Thermodynamic Analysis of Transcritical CO_2 Refrigeration System Used For Simultaneous Heating and Cooling Application

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Abstract* – CO_2 has low critical pressure and temperature. This gives an opportunity for CO_2 cycles to work in a transcritical nature where heat rejection and absorption are done at supercritical and subcritical conditions, respectively. However, this characteristic poses some performance issues for CO_2 refrigeration cycle such as the pressure and temperature of CO_2 becomes independent of one another above the critical point thus specifying the operating conditions would be tough. It is also important to identify the optimum cooler pressure and control it in order to get high cycle coefficient of performance (COP). Thus, the objective of this paper is to investigate the performance of a transcritical CO_2 compression refrigeration cycle for different parameters and evaluate its COP. To achieve that, a refrigeration cycle was modelled using thermodynamic concepts. Then, the model was simulated for various parameters that were manipulated to investigate the cycle performance. Maintaining other operating parameters constant the highest COP was 5.6 at 100KPa gas Cooler pressure. It was also observed that the cycle is suitable for air-condition application than refrigeration cycle, as COP increases when the evaporator increases. Simulations were conducted using EXCEL developed program. The results can be used in the design of CO_2 refrigeration cycle.

***Keywords—* CO_2; Transcritical cycle; Refrigeration; Optimum pressure; COP; Supercritical**

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INTRODUCTION

CO_2 has low critical pressure and temperature which are 73.6 KPa and 31.1^{o}C, respectively. The low critical temperature causes the heat rejection process to occur above the critical point and heat absorption process to happen below the critical point. Figure 1 represents a p-h phase diagram of CO_2 transcritical refrigeration cycle. Heat is rejected at supercritical pressure and the fluid will exist in the superheated region. If we are able to maintain optimum Gas pressure we can get maximum C.O.P. During heat rejection process, the refrigerant experiences large temperature glide. One of the challenges of this cycle is, due to high pressure level there is a need to control the pressure. One method is to adopt dynamic pressure control [1]. This pressure influences the highest COP value; the cycle can produce [2]. Thus, having the ability to control high-side pressure will provide optimum COP.

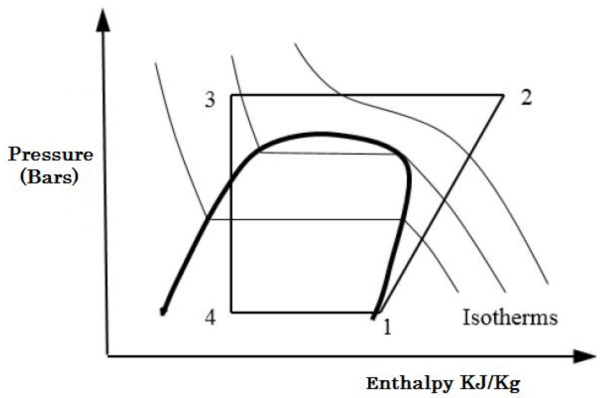

Figure1: P-H DIAGRAM FOR CO_2

However, practically this pressure changes as it is influenced by various operating parameters of the cycle. With this respect, McEnaney [3], investigated CO_2 for mobile air conditioning application and his studies showed that maximum COP was obtained as a function of various operating parameters. Kim et al. [4] reviewed many research works and explained this cycle COP is optimum at a specific operating parameters combination. Sarkar [5] explained that maximum COP occurred at specific gas cooler pressure which in turn is affected by evaporator temperature (T_1), gas cooler exit temperature (T_3), and components' efficiency. Moreover, Perez-Garcia et al. and Xue et al. [6, 7], supports that compressor inlet temperature (T_1) influences the COP and added another variable which is compressor efficiency. Thus, in order to obtain the maximum value of COP, the p_{opt} for the system must be achieved and controlled. Since the pressure is not constant and influenced by other working parameters, the relationship between the parameters and its influence on the system COP must be understood. With this understanding only the parameters that significantly affect the refrigeration cycle of COP could be controlled and CO_2 refrigeration system COP can be improved. Thus, the objective of the paper is to understand operating parameter changes on each and the subsequent devices and on the cycle performance. In order to do that, a system used for simultaneous cooling and heating application is designed & model was developed using thermodynamic concepts. Then, the model was evaluated to calculate C.O.P of the system first by varying one parameter and maintaining other parameters constant and in second case by varying 2 parameters and maintaining remaining parameters constant likewise study is carried out to find the effect of various parameters on C.O.P of system

Nomenclature

W	: Power
Q	: Heat transfer rate
$\dot{m}$	: Mass flow rate
h	: Enthalpy
p	: Pressure
T	: Temperature
C_p	: Specific heat
P_{opt}	: Gas cooler optimum pressure
X	: Quality of refrigerant
η_{ise}	: Isentropic efficiency of compressor

Subscripts

R	: Refrigerant
1	: Exit of evaporator
2	: Exit of compressor
3	: Exit of gas cooler
4	: Exit of throttling valve

METHODOLOGY

Modelling Components of the System

Each process that represent the transcritical CO_2 refrigeration cycle was identified. With some assumptions each component process was modeled thermodynamically.

Evaporator

Inside the evaporator, the refrigerant absorbs heat from the Air Conditioned space and the amount of heat absorbed is calculated as

$$Q_e = \dot{m}_r \times (h_1 - h_4) \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(I)}$$

Whrere, Q_e represents the amount of Heat absorbed in Evaporator

Compressor

Once the refrigerant exits the evaporator, it flows into the compressor where it is compressed to superheated state. Compressor is power consuming device and the amount of power input required to drive compressor is calculated as

$$W_{1\text{-}2a} = \dot{m}_r (h_{2a} - h_1) \quad \ldots\ldots\ldots\ldots\ldots\ldots \text{(II)}$$

Gas Cooler

The fluid flows into the gas cooler where heat rejection is done. Here the refrigerant will experience large temperature glide and exits the gas cooler at slightly higher than the coolant temperature. This high temperature glide can be effectively used for water heating in the gas cooler, the heat rejection process occurs at constant pressure. The heat rejection in this component or Amount of heat gain by water can be quantified by using:

$$Q_{3\text{-}2a} = \dot{m}_r (h_{2a} - h_3) \quad \ldots\ldots\ldots\ldots \text{(III)}$$

The value of h_{2a} is influenced by the value of compressor efficiency. For COP calculation, actual compressor exit enthalpy (h_{2a}) was used. And it was calculated by Eq. (IV).

$$h_{2a} = h_1 - \left\{\frac{h_2 - h_1}{\eta_{ise}}\right\} \ldots\ldots\text{(IV)}$$

Expansion Valve

Then, the refrigerant enters into the throttling device. In this system expansion valve is used as throttling device where it was expanded and experienced isenthalpic process. The enthalpies of the refrigerant both at gas cooler exit and evaporator inlet are equal as represented by Eq. (v).

$h_3 = h_4$ ………… (V)

Enthalpy at point three is a function of both gas cooler pressure and exit temperature. Whereas, the enthalpy at point four is a function of the evaporator pressure and the Quality at the expansion valve exit. If T_3 and cooler exit pressure is known, then the enthalpy is obtained from CO_2 property tables. When x_4 was used as the input parameter, the value was obtained by using Eq. (VI) at the given evaporator pressure

$h_4 = (h_f)\ 4 + x_4\ (h_{fg})_4$ ……..(VI)

Finally the coefficient of performance (C.O.P) of the System was calculated

$$\text{C.O.P} = \left\{ \frac{\text{Useful Heat Gain}}{\text{Compressor Work}} \right\}$$

Here system is used for both heating and cooling so useful heat gain will be addition of heat gain in Evaporator and Heat rejected in Gas Cooler therefore C.O.P can be written as

$$\text{C.O.P} = \left\{ \frac{\text{heat gain in Evap+ Gas Cooler}}{\text{Compressor Work}} \right\}$$

$$= \left\{ \frac{(h2-h3)+(h1-h4)}{(h2-h3)} \right\}$$

Once each process is represented mathematically the C.O.P for system can be calculated using Microsoft Excel. For the study of effect of each parameter and to evaluate the COP of the cycle practical operating parameters were used. Gas cooler pressure and its exit temperature were varied from 80 to 130 KPa and 35 to 50^{o}C, respectively. The evaporator exit temperature was varied from -15^{o}C to 15^{o}C, whereas its pressure was maintained at 40KPa. The compressor efficiency was assumed to be 100%.

Varying the Cycle Parameters

At this stage, various input parameters were manipulated and analysed to understand their influence on the COP. First, only one parameter was varied to see its effect on COP and then two parameters were manipulated to see the influence of their relationship on the cycle COP.

RESULTS AND DISCUSSION

Figure 2 shows the COP versus gas cooler pressure (P_2). For this Calculation, the input parameters that were maintained are P_1= 40 KPa, T_3 = 35^{0}C, η_{ise} =100 % and P_2 was varied from 80 KPa to 130 KPa The graph shows as P_2 increases initially the COP increases, reach maximum and reduces. At this given condition the optimum pressure is 100 KPa and the corresponding highest COP is 5.61. Increasing the pressure, increases the COP initially, however the added capacity is no longer able to compensate compressor additional work thus the COP value decreases. The initial COP shows that as the p value is close to the critical pressure the COP is less and hence to improve COP value, P_2 should not be too close to the P_{cr}

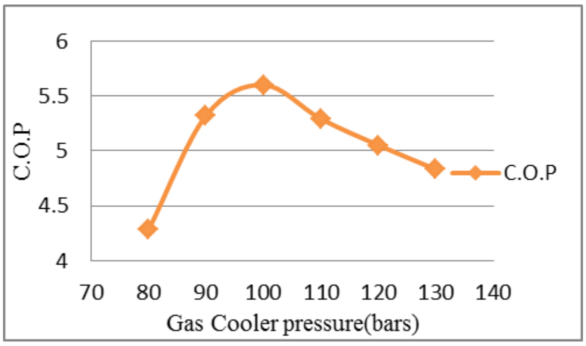

Fig 2: Variation of C.O.P with Gas Cooler pressure

Figure 3 shows the variation of COP, at 40 KPa and 90KPa evaporator and gas cooler pressures, respectively, for different CO_2 gas cooler exit temperatures. Almost linear relationship is observed between the COP and T_3. The highest value of COP was 5.31 at 35^{o}C which is the heat sink temperature. The smaller the refrigerant temperature leaving the gas cooler, the bigger will be the COP, but this is limited by the heat sink temperature. At temperature more than 50^{o}C, the COP value is less than one

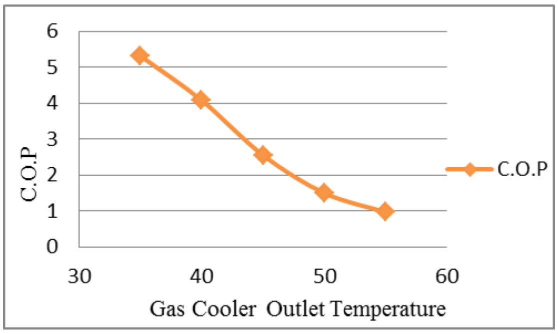

Fig 3: variation of C.O.P with Gas Cooler Outlet Temperature

The effect of the evaporator temperature was investigated while other parameters are maintained constant. Here, the constant parameters were P_2 = 90 KPa, T_3 = 35°C and at η_{isc} = 100%. It can be seen in Figure 4 as T_1 increases the cycle COP value increases. However, the temperature of the evaporator is determined by the space to be cooled. This result also showed that CO_2 refrigeration cycle is suitable for air conditioning purpose than for refrigeration application

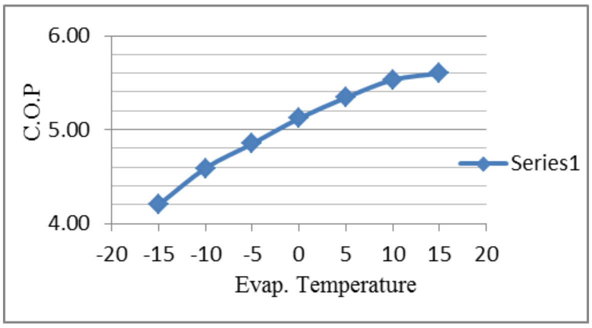

Fig 4: Variation of C.O.P with evaporator temperature

Effect of two parameters change on COP was investigated. Fig.5 shows the COP variation against the gas cooler pressure for different gas cooler exit temperatures. At a given gas cooler pressure the smaller the gas cooler temperature, the higher will be the COP. For the pressure range analysed, maximum COP was observed at 35°C and 40°C gas cooler exit temperatures at unique pressure (Popt). However, for T_3 45°C and above, the COP initially increases and then becomes flat. Here it can be deduced that T_3 has a significant effect on the p_{opt}. Moreover, it would take higher pressure for the system to achieve the highest COP as the gas cooler exit temperature increases.at 35°C, the value suddenly increased to 5.61. This was due to the effect of enthalpy value at h_4. At higher T_3, the value was bigger compared to enthalpy at 35°C, thus enthalpy difference at refrigerating capacity was smaller (even negative) as the temperature increases. Thus appropriate gas cooler pressure must be used for a given gas cooler exit temperature.

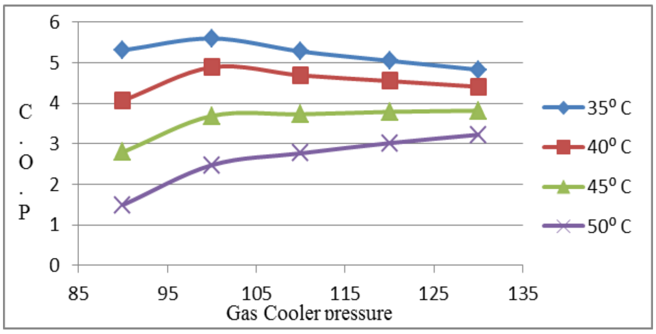

Fig 5: COP variation against the gas cooler pressure for different gas cooler exit temperatures

In Fig. 6 the COP value is plotted against gas cooler pressure (P_2) for different evaporator temperatures maintaining other parameters constant. At a given gas cooler pressure as the evaporator temperature increases the COP increases. Apart from that, by varying P_2, maximum COP is observed at p_{opt} and this is more distinct at higher T_1 especially at 0°C and above. This figure also shows that maximum COP occurred almost at the same optimum pressure. Hence the effect of evaporator temperature on the optimum gas cooler pressure for maximum COP is not significant as compared to the gas cooler exit temperature which is shown in Fig. 6.

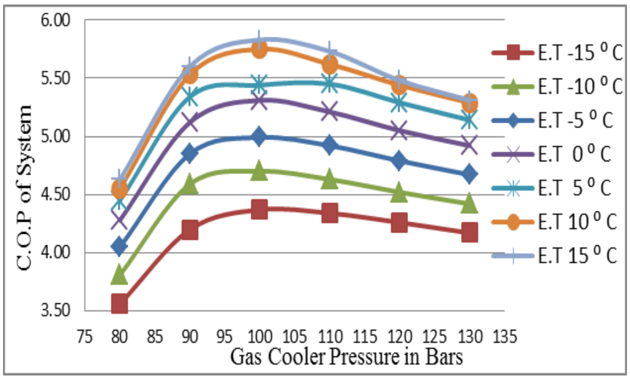

Fig 6 : Variation of C.O.P with Evaporator temperature and Gas Cooler pressure

CONCLUSION

Model of CO_2 transcritical refrigeration cycle was developed thermodynamically. The model was used to investigate the effect of the various parameters on the System COP and to identify the combined effect for optimum COP. The following were drawn from the investigation. Transcritical CO_2 refrigeration cycle has specific gas cooler pressure (P_{opt}) that gives maximum COP. This pressure is not constant and varies when the rest of the cycle operating parameters change. Moreover, gas cooler exit temperature and evaporator temperature have significant effect on the cycle pressure that gives maximum COP.

In general, higher evaporator and smaller gas cooler exit temperatures would give better cycle COP. The best combinations of these parameters can be obtained by analysing the cycle for the given parameters. It was also observed that the cycle is suitable for air-condition application than refrigeration cycle, as COP increases when the evaporator temperature increases. Based on these outcomes, it is hoped that a better understanding of controlling CO_2 transcritical refrigeration cycle COP can be achieved. Apart from that, with the identification of the parameters that affect the COP significantly, it is hoped that future design of CO_2 refrigeration cycle can be improved.

ACKNOWLEDGEMENT

The authors wish to thank Shree Refrigeration Pvt.Ltd. Ogalewadi for their constant support and Prof. A.R Acharya (Dean Industry Institution interaction).

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