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Thermodynamic analysis of two-stage transcritical Ethane refrigeration cycle with expander

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Abstract - *In this paper, thermodynamic performance comparison of the following refrigeration cycles: Two Stage with Expander (TSE) and Two Stage with an Expansion Valve (TSEV) using Ethane has been carried out. The effect of operating parameters such as gas cooler exit and evaporator temperatures on optimum values of gas cooler and intermediate pressure and on the coefficient of performance (COP) has been investigated and comparison is done between both cycles. The TSE cycle had higher COP over the TSEV cycle for all studied conditions. Replacing the expansion valve with an expander resulted in lower gas cooler and intermediate pressure for all studied conditions. The temperature of refrigerant at the exit of high-pressure stage compressor is almost similar for both the cycles. The analysis is performed while simultaneously optimising the gas cooler and intermediate pressures. Temperature of evaporator and gas cooler exit is differed between 0 ℃ to -30 ℃ and 35 - 50 ℃, respectively. An optimum gas cooler and intermediate pressure can be obtained to maximise the performance of cycle. The TSE cycle yields average 31.73% and highest 38.64% COP improvement over the TSEV cycle. The gas cooler exit temperature and efficiency of compressor majorly determine the performance of the cycle. A performance improving device should minimize the gas cooler outlet temperature and reduce cycle's operating pressures which would also help ensure safer operation. The results of this analysis are aimed to provide basis for system design and optimization of Ethane based refrigeration system.*

Key Words: Two-stage, transcritical, refrigeration, expander, ethane,

Nomenclature

COP	coefficient of performance
h	specific enthalpy (kJ kg^{-1})
HP	high pressure
LP	low pressure
P	pressure (MPa)
q	specific heat transfer (kJ kg^{-1})
t, T	temperature (°C)
w	specific work (kJ kg^{-1})
η	efficiency

Subscripts

1-6	refrigerant state points
2s, 4s	isentropic compression state
c	compressor
e	evaporator
gc	gas cooler
im	intermediate
is	isentropic
ex	expander

1.INTRODUCTION

The demand for alternate refrigerants to reduce energy consumption has become a worldwide concern. One way of reducing energy consumption is by enhancing the performance of refrigeration cycle. Refrigeration systems and use of synthetic refrigerants contribute to worldwide global warming. Alternatively, natural refrigerants such as ethane can be used which has very low Global Warming Potential (GWP) and zero Ozone Depletion Potential (ODP). Ethane (C_2H_6) has similar properties as CO_2 [1]. The critical temperature of Ethane is 32.2°C which is very close to critical temperature of carbon dioxide i.e. 31.1°C. Besagni et al. [2] stated that an appropriate working fluid for refrigeration system is one which provides high performance in all working environment. To achieve optimal performance, careful consideration is given to the refrigerant thermo-physical properties. Calm et al. [3] studied vapor compression-based refrigeration systems to be used with next generation of refrigerants. Nasruddin et al. [4] revealed that a temperature of -80°C can be achieved in ethane-based refrigeration cycles. Agrawal et al [5] studied CO_2 and N_2O based two-stage transcritical cycle with intercooling and revealed staging of compression resulted in performance enhancement. Baek J et al. [6] suggested equipping an expander to produce work to drive the high stage compressor to maximise performance of two-stage cycle with intercooling. Yang et al. [7] concluded a maximum improvement of 33% in COP of expander based single stage transcritical cycle compared to expansion valve-based cycle. Also, Yang et. al [8] concluded that optimum intermediate pressure of the two-stage transcritical refrigeration cycle with expander varied from the geometric mean of gas cooler and evaporator pressure. Cecchinato et al. [9] compared various two-stage CO_2 cycles, their analysis employed expansion valves. This study explores the possibility of enhancing performance in such systems by replacing the expansion valve with an expander. Arash Nemati et al. [10] studied ethane as refrigerants along with CO_2 and N_2O in two stage ejector-expansion based transcritical refrigeration cycle and concluded that compressor pressures and cycle temperature level are much lower compared to CO_2 and N_2O. This ensures higher safety and longer life cycles. Zhang et al. [11] studied and compared four different double-compression transcritical refrigeration cycle based on CO_2 and concluded that cycle with external intercooler with an expander yielded 29.2% COP improvement over the base cycle employing an expansion valve. Kasi et al. [12] studied cascade refrigeration system using ethane and achieved very low temperature range of -70°C to -50°C. Nehdi et al. [13] compared different refrigerants and concluded that R141b yielded 22% COP improvement among all others. Ersoy and Sag [14] found that an R134a EERS system can achieve a coefficient of performance (COP) that is 6.2-14.5% higher than a conventional system, depending on the specific operating conditions. A comprehensive literature review, including the studies mentioned, revealed a gap in research regarding the thermodynamic comparison of ethane-based expander and expansion valve cycles. To overcome this research gap, in the present study performance of ethane based two stage transcritical refrigeration cycle with an expander has been compared to a cycle having an expansion valve. Engineering Equation Solver [15] has been used for energy analysis.

Table -1: Properties of Ethane refrigerant

Properties	Ethane
ODP	0
GWP	8.4
Toxicity (ppm)	3000
Boiling temperature (°C)	-89
Critical temperature (°C)	32.2
Critical pressure (bar)	48.72

2. SYSTEM MODEL AND ANALYSIS

The schematics for the two-stage transcritical cycle with expander (TSE) and the corresponding pressure-enthalpy diagram are presented in Figures 1 and 2, respectively. Similarly, Figures 3 and 4 illustrate the two-stage transcritical cycle with expansion valve (TSEV) and its pressure-enthalpy diagram.

2.1 Two stage cycle with Expander (TSE):

The cycle begins with saturated vapor at state 1, corresponding to the evaporator pressure (P_e). This vapor is compressed in the low-pressure compressor to state 2 at an intermediate pressure (P_{im}). Intercooling reduces the temperature of the high-pressure vapor to state 3. In the subsequent stage, the superheated vapor undergoes further compression in the high-pressure compressor, reaching state 4. The gas cooler then cools the vapor to state 5. Finally, expansion in the expander (state 6) reduces the pressure back to the evaporator pressure, recovering work that contributes to driving the high-pressure compressor.

2.2 Two stage cycle with Expansion valve (TSEV):

The processes from state 1 to state 5 in the TSEV cycle remain identical to those in the TSE cycle. However, the difference exists from state 5 to state 6. Unlike the TSE cycle's expander, the TSEV cycle utilizes an expansion valve to achieve pressure reduction from state 5 to state 6, bringing the vapor back to the evaporator pressure.

Both cycles were investigated under following assumptions:

- Cycle operates at steady state.
- At evaporator outlet, vapor is dry saturated.
- Processes in intercooler, gas cooler and evaporator are isobaric.
- At the intercooler outlet, vapor is superheated.

The values for the isentropic efficiencies of the low-pressure and high-pressure compressors are determined through:

$$\eta_{is,c,LP} = \frac{(h_{2s} - h_1)}{(h_2 - h_1)} \; and \; \eta_{is,c,HP} = \frac{(h_{4s} - h_3)}{(h_4 - h_3)} \quad (1)$$

where the efficiency for both low-pressure and high-pressure compressor are assumed as 80%.

Expander efficiency is assumed as:

$$\eta_{ex} = 0.5 \quad (2)$$

Evaporator cooling capacity can be observed as:

$$q_e = h_1 - h_6 \quad (3)$$

The work done in both Low-pressure and High-pressure compressor is given as:

$$w_{c1} = h_2 - h_1 \quad (4)$$

$$w_{c2} = h_4 - h_3 \quad (5)$$

The work done in expander:

$$w_{ex} = h_5 - h_6 \quad (6)$$

The COP of TSE cycle can be defined as:

$$COP = \frac{q_e}{w_{c1} + w_{c2} - w_{ex}} \tag{7}$$

Similarly, COP of TSEV cycle can be defined as:

$$COP = \frac{q_e}{w_{c1} + w_{c2}} \tag{8}$$

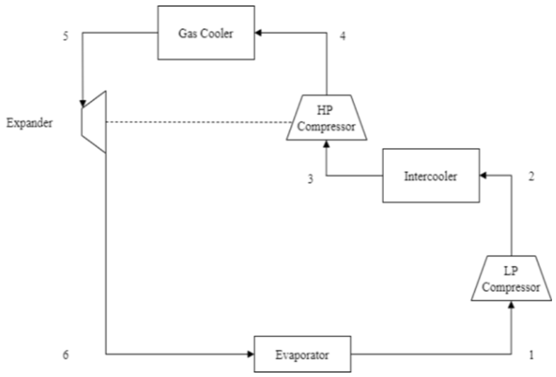

Figure -1: Schematic diagram of two-stage transcritical cycle with expander

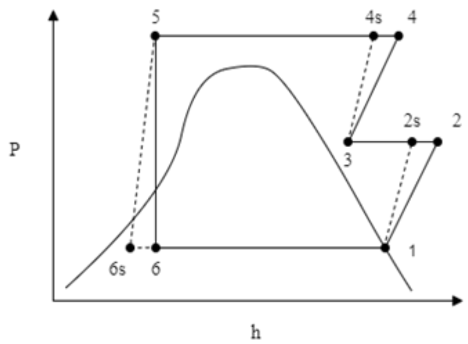

Figure -2: P-h diagram of two-stage transcritical cycle with expander

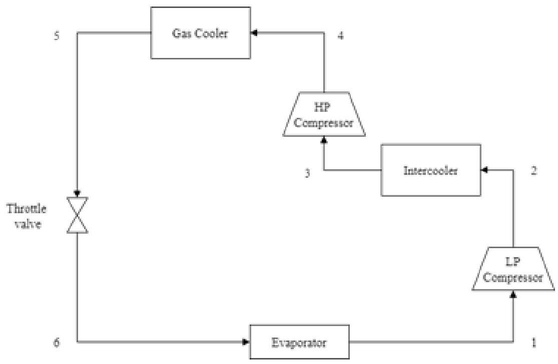

Figure -3: Schematic diagram of two-stage transcritical cycle with expansion valve

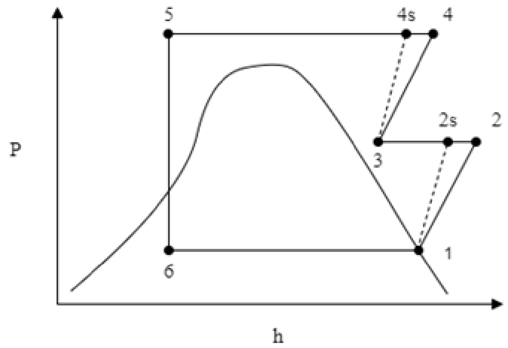

Figure -4: P-h diagram of two-stage transcritical cycle with expansion valve

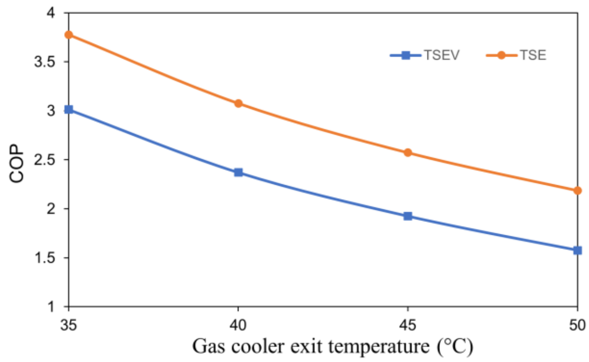

Figure -5: Effect of gas cooler exit temperature on COP

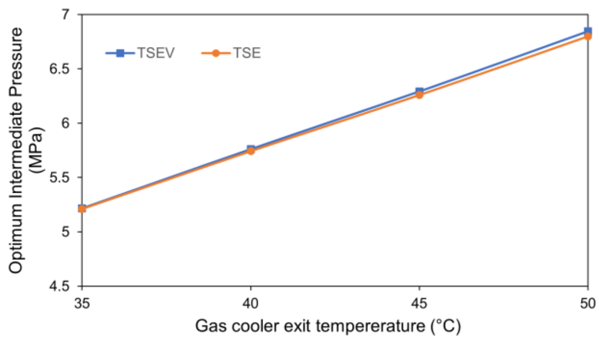

Figure -6: Optimum intermediate pressure at different gas cooler exit temperatures

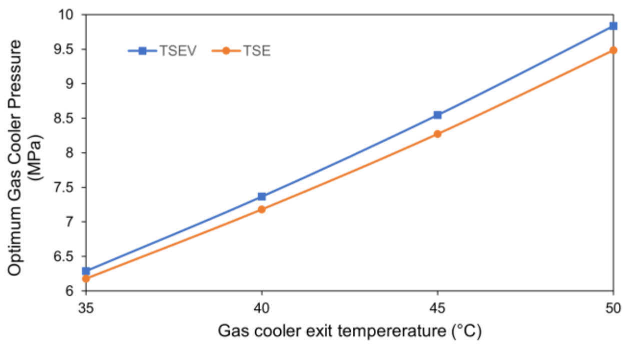

Figure -7: Optimum gas cooler pressure at different gas cooler exit temperatures

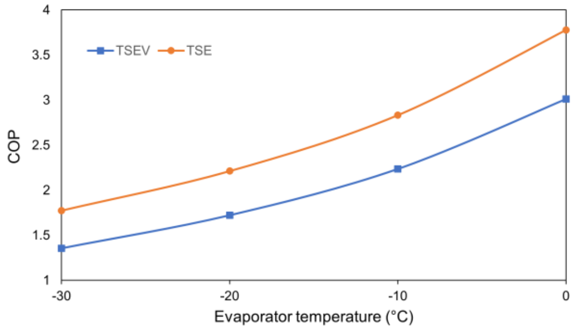

Figure -8: Effect of evaporator temperature on COP

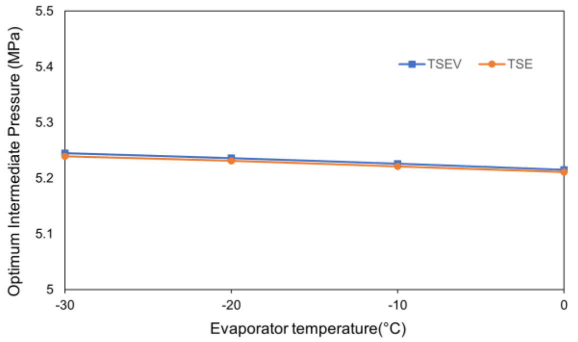

Figure -9: Optimum intermediate pressure at different evaporator temperature

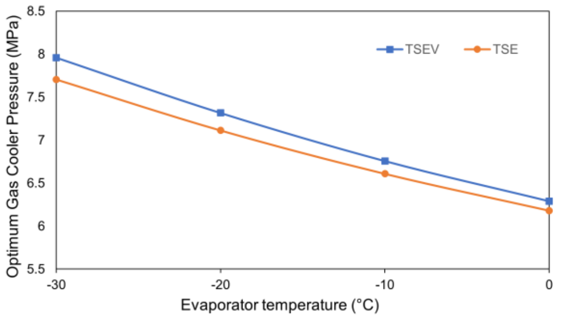

Figure -10: Optimum gas cooler pressure at different evaporator temperature

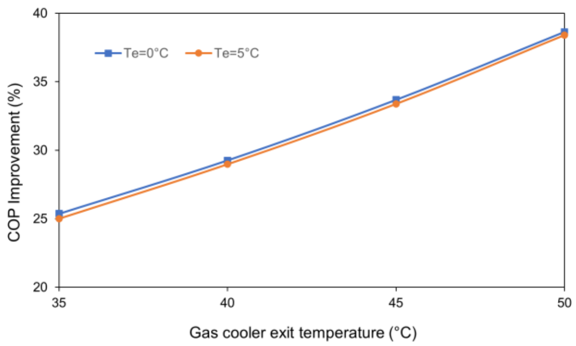

Figure -11: Effect of gas cooler exit temperature on COP improvement

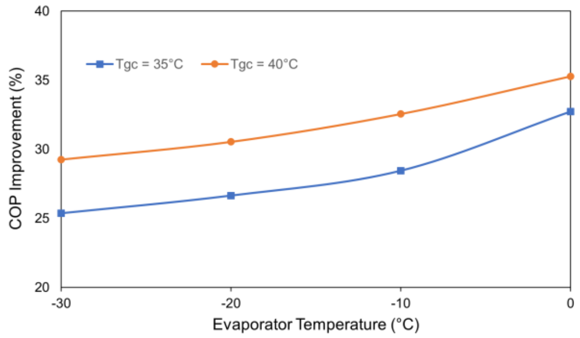

Figure -12: Effect of evaporator temperature on COP improvement

3. RESULTS AND DISCUSSION

Both the low-pressure and high-pressure compressor efficiency have been considered as 80% for the analysis. The gas cooler exit temperature and intercooler temperature are taken as equal and has been assumed as 35°C and evaporator temperature has been assumed to be 0°C unless mentioned otherwise.

The COP of the TSEV and TSE cycle, where the intercooler pressure and gas cooler pressure are simultaneously varied, is displayed in Figure 5. The intermediate pressure is varied between 2 - 8 MPa, whereas the gas cooler pressure is varied between 6 - 12 MPa. The gas cooler exit temperature varies between 35 - 50°C, the evaporator temperature is assumed to be 0°C. With reference to Figure 5, it is evident that both cycles exhibit similar behaviour, with the COP initially reaching its maximum and thereafter decreasing as the temperature of the gas cooler outlet increases. The graph shows that the COP of cycle having an expander (TSE) was greater compared to cycle having expansion valve.

The relation between the optimum intermediate pressure and changes in the gas cooler exit temperature for the TSE and TSEV cycles, when the gas cooler pressure is optimized, is depicted in Figure 6. The range of the intermediate pressure is observed between 5 - 7 MPa. The gas cooler exit temperature varies between 35 - 50°C, the evaporator temperature is assumed to be 0°C. With reference to Figure 6, it is evident that both cycles act in a manner consistent with an increase in the gas cooler outlet temperature followed by rise in the optimum intermediate pressures. The graph indicates that the optimum intermediate pressure values for a cycle with an expansion valve (TSEV) is almost similar but slightly higher than that of expander (TSE).

The variation of the optimum gas cooler pressure for the TSE and TSEV cycle, where the intermediate pressure is maximized, with changes in gas cooler exit temperature is depicted in Figure 7. The gas cooler pressure range is observed to be between 6 - 10 MPa. The gas cooler exit temperature ranges between 35 - 50°C and the evaporator temperature is assumed to be 0°C. With reference to Figure

7, it is evident that both cycles exhibit a similar behaviour, whereby rise in the gas cooler outlet temperature corresponds with an increase in the optimum pressure of gas cooler. The graph indicates that the optimum gas cooler pressure levels found for two-stage transcritical cycle with expansion valve is higher as compared to cycle with expander (TSE). Cycle's operating pressures drops when expander is used in place of expansion valve.

When the gas cooler and intermediate pressures simultaneously varied, Figure 8 displays a variation of COP with evaporator temperature for TSE and TSEV. The gas cooler pressure ranges between 6 - 12 MPa, while the intermediate pressure varies between 2 - 8 MPa. The temperature of the evaporator varies between -30°C to 0°C, the gas cooler exit temperature is presumed as 35°C. With reference to Figure 8, it is evident that both cycles exhibit a similar behaviour, whereby the COP increases in synchrony with the evaporator temperature. The graph indicates that the COP values for the two-stage transcritical cycle with expander is higher than two-stage transcritical cycle with expansion valve.

The variation of the optimum intermediate pressure with the temperature in evaporator for both TSE and TSEV cycle, when the gas cooler pressure is varied, is depicted in Figure 9. The optimum intermediate pressure is observed at 5.25 MPa when the evaporator temperature ranges between -30°C to 0°C, the gas cooler exit, and intercooler temperature is assumed to be 35°C. With reference to Figure 9, it is evident that both cycles exhibit a similar behaviour whereby the optimum intermediate pressure reduces as the evaporator temperature increases. The graph indicates that, in comparison to two-stage transcritical cycle with expander, the values of optimum intermediate pressure achieved for cycle having expansion valve are slightly higher.

The Figure 10 illustrates the relationship between optimal gas cooler pressure and evaporator temperature for TSE and TSEV cycles, considering the concurrent optimization of intercooler pressure. The optimum gas cooler pressure for both cycles is observed between 6 - 8 MPa. The temperature of evaporator varies between -30°C to 0°C, the gas cooler exit and intercooler temperature is considered as 35°C. With reference to Figure 10, it is evident that both cycles exhibit a similar behaviour, whereby the optimum gas cooler pressure decreases as the evaporator temperature increases. The graph indicates that, in comparison to two-stage transcritical cycles with expander, the values of optimum gas cooler pressure obtained for the cycle with expansion valve are higher.

Figure 11 illustrates the percentage increase in cycle COP that occurs when an expander is used in place of an expansion valve, with the temperature variation of the gas cooler exit and for the fixed evaporator temperature. the intermediate and gas cooler pressures are concurrently optimised. The intercooler pressure varies from 2 - 8 MPa, while the gas cooler pressure fluctuates between 6 - 12 MPa. The temperature at gas cooler outlet varies between 35 - 50°C, the evaporator temperatures at which comparison is done are 0°C and 5°C. With reference to Figure 11, it is evident that the percentage improvement in COP increases with increase in gas cooler exit temperature for both cycles. It can also be concluded from the graph that as the evaporator temperature decreases the percentage improvement in COP increases.

The percentage increase in cycle COP that results from replacing the expander with an expansion valve and varying the evaporator temperature for fixed gas cooler temperatures is displayed in Figure 12. In this also, the gas cooler and intermediate pressures are concurrently optimised. The intermediate pressure varies from 2 - 8 MPa, while the gas cooler pressure is varied between 6 - 12 MPa. The evaporator temperature ranges between -30°C to 0°C, the gas cooler temperature is assumed to be 35°C and 40°C. With reference to Figure 12, it can be observed that in both cycles, the percentage increase in COP rises with an increase in evaporator temperature. Additionally, the analysis suggests that as the gas cooler outlet temperature rises, the percentage improvement in COP increases as well.

4. CONCLUSIONS

In this study, the energy analysis based on first law of thermodynamics of the cycles: Two Stage with Expander (TSE) and Two Stage with Expansion Valve (TSEV) has been studied and compared. These cycles are based on refrigerant Ethane (R170). Following conclusions are made through thermodynamic modelling and analysis through EES software:

- The COP of TSEV significantly improved when expansion valve is replaced by expander as observed in TSE cycle.
- As gas cooler exit and intercooler temperature rises, COP of both TSE and TSEV cycles.
- For both TSE and TSEV cycles, as the evaporator temperature increases, the COP of both cycles increases.
- Through simultaneous optimisation of gas cooler and intercooler pressures, an optimum pressure is determined which maximises COP.
- The optimum intermediate and gas cooler pressure increases with increase in gas cooler exit and intercooler temperatures.
- With increase in evaporator temperature, optimum intermediate and gas cooler pressure decreases.
- The percentage improvement in COP of TSE cycle compared with TSEV cycle increases with rise in gas cooler exit and evaporator temperatures.

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