Investigating the effect of interstage pressure on cooling performance of a real-world CO$_2$ heat pump system

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Abstract. The study reveals the effect of interstage pressure with the consideration of gas cooler outlet temperature on the cooling performance of an operating CO$_2$ heat pump system currently in use at a dairy plant. Based on the numerical model and CO$_2$ properties, it was found that the maximum mass fraction of liquid allocated for cooling can be achieved in the middle of the operating range of interstage pressure when the enthalpy at the location of the gas cooler outlet is between 341 kJ/kg. K and 399 kJ/kg. K (that is, the gas cooler outlet temperature is between 37.34 °C and 43.02 °C with a fixed discharge pressure of 85 bar), however, the maximum liquid fraction can be achieved at the upper limit of the pressure when the enthalpy is less than 341 kJ/kg. K and can be achieved at the lower limit of the pressure when the enthalpy is higher than 399 kJ/kg. K, respectively. In the comparative study on fixed and variable interstage pressure, it was found that there is small difference in cooling COP when the gas cooler outlet temperature is lower than 38.5 °C, while the COPs with variable interstage pressures surpass those with fixed pressures when the gas cooler outlet temperatures are even higher.

Nomenclature

| Symbol | Description                          | Subscripts |
|--------|--------------------------------------|------------|
| $h$    | enthalpy, kJ/kg                      | min        |
| $m$    | mass flow rate, kg/s                | MT         |
| $P$    | pressure, bar                        | ODP        |
| $\Delta T$ | temperature variation, °C           | Subscripts |
| $T$    | temperature, °C                      | cool       |
| $V_s$  | compressor displacement rate, m$^3$/s| comp       |
| $W_{comp}$ | compression power consumption, kW    | dis        |
| $x$    | mass fraction of liquid              | ev         |
| $\eta$ | efficiency                           | inter      |

Greek symbols

- $\eta$ - efficiency
- $\eta_{inter}$ - interstage 
- $\eta_{isentr}$ - isentropic

Nomenclature

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- $x$ - mass fraction of liquid
- $\eta$ - efficiency

Subscripts

- min - minimum
- MT - Medium temperature
- ODP - Ozone depletion potential
- cool - cooling
- comp - compressor
- dis - discharge
- ev - evaporation
- inter - interstage
- isentr - isentropic

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1. Introduction

In the context of the Montreal Protocol, synthetic refrigerants are being phased out due to their high Global Warming Potential (GWP) and high Ozone Depletion Potential (ODP). CO$_2$ is considered as an appropriate natural refrigerant with ODP of zero and GWP of one. CO$_2$ is neither toxic nor flammable, compared with other natural refrigerants. For instance, ammonia is not recommended in small systems due to the toxicity and flammability of the substance, as well as its high capital and operational cost. Due to the numerous advantages of vapour-compression systems (e.g., cost-effectiveness, high-efficiency and energy-saving) [1-3], CO$_2$ heat pump systems have been investigated extensively and adopted either in residential, commercial, or industrial sectors.

CO$_2$ refrigerant has a low critical point of 31°C. Operating the high side at supercritical region will generate heat at up to 120°C efficiently. High heat transfer coefficient and specific heat of supercritical CO$_2$ enables effective heat transfer with water in a relatively small heat exchanger. On the cooling side, COP of 3 can always be expected. An adiabatically cooled gas cooler can enable subcritical operation in hotter ambient conditions. The transcritical CO$_2$ system currently used in industry is operated with the planning of flash gas parallel or bypass depending on the actual purposes, as the system cooling performance can be enhanced with a larger cooling capacity due to this arrangement [4]. The working principle of this design is to bypass the mass fraction of refrigerant vapour before entering to the evaporator where there is a mixture of vapour and liquid. Three main advantages have been proved experimentally and summarized as [5]: refrigerant distribution can be improved, pressure drop will be diminished, and heat transfer coefficient will be improved.

Based on the publication and current industry [6], even the theory of the flash gas bypass has been employed in most of existing installations, the effect of interstage pressure (or receiver pressure) with the consideration of the gas cooler outlet temperature on the cooling performance has not been fully considered. With a random gas cooler outlet temperature, there should be a corresponding interstage pressure providing the maximum mass fraction of liquid allocated for cooling. Additionally, there should be an allowable operating range of the receiver pressure (or interstage pressure) with various operating conditions for a real-world system.

In this study, a simplified configuration of a transcritical CO$_2$ heat pump system recently constructed is presented, with the addition of a thermodynamic cycle that is established depending on the real situations. The mathematical model for the cooling performance with flash gas bypass has been developed. The allowable range of the receiver pressure (or interstage pressure) is generated considering the industrial applicability of the system. Based on the model and CO$_2$ properties, the simulated liquid mass fraction allocated for cooling is studied with various interstage pressures and gas cooler outlet temperatures. Lastly, a desktop study has been conducted to analyze the cooling performance with fixed and variable interstage pressures.

2. System configuration and thermodynamic cycle

The simplified configuration of a real-world CO$_2$ heat pump system integrated with a thermal energy storage (TES) is shown in Figure 1, utilizing for simultaneous water heating and refrigeration (or cooling capacity storage). In this system, the heat rejection process is utilized for water heating while the evaporation process is used to refrigerate dynalene (which is a heat transfer fluid), and then the cold dynalene will freeze a phase change material (PCM) in a TES.
The working principle of the CO$_2$ system can be described with the help of a thermodynamics cycle (P-h cycle), as shown in Figure 2. The system operates with the arrangement of flash gas bypass, where two compressors (one main and one auxiliary compressors) work in a Medium-Temperature (MT) cycle, i.e., compressing CO$_2$ vapour to a supercritical state (state 2 and 3, respectively), and then mixing at state 4. During the heat rejection process from state 4 to 5 (between sCO$_2$ and water), low-temperature water is heated up through a plate heat exchanger, followed by a subsequent cooling process through ambient air till state 6. An expansion valve has been utilized to reduce the CO$_2$ pressure before going through the liquid receiver, which is from state 6 to 7. In the next, the two phases (vapour and liquid mixture) are divided physically and proportionally in a liquid receiver during the gas flashing process, which implies the movement from state 7 to state 8 and from state 7 to state 9, respectively. Afterwards, the saturated liquid will pass through another valve, and then to refrigerate dynalene from state 10 to state 11. At the meanwhile, the saturated vapour coming out of the liquid receiver becomes to be the mixture of liquid and vapour after a throttling process. The CO$_2$ vapour coming out of the evaporator blends with the mixture of liquid and vapour, followed by a moisture and dust filtering process through a suction line filter. Finally, the mixture of CO$_2$ is at a pure vapour state (state 1).

The cooling capacity of a real-world CO$_2$ heat pump system can be determined by the amount of energy required to cool the milk in the shortest possible time. The highest rate that milk can be cooled to a temperature is the rate at which it produces. In terms of heating capacity, up to 800 liters of hot water is required for cleaning purposes every day. Taking an average value of 15°C for the creek water, around 60.7 kWh of heating energy is required accumulatively every day to produce the required hot water. Consequently, the heat pump sized to deliver the cooling power can supply enough capacity for the heating requirement of the process. The produced hot water should be stored in an insulated container so that it is readily available when there is hot water demand from the process.

**Figure 1.** A simplified configuration of a real CO$_2$ heat pump.
3. Mathematical modelling

The numerical model was established considering the real design and operating conditions of a heat pump system currently operating at a dairy plant. Several main assumptions are considered, as the study focuses on the cooling performance with the flash gas bypass, namely:

1. There is an isenthalpic process going through an expansion valve.
2. A semi-reciprocating compressor operates in an adiabatic process.
3. The effect of pressure drop during the heat exchange process is negligible.
4. The power consumption of the gas-cooling fan is neglected.
5. There is no heat loss during the heat exchange process between CO$_2$ and dynalene.

3.1. Compressor model

The numerical model for a semi-reciprocating compressor was developed and validated [7], in which the mass flow rate of a semi-reciprocating compressor can be described as:

$$m_{\text{comp}} = \frac{\eta_v V_s}{v_{suc}}$$  \hspace{1cm} (1)

where $V_s$ is the Bitzer compressor displacement rate, depending on the actual size and type of the compressor, $\eta_v$ is the volumetric efficiency of a compressor, of which correlation on compression ratio is generated based on the experiment, and $v_{suc}$ is a specific volume for vapour at suction, considering the parameters of pressure ($P_{suc}$) and temperature ($T_{suc}$) at the suction state.

The energy power consumed for a single semi-reciprocating compressor can be worked out from [8]:

$$W_{\text{comp}} = \frac{m_{\text{comp}}(h_{\text{dis,isent}}-h_{\text{suc}})}{\eta_{\text{isent}}},$$  \hspace{1cm} (2)

where $h_{\text{dis,isent}}$ is a specific enthalpy at the discharge state through an isentropic process, $h_{\text{suc}}$ is the specific enthalpy of the vapour at suction, and $\eta_{\text{isent}}$ is the isentropic efficiency of the compressor.

The discharge temperature ($T_{\text{dis}}$) can be worked out when the parameters of discharge pressure and enthalpy have been known, of which equation can be generated with CO$_2$ properties [9], namely:

$$T_{\text{dis}} = f_1(h_{\text{suc}} + \frac{W_{\text{comp}}}{m_{\text{comp}}}, P_{\text{dis}})$$  \hspace{1cm} (3)
3.2. Liquid receiver and evaporation
During the gas flashing process, the liquid mass fraction ($x$) allocated for cooling via the liquid receiver will be:

$$x = \frac{h_9 - h_7}{h_9 - h_8}$$

(4)

where $h_7$ is the enthalpy for a liquid-vapour mixture, and $h_8$ and $h_9$ are the enthalpies at saturated states of liquid and vapour, respectively.

So, the enthalpies at states 8 and 9 ($h_8$ and $h_9$) with a specific interstage pressure ($P_{\text{inter}}$) can be determined from [10, 11]:

$$h_8 = f_2(P_{\text{inter}})_{\text{sat,liq}}$$

(5)

$$h_9 = f_3(P_{\text{inter}})_{\text{sat,vap}}$$

(6)

As there is no enthalpy change during the throttling process, the enthalpy at state 7 ($h_7$) is the same as the enthalpy at the exit of the gas cooler ($h_6$), namely:

$$h_7 = h_6 = f_4(T_6, P_{\text{dis}})_{\text{lip,vap}}$$

(7)

where $T_6$ is the gas cooler outlet temperature.

As the system is operated with flash gas bypass, two streams will be separated after the liquid receiver and being mixed at the suction state. The change of interstage pressure ($P_{\text{inter}}$) will affect the fraction amount of the liquid mass ($x$) and then will alter the temperature at the suction state ($T_{\text{suc}}$).

Based on the theory of the energy balance, the enthalpy after mixing of two streams ($h_1$) is generated as:

$$h_1 = h_{12} + x(h_{11} - h_{12})$$

(8)

In practice, there is an allowable operating range of interstage pressure when the system strictly operates as a transcritical cycle, i.e.:

$$P_{\text{inter}} \in [P_{\text{suc}} + 5 \text{ bar}, 60 \text{ bar}]$$

(9)

where the lower limit of the interstage pressure ($P_{\text{inter}}$) is set as default, which should be at least 5 bar higher than $P_{\text{suc}}$, in order to generate enough mass flow rate due to the minimum pressure difference between the suction pressure ($P_{\text{suc}}$) and interstage pressure ($P_{\text{inter}}$), and the upper limit of the interstage pressure is normally controlled around 60 bar for an industrial heat pump, due to the critical point of CO$_2$ and the industrial applicability of the liquid receiver.

Based on the specific operating conditions of the system, there can be an upper-pressure or lower-pressure limit to make the suction temperature always higher than or equals to the evaporation temperature, which is $T_{\text{suc}} = T_{\text{ev}}$ when a vapour is at the saturated state. It is used to ensure that CO$_2$ mixture is at a strictly vapour state before being suctioned to the compressor. The operating range for this system, therefore, can be described as:

$$P_{\text{inter}} \in [P_{\text{suc}} + 5 \text{ bar}, P_{(T_{\text{suc}}=T_{\text{ev}})_{\text{sat,vap}}}]$$

(10)

Only if

$$P_{(T_{\text{suc}}=T_{\text{ev}})_{\text{sat,vap}}} \leq 60 \text{ bar}$$

(11)

Or

$$P_{\text{inter}} \in [P_{(T_{\text{suc}}=T_{\text{ev}})_{\text{sat,vap}}}, 60 \text{ bar}]$$

(12)

Only if

$$P_{(T_{\text{suc}}=T_{\text{ev}})_{\text{sat,vap}}} \geq P_{\text{suc}} + 5 \text{ bar}$$

(13)
With a random gas cooler outlet temperature, there exists one interstage pressure which can be applied to achieve the maximum mass fraction of the liquid for cooling, which is:

\[ x_{\text{max}} = \max(x | P_{\text{inter}}) \]  

(14)

Then, the cooling COP can be determined considering other design conditions, e.g., suction pressure \((P_{\text{suc}})\) and evaporator superheat \((\Delta T_{\text{ev}})\), i.e.:

\[ \text{COP}_{\text{cool}} = \frac{m_{\text{ev}}(h_{11} - h_{10})}{\Sigma W_{\text{comp}}} \]  

(15)

4. Preliminary simulation results

Based on the numerical model and \(\text{CO}_2\) properties, the simulation results of the mass fraction of liquid allocated for cooling is shown in Figure 3. Each line represents one enthalpy at the gas cooler outlet (with the interval of 0.5 kJ/kg K), and the red points are the maximum mass fraction achieved at the various interstage pressure versus the enthalpy at the gas cooler outlet, regardless of the discharge pressure or the compressor performance. In the figure, “position A” represents when the enthalpy is around 341 kJ/kg K (e.g., 37.34 °C at the discharge pressure of 85 bar) and “position B” indicates when the enthalpy is 399 kJ/kg K (which can be 43.02 °C at the discharge pressure of 85 bar).

In conclusion, the maximum mass fraction can be achieved at the middle of the operating range of interstage pressure when the enthalpy at the gas cooler outlet ranged from 341 kJ/kg K to 399 kJ/kg K, while the maximum fraction can be achieved at the upper limit of the pressure when the enthalpy is less than 341 kJ/kg K and can be achieved at the lower limit of the pressure when the enthalpy is higher than 399 kJ/kg K, respectively.

5. Comparative study

The effect of the interstage pressure on the cooling performance can be studied with the consideration of real and actual conditions at a dairy plant, as described in Table 1.

There are two situations considered to compare the cooling performance, including 1) when the interstage pressure is fixed as 45 bar, and 2) when the interstage pressure is varied with the gas cooler outlet temperature, in order to always achieve the maximum mass fraction. Therefore, the simulated cooling COPs are achieved and described in Figure 4, in which there is small difference in cooling COP when the gas cooler outlet temperature is lower than 38.5 °C, while the cooling performance with
variable interstage pressure surpasses the performance with fixed pressure in the liquid receiver when the exit temperature of the gas cooler exceeds 38.5°C.

**Table 1.** Design conditions of a CO2 heat pump currently in use at a dairy plant.

| Parameter                              | Value               |
|----------------------------------------|---------------------|
| Evaporation temperature, $T_{ev}$      | -12.5 °C            |
| Evaporator superheat, $\Delta T_{ev}$  | 20 K                |
| Discharge pressure, $P_{dis}$          | 85 bar              |
| Gas cooler outlet temperature, $T_{gc}$| between 20°C to 40°C|
| Types and numbers of compressors       | 4HTC-20K-60Hz × 1   |
|                                        | 2MTE-5K-60Hz × 1    |

![Cooling COPs with variable or fixed interstage pressure.](image)

**Figure 4.** Cooling COPs with variable or fixed interstage pressure.

The other simulated results including compressor power consumption and interstage pressures are described in Figure 5. It can be noted that the calculated power of two compressors with the variable interstage pressure remains almost the same with various gas cooler outlet temperature, while the compressor power with fixed interstage pressure decreases all the way down when the exit temperature of the gas cooler is getting higher than 38.5 °C. In terms of variable interstage pressures, it remains the same at 60 bar from 25°C to 36.25°C and 29.65 bar from 43°C to 45°C, respectively.

![Interstage pressure and total power consumption of two compressors with variable or fixed interstage pressure.](image)

**Figure 5.** Interstage pressure and total power consumption of two compressors with variable or fixed interstage pressure.
6. Conclusions
Base on the numerical analysis, the following insights can be achieved and concluded from this study:

- The simplified configuration of a CO₂ heat pump recently operating at dairy plant is presented, with the addition of a thermodynamic cycle that is developed considering real situations.
- The mathematical model for the cooling performance with flash gas bypass has been developed. The allowable operating range of the receiver pressure (or interstage pressure) is generated considering the industrial applicability.
- Based on the numerical model and CO₂ properties, the simulated results of the maximum mass fraction of liquid allocated for cooling with various gas cooler outlet temperature has been revealed.
- Mathematical modelling has been conducted to investigate the cooling performance with fixed and variable interstage pressures. It was found that the cooling performance with variable interstage pressure surpasses the performance with fixed receiver pressure (or interstage pressure) under the condition that the exit temperature of gas cooler exceeds 38.5°C.

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