Thermodynamic Assessment of Single-Stage and Two-Stage Vapour Compression Refrigeration Systems: A Simulation-Based Study for Humid Climate Applications

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Abstract: - Vapor Compression Refrigeration (VCR) systems are commonly used for Air Conditioning (AC) purposes in residential, commercial, and industrial applications in both dry and wet climatic conditions. Most of these applications are achieved with the conventional single-stage Vapor Compression Refrigeration systems; however, the systems possess lower cooling capacity and coefficient of performance at low evaporator temperatures and higher-pressure ratios, and this situation worsens in the tropics. This study compares the performances of single and two-stage Vapor Compression Refrigeration systems using sustainable refrigerants in a climate region.

Performance of single-stage and two-stage Vapor Compression Refrigeration (VCR) systems containing the new generation refrigerants of R-600a (isobutane), R-600 (n-butane), R-717 (ammonia), and R-290 (propane) was studied theoretically via performance simulation. Results were discussed, changing evaporator temperature, condenser temperature, degree of superheat, subcooling, and mass flow rate of refrigerant. Thermodynamic analysis of VCR systems, including all refrigerants, was conducted using the Engineering Equation Solver (EES) software.

A comparative performance study was conducted between a two-stage vapor compression refrigeration (VCR) system and a corresponding single-stage vapor compression refrigeration system. The performance of the two-stage V.C.R. system is higher than that of the single-stage system and maximum at low evaporating temperature. Maximum COP of 7.636 was obtained with R-717 in a two-stage VCR system at evaporating temperature of 0 °C. Minimum COP of 0.948 was obtained with R-290 in a single-stage VCR system at a condensing temperature of 80 °C. The performance of the two-stage system is better, and R-717 is found to be the best refrigerant among the refrigerants studied in this research.

The findings of this study will provide an experimental basis for future investigations of new systems and designs. This would enable design engineers to assess and determine improvements to air conditioning and refrigeration systems, and refrigerants appear to be economically worthwhile.

Key-Words: - Vapour compression refrigeration, Single-stage system, Two-stage system, Coefficient of performance, Eco-friendly refrigerants, Ammonia

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1 Introduction

Refrigeration is a technology used in the preservation of food and drugs, in industrial applications, and for air conditioning. It is a very essential technology in our daily living and that of our animals. It is a very dominant technology in Nigeria and other developing countries. The vapour compression refrigeration system (VCRS) is the most efficient of all the refrigeration systems in the world. This is because VCRS performance is heavily affected by the ambient temperature, design loads, and the energy source. In tropical countries like Nigeria, these parameters are often very high, and therefore, there is a need to improve the VCRS technology by introducing new technology or by improving the existing technology. The conventional single-stage VCRS consists of four major components, namely: compressor, condenser, expansion valve, and evaporator. The VCRS uses the vapour compression cycle, which has five refrigerant states, namely, high pressure and temperature after compression, low pressure and temperature after expansion, and also evaporation of the refrigerant at the evaporator for absorption of heat from the cold reservoir. Although the single-stage VCRS is very simple, of small size, and also cheap, it does not perform efficiently at low evaporator temperatures, which is a characteristic of tropical climates. This is as a result of a large pressure ratio and also large work input of the compressor (Mehrpooya et al. 2020, Librado et al. 2018, and Bhattad et al. 2017). In general, a standard VCRS consists of a compressor, a condenser, an expansion valve, and an evaporator. This system compresses the refrigerant, condenses it, expands it through an expansion valve for throttling, and evaporates it for heat absorption. Although the VCRS is the simplest, the smallest in size, and the cheapest of all systems, it generally performs poorly under low evaporator temperatures and high-pressure ratios. These conditions are, however, very common in tropical climates like that of Nigeria.

A new two-stage vapor compression refrigeration system has been developed to raise the performance of the system. The first stage is the same as the conventional system. The second stage has a cooling section, which can be a cooling chamber or a flash chamber. After cooling in this section, the high-pressure and low-temperature vapor of the refrigerant is further cooled so that the cooling capacity of the system is raised, the work of the compressor is reduced, and the coefficient of performance of the system and the economy are also improved. The new system is especially suitable for the new type of refrigerators, air conditioners, and industrial refrigeration systems. The two-stage system is more complicated than the conventional system. The new components are more expensive and more difficult to operate and maintain. The results of numerical calculations of the various parameters of the new system show that the new system can have a significantly better performance in some conditions.

Performance evaluation of the new system of air conditioning chillers indicated that two-stage systems have a certain application. Two-stage systems are recommended for low ambient temperatures and low input energy with high cooling coefficient values. Single-stage systems were found to be economical and require no additional cooling capacity. Therefore, they can be recommended for moderate ambient temperatures. The research work on the enhancement of air conditioning system performance started in 2018 and is still ongoing by Librado et al (2019).

Apart from the general problems of refrigeration in Nigeria, such as high ambient temperature, poor electricity supply, lack of technical skills, and high cost of system components, there are other parameters to be considered. Some of them are: whether to use a two-stage or single-stage compression system. While the two-stage system provides higher cold capacity, which is required in some applications such as vaccine storage and ice production, its drawback is the complexity of the system, higher initial capital cost, etc. The single-stage system, on the other hand, is simple, easy to handle and maintain, has a lower initial capital cost, etc. It is therefore recommended for residential and small commercial buildings. This paper reviews the solar-powered adsorption refrigeration system recently proposed by Sangotayo et al as an alternative means of meeting cold chain requirements in Nigeria. The working pairs were activated carbon and methanol. Models for the system were developed, and a computer program was written to investigate the performance of the system under varying evaporation temperature and desorption temperature. The result indicated that the performance of the system is largely dependent on the desorption temperature and that optimal performance is achieved when the desorption temperature is less than 120 °C.

Performance evaluation of single-stage and two-stage vapor compression refrigeration systems in Nigeria was done using a computer simulation model and analysis. The performance evaluation was done by varying the evaporator temperature, type of refrigerant, and cooling capacity. The main objective of this work is to guide the design engineer in his choice of an efficient system for a particular application. The design engineer will base his choice on some deciding parameters such as cooling capacity, coefficient of performance, capital and operating cost, etc. The performance evaluation of the various systems under the various conditions provides enough information to enable the design engineer to select an appropriate system for an application in Nigeria.

2 Methodology

The coefficient of performance (COP) of a refrigeration system is influenced by some parameters. These parameters include work done in compressing the refrigerant at the given pressure drop between the condenser and evaporator, as well as the amount of cooling provided. The EES (Engineering Equation Solver) software was used to determine the thermodynamic properties of the systems. The calculated values were used in determining the COP of the refrigeration systems.

Refrigeration System Performance. In this study, an investigation of the performance of different refrigeration systems at various temperatures has been made. The experimental range of temperatures is from -50 to +80 degrees centigrade. Single-stage vapour compression system and two-stage vapour compression system of four refrigerants have been studied to find out the best refrigerant for the purpose. From the research, the performance of different refrigeration systems at various temperatures has been identified clearly. Thus, this research helps a great deal in designing an efficient refrigeration system for the required purpose. The performance of the refrigeration system at different temperatures helps to know the performance of the system in various applications.

2.1 Selection of Refrigerants

Factors considered in the suitability of the refrigerant used are:

i. Availability
ii. Global warming potential (GWP)
iii. Ozone depletion potential (ODP)
iv. Good performance properties

2.2 Refrigerants

Refrigerant is the working fluid in the refrigerating system. The refrigerants employed for this work are R-600a (isobutane), R-600 (butane), R-717 (Ammonia), and R-290 (propane). Table 1 shows the thermophysical and environmental properties of the refrigerants used.

2.3 Performance Analysis

The performance analysis is a measure of testing the effectiveness of the refrigeration system. The analysis is done to be able to identify the system with the highest COP by varying the operating conditions and properties.

2.3.1 Theoretical description of Single Stage (SS) vapour compression refrigeration cycles

Comparison of Four Different Refrigeration Cycles using R-600a, R-600, R-717 and R-290 as Refrigerants. Figure 1 Single-stage refrigeration cycle. Figure 2 T-S diagram for a single-stage refrigeration cycle. In this paper, four different cycles have been investigated with refrigerants R-600a, R-600, R-717 and R-290. At point 1, the refrigerant enters the compressor as saturated vapour and this saturated vapour is being compressed isentropically from point 1 to 2. A portion of the vapour refrigerant from point 2 to 3 comes in contact with a portion of the surface area of the condenser exposed outside. At points 3 and 4, the vapour refrigerant comes in contact with the expansion device (in this illustration it is an expansion valve) and a sharp decrease in pressure occurs. This results in an adiabatic flash and the auto- refrigeration of a portion of the liquid. The resulting low-quality (saturated) liquid at point 4 is then returned to the evaporator where the process is continued between points 4 and 1. There, the liquid refrigerant is vapourised, removing heat from the cold space. The resulting vapour is then returned to the compressor to start the cycle all over again.In the performance analysis of this system, the parameters given in Table 2 were assumed: Table 2. Parameters assumed in performance analysis.

T_C - Condenser Temperature. Increased from base line (original 50 °C) by +30 °C to +80 °C.

Evaporator Temperature (T_E) The baseline for the T_E test point was established as -20 °C with a measurement range of 0 °C to -50 °C.

The Superheating Temperature (T_{SUP}) has a base value of 4 °C. The measuring range is from 0 ° C to 20 ° C.

Temperature for sub-cooling ($_{TSUB}$): 4° C. The sub-cooling is changed by ±3° C from base case (nominal value). For this test condition, test points at -3° C, -6° C and -9° C shall be used.

The base case mass-flow rate of the refrigerant ($\dot{m}$) was pre-set to 0.05 kg/sec and subsequently ramped up and down through the range 0.01-0.11 kg/sec in order to test extreme conditions.

Table 1. Thermophysical and environmental properties of the selected refrigerants

Properties	**R-6OO**	**R-6OOa**	**R-717**	**R-290**
Chemical Formula	C_4H_{10}	C_3H_8	NH_3	C_4H_{10}
Molar Mass	58.12g/mol	44.10g/mol	17g/mol	58.12g/mol
Colour	Colourless	Colourless	Colourless	Colourless
Density	2.0098kg/m^3	2.0098kg/m^3	1.61g/l	2.51mg m/l
Boiling Point	-42.25°C	-42.25°C	-33°C	-11.7°C
Melting Point	-187.7°C	-187.7°C	-77.7°C	-159.42°C
Odour	Odourless	Odourless	Pungent	Odourless
Vapour Pressure	853.16KPa	853.16KPa	308.61KPa	204.8KPa
Ozone Depletion Potential (ODP)	0	0	0	0
Global Warming Potential (GWP)	3	3	0	3

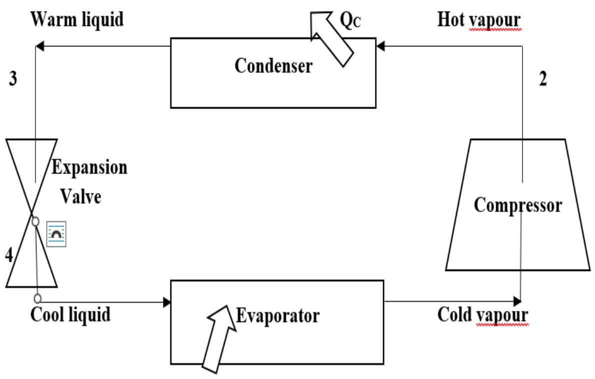

Fig. 1: Schematic Diagram of a Single-Stage Refrigeration System

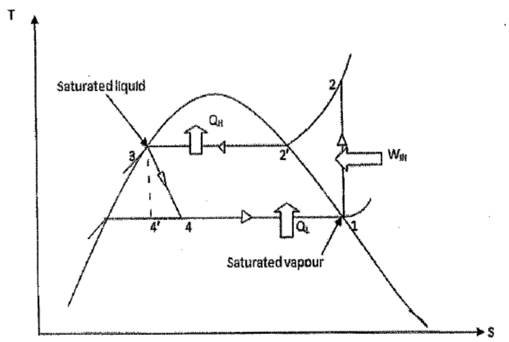

Fig. 2: T-S Diagram of a Single-Stage Refrigeration System

About these assumed parameters, thermodynamic analysis is performed using the EES software, and the balanced equations are applied to derive the work input into the compressor, the refrigeration capacity, and the COP of the system. The analysis is based on the first law of thermodynamics using the energy equation at steady state and assuming that changes in kinetic and potential energies are negligible.

Mass balance

$$\sum_{in} \dot{m} = \sum_{out} \dot{m} \qquad 1$$

Energy balance

$$Q - W + \sum_{in} \dot{m}h - \sum_{out} \dot{m}h = 0 \qquad 2$$

The refrigeration capacity is defined by

$$Q_L = \dot{m}(h_1 - h_4) \qquad 3$$

Work input into the compressor is given by

$$W_{in} = \dot{m}(h_2 - h_1) \qquad 4$$

Condenser load is calculated by

$$Q_H = \dot{m}(h_3 - h_2) \qquad 5$$

For the throttling process

$$\dot{m}h_3 = \dot{m}h_4 \qquad 6$$

The coefficient of performance of the cycle is

$$COP = {Q_L}/{W_{in}} \qquad 7$$

Table 2. Thermodynamic Operating Parameters

Properties	**Base Value**	**Variation Difference**	**Variation Range**
Evaporating Temperature (T_C)	-20 ˚C	-5	0-30
Condensing Temperature (T_E)	50 ˚C	+5	30- 80
Superheating Temperature (T_{SUP})	4 ˚C	+2	0- 20
Subcooling Temperature (T_{SUP})	4 ˚C	+2	0- 20
Mass Flow Rate ($\dot{M}$)	0.01kg/s	0.01	0.01-0.11

2.3 Two-Stage (TS) Vapour Compression Refrigeration Cycle

Fig. 3: Cascade Refrigeration System Figure 4 Cascade Refrigeration System High-stage compressor, low-stage compressor, inter-stage pressure (pressure between high-stage discharge pressure and low-stage suction pressure) are featured in this system. From Figs. 3 and 4, the vapour refrigerant at the dry saturated state point (I) is entered into the first stage of the compressor. The compressed vapour refrigerant is carried at point (2) to the inter-stage pressure Pi. The mixture at point (9) is entered into the second stage of the compressor. The compressed vapour refrigerant is carried from point (9) to (4) by means of a compressor. The refrigerant at point (4) is entered into the condenser. The refrigerant at point (4) condenses into a liquid at point (5). The sub-cooled liquid refrigerant at point (5) is entered into the high-pressure side flow control device called the expansion valve. In the expansion valve, the liquid refrigerant at point (5) is expanded into (6). A part of the liquid refrigerant is vaporised at (6) to (7) in the flash cooler. The flashed refrigerant is used for cooling the remaining liquid refrigerant of the inter-stage pressure at saturated temperature. The liquid refrigerant at point (7) is entered into the low-pressure expansion valve at point (8). A small portion of the liquid refrigerant is pre-flashed, and then all the liquid refrigerant is evaporated into vapour in the evaporator at point (8) to (1)

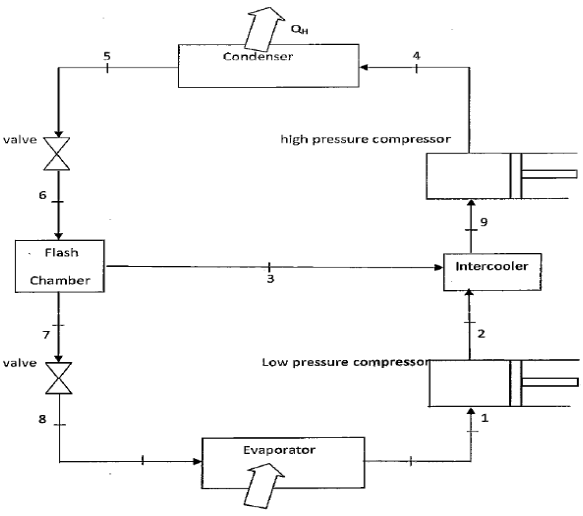

Fig. 3: The Schematic Diagram of Two-Stage Vapour Compression Cycle

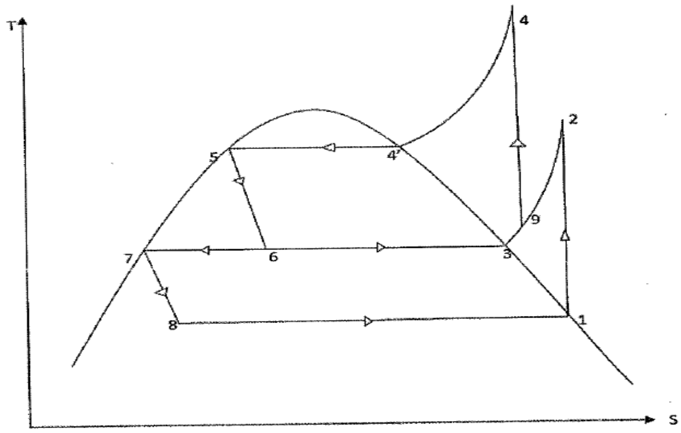

Fig. 4: T-S Diagram of Two-Stage Vapour Compression Cycle

2.3 Thermodynamic Analysis

The thermodynamic analysis of a two-stage vapour compression refrigeration system is performed based on the following assumption:

i. Isenthalpic expansion of refrigerant in the expansion valves,

ii. Negligible changes in kinetic and potential energy.

iii. Negligible pressure and heat loss or gain in the pipe or other components,

iv. The compressor process is irreversible and adiabatic

The following parameters were assumed:

Condenser Temperature (T_C): 50 ˚C is used as the baseline and varied from 30^{0}C °C to 80 ˚C.

Evaporator Temperature (T_E): -20 ˚C is used as the baseline and varied from 0 ˚C to -50 ˚C.

Superheating Temperature (T_{SUP}): 4 ˚C is used as the baseline and varied between 0 ˚C and 20 ˚C.

Sub-cooling Temperature (T_{SUB}): 4°C is used as the baseline and varied between 0°C and 20°C

Mass flow rate of the refrigerant ($\dot{m}$): 0.05kg/sec is used as baseline and varied between 0.01kg/sec and 0.11kg/sec.

Isentropic efficiency (ή) = 0.85

Inter stage pressure, $p_i = \sqrt{P_C \times P_E}$

R-600a, R-600, R-717, and R-290 are employed as the working fluid in both high temperature (I) and low temperature (II) stages. The capacity of evaporator I is defined as:

$$Q_{el} = \dot{m}_I(h_1 - h_8) \qquad 9$$

The capacity of the evaporator (flash chamber) is given by:

$$Q_{ell} = \dot{m}_{II}(h_3 - h_6) \qquad 10$$

The work input into the low-stage compressor is defined by:

$$W_{CI} = \dot{m}_I(h_2 - h_1) \qquad 11$$

The work input into the high-stage compressor can be found by:

$$W_{CII} = \dot{m}_{II}(h_4 - h_3) \quad 12$$

The mass flow rate of the low temperature stage can be obtained by:

$$\dot{m}_I = {Q_{eI}}/{(h_1 - h_8)} \quad 13$$

Mass flow rate of the high-temperature refrigerant:

$$\dot{m}_{II} = {Q_{eII}}/{(h_3 - h_6)} \quad 14$$

The system COP is obtained by:

$$COP = \frac{Q_{eI} + Q_{eII}}{W_{CI} + W_{CII}} \quad 15$$

The mass flow rate of refrigerant through the high-stage compressor can be obtained by taking a control volume that includes the flash tank and high-temperature evaporator. This can be obtained by applying the mass and energy balance.

Mass balance:

$$\dot{m}_5 + \dot{m}_2 = \dot{m}_7 + \dot{m}_3; \dot{m}_5 = \dot{m}_{II} = \dot{m}_3 \& \dot{m}_2 = \dot{m}_I = \dot{m}_7 \quad 16$$

Energy balance:

$$\dot{m}_5 h_5 + \dot{m}_2 h_2 + Q_{eII} = \dot{m}_7 h_7 + \dot{m}_3 h_3 \quad 17$$

2.6 Numerical Solution Procedure

The nonlinear algebraic equations were solved using the Newton–Raphson iterative scheme embedded in EES.

The convergence criteria are determined using max. $| R_i | < 10^{-6}$. Additionally: $| COP^{(k+1)} - COP^{(k)} | < 10^{-7}$ All simulations converged within 8–15 iterations.

2.7 Grid Independence

An additional evaporator temperature sweep test was conducted to validate the test results. For this test, three different ΔT's were used including 5°C, 2°C and 1°C. The maximum variation in COP is <0.5%.

Thus, ΔT = 5°C was adopted as grid-independent.

2.8 Validation Strategy

Model results were validated using:

Carnot COP limit:

$$COP_{Carnot} = \frac{T_E}{T_C - T_E}$$

Ideal isentropic cycle comparison.

Relative deviation:

$$\%Error = \frac{| COP_{ideal} - COP_{sim} |}{COP_{ideal}} \times 100$$

Maximum deviation across cases: <5%

3 Results and Discussion

2.7 Stability and Grid Independence

The impact of the numerical instability on the reliability of the results has to be evaluated and controlled in order to get reliable simulation results. In the following, a grid independence and step-size sensitivity analysis is carried out for the evaporator temperature sweep. The value of the evaporator temperature increment (ΔT) is changed using three step sizes of 5 °C, 2 °C, and 1 °C. Then, the whole thermodynamic model is run again for the new value of the temperature step, and corresponding values of the COP are calculated.

The maximum deviation of the COP between the very small time steps (ΔT=1 °C) and the relatively large time steps (ΔT=5 °C) is less than 0.5% for all refrigerants and system configurations. Hence, the solution is regarded as grid independent with an error of less than 0.5%.

Hence, a ΔT = 5 °C was selected for the parametric study in order to provide the best compromise between precision of calculations and associated computational efforts and cost.

2.8 Validation Strategy

To verify the accuracy and thermodynamic consistency of the developed model, validation was performed using two independent approaches.

First, the simulated results were compared with the theoretical upper performance limit defined by the Carnot refrigeration cycle. The Carnot coefficient of performance is given by:

$$COP_{Carnot} = \frac{T_E}{T_C - T_E}$$

where T_E is the evaporator and $T_C is$ condenser temperatures in Kelvin.

Secondly, the findings from the model were then compared to an ideal vapour compression refrigeration cycle, with isentropic compression being assumed and all other forms of irreversibility neglected. The relative deviation between the ideal COP and the simulated COP was calculated using:

$$\%Error = \frac{| COP_{ideal} - COP_{sim} |}{COP_{ideal}} \times 100$$

Across all refrigerants and operating conditions examined, the maximum deviation was found to be less than 5% (%Error_max < 5%). This level of agreement falls within acceptable engineering tolerance limits and confirms the accuracy of the numerical implementation in Engineering Equation Solver (EES).

Conclusion of Validation

The combined grid-independence assessment and thermodynamic validation demonstrate that the numerical solution is stable, the selected temperature increment (ΔT = 5 °C) is sufficiently accurate for parametric analysis, and the model predictions are physically consistent and thermodynamically sound. Furthermore, the maximum prediction error remains within acceptable engineering limits (<5%), thereby confirming the reliability of the simulation framework used in this study

3.1 Effect of Evaporating Temperature (T_E)

Fig. 5: Effect of evaporating temperature on evaporating pressure, refrigerating effect, compressor work, and coefficient of performance for single stage and two stage vapor compression refrigeration systems for R-600a, R-600, R-717, and R-290 Figure 5 shows that as the evaporating temperature decrease, the refrigerating effect (RE) decreases in both single stage and two stage VCRS systems for all the refrigerants. Also, as the evaporating temperature (TE) decreases, the efficiency of the compressor (QE) also decreases

Figure 1 presents an overview of the influence of evaporator temperature on COP_{REF} of the four working fluids in a single-stage and two-stage vapor compression refrigeration system. Figure 1 Variation of COPREF with evaporator temperature of TE for two-stage and single-stage cycles with different refrigerants. From the Figure 1, it can be seen that when the evaporator temperature of T_E was 0 ºC, the COP_{REF} of R-717 (two-stage) was about 7.636, the maximum coefficient of performance among the other eight cycles. At the same evaporator temperature of T_E was 0 ºC, the maximum COP_{REF} value for the refrigerant R-290 (two-stage) was about 4.701, larger than all other cycles except R-717 (two-stage). There is quite a large difference between the COP_{REF} of R-717 (two-stage) and the other eight cycles. The COP_{REF} of the refrigerant R-600a in the single-stage and two-stage vapor compression refrigeration system at 0 ºC are about 3.682 and 4.009, respectively. At -50 ºC, the COP_{REF} values are only about 1.078 and 1.3, respectively, which is reduced by about 70.72% and 67.57%, respectively. Similarly, the COP_{REF} of R-600 at different evaporator temperatures, at 0 ºC, the COP_{REF} values for single-stage and two-stage vapor compression refrigeration systems are about 3.827 and 4.183, respectively. At -50 ^{0}C, the COP_{REF} values for single-stage and two-stage vapor compression refrigeration systems are only about 1.18 and 1.415, respectively. The reduction of COP_{REF} in the single-stage and two-stage cycles is about 69.17% and 66.17%, respectively. The reduction of COP_{REF} for R-600 in single-stage and two-stage cycles is closer to the COP_{RE}F values of R-600a in single-stage and two-stage cycles.

Experimental results show that for R-290, the COP of single-stage and two-stage VCR system at 0°C is 3.555 and 4.071 respectively, and at -50 °C is 1.078 and 1.436 respectively. Hence the percentage decrease in COP of single-stage VCR system is 69.68% and for two-stage VCR system is 69.45% (Fig.2). For R-717, the COP of single and two-stage VCR systems at 0°C is 3.858 and 7.636 respectively, and at -50°C is 1.278 and 2.424 respectively. Hence the percentage decrease in COP of single-stage VCR system is 66.87%, and the percentage decrease in COP of two-stage VCR system is 68.25%. The decrease in the COP of evaporator temperature in the systems under investigation is almost the same.

Fig. 2: Values of COP for TE = -50 °C. Also from Figure 2, it is found that all COP values for the evaporating temperature of TE is -50 °C are less than unity. Hence, the two-stage VCRS has higher COP values than the other systems except for R-600a (TS) and R-600 (TS), which are almost the same as the corresponding single-stage VCRS. Moreover, it is also concluded from the above Figure that, as the evaporating temperature is decreased, the coefficient of performance of the VCRS also decreases, which is in good agreement with the same conclusion given by Baakeem et al. (2018) and Argawal et al. (2019)

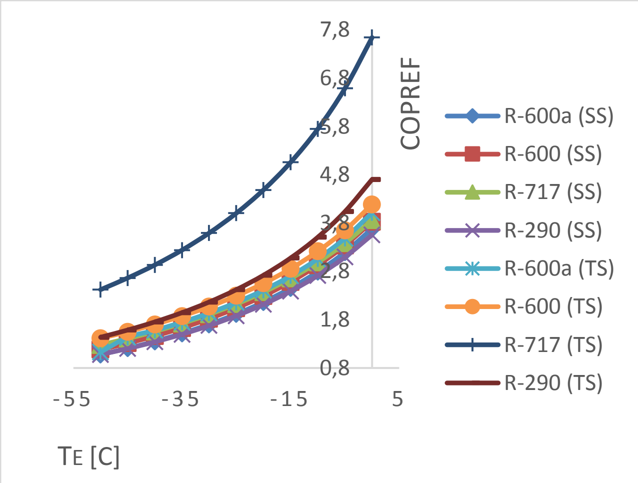

Fig. 5: Comparison of the Effect of TE on COPREF of Single and Two Stage Vapour Compression Refrigeration Systems

3.2 Effect of Condenser Temperature (T_C)

The influence of condenser temperature (T_C) variation on the efficacy of single-stage and two-stage vapor compression systems is depicted in Figure 6. It illustrates the impact of changing condenser temperature, TC, on the coefficient of performance, COP_{REF}. The highest COP for R-717 (two-stage) was 6.708 among the eight test cases. The COPs for R-717, R-600 and R-290 at 30°C are nearly the same. The differences between the COP of R-717 (two-stage) and the other seven cases are very large. R-717 has 46.18% higher efficiency than R-290 (two-stage), which is the second highest efficiency among all the vapor compression refrigeration cycles tested. At 55 °C, the coefficient of performance (COP) of the VCR systems approaches convergence, except R-717, which continues to exhibit a notably superior COP compared to the others. The close correspondence in the performance of R-600a in both single and two-stage VCR systems indicates an overall reduction of 70.20% and 72.43%, respectively, in their COP_{REF} as the temperature changes from 30°C to 80°C. Similarly, a thorough analysis of R-600's behaviour indicates an overall reduction of 67.19% in the Coefficient of Performance (COP) for the single-stage cycle, and a 69.43% overall decrease in the COP for the two-stage cycle. Additionally, when R-717 working fluid is taken into account, the results demonstrate an approximate 59.09% and 58.04% overall reduction in the COP of the single-stage and two-stage VCR systems, respectively. Similarly, there is approximately a 72.73% and a 73.81% reduction in the COP of single-stage and two-stage VCR systems, respectively, when utilizing R-290 as the working fluid. Performance of single stage cycles using R-290 as working fluid at condenser temperature of 80 °C. Performance is not adequate since the single stage COP is below unity. Also the refrigeration coefficient of performance decreases with the increase of condensing temperature. This pattern aligns with those documented in the research conducted by Baakeem et al. (2018).

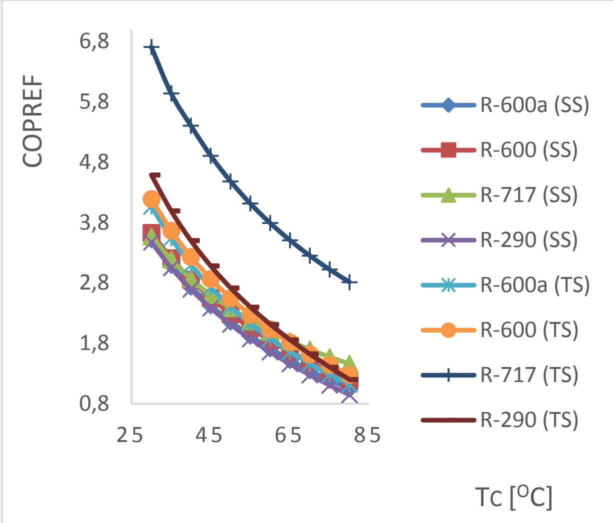

Fig. 6: Effect of TC on COPREF of Single and Two Stage Vapour Compression Refrigeration Systems

3.3 Effect of Superheating Temperature (T_{SUP})

The influence of T_{SUP} on COP_{REF} is depicted in Figure 7. The COP_{REF} of R-600a and R-290 remains nearly constant across the range of superheating temperature, T_{SUP}. For the single- and two-stage vapor compression refrigeration systems, except for the two natural refrigerants R-717 (TS) and R-600a (SS), the Coefficient of Performance, COP, at T_{SUP} = 0°C varied from 2.0 to 2.5. At T_{SUP} = 120°C, the COP for R-600a (SS) and R-600a (TS) were the same, and at maximum COP, T_{SUP} = 200 °C. The performance for R-600a (TS) and R- 600 (TS) was nearly equivalent. As shown in Figure 3, it can be observed that the single vapor compression refrigeration systems maintain a nearly constant COP, despite increases in superheating temperature. However, the COP of the two-stage counterparts rises as T_{SUP} increases. To achieve an improved coefficient of performance in a two-stage VCRS, it is necessary to sustain a higher superheating temperature, T_{SUP}.

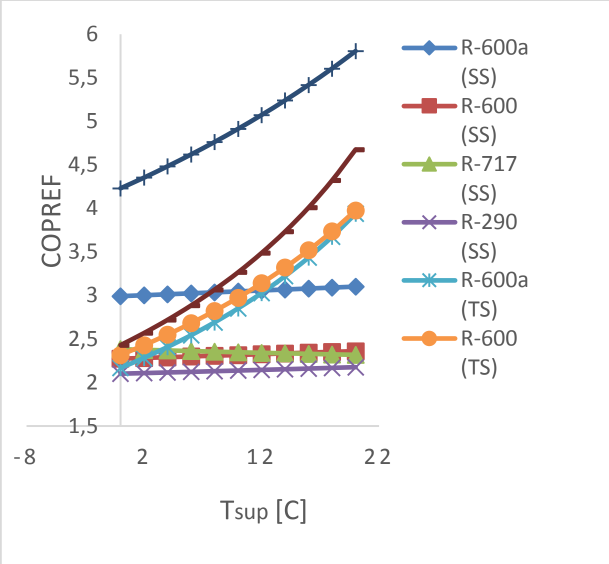

Fig. 7: Effect of TSUP [°C] on COPREF using different refrigerants

3.4 Effect of Subcooling Temperature (T_{SUB})

Figure 8 illustrates the impact of varying the subcooling temperature, T_{SUB}, on the refrigeration COP. It can be observed that as T_{SUB} increases, the COP correspondingly rises, as anticipated. The graph indicates that, for all cycles except R-717(SS), the relationship between the COP and subcooling temperature is linear. As T_{SUB} ranges from 00C to 200C, the COP of the cycles exhibits a consistent increase, with the overall enhancement in R-717 (TS) COP amounting to 10.06%. It indicates that the COP does not experience a substantial increase. At T_{SUB} = 0^0C, the vapor compression refrigeration systems exhibit COP values ranging from 2 to 2.5, except R-290 (TS) and R-717, which have respective COP values of 2.593 and 4.394. At 100°C, R-600(SS) and R-717(SS) exhibit nearly identical coefficients of performance. At 200°C, the coefficient of performance of R-290 and R-717 is nearly identical.

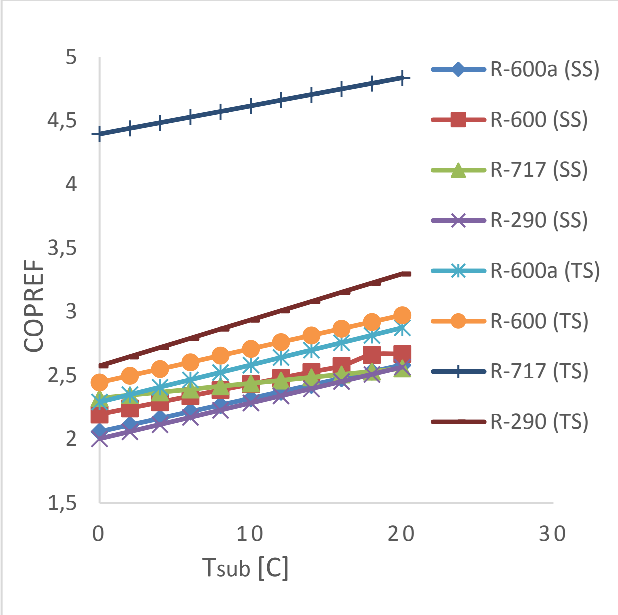

Fig. 8: Effect of TSUB [°C] on COPREF using different refrigerants

3.5 Effect of Refrigerant Mass Flow Rate ($\dot{M}$)

Figure 9 presents the effect of mass flow rate on COP. The highest COP is observed in the case of R-717 (TS), and the second highest COP is observed in the case of R-290 (TS). In the case of single-stage and two-stage VCR systems, the COP values for R-717 refrigerant are 2.368 and 4.483, respectively. Although the COP values are almost constant for all the mass flow rates, the COP is much higher in this condition. The COP values for R-290 in the case of single-stage and two-stage VCR systems are almost 2.115 and 2. Compared with single-stage VCR, the COPs of two-stage VCR are about 28.55% higher. The constant COPs for single-stage and two-stage VCR with R-600a are 2.163 and 2.403, respectively. The enhancement in COP in these two systems is not much. In the case of R-600 as the working fluid, the COP values for the single-stage and two-stage VCR systems are almost 2.29 and 2.55, respectively.

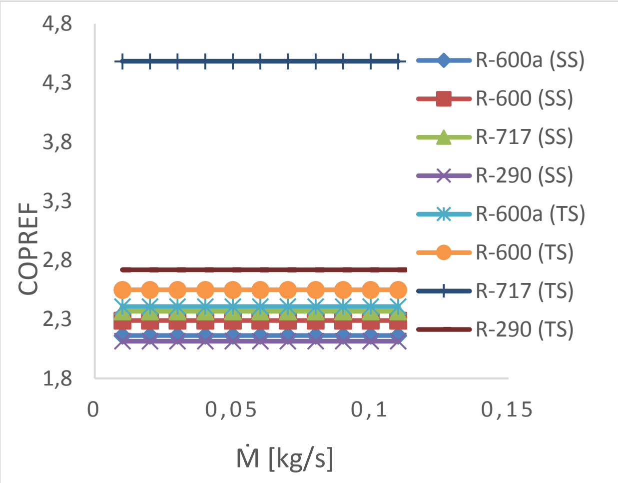

Fig. 9: Effect of Ṁ [kg/s] on COPREF using different refrigerants

4 Conclusion

The following conclusions were drawn from the evaluation of single-stage and two-stage vapour compression refrigeration systems: a simulation-based study for humid climate applications using eco-friendly refrigerants such as isobutane, n-butane, propane, and ammonia. Performance parameters are the coefficient of performance (COP), refrigeration effect, compressor work, and their sensitivity. The effects of varying parameters on the performance of VCR systems are examined. The influences of system configuration and operating variables on refrigeration efficiency are clearly brought out.

The effect of VCR system configuration on the performance of ammonia refrigeration systems is studied. A two-stage VCR system is compared with a single-stage VCR system for the two working fluids R-717 and R-290. Results show that the performance of the two-stage VCR system is higher than that of the single-stage VCR system for all cases. R-717 gives the maximum value of COP among the three refrigerants, hence it is recommended for the VCR system. R-290 gives better performance with a two-stage VCR system than with a single-stage VCR system over the range of operating conditions considered.

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