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Testing of tri-partite CO_2 gas cooler prototype for domestic hot water and space heating

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ABSTRACT

The aim of this study was to test a tri-partite gas cooler for CO_2 heat pumps (5-13 kW). The proposed heat exchanger allowed the simultaneous and separate production of space heating and domestic hot water with an integrated design, simplifying the heat pump layout and piping requirements. The experimental campaign was dedicated to the performance of the gas cooler at various operation conditions. These operation modes are: i) only space heating, ii) only domestic hot water iii) simultaneous production of domestic hot water and space heating. The results of the test campaigns were analysed in order to determine potential improvements or required redesign of the heat exchanger. The heat exchanger solution was integrated to novel prototype of tri-generation CO_2 heat pump system.

Keywords: CO_2, domestic hot water, space heating, pressure drop, heat transfer

1. INTRODUCTION

R744 is used in both heating and refrigeration applications. It is an alternative to HFCs in several sectors, e.g. supermarket, transportation, domestic hot water (DHW) heat pumps and industrial processes (Gullo et al., 2018; Hafner, 2015). The efficient heating of water even up to temperatures of 90 ◦C is possible due to temperature glide in trans-critical region (Bamigbetan et al., 2017). The R744 systems prove the potential benefits of implementing integrated R744 in buildings with large DHW demands, such as residential buildings. The energy performances of R744 and R407A systems were comparable (Byrne et al., 2009). The recent research of Minetto et al. (2016) showed high efficient operation of R744 system during DHW production. At the same time, COP was significantly reduced during the space heating (SH) heating mode, due to high return temperatures from the heating system.

The solution, which is investigated in this study, allows maintaining high efficiency of the system even at high return temperatures by split of the CO_2 gas-cooler by three parts: pre-heating of DHW, SH and final heating of DHW. Thus, DHW and SH can be produced simultaneously. The first prototype of this configuration was previously described by Stene (2005). The system performed very well for supplying DWH and SH for residential building. The main aim of the current research was investigation of a new generation of a compact tri- partite gas-cooler, which consists of plate heat exchangers. The experiments were conducted at design and off-design condition.

2. METHODS

2.1. System description

The test facility for the evaluation of the tri-partite gas coolers was located at NTNU University (in Trondheim, Norway). The tri-partite gas cooler consists of three individual gas coolers AXP14 (ALFA LAVAL, Sweden) and hydraulic system, Figure 1. Alfa Laval AXP is a brazed plate heat exchanger, which was designed for high-pressure application. The domestic hot water (DHW) production was split into a preheating (HX3 – 14 plates) heat exchanger and a reheating (HX1- 34 plates) heat exchanger. Additional gas cooler (HX2-50 plates) was used for space heating. The counter flow of fluids arrangement was utilized for all the heat exchangers.

The tripartite gas cooler was connected to a test facility for multi-ejectors and was placed right after the compressors. The refrigerant loop of the multi-ejector test rig contains three compressors in parallel, three gas coolers in series, the multi ejectors, a liquid receiver tank, a suction accumulator, two evaporators and two internal heat exchangers. The secondary fluid of the evaporators and one of the gas coolers was glycol, which was installed as a closed loop with a tank. The secondary fluids of the other gas coolers were water and glycol, but they are connected to the house glycol/water supply system. The tripartite gas cooler was placed after the compressors and to be able to control the mass flow precisely, a bypass over the tripartite gas cooler loop was installed as well. Valves which are important for controlling the system are labelled, whereas the other non-labelled valves were required for measuring the pressure drop of the heat exchangers.

2.2. System control

Controlling CO_2 side inlet conditions. The mass flow of CO_2 was regulated by the bypass valve and V1. CO_2 discharge temperature: The inlet temperature was controlled by the set point of the evaporation temperature of heat pump. It depends on the set point of the pressure and the outlet conditions of the gas coolers after the tripartite gas coolers as well. The pressure was controlled by the expansion device which was controlled automatically.

Controlling DHW conditions. The mass flow was controlled, similar to the mass flow of CO2, by changing the two inlet valves V2 and V3. V4 influences the mass flow as well, but is mostly used to maintain a higher pressure than before the pump to be able to mix the two streams in order to control the temperature. The inlet temperature T5 is controlled by changing the opening degree of V3 and V2. V3 regulates the hot stream and V2 regulates the cold stream and by mixing the streams, the required inlet temperature can be reached.

Controlling SH conditions. The mass flow is regulated exactly the same way as DHW, the valve for the cold stream is V5 and for the hot stream is V6. V7 is used to maintain a high enough outlet pressure. The inlet temperature is regulated by changing V5 and V6 until the required inlet temperature is reached.

When performing tests and steady state conditions are reached, the operator has to change the measuring valves for the pressure loss to the required positions in order to measure the pressure loss over the right heat exchanger. It is mandatory to keep track of the specific time when which pressure loss is measured.

2.3. Experiment design

The operation of system was tested in three modes:

DHW - production of domestic hot water mode, there was no water circulation in HX2. The capacity of the system, Q_{DHW}, was 9.0 kW at the design point, when inlet DHW temperature T_{inDHW}=11.0 ºC and outlet DHW temperature T_{outDHW}=70.0 ºC. The design condition for the DHW operation is: P_{GC}=100 bar, T_{dis}, =94.0 °C, m_{DHW} =2.2 kg min^{-1}, m_{co2}=2.12 kg min^{-1}.

SH - space heating mode - there was no water circulation in the DHW circuit, and only HX2 was operative. The capacity of the system, Q_{SH}, was 8.0 kW. The design condition for the SH operation is: the gas cooler pressure P_{GC}=85 bar, discharge temperature, T_{dis}, =79.7 °C, CO_2 mass flow rate m_{CO2} =2.5 kg min^{-1}, SH mass flow rate m_{SH} =22.93 kg min^{-1}, and SH inlet temperature T_{in_SH}=30 ºC.

DHW+SP – space heating and domestic hot water production mode, all the gas-coolers were in operation. The capacity of the system was 4.0 kW SH and 6.0 kW DHW production. The design condition for DHW+SH operation is: P_{GC}=85 bar, T_{dis}, =79.7 °C, m_{SH} =11.46 kg min^{-1}, m_{DHW}=1.43 kg min^{-1}, T_{in_SH}=30 ºC, and T_{in_DHW}=10 ºC, m_{CO2} =2.4 kg min^{-1}.

The experimental tests for the performance evaluation of the tri-partite gas cooler have been evaluated under design and off-design conditions. The parameter values have been recorded as a set of steady data once all parameters attained steady state operating conditions. The variation of the influencing factors is introduced on Table 1.

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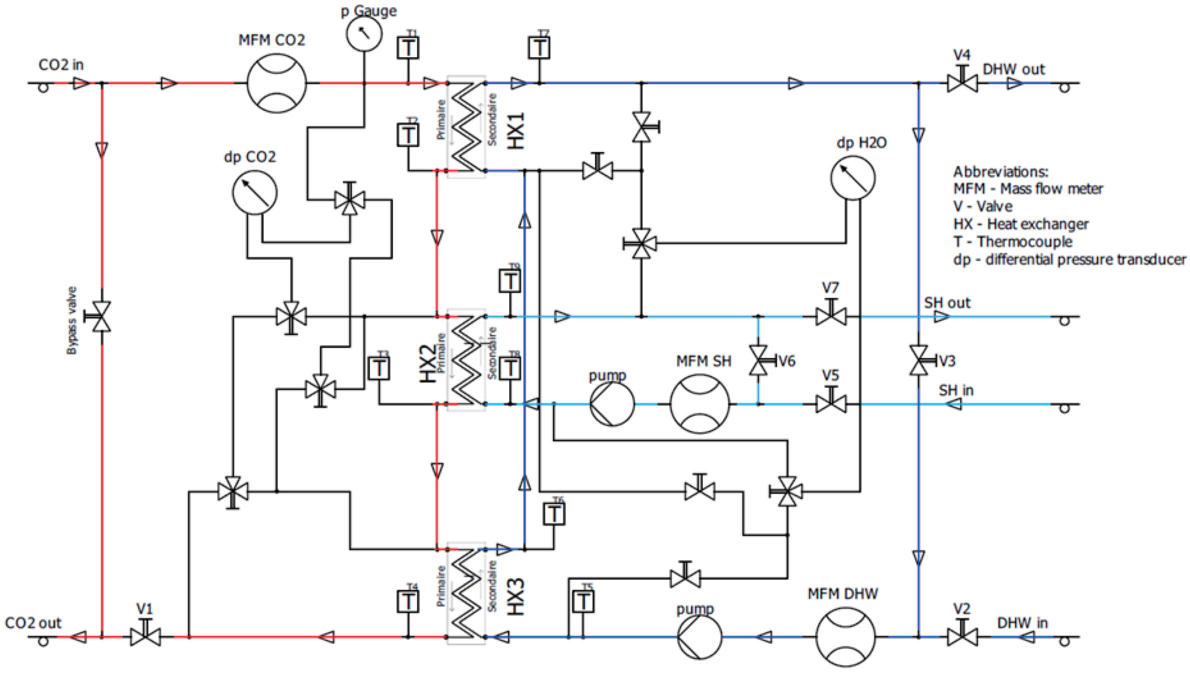

Figure 1. Experimental setup configuration

Table 1. Experimental tests of tri-partite gas cooler

Mode	Gas-cooler pressure, bar	CO_2 discharge T, ^oC	DHW inlet T, ^oC	SH inlet T, ^oC	CO_2 mass flow, kg min^{-1}	SH, kg min^{-1}	DHW, kg min^{-1}
DHW	90.0-100.0	91.0-101.0	10.5-19.7	-	1.8-2.2	-	1.4-2.3
SH	77.5-91.5	74.5-93.2	-	23.2-35.5	1.2-3.4	19.5-25.9	-
DHW+SH	79.3-89.3	68.1-89.0	11.2- 18.9	26.4-37.4	2.1-3.4	9.4-13.9	1.2-2.2

3. RESULTS AND DISCUSSION

3.1. Space heating operation

Two-dimensional contour plots of the HX2 performance in space heating mode is shown on Fig. 2. Each plot demonstrates the interaction effect of two parameters whilst other parameters remain were fixed. The required performance and temperature can be easily reached at nominal conditions.

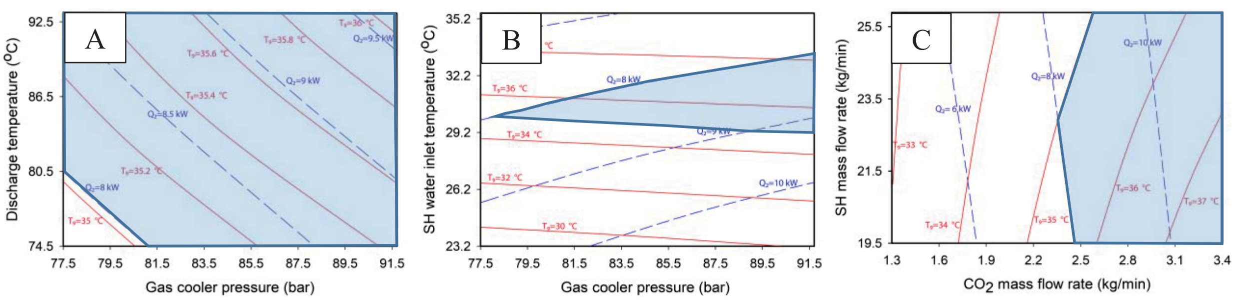

Figure 2. Contour plots of HX2 performance at different operation conditions in SH mode. Blue regions introduces the sufficient performance and outlet SH temperature. A - m_{CO2} =2.5 kg min^{-1}, m_{SH} =22.93 kg min^{-1}, T_{in_SH}=30 ^oC.; B- T_{dis}, =79.7 °C, m_{CO2} =2.5 kg min^{-1}, m_{SH} =22.93 kg min^{-1}; C- mCO_2 =2.5 kg min^{-1}, m_{SH} =22.93 kg min^{-1}, Tin_SH=30 ^oC.

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The gas cooler pressure over 77.5 bar and discharge temperature over 80.5 ºC allow to maintain the required SH return temperature and SH capacity. The subsequent increasing of the CO_2 pressure will increase the performance of the SH but influence badly on heat pump efficiency in general.

The CO_2 mass flow rate and SH water inlet temperature is the most influencing factors for correct maintenance of the required capacity of 8.0 kW and return temperature of 35.0 ºC. The SH inlet temperature should not drop below 29.2 ºC, when the mass flow rates are kept at nominal values. The CO_2 mass flow should maintain at 2.4 kg min^{-1} and higher, while SH water mass flow rate may vary in a high range without significant impact on the design performance of the gas cooler.

3.2. DHW operation

Two-dimensional contour plots of the HXs performance in DHW mode is shown on Fig. 3. The Fig. 3 illustrates the combined effect of two heat exchangers: HX1 and HX3 (pre-heater and re-heater respectively). The nominal gas-cooler pressure of 100 bar was found to be insufficient for maintenance of nominal heating capacity and DHW temperature of 70.0 ºC. The gas-cooler was very sensitive to change of inlet DHW temperature, when even a slight increasing of the DHW inlet temperature over the nominal values led to significant decreasing of the heating capacity or decreasing the DHW outlet temperature. The effective operation of a heat pump in DHW mode will require specific gas-cooler pressure strategy or increasing the mass flow through the HX1 and HX3 at the values similar to SH mode.

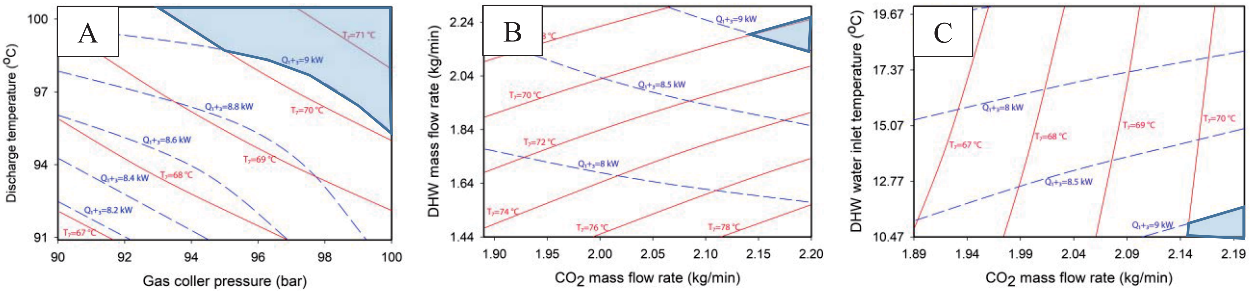

Figure 3. Contour plots of HX1+HX3 performance at different operation conditions in DHW mode. Blue regions introduce the sufficient performance and outlet DHW temperature. A - m_{CO2} =2.12 kg min^{-1}, m_{DHW} =2.2 kg min^{-1}, T_{in_DHW}=11 ºC.; B- T_{dis}, =94 °C, P_{GC} =100 bar, T_{in_DHW} =11 ºC.; C - T_{in_DHW} =11 ºC, P_{GC} =100 bar, T_{dis}, =94 °C.

3.3. DHW and SH operation

Two-dimensional contour plots of the HX2 performance in DHW+SH mode is shown on Fig. 4. The gas-cooler could not provide the required space heating capacities at nominal conditions (see. Fig.3B). The increasing of gas-cooler pressure over 87 bar and CO_2 discharge temperatures over 88.5 ºC or CO_2 mass flow over 2.5 kg min^{-1} can be an effective measure to avoid such mismatching.

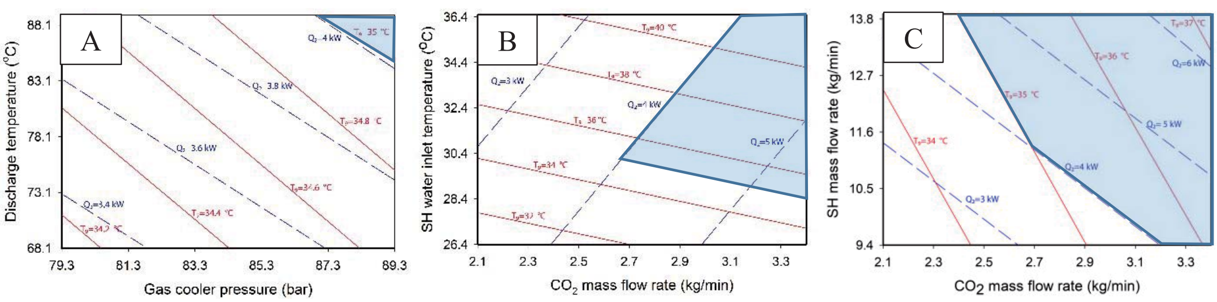

Figure 4. Contour plots of HX2 performance at different operation conditions in DHW+SH mode. Blue regions introduces the sufficient performance and outlet SH temperature. A - m_{CO2} =2.4 kg min^{-1}, m_{SH} =11.46 kg min^{-1}, T_{in_SH}=30 ºC m_{DHW}=1.43 kg min^{-1}; B- T_{dis}, =79.7 °C, m_{CO2} =2.4 kg min^{-1}, m_{SH} =11.46 kg min^{-1}, m_{DHW}=1.43 kg min^{-1}; C- m_{CO2} =2.4 kg min^{-1}, m_{SH} =11.46 kg min^{-1}, T_{in_SH}=30 ºC.

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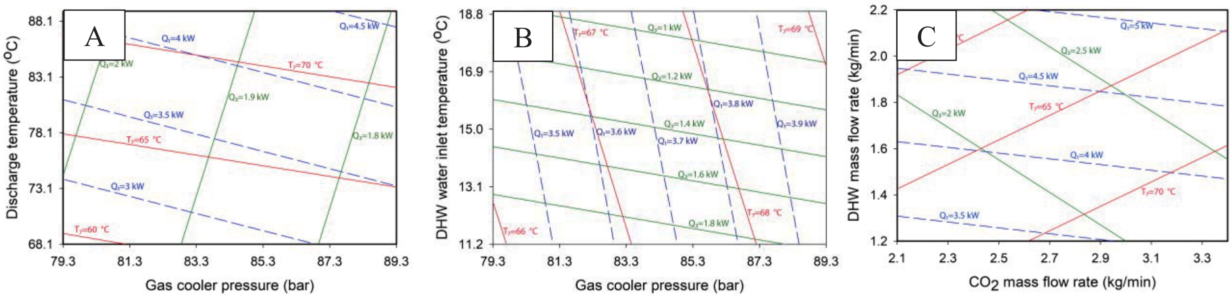

Figure 5. Contour plots of HX3 and HX1 performance at different operation conditions in DHW+SH mode. Blue regions introduces the sufficient performance and outlet DHW temperature. A - m_{CO2} =2.4 kg min^{-1}, m_{SH} =11.46, m_{DHW} =1.43 kg min^{-1}, T_{in_SH}=30 ºC, T_{in_DHW}=11 ºC; B- T_{dis}, =79.7 °C, m_{CO2} =2.4 kg min^{-1}, m_{SH} =11.46 kg min^{-1}, m_{DHW}=1.43 kg min^{-1}, T_{in_SH}=30 ºC.; C- $P_{GC=}$ 85 bar, T_{in_DHW}=11 ºC, m_{SH} =11.46 kg min^{-1}, T_{in_SH}=30 ºC.

The impact of nominal and off-design condition of the DHW heating and outlet temperature is more complicated as soon as the two heat exchangers HX1 and HX3 are split by HX2, Fig.5. The decreasing of gas-cooler pressure increases the performance of HX3 due to the lower load on HX2 and HX1 respectively (Fig. 4A and 5A). Overall performance of HX1 and HX3 is introduced on Fig. 6.

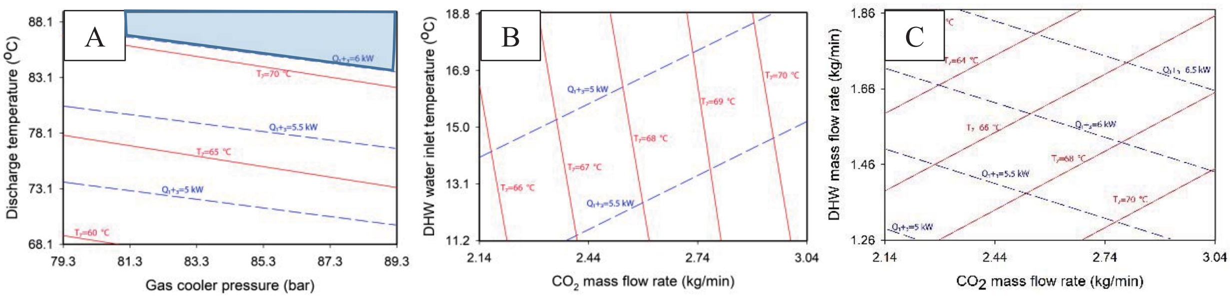

Figure 6. Contour plots of combined HX3 and HX1 performance at different operation conditions in DHW+SH mode. Blue regions introduces the sufficient performance and outlet DHW temperature. A - m_{CO2} =2.4 kg min^{-1}, m_{SH} =11.46, m_{DHW} =1.43 kg min^{-1}, T_{in_SH}=30 ºC, T_{in_DHW}=11 ºC; B- $P_{GC=}$ 85 bar, T_{dis}, =79.7 °C, m_{SH} =11.46 kg min^{-1}, m_{DHW}=1.43 kg min^{-1}, T_{in_SH}=30 ºC.; C- $P_{GC=}$ 85 bar, T_{in_DHW}=11 ºC, m_{SH} =11.46 kg min^{-1}, T_{in_SH}=30 ºC.

The required performance of 6 kW can be achieved only at high discharge temperatures and gas-cooler pressure (Fig.6A). The increasing of CO_2 mass flow over 2.5 kg min^{-1} , which is sufficient to provide the required space heating, makes positive effect but insufficient to rich DHW temperature of 70.0 ºC. The gas-cooler pressure regulation and CO_2 mass flow rate strategy is of a high importance for effective operation of such type of equipment. The same conclusion was obtained for the industrial CO_2 gas-cooler for DHW production (Berntsen et al., 2014).

3.4. Pressure drop at CO_2 side

The increasing of the CO_2 mass flow rate will influence the heat transfer and pressure drop in the tri-partite gas-cooler. The compact design and high mass flow can lead to high losses during the operation. The gas-cooler pressure drop in all the segments of the tri-partite gas-cooler did not exceed 0.35 bar. The maximum pressure drop was measured for HX3, which consists of 14 plates. This HX showed gradually increasing of pressure with respect to bulk temperature for DHW and DHW+SH operation modes. The HX1 and HX2 did not show strong dependence between the pressure drop and bulk temperature, while the strong correlation between mass flow rate and pressure drop was detected.

Table 2. Pressure drop at CO_2 in Tri-partite gas-cooler

Heat exchanger	Pressure drop, bar		
	DHW mode	SH mode	DHW+SH
HX1, re-heating DHW	0.08±0.04	NA	0.1±0.02
HX2, SH	0.03±0.01	0.03±0.001	0.03±0.001

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HX3, pre-heating DHW	0.05..0.35	NA	0.05..0.16

4. CONCLUSIONS

This study evaluates experimentally the performance of a trans-critical CO_2 tri-partite gas cooler (heat pump water heater) for space heating and domestic hot water production. The CO_2 tri-partite gas cooler was tested in three different modes: domestic hot water production, space heating operation and simultaneous space heating and domestic hot water operation. The effects of each input parameters have been illustrated on the 2D contour plots, presenting the pairwise comparisons between the operating parameters. From the outcome of this research, the following conclusions are drawn:

- the current design of the tri-partite gas-cooled showed high flexibility in SH mode, when the sufficient amount space heating can be produced both at nominal points and off-design operation.

- the production of DHW of 70.0 ºC is possible at discharge temperature higher than 94.0 ºC. The impact of gas-cooled pressure and discharge temperature is even higher when compared with the increasing of the CO_2 mass flow rate.

- the DHW+SH mode requires gentle regulation of the discharge pressure (CO_2 discharge temperature). The design capacity cannot be achieved at the nominal operation.

- pressure drop in the heat exchangers is relatively small for the compact system, which gives opportunity to manipulate with mass flow rates of CO_2 when achieving the desired performance and outlet temperatures.

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