

Exergy-Based Assessment and Annual Performance of Rotary Pressure Exchanger–Integrated Transcritical CO₂ Refrigeration in Extreme Hot Climates

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Abstract- Transcritical CO₂ systems lose efficiency in extreme heat through high-pressure expansion and flash-gas recompression. A rotary pressure exchanger (PX) recovers part of the expansion work by using the dense gas-cooler outlet stream to compress receiver vapour inside rotating ducts. This study evaluates a supermarket booster with 100 kW medium-temperature and 25 kW low-temperature loads in four configurations: flash-gas bypass (FGB), parallel compression (PC), PX with flash-gas bypass (PX), and PX combined with parallel compression (PX + PC). A reduced-order model based on constant-pressure displacement of the driven vapour and the Span–Wagner equation of state is coupled to the booster model. High-side and receiver pressures are optimised from 0 to 52 °C. Component exergy destruction is evaluated for each layout, while 8784 hourly weather observations from 2024 at Kuwait International Airport and Ali Al Salem, Kuwait, provide the basis for annual performance. At 45 °C, PX + PC achieves a COP of 1.66, compared with 1.37 for PC and 1.14 for FGB, while total exergy destruction falls from 85.7 kW in FGB to 51.8 kW. Annual electricity use falls by 11.7% relative to PC and 24.0% relative to FGB. Peak power is reduced by up to 24.2% relative to PC, and hours of operation at the 120 bar high-side limit fall by 87–88%. Volumetric effectiveness has the greatest influence on annual savings: effectiveness of 0.7–1.0 gives savings of 9.2–13.0% relative to PC. The results support pairing a PX with parallel compression for efficient CO₂-only commercial refrigeration in extreme heat.

Index Terms- Exergy analysis, Expansion work recovery, Hot climate, Rotary pressure exchanger, Supermarket refrigeration, Transcritical CO₂

Highlights

- A reduced-order displacement model of the rotary pressure exchanger (PX) is coupled with a CO₂ booster model.
- Hourly optimisation of four booster layouts uses 2024 station observations from Kuwait International Airport and Ali Al Salem, Kuwait.
- Adding PX to parallel compression reduces annual electricity consumption in Kuwait by 11.7%.
- At 45 °C ambient, PX removes about 47% of high-pressure valve expansion exergy destruction.
- Annual savings depend chiefly on PX volumetric effectiveness; other device parameters change them by less than 1.5 points.

NOMENCLATURE

$\dot{E}$	exergy rate (kW)	ρ	density (kg m ⁻³)
E	annual electricity (MWh)	ψ	specific flow exergy (kJ kg ⁻¹)
h	specific enthalpy (kJ kg ⁻¹)	amb, 0	ambient, dead state
$\dot{m}$	mass flow rate (kg s ⁻¹)	D	destruction
MBR	PX mass boost ratio (–)	gc	gas cooler
p	pressure (bar)	HP, LP	high-, low-pressure port
$\dot{Q}$	heat transfer rate (kW)	LT, MT	low-, medium-temperature
r	pressure ratio (–)	rec	receiver
s	specific entropy (kJ kg ⁻¹ K ⁻¹)	COP	coefficient of performance
T	temperature (°C or K)	EEV	electronic expansion valve
$\dot{W}$	electrical power (kW)	FGB	flash-gas bypass

x	vapour quality (–)	HPV	high-pressure valve
ΔT_{app}	gas-cooler approach (K)	LLB	low-lift booster
Δp_{PX}	PX pressure loss (bar)	PC	parallel compression
η	efficiency (–)	PX	rotary pressure exchanger
λ	PX volumetric effectiveness (–)	SCOP	seasonal COP
ξ	rotor-drive power fraction (–)		

I. INTRODUCTION

The regulatory pressure upon high-GWP hydrofluorocarbons (HFCs) has pushed supermarket refrigeration toward CO₂-only (R744) working fluid systems [1,2]. CO₂ is nonflammable and nontoxic and has a GWP of one, much lower than HFCs; therefore, interest in the use of CO₂ as a refrigerant was revived three decades ago [3,4]. However, because of the thermophysical characteristics of CO₂, the performance of a CO₂ vapor compression refrigeration system is highly dependent upon the heat rejection temperature. With a critical temperature of 30.98 °C and a critical pressure of 73.77 bar [5], CO₂ rejects heat above the critical point when the ambient temperature exceeds roughly 25–28 °C. Therefore, it becomes necessary to control the gas-cooler pressure to an optimum value that increases rapidly with increasing gas-cooler outlet temperature [6,7]. The expansion valve decreases the pressure of the refrigerant from the high side-pressure (90–120 bar) down to the receiver pressure. The specific enthalpy remains unchanged during this throttling process, which destroys a large share of the exergy supplied by the compressors. In addition, the expansion generates flash gas, and as the gas cooler outlet temperature increases so does the flash-gas fraction. Consequently, the compressors are forced to re-compress all this flash gas while still not generating any additional refrigeration [1,8].

The issue is most pronounced in the Arabian Peninsula. In a World Meteorological Organization evaluation, Merlone et al. [9] confirmed 53.9 °C, measured at Mitribah, Kuwait, on 21 July 2016, as the highest temperature recorded in Asia. Also, climate models predict an increase in extreme summer heat for southwest Asia this century [10]. This study used annual hourly temperature observations for 2024: the ambient temperature exceeded 40 °C for 1,496 h at Kuwait International Airport and 1,494 h at Ali Al Salem, and reached 51.0 °C at Kuwait International Airport. Such conditions are indicative of lengthy durations of trans-critical operations at or very close to the upper (high) side design pressure of a CO₂ rack.

Several solutions exist for warmer climates. Parallel compression allows flash gas to be drawn directly from the receiver and compressed from receiver pressure. Thus it avoids reducing that gas to the medium-temperature suction pressure at which the booster operates before compression. This results in less power consumption than with standard boosters, by close to one-fifth at 30 °C [11,12]. Ejectors partially recover expansion work, using the recovered work to raise the suction pressure of a portion of the refrigerant flow through multi-ejector racks and allow flooded evaporators [13–16]. Dedicated or integrated mechanical subcooling lowers the vapour quality after expansion [17,18]. There are proposals for fully integrated CO₂ systems that provide refrigeration, space heating and air conditioning [19], including systems designed for hot countries [2]. Studies comparing various booster designs used in supermarkets ranked parallel-compression and ejector-supported designs above traditional booster designs when operating in warmer climates [20,21]. High ambient temperature studies were conducted in the Middle East and India by Purohit et al. [22]. The study indicated that in mild-warm climates an all-CO₂ multi-ejector unit remained competitive with an NH₃/CO₂ cascade system. The cascade gained up to 12.23% in the hottest locations. An expansion turbine can raise the efficiency of a transcritical cycle markedly [23], but two-phase CO₂ expanders remain rare in commercial racks.

The rotary pressure exchanger (PX) is another way to accomplish this. It was originally designed for use in a seawater reverse osmosis process. The seawater is forced through a ceramic rotor that has axial flow passages. The pressure of the reject brine is transferred to the incoming seawater [24]. This is the principle behind the wave-rotor designs being considered for topping gas turbines. In these devices, pressure waves are created in rotating channels and they transfer energy between two gas streams [25]. The dense fluid exiting the gas cooler in a carbon dioxide refrigeration system is under high pressure as it enters the pressure exchanger. As this high pressure fluid flows into the rotor of the pressure exchanger it compresses the receiver vapor inside the rotor ducts from low pressure to higher pressure. Simultaneously, the compressed vapor is displaced out of the rotor and exits through the high pressure exit. As this occurs the original high pressure dense fluid expands down to receiver pressure. Therefore the pressure exchanger can be said to perform both an expansion and compression in one rotor. Elatar et al. [26] used two-dimensional CFD to examine the rotary pressure exchanger design and found their highest compression effectiveness, up to 42.8%, at the lowest rotational speed they tested. Thatte [27] reported mass boost ratios of 0.27 and 0.40 with receiver pressures of 37 and 48 bar, respectively, in a heat-pump application. Sarawate et al. [28,29] discussed possible system architectures for this unit. They noted that an auxiliary low-lift device or a secondary gas cooler would have to compensate for the small pressure loss across the rotor. In addition, they constrained mass flow rates based on the density ratio of the two inlet streams. Saeed et al. [30] predicted COP gains of 17.7–23.5% over a standard booster and 16.3–20.3% over parallel compression for an evaporation temperature of 0 °C and gas-cooler outlet temperatures of 33–37 °C. In addition, their experiments [31] showed that free compression was possible over about 70 bar, along with a COP improvement of up to 18.2%. Sengupta and Dasgupta [32,33] developed correlations for estimating performance based on CFD modelling of the PX. They predicted annual savings of 4.16–6.57% relative to a multi-ejector system across climates that included Riyadh, for ambient temperatures up to

40 °C. The literature includes studies using the PX combined with ejectors [34] and a commercial case study [35]. Field measurement campaigns at supermarkets in North America and Europe report peak COP gains of up to 30% [36].

Three gaps remain for extreme hot climates. First, existing system studies either stop at 40 °C or examine a few design points. Gulf cities spend hundreds of hours between 40 and 50 °C; at those times, high-side pressure limits and flash-gas generation dominate. Second, although energy gains are documented, their relationship to component-level exergy destruction has rarely been traced across the full ambient-temperature range. This shows whether the PX reduces the throttling loss or simply relocates it. Exergoeconomic methods have been applied to transcritical CO₂ machines [37]. Exergy analyses of parallel-compression cycles identify the gas cooler and expansion process among the largest sources of irreversibility [38]. More advanced exergy analyses of CO₂ boosters using parallel compression have identified the portion of these losses that could be avoided [39,40]. None of these exergy studies includes a PX. Third, system-level studies need a transparent reduced-order PX model with physically meaningful parameters that can be varied in a sensitivity analysis. The current research addresses these needs. Its contributions are:

- a reduced-order PX model using constant pressure to displace the driven vapour. State densities determine the mass boost ratio, driven-gas enthalpy increase and motive-stream expansion. Volumetric effectiveness, pressure loss and rotor-drive power are represented as explicit loss parameters;
- component-level exergy analysis of the FGB, PC, PX and PX + PC layouts over the ambient-temperature range from 0 to 52 °C, with optimal high-side and receiver pressures and a maximum allowable high-side pressure of 120 bar;
- an annual assessment based upon 2024 hourly station data for Kuwait International Airport and Ali Al Salem in Kuwait. The results include annual electricity consumption, seasonal COP, seasonal exergy efficiency, peak power demand and hours at the pressure limit;
- a sensitivity analysis assessing the impact of changes in both the PX loss parameters and the main system assumptions on the annual saving.

Section II describes the system, Section III the models and data, Section IV the results, and Section V the conclusions.

II. SYSTEM DESCRIPTION

The reference plant is a centralised CO₂ booster for a medium-size supermarket, with medium-temperature (MT) refrigeration duty of 100 kW at −10 °C and low-temperature (LT) duty of 25 kW at −35 °C. This range of evaporation levels has commonly been used in supermarket booster studies [41–44]. Fig. 1 illustrates the PX + PC configuration and the state points used in the analysis. The LT compressors discharge into the MT suction line (state 12), where the stream combines with the MT evaporator outlet (state 9). In layouts without a parallel compressor, it also mixes with flash gas throttled through the flash-gas bypass (FGB) valve. The MT compressors deliver the mixture to the gas cooler (state 2). Saturated liquid leaves the receiver (state 5) and feeds both the MT and LT electronic expansion valves (EEVs).

The four booster configurations vary solely in how the gas-cooler outlet stream (state 3) is expanded and how the receiver vapour (state 6) is processed:

- **FGB (reference booster):** the entire gas-cooler flow is expanded into the receiver using the high-pressure valve (HPV), and the flash gas is then throttled down to the MT suction pressure.
- **PC:** the HPV expands the gas-cooler flow, while a parallel compressor draws off the flash gas at receiver pressure and discharges it to the gas cooler (state 14).
- **PX:** the entire gas-cooler flow enters the pressure exchanger via its high-pressure inlet (HP-in), as in the field installations where the HPV stays closed while the PX operates [36]. The pressure exchanger expands this stream to receiver pressure through the low-pressure outlet (LP-out, state 4). Simultaneously, it compresses receiver vapour drawn through the low-pressure inlet (LP-in). The compressed vapour then exits through the high-pressure outlet (HP-out, state 7). A low-lift booster (LLB) raises the HP-out pressure by an amount equivalent to the PX pressure loss before the stream joins the gas-cooler inlet (state 7b). This follows the configuration tested by Saeed et al. [31]. Flash gas not accepted by the PX is diverted to the MT suction line.
- **PX + PC:** as PX, but the residual flash gas is compressed by a parallel compressor.

Fig. 2 illustrates the duct cycle by viewing the process on an unwrapped rotor. A duct filled with receiver vapour is exposed to the HP-in port. A compression wave raises the vapour pressure, and the incoming dense fluid pushes the compressed vapour out through HP-out. The duct then passes a sealed sector before opening to the LP-out port. An expansion wave reduces the pressure of the trapped dense fluid, which exits as a two-phase mixture. Fresh vapour flows into the duct through LP-in during this time. The manufacturer describes the depressurisation as lying between isentropic and isenthalpic, with the HP-out pressure about 1–2 bar below the HP-in pressure [36].

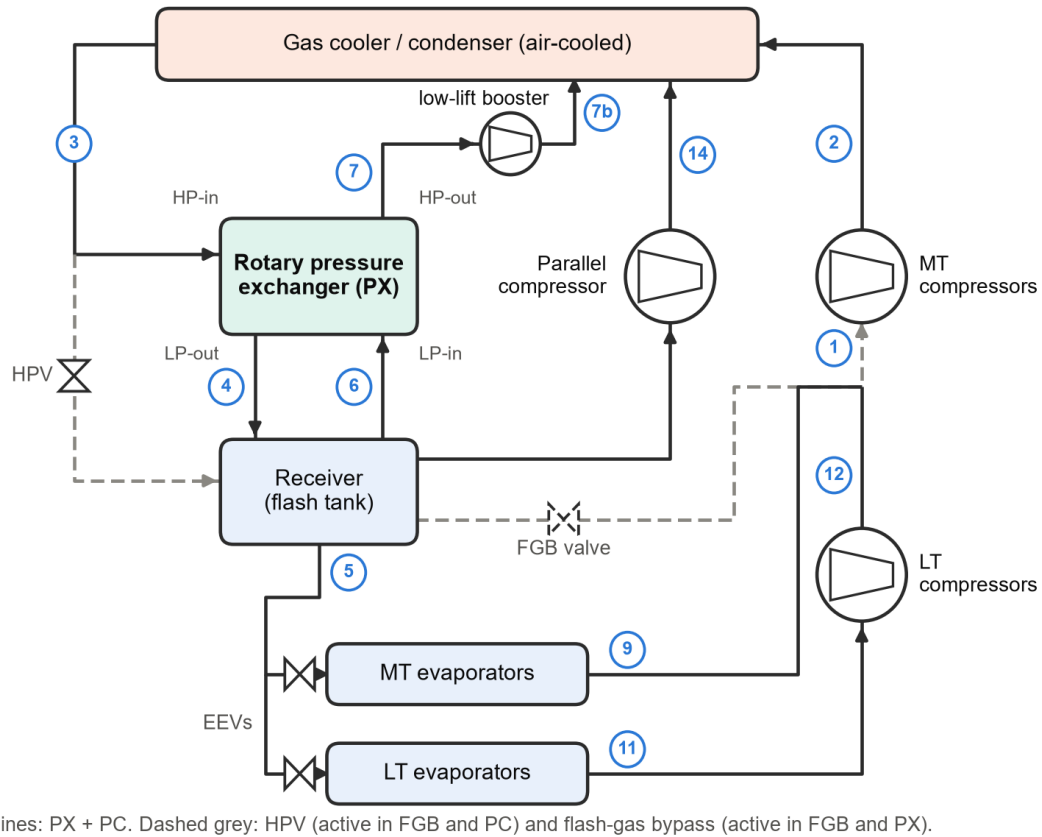


Fig. 1. Schematic of the PX + PC booster with state points. The high-pressure valve (HPV) and flash-gas bypass (FGB) branches, shown dashed, are active in the reference layouts.

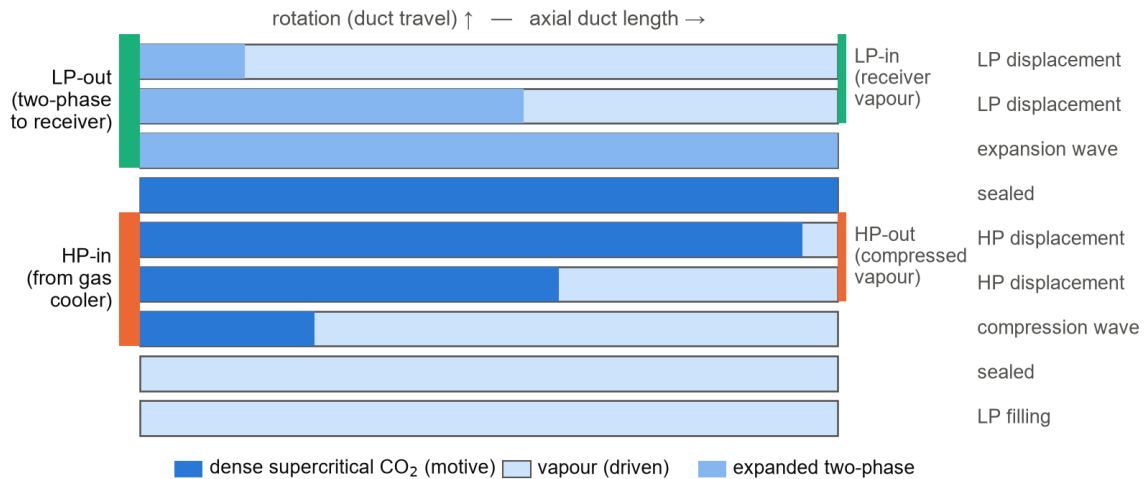


Fig. 2. Duct cycle of the rotary PX on an unwrapped rotor (rotation upwards). The HP ports sit on the compression side and the LP ports on the expansion side.

III. METHODOLOGY

A. Modelling assumptions

Each layout is represented by a steady-state, lumped-parameter model with the following assumptions:

- steady operation is assumed, while pressure losses through heat exchangers and piping are ignored except for the pressure loss through the PX;

- the gas-cooler or condenser exit temperature equals ambient temperature plus an approach $\Delta T_{app} = 3$ K; the condensing temperature cannot fall below 10 °C (45.0 bar);
- saturated liquid and saturated vapour leave the receiver, and a fixed superheat of 5 K applies at the evaporator outlets;
- throughout the year, evaporator duties remain constant because the display cabinets are located within an air-conditioned sales area;
- compressor operation is adiabatic, with isentropic efficiency depending on pressure ratio; an electro-mechanical efficiency of 0.90 converts required shaft power to electrical input;
- the high-side pressure is capped at 120 bar, and limits of 110 and 130 bar are evaluated in the sensitivity analysis;
- CO₂ properties are computed with the Span–Wagner equation of state [5] as implemented in CoolProp [45].

Table 1 lists the parameter values.

Table 1. Design and operating parameters of the reference plant and of the PX model.

Parameter	Symbol	Value
MT / LT evaporator duty	$\dot{Q}_{MT}, \dot{Q}_{LT}$	100 / 25 kW
MT / LT evaporation temperature	T_{MT}, T_{LT}	−10 / −35 °C
MT / LT evaporation pressure	p_{MT}, p_{LT}	26.49 / 12.02 bar
Useful superheat at evaporator outlets	—	5 K
Gas-cooler approach	ΔT_{app}	3 K
Minimum condensing temperature	—	10 °C
Maximum high-side pressure	$p_{gc,max}$	120 bar
Receiver pressure, FGB	p_{rec}	35 bar (fixed)
Receiver pressure range, PC, PX, PX + PC	p_{rec}	30–45 bar (optimised)
Electro-mechanical efficiency	η_{em}	0.90
PX volumetric effectiveness	λ	0.90
PX pressure loss	Δp_{PX}	1.5 bar
Low-lift booster isentropic efficiency	η_{LLB}	0.50
Rotor-drive power fraction	ξ	0.05
Refrigerated-space temperatures (exergy product)	$T_{sp,MT}, T_{sp,LT}$	0 / −25 °C

B. Component models

The refrigerant mass-flow rates in the MT and LT evaporators are determined from energy balances using saturated receiver liquid at the inlet:

$$\dot{m}_{MT} = \frac{\dot{Q}_{MT}}{h_9 - h_5}, \quad \dot{m}_{LT} = \frac{\dot{Q}_{LT}}{h_{11} - h_5} \quad (1)$$

Each compressor is modelled through its isentropic efficiency using the pressure-ratio correlation of Brunin et al. [46], which has been widely used for CO₂ compressors [7,11]:

$$\eta_{is} = 0.815 + 0.022r - 0.0041r^2 + 0.0001r^3 \quad (2)$$

$$W_{comp} = \frac{\dot{m}(h_{out,s} - h_{in})}{\eta_{is}\eta_{em}}, \quad h_{out} = h_{in} + \frac{h_{out,s} - h_{in}}{\eta_{is}} \quad (3)$$

where r is the ratio of discharge pressure to suction pressure, and $h_{out,s}$ is the isentropic discharge enthalpy. The gas-cooler exit state is specified by the high-side pressure p_{gc} and exit temperature $T_3 = T_{amb} + \Delta T_{app}$. Below the critical pressure, the lower bound of p_{gc} is the saturation pressure at T_3 , so the fluid leaves as liquid. Valves throughout the system operate at constant enthalpy. The receiver separates the incoming two-phase flow of quality x_4 into saturated liquid and vapour, so that

$$\dot{m}_{gc} = \frac{\dot{m}_{MT} + \dot{m}_{LT}}{1 - x_4}, \quad \dot{m}_{fg} = x_4 \dot{m}_{gc} \quad (4)$$

In FGB, the flash gas $\dot{m}_{fg}$ is throttled directly to MT suction. In PC, a parallel compressor compresses it from p_{rec} to p_{gc} . An adiabatic mixing balance determines the MT suction state. COP is defined as total cooling divided by total electrical input:

$$\text{COP} = \frac{\dot{Q}_{\text{MT}} + \dot{Q}_{\text{LT}}}{\dot{W}_{\text{LT}} + \dot{W}_{\text{MT}} + \dot{W}_{\text{PC}} + \dot{W}_{\text{LLB}} + \dot{W}_{\text{drv}}} \quad (5)$$

C. Reduced-order model of the rotary pressure exchanger

The PX model views each duct as a control volume filled alternately by the motive and driven streams. During the high-pressure portion of the cycle, dense motive fluid (state 3) enters a duct of volume V containing receiver vapour (state 6) and forces the compressed vapour out at the HP-out pressure $p'_{\text{HP}} = p_{\text{gc}} - \Delta p_{\text{PX}}$. During the low-pressure portion, vapour enters through LP-in and forces the expanded motive fluid out to the receiver. Ideally, the volume of motive fluid entering the duct per cycle equals the volume of vapour admitted. The mass-flow ratio is therefore set by the inlet densities. Mixing at the fluid interface, leakage through rotor clearances and incomplete scavenging decrease this ratio. A volumetric effectiveness λ accounts for these losses:

$$\text{MBR}_V = \frac{\dot{m}_{\text{LP,in}}}{\dot{m}_{\text{HP,in}}} = \lambda \frac{\rho_6}{\rho_3} \quad (6)$$

The incoming dense fluid compresses the vapour over a time short compared with the duct residence time. According to the manufacturer, this process is an acoustic compression wave [36]. The vapour therefore undergoes compression against the full HP pressure from the beginning. For an adiabatic control mass compressed at constant external pressure p'_{HP} and then displaced at that pressure, an energy balance determines the enthalpy rise of the driven stream:

$$h_7 - h_6 = \frac{p'_{\text{HP}} - p_{\text{rec}}}{\rho_6} \quad (7)$$

The same result can be obtained from a shock-compression balance in which the kinetic energy imparted to the gas is dissipated further downstream. Eq. (7) gives a larger enthalpy rise than isentropic compression; the difference represents the intrinsic irreversibility of pressure exchange by displacement. With the rotor treated as adiabatic, the motive stream supplies the energy absorbed by the driven stream:

$$h_4 = h_3 - \text{MBR}(h_7 - h_6) \quad (8)$$

Combining Eqs. (6)–(8) at ideal volumetric effectiveness, $\lambda = 1$, shows that the specific enthalpy drop in the motive stream equals $(p'_{\text{HP}} - p_{\text{rec}})/\rho_3$, the displacement or flow work of the dense fluid. As the specific volume $1/\rho$ increases during expansion, this enthalpy drop lies between zero (isenthalpic throttling) and the isentropic enthalpy drop, consistent with the qualitative explanation of the device. A PX cannot extract more vapour from its receiver than the receiver produces. Increasing the mass boost ratio lowers the receiver inlet vapour quality x_4 . The availability limit is therefore determined by solving $\text{MBR} = x_4(\text{MBR})$:

$$\text{MBR}_{\text{av}} = \frac{h_3 - h_5}{h_6 - h_5 + (h_7 - h_6)}, \quad \text{MBR} = \min(\text{MBR}_V, \text{MBR}_{\text{av}}) \quad (9)$$

The high-pressure outlet flow leaves the PX at p'_{HP} and is raised to p_{gc} by the low-lift booster. The performance of this unit is quantified by its isentropic efficiency η_{LLB} . The rotor drive controls rotational speed and consequently the gas-cooler pressure [31]. Its electrical input is expressed as a fraction ξ of the work transferred to the driven stream:

$$\dot{W}_{\text{LLB}} = \frac{\dot{m}_{\text{PX}}(h_{7b} - h_7)}{\eta_{\text{em}}}, \quad \dot{W}_{\text{drv}} = \xi \dot{m}_{\text{PX}}(h_7 - h_6), \quad \dot{m}_{\text{PX}} = \text{MBR} \dot{m}_{\text{gc}} \quad (10)$$

The residual flash gas, $(x_4 - \text{MBR})\dot{m}_{\text{gc}}$, is directed to the MT suction in the PX configuration and compressed by the parallel compressor in PX + PC. The model includes three parameters describing physical losses: λ , Δp_{PX} and ξ . Each is varied in the sensitivity analysis in Section IV-F. At 45 °C ambient and a receiver pressure of 45 bar, the inlet density ratio ρ_6/ρ_3 is 0.235, comparable in magnitude to the mass boost ratios measured by Thatte [27].

D. Optimal operation

Numerical optimisation for each layout and ambient temperature determines the controller settings that maximise the coefficient of performance. An initial coarse search is followed by bounded Brent refinement within the admissible high-side pressure range: the saturation pressure at T_3 or 73.8 bar, as applicable, up to 120 bar. In PC, PX and PX + PC, receiver pressure is optimised in an outer loop between 30 and 45 bar. It remains fixed at 35 bar in the FGB layout because it has little effect on that cycle. The optimisation uses a 0.5 K grid from 0 to 52 °C; linear interpolation of the resulting performance maps supplies the hourly simulation.

E. Exergy analysis

The hourly ambient temperature defines the exergy dead state, $T_0 = T_{\text{amb}}$, at $p_0 = 1.013$ bar. The specific flow exergy and the exergy destruction rate for component k are

$$\psi = (h - h_0) - T_0(s - s_0) \quad (11)$$

$$\dot{E}_{D,k} = \sum_{in} \dot{m}\psi - \sum_{out} \dot{m}\psi + \dot{W}_k - \dot{E}_{Q,k} \quad (12)$$

using the conventions of Bejan [47] and Tsatsaronis [48]. For compressors and the low-lift booster, $\dot{W}_k$ denotes electrical input, so the destruction includes losses associated with the motors. All exergy released with the gas-cooler heat is counted as lost to the ambient. The plant's exergy product is the heat extracted from the refrigerated spaces, expressed as exergy:

$$\dot{E}_P = \dot{Q}_{MT} \left(\frac{T_0}{T_{sp,MT}} - 1 \right) + \dot{Q}_{LT} \left(\frac{T_0}{T_{sp,LT}} - 1 \right) \quad (13)$$

so the temperature difference between space and refrigerant contributes to exergy destruction in the evaporator. The plant exergy efficiency and balance are

$$\eta_{ex} = \frac{\dot{E}_P}{\dot{W}_{tot}}, \quad \dot{W}_{tot} = \dot{E}_P + \sum_k \dot{E}_{D,k} \quad (14)$$

The components are the LT and MT compressors, the parallel compressor, the low-lift booster, the gas cooler, the HPV or PX, the MT and LT EEVs, the evaporators, the FGB valve, the two mixing junctions and the receiver.

F. Climate data and annual integration

Hourly dry-bulb temperature measurements for 2024 were obtained from the Integrated Surface Database of the US National Centers for Environmental Information [49]. The stations are Kuwait International Airport (WMO 40582), about 17 km south of Kuwait City centre, and Ali Al Salem (ICAO OKAS), about 44 km west of it; they are the two Kuwaiti stations with 2024 records in the database. The stations represent the metropolitan area near the Gulf coast and the inland desert, respectively; both fall within the hot-desert class of the Köppen–Geiger classification [50]. Synoptic (FM-12) reports at the full hour were preferred for their 0.1 K resolution, with METAR (FM-15) reports as the secondary choice, followed by the nearest observation within ± 30 min. Only observations passing the database quality checks were selected. Gaps of up to 6 h were estimated through linear interpolation; longer gaps used the average for the same hour on neighbouring days. Ali Al Salem has no synoptic reports; its METAR reports, issued at 55 min past each hour, were assigned to the following full hour.

The annual electricity use, seasonal COP and seasonal exergy efficiency are

$$E = \sum_{t=1}^{8784} \dot{W}_{tot}(T_{amb,t}) \Delta t \quad (15)$$

$$SCOP = \frac{\sum_t (\dot{Q}_{MT} + \dot{Q}_{LT}) \Delta t}{E}, \quad \eta_{ex,s} = \frac{\sum_t \dot{E}_{P,t} \Delta t}{E} \quad (16)$$

with $\Delta t = 1$ h. Annual electricity savings are valued at the commercial tariff of 0.025 KWD kWh⁻¹, equivalent to US\$0.083 and applicable to the government and commercial sectors since 2017 [51]. Avoided emissions use the 2024 carbon intensity of Kuwaiti electricity generation, 635 g CO₂-eq kWh⁻¹ [52].

G. Verification and benchmarking

The gas-cooler energy balance and the exergy balance of Eq. (14) close to within 10⁻¹⁴ of the input power throughout the maps, supporting the conclusion that no streams or components have been omitted. An additional test against literature data uses the single-evaporator configuration reported by Saeed et al. [30], with evaporation at 0 °C and gas-cooler outlet temperatures of 33–37 °C. Table 2 compares the COP gains: at receiver pressures of 38–40 bar, the model predicts 20.4–23.5% over the standard booster and 13.2–17.3% over parallel compression, compared with 17.7–23.5% and 16.3–20.3% in the reference study. These ranges overlap, although the compressor data, receiver control and auxiliary devices differ. The calculated COPs correspond to efficient supermarket compressors without pressure drops across heat exchangers. Lower absolute COP values have been measured for a small laboratory plant employing parallel compression [53]. Comparisons throughout this paper therefore use identical assumptions for all layouts; Section IV-F also shows that the relative savings are insensitive to the compressor efficiency level.

Table 2. COP gain of the PX layout predicted by the present model and reported by Saeed et al. [30] (evaporation at 0 °C).

Case	$T_{gc,out}$ 33 °C	$T_{gc,out}$ 35 °C	$T_{gc,out}$ 37 °C	Reference study
PX vs. booster, $p_{rec} = 38$ bar	20.4%	21.5%	22.5%	17.7–23.5%
PX vs. booster, $p_{rec} = 40$ bar	21.4%	22.5%	23.5%	17.7–23.5%
PX vs. parallel, $p_{rec} = 38$ bar	15.1%	16.2%	17.3%	16.3–20.3%
PX vs. parallel, $p_{rec} = 40$ bar	13.2%	14.3%	15.4%	16.3–20.3%

Table 3 places the present work among the published PX studies.

Table 3. Published studies of the rotary pressure exchanger in CO₂ systems and the present work.

Study	Approach	Main finding
Elatar et al. [26]	2D CFD, nine cases	Compression effectiveness 25.7–42.8%; temperature rise 1–17 K
Thatte [27]	CFD and cycle model, heat pump	MBR 0.27 and 0.40 at 37 and 48 bar receiver; COP lift up to 40% at –10 °C evaporation
Sarawate et al. [28,29]	Cycle modelling, architectures	Density-ratio flow constraint; auxiliary lift or secondary gas cooler needed; one PX adds about 20% capacity to a 150 kW store
Sengupta and Dasgupta [32]	CFD correlations, 30–40 °C	PX system outperformed parallel compression and multi-ejector systems
Saeed et al. [30]	EES model, 0 °C evaporation	COP +17.7–23.5% vs. booster, +16.3–20.3% vs. parallel, +2.4–5.5% vs. ejector
Saeed et al. [31]	Laboratory experiments	Free lift about 70 bar; 1–2 bar loss covered by boosters; COP up to +18.2%
Sengupta and Dasgupta [33]	CFD-based model, annual, 13–40 °C	Annual saving 4.16–6.57% vs. multi-ejector; highest in Chennai and Riyadh
Field campaigns [36]	Measurement at seven stores	Peak COP gain up to 30%; flash-gas valve opening reduced by 21–22% at 24–27 °C
Present work	Displacement model, exergy, hourly 2024 data, 0–52 °C	Component exergy destruction and annual performance at two Kuwaiti sites

IV. RESULTS AND DISCUSSION

A. Climate of the two sites

Table 4 summarises the 2024 temperature series; Fig. 3 presents the duration curves and monthly means of daily maxima. Kuwait International Airport and Ali Al Salem have similar yearly averages (28.3 and 28.0 °C) and considerable seasonal variation: winter nights reach 3.0–3.2 °C, while daily maxima reach 45 °C or more on about 100 days. Ali Al Salem recorded 630 h above 45 °C, compared with 543 h at Kuwait International Airport. The long hot season keeps CO₂ racks in transcritical operation for almost half of the year: T_3 exceeds the critical temperature for 4,165 and 4,245 h, respectively. The maximum temperature reached 51.0 °C at Kuwait International Airport and 50.4 °C at Ali Al Salem.

Table 4. Hourly dry-bulb temperature data for 2024 (NOAA Integrated Surface Database).

Site (station)	Valid hours (%)	Mean (°C)	Max (°C)	Min (°C)	h > 35 °C	h > 40 °C	h > 45 °C
Kuwait International Airport (40582)	95.3	28.3	51.0	3.0	2,613	1,496	543
Ali Al Salem (OKAS)	95.2	28.0	50.4	3.2	2,577	1,494	630

Valid hours: share of the 8784 hours with a quality-controlled observation assigned to the full hour; the remainder was gap-filled (Section III-F).

B. Cycle performance at design conditions

Fig. 4(a) plots the optimised COP of the four layouts against ambient temperature, with the corresponding high-side pressures in Fig. 4(b). Below about 7 °C, all layouts operate at the 10 °C condensing floor; differences in performance arise solely from the flash gas formed between 45.0 bar and the receiver pressure. As ambient temperature increases, the COP of the FGB reference booster decreases from 2.47 at 25 °C to 1.61 at 35 °C and 1.14 at 45 °C. Its optimum high-side pressure reaches the 120 bar design limit at about 42 °C. Beyond this point, COP declines more rapidly because pressure can no longer be increased to lower the gas-cooler outlet enthalpy.

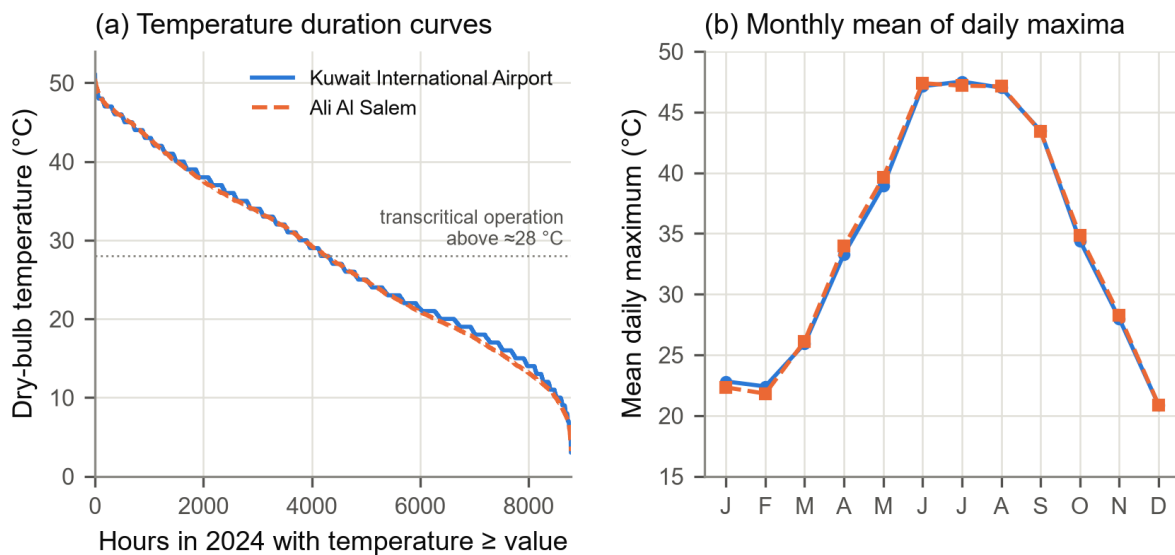


Fig. 3. Ambient temperature in 2024: (a) duration curves; (b) monthly mean of daily maximum temperature.

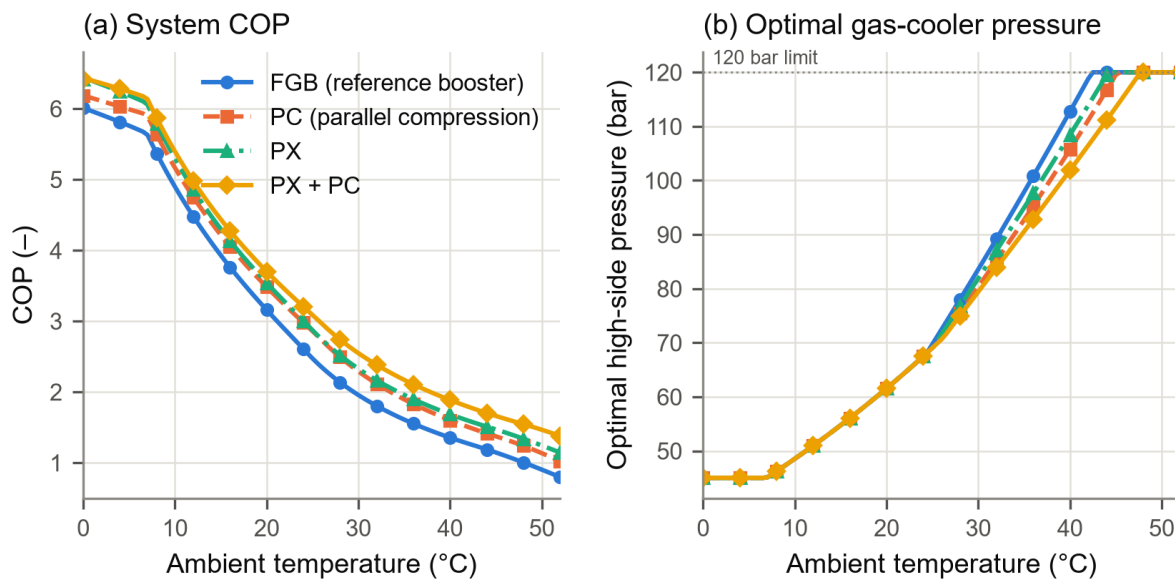


Fig. 4. (a) COP of the four layouts and (b) optimal high-side pressure versus ambient temperature.

Parallel compression improves COP by 17.7% at 35 °C and 19.9% at 45 °C, consistent with the reductions reported for integrated racks [12]. The PX configuration has a slight performance advantage over PC (1.47 versus 1.37 at 45 °C), because it decreases vapour quality after expansion while compressing a portion of the flash gas without a compressor. Combining the devices produces the greatest gain: at 45 °C, PX + PC achieves a COP of 1.66, exceeding PC by 21.0% and FGB by 45.1%; at 50 °C, the margins increase to 29.1% and 62.4%. The combined layout also reduces the optimum high-side pressure to 101.9 bar at 40 °C, versus 105.7 bar for PC and 112.7 bar for FGB. It therefore reaches the 120 bar limit only above about 47 °C.

The pressure–enthalpy diagram at 45 °C (Fig. 5) illustrates this improvement. The FGB reference booster employs isenthalpic expansion from 120 bar to 35 bar, producing a receiver inlet quality of 0.551. Flash gas returning to the MT compressors accounts for 0.633 of the 1.150 kg s⁻¹ leaving the gas cooler. In PX + PC, the optimiser lowers the high-side pressure to 113.52 bar and increases receiver pressure to 45 bar. Throttling from this state would yield a quality of 0.559. PX expansion reduces the motive-stream enthalpy by 10.5 kJ kg⁻¹, lowering receiver inlet quality to 0.505; the enthalpy reduction equals 66% of the isentropic drop of 15.9 kJ kg⁻¹. The rotor also compresses 0.247 kg s⁻¹ of receiver vapour to 112.02 bar and 87.7 °C, corresponding to an effective compression efficiency of 68%. The parallel compressor handles the remaining 0.343 kg s⁻¹, while the MT compressors process only the evaporator flows. Table 5 gives the state points for both cycles.

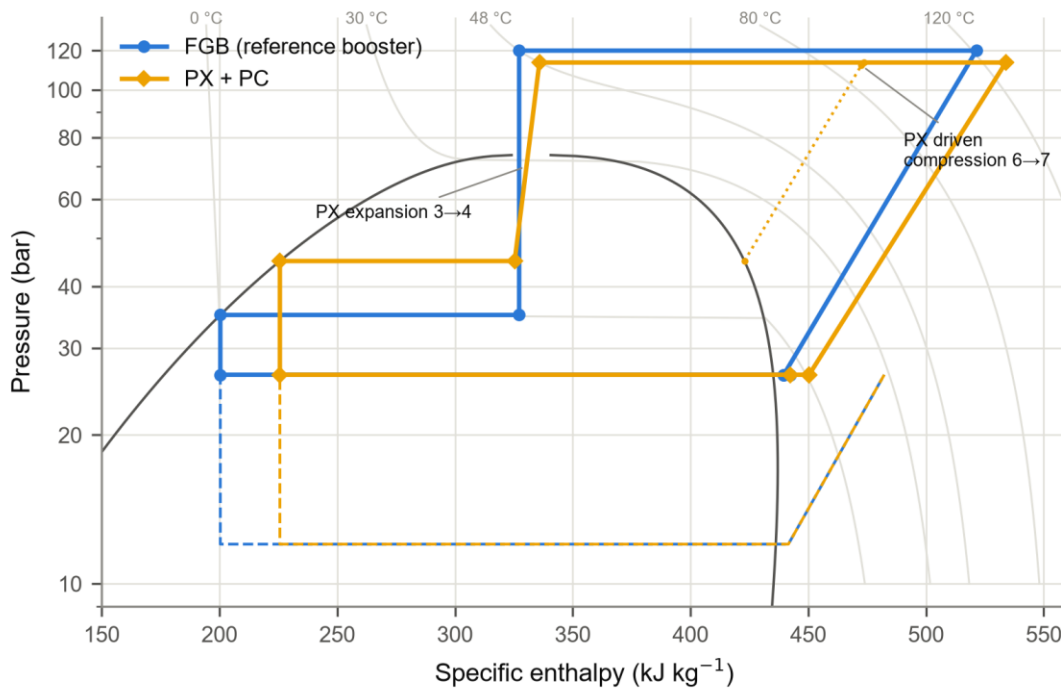


Fig. 5. Pressure–enthalpy diagram of the FGB and PX + PC cycles at 45 °C ambient (LT stages dashed).

Table 5. State points of the FGB and PX + PC layouts at 45 °C ambient (optimised pressures).

State	Location	FGB p (bar)	FGB T (°C)	FGB h (kJ kg ⁻¹)	PX + PC p (bar)	PX + PC T (°C)	PX + PC h (kJ kg ⁻¹)
1	MT compressor suction	26.49	-7.0	439.6	26.49	1.0	450.4
2	MT compressor discharge	120.00	121.1	521.4	113.52	126.9	533.9
3	Gas-cooler outlet	120.00	48.0	327.3	113.52	48.0	335.9
4	Receiver inlet (HPV or PX LP-out)	35.00	0.2	327.3	45.00	10.0	325.4
5	Receiver liquid	35.00	0.2	200.4	45.00	10.0	225.7
6	Receiver vapour	35.00	0.2	430.8	45.00	10.0	422.9
7	PX HP-out	–	–	–	112.02	87.7	472.5
7b	Low-lift booster outlet	–	–	–	113.52	89.1	473.7
9	MT evaporator outlet	26.49	-5.0	442.4	26.49	-5.0	442.4
11	LT evaporator outlet	12.02	-30.0	441.6	12.02	-30.0	441.6
12	LT compressor discharge	26.49	28.1	482.3	26.49	28.1	482.3
14	Parallel compressor discharge	–	–	–	113.52	84.0	463.7

Mass flow rates (kg s⁻¹): FGB gas cooler 1.150, flash-gas bypass 0.633; PX + PC gas cooler 1.167, PX driven stream 0.247, parallel compressor 0.343. COP: 1.144 (FGB) and 1.660 (PX + PC).

Discharge temperatures remain within the usual limits for CO₂ compressors. At 45 °C, MT discharge reaches 121.1 °C in FGB and 126.9 °C in PX + PC. The combined layout has warmer MT suction because cold flash gas no longer dilutes the LT discharge.

C. Operating characteristics of the pressure exchanger

Fig. 6(a) plots the mass boost ratio for both PX layouts alongside the inlet density ratio; Fig. 6(b) gives the share of flash gas compressed by the PX. Above about 25 °C, the mass boost ratio settles between 0.19 and 0.21, controlled by the displacement limit of Eq. (6) because the receiver generates more vapour than the rotor can extract. It increases again above 47 °C, when high-side pressure is held at 120 bar and the gas-cooler outlet density falls. In PX + PC, the PX compresses about 47% of the flash gas at 35 °C and 42% at 45 °C; the parallel compressor handles the remainder. Without parallel compression, increasing receiver pressure between about 13 and 20 °C allows the PX to capture nearly all the flash gas and eliminate bypass throttling. At higher temperatures, flash-gas production exceeds the displacement limit, leaving a residual flow to bypass. Consequently, PX alone loses performance relative to the combined layout in hot weather.

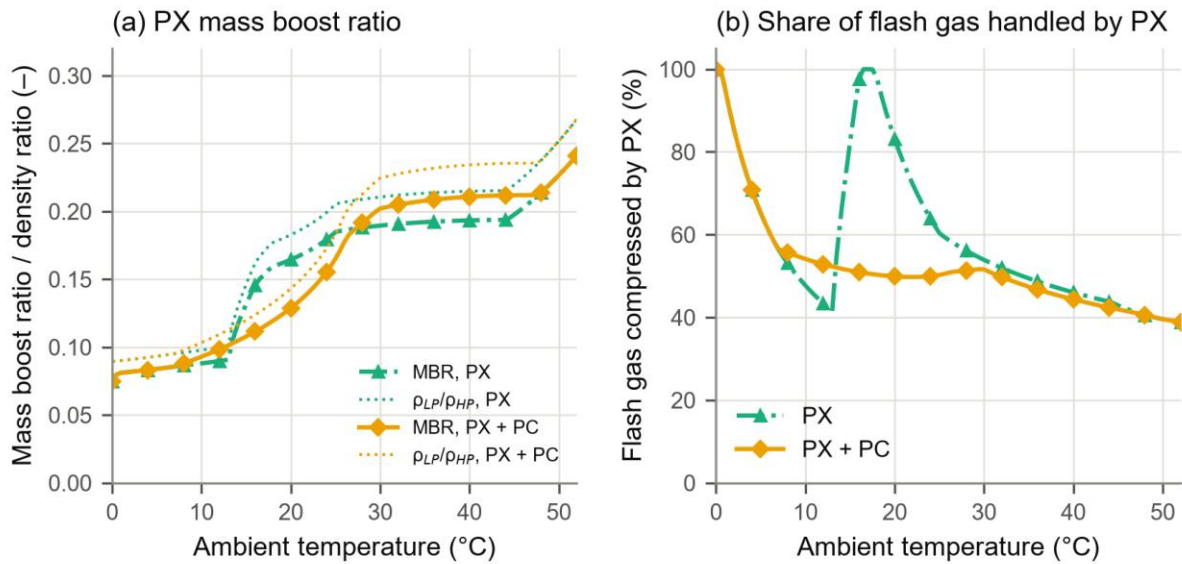


Fig. 6. PX operating characteristics: (a) mass boost ratio and inlet density ratio; (b) share of the flash gas compressed by the PX.

This behaviour provides a design rule for hot climates. The PX mass boost ratio saturates at about 0.2, whereas the flash-gas fraction exceeds 0.5 at 45 °C. A parallel compressor is therefore required at peak conditions, although it can be sized for roughly half the flash-gas flow that a plant without a PX would need to recompress.

D. Exergy destruction at design conditions

Fig. 7 separates the exergy destruction rates at 35 and 45 °C by component; Table 6 lists the values at 45 °C. Because product exergy is identical among all configurations (23.53 kW at 45 °C), every kilowatt reduction in total destruction produces an equal reduction in electrical input.

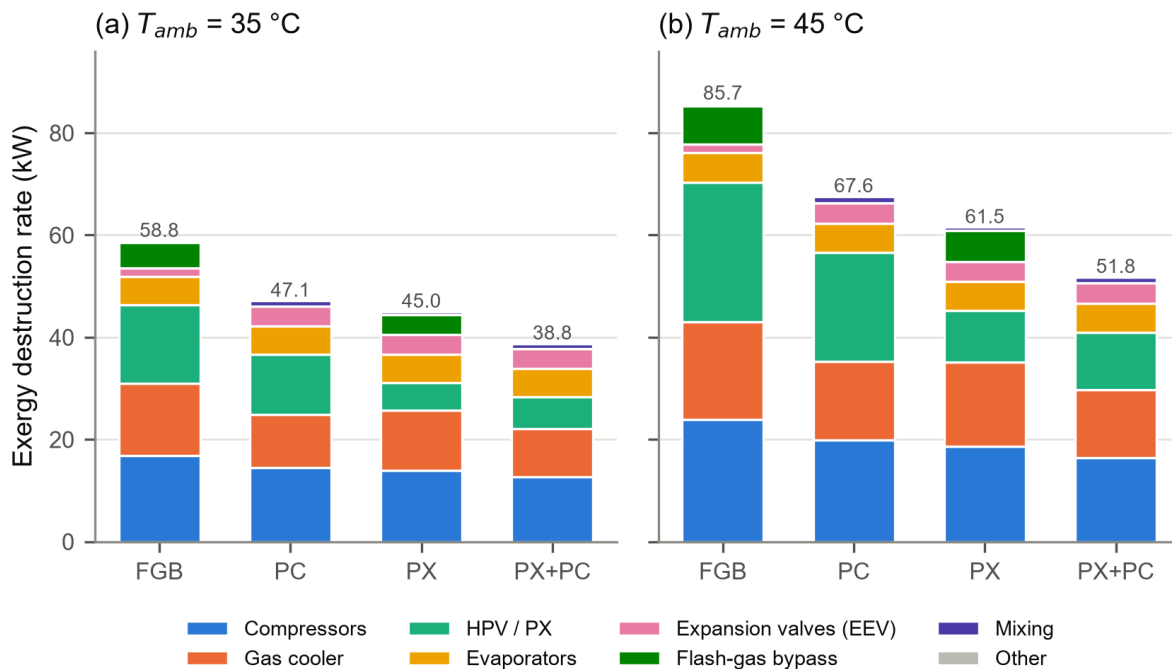


Fig. 7. Exergy destruction rate by component group at (a) 35 °C and (b) 45 °C ambient. Values above the bars are totals in kW.

Table 6. Component exergy destruction rates (kW) at 45 °C ambient and optimised pressures.

Component	FGB	PC	PX	PX + PC
LT compressors	1.17	1.30	1.30	1.30
MT compressors	22.75	12.00	17.14	11.49
Parallel compressor	–	6.64	–	3.48
Low-lift booster	–	–	0.14	0.17
Gas cooler	19.11	15.33	16.51	13.35
HPV (FGB, PC) or PX (PX, PX + PC)	27.31	21.34	10.09	11.17
MT and LT expansion valves	1.70	3.97	3.97	3.97
Evaporators (incl. space ΔT)	5.73	5.73	5.73	5.73
Flash-gas bypass valve	7.58	–	6.02	–
Mixing junctions	0.37	1.26	0.64	1.13
Total destruction	85.71	67.57	61.52	51.78
Exergy product	23.53	23.53	23.53	23.53
Electrical input	109.24	91.09	85.05	75.31
Exergy efficiency (%)	21.5	25.8	27.7	31.2

Adiabatic phase separation in the receiver destroys no exergy in the model and is omitted.

Three effects explain the reduction. Expansion is the largest individual source of exergy destruction in the reference booster: at 45 °C, the HPV destroys 27.31 kW, or 25.0% of electrical input. Parallel compression reduces this loss to 21.34 kW, primarily through increased receiver pressure. In PX + PC, pressure exchange reduces expansion destruction further to 11.17 kW, or 11.33 kW including the low-lift booster, about 47% below the PC layout's HPV. The loss does not disappear: displacement in Eq. (7) is inherently irreversible and transfers roughly two-thirds of the isentropic expansion work. The benefits also extend to the compressors and gas cooler. Less flash gas at the MT suction inlet and lower high-side pressure reduce MT compressor destruction from 22.75 kW in FGB to 11.49 kW. A smaller circulating flow at lower pressure cuts gas-cooler destruction from 19.11 to 13.35 kW. However, raising receiver pressure to 45 bar increases destruction in the MT and LT expansion valves to 3.97 kW in PX + PC, versus 1.70 kW in FGB, because liquid is throttled over a larger pressure difference. Total destruction in PX + PC is 51.78 kW, 23.4% below PC; the PX-only layout achieves a reduction of 28.2% relative to FGB.

Fig. 8 shows that all layouts reach peak exergy efficiency between 20 and 25 °C, as the cycle becomes transcritical while the exergy product continues to grow with ambient temperature. Above this range, FGB efficiency falls from 28.1% to 18.7% at 50 °C. The combined configuration retains a relatively level exergy-efficiency profile of 30.3–35.0%. This reflects its lower sensitivity to heat, particularly relevant in Kuwait, where both stations recorded about 1,500 h above 40 °C in 2024.

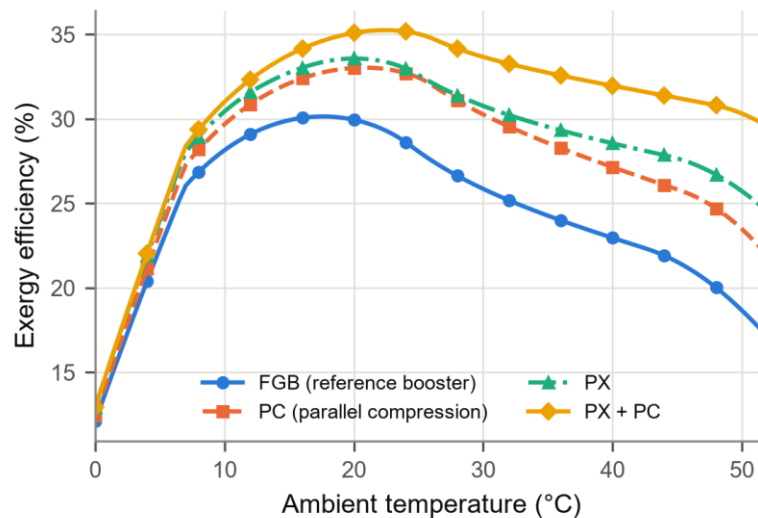


Fig. 8. Exergy efficiency of the four layouts versus ambient temperature.

E. Annual energy and exergy performance

Table 7 reports annual performance for the two sites, and Fig. 9 compares electricity consumption. Each plant supplies 1,098 MWh of cooling annually, while the reference booster consumes 551.0 MWh at Kuwait International Airport and 546.0 MWh at Ali Al Salem. A warmer winter and July, rather than summer peaks, explain the greater consumption at the airport: Ali Al Salem uses 5.0 MWh less

despite 87 more hours above 45 °C. Relative to FGB, annual savings are 13.9% for PC, 16.9% for PX and 24.0% for PX + PC at both sites. Compared with PC, the realistic alternative for a new rack in a hot climate, the combined layout saves 11.7%, equivalent to 55.7 MWh at Kuwait International Airport and 55.1 MWh at Ali Al Salem. Its seasonal COP is 2.62 and 2.65, respectively, and its seasonal exergy efficiency is 33.0%, versus 25.1% for FGB.

Table 7. Annual performance of the four layouts with 2024 hourly weather data.

Site	Layout	Electricity (MWh)	SCOP	$\eta_{ex,s}$ (%)	Peak power (kW)	Hours at 120 bar
Kuwait International Airport	FGB	551.0	1.99	25.1	147.0	1,056
	PC	474.2	2.32	29.1	115.8	528
	PX	457.6	2.40	30.2	104.7	714
	PX + PC	418.5	2.62	33.0	87.9	139
Ali Al Salem	FGB	546.0	2.01	25.1	141.8	1,075
	PC	470.1	2.34	29.1	112.3	572
	PX	453.7	2.42	30.1	102.2	748
	PX + PC	414.9	2.65	33.0	86.3	130

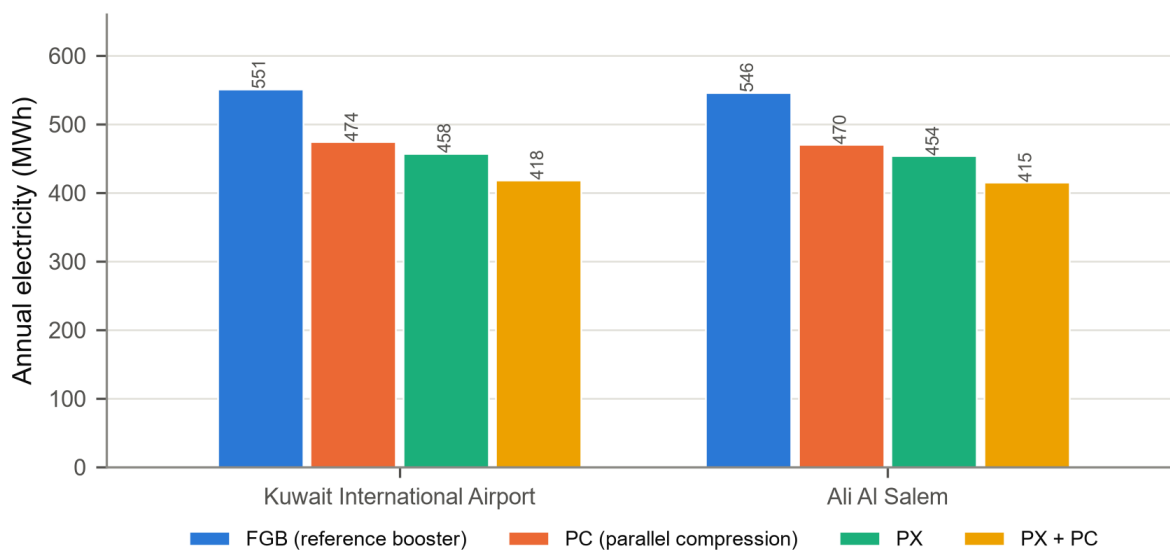


Fig. 9. Annual electricity use of the four layouts at the two sites (2024 weather).

Annual relative savings are nearly identical at the two sites despite their different locations. The monthly data in Fig. 10 explain this result. At both sites, about three-quarters of the saving relative to PC occurs from May to September, when electricity consumption is greatest and the PX benefit per kilowatt-hour increases with ambient temperature. Both consumption and relative gains are smaller in winter. The two sites share the same seasonal pattern, so their monthly savings follow almost the same curve. Kuwait International Airport achieves the larger absolute saving relative to PC, at 55.7 MWh.

Peak electrical demand is reduced by a greater percentage than annual energy use. In the hottest hour at Kuwait International Airport, the reference booster draws 147.0 kW; PC lowers this peak to 115.8 kW, and PX + PC reduces it further to 87.9 kW. The combined layout therefore cuts the peak by 24.2% relative to PC and 40.3% relative to FGB. Hours at the 120 bar high-side limit also decline markedly: FGB reaches the limit for 1,056 h at Kuwait International Airport and 1,075 h at Ali Al Salem, compared with 139 and 130 h, respectively, for PX + PC. Field campaigns identify the resulting pressure margin during heat waves as a practical benefit alongside energy savings [36].

Annual exergy destruction at Kuwait International Airport falls from 412.8 MWh with FGB and 336.0 MWh with PC to 280.3 MWh with PX + PC. Most of this reduction is attributable to the expansion device: the PC layout's HPV destroys 78.4 MWh, versus 44.1 MWh for the PX and its booster, a reduction of 44%. The gas cooler (69.5 → 65.2 MWh) and compressors (104.8 → 92.5 MWh) account for most of the remainder. Evaporator destruction stays at 47.7 MWh for every layout because it depends only on the cooling loads and temperature differences to the refrigerated spaces.

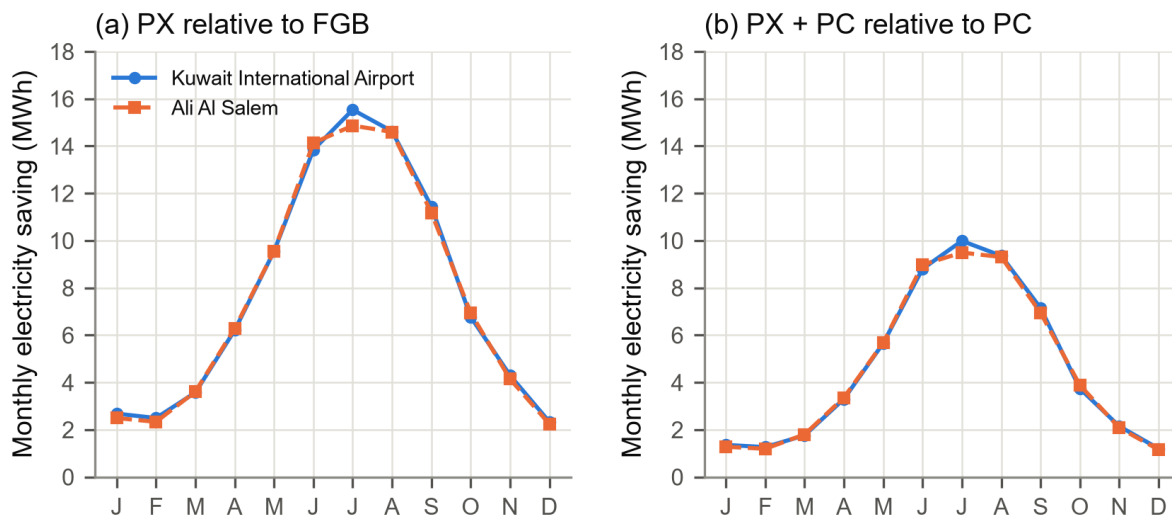


Fig. 10. Monthly electricity saving: (a) PX relative to FGB; (b) PX + PC relative to PC.

F. Sensitivity analysis

Fig. 11 ranks the effects of PX loss parameters and system assumptions on the annual electricity saving of PX + PC relative to PC at Kuwait International Airport. Volumetric effectiveness has the greatest influence: savings fall to 9.2% at $\lambda = 0.7$ and rise to 13.0% at $\lambda = 1.0$, because λ determines how much mass the PX compresses without a compressor. Rotor-drive power and pressure loss have smaller effects. Raising ξ from 0 to 0.15 reduces savings from 12.2% to 10.8%; raising Δp_{PX} from 0.5 to 3.0 bar reduces them from 12.1% to 11.2%. Booster efficiency has an even smaller influence (11.4–11.9%), as the booster raises a small flow through only 1.5 bar.

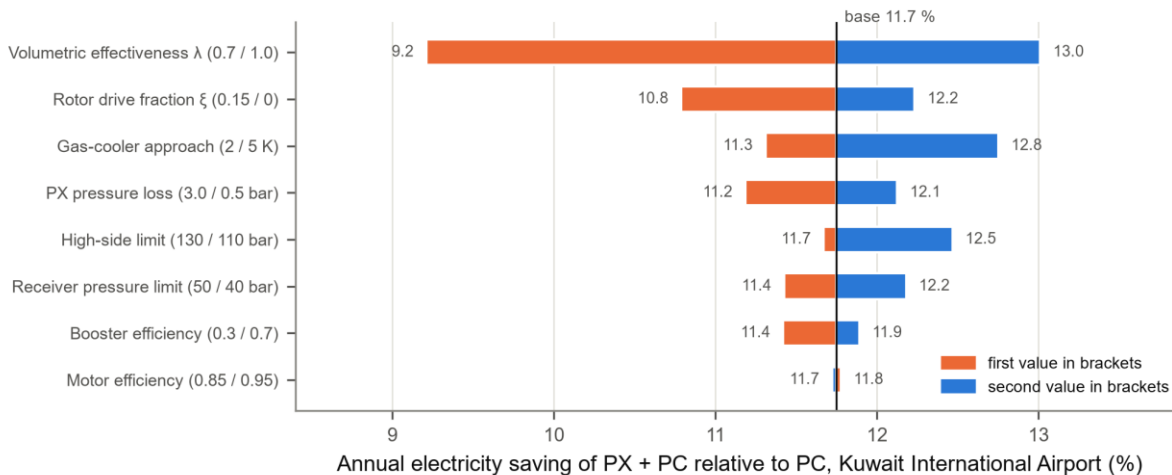


Fig. 11. Sensitivity of the annual electricity saving of PX + PC relative to PC at Kuwait International Airport. Bars span the two parameter values given in brackets; the vertical line marks the base case.

Increasing the gas-cooler approach temperature produces more flash gas for the PX to process and raises the annual saving from 11.3% at 2 K to 12.8% at 5 K. Allowing receiver pressure up to 50 bar slightly lowers the relative saving to 11.4%, because PC benefits more from the higher pressure than PX + PC. Electro-mechanical efficiency scales absolute consumption proportionally in every layout, leaving relative savings practically unchanged: 11.8% at 0.85 and 11.7% at 0.95. This supports relative savings as the main outcome. The parameter ranking is almost the same at Ali Al Salem, where savings span 9.2–13.0% across the range of λ . Even at $\lambda = 0.7$, PX + PC saves 9.2% relative to PC and PX alone saves 13.1% relative to FGB. The conclusions therefore do not require an optimistic PX assumption.

G. Economic and environmental indicators

Table 8 expresses annual electricity savings as cost reductions and avoided emissions. For a store of this size, a commercial tariff of 0.025 KWD kWh⁻¹ gives PX + PC annual savings of 1,378–1,393 KWD relative to PC and 3,277–3,312 KWD relative to FGB. Applying the 2024 grid intensity of 635 g CO₂-eq kWh⁻¹ [52], the saving relative to PC avoids 35.0–35.4 t CO₂-eq annually. These

results scale with tariff and compressor efficiency. Less efficient compressors would increase both reference consumption and absolute savings, making Table 8 conservative for installations with older compressors.

Table 8. Annual electricity, cost and emission savings (tariff 0.025 KWD kWh⁻¹; 635 g CO₂-eq kWh⁻¹).

Site	Comparison	Saving (MWh)	Saving (%)	Cost (KWD)	Emissions (t CO ₂ -eq)
Kuwait International Airport	PX vs. FGB	93.3	16.9	2,333	59.3
	PX + PC vs. PC	55.7	11.7	1,393	35.4
	PX + PC vs. FGB	132.5	24.0	3,312	84.2
Ali Al Salem	PX vs. FGB	92.3	16.9	2,309	58.7
	PX + PC vs. PC	55.1	11.7	1,378	35.0
	PX + PC vs. FGB	131.1	24.0	3,277	83.3

H. Limitations

The steady-state model assumes constant loading, excluding part-load compressor cycling, defrost and daily variations in cabinet loads. Instead of fitting a specific rotor design, the reduced-order displacement model varies the PX loss parameters. Predicted mass boost ratios agree in magnitude with published measurements, but a validated map of volumetric effectiveness λ against rotor speed and inlet conditions would improve prediction. Heat-exchanger pressure drops, gas-cooler fan power and adiabatic pre-cooling are excluded. Pre-cooling, widely used in hot climates, leaves less flash gas available to the PX; savings may therefore be lower when a PX is installed alongside an adiabatic gas cooler. The two stations do not represent the hotter northern desert, where the Mitribah record was set [9]. Public information on PX capital and maintenance costs is unavailable, so these costs are excluded from the economic indicators. The annual cost savings in Table 8 define the installation budget consistent with a chosen payback period.

V. CONCLUSIONS

This study evaluates a rotary pressure exchanger for recovering expansion work in transcritical CO₂ supermarket racks under Kuwait's extreme heat. A reduced-order model based on constant-pressure displacement of the driven vapour is coupled with a booster model. High-side and receiver pressures are optimised over 0 to 52 °C, and hourly weather-station data from 2024 provide annual performance estimates for Kuwait International Airport and Ali Al Salem. The main conclusions follow:

- The displacement model predicts a hot-weather mass boost ratio of about 0.2. At 45 °C, its effective expansion and compression efficiencies are 66% and 68%, respectively, placing PX performance between an isenthalpic valve and an ideal expander-compressor.
- At 45 °C, the PX removes about 47% of the exergy destruction associated with high-pressure valve expansion. Combining it with parallel compression reduces total destruction from 85.7 kW in FGB and 67.6 kW in PC to 51.8 kW, while increasing COP to 1.66, or 21.0% above PC.
- The combined configuration maintains exergy efficiency at 30.3–35.0% from 25 to 50 °C, while reference-booster efficiency falls from 28.1% to 18.7%.
- At both Kuwaiti sites, PX + PC reduces annual electricity consumption by 11.7% relative to PC and 24.0% relative to FGB. It also lowers peak demand by up to 24.2% relative to PC and cuts the hours at the 120 bar high-side limit by 87–88%.
- PX alone cannot capture all the flash gas generated in extreme heat: its mass boost ratio saturates near 0.2, while the flash-gas fraction exceeds 0.5 at 45 °C. It is therefore best combined with a parallel compressor, which can be sized for a lower capacity.
- Volumetric effectiveness is the main determinant of savings; PX + PC still saves 9.2% relative to PC at $\lambda = 0.7$.

Future work should calibrate λ using measurements resolved by rotor speed, incorporate part-load compressor maps and adiabatic gas coolers into the annual model, and divide exergy destruction into avoidable and unavoidable parts. This would rank the remaining improvement potential of the PX against that of the gas cooler and compressors.

DECLARATION OF COMPETING INTEREST

The author declares no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

DATA AVAILABILITY

The weather data are openly available from the NOAA Integrated Surface Database. Model results are available from the corresponding author on reasonable request.

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