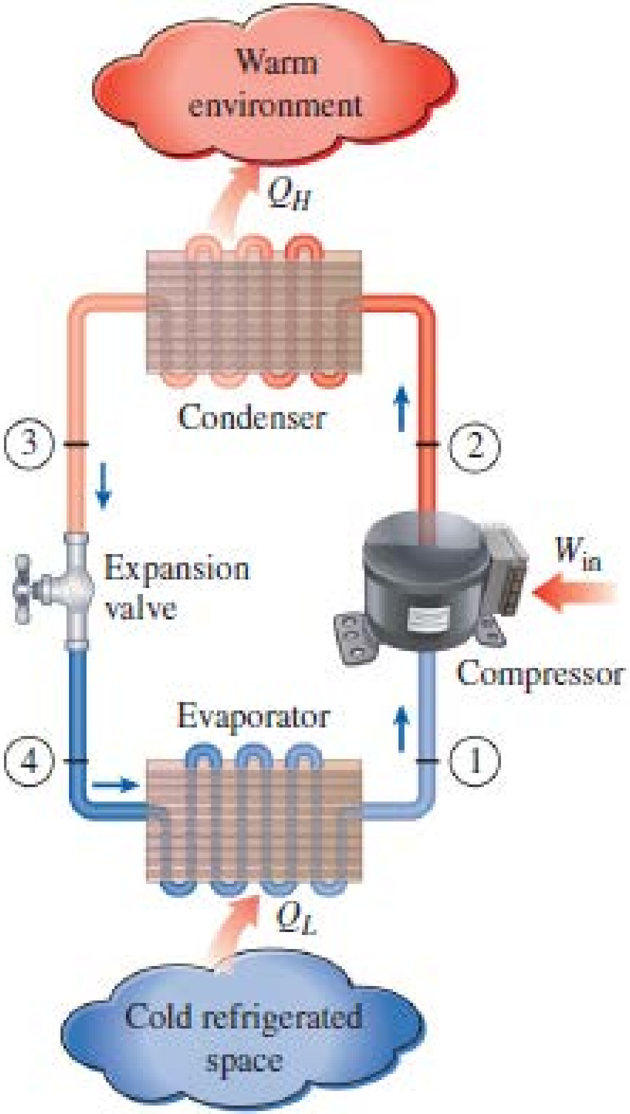


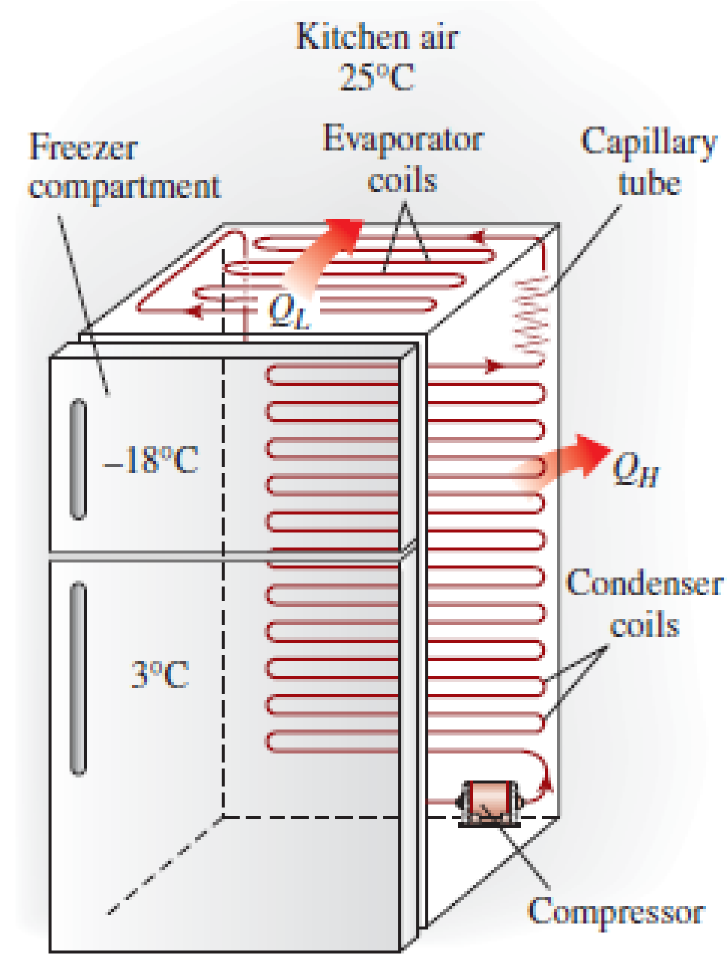


$$\mathrm{COP}_{\mathrm{R,Carnot}} = \frac{1}{T_H/T_L - 1}$$

Ref. Cengel and Boles, 8th Edition (2015)

THE IDEAL VAPOR-COMPRESSION REFRIGERATION CYCLE





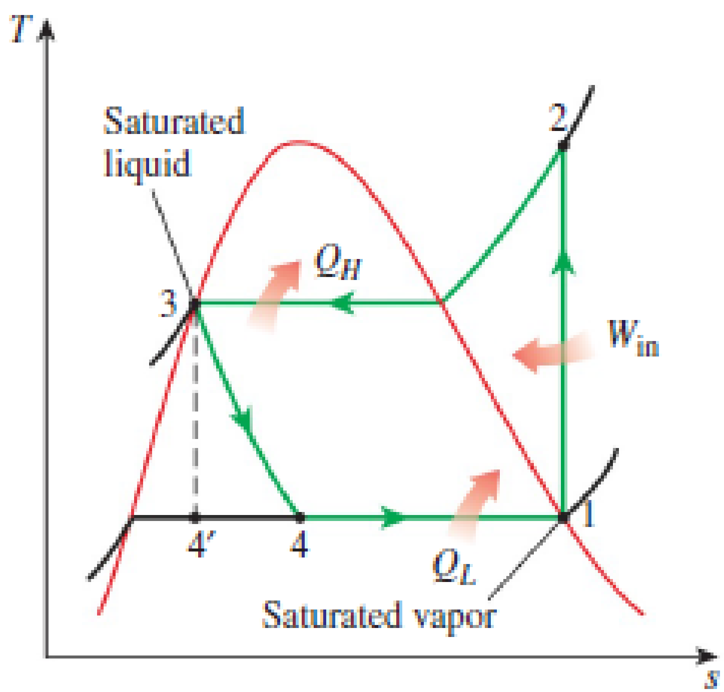
Ref. Cengel and Boles, 8th Edition (2015)

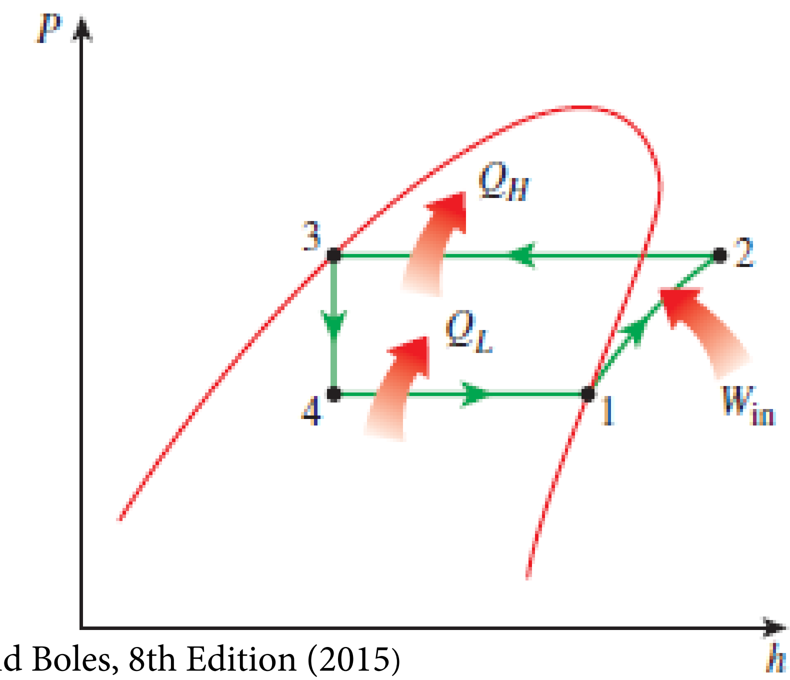
1-2	Isentropic compression in a compressor
2-3	Constant-pressure heat rejection in a condenser
3-4	Throttling in an expansion device
4-1	Constant-pressure heat absorption in an evaporator

$$\mathrm{COP_R} = \frac{q_L}{w_{\mathrm{net,in}}} = \frac{h_1 - h_4}{h_2 - h_1}$$

$$\mathrm{COP_{HP}} = \frac{q_H}{w_{\mathrm{net,in}}} = \frac{h_2 - h_3}{h_2 - h_1}$$

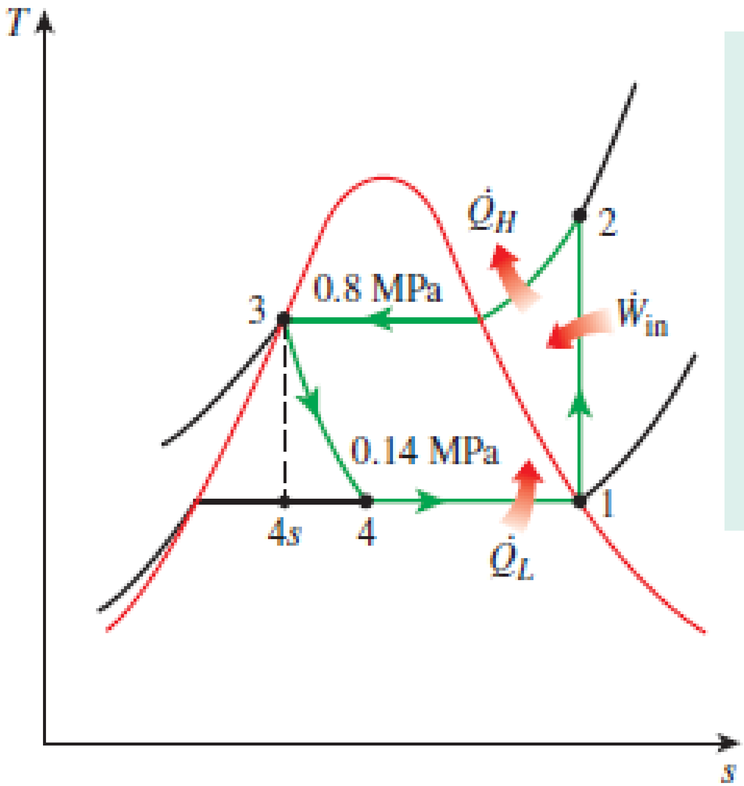
where $h_1 = h_{g\,@\,P_1}$ and $h_3 = h_{f\,@\,P_3}$ for the ideal case.





Ref. Cengel and Boles, 8th Edition (2015)

A refrigerator uses refrigerant-134a as the working fluid and operates on an ideal vapor-compression refrigeration cycle between 0.14 and 0.8 MPa. If the mass flow rate of the refrigerant is 0.05 kg/s, determine (*a*) the rate of heat removal from the refrigerated space and the power input to the compressor, (*b*) the rate of heat rejection to the environment, and (*c*) the COP of the refrigerator.



$$P_1 = 0.14 \text{ MPa} \longrightarrow h_1 = h_{g\,@\,0.14\text{ MPa}} = 239.19 \text{ kJ/kg}$$

$$s_1 = s_{g\,@\,0.14\text{ MPa}} = 0.94467 \text{ kJ/kg·K}$$

$$\left.\begin{aligned} P_2 &= 0.8 \text{ MPa} \\ s_2 &= s_1 \end{aligned}\right\} h_2 = 275.40 \text{ kJ/kg}$$

$$P_3 = 0.8 \text{ MPa} \longrightarrow h_3 = h_{f\,@\,0.8\text{ MPa}} = 95.48 \text{ kJ/kg}$$

$$h_4 \cong h_3 \text{ (throttling)} \longrightarrow h_4 = 95.48 \text{ kJ/kg}$$

Ref. Cengel and Boles, 8th Edition (2015)

(*a*) The rate of heat removal from the refrigerated space and the power input to the compressor are determined from their definitions:

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})[(239.19 - 95.48) \text{ kJ/kg}] = \mathbf{7.19 \text{ kW}}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})[(275.40 - 239.19) \text{ kJ/kg}] = \mathbf{1.81 \text{ kW}}$$

(*b*) The rate of heat rejection from the refrigerant to the environment is

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.05 \text{ kg/s})[(275.40 - 95.48) \text{ kJ/kg}] = \mathbf{9.00 \text{ kW}}$$

It could also be determined from

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 7.19 + 1.81 = 9.00 \text{ kW}$$

(*c*) The coefficient of performance of the refrigerator is

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.19 \text{ kW}}{1.81 \text{ kW}} = \mathbf{3.97}$$

That is, this refrigerator removes about 4 units of thermal energy from the refrigerated space for each unit of electric energy it consumes.

Discussion It would be interesting to see what happens if the throttling valve were replaced by an isentropic turbine. The enthalpy at state 4*s* (the turbine exit with P_{4s} = 0.14 MPa, and $s_{4s} = s_3$ = 0.35408 kJ/kg·K) is 88.95 kJ/kg, and the turbine would produce 0.33 kW of power. This would decrease the power input to the refrigerator from 1.81 to 1.48 kW and increase the rate of heat removal from the refrigerated space from 7.19 to 7.51 kW. As a result, the COP of the refrigerator would increase from 3.97 to 5.07, an increase of 28 percent.

Ref. Cengel and Boles, 8th Edition (2015)

ACTUAL VAPOR-COMPRESSION REFRIGERATION CYCLE

Refrigerant leaves the evaporator in slightly superheated state

Pipes connecting the evaporator exit to compressor inlet is relatively long. This leads to pressure drop due to fluid friction and heat transfer from surroundings

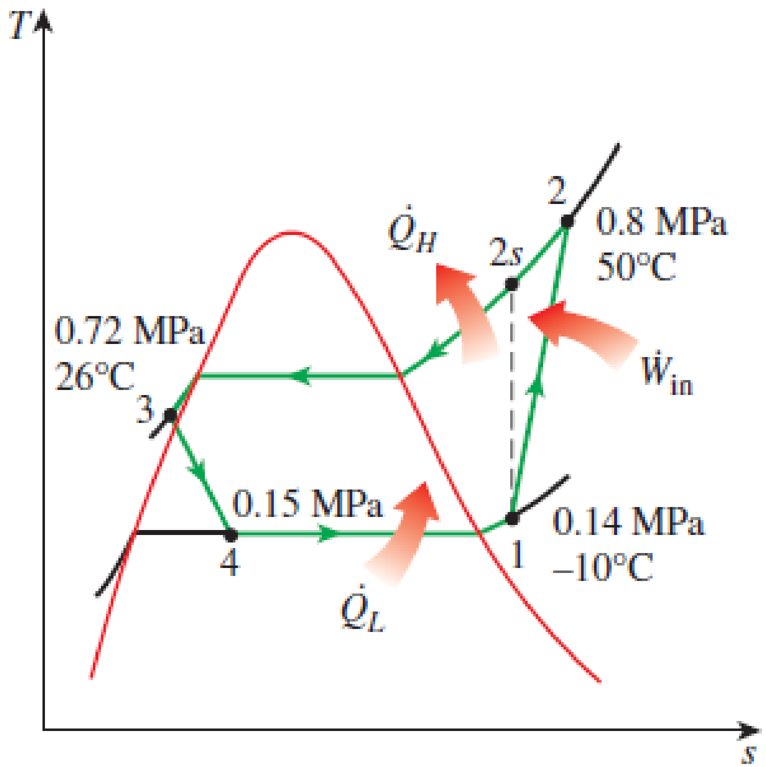
As a result specific volume of the vapor entering the compressor is higher. This leads to higher power requirement in compressor.

There are frictional effects in compressor as well as heat transfer to or from surroundings during compression. As a result, compression process in not isoentropic.

There is a subcooling of the liquid at the exit of the condenser as a result of fluid friction in condenser, and in pipes connecting the condenser to compressor and throttling valve.

Ref. Cengel and Boles, 8th Edition (2015)

Refrigerant-134a enters the compressor of a refrigerator as superheated vapor at 0.14 MPa and -10°C at a rate of 0.05 kg/s and leaves at 0.8 MPa and 50°C. The refrigerant is cooled in the condenser to 26°C and 0.72 MPa and is throttled to 0.15 MPa. Disregarding any heat transfer and pressure drops in the connecting lines between the components, determine (*a*) the rate of heat removal from the refrigerated space and the power input to the compressor, (*b*) the isentropic efficiency of the compressor, and (*c*) the coefficient of performance of the refrigerator.



$$\left.\begin{aligned} P_1 &= 0.14 \text{ MPa} \\ T_1 &= -10°\text{C} \end{aligned}\right\} \quad h_1 = 246.37 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_2 &= 0.8 \text{ MPa} \\ T_2 &= 50°\text{C} \end{aligned}\right\} \quad h_2 = 286.71 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 0.72 \text{ MPa} \\ T_3 &= 26°\text{C} \end{aligned}\right\} \quad h_3 \cong h_{f\,@\,26°\text{C}} = 87.83 \text{ kJ/kg}$$

$$h_4 \cong h_3 \text{ (throttling)} \longrightarrow h_4 = 87.83 \text{ kJ/kg}$$

Ref. Cengel and Boles, 8th Edition (2015)

(*a*) The rate of heat removal from the refrigerated space and the power input to the compressor are determined from their definitions:

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05\ \text{kg/s})[(246.37 - 87.83)\ \text{kJ/kg}] = \mathbf{7.93\ kW}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.05\ \text{kg/s})[(286.71 - 246.37)\ \text{kJ/kg}] = \mathbf{2.02\ kW}$$

(*b*) The isentropic efficiency of the compressor is determined from

$$\eta_C \cong \frac{h_{2s} - h_1}{h_2 - h_1}$$

where the enthalpy at state 2*s* (P_{2s} = 0.8 MPa and $s_{2s} = s_1$ = 0.9724 kJ/kg·K) is 284.20 kJ/kg. Thus,

$$\eta_C = \frac{284.20 - 246.37}{286.71 - 246.37} = \mathbf{0.938} \text{ or } \mathbf{93.8\%}$$

(*c*) The coefficient of performance of the refrigerator is

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.93\ \text{kW}}{2.02\ \text{kW}} = \mathbf{3.93}$$

Discussion This problem is identical to the problem in the previous example, except that the refrigerant is slightly superheated at the compressor inlet and subcooled at the condenser exit. Also, the compressor is not isentropic. As a result, the heat removal rate from the refrigerated space increases (by 10.3 percent), but the power input to the compressor increases even more (by 11.6 percent). Consequently, the COP of the refrigerator decreases from 3.97 to 3.93.

Ref. Cengel and Boles, 8th Edition (2015)