3
CO_2 booster refrigeration system

In similar fashion, a two-stage transcritical cycle operating with CO_2 as a refrigerant is modeled and simulated.

3.1
Overview

Sixteen control volumes comprise the CO_2 booster refrigeration system, Figure 3.1, namely: high stage compressor (*cp,hs*), discharge line (*dl*), gas cooler or condenser (*gc*), liquid line upstream (*ll,uhx*) and downstream (*ll,dhx*) heat exchanger, high pressure control valve (*xd,cv*), receiver (*rec*), medium temperature (*xd,mt*) and low temperature (*xd,lt*) expansion valves, medium temperature (*ev,mt*) and low temperature (*ev,lt*) evaporators, low stage compressor (*cp,ls*), bypass valve (*xd,bv*), suction line upstream (*sl,uhx*) and downstream (*sl,dhx*) heat exchanger, and suction line heat exchanger (*shx*).

The refrigerant enters the high stage compressor (*cp,hs*) as superheated vapor at state 1, being compressed to a high pressure refrigerant at state 2. During this process, the temperature of the refrigerant increases to well above the temperature of the surroundings. When the high side pressure is above the critical value, the system operates in the transcritical region. Flowing through the discharge line (*dl*), the refrigerant experiences pressure drop and heat transfer, exiting the device at the condenser or gas cooler inlet pressure, depending on ambient conditions.

The refrigerant, then, enters the gas cooler or condenser (*gc*) at state 3, rejecting heat to the surrounding medium and leaving at state 4. The temperature of the surroundings at this state is still below that of the refrigerant. When passing through the liquid line upstream (*ll,uhx*) and downstream (*ll,dhx*) the heat exchanger, the refrigerant experiences once again pressure drop and heat transfer. At the suction line heat exchanger (*shx*), the high pressure refrigerant entering at state 5 is further cooled to state 6 by the low temperature superheated vapor on the other side, which, by its turn, is heated from state 19 to 20.

The high pressure refrigerant at state 7 is, then, throttled to the receiver pressure as it passes through a high pressure control valve (xd,cv). During this process, the pressure of the refrigerant drops below the critical value, in case the system operates in the transcritical region in the high pressure section. The liquid refrigerant expands in the high pressure control valve (xd,cv) to the receiver (rec) pressure, with part of the liquid vaporizing during this process.

This saturated vapor (state 9) expands through the bypass valve (xd,bv) and is mixed with the superheated vapor (state 17), with the resulting mixture entering the suction line at state 18. The refrigerant, then, experiences another pressure drop and temperature change when passing through the suction line upstream (sl,uhx) and downstream (sl,dhx) the heat exchanger (sl,ahx), and the outlet superheated vapor, after being heated at the suction heat exchanger (shx) in between the lines, enters the high stage compressor at state 1.

The saturated liquid (state 11) expands through the medium temperature (xd,mt) and low temperature (xd,lt) expansion valves, states 12 and 14, into the medium temperature (ev,mt) and low temperature (ev,lt) evaporators, respectively, where it picks up heat from the refrigerated space. The low pressure vapor (state 15) leaving the low temperature evaporator enters, next, the low stage compressor (cp,ls), and is mixed with the medium temperature vapor leaving the medium temperature evaporator (state 13). Then, as previously described, this superheated vapor is mixed with the refrigerant leaving the bypass valve at state 10.

Figure 3.2 presents the pressure-enthalpy diagram of a CO_2 booster thermodynamic cycle. The gas cooler pressure is 8000 kPa, receiver pressure is 5000 kPa and evaporating temperatures are −5.6 and −27.8°C.

3.2 Mathematical model

The mathematical model for the CO_2 booster refrigeration system is outlined next, starting with the detailed description of the control volumes, followed by the modeling of each component. It is worth mentioning that, similarly to the pumped CO_2, although the carbon dioxide is constantly referred to as the operating refrigerant, the model is capable of simulating the booster technology for different fluids.

3.2.1 Compressors

Similarly to the strategy employed for the pumped CO_2 device, Section 2.2.1, an efficiency-based model is considered for the low stage, Eqs.(3.1) to

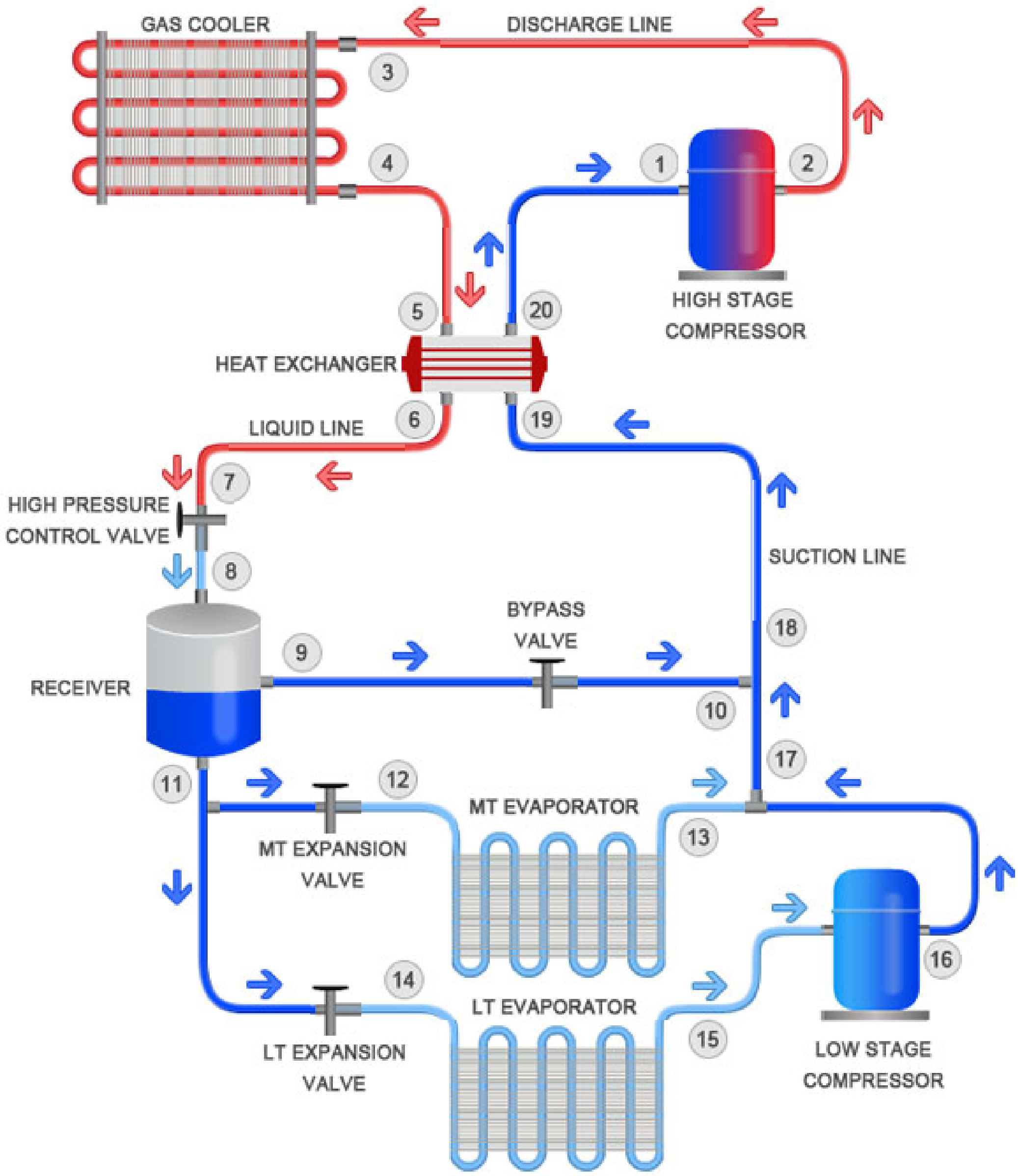


Figure 3.1: Scheme comprising the sixteen control volumes of the CO_2 booster refrigeration system [130].

(3.4), and the high stage, Eqs.(3.5) to (3.8), compressors of the CO_2 booster cycle, respectively.

$$\eta_{s,cp,ls} = \frac{(h_{16i} - h_{15})}{(h_{16} - h_{15})} \tag{3.1}$$

$$s_{16i} = s_{15} \tag{3.2}$$

$$\eta_{v,cp,ls} = \frac{v_{15}\dot{m}_{lt}}{\dot{V}_{cp,ls}} \tag{3.3}$$

$$\dot{W}_{cp,ls} = \dot{m}_{lt}\ (h_{16} - h_{15}) \tag{3.4}$$

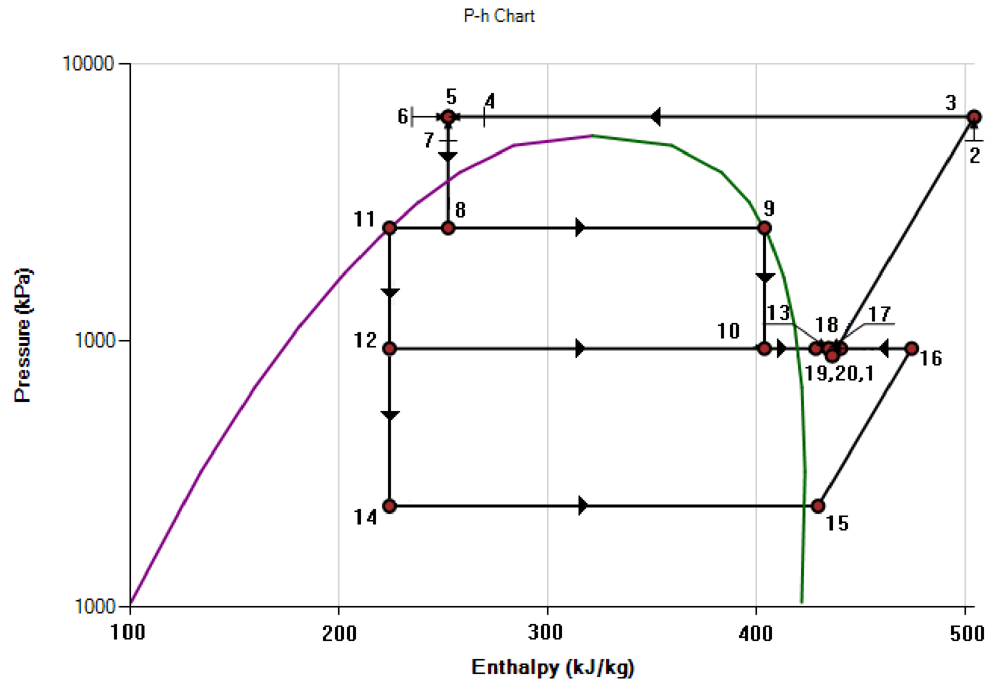


Figure 3.2: P-h diagram of a CO_2 booster cycle, with gas cooler pressure of 8000 kPa, receiver pressure of 5000 kPa and evaporating temperatures of −5.6 and −27.8°C [130].

$$\eta_{s,cp,hs} = \frac{(h_{2i} - h_1)}{(h_2 - h_1)} \tag{3.5}$$

$$s_{2i} = s_1 \tag{3.6}$$

$$\eta_{v,cp,hs} = \frac{v_1 \dot{m}_{rf}}{\dot{V}_{cp,hs}} \tag{3.7}$$

$$\dot{W}_{cp,hs} = \dot{m}_{rf} \, (h_2 - h_1) \tag{3.8}$$

3.2.2 Gas cooler or condenser

The pressure drop at gas cooler or condenser (depending on ambient temperature) is described by Eqs.(3.9) and (3.10), whereas the temperature at outlet of the device is represented by Eq.(3.11). Finally, Eq.(3.12) shows the energy balance for the refrigerant in the device.

$$P_3 = P_{gc} + \frac{\Delta P_{gc}}{2} \tag{3.9}$$

$$P_4 = P_{gc} - \frac{\Delta P_{gc}}{2} \tag{3.10}$$

$$T_4 = T_{gc,out} \tag{3.11}$$

$$\dot{Q}_{gc} = \dot{m}_{rf}\ (h_3 - h_4) \tag{3.12}$$

3.2.3 Suction line heat exchanger

The temperature change for transcritical or liquid refrigerant at the suction heat exchanger is considered in Eq.(3.13). Alternatively, it is possible to consider the temperature change of the superheated vapor flowing on the other side of the component instead, or other descriptive parameters which have the same applicability, as explained in Section 3.3, to follow.

$$T_6 = T_5 - \Delta T_{shx,liq} \tag{3.13}$$

Eqs.(3.14) and (3.15) describe the pressure drop of transcritical or liquid refrigerant and superheated vapor through the heat exchanger, respectively.

$$P_6 = P_5 - \Delta P_{shx,liq} \tag{3.14}$$

$$P_{20} = P_{19} - \Delta P_{shx,vap} \tag{3.15}$$

Energy balance at the suction line heat exchanger is described by Eq.(3.16).

$$h_5 + h_{19} = h_6 + h_{20} \tag{3.16}$$

3.2.4 Expansion devices

Analogously to the assumption considered for the pumped CO_2 cycle, expansion processes for all CO_2 booster expansion valves are assumed isenthalpic [129].

$$h_8 = h_7 \tag{3.17}$$

$$h_{12} = h_{11} \tag{3.18}$$

$$h_{14} = h_{11} \tag{3.19}$$

$$h_{10} = h_9 \tag{3.20}$$

3.2.5 Receiver

The receiver can be controlled so that its pressure is kept at a set reference point [70]. The pressure at receiver inlet and outlet thermodynamic states are defined as follows, assuming no pressure drop:

$$P_8 = P_{rec} \tag{3.21}$$

$$P_9 = P_{rec} \tag{3.22}$$

$$P_{11} = P_{rec} \tag{3.23}$$

3.2.6 Evaporators

Similarly to the approach for the pumped CO_2 heat exchangers, nominal pressure is taken as the middle pressure in the coils. Pressure drops at the medium temperature and low temperature evaporators are described from Eqs.(3.24) to (3.25), and from Eqs.(3.28) to (3.29), respectively. Evaporator outlet degree of superheat is outlined in Eqs.(3.26) and (3.30). Finally, Eqs.(3.27) and (3.31) represent the energy balance expressions for the refrigerant in each evaporator.

$$P_{12} = P_{ev,mt} + \frac{\Delta P_{ev,mt}}{2} \tag{3.24}$$

$$P_{13} = P_{ev,mt} - \frac{\Delta P_{ev,mt}}{2} \tag{3.25}$$

$$T_{13} = T_{ev,mt,dew} + \Delta T_{ev,mt,sh} \tag{3.26}$$

$$\dot{Q}_{ev,mt} = \dot{m}_{mt} \left(h_{13} - h_{12}\right) \tag{3.27}$$

$$P_{14} = P_{ev,lt} + \frac{\Delta P_{ev,lt}}{2} \tag{3.28}$$

$$P_{15} = P_{ev,lt} - \frac{\Delta P_{ev,lt}}{2} \tag{3.29}$$

$$T_{15} = T_{ev,lt,dew} + \Delta T_{ev,lt,sh} \tag{3.30}$$

$$\dot{Q}_{ev,lt} = \dot{m}_{lt} \left(h_{15} - h_{14}\right) \tag{3.31}$$

3.2.7 Mixing stages

Eqs.(3.32) to (3.35) set pressure values for the thermodynamic states before and after the two mixing points.

$$P_{16} = P_{13} \tag{3.32}$$

$$P_{17} = P_{13} \tag{3.33}$$

$$P_{10} = P_{17} \tag{3.34}$$

$$P_{18} = P_{17} \tag{3.35}$$

Energy balance for the mixing between the refrigerant leaving the low stage compressor and the refrigerant exiting the medium temperature evaporator is represented by Eq.(3.36).

$$\dot{m}_{mt} \cdot h_{13} + \dot{m}_{lt} \cdot h_{16} = \dot{m}_{liq} \cdot h_{17} \tag{3.36}$$

Eq.(3.37) describes the energy balance when the resulting refrigerant at state 17 is mixed with the refrigerant leaving the bypass valve.

$$x_8 \cdot h_{10} + (1 - x_8)\ h_{17} = h_{18} \tag{3.37}$$

3.2.8 Lines

Analogously to the pumped CO_2 system modeling, Section 2.2.9, simple models for pressure drop and heat transfer are assumed for the refrigerant as it passes through the different lines in the CO_2 booster cycle.

Pressure losses in discharge line, liquid line upstream and downstream heat exchanger, and suction line upstream and downstream heat exchanger, are described by, respectively, Eqs.(3.38) to 3.42.

$$P_2 = P_3 + \Delta P_{dl} \tag{3.38}$$

$$P_5 = P_4 - \Delta P_{ll,uhx} \tag{3.39}$$

$$P_7 = P_6 - \Delta P_{ll,dhx} \tag{3.40}$$

$$P_1 = P_{20} - \Delta P_{sl,uhx} \tag{3.41}$$

$$P_{19} = P_{18} - \Delta P_{sl,dhx} \tag{3.42}$$

Regarding heat transfer processes, temperature changes are represented in Eqs.(3.43) to 3.47 for, respectively, discharge line, liquid line upstream and downstream heat exchanger, and suction line upstream and downstream heat exchanger.

$$T_3 = T_2 - \Delta T_{dl} \tag{3.43}$$

$$T_5 = T_4 - \Delta T_{ll,uhx} \tag{3.44}$$

$$T_7 = T_6 - \Delta T_{ll,dhx} \tag{3.45}$$

$$T_1 = T_{20} + \Delta T_{sl,uhx} \tag{3.46}$$

$$T_{19} = T_{18} + \Delta T_{sl,dhx} \tag{3.47}$$

3.2.9 Refrigerant properties

Calculation of thermodynamic refrigerant properties, Eqs.(3.48) to (3.75), follows the same pattern developed in Section 2.2.10 for the pumped CO_2 refrigeration system.

Algorithm P_{evap} $(T_{evap}, \Delta P)$, detailed in Appendix B and applied in Eqs.(3.76) and (3.77), is employed to obtain average pressure in terms of average temperature and pressure drop.

$$h_1 = \underline{h}\ (T_1, P_1) \tag{3.48}$$

$$s_1 = \underline{s}\ (T_1, P_1) \tag{3.49}$$

$$v_1 = \underline{v}\ (T_1, P_1) \tag{3.50}$$

$$h_{2i} = \underline{h}\ (s_{2i}, P_2) \tag{3.51}$$

$$T_2 = \underline{T}\ (h_2, P_2) \tag{3.52}$$

$$h_3 = \underline{h}\ (T_3, P_3) \tag{3.53}$$

$$h_4 = \underline{h}\ (T_4, P_4) \tag{3.54}$$

$$h_5 = \underline{h}\ (T_5, P_5) \tag{3.55}$$

$$h_6 = \underline{h}\ (T_6, P_6) \tag{3.56}$$

$$h_7 = \underline{h}\ (T_7, P_7) \tag{3.57}$$

$$x_8 = \underline{x}\ (h_8, P_8) \tag{3.58}$$

$$h_9 = \underline{h}\ (x = 1, P_9) \tag{3.59}$$

$$T_{10} = \underline{T}\ (h_{10}, P_{10}) \tag{3.60}$$

$$h_{11} = \underline{h}\ (x = 0, P = P_{11}) \tag{3.61}$$

$$T_{12} = \underline{T}\ (h_{12}, P_{12}) \tag{3.62}$$

$$h_{13} = \underline{h}\ (T_{13}, P_{13}) \tag{3.63}$$

$$T_{14} = \underline{T}\ (h_{14}, P_{14}) \tag{3.64}$$

$$h_{15} = \underline{h}\ (T_{15}, P_{15}) \tag{3.65}$$

$$s_{15} = \underline{s}\ (T_{15}, P_{15}) \tag{3.66}$$

$$v_{15} = \underline{v}\ (T_{15}, P_{15}) \tag{3.67}$$

$$h_{16i} = \underline{h}\ (s_{16i}, P_{16}) \tag{3.68}$$

$$T_{16} = \underline{T}\ (h_{16}, P_{16}) \tag{3.69}$$

$$T_{17} = \underline{T}\ (h_{17}, P_{17}) \tag{3.70}$$

$$T_{18} = \underline{T}\ (h_{18}, P_{18}) \tag{3.71}$$

$$h_{19} = \underline{h}\ (T_{19}, P_{19}) \tag{3.72}$$

$$T_{20} = \underline{T}\ (h_{20}, P_{20}) \tag{3.73}$$

$$T_{ev,mt,dew} = \underline{T}\ \left(x = 1, P = P_{ev,mt} - \frac{\Delta P_{ev,mt}}{2}\right) \tag{3.74}$$

$$T_{ev,lt,dew} = \underline{T}\ \left(x = 1, P = P_{ev,lt} - \frac{\Delta P_{ev,lt}}{2}\right) \tag{3.75}$$

$$P_{ev,mt} = P_{evap}\ (T_{ev,mt}, \Delta P_{ev,mt}) \tag{3.76}$$

$$P_{ev,lt} = P_{evap}\ (T_{ev,lt}, \Delta P_{ev,lt}) \tag{3.77}$$

3.2.10 Refrigerating capacity

Capacity parameters relating the medium or low temperature cooling loads to the total load are defined as follows:

$$\dot{Q}_{ev,mt} = r_{mt} \cdot \dot{Q}_{ev} \tag{3.78}$$

$$\dot{Q}_{ev,lt} = r_{lt} \cdot \dot{Q}_{ev} \tag{3.79}$$

They are associated to one another as expressed below:

$$r_{lt} + r_{mt} = 1 \tag{3.80}$$

Eqs.(3.81) to (3.83) express the algebra associated with the mass flow rates observed to be part of the model.

$$\dot{m}_{lt} + \dot{m}_{mt} = \dot{m}_{liq} \tag{3.81}$$

$$\dot{m}_{liq} + \dot{m}_{vap} = \dot{m}_{rf} \tag{3.82}$$

$$\dot{m}_{liq} = (1 - x_8)\ \dot{m}_{rf} \tag{3.83}$$

3.3 Input data

A detailed description of the set of input data required to simulate the model is outlined. For each parameter, at least one alternative may be selected as input information. All options are defined and explained next.

Although the computational model is capable of working with any combination of input information, similarly to the pumped CO_2 system, the mathematical model here described considers one specific set of input data. Selecting different types of input for a given parameter would, therefore, require small adaptations to the set of expressions presented.

With that in mind, once again, the alternatives of input information associated to the mathermatical model developed in Section 3.2 are highlighted in bold.

1. First refrigerating capacity parameter
   (a) Cooling load at medium temperature evaporator ($\dot{Q}_{ev,mt}$): cooling capacity rate at medium temperature evaporator; or
   (b) **Total cooling load ($\boldsymbol{\dot{Q}_{ev}}$)**: total cooling capacity rate (low temperature plus medium temperature).
2. Second refrigerating capacity parameter
   (a) Cooling load at low temperature evaporator ($\dot{Q}_{ev,lt}$): cooling capacity rate at low temperature evaporator; or
   (b) **Ratio of medium temperature cooling load over total cooling load ($\boldsymbol{r_{mt}}$)**: ratio of cooling load at medium temperature evaporator to total cooling load; or
   (c) Ratio of low temperature cooling load over total cooling load (r_{lt}): ratio of cooling load at low temperature evaporator to total cooling load.
3. **High stage compressor isentropic efficiency ($\boldsymbol{\eta_{s,cp,hs}}$)**
4. **High stage compressor volumetric efficiency ($\boldsymbol{\eta_{v,cp,hs}}$)**

5. Low stage compressor isentropic efficiency ($\boldsymbol{\eta_{s,cp,ls}}$)

6. Low stage compressor volumetric efficiency ($\boldsymbol{\eta_{s,cp,ls}}$)

7. Temperature setting in discharge line

   (a) Condenser or gas cooler inlet temperature ($T_{dl,out}$): the temperature at inlet of the condenser or gas cooler; or

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

8. Pressure drop in discharge line ($\boldsymbol{\Delta P_{dl}}$)

   The difference between pressure at compressor discharge and condenser or gas cooler inlet.

9. Condenser or gas cooler pressure ($\boldsymbol{P_{gc}}$)

10. Condenser or gas cooler outlet temperature ($\boldsymbol{T_{gc,out}}$)

11. Pressure drop at condenser or gas cooler ($\boldsymbol{\Delta P_{gc}}$)

    The difference in pressure between inlet and outlet of the condenser or gas cooler.

12. Temperature setting in liquid line upstream heat exchanger

    (a) Liquid line temperature at inlet heat exchanger ($T_{ll,uhx,out}$); or

    (b) Temperature change in liquid line upstream heat exchanger ($\boldsymbol{\Delta T_{ll,uhx}}$): the difference between temperature at condenser or gas cooler outlet and heat exchanger inlet.

13. Pressure drop in liquid line upstream heat exchanger ($\boldsymbol{\Delta P_{ll,uhx}}$)

    The difference between pressure at condenser or gas cooler outlet and heat exchanger inlet.

14. Temperature setting in liquid line downstream heat exchanger

    (a) Temperature at inlet expansion device ($T_{ll,dhx,out}$); or

    (b) Temperature change in liquid line downstream heat exchanger ($\boldsymbol{\Delta T_{ll,dhx}}$): the difference between temperature at heat exchanger outlet and expansion device inlet.

15. Pressure drop in liquid line downstream heat exchanger ($\boldsymbol{\Delta P_{ll,dhx}}$)

    The difference between pressure at heat exchanger outlet and expansion device inlet.

16. Temperature setting in suction line downstream heat exchanger

    (a) Suction line temperature at inlet heat exchanger ($T_{sl,dhx,out}$); or

    (b) Temperature rise in suction line before heat exchanger ($\boldsymbol{\Delta T_{sl,dhx}}$): the difference between temperature at heat exchanger inlet and suction line inlet.

17. Pressure setting in suction line downstream heat exchanger

    (a) Pressure drop in suction line downtream heat exchanger ($\boldsymbol{\Delta P_{sl,dhx}}$): the difference between pressure at suction line inlet and heat exchanger inlet; or

    (b) Reduction of saturation temperature in suction line downstream heat exchanger ($\Delta T_{sl,dhx,sat}$): the difference between saturation temperature at suction line inlet and heat exchanger inlet.

18. Temperature setting in suction line upstream heat exchanger

    (a) Temperature at high stage compressor inlet ($T_{sl,ahx,out}$); or

    (b) Temperature rise in suction line upstream heat exchanger ($\boldsymbol{\Delta T_{sl,uhx}}$): the difference between temperature at compressor inlet and heat exchanger outlet.

19. Pressure setting in suction line upstream heat exchanger

    (a) Pressure drop in suction line upstream heat exchanger ($\boldsymbol{\Delta P_{sl,uhx}}$): the difference between pressure at heat exchanger outlet and compressor inlet; or

    (b) Reduction of saturation temperature in suction line upstream heat exchanger($\Delta T_{sl,uhx,sat}$): the difference betweenn saturation temperature at heat exchanger outlet and compressor inlet.

20. Suction line heat exchanger efficiency

    (a) Suction line heat exchanger effectiveness (ϵ_{shx}): ratio of the actual rate of heat transfer in the given exchanger to the hypothetical rate of heat transfer if the area of the exchanger were infinite; or

(b) Liquid or transcritical refrigerant temperature change ($\boldsymbol{\Delta T_{shx,liq}}$): the difference between temperature at inlet and outlet of heat exchanger for liquid or transcritical refrigerant; or

(c) Liquid or transcritical refrigerant outlet temperature ($T_{shx,liq,out}$); or

(d) Superheated vapor temperature change ($\Delta T_{shx,vap}$): the difference between temperature at outlet and inlet of heat exchanger for superheated vapor; or

(e) Superheated vapor outlet temperature ($T_{shx,vap,out}$).

21. Liquid or transcritical refrigerant pressure drop ($\boldsymbol{\Delta P_{shx,liq}}$)

    The difference between inlet and outlet pressures of liquid or transcritical refrigerant through the heat exchanger.

22. Pressure setting for superheated vapor at heat exchanger

    (a) Superheated vapor pressure drop at heat exchanger ($\boldsymbol{\Delta P_{shx,vap}}$): the difference between inlet and outlet pressures of superheated vapor through the heat exchanger; or

    (b) Superheated vapor drop of saturation temperature at heat exchanger ($\Delta T_{shx,vap,sat}$): the difference between inlet and outlet saturation temperatures of superheated vapor through the heat exchanger.

23. Receiver pressure ($\boldsymbol{P_{rec}}$).

24. Medium temperature evaporating parameter

    (a) Medium evaporating temperature ($\boldsymbol{T_{ev,mt}}$): the average temperature inside the medium temperature evaporator; or

    (b) Medium evaporating pressure ($P_{ev,mt}$).

25. Medium temperature evaporator temperature setting

    (a) Medium temperature evaporator outlet temperature ($T_{ev,mt,out}$); or

    (b) Medium temperature evaporator outlet superheat ($\boldsymbol{\Delta T_{ev,mt,sh}}$): the difference between refrigerant temperature and saturation temperature, both at medium temperature evaporator outlet.

26. Medium temperature evaporator pressure setting

    (a) Pressure drop in medium temperature evaporator ($\boldsymbol{\Delta P_{ev,mt}}$): the difference between pressure at inlet and outlet of the medium temperature evaporator; or

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

27. Low temperature evaporating parameter

(a) Low evaporating temperature ($\boldsymbol{T_{ev,lt}}$): the average temperature inside the low temperature evaporator; or

(b) Low evaporating pressure ($P_{ev,lt}$).

28. Low temperature evaporator temperature setting

(a) Low temperature evaporator outlet temperature ($T_{ev,lt,out}$); or

(b) Low temperature evaporator outlet superheat ($\boldsymbol{\Delta T_{ev,lt,sh}}$): the difference between refrigerant temperature and saturation temperature, both at low temperature evaporator outlet.

29. Low temperature evaporator pressure setting

(a) Pressure drop in low temperature evaporator ($\boldsymbol{\Delta P_{ev,lt}}$): the difference between pressure at inlet and outlet of the low temperature evaporator; or

(b) Drop of saturation temperature in low temperature evaporator ($\Delta T_{ev,lt,sat}$): the difference between saturation temperature at inlet and outlet of the low temperature evaporator.

3.4 Numerical solution and computational code

Figure 3.3 summarizes the sequence pattern considered for the solving process. In the case of the CO_2 booster, three iteration processes can occur during the solution of the model. Two of them refer to algorithm P_{evap} (T_{evap}, ΔP), taking place in each of the evaporators, and thoroughly described in Appendix B.

The other iterative procedure is associated with the efficiency parameter for suction line heat exchanger. In the situation where the temperature difference for the liquid side or the liquid outlet temperature are entered as input, no iteration process is necessary: the flow chart represented in Figure 3.3 is executed only once. That is not the case when one of the following paremeters is prescribed as input: temperature difference for the vapor side, vapor outlet temperature or heat exchanher effectiveness. Then, when any of these three parameters is selected for describing the suction heat exchanger,

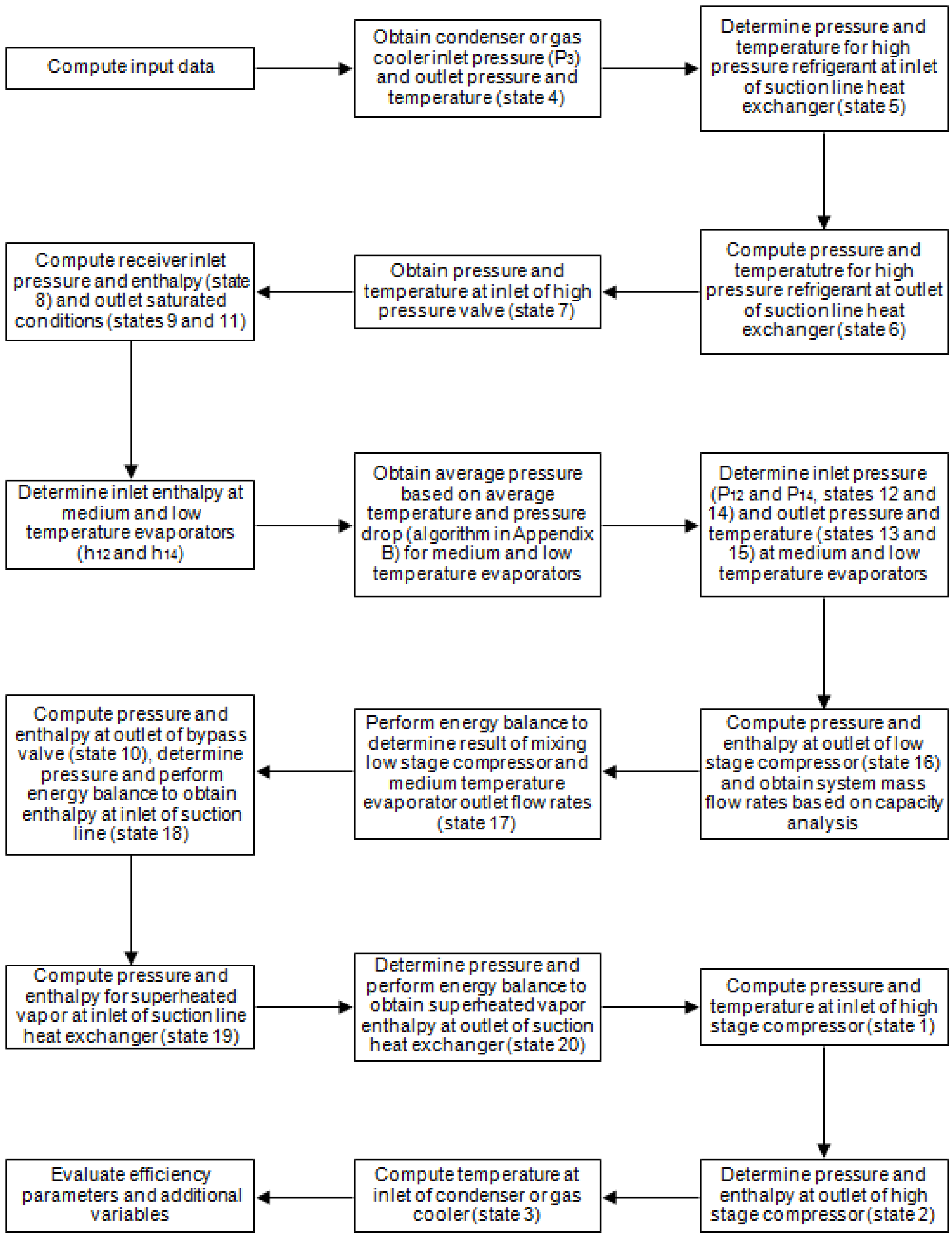


Figure 3.3: Computational sequence of the CO_2 booster model solution.

the routine represented in Figure 3.4, which requires repetition of the sequence in Figure 3.3 until convergence, is executed.

Similarly to the pumped CO_2 model solution, a Fortran code was developed to solve the set of equations presented, coupled with REFPROP 9.0 library for properties calculation. Once again, results presented are overall performance parameters associated to the refrigeration cycle and refrigerant states at inlet and outlet for each device.

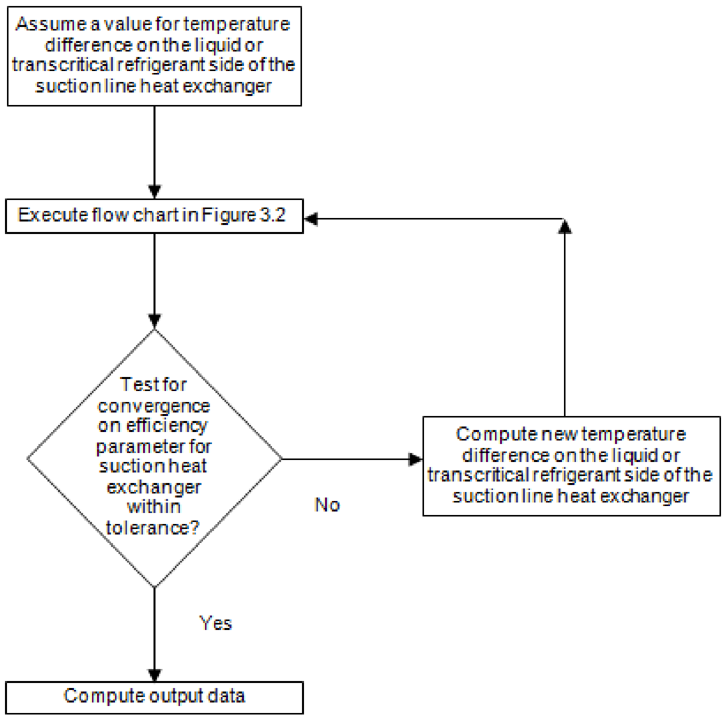


Figure 3.4: Iteration process related to the suction heat exchanger parameter.

3.5 Validation

Experimental data of the CO_2 booster refrigeration system matching the cycle configuration considered is still, to the author's knowledge, non-existent in the literature.

Notwithstanding, following the same methodology considered for the pumped CO_2 system, predictable trends associated with the CO_2 booster refrigeration technology are evaluated.

The coefficient of performance is, again, the performance indicator plotted against four different parametric variables: medium evaporating temperature, low evaporating temperature, gas cooler pressure and receiver pressure. For the CO_2 booster refrigeration system, application of the COP definition in Eq.(2.68) has the following expression as a result:

$$COP_{Booster} = \frac{\dot{Q}_{ev,mt} + \dot{Q}_{ev,lt}}{\dot{W}_{cp,hs} + \dot{W}_{cp,ls}} \tag{3.84}$$

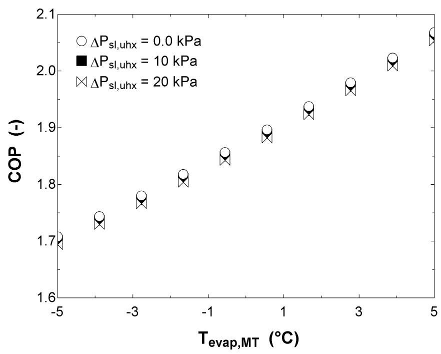
For the parametric analysis here described, the input data considered is presented in Table 3.1. Highlighted parameters are varied in the case their influence is to be studied. Figure 3.5 illustrates the parametric analysis.

As predicted, the trends follow what is expected of the model. The COP decreases as lower evaporating temperatures, Figures 3.5(a) and 3.5(b), are considered. Furthermore, there is a maximum COP when the gas cooler

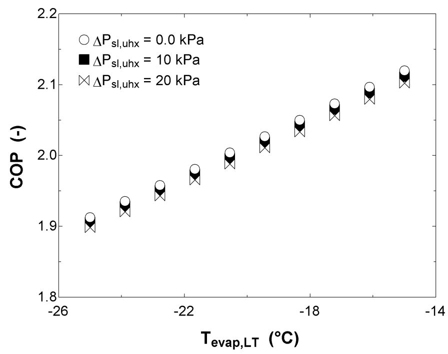
Table 3.1: Input data for CO_2 booster model goodness verification. Highlighted parameters are varied when their influence is studied.

Parameter	Value
Total cooling load	100 kW
Ratio of medium temperature cooling load over total cooling load	0.65
High stage compressor isentropic efficiency	0.65
High stage compressor volumetric efficiency	1.00
Low stage compressor isentropic efficiency	0.65
Low stage compressor volumetric efficiency	1.00
Temperature change in discharge line	0.5°C
Pressure drop in discharge line	1 kPa
Gas cooler pressure	8700 kPa
Gas cooler outlet temperature	34.9°C
Gas cooler pressure drop	5 kPa
Temperature change in liquid line upstream heat exchanger	0.5°C
Pressure drop in liquid line upstream heat exchanger	1 kPa
Temperature change in liquid line downstream heat exchanger	0.5°C
Pressure drop in liquid line downstream heat exchanger	1 kPa
Temperature rise in suction line downstream heat exchanger	0.5°C
Pressure drop in suction line downstream heat exchanger	1 kPa
Temperature rise in suction line upstream heat exchanger	0.5°C
Pressure drop in suction line upstream heat exchanger	1 kPa
Suction line heat exchanger outlet temperature for transcritical refrigerant	32.9°C
Suction line heat exchanger pressure drop for transcritical refrigerant	2.5 kPa
Suction line heat exchanger pressure drop for superheated vapor	2.5 kPa
Receiver pressure	5000 kPa
Medium evaporating temperature	1°C
Medium temperature evaporator outlet superheat	3°C
Medium temperature evaporator pressure drop	5 kPa
Low evaporating temperature	−25°C
Low temperature evaporator outlet superheat	3°C
Low temperature evaporator pressure drop	5 kPa

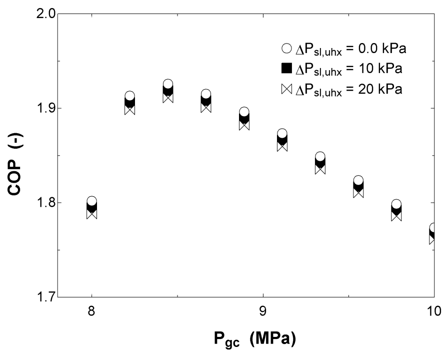
pressure is varied, Figure 3.5(c), as also reported by Ge and Tassou [69]. Danfoss [70] suggested that the performance of the system, as it would be expected in conventional two-stage cycles, increases with lower receiver pressures, which is verified in Figure 3.5(d). However, the receiver pressure always has to be set to a value larger than the medium evaporating pressure [70]. Finally, increasing pressure drop at the suction line reflects in lower performance, Figures 3.5(a) to 3.5(d).



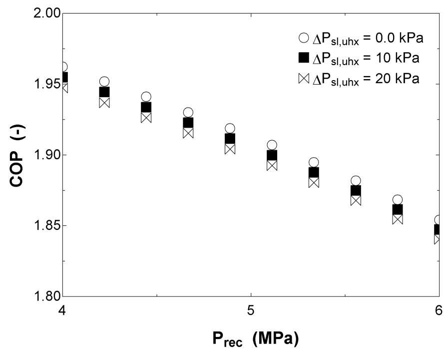
3.5(a): COP against average temperature in the medium temperature evaporator.



3.5(b): COP against average temperature in the low temperature evaporator.



3.5(c): COP against gas cooler pressure.



3.5(d): COP against receiver pressure.

Figure 3.5: Parametric analysis for the CO_2 booster refrigeration system.