Université de Batna 2 Année universitaire 2019-2020

Faculté de Technologie Département de Mécanique

Machines frigorifiques et pompes à chaleur L3ENG

Série d'exercices n° 01

Exercice 01 :

T, °C	*P*, kPa	v, m^3 / kg	Phase description
50	***12.35***	7.72	***Saturated mixture***
143.6	400	***0.4624***	Saturated vapor
250	500	***0.4744***	***Superheated vapor***
110	350	***0.001051***	***Compressed liquid***

Exercice 02 :

T, °C	P, kPa	*h*, kJ / kg	*x*	Phase description
120.21	200	***2045.8***	0.7	***Saturated mixture***
140	***361.53***	1800	***0.565***	***Saturated mixture***
177.66	950	***752.74***	0.0	***Saturated liquid***
80	500	***335.37***	**- - -**	***Compressed liquid***
350.0	800	3162.2	**- - -**	***Superheated vapor***

Exercice 03 :

T, °C	P, kPa	v, m^3 / kg	Phase description
-8	320	***0.0007569***	***Compressed liquid***
30	***770.64***	0.015	***Saturated mixture***
-12.73	180	***0.11041***	Saturated vapor
80	600	***0.044710***	***Superheated vapor***

Exercice 04 :

T, °C	P, kPa	*u*, kJ / kg	Phase description
20	***572.07***	95	***Saturated mixture***
-12	***185.37***	***35.78***	Saturated liquid
86.24	400	300	***Superheated vapor***
8	600	***62.26***	***Compressed liquid***

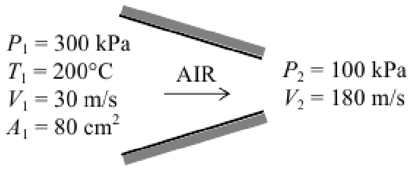
Série d'exercices n° 02

Exercice 01 :

Air is accelerated in a nozzle from 30 m/s to 180 m/s. The mass flow rate, the exit temperature, and the exit area of the nozzle are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Air is an ideal gas with constant specific heats. **3** Potential energy changes are negligible. **4** The device is adiabatic and thus heat transfer is negligible. **5** There are no work interactions.

Properties The gas constant of air is 0.287 kPa.m^3/kg.K (Table A-1). The specific heat of air at the anticipated average temperature of 450 K is c_p = 1.02 kJ/kg.°C (Table A-2).



Analysis (*a*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Using the ideal gas relation, the specific volume and the mass flow rate of air are determined to be

$$v_1 = \frac{RT_1}{P_1} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(473 \text{ K})}{300 \text{ kPa}} = 0.4525 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{v_1}A_1V_1 = \frac{1}{0.4525 \text{ m}^3/\text{kg}}(0.008 \text{ m}^2)(30 \text{ m/s}) = \mathbf{0.5304 \text{ kg/s}}$$

(*b*) We take nozzle as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}(h_1 + V_1^2/2) = \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \dot{W} \cong \Delta\text{pe} \cong 0)$$

$$0 = h_2 - h_1 + \frac{V_2^2 - V_1^2}{2} \longrightarrow 0 = c_{p,ave}(T_2 - T_1) + \frac{V_2^2 - V_1^2}{2}$$

Substituting, $$0 = (1.02 \text{ kJ/kg}\cdot\text{K})(T_2 - 200^\circ\text{C}) + \frac{(180 \text{ m/s})^2 - (30 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right)$$

It yields T_2 = **184.6°C**

(*c*) The specific volume of air at the nozzle exit is

$$v_2 = \frac{RT_2}{P_2} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(184.6 + 273 \text{ K})}{100 \text{ kPa}} = 1.313 \text{ m}^3/\text{kg}$$

$$\dot{m} = \frac{1}{v_2}A_2V_2 \longrightarrow 0.5304 \text{ kg/s} = \frac{1}{1.313 \text{ m}^3/\text{kg}}A_2(180 \text{ m/s}) \rightarrow A_2 = 0.00387 \text{ m}^2 = \mathbf{38.7 \text{ cm}^2}$$

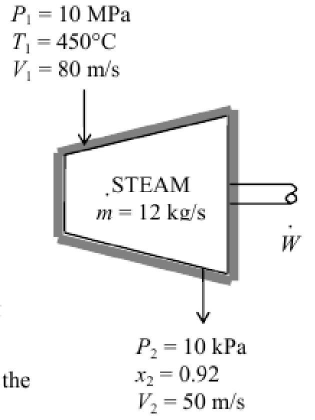
Exercice 02 :

Steam expands in a turbine. The change in kinetic energy, the power output, and the turbine inlet area are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Potential energy changes are negligible. **3** The device is adiabatic and thus heat transfer is negligible.

Properties From the steam tables (Tables A-4 through 6)

$$\left.\begin{array}{l} P_1 = 10 \text{ MPa} \\ T_1 = 450°\text{C} \end{array}\right\} \begin{array}{l} \boldsymbol{v}_1 = 0.029782 \text{ m}^3/\text{kg} \\ h_1 = 3242.4 \text{ kJ/kg} \end{array}$$



and

$$\left.\begin{array}{l} P_2 = 10 \text{ kPa} \\ x_2 = 0.92 \end{array}\right\} h_2 = h_f + x_2 h_{fg} = 191.81 + 0.92 \times 2392.1 = 2392.5 \text{ kJ/kg}$$

Analysis (a) The change in kinetic energy is determined from

$$\Delta ke = \frac{V_2^2 - V_1^2}{2} = \frac{(50 \text{ m/s})^2 - (80 \text{ m/s})^2}{2}\left(\frac{1 \text{ kJ/kg}}{1000 \text{ m}^2/\text{s}^2}\right) = -1.95 \text{ kJ/kg}$$

(b) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. We take the turbine as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as

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

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

$$\dot{m}(h_1 + V_1^2/2) = \dot{W}_{\text{out}} + \dot{m}(h_2 + V_2^2/2) \quad (\text{since } \dot{Q} \cong \Delta\text{pe} \cong 0)$$

$$\dot{W}_{\text{out}} = -\dot{m}\left(h_2 - h_1 + \frac{V_2^2 - V_1^2}{2}\right)$$

Then the power output of the turbine is determined by substitution to be

$$\dot{W}_{\text{out}} = -(12 \text{ kg/s})(2392.5 - 3242.4 - 1.95)\text{kJ/kg} = \mathbf{10.2 \ MW}$$

(c) The inlet area of the turbine is determined from the mass flow rate relation,

$$\dot{m} = \frac{1}{\boldsymbol{v}_1} A_1 V_1 \longrightarrow A_1 = \frac{\dot{m}\boldsymbol{v}_1}{V_1} = \frac{(12 \text{ kg/s})(0.029782 \text{ m}^3/\text{kg})}{80 \text{ m/s}} = \mathbf{0.00447 \ m^2}$$

Exercice 03 :

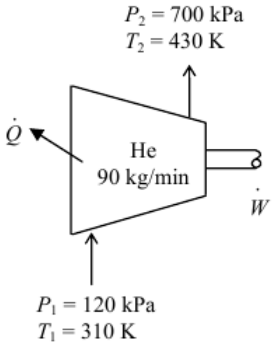
Helium is compressed by a compressor. For a mass flow rate of 90 kg/min, the power input required is to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** Helium is an ideal gas with constant specific heats.

Properties The constant pressure specific heat of helium is c_p = 5.1926 kJ/kg·K (Table A-2a).

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

We take the compressor as the system, which is a control volume since mass crosses the boundary. The energy balance for this steady-flow system can be expressed in the rate form as



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

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

$$\dot{W}_{in} + \dot{m}h_1 = \dot{Q}_{out} + \dot{m}h_2 \quad (\text{since } \Delta ke \cong \Delta pe \cong 0)$$

$$\dot{W}_{in} - \dot{Q}_{out} = \dot{m}(h_2 - h_1) = \dot{m}c_p(T_2 - T_1)$$

Thus,

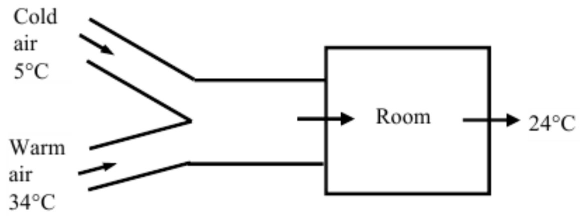
$$\dot{W}_{in} = \dot{Q}_{out} + \dot{m}c_p(T_2 - T_1)$$
$$= (90/60 \text{ kg/s})(20 \text{ kJ/kg}) + (90/60 \text{ kg/s})(5.1926 \text{ kJ/kg} \cdot \text{K})(430 - 310)\text{K}$$
$$= \mathbf{965 \ kW}$$

Exercice 04 :

Two streams of cold and warm air are mixed in a chamber. If the ratio of hot to cold air is 1.6, the mixture temperature and the rate of heat gain of the room are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** The device is adiabatic and thus heat transfer is negligible.

Properties The gas constant of air is $R = 0.287$ kPa.m^3/kg.K. The enthalpies of air are obtained from air table (Table A-17) as



$$h_1 = h_{@278 \text{ K}} = 278.13 \text{ kJ/kg}$$
$$h_2 = h_{@ 307 \text{ K}} = 307.23 \text{ kJ/kg}$$
$$h_{room} = h_{@ 297 \text{ K}} = 297.18 \text{ kJ/kg}$$

Analysis (*a*) We take the mixing chamber as the system, which is a control volume since mass crosses the boundary. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance:

$$\dot{m}_{in} - \dot{m}_{out} = \Delta \dot{m}_{system} {}^{\nearrow 0 \text{ (steady)}} = 0 \rightarrow \dot{m}_{in} = \dot{m}_{out} \rightarrow \dot{m}_1 + 1.6\dot{m}_1 = \dot{m}_3 = 2.6\dot{m}_1 \quad \text{since} \quad \dot{m}_2 = 1.6\dot{m}_1$$

Energy balance:

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

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

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

Combining the two gives $\dot{m}_1 h_1 + 1.6\dot{m}_1 h_2 = 2.6\dot{m}_1 h_3$ *or* $h_3 = (h_1 + 1.6h_2)/2.6$

Substituting,

$$h_3 = (278.13 + 1.6\times 307.23)/2.6 = 296.04 \text{ kJ/kg}$$

From air table at this enthalpy, the mixture temperature is

$$T_3 = T_{@\, h = 296.04 \text{ kJ/kg}} = 295.9 \text{ K} = \mathbf{22.9°C}$$

(b) The mass flow rates are determined as follows

$$\boldsymbol{v}_1 = \frac{RT_1}{P} = \frac{(0.287 \text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(5 + 273 \text{ K})}{105 \text{ kPa}} = 0.7599 \text{ m}^3/\text{kg}$$

$$\dot{m}_1 = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{1.25 \text{ m}^3/\text{s}}{0.7599 \text{ m}^3/\text{kg}} = 1.645 \text{ kg/s}$$

$$\dot{m}_3 = 2.6\dot{m}_1 = 2.6(1.645 \text{ kg/s}) = 4.277 \text{ kg/s}$$

The rate of heat gain of the room is determined from

$$\dot{Q}_{cool} = \dot{m}_3(h_{room} - h_3) = (4.277 \text{ kg/s})(297.18 - 296.04) \text{ kJ/kg} = \mathbf{4.88\ kW}$$

<u>Exercice à domicile :</u>

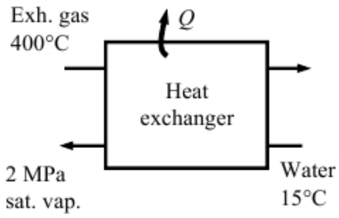
A heat exchanger that is not insulated is used to produce steam from the heat given up by the exhaust gases of an internal combustion engine. The temperature of exhaust gases at the heat exchanger exit and the rate of heat transfer to the water are to be determined.

Assumptions **1** This is a steady-flow process since there is no change with time. **2** Kinetic and potential energy changes are negligible. **3** There are no work interactions. **4** Exhaust gases are assumed to have air properties with constant specific heats.

Properties The constant pressure specific heat of the exhaust gases is taken to be c_p = 1.045 kJ/kg·°C (Table A-2). The inlet and exit enthalpies of water are (Tables A-4 and A-5)

$$\left.\begin{array}{l} T_{w,in} = 15°C \\ x = 0 \text{ (sat. liq.)} \end{array}\right\} h_{w,in} = 62.98 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{w,out} = 2 \text{ MPa} \\ x = 1 \text{ (sat. vap.)} \end{array}\right\} h_{w,out} = 2798.3 \text{ kJ/kg}$$



Analysis We take the entire heat exchanger as the system, which is a control volume. The mass and energy balances for this steady-flow system can be expressed in the rate form as

Mass balance (*for each fluid stream*):

$$\dot{m}_{in} - \dot{m}_{out} = \Delta\dot{m}_{system}^{\nearrow 0 \text{ (steady)}} = 0 \longrightarrow \dot{m}_{in} = \dot{m}_{out}$$

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

$$\dot{m}_{\text{exh}} h_{\text{exh,in}} + \dot{m}_{\text{w}} h_{\text{w,in}} = \dot{m}_{\text{exh}} h_{\text{exh,out}} + \dot{m}_{\text{w}} h_{\text{w,out}} + \dot{Q}_{\text{out}} \quad (\text{since } \dot{W} = \Delta\text{ke} \cong \Delta\text{pe} \cong 0)$$

or

$$\dot{m}_{\text{exh}} c_p T_{\text{exh,in}} + \dot{m}_w h_{\text{w,in}} = \dot{m}_{\text{exh}} c_p T_{\text{exh,out}} + \dot{m}_w h_{\text{w,out}} + \dot{Q}_{\text{out}}$$

Noting that the mass flow rate of exhaust gases is 15 times that of the water, substituting gives

$$15\dot{m}_{\text{w}}(1.045 \text{ kJ/kg.°C})(400\text{°C}) + \dot{m}_{\text{w}}(62.98 \text{ kJ/kg}) = 15\dot{m}_{\text{w}}(1.045 \text{ kJ/kg.°C})T_{\text{exh,out}} + \dot{m}_{\text{w}}(2798.3 \text{ kJ/kg}) + \dot{Q}_{out} \quad (1)$$

The heat given up by the exhaust gases and heat picked up by the water are

$$\dot{Q}_{exh} = \dot{m}_{\text{exh}} c_p (T_{\text{exh,in}} - T_{\text{exh,out}}) = 15\dot{m}_{\text{w}}(1.045 \text{ kJ/kg.°C})(400 - T_{\text{exh,out}})\text{°C} \quad (2)$$

$$\dot{Q}_w = \dot{m}_{\text{w}}(h_{\text{w,out}} - h_{\text{w,in}}) = \dot{m}_{\text{w}}(2798.3 - 62.98)\text{kJ/kg} \quad (3)$$

The heat loss is

$$\dot{Q}_{out} = f_{\text{heat loss}} \dot{Q}_{exh} = 0.1\dot{Q}_{exh} \quad (4)$$

The solution may be obtained by a trial-error approach. Or, solving the above equations simultaneously using EES software, we obtain

$$T_{\text{exh,out}} = \mathbf{206.1°C}, \dot{Q}_{\text{w}} = \mathbf{97.26\ kW}, \ \dot{m}_{\text{w}} = 0.03556 \text{ kg/s}, \ \dot{m}_{\text{exh}} = 0.5333 \text{ kg/s}$$

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Série d'exercices n° 03

Exercice 01 :

A commercial refrigerator with refrigerant-134a as the working fluid is considered. The quality of the refrigerant at the evaporator inlet, the refrigeration load, the COP of the refrigerator, and the theoretical maximum refrigeration load for the same power input to the compressor are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From refrigerant-134a tables (Tables A-11 through A-13)

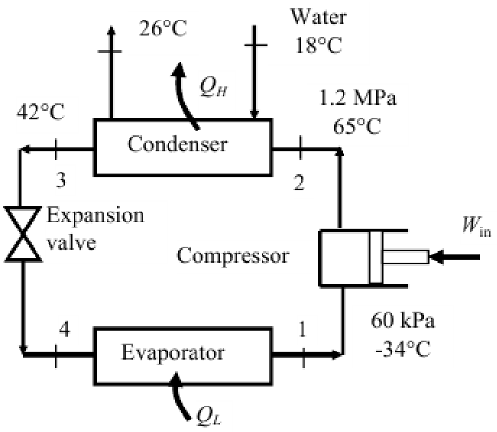
$$\left.\begin{array}{l} P_1 = 60\text{ kPa} \\ T_1 = -34°\text{C} \end{array}\right\} h_1 = 230.03\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1200\text{ kPa} \\ T_2 = 65°\text{C} \end{array}\right\} h_2 = 295.16\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1200\text{ kPa} \\ T_3 = 42°\text{C} \end{array}\right\} h_3 = 111.23\text{ kJ/kg}$$

$$h_4 = h_3 = 111.23\text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 60\text{ kPa} \\ h_4 = 111.23\text{ kJ/kg} \end{array}\right\} x_4 = \mathbf{0.4795}$$



Using saturated liquid enthalpy at the given temperature, for water we have (Table A-4)

$$h_{w1} = h_{f@18°\text{C}} = 75.47\text{ kJ/kg}$$

$$h_{w2} = h_{f@26°\text{C}} = 108.94\text{ kJ/kg}$$

(*b*) The mass flow rate of the refrigerant may be determined from an energy balance on the compressor

$$\dot{m}_R(h_2 - h_3) = \dot{m}_w(h_{w2} - h_{w1})$$

$$\dot{m}_R(295.16 - 111.23)\text{kJ/kg} = (0.25\text{ kg/s})(108.94 - 75.47)\text{kJ/kg}$$

$$\longrightarrow \dot{m}_R = 0.0455\text{ kg/s}$$

The waste heat transferred from the refrigerant, the compressor power input, and the refrigeration load are

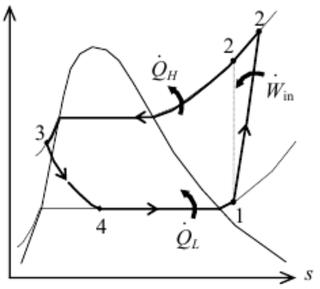
$$\dot{Q}_H = \dot{m}_R(h_2 - h_3) = (0.0455\text{ kg/s})(295.16 - 111.23)\text{kJ/kg} = 8.367\ \text{ kW}$$

$$\dot{W}_{\text{in}} = \dot{m}_R(h_2 - h_1) - \dot{Q}_{\text{in}} = (0.0455\text{ kg/s})(295.16 - 230.03)\text{kJ/kg} - 0.45\text{ kW} = 2.513\text{ kW}$$

$$\dot{Q}_L = \dot{Q}_H - \dot{W}_{\text{in}} = 8.367 - 2.513 = \mathbf{5.85\ kW}$$

(*c*) The COP of the refrigerator is determined from its definition

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{5.85}{2.513} = \mathbf{2.33}$$



(*d*) The reversible COP of the refrigerator for the same temperature limits is

$$\text{COP}_{\max} = \frac{1}{T_H / T_L - 1} = \frac{1}{(18 + 273)/(-30 + 273) - 1} = 5.063$$

Then, the maximum refrigeration load becomes

$$\dot{Q}_{\text{L,max}} = COP_{\max}\dot{W}_{\text{in}} = (5.063)(2.513\text{ kW}) = \mathbf{12.72\ kW}$$

Exercice 02 :

An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the rate of heat rejection to the environment, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

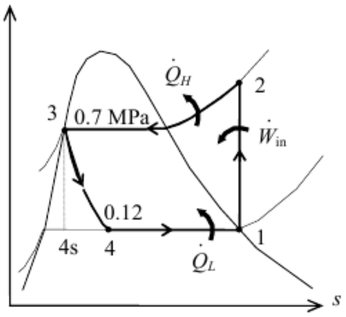
Analysis (*a*) In an ideal vapor-compression refrigeration cycle, the compression process is isentropic, the refrigerant enters the compressor as a saturated vapor at the evaporator pressure, and leaves the condenser as saturated liquid at the condenser pressure. From the refrigerant tables (Tables A-12 and A-13),

$$\left.\begin{array}{l} P_1 = 120 \text{ kPa} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} h_1 = h_{g\,@\,120\text{ kPa}} = 236.97 \text{ kJ/kg} \\ s_1 = s_{g\,@\,120\text{ kPa}} = 0.94779 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 273.50 \text{ kJ/kg} \ (T_2 = 34.95°\text{C})$$

$$\left.\begin{array}{l} P_3 = 0.7 \text{ MPa} \\ \text{sat. liquid} \end{array}\right\} h_3 = h_{f\,@\,0.7\text{ MPa}} = 88.82 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 88.82 \text{ kJ/kg (throttling)}$$



Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})(236.97 - 88.82) \text{ kJ/kg} = \mathbf{7.41\ kW}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})(273.50 - 236.97) \text{ kJ/kg} = \mathbf{1.83\ kW}$$

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

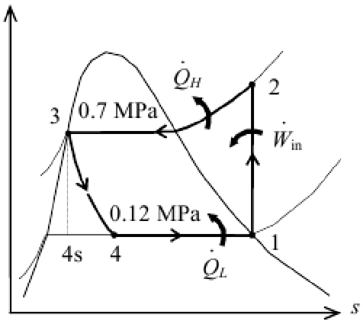
$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 7.41 + 1.83 = \mathbf{9.23\ kW}$$

(*c*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.41 \text{ kW}}{1.83 \text{ kW}} = \mathbf{4.06}$$

Exercice 03 :

An ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is considered. The throttling valve in the cycle is replaced by an isentropic turbine. The percentage increase in the COP and in the rate of heat removal from the refrigerated space due to this replacement are to be determined.



Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis If the throttling valve in the previous problem is replaced by an isentropic turbine, we would have $s_{4s} = s_3 = s_{f @ 0.7 \text{ MPa}} = 0.33230$ kJ/kg·K, and the enthalpy at the turbine exit would be

$$x_{4s} = \left(\frac{s_3 - s_f}{s_{fg}} \right)_{@ 120 \text{ kPa}} = \frac{0.33230 - 0.09275}{0.85503} = 0.2802$$

$$h_{4s} = \left(h_f + x_{4s} h_{fg} \right)_{@ 120 \text{ kPa}} = 22.49 + (0.2802)(214.48) = 82.58 \text{ kJ/kg}$$

Then,

$$\dot{Q}_L = \dot{m}(h_1 - h_{4s}) = (0.05 \text{ kg/s})(236.97 - 82.58) \text{ kJ/kg} = 7.72 \text{ kW}$$

and

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_\text{in}} = \frac{7.72 \text{ kW}}{1.83 \text{ kW}} = 4.23$$

Then the percentage increase in $\dot{Q}$ and COP becomes

$$\text{Increase in } \dot{Q}_L = \frac{\Delta \dot{Q}_L}{\dot{Q}_L} = \frac{7.72 - 7.41}{7.41} = \mathbf{4.2\%}$$

$$\text{Increase in COP}_\text{R} = \frac{\Delta \text{COP}_\text{R}}{\text{COP}_\text{R}} = \frac{4.23 - 4.06}{4.06} = \mathbf{4.2\%}$$

Exercice 04 :

A refrigerator with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the isentropic efficiency of the compressor, and the COP of the refrigerator are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the refrigerant tables (Tables A-12 and A-13),

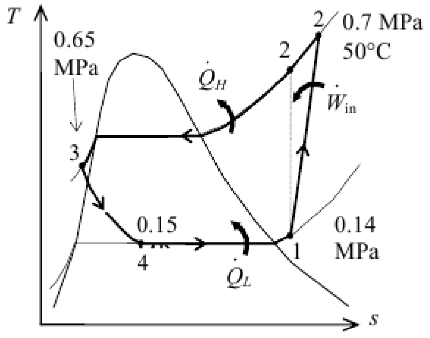
$$\left.\begin{array}{l} P_1 = 0.14 \text{ MPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 246.36 \text{ kJ/kg} \\ s_1 = 0.97236 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7 \text{ MPa} \\ T_2 = 50°\text{C} \end{array}\right\} h_2 = 288.53 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{2s} = 0.7 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 281.16 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.65 \text{ MPa} \\ T_3 = 24°\text{C} \end{array}\right\} h_3 = h_{f\,@\,24°\text{C}} = 84.98 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 84.98 \text{ kJ/kg (throttling)}$$



Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.12 \text{ kg/s})(246.36 - 84.98) \text{ kJ/kg} = \mathbf{19.4\ kW}$$

and

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = (0.12 \text{ kg/s})(288.53 - 246.36) \text{ kJ/kg} = \mathbf{5.06\ kW}$$

A refrigerator with refrigerant-134a as the working fluid is considered. The rate of heat removal from the refrigerated space, the power input to the compressor, the isentropic efficiency of the compressor, and the COP of the refrigerator are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the refrigerant tables (Tables A-12 and A-13),

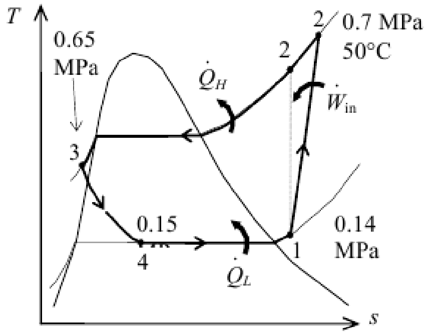
$$\left.\begin{array}{l} P_1 = 0.14 \text{ MPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 246.36 \text{ kJ/kg} \\ s_1 = 0.97236 \text{ kJ/kg}\cdot\text{K} \end{array}$$

$$\left.\begin{array}{l} P_2 = 0.7 \text{ MPa} \\ T_2 = 50°\text{C} \end{array}\right\} h_2 = 288.53 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_{2s} = 0.7 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 281.16 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.65 \text{ MPa} \\ T_3 = 24°\text{C} \end{array}\right\} h_3 = h_{f\,@\,24°\text{C}} = 84.98 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 84.98 \text{ kJ/kg (throttling)}$$



Then the rate of heat removal from the refrigerated space and the power input to the compressor are determined from

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.12 \text{ kg/s})(246.36 - 84.98) \text{ kJ/kg} = \mathbf{19.4\ kW}$$

and

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = (0.12 \text{ kg/s})(288.53 - 246.36) \text{ kJ/kg} = \mathbf{5.06\ kW}$$

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

$$\eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{281.16 - 246.36}{288.53 - 246.36} = \mathbf{82.5\%}$$

(*c*) The COP of the refrigerator is determined from its definition,

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{19.4 \text{ kW}}{5.06 \text{ kW}} = \mathbf{3.83}$$

Travail à domicile :

A refrigerator with refrigerant-134a as the working fluid is considered. The power input to the compressor, the rate of heat removal from the refrigerated space, and the pressure drop and the rate of heat gain in the line between the evaporator and the compressor are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis (*a*) From the refrigerant tables (Tables A-12 and A-13),

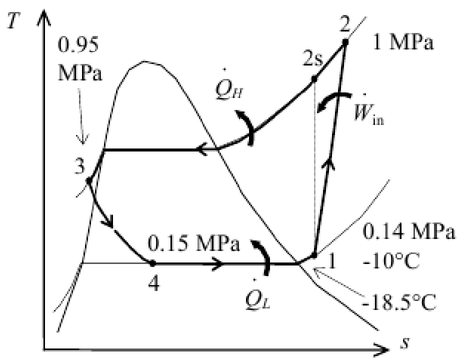
$$\left.\begin{array}{l} P_1 = 140 \text{ kPa} \\ T_1 = -10°\text{C} \end{array}\right\} \begin{array}{l} h_1 = 246.36 \text{ kJ/kg} \\ s_1 = 0.97236 \text{ kJ/kg}\cdot\text{K} \\ \boldsymbol{v}_1 = 0.14605 \text{ m}^3/\text{kg} \end{array}$$

$$\left.\begin{array}{l} P_2 = 1.0 \text{ MPa} \\ s_{2s} = s_1 \end{array}\right\} h_{2s} = 289.20 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 0.95 \text{ MPa} \\ T_3 = 30°\text{C} \end{array}\right\} h_3 \cong h_{f\,@\,30\,°\text{C}} = 93.58 \text{ kJ/kg}$$

$$h_4 \cong h_3 = 93.58 \text{ kJ/kg (throttling)}$$

$$\left.\begin{array}{l} T_5 = -18.5°\text{C} \\ \text{sat. vapor} \end{array}\right\} \begin{array}{l} P_5 = 0.14165 \text{ MPa} \\ h_5 = 239.33 \text{ kJ/kg} \end{array}$$



Then the mass flow rate of the refrigerant and the power input becomes

$$\dot{m} = \frac{\dot{\boldsymbol{V}}_1}{\boldsymbol{v}_1} = \frac{0.3/60 \text{ m}^3/\text{s}}{0.14605 \text{ m}^3/\text{kg}} = 0.03423 \text{ kg/s}$$

$$\dot{W}_{in} = \dot{m}(h_{2s} - h_1)/\eta_C = (0.03423 \text{ kg/s})[(289.20 - 246.36) \text{ kJ/kg}]/(0.78) = \mathbf{1.88 \text{ kW}}$$

(*b*) The rate of heat removal from the refrigerated space is

$$\dot{Q}_L = \dot{m}(h_5 - h_4) = (0.03423 \text{ kg/s})(239.33 - 93.58) \text{ kJ/kg} = \mathbf{4.99 \text{ kW}}$$

(*c*) The pressure drop and the heat gain in the line between the evaporator and the compressor are

$$\Delta P = P_5 - P_1 = 141.65 - 140 = \mathbf{1.65}$$

and

$$\dot{Q}_{gain} = \dot{m}(h_1 - h_5) = (0.03423 \text{ kg/s})(246.36 - 239.33) \text{ kJ/kg} = \mathbf{0.241 \text{ kW}}$$