Evaluation on Expressions for Optimum Intermediate Condition of Two-Stage Vapor Compression Refrigeration Cycle

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***Abstract*—Mathematical expressions for estimating the optimum intermediate condition of two-stage vapor compression refrigeration cycle have been investigated. The objective is to evaluate the expressions for optimum intermediate condition (pressure or temperature) for maximum COP. A set of governing equations on two-stage vapor compression refrigeration cycle are developed and solved numerically. The two-stage cycle is analyzed using commonly used refrigerants for air-conditioning use, they are R12, R22, and R134a. There are six expressions for optimum intermediate condition found in literature. These expressions are divided into two group, expression for optimum pressure and expression for optimum temperature. These expressions are evaluated using the developed model. The results show that deviation of the expressions for optimum pressure can be up to 18.38%. On the other hand, the maximum deviation for optimum temperature is only 6.74%. This fact suggests that expressions for optimum temperature are better than pressure one. However, the expressions found in literature only specific for a particular refrigerant. Those can't be used for all refrigerants.**

Keywords—intermediate pressure, intermediate temperature, two-stage refrigeration cycle, vapor compression cycle.

I. INTRODUCTION

Greenhouse gas (GHG) emission is a big problem due to climate change which can be a catastrophe for the human being in the future. Many countries have released the target to reduce GHG emissions to a certain level. For instance, Indonesia targets to reduce its GHG emission up to 29% from level business-as-usual by the year 2030 by using its resources and the target can be increased up to 41% by international support. One of the solutions to reduce GHG emission is sustainable energy policy. The sustainable energy policy has two components; the use of renewable energy and using energy efficiently. In addition, enhancing the use of technologies with high energy efficiency is a must for reducing emissions. Implementation of energy efficient technology in vapor compression refrigeration cycle will give significant impact on emission reduction. Application of the vapor compression refrigerant cycle is a mature technology. The development of economy, will increase the consumption of energy by vapor compression (VC) cycle. This motivates researchers to investigate energy efficient technologies in the vapor compression refrigeration cycle. Several studies related to enhancement of energy efficiency of VC refrigeration cycle can be found in literature. Some techniques are heat recovery equipment [1,2], installing internal heat exchanger, multi-stages cycle, etc.

In this work, we focus on increasing efficiency of VC refrigeration cycle by using a multi-stage technique. The number of stages in the VC refrigeration cycle has been optimized by Bilge and Temir [3]. The study carried out on a VC refrigeration cycle with cooling capacity of 600 kW and using ammonia as a working fluid.

The correlation of inter-stage pressure and performance has been explored by Purohit et al. [4] on a two-stage VC refrigeration cycle coupled with an intercooler. In the tested system, six refrigerants have been tested. They are R22, R143a, propane, carbon dioxide and nitrous oxide. The results show that inter-stage pressure should be more considered in the areas with higher daily temperature. Two-stages VC refrigeration cycle has been analysed by using energy, exergy and sustainability concept [5]. Multistage VC refrigeration cycle has been optimized using general staging model [6]. Torrella et al. [7] performed experimental work on two-stage VC cycle that employing a double compressor. The objective was to detail an analysis of the inter-stage working conditions of a two-stage vapor compression facility equipped with a compound compressor with refrigerant R 404a. The results show the inter-stage working temperature/pressure obtained in the test has been contrasted with the two usual criterions of the optimum operational conditions definition: the arithmetical mean of the refrigerant condensing and evaporating temperatures and the criterion of equal pressure ratios in both stages.

Some researchers focus on proposing mathematical expression to assess the optimum intermediate pressure. In a two-stage vapor compression refrigerant cycle, the selection of intermediate pressure is critical for system parameter design. In designing a multi-stage cycle, the evaporating temperature, condensing temperature, and intermediate pressure must be determined first. Based on these values, the compressor and heat exchangers are selected. This fact shows that intermediate pressure is a critical value in designing a multi-stage VC refrigeration cycle. This motivates researcher to carry out studies on the appropriate expression to estimate an intermediate pressure of two-stages VC refrigeration cycle. The commonly used intermediate pressure for two-stage VC refrigeration cycle is the geometric mean of the condenser and evaporating pressure [8]. This was derived based on two assumptions, they are refrigerant obeys the ideal gas law and the temperature at suction of the high-stage compressor is took down to that of the low-stage one. Chandra [9] proposed an expression by taking the temperature of the high compressor to develop correction factor to the intermediate pressure equation. However, this equation indicated deviations between experimental

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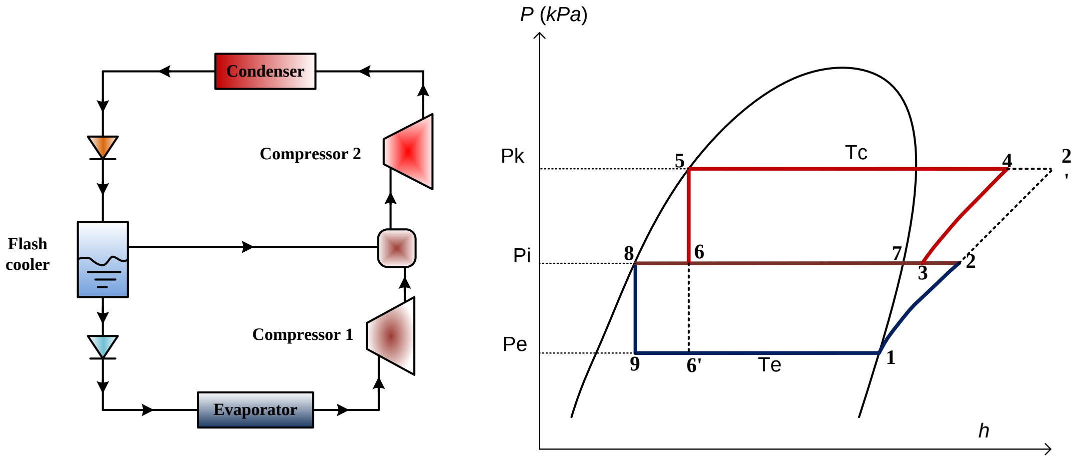

Figure 1. Schematic and *P*-*h* diagrams of the system.

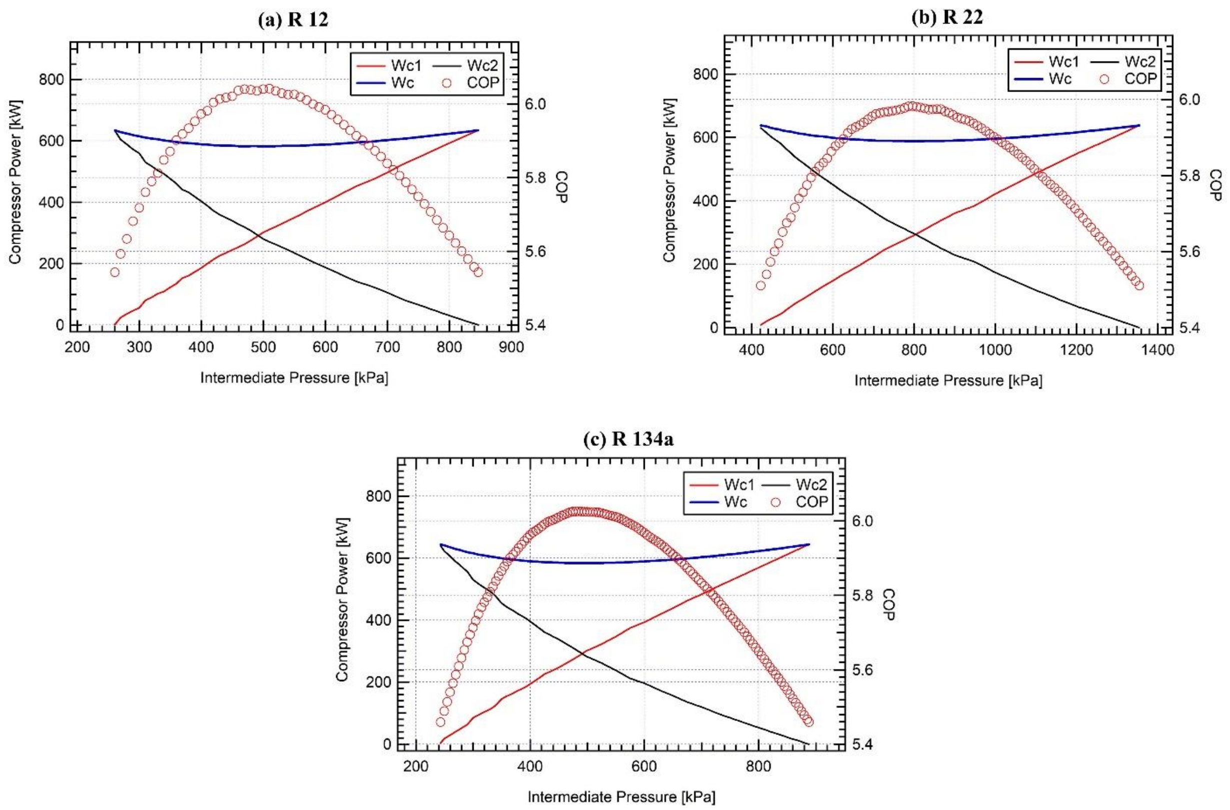

Figure 2. Performance of two-stage vapor compression cycle for case 1.

and simulation as reported by Jekel and Reindl [10] and Jin et al. [11]. In the further study, Chandra [9] proposed a new correlation of the optimum intermediate pressure in some cases. In the correlation temperature condensation and evaporation are taken into account. Domanski [12] investigated 38 refrigerants that has dissimilar molar weights based on the flash tank cycle. The results revealed that the optimum intermediate temperatures are fairly similar for the fluids considered and can be well calculated by using the mean temperatures between the condenser and evaporator. This fact is also concluded by Zubair et al. [13]. As a note, the proposed equation is in temperature intermediate term. Even though the proposed correlation shows a good agreement, however, if it is converted into pressure, the difference between the estimated optimum intermediate pressure and the actual one may not be neglected and needs should be taken into account. This deviation is due to the approximate exponential relationship between the saturation pressure and its corresponding saturation temperature [14]. Tiedeman and Sherif [15] reported their study on the optimization of two-stage flash intercooler system based on the COP and exergetic efficiency and presented a corrected correlation for estimating the optimum intermediate pressure. The equation related to an intermediate temperature that is maximizing both COP and the exergetic efficiency. In addition, the optimum intermediate pressure is closer to the geometric mean of the condensing and evaporating temperatures when sub-cooling, superheating and isentropic efficiencies of the compressors are taken into consideration simultaneously. The above literatures show that there exists an optimum intermediate pressure for maximum COP of a two-stage VC refrigeration cycle. However, still, there many

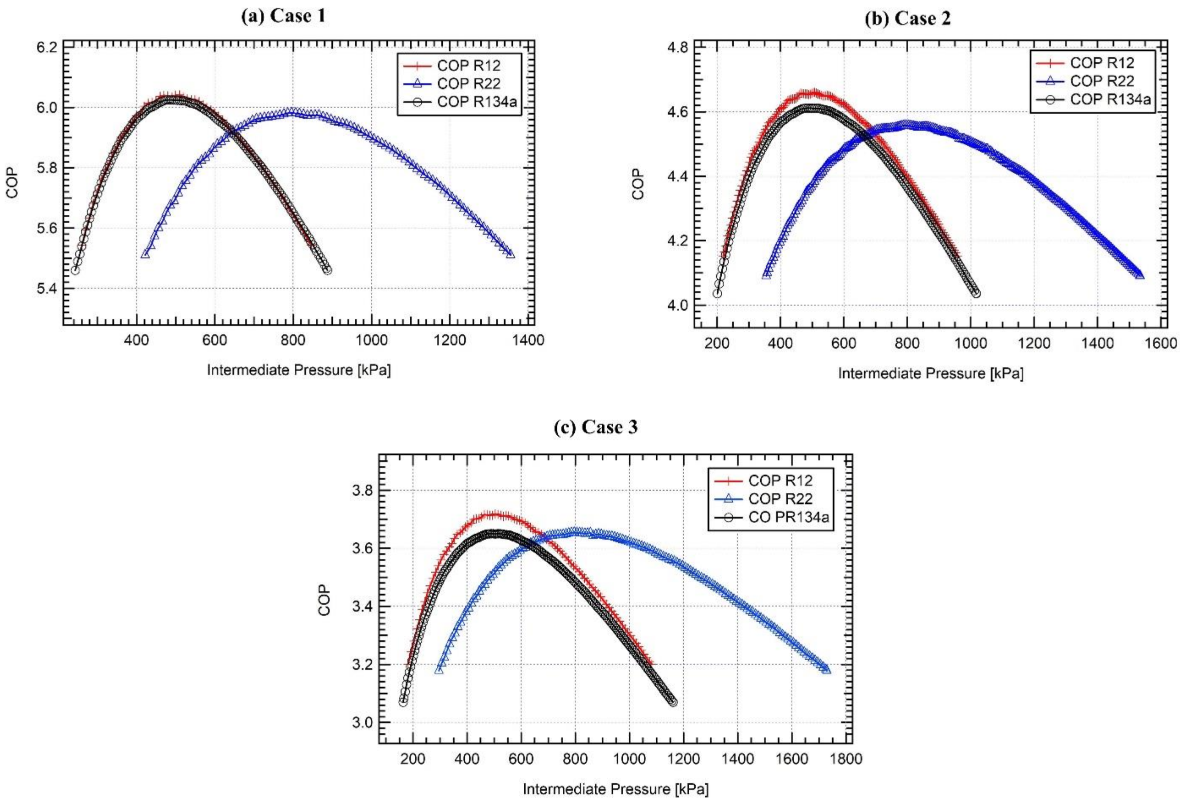

Figure 3. Effect of intermediate pressure on COP.

parameters or issues need to be clarified. The proposed equations are mainly included the effects of the condensing and evaporating parameters without any consideration on inter-stage configurations [7], [16]. Generally, the isentropic efficiency of the compressor is taken into account. However, it is assumed to be constant for all compressor power. As a note, the first and the basic equation of the optimum intermediate pressure is the geometrical mean of the evaporating and condensing pressure. Several modified equations have been proposed. Those equations can be divided into three different trends. The first shows that the optimum intermediate pressure is above the geometrical mean of the evaporating and condensing pressure [11], [17]. In contrast, the second shows that the optimum intermediate pressure is significantly below the geometrical mean [18]. The third is that COP changes are small when the intermediate pressure close to the optimum one [13], [15]. Recently, Jian et al. [19] explored the role of optimum intermediate pressure in the design of two-stage vapor compression systems as a further investigation. They analysed six types of two-stage VC refrigeration cycle where R22 and Ammonia as the refrigerants.

Those studies show that multi-stage strategy to search the optimum VC refrigeration cycle has come under scrutiny. However, many issues need further investigation. In the present work, evaluation of the optimum intermediate pressure for maximum COP of the two-stage vapor compression refrigeration cycle is carried out. The evaluation is performed using refrigerants R12, R22 and R134a. As a note, R134a is a typical refrigerant which is used in VC refrigeration cycle in the air-conditioning application. However, the study on the optimization of VC refrigeration cycle that uses R134a is very limited. In this work refrigerant R134a will be analysed in a two-stage VC refrigeration cycle. The objective is to explore the equations of the optimum intermediate pressure of two-stage VC refrigeration cycle. The results are expected to supply the necessary information on developing highly efficient vapor compression refrigeration cycle.

II. Method

In this study, governing equations are formulated and solved numerically. Figure **1** shows the outline of the two-stage VC refrigeration cycle. As shown, the *p-h* diagram is shown as well. The figure shows that the system consists of high pressure and low pressure compressors, a condenser, a flash cooler, and an evaporator. Several assumptions are made. The resistance of the refrigerant flow is negligible. The pressure loss in evaporator and condenser are neglected. The differences of potential energy and kinetic energy of the refrigerant are ignored. The condition of working fluid in the container is constant. The suction and exhaust fluctuations of compressors are neglected.

A. Governing equations

The equations that generated from the conservations law are formulated in the following. The rate energy in the first compressor ($\dot{W}_{c1}$) and in the second condenser ($\dot{W}_{c2}$) are given by

$$\dot{W}_{c1} = \dot{m}_1(h_2 - h_1) \quad (1)$$

$$\dot{W}_{c2} = \dot{m}_2(h_4 - h_3) \quad (2)$$

TABLE 1.
EVALUATION THE OPTIMUM INTERMEDIATE PRESSURE OF R12

Parameter	**Case 1**	**Case 2**	**Case 3**
Condensing Pressure [kPa]	846.2	958.8	1082
Condensing Temperature [K]	308	313	318
Evaporating Pressure [kPa]	260.6	218.8	182.3
Evaporating Temperature [K]	268	263	258
Opt. intermediate pressure [kPa]	469.59	503.02	544.12
Opt. intermediate temperature [K]	286.5	288.8	291.57
Evaluation based on Pressure			
Equation (8) [kPa]	469.60	458.02	444.13
Deviation	0.00%	8.95%	18.38%
Equation (9) [kPa]	503.42	499.67	493.07
Deviation	7.20%	0.67%	9.38%
Equation (10) [kPa]	469.94	458.37	444.48
Deviation	0.08%	8.88%	18.31%
Evaluation based on Temperature			
Equation (12) [kPa]	287	286	285
Deviation	0.17%	0.97%	2.25%
Equation (13) [kPa]	287.30	286.91	286.43
Deviation	0.28%	0.65%	1.76%
Equation (14) [kPa]	288	288	288
Deviation	0.52%	0.28%	1.22%

respectively. Here, h_1, h_2, h_3, and h_4 are enthalpy of the refrigerant when entering and the leaving from lower compressor and at entering and leaving of the higher compressor, respectively. The rate of energy to the VC refrigeration cycle is given by

$$\dot{W}_c = \dot{W}_{c1} + \dot{W}_{c2} \quad (3)$$

The total of heat released by the VC refrigeration cycle to ambient is calculated by equation (4).

$$\dot{Q}_c = \dot{m}_2(h_4 - h_5) \quad (4)$$

where h_4 and h_5 are enthalpy of the refrigerant at the intake and the outlet of the condenser, respectively. Here $\dot{m}_c$ is formulated as the flow rate of the refrigerant flows into the condenser. The cooling that can be drawn by the evaporator, $\dot{Q}_e$ is calculated by:

$$\dot{Q}_e = \dot{m}_1(h_1 - h_9) \quad (5)$$

Performance of VC refrigeration cycle is examined by COP and it is calculated by

$$COP = \frac{\dot{Q}_e}{\dot{W}_{tot}} \quad (6)$$

A FORTRAN code has been formulated to analyse the governing equations presented above. The data drawn from ASHRAE (American Society of Heating, Refrigeration, and Air Conditioning Engineers) are used to model the working fluid properties.

B. *Optimum intermediate pressure*

The intermediate pressure is the working pressure of exhaust low compressor and assumed to be similar to suction pressure of high compressor. Thus, the optimum intermediate pressure is defined as the intermediate saturation pressure which results in maximum the system COP. It is searched to meet the below condition.

$$\frac{d(COP)}{dp} = 0 \quad (7)$$

In the literature, as reviewed in the first section, several equations have been proposed. Gosney [20] reported that the commonly used equation for optimum intermediate pressure is the geometric mean of the condensing and evaporating pressure.

$$P_i = \sqrt{P_e \times P_c} \quad (8)$$

Where P_e and P_c are the saturation pressure at the evaporator and the saturation pressure at the condenser, respectively. Chandra [9] proposed a correction to calculate the optimum intermediate pressure as given by the equation below.

$$P_i = \sqrt{P_e \times P_c \frac{T_c}{T_e}} \quad (9)$$

Here, T_e [K] and T_c [K] are the evaporating temperature and condensing temperature at saturation, respectively. In addition, De Lepeleire [21] proposed an additional correction to the equation as shown in the below equation.

$$P_i = \sqrt{P_e \times P_c} + 0.35 \quad (10)$$

intermediate temperature instead of pressure. Behringer [22] proposed the below equation to calculate the optimum intermediate temperature.

$$T_i = T_{\sqrt{P_e P_c}} + 5 \quad (11)$$

In this equation, the optimum intermediate pressure is still used, but it is only to be converted into intermediate temperature and to get optimum one it is corrected by adding 5 K. Rasi [23] also proposed the optimum intermediate temperature as the below equation.

$$T_i = 0.4T_c + 0.6T_e + 3 \quad (12)$$

Similar to the geometric mean pressure equation, Czaplinski [24] proposed geometric mean temperature to calculate the optimum one.

$$T_i = \sqrt{T_e \times T_c} \quad (13)$$

And Domanski [12] proposed the arithmetic mean temperature to calculate the optimum intermediate temperature.

$$T_i = \tfrac{1}{2}(T_c + T_e) \quad (14)$$

These optimum intermediate pressure and temperature are evaluated in this work.

III. RESULTS AND DISCUSSION

Three commonly used refrigerants in the vapor compression refrigeration cycle will be used; they are R11, R22 and R134a. These refrigerants are employed for three different operational conditions, respectively. The first group is with evaporator at temperature -5^{0}C and condenser at temperature 35^{0}C. The second group is with evaporator at temperature -10^{0}C and condenser at temperature 40^{0}C. The third group is with evaporator at temperature -15^{0}C and condenser at temperature 45^{0}C. The condensing and evaporating temperature difference of the first, second and third case are 40^{o}C, 50^{o}C, and 60^{o}C, respectively. For VC refrigeration cycle of air conditioning unit, these temperature difference can be categorized as the representation of low, medium, and high-temperature difference, respectively. Thus, a total of 9 simulations are carried out. In every simulation, the intermediate pressure is varied from evaporating pressure to condensing pressure. In every intermediate pressure, the performance of the system is analysed, and the results are plotted. In all groups, the load of the evaporator is assumed to be constant at 1000 Ton. Several researchers also proposed the optimum

A. Characteristic of intermediate pressure

Figure 2 shows the power at lower and upper compressors for case 1. It can be seen, the total of energy rate of compressor and COP of the system are also presented. It can be seen that all refrigerants show the similar trend. Increasing intermediate pressure from evaporating pressure to condensing pressure results in increased power to the low-pressure compressor and at the same time decreasing power to the high-pressure compressor. This is because a higher intermediate pressure increases the pressure difference in the lower compressor and decreases the pressure difference in the upper compressor. As a note, the total power to the system is the sum of the power at lower and upper compressors. In the figure, the total compressor power is shown by the solid blue line. The figure shows that, at low intermediate pressure, total compressor power decreases with increasing intermediated pressure. However, after reaching the minimum value total compressor power growing up with raising intermediate pressure.

The trend of the compressor power affects the COP of the system. In the figure, the COP of the system is shown

TABLE 2.
EVALUATION THE OPTIMUM INTERMEDIATE PRESSURE OF R22

Parameter	Case 1	Case 2	Case 3
Condensing Pressure [kPa]	1355	1534	1729
Condensing Temperature [K]	308	313	318
Evaporating Pressure [kPa]	421.8	354.8	296.2
Evaporating Temperature [K]	268	263	258
Opt. intermediate pressure [kPa]	856	792.74	795.6
Opt. intermediate temperature [K]	290.8	288.2	305.6
Evaluation based on Pressure			
Equation (8) [kPa]	756.00	737.74	715.63
Deviation	11.68%	6.94%	10.05%
Equation (9) [kPa]	810.46	804.82	794.50
Deviation	5.32%	1.52%	0.14%
Equation (10) [kPa]	756.35	738.09	715.98
Deviation	11.64%	6.89%	10.01%
Evaluation based on Temperature			
Equation (12) [kPa]	287	286	285
Deviation	1.31%	0.76%	6.74%
Equation (13) [kPa]	287.30	286.91	286.43
Deviation	1.20%	0.45%	6.27%
Equation (14) [kPa]	288	288	288
Deviation	0.96%	0.07%	5.76%

TABLE 3.
EVALUATION THE OPTIMUM INTERMEDIATE PRESSURE OF R134A

Parameter	Case 1	Case 2	Case 3
Condensing Pressure [kPa]	887.91	1017.61	1161.01
Condensing Temperature [K]	308	313	318
Evaporating Pressure [kPa]	243.39	200.6	163.9
Evaporating Temperature [K]	268	263	258
Opt. intermediate pressure [kPa]	484.8	486.8	521.8
Opt. intermediate temperature [K]	287.7	287.9	290.07
Evaluation based on Pressure			
Equation (8) [kPa]	464.87	451.81	436.22
Deviation	4.11%	7.19%	16.40%
Equation (9) [kPa]	498.36	492.89	484.30
Deviation	2.80%	1.25%	7.19%
Equation (10) [kPa]	465.22	452.16	436.57
Deviation	4.04%	7.12%	16.33%
Evaluation based on Temperature			
Equation (12) [kPa]	287	286	285
Deviation	0.24%	0.66%	1.75%
Equation (13) [kPa]	287.30	286.91	286.43
Deviation	0.14%	0.34%	1.25%
Equation (14) [kPa]	288	288	288
Deviation	0.10%	0.03%	0.71%

by red circle mark. The figure shows that, in the beginning, increasing intermediate pressure increases COP of the system. After reaching a maximum value, COP decreases with raising intermediate pressure. This is because COP of a system is the inverse proportion to the compressor power. In other words, COP increases with decreasing compressor power. This fact suggests that there exists an optimum intermediate pressure for maximum COP.

As a note, the present system is operated at the same evaporating and condensing temperature also the same cooling load. The COP maximum of the system for refrigerant R12, R22 and R134a are 6.04, 5.98, and 6.03, respectively. This fact suggests that the highest COP maximum is for R12 and followed by R134a and the lowest is for R22.

Figure 3 shows the effect of intermediate pressure on COP of the system for all refrigerants and cases. The figure shows that the characteristic of refrigerant R12 is similar to R134a. It can be seen that the pressure range and COP of both refrigerants are almost the same. This is because R134a is designed as the replacement for R12. However, at high and low-temperature differences there is a small discrepancy between refrigerant R12 and R134a. On the other hand, R22 shows the similar trend but different value. The maximum COP of refrigerant R22 is lower than R12 and R134a. In addition, the pressure of R22 is higher than R12 and R134a at the same operational temperature. Effect of temperature range between evaporator and condenser can be examined by using Figure 3. The higher temperature range results in a lower maximum COP. For instance, refrigerant R134a, the maximum COP for case 1, case 2, and case 3 are 6.03, 4.61, and 3.65, respectively.

A. Evaluation of intermediate pressure

The main objective of the present work is to evaluate the correlations proposed by previous researchers. The optimum intermediate pressure and temperature are determined from the simulation. The simulation results are compared with the proposed correlation. There are six correlations can be used to estimate the optimum intermediate operational condition. They are divided into optimum intermediate pressure and optimum intermediate temperature. The optimum intermediate pressure can be calculated using equation (8), equation (9), and equation (10). On the other hand, the optimum intermediate temperature can be estimated using equation (12), equation (13), and equation (14). The results of these equations are compared with the present simulation results.

Table 1 shows the comparison of the simulation and correlation results for refrigerant R12. For case 1, the optimum intermediate pressure and temperature are 469.59 kPa and 286.5 K, respectively. The estimated optimum intermediate pressure calculated by using equation (8), equation (9), and equation (10) are 469.6 kPa, 503.42 kPa, and 469.94 kPa, respectively. The maximum deviation is shown by equation (9), it is 7.2%. On the other hand, the optimum intermediated temperature calculated by using equation (12), equation (13) and equation (14) are 287 K, 287.3K, and 288 K, respectively. The deviation varies from 0.17% to 0.28%. These values suggest that the optimum intermediate temperature equation predict the optimum condition better than intermediate pressure. Similar comparisons are made for case 2 and case 3 for refrigerant R12. If the pressure equations are used, the deviation varies from 0.67% to 18.38%. On the other hand, if the correlations of temperature are used the deviation varies from 0.28% to 1.25%. It can be said that the deviations resulted from intermediate temperature equations are lower than intermediate pressure.

The estimated optimum intermediate pressure and temperature from the present simulation and the correlation for refrigerant R22 are shown in Table 2. Data of the table shows that for case 1, the deviation resulted by correlation of optimum intermediate pressure varies from 5.32% to 11.68%. The deviation resulted by case 2 and case 3 is from 1.52% to 6.9% and from 0.14% to 10.05%, respectively. On the other hand, the optimum intermediate temperature for case 1, case 2, and case 3 varies from 0.96% to 1.31%, from 0.07% to 0.76%, and from 5.76% to 6.74%, respectively. These values reveal that for the refrigerant R22 the correlation for optimum intermediate temperature predicts the optimum intermediate condition is better than pressure one.

Table 3 shows the comparison of optimum intermediate pressure and temperature resulted by the present simulation and the correlations for refrigerant R134a. The optimum intermediate pressure and temperature resulted by the present simulation for case 1, case 2, and case 3 are 484.8 kPa and 287.7K, 486.8 kPa and 287.9 K, and 521.8 kPa and 290.07 K, respectively. For case 1, the correlations of pressure result in deviation varies from 2.8% to 4.11%. For case 2 and case 3, the deviation varies from 1.25% to 7.19% and from 7.19% to 16.4%, respectively. It is clear that the deviation is still significant it is over than 10%. On the other hand, if the correlations of temperature are used, for case 1 the deviation varies from 0.1% to 0.24%. And for case 2 and case 3, the deviations vary from 0.03% to 0.66% and from 0.71% to 1.75%, respectively. These values also show that deviation from temperature correlation is better than pressure correlations.

Those facts show that it is better to use temperature correlation to predict the optimum intermediate condition for maximum coefficient of performance of a two-stage VC refrigeration cycle. Furthermore, the results also show that there is no specific correlation can be employ to predict an accurate optimum intermediate condition.

IV. CONCLUSIONS

The optimum intermediate operational condition of two-stage vapor compression refrigeration cycle has been investigated. Numerical simulations have been carried out to investigated the optimum intermediate pressure and intermediate temperature for refrigerant R12, R22, and R134a. The results of the present study are compared with the correlation proposed in the literature. The conclusions of this studies are as follows.

- The literature review reveals that there are two types of correlation of optimum intermediate conditions for maximum COP found in the literature. The first is optimum intermediate pressure and the second is intermediate temperature.
- For the used refrigerants, the correlations of optimum intermediate temperature show a better estimation in

comparison with intermediate temperature. Which shows a lower deviation.

The correlations found in literature only specific to a particular refrigerant. It cannot be used for all refrigerants.

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