**Thermoelectric and Solar Photovoltaic Synergy for Optimized Trans-critical CO₂ Refrigeration in Hot Climates**

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**Abstract**

Traditional trans-critical CO2 refrigeration cycles are energy-intensive, and their efficiency is influenced by outdoor conditions. This study presents a novel technique to enhance the efficiency of these cycles by integrating a thermoelectric sub-cooler tailored to Jordan's climate. The trans-critical CO2 refrigeration cycle, with a nominal refrigeration capacity of 14 kW, was modeled using Engineering Equation Solver (EES) software. A key aspect of this study is the incorporation of solar energy through a custom-designed photovoltaic (PV) system to power the refrigeration cycle, contributing to sustainable cooling technology. Key performance indicators, including refrigeration capacity, power consumption, and coefficient of performance (COP), were thoroughly investigated across varying parameters such as gas cooler pressure (8,000–13,000 kPa), evaporation temperature (-15 to 15°C), ambient temperature (28–40°C), current supply (5–15A), and the number of thermoelectric pairs (50–150). Results showed that increasing gas cooling pressure increased refrigeration capacity by approximately 79%. At a gas cooling pressure of 9,000 kPa, the thermoelectric sub-cooler increased refrigeration capacity by 55%. Increasing the evaporation temperature improved the COP by approximately 125% and reduced power consumption by 67%. At an evaporation temperature of -15°C, the thermoelectric sub-cooler improved performance by 7.5%. Lowering the ambient temperature also enhanced COP by 60% and reduced power consumption by 33%. At a 40°C ambient temperature, the sub-cooler improved COP by 7.6%. Experimental validation showed a 6% average deviation between simulation and experimental results for COP. The on-grid PV system designed with PVsyst software successfully met the cycle's energy demands, achieving 45.3% energy savings.

**Key words: Trans-critical CO2 cycle, Thermoelectric sub-cooler, Photovoltaic, Refrigeration, Coefficient of performance**

**Nomenclature**

|  |  |  |  |
| --- | --- | --- | --- |
| **Symbols** | **Explanation** | **Symbols** | **Explanation** |
| A | Current supply (A) |  | Refrigeration capacity (kW) |
| avg | Average |  | Heat Rejection Rate (kW) |
| amb | Ambient | ref | Reference |
|  | PV module Area (**m2**) | s | Entropy (J/k) |
|  | Temperature coefficient | T | Temperature |
| COP | Coefficient of performance | TESC | Thermoelectric sub-cooling |
| EES | Engineering equation solver | VW | Wind speed |
| EC | Energy consumption (kWh) |  | Power consumption of basic cycle (kW) |
| evap | Evaporation |  | Power consumption of the compressor for thermoelectric sub-cooling cycle (kW) |
| enh | Enhancement |  | Power input into thermoelectric sub-cooler (kW) |
| GWP | Global warming potential |  | Total power consumption in thermoelectric sub-cooling cycle (kW) |
| G | Total solar irradiation (W/ **m2**) | X | Quality |
| gc | Gas cooler | z | Seebeck coefficient (V/k) |
| h | Specific enthalpy (kJ/kg) |  | Isentropic efficiency of the compressor |
| IHEX | Internal heat exchanger | 𝜆 | Thermoelectric element length to cross sectional area ratio (m–1) |
|  | Mass flow rate of the refrigerant (kg/s) | 𝜌 | Specific electrical resistance (Ωm) |
| N | Number of thermoelectric pairs | ΔTsc | Degree of sub-cooling (°C) |
| ODP | Ozone depletion potential | ΔTsh | Degree of super heat (°C) |
| P | Pressure (kPa) | 𝛼 | Surface absorptivity |
| PV | Photovoltaic |  |  |
|  | Power generated by the photovoltaic module |  |  |

**Introduction**

The refrigeration industry has significantly transformed in recent years, driven by the need for energy-efficient and environmentally responsible cooling solutions [1,2]. Several sub-cooling techniques have been investigated to enhance the efficiency of the refrigeration cycles. Thermoelectric sub-cooling (TESC) is a novel approach that uses thermoelectric devices to enhance sub-cooling in trans-critical CO2 refrigeration systems. environment. Thermoelectric sub-cooling offers advantages such as precise temperature control and compatibility with renewable energy sources, making it a promising avenue for further research and development in trans-critical CO2 refrigeration. On the other hand, the geographical location of Jordan, which receives abundant solar radiation, presents a unique opportunity for the cost-effective generation of solar energy [3]. Leveraging available solar energy as a power source for trans-critical carbon dioxide refrigeration cycle compressors could prove highly beneficial, particularly in regions with high ambient temperatures [4]. This simulation research studies the effect of thermoelectric sub-cooler (TESC) on the trans-critical CO2 refrigeration cycle of **14 kW** nominal capacity. Also, on-grid PV solar systems are designed to meet the energy demands of the basic and TESC cycles. Recent studies have explored the potential of thermoelectric sub-cooling and other techniques and have reported improvements in COP and cooling capacity.

Llopis et al. (2015) [5] embarked on a thermodynamic analysis to evaluate the performance of carbon dioxide (CO2) trans-critical refrigeration cycles featuring a mechanical sub-cooler. Their research aimed to shed light on how such sub-cooling methods could impact key performance metrics like the Coefficient of Performance (COP) and cooling capacity. When the mechanical sub-cooler was incorporated, COP increased by up to 20%. Furthermore, the cooling capacity saw an even more impressive surge, increasing to 28.8%.

Sarkar, J. (2013) [6] presented thermodynamic analyses and optimizations of trans-critical CO2 refrigeration cycle with thermoelectric sub-cooler. Results showed that thermoelectric current supply, COP improvement, and discharge pressure reduction increase with increase in cycle temperature lift, with maximum values of 11 A, 25.6%, and 15.4%, respectively, for studied ranges.

Santosa et al. (2018) [7] assessed the efficiency of a trans-critical CO2refrigeration system with an Internal Heat Exchanger (IHEX) under tropical conditions. By comparing it to a conventional CO2 system and using Engineering Equation Solver (EES) software for simulations, they found that the IHEX improves sub-cooling or superheating. This enhancement suggests that IHEX, alone or combined with other methods, boosts refrigeration system performance in hot climates.

Sánchez et al. (2020) [8] evaluated energy improvements in a CO2 trans-critical refrigeration plant by adding a thermoelectric sub-cooling system. Tests showed that at an optimal 2 VDC, cooling capacity increased by +10.7% at 25ºC and +16.0% at 30ºC. COP improvements over the base cycle reached +6.3% at 25ºC and 75.3 bar, and +9.9% at 30ºC and 83.3 bar.

Aranguren et al. (2021) [9] experimentally tested a real trans-critical CO2 refrigeration cycle vapor compression cycle that includes a thermoelectric sub-cooler. The results demonstrate an 11.3% improvement in COP and a 15.3% increase in cooling capacity when the thermoelectric modules are powered by 2 V and the fans by 9 V.

Megdouli et al. (2019) [10] studied three trans-critical CO2 refrigeration cycles to find the most efficient setup. They discovered that a new combined cycle, designed for both power generation and cooling, achieved a lower optimal gas cooler pressure, boosting its COP. This cycle's COP was 110% higher than the vapor refrigeration cycle and 58% higher than the combined cycle.

Shan (2020) [11] conducted a detailed review of CO2 as a natural refrigerant across various applications, focusing on technological advancements to overcome CO2 limitations and boost energy efficiency. The review covered core CO2 cooling cycle principles, key performance traits, and modifications aimed at improving CO2 refrigeration, discussing their operational, technical, and performance aspects.

Yadav & Sarkar (2021) [12] embarked on a mission to propose and scrutinize various ejector refrigeration-based sub-cooling methods in the context of CO2 refrigeration systems. The results showed that adopting sub-cooling methods led to overarching enhancements in CO2 refrigeration plant performance, spanning an impressive range from 12% to 30.3%.

Liu et al. (2019) [13] studied five mechanical sub-cooling systems to maximize COP, finding the two-throttling, two-stage compression high-pressure system performed best. At -30°C evaporation, it improved COP by 76.74%, reaching 1.52 at 40°C ambient temperature—a 21.87% increase over other systems.

Tarawneh, M. (2019) [14] experimentally studied the effect of a porous sub-cooler on the performance of a refrigeration system using R422A refrigerant. They observed that increasing sub-cooling and decreasing sub-cooler porosity resulted in considerable increases in refrigeration capacity, coefficient of performance and relative capacity index by (38.5%), (48%) and (58%) respectively. Furthermore, decreasing porosity from 43% to 40% resulted in a 9.5% reduction in electricity consumption per ton of refrigeration.

In a 2022 study, Tarawneh, M. [15] experimentally investigated the impact of adding porous materials to evaporator pipes to enhance the performance of split air conditioners. The findings showed that reducing porosity from 100% to 33% led to an average increase of 84.3% in the refrigeration system’s coefficient of performance and a 27% rise in power consumption during compression.

In a 2024 simulation study, Tarawneh et al. [16] introduced a novel approach to a solar-powered refrigeration system incorporating a mechanical porous sub-cooler. The findings demonstrated that lowering both porosity and condensing pressure considerably improved the coefficient of performance by about 84% and decreased power consumption by 30.2%.

In 2023, Tarawneh, M. [17] used a porous internal heat exchanger to enhance the performance of a trans-critical CO₂ refrigeration cycle. By decreasing the gas cooler discharge temperature from 53°C to 34°C and reducing the porosity from 100% to 35%, the refrigeration capacity and coefficient of performance increased by 49.7% and 93%, respectively. Additionally, lowering the internal heat exchanger’s porosity to 35% led to a 29.6% reduction in the compressor's power consumption.

.Karampour and Sawalha (2014) [18] aimed to evaluate the impact of Integrated Heat Exchanger (IHEX) positions within a two-stage booster refrigeration system with heat recovery. The study's results indicated that, when focusing solely on cooling demand, the IHEX positions did not significantly improve refrigeration COP. However, the research uncovered a noteworthy finding when considering simultaneous refrigeration and heat recovery. In this scenario, the IHEX located at the gas-cooler output predicted a substantial 12% boost in system efficiency.

Barghash et al. (2021) [19] extensively examined the performance of a solar-powered mechanical sub-cooling air conditioning cycle to boost refrigeration system efficiency in hot climates. The mechanical sub-cooling system increased the overall system COP by approximately 5% to 16% compared to traditional refrigeration systems.

Khalilzadeh et al. (2019) [20] proposed an innovative integrated system that harnessed solar energy to enhance the performance of a cascade refrigeration cycle. Through the integration of solar energy, the researchers achieved a remarkable COP of 4.233, which was five times higher than that of conventional cascade refrigeration cycle systems. Additionally, the proposed integrated system significantly reduced energy consumption by an impressive 84.53% compared to conventional systems.

The manuscript addresses a critical gap in current research on CO₂ trans-critical refrigeration systems, particularly under high-temperature conditions. Existing literature focuses on enhancing CO₂ systems' COP and efficiency through methods such as mechanical and thermoelectric sub-cooling, IHEX integration, and solar-powered cycles. However, these studies examine either thermoelectric sub-cooling or solar integration in isolation, with limited exploration of their combined potential.

This manuscript fills the gap by proposing a synergistic approach, integrating both thermoelectric sub-cooling and solar photovoltaic (PV) technology, to optimize trans-critical CO₂ refrigeration performance under Jordan climate. This dual approach is expected to advance efficiency gains beyond what single-modality methods have achieved, addressing both energy efficiency and COP improvement in a comprehensive, scalable solution for hot climates.

**Thermodynamic cycles**

**Figure 1** demonstrates the schematic diagram of a **4-ton (14 kW)** trans-critical CO2 refrigeration cycle with thermoelectric sub-cooler investigated in this simulation. This cycle consists of solar powered compressor, gas cooler, thermoelectric sub-cooler for additional cooling before expansion valve and evaporator. **Figure 2** exhibits the P-h diagram of thermoelectric sub-cooling (TESC) cycle.

Carbon dioxide (CO2) is used as the working refrigerant in the cycle. **Tabel 1** summarizes thermophysical properties of CO2 refrigerant and shows that refrigerant have zero ozone depletion potential (ODP) and a low global warming potential (GWP).

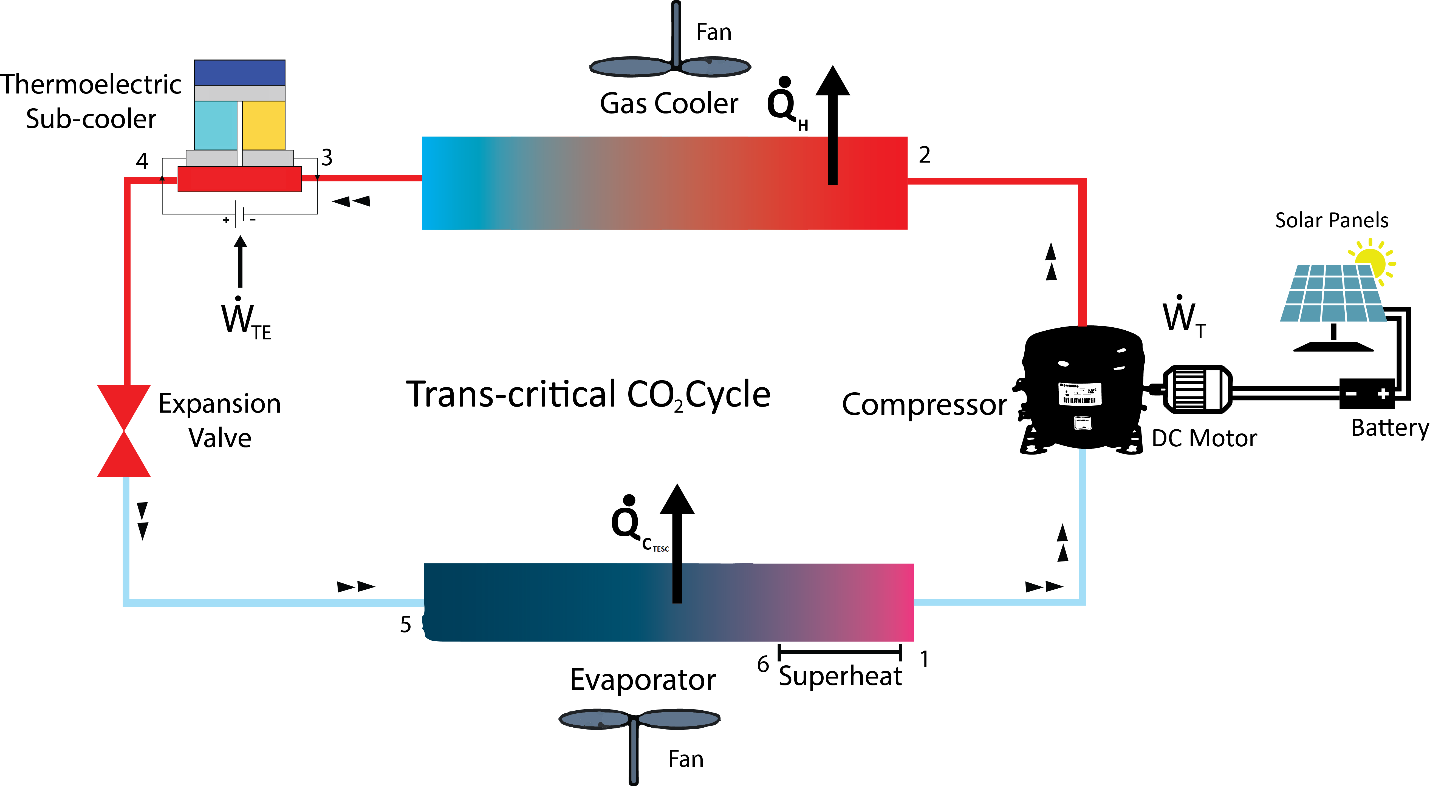


Figure 1: Schematic diagram of solar-powered CO2 refrigeration cycle with thermoelectric sub-cooler

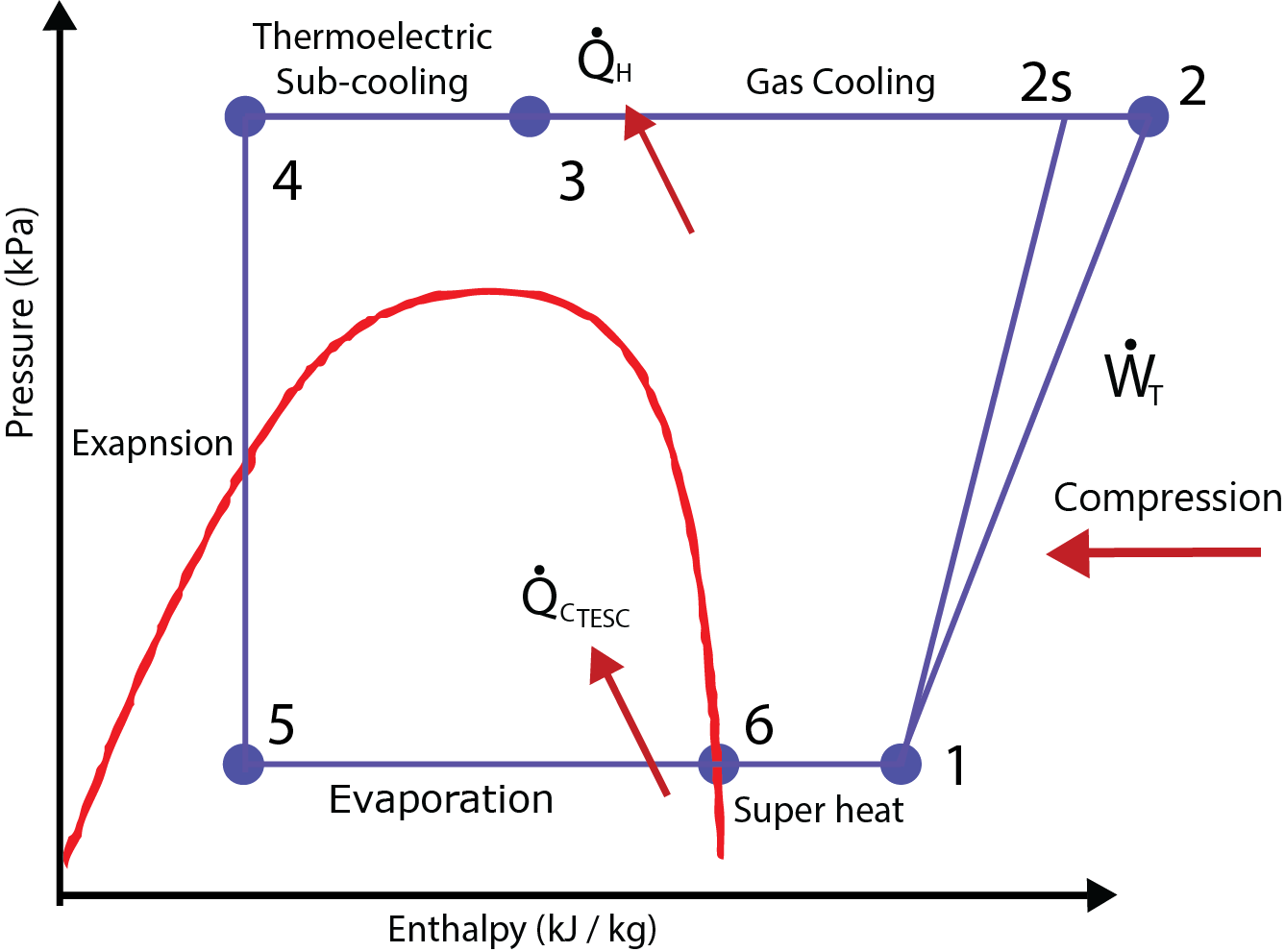


Figure 2: P-h diagram of solar-powered CO2 refrigeration cycle with thermoelectric sub-cooler

Table 1: Thermophysical properties of carbon dioxide refrigerant

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| Refrigerant | ODP | GWP | Boiling point (°C)  (At 1 Atmosphere) | Critical  Temperature (°C) | Critical  Pressure (bar) | Critical  Density (kg/m3) |
| CO2(R744) | 0 | 1 | -78.46 | 30.97 | 73.77 | 467.6 |

**Modeling**

The thermodynamic laws are used to model the trans-critical CO2 refrigeration cycles. According to **Figures** **1** and **2**, the following equations are obtained. The simulation equations for each refrigeration cycle were solved using the Engineering Equation Solver (EES) software [21]. Which is also used to determine the characteristics of refrigerant and perform parametric analyses.

**Basic cycle**

The mass flow rate of CO2 refrigerant () in (kg/s) is calculated as shown in equation (1) [22]:

Where: is the refrigeration capacity in (kW), is the specific enthalpy of CO2 at inlet of the compressor in (kJ/kg), is the specific enthalpy of CO2 at inlet of the evaporator in (kJ/kg).

The refrigeration capacity () in (kW) is calculated as shown in equation (2) [22]:

=

Where: is the average mass flow rate of refrigerant (kg/s).

The power consumption of the compressor of the basic cycle () in (kW) is calculated as shown in equation (3) [22]:

=

Where: is the specific enthalpy of CO2 at inlet of gas cooler in (kJ/kg).

The isentropic efficiency () of the compressor is defined as per equation (4) [22]:

=

Where: is the specific enthalpy of CO2 at the exit of the isentropic compression process in (kJ/kg).

The coefficient of performance (COPBasic) can be calculated by equation (5) [22]:

The equations used to model thermodynamic properties of the basic cycle are listed in **Table 2:**

*Table 2: Modeling equations of thermodynamic properties for basic cycle at each state*

|  |  |  |  |
| --- | --- | --- | --- |
| State 1 | State 2 | State 3 | State 4 |
|  | = + | = h () | = |

Where: , are the evaporation and gas cooler outlet temperatures in (°C), is the quality at state 1, P1, P2, P3 are the pressures in (kPa) at state 1, 2, 3 respectively, S1 is the entropy in (J/k) at state 1, S2S is the entropy at the end of the isentropic compression process in (J/k), T3 is the temperature in (°C) at state 3.

**Thermoelectric cycle (TESC)**

The refrigeration capacity for thermoelectric sub-cooling cycle () is calculated by equation (6) [23]:

(

Where: , are the specific enthalpies in(kJ/kg) at the inlets of the compressor and the evaporator, respectively.

The power consumption of the compressor for thermoelectric sub-cooling cycle () is calculated as per equation (7) [23]:

The power input into thermoelectric sub-cooler () is calculated by using equation (8) [23]:

Where: is the number of thermoelectric pairs, is the seebeck coefficient (V/k), , are the temperatures of hot and cold ends of thermoelectric device in (K), is the current supply in (Amp), is the thermoelectric element length to cross sectional area ratio in (m–1), is specific electrical resistance in (Ωm).

The total power consumption in thermoelectric sub-cooling cycle () is calculated as per equation (9) [23]:

=

The isentropic efficiency of the compressor for thermoelectric sub-cooling cycle () is defined as equation (10) [23]:

The coefficient of performance for thermoelectric sub-cooling cycle () can be calculated by using equation (11) [23]:

=

The enhancement in coefficient of performance () can be calculated by using equation (12):

The equations used to model thermodynamic properties of thermoelectric cycle are listed in **Table 3:**

*Table 3: Modeling equation of thermodynamic properties for thermoelectric sub-cooling cycle at each state*

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| State 1 | State 2 | State 3 | State 4 | State 5 | State 6 |
| P1 = P6  T1 = T6 + ΔTsh  P1 , T1 )  P1 , T1 ) | = + | = h () | ΔTsc  = h () | = | The same state 1 in basic cycle |

Where: P1, P2, P3, P4, P6 are the pressures in (kPa) at state 1, 2, 3, 4, 6 respectively, T1, T3, T4, T6 are the temperatures in (°C) at state 1, 3, 4, 6 respectively, ΔTsc, ΔTsh are the degree of sub-cooling and superheating in (°C) respectively, S1 is the entropy in (J/k) at state 1.

The study factored in the variability of solar irradiance, especially relevant to Jordan's climate, and the conversion efficiency of solar panels. The objective was to accurately reflect the potential impact of solar energy on the cycle's performance. A new model utilizing solar PV energy is introduced to assess the actual energy consumption of the compressor throughout the year, taking into account the variation of ambient temperatures in Jordan, as well as the energy generated from the PV system. In this study, the PV module model PS-M72-380 is used to construct the solar system.

The power generated by the photovoltaic module () is calculated by using equation (13) [24]:

Where: is the efficiency of PV module, is the PV module Area (m2), is the total solar irradiation (W/m2), is the surface absorptivity.

The temperature of PV cell () in (°C) can be calculated by equation (14) [25]:

Where: is the ambient temperature (°C), is the wind speed in (m/s).

The efficiency of PV module () is defined according to equation (15) [24]:

Where: is the temperature coefficient, is the reference efficiency of PV module under standard test conditions [ G = 1000 W/m2, air mass 1.5 and is assumed to be equal to ( 25 °C). and depend on the PV material.

The area under the curve representing the energy consumption () of the compressor in (kWh) is calculated using trapezoidal rule equation (16):

Where: , are the initial and final time in hours, respectively, is the power consumption at initial and final time, respectively.

The grid-connected photovoltaic system design load should be larger than the actual load to account for component losses. The designed load was determined based on a factor of 1.2 for the existing photovoltaic system.

The PV array size () can be calculated by equation (17) [27]:

Where: is the average energy consumption (kWh), is the solar irradiation (W/m2).

The area of the PV array () in (m2) can be calculated by equation (18) [27]:

Where: is the peak solar irradiation (1000 W/m2).

The inverter capacity can be evaluated as equation (19) [27]:

The study focused on a set of parameters known to significantly influence the cycle's efficiency and overall performance. **Table 4** contains a list of simulation's parameter values.

*Table 4: Simulation parameters*

|  |  |  |
| --- | --- | --- |
| **Parameters** | **Range value** | **Constant value** |
| Gas cooler pressure (Pgc) | 8000 – 13000 kPa | - |
| Evaporation temperature (Tevap) | (-15) – 15 °C | 5 °C |
| Ambient temperature (Tamb) | 28 – 40 °C | 35 °C |
| The number of thermoelectric pairs (N) | 50 - 150 | 50 |
| The current supply | 5 – 15 A | 5 A |
| Degree of sub-cooling (ΔTsc) | - | 5 °C |
| Degree of super-heat (ΔTsh) | - | 5 °C |

**Results and discussion**

In this simulation study, various thermodynamic processes have been investigated in accordance with the laws of thermodynamics. The performance of refrigeration cycle has been evaluated, with a particular emphasis on the impact of thermoelectric sub-cooling. The study evaluates the effect of several parameters such as gas cooler pressure, evaporation temperature, ambient temperature, the current supply and the number of thermoelectric pairs on the refrigeration capacity, power consumption and coefficient of performance of the cycle. The Engineering Equation Solver (EES) software [21] is employed to analyze these parameters. On the other hand, on-grid solar PV systems have been designed to meet the cycles energy consumption. In case of optimal gas cooling pressure (), the relationship between several parameters and was studied. In comparison with basic cycle, the correlations between average optimal gas cooler pressure were as follows in **Equations 20, 21, 22**:

**Figures 3, 4, and 5** provide a comprehensive analysis of the refrigeration capacity (), power consumption () and coefficient of performance (COP) across varying gas cooler outlet pressures (Pgc) for both the basic refrigeration cycle and the thermoelectric sub-cooling cycle (TESC). Under specified conditions (Tevap = 5°C, Tgc = 40 °C, = 1, ΔTsc = 5°C), these figures allow for a comparative analysis of performance trends and efficiency implications.

In **Figure 3**, the refrigeration capacity shows a progressive increase with increasing Pgc for both cycles. However, while the basic cycle reaches a plateau, signaling diminishing returns in efficiency at elevated pressures, the TESC cycle begins with a notably higher at the lowest pressure and maintains a steady, advantageous growth across all tested pressures. At specific Pgc values—9000 kPa and 13000 kPa—the TESC cycle enhances by 55% and 18%, respectively, compared to the basic cycle. Additionally, within the Pgc range of 8000 kPa to 10000 kPa, the TESC cycle demonstrates a significant increase of approximately 79% in refrigeration capacity.

**Figure 4** indicates that the power consumption decreases linearly with decreasing in gas cooler outlet pressure, indicating that higher pressures require more energy input to maintain the desired refrigeration output. In comparison with basic cycle, thermoelectric sub-cooler cycle consumed more power by approximately 16%. When Pgc decreased from 13000 kPa to 8000 kPa, the power consumption for thermoelectric cycle decreased by approximately 42%.

**Figure 5** shows that at 9111 kPa, the TESC cycle achieves a performance improvement of approximately 31.5% compared to the basic cycle. Additionally, as Pgc increases from 8000 kPa to 9111 kPa, the COP of the TESC cycle improves by 38.4%.

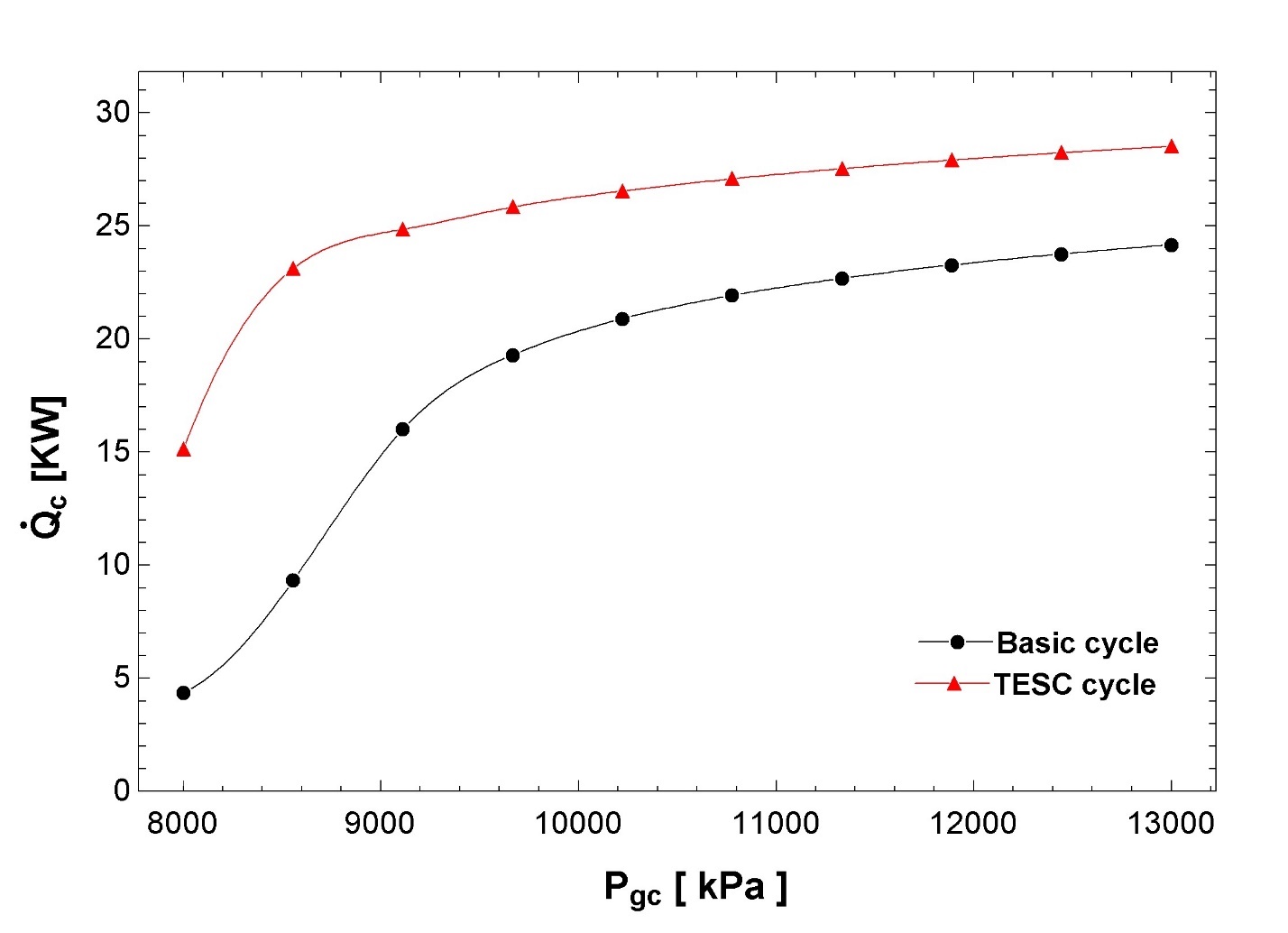


Figure 3: Comparison of refrigeration capacity between basic and thermoelectric sub-cooling cycles as a function of gas cooler outlet pressure

A diagram of a basic cycle

Description automatically generated

Figure 4: Comparison of power consumption between cycles as a function of gas cooler outlet pressure

A graph of a normalized cycle

Description automatically generated with medium confidence

Figure 5: Comparison of COP between cycles as a function of gas cooler outlet pressure

**Figures 6, 7** and **8** show the impact of varying evaporation temperature (Tevap) on the refrigeration capacity (), the power consumption () and coefficient of performance (COP) for basic cycle and thermoelectric sub-cooling cycle (TESC), respectively. Under specified conditions (Tgc = 40 °C, = 1, ΔTsc = 5°C), these figures allow for a comparative analysis of performance trends and efficiency implications.

**Figure 6** demonstrates a pronounced increase in refrigeration capacity as the evaporation temperature decreases from 15°C to -15°C. TESC cycle increased the refrigeration capacity () by approximately 25% at Tevap = 0°C. When evaporation temperature decreased from 15°C to 0°C, the refrigeration capacity increased by 8%.

**Figure 7** shows a linear decrease in power consumption as evaporation temperature increases. TESC cycle consistently shows higher power consumption across all temperatures compared to the basic cycle (12% at Tevap = -15°C, 33% at Tevap = 15°C). When evaporation temperature increases from -15°C to 15°C, the power consumption for basic cycle decreased by approximately 67%.

**Figure 8** displays a continuous increase in COP, indicating improving efficiency as the evaporation temperature increases, typical for refrigeration cycles where higher temperatures reduce the workload on the compressor. TESC cycle enhanced the COP by 7.5% at Tevap = -15°C and 4% at Tevap = 15°C compared with basic cycle.

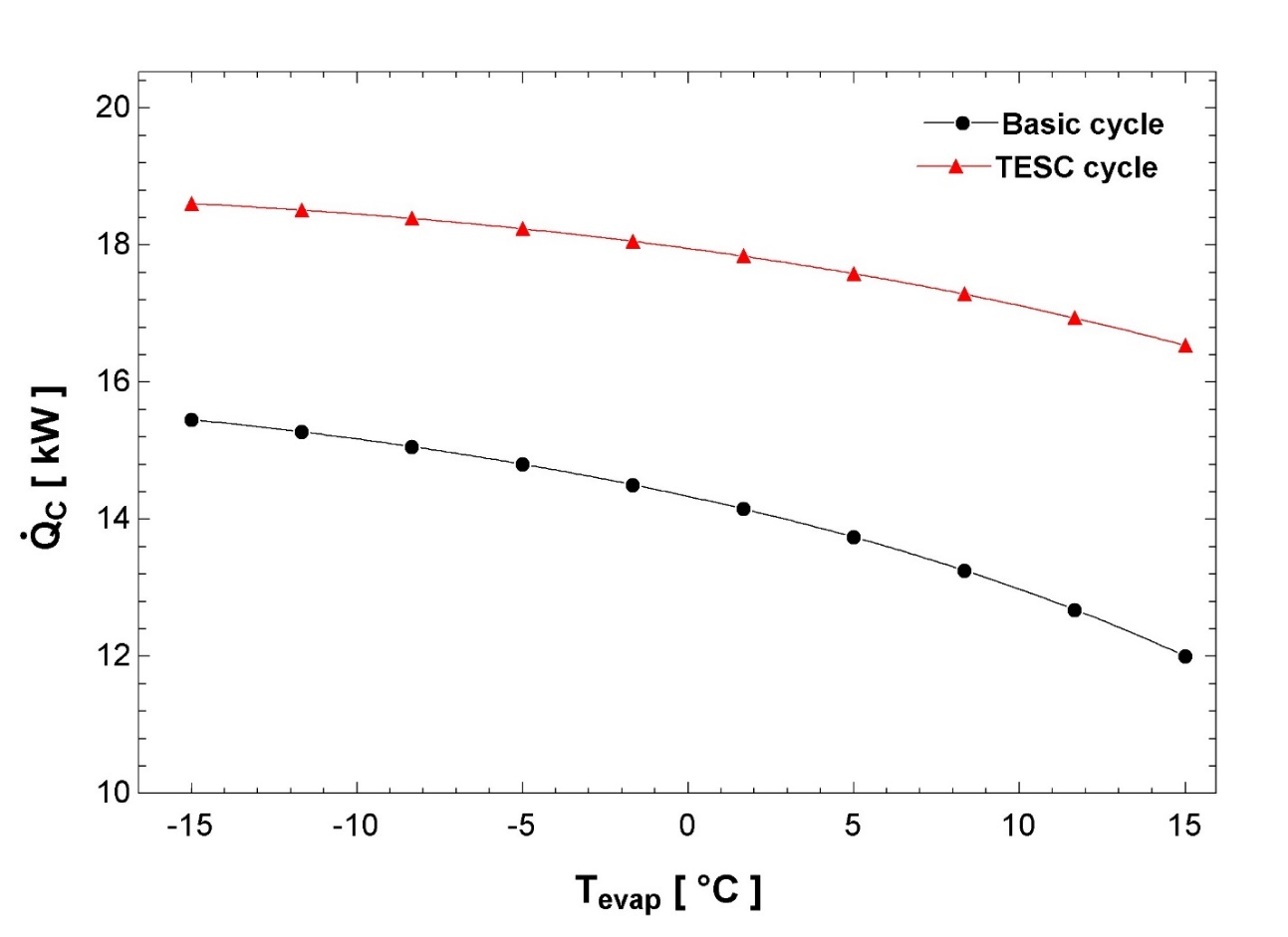


Figure 6: Comparison of refrigeration capacity between basic and thermoelectric sub-cooling cycles as a function of evaporation temperature

A graph of a basic cycle

Description automatically generated

Figure 7: Comparison of power consumption between cycles as a function of evaporation temperature

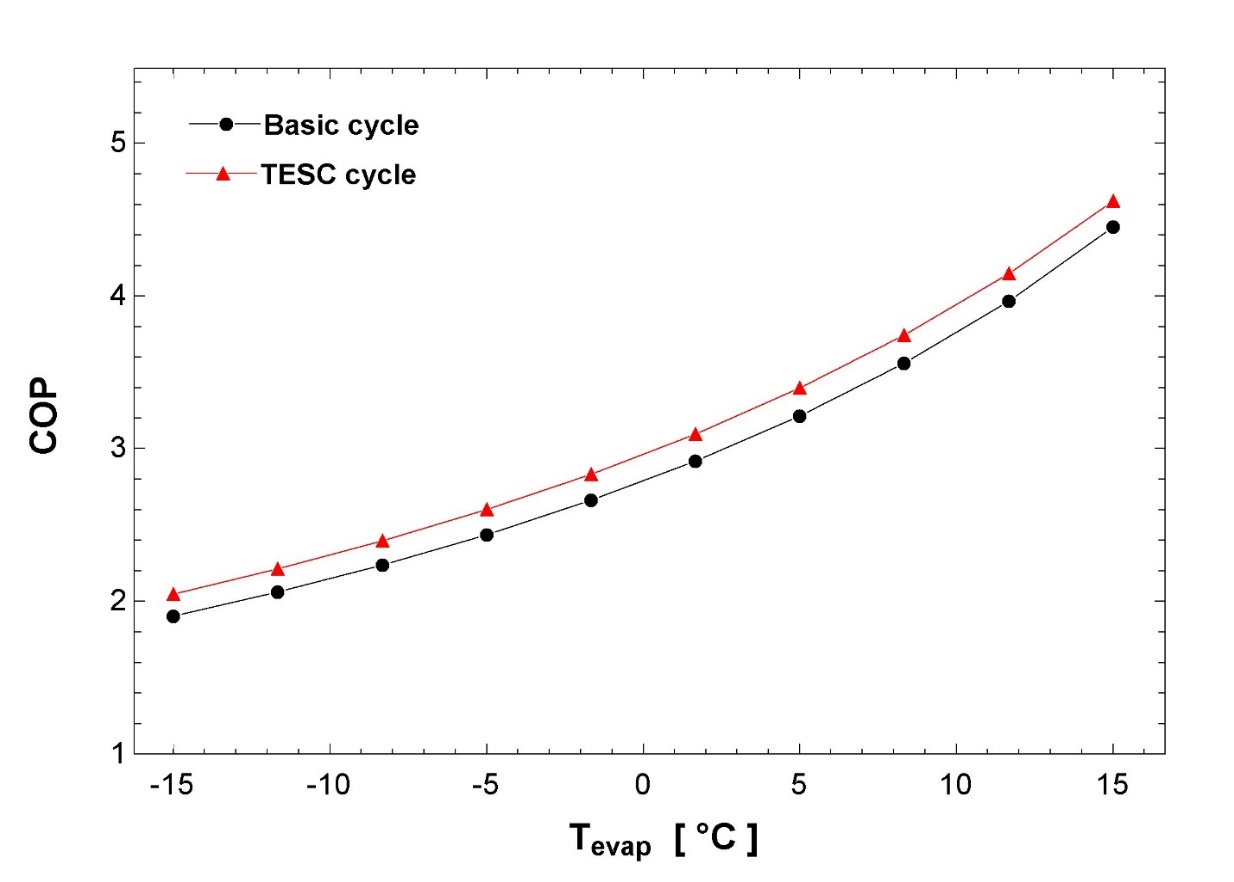


Figure 8: Comparison of COP between cycles as a function of evaporation temperature

**Figures 9, 10** and **11** illustrate the refrigeration capacity (), the power consumption () and coefficient of performance (COP) for basic cycle and thermoelectric sub-cooling cycle (TESC), plotted against the ambient temperature (Tamb) from 28°C to 40°C, respectively. Under specified conditions (Tevap = 5°C, Tgc = 40 °C, = 1, ΔTsc = 5°C), these figures allow for a comparative analysis of performance trends and efficiency implications.

**Figure 9** shows that the refrigeration capacity increases with decreasing ambient temperature. At Tamb = 35°C, TESC cycle increased the by 28.5% compared with basic cycle. When Tamb decreased from 40°C to 28°C, the of TESC cycle increased by 18%.

**Figure 10** illustrates a linear decrease in power consumption as ambient temperatures decrease. At 28°C and 40°C, the power consumption for TESC cycle increased by 26% and 19% compared with basic cycle, respectively. When ambient temperature decreased from 40°C to 28°C, the power consumption for TESC cycle decreased by 33%.

**Figure 11** displays that the COP values increase as ambient temperatures decrease. At 28°C and 40°C, the COP for TESC cycle enhanced by 3.4% and 7.6%, respectively. When Tamb decreases from 40°C to 28°C, TESC cycle performance enhances by approximately 60%.

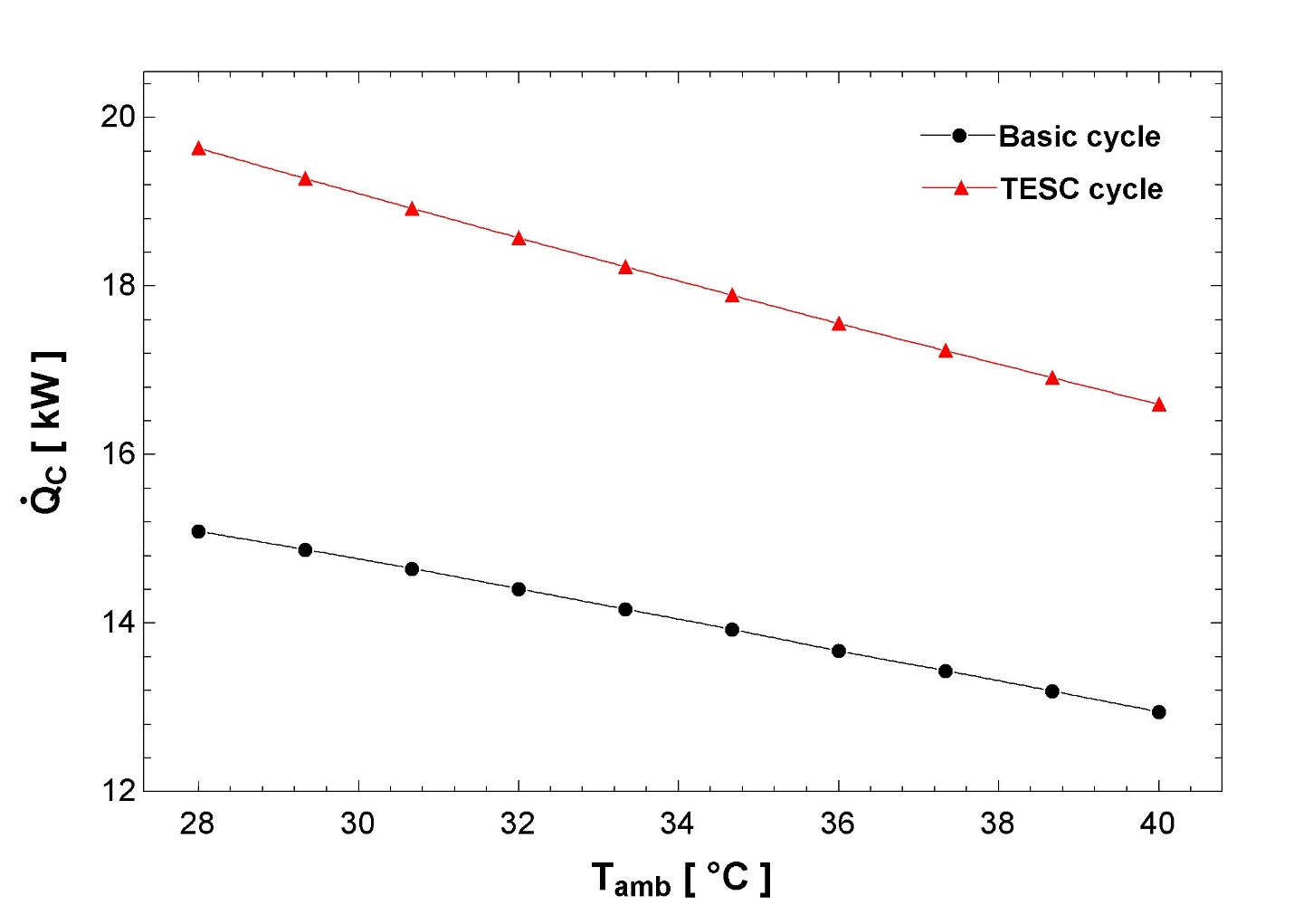


Figure 9: Comparison of refrigeration capacity between basic and thermoelectric sub-cooling cycles as a function of ambient temperature

A diagram of a basic cycle

Description automatically generated

Figure 10: Comparison of power consumption between cycles as a function of ambient temperature

A graph of a diagram

Description automatically generated with medium confidence

Figure 11: Comparison of COP between cycles as a function of ambient temperature

**Figures 12 and 13** demonstrate the relationship between the current supplied (in Amperes) and the power consumption () and coefficient of performance (COP) of a TESC cycle, across different numbers of thermoelectric pairs (N). Three different curves corresponding to different numbers of pairs were presented: 50, 100, and 150 pairs. Under specified conditions (Pgc= 9000 kPa, Tevap = 5°C, Tgc= 40°C, ΔTsc = 5°C), these figures allow for a comparative analysis of performance trends and efficiency implications.

**Figure 12** indicates that as the current increases from 5 A to 15 A, the power consumption increases linearly. It also shows that the power consumption increases more steeply with a higher number of pairs. When current decreased from 15A to 5 A, the power consumption for N=50, N=100, N=150 decreased by approximately 24.7%, 37% and 43%, respectively. At 10A, TESC cycle with N=50 pairs consumed less power than N=100 and N=150 by 20% and 34%, respectively. These results also illustrate that while increasing the number of thermoelectric pairs can potentially enhance the sub-cooling effect, it also significantly raises the power consumption of the system. **Figure 13** indicates that the COP increases as the current decreases. The highest number of pairs (N=150) shows the steepest decline in COP with increasing current, indicating a diminishing return on efficiency with more pairs under higher current loads. At 15A, the TESC cycle with N=50 enhanced the COP more than TESC cycle with N=100 and N=150 by 27% and 67%, respectively. When decreasing the current from 15A to 5A, the COP of TESC cycle with N=50 enhanced by 33%.

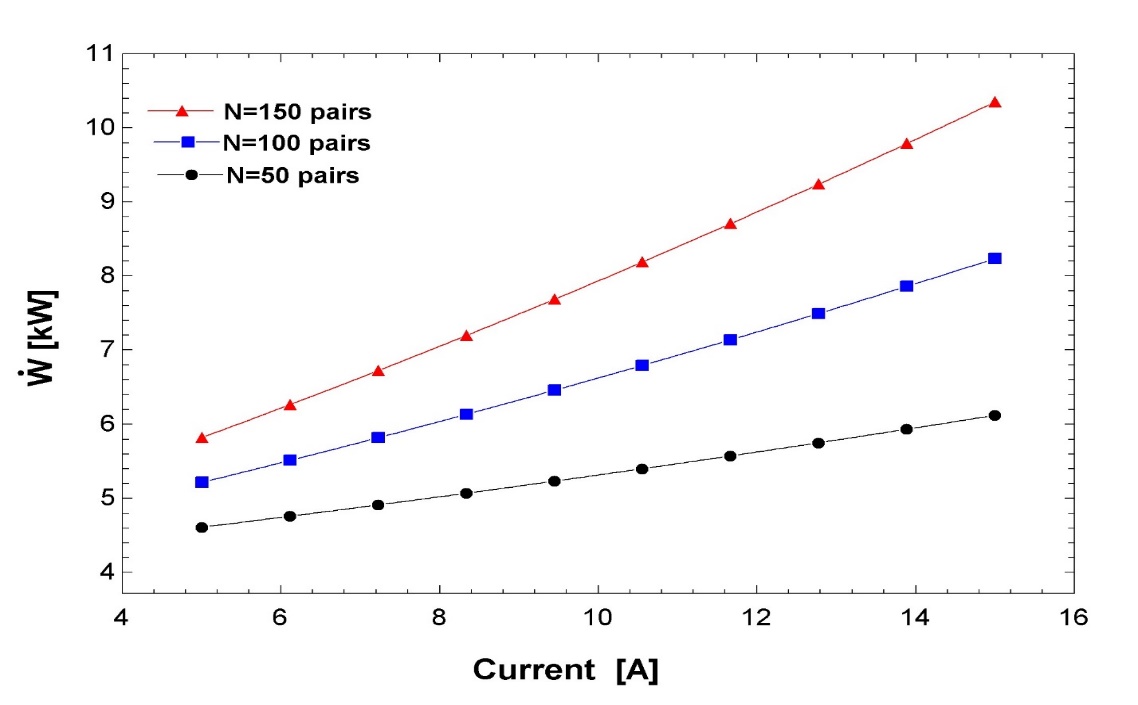


Figure 12: The effect of current supply on the power consumption of thermoelectric sub-cooling cycle at different number of pairs

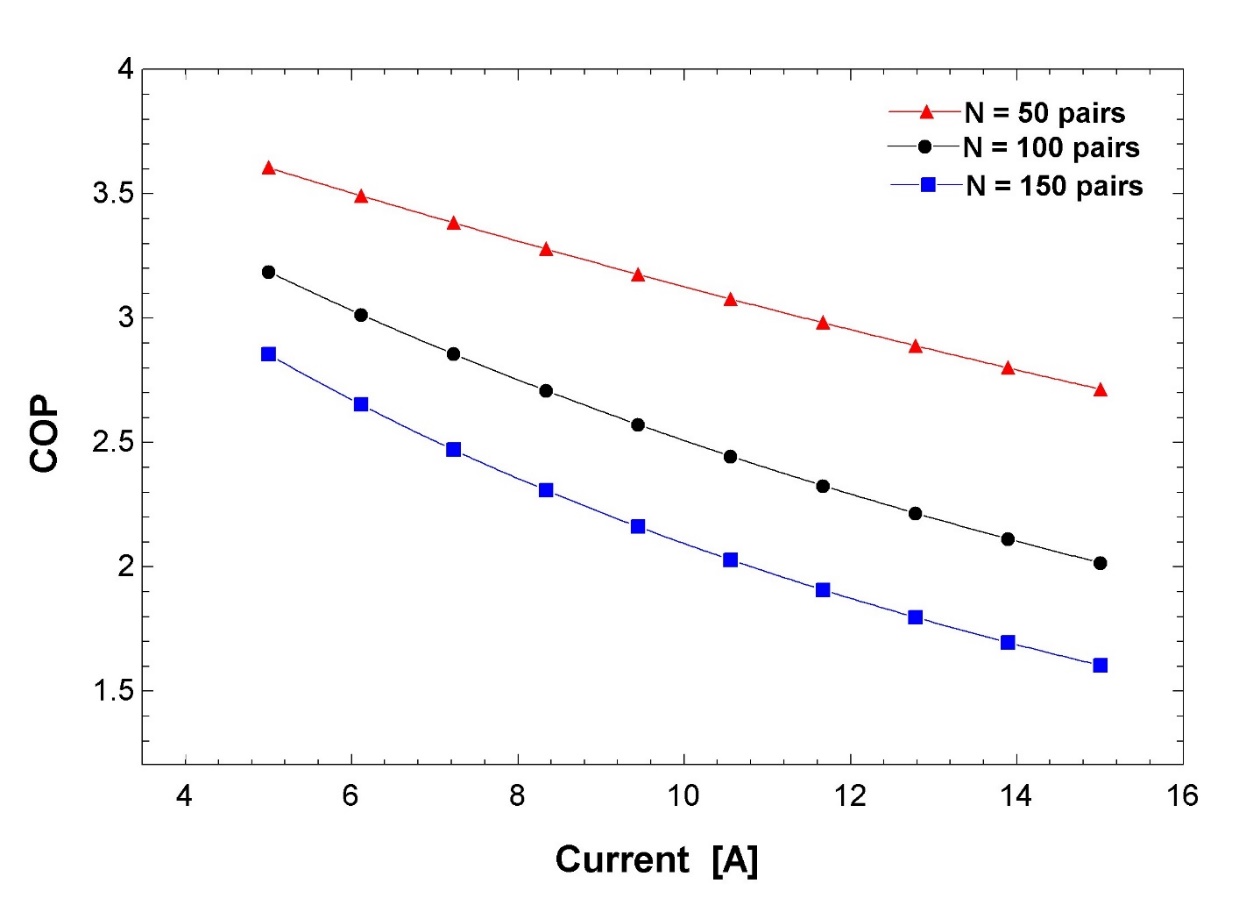


Figure 13: The effect of current supply on COP of thermoelectric sub-cooling cycle at different number of pairs

**Model validation**

The numerical simulation model for thermoelectric sub-cooling cycle has been validated through experimental work conducted by (Alvaro Casi et al., 2022) [28].

This research presented an experimental study to improve performance of refrigeration cycle with thermoelectric sub-cooler. The simulation results for the COP of thermoelectric sub-cooling cycle are validated by comparing them to the experimental data presented by (Alvaro Casi et al., 2022) [28] at (Pgc = 8200 – 8800 kPa, Tevap = -10 °C, Tamb = 30 °C and mass flow rate of 0.0009 – 0.0011 kg/s) as shown in **Figure 14**. The average deviation between Simulation and Experimental results of COP for thermoelectric cycle is about 6%.

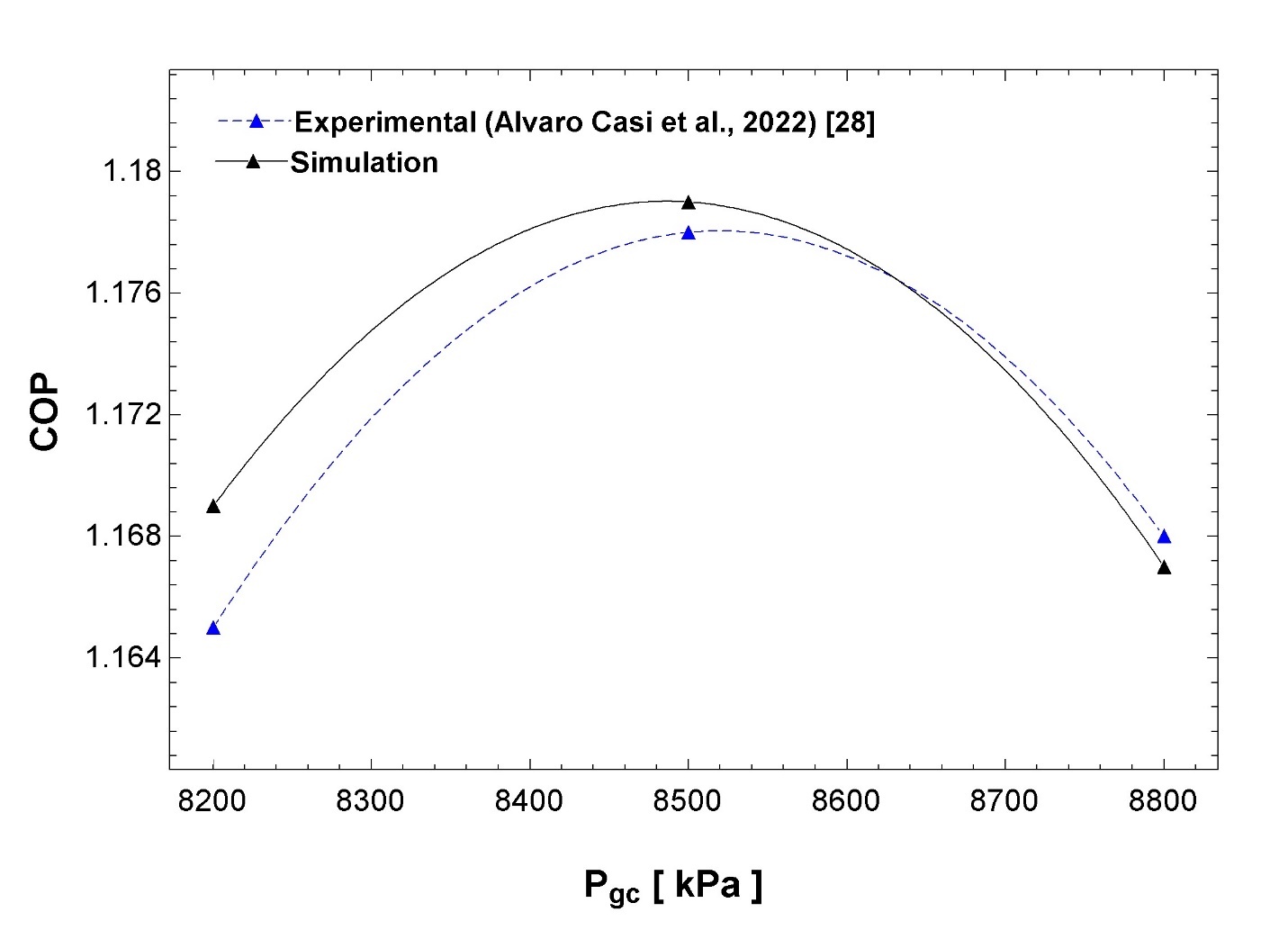


Figure 14: Variation of COP with gas cooler outlet pressure for the experimental and simulated studies (thermoelectric sub-cooling cycle)

**Solar system analysis**

The power consumption of the compressor based on the ambient temperature of every hour of the average day for each month [29] for the basic cycle and thermoelectric sub-cooling cycle are shown in **Figure** **15, 16** at the evaporating temperature of 5°C and compressor efficiency of 1. These figures indicate that during the night hours, the power consumption of the compressor decreases due to lower ambient temperatures. As the sun rises, power consumption increases, peaking in the afternoon before declining at sunset. The highest power consumption occurs in August, while the lowest is recorded in January. The maximum power consumption in August for the basic cycle and thermoelectric sub-cooling cycle are 3.598 kW and 4.457 kW respectively. Also, the maximum power consumption in January for the basic cycle and thermoelectric sub-cooling cycle are 0.7444 kW and 1.403 kW respectively. The results show that the maximum power consumption of the compressor when operating in the subcritical mode in January is approximately 80% lower than in the trans-critical mode in August.

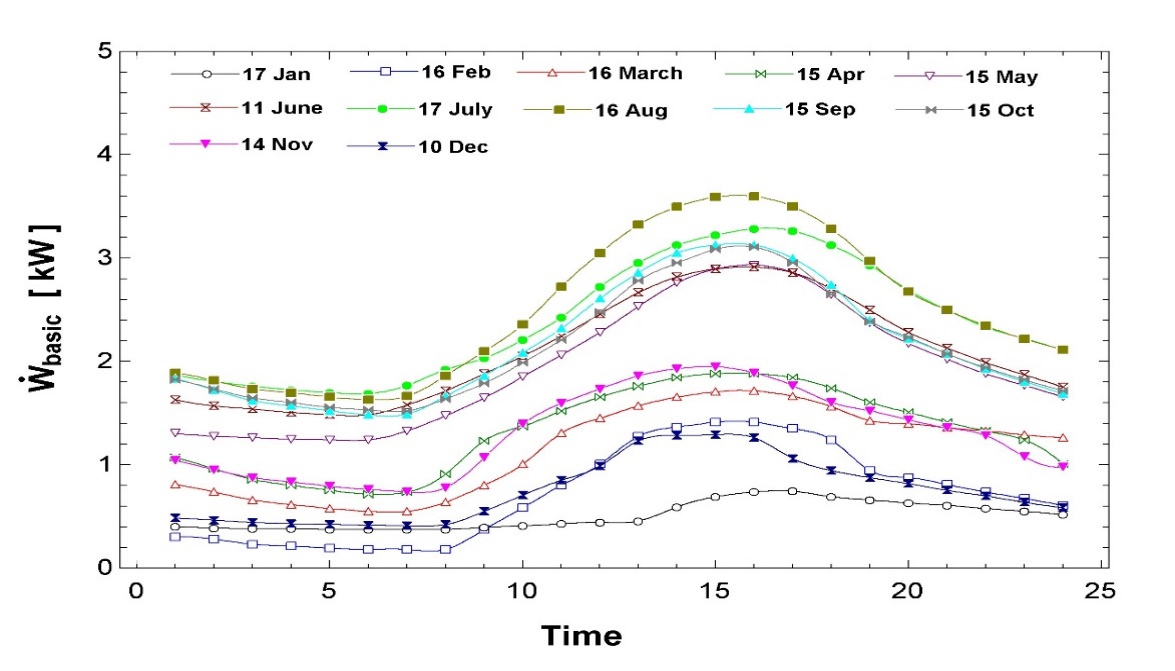


Figure 15: The power consumption for the basic cycle versus time for the average day of the month

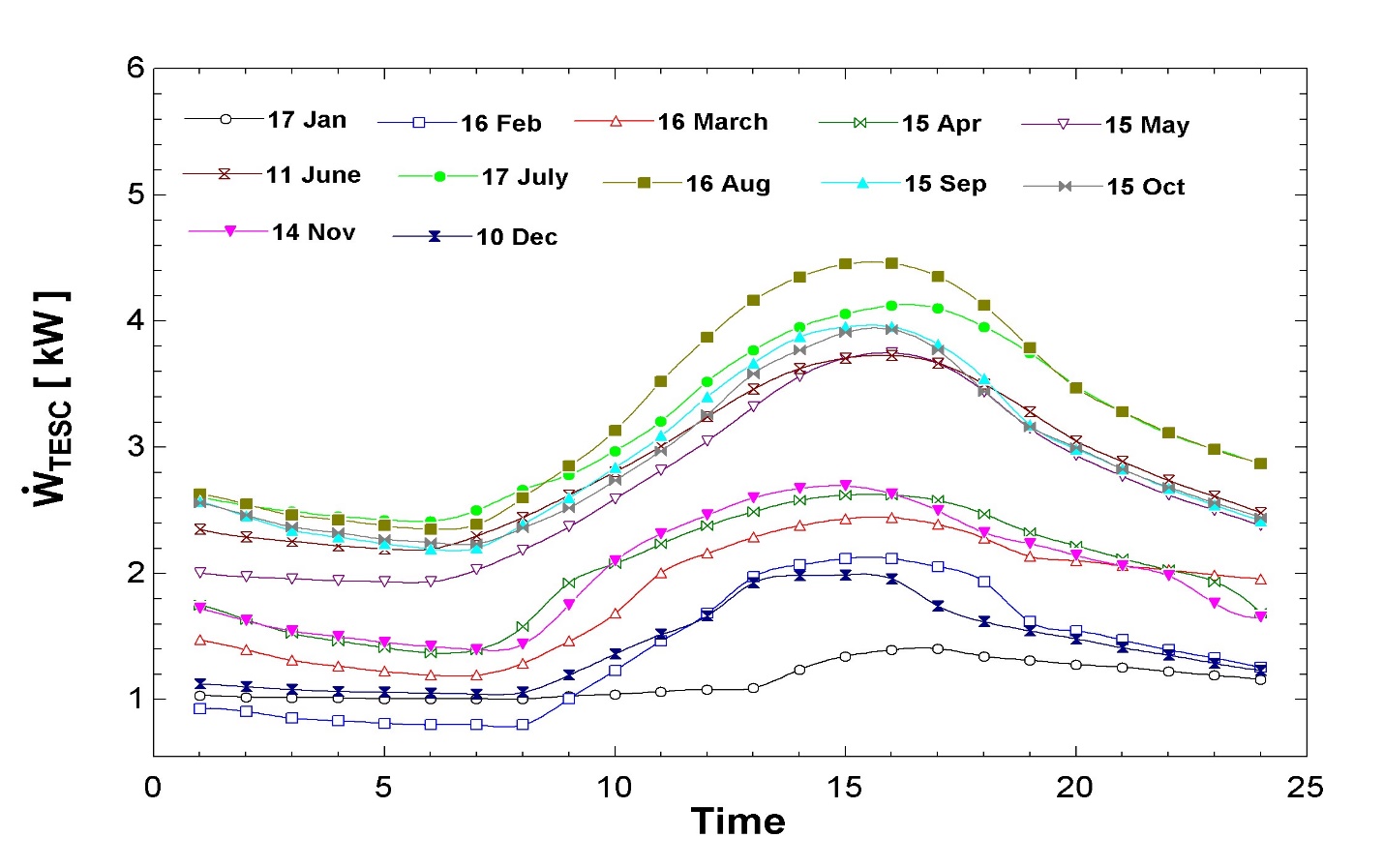


Figure 16: The power consumption for the thermoelectric sub-cooling cycle versus time for the average day of the month

A grid-connected PV system was chosen for the cycles. At ambient temperature, wind speed, and solar irradiation under climate of Jordan, the power generated by the PV module every hour of the average day for each hot month and the power consumption of the compressor for the basic cycle and thermoelectric sub-cooler are depicted in **Figures** **17**, **18** respectively. The figures show that the PV module generated the most power during the middle of the day. At the time of 12, it recorded the highest solar output in May and the lowest in July. Additionally, the PV module does not generate power at night, so any shortfall is supplemented by the electricity grid.

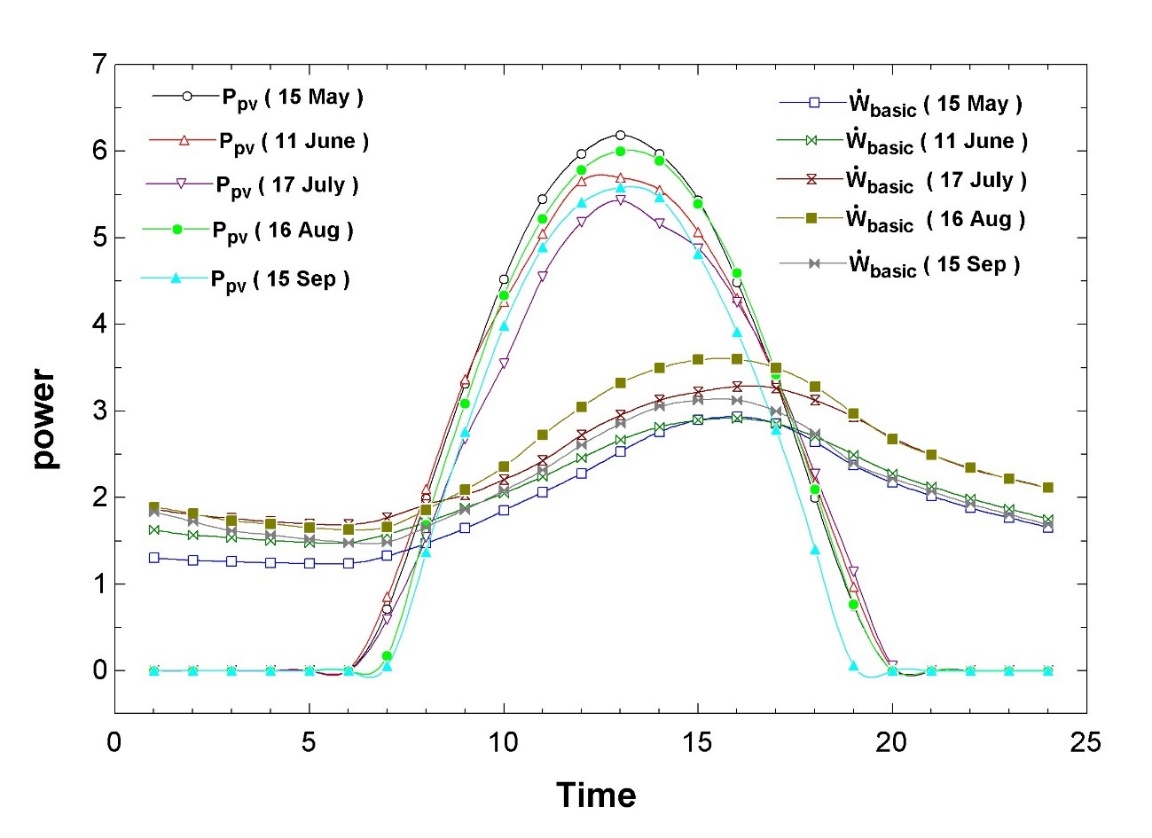


Figure 17: The power consumption and the power generated by the PV module for the basic cycle versus time for the average day of the hot months

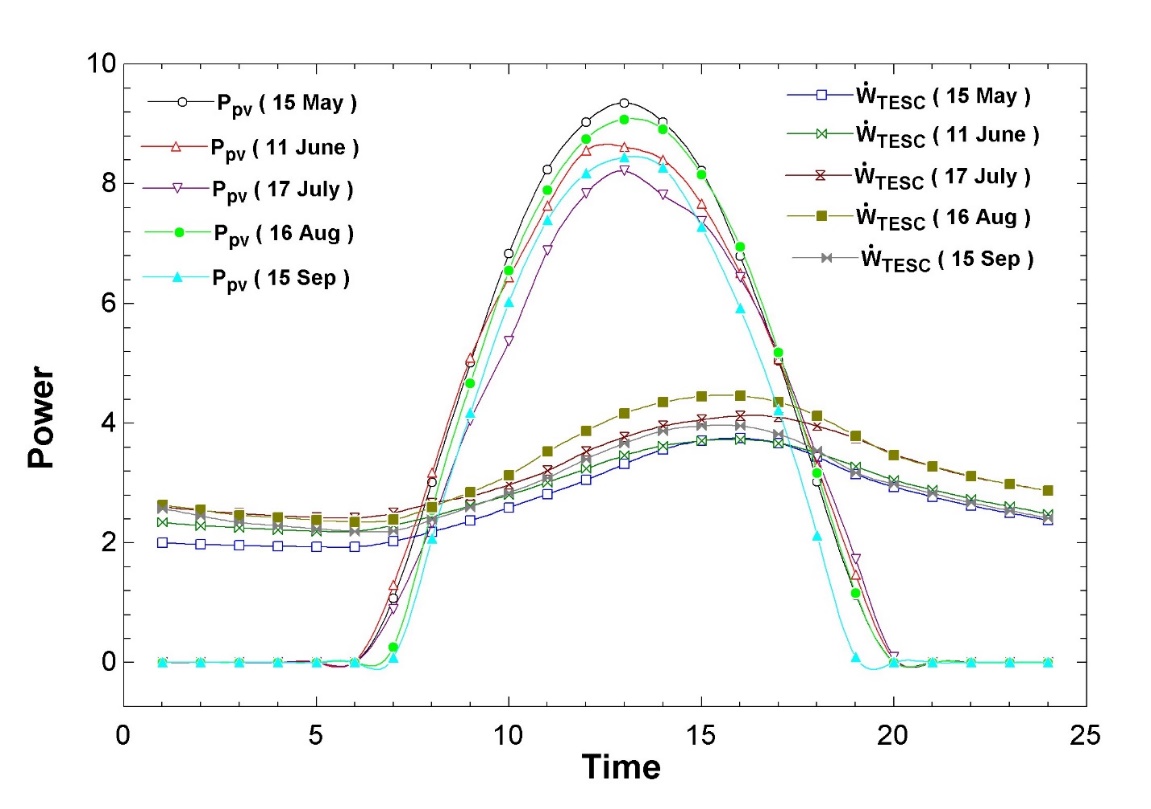


Figure 18: The power consumption and the power generated by the PV module for the thermoelectric sub-cooling cycle versus time for the average day of the hot months

The size of the PV array to meet the power demand for the basic cycle and thermoelectric sub-cooling cycle was calculated by **equation 16**. Also, the area of the PV array for these cycles was calculated by **equation 17**. As shown in **Table 5**.

Table 5: The sizing of the on-grid PV system

|  |  |  |
| --- | --- | --- |
|  | Basic cycle | TESC cycle |
| PV size (KW) | 7.99 | 11.6 |
| Area (m2) | 41 | 62 |
| Inverter (KW) | 8 | 12 |

An annual simulation using PVsyst software [30] was performed to evaluate the performance of the on-grid system, which was designed with a tilt angle of 30° and an azimuth of 0 for south-facing module orientation. The main results for the basic cycle and TESC cycle are presented in **Tables** **6**, **7** respectively. These tables indicate that the highest solar irradiation occurs in June at 216.2 kWh/m², while the lowest is recorded in February at 136.5 kWh/m². In June, the maximum energy powered by PV module was 1538 kWh in basic cycle and 2238 kWh in TESC cycle. The maximum saved energy was 44.69% in the basic cycle and 45.3% in TESC cycle.

Table 6: Results of simulating the basic cycle using PVsyst software

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| Month | Total  irradiation  kWh/m2 | EC  kWh | Epv  kWh | EC from  the sun  kWh | Energy  injected  into grid  kWh | Energy  from  the grid  kWh | Saved  energy  during  day light |
| Jan | 144.2 | 362 | 1128 | 133.8 | 965 | 229 | 36.9613% |
| Feb | 136.5 | 469 | 1061 | 189 | 844.6 | 280 | 40.2985% |
| Mar | 178.7 | 824 | 1351 | 353.4 | 962.6 | 471 | 42.8883% |
| April | 191.7 | 917 | 1420 | 414.7 | 968 | 503 | 43.2236% |
| May | 204.7 | 1402 | 1489 | 624.3 | 825.9 | 778 | 44.5292% |
| June | 216.2 | 1465 | 1538 | 654.8 | 844 | 810 | 44.6962% |
| July | 214.3 | 1726 | 1513 | 749.8 | 723.2 | 977 | 43.4415% |
| Aug | 214.1 | 1791 | 1511 | 760 | 711.1 | 1031 | 42.4344% |
| Sep | 203 | 1501 | 1441 | 628.7 | 775.5 | 873 | 41.8854% |
| Oct | 281 | 1533 | 1320 | 597.6 | 688.6 | 935 | 38.9824% |
| Nov | 150.7 | 906 | 1146 | 341 | 774.8 | 565 | 37.638% |
| Dec | 140.9 | 541 | 1098 | 197.5 | 873.1 | 344 | 36.5065% |
| Year | 2175.8 | 13438 | 16015 | 5644.6 | 9956.4 | 7793 | 42.0048% |

Table 7: Results of simulating the thermoelectric sub-cooling cycle using PVsyst software

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| Month | Total  irradiation  kWh/m2 | EC  kWh | Epv  kWh | EC from  the sun  kWh | Energy  injected  into grid  kWh | Energy  from  the grid  kWh | Saved  energy  during  day light |
| Jan | 144.2 | 820 | 1641 | 299 | 1310 | 521 | 36.4634% |
| Feb | 136.5 | 892 | 1543 | 356 | 1157 | 536 | 39.9103% |
| Mar | 178.7 | 1314 | 1965 | 561 | 1366 | 754 | 42.6941% |
| April | 191.7 | 1400 | 2065 | 633 | 1392 | 767 | 45.2143% |
| May | 204.7 | 1933 | 2165 | 872 | 1252 | 1061 | 45.1112% |
| June | 216.2 | 1986 | 2238 | 901 | 1293 | 1085 | 45.3676% |
| July | 214.3 | 2270 | 2200 | 1005 | 1152 | 1266 | 44.2731% |
| Aug | 214.1 | 2349 | 2198 | 1014 | 1140 | 1335 | 43.1673% |
| Sep | 203 | 2025 | 2096 | 860 | 1195 | 1165 | 42.4691% |
| Oct | 181 | 2073 | 1920 | 817 | 1067 | 1256 | 39.4115% |
| Nov | 150.7 | 1388 | 1667 | 523 | 1112 | 866 | 37.6801% |
| Dec | 140.9 | 1011 | 1597 | 367 | 1201 | 645 | 36.3007% |
| Year | 2175.8 | 19463 | 23295 | 8207 | 14635 | 11256 | 42.1672% |

Where: EC is the energy consumption in (kWh), Epv is the energy generated from PV system in (kWh), EC from the sun is the energy consumption directly during daylight in (kWh).

**Conclusions**

1. Integrating a thermoelectric sub-cooler improved the refrigeration capacity by **55%** at a gas cooling pressure of 9,000 kPa and enhanced performance by **7.5%** at an evaporation temperature of -15°C under Jordan's climate conditions.
2. Solar energy, harnessed through a custom-designed PV system, effectively powered the refrigeration cycle, achieving up to **45.3% energy savings** when the thermoelectric sub-cooler was used, advancing sustainable cooling technology.
3. Key performance parameters significantly influenced system efficiency: increasing gas cooling pressure from **8,000 to 13,000 kPa** resulted in a **79% increase in refrigeration capacity**, while raising evaporation temperature boosted COP by **125%** and reduced power consumption by **67%**. Lower ambient temperatures improved COP by **60%** and reduced power consumption by **33%**.
4. Experimental validation demonstrated the model’s reliability, with a deviation of only **6%** between simulated and actual COP results, underscoring the accuracy of the modeled system.

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