The optimum intermediate pressure of two-stages vapor compression refrigeration cycle for Air-Conditioning unit

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Abstract. Vapor compression cycle is mainly employed as a refrigeration cycle in the Air-Conditioning (AC) unit. In order to save energy, the Coefficient of Performance (COP) of the need to be improved. One of the potential solutions is to modify the system into multi-stages vapor compression cycle. The suitable intermediate pressure between the high and low pressures is one of the design issues. The present work deals with the investigation of an optimum intermediate pressure of two-stages vapor compression refrigeration cycle. Typical vapor compression cycle that is used in AC unit is taken into consideration. The used refrigerants are R134a. The governing equations have been developed for the systems. An inhouse program has been developed to solve the problem. COP, mass flow rate of the refrigerant and compressor power as a function of intermediate pressure are plotted. It was shown that there exists an optimum intermediate pressure for maximum COP. For refrigerant R134a, the proposed correlations need to be revised.

1. Introduction
To avoid the globe from global warming, many countries have released the target on reducing Greenhouse gas (GHG) emissions. There are two strategies that can be used to mitigate GHG emissions. They are deforestation and sustainable energy development. In the sustainable energy development consists of enhancing renewable energy utilization and energy efficiency technology. Thus, increasing the energy efficient technology is extremely needed in the mitigation actions to reduce GHG emission. One of the potential application of energy efficient technology is vapor compression refrigeration cycle. As a note, vapor compression refrigeration cycle typically employed to provide cooling for air-conditioning unit. Considering the growth of economy and living standard, it is estimated that vapor compression cycle will consume huge amount of electricity. There are several works related to increasing energy efficiency of a vapor compression cycle such as using heat recovery technique [1,2], multi-stages cycle, etc.

The present paper focus in the multi-stages strategy. Bilge and Temir [3] reported the study on the optimum numbers of stages in the vapor compression refrigeration system. The cases of single-stage and multi-stages compression have been compared on a system with 600 kW cooling capacity and the working fluid is ammonia. Goodarzi et al. [4] studied the comparative analysis of the improved two-stage multi-inter-cooling ejector-expansion trans-critical CO₂ refrigeration cycle. In the study, an
internal heat exchanger has been included within the system for improvement in the cooling capacity. Purohit et al. [5] reported the study on the effect of inter-stage pressure on the performance of a two stages refrigeration cycle using intercooler. Six common refrigerants are selected for the analyses include R22, R143a, propane, carbon dioxide and nitrous oxide. It was shown that the effect of inter stage pressure is expected to be an important consideration in warmer climate condition. Jian et al [6] investigated the role of optimum intermediate pressure in the design of two-stage vapor compression systems as a further investigation. Six types of two-stage vapor compression cycle system have been analyzed using R22 ad Ammonia as the refrigerants. Chopra et al. [7] performed energy, exergy and sustainability analysis of two-stage vapour compression refrigeration system. Eight types of refrigerants have been analyzed on the basis of energetic and exergetic performance. Xuan [8] developed a general staging model to optimize multistage exo-reversible refrigeration system affected solely by internal irreversibilities.

Those studies show that multi-stage technology to optimize the vapor compression refrigeration cycle has come under scrutiny. The present work focuses on the investigation of optimum intermediate pressure of two-stage vapor compression refrigeration cycle. The used refrigerant here is R134a which is a common type of refrigerant in the air-conditioning application. The literature shows that very limited studies reported the using of R134a in the optimization of multi-stage vapor compression refrigeration cycle. The objective is to explore the maximum coefficient of performance (COP) of the system at several different working pressures of evaporator and condenser. The results are expected to supply the necessary information on development highly efficient vapor compression refrigeration cycle.

2. Solution Method

The schematic diagram of two-stage vapor compression refrigeration cycle is shown in figure 1. In the figure, the p-h diagram is also presented. It can be seen that the component of the system consists of two compressors, condenser, flash cooler, and evaporator.

![Schematic diagram of two-stage vapor compression refrigeration cycle](image)

**Figure 1.** Schematic and P-h diagrams of the system

The governing equations of the systems are developed as follows. The power in the first ($\dot{W}_{c1}$) and second compressors ($\dot{W}_{c2}$) are calculated by

$$\dot{W}_{c1} = m_1 (h_2 - h_1)$$  \hspace{1cm} (1)

$$\dot{W}_{c2} = m_2 (h_4 - h_3)$$  \hspace{1cm} (2)

where $h_1$, $h_2$, $h_3$, and $h_4$ are enthalpy of the refrigerant at the inlet and the exit of first compressor and at inlet and the exit of the second compressor, respectively. The total of the power to system is calculated by
\[ \dot{W}_{tot} = \dot{W}_{c1} + \dot{W}_{c2} \]  

(3)

The heat release by the system to ambient is given by equation (4).

\[ \dot{Q}_{e} = \dot{m}_{e}(h_1 - h_5) \]  

(4)

where \( h_4 \) and \( h_5 \) are enthalpy of the refrigerant at the inlet and the exit of the condenser. Here \( \dot{m}_{e} \) is defined as the flow rate of the refrigerant entering the condenser. The heat absorbed by the refrigerant in the evaporator, \( \dot{Q}_{e} \) is given by:

\[ \dot{Q}_{e} = \dot{m}_{e}(h_1 - h_5) \]  

(5)

Refrigeration effect (ER) of the system is calculated by equation (6).

\[ ER = h_1 - h_5 \]  

(6)

The coefficient of performance of the system (COP) is given by

\[ COP = \frac{\dot{Q}_{e}}{\dot{W}_{tot}} \]  

(7)

In the literature it is very limited equation can be used to estimate the optimum intermediate pressure (\( P_i \)) in terms of suction and discharge pressures. To the best knowledge of the author, only the below equation is proposed to estimate the optimum intermediate pressure [12].

\[ P_i = \sqrt{P_e \times P_c} \]  

(8)

Where \( P_e \) and \( P_c \) are the pressure of the evaporator and the condenser. A computer program has been developed to solve the above equations. The properties of the refrigerant is modeled using the data provided by American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE).

3. Results and Discussions

The analysis is carried out for refrigerant R134a. Three operating conditions, hereafter named as case, are analyzed, the first case is at temperature evaporation of \(-15^\circ\)C and temperature condensation of \(35^\circ\)C, the second case is at temperature evaporation of \(-10^\circ\)C and temperature condensation of \(40^\circ\)C, and the third case is at temperature evaporation of \(-5^\circ\)C and temperature condensation of \(45^\circ\)C. For all cases the cooling load is assumed to be constant at 1000 Ton.

3.1. Mass flow rate of the refrigerant

The mass flow rate of the refrigerant from in the system for case 1 and case 2 are shown in figure 2. In the figure, the mass flow rate from the evaporator is shown by green line, while mass flow rate from the flash cooler to the second compressor line is shown by the blue line. The total mass flow rate in the system shown by red line. The figure shows the same trend for all cases. The intermediate pressure has no effect to the mass flow rate from the evaporator. However, mass flow rate from the flash cooler increases with increasing intermediate pressure. As a result, the mass flow rate in the system increases with increasing intermediate pressure.

![Figure 2. Mass flow rate in the system for case 1 and case 2](image-url)
3.2. Compressor Power

Figure 3 shows the compressor power of the system for all cases. The figure shows the almost similar trend. In the first compressor, the power increases with increasing intermediate pressure. This is because of increasing intermediate compressor will increase the pressure difference between the inlet and the exit of the compressor. The higher pressure difference results in higher compressor power. Different phenomenon is shown by the second compressor. The power of the second compressor decreases with increasing intermediate pressure. As a note, the top pressure and low pressure of the present system is fixed. This is because the evaporation temperature and condensing temperature are fixed. Thus, increasing the pressure intermediate means increasing pressure difference in the first compressor. Since the top and low pressures are fixed, pressure difference in the second compressor will decrease. This fact is clearly shown in figure 3.

As a system the power of the system is calculated by using equation (3). In the figure, the total of the compressor power in the system is shown by red line. It was clearly shown that there exists a minimum compressor power for an optimum intermediate pressure. If the intermediate pressure lower that its optimum value, increasing the intermediate pressure will decrease the total compressor power. But, if the intermediate pressure over than optimum one, increasing intermediate pressure will increase the compressor power. This fact reveals that there will be exist an optimum performance of the system. This will be discussed in the next section.

![Case 1 Compressor Power](image1.png)

![Case 2 Compressor Power](image2.png)

![Case 3 Compressor Power](image3.png)

**Figure 3.** Compressor power for all cases

3.3. Coefficient of Performance

Figure 4 shows the performance of the system for all cases. The figure shows that, in general, the COP of the system increases with decreasing pressure difference of the system. As a note, the highest pressure difference in the system is for case 1 and followed by case 2 and case 3. It can be seen that COP of case 3 is higher than case 2 and case 1. Also, the COP of case 2 is higher than case 1. As expected, there exists an optimum intermediate pressure for maximum COP. If the system is operated at its optimum intermediate pressure, increasing and decreasing the intermediate pressure will decrease the COP of the system.
In order to provide a convenient discussion, the value of maximum COP and the optimum intermediate pressure are presented in Table 1. The theoretical optimum intermediate pressure suggested by Arora [12] is calculated using equation (8) is also presented in the table. The data reveals that there is a discrepancy of the present optimum intermediate pressure in comparison with theoretical one. The difference varies from 7.19% to 11.3%. This fact suggest that equation (8) need to be revised by proposing an correction factor.

| Table 1. Optimum intermediate pressure |
|----------------------------------------|
| COPmax | Intermediate pressure [kPa] | Discrepancy [%] |
|        | Present | Theory Equation (8) |                |
| Case 1 | 3.65    | 491.80              | 436.22          | 11.30          |
| Case 2 | 4.61    | 486.80              | 451.81          | 7.19           |
| Case 3 | 6.02    | 509.80              | 464.87          | 8.81           |

4. Conclusions
In this work the optimum intermediate pressure of two-stage vapor compression refrigeration cycle has been explored. The conclusions are as follows.

a. The intermediate pressure has no effect to the mass flow rate from the evaporator. However, mass flow rate from the flash cooler increases with increasing intermediate pressure. As a result, the mass flow rate in the system increases with increasing intermediate pressure.

b. There exists a minimum compressor power for an optimum intermediate pressure. This fact reveals that there will be exist an optimum performance of the system.

c. There exists an optimum intermediate pressure for maximum COP.

d. There is a discrepancy of the present optimum intermediate pressure in comparison with theoretical one. The difference varies from 7.19% to 11.3%.

Figure 4. Coefficient of the performance
e. It is suggested to perform further analysis to propose a correction factor for the optimum intermediate pressure.

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