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CO_2 transcritical plant with optimized dedicated mechanical subcooling

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ABSTRACT

In CO_2 refrigerating system, one of the most effective techniques for the improvement of energy efficiency is subcooling the fluid at the exit of the gas cooler. This can be achieved by taking advantage of cooling capacity supplied by HVAC chillers or dedicated mechanical refrigeration units. Furthermore, in the presence of subcooling the settings of the CO_2 system can be modified in order to further reduce the energy use of the plant. In this paper, the effect of Dedicated Mechanical Subcooling on the performance of a CO_2 booster system is investigated, and compared to a plant scheme where subcooling is performed via the HVAC, at two different climate conditions. The effect of optimisation of the gas cooler pressure is also investigated, and shows to be effective especially at hot climate conditions.

Keywords: Refrigeration, Carbon Dioxide, Subcooling, Energy Efficiency

1. INTRODUCTION

CO_2 direct expansion refrigerating systems are currently popular solutions for commercial refrigeration and the number of stores using CO_2 transcritical refrigeration technology has been increasing substantially all over the world, especially in Europe, where 14,000+ stores (Shecco, 2018) are equipped with this type of system. Subcooling of the fluid exiting the gas cooler is one of the most promising solutions to improve the plant performance. It can be performed by an internal heat exchanger, by coupling the refrigeration and the HVAC systems (Cortella et al. 2014a, D'Agaro et al., 2018), or by benefitting of cold water storage (Polzot et al., 2016). Dedicated Mechanical Subcooling (DMS), which consists in using an additional vapour compression cycle, sometimes with a secondary fluid, to provide subcooling, is being increasingly used.

Several studies are available in the literature on DMS systems, and Llopis et al. (2018) have recently presented a detailed review work on subcooling techniques. Mainly systems with single compression and one temperature level are considered for the evaluation of benefits due to subcooling, while up to now only very few systems with two evaporating temperature levels are investigated (Catalan-Gil et al., 2019).

In this paper, the performance of a CO_2 booster system with a R1234yf DMS is analysed, applied to a commercial refrigeration plant. The optimal operating conditions of the overall system are identified and the energy saving produced by the control of the gas cooler pressure, as a function of the subcooling degree in addition to the outdoor temperature, has been estimated. The performance of the CO_2 booster with DMS is compared to alternative plant schemes for the same reference supermarket, here included subcooling via the HVAC, underlining how the benefits change with climate. The comparison in terms of annual energy demand among the solutions allows the evaluation of the cost effectiveness of a dedicated subcooler or of an equivalent oversizing of the HVAC chiller.

2. BOOSTER SYSTEM WITH DEDICATED MECHANICAL SUBCOOLING

The commercial refrigeration system considered in this work is an existing monitored one in operation in a small supermarket, of approximately 1200 m^2, located in Modena (northern Italy). It is a transcritical CO_2 booster system with liquid receiver and flash gas expansion valve; the peak cooling capacity of display cabinets and cold rooms is equal to 45 kW for the Medium Temperature (MT) level and to 7 kW for the Low Temperature (LT). The actual plant allows subcooling at the exit of the gas cooler by the chilled water provided by the HVAC plant (Cortella et. al, 2020). In the present work, a Dedicated Mechanical Subcooler (DMS) through a single-stage cycle working with R1234yf as refrigerant has been considered for the same plant and included in a model, in order to exploit the energy improvements from the use of a dedicated subcooler, free from the penalties of HVAC coupling.

2.1. Thermodynamic cycle and model

In Fig. 1 the thermodynamic cycle of the transcritical basic CO_2 booster system (B) is compared to that of the booster with subcooling Δh_{sub} at the exit of the gas cooler (BDMS). Subcooling reduces the exit temperature at the gas cooler pressure and in turn the vapor quality in the receiver. Furthermore, suitable control rules of the high pressure p_{GC} in transcritical conditions can be adopted in order to maximize the COP of the overall system (CO_2 booster and subcooler) leading to the full exploitation of the subcooling by DMS. Typically, the optimal p_{GC} is reduced, as well as the specific work of high stage compressors, as it is qualitatively sketched in Fig. 1 for the optimized cycle OBDMS vs the BDMS one.

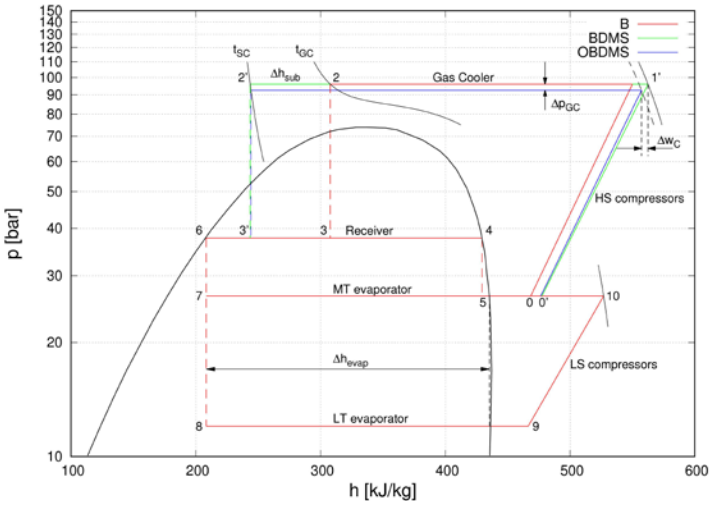

Figure 1: Thermodynamic cycle in a (*p-h*) chart of a transcritical CO_2 booster system in the configurations B: Booster with flash gas valve; BDMS: Booster with DMS; OBDMS: Optimised Booster with DMS.

Table 1. Main design parameters for the CO_2 Booster System and for DMS unit

CO_2 Booster System	
LT evaporating temperature	-35 °C
MT evaporating temperature	-10 °C
Minimum condensing temperature at subcritical conditions	6 °C
Liquid receiver pressure p_{INT}	35 bar
Subcooling at subcritical conditions	3 K
Gas cooler/Condenser approach temperature difference	4 K
LT superheating (up to actual suction temperature)	30 K
MT superheating (up to actual suction temperature)	20 K
Temperature set point at the exit of the subcooler	15 °C
LS compressors	Bitzer 2JSL-2K + Bitzer 2KSL-1K
HS compressors	Bitzer 4JTC-15K + Bitzer 4FTC-20K
DMS unit	
Evaporator approach temperature (minimum value)	5 K
Condenser approach temperature	10 K
Superheating	10 K
Compressor total efficiency	$\eta_{DMS} = -0,07\beta^2 + 0.4796\beta + 0.1234$

The CO_2 refrigeration unit has been modelled by in-house routines developed in the TRNSYS environment as described in detail in Polzot et al. (2016). The refrigerant properties are calculated by linking our in-house routines to the CoolProp libraries (Bell et al., 2014); the instantaneous mass flow rate is calculated from the cooling load estimated by time dependent models of the display cabinets and cold rooms; the compressors have been described using the manufacturer correlations. The detailed description of the refrigeration system, including information on the configuration of the LS and HS compressor racks and activation rules, is given in D'Agaro et al. (2019), where a thorough calibration and validation process of the model and control rules has been carried out against the yearly field data available from the real plant. The values of the main design parameters and settings are recalled in Table 1.

The thermodynamic cycle of the subcooler has been modelled according to the parameter values reported in Table 1 (DMS unit) and taking into account the compressor operating limits. Standalone simulations have been carried out in order to infer the COP_{DMS} as a continuous function of the outdoor temperature and evaporating level.

In the coupling to the main cycle, the DMS evaporating temperature depends on p_{GC}, the approach temperature at the subcooler and the DMS compressor operating limits. The set point temperature of the CO_2 at the exit of the subcooler is fixed at 15°C. Once the DMS size is chosen, a check is carried out to verify if the outdoor temperature and the available DMS cooling capacity allow to reach the set point value, otherwise the achievable CO_2 exit temperature is calculated.

2.2. Optimization

As stated above, for the full exploitation of subcooling the control rules on the high pressure p_{GC} in transcritical conditions can be adopted in order to maximize the COP of the overall system (CO_2 booster plus subcooler), which can be expressed as follows:

$$COP = \frac{\Delta h_{evap}}{\varphi w_{LS} + \frac{1}{\psi}\left(w_{HS} + \frac{\Delta h_{sub}}{COP_{DMS}} \right)} \qquad \text{Eq. (1)}$$

where Δh_{evap} is the enthalpy difference at the evaporator which, with good approximation, has the same value at the two evaporating levels (Fig.1), w_{LS} and w_{HS} are the specific compressor works of the two stages, Δh_{sub} is the subcooling degree, COP_{DMS} is the coefficient of performance of the DMS unit and the non-dimensional parameters φ and ψ are defined as:

$$\varphi = \frac{\dot{m}_{LT}}{\dot{m}_{LT} + \dot{m}_{MT}} \qquad \text{Eq. (2)}$$

$$\psi = 1 - x_{rec} \qquad \text{Eq. (3)}$$

where $\dot{m}_{LT}$ and $\dot{m}_{MT}$ is the refrigerant mass flow at the corresponding evaporating levels and ψ is the complementary to the vapour quality x_{rec} at the receiver inlet.

Once the plant operating conditions are defined (i.e. the parameters of Table 1, the compressors' efficiency and the φ value which, on yearly averaged basis, is set to 0.176), the COP defined in Eq. (1) essentially depends on three variables:

$$COP = f(p_{GC}, \Delta h_{sub}, t_{ext}) = f(p_{GC}, \Delta t_{sub}, t_{ext}) \qquad \text{Eq. (4)}$$

i.e. the outdoor temperature t_{ext}, which is the heat rejection temperature for both booster and DMS cycles, the gas cooler pressure p_{GC} and the subcooling degree Δt_{sub}.

Simulations have been carried out for outdoor temperature t_{ext} ranging from 26°C to 38°C with a 1 K step, in order to identify the couple of controllable variables $(p_{GC}, \Delta t_{sub})_{opt}$ that maximizes the COP in transcritical regime. The following ranges have been considered: p_{GC} between 75 bar and 110 bar with a 0.5 bar step; Δt_{sub}, with a 2.5 K step, between the lower value of 2.5 K and an upper value, which is the one necessary to approach the subcooler outlet set point temperature, consequently depending on the outdoor temperature.

As an example, Fig. 2 depicts the COP as a function of the gas cooler pressure p_{GC} for a given subcooling degree (10 K) for a set of outdoor temperatures. The COP shows a maximum in correspondence of the value of gas cooler pressure marked in empty dots. For the sake of comparison, the values of the pressure optimized for the basic transcritical booster cycle without DMS (Polzot et al., 2016) are marked in solid dots. The predicted COP increments (OBDMS vs BDMS) range from 0.40%, at 26°C of outdoor temperature, to 3.2% at 38°C. Fig. 2b shows the COP as a function of the subcooling degree for several values of the outdoor temperature. It can be observed that in this plant the optimal subcooling degree is obtained at the upper limit considered in the simulations, given by a subcooler outlet set point temperature of 15 °C.

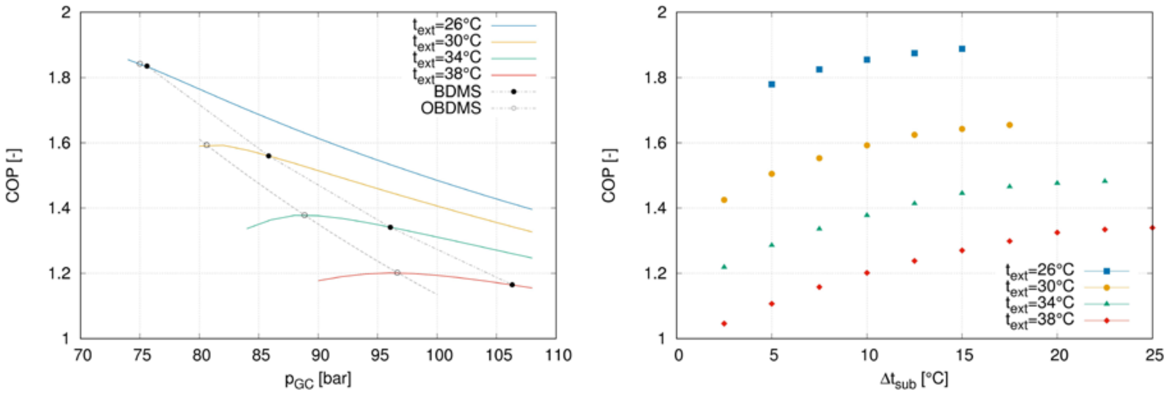

Figure 2: Overall plant COP vs. gas cooler pressure for Δt_{sub}=10 K (a) and vs. Δt_{sub} at the optimum gas cooler pressure (b) for a set of outdoor temperatures.

3. DMS PERFORMANCE COMPARISON AT MILD CLIMATE

Once the optimization of the heat rejection pressure has been performed, the comprehensive model has been used to carry out simulations for a whole year, with an hourly time step, to predict the annual electrical energy demand. The mild weather data typical of Modena (Italy) has been considered. For the sake of annual energy demand prediction, the booster model includes the electrical demand for auxiliaries, which has been calibrated against monitored data and accounts on average around 3.2 kW (D'Agaro et al., 2019). This leads to a reduction of 31.6% for the COP at optimal conditions $(p_{GC}, \Delta t_{sub})_{opt}$ with respect to the value of Eq. (1) at 26°C, and of 26.2% at 38°C. At first a threshold size of the DMS unit has been chosen and then the performance of the OBDMS solution has been compared to that of other system layouts operating under the same conditions of time dependent cooling load (from the simulation of display cabinets and cold rooms) and weather.

3.1 DMS unit size

The DMS unit size has been chosen on the basis of the annual energy saving achievable with respect to the total energy demand of the system without DMS. Simulations have been carried out for different values of the DMS maximum cooling capacity and the results reported in Fig. 3. Since no further significant gains are achieved by increasing the cooling capacity over 18 kW, this size is considered as a favourable choice.

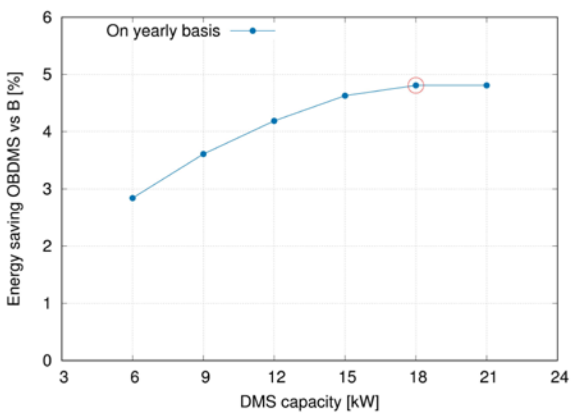

Figure 3: Annual energy saving of OBDMS vs. case B for a set of maximum cooling capacity of the DMS unit

3.2 Layout comparison

In order to assess the performance of the DMS solutions in comparison to alternative plant schemes, the following cases are considered:

- B (reference case): basic booster system with flash gas valve, whose main operating parameters are shown in Table 1;

- SC: sub-cooler heat exchanger applied to case B (Cortella et al., 2000). It is the actual plant scheme: the subcooling is provided by the chilled water from the HVAC plant. The control of the gas cooler pressure is the same as in case B;

- BDMS: Dedicated Mechanical Subcooling applied to B case. *Ad hoc* optimisation for the DMS solution is not implemented and the control of gas cooler pressure is the same as in case B;

- OBDMS: DMS with gas cooler pressure optimization to maximize the COP of the overall system.

The subcooler maximum cooling capacity is set to 18 kW, according to the sizing results from the previous section. For the sake of comparison, the same temperature set point, i.e. 15°C, is set at the exit of the subcooler and the same activation outdoor temperature, i.e. 19°C, have been chosen for all the cases.

The electrical energy use of the commercial refrigeration unit on yearly basis is reported in Table 2 (mild climate). The DMS solutions allows a reduction of the annual energy demand with respect to the transcritical booster system from 4.4% in the non optimized case (BDMS) to 4.9% in the optimized one (OBDMS). In the mild climate, the definition of a specific control rule for the gas cooler pressure gives a minor advantage (saving lower than 1% on annual basis with respect to BDMS). On the other hand, a dedicated subcooler with a variable evaporating temperature allows a global saving around 2% with respect to the subcooling by the HVAC chiller whose evaporating temperature is set at 2°C, since its COP is obviously higher. The saving in the energy use for subcooling is 26.5%.

4. DMS PERFORMANCE COMPARISON AT HOT CLIMATE

It is well known from the literature (Llopis et al., 2018) that the DMS solution is more effective at hot climates, i.e. at higher rejection temperature. A simulation has been carried out for the cases considered at the hot weather condition of Bangkok, where the outdoor temperature is above 19°C, thus the subcooler is active for 99.6% of the year, against 36% of Modena mild climate. The convenience threshold for DMS size is again 18 kW. In the hot climate, the optimized case (OBDMS) gives a much larger saving with respect both to the reference case B (+11.5%) and to the non-optimized BDMS (around 2%). Furthermore, the higher flexibility of the DMS solution versus the SC ones gives around 5% reduction of the annual energy demand.

Table 2. Annual Electrical Energy Demand including auxiliaries for the cases analysed and saving vs case B

Mild climate (Modena, northern Italy)				
Case	***Booster [MWh]***	***Subcooler [MWh]***	***Total [MWh]***	***Energy saving [%]***
B	128.7	0.0	128.7	0.0
SC	117.1	7.9	125.0	2.9
BDMS	117.1	5.9	123.0	4.4
OBDMS	116.6	5.8	122.4	4.9
Hot climate (Bangkok, Thailand)				
B	184.0	0.0	184.0	0.0
SC	147.9	23.3	171.2	7.0
BDMS	147.9	17.9	165.8	9.9
OBDMS	144.9	17.8	162.7	11.5

5. CONCLUSIONS

In an actual CO_2 booster plant, a subcooling scheme has been adopted taking advantage of some excess cooling power from the HVAC system, experiencing around 3% advantage in terms of energy use. In order to compare the profitability of such a scheme with a dedicated mechanical subcooling scheme, a comparison has been performed by simulations at mild and hot climate conditions, employing a R1234yf direct expansion unit as subcooler. The higher evaporating temperature in the case of the DMS unit allows to increase significantly (about 40 to 50 %) the energy saving compared to the HVAC chiller. A further increase (additional 10%) can be obtained by optimizing the gas cooler pressure in the presence of subcooling. As it was expected, this advantage is higher at hot climate conditions, where the subcooler operation is expected almost all year long. Such a comparison allows an economic analysis to find out the most profitable scheme in case both solutions are feasible.

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NOMENCLATURE

B	Basic Transcritical Booster System	$\dot{m}$	Refrigerant flow rate (kg/s)
BDMS	B system with DMS	OBDMS	Optimised BDMS
COP	Coefficient Of Performance	p_{GC}	Gas Cooler pressure (bar)
DMS	Dedicated Mechanical Subcooling	SC	Sub Cooler via HVAC
GC	Gas Cooler	t_{ext}	Outdoor temperature (°C)
HS	High pressure side	w	Specific compressor work (kJ/kg)
LS	Low pressure side	x_{rec}	Receiver inlet vapour quality (-)
LT	Low Temperature	Δh_{sub}	Subcooling degree, enthalpy (kJ/kg)
MT	Medium Temperature	Δt_{sub}	Subcooling degree, temperature (K)

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