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Thermodynamical Analysis of a Two Stage Mixing Chamber R-717 Refrigeration Cycle

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In the process of abandoning the refrigerants which cause the greenhouse effect, R-717 has regained importance. However, energy efficiency of these refrigerant systems should be enhanced. To generalize more competitive and environmentalist systems, thermodynamic analysis of a two stage ammonia refrigeration system was performed. By using ammonia (R-717) refrigerant, energy and exergy analysis of single stage and two stage refrigeration systems were compared. In analyses EES program was used. To enhance efficiency in two stage compressions, two stage expansion and bubble through was used. Analyses were carried out with different step pressures and varied system temperatures. When the two systems were compared with each other, it was confirmed that refrigeration efficiency coefficient of two stage system is higher than that of the single stage system.

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PACS/topics: R-717, refrigeration, mixing chamber

1. Introduction

Protocols which are targeting to decrease emissions and which became the agenda of the refrigerating sector have become the cause for new systems design and for discussions about the refrigerant selection. The most known of these protocols are Montreal and Kyoto protocols. In this process, it was planned to put an end to usage of the refrigerants which are harmful for ozone layer. This plan was implemented. At present, regulations have been published which bring limitations of usage of ozone friendly refrigerants which have high global warming potential (GWP). European Union has brought limitations to the usage of refrigerants which have GWP value bigger than 150 (like R-134a), and was planning to gradually stop the usage of these refrigerants, according to Dang, et al. [1].

Among the alternative refrigerants, the ammonia was preferred because of its high evaporation temperature, quite low steam specific volume and the advantage of low refrigerant amount which has to be circulated through the system. Although ammonia's toxicity, substantial inflammability and explosiveness looks like disadvantages, it is used in various applications because of its perfect thermal properties. High temperatures are reached as a result of compression at single stage refrigeration processes. For this reason, abrasions can occur at compressor's components. For the purpose to prevent the temperature rise at the end of the compression process and to eliminate disadvantages of temperature rise, staged compression method was used. When the two stage compression was compared to the one stage, some advantages were found, such as compression by consuming less

work, lesser operating costs and bigger coefficient of performance (COP). The only disadvantage is the higher installation cost.

There are quite many reports considering these systems. Rasi [2], Czaplinsky [3], Ouadha, et al. [4], Zubair, et al. [5], Domanski [6] and Arora and Dhar [7] have studied these systems. Research for two stage compression cycles was generally performed for CO₂ applications in HVAC systems by Cavallini, et al. [8], Celik [9], and for hydro fluorocarbons applications in commercial refrigeration by Torrella, et al. [10, 11], Llopis, et al. [12, 13], Ozgur and Tosun [14].

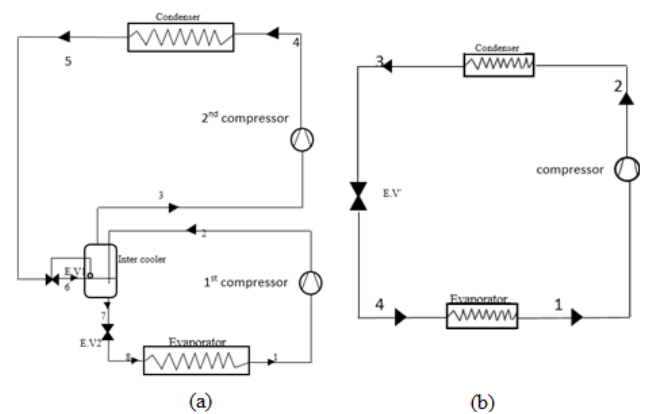


Fig. 1. Two stage (a) and single stage (b) refrigeration cycles.

2. Theoretical analysis

In two stage compression systems with a liquid sub-cooler there is a disadvantage of the cycles that the inlet gas temperature to the second compression stage may be relatively high. Especially with ammonia as refrigerant this is sometimes a problem, as was found by Granryd, et al. [15]. To solve this problem, there is an inter-cooling arrangement called "bubble through". The components of

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two stage bubble through refrigeration cycle and single stage system using R-717 are shown in Fig. 1.

In order to calculate the parameters of the refrigeration cycles some assumptions were made. These assumptions are listed below.

- For two stage system; $P_3 = P_2$, $P_4 = P_5$, $P_6 = P_{INT} = P_7 = P_2$ and $s_1 = s_{2s}$, $s_3 = s_{4s}$ and $h_5 = h_6$, $h_7 = h_8$.
- For two stage system; $P_{INT} = 600$ kPa, $T_E = 35$ °C, $T_c = 50$ °C, $Q_E = 100$ kW.
- Pressure loss in cycles was neglected.
- Heat transfers from or to the systems and components were neglected.
- Superheating value of evaporator outlet was 4 °C in the cycles.

- Efficiency value for the cycles was accepted as 0.75.

In these assumptions, P is pressure (Nm^{-2}), s is entropy (JK^{-1}), h is enthalpy (kJkg^{-1}), Q is heat energy (J or kW), T is temperature (K or $^{\circ}\text{C}$) respectively.

Using isentropic efficiency values, enthalpy values of refrigerant at compressors' outlet obtained from the expressions are given below as Eqs. (1) and (2):

$$\eta_{1k} = (h_{2s} - h_1)/(h_2 - h_1), \quad (1)$$

$$\eta_{2k} = (h_{4s} - h_3)/(h_4 - h_3). \quad (2)$$

Balance equations of system components.

TABLE I

Component	Energy balance	Exergy balance	Mass balance
1st compressor	$W_1 = m_1(h_2h_1)$	$W_1 + m_1e_1 = m_1e_2 + I_{K1}$	$m_2 = m_1$
Inter cooler	$m_3h_3 + m_1h_7 = m_3h_6 + m_1h_2$	$m_3e_3 + m_1e_7 = m_3e_6 + m_1e_2 + I_{AS}$	$m_3 = m_6; m_1 = m_2 = m_7$
2nd compressor	$W_2 = m_3(h_4h_3)$	$W_2 + m_3e_3 = m_3e_4 + I_{K2}$	$m_3 = m_4$
Expansion valve 1	$h_5 = h_6$	$m_6e_6 + I_{KV1} = m_5e_5$	$m_6 = m_5$
Expansion valve 2	$h_7 = h_8$	$m_8e_8 + I_{KV2} = m_7e_7$	$m_7 = m_8$
Condenser	$Q_C = m_3(h_4h_5)$	$Q_C + m_3e_5 = m_3e_4 + I_C$	$m_4 = m_5$
Evaporator	$Q_E = m_1(h_1h_8)$	$Q_E + m_1e_8 = m_1e_1 + I_E$	$m_1 = m_8$

Using the balance equations given in Table I, compressor work, heat rejected from condenser, and the coefficient of performance for cooling (COP_c) we calculated. COP_c for the cycles were obtained from Eq. (3):

$$\text{COP}_c = \frac{Q_E}{W_1 + W_2}. \quad (3)$$

For the calculation of two stage ‘‘bubble through’’ refrigerating cycle by using R-717 (ammonia), EES software was used [16]. In our notation η represents efficiency, 2 represents second (law), ‘‘c’’ stands for cooling, ‘‘C’’ stands for condenser, ‘‘E’’ for evaporator, ‘‘h’’ for heating and ‘‘INT’’ stands for intercooler in the subscripts.

3. Results

Obtained results are presented graphically in Figs. 2–6. If the results are examined it is seen that η and COP_c values of two stage cycles are better than those of single stage cycles.

In Fig. 2, η_2 decreases more rapidly with increasing condensing temperature than that of the two stage cycle. The highest η_2 was obtained between 30 and 40 °C for both systems. Figure 3 shows COP_c variation with the condenser temperature. COP_c value decreases with increasing condensing temperature. Figure 4 shows COP_c variations with respect to evaporator temperature. COP_c value increases with increasing evaporation temperature. It is clearly seen that two stage compression cycle is more advantageous among the considered two cycles.

Figure 5 shows η_2 as function of evaporator temperature. η_2 increases with increasing evaporation temper-

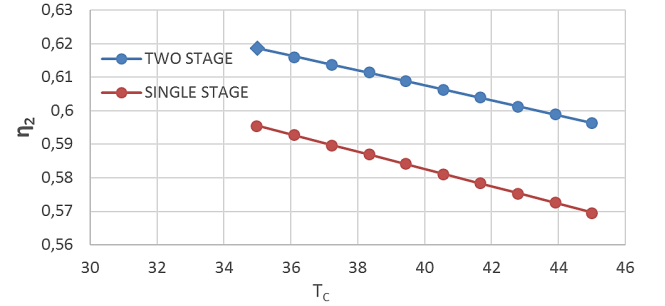


Fig. 2. Efficiency as function of condenser temperature.

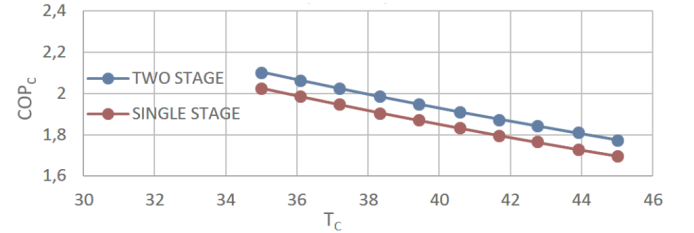


Fig. 3. COP_c variation with the condenser temperature.

ature. Figure 6 shows COP_c variations with the inter-stage pressure at various evaporation temperatures, while condensing temperature is kept at 50 °C for two stages cycle. For finding the change of COP_c , the inter-stage pressure was varied in the range of 400–1600 kPa. It is seen that there is an optimum value range between 600–800 kPa. This value could be important for these kinds of systems which have large refrigeration capacities. These

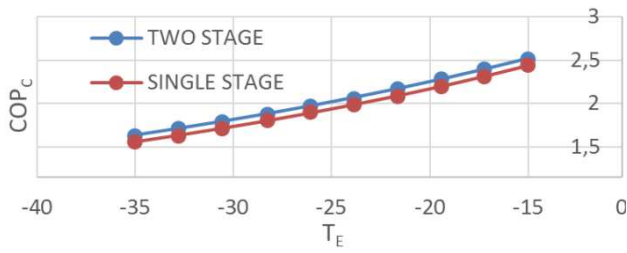


Fig. 4. COP variation with the evaporator temperature.

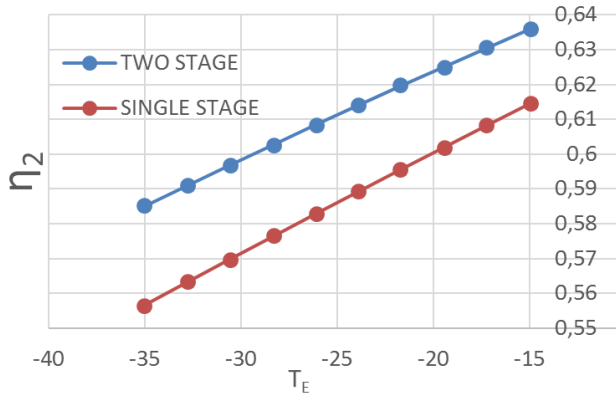


Fig. 5. Efficiency as function of evaporator temperature.

values could be considered for the selection of inter-stage pressure.

4. Conclusions

A comparative study was presented for two different cycles of ammonia. It was found that with the increasing evaporation temperatures COPc values are also increasing, however exergy efficiency values were decreasing with the condensing temperature. For lower condensing temperatures exergy efficiency reaches higher values for both cycles. Furthermore, there was an optimum value range. It was clearly seen that, “bubble through” two stage compression cycle is advantageous compared to single stage cycle. Such a comparison of energetic and exergetic performance for two stage “bubble through” R-717 refrigeration cycle gives valuable and practical knowledge for designers from the refrigeration sector.

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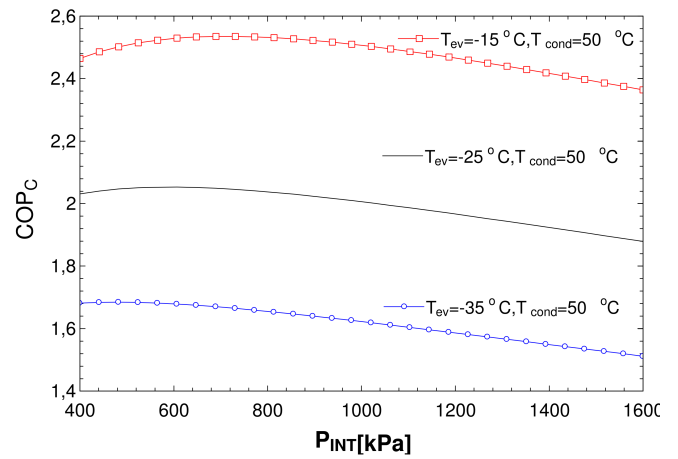


Fig. 6. COPc variation as function of inter-stage pressure for two stage cycle.

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