Part I

Thermodynamic Models

2 Pumped CO_2 refrigeration system

A thermodynamic model for the secondary coolant system operating with CO_2 as a two-phase secondary refrigerant is developed and simulated.

2.1 Overview

Twelve control volumes can comprise the pumped CO_2 thermodynamic cycle, Figure 2.1, namely: compressor (*cp*), discharge line (*dl*), condenser (*cd,rf*), liquid line (*ll,rf*), expansion device (*xd*), and suction line (*sl*), for the main cycle; pump (*pp*), high pressure liquid line (*ll,sf*), evaporator (*ev,sf*), low pressure vapor line (*vl*), and receiver (*rec*), for the secondary loop; and an intermediate heat exchanger (*ev,rf* for the refrigerant side and *cd,sf* for the secondary fluid side) connecting both circuits.

In the main cycle, the refrigerant enters the compressor (*cp*) as saturated or superheated vapor at state 8, leaving as superheated vapor at state 9. As it flows through the discharge line (*dl*), the refrigerant experiences pressure drop and temperature change, due to heat transfer with surroundings, exiting the device at the condenser inlet pressure.

Refrigerant, then, enters the condenser (*cd,rf*) as superheated vapor at state 10, with saturated or subcooled liquid leaving at state 11, as a result of heat rejection to the cooling medium. The temperature of the surroundings at this state is still below that of the refrigerant. When passing through the liquid line (*ll,rf*), the refrigerant experiences, once again, pressure drop and temperature change, exiting as subcooled liquid.

The liquid refrigerant at state 12 is throttled to the intermediate heat exchanger inlet pressure by flowing through an expansion valve or capillary tube (*xd*). During this process, the temperature of the refrigerant drops below that of the secondary fluid leaving the heat exchanger on the other side. The refrigerant, then, enters the intermediate heat exchanger (*ev,rf*) as a low quality saturated mixture at state 13, evaporating completely by absorbing heat from the secondary fluid, and exiting as superheated vapor, at state 14. Pressure drop and temperature change in the suction line (*sl*) complete the primary cycle, with the refrigerant leaving it as low pressure vapor and reentering the compressor at state 8.

In the secondary cycle, the secondary fluid enters the pump (*pp*) as saturated liquid at state 1, exiting as high pressure liquid at state 2. The temperature of the outlet liquid is well below the temperature of the surroundings. The fluid, then, experiences pressure drop and heat transfer when passing through the high pressure liquid line (*ll,sf*), leaving the device at the evaporator inlet pressure.

Next, the secondary fluid enters the evaporator as subcooled liquid at state 3, evaporating partially or completely as heat is transfered from the refrigerated space. High quality saturated mixture or vapor exit, thus, the evaporator at state 4. In the low pressure vapor line, the secondary fluid expands to a saturated or superheated vapor at the receiver pressure, which is equal to the intermediate heat exchanger pressure.

This vapor (state 5) is mixed with the liquid from the intermediate heat exchanger (state 7) in the receiver (*rec*), with saturated vapor (state 6) and saturated liquid (state 1) exiting the reservatory. Entering the intermediate heat exchanger (*cd,sf*), the saturated vapor at state 6 condenses as it rejects heat to the refrigerant, leaving it as saturated or subcooled liquid at state 7.

Figure 2.2 presents the pressure-enthalpy diagram of a pumped CO_2 thermodynamic cycle with R404A refrigerant. The refrigerant condensing temperature is 45.0°C, CO_2 evaporating temperature is −27.8°C and intermediate temperature difference is 5.6°C.

2.2 Mathematical model

The mathematical model for the pumped CO_2 refrigeration system is now outlined, starting with the detailed description of the control volumes, followed by the modeling of each component.

It is worth mentioning that, although the carbon dioxide is constantly referred to as the operating fluid in the secondary cycle, the model is capable of simulating the secondary-coolant technology for different secondary fluids.

2.2.1 Compressor

A simple efficiency-based model is employed for the compressor [129]. The isentropic and volumetric efficiencies, assumed as constant empirical values, are related to compressor suction and discharge thermodynamic states, as well as to refrigerant mass flow rate, as expressed by Eqs.(2.1) and (2.3), respectively.

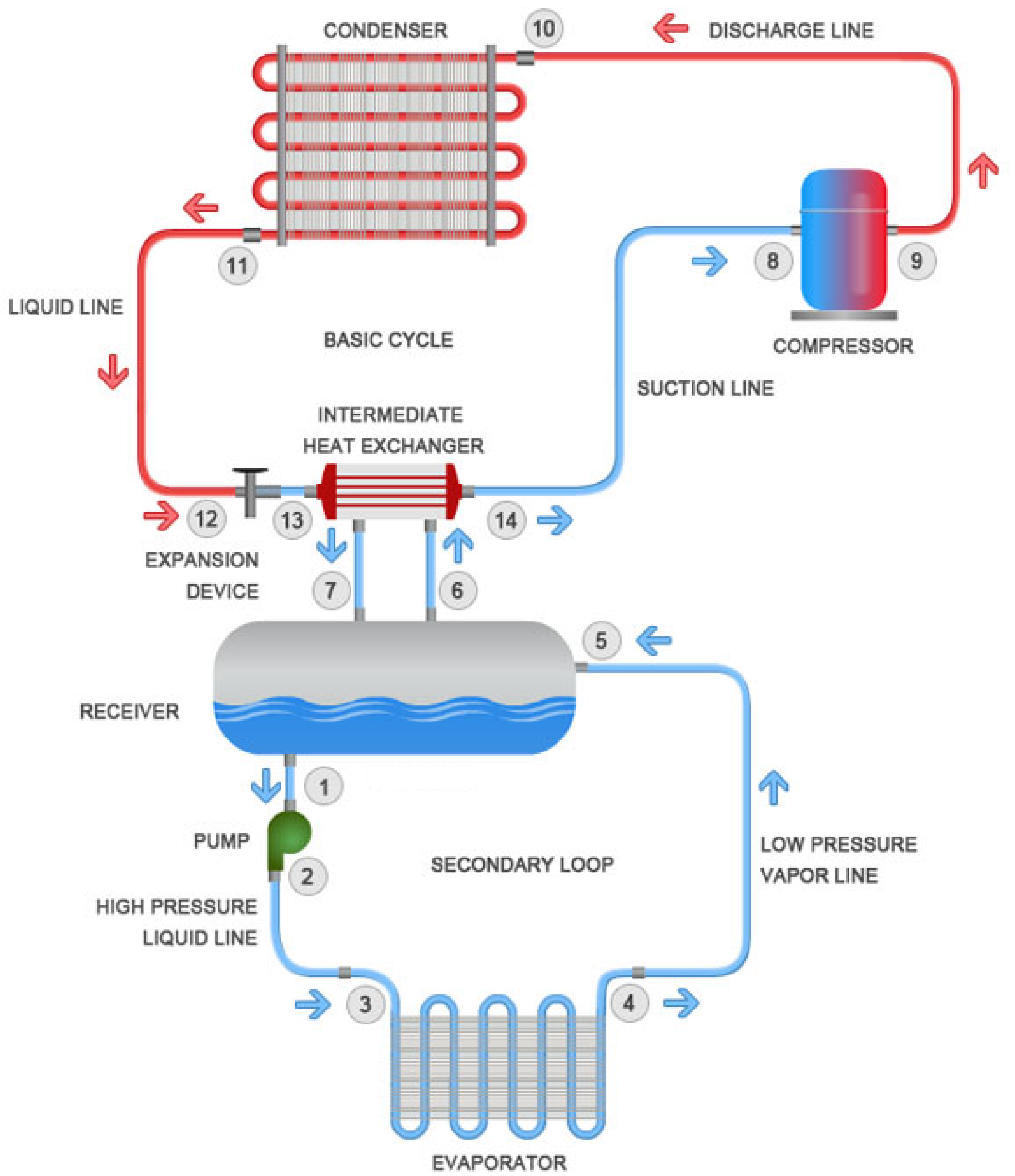


Figure 2.1: Scheme comprising the twelve control volumes of the pumped CO_2 refrigeration system [130].

Compressor adiabatic power consumption is, ultimately, calculated for the device, Eq.(2.4).

$$\eta_{s,cp} = \frac{(h_{9i} - h_8)}{(h_9 - h_8)} \tag{2.1}$$

$$s_{9i} = s_8 \tag{2.2}$$

$$\eta_{v,cp} = \frac{v_8 \dot{m}_{rf}}{\dot{V}_{cp,rf}} \tag{2.3}$$

$$\dot{W}_{cp} = \dot{m}_{rf}\ (h_9 - h_8) \tag{2.4}$$

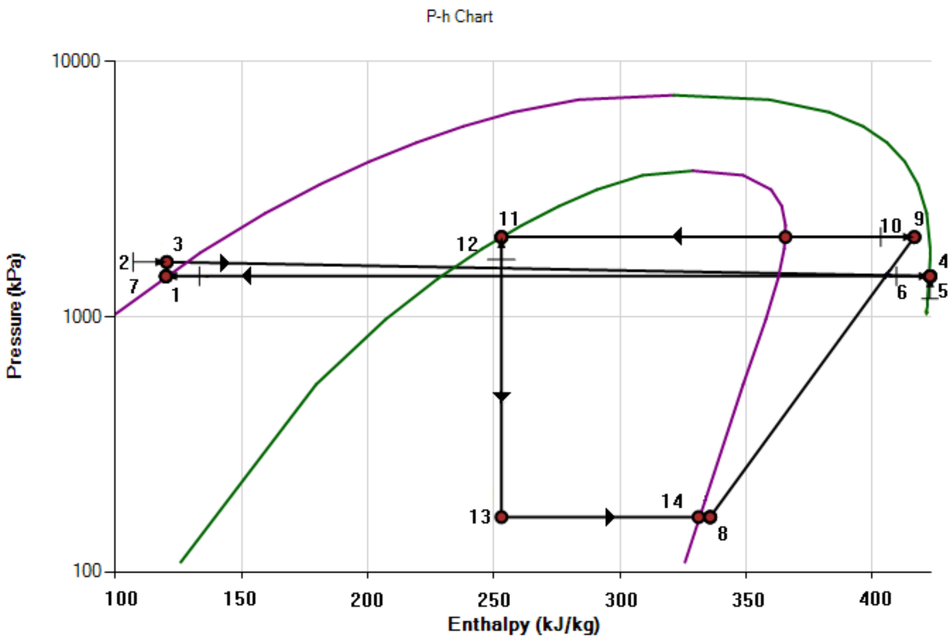


Figure 2.2: P-h diagram of a pumped CO_2 cycle, with R404A refrigerant condensing temperature of 45.0°C, CO_2 evaporating temperature of −27.8°C and intermediate temperature difference of 5.6°C [130].

2.2.2 Refrigerant condenser

Taking the middle pressure in the outdoor coil as the nominal pressure, pressure drop at the refrigerant condenser can be described by Eqs.(2.5) and (2.6).

$$P_{10} = P_{cd,rf} + \frac{\Delta P_{cd,rf}}{2} \tag{2.5}$$

$$P_{11} = P_{cd,rf} - \frac{\Delta P_{cd,rf}}{2} \tag{2.6}$$

The difference between refrigerant temperature at condenser outlet and saturation temperature is represented by Eq.(2.7).

$$T_{11} = T_{cd,rf,bub} - \Delta T_{cd,rf,sc} \tag{2.7}$$

Eq.(2.8) describes the refrigerant energy balance for the device.

$$\dot{Q}_{cd,rf} = \dot{m}_{rf} \left(h_{10} - h_{11} \right) \tag{2.8}$$

2.2.3 Expansion device

Given that there is no work done and neglecting heat transfer from the environment [129], from the energy conservation equation, the expansion process is isenthalpic.

$$h_{13} = h_{12} \tag{2.9}$$

2.2.4 Intermediate heat exchanger – refrigerant evaporator

The average evaporating temperature for the refrigerant in the main cycle is obtained using Eq.(2.10), considering the difference between condensing and evaporating temperatures in the intermediate heat exchanger, $[T_{cd,sf} - T_{ev,rf}]$, as a known parameter. Section 2.3 presents a more detailed description of different alternatives of input data for the model.

$$T_{ev,rf} = T_{cd,sf} - [T_{cd,sf} - T_{ev,ref}] \tag{2.10}$$

Refrigerant pressure drops at the intermediate heat exchanger are described by Eqs.(2.11) and (2.12).

$$P_{13} = P_{ev,rf} + \frac{\Delta P_{ev,rf}}{2} \tag{2.11}$$

$$P_{14} = P_{ev,rf} - \frac{\Delta P_{ev,rf}}{2} \tag{2.12}$$

Evaporator outlet degree of superheat is represented by Eq.(2.13).

$$T_{14} = T_{ev,rf,dew} + \Delta T_{ev,rf,sh} \tag{2.13}$$

Eq.(2.14) presents the energy balance for the refrigerant side of the device.

$$\dot{Q}_{ev,rf} = \dot{m}_{rf}\ (h_{14} - h_{13}) \tag{2.14}$$

2.2.5 Pump

A simple efficiency-based model is employed for the secondary fluid pump. The isentropic efficiency is related to pump suction and discharge thermodynamic states by Eq.(2.15). Pump adiabatic power consumption is, ultimately, calculated for the device, Eq.(2.17).

$$\eta_{s,pp} = \frac{(h_{2i} - h_1)}{(h_2 - h_1)} \tag{2.15}$$

$$s_{2i} = s_1 \tag{2.16}$$

$$\dot{W}_{pp} = \dot{m}_{sf,ev}\ (h_2 - h_1) \tag{2.17}$$

2.2.6 Secondary fluid evaporator

Pressure drop at the secondary fluid evaporator is described by Eqs.(2.18) and (2.19), while the difference between secondary fluid temperature at evaporator outlet and saturation temperature is represented by Eq.(2.20). Analogously to the condenser, nominal pressure in the evaporator is taken at the middle pressure.

$$P_3 = P_{ev,sf} + \frac{\Delta P_{ev,sf}}{2} \tag{2.18}$$

$$P_4 = P_{ev,sf} - \frac{\Delta P_{ev,sf}}{2} \tag{2.19}$$

$$T_4 = T_{ev,sf,dew} + \Delta T_{ev,sf,sh} \tag{2.20}$$

Eq.(2.21) presents the secondary fluid energy balance for the device.

$$\dot{Q}_{ev,sf} = \dot{m}_{sf,ev} \, (h_4 - h_3) \tag{2.21}$$

2.2.7 Receiver

The receiver can be controlled so that its pressure is kept at a set reference point, which is considered to be identical to that of the intermediate heat exchanger. The pressure at the receiver inlet and outlet thermodynamic states is, thus, defined as follows, considering no pressure drop across it:

$$P_{rec} = P_5 \tag{2.22}$$

$$P_6 = P_{rec} \tag{2.23}$$

$$P_1 = P_{rec} \tag{2.24}$$

Further, the energy balance at the device is:

$$\dot{m}_{sf,ev} \cdot h_5 + \dot{m}_{sf,cd} \cdot h_7 = \dot{m}_{sf,ev} \cdot h_1 + \dot{m}_{sf,cd} \cdot h_6 \tag{2.25}$$

2.2.8 Intermediate heat exchanger – secondary fluid condenser

Eqs.(2.26) and (2.27) set the pressure level related to the condenser in the secondary cycle. Pressure drop is neglected at this part of the device.

$$P_{cd,sf} = P_{rec} \tag{2.26}$$

$$P_7 = P_{rec} \tag{2.27}$$

Condenser outlet degree of subcooling is represented by Eq.(2.28).

$$T_7 = T_{cd,sf,bub} - \Delta T_{cd,sf,sc} \tag{2.28}$$

Eq.(2.29) describes the secondary fluid energy balance for the component.

$$\dot{Q}_{cd,sf} = \dot{m}_{sf,cd}\,(h_6 - h_7) \tag{2.29}$$

2.2.9 Lines

Regarding the modeling of the connecting lines, the refrigerant flow is considered to endure pressure drop and heat transfer. For the latter, to preserve the "thermodynamic" nature of the model, a simple $\Delta T = T_{out} - T_{in}$ description is here employed, as a less simplified model would, otherwise, require specific heat transfer information.

The pressure drops in the discharge, liquid, suction, high pressure liquid and low pressure valor lines are represented from Eqs.(2.30) to (2.34), respectively.

$$P_9 = P_{10} + \Delta P_{dl} \tag{2.30}$$

$$P_{12} = P_{11} - \Delta P_{ll,rf} \tag{2.31}$$

$$P_8 = P_{14} - \Delta P_{sl} \tag{2.32}$$

$$P_2 = P_3 + \Delta P_{ll,sf} \tag{2.33}$$

$$P_5 = P_4 - \Delta P_{vl} \tag{2.34}$$

For heat transfer, in a similar fashion, temperature changes are described in Eqs.(2.35) to (2.39) for, respectively, discharge line, liquid line, suction line, high pressure liquid line and low pressure valor line.

$$T_{10} = T_9 - \Delta T_{dl} \tag{2.35}$$

$$T_{12} = T_{11} - \Delta T_{ll,rf} \tag{2.36}$$

$$T_8 = T_{14} + \Delta T_{sl} \tag{2.37}$$

$$T_3 = T_2 + \Delta T_{ll,sf} \tag{2.38}$$

$$T_5 = T_4 + \Delta T_{vl} \tag{2.39}$$

2.2.10 Refrigerant and secondary fluid properties

Eqs.(2.40) to (2.62), below, describe schematically property functions that calculate thermodynamic refrigerant properties, for a given thermodynamic state, as a function of two other intensive properties. They are: specific enthalpy, $\underline{h}$, specific entropy, $\underline{s}$, temperature, $\underline{T}$, specific volume, $\underline{v}$, and vapor quality, $\underline{x}$.

Algorithms T_{cond} $(P_{cond}, \Delta P)$, P_{evap} $(T_{evap}, \Delta P)$, P_{cond} $(T_{cond}, \Delta P)$ and P_{evap} $(T_{evap}, \Delta P, h_{in})$, applied in Eqs.(2.63) to (2.66), are employed to obtain the average pressure or average temperature in terms of one another and of other parameters. Appendix B presents a detailed explanation of the structure of such algorithms.

$$h_1 = \underline{h}\ (x = 0, P_1) \tag{2.40}$$

$$s_1 = \underline{s}\ (x = 0, P_1) \tag{2.41}$$

$$h_{2i} = \underline{h}\ (s_{2i}, P_2) \tag{2.42}$$

$$T_2 = \underline{T}\ (h_2, P_2) \tag{2.43}$$

$$h_3 = \underline{h}\ (T_3, P_3) \tag{2.44}$$

$$h_4 = \underline{h}\ (T_4, P_4) \tag{2.45}$$

$$h_5 = \underline{h}\ (T_5, P_5) \tag{2.46}$$

$$h_6 = \underline{h}\ (x = 1, P_6) \tag{2.47}$$

$$h_7 = \underline{h}\ (T_7, P_7) \tag{2.48}$$

$$h_8 = \underline{h}\ (T_8, P_8) \tag{2.49}$$

$$s_8 = \underline{s}\ (T_8, P_8) \tag{2.50}$$

$$v_8 = \underline{v}\ (T_8, P_8) \tag{2.51}$$

$$h_{9i} = \underline{h}\ (s_{9i}, P_9) \tag{2.52}$$

$$T_9 = \underline{T}\ (h_9, P_9) \tag{2.53}$$

$$h_{10} = \underline{h}\ (T_{10}, P_{10}) \tag{2.54}$$

$$h_{11} = \underline{h}\ (T_{11}, P_{11}) \tag{2.55}$$

$$h_{12} = \underline{h}\ (T_{12}, P_{12}) \tag{2.56}$$

$$T_{13} = \underline{T}\ (h_{13}, P_{13}) \tag{2.57}$$

$$h_{14} = \underline{h}\ (T_{14}, P_{14}) \tag{2.58}$$

$$T_{cd,rf,bub} = \underline{T}\ \left(x = 0, P = P_{cd,rf} - \frac{\Delta P_{cd,rf}}{2} \right) \tag{2.59}$$

$$T_{ev,rf,dew} = \underline{T}\ \left(x = 1, P = P_{ev,rf} - \frac{\Delta P_{ev,rf}}{2} \right) \tag{2.60}$$

$$T_{cd,sf,bub} = \underline{T}\ \left(x = 0, P = P_{cd,sf} \right) \tag{2.61}$$

$$T_{ev,sf,dew} = \underline{T}\ \left(x = 1, P = P_{ev,sf} - \frac{\Delta P_{ev,sf}}{2} \right) \tag{2.62}$$

$$T_{cd,sf} = T_{cond}\ (P_{cd,sf}, \Delta P_{cd,sf}) \tag{2.63}$$

$$P_{ev,sf} = P_{evap}\ (T_{ev,sf}, \Delta P_{ev,sf}) \tag{2.64}$$

$$P_{cd,rf} = P_{cond}\ (T_{cd,rf}, \Delta P_{cd,rf}) \tag{2.65}$$

$$P_{ev,rf} = P_{evap}\ (T_{ev,rf}, \Delta P_{ev,rf}, h_{13}) \tag{2.66}$$

2.2.11 Refrigerating capacity

Eq.(2.67) characterizes the heat transfer rate taking place in the intermediate heat exchanger, for capacity analysis. Heat losses at the intermediate heat exchanger are, in this case, neglected.

$$\dot{Q}_{ev,rf} = \dot{Q}_{cd,sf} \tag{2.67}$$

2.3 Input data

A detailed description of the set of input data required for the simulation of the model developed is presented. For each parameter, there is at least one option to choose as possible input information. All alternatives for each paremeter are, hereafter, defined and explained.

Although the computational model is capable of working with any combination of input information, and the set of physical phenomena is exactly the same, due to shortage of space, the mathematical model presented in

Section 2.2 considers only one specific set of input data. Thus, choosing different types of input for a given parameter would require small adaptations to the equations exposed.

With that in mind, in order to clarify the set of data considered, the options of input information associated to the mathematical model presented are highlighted in bold.

1. Compressor isentropic efficiency ($\boldsymbol{\eta_{s,cp}}$)

   Ratio of the work input for an isentropic process to the work input for an actual process under the same inlet conditions and outlet pressure.

2. Compressor volumetric efficiency ($\boldsymbol{\eta_{v,cp}}$)

   Ratio of the delivered mass flow rate to the mass flow rate should refrigerant occupy the entire displaced volume at suction state.

3. Temperature setting in discharge line

   (a) Refrigerant temperature at condenser inlet ($T_{dl,out}$): the refrigerant temperature at inlet of the condenser; or

   (b) Temperature change in discharge line ($\boldsymbol{\Delta T_{dl}}$): the difference in refrigerant temperature between compressor discharge and condenser inlet.

4. Pressure setting in discharge line

   (a) Pressure drop in discharge line ($\boldsymbol{\Delta P_{dl}}$): the difference between refrigerant pressure at compressor discharge and condenser inlet; or

   (b) Reduction of saturation temperature in discharge line ($\Delta T_{dl,sat}$): the difference between refrigerant saturation temperature at compressor discharge and condenser inlet.

5. Refrigerant condensing parameter

   (a) Refrigerant condensing temperature ($\boldsymbol{T_{cd,rf}}$): the average refrigerant temperature inside the refrigerant condenser; or

   (b) Refrigerant condensing pressure ($P_{cd,rf}$).

6. Refrigerant condenser temperature setting

   (a) Refrigerant temperature at condenser outlet ($T_{cd,rf,out}$): the refrigerant temperature at exit of the condenser; or

   (b) Refrigerant subcooling at condenser outlet ($\boldsymbol{\Delta T_{cd,rf,sc}}$): the difference between refrigerant saturation temperature at the exit of the condenser and refrigerant temperature at condenser outlet.

7. Condenser pressure setting

   (a) Pressure drop at refrigerant condenser ($\boldsymbol{\Delta P_{cd,rf}}$): the difference between refrigerant pressures at inlet and outlet of the condenser; or

   (b) Reduction of saturation temperature at refrigerant condenser ($\Delta T_{cd,rf,sat}$): the difference between refrigerant saturation temperatures at inlet and outlet of the condenser.

8. Temperature setting in liquid line

   (a) Expansion device inlet temperature ($T_{ll,rf,out}$); or

   (b) Refrigerant temperature change in liquid line ($\boldsymbol{\Delta T_{ll,rf}}$): the difference in refrigerant temperature at condenser outlet and expansion device inlet; or

   (c) Subcooling at expansion device inlet ($\Delta T_{xd,sc}$): the difference between saturation temperature and refrigerant temperature, both at expansion device inlet.

9. Pressure setting in liquid line

   (a) Refrigerant pressure drop in liquid line ($\boldsymbol{\Delta P_{ll,rf}}$): the difference in refrigerant pressure at condenser outlet and expansion device inlet; or

   (b) Drop of refrigerant saturation temperature in liquid line ($\Delta T_{ll,rf,sat}$): the difference in refrigerant saturation temperature at condenser outlet and expansion device inlet.

10. Refrigerant superheat at evaporator outlet ($\boldsymbol{\Delta T_{ev,rf,sh}}$)

    The difference between the refrigerant temperature at evaporator outlet and the corresponding saturation temperature.

11. Refrigerant evaporator pressure setting

    (a) Refrigerant pressure drop through evaporator ($\boldsymbol{\Delta P_{ev,rf}}$): the difference between refrigerant pressure at inlet and outlet of the evaporator; or

(b) Reduction of saturation temperature at refrigerant evaporator ($\Delta T_{ev,rf,sat}$): the difference between refrigerant saturation temperatures taken at the inlet and outlet of the evaporator.

12. Temperature change in suction line ($\boldsymbol{\Delta T_{sl}}$)

    The difference between refrigerant temperature at compressor inlet and evaporator outlet.

13. Pressure setting in suction line

    (a) Pressure drop across suction line ($\boldsymbol{\Delta P_{sl}}$); or

    (b) Reduction of saturation temperature in suction line ($\Delta T_{sl,sat}$): the difference between refrigerant corresponding saturation temperature at evaporator outlet and compressor inlet.

14. Cooling load ($\boldsymbol{\dot{Q}_{ev,sf}}$)

    Total refrigerating capacity rate.

15. Pump isentropic efficiency ($\boldsymbol{\eta_{s,pp}}$)

    Ratio of the work input for an isentropic process to the work input for an actual process under the same inlet conditions and outlet pressure.

16. Temperature setting in high pressure liquid line

    (a) Secondary fluid temperature at liquid line outlet ($T_{ll,sf,out}$): the secondary fluid temperature at evaporator inlet; or

    (b) Secondary fluid temperature rise in liquid line ($\boldsymbol{\Delta T_{ll,sf}}$): the difference in secondary fluid temperature between evaporator inlet and pump discharge.

17. Pressure setting in high pressure liquid line

    (a) Secondary fluid pressure drop in liquid line ($\boldsymbol{\Delta P_{ll,sf}}$): the difference between secondary fluid pressure at pump discharge and evaporator inlet; or

    (b) Secondary fluid drop of saturation temperature in liquid line ($\Delta T_{ll,sf,sat}$): the difference between secondary fluid corresponding saturation temperature at pump discharge and evaporator inlet.

18. Secondary fluid evaporating parameter

    (a) Secondary fluid evaporating temperature ($\boldsymbol{T_{ev,sf}}$): the average temperature inside the secondary fluid evaporator; or

   (b) Secondary fluid evaporating pressure ($P_{ev,sf}$).

19. Secondary fluid evaporator temperature setting

   (a) Secondary fluid temperature at evaporator outlet ($T_{ev,sf,out}$); or

   (b) Secondary fluid superheat at evaporator outlet ($\boldsymbol{\Delta T_{ev,sf,sh}}$): the difference between secondary fluid temperature and saturation temperature, both at evaporator outlet; or

   (c) Secondary fluid vapor quality at evaporator outlet ($x_{ev,sf,out}$).

20. Secondary fluid evaporator pressure setting

   (a) Pressure drop at secondary fluid evaporator ($\boldsymbol{\Delta P_{ev,sf}}$): the difference between secondary fluid pressure at both inlet and outlet of the evaporator; or

   (b) Reduction of saturation temperature at secondary fluid evaporator ($\Delta T_{ev,sf,sat}$): the difference between secondary fluid saturation temperature at inlet and outlet of the evaporator.

21. Temperature setting in low pressure vapor line

   (a) Vapor line outlet temperature($T_{vl,out}$): the vapor temperature at inlet of the receiver.

   (b) Temperature rise in vapor line ($\boldsymbol{\Delta T_{vl}}$): the difference between vapor temperature at receiver inlet and evaporator outlet; or

   (c) Vapor line outlet vapor quality ($x_{vl,out}$).

22. Pressure setting in low pressure vapor line

   (a) Pressure drop in vapor line ($\boldsymbol{\Delta P_{vl}}$): the difference between vapor pressure at evaporator outlet and receiver inlet; or

   (b) Reduction of saturation temperature in vapor line ($\Delta T_{vl,sat}$): the difference between vapor saturation temperature at evaporator outlet and receiver inlet.

23. Secondary fluid condenser temperature setting

   (a) Secondary fluid temperature at condenser outlet ($T_{cd,sf,out}$): the secondary fluid temperature at exit of the condenser; or

   (b) Secondary fluid subcooling at condenser outlet ($\boldsymbol{\Delta T_{cd,sf,sc}}$): the difference between saturation temperature and secondary fluid temperature, both at condenser outlet.

24. Intermediate temperatures
    (a) Temperature difference in intermediate heat exchanger ($[\boldsymbol{T_{cd,sf} - T_{ev,ref}}]$): the difference between secondary fluid condensing temperature and refrigerant evaporating temperature; or
    (b) Refrigerant evaporating temperature ($T_{ev,rf}$): the average evaporating temperature in the refrigerant evaporator.

2.4 Numerical solution and computational code

Solution of the above system of equations follows the sequence pattern illustrated in Figure 2.3. In addition, without loss of generality, four iteration processes may occur during the solution of the model. They take place inside the heat exchangers considered, and refer to algorithms $T_{cond}\ (P_{cond}, \Delta P)$, $P_{evap}\ (T_{evap}, \Delta P)$, $P_{cond}\ (T_{cond}, \Delta P)$ and $P_{evap}\ (T_{evap}, \Delta P, h_{in})$, which are further detailed in Appendix B.

A computational code comprising the numerical model described was developed in Fortran language. Thermodynamic and transport properties of the refrigerants considered were calculated with REFPROP 9.0 [131] and coupled to the code. REFPROP uses the reportedly most accurate equations available worldwide to calculate thermophysical properties for pure fluids and their mixtures [131, 132].

Results of the simulation are overall performance parameters associated with the refrigeration cycle, as well as detailed information of inlet and outlet fluid states for each of the components.

2.5 Validation

Experimental data of the pumped CO_2 system configuration modeled is still, to the author's knowledge, non-existent in the literature. Theoretical and experimental results may be found, however, for secondary coolant refrigeration systems with arrangements somewhat different from that of the pumped CO_2, for instance, those provided by Sawalha [37] and Hinde et al. [41].

Although specifically discussing the CO_2 secondary technology modeled in this work, Sawalha [37] chose to develop a computer simulation model of a different indirect system. As depicted in Figure 1.16, the author opted to simulate the circuit design in 1.16(a) instead of that in 1.16(b), the latter corresponding to the model studied here, which prevents any further theoretical comparisons to be produced.

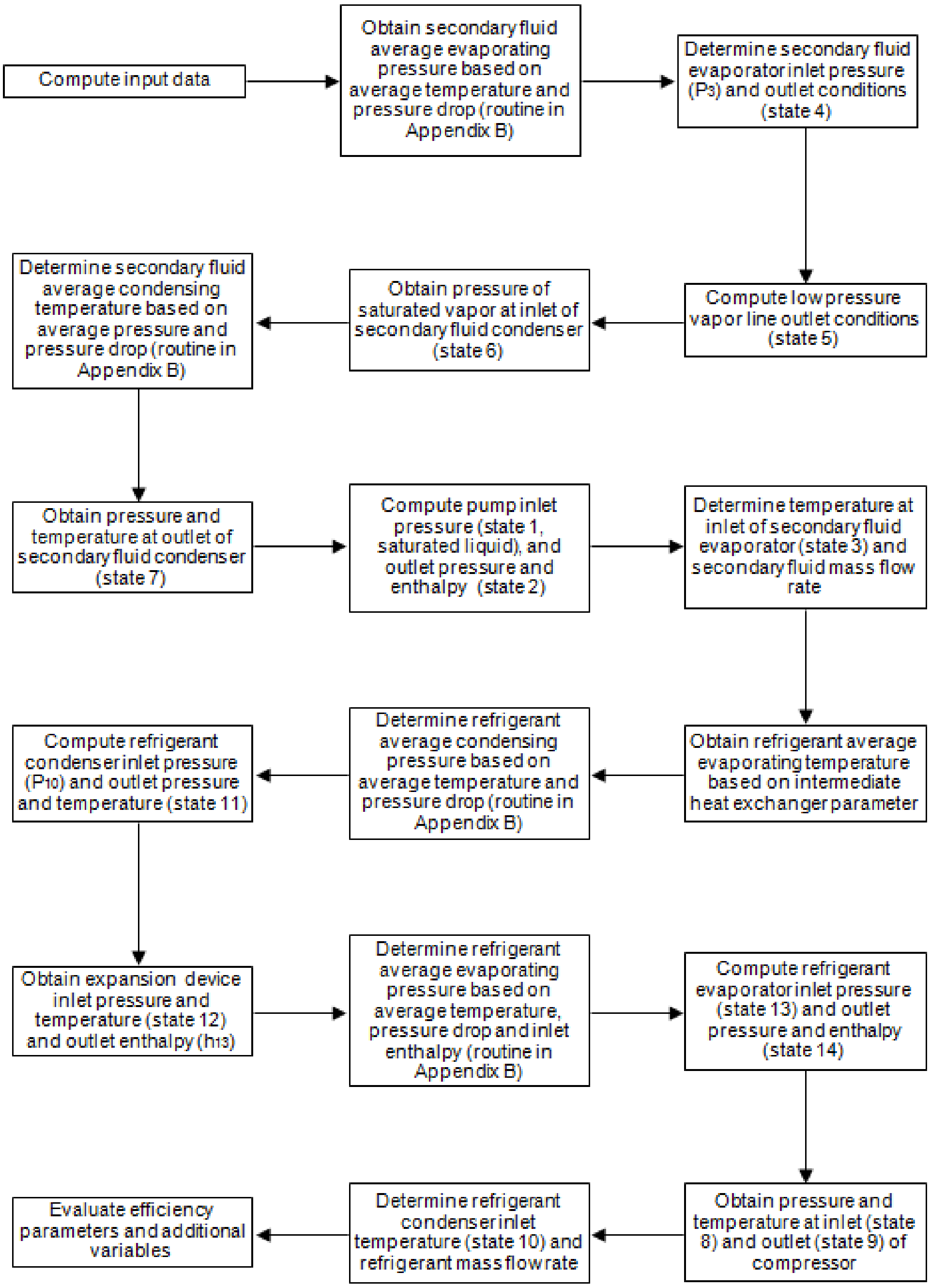
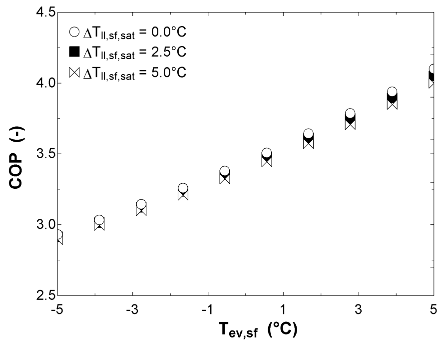


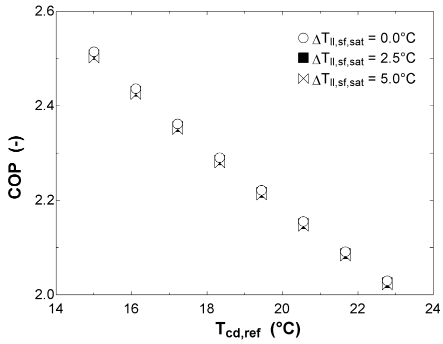
Figure 2.3: Computational sequence of the pumped CO_2 model solution.

Hinde et al. [41] presented simulation results of a system operating with CO_2 as secondary fluid for low temperature applications. The authors showed the total annual system energy consumption for secondary coolant systems at selected locations. However, the arrangement of the secondary circuit was different from that of the pumped CO_2, thus precluding, once again, a more detailed comparison.

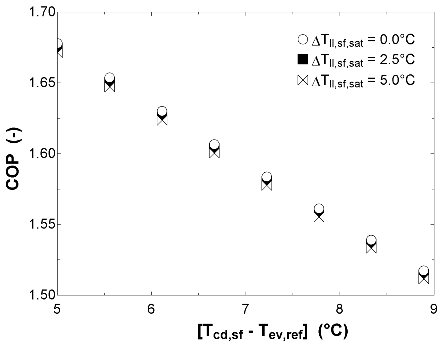
Nevertheless, a number of predictable trends associated with the pumped CO_2 refrigeration system are evaluated, in order to verify the goodness of the model. Thus, it is possible to check if well known conditions related to



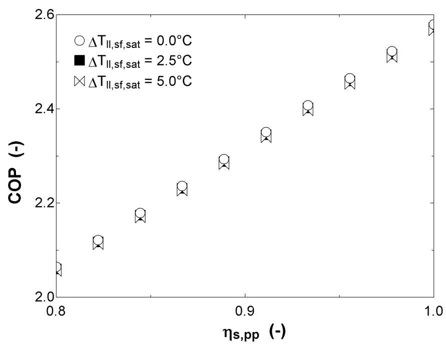
2.4(a): COP against secondary fluid average temperature in the evaporator.



2.4(b): COP against refrigerant average temperature in the condenser.



2.4(c): COP against temperature difference at the intermediate heat exchanger.



2.4(d): COP against compressor isentropic efficiency.

Figure 2.4: Parametric analysis for the pumped CO_2 refrigeration system.

the technology considered are present. For this, the coefficient of performance (COP) was plotted against four different parametric variables: average temperature in the secondary fluid evaporator, average temperature in the refrigerant condenser, compressor isentropic efficiency and temperature difference in the intermediate heat exchanger.

The performance of refrigeration system can be expressed in terms of the COP, defined by Eq.(2.68).

$$COP = \frac{\text{cooling capacity}}{\text{power input}} \tag{2.68}$$

For the pumped CO_2 technology, application of this definition results in Eq.(2.69).

$$COP_{Pumped} = \frac{\dot{Q}_{ev,sf}}{\dot{W}_{cp} + \dot{W}_{pp}} \tag{2.69}$$

The input data considered for such analysis are presented in Table 2.1. Highlighted parameters are varied when their influence is to be studied. The main cycle operates with R404A, whilst CO_2 is used as secondary fluid. Figure

Table 2.1: Input data for pumped CO_2 model goodness verification. Highlighted parameters are varied when their influence is studied.

Parameter	Value
Compressor isentropic efficiency	0.65
Compressor volumetric efficiency	1.00
Temperature change in discharge line	0°C
Pressure drop in discharge line	0 kPa
Refrigerant condensing temperature	30°C
Refrigerant subcooling at condenser outlet	0°C
Pressure drop at refrigerant condenser	0 kPa
Temperature change in liquid line	5°C
Pressure drop in liquid line	0 kPa
Refrigerant superheat at evaporator outlet	5°C
Pressure drop at refrigerant evaporator	0 kPa
Temperature change in suction line	5°C
Pressure drop in suction line	0 kPa
Cooling load	100 kW
Pump isentropic efficiency	0.60
Temperature rise in high pressure liquid line	0°C
Drop of saturation temperature in high pressure liquid line	0.1°C
Secondary fluid evaporating temperature	−30°C
Secondary fluid superheat at evaporator outlet	3°C
Drop of saturation temperature at secondary fluid evaporator	0.5°C
Temperature rise in low pressure vapor line	5°C
Drop of saturation temperature in low pressure vapor line	2°C
Secondary fluid subcooling at condenser outlet	3°C
Temperature difference between refrigerant and secondary fluid at intermediate heat exchanger	10°C

2.4 presents the parametric analysis.

The trends follow what is expected of the model. The COP decreases as lower secondary fluid evaporating temperatures, Figure 2.4(a), and higher refrigerant condensing temperatures, Figure 2.4(b), are considered. Further, increasing the temperature difference in the intermediate heat exchanger results in lower values of COP, Figure 2.4(c), whereas the increase in compressor isentropic efficiency influences positively the system performance, Figure 2.4(d). Finally, one can observe that lower performance is obtained when greater reductions of the secondary fluid saturation temperatre in the liquid line are considered, Figures 2.4(a) to 2.4(d).