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A Comparative Performance study of Combined Cooling Heating and Power Transcritical CO_2 and N_2O Cycles

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Abstract **- Steady state thermodynamic performance of CO_2 and N_2O transcritical combined cooling heating and power cycle have been studied. The effect of the influencing operating parameters such as evaporator temperature, gas cooler outlet temperature, turbine inlet temperature and gas heater pressure on COP and exergetic performance has been investigated. It is found that the performances of transcritical N_2O combined cooling heating and power cycle is better than transcritical CO_2 combined cooling heating and power cycle under all operating conditions. It is also noticed that there is optimum exergetic efficiency at a particular evaporator temperature. A Grassmann diagram for both CO_2 and N_2O combined cooling heating and power cycles presented at a given operating condition.**

Keywords **- Cooling heating and power cycle, grassmann diagram, energy, exergy, transcritical.**

I. INTRODUCTION

The use of natural refrigerants like hydrocarbon, water, carbon dioxide, ammonia etc. are preferred over halogenated refrigerants CFCs and HCFCs ,due to the two menaces; global warming and ozone layer depletion. In recent years the researchers and environmentalist attempted to revive carbon dioxide as one of the promising natural refrigerant and succeeded to draw attention around the world.

Lorenzen et al. [1-3] successfully demonstrated the use of natural refrigerant CO_2 in transport air conditioning with renewed approach. Further certain excellent thermo physical and heat transfer properties coupled with cheap cost, ease of availability, non-toxicity, non-flammability characteristics have made CO_2 as one of the formidable and preferred alternate refrigerant. Kim et al. [4] demonstrated the system design issues and cycle modifications of CO_2 as refrigerant and explained the advantageous side of CO_2 of low-pressure ratio and high volumetric capacity.

Kauf [5] and Liao et al. [6] explained that there exists an optimum gas cooler pressure for a given gas cooler outlet temperature in a transcritical CO_2 vapour compression cycle for maximum COP.

Sarkar et al. [7] demonstrated the correlations for simultaneous cooling and heating applications for optimum gas cooler pressure in transcritical CO_2 heat pump system. Then later, Bhattacharyya et al. [8] carried out optimization studies on a CO_2-C_3H_8 cascade cycle and showed the existence of an optimum pressure in heating and cooling applications.

Agrawal et al. [9] carried out three different two-stage transcritical CO_2 heat pump cycles and optimized the inter stage pressure and gas cooler pressure simultaneously. They proposed empirical correlations to obtain optimum value of inter stage pressure and gas cooler pressure. Lu et al. [10] studied novel combined power - refrigeration cycle using ammonia-water binary mixture as a working fluid and optimized the thermal performances. Agrawal et al. [11] studied based on steady state operation of combined power refrigeration transcritical CO_2 cycle. Stegou-Stagia et al. [12] studied exergy loss analysis using different mixtures such as (R-404 A, R 410-A, R 410-B and R 507). Yang et al. [13] studied the energy and exergy analysis of a simple refrigeration transcritical carbon dioxide cycle using throttle valve and expander.

A comparison was made between transcritical carbon dioxide power cycle and an organic rankine cycle (ORC) using R 123 as working fluid by Chen .Y. et al. [14]. It is found that the transcritical carbon dioxide power cycle with low-grade waste is relatively better than the organic Rankine cycle at the same mean temperature of heat rejection.

Chen et al. [15], Torres-Reyes et al. [16], Krakow [17] and Yaqub et al. [18] effectively studied exergy analysis of refrigeration and heat pump system get the true insight of the system. A comparative study of two-stage transcritical N_2O and CO_2 cycles have been carried out by Agrawal et al. [19]. Based on cycle simulation, the results are presented for simultaneous optimization of intercooler and gas cooler pressures.

Sarkar [20] studied transcritical N_2O refrigeration system with internal heat exchanger and simultaneously optimized the discharge pressure. Based on different operating parameters a comparative study has been made with its counterpart transcritical CO_2 system. The result shows the use of internal heat exchanger is less profitable in comparison with CO_2 system, considering both COP improvement and high side pressure reduction at optimum condition.

Chen et al. [21] studied energetic analysis of a transcritical CO_2 combined cooling and power cycle using EES for automobile air conditioning. Gases exhausted from automobile are used as a heat generating source for gas heater. It is found that optimum gas cooler pressure for combined cooling and power cycle more or less remains same as that for basic cooling cycle. It is also observed that there exists an optimum gas heater pressure in both basic cycle and combined cooling and power cycle. They observed 40% increase in COP in case of combined cooling and power cycle. Kruse et al. [22] observed that a transcritical CO_2 topping cycle and N_2O bottoming cycle shows poor system performance as compared to an equivalent R23 - R134 cascade system. They also carried out studies on two-stage N_2O systems with flash intercooler and with intermediate liquid injection and also found that using flash intercooler for mean temperatures below -10 ^{0}C system COP is improved as compared to cascade system.

Di Nicola et al. [23] carried out an experimental study on a system with CO_2 and N_2O binary mixture in the low temperature cycle and R404a in the high temperature cycle. Simultaneously the results were compared with a cascade system incorporating R23 in low temperature cycle and R404a in high temperature cycle. Sarkar et al.[24] in their pioneering study has shown that transcritical N_2O cycle performs better than that of transcritical CO_2 cycle with respect to system energetic and exergetic performance and system pressure. Bhattacharyya et al. [25] modeled and comprehensively analyzed a N_2O - CO_2 cascade system. Anjan et al. [26] carried out parametric steady state first law and second law analysis of simple and combined power refrigeration transcritical CO_2 cycles.

The present work is aimed to carry out the theoretical thermodynamic analysis of transcritical CO_2 and N_2O combined cooling heating and power cycle and draw a comparison between them.

II. Mathematical Modeling

2.1. Process analysis of combined cooling heating and power cycle

Fig. 1 shows the schematic diagram of combined cooling heating and power system consisting of two internal heat exchangers, turbine, gas heater and other conventional refrigeration cycle components. Corresponding T-s diagram of combined cooling heating and power cycle is shown in Fig. 2.

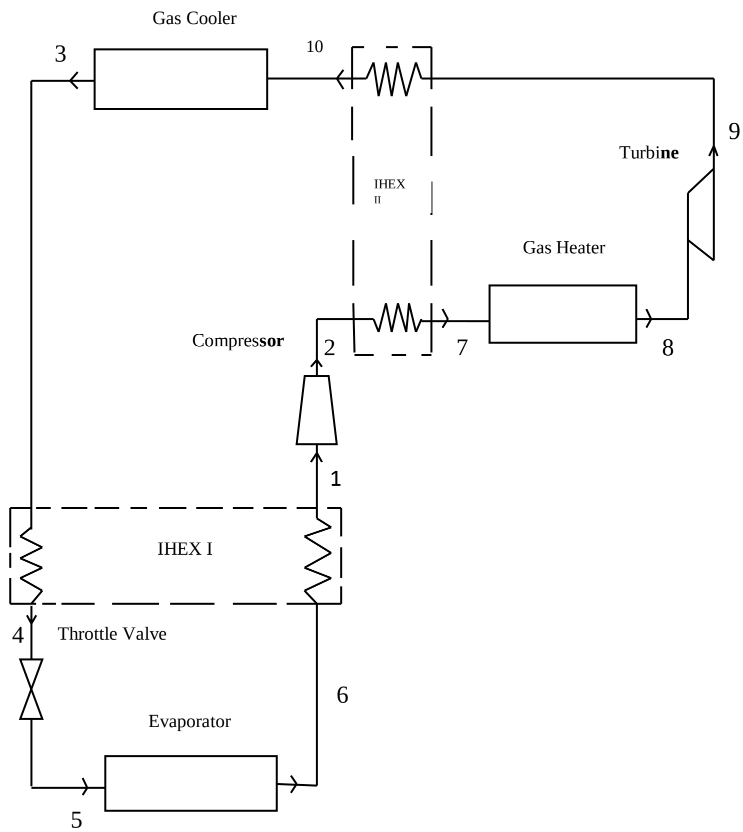

Fig.1. Schematic Diagram of Transcritical Combined cooling heating and power cycle [26]

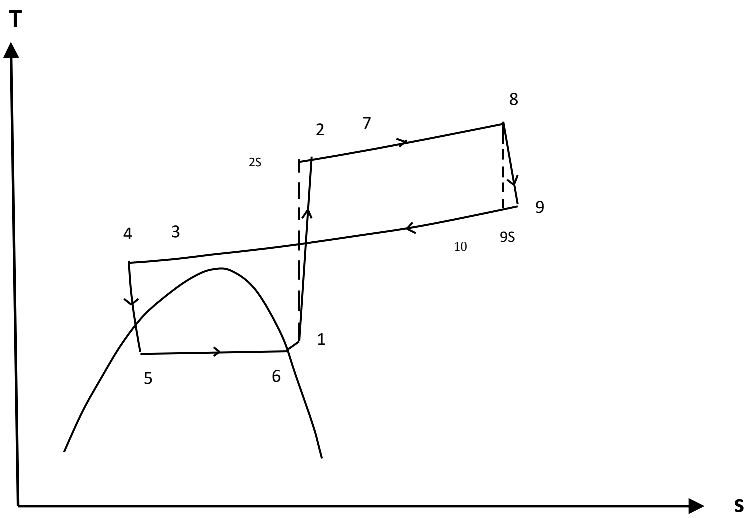

Fig.2. T-s Diagram of Transcritical Combined cooling heating and power cycle [26]

The following assumptions have been made in the analysis.

- Heat transfer with ambient is negligible
- Compression process in compressor is adiabatic but not isentropic.
- Expansion process in turbine is adiabatic but not isentropic.
- Expansion process in throttle valve is constant isenthalpic.
- Evaporation, gas cooling, gas heating and heat exchanger processes are isobaric.

2.2. Specific energy analysis of combined cooling heating and power cycle

Refrigerating effect of evaporator

$$q_{evap} = h_6 - h_5 \tag{1}$$

Work input to compressor

$$w_{comp} = h_2 - h_1 \tag{2}$$

Heat rejected in gas cooler

$$q_{gc} = h_{10} - h_3 \tag{3}$$

Heat supplied in gas heater

$$q_{gh} = h_8 - h_7 \tag{4}$$

Work output in turbine

$$w_{tur} = h_8 - h_9 \tag{5}$$

Isentropic efficiency of compressor

$$\eta_{is.comp} = \frac{h_{2S} - h_1}{h_2 - h_1} \tag{6}$$

Isentropic efficiency of turbine

$$\eta_{is.tur} = \frac{h_8 - h_9}{h_8 - h_{9S}} \tag{7}$$

Effectiveness of internal heat exchanger II

$$\varepsilon_{ihex\ \mathrm{II}} = \frac{h_7 - h_2}{h_9 - h_2} \tag{8}$$

Energy balance in Internal heat exchanger I

$$h_3 - h_4 = h_1 - h_6 \tag{9}$$

COP of combined cooling heating and power cycle

$$COP_{combined} = \frac{q_{evap} + q_{gc}}{(w_{comp} - w_{tur}) + q_{gh}} \tag{10}$$

2.3. Specific exergy loss in combined cooling heating and power cycle

Exergy loss in the compressor

$$i_{comp} = T_o\,(s_2 - s_1) \tag{11}$$

Exergy loss in the evaporator

$$i_{evap} = T_o\,(s_6 - s_5) - \frac{T_o}{T_{evap}}\,(h_6 - h_5) \tag{12}$$

Exergy loss in the throttle valve

$$i_{tv} = T_o\,(s_5 - s_4) \tag{13}$$

Exergy loss in the gas cooler

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$$i_{gc} = (h_{10} - h_3)\frac{T_o}{T_{gc}} - T_o(s_{10} - s_3) \quad (14)$$

Exergy loss in the gas heater

$$i_{gh} = T_o(s_8 - s_7) - \frac{T_o}{T_{gh}}(h_8 - h_7) \quad (15)$$

Exergy loss in turbine

$$i_{tur} = T_o(s_9 - s_8) \quad (16)$$

Exergy loss in internal heat exchanger I

$$i_{hex\,I} = T_0\{(s_1 + s_4) - (s_3 + s_6)\} \quad (17)$$

Exergy loss in internal heat exchanger II

$$i_{hex\,II} = T_0\{(s_7 + s_{10}) - (s_2 + s_9)\} \quad (18)$$

2.4. Specific exergy change in combined cooling heating and power cycle

Exergy change in gas cooler

$$e_{gc} = (h_{10} - h_3) - T_o(s_{10} - s_3) \quad (19)$$

Exergy change in the evaporator

$$e_{evap} = (h_6 - h_5) - T_o(s_6 - s_5) \quad (20)$$

Exergy change in the compressor

$$e_{comp} = (h_2 - h_1) - T_o(s_2 - s_1) \quad (21)$$

Exergy change in turbine

$$e_{tur} = (h_8 - h_9) - T_o(s_8 - s_9) \quad (22)$$

Exergy change in the gas heater

$$e_{gh} = (h_8 - h_7) - T_o(s_8 - s_7) \quad (23)$$

Actual energy supplied to the compressor

$$w_{act,comp.} = \frac{h_2 - h_1}{\eta_{mech}} \quad (24)$$

II law efficiency (η_{II}) in

$$\eta_{II} = \frac{\text{Exergy output}}{\text{Exergy input to the system}} \times 100\ \%$$

$$= \frac{e_{evap} + e_{gc} + e_{tur}}{(w_{act.comp.} - w_{tur}) + e_{gh}} \times 100\ \% \quad (25)$$

The working condition of combined cooling heating and power cycles are listed in Table.1.

Table1
Combined Cooling Heating And Power Cycles Operating Parameters

Operating Value		Unit
Ambient Temperature (T_o)	303	K
Evaporator Temperature for combined cycle (T_{evap})	270	K
Super heat after evaporator for Combined cycles	5 (Fixed Value)	K
Gas cooler exit Temperature (T_{gc})	308	K
Gas Heater Pressure (P_{gh})	140	Bar
Gas Heater temperature (T_{gh})	460	K
Compressor Isentropic efficiency for Combined cycle ($\eta_{is.comp}$)	0.75	---
Turbine isentropic efficiency ($\eta_{is.tur}$)	0.8	---
Mechanical efficiency of (η_{mech})	0.8	---
Effectiveness of IHEX II ($\varepsilon_{ihex\ ii}$)	0.7	---

A computer code has been developed for the energy and exergy analysis of combined cooling heating and power cycle for various operating parameters. Sub-critical and super-critical thermodynamic and transport properties of CO_2 and N_2O are calculated employing CO_2 property code CO2PROP [7] and N_2O property code N2OPROP.

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III. Results And Discussion

The performance comparison of the CO_2 and N_2O combined cooling heating and power cycle is presented based on the operating conditions such as evaporator temperature, gas cooer outlet temperature, gas heater pressure and turbine inlet temperature. The isentropic efficiency of the compressor and the turbine are taken as 75% and 80%, respectively. The effectiveness of IHEX-II is taken to be 0.7 and the mechanical efficiency of the compressor is taken as 80%. It is assumed that the gas is heated by 5^0 C in the IHEX-I as referred to [21]. The intermediate gas cooler pressure is taken as the geometric mean of inlet and outlet pressure of compressor.

3.1. Effect of evaporator temperature on the performances of CO_2 and N_2O cycles

Fig. 3 presents the variation of COP with evaporator temperature considering T_3 = 308 K, T_8 = 500 K and P = 170 bar respectively. As evaporator temperature increases COP increases for both CO_2 and N_2O combined cooling heating and power cycles. However, the increase in COP of N_2O cycle is marginal better in comparison with CO_2 cycle.

Fig. 4 shows the variation of second law efficiency of CO_2 and N_2O combined cooling heating and power cycle. It can be observed that the second law efficiency initially increases and attains maximum values of 37.21% and 38.12% for CO_2 and N_2O cycles respectively and then starts decreasing with evaporator temperature.

3.2. Effect of gas cooler outlet temperature on the performances of CO_2 and N_2O cycles

Fig.5. presents the variation of COP with gas cooler outlet temperature. Reduction in refrigerant effect decreases the COP with increase in gas cooler outlet temperature for both the combined cooling heating and power CO_2 and N_2O cycles. However, the variation in COP for CO_2 cycle is relatively less in comparison to N_2O cycle.

Fig.6 presents the variation of exergetic efficiency with gas cooler outlet temperature for both, CO_2 and N_2O combined cooling heating and power cycles. It is observed that the exergy loss in components increases with gas cooler outlet temperature for both the cycles. In N_2O combined cooling heating and power cycle, the exergetic efficiency percentage decreases significantly in comparison to CO_2 combined cycle.

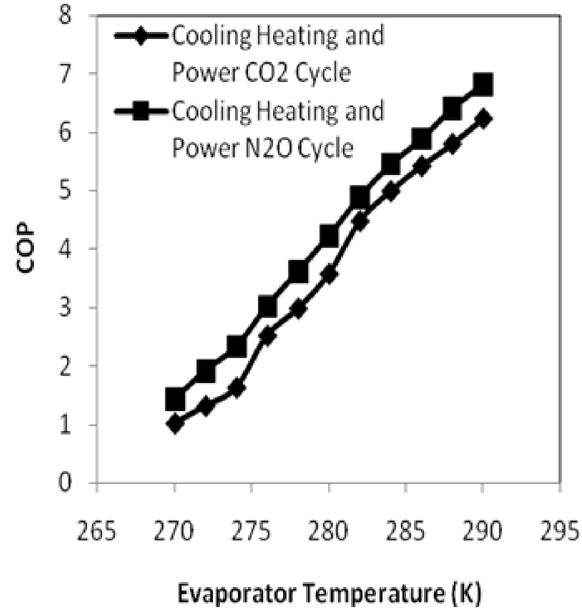

Fig.3. Variation of COP with evaporator temperature for T_3 = 308 K, T_8 = 500 K and P = 170 Bar.

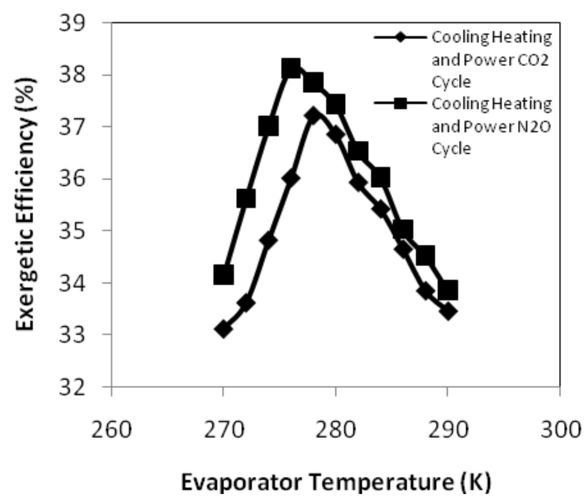

Fig.4. Percentage Exergetic efficiency with evaporator temperature for T_3 = 308 K, T_8 = 500 K and P = 170 Bar.

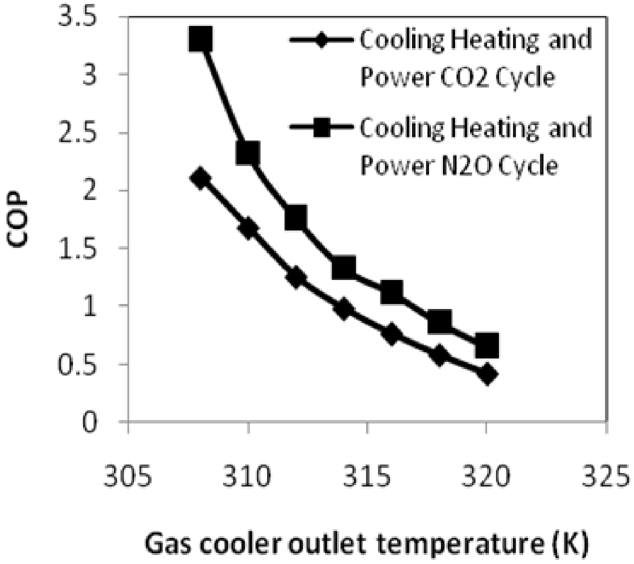

Fig.5. Variation of COP with gas cooler outlet temperature for T_{evap} = 275 K, T_8 = 500 K and P = 170 Bar.

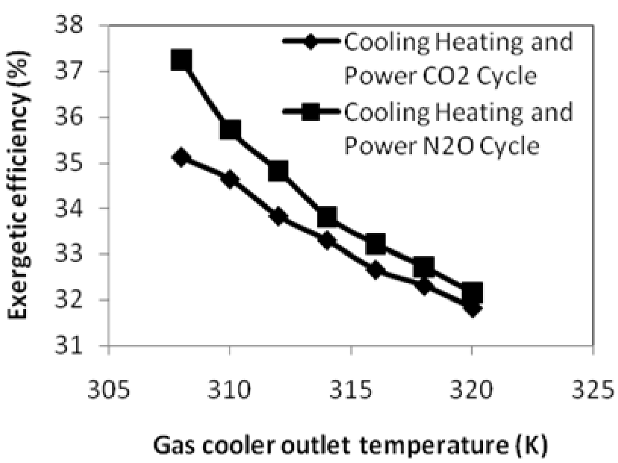

Fig.6. Percentage Exergetic efficiency with gas cooler outlet temperature for T_{evap} = 275 K, T_8 = 500 K and P = 170 Bar.

3.3. Effect of turbine inlet temperature on the performances of CO_2 and N_2O cycles

Fig. 7 presents the variation of COP with turbine inlet temperature. As turbine inlet temperature increases 460 K to 600 K, the turbine output increases in turn COP increases monotonically for both the combined cooling heating and power cycles. However, the COP is significantly better for N_2O combined cycle.

Fig. 8 shows the effect of turbine inlet temperature on the exergetic efficiency on the combined cooling heating and power cycles. The trend of both cycles is same, increases monotonically with turbine inlet temperature. However, the combined exergetic efficiency is relatively better for N_2O combined cycle.

3.4. Effect of gas heater pressure on the performances of CO_2 and N_2O cycles

Gas heater pressure which is the discharge pressure of the compressor is an important parameter. Fig. 9 depicts the variation of COP with gas heater pressure. As the gas heater pressure increases COP increases monotonically for both the combined CO_2 and N_2O cycles. However, the rise of COP for N_2O combined cycle is quite better than CO_2 cycle. The similar trend is observed for exergetic efficiency with gas heater pressure as shown in Fig. 10.

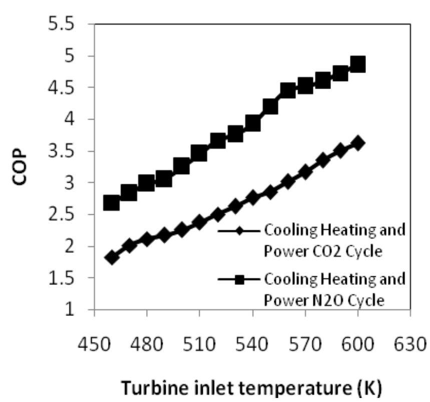

Fig.7. COP with turbine inlet temperature for T_{evap} = 275 K, T_3 = 308 K and P = 175 Bar

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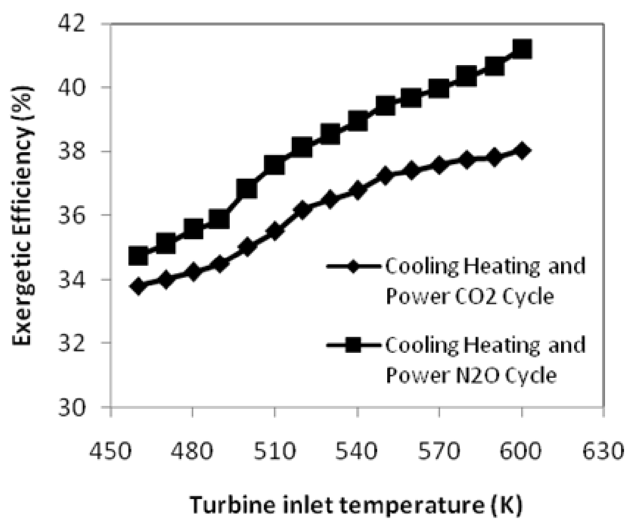

Fig.8. Percentage exergetic efficiency with turbine inlet temperature for T_{evap} = 275 K, T_3 = 308 K and P = 175 Bar

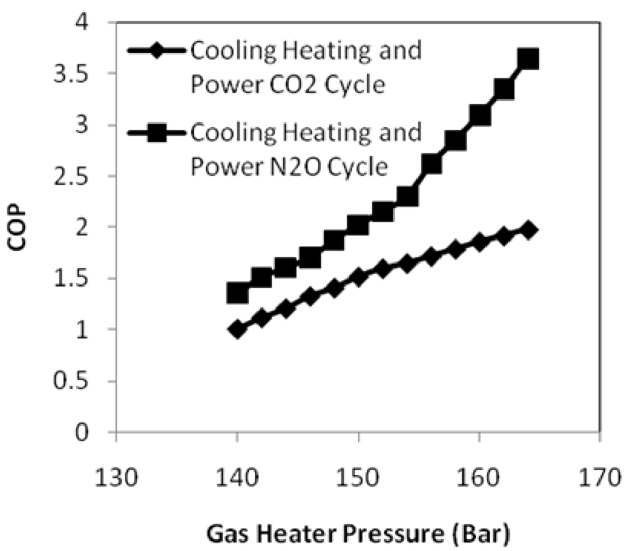

Fig.9. COP variation with Gas heater pressure for T_{evap} = 275 K, T_3 = 308 K and T_8 =500 K.

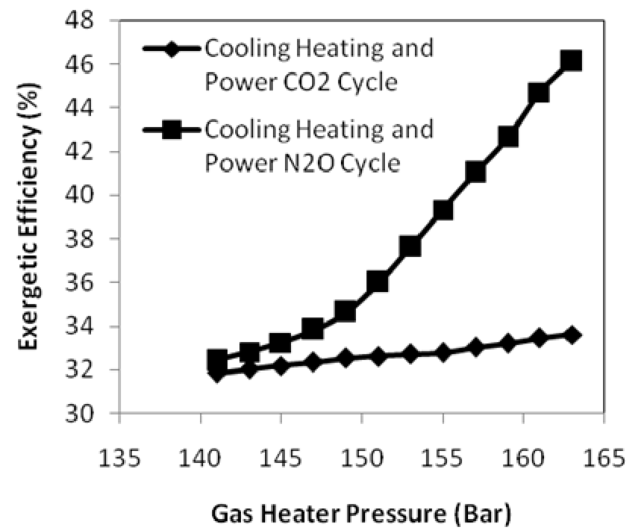

Fig.10. Variation of percentage exergetic efficiency with gas heater pressure for T_{evap} = 275 K, T_3 = 308 K and T_8 = 500 K.

3.5. Grassmann diagram for CO_2 and N_2O cooling heating and power cycles

Figs. 11 and 12 present the percentage exergy loss in components for CO_2 and N_2O combined cooling heating and power cycles, respectively. It is defined as the % exergy loss in components with respect to exergy input (100 %) to the compressor and gas heater. The operating conditions are taken as T_{evap} = 275 K, T_3 = 308 K, T_8=500 K and P = 175 bar for both the cycles. It is found that 60.35 % and 61.86 % exergy recovered at turbine, gas cooler and evaporator for CO_2 and N_2O combined cycles.

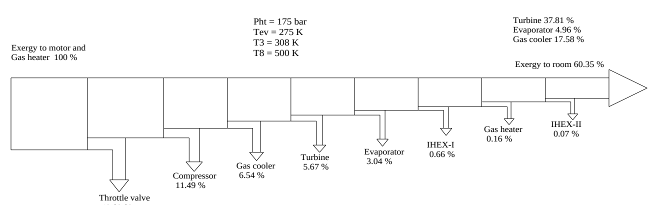

Fig. 11. Grassmann diagram for CO_2 combined cooling heating and power cycle

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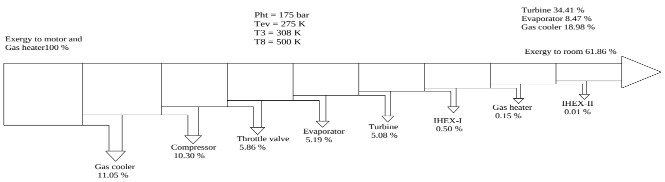

Fig. 12. Grassmann diagram for N_2O combined cooling heating and power cycle

IV. CONCLUSIONS

A comparative study of transcritical CO_2 and N_2O combined cooing heating and power cycle is carried out based on steady state thermodynamics analysis. Some of the conclusions are outlined below based on the analysis.

1. The thermodynamic performances like COP and exergetic efficiency of N_2O combined cooling heating and power cycle is better than CO_2 combined cooling heating and power cycle for all operating conditions.
2. There is an existence of optimum exergetic efficiency for both CO_2 and N_2O combined cooling, heating and power cycle at evaporator temperature.
3. The COP and exergetic efficiency are in decreasing trend for both CO_2 and N_2O combined cooling heating and power cycle with gas cooler outlet temperature.
4. The COP and exergetic efficiency strongly influenced by gas heater pressure and turbine inlet temperature. It is observed that COP and exergettic efficiency increases with gas heater pressure and turbine inlet temperature for both the cycles. However, the increase of COP and exergetic efficiency is more significant in N_2O cycle.

Nomenclature:

h	specific enthalpy (kJ /kg)
q	specific heat transfer (kJ /kg)
i	specific exergy loss (kJ/kg)
T	temperature (K)
e	specific exergy (kJ /kg)
w	specific work transfer (kJ /kg)
s	specific entropy (kJ /kg.K)
P	pressure (bar)
IHEX	internal heat exchanger
COP	coefficient of performance

Greek symbols

ε	effectiveness
η	efficiency

Subscripts

1 – 10	state points of refrigerant
comp	compressor
evap	evaporator
tur	turbine
gc	gas cooler
comb	combined cycle
gh	gas heater
tv	throttle valve
is	isentropic
mech	mechanical efficiency
o	ambient
II	second law
ht	high temperature

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