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PERFORMANCE ANALYSIS OF A NOVEL ENHANCED TRANSCRITICAL N2O REFRIGERATION CYCLE FOR POWER AND COLD GENERATION

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CANDIDATE'S DECLARATION

I RICHA KESARWANI, ROLL No-2k19/THE/17 student of M.Tech (Thermal Engineering), hereby declare that the project dissertation titled "Performance Analysis Of A Novel Enhanced Transcritical N2O Refrigeration Cycle For Power And Cold Generation" which is submitted by me to the Department of Mechanical Engineering, Delhi Technological University, Delhi in partial fulfilment of the requirement for the award of the degree of the Master of technology, is original and not copied from any source without proper citation. This work has not previously formed the basis for the award of any Degree, Diploma Associateship, Fellowship or other similar title or recognition

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CERTIFICATE

I hereby certify that the project Dissertation titled "Performance Analysis Of A Novel Enhanced Transcritical N2O Refrigeration Cycle For Power And Cold Generation" which is submitted by [RICHA KESARWANI], Roll no 2k19/THE/17 [Department of mechanical Engineering], Delhi Technological University, Delhi in partial fulfilment of the requirement for the award of degree of Master of Technology, is a record of the project work carried out by the student under my supervision. To the best of my knowledge this work has not been submitted in part or full for any Degree or Diploma to this university or elsewhere.

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ABSTRACT

Performance evaluation of the three special transcritical refrigeration cycles, specifically the vapor compression cycle (VCC), the combined refrigeration cycle (CRC) and the new combined refrigeration cycle for power and cold generation (NCRCPC), are studied. The principles of mass and energy conservation are implemented to every devices of the cycle, and the resulted linear system of equations was numerically solved. It was investigated that the new proposed cycle had a lesser optimum gas cooler pressure than VCC and CRC refrigeration cycles, and this would enhance the lifetime and safety of the system. It was found that the COP of the new cycle is more than that of the vapor compression and combined refrigeration cycles 118.73% and 49.146%, respectively. Besides, The exergy efficiency was approximately 64.55% and 56.68% higher than that of the vapor and combined refrigeration cycles, respectively.

If we compare between N2O and CO2 refrigerant, the N2O based refrigeration cycle is found to have better coefficient of performance (COP), lower gas cooler pressure and higher entrainment ratio (u). For N2O based refrigeration cycle maximum COP is found to be more than that of CO2 based refrigeration cycle. Exergetic output of N2O ejector cycle is better while losses passed off because of irreversibility during expansion is lower.

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Nomenclature

Symbols

T = temperature

P = pressure

COP = coefficient of performance

h = specific enthalpy(kJ/kg

m= mass flow rate (kg/s)

Q = heat transfer rate (kJ/kg)

S = specific entropy (kJ/kgK)

U = entrainment ratio

W = electrical power (kW)

E = exergy rate (kW)

$ED,_{tot\ total}$ = exergy destruction cost rate (kW)

Greek Symbols

η = efficiency (%)

Abbreviations

is	Isentropic
gc	gas cooler
EV	expansion valve
evap	Evaporator
comp	compressor
cond	Condenser
Turb	Turbin
tot	total value

Subscripts

0 = ambient condition

t = total

1,2,3,4... = cycle locations (kJ/kg)

k = each component

in = Inlet

out = Outlet

F = Fuel

P = Product

CHAPTER 1
INTRODUCTION

INTRODUCTION

Combined cycles for refrigeration and producing energy had been highlighted for the final a long time for growing structures efficiency. Different combined cycle kinds for cooling and power production were proposed to obtain higher overall performance with a considerable reduction in electrical energy consumption. The combined cycles for refrigeration and power generation have been appreciably studied via way of means of numerous researchers with inside the beyond a long time.

The Goswami cycle that combined a power Rankine cycle with absorption cycles for producing energy and refrigeration turned into proposed and studied [1]. Many research works presented a parametric, exergy analysis for this cycle [2–4], and its feasibility was experimentally validated [5-6]. Another single-stage combined cycle with higher exergy and energy efficiencies was proposed and offered with inside the literature [7-8].

Many studies paintings has been carried out on using ejector technology and their applications in thermal systems, and a detailed assessment of such works turned into offered [9]. Other combined cycle for higher energy and exergy efficiencies and better cooling capacity was proposed the use of the Kalina thermodynamic cycle [10–14]. Even though the existed combined refrigeration cycles can could have reasonably higheroverall performance and energy efficiencies than the one of the traditional cycles, they may be nonetheless complex and pertain excessive capital cost.

Ejector refrigeration cycles (ERC) are any other thermally pushed refrigeration technology used in lot of co-generation and tri-generation systems with the benefits of ease of conception, low running costs since there are no moving parts and they may be operated the usage of numerous refrigerants.

A combined Rankine and ejector cycles had been advanced and notably studied to make use of the low grade and wasted heat. Exergy and thermodynamic evaluation of those cycles and systems indicated that the system component with the maximum exergy destruction is the ejector [15]. New system configurations evaluation of those combined cycle and better values of thermal and exergy performance [16]. Besides, hybrid vapor compression refrigeration cycles and systems had been delivered and examined, and a parametric take a look at in their thermodynamic overall performance the usage of energy and exergy evaluation changed into conducted [17]. Some devices with excessive exergy destruction were either removed or improved to enhance the system efficiency and cycle overall performance. Organic Rankine cycle (ORC) was integrated into a system with ERC running with distinctive sorts of refrigerants for power generation and enhancement in the refrigeration system thermodynamic overall performances [18–21].

Several research works had been carried out to study the effects of working fluid properties on the performance characteristics and efficiencies of the combined cycle. Also, the selection of the best working fluid for a given refrigeration cycle turned into based on specific criteria including but not limited to safety, high COP, low Global Warming Potential (GWP), toxicity, low Ozone Depletion Potential (ODP), and environment-friendly properties [22-23]. In this regard, the N2O refrigeration systems are in a enormous expansion stage to substitute artificial refrigerants currently applied inthe refrigeration and air conditioning applications. Also, the main advantage of using N2O as a refrigerant is that it has a very low GWP, ODP and nontoxic. The advancement in developing refrigeration cycles working with N2O as a refrigerant made it more profitable, especially with the integration of ejector and expanders into the refrigeration system, which made it possible to utilize the turned intoted low-grade heat and reduce the compressor work and, thus improving the cycle efficiency and performance [24–27].

Several research had been provided to analyze the opportunity of usage of this turned intoted heat and obtain an development for the overall cycle performance

[18-20].

For further improvement in efficiency and overall performance of the cycle, an ejector refrigeration cycle is incorporated with the transcritical refrigeration cycle with N2O asa refrigerant and replacing throttle valve with turbine which generate energy and cooling might be a more effective method. To the pleasant of the author's information and with the aid of using surveying the above-stated literature review, the combination of the vapor compression cycle and the ejector refrigeration cycle to simultaneously generate power and cooling is not so far investigated. The present research is carried outin the continuation of our preceding work [18]. In this work, a new combined refrigeration cycle for power and cold (NCRCPC) is presented to improve the economic and thermodynamic characteristics in the traditional and combined N2O refrigeration cycles. Also, the thermodynamic performance of the NCRCPC will be investigated centered on energy and exergy inspection. A 1D constant area ejector, which works in the transition regime model, is adopted to calculate its performance. The ejector performance in the transition regime turned into considerably studied and proved to be an optimal operating regime [28–30]. The energetic and exergetic results of the three studied cycles are compared to find the best operating cycle.

The Description Of The Cycles

The N2O VCC schematic and p-h diagrams are presented in Fig. 1.

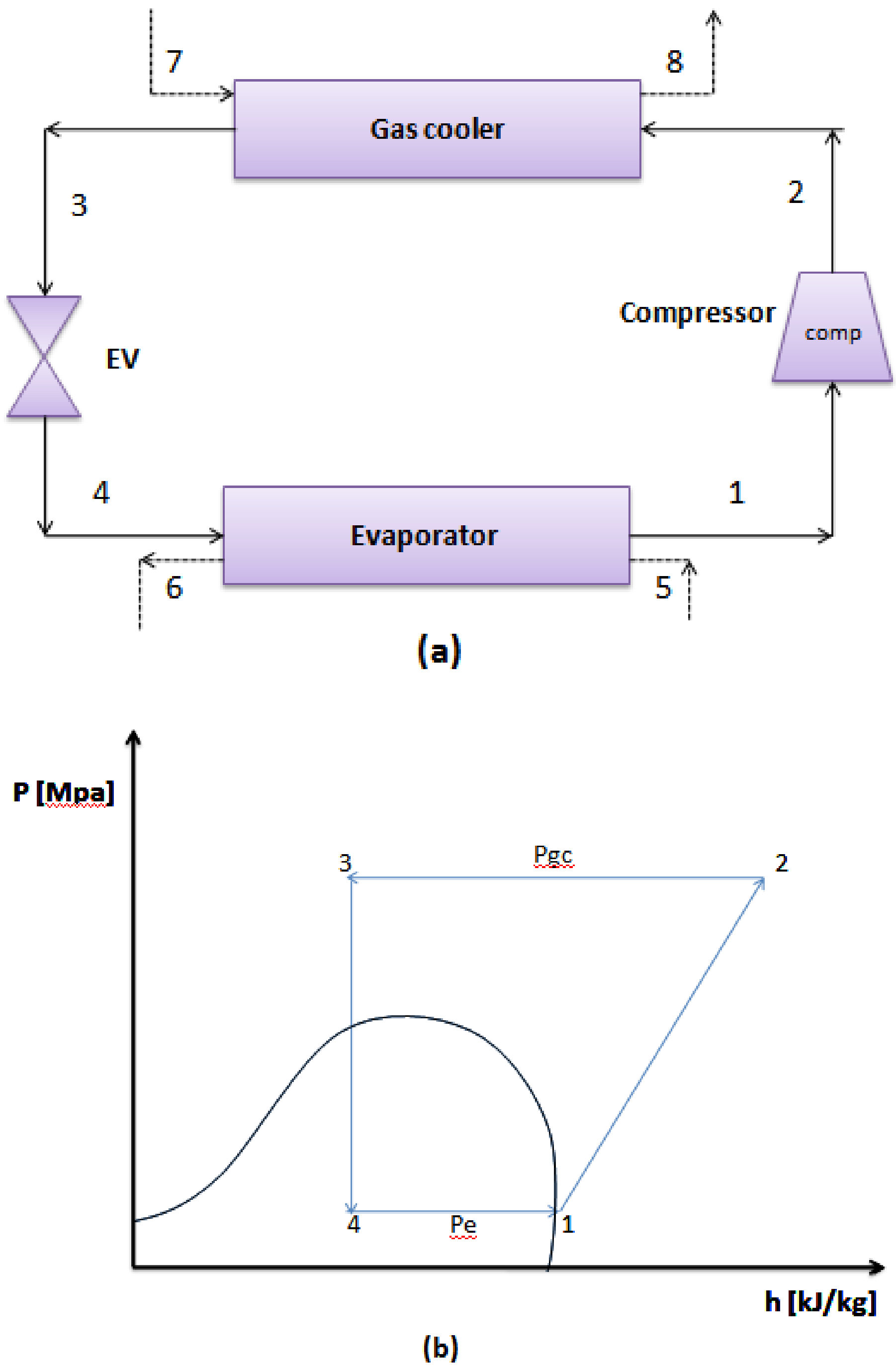
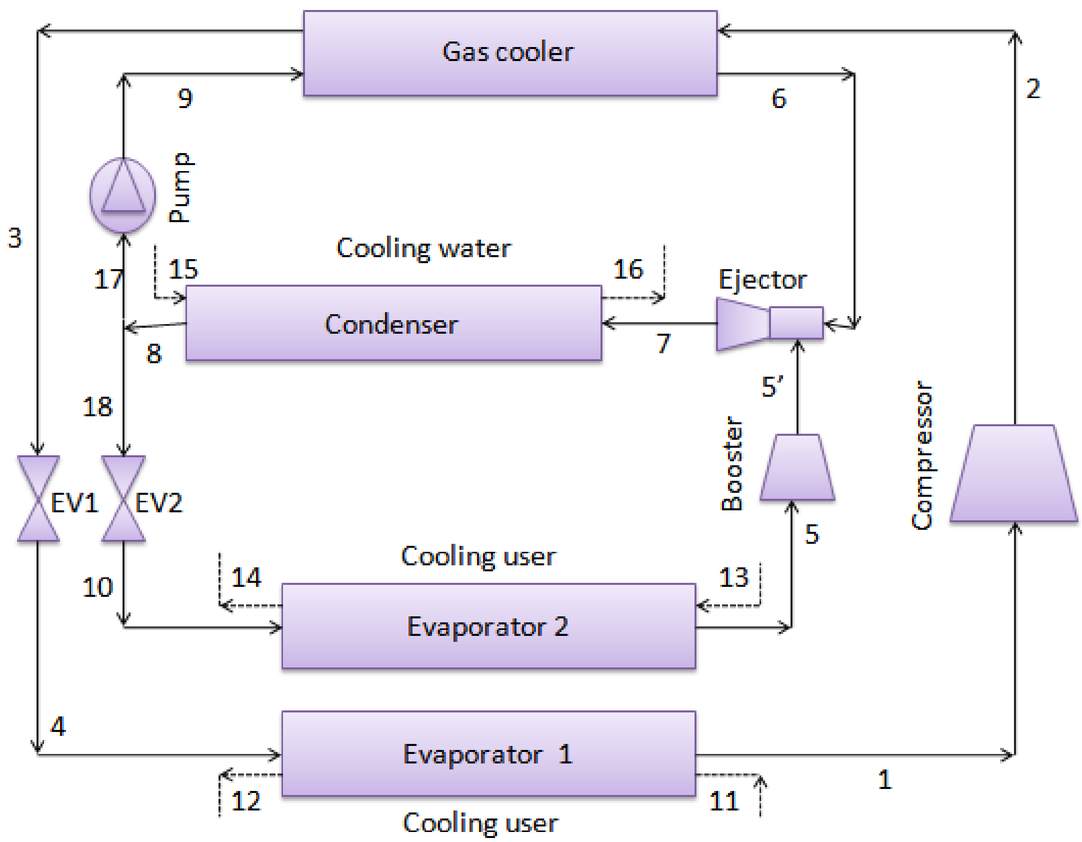


Fig. 1. (a) VCC system, (b) VCC P-h diagram.

The components of this system are the evaporator, compressor, throttling valve, and a gas cooler. The wasted low-grade energy from the gas-cooler is utilized by incorporating an ERC into the transcritical refrigeration system, which will result in an enhancement in the performance of the system, production of cooling effect and obtaining dual temperature refrigeration system. The schematic diagram of the N2O CRC and its p-h diagram are presented in Fig. 2.



(a)

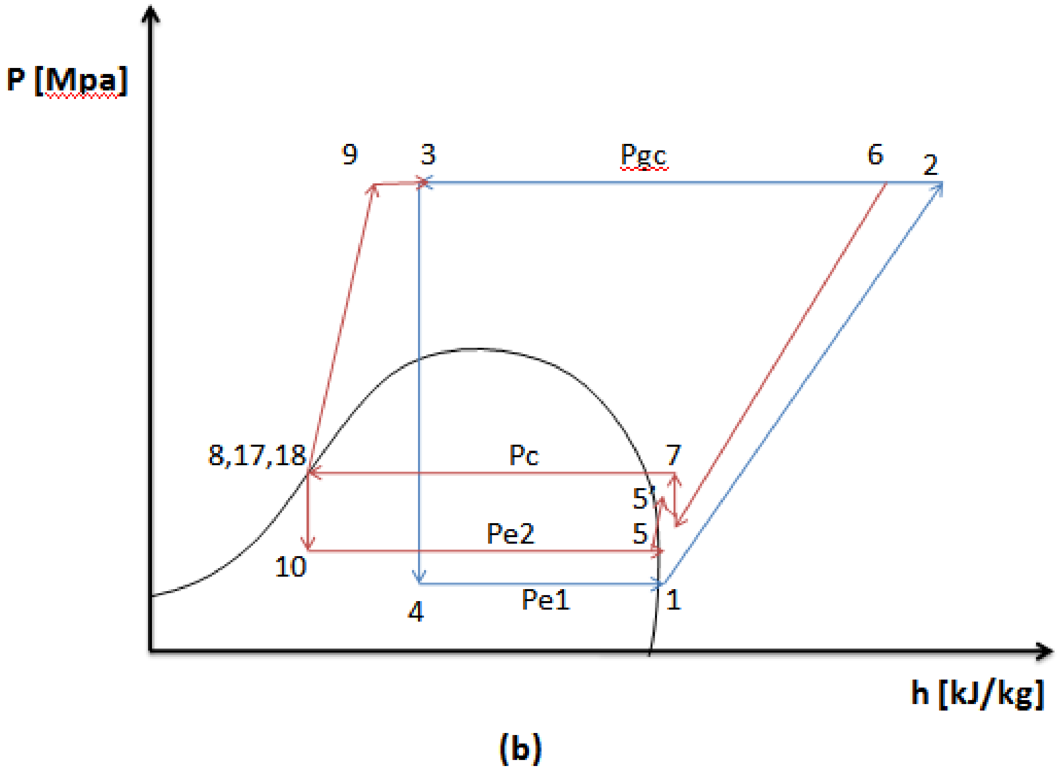


Fig. 2. (a) CRC system, (b) CRC P-h diagram.

The existence of the throttling device in the CRC results in high losses and thus reduces the cycle total thermal efficiency. Therefore, the replacement of the inefficient expansion valve by a turbine that will produce power will solve this problem. By introducing a turbine in the CRC, a novel combined refrigeration cycle for producing a refrigerating effect and generating power termed (NCRCPC) is achievable. The NCRCPC schematic and p-h diagrams are presented in Fig. 3.

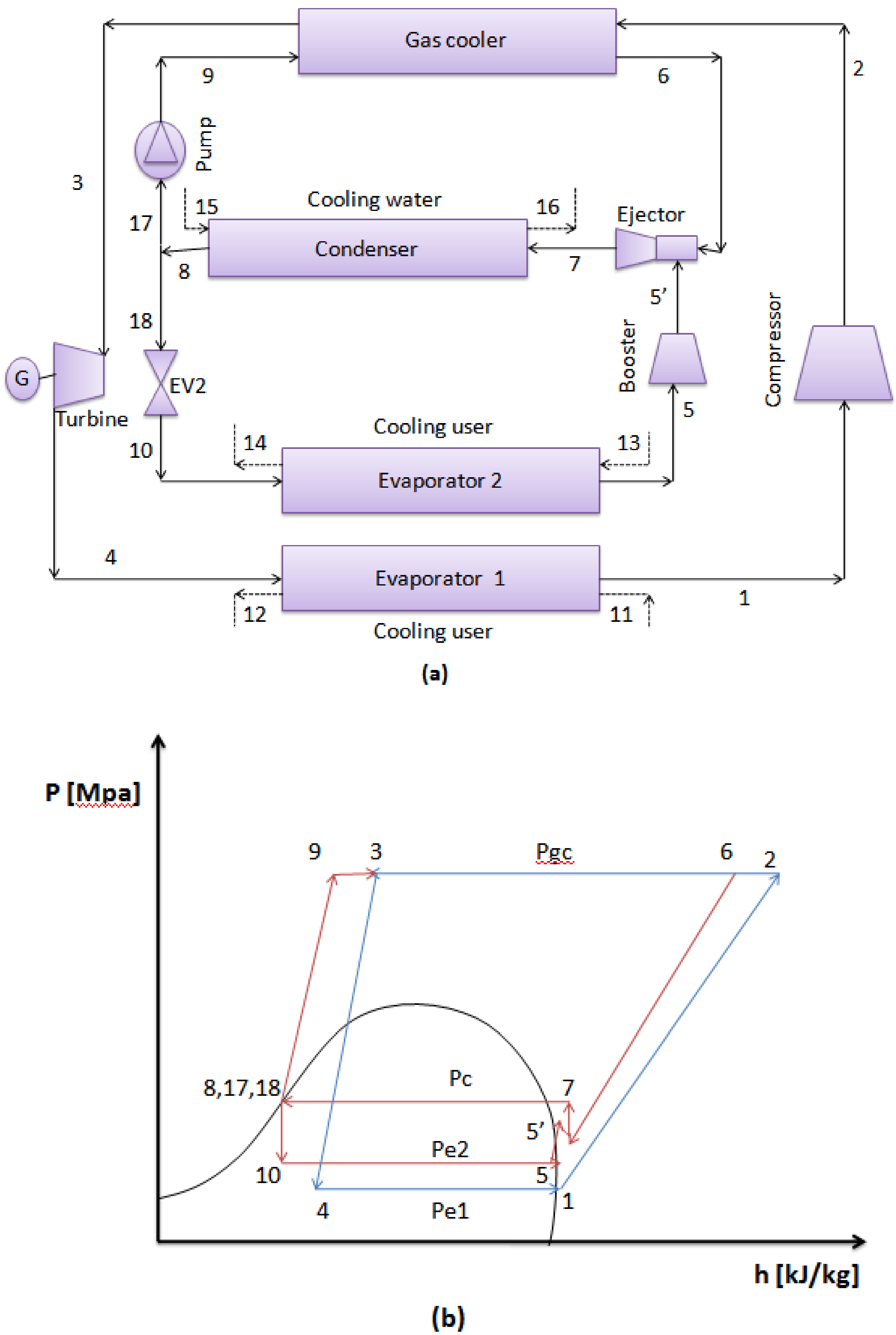


Fig. 3. (a) NCRCPC system, (b) NCRCPC P-h diagram.

The refrigeration process of NCRCPC can be described by starting at any point in the system and coming back to this point. The wasted heat in the gas cooler is used as a driving source of the ERC integrated into the system. The heat rejected by the gas cooler is transferred to the liquid refrigerant entering at state 9 and leaving at state 6 in superheated state conditions.

This superheated refrigerant enters the ejector nozzle and acts as a primary flow, whereit continues to gain speed while flowing through the converging-diverging nozzle to a higher speed and thus creating a suction effect, where the low pressure refrigerant is entrained from the evaporator at state 5′. A booster is installed to higher the refrigerant pressure after the evaporator from state 5 to state 5′. The entrained refrigerant is mixed with the high pressure flow in the constant area mixing chamber in the ejector and is directed to the ejector diffuser section, where its pressure is partially recovered to state

The mixed refrigerant enters the condenser, rejects heat to the environment and condenses to a liquid at state 9. Then, the liquid refrigerant in the condenser is pumped to the gas cooler and to an expansion valve at state 18, where the pressure is reduced to the evaporator pressure at state 10 and is delivered to evaporator 2. The low- pressure mixed refrigerant enters evaporator 2 at state 10 and evaporates to state 5, producing a refrigerating effect. The refrigerant vapor is delivered to the booster, where its pressure is increased to state 5′, and the cycle is complete.

Selection Of Refrigerant

The selection of the exceptional working fluid for a given refrigeration cycle was based on specific criteria including no longer restricted to safety, high COP, low Global Warming Potential (GWP), toxicity, flammability, low Ozone Depletion Potential (ODP), and environment-friendly properties.

A wide variety of natural refrigerants are to be had to be used in refrigeration and air conditioning applications. Using the natural refrigerants like carbon dioxide, nitrous oxide, air, ammonia, isobutene, propane, water etc. benefitted us as they have zero ODP and low GWP and low cost as well. In colder countries CO2 based cycles have already received fairly right compliance in refrigeration applications. N2O as a refrigerant is yet to be completely investigated. As N2O and CO2 has comparable properties like critical pressure and critical temperature and molecular weight. Although, it's far much less beneficial in phrases of GWP in comparison to CO2. Important properties of N2O and CO2 are compared in the table shown below.

Table 1

Properties of **CO2** and **N2O** shown.

Propoties	**CO2**	**N2O**
Critical pressure (MPa)	7.377	7.245
Critical temperature (°C)	31.1	36.4
Boiling point (°C)	-78.4	-88.47
Triple point temperature (°C)	-56.55	-90.82
Toxicity (ppm)	5000	1000
Molecular weight (kg/kmol)	44.01	44.013
GWP	1	240
ODP	0	0

Latent heat of vaporization (kJ/kg)	574	374.28

The development in growing refrigeration cycles running with N_2O as a refrigerant made it extra profitable, in particular with the integration of ejector and expanders into the refrigeration system, which made it feasible to make use of the wasted low-grade heat and decrease the compressor work and, consequently enhancing the cycle efficiency and overall performance.

Conclusion

1) For N_2O based refrigeration cycle maximum coefficient of performance is more than that of CO_2 based refrigeration cycle for the same evaporator temperature.
2) N_2O based refrigeration cycle has higher cycle performance than CO_2 based refrigeration cycle at lesser gas cooler exit pressure.
3) For N_2O based refrigeration cycle 2nd law efficiency is more at lesser gas coolerexit pressure comparison of CO_2 based refrigeration cycle.
4) N_2O based refrigeration cycle has lower pressure recovery ratio and higher entrainment ratio compared to CO_2 based refrigeration cycle for same gas coolerexit pressure.
5) Exergy output for N_2O based refrigeration cycle is premier as compare to CO_2 based refrigeration cycle at maximum COP condition.
6) N_2O based refrigeration cycle have less irreversibility in Ejector compared to CO_2 based refrigeration cycle.

CHAPTER2
LITERATURE SURVEY

LITERATURE REVIEW AND RESEARCH GAP

With the development of global technology, the demand for energy is increasing. The prosperity of a country depends on its production and development of energy.

Arora and Kaushik [1] Descriptive exergy analysis of the real vapor compression refrigeration cycle has been proposed. They compared the performance of R502, 404A and R507A refrigerants. They observed that R507A is better than R404A in replacing R502.

Hwang and Jeong [2] The effect of the refrigeration system parameters using R600a on the non-equilibrium subcooled two-phase flow of refrigerant was observed. They studied the influence of the operating conditions of the refrigeration system on the unbalanced gas content of R600a through experiments. They found that the system parameters changed the porosity, as well as the pressure and temperature.

Bolaji [3] In household refrigerators, an experimental study was carried out to replace R134a with R152a and R32, and the performance of R152a and R32 with R134a was compared. He pointed out that for R152a and R134a refrigerants, the extraction time and design temperature set by the International Organization for Standardization (ISO) for (small) refrigerators are higher than those of R32. The average coefficient of performance (COP) obtained by R134a is 4.7% lower than that of R152a, and the average coefficient of performance (COP) of R152a is 8.5% higher than that of R32.

For R152a, the system consumes less energy. He added that under all working conditions, the performance of R152a is better than that of R134a and R32 in domestic refrigerators. Therefore, in domestic refrigerators, R152a can be used as a substitute for R134a.

Alavi et al. [4] The power regeneration technology used for VCR equipment to upgrade existing equipment is introduced. The main compressor, evaporator, condenser and main expansion valve are allocated according to the characteristics of each equipment. They observed that for a given main screw compressor, the COP and cooling capacity were increased by 80% and 100%, respectively. They added that for the constant cold power plant, the main compressor throughput is reduced to 50%, while the COP is increased to 70%.

Humane et al. [5] The performance of low GWP refrigerants was compared for a micro vapor compression system (VCS) integrated with improved phase change materials. They compared the performance of R1234ze (E), R1234yf and R32 and R290. They concluded that compared with R134a, the system load reduction of R32, R1234ze (E), R290 and R1234yf is 1%, 30%, 52% and 11%, respectively. The COP changes of R32, R290, R1234yf, and R1234ze (E) were + 8%, 3%,-5%, and-12%, respectively. The revolutions per minute (RPM) of the compressor are R90, R1234ze(E), R1234yf, R32 and R134, which are 1700, 2900, 2400, 900 and 2100 respectively.

Khan and Zubair [6] Engaged in the design and classification of mechanically integrated subcooled vapor compression refrigeration systems. They developed a thermodynamic model of the integrated mechanical supercooling system to simulate the actual performance of the supercooling system. They predict that the performance of an integrated vapor compression refrigeration cycle is better than a simple cycle that uses mechanical subcooling.

Zubair and Khan [7] conducted thermodynamic analysis based on the first and second laws of the two-stage and mechanical subcooling refrigeration cycle. They analyzed the irreversible loss of each component of the system. They compared the theoretical performance results of the two-stage refrigeration system with the experimental values of the CFC22 system.

Fontalvo A [8] In this work, the ammonia-water mixture is used as the working fluid, combined with the Rankine power and the absorption refrigeration cycle and the exergy analysis of the combined refrigeration cycle. Find out the influence of the pressure ratio, the ammonia mass fraction in the absorber, and the turbine efficiency on the total exergy destruction of the cycle.

Lopez-Villada J [9] By using an ammonia-based working fluid mixture, different solar absorption power-cooling systems are simulated, which simultaneously generate mechanical energy and cooling energy through a single system.

Zheng D [10] A combined absorption/cooling power cycle is proposed, and the cycle is thermodynamically analyzed using log p-T, log p-h, and T-s diagrams. Find the exergy efficiency and thermal efficiency of the cycle.

Cao L [11] This article looks at the Kalina-based Combined Power and Cooling Cycle driven by a low-grade heat source and an absorption refrigeration cycle. Through the establishment of mathematical models, numerical simulation and parameter analysis, to investigate the influence of the best exergetic efficiency and the best inlet pressure of the expander.

Megdouli K [12] In the current work, a new CO2 refrigeration cycle is studied, in which the ejector expansion cycle is used as a tool to collect waste heat from the gas cooler. Basic refrigeration cycle, thereby reducing labor consumption, thereby improving the performance characteristics of the entire system. The operation and performance characteristics of the system are theoretically studied using energy and exergy methods.

Lontsi F [13] In this study using environmentally friendly refrigerants, a combined compression / expulsion refrigeration cycle designed to simultaneously generate refrigeration for refrigeration and freezing, eight candidate fluids were selected.

Singh S [14] Using an expander for work recovery instead of an expansion valve is a promising option for achieving commercial success in transcritical

CO2 refrigeration and air conditioning systems by increasing the COP.and overall performance of the system.

Conclusion Of Literature Review

It has been concluded that NCRCPC has been worked already by using CO_2 as a refrigerant. In colder countries CO_2 based cycles have already received fairly right compliance in refrigeration applications. N_2O as a refrigerant is yet to be completely investigated. As N2O and CO_2 has comparable properties like critical pressure and critical temperature and molecular weight. Although, it's far much less beneficial in phrases of GWP in comparison to CO_2.

Research Gap

1) The work is yet to done for Vapor Compression Cycle (VCC) using N_2O refrigerant.
2) The work is yet to done for Combined Refrigeration Cycle (CRC) using N_2O refrigerant.
3) The work is yet to done for New Combined Refrigeration Cycle For Power And Cold (NCRCPC) using N_2O refrigerant.

Objective

1. To carry out energy and exergy analysis for Vapor Compression Cycle (VCC) with N_2O refrigerant and compare it with CO_2.
2. To carry out energy and exergy analysis for Combined Refrigeration Cycle (CRC) with N_2O refrigerant and compare it with CO_2.
3. To carry out energy and exergy analysis for New Combined Refrigeration Cycle For Power And Cold (NCRCPC) with N_2O refrigerant and compare it with CO_2.

CHAPTER3
METHODOLOGY

System Mathematical Modeling

A mathematical model for the system is presented to analyze the proposed cycles. Theejector performance is evaluated by a 1D constant area ejector model.

Assumptions

The following assumptions will be considered to simplify the mathematical modelingand thermodynamic analysis of the proposed NCRCPC [13-14].

1. Steady-state conditions.
2. Frictionless flow in the components except for the ejector.
3. Adiabatic process in the ejector.
4. The mixing chamber, nozzle and diffuser losses are accounted for by the efficiency of these components.
5. Pressure ratio of booster is taken as 1.1
6. The isenthalpic process through the expansion valve.
7. The change in kinetic energy in the ejector is neglected.
8. Saturated state of vapor and liquid are assumed at the exit of evaporators andcondenser.
9. Constant isentropic efficiency for the booster, turbine, and pump at 85%, 95%,and 90%, respectively [29].
10. The isentropic compressor efficiency is estimated as

    $$\eta_{com} = 0.9343 - 0.04478(P2/P1)\ (n!/r!\ (n - r)!) \quad [18].$$
11. A constant cooling capacity of 100 kW for the system.

12. Temperature difference for the gas cooler is assumed to be five degrees ($\Delta T = T2 - T6 = 5$) [18].
13. The ambient reference temperature and pressure are 293.15 K and 0.101 MPa,respectively [18].

1 The system energy analysis

In the new proposed NCRCPC, the energy conservation principle is applied to every component in the steady-state regime, and the modelequations are derived with the application of conservation principles. Here, the non-transient conservation of mass and energy equations are:

$$\sum min - \sum mout = 0 \tag{1}$$

$$\sum (mh)in - \sum (mh)out + \sum Qin + \sum Qout + W = 0 \tag{2}$$

In the ERC, the mass flow ratio of the refrigerant entrained from the evaporator, (ms) to the refrigerant flowing from the gas cooler, (mp) is called the entrainment ratio. It is an essential characteristic that profoundly affects the performance of the ERC, and it has the following definition:

$$u = ms/mp \tag{3}$$

Table 2 presents the model equation for every cycle component.

Table2

Energy balance and thermodynamic equations for VCC, CRC and NCRCPC cycles.

a) VCC

Component	**Equation**
Evaporator	$Qe = m(h1 - h4)$
Compressor	$Wcom = m(h2 - h1)$
Gas cooler	$Qgc = m(h2 - h3)$
Expansion valve	$h3 = h4$
Coefficient of performance (COP)	$COP = Qe/Wcom$

b) CRC

Component	Equation
Evaporator 1	$Qe1 = m1\ (h1 - h4)$
Evaporator 2	$Qe2 = m5\ (h5 - h10)$
Compressor	$Wcom = m1\ (h2 - h1)$
Expansion valve 1	$h3 = h4$
Gas cooler $m1$	$(h2 - h3) = m6\ (h6 - h9)$
Condenser	$Qcd = m7\ (h7 - h8)$
Pump	$Wp = m9\ (h9 - h8)$
Ejector	$h[7]=(h[6]+(u*h[5']))/(1+u)$
Booster	$Wb = m5\ (h5' - h5)$
Expansion valve 2	$h8 = h10$
Coefficient of performance	$COP= (Qe1 + Qe2)/(Wnet)$, $Wnet = Wcom + Wp + Wb$

c) NCRCPC

Component	Equation
Evaporator 1	$Qe1 = m1\ (h1 - h4)$
Evaporator 2	$Qe2 = m5\ (h5 - h10)$
Compressor	$Wcom = m1\ (h2 - h1)$
Expansion valve 1	$h3 = h4$
Gas cooler $m1$	$(h2 - h3) = m10\ (h11 - h9)$
Condenser	$Qcd = m7\ (h7 - h8)$
Pump	$Wp = m9\ (h9 - h8)$
Ejector	$h[7]=(h[6]+(u*h[5']))/(1+u)$
Expansion valve 2	$h8 = h10$
Turbine	$Wt = m3\ (h3 - h4)$
Booster	$Wb = m5\ (h5' - h5)$
Coefficient of performance	$COP = (Qe1 + Qe2)/(Wnet)$, $Wnet = Wcom + Wp + Wb - Wt$

1 The system exergetic analysis

Several definitions exist in the literature for exergy. One of these definitions is the maximum theoretical work delivered by the system with its conditions to the environment equilibrium [33]. Neglecting the nuclear, electrical, magnetic and other less predominant effects, the rate of system total exergy (Etotal) constitutes of the kinetic energy, physical, chemical and potential exergy rates (EKN, EPH, ECH, EPT) [33]:

$$E_{total} = E_{ph} + E_{kn} + E_{pt} + E_{ch} \quad (4)$$

The physical exergy rate of any closed system is related to the entropy and enthalpy differences of the refrigerant for a specified reference of state of a given temperature and pressure (T0, P0), and is estimated as:

$$Eph = m\ [(h\ h0)\ T0\ (s\ s0)] \quad (5)$$

The fuel and product (Ef , Ep) (exergies concepts are introduced for system components to conduct the system exergy analysis. These two concepts represent either the desired output exergy from the system (product) or the needed exergy to generate the product (fuel). Furthermore, an exergy term that relates to the performance and inefficiency of the system component is described by the component exergy destruction rate ED . The rate of system exergy losses (Ef , Ep) represents the system component losses.

Therefore, for any k component in the system, the exergy balance equation is [33]:

$$EFk = EPk + EDk + Elk \quad (6)$$

The exergy analysis is based on exergy balances written for the overall system

$$EF, tot = EP, tot + \sum K\ ED,k + EL,tot \quad (7)$$

The exergy efficiency of element k, (εk) can be estimated as the ratio of the exergy produced, (EP,k) to the exergy needed, (EF,k) [33].

$$\varepsilon k = EP,k / EF,k \tag{8}$$

The exergy efficiency of the overall refrigeration system is:

$$\varepsilon tot = EP,tot / EF,tot \tag{9}$$

The inefficiency of the system is defined by the ratio of the element rate of exergy destruction, (ED,k) to the system rate of exergy destruction,ED,tot as follows:

$$yD,k = ED,k / ED,tot \tag{10}$$

Table 3 presents the equations of exergy for the system components of the cycles under consideration.

Table 3

VCC, CRC an NCRCPC exergy balance equations.

a) VCC

Component	**Generated product rate ($E_{P,k}$)**	**Fuel supplied rate ($E_{F,k}$)**	**Exergy destruction rate ($E_{D,k}$)**
Evaporator	$E_{P,e} = E_6 - E_5$	$E_{F,e} = E_4 - E_1$	$E_{D,e} = E_{F,e} - E_{P,e}$
Compressor	$E_{P,com} = E_2 - E_1$	$E_{F,com} = W_{com}$	$E_{D,com} = E_{F,com} - E_{P,com}$
Gas cooler	$E_{P,gc} = E_8 - E_7$	$E_{F,gc} = E_2 - E_3$	$E_{D,gc} = E_{F,gc} - E_{P,gc}$
Expansion valve	$E_{P,ev} = E_4$	$E_{F,ev} = E_3$	$E_{D,ev} = E_{F,ev} - E_{P,ev}$
Total system	$E_{P,tot} = E_{P,e}$	$E_{F,total} = E_{P,total} + \sum_k E_{D,k} + E_{P,gc}$	$E_{D,tot} = \sum^4_{k=1} E_{D,k}$

b) CRC

Component	**Generated product rate (EP,k)**	**Fuel supplied rate (EF,k)**	**Exergy destruction rate (ED,k)**
Evaporator 1	EP,e1 = E12 - E11	EF,e1 = E4 - E1	ED,e1 = EF,e1 - EP,e1
Evaporator 2	EP,e2 = E14 - E13	EF,e2 = E10 - E5	ED,e2 = EF,e2 - EP,e2

Compressor	EP,com = E2 - E1	EF,com = Wcom	ED,com = EF,com - EP,com
Expansion valve 1	EP,ev1 = E4	EF,ev1 = E3	ED,ev1 = EF,ev1 - EP,ev1
Gas cooler	EP,gc = E6 - E9	EF,gc = E2 - E3	ED,gc = EF,gc - EP,gc
Condenser	EP,cd = E16 - E15	EF,cd = E7 - E8	ED,cd = EF,cd - EP,cd
Pump	EP,p = E9 -	EF,p = Wp	ED,p = EF,p - Ep,p
	(m9/m8)E8		
Ejector	EP,ej = E7	EF,ej = E5 + E6	ED,ej = EF,ej - EP,ej
Expansion valve 2	EP,ev2 = E10	EF,ev2 = (m10/m8)E8	ED,ev2 = EF,ev2 - EP,ev2
Booster	EP,b = E5 - E5	EF,b = Wb	ED,b = EF,b - EP,b
Total system	EP,tot = EP,e1 + EP,e2	EF,total = EP,total + ∑k ED,k + EP,cond	ED,tot = ∑k=1 ED, k

c) NCRCPC

Component	**Generated product rate (EP,k)**	**Fuel supplied rate (EF,k)**	**Exergy destruction rate (ED,k)**
Evaporator 1	EP,e1 = E12 - E11	EF,e1 = E4 - E1	ED,e1 = EF,e1 - EP,e1
Evaporator 2	EP,e2 = E14 - E13	EF,e2 = E10 - E5	ED,e2 = EF,e2 - EP,e2
Compressor	EP,com = E2 - E1	EF,com = Wcom	ED,com = EF,com - EP,com
Gas cooler	EP,gc = E6 - E9	EF,gc = E2 - E3	ED,gc = EF,gc - EP,gc
Condenser	EP,cd = E16 - E15	EF,cd = E7 - E8	ED,cd = EF,cd - EP,cd
Pump	EP,p = E9(m9/m8) - E8	EF,p = Wp	ED,p = EF,p - EP,p
Ejector	EP,ej = E7	EF,ej = E5 + E6	ED,ej = EF,ej - EP,ej
Expansion valve	EP,ev2 = E10	EF,ev =(m10/m8)E8	ED,ev2 = EF,ev2 - EP,ev2
Turbine	EP,t = Wt	EF,t = E3 - E4	ED,t = EF,t - EP,t
Booster	EP,b = E5 - E5	EF,b = Wb	ED,b = EF,b - EP,b

Total system	EP,tot = EP,e1 + EP,e2	EF,total = EP,total + ∑ k ED,k + EP,cond	ED,tot = ∑k=1 ED, k

Table 4

Operating conditions and reference parameters for the analysis of cycles performanc

Parameter	**Value**
Evaporator 1,Te1 [°C]	−30 to −10 °C
Evaporator 2, Te2 [°C]	0
Gas cooler temperature, Tgc [°C]	35 to 50
Gas cooler pressure, Pgc [MPa]	8 to 14
Pump isentropic efficiency, ηis,p (%)	90
Turbine isentropic efficiency, ηis,t (%)	95
Booster isentropic efficiency, ηb (%)	85
Diffuser efficiency, ηd (%)	85
Isentropic efficiency of primary flow, ηp %	95
Isentropic efficiency of secondary flow, ηs %	85
The temperature of reference state, T0 [°C]	20
Pressure of reference state, P0 [MPa]	0.101
Cooling capacity [kW]	100

Calculated properties at the state points for every cycle is presented in Table 5

Table 5

VCC, CRC, and NCRCPC thermodynamic properties.

a) VCC

Te =- 25°C, Tgc = 40°C, Pgc = 9 MPa

S. No.	**T (K)**	**P (MPa)**	**h (kJ/kg)**	**s (kJ/kg/K)**	***m* (kg/s)**	**E(kW)**
1	248.15	1.543	400	1.676	0.7513	112.9
2	411	9	523.4	1.676	0.7513	205.7
3	313.15	9	266.9	1.034	0.7513	154.2

4	248.15	1.543	266.9	1.14	0.7513	131
5	258.15	0.101	434	2.298	19	36.61
6	253.15	0.101	429.8	2.282	19	48.28
7	293.15	0.101	464.1	2.408	2	0
8	311.15	0.101	480.1	2.461	2	0.96

b) CRC

Te =- 25°C, Te2 = 0°C, Tgc = 40°C, Pgc = 8.54MPa

S. No.	**T (K)**	**P (MPa)**	**h (kJ/kg)**	**s(kJ/kg/K)**	***m* (kg/s)**	**E(kW)**
1	248.15	1.543	*400*	*1.676*	*0.511*	76.91
2	405.2	8.54	518.9	1.676	0.511	137.7
3	313.15	8.54	273.1	1.056	0.511	104.8
4	248.15	1.543	273.1	1.164	0.511	88.63
5	273.15	3.122	398.2	1.567	0.232	41.93
5’	279.7	3.435	401.8	1.567	0.232	42.75
6	400.2	8.54	512.4	1.763	0.512	121.5
7	359.8	6.307	476.8	1.716	0.743	160.1
8	303.15	6.307	246.7	0.9817	0.743	149.1
9	308.2	8.54	250	0.9819	0.512	104.3
10	273.15	3.122	246.7	1.012	0.232	44.49
11	258.15	0.101	434	2.299	13.64	26.28
12	253.15	0.101	429.8	2.283	13.64	34.66
13	283.15	0.101	455.4	2.378	6.24	0.9214
14	278.15	0.101	451.1	2.363	6.24	2.111
15	293.15	0.101	464.2	2.408	5	0
16	301.15	0.101	471.2	2.432	5	0.4915
17	303.15	6.307	246.7	0.9817	0.512	102.7
18	303.15	6.307	246.7	0.9817	0.232	46.54

c) NCRCPC

Te1 = -25°C, Te2 = 0°C, Tgc = 40°C, Pgc = 8 MPa

S. No.	T (K)	P (MPa)	h (kJ/kg)	s(kJ/kg/K)	***m*** **(kg/s)**	E(kW)
1	248.15	1.543	400	1.676	0.485	72.99
2	398.1	8	513.3	1.676	0.485	127.9
3	313.15	8	291.1	1.117	0.485	99.63
4	248.15	1.543	261.2	1.117	0.485	85.15
5	273.15	3.122	398.2	1.567	0.2158	39.01
5’	279.7	3.434	401.8	1.567	0.2158	39.77
6	393.1	8	506.8	1.76	0.479	111.5
7	358.1	6.307	474.5	1.71	0.7008	150.7
8	303.15	6.307	246.7	0.9817	0.7008	140.6
9	307.1	8	249.2	0.9818	0.479	97.25
10	273.15	3.122	246.7	1.012	0.2158	41.38
11	258.15	0.101	434	2.299	14.44	27.83
12	253.15	0.101	429.8	2.283	14.44	36.71
13	283.15	0.101	455.4	2.378	5.44	0.8038
14	278.15	0.101	451.1	2.363	5.44	1.842
15	293.15	0.101	464.2	2.408	4	0
16	301.15	0.101	471.2	2.432	4	0.3932
17	303.15	6.307	246.7	0.9817	0.479	96.09
18	303.15	6.307	246.7	0.9817	0.2158	43.29

Table 6

Exergetic parameters of VCC, CRC and NCRCPC components.

a) VCC

Te = -25°C, Tgc = 40°C, Pgc = 9[MPa]

Component	$E_{P,k}$(kW)	$E_{F,k}$(kW)	$E_{D,k}$(kW)	$y_{D,k}$ (%)	ε_k (%)
Evaporator	11.67	18.1	6.43	8.79	64.47
Compressor	92.8	93	0.2	0.27	99.78

Gas cooler	0.96	51.5	50.54	68.91	18.6
Expansion valve	131	154.2	16.17	22.047	84.95
Total system	11.67	85.77	73.34	-	13.606

b) CRC

Te1 = -25°C, Te2 = 0°C, Tgc = 40°C, Tcd = 30°C, Pgc = 8.54[MPa]

Component	**$E_{P,k}$(kW)**	**$E_{F,k}$(kW)**	**$E_{D,k}$(kW)**	**$y_{D,k}$ (%)**	**ε_k (%)**
Evaporator 1	8.38	11.72	3.34	5.87	71.501
Evaporator 2	1.189	2.56	1.371	2.41	2.12
Compressor	60.76	65	4.24	7.45	0.934
Expansion valve 1	88.63	104.8	16.17	28.42	84.57
Gas cooler	17.2	32.9	15.7	27.59	52.27
Condenser	0.491	11	10.509	18.47	4.46
Pump	1.555	1.662	0.107	0.188	9.35
Ejector	160.1	163.43	3.33	5.85	9.79
Expansion valve 2	44.49	46.55	2.06	3.62	95.57
Booster	0.82	0.88	0.06	0.105	93.18
Total system	9.569	66.947	56.887	-	14.29

c) NCRCPC

Te1 = -25°C, Te2 = 0°C, Tgc = 40°C, Tcd = 30°C, Pgc = 8[MPa]

Component	**$E_{P,k}$(kW)**	**$E_{F,k}$(kW)**	**$E_{D,k}$(kW)**	**$y_{D,k}$ (%)**	**ε_k (%)**
Evaporator 1	8.88	12.16	3.28	9.65	73.026
Evaporator 2	1.038	2.37	1.332	3.92	43.79
Compressor	54.91	58	3.09	9.09	94.67
Gas cooler	14.25	28.27	14.02	41.27	50.40
Condenser	0.3932	10.1	9.706	28.57	3.89
Pump	1.149	1.179	0.03	0.088	97.455
Ejector	150.7	151.27	0.57	1.67	99.62

Expansion valve	41.38	43.295	1.915	5.63	95.57
Turbine	14.47	14.48	0.01	0.029	99.9
Booster	0.76	0.78	0.02	0.058	97.43
Total system	9.918	44.28	33.97	-	22.39

CHAPTER4
RESULTS

The EES (Engineering Equation Solver) programs is used to decide and calculate the refrigerant thermodynamic properties. The effects are used to evaluate the proposed cycles and calculate their overall performance for in addition investigation. The boundary and working situations used to simulate the cycles are offered in Table 3.

The computer program is used to calculate the cycles thermodynamic characteristics at all state points. Based on those effects, power evaluation can be carried out. The cycles under consideration in this work are the VCC, CRC, and the NCRCPC with N2O as the refrigerant.

❖ Energy results

The energy performance of the investigated cycles in terms of COP, net work input, entrainment ratio, and gas cooler optimum pressure are presented in Table 7 for the cycles under consideration in this work. The cooling capacity was fixedfor the three cycles at a constant value of 100 kW for comparison purposes. It can be seen from the results that the traditional VCC had the minimum COP since the net power consumed in this cycle is comparatively high. The COP of the proposed cycle is 1.49 times more than that of the CRC and 2.18 times more the value of the traditional VCC. The high value of COP in the new cycle is attributed to the integration of the turbine and the ERC into the VCC, where the turbine produced power to cover part of the compressor and pump required net power. Therefore, the new proposed NCRCPC has the advantage from an energetic point of view and cycle performance.

Table 7

VCC, CRC and NCRCPC energy performance characteristics.

Te1 =-25°C, Te2 = 0°C, Tgc = 40°C, Tcd = 30°C

Parameter	Basic VCC	CRC	NCRCPC
The net power rate (kW)	92.785	63.23	42.41
The produced cooling rate (kW)	100	100	100
U	–	0.454	0.445
COP	1.078	1.581	2.358
Popt,gc[MPa]	9	8.54	8

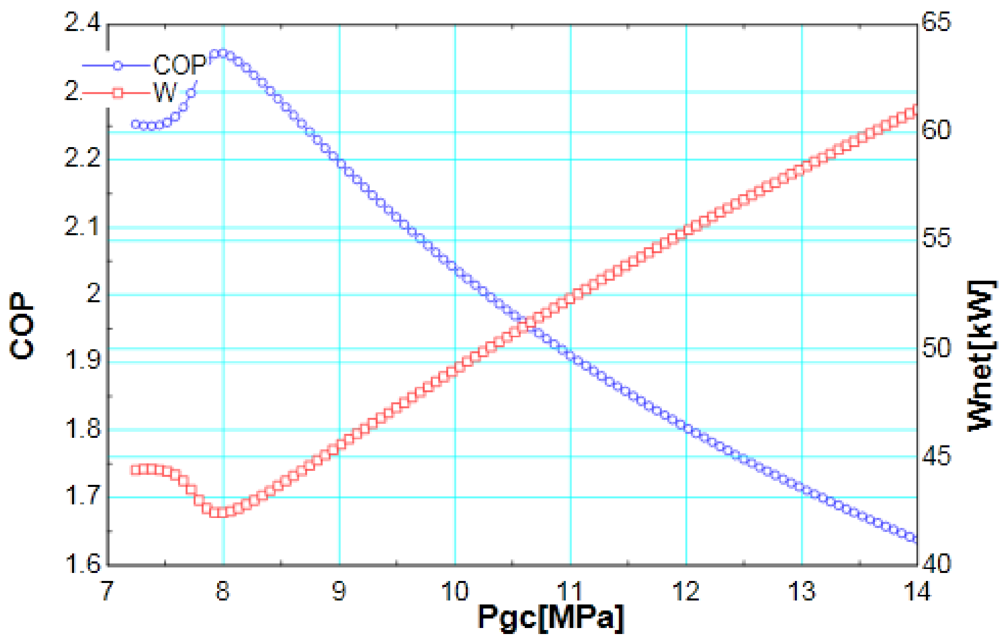


Fig. 4. The effect of pressure variation in the gas cooler on the net work input and COP of

the NCRCPC.

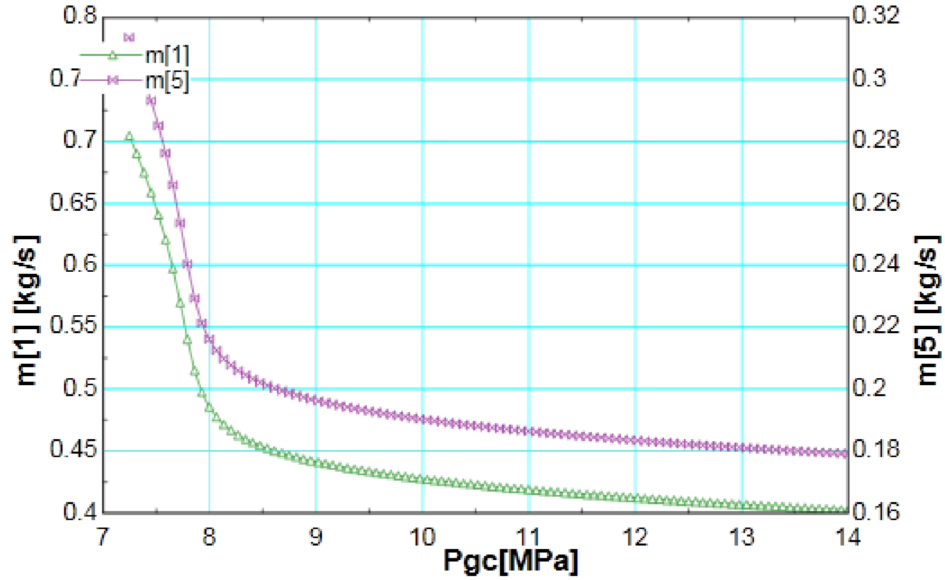


Fig. 5. The effect of pressure variation in the gas cooler on the motive fluid (m1) andSecondary fluid (m5) of the NCRCPC.

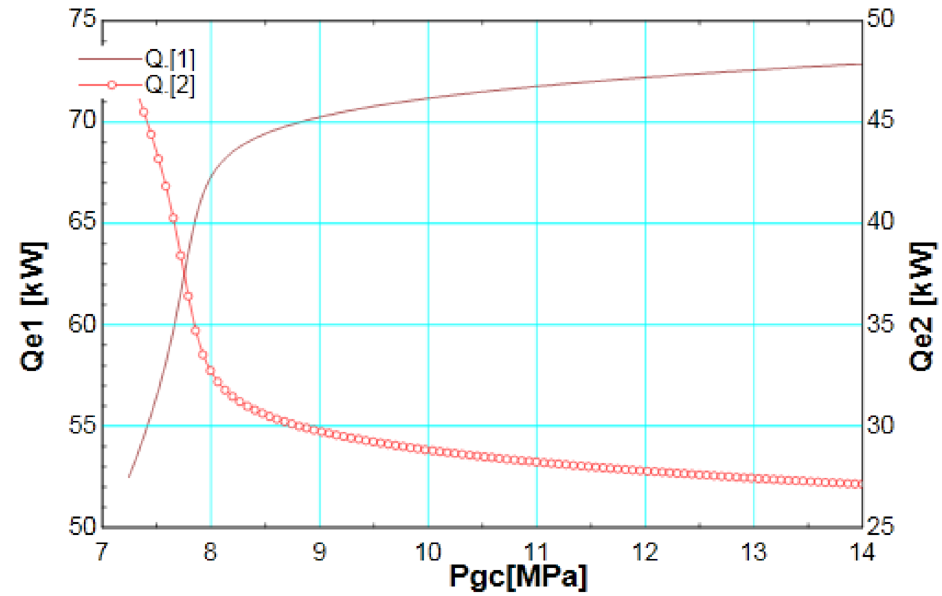


Fig. 6. The effect of pressure variation in the gas cooler on the Qe1 and Qe2 of the NCRCPC.

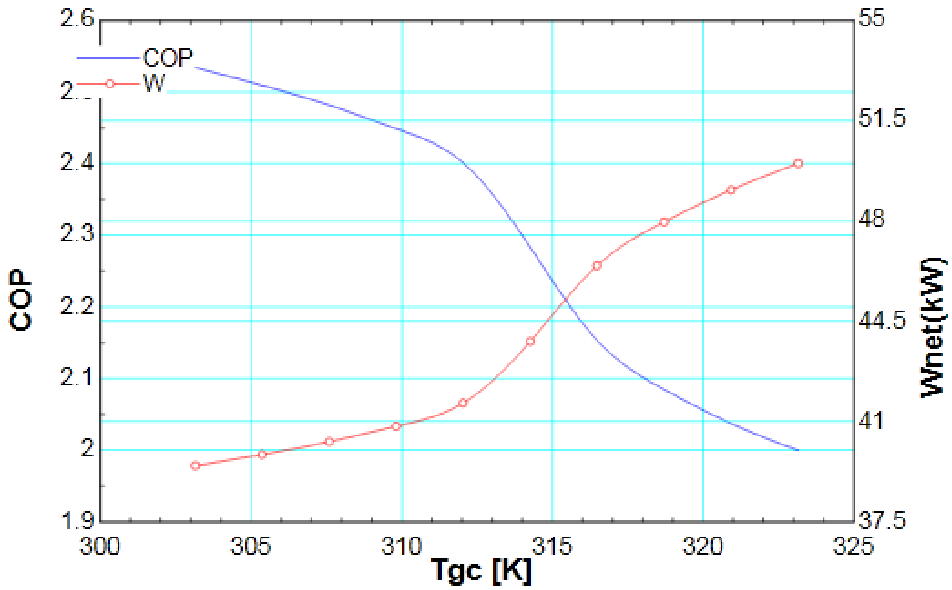


Fig. 7. The effect of Temperature variation in the gas cooler on the net work input and COP of the NCRCPC.

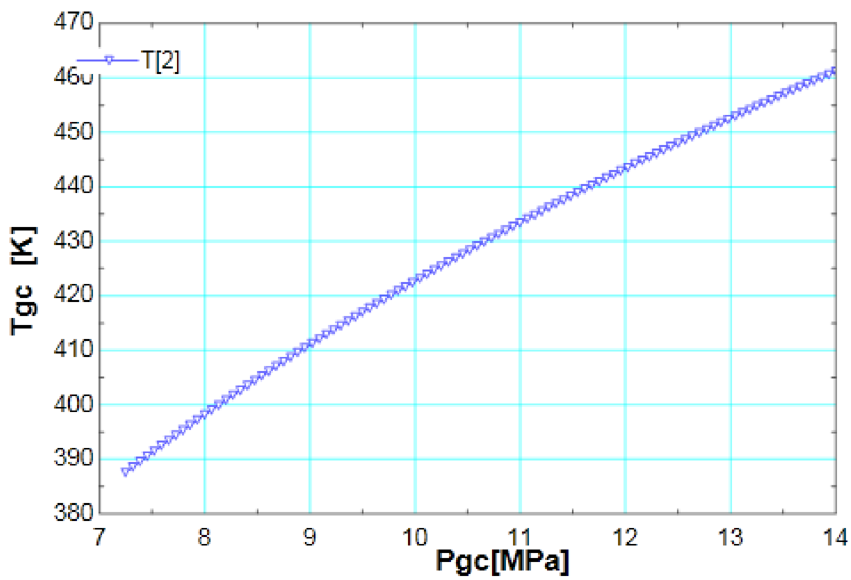


Fig. 8. The effect of pressure and Temperature variation in the gas cooler of the NCRCPC.

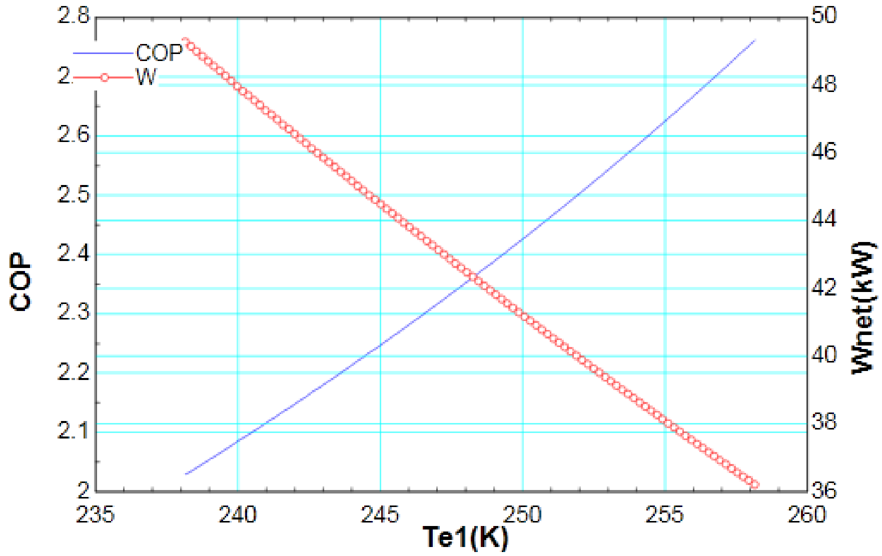


Fig. 9. The effect of Temperature variation in the evaporator1 on the net work input and COP of the NCRCPC.

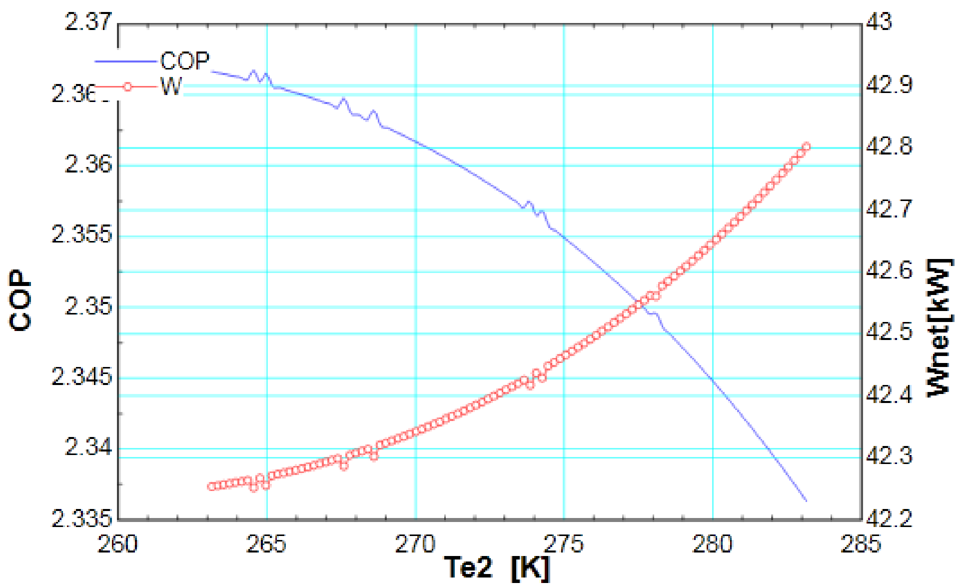


Fig. 10. The effect of Temperature variation in the evaporator2 on the net work input and COP of the NCRCPC.

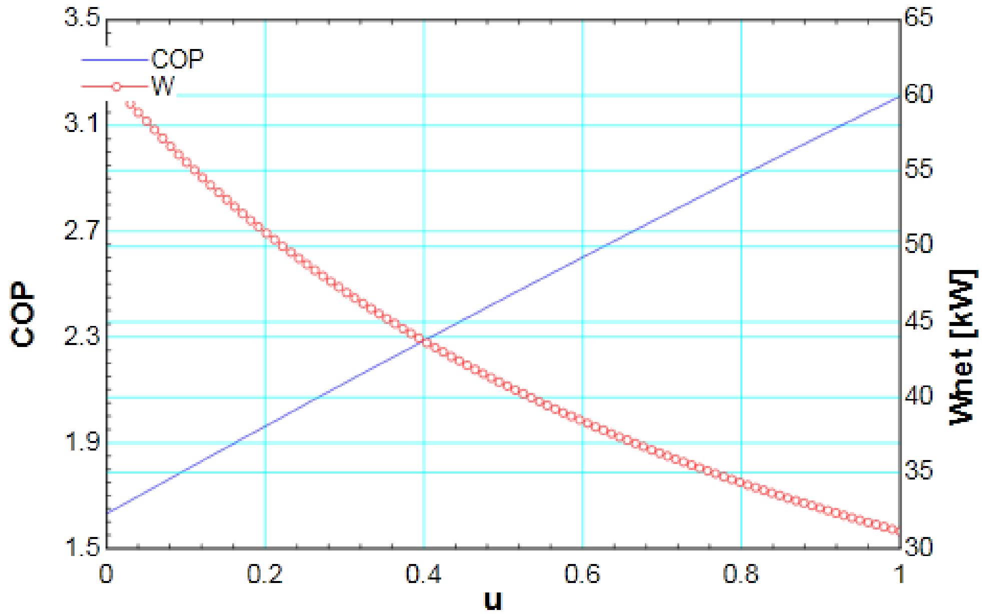


Fig. 11. The effect of entrainment ratio variation on the net work input and COP of the NCRCPC.

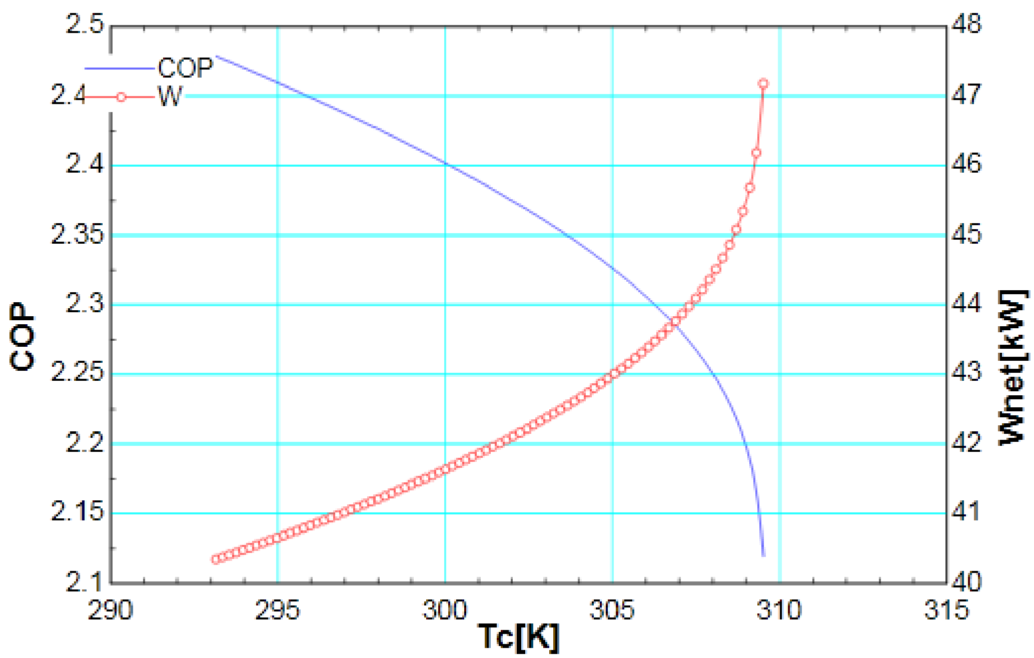


Fig. 12. The effect of Temperature variation in the condenser on the net work input and COP of the NCRCPC.

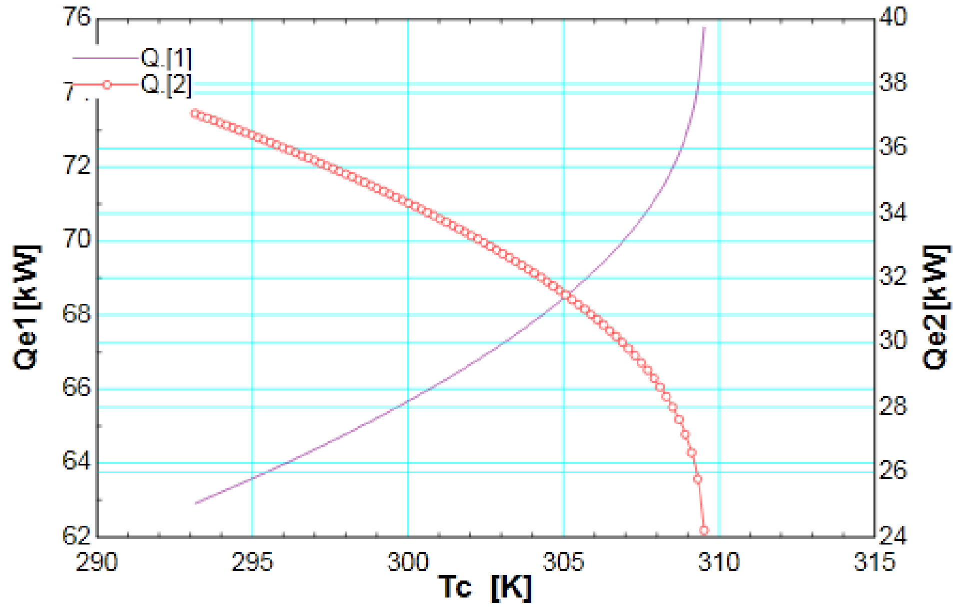


Fig. 13. The effect of Temperature variation in the condenser on the Qe1 and Qe2 of the NCRCPC.

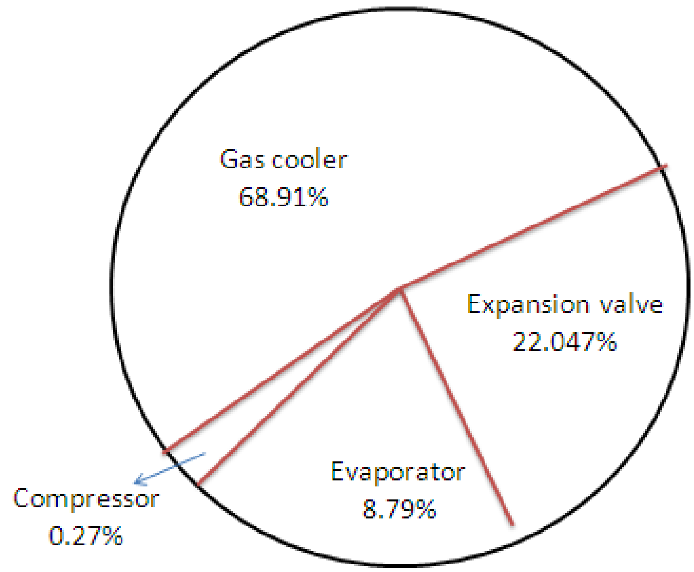
❖ Exergy Results

The Automotive sector largely depends on additive manufacturing technology. This The exergetic analysis results for the cycles under consideration are presented in Table 5. The results data were obtained for a fixed cooling capacity of 100 kW and with the corresponding gas cooler optimum pressure value for each cycle. The presented data indicated that the NCRCPC had higher exergy efficiency values than the VCC and CRC. The second law efficiency value of the NCRCPC exceeded those of the VCC and CRC by 64.55% and 56.68%, respectively.

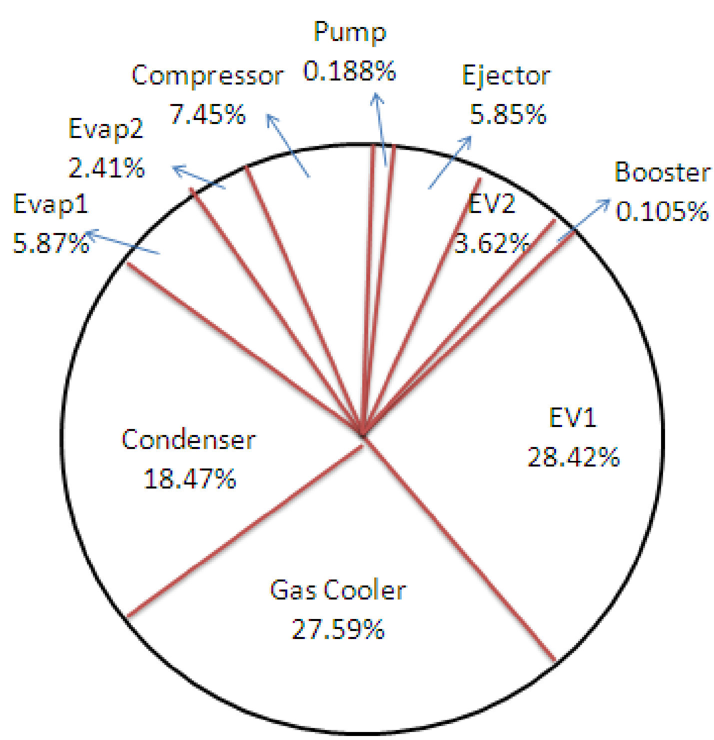
Moreover, it can be seen that the total exergy destruction rate for the new cycle it was decreased and it was only 53.68% of the VCC value and 22.43% of the CRC value. The exergy destruction of the components of the proposed refrigeration cycles is presented in Fig. 4. It can be seen that the expansion valve in the VCC had higher values of the exergy destruction rate. Besides, the percentages of the total exergy destruction for the gas cooler, expansion valve, compressor, and evaporator are 68.91%, 22.047%, 0.27%, and 8.79%, respectively in Fig. 4a. These results indicated that efforts must be made for the reduction of irreversibilities in the VCC components with high rate of exergy destruction to enhance its performance. The results presented in Fig. 4b showed that the expansion valve component had 28.42% of the total exergy destruction, while the compressor and condenser had 7.45% and 18.47%, respectively. The exergy destruction rate in the CRC gas cooler component was reduced by almost 59.96%. This reduction affirmed the fact that the use of the expansion valve in the CRC must be carefully considered to enhance CRC performance.

To accomplish the objective of reducing the exergy destruction rate in the expansion valve and improve the cycle performance, a turbine is integrated into the new proposed cycle as a replacement to the expansion valve. It is clear from Table 8 that using the turbine will enhance the cycle exergy efficiency by 56.68%. Therefore, the replacement of the expansion valve by the turbine in the new proposed NCRCPC will decrease the rate of exergy destruction and increase the exergy efficiency. The proposed NCRCPC has a maximum exergy performance.

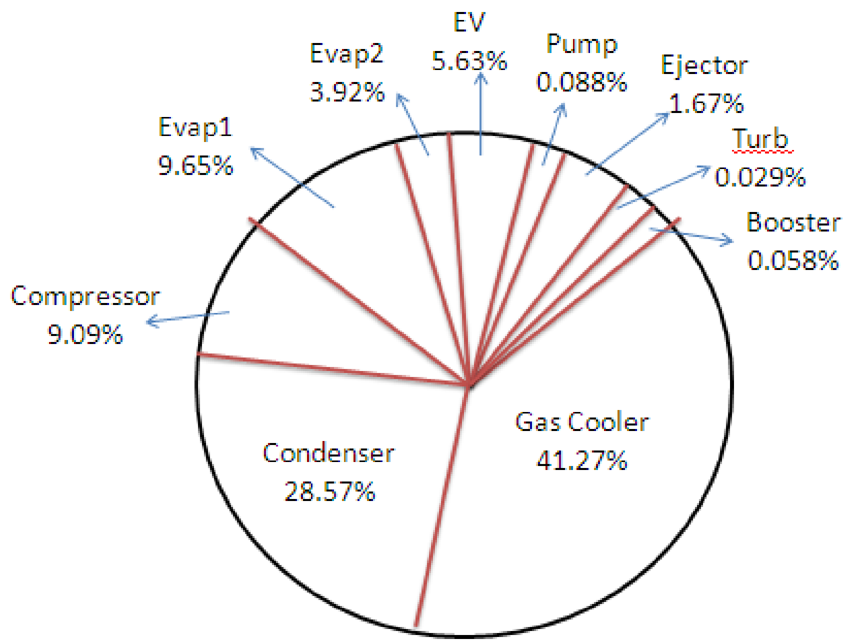
The best and more effective refrigeration cycle cannot be selected based only on exergy analysis.


Gas cooler
68.91%
Expansion valve
22.047%
Evaporator
8.79%
Compressor
0.27%

4 (a)


Pump
0.188%
Compressor
7.45%
Ejector
5.85%
Evap2
2.41%
Booster
0.105%
Evap1
5.87%
EV2
3.62%
EV1
28.42%
Condenser
18.47%
Gas Cooler
27.59%

4(b)



4 (c)

Fig. 14. The percentages of the components' contribution to the total amount of ED,totparameter for the cycles (a) VCC (b) CRC and (c) NCRCPC.

Table 8

Exergy evaluation results for various refrigeration cycles.

Te1 = -25°C, Te2 = 0°C, Tgc = 40°C, Tcd = 30°C

Parameter	**Basic VCC**	**CRC**	**NCRCPC**
Exergy destruction rate [kW]	73.34	56.887	33.97
Second law efficiency [%]	13.606	14.29	22.39

Table 9

Comparision of Energy Results of The Refrigerant N2O And CO2 For NCRCPC

Parameter	**N2O**	**CO2**
The net power rate (kW)	42.41	47.76
The produced cooling rate (kW)	100	100
U	0.445	0.435

COP	2.358	2.094
$P_{opt,gc}$[MPa]	8	9.2

Table 10

Comparision of Exergy Results of The Refrigerant N2O And CO2 For NCRCPC

Parameter	**N2O**	**CO2**
Exergy destruction rate [kW]	33.97	34.094
Second law efficiency [%]	22.39	24.86

CHAPTER 5
CONCLUSIONS

In the present work, a comprehensive study of three refrigeration cycles, the VCC, the CRC, and the new proposed NCRCPC based on energy analysis was conducted. Performance comparison of the three refrigeration cycles was presented. The value of the optimum pressure in the gas cooler was determined, and the cooling capacity was fixed at a constant value of 100 kW. It was found that the minimum value of the optimum pressure was 11.11% and 6.32% lower for the new NCRCPC than those values for the VCC and CRC, respectively. The decrease in pressure increases the safety of the system and its lifetime operation. The value of the COP for the new NCRCPC was found to be 2.18 times the value of the VCC and 1.49 times the value of the CRC, which indicated the energetic performance superiority of the new cycle. The obtained results revealed that NCRCPC has the lowest total exergy destruction. This value for NCRCPC is about 42% lower than that of VCC and CRC. Furthermore, it was found that the highest three components in exergy destruction for the first VCC were the expansion valve with 16.17 kW, the gas cooler with 50.54 kW and the evaporator with 6.43 kW. While, in the NCRCPC the top three components were the gas cooler with 14.02 kW, the condenser with 9.706 kW and the evaporator 1 with 3.28 kW. The new NCRCPC exergy efficiency (ε) was found to be more than that of the VCC and CRC. Finally, as the temperature of the gas cooler increased, the irreversibilities of the refrigeration system increased, which would lead to the decrements in exergy efficiency.

When the evaporator temperature increased, an improvement in energy and exergy efficiency was achieved, and the TUPC might be reduced. The effects of this work are inspire and sell the in addition improvement of NCRCPC.

REFERENCES

- Islam Md, Huda N, Abdullah A, Saidur R. A comprehensive review of state-of- the-art concentrating solar power (CSP) technologies: Current status and research trends. Renew Sustain Energy Rev 2018; 91(C):987–1018.
- Padilla R, Demirkaya G, Goswami D, Stefanakos E, Rahman M. Analysis of power and cooling cogeneration using ammonia-water mixture. Energy 2010;35(12):4649–57.
- Fontalvo A, Pinzon H, Duarte J, Bula A, Quiroga A, Padilla R. Exergy analysis of a combined power and cooling cycle. Appl Therm Eng 2013;60(1–2):164–71.
- Zare V, Mahmoudi S, Yari M, Amidpour M. Thermoeconomic analysis and optimization of an ammonia-water power/cooling cogeneration cycle. Energy 2012;47(1):271–83.
- Chen X, Su Y, Omer S, Riffat S. Theoretical investigations on combined power and ejector cooling system powered by low-grade energy source. International Journal of Low-Carbon Technologies 2016;11(4):466–75.
- Tamm G, Goswami D. Novel combined power and cooling thermodynamic cycle for low temperature heat sources, part II: Experimental investigation. J SolEnergy Eng 2003;125:223–9.
- Lopez-Villada J, Ayou D, Bruno J, Coronas A. Modelling, simulation and analysis of solar absorption power-cooling systems. Int J Refrig 2014;39:125– 36.

- ❖ Zheng D, Chen B, Qi Y, Jin H. Thermodynamic analysis of a novel absorption power/cooling combined cycle. Appl Energy 2006;83:311–23.
- ❖ Tashtoush B, Al-Nimr M, Khasawneh M. A comprehensive review on ejector design, performance and applications. Appl Energy 2019;240:138–72.
- ❖ Jing X, Zheng D. Effect of cycle coupling-configuration on energy cascade utiliza- tion for a new power and cooling cogeneration cycle. Energy Convers Manage 2014;78:58–64.
- ❖ Zare V. Comparative thermodynamic analysis of two tri-generation systems uti- lizing low grade geothermal energy. Energy Convers Manage 2016;118:264–74.
- ❖ Cao L, Wang J, Wang H, Zhao P, Dai Y. Thermodynamic analysis of a Kalina based combined cooling and power cycle driven by low-grade heat source. Appl Therm Eng 2017;111:8–19.
- ❖ Rashidi J, Ifaei P, Esfahani I, Ataei A, Yoo C. Thermodynamic and economic studies of two new high efficient power-cooling cogeneration systems based on Kalina and absorption refrigeration cycles. Energy Convers Manage 2016;127:170–86.
- ❖ Gandeniz S. Thermodynamic analysis of a combined power/refrigeration cycle: Combination of Kalina cycle and ejector refrigeration cycle. Energy Convers Manage 2018;157:631–43.
- ❖ Alexis G. Performance parameters for the design of a combined refrigeration and electrical power cogeneration system. Int J Refrig 2007;30:1097–103.
- ❖ Wang J, Dai Y, Sun Z. A theoretical study on a novel combined power

and ejector refrigeration cycle. Int J Refrig 2009;32:1186–94.

- Wang J, Dai Y, Gao L, Ma S. A new combined cooling, heating and power system driven by solar energy. Renewable Energy 2006;34:2780–8.

- Megdouli K, Tashtoush B, Nahdi E, Elakhdar M, Mhimid A, Kairouani L. Performance analysis of a combined vapour compression cycle and ejector cyclefor refrigeration cogeneration. Int J Refrig 2016;74:517–27.

- Zheng B, Weng Y. A combined power and ejector refrigeration cycle for low tem-perature heat sources. Sol Energy 2010;84:784–91.

- Megdouli K, Tashtoush B, Ezzaalouni Y, Nahdi E, Mhimid A, Kairouani L. Performance analysis of a new ejector expansion refrigeration cycle (NEERC) for power and cold: Exergy and energy points of view. Appl Therm Eng2017;122:39–48.

- Lemmon E, McLinden M, Huber M. NIST standard reference database 23: reference fluid thermodynamic and transport properties-REFPROP, version 9.0. Gaithersburg, USA: National Institute of Standards and Technology, Standard Reference Data Program; 2010.

- Mosaffa A, Farshi G. Thermodynamic and economic assessments of a novel CCHP cycle utilizing low-temperature heat sources for domestic applications. Renewable Energy 2018;120:134–50.

- Bejan A, Tsatsaronis G. Thermal design and optimization. John Wiley & Sons; 1996.p.9.

- Shengya H, Yaodong Z, Lijun Y, Fengyuan Z. Sheng C. Optimization of the combined supercritical CO2 cycle and organic Rankine cycle using zeotropic mixtures for gas turbine waste heat recovery. Energy Convers Manage 2018;160:313–25.

- Towhid P, Hossein A, Reza B, Hadi G, Javad J. Thermodynamic and thermo- economic analysis of a novel ammonia water mixture combined cooling, heating, and power (CCHP) cycle. Renewable Energy 2020:1451158–75.

- Selvaraju A, Mani A. Experimental investigation on R134a vapour ejector re- frigeration system. Int J Refrig 2006;36:1160–6.
- Farivar F, Tatiana M. Exergoeconomic analysis of carbon dioxide transcritical re-frigeration machines. Int J Refrig 2014;38:128–39.
- Ghaebi H, Parikhani T, Rostamzadeh H. A novel trigeneration system using geothermal heat source and liquefied natural gas cold energy recovery: Energy, exergy and exergoeconomic analysis. Renewable Energy 2018;119:513–27.
- Rostamzadeh H, Ghaebi H, Parikhani T. Thermodynamic and thermoeconomic analysis of a novel combined cooling and power (CCP) cycle. Appl Therm Eng 2018;139:474–87.
- Lontsi F, Hamandjoda O, Maya O, Kemajou A. Development and performance analysis of a multi-temperature combined compression/ejection refrigeration cycle using environment friendly refrigerants. Int J Refrig 2016;69:42–50.
- Tashtoush B, Bani Younes M. Comparative Thermodynamic Study of Refrigerants to Select the Best Environment-Friendly Refrigerant for Use in a Solar Ejector Cooling System. Arab J Sci Eng 2019;44(2):1165–84.
- Tim E. Refrigeration in the global economy. Australian and New Zealand Grapegrower and Winemaker 2016;625:68–9.
- Singh S, Dasgupta M. Evaluation of research on CO2 transcritical work recovery expander using multi attribute decision making methods. Renew Sustain Energy Rev 2016;59:119–29.

- Megdouli K, Tashtoush BM, Nahdi E, Elakhdar M, Kairouani L, Mhimid A. Thermodynamic analysis of a novel ejector cascade refrigeration cycles for freezing process applications and air-conditioning. Int J Refrig 2016;70:108–18.

- Elbel S, Lawrence N. Review of recent developments in advanced ejector tech- nology. Int J Refrig 2016;62:1–18.

- Elakhdar M, Tashtoush B, Nehdi E, Kairouani L. Thermodynamic analysis of a novel Ejector Enhanced Vapor Compression Refrigeration (EEVCR) cycle. Energy 2018;163:1217–30.

- Antonio Y, Périlhon C, Descombes G, Chacoux C. Thermodynamic Modelling of an Ejector with Compressible Flow by a One-Dimensional Approach. Entropy 2012;14:599–613.

- Tashtoush B, Alshare A, Alrifai S. Performance study of ejector cooling cycle at critical mode under superheated primary flow. Energy Convers Manage 2015;94:300–10.