Indian Journal of Science and Technology, Vol 8(31), DOI: 10.17485/ijst/2015/v8i31/84313, November 2015 ISSN (Print) : 0974-6846
ISSN (Online) : 0974-5645

Experimental Investigation of a Combined Power Refrigeration Transcritical CO_2 Cycle

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Abstract

Optimization studies of the CO_2 transcritical combined power refrigeration cycle are presented here. Steady state thermodynamic analysis including energy is carried out at various operating conditions employing a precision in-house property code CO_2 PROP. CO_2 transcritical combined power refrigeration cycle exhibits better performance than that of CO_2 transcritical refrigeration cycle alone. However, optimum gas cooler pressure does not change with the type of the system; it remains same for both the system; refrigeration and combined cycle. However, system performance is significantly affected by the evaporator pressure and compressor isentropic efficiency unlike the CO_2 transcritical refrigeration system Combined cycle exhibits significant improvement in COP at relatively higher evaporative temperature. However, at lower evaporator temperature cooling cycle shows better performance.

Keywords: CO_2, Combined Cycle, Transcritical

1. Introduction

Environmentalists around the world have been seeking more environment friendly refrigerants to replace halogenated refrigerants such as CFCs and HCFCs owing to their global warming, ozone depletion and atmospheric pollution problems. Natural refrigerants such as water, carbon dioxide, ammonia are obvious choice. While a renewed interest has already been observed in environmentally benign natural refrigerants[1–3], such findings are likely to accelerate a switch to these alternates. Carbon dioxide is a preferred choice over other natural refrigerants with its excellent thermo physical and heat transfer properties. In addition, its low price, easy availability, non-toxicity, non-flammability, low pressure ratio and high volumetric capacity make it a promising alternative[4].

Previous studies[5–7] show that an optimum gas cooler pressure exists for the transcritical CO_2 cycle where it exhibits the maximum COP for a given cooler outlet temperature. This can be attributed to the unique behavioral pattern of CO_2 properties around the critical point and beyond, where the slope of the isotherms is quite modest for a specific pressure range; at pressures above and below this range, the isotherms become much steeper. Bhattacharyya, et al.[8] have shown in the similar studies that there exist optimum pressures for a CO_2-C_3H_8 cascade system used for cooling and heating. Agrawal, et al.[9] presented optimization studies for two-stage transcritical CO_2 heat pump systems. Correlations for optimum inter stage pressure and gas cooler pressure is obtained through their studies.

A novel combined power-refrigeration thermodynamic cycle has also been optimized for thermal performance using ammonia-water binary mixture as a working fluid[10]. Sarkar and Bhattacharyya[7] optimized transcritical CO_2 heat pump system for simultaneous cooling and heating applications and presented the correlations for optimum gas cooler pressure. But the optimum condition of a combined CO_2 cooling and power cycle has not been reported yet. Owing to unique thermodynamic properties of CO_2, it can be used in the combined cycle where refrigeration and power obtain simultaneously. Such type of system is suitable for several applications, where both power and cooling are needed (e.g. automobile applications, in which the cycle can utilize the energy

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produced in the engine exhaust gasses to produce power and provide cooling/heating to the mobile compartment room at the same time). Being the operation in transcritical mode an optimum gas cooler do exist similar to CO_2 trans critical vapor compression cycle. To the best of the authors' knowledge optimization of combined cycle has not been reported in open literature. A detailed energetic analysis is presented in this paper. Cycle performance is evaluated varying operating parameters such as gas cooler temperature, evaporator pressure, compressor and power turbine isentropic efficiency and internal heat exchanger effectiveness.

2. Mathematical Model

The carbon dioxide cooling and power combined system is mainly composed of six parts, namely: An evaporator, compressor, gas heater and expander, gas cooler and throttling valve[1]. A schematic diagram of the proposed CO_2 power refrigeration cycle and the corresponding T-s diagram have been illustrated in Figures 1 and 2.

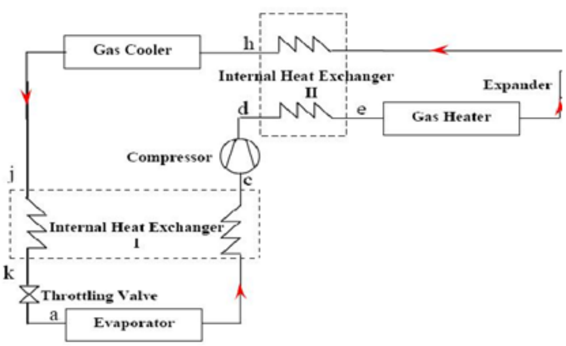

Figure 1. Transcritical CO_2 cooling and power combined system schematic layout.

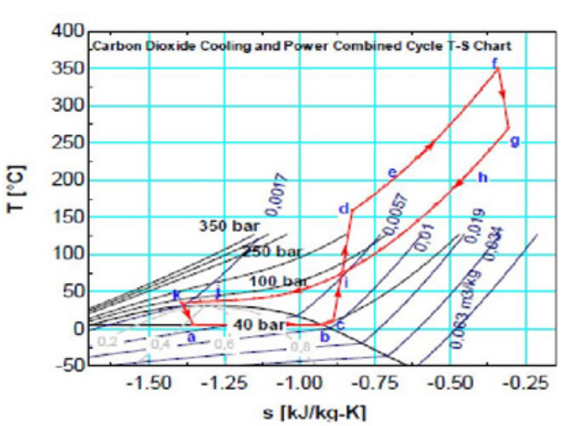

Figure 2. Transcritical CO_2 cooling and power combined cycle T-S chart.

In the entire system, each component is modeled based on steady flow energy balance. The following assumptions have been made to simplify the analysis:

- Heat transfer with the ambient is negligible.
- Single-phase heat transfer occurs for the external fluid.
- Compression process is adiabatic but non-isentropic.
- Evaporation, gas cooling and inter cooling processes are isobaric.

Modular mathematical model is presented below:

2.1 Energy Analysis for the Combined Power Refrigeration Cycle

The energy balance equation for the components of the cycle is given by

$$q_{evr} = h_2 - h_1 \tag{1}$$
$$h_3 - h_2 = h_9 - h_{10} \tag{2}$$
$$\epsilon_I = (h_3 - h_2)/(h_9 - h_2) \approx (T_3 - T_2)/(T_9 - T_2) \tag{3}$$
$$qcom = h4 - h3 \tag{4}$$
$$\eta_{com} = (h_4 s - h_3)/(h_4 - h_3) \tag{5}$$
$$h_5 - h_4 = h_7 - h_8 \tag{6}$$
$$\epsilon_{II} = (h_5 - h_4)/(h_7 - h_4) \approx (T_5 - T_4)/(T_7 - T_4) \tag{7}$$
$$q_{gh} = h_6 - h_5 \tag{8}$$
$$w_{exp} = h_6 - h_7 \tag{9}$$
$$\eta_{exp} = (h_6 - h_7)/(h_6 - h_7 s) \tag{10}$$
$$q_{gc} = h_8 - h_9 \tag{11}$$
$$h_{10} = h_1 \tag{12}$$

Traditionally, the COP of a vapor compression refrigeration system is defined as:

$$COP\ cool = Qcool / Wcom \tag{13}$$

Where, $Q_{cooling}$ is the refrigerating capacity of the system and W is the required compression work of the compressor which compresses up to gas cooler pressure[2]. It is assumed that waste heat is utilized to increase the temperature and corresponding enthalpy of the compressed CO_2 in the gas heater prior to expand in the turbine for power generation. If the work output of the turbine is W output and the combine work input to the compressor is W_{tot}, the new COP of the combined cycle cooling part can be defined as:

$$COPcom = Qcool / Wnet = Qcool / (Wcom - Wtur) \tag{14}$$

Where $Q_{cooling}$ is the required cooling capacity, W_{new} is the new compression work after taking away the energy gained by the combined cycle power part, W_{tot} is the combine compression work, W_{output} is the work output from the combined cycle power part. It may be noted that W_{tot} will be higher than the work required when the system operates only as simple refrigeration cycle since the refrigerant has out to be compressed up to higher pressure for combined cycle which may be compensated by the turbine work out which has to look into[3].

3. Solution Procedure

A computer code has been developed for the energy and exergy analysis of combined power refrigeration cycle for various operating conditions. Employing the new equation of state for CO_2 and transport property correlations available in the literature, a separate property code CO_2PROP, employing a technique based on derivatives of Helmholtz free energy function using efficient iterative procedures has been developed to calculate sub-critical and super-critical thermodynamic and transport properties of CO_2[7].

4. Results and Discussion

Optimization of the combined transcritical CO_2 cycle is studied under various operating conditions. Heat is supplied to the gas heater at 200 bar to raise the temperature of the working fluid CO_2 up to 350 °C prior to expansion in the power turbine. Gas cooler pressure is optimized considering gas cooler exit temperature at 35°C. Results are generated considering evaporator pressure and temperature are 40 bar and 5.3 0C, respectively for various operating conditions. Until otherwise it is mentioned the compressor efficiency is assumed to be 0.75 and expander efficiency is assumed to be 0.80[4].

4.1 Effect of Gas Cooler Pressure

It is shown by the various researchers that there exists an optimum gas cooler pressure for the CO_2 transcritical refrigeration cycle at which system operates at its best COP for the given gas cooler exit temperature[5–7]. Figure 3 depicts the variation of COP with gas cooler pressure for cooling and combined cycle. It can be seen that optimum pressure for both the cycle is almost same. For both transcritical refrigeration and combined power refrigeration cycles, COP increases initially with gas cooler pressure and beyond a certain value 85 bar, decreases as it is evident from Figure 3. Both the system attains a maximum COP at about 85 bar[5]. However, COP of the combined power refrigeration cycle is significantly higher than that of COP of the transcritical refrigeration cycle). This can be attributed to the fact that despite increase in compressor work due to high operating pressure, higher turbine work output brings down the total compressor work and increase the COP.

4.2 Effect of Compressor Efficiency

Figure 4 presents the variation of COP of the optimized cooling and combined systems against the efficiency of the compressor. Efficiency of the compressor is varied from 0.60 – 0.85. It can be seen that both the system's COP increases with increasing efficiency of the compressor. However, rise in COP is not significant for refrigeration cycle as like shown by previous authors[7] whereas there is significant rise in COP of combined cycle. Further, difference in COP of these two cycles increases as compressor efficiency increases. This may due to the higher operating pressure of the power cycle where entropy lines are diverged in a greater extent which brings significant drop in compressor power due to reduction in compressor exit enthalpy[6].

Figures 4-7 show the effects of different operating conditions for optimized systems. Parameters such as compressor and expander efficiencies, evaporator pressure and temperature, effectiveness of heat exchangers, gas cooler outlet temperature, expander inlet temperature, gas heater pressure are varied within specified range[8].

4.3 Effect of Gas Cooler Outlet Temperature

Variation of COP of both systems with the gas cooler outlet temperature is shown in Figure 5. Outlet temperature of the gas cooler is varied from 303 – 323 K. It can be that COP of both the systems decreases drastically as the outlet temperature of the gas cooler increases due to decrease in refrigerating effect at higher gas cooler outlet temperature. The COP trend for both the cycles is similar with the variation in gas cooler exit temperature.

Figure 6 shows the COP of two systems for different values of the evaporator pressure[12]. For both the systems, increase in evaporator pressure increases the COP. However, the rise in COP for the power cycle is significantly higher that of the cooling cycle as the evaporator pressure increases and increases rapidly beyond a certain evaporator pressure. This is because due to unique temperature lines which bring down the work input to the compressor operates at higher Pressure[9–11].

4.4 Effect of Expander Inlet Temperature

Variation of combined cycle COP with inlet temperature of the turbine is exhibited in Figure 7. The COP of the combined power refrigeration cycle increases almost linearly with the increase in the inlet temperature of the expander. At higher inlet temperature turbine work increases without affecting compressor work which increase the COP.

4.5 Effect of Expander Efficiency

Variation of combined cycle COP is shown in Figure 8. As shown in the figure, COP increases with the increase in efficiency of the expander. This can be attributed to fact that at higher expander efficiency work out is more which increase COP. Cycle COP is more sensitive at higher expander efficiency, relatively increases at faster rate with expander efficiency.

5. Conclusions

A comparative study based on first law of thermodynamics is presented here for the CO_2 transcritical refrigeration cycle and CO_2 trans critical combined power refrigeration cycle. Optimum gas cooler remains unaffected with the type of cycle, refrigeration or combined cycle. However, combined cycle shows higher COP due to significantly gain in turbine work. Compressor and expander isentropic efficiencies are the influential parameters on COP of combined cycle unlike the simple refrigeration cycle. As ike refrigeration system, higher gas cooler exit temperature and lower evaporator pressure brings down the combined cycle COP.

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