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# Experimental analysis of a small-scale dynamic vapor injection $CO_2$ transcritical system operating under extreme conditions

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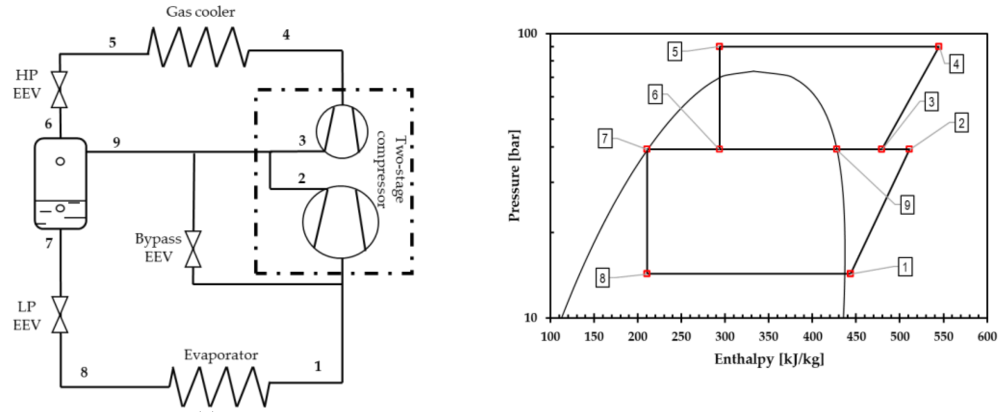
## 1. Introduction

Currently, negotiations are underway for the review of the EU F-gas Regulation to update the Regulation (EU) No. 517/2014 [1]. The transition to natural refrigerants is a reality, although no natural refrigerant is ideal, as both $CO_2$, R290, and R717 have their advantages and disadvantages. Due to the flammability of propane and the toxicity of ammonia, its use in split systems is a solution that requires additional efforts to achieve safety standards. Additionally, propane has charge limits as it is classified as an A3 refrigerant. This limitation is critical in split systems with direct expansion, as the charge needs to be increased due to the liquid lines to feed the evaporator, whose length will depend on the installation requirements. As a refrigerant classified as A1, $CO_2$ has an advantage in split systems, as it allows for a higher amount of charge. The efficiency is poor if the $CO_2$ base cycle is used due to its low critical temperature of 31°C, forcing the systems to operate under transcritical conditions with consequent high pressures and exergy losses. Therefore, $CO_2$ requires advanced cycles, such as the one presented. As a consequence, professionals and researchers in this field have gathered to find technical solutions to improve the efficiency of $CO_2$ systems.
Regarding small-scale applications, among the $CO_2$ cycles the two-stage compression system with two-phase separation in a flash tank and vapor injection (FTVI) is gaining attention [2]. Furthermore, the emergence of two-stage compressor technology has improved the simplicity and competitiveness of FTVI systems by eliminating one of the two compressors [3]. In early research, Cho et al. [4] carried out an experimental test of a two-stage transcritical $CO_2$ system with a flash tank and IHX establishing that the cooling COP of this system with gas injection was maximally enhanced by 16.5%. It was also shown that optimum control of the two EEVs was necessary to achieve high performance and system reliability. Chung et al. [5] experimentally optimized a $CO_2$ heat pump with gas injection in heating and cooling mode using a dual rotary compressor with a volumetric ratio of 0.7. In recent works (Citarella et al. [6]), an experimental test of a MT small two-stage $CO_2$ condensing unit with intercooling using ambient temperature and also gas injection is studied. The FTVI system with intercooling was found to meet the Ecodesign requirements for MT applications.
As highlighted by the literature surveyed, there is a growing interest in the use of CO2 in two-stage transcritical systems in heating, cooling, and refrigeration applications. Several researchers agree with the greater potential of flash tank gas injection in the two-stage compression configuration. Although there are some studies for the FTVI configuration with $CO_2$, to the best knowledge of the authors, the information is limited when it comes to experimental and validated data at low evaporation temperatures (-20°C, -40°C) and heat rejection conditions typical of warm climates (30°C, 40°C). The present work contributes to filling this knowledge gap by experimentally testing a commercial FTVI refrigeration prototype with an optimal gas cooler pressure control under extreme conditions.
The document is carried out as follows: first, the FTVI system is described. Subsequently, a thermodynamic model of the cycle is developed to establish the optimized design and operating parameters of the system. Then, the system will be manufactured according to the values obtained in the parametric analysis and experimentally tested, with special attention to the control of the gas cooler pressure. Finally, the experimental operation of the system is presented, along with the accuracy of the implemented control.

## 2. System description

The FTVI system, along with its p-h diagram, is illustrated in Figure 1. The system consists of a condenser, an evaporator, a two-stage compressor, a flash tank, and three expansion valves. The LP valve is responsible for expanding the refrigerant to the evaporating pressure and controlling the superheating in the evaporator. The HP valve manages the gas cooler pressure on its degree of opening, in addition to expanding the refrigerant to the intermediate pressure. The bypass valve relieves the pressure in the flash tank when it rises to values that compromise the safety of the vessel. The gas generated in the flash tank is drawn into the second stage of the compressor, which operates similarly to a parallel compressor.

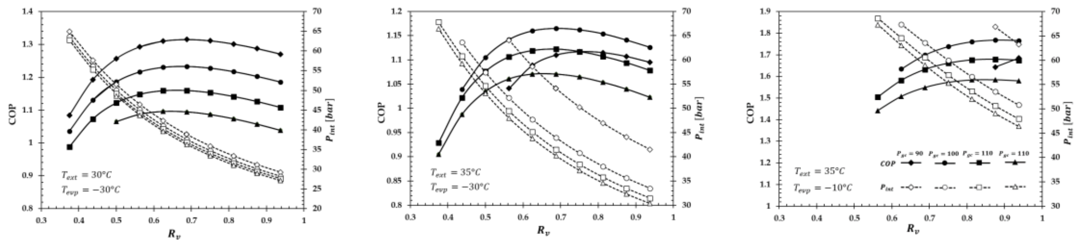



(a) (b)

Fig. 1. Schematic (a) and $p - h$ (b) diagrams of the system

## 3. Thermodynamic analysis and optimization

Based on the laws of conservation of mass and energy, a thermodynamic model of the FTVI system has been established using the Engineering Equation Solver (EES) software [7]. The superheat of the evaporator and the approach between the outside temperature and the gas cooler outlet temperature have been set at 6K and 3K, respectively. The flash tank has been modeled as an ideal two-phase separator with 100% efficiency. Volumetric efficiencies have been assumed constant with a value of 0.85 for both stages. The isentropic efficiencies of the first and second stages have been calculated using polynomials applied to subcritical and transcritical $CO_2$ compressors proposed by Gullo et al. [8]. First, a parametric analysis of the volumetric ratio is carried out by varying the displacement of the second stage, keeping the swept volume of the first stage constant. The volumetric displacement ratio $R_v$, defined as the ratio between the swept volume of the second stage and the swept volume of the first stage, is a valuable parameter to evaluate how the geometrical characteristic of the compressor affects system performance. A parametric analysis of the volumetric ratio is carried out by varying the displacement of the second stage, keeping the swept volume of the first stage constant.




(a) (b) (c)

Fig. 3. Variations of COP and intermediate pressure with the volumetric ratio for different operational conditions

Fig. 2 shows the effect of varying the volumetric ratio on the intermediate pressure and cycle performance for different operating conditions. Regardless of the operating conditions ($T_{ext}, T_{evp}, P_{gc}$), increasing the $R_v$ results in a decrease in the intermediate pressure. This is due to the larger volume available in the second stage suction to manage the gas generated in the flash tank. This decrease in pressure in the intermediate stage has two opposing effects on the cycle performance: (i) an increase in the refrigeration capacity and (ii) an increase in the vapor fraction in the flash tank, resulting in a higher mass flow rate and higher power consumption in the second stage of the compressor. Therefore, for a given set of operating conditions, there exists an optimal compressor geometry linked to the intermediate pressure, as observed in Fig. 2.

The optimal volumetric ratio tends to increase with higher evaporating temperatures, as illustrated in Figs. 2b and 2c. Higher evaporating temperatures lead to a reduced volume available for suction in the second stage. This decrease in the intermediate suction capacity results in an increase in intermediate pressure, which ultimately penalizes the COP of the cycle. The optimal $R_v$ is not significantly affected by changes in outside temperature, as observed in Figures 2a and 2b, where the optimal volumetric ratio remains relatively constant and approximately equal to 0.7. The increase in outside temperature leads to an expansion with a higher vapor fraction in the flash tank, resulting in an increase in intermediate pressure and a subsequent increase in the optimal volumetric displacement ratio. However, by raising the gas cooler pressure, the expansion can be shifted to lower vapor fractions, achieving better performance keeping constant $R_v$ but with a higher gas cooler pressure.

Considering the LT application of the designed system and the information brought by Fig. 2, a two-stage rolling piston compressor with a volumetric ratio of 0.7 has been selected. The correlations for the isentropic and volumetric efficiencies of the compressor have been introduced into the thermodynamic model. These correlations are calculated on the experimental results provided by the manufacturer.

Regarding the pressure of the gas cooler, several authors agree on the key parameters that influence the optimal value, namely the outside temperature (through the direct relationship with the maximum possible cooling of $CO_2$ in the gas cooler), the evaporating conditions and the compressor characteristics [9], [10]. Thus, a correlation of the optimal heat rejection pressure as a function of the outside temperature and evaporating pressure has been proposed for the FTVI system. The optimization simulations are carried out with the thermodynamic model in EES applying the least squares type criteria. The correlation polynomial and the coefficients are shown in Eq. (1) and Table 1, respectively, with an $R^2$ of 99.94 %. It should be noted that the compressor efficiencies do not appear in Eq. (1) as they are implicitly in the thermodynamic model used to establish the correlation.

$$P_{gc}^{optimal} = a_0 + a_1 \cdot T_{ext} + a_2 \cdot T_{ext}^2 + a_3 \cdot P_{evp} + a_4 \cdot P_{evp}^2 + a_5 \cdot T_{amb} \cdot P_{evp} \tag{1}$$

Table 1. Coefficients and accuracy of the proposed optimal gas cooler correlation

| $a_0$ | $a_1$ | $a_2$ | $a_3$ | $a_4$ | $a_5$ | $R^2$ |
|---|---|---|---|---|---|---|
| -1.2011E+03 | 7.0810E+00 | -9.3737E-03 | -1.1408E+01 | 8.8572E-03 | 3.6583E-02 | 99.94 % |

## 4. Results and Conclusions

A prototype was manufactured based on the schematic shown in Figure 1, including the double-stage compressor selected based on the findings of the previous section. The flash tank is a separator vessel equipped with a safety valve set at 60 bar. The three expansion valves are governed by their own controller. The proposed correlation for optimizing the gas cooler pressure has been implemented in the equipment control system, serving as a reference value for the PID control that governs the HP valve. The variable speed drive allows the compressor to operate at a maximum rotational speed of 80 rps (160 Hz) and a minimum rotational speed of 40 rps (80 Hz).

An experimental test has been conducted, varying different operating conditions to verify the reliability and accuracy of the implemented control over the gas cooler pressure set point. The compressor speed has been varied between 50 and 80 rps, the outside temperature has been varied between 30°C and 40°C, and the evaporating temperature has stabilized at about -25°C, -30°C, and -40°C, as shown in Figure 3b.

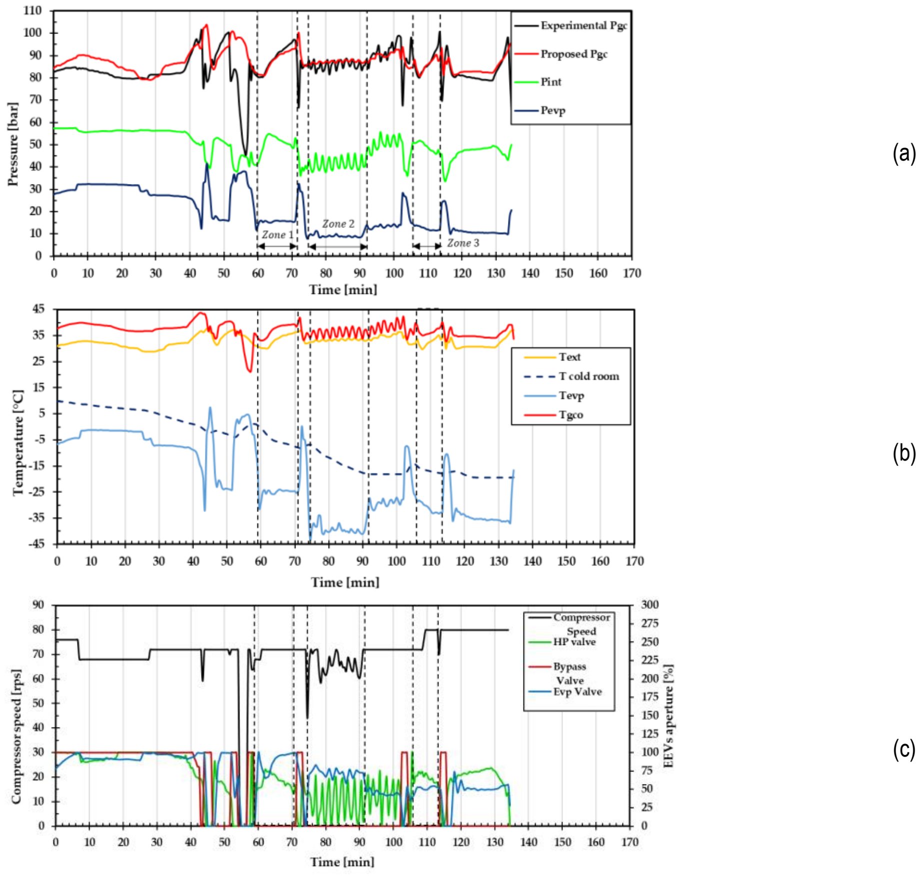



Fig. 3. Experimental results of the test

Due to the scope of this work, the complete test is shown without interruptions, although there are transient zones where valid conclusions cannot be drawn. For example, sudden drops in the experimental gas cooler pressure are observed, which are not due to the action of the implemented control. These drops are caused either by compressor shutdown or by the opening of the forced bypass valve to relief the flash tank pressure, as observed in Fig. 3c. In Figure 3a, the stationary zones of the greatest interest in the test have been marked. In Zone 1, the evaporating temperature and compressor capacity are maintained at -25°C and 80%, respectively. The outside temperature progressively increases from 30°C to 36°C, resulting in an increase in the optimal gas cooler pressure due to the degree of closure of the HP valve. In Zone 3, the control of the gas cooler pressure is verified by stimulating the system with a decrease of 3.6K and an increase of 5.2K in the outside temperature. The compressor capacity varies between 80% and 100% and the evaporating temperature slightly decreases in this range, reaching a minimum of -33°C. In Zone 2, significant oscillation in the opening of the HP valve is observed, caused by the constant change in the outside temperature, which, in turn, modifies the gas cooler pressure set point. Although the control exhibits some instability, the maximum and average errors in the gas cooler pressure set point are acceptable, as shown in Table 2. During this test, the evaporating temperature remained around -40°C, and the compressor operated between 50% and 80% of its capacity. It should be noted that such abrupt changes in the outside temperature do not occur in the real operation of these systems. However, subjecting the system to such adverse conditions is a way to test the reliability of the suggested control.

Table 2. Maximum error and mean error of the $P_{gc}^{opt}$ set value

| | Zone 1 | Zone 2 | Zone 3 |
|---|---|---|---|
| $P_{gc}$ maximum error [%] | 4.4 | 6.3 | 7.9 |
| $P_{gc}$ mean error [%] | 1.7 | 2.5 | 1.94 |

In summary, this work has calculated the optimal compressor geometry for a two-stage $CO_2$ refrigeration system for low-temperature small-scale applications using a thermodynamic model. A correlation has been successfully proposed and implemented to calculate the set point of the optimal gas cooler pressure. The accuracy of the control has been experimentally validated through tests at different evaporating temperatures (-20°C, -40°C) while varying the outside temperature between 30°C and 40°C.

## 5. Acknowledgments

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