Transcritical-Transcritical CO_2 High-Temperature Heat Pump Cycle for Dairy Spray Dryers

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Keywords:
High-temperature heat pump (HTHP), transcritical, CO_2, R744, spray drying, two-stage cycle, cascade, internal heat exchanger, efficiency

Extended Abstract

Introduction:

An innovative high-temperature heat pump (HTHP) concept [1], [2] has been identified, enabling the production of high-temperature air across a range of process temperature levels. Compared to conventional multi-compression cycles, this cycle configuration allows for achieving an equivalent maximum supply temperature at lower operating pressures (15 MPa) and offers better temperature profile matching on the heat sink. This approach demonstrates potential for application in two-stage dairy spray dryers, providing drying air to the spray-dryer inlet and the fluidized bed.
The so-called transcritical-transcritical cascade CO_2 cycle [2] (Figure 1) can attain heat sink outlet temperatures exceeding 200 °C, surpassing conventional HTHP limits, and provides significant temperature glides (e.g., from 15 °C inlet air) that facilitate the simultaneous supply of process air at 102 °C and 200 °C in two gas coolers.

CO_2 is non-flammable, has zero ODP, and a low GWP. For large-scale applications such as integration into dairy spray dryers (e.g., 5–20 MW_{th}), oil-free centrifugal compressors are envisaged [4]. Transcritical CO_2 heat pumps can achieve higher supply temperatures than other refrigerants because CO2's thermal stability and supercritical heat transfer in the gas coolers enable higher temperatures. There are currently a limited number of CO_2 HTHPs. The current state-of-the-art for CO_2 HTHPs can achieve temperatures up to 150 °C and supply heating from 10 to 50 MW. Two pilot-scale projects have also been reported, targeting sink temperatures of up to 275 °C [3] and 400 °C [4], respectively. Both systems are at a TRL of approximately 4 to 6.

In the literature, Steinberg et al. (2024) [5] proposed using a CO_2 HTHP for producing 300 °C steam, which featured both sink preheating and preheating at the compressor suction inlet as performance-improving aspects of the cycle. In general, the system's COP will be higher if the compressor discharge pressure is not limited. Sarkar et al. (2007) [6] modelled a simple transcritical cycle for sink heating up to 200 °C and a maximum pressure of 20 MPa, only achieving a COP of 1.67. For similar target conditions, but with a maximum pressure of 30 MPa, Held et al. (2024) [7] achieved a COP of approximately 2.85. The cycle was similar in concept to that of Steinberg et al. (2024) but had a COP ratio of 1.05 compared to a transcritical cycle with an internal heat exchanger.

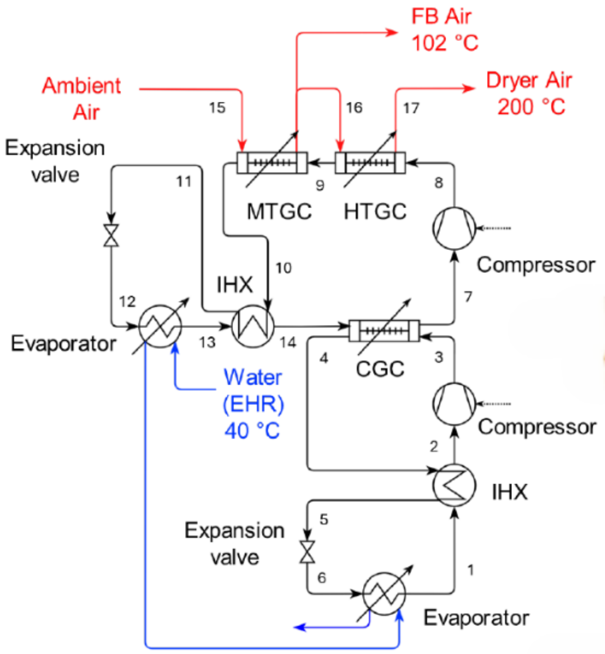

Figure 1: Transcritical-transcritical CO_2 heat pump cycle [2].

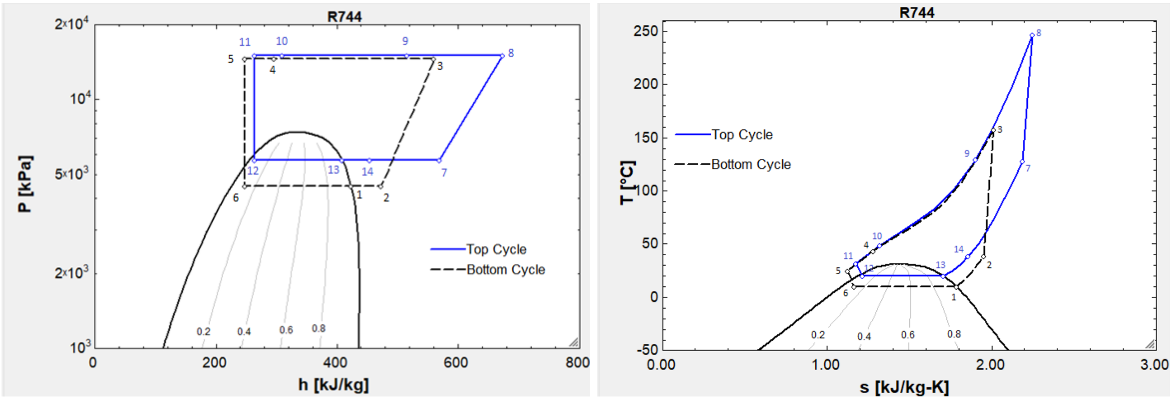

Figure 2: Log-p-h-diagram (left) and T-s diagram (right) of the transcritical-transcritical CO_2 heat pump cycle at reference point conditions.

The use of CO_2 as the working fluid ensures compliance with current low Global Warming Potential (GWP) refrigerant regulations and food applications [8]. When coupled with modern CO_2 compressor technology (e.g., oil-free piston or turbo compressors), the system achieves a COP of 2.55 (Figure 3) under the specified reference conditions (heat source water inlet/outlet: 40/20 °C, heat sink air inlet: 15 °C, heat sink air outlets 102 °C and 200 °C, heating capacities: 472.7 kW and 360.1 kW) which is 4 % higher than the COP obtained with the original optimized Excel model using CoolProp property data [9].

This heat pump cycle has potential applications for two-stage dairy spray dryers. Today, fossil-fuel or electric heaters are used to provide heat for such air-drying processes. Within the framework of decarbonization, renewable energy sources are increasingly deployed, including electrification-based technologies and biomass.

Previous studies [1], [2] modelled the feasibility of using the transcritical-transcritical cascade cycle for spray dryer air heating up to 200 °C. Sixteen applications, including coffee, starch, and paper drying, were also identified as suitable for the system to approximate the potential addressable heat

(>300 PJ/year) that could benefit from the development of a CO_2 HTHP. COPs ranging from 1.7 to 4.0 were estimated for the sixteen applications.

Compared to the previous studies [1], [2], this work extends the modelling by conducting a parameter study of the multi-temperature heat delivery configuration, varying the mass flow ratio, and demonstrates that there is no tight operating window for controlling the cycle.

Method:

This cycle was modelled using Engineering Equation Solver (EES) Academic Commercial Version 10 software and analysed through a comprehensive parametric study. A static model was created based on the following assumptions:

- The compressors have a constant isentropic efficiency
- A pinch model was used for all heat exchangers
- The pinch point in the mid-temperature gas cooler (MTGC) is smaller than that in the high-temperature gas cooler (HTGC)
- The MTGC pinch point is set to a constant, while the pinch point of the HTGC is calculated
- The refrigerant leaves the evaporator as a saturated vapour in both stages
- The low pressure is determined by the temperature in the evaporators
- The cascade gas cooler functions essentially as a superheater for the second stage.

Although the EES model allows the HTGC pinch point to be smaller than that of the MTGC, checks ensure it is never more than 3 K below the MTGC pinch point.

The reference values were obtained from the two publications on which the model is based, as well as directly from the authors of those papers [1], [2], [9]. The reference values are listed in Table 1. During the parametric studies, one variable was varied while the other variables were kept constant. Values that were varied have been marked as "Variable" in Table 1.

Table 1: EES-model parameters

Parameter	Reference value	Constant or variable
Refrigerant for both stages	R744	Constant
Heat source medium	Water	Constant
Heat sink medium	Air	Constant
Source medium temperature	**40 °C**	**Constant**
Ambient pressure	101.3 kPa	Constant
Temperature, air outlet 1	**102 °C**	**Constant**
Pinch Point, CGC	5 K	Variable
Pinch Point evaporator (both)	2 K	Variable
Pinch Point, MTGC	20 K	Variable
Pinch Point, IHX, first stage	5 K	Variable
Pinch Point, IHX, second stage	10 K	Variable
Isentropic efficiency, compressor	0.7	Variable
Heat sink capacity ratio	0.57	Variable
Refrigerant mass flow ratio	0.437	Variable
High pressure, first stage	14.62 MPa	Variable
High pressure, second stage	15 MPa	Variable
Total heating capacity of the heat sink	832.8 kW	Variable
Evaporation temperature, first stage	**10 °C**	**Variable**
Evaporation temperature, second stage	**20 °C**	**Variable**
Temperature, air inlet	**15 °C**	**Variable**
Temperature, air outlet 2	**200 °C**	**Variable**

Results:

Results demonstrate that the cycle exhibits strong resilience against fluctuations in supply air temperature, rendering the operation seasonally independent under moderate climatic conditions (air inlet temperature between 0 °C and 28 °C).

Figure 3 shows a plot of the COP as a function of the air inlet temperature. Compared to the reference case (15 °C, shown in blue, COP 2.55), the maximum deviation of the COP is 5 % at 28 °C, which is considered acceptable.

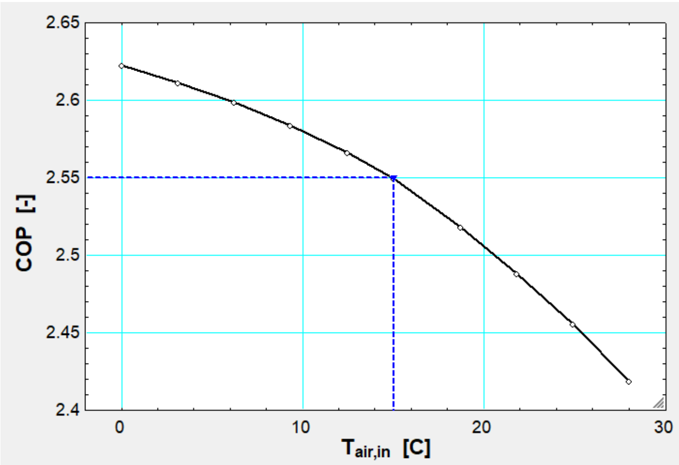

Figure 3: Plot of the COP of the transcritical-transcritical CO_2 heat pump cycle as a function of the air inlet temperature at the heat sink. The reference point (15 °C) is marked in blue.

Moreover, the theoretical analysis revealed that higher discharge pressures lead to an improved COP, although the lower cycle imposes an upper limit (Figure 4).

Note: To show a comparison, two separate simulations were plotted in Figure 4. For each simulation, one discharge pressure was kept constant at the reference value, while the other was varied.

Figure 4 shows that an optimum is found for the first stage (lower cycle) at 14.62 MPa, consistent with the value reported in the original study [1], [9]. For the second stage (upper cycle), the hardware limit (pressure) was reached before an optimum could be found. A hardware limit of 15 MPa was defined by the original study [2, p. 8]. Further research has found that commercially available oil-free CO_2 centrifugal compressors can indeed be operated at these pressures [10].

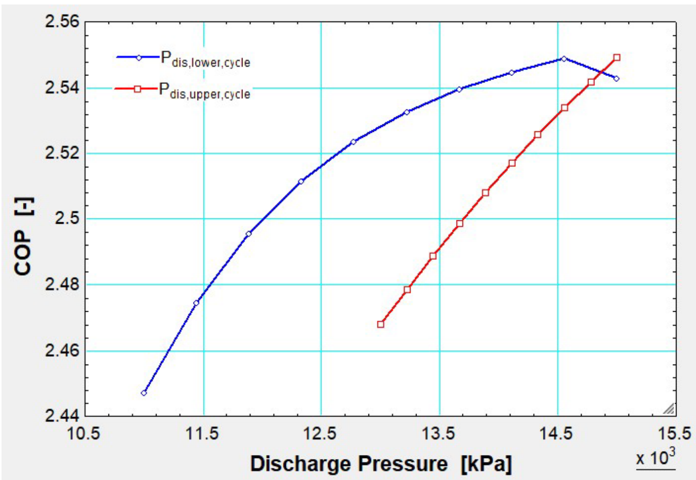

Figure 4: COP for different compressor discharge pressures (high-pressure) for the lower and upper cycles, or rather the first and second stages.

When solving the system of equations in EES, multiple solutions can be found for varying refrigerant mass flow ratios. This ratio is defined as $\phi_{\dot{m},ref} = \dot{m}_{ref,first\ stage}/\dot{m}_{ref,second\ stage}$. When the mass flow in the first stage is increased, the load on the compressor and, consequently, on the cascade gas cooler (CGC) increases. This results in the refrigerant being superheated to a higher temperature after leaving the CGC in the second stage, raising the hot-gas temperature. As a result, to maintain a constant heat sink capacity, the mass flow in the second stage can be reduced.

Figure 5 illustrates that for this model, the optimal COP was found at a mass flow ratio of 0.437. This is the same ratio identified by the original study [9]. The study was not performed at ratios below 0.2 because the HTGC pinch point is negative at these ratios.

Figure 5 also shows that the COP has a maximum deviation of 2% over the range 0.34 to 0.56. This is a positive outcome, as it indicates that although correctly matching the compressors is critical for process stability, some tolerance is available in real-world applications.

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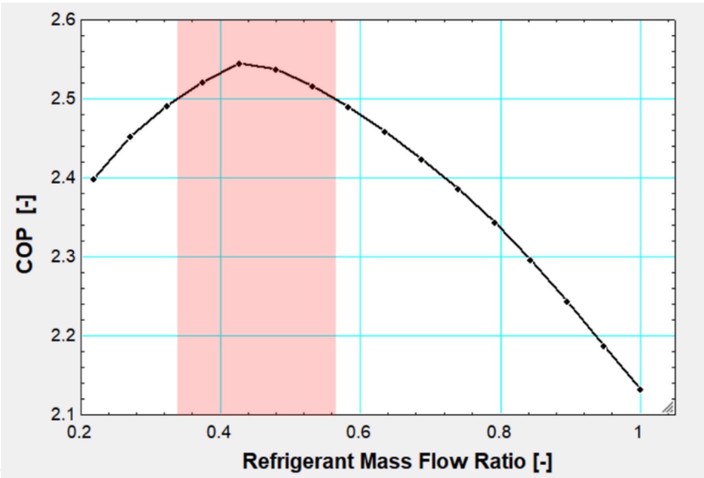


Figure 5: COP as a function of the refrigerant mass flow ratio between the upper and lower cycles.

Moreover, the effect of shifting the heating capacity ratio between the MTGC and the HTGC, while keeping the air inlet and outlet temperatures constant, on the COP and the HTGC pinch point was examined (Figure 6). This ratio ranges from 0 to 1. A value of 0 means all heating is supplied by the HTGC, while a value of 1 means all heating is supplied by the MTGC.

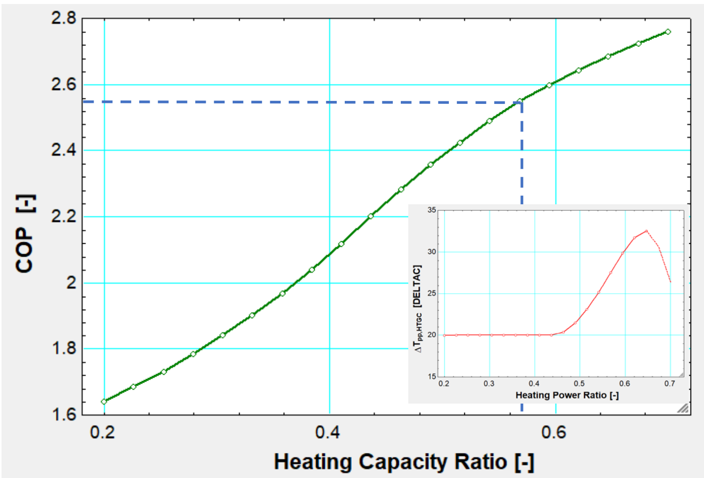
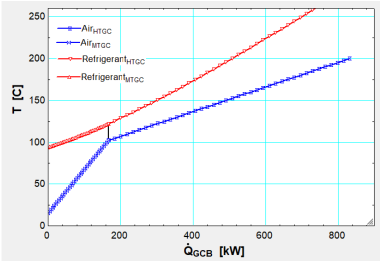
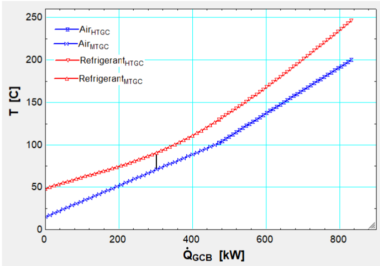


Figure 6: COP vs. heating capacity ratio between MTGC and HTGC.

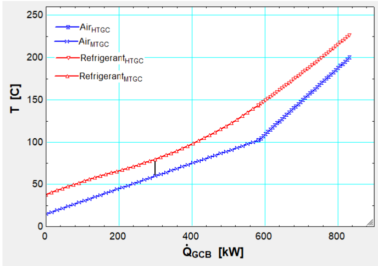
The purpose of having two gas coolers is to optimise the COP by achieving a better temperature-profile matching, i.e., by closely following the change in temperature along the refrigerant isobar with the change in temperature of the heated medium, in this case, air (Figure 7).



(a) Profile for a heating capacity ratio of 0.2.



(b) Profile for a heating capacity ratio of 0.57 (reference).



(c) Profile for a heating capacity ratio of 0.7.

Figure 7: Temperature profile plots in the MTGC and HTGC for different heating capacity ratios.

Because the inlet and outlet air temperatures remain constant, it is logical that the COP will decline rapidly if the heating capacity ratio is shifted too far towards either the HTGC or the MTGC, as this would significantly impair profile matching. Therefore, the analysis was performed only for the range 0.2 to 0.7. Figure 7 also shows that the COP can be improved slightly by increasing the ratio, but this comes at the expense of an ever-increasing pinch point in the HTGC, thereby impairing the efficiency of the heat transfer of this heat exchanger. At a ratio of 0.65, the model reaches its limits because the assumption that the smallest pinch point is in the MTGC no longer holds (see the subplot in Figure 6); it has shifted to the HTGC.

Discussion:

The motivation for this cycle is based on previous studies [1], [2], and the industry's need to electrify and decarbonize process heat. The cycle facilitates the production of two high-temperature air streams across a range of process temperature levels. Compared to conventional multi-compression cycles, this configuration allows for achieving an equivalent maximum supply temperature at lower operating pressures (15 MPa) and offers better temperature profile matching on the heat sink.

The EES model developed for this paper shows results similar to those of the Excel model created by Kong et al. (2025) [1], [9], with the only notable difference being a slightly higher COP (4 %). While the Excel model was designed to find optima, the EES model was created to analyse the cycle's behaviour when selected parameters were changed. This process is well-suited to dairy two-stage spray dryers with fluid beds (Figure 8).

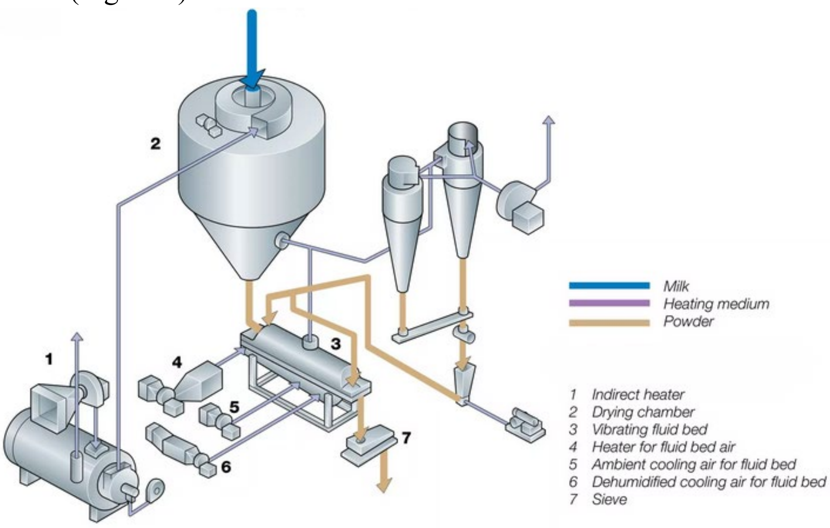


Figure 8: Two-stage spray dryer with fluid bed for the dairy industry [12].

The EES model could be further improved by implementing an option to shift the air volume ratio to compensate for variations in ambient air temperature and humidity. Additionally, a function might be integrated that allows the pinch point to switch between the two heat sinks. The effects of changing both outlet air temperatures could also be investigated.

Large spray dryers (e.g., 5 to 20 MW_{th}) typically operate for 6,000 hours a year [11] at constant process parameters, processing whole milk and skimmed milk, so some changes might be desired. Smaller plants that use batch processing would be more interested in being able to adjust outlet temperatures.

To this end, the EES model could be applied to other milk products at different air temperatures and airflow ratios, as well as other processes. Because transcritical heat pump cycles perform better with lower heat sink inlet temperatures, examining exhaust heat recovery is not anticipated. In fact, if an existing cycle that uses heat recovery were replaced by this transcritical-transcritical cycle, it would be advisable to use the recovered heat in other parts of the plant, such as for hot water. Research might also be conducted on different refrigerants or refrigerant mixtures.

Conclusions:

The proposed transcritical-transcritical CO_2 heat pump cycle can achieve very high supply temperatures exceeding 200 °C at reduced discharge pressures (below 150 bar), compared to conventional multi-compression cycles, and, by using two gas coolers for the heat sink, it offers a better temperature profile matching.

Careful selection of appropriate compressor types is required, for instance, oil-free centrifugal CO_2 compressors for large-scale applications such as dairy spray dryers. CO_2 has appropriate thermodynamic properties for these high supply temperatures.

For a stable heat pump cycle, the refrigerant mass flow ratio between the two compressors should remain stable as well. There is, however, some margin for manoeuvre. Having two gas coolers on the heat sink provides two air streams at different temperature levels simultaneously (as required in dairy spray dryers with fluidized beds). However, altering the heating capacity ratio between the two gas coolers significantly affects the pinch points of both heat exchangers.

The cycle is highly resistant to changes in supply air temperatures and is therefore season-independent in moderate climates.

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