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Research Article

Performance study of sub-cooled CO_2 trans-critical air conditioning cycle: The combined effect of vapor quality and compressor efficiency

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ABSTRACT

Carbon dioxide, one of the most critical potential refrigerants, has little impact on the environment. CO_2 trans-critical cycles are an essential topic in air conditioning. The current study investigates the performance of the CO_2 trans-critical air conditioning cycle for various parameters. The distinct contribution of this work arises from its emphasis on the interrelated nature of the combined effect of compressor efficiency and vapor quality at the evaporator inlet on the overall performance of the CO_2 trans-critical cycle; by filling this knowledge gap, the research endeavours to comprehensively understand the system's behavior under a wide range of operation conditions. The cycle has been modelled using Engineering Equation Solver (EES) and MATLAB codes and validated against an experimental study. The results showed that the cooling capacity increases by 66% when gas-cooling pressure rises from 100 to 150 bar. Raising vapor quality from 0.1 to 0.5 and lowering the degree of superheat from 12 to 0 °C reduces the cooling capacity by 52.4% and increases the coefficient of performance by 87%. Power consumption of the compressor decreases by 50% by increasing compressor efficiency from 70% to 100% and lowering gas cooling pressure from 110 to 80 bar. While the coefficient of performance of the cycle increases by 111.7% by increasing compressor efficiency from 70 to 100% with a degree of sub-cool from 0 to 6 °C and a degree of superheat from 0 to 12 °C.

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INTRODUCTION

Carbon dioxide (CO_2) has been increasingly proposed as an efficient alternative refrigerant with zero-ozone depletion and suitable thermophysical characteristics such as low critical temperature (31.1 °C) and high gas cooling pressure (73.8 bar). The growing environmental worries over using conventional refrigerants, i.e., chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HFCs) in refrigeration systems, have directed research toward finding an adequate replacement for the traditional refrigerants with low environmental impacts. For the CO_2 trans-critical

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cycle, enhanced effectiveness has always been the primary concern. Numerous studies investigated the effect of different parameters on the performance of the CO_2 trans-critical cycle. The novelty of this research lies in addressing a specific gap in existing literature regarding the CO_2 trans-critical cycle. While previous studies have explored various aspects of its performance, the combined effect of two crucial factors, compressor efficiency, and evaporator inlet vapor quality, has not been thoroughly investigated. This research aims to comprehensively examine the synergistic effects of these two parameters through a systematic simulation study.

The influence of various gas cooler pressures, gas cooler outlet temperature, and evaporation temperature on the coefficient of performance (COP) has been investigated. Okasha and Müller [1] modelled and simulated a single-stage CO_2 trans-critical heat pump cycle using MATLAB code integrated with the NIST REFPROP thermodynamic database. The isentropic and volumetric efficiency correlations for the compressor were developed at three different evaporation temperatures. According to the authors [1], improved COP of the cycle can be attained by optimizing the gas cooler pressure, which is mainly influenced by the gas cooler outlet temperature. The optimum gas cooler pressure has been correlated with the gas cooler outlet temperature and validated with the ones available in the literature. Using the EXCEL program, the performance of the CO_2 trans-critical cycle for various parameters was investigated by Baheta et al. [2]. They concluded that for maximum COP, an optimum gas cooler pressure is significantly affected by gas cooler outlet temperature and evaporation temperature. It was also observed that increasing the evaporation temperature improves the COP of the cycle, which makes it a better candidate for air-conditioning systems than refrigeration systems.

The effect of high outdoor temperature on the performance of the CO_2 trans-critical cycle has been studied theoretically and numerically using Engineering Equation Solver software (EES) by Santosa and Waisnawa [3]. The main conclusion showed that an increase in the outdoor temperature between 25 °C and 45 °C reduces the overall COP of the refrigeration cycle by 3%. Another study by Singh et al. [4] examined six different modifications to the basic CO_2 trans-critical refrigeration cycle under high outdoor temperature conditions. Their results showed that the performance of systems with internal heat exchanger (IHX) is relatively better (highest COP) at higher ambient temperature.

Using energy and exergy analysis, the effectiveness of the CO_2 trans-critical two-stage compression refrigeration cycle was simulated by Sun et al. [5]. Their results showed that by employing an auxiliary gas cooler to the cycle, a higher COP, lower compression power, and less exergy destruction could be achieved. A comparative numerical and experimental investigation on the effectiveness of the CO_2 trans-critical cycle with an emphasis on the usage of hermetic compressors was conducted by Rigola et al. [6]. They concluded that CO_2 is a suitable replacement for conventional refrigerants under trans-critical operating conditions.

To reduce irreversibly and improve the COP of the CO_2 trans-critical cycle, a single vortex tube replaced with the conventional expansion valve was proposed by Liu et al. [7]. The modified cycle has been compared with the traditional cycle under various operating conditions. Their results showed a significant improvement in COP, mainly with the vortex tube's inlet temperature and discharged pressure. Moreover, for best improvement in COP, an optimal discharged pressure of the vortex tube has been correlated.

Using a computer simulation model, the two main CO_2 trans-critical cycle alternatives in supermarket refrigeration applications, a centralized system with an accumulation tank at the medium temperature level and a parallel system with two separate circuits for low and medium temperature, levels were investigated by Sawalha [8]. He showed that a two-stage centralized system provides the best COP for the selected ambient temperature range. The effect of adding a subcooled compression on the performance of a CO_2 trans-critical cycle parallel compression system for supermarket refrigeration applications under different ambient conditions was investigated by Wang et al. [9]. Their simulation showed a 7.1% reduction in total energy consumed by the system when the gas cooler and receiver pressure were optimum.

The performance of a CO_2 trans-critical cycle heat pump for a detailed geometrical variation in the gas cooler and evaporator was simulated by Lin et al. [10]. The effect of dry bulb temperature, relative humidity, inlet water temperature, compressor speed, and the capillary tube length were investigated and reported. They concluded that the COP of the cycle improves by increasing the dry bulb temperature or the evaporator's inlet relative humidity but reduces with compressor speed.

A comprehensive overview of many advancements in modifying the essential CO_2 trans-critical cycle performance under different operating parameters and conditions has been summarized by Shan [11]. The basic principle of the CO_2 trans-critical cycle and the significance of each operating parameter on the performance have also been discussed. Another review study by Ma et al. [12] presented an overview of the CO_2 trans-critical refrigeration cycle and heat pump systems, analyzed some essential cycle characteristics, and provided a comparative performance analysis of several novel trans-critical cycles.

The effectiveness of various CO_2 trans-critical cycle configurations, i.e., an internal heat exchanger, a parallel compression system, a two-stage compression system, and a system with mechanical subcooling after the gas cooler, has been investigated and compared with the essential CO_2 trans-critical cycle by Bellos and Tzivanidis [13]. They concluded that the system with mechanical subcooling is the most efficient configuration (with the highest COP) among the others. Nakagawa et al. [14] showed that an enhancement of 27% in COP over the basic cycle could be achieved when employing IHX in the CO_2 trans-critical refrigeration cycle with a two-phase ejector.

Despite these documented studies on the performance of the CO_2 trans-critical cycle, the combined effect of compressor efficiency and evaporator inlet vapor quality has not been comprehensively investigated. This paper aims to fill this knowledge gap by providing a systematic simulation study on the combined effect of compressor efficiency (η_C) and vapor quality at the evaporator inlet. Moreover, the effects of evaporation temperature (T_{evap}), degree of subcooling (ΔT_{sub}), degree of superheating (ΔT_{sup}), gas cooler pressure (P_{gc}), gas cooler outlet temperature (T_{gco}), and ambient temperature (T_{amb}) on the cycle COP, cooling capacity (CC), and compressor power consumption (PC) will be investigated and analyzed. Hence, the current study will contribute to a fundamental understanding of the CO_2 trans-critical cycle under various working parameters. In the following, we briefly describe the implementation of our model to the present cycle, discuss and conclude the simulation results, and elucidate the interplay effects of the cycle variables.

CYCLE MODELLING AND VALIDATION

In this section, the CO_2 trans-critical cycle for air conditioning applications is modelled and validated against previous experimental study [14]. The proposed cycle, shown schematically in Figure 1, includes a compressor, gas cooler with subcooling, expansion valve, and evaporator with superheating. The mathematical model satisfies mass, momentum, and energy conservation principles through the cycle under various steady-state flow operating conditions.

The pressure-enthalpy (P-h) diagram and the temperature-entropy (T-s) diagram for the CO_2 trans-critical cycle with the main states are shown in Figure 2 and Figure 3, respectively. The cycle consists of a low-pressure side where two-phase evaporation takes place and a trans-critical high-pressure side where single-phase gas cooling takes place at pressures independent of the outlet temperature of the gas cooler.

During compression process the total amount of power consumed (PC) (in kW) can be given by:

$$PC = \dot{m}_R \left(h_{2'} - h_{1'}\right) / \eta_C \tag{1}$$

In Equation 1 $\dot{m}_R$ represents the amount of refrigerant flow rate (in kg/s), η_C is the isentropic compressor efficiency, $h_{1'}$ and $h_{2'}$ are the enthalpies of the refrigerant at the inlet (state 1') and the exit (state 2') to the compressor (in kJ/kg) respectively. The cooling capacity (CC) of the cycle evaporator with subcooling and super-heating (in kW) can be given by Equation 2:

$$CC = \dot{m}_R \left(h_{4'} - h_{1'}\right) \tag{2}$$

Where $h_{4'}$ represents enthalpy of the refrigerant (in kJ/kg) at the evaporator inlet (state 4'). However, the CC of the cycle evaporator without subcooling and superheating can be written as:

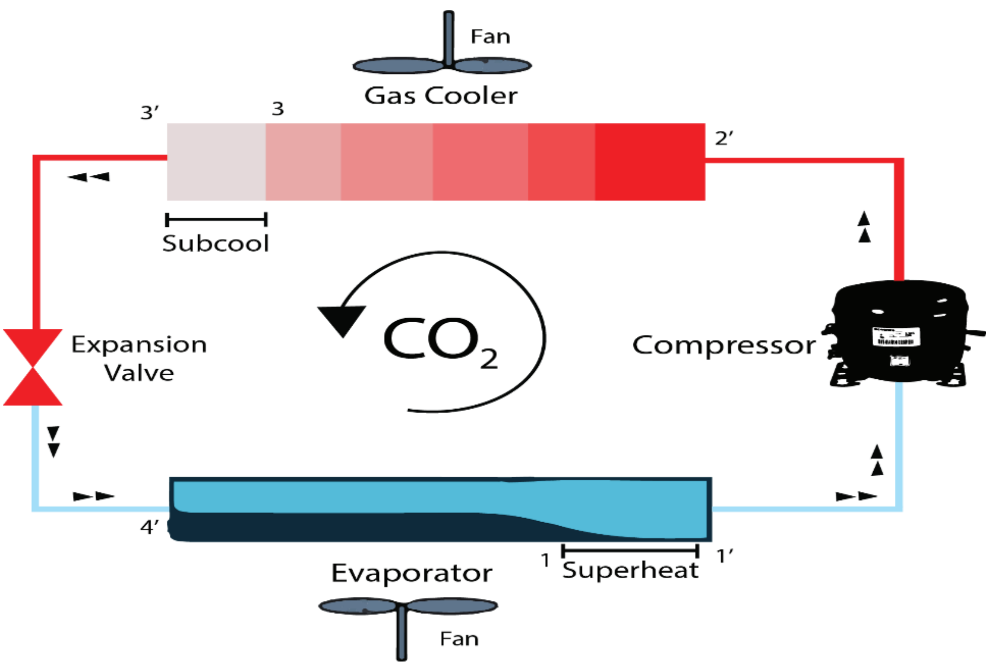

Figure1. Schematic diagram of the CO_2 trans-critical cycle.

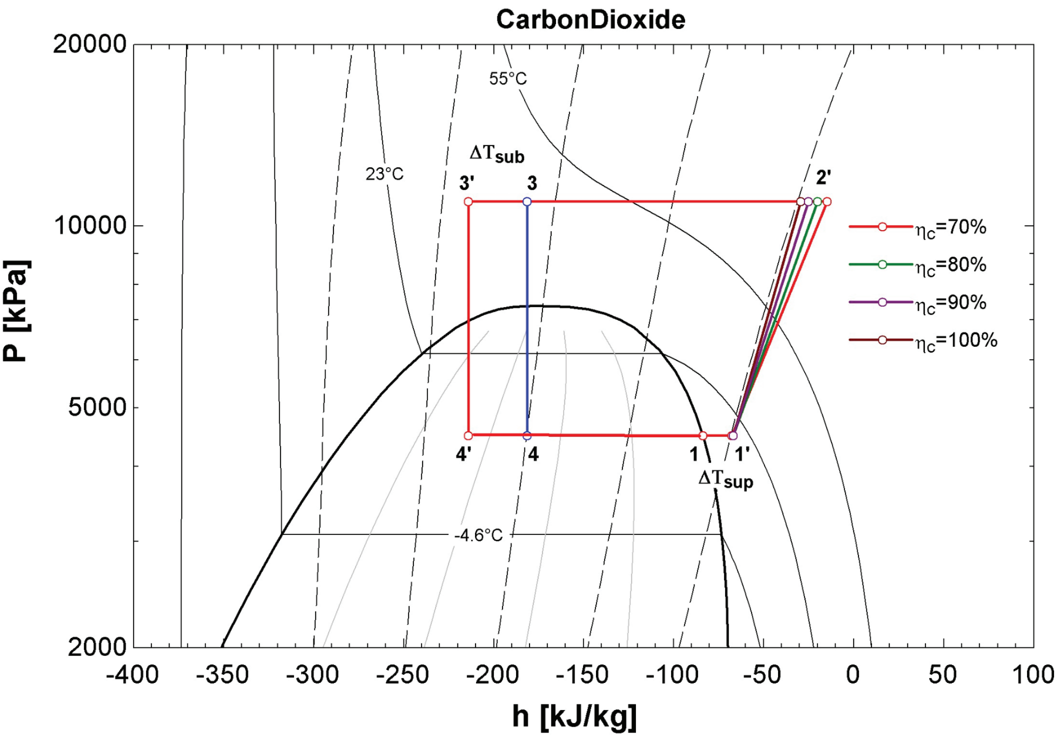

Figure 2. The P-h diagram of the CO_2 trans-critical cycle.

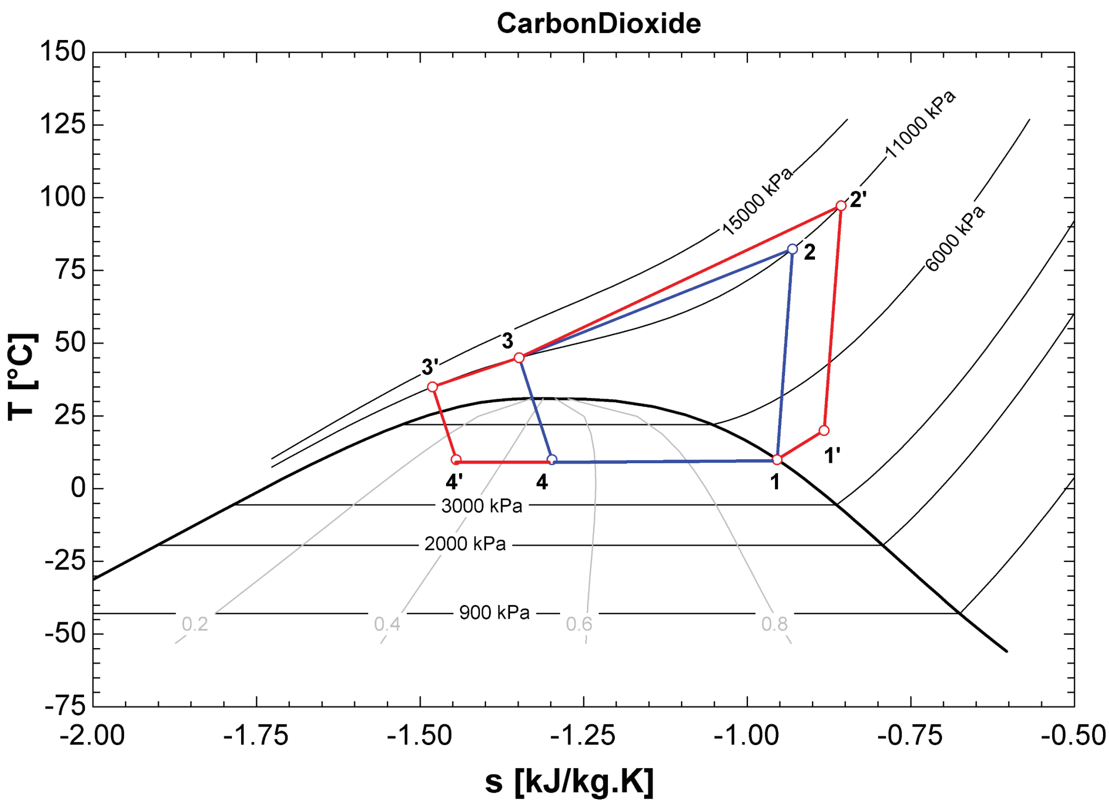

Figure 3. The T-s diagram of the CO_2 trans-critical cycle.

$$\mathrm{CC} = \dot{m}_R \left(h_4 - h_1\right) \tag{3}$$

In Equation 3 above h_1 and h_4 represent the enthalpies of the refrigerant (in kJ/kg) at inlet and exit to the evaporator without subcooling and superheating (state 1 and 4) respectively. The degrees of superheat (ΔT_{sup}) and sub-cool (ΔT_{sub}) can be respectively written as:

$$\Delta T_{\mathrm{sup}} = \left(T_{1'} - T_1\right) \tag{4}$$

$$\Delta T_{\mathrm{sub}} = (T_3 - T_{3'}) \tag{5}$$

The COP of the CO_2 cycle which represents the ratio between CC and PC can be calculated as:

$$\mathrm{COP} = \frac{\mathrm{CC}}{\mathrm{PC}} = \frac{\eta_C \left(h_{4'} - h_{1'}\right)}{\left(h_{2'} - h_{1'}\right)} \tag{6}$$

Vapor quality (x_4) at the inlet of the evaporator can be evaluated using the following equation:

$$x_4 = \left(h_{4'} - h_f\right) / h_{fg} \tag{7}$$

In the previous Equation 7, h_f and h_{fg} represent saturated liquid enthalpy and enthalpy of vaporization at evaporator pressure, respectively.

The approach temperature (T_{app}) is defined as the difference between the ambient temperature (T_{amb}) and the CG outlet temperature (T_{gco}):

$$T_{app} = T_{amb} - T_{gco} \tag{8}$$

For all simulations, enthalpic expansion is assumed i.e., $h_3 = h_4$ and $h_{3'} = h_{4'}$.

Using Engineering Equation Solver (EES) and MATLAB codes, the mathematical formulations presented previously have been employed to correctly simulate the combined effect of the compressor isentropic efficiency and vapor quality on the cycle performance. The amount of CC, PC, and the COP of the cycle at different values of gas cooler outlet temperatures and pressures, evaporation temperatures, and ambient temperatures are simulated. Moreover, the effect of various amounts of superheating and subcooling has also been presented. Table 1. summarizes thermodynamics constraints used in the EES model, assuming that steady flow processes through the cycle, no pressure drops are presented in the evaporation and the gas cooler, and negligible heat loss to the surrounding. The customized operating conditions are listed in Table 2.

The present model has been validated against an experimental study by Nakagawa et al. [14]. Figure 4 compares the COPs for conventional CO_2 trans-critical cycle obtained from the current model and experiment at different values

Table 1. Thermodynamics constraints used in EES model to simulate the CO_2 trans-critical cycle

State 1	State 2	State 3	State 4
$T_1 = T_{evap}$	$s_{2s} = s_{1'}$	$T_3 = T_{gco}$	$h_4 = h_{3'}$
$x_1 = sat.vapor$	$P_2 = P_{gc}$	$P_3 = P_2$	$P_4 = P_1$
$P_1 = P\left(T_{evap}, x_1\right)$	$h_{2s} = h\left(s_{2s}, P_2\right)$	$h_3 = h\left(T_3, P_3\right)$	$T_4 = T_1$
$h_1 = h\left(P_1, x_1\right)$	$h_2 = h_{2'} + \left(h_2 - h_{2'}\right) / \eta_C$	$s_3 = s\left(T_3, P_3\right)$	$x_4 = x\left(T_4, h_4\right)$
$s_1 = s\left(P_1, x_1\right)$	$s_2 = s\left(h_2, P_2\right)$	$T_{3'} = T_3 + \Delta T_{sub}$	$s_4 = s\left(T_4, h_4\right)$
$T_{1'} = T_1 + \Delta T_{\mathrm{sup}}$	$T_2 = T\left(s_2, P_2\right)$	$P_{3'} = P_3$	
$P_{1'} = P_1$		$h_{3'} = h\left(T_{3'}, P_{3'}\right)$	
$h_{1'} = h\left(T_{1'}, P_{1'}\right)$			
$s_{1'} = s\left(T_{1'}, P_{1'}\right)$		$s_{3'} = s\left(T_{3'}, P_{3'}\right)$	

Table 2. Customized operating conditions

Par.	**CC (kW)**	T_{evap} **(°C)**	ΔT_{sup} **(°C)**	P_{gc} **(kPa)**	T_{gco} **(°C)**	T_{amb} **(°C)**	ΔT_{sub} **(°C)**	η_C **(%)**
Value	1	0 ~ 20	0 ~ 18	7500 ~ 17000	35- 45	30 ~ 40	0 ~ 24	70 ~ 100

gas cooler pressures. Under identical operation conditions i.e., T_{evap} = 0 °C, T_{gco} = 42 °C, ΔT_{sub} = ΔT_{sup} =0 °C, and η_C = 50%, our model displays a good agreement in COP behaviour to the one exists in the experiment with slightly higher COP than the experimental values for P_{gc} below 9800 kPa and slightly lower COP when the P_{gc} higher that 10100 kPa. An average inaccuracy of about 5.74% is observed when compared to experiment.

To further validate our model, we also compared the variation of COP with gas cooler outlet temperatures (T_{gco}) under similar operating conditions. Figure 5 shows identical trend of our model when compared to experiment with

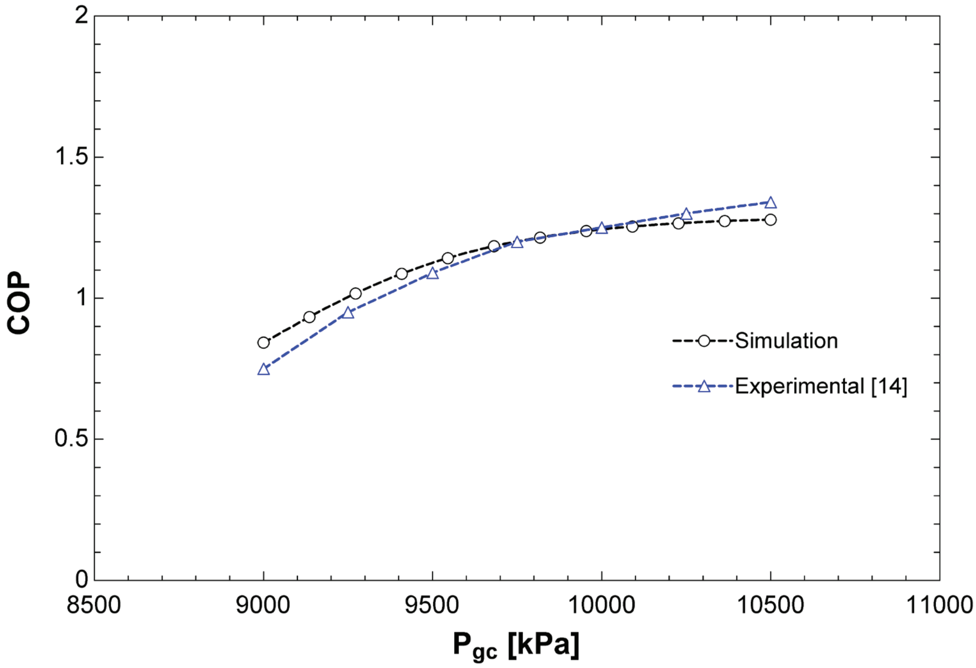

Figure 4. Variation of COP of the cycle versus P_{gc} compared to experiment by Nakagawa et al. [14].

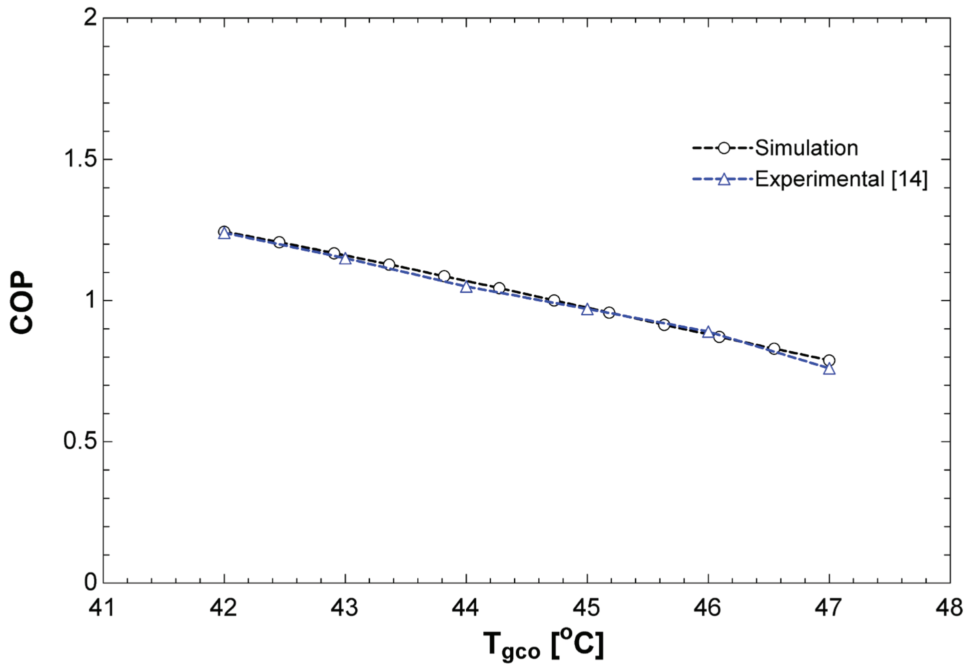

Figure 5. Variation of COP of the cycle versus T_{gco} compared to experiment by Nakagawa et al. [14].

an average error of 1.93%. The slight deviation between the present model and experimental data may be attributed to specific circumstances, conditions, or factors aligning with the assumptions or simplifications embedded in the model. Also, the experimental study and the model concentrate on a limited range of parameters where irreversibility effects are marginal, the model's alignment with experimental data could be happenstance within that specific range. The Experimental setups, intentionally or not, might operate in a manner that minimizes the impact of irreversibilities. This can lead to the surprisingly accurate performance of the simplified model within the specific experimental context. On the other hand, certain operational or design aspects dominating the system's overall performance could diminish the influence of irreversibilities. In such scenarios, the model might effectively capture the primary factors steering variations in COP.

RESULTS AND DISCUSSION

Despite earlier investigations into different facets of its performance, the combined influence of two essential factors specifically, compressor efficiency and evaporator inlet vapor quality has not received comprehensive scrutiny in prior studies. This research distinguishes itself by seeking to conduct a thorough examination of the synergistic effects produced by these two parameters through a systematic simulation study. Using the current model, the combined effect of compressor efficiency and vapor quality to the evaporator inlet has been comprehensively studied under various operating conditions (Table 1, 2). Figure 6 shows the effect of P_{gc} on the CC of the cycle at different values T_{evap}. The magnitude of CC gradually increases with P_{gc}, after that when P_{gc} > 11000 kPa the enhancement in CC slows down. For different values of T_{evap}, the CC of the cycle with P_{gc} demonstrates identical trend with enhanced CC as T_{evap} reduces. By raising P_{gc} from 100 to 150 bar and lowering T_{evap} from 18 to 10 °C, a 66% rise in CC is seen. In contrast, the CC of the cycle decreases with vapor quality entering the evaporator. Figure 7 clearly shows that CC of cycle linearly drops versus vapor quality to the evaporator at different values of T_{sup} with enhanced CC at higher values of T_{sup}. By lowering ΔT_{sup} from 12 to 0 °C and raising vapor quality from 0.1 to 0.5, a 52.4% reduction in CC occurs.

Figure 8 displays PC during compression process versus compressor efficiency at various values of P_{gc}, increasing compressor efficiency and reducing P_{gc} will reduce the total amount of PC during compression process. Similar trend in PC variation against compressor efficiency can be seen for all values of P_{gc}. By increasing compressor efficiency from 70 to 100% and decreasing P_{gc} from 110 to 80 bar, an approximate 50% reduction in PC is seen. However, the total amount of PC during compression process can be reduced by increasing T_{evap}. Figure 9 shows PC versus compressor efficiency at various T_{evap}, it is clear that by raising T_{evap} from 10 to 18 °C and efficiency from 70 to 100%, it is discovered that PC will drop by 42.7% of its original value.

An improvement in the cycle COP with higher compressor efficiency and higher values of ΔT_{sub} is clearly seen in Figure 10. Both higher values of compressor efficiency and higher values of ΔT_{sub} have favourable effects on the cycle COP i.e., increasing compressor efficiency reduces PC

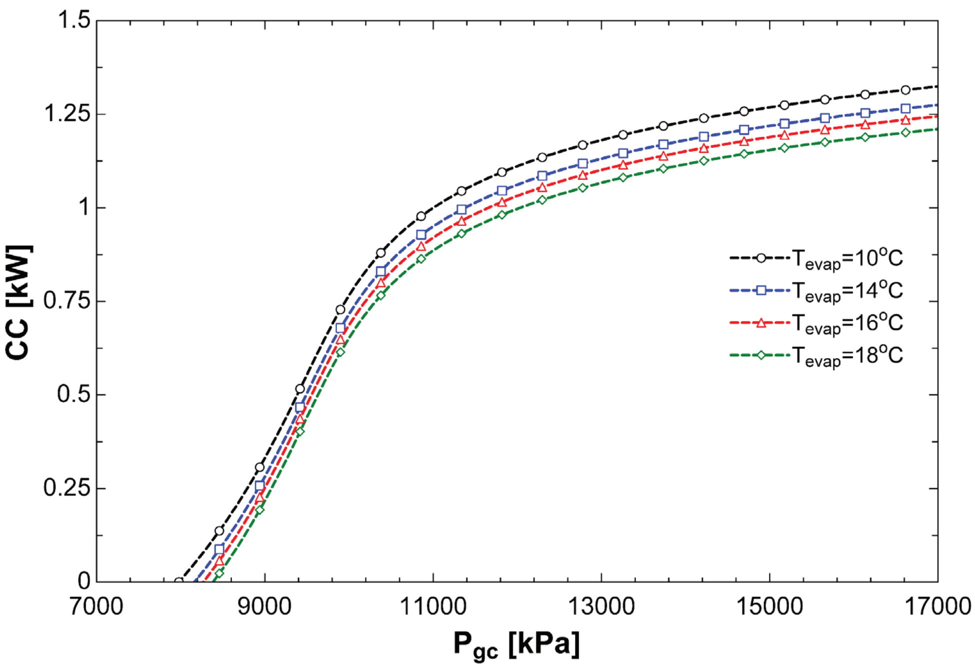

Figure 6. Variation of CC versus P_{gc} at different value of T_{evap}.

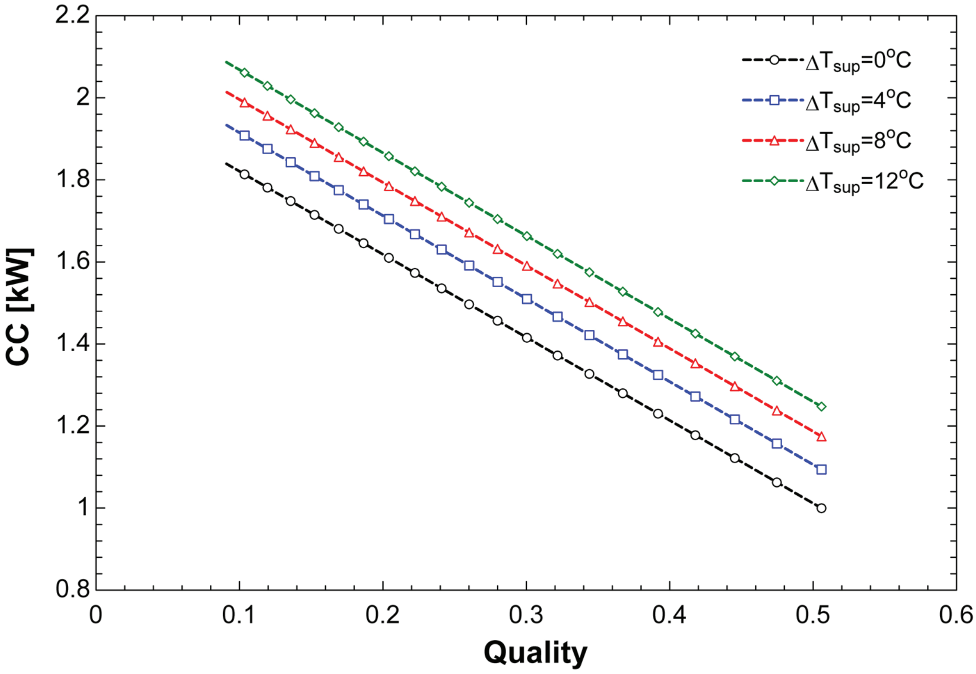

Figure 7. Variation of CC versus quality to evaporator at different values of ΔT_{sup}.

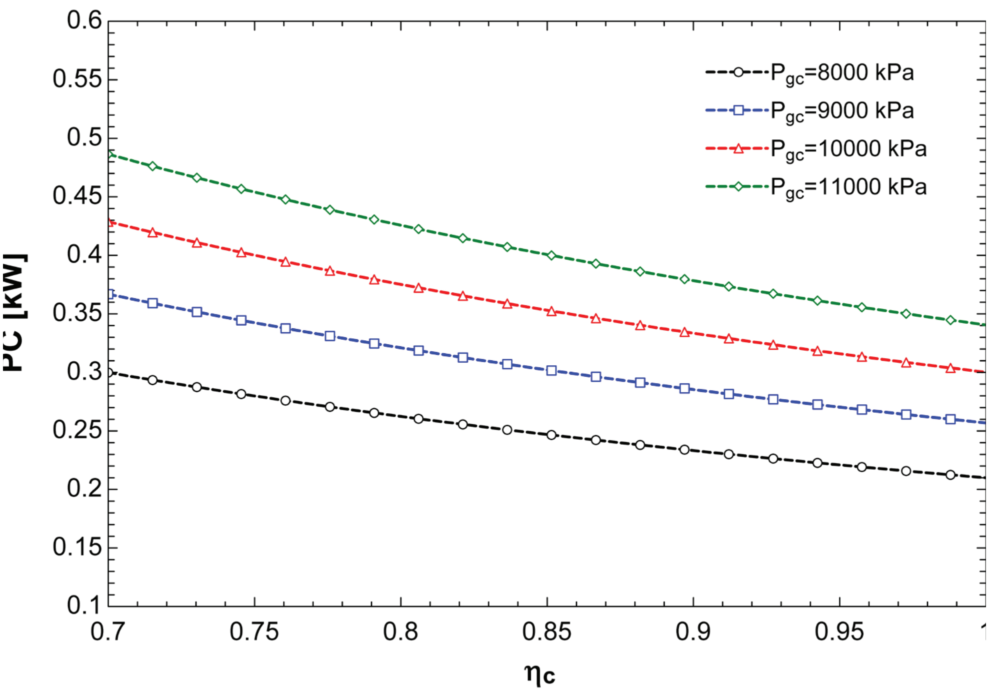

Figure 8. Total amount of PC during compression process versus compressor isentropic efficiency at different values of P_{gc}.

and increasing ΔT_{sub} increases CC. On the other hand, the improvement in CC with increasing ΔT_{sup} is overweighted by the increase in PC during compression process which will reduces the overall COP of the cycle. It shows that by increasing compressor efficiency from 70 to 100%, raising ΔT_{sub} from 0 to 6 °C, and lowering ΔT_{sup} from 12 to 0 °C, an improvement in COP of roughly 111.7% is seen.

The effect of compressor efficiency and T_{gco} is shown in Figure 11. Since reducing T_{gco} increases the cycle CC while increasing compressor efficiency reduces the total

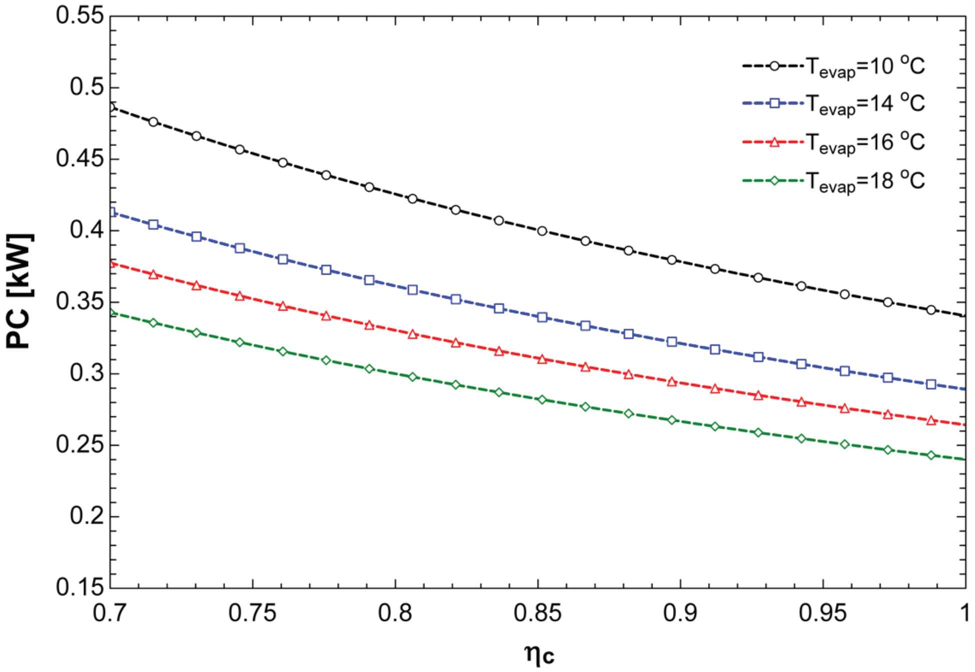

Figure 9. Total amount of PC during compression process versus compressor isentropic efficiency at different values of T_{evap}.

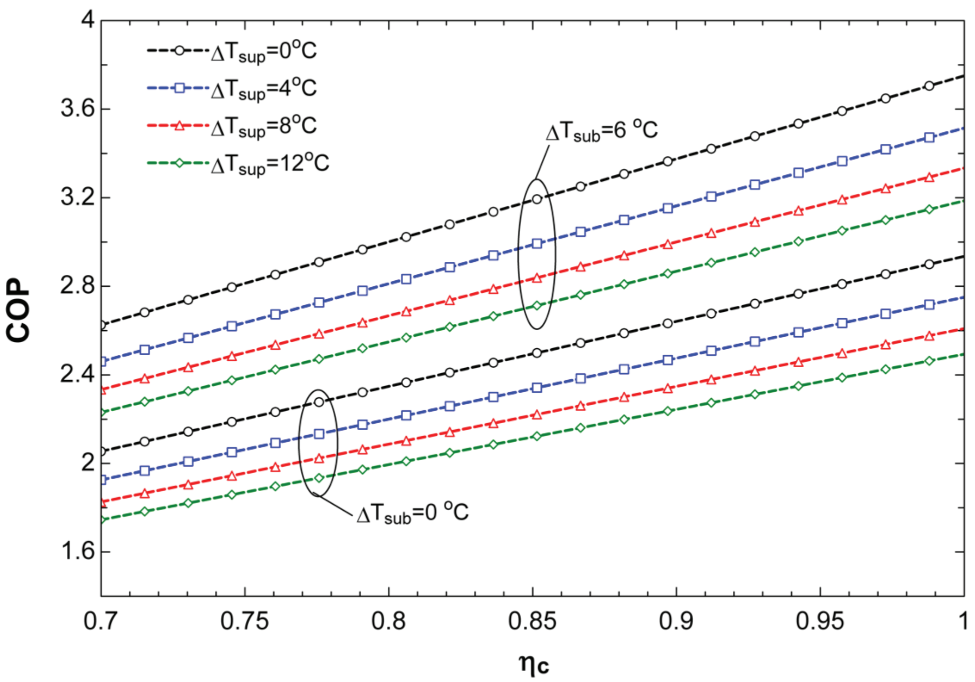

Figure 10. Cycle COP versus compressor efficiency at different values of ΔT_{sub} and ΔT_{sup}.

PC during compression process then both will positively enhance the COP of the cycle. it can be inferred that lowering T_{gco} and raising compressor efficiency will increase the COP of the cycle. Also, when T_{gco} is reduced from 45 to 37.5 °C and compressor efficiency is raised from 70% to 100%, cycle COP is observed to increase by nearly 90%.

The effect of P_{gc} on cycle COP at different values of compressor efficiency and ΔT_{sub} is shown in Figure 12. For different values of compressor efficiency and ΔT_{sub}, an optimum value of P_{gc} for maximum COP can be observed. For all simulations in Figure 12 the COP of the cycle enhances rapidly when P_{gc} is less than the optimum value and then decline

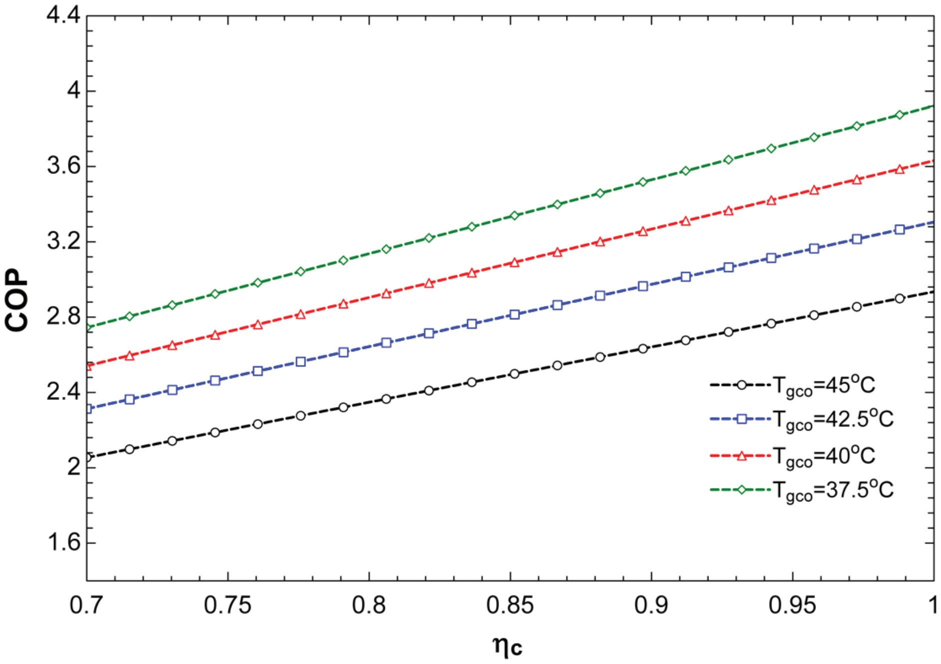

Figure 11. Cycle COP versus compressor efficiency at different values of T_{gco}.

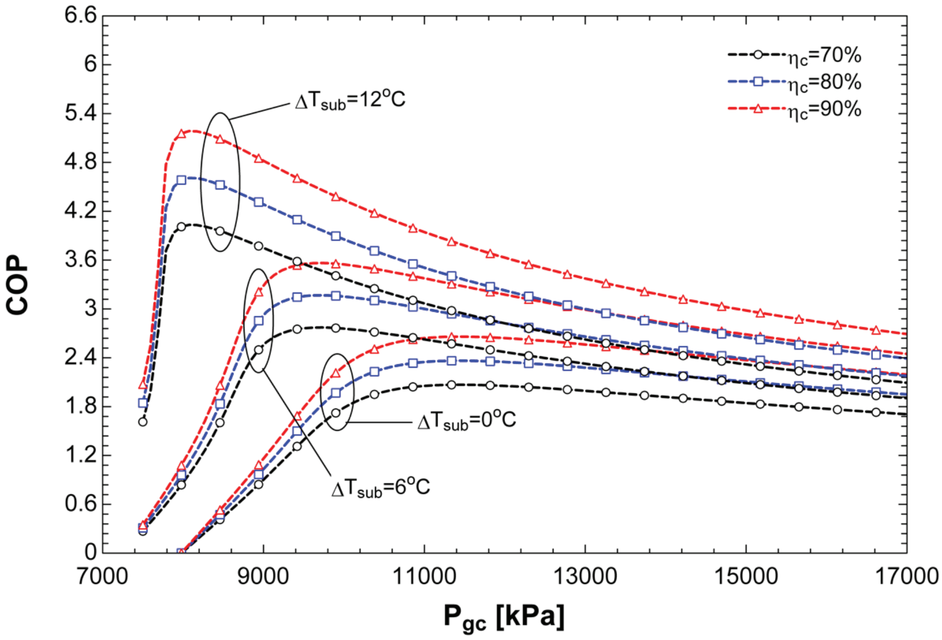

Figure 12. Cycle COP versus P_{gc} at different values of compressor efficiency and ΔT_{sub}.

gradually when P_{gc} is greater than the optimum value of P_{gc}. Furthermore, this trend of the cycle COP with P_{gc} is more obvious for higher values of ΔT_{sub}. By increasing P_{gc} from 75 to 85 bar, ΔT_{sub} from 0 to 12 °C, and compressor efficiency from 70% to 100%, a value of COP of 5.2 can be attained.

Figure 13 illustrates how COP varies with vapor quality for various ΔT_{sup} both increasing vapor quality and ΔT_{sup} reduce COP. It is shown that by raising vapor quality from 0.1 to 0.5 and increasing ΔT_{sup} from 0 to 12 °C, a drop in COP of roughly 87% is seen.

Figure 14 displays the combined effect of vapor quality and P_{gc} on COP for various compressor efficiencies. Increasing vapor quality and P_{gc} both result in a drop in cycle COP. The improvement of COP with compressor

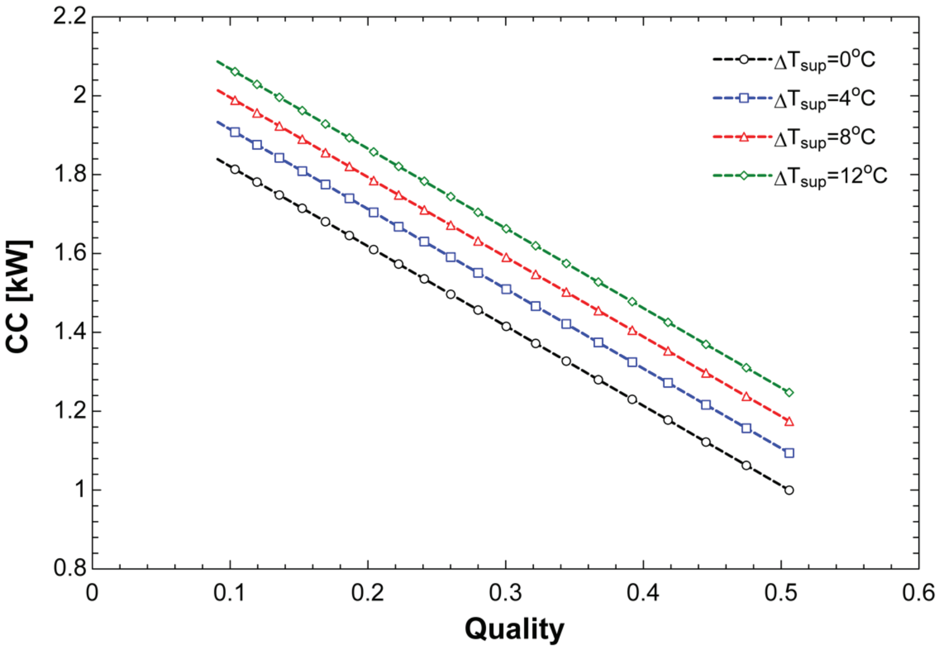

Figure 13. Variation of COP with vapor quality for different ΔT_{sup} in (° C).

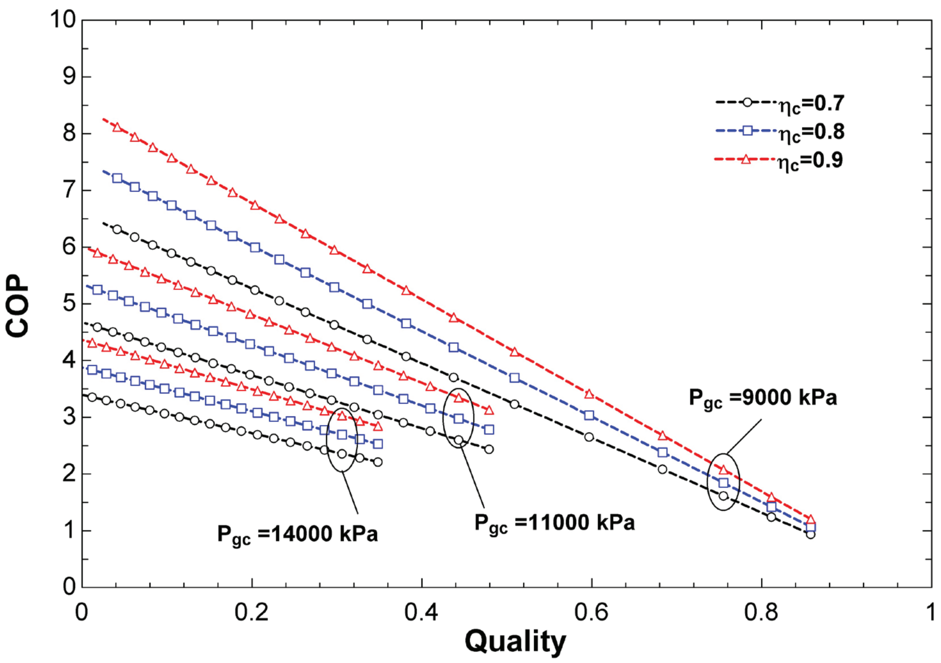

Figure 14. Cycle COP versus vapor quality at different P_{gc} and compressor efficiency.

efficiency become smaller by increasing quality and reducing P_{gc}. But the significant improvement of COP is shown by increasing compressor efficiency.

The COP of the cycle can be changed by raising either ΔT_{sup} or ΔT_{sub} respectively. It is obvious from Figure 15, the combined effect of ΔT_{sub} and ΔT_{sup} on cycle COP. The rising ΔT_{sup} from 0 to 18 °C and lowering ΔT_{sub} from 18 to 0 °C shows a drop of roughly 42.1%.

Figure 16 displays the combined variations of COP with ΔT_{sub} and compressor efficiency. Increasing both ΔT_{sub} and compressor efficiency increase cycle COP. Enhancing COP in a trans-critical refrigeration cycle involves optimizing

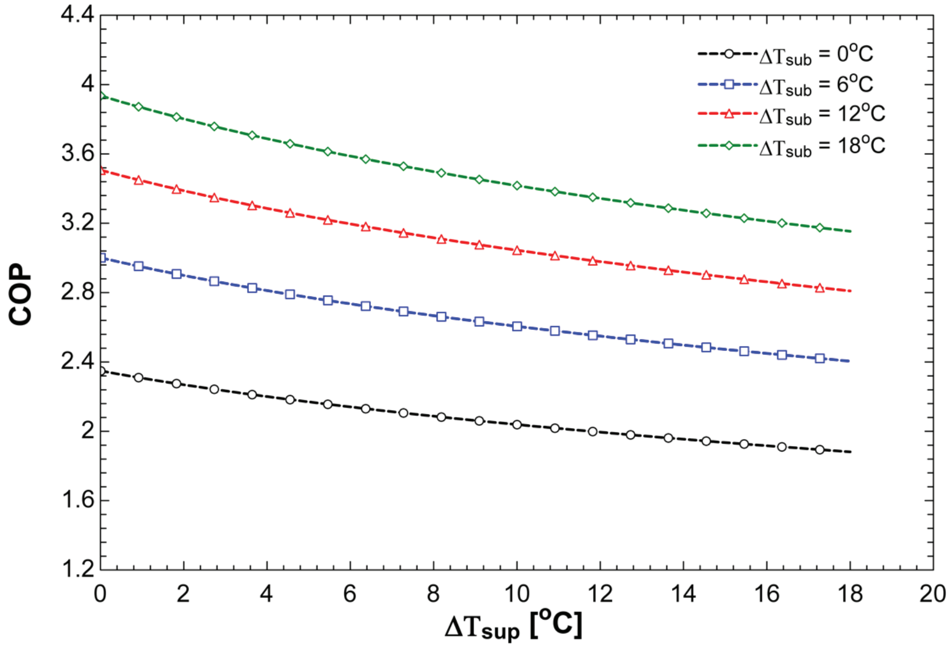

Figure 15. Cycle COP versus ΔT_{sub} at different values of ΔT_{sup}.

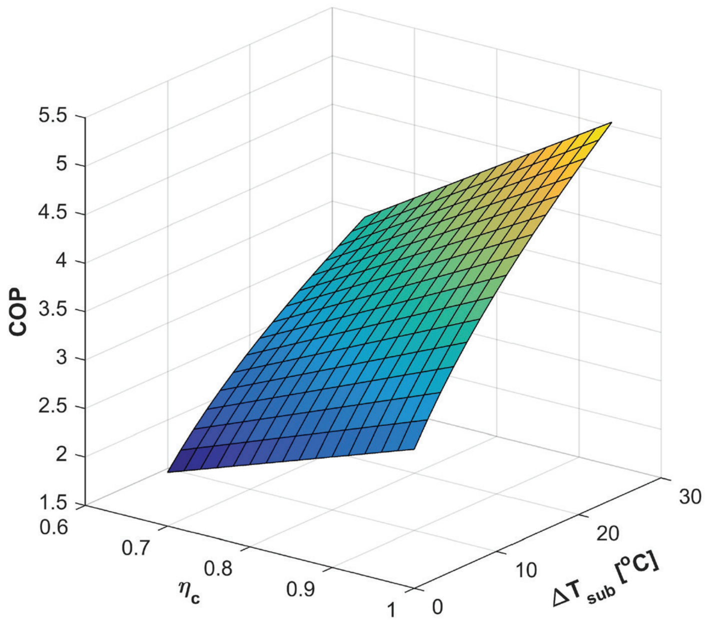

Figure 16. Variation of COP with ΔT_{sub} and isentropic compressor efficiency.

both compressor efficiency and the degree of subcooling. Greater compressor efficiency reduces energy consumption, allowing the system to generate more cooling output with the same energy input, thus increasing the COP. Increased subcooling is advantageous as it improves the overall heat rejection process in the condenser, leading to higher liquid density. This, in turn, enables the refrigerant to absorb more heat during the evaporation process in the evaporator. Simultaneously improving both compressor efficiency and subcooling creates a synergistic effect, with the compressor operating more efficiently to raise refrigerant pressure, and heightened subcooling optimizing heat transfer characteristics. This combined enhancement results in a more effective refrigeration cycle, yielding a higher COP. The system attains improved energy efficiency by maximizing cooling performance for a given energy input.

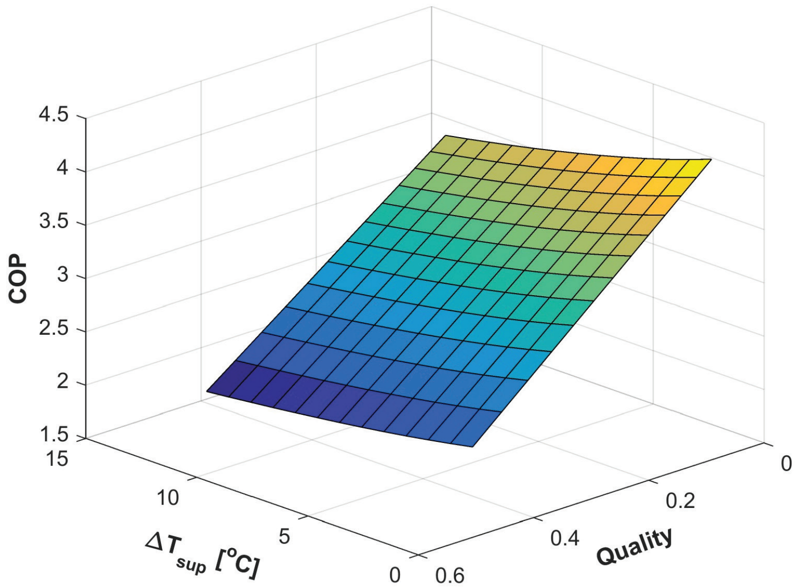

Figure 17. Variation of COP with ΔT_{sup} in ($^\circ$C) and vapor quality.

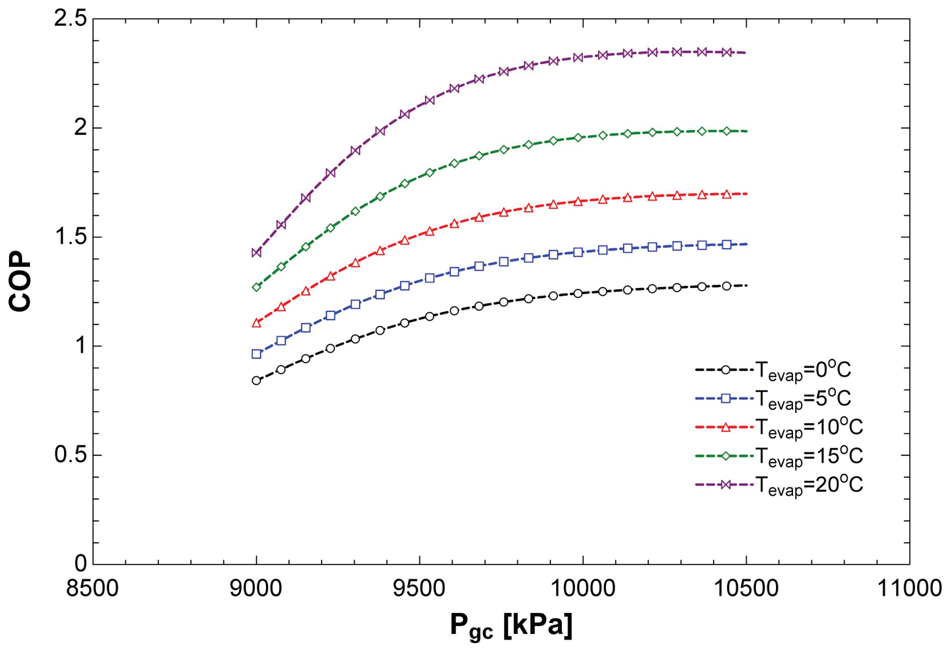

Figure 18. Variation of COP with P_{gc} for different values of T_{evap}.

Figure 17 shows the combined effect of ΔT_{sup} and vapor quality on cycle COP. It can be noticed from this figure that increasing both ΔT_{sup} and vapor quality reduces cycle COP. Superheating, when excessive, can lead to an increase in the energy required to bring the refrigerant back to its saturated state during the cooling process. This additional energy input without a corresponding increase in useful cooling output reduces the overall efficiency of the refrigeration cycle, leading to a decrease in COP. Higher vapor quality implies a greater proportion of vapor and less liquid in the refrigerant mixture. While some degree of vaporization is necessary for effective heat absorption, excessively high vapor quality can result in reduced mass flow rate of refrigerant, leading to a decline in overall heat transfer efficiency. This inefficiency contributes to a decrease in COP. Excessive superheating increases the energy required for the phase change during cooling, while high vapor quality diminishes the efficiency of heat absorption. The combined

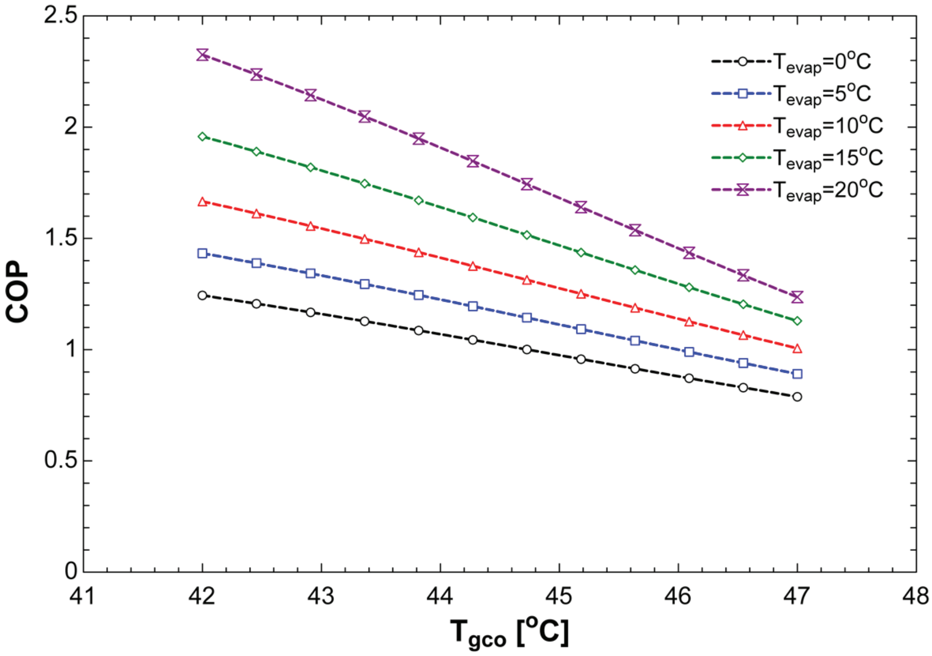

Figure 19. Variation of COP with T_{gco} for different values of T_{evap}.

result is a less efficient refrigeration cycle, translating to a reduced COP. The system expends more energy for a given cooling output, indicating decreased overall performance.

Figure 18 illustrates the variation in COP as a function of P_{gc} for various T_{evap}; it could be inferred that raising P_{gc} and T_{evap} will enhance the cycle COP. Figure 19 displays the variation of COP with T_{gco} for various T_{evap}; it shows that raising T_{gco} causes a decrease in cycle COP, while raising T_{evap} causes an increase.

CONCLUSION

Fundamentally, the novelty of this research lies in its targeted exploration of an underexplored aspect within the realm of the CO_2 trans-critical cycles, thereby making a significant contribution to the advancement of knowledge in this field. The combined effect of compressor efficiency and vapor quality on the CO_2 trans-critical cycle performance has been investigated under various operating conditions. This simulation study showed that for a given combination of performance and operational parameters, there are optimal gas cooling pressure values at which the cycle COP attains a maximum value. At P_{gc} of 80 bars, ΔT_{sub} of 12 °C, and compressor efficiency of 90%, a maximum value of COP of 5.2 is attained. A significant boost in COP was produced by lowering vapor quality, raising compressor efficiency, and raising sub-cooling levels. An increase in the degree of superheating, an increase in vapor quality, and a decrease in isentropic compressor efficiency caused a significant drop in COP. While increasing gas cooling pressure and lowering vapor quality are helping boost cycle cooling capacity. In addition to the rising evaporation temperature, improving the compressor's efficiency and lowering the gas cooling pressure will reduce the cycle power consumption. Increasing compressor efficiency from 70 to 100% and lowering gas cooling pressure from 110 to 80 bar reduces in power consumption by about 50%. Overall, in this study an increase in COP of approximately 111.7% is observed.

NOMENCLATURE

C	Celsius
CC	Cooling Capacity (kW)
COP	Coefficient of performance
h	Enthalpy (kJ/kg)
$\dot{m}_R$	Refrigerant mass flow rate in (kg/s)
P	Pressure (kPa)
PC	Power Consumption (kW)
R	Refrigerant
s	Entropy (kJ/kg.°C)
T	Temperature (°C)
x	Quality (%)

Superscript

°	Degree

Greek Letters

Δ	Difference
η	Efficiency

Subscripts

amb	Ambient
app	Approach

evap Evaporation
gc Gas Cooler
gco Gas Cooler Outlet
isc Isentropic efficiency of compressor
sub Subcooling
sup Superheating

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AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

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