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Simultaneous implementation of rotary pressure exchanger and ejectors for CO_2 refrigeration system

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ABSTRACT

Natural refrigerant CO_2 has become a viable choice for refrigeration units. The CO_2 systems are working efficiently on land-based facilities, and their demand is increasing for offshore applications, e.g., cruise ships and fishing vessels, due to their environment-friendly nature and compactness. The investigated application of the CO_2 system in this work is a single-stage system for air conditioning and a two-stage system for provision refrigeration at high heat rejection temperatures. The CO_2 transcritical cycle allows operating in higher ambient temperatures and in a colder climate with significant heat recovery. However, the system efficiency decreases in higher ambient conditions due to the high-pressure ratio and expansion losses. Therefore, ejectors are implemented to boost the cycle efficiency at high heat rejection temperature conditions. The pressure exchanger (PX) device recently came up and claimed to be an option to recover expansion work in CO_2 systems. PX is already in use for reverse osmosis (RO) desalination units to recover pressure work from the high pressure reject concentrate to low-pressure seawater. This work theoretically investigates the implementation of a CO_2-PX for transcritical CO_2 systems combined with ejectors and compressors. The energy efficiency of alternative system configurations is evaluated for various operating conditions.

Keywords: Refrigeration, Carbon Dioxide, Transcritical, Pressure Exchanger, Ejector, Energy Efficiency.

1. INTRODUCTION

High GWP (Global Warming Potential) refrigerants are prohibited under different international and national regulations. As a result, instead of searching for new synthetic refrigerants, there has been a surge of interest in natural working fluids, such as hydrocarbons, water, carbon dioxide (CO_2), and ammonia. CO_2 has been used as a working fluid in the past, but the low critical point made operation difficult, particularly in hot climes. Lorentzen et al. (1994) reconsider CO_2 in a transcritical mode as a possible alternative refrigerant to combat global warming and ozone depletion. Today, such systems operate successfully in a transcritical mode.

However, a transcritical operation's heat absorption and rejection sides have a high-pressure difference. In addition, a traditional vapor compression cycle results in a significant energy loss in the expansion valve. As a result, there is a lot of interest in adopting alternate expansion devices to make a CO_2 system's Coefficient of Performance (COP) comparable with systems employing standard fluorinated working fluids.

The simple transcritical cycle can be improved by reducing the throttling losses of the transcritical CO_2 cycle (Elatar et al., 2021). Because of the characteristics of CO_2, one viable alternative is to recover the expansion work directly using an expansion machine. The COP may increase between 14% to 17% (Groll et al., 2007) by employing an expansion turbine. Another renowned way to improve CO_2 cycle efficiency is by ejectors. Both theoretically and empirically, the benefit of utilizing ejectors in transcritical CO_2 refrigeration systems has

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been verified. According to several simulation assessments, the COP of the ejector expansion transcritical CO_2 cycle can be about 20% greater than that of a basic transcritical cycle (Zhu et al., 2017).

This work investigates a new work recovery device for CO_2 systems, a pressure exchanger (PX). A pressure exchanger is made up of a cylindrical rotor with a series of channels positioned around the rotor's axis. The rotor rotates between two fixed end plates with ports for directing fluid flow into and out of the rotor channels on either side. The channel ends are regularly exposed to varying port pressures as the rotor rotates, causing compression and expansion inside the rotor channels. As a result, the pressure of a high-pressure stream can be transferred to a low-pressure stream, boosting the low-pressure stream's pressure while lowering that of the high-pressure stream (Fricke et al., 2019, Energy Recovery, 2017). The detailed working principle of pressure exchangers is explained in Thatte (2018) and Thatte (2019). The internal mechanism of the device is shown in Figure 1.

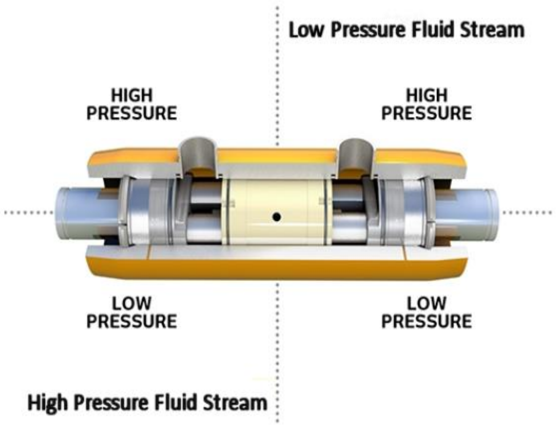

Figure 1: Cut section of PX and rotor (Fricke et al., 2019, Thatte 2018, Thatte 2019).

Figure 2 shows the four ports of the PX. High-pressure CO_2 enters from the top left port and, after decrementing in pressure, goes through the bottom-left port. The flash gas from the phase separator enters from the bottom right port and, after gaining pressure, leaves through the top right port. This work theoretically investigates different layouts for CO_2 systems to implement PX, including ejectors, and this will be verified experimentally in further work.

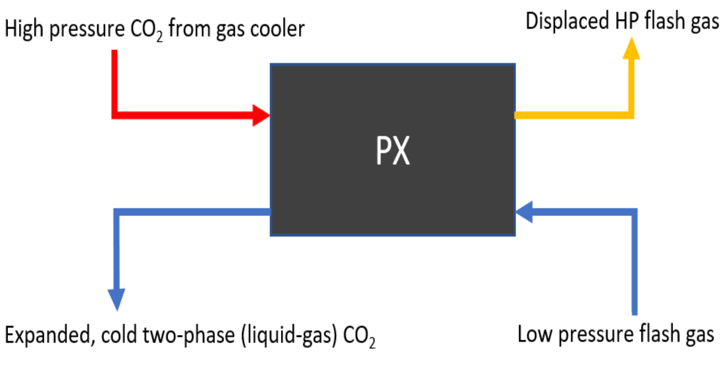

Figure 2: Two inlets and outlets of PX

2. METHODS AND DATA

The PX device can be implemented in the single-stage and two-stage refrigeration system, and the application can be an offshore or onshore installation. These two different layouts are analysed (Figure 4 and Figure 5). Figure 4 is a single-stage system producing chilled water at 5 °C for air conditioning purpose. Figure 5 shows

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a typical refrigeration system for supermarket/onboard provision with LT evaporator at -25 °C and MT evaporator 0 °C. The focus of this work is on PX and how it can best integrate into the proposed layouts. The flash gas from the phase separator needs a pressure boost before entering the PX. For this purpose, an ejector is used to lift the pressure of flash gas, and then the PX further increases the pressure. A second ejector is implemented to further boost the flash gas pressure at the PX outlet to the gas cooler pressure. The efficiencies for both ejectors were fixed to 30%. The PX's mass boost ratio can be calculated using Eq. 1 and Eq. 2. For calculation purposes, some simplifications were made, shown in Eq. 3 and Eq. 4. The entropy generation can be calculated using Eq. 6.

$$m_{BR} = \frac{\rho_{LP_{in}}}{\rho_{HP_{in}}} \quad \text{Eq. (1)}$$

$$m_{BR} = \frac{m_{LP_{in}}}{m_{HP_{in}}} \quad \text{Eq. (2)}$$

$$m_{LP_{out}} = m_{HP_{in}} \quad \text{Eq. (3)}$$

$$m_{HP_{out}} = m_{LP_{in}} \quad \text{Eq. (4)}$$

$$m_{BR} = \frac{h_{LP_{out}} - h_{HP_{in}}}{h_{LP_{in}} - h_{HP_{out}}} = \frac{h_{HP_{in}} - h_{LP_{out}}}{h_{HP_{out}} - h_{LP_{in}}} \quad \text{Eq. (5)}$$

$$E_g = m_{BR}.\left(S_{HP_{out}} - S_{LP_{in}}\right) + \left(S_{LP_{out}} - S_{HP_{in}}\right) \quad \text{Eq. (6)}$$

Where m is mass flow rate (kg/s), ρ is density (kg/m^3), h is enthalpy (kJ/kg), S is entropy (J/kg. K), and E_g is entropy generation (J/kg. K). The calculation models were established on an Engineering equation solver (EES) under various boundary conditions. Using Eq. 1 to Eq. 6, different combinations of compression and expansion efficiencies were found that satisfied the mass boost ratio. The mass boost ratios for both cases are presented in Table 3 and Table 4. Figure 3 shows the efficiency curves of PX for single stage and two stage. Each case is analysed with two temperatures of 33 °C and 35 °C (outlet temperature of gas cooler). Although 100% compression or expansion efficiency is not practically possible, the median efficiencies were used as an input for modelling.

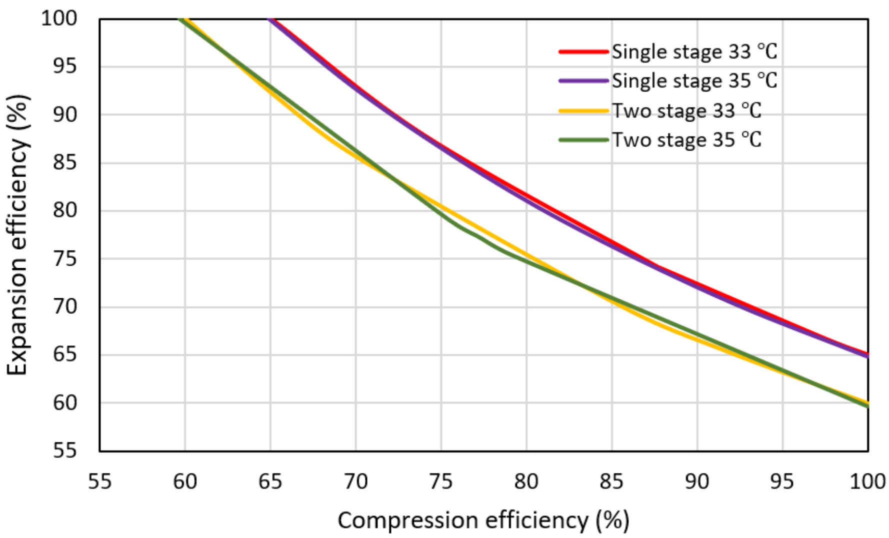

Figure 3: Efficiency curves for different operating conditions

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The boundary conditions for the single-stage system with PX is tabulated in Table 1, and the corresponding system layout is shown in Figure 4. The PX system layouts are compared with parallel compression to evaluate the efficiency of the system. Due to system's flow resistance (pressure losses in piping, and a small pressure loss in PX (~ 1 bar) etc.), the high-pressure outlet of the PX is ~ 2 bar lower than the gas cooler pressure. The value of 2 bar is selected based on the discussions with PX manufacturer. To overcome this small pressure loss, an ejector is used in conjunction with PX to boost the PX exit flow by additional 2 bar. Another ejector is implemented between the separator and PX to raise the flash gas pressure by 2 bar. After the gas cooler, the mass flow rate was kept constant in all cases, but two different temperatures were investigated. The temperature range will extend in further work. A suction gas heat exchanger is used to superheat the refrigerant gas by 5 K coming from the evaporator and by-pass valve. The compressor efficiency was kept constant.

Table 1. Boundary conditions for the single stage system

Parameters	Single stage (33 °C)	Parallel compression (33 °C)	Single stage (35 °C)	Parallel compression (35 °C)
Compressor outlet (bar)	86	84	91	89
Gas cooler pressure (bar)	84	84	89	89
Mass flow gas cooler (kg/s)	0.5	0.5	0.5	0.5
Liquid receiver pressure (bar)	55	55	55	55
Gas cooler outlet temperature (°C)	33	33	35	35
Compressor efficiency (%)	70	70	70	70
PX compression efficiency (%)	74.14	-	77.44	-
PX expansion efficiency (%)	87.7	-	83.78	-

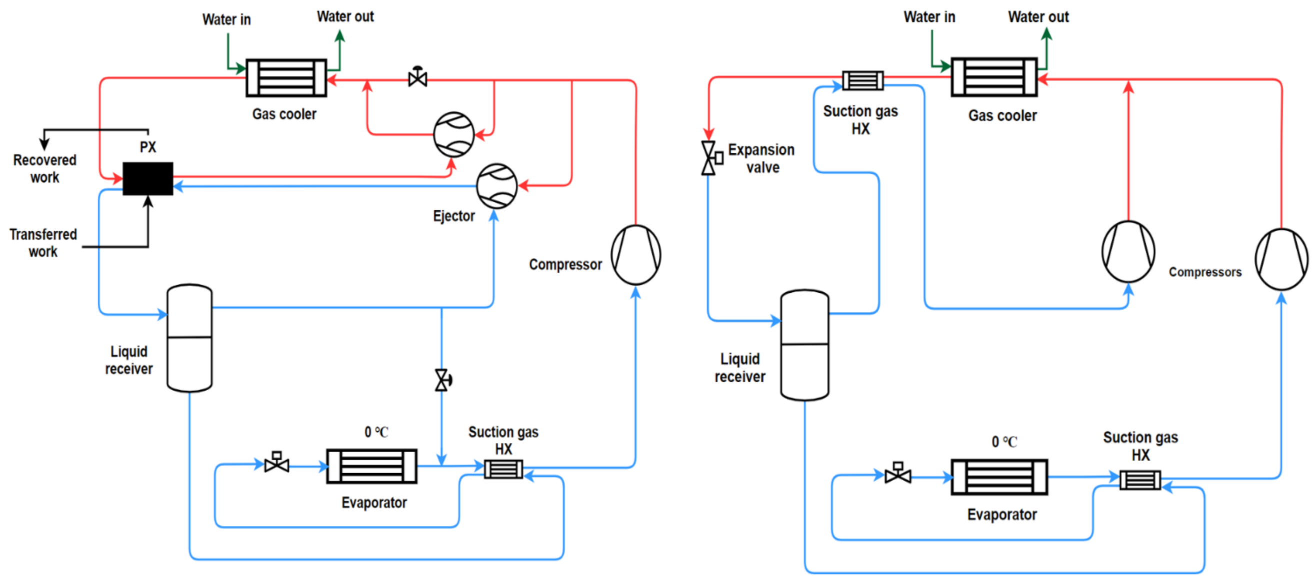

Figure 4: System layouts for single stage modelling, PX integration (left), Parallel compression (right)

The optimal (max COP) gas cooler pressure with their corresponding temperature was used in the models. The optimal pressure was also followed for the flash tank pressure. In the two-stage model, the flash tank

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pressure is 5 bar lower than in the single-stage system. The by-pass valve is also implemented in both layouts to ensure the removal of remaining gas that cannot handle by PX or ejector. The two-stage system layout with PX and ejectors is shown in Figure 5. All the parameters that were used for the two-stage modelling purpose are tabulated in Table 2.

Table 2. Boundary conditions for the two-stage system

Parameters	Two stage (33 °C)	Parallel compression (33 °C)	Two stage (35 °C)	Parallel compression (35 °C)
Compressor outlet (bar)	86	84	91	89
Gas cooler pressure (bar)	84	84	89	89
Mass flow gas cooler (kg/s)	0.5	0.5	0.5	0.5
Liquid receiver pressure (bar)	50	50	50	50
Gas cooler outlet temperature (°C)	33	33	35	35
Compressor efficiency (%)	70	70	70	70
PX compression efficiency (%)	70.41	-	75.42	-
PX expansion efficiency (%)	85.21	-	79.09	-

The refrigerant gas from the LT evaporator was superheated by 5 K, by exchanging heat in the liquid receiver. The refrigerant flow from the liquid line of the receiver is equally divided between the LT and MT evaporator. The LT compressor lifts the pressure to 34.85 bar, which is the same pressure as for the MT evaporator. The MT compressor lifts the gas from LT evaporator, by-pass valve, and MT evaporator to the designed outlet pressure of the MT compressor.

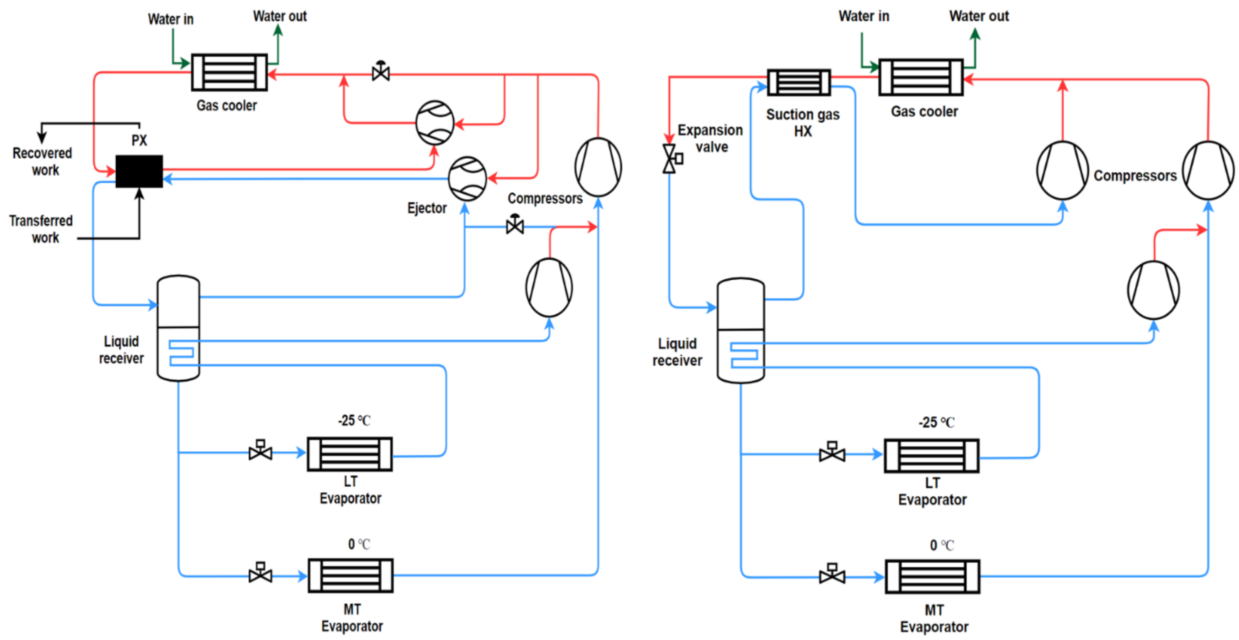

Figure 5: System layouts for two stage modelling, PX integration (left), Parallel compression (right)

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3. RESULTS AND DISCUSSIONS

Figure 6 shows the COP analysis of the single-stage system. The comparison of different system layouts indicates that COP of the PX system is 6% higher compared to parallel compression with 33 ℃ gas cooler outlet temperature. The COP difference increased to 7.5% when the gas cooler outlet temperature was 35 ℃.

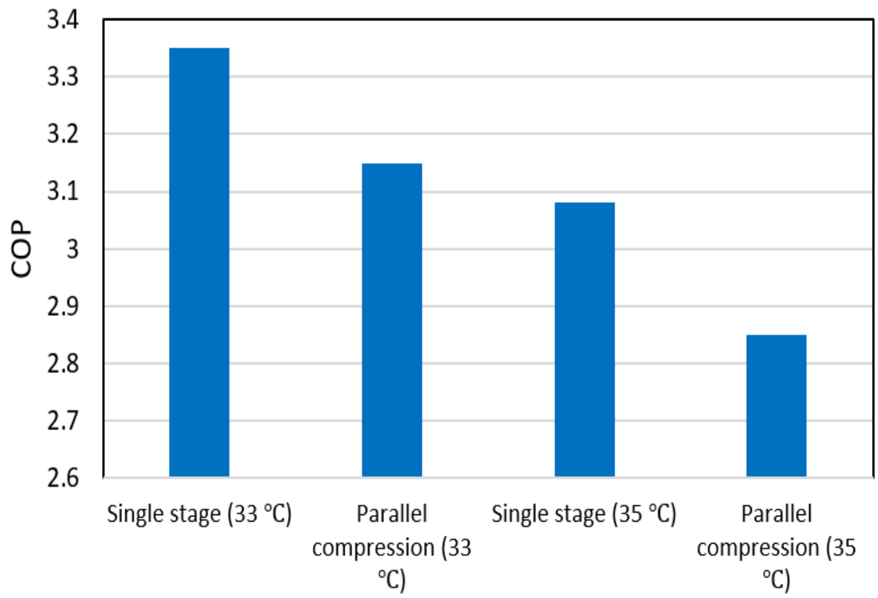

Figure 6: COP comparison between single-stage PX layout and parallel compression

The remaining modelling results for the single-stage system is presented in Table 3. The evaporation capacity in the PX case is higher than in the parallel compression case due to the less vapor fraction in the liquid receiver. An average 4.9% higher evaporation capacity was achieved in both PX cases. The recovered work from the PX expansion equals 10.4% of the compression work, but it is not enough to completely remove the flash gas. As a result, 15.5% of the flash gas is by-passed in the 33 ℃ case and 19.7% in 35 ℃ case.

Table 3. Modelling results for the single-stage system

Parameters	Single stage (33 ℃)	Parallel compression (33 ℃)	Single stage (35 ℃)	Parallel compression (35 ℃)
Evaporator load (kW)	70.93	67.77	68.85	65.11
Compression work (kW)	21.15	18.77	22.37	19.36
Parallel compressor (kW)	-	2.752	-	3.465
PX expansion (kW)	2.208	-	2.508	-
PX expansion theoretical (kW)	2.517	-	2.993	-
PX compression (kW)	2.205	-	2.504	-
Vapor quality	0.257	0.284	0.282	0.313
Mass boost ratio	0.258	-	0.263	-
Flash gas by-pass (%)	15.5	-	19.7	-

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Figure 7 shows the COP results of the two-stage system. In these scenarios, the COP of the PX system (33 ℃) case is 14.2% higher than the parallel compression under similar conditions. The COP takes further lead to 16.5% in the 35 ℃ case. The increase in COP is mainly due to the removal of flash gas by the free expansion work of PX.

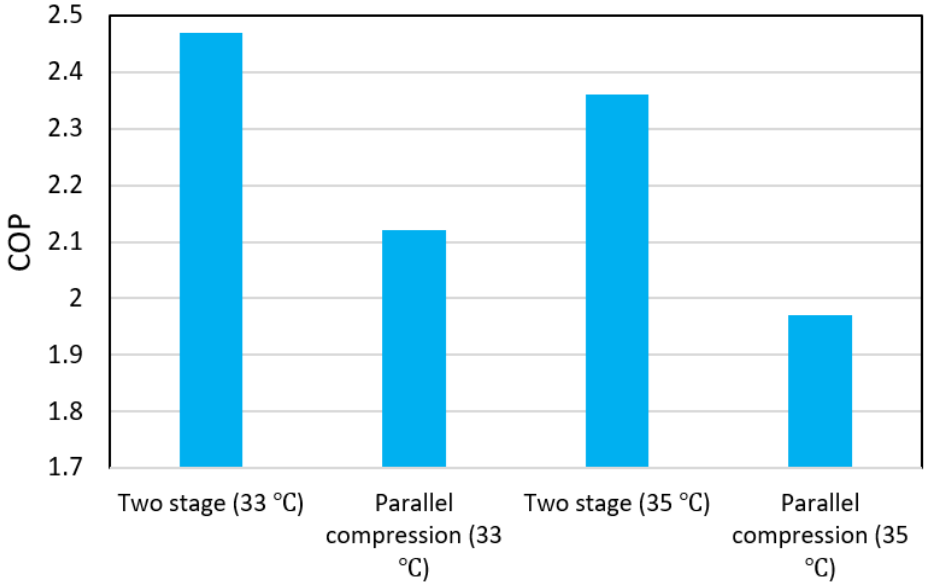

Figure 7: COP comparison between two stage PX layout and parallel compression

The modelling results for the two-stage system is presented in Table 4. In addition to the increased COP, the average evaporation capacity of both cases with PX is 4.4% higher than the parallel compression. The average vapor fraction difference between PX and parallel compression cases is 9.2%. The expansion follows the entropy lines in the PX cases, resulting in less vapor fraction than the isenthalpic process. The recovered work from the PX expansion equals 9.3% of the total compression work.

The results of this modelling work depend on the compression and expansion efficiency of the PX device. In further work, the experiments will be performed according to the system layouts highlighted in this work, which will act as a benchmark for extended applications of PX.

Table 4. Modelling results for the two-stage system

Parameters	Two stage (33 ℃)	Parallel compression (33 ℃)	Two stage (35 ℃)	Parallel compression (35 ℃)
Evaporator load LT (kW)	35.64	34.2	34.55	32.93
Evaporator load MT (kW)	34.56	33.16	33.5	31.92
Compression LT (kW)	7.91	7.59	7.667	7.307
Compression MT (kW)	20.42	20.08	21.13	20.75
Parallel compressor (kW)	-	4.068		4.911
PX expansion (kW)	2.68	-	2.884	-
PX expansion theoretical (kW)	3.146	-	3.646	-
PX compression (kW)	2.679	-	2.883	-
Vapor quality	0.2934	0.3232	0.3163	0.3484
Mass boost ratio	0.2243	-	0.2283	-
Flash gas by-pass (%)	34.4	-	37.1	-

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4. CONCLUSION

In this work, thermodynamic analysis has been performed to investigate the integration of a pressure exchanger device into the CO_2 transcritical refrigeration system. The numerical models were established on Engineering Equation Solver to evaluate the potential of PX to enhance the energy efficiency of the CO_2 system. Two different layouts were investigated, including PX and ejectors: a single stage, and a two-stage system. Comparison of the systems was made with a system based on parallel compression. The refrigerant flow was constant after the gas cooler in all cases, but two temperatures,33 ℃ and 35 ℃, were analyzed. Results for the single-stage system show that COP could improve by 6% to 7.5% under investigated conditions. In addition, the evaporation capacity was 4.9 % higher than the parallel compression case. The investigation of the two-stage system shows that COP could improve by 14 % to 16.5 % and with an increased evaporation capacity of 4.4%. In further work, experiments will be performed with the proposed layouts to verify the efficiency of PX.

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NOMENCLATURE

LP	Low pressure	*PX*	Pressure exchanger
HP	High pressure	*h*	Enthalpy (KJ/kg)
E_g	Entropy generation (J/kg. K)	*S*	Entropy (J/kg. K)
BR	Boost ratio	*in*	Inlet
out	Outlet	*LT*	Low temperature
MT	Medium temperature		

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