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Performance investigation of N_2O transcritical refrigeration cycle using dedicated mechanical subcooling

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ABSTRACT

Thermodynamic analysis of N_2O transcritical refrigeration cycle using dedicated mechanical subcooling cycle has been carried out in the present work. The transcritical cycle with the mechanical subcooling is evaluated for three different evaporator temperatures 5, -5 and -30 ºC with different degrees of subcooling and for the environment temperatures range from 30 to 40 ºC using propane as refrigerant for the subcooling cycle. Performance of N_2O transcritical cycle has been compared with CO_2transcritical cycle. The results show that N_2O transcritical cycle is better than CO_2 cycle for environment temperature above 30 °C, which is an important fact for countries having tropical climate.

Keywords: N_2O, Transcritical Cycle, Performance Investigation, Dedicated Mechanical Subcooling.

I INTRODUCTION

World is facing energy crisis and environmental challenges due to increasing demand of power on account of rapid growth in population and industrial development. Refrigeration and air conditioning sector is also adding to this problem due to high power consumption and environmental issues related to usage of different refrigerants. To solve this issue, the refrigeration sector is undergoing a phase of transition from traditional refrigerants, with high environmental impact, to environmental friendly refrigerants. Due to the zero ozone layer depletion potential and low global warming potential, several natural refrigerants are considered to be better than synthetic refrigerants. As a result nitrous oxide and carbon dioxide are again being considered as future refrigerants. The CO_2 based systems have already gained large acceptance in simultaneous cooling and heating applications, N_2O still remains mostly unexplored. However, CO_2 can be used down to an evaporator temperature of –55 °C and further lowering of temperature cannot be achieved since the triple point of CO_2 is –56.6 °C. On the other hand, N_2O has a triple point temperature of –90.82 °C with a boiling point temperature of –88.47 °C and hence it can be used in the region below the application range of CO_2. Furthermore, similarity between critical temperature, pressure and molecular weight of N_2O and CO_2 (Kruse and Russmann, 2006)[1] causes nearly similar behavior with respect to system temperature and pressure. It must be pointed out that N_2O exhibits five times lower toxicity than CO_2 but its GWP is significantly higher than CO_2. Still it comes under the consideration of low GWP category which is currently considered to have an upper limit of 300. Recent study on transcritical N_2O cycle (Sarkar and Bhattacharyya, 2008)[2] showed higher COP and lower high side pressure compared to CO_2, which motivates the further detailed study.Sarkar et al.(2010)[3]observed that the effect of using internal heat exchanger in the transcritical N_2O cycle on the maximum cooling COP improvement is more significant at lower evaporator temperature and less significant for higher evaporator temperature, whereas the effect on optimum discharge pressure reduction is less significant compared to its counterpart CO_2 cycle. In the present study, the thermodynamic analysis for the performance investigations of N_2O transcritical refrigeration cycle using dedicated mechanical subcooling as well as performance comparison between N_2O and CO_2transcritical cycle has been presented.

II THERMODYNAMIC ANALYSIS

The process layout of N_2O transcritical refrigeration cycle with dedicated mechanical subcooling isshown in Fig

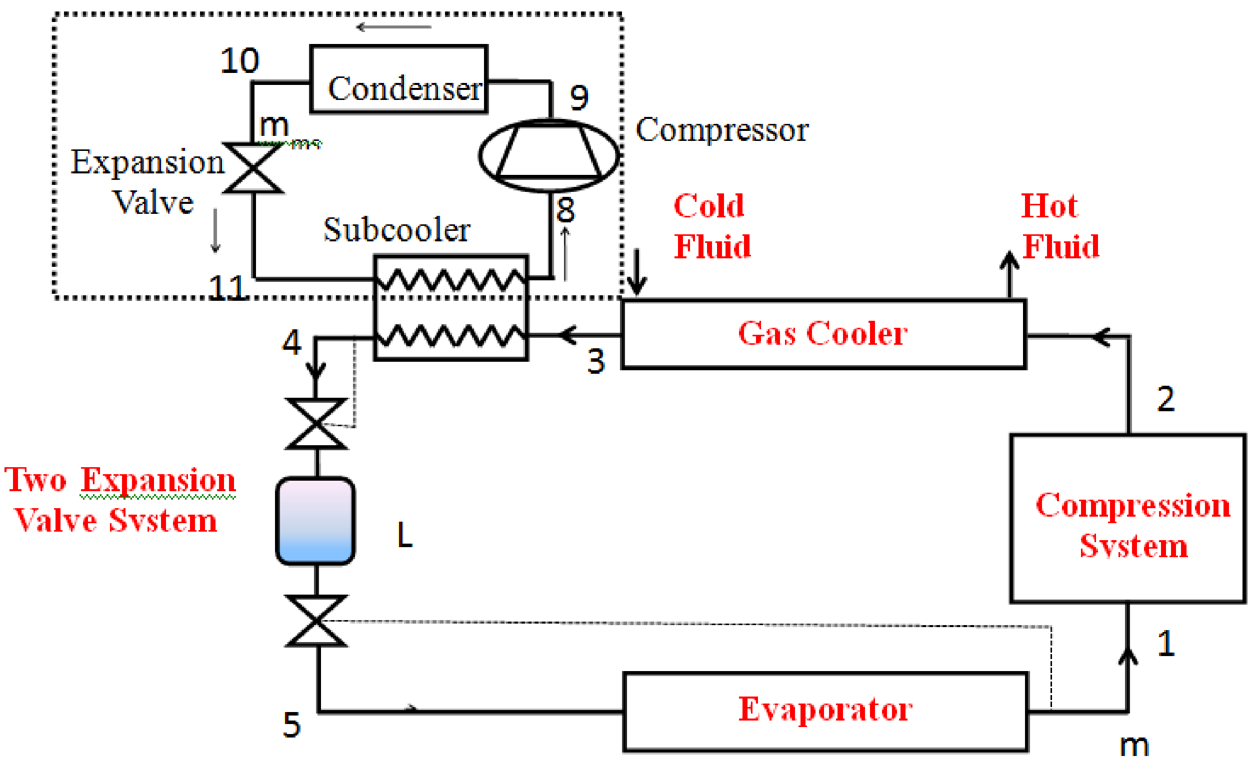

Fig.1 Schematic representation of N_2O transcritical refrigeration cycle with dedicated mechanical subcooling.

To analyze the opportunity of enhancing the performance of N_2O transcritical refrigeration cycle using thededicated mechanical subcooling cycle (the cycle shown in Fig. 1) has been considered. The CO_2transcritical cyclewith a double-stage expansion system (Cabello et al., 2008) [4] with an additional subcooler at the exit of thegas-cooler, where the subcooling is provided by a single-stage compression system, is called the MS cycle. Thetranscritical cycle incorporates a device to regulate the heat rejection pressure and another to control theevaporating process. Here for the evaluation of N_2O transcritical cycle, two compression systems are considered in the main cycle, single-stage for medium and high evaporating temperatures and two-stage with intercooling for low evaporating temperatures. Both cycles perform the heat rejection (in condenser of the MS cycle, in the gas-cooler and in the intercooler of the transcritical cycle) to the same high temperature i.e. the environment temperature.To evaluate the performance of N_2O transcritical cycle, pressure losses and heat transfer to the environment are neglected. Evaporator temperature and a constant degree of superheat in the evaporator of 10 °Chas been fixed. All the calculations are made for the optimum heat rejection pressure, which is calculated by an iteration process.

In this work, thermodynamic analysis for the performance improvements of N_2O transcritical refrigeration cycle using dedicated mechanical subcooling as well as comparison with CO_2 cycle are presented. For the analysis of the transcritical cycle, thermodynamic models have been developed in Engineering Equation Solver software.

2.1. N_2O transcritical refrigeration cycle

Considering the Fig. 1, if $q_{0,r}$ denotes the specific cooling capacity and $w_{c,r}$ is specific compression work of the main cycle than the COP of the main cycle is given by Eq. (1).

$$\mathrm{COP} = \frac{q_{o,r}}{w_{c,r}} \qquad (1)$$

The relation of refrigerant mass flow rates of the cycles is obtained with the energy balance in the subcooler, if Δh_{sub} is the enthalpy difference of the main cycle in subcooler and $q_{0,ms}$ is the specific cooling capacity of MS cycle than the relation of mass flow rates is obtained by Eq. (2).

$$m_{ms} * q_{0,ms} = m_r * \Delta h_{sub} \qquad (2)$$

To obtain gas-cooler exit temperature, Eq. (3), where an approach temperature of 5 °C has been assumed, due to the high heat transfer rates of N_2O in transcritical conditions (Kim et al., 2004). Further, the outlet temperature of subcooler is obtained by considering the subcooling degree in the subcooler, Eq. (4), which is varied in the analysis. The enthalpy difference of N_2O in the subcoolerΔh_{sub} is evaluated by Eq. (5).

$$T_3 = T_{env} + 5\ °C \qquad (3)$$

$$T_4 = T_3 - SUB \qquad (4)$$

$$\Delta h_{sub} = h_3 - h_4 \qquad (5)$$

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The specific cooling capacity is calculated by Eq. (6).

$q_{o,r} = h_1 - h_4 = h_1 - h_3 + \Delta h_{sub}$ (6)

To consider a simple but realistic approach, the compressors are modelled using the same equation of isentropic efficiency, Eq. (7), close to one obtained for a transcritical CO_2 compressor (Sánchez et al., 2010). Here we are using same expression for transcritical N_2O compressor.

$\eta_i = 0.95 - 0.1*d$ (7)

2.1.1 Single stage compression system

The single stage compressor is used for high and medium evaporating temperatures, the specific compression work is calculated by Eq. (8). Energy flow diagram of main compressor is shown in Fig. 2a.

$w_{c,r} = \frac{h_2 - h_1}{\eta_{i,single}}$ (8)

2.1.2 Double stage compression system

For the double stage compression system, booster with intercooler is used. Here the geometric mean of gas-cooler and evaporator pressure is used for the intermediate stage pressure, Eq. (9). Energy flow diagram for double stage compression system is shown in Fig. 2b

$p_6 = p_7 = \sqrt{p_1 * p_2}$ (9)

The specific compression work in the first compressor is calculated by Eq. (10). The isentropic efficiency for the first compressor is calculated by using the pressures p_1 and p_6, it is denoted by $\eta_{i,l}$

$w_{c,l} = \frac{h_6 - h_1}{\eta_{i,l}}$ (10)

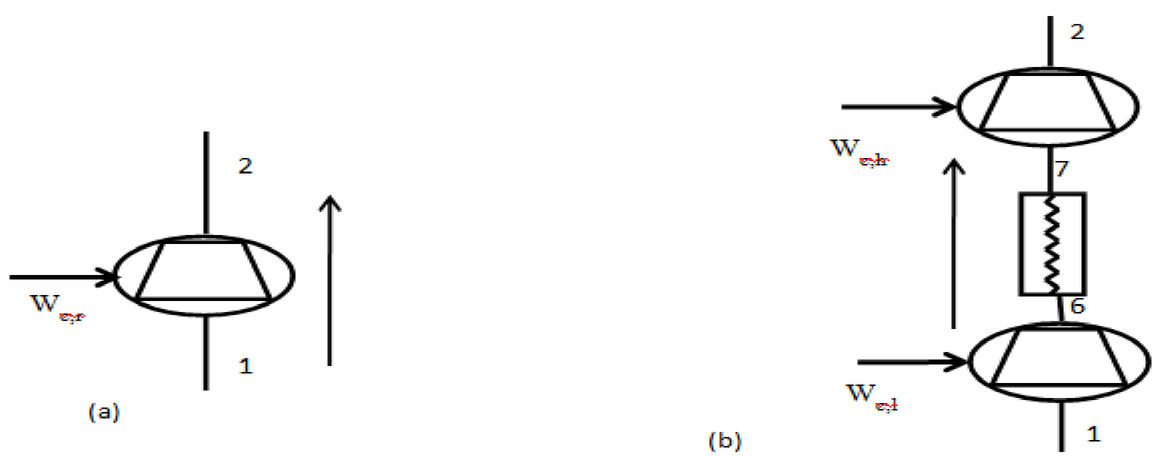

Fig. 2 Single stage compression anddouble stage compression booster with intercooling.

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To obtain the inlet temperature of the second compression stage, an approach temperature with environment of 5 °C has been assumed. It is expressed by Eq. (11).

$$T_7 = T_{env} + 5\ °C \quad (11)$$

The specific compression work for the second compressor is calculated by Eq. (12). The isentropic efficiency for the second compressor is calculated by using the pressures p_7 and p_2.

$$w_{c,h} = \frac{h_2 - h_7}{\eta_{i,h}} \quad (12)$$

The total specific compression work for the double stage compression system is given by Eq. (13).

$$w_{c,r} = w_{c,l} + w_{c,h} \quad (13)$$

2.2 Dedicated mechanical subcooling cycle (MS cycle)

For this cycle only subcritical fluids are considered. Its condensing pressure is evaluated considering a temperature difference with the environment of 10 °C. The refrigerant at the exit of the condenser is in saturated state. The evaporating temperature of the MS cycle can be fixed to be equal to saturation temperature corresponding tomaximum operating pressure of the high temperature compressors. In this case it is 10 ºC.In the mechanical subcooling cycle, the specific cooling capacity of the MS cycle is expressed by Eq. (14).

$$q_{0,ms} = h_8 - h_{10} \quad (14)$$

To find the specific compression work, same expression of isentropic efficiency, i.e. Eq. (7) has been considered. Compression work is given by Eq. (15) .

$$w_{c,ms} = \frac{h_9 - h_8}{\eta_{i,ms}} \quad (15)$$

2.3 Complete system

The performance of the combined cycle (Fig. 3.1) can be expressed using the heat balance in the subcooler,Eq. (2), which provides the relation between the refrigerant mass flow rates in the cycles. Using it,the ratio between the power consumption of the cycles is expressed by Eq. (16), and the overallCOP by Eq. (17).

$$\frac{P_{c.ms}}{P_{c,r}} = \frac{m_{ms} * w_{c,ms}}{m_r * w_{c,r}} = \frac{\Delta h_{sub} * w_{c,ms}}{q_{0,ms} * w_{c,r}} \quad (16)$$

$$COP = \frac{m_r * q_{0,r}}{m_r * w_{c,r} + m_{ms} * w_{c,ms}} = \frac{q_{0,r}}{w_{c,r} + \frac{\Delta h_{sub} * w_{c,ms}}{q_{0,ms}}} \quad (17)$$

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III RESULTS AND DISCUSSION

Variation of COP with environment temperature at three different evaporator temperatures 5, -5 and -30 °C has been shown in Fig. 3. Here the gas-cooler exit temperature is fixed at 5 °C greater than the environment temperature. It has been observed that for evaporator temperatures of 5 and – 5 °C the COP of the N_2O transcritical cycle is greater than that of CO_2transcritical cycle if the environment temperature is more than 26 °C or the gas-cooler exit temperature is more than 31 °C. For evaporator temperature below -30 °C, the COP of N_2O transcritical cycle is more than that of CO_2transcritical cycle even at environment temperature below 25 °C. The reason of this variation is optimum heat rejection pressure, at the environment temperature of 20 °C, both refrigerants have same value of optimum heat rejection pressure, but if environment temperature increases, the optimum heat rejection pressure will increase. The increment in optimum heat rejection pressure with environment temperature is higher in case of CO_2 than N_2O. Hence, the pressure ratio for CO_2 will be higher than N_2O, which increases the compressor work and decreases the COP of CO_2transcritical cycle.

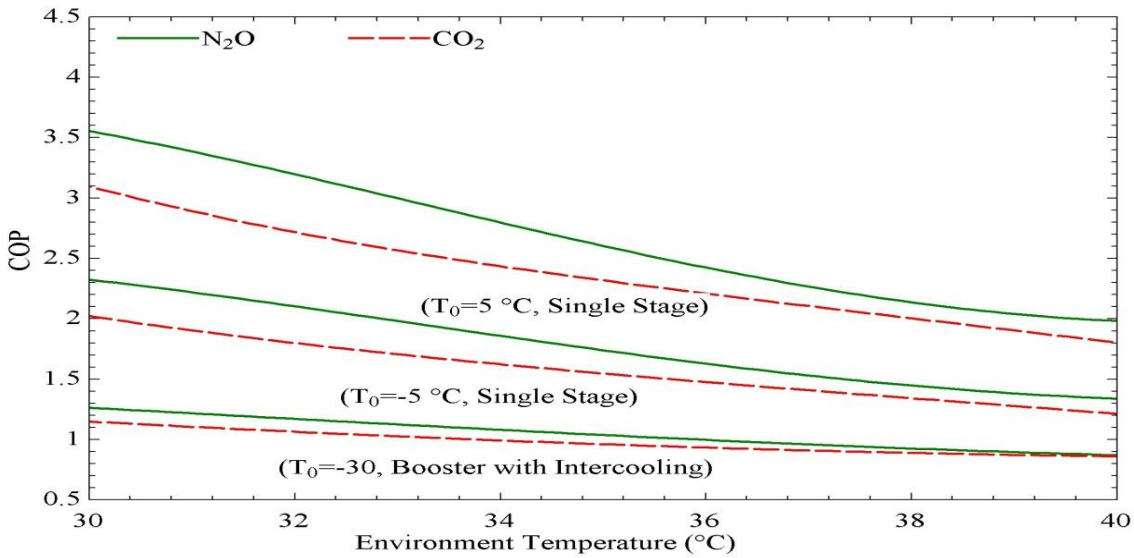

Fig. 3 Variation in COP with environment temperature at different evaporator temperatures.

Variation of COP with environment temperature for different degrees of subcooling and for three different evaporator temperatures 5, -5 and -30 °C has been shown in Fig. 4. It has been observed that the COP of the N_2O cycle decreases as environment temperature increases for all evaporator temperatures, and also the COP of the cycle decreases as evaporator temperature decreases. Using dedicated mechanical subcooling it is clearly seen that for all environment temperatures, the COP of the cycle improves at all evaporator temperatures because of the increment in specific cooling capacity.

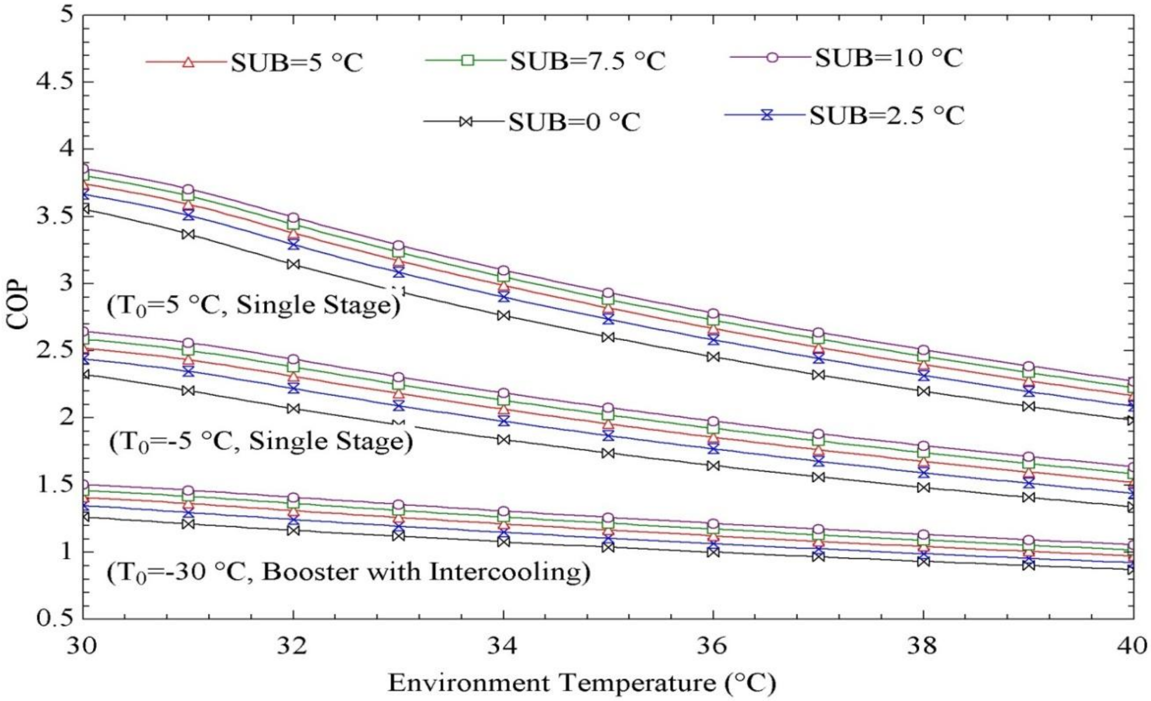

Fig. 4Variation in COP with environment temperature for different degrees of subcooling at different evaporator temperatures

IV CONCLUSIONS

The thermodynamic analysis for the performance improvements of N_2O transcritical refrigeration cycle using dedicated mechanical subcooling cycle has been carried out in this work. range. For this analysis single stage compression system has been used for the evaporator temperatures of 5 °C and -5 °C, and double stage compression system has been used for the evaporator temperature of -30 °C. For this study, thermodynamic models have been developed in Engineering Equation Solver software. Following conclusions have been drawn from the present study:

1. It has been observed that the COP of N_2O transcritical cycle is more than that of CO_2transcritical cycle for the environment temperature above 30 °C and the evaporator temperatures of 5 °C, -5 °C and -30 °C.
2. While using MS cycle the optimum pressure of the transcritical cycle can be reduced. Use of MS cycle allows an increment in the COP and the specific cooling capacity of the system for all operating conditions considered in this work.

3. More benefit in COP is obtained using the MS cycle for environment temperature above 30 ºC for all the evaporator temperatures. But the increments in COP are higher for -5 °C and -30 °C evaporator temperatures.
4. It is observed that the value of power consumption required for MS cycle is less than 15% of the power consumption of main cycle at the optimum conditions, but it increases significantly for pressures below the optimum value.

Finally, based on the conclusion of the results, the use of dedicated mechanical subcooling cycle is an efficient way of improving the performance of N_2O transcritical refrigeration cycle especially for tropical countries.

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