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RESEARCH ARTICLE

# Two-stage compression coupled with mechanical sub-cooling in trans-critical $CO_2$ refrigeration system development

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## Abstract

This study presents a platform that focuses on the improvement of the performance of trans-critical $CO_2$ refrigeration systems to face the big demand of economic and friendly cooling technologies, reduce environmental impact, and reduce operational costs. Four different refrigeration cycles were studied: abase cycle with 14 kW nominal capacity, a two-stage compression cycle, a cycle with R448a mechanical subcooling and a hybrid cycle that incorporates the two modifications. The Engineering Equation Solver (EES) software is used to analyze the different cycles under different operating parameters that includes gas cooling outlet temperature (35–50°C), evaporation temperatures (0–10 °C), compressor efficiencies (0.6–0.9), and gas cooler pressures (8,000–12,000 kPa). All modified cycles show more enhanced performance than base cycle. The hybrid cycle reached the highest coefficient of performance with an average enhancement of 55.3%, followed by mechanical sub-cooling cycle with (46.1%) average enhancement and two-stage compression cycle with (7%) average enhancement. The analysis showed that the hybrid cycle reduced power consumption by about 7.1% compared to mechanical sub-cooling cycle and by about 6% compared to the base cycle. Also, the mechanical sub-cooling showed an increase of power consumption of 13.2% over the base cycle. The increase in ambient temperature from 35 °C to 50 °C resulted in about 35.5% increase in power consumption and a 40.1% decrease in coefficient of performance (COP). This study is novel because it evaluates the combination of two-stage compression refrigeration system cycle and a mechanical sub-cooling cycle under the effect of Jordanian climate conditions, conditions, which resulted in achieving higher performance, lower environmental impact, shorter payback period, and higher return on investment (ROI).

**Keywords:** Refrigeration, two-stage compression, cooling, performance, hybrid systems

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## 1. Introduction

Carbon dioxide is a natural refrigerant extensively used in refrigeration and air conditioning because of its zero-ozone depletion potential (ODP) and minimal global warming potential (GWP) of 1[15]. It is abundant, cost-effective, non-toxic, non-flammable, and efficient at high pressures and temperatures [15]. Trans-critical $CO_2$ refrigeration cycles are particularly effective for modern refrigeration and air conditioning demands [6].

This study tests a novel hybrid cycle that combines two-stage compression and mechanical subcooling to enhance performance of trans-critical carbon dioxide cycles. The simulation study evaluates the effects of this combination on the cycle performance and power consumption compared to a base cycle under Jordanian climate conditions. Four different refrigeration cycles were studied: abase cycle with 14 kW nominal capacity, a two-stage compression cycle, a cycle with R448a mechanical subcooling and a hybrid cycle that incorporates the two modifications. The outputs and findings of this novel study can be used and applied to other countries that have the climates, which gives serious insights for advancing sustainable refrigeration technologies worldwide.

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Many studies found in literature highlight the increasing interest in perfecting the performance of trans-critical carbon dioxide cycles through such modified advanced hybrid techniques.

Yuyao Sun et al. [1] conducted a simulation study to perfect the performance of the trans-critical $CO_2$ two-stage compression refrigeration system when changing the gas cooler outlet temperature in the range of (30 °C-12 °C). They concluded that, the implementation of the auxiliary gas cooler enhances the efficiency of the trans-critical $CO_2$ two-stage compression refrigeration system.

Shengchun Liu and colleagues [2] conducted a comparative analysis of the thermodynamic performance of six $CO_2$ trans-critical two-stage refrigeration systems incorporating expanders. They showed that, under specified conditions, a two-stage expander with partial intercooling leads to significant advantages in system performance and equipment choice when working under high-pressure stage.

Ye Liu et al. [3] presented a two-stage compression trans-critical $CO_2$ dual-evaporator refrigeration cycle that uses an ejector. Analysis of the system studied showed that it performs better than the conventional system using all operating conditions. Specifically, under normal operating conditions, the presented system reached a 19.6% and 15.9% enhancement in COP and exergy efficiency compared to the conventional system.

Benlin Shi et al. [4] Proposed a novel $CO_2$ trans-critical refrigeration system that integrates an internal heat exchanger, expander, and two-stage compression cycle. The researchers evaluated the effect of using the expander, internal heat exchanger, and intercooling on cycle performance through different operating conditions, focusing on energy and exergy analyses based on the effect of best discharge pressure. In comparison to the single-stage compression cycle using an expander, the studied system showed a 2–15.7% decrease in cycle power consumption and a 2.93–6.93% improvement in COP.

Paride Gullo et al. [5] analyzed a $CO_2$-based supermarket refrigeration cycle that uses R290 mechanical subcooling. The researcher conducted advanced exergy analysis and performance enhancements for the main cycle components like compressors and heat exchangers.

Evangelos Bellos et al. [6] compared different trans-critical $CO_2$ refrigeration cycles to predict most efficient layouts. The base cycle which incorporates (evaporator, compressor, gas cooler, expansion valve) is compared with cycles incorporating internal heat exchangers, parallel compression, two-stage compression, and mechanical subcooling. Their analysis based on evaporation temperatures ranged from −35 °C to +5 °C for refrigeration and condensing temperatures ranged from 35 °C to 50 °C for the condenser. They concluded that the mechanical subcooling and two-stage compression cycles produced the highest performance, with COP enhancements which reached about 75.8% and 49.8%, respectively.

Yuangang Lu et al. [7] studied the enhancing $CO_2$ cycle performance by using a second subcooling system. They found that the coefficient of performance exceeds that of the base cycle and the use of subcooling under trans-critical conditions is very useful in enhancing the cycle performance and power saving.

Alberto Cavalliniet al. [8] studied experimentally and theoretically, the potential for enhancing refrigeration cycle performance through using two-stage compression with inter cooling under operating conditions that are the same used in air conditioning cycles. The findings confirm that, in trans-critical operation, there is a best upper cycle pressure that maximizes energy efficiency.

Daniel Sánchez et al. [9] conducted an experimental test on the performance of a carbon dioxide trans critical refrigeration cycle integrated with dedicated mechanical subcooling cycle that uses R600a as refrigerant under different operating conditions. They compared their results with those obtained from the refrigeration cycle that uses a suction-to-liquid internal heat exchanger to assess the performance enhancements obtained with the mechanical subcooling cycle.

Mohammad Tarawneh et al. [10] studied the performance enhancement of a trans-critical $CO_2$ refrigeration cycle that uses a porous internal heat exchanger. Author showed that there are considerable enhancements in refrigeration capacity and COP. The use of porous internal heat exchanger resulted in improved heat transfer performance, reduced energy consumption. This study shows the importance of using porous heat exchangers in perfecting $CO_2$ refrigeration cycle performance.

Yang Li et al. [11] offers a trans-critical $CO_2$ two-stage compression system incorporating a two-phase ejector for maritime cold storage units. Simulations were performed using EES to analyze the effects of exhaust pressure, intermediate pressure and evaporation temperature. The results showed that the coefficient of performance (COP) increased with the increase of the evaporation temperature from -35°C to -25°C but decreased with the increase of the exhaust pressure at -35°C. The best intermediate pressure was found to be 4.75 MPa at -35°C.

Evangelos Bellos et al. [12] conducted their studies under varied operating conditions and proved that the employment of an absorption chiller can result in a decrease of 54% in electricity consumption. Mechanical sub-cooling also results in a 41% reduction in energy use compared to the conventional cycle. Although the mechanical subcooling has better exergy efficiency, the overall exergy performance of the system is enhanced by both methods.

Rodrigo Llopis et al. [13] conducted an analytical study on the performance of carbon dioxide trans critical cycle that uses mechanical subcooling. Author used simplified models and evaluated the effect of the propane integrated mechanical subcooling on the operating conditions at evaporating temperatures of (5, -5, and -30 °C) and ambient temperatures ranged from (20 to 35 °C). They concluded that the cycle COP increases up to 20% and the cycle refrigeration capacity increases up to about 28.8%, with higher gains at higher temperatures. Authors showed that the modified cycle is more efficient for ambient temperatures above 25 °C

Ahmad Bani Yaseen et al. [14] studied the effect of compressor performance and vapor quality on the $CO_2$ trans-critical air conditioning cycle. Utilizing EES and MATLAB, corroborated by experimental data, results showed a 66% enhancement in cooling capacity when gas-cooling pressure increased from 100 to 150 bar. The cooling capacity decreased by 52.4% when the superheat was reduced from 12°C to 0°C and the vapor quality increased from 0.1 to 0.5, while the coefficient of performance (COP) increased by 87%. The compressor's power was reduced by 50%, while its efficiency increased from 70% to 100%, and the gas-cooling pressure lowered from 110 to 80 bar.

Although of huge research on the performance of trans-critical $CO_2$ refrigeration cycles, there is still a gap that persist in studying the effect of combining two-stage compression and mechanical subcooling within the same cycle on the overall cycle performance. The present study fills this gap by simulating trans-critical $CO_2$ cycles for a wide range of different parameters, including ambient air temperature, compressor efficiency, evaporating temperature, gas cooling outlet temperature, and pressure. The novelty of this study comes from the novel combination of two-stage compression and mechanical subcooling in single cycle to enhance its performance. This research study introduces comprehensive analysis, which offers novel insights into enhancing refrigeration performance, reducing power consumption and reducing environmental impacts. All these results and findings can help in advancing the sustainable development of trans-critical $CO_2$ refrigeration systems.

## 2. Methodology

The method of this study can be presented by using the following steps and procedures:

### 2.1. Methodology overview

- Background: The trans-critical $CO_2$ refrigeration cycles have lower performance when using single-stage compression systems and high operating pressures. Coefficient of performance and refrigeration capacity can be increased by using mechanical sub-cooling and two-stage compression.
- Purpose: This research evaluates the thermodynamic modified cycle performance, presents model validation, and conduct economic evaluation of the modified cycle that forms a two-stage compressor with mechanical sub-cooling in a trans-critical $CO_2$ cycle.

### 2.2. Literature review

- Trans-Critical $CO_2$ Cycles: Carbon dioxide can be considered as low impact refrigerant when compared to other refrigerants is a low-impact refrigerant, and its performance is easily affected by the variations of pressure and ambient conditions.
- Two-Stage Compression: Most of the studies in literature prove that the use of two-stage compressors enhances cycle efficiency by decreasing power consumption and increasing coefficient of performance.
- Mechanical Sub-Cooling: The sub-cooling technique can improve the cycle refrigeration capacity and COP in in trans critical $CO_2$ systems.
- Integration of Techniques: The combination of two-stage compression with mechanical sub-cooling in trans-critical carbon dioxide refrigeration cycles can be considered as an area with little literature research, accordingly, this study comes to fill this gap in literature.

### 2.3. Study overview

This simulation study is carried out to evaluate the trans-critical $CO_2$ cycle thermodynamic performance supported with model validation and economic viability when a two-stage compression system is integrated with mechanical sub-cooling.

### 2.4. Thermodynamic cycle and system design

A comprehensive thermodynamic cycle is presented with integrated two-stage compressor and mechanical sub-cooler.

- P-h Diagrams: These diagrams show the processes of compression, expansion and cooling in the trans critical carbon dioxide cycle and the mechanical subcooling cycle.
- Schematic Diagrams: This diagram shows full representation of the studied cycle with all its components.

### 2.5. Modeling approach

The refrigeration cycles are modeled by using the thermodynamic first and second laws.

### 2.6. Key assumptions

The main assumptions that are used during this simulation study include steady-state operation and neglecting heat losses with focusing on the main parameters like pressure, temperature, enthalpy, and COP.

### 2.7. Simulation setup

- The cycles are modeled and simulated utilizing Engineering Equation Solver software [EES].

- Input Parameters: The thermophysical and thermodynamic properties of $CO_2$ are obtained from EES libraries, and the boundary conditions which include temperatures, pressures, and flow rates are put based on the standard refrigeration conditions.
- Output Metrics: The Coefficient of Performance (COP) and power consumption are evaluated for different cycle configurations.

## 2.8. Model validation

Model The accuracy of the used model is confirmed by comparing selected results of this study with that of similar systems from literature.

## 2.9. Economic evaluation

Operational Costs: The calculations of power consumption are on simulated performance of the system studied.

Payback Period and ROI: Payback period and ROI calculations. Are used for the evaluation of the economic feasibility of the studied cycles.

## 2.10. Limitations

The main limitations of this study are:

- The assumption of steady state operation
- Neglecting the transient effects
- The study is based on specific configurations, so the results may vary with different setups or different refrigerants.

## 2.11. Constraints

Compression Pressure Range: limited pressure range throughout the study.

Operating Temperature Limits: To achieve reliable cycle operation and to get efficient heat transfer, the operating temperatures at the different cycle states must be constrained in specified operational limits.

Mass Flow Rate of Refrigerants: The mass flow rates of the main $CO_2$ refrigerant in the trans critical and sub-cooling cycle should be consistent with the cycle's refrigeration capacity.

Thermodynamic Consistency: To keep energy balance, enthalpy, pressure, and temperature must be aligned within the expected states to keep energy balance.

Steady state assumption is considered throughout the analysis of this study.

## 2.12. Thermodynamic cycles

The schematic diagram of the studied cycle is shown in Figure 1. The top cycle uses R448A as the refrigerant, while the bottom, trans-critical cycle uses carbon dioxide. Two- stage compression with an intercooler is incorporated into the trans-critical carbon dioxide system, which is modeled with a nominal cooling capacity of 4 tons (14 kW). A heat exchanger links the two cycles and sub-cools the refrigerant exiting the gas cooler. The upper cycle functions as a mechanical sub-cooling system. A thorough thermodynamic description of the main cycle's pressure-enthalpy (P-h) diagram is shown in Figure 2. The P-h diagram of the mechanical sub-cooling cycle is shown in Figure 3. The thermophysical properties of the tested refrigerants are presented in Table 1, showing the refrigerants' low global warming potential (GWP) and zero ozone depletion potential (ODP).

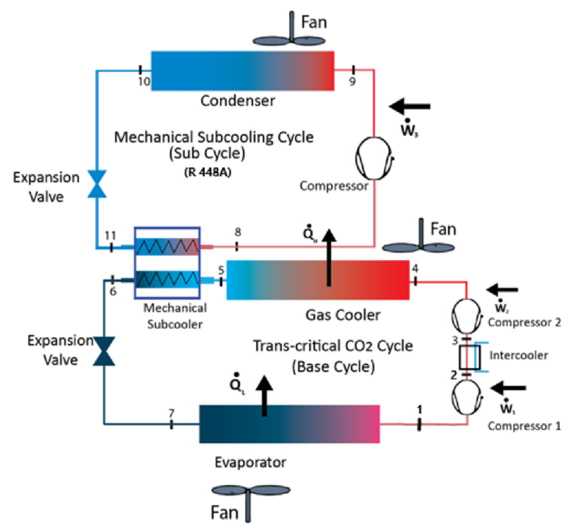


**Figure 1**. Schematic representation of the refrigeration system

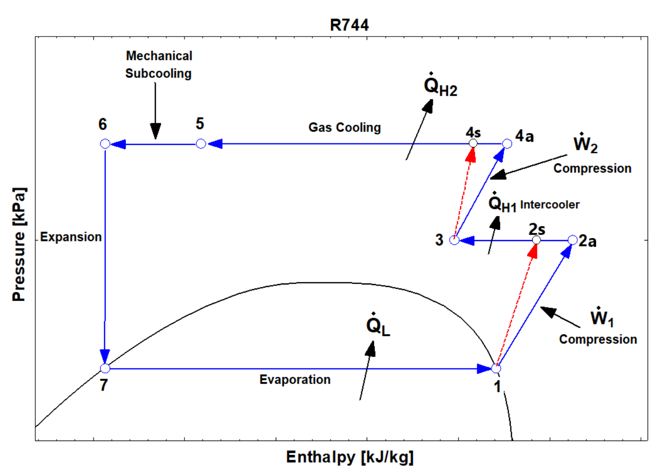


**Figure 2.** P-h diagram of a two-stage compression trans-critical cycle

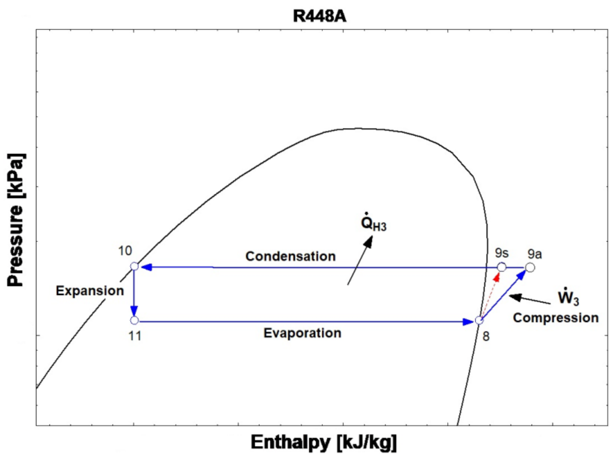


**Figure 3**. P-h diagram of the mechanical sub-cooling cycle

**Table 1.** Thermophysical properties of the tested refrigerants [15]

| ASHRAE Classification | ODP | GWP | Critical Pressure/bar | Critical Temperature/°C | Boiling Point/°C |
|---|---|---|---|---|---|
| R744 ($CO_2$) | 0 | 1 | 73 | 31.0 | – 56.6 at Pressure of (5.2 bars) |
| R448A | 0 | 1360 | 45.945 | 82.7 | -46.1 at Pressure of (1 bars) |

## 3. Modeling

The trans-critical cycle is modeled using the laws of thermodynamics. Several hypotheses are typically made to simplify the analysis:

- Steady-State Operation.
- Isentropic Compressor Efficiency.
- Negligible Pressure Drops.
- Thermal Equilibrium in Heat Exchangers.
- Perfect Heat Transfer in Subcooling System.
- Negligible Superheating.
- Constant ambient conditions for each individual run.

The following set of equations are derived with reference to Figures 1,2 and 3. The actual specific enthalpies of the superheated refrigerant in (kJ $kg^{-1}$) at the exit of the first and second stages of compression processes of the trans-critical cycle are shown in equations 1 and 2, respectively [16].

$$h_{2a} = \frac{h_{2s} - h_1}{\eta_{Comp1}} + h_1 \tag{1}$$

$$h_{4a} = \frac{h_{4s} - h_3}{\eta_{Comp2}} + h_3 \tag{2}$$

Where, $h_1$ and $h_{2s}$ are the enthalpies in (kJ $kg^{-1}$) at the inlet and exit of the first stage isentropic compression process in the trans-critical cycle. $h_3$ and $h_{4s}$ are the enthalpies in (kJ $kg^{-1}$) at the inlet and exit of the second stage isentropic compression process in the trans-critical cycle. $\eta_{Comp1}$ and $\eta_{Comp2}$ are the isentropic efficiences of the first and second stage copmpression processes in the transcritcal cycle. The actual specific enthalpy of the superheated refrigerant in (kJ $kg^{-1}$) at the exit of the compression process of the mechanical sub-cooling cycle is shown in equation 3 [16].

$$h_{9a} = \frac{h_{9s} - h_8}{\eta_{Comp-Mech}} + h_8 \tag{3}$$

Where, $h_8$ and $h_{9s}$ are the enthalpies in (kJ $kg^{-1}$)at the inlet and exit of the isentropic compression process in the mechanical sub-cooling cycle and $\eta_{Comp-Mech}$ is the isentropic efficiency of the compression process in the mechanical sub-cooling cycle .

Equation (4) [16] is used to calculate the mass flow rate of the refrigerant in mechanical sub-cooling cycle ($\dot{m}_{mech}$) in (kg $s^{-1}$).

$$\dot{m}_{base}(h_6 - h_5) = \dot{m}_{mech}(h_8 - h_{11}) \tag{4}$$

Where, $h_5$ and $h_6$ are the enthalpies in (kJ $kg^{-1}$) at the inlet and exit of the mechanical sub-cooling process in the trans-critical cycle whereas, $h_8$ and $h_{11}$ are the enthalpies in (kJ $kg^{-1}$) at the inlet and exit of the evaporation process in the mechanical sub-cooling cycle, $\dot{m}_{base}$ is the mass flow rate in (kg $s^{-1}$) of the refrigerant in the base trans-critical cycle.

The power consumption of the first stage compression process ($\dot{W}_1$) and the second stage compression process ($\dot{W}_2$) and the power consumption of compression process of the mechanical sub-cooling cycle ($\dot{W}_3$) are calculated in (kW) as per equations 5,6 and 7 respectively.

$$\dot{W}_1 = \dot{m}_{base}(h_{2a} - h_1) \tag{5}$$

$$\dot{W}_2 = \dot{m}_{base}(h_{4a} - h_3) \tag{6}$$

$$\dot{W}_3 = \dot{m}_{mech}(h_{9a} - h_8) \tag{7}$$

The power consumption ($\dot{W}_{net}$) in (kW) of the basic cycle, two-stage cycle, mechanical cycle and combined cycle can be calculated according to equation 8,9,10 and 11, respectively [16].

$$\dot{W}_{net,basic} = \dot{W}_1 \tag{8}$$

$$\dot{W}_{net,multi} = \dot{W}_1 + \dot{W}_2 \tag{9}$$

$$\dot{W}_{net,mech} = \dot{W}_1 + \dot{W}_3 \tag{10}$$

$$\dot{W}_{net,combined} = \dot{W}_1 + \dot{W}_2 + \dot{W}_3 \tag{11}$$

The heat rejection in (kW) from the intercooler in the two-stage compression trans-critical cycle, the gas cooler in basic trans-critical cycle and the condenser in the mechanical sub-cooling cycle are calculated as per equations 12, 13 and 14, respectively [16].

$$\dot{Q}_{H1} = \dot{m}_{basic}(h_{2a} - h_3) \quad (12)$$

$$\dot{Q}_{H2} = \dot{m}_{basic}(h_{4a} - h_5) \quad (13)$$

$$\dot{Q}_{H3} = \dot{m}_{mech}(h_{9a} - h_{10}) \quad (14)$$

Where, $h_{10}$ is the enthalpy in (kJ $kg^{-1}$) at the exit of the condenser of the mechanical sub-cooling cycle. The heat rejection in ($\dot{Q}_H$) in (kW) of the basic cycle, two-stage cycle, mechanical cycle and combined cycle can be calculated according to equation 15,16,17 and 18, respectively [16].

$$\dot{Q}_{H,basic} = \dot{Q}_{H2} \quad (15)$$

$$\dot{Q}_{H,multi} = \dot{Q}_{H1} + \dot{Q}_{H2} \quad (16)$$

$$\dot{Q}_{H,mech} = \dot{Q}_{H1} + \dot{Q}_{H3} \quad (17)$$

$$\dot{Q}_{H,combined} = \dot{Q}_{H1} + \dot{Q}_{H2} + \dot{Q}_{H3} \quad (18)$$

The refrigeration capacity ($\dot{Q}_L$) in (kW) of the trans-critical basic cycle can be calculated according to the following equation (19) [16]:

$$\dot{Q}_L = \dot{m}_{basic}(h_1 - h_7) \quad (19)$$

Where, $h_7$ is the enthalpy of the refrigerant in (kJ $kg^{-1}$) at inlet of the evaporator in the trans-critical cycle. The coefficient of performance of the trans-critical cycle can then be calculated as per equation (20) [16]:

$$COP = \frac{\dot{Q}_L}{\dot{W}_{net}} \quad (20)$$

Table 2 shows the considered ranges of operating parameters and boundary conditions of the studied refrigeration cycles with short explanation of the justifications of the selected values.

**Table 2. Range of the boundary conditions with justifications**

| Parameter | Range/Value | Justifications | Constant |
|---|---|---|---|
| Gas cooler outlet temperature, $T_{gc}$ in (°C) | 35-50 | The range Represents typical gas cooler temperatures depending on the expected heat rejection and ambient conditions | 35 |
| Evaporation temperature, $T_{evap}$ in (°C) | 0-10 | Typical evaporation temperatures in for refrigeration cycles. | 5 |
| Gas cooler pressure, $P_{gc}$ in (kPa) | 8000-12000 | Typical operating pressure for the gas cooler in $CO_2$ trans-critical systems to keep $CO_2$ refrigerant in the supercritical state. | 8500 |
| Compressor efficiency ($\eta_{comp}$) | 0.6-0.9 | Typical for the compressors used in modern refrigeration cycles | 0.85 |
| Ambient temperature, $T_{Ambiient}$ in (°C) | 30-45 | Suitable for hot climate conditions | 30 |
| Mass flow rate(kg/s) (Base cycle) | 0.06-0.12 | Ensures efficient heat transfer, aligns with system capacity (14 kW), matches compressor design, and supports $CO_2$'s thermodynamic properties for stable, energy-efficient operation. | 0.12 |
| Subcooling Temperature (°C) | 10 | To avoid flash gas formation in the expansion valve. | 10 |
| Intercooling Temperature (°C) | 5-20 | Important for the optimization of compressor efficiency and cycle stability during the reducing of compressor input work. | 10 |

## 4. Result and discussion

In this performance study four thermodynamic cycles are examined: the basic trans-critical cycle, two-stage compression, mechanical sub-cooling, and a combined cycle integrating both. Simulations, conducted using EES software [17], compare these cycles to decide the highest COP and lowest net power consumption. Figures 4 and 5 illustrate the fluctuations in the coefficient of performance (COP) and net power consumption ($\dot{W}_{net}$) of trans-critical carbon dioxide cycles with respect to gas cooler temperature. Utilizing the designated parameters ($T_{evap}$=5 ,$\eta_{comp}$=0.85). Figure 4 illustrates that increasing the gas cooler temperature reduces the COP due to higher heat rejection losses and reduced system efficiency. COP enhancements of about 45.6%, 54.9%, and 7% are achieved over that of the base cycle when the mechanical sub-cooling cycle, combined cycle, and multistage cycle are used, respectively. This is because of the better heat rejection and lower compression work when using the combined cycles. Figure 5 shows the effect of the variation of gas cooler outlet temperature on the compressor power consumption and cycle performance. It is very clear from this figure that power consumption increases as the gas cooler outlet temperature increases because the compressor needs more power to overcome the increase in heat rejection and to keep cycle performance.

It can be concluded from Figure 5 that when using two-stage compression cycle, the compressor power consumption can be reduced by about 6.5% compared to the base cycle. The reason for that is the enhanced distribution of compression power and reduced power consumption during each compression stage. On the other hand, the mechanical sub-cooling cycle and the combined cycle show increases in compressor power consumption by about 12.5% and 7.8%, respectively. It can be noted from Figure 5 that decreasing the gas cooler outlet temperature from 50°C to 35°C resulted in a significant decrease in compressor power consumption by an average value of about 35.5%, because of running the cycle under low thermal load. This reduction in power consumption resulted in enhancing the cycle coefficient of performance by an average value of about 40.1%.

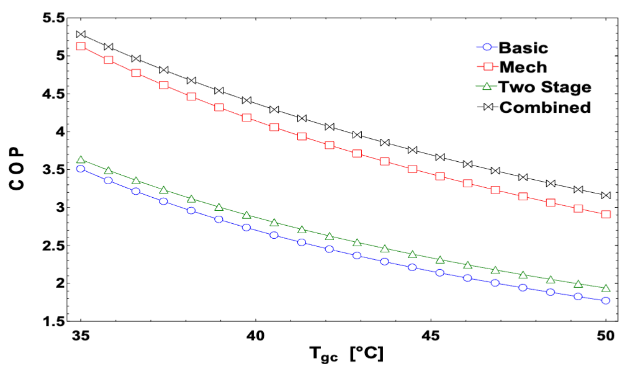


**Figure 4.** Variation of COP with gas cooler outlet temperature

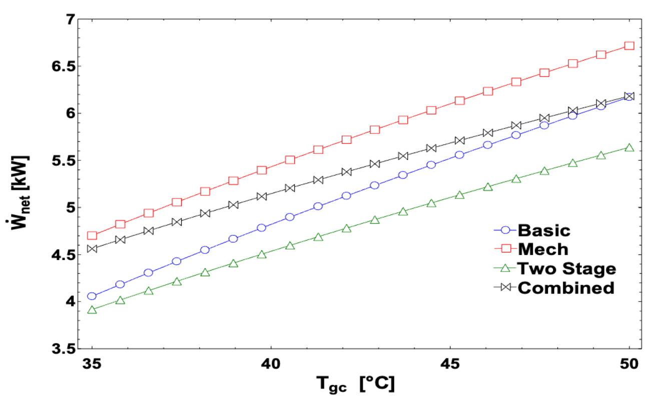


**Figure 5.** Variation of power consumption with gas cooler outlet temperature

The variations of the coefficient of performance (COP) and the net power consumption ($\dot{W}_{net}$) of trans-critical carbon dioxide cycles with evaporator temperature for the different cycle configurations are shown in Figures 6 and 7, respectively. Under the specified conditions ($T_{gc}$=35 °C, $\eta_{comp}$=0.85), $P_{gc}$= 8500 kPa), Figure 6 shows that the increase in evaporator temperature resulted in a considerable enhancement of COP because of decreasing the compressor power consumption and enhancing the cycle thermodynamic efficiency. An enhancement of about 55.3% compared to the basic cycle is achieved when using the combined cycle because of the considerable benefits from the use of optimized subcooling and compression processes. The mechanical sub-cooling and two-stage compression cycles also achieved significant COP increases of about 45.9% and 7 %, respectively, because of improved heat exchange and reduced power consumption. Figure 7 shows that net power consumption is decreased by increasing the evaporation temperature. The two-stage compression cycle reaches the lowest power consumption of about 3.69% decrease below the base cycle, because of the enhanced compression efficiency. On the other hand, the mechanical sub-cooling cycle reaches the highest power consumption of about 16% above the basic cycle, because of extra power demands of the sub-cooling process. It can be concluded from Figure 7 that increasing the evaporator temperature in the range from 0°C to 10°C results in an average percentage enhancement of about 44% in COP and an average percentage decrease of about 33% in the net power consumption. An enhancement in COP of about 49.8% was achieved by [6] and an average percent reduction in power consumption of about 41% when using mechanical subcooling is achieved by [12].

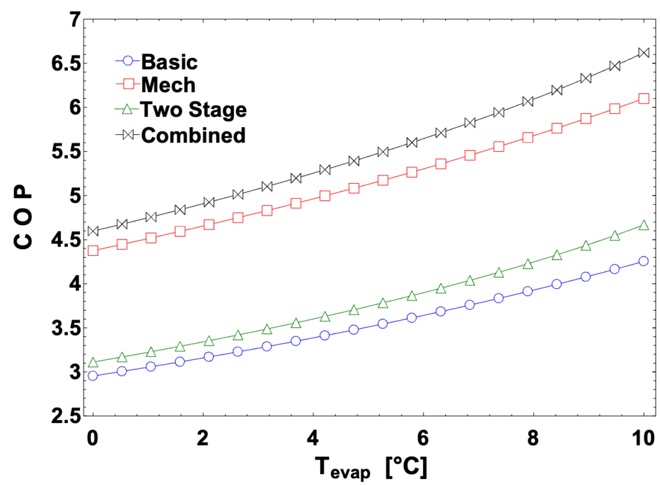


**Figure 6.** Variation of COP with evaporation temperature

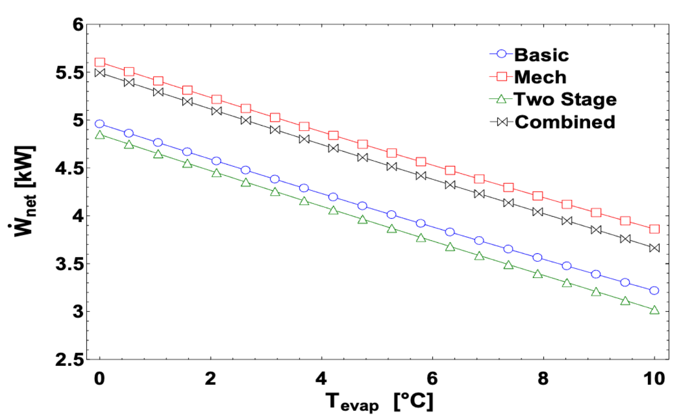


**Figure 7**. Variation of power consumption with evaporation temperature

Figures 8 and 9 depict the variations in the coefficient of performance (COP) and net power consumption ($\dot{W}_{net}$) for trans-critical carbon dioxide cycles as functions of gas cooler outlet pressure within the range of 8,000–120,000 kPa. Figure 8 shows the impact of gas cooler pressure on COP under the specified conditions ($T_{evap}$ =5°C, $T_{gc}$=35°C, $\eta_{comp}$=0.85). The cycle thermodynamic efficiency is affected by the gas cooler outlet pressure. Higher gas cooler pressures enhance heat rejection, resulting in more effective cooling and good use of the refrigerant properties. The combined cycle achieves an average percentage enhancement in COP by about 52.6% over that of the basic cycle, because the integration of two-stage compression and mechanical sub-cooling perfects both heat rejection and compression efficiency. A percentage increase in COP of about 44.4% is noticed when the mechanical sub-cooling cycle is compared with the base cycle. The use of two-stage compression cycle resulted in 6.4% enhancement in COP because of the reduced compression work. The resulting findings confirm the importance of perfecting gas cooler outlet pressure in enhancing cycle performance.

Figure 9 shows that increasing gas cooler pressure increases net power consumption due to higher compressor workload. Despite this, the combined cycle reduces power consumption by 7.1% compared to the mechanical sub-cooling cycle, and the two-stage compression cycle achieves a 6% reduction compared to the basic cycle. However, the mechanical sub-cooling cycle increases power consumption by 13.2% compared to the base cycle. Across the pressure range studied, power consumption increases by an average of about 46.8%, while COP improves by about 13.5%, highlighting a trade-off between energy use and efficiency gains.

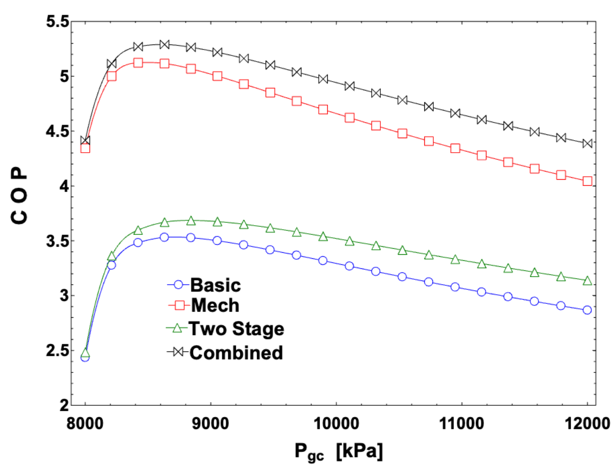


**Figure 8**. Variation of COP with gas cooler pressure

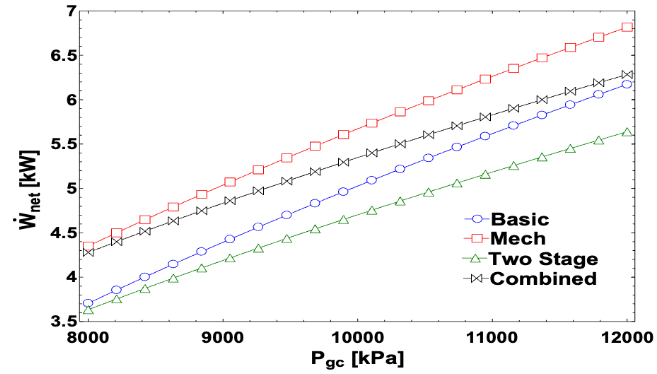


**Figure 9.** Variation of power consumption with gas cooler pressure

Figures 10 and 11 illustrate the variations in the coefficient of performance (COP) and power consumption across trans-critical carbon dioxide cycles considering compressor efficiencies for the different configurations ranging from 60% to 90%. Under the specified conditions ($T_{evap}$=5°C,$P_{gc}$=8500kpa, $T_{gc}$=35°C ) Figures 10 illustrate that increasing compressor efficiency significantly boosts COP by reducing energy losses during compression. The combined cycle reaches the highest enhancement, with a about 54.13% increase in COP compared to the base cycle, because of the addition of the optimized advanced processes. The mechanical sub-cooling cycle shows an average increase of about 46.1% because of the enhanced refrigeration capacity. On the other hand, the two-stage compression cycle shows the smallest increase in COP of about 6%. It can be noted from Figure 11 that increasing compressor efficiency results in notable decrease in power consumption, because the refrigerant needs less energy to be compressed. The combined cycle reaches a 13.1% increase in power consumption compared to base cycle. This is because of the addition of the advanced systems that require added energy. The two-stage compression cycle shows an average percentage decrease of about 2.7% in power consumption because of the more efficient compression work. On the other hand, the mechanical sub-cooling cycle increases power consumption by about 15.8%, because of the need for more energy for the added cooling system. Throughout all cycles, increasing compressor efficiency resulted in a substantial average increase in COP of about 52.9% and considerable reduction in power consumption of about 34.5%. This means that the increase in compressor efficiency enhances performance and helps in reducing overall energy needs and power consumption, especially for refrigeration systems that undergo optimized compression stages and better heat management.

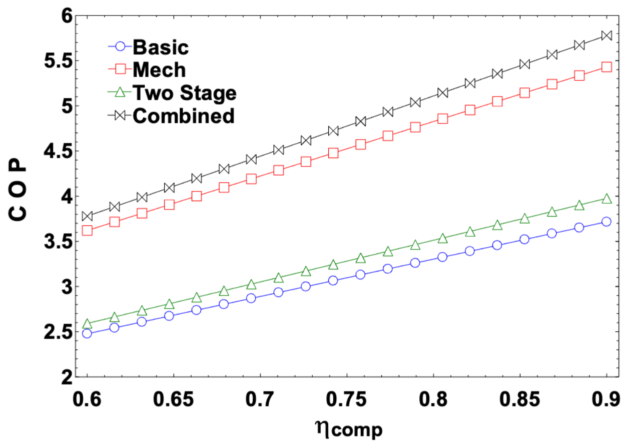


**Figure 10.** Variation of COP with compressor efficiency

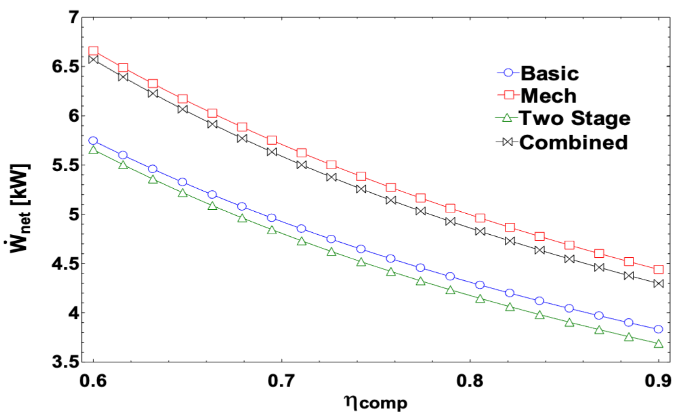


**Figure 11**. Variation of power consumption with compressor efficiency

Figures 12 and 13 depict the variations in COP and power consumption across the studied trans-critical carbon dioxide cycles within an ambient temperature range of 30°C to 45°C. Under the specified conditions ($T_{evap}$=5°C, $\eta_{comp}$=0.85), Figure 12 proves that as ambient temperature increases, the COP of the refrigeration system decreases because of higher temperature difference between the refrigerant and the surroundings which makes it harder for the system to reject heat and keep efficiency. The combined cycle reaches the highest COP enhancement of about 54.9% over the base cycle, because of the efficient integration of two-stage compression and mechanical sub-cooling, which helps keep performance under higher ambient temperatures. The two-stage compression and mechanical sub-cooling cycles also show enhancements of about 7% and 45.6%, respectively because of the optimized compression stages and enhanced heat exchange processes. These results show the importance of the use of advanced cycle configurations in controlling the impact of increasing ambient temperatures on the refrigeration system performance. Figure 13 implies that the cycle power consumption is increased at higher ambient temperatures as the system works harder to reject heat. The combined cycle shows a power consumption increase of about 5.9% compared to the basic cycle, while the two-stage compression cycle shows a reduction in power consumption by about 6.5%, because of more efficient compression process. The mechanical sub-cooling cycle achieves a power consumption increase of about 12.5% compared to basic cycle, because of its added complexity. Overall, the results show a 35.5% increase in power consumption and a 40.1% decrease in COP as ambient temperature increases, showing the challenges of keeping efficiency under higher temperatures. An increase in COP of up to 20% was reported by [13] when the ambient temperature decreased from 35 °C to 30 °C in a mechanical subcooling cycle, while the present study achieved a COP increase of approximately 18.6% under the same conditions.

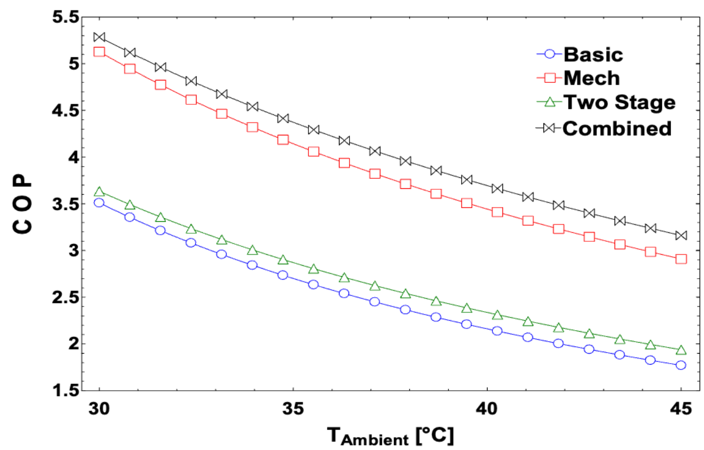


**Figure 12**. Variation of COP with ambient temperature

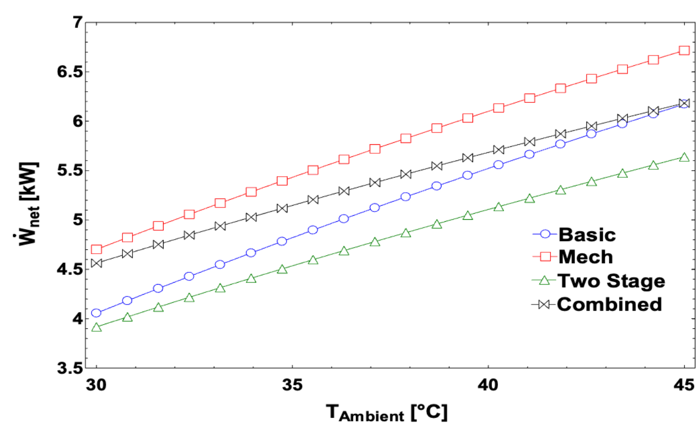


**Figure 13.** Variation of power consumption with ambient temperature

Figures 14 and 15 present the impact of intercooling between two stage compression processes at various gas cooler pressures on COP and power consumption. Under the specified conditions ($T_{evap}$=5°C, $\eta_{comp}$=0.85, Tgc=35°C), Figure 14 shows that increasing the degree of intercooling and decreasing the gas cooler pressure resulted in COP enhancement by enhancing the cycle efficiency. Reducing the gas cooler pressure from 12,500 kPa to 8,500 kPa resulted in an average percentage increase in COP of about 6.3%, because lower pressure leads to better heat rejection and reduces the power consumption of the compressor, thus improving overall coefficient of performance of the cycle. Figure 15 emphasizes that increasing the degree of intercooling between the two compression stages reduces power consumption by an average percentage of about 6.8%. Additionally, raising the intercooler temperature within the specified range results in a 10.5% increase in COP and a 9.5% decrease in power consumption, as it enhances system efficiency by reducing the compressor work and perfecting heat transfer between stages. A 2.93–6.93% improvement in COP is achieved by [4].

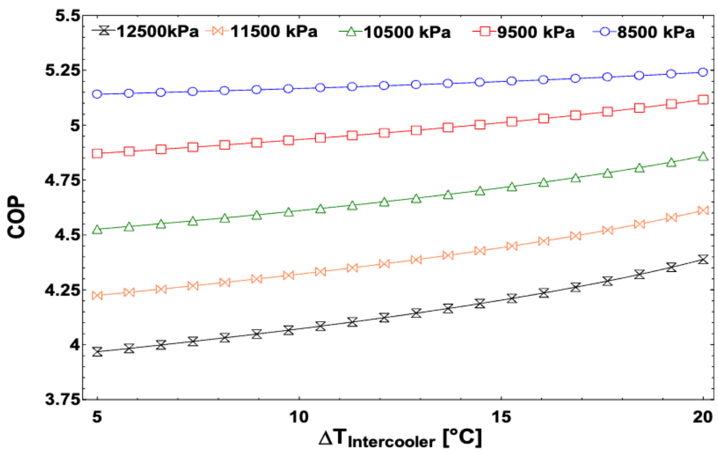


**Figure 14**. Variation of COP with $\Delta T_{intercooler}$

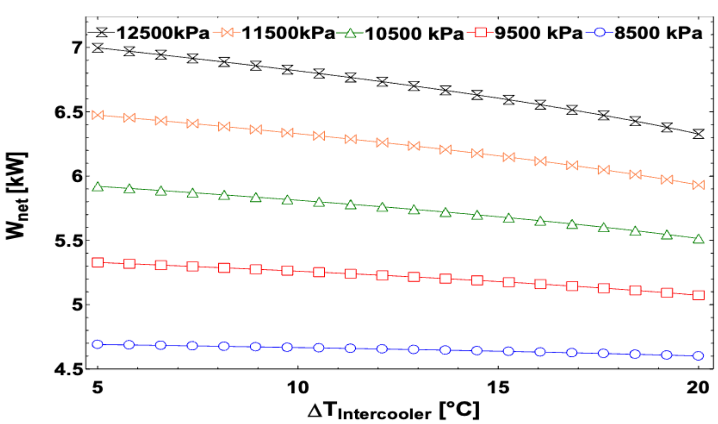


**Figure 15**. Variation of power consumption with $\Delta T_{intercooler}$

## 5. Economic study

This section presents an economic analysis of a modified 14-kW refrigeration system that is used to improve energy efficiency under Jordan operating conditions [18,19]. The analysis compares the performance and cost-effectiveness of a base cycle with that of a modified cycle. Through the examining of power consumption, maintenance costs, and potential financial savings, the study focusses on evaluation of the feasibility of the added modifications in terms of the reduction of operational cost and on the return on investment. The results and findings of this analysis provide valuable insights for decision-makers who seek to make balance between the first investment and long-term economic and environmental benefits.

### 5.1. Assumptions and variables

Refrigeration Capacity: 14 kW for base cycle

COP: Calculated in the study.

Base System: Varies from 2.5 to 3.5.

Modified System: Varies from 3.5 to 5.6.

Annual Maintenance Cost:

Base System: $150/year.

Modified System: $200/year.

Operation Hours per Year: 2,000 hours (standard assumption for industrial use).

Electricity Cost: The cost of electricity in Jordan is used to calculate energy consumption. The average price is typically around $0.15 per kWh.

Discount Rate: Considering future costs and savings, a discount rate of 5% per year may be applied (standard for financial analysis).

Systems first cost:

Modified system first cost: $12,000

Base system first cost: $ 10,000

Operational lifespan for both systems: 10 years

The annual energy consumed by each system can be calculated as per equation (21) [18,19]:

Energy Consumption (kWh)= (Refrigeration Power (kW)/COP) × Operating Hours

(21)

The annual energy cost can be calculated according to equation (22) [18,19]:

The annual energy cost=energy consumption (kWh)×cost per kWh (22)

The total annual cost is calculated as per equation (23) [18,19]:

Total annual cost = energy cost+ maintenance cost (23)

The first investment difference between the two systems can be calculated as per equation (24) [18,19]:

Initial investment difference= Modified system first cost - Base system first cost

(24)

The payback period, which is the time it takes for the annual savings to cover the first investment difference is calculated by using equation (25) [18,19]:

Payback Period= First Investment Difference/ Annual Savings

(25)

The Return on Investment (ROI) is the return which will earn from the modified system compared to the first investment. ROI can be calculated by using equation (26) [19]:

ROI= (Annual Savings /Initial Investment Difference)×100

(26)

Depending on the assumptions above and using equations (21–26), The main economic parameters, including energy consumption, annual operating costs, maintenance expenses, payback period, and return on investment, have been calculated. The analysis based on the variations in COP, in the ranges from 2.5 to 3.5 for the basic cycle and from 3.5 to 5.6 for the modified cycle. Comparative analysis of the two cycles is shown in Table 3.

**Table 3.** Results of economic analysis

| S. N | Parameter | Base Cycle COP | | Modified Cycle COP | |
|---|---|---|---|---|---|
| | | 2.5 | 3.5 | 3.5 | 5.6 |
| 1 | Energy Consumption (kWh) | 11200 | 8000 | 8000 | 5000 |
| 2 | Energy cost (USD) | 1680 | 1200 | 1200 | 750 |
| 3 | Total Annual Cost (USD) | 1830 | 1350 | 1400 | 950 |
| 4 | Annual saving (USD) | For COP = 2.5 (Base System) vs COP = 3.5 (Modified System): 430<br>For COP = 3.5 (Base System) vs COP = 5.6 (Modified System): 400 | | | |
| 5 | Payback Period (Year) | For COP = 3.5 (Base) vs COP = 3.5 (Modified): 4.65<br>For COP = 3.5 (Base) vs COP = 5.6 (Modified): 5 | | | |
| 6 | Return on Investment (ROI) | For COP 3.5 (Base) vs COP 3.5 (Modified): 21.5%<br>For COP 3.5 (Base) vs COP 5.6 (Modified): 20% | | | |

Table 3 illustrates that the first case proves a higher ROI (COP 3.5 compared to COP 3.5), showing that the investment is expected to yield returns more swiftly. A 20–21% annual return on investment (ROI) for energy-saving systems is viewed as helpful, despite the system's lifespan being limited to only 10 years. The modified system continues to generate revenue beyond the payback period, showing its sustained profitability.

The duration of the payback period ranges from 4.65 to 5 years, as illustrated in Table 3. Subsequently, the accumulated savings on energy expenses amount to approximately $400 to $430 annually. The modified system is projected to yield significant cost savings over its ten-year operational lifespan. The reduction in energy costs leads to enhanced financial stability and better forecasting, positioning it as a wise long-term investment.

The improved system's increased efficiency uses less electricity, which helps lower carbon emissions and supports sustainability goals. When systems run more efficiently, they generally incur lower operational costs and exert a reduced impact on the environment, particularly in regions reliant on fossil fuels.

Table 3 proves that the modified refrigeration system serves as a significant investment for companies seeking to reach a balance between operational efficiency, cost savings, and environmental responsibility over the next decade. This can be attributed to its ability to conserve energy, provide a favorable return on investment, and contribute positively to the environment.

The upgraded refrigeration system has a higher coefficient of performance (COP) that results in less energy consumption and so less greenhouse gases emissions. This is especially true in countries like Jordan, which rely on fossil fuels to meet their energy needs. In the last ten years, the environmental impact that has been noted has been significant and it helps to conserve electricity. This is in line with sustainability goals and helps to ease some of the pressure on the national grid.

This analysis provides stakeholders with valuable insights into the long-term cost-effectiveness and economic advantages of using energy-efficient refrigeration equipment.

## 6. Model validation

The existing model is confirmed using the method described by Llopis et al. [13]. For the trans critical carbon dioxide cycle the gas cooler outlet temperature, gas cooler outlet pressure and evaporation temperature are set to 35 °C, 8500 kPa and 5 °C, respectively. The nominal refrigerating capacity of the base cycle is kept at 14 kW. The degree of subcooling is kept at supported at 5 °C and 7.5 °C, while the compressor's isentropic efficiency for both the basic cycle and sub-cooling cycle is assumed to be 85%. The ambient temperature ($T_{ambient}$) varies between 20 °C and 35 °C. R290 refrigerant is employed exclusively in the subcooling cycle across the two models. The validation process assesses the performance of the dedicated mechanical sub-cooling vapor-compression system using EES software.

Figures 16 and 17 Compare the coefficient of performance (COP) of the trans-critical cycle of the current model and the data from Llopis et al. [13]. The values of COP are evaluated through the variation of ambient and subcooling temperatures. The ambient temperature ranges from 20 °C to 35 °C, while the subcooling temperatures are fixed at 5 °C in Figure 16 and 7.5 °C in Figure 17. The validation results show that the current EES model accurately predicts the COP across different ambient and subcooling conditions. Figure 16 shows an average deviation of approximately 5.8%, while Figure 17 shows an average deviation of about 5.1%.

Figure 18 compared the accuracy in COP comparison for the two models. It can be noted from Figure 18 that there is a strong agreement in performance between the current model and the reference model [13].

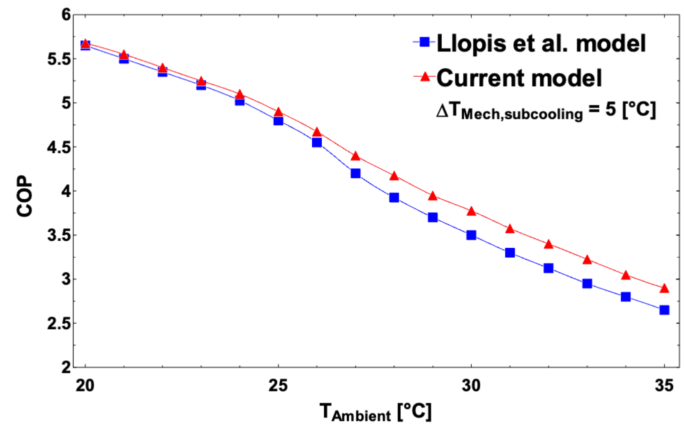


**Figure 16.** Variation of COP with $T_{Ambient}$ for current study and Llopis et al. study [13] at degree of mechanical subcooling of 5 °C

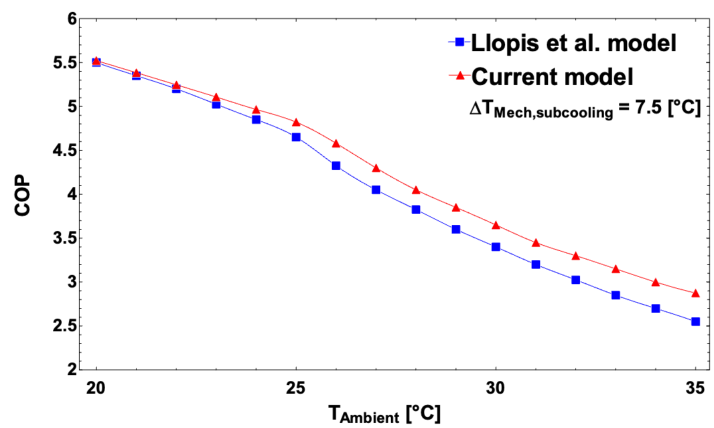


**Figure 17.** Variation of COP with $T_{Ambient}$ for current study and Llopis et al. study [13] at degree of mechanical subcooling of 7.5 °C

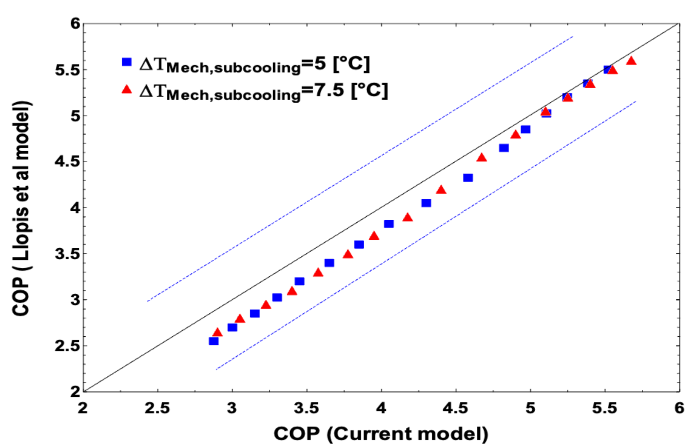


**Figure 18.** Accuracy performance analysis on the values of COP

## 7. Applications for the work

This research has the potential to improve sustainable refrigeration technologies, especially in arid regions like Jordan. This framework provides a new approach to enhance the efficiency and performance of trans-critical CO2 systems through two-stage compression and mechanical subcooling. The results are positive for the deployment of energy efficient and environmentally sound solutions applicable in many sectors, such as commercial refrigeration, industrial cooling and HVAC systems. This approach contributes to global energy efficiency and climate change mitigation goals through reducing electricity consumption and improving the coefficient of performance (COP).

## 8. Conclusion

1. Hybrid System: The hybrid system, which incorporates two-stage compression alongside mechanical sub-cooling, proved a 55.3% improvement in the coefficient of performance (COP), thereby positioning it as the most efficient configuration assessed in the research. This significant enhancement proves the potential inherent in the integration of both technologies to enhance system performance.
2. Energy Savings: The hybrid system showed a 7.1% reduction in power consumption in comparison to the mechanical sub-cooling system, along with a 6% decrease compared to the reference system. The findings show that the hybrid system enhances efficiency and concurrently conserves significant amounts of energy, thereby reducing operating costs.
3. Mechanical Sub-Cooling: In comparison to the reference system, mechanical sub-cooling increased the coefficient of performance (COP) by 46.1% and power consumption by 13.2%. This shows that while sub-cooling enhances performance, the added energy requirements may diminish certain advantages in specific instances.
4. Ambient Temperature Impact: The research showed that elevating ambient temperatures from 35 °C to 50 °C led to a 35.5% rise in power consumption and a 40.1% decrease in COP. This underscores the necessity of considering environmental conditions, as elevated temperatures substantially impair system performance and efficiency.
5. The hybrid system is better for the environment and the economy since it is a more sustainable way to cool things down and promises a faster payback period and a better return on investment (ROI). The hybrid system is a great option for saving money on operations in the long run.
6. The hybrid system is an excellent solution for developing eco-friendly refrigeration technology because it works better and uses less energy. To encourage environmentally friendly practices in the refrigeration industry, it's necessary to be able to produce good outcomes while also having less influence on the environment.
7. Novelty: it uses two-stage compression and mechanical sub-cooling in a single system in a new way. This research offers a unique insight by conducting a comprehensive evaluation of the hybrid system in relation to the climate conditions of Jordan. This research offers valuable perspectives focused on enhancing the efficiency, sustainability, and cost-effectiveness of trans-critical $CO_2$ systems running under challenging environmental conditions.

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## Author contributions

Mohammad Tarawneh: Conceptualization, method, writing original drafts, interpretation of results, supervision, formal analysis, and corresponding author, reviewed and approved the final version of the manuscript. Sohaib AL bayyari: Software, formal analysis, writing, review & editing, critical revision of the manuscript, interpretation of results, reviewed and approved the final version of the manuscript. Mustafa AL hourani: Contributed to method, investigation, validation, interpretation of results, formal analysis, writing, review & editing, reviewed and approved the final version of the manuscript.

## Conflict of interest

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

## Data availability

The data that support the findings of this study are not publicly available, and the authors do not intend to share the datasets or simulation models generated during the study.

## Declaration of generative AI and AI-assisted technologies in the writing process

The authors declare that no generative AI or AI-assisted technologies were used during the preparation, writing, or editing of this manuscript.

## Nomenclature

| | |
|---|---|
| CFC | Chlorofluorocarbon |
| COP | Coefficient of Performance |
| EES | Engineering Equation Solver |

| | |
|---|---|
| GWP | Global Warming Potential |
| h | Enthalpy (kJ kg−1) |
| HCFC | Hydrochlorofluorocarbon |
| HFO | Hydro-Fluoro-Olefin |
| $\dot{m}$ | Refrigerant mass flow rate (kg s −1) |
| Mech | Mechanical Subcooling |
| ODP | Ozone Depletion Potential |
| $P_{gc}$ | Pressure gas coolant (kPa) |
| $\dot{Q}_H$ | heat rejection rate (kW) |
| $\dot{Q}_I$ | Refrigeration capacity (kW) |
| R744 | Refrigerant 744 (Carbon Dioxide) |
| $T_{evap}$ | Evaporation temperature (°C) |
| Two-Stage | Two stage compression |
| $T_{gc}$ | Gas cooler outlet temperature (°C) |
| $\dot{W}_{net}$ | Power consumption in (kW) |

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