# Annex I

In this section, the calculations needed to obtain the COP (Eq. (I. 1)) of each of the cycles, defined as objective function, are presented. The cooling capacity (Eq. (I. 2)) corresponds to the refrigerant mass flow rate multiplied by the enthalpy difference in the evaporator, where the inlet enthalpy () depends on the considered cycle. The net cycle electrical power consumption (Eq. (I. 3)) is the sum of the input to the compressor and to the eC device minus the electrical energy recovered by the expander.

|  |  |
| --- | --- |
|  | (I. 1) |
|  | (I. 2) |
|  | (I. 3) |

Three evaluated cycles have the following common thermodynamic states: the enthalpy at compressor suction (point 1 in Fig. 2) is evaluated by the evaporating pressure and with a fixed degree of superheat (); the discharge enthalpy (point 2 in Fig. 2) is evaluated by considering the overall effectiveness of the compressor and the isentropic discharge enthalpy using Eq. (I. 6) [1], where the overall effectiveness of the compressor (Eq. (I. 5)) was obtained by the same author from experimental data [2, 3]. Temperature at the exit of the gas-cooler is equal to the heat rejection temperature plus the approach temperature in the heat exchanger (Eq. (I. 7)) [4].

|  |  |
| --- | --- |
|  | (I. 4) |
|  | (I. 5) |
|  | (I. 6) |
|  | (I. 7) |

The electrical power input to the compressor is defined as.

|  |  |
| --- | --- |
|  | (I. 8) |

Each of the evaluated cycles shown in Fig. 2 have the following specific thermodynamic relations:

**- The baseline cycle (Fig. 2a).** Expansion processes are considered isenthalpic, therefore enthalpy at the inlet of the evaporator is equal to that at the exit of the gas-cooler (Eq. (I. 9)). Combining Eq. (I. 1), (I. 2) and (I. 8), the COP for the baseline cycle can be written by Eq. (I. 10).

|  |  |
| --- | --- |
|  | (I. 9) |
|  | (I. 10) |

**- Cycle with elastocaloric subcooler (Fig. 2b).** The eC subcooler cools down the CO2 at the exit of the gas-cooler. The subcooling degree is expressed by Eq. (I. 11) and the enthalpy at the exit of the subcooler (point 4b in Fig. 2) is evaluated using Eq. (I. 12), where is the specific heat extracted with the subcooling system. Accordingly, the cooling capacity of the cycle can be expressed by Eq. (I. 13).

|  |  |
| --- | --- |
|  | (I. 11) |
|  | (I. 12) |
|  | (I. 13) |

The net power input of this configuration is the sum of the compressor power consumption (Eq. (I. 8)) and the power input to the eC device (Eq. (I. 14)). The latter is calculated through the experimentally obtained COP values of the eC device (Table 1). Therefore, the COP of the cycle with eC subcooler can be expressed by Eq. (I. 15).

|  |  |
| --- | --- |
|  | (I. 14) |
|  | (I. 15) |

**- Cycle with expander at the 2nd expansion stage and elastocaloric subcooler (Fig. 2c).** Finally, this configuration uses both sub-systems: energy recovery in the 2nd expansion stage through a generic expander and electric generator that is used to drive the eC subcooler, which further cools down the refrigerant at the exit of the gas-cooler for a certain subcooling degree as shown by Eq. (I. 16). The enthalpy at the exit of the subcooler is quantified with Eq. (I. 17). Considering the expansion in the back-pressure as isenthalpic (), the enthalpy at the inlet of the evaporator and the cooling capacity can be calculated using Eq. (I. 18) and Eq. (I. 19), respectively.

|  |  |
| --- | --- |
|  | (I. 16) |
|  | (I. 17) |
|  | (I. 18) |
|  | (I. 19) |

This cycle assumes that the energy recovered by the expander after conversion to electric power (Eq.(I. 20)) is used to partially drive the eC device. Since the eC device requires larger input power to provide the desired subcooling power than provided by the expander, the net electrical input work to this cycle can be evaluated using Eq. (I. 21).

|  |  |
| --- | --- |
|  | (I. 20) |
|  | (I. 21) |

Combining Eq. (I. 17) to (I. 21), the COP of the cycle with expander at the 2nd expansion stage and elastocaloric subcooler can be expressed with Eq. (I. 22).

|  |  |
| --- | --- |
|  | (I. 22) |

[1] L. Nebot-Andrés, D. Calleja-Anta, D. Sánchez, R. Cabello, R. Llopis. Thermodynamic analysis of a CO2 refrigeration cycle with integrated mechanical subcooling. Energies. 13 (2019).

[2] L. Nebot-Andrés, D. Sánchez, D. Calleja-Anta, R. Cabello, R. Llopis, Experimental determination of the optimum working conditions of a commercial transcritical CO2 refrigeration plant with a R-152a dedicated mechanical subcooling, International Journal of Refrigeration, 121 (2021) 258-268.

[3] L. Nebot-Andrés, J. Catalán-Gil, D. Sánchez, D. Calleja-Anta, R. Cabello, R. Llopis, Experimental determination of the optimum working conditions of a transcritical CO2 refrigeration plant with integrated mechanical subcooling, International Journal of Refrigeration, 113 (2020) 266-275.

[4] N. Purohit, V. Sharma, S. Sawalha, B. Fricke, R. Llopis, M.S. Dasgupta, Integrated supermarket refrigeration for very high ambient temperature, Energy, 165 (2018) 572-590.