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Article

Impact of Wire Condenser Surface Area on Thermal performance of Household Refrigerators Using R134a and R600a under Hot Climates Conditions

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Abstract: This research is an experimental study that will examine the thermal performance of a domestic refrigerator used three-wire tube condenser setup with varying fin numbers. Under Natural Convection, the efficiency of condensers is calculated. They work with two refrigerants: R134a and R600a. The condensers were tested at ambient temperatures of 20°C, 30°C, and 40°C. The performance of the refrigeration system was evaluated based on refrigeration effect, compressor work, pressure ratio, and compressor COP. The results have demonstrated that the cooling performance of the refrigeration systems decreases and the COP decreases with increasing ambient temperature. There is a higher compressor load and condensing pressure for both refrigerants. Among the condensers, the conventional condenser had the highest COP in all the cases. A larger effective value of k resulted in values and better thermal performance. heat transfer area. R600a showed the best results under moderate ambient conditions, and R134a operated more smoothly at high-temperature conditions. Overall, it is seen that the surface area of the condenser has a considerable effect on the thermal efficiency of the refrigerator in a hot climate.

Keywords: Wire & tube condenser, Effect of ambient temperature, , Condenser Fins, R-600a, Condenser performance, domestic refrigerator, condenser surface areas, home refrigeration.

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Introduction

The study of the energy efficiency of domestic refrigeration systems has become an important objective in order to meet the growing energy demand and address environmental concerns [1]. The thermal performance and energy efficiency of refrigeration systems are among the most important issues in the mechanical and thermal engineering field and one of the most energy-intensive household appliances. The type of refrigerant and the design of the major components of the refrigeration system significantly affect the performance of vapor-compression systems, especially the condenser, where heat is rejected to the environment. In recent years, the move has been toward environmentally-friendly refrigerants with low ODP (Ozone Depletion Potential) and GWP (Global Warming Potential). In this context, R600a (isobutane) has been extensively studied as an alternative to the conventional refrigerants like R134a, because of its proper thermodynamic properties and low environmental impact. A number of experimental investigations have been conducted to measure the performance of other refrigerants in domestic refrigerating units. In addition to the improvement in energy efficiency with R600a under some operating conditions, as mentioned in the above studies [2],[3] and [4] R600a can achieve higher coefficients of performance (COP) it also has the potential to achieve higher COP and lower .energy consumption than R134a.

On the other hand, other investigations [5], [6] indicated that the performance of R600a may be comparable to or slightly lower than that of R134a, depending on system configuration, compressor capacity, and operating conditions, although it requires a significantly lower refrigerant charge. Furthermore, ambient conditions play a crucial role in determining system performance, as demonstrated in a study that showed that increasing ambient temperature leads to higher power consumption and reduced cooling capacity, highlighting the importance of evaluating refrigeration systems under different environmental conditions. The current research into the use of low-GWP refrigerants and system design parameters has been widely conducted. As an example, [7], [8] studied the use of R600a for replacing R134a and found that the system efficiency is the same with a lower environmental impact. Likewise, the performance of domestic refrigerators operating with R600a with different ambient temperatures was investigated by [9] and it was found that R600a is highly energy efficient at moderate and high ambient temperatures. A study, [10] investigated he effect of the design parameters of the condenser on the heat rejection rate and concluded that the heat rejection rate gets enhanced significantly with an increase in surface area. In Study [11] the authors focus on the natural convection wire-tube condenser and confirm the effect of fin configuration on the thermal performance of the condenser.Furthermore, [12] tested the performance of a refrigeration system with high ambient temperature and highlighted the reduction of COP along with higher temperature.

Finally, the combined effect of refrigerant type and heat exchanger design was investigated in [13], and it was found that the optimization of the heat exchanger system should be done simultaneously, considering both the refrigerant type and the heat exchanger design. Although an effort has been made in these previous studies, the majority have focused on refrigerant comparison and optimization of refrigerant charge, with little study on the effect of a condenser design, especially for systems where a condenser is used with a wire-type condenser and natural convection. The effect of fin density (which is directly related to the effective heat transfer surface area) has not been studied sufficiently for domestic refrigeration applications. Furthermore, the combined effect of condenser surface characteristics and the refrigerant type is not covered systematically in the literature. Finally, the combined effect of refrigerant type and heat ex changer design was investigated in [13] , and it was found that the optimization of the heat exchanger system should be done simultaneously, considering both the refrigerant type and the heat exchanger design. Although an effort has been made in these previous studies, the majority have focused on refrigerant comparison and optimization of refrigerant charge, with little study on the effect of a condenser design, especially for systems where a condenser is used with a wire-type condenser and natural convection. The effect of fin density (which is directly related to the effective heat transfer surface area) has not been studied sufficiently for domestic refrigeration applications. Furthermore, the combined effect of condenser surface characteristics and the refrigerant type is not covered systematically in the literature. Besides, the performance of the R600a and R134a refrigerants has not been sufficiently investigated in hot and semi-hot climates like Iraq, as the ambient temperature plays a significant role in the efficiency of refrigeration systems. In the present, the variation of condenser surface area (no fins, medium fins, and standard fins) is of special significance and is investigated in detail. Thus, the work in this study experimentally compares the performance of a domestic refrigerator using R600a and R134a with different ambient temperatures of 20°C, 30°C, and 40°C, with the use of three different wire-tube condensers with varying fin, to analyze the effect of condenser surface area on system performance. The performance of the refrigerator is evaluated in terms of COP, cooling capacity, consumption of the compressor power, and heat rejection characteristics, and to find the best condenser configuration for hot climate operation. The current study is mainly focused on the study of the effect of the condenser surface area, which is varied in the study by altering the fin in the following three configurations by using a wire-tube condenser. The system performance is evaluated in terms of coefficient of performance (COP), cooling capacity, power consumption, and heat rejection characteristics, with the objective of identifying the optimal combination of refrigerant type and condenser design that ensures improved energy efficiency and thermal performance, particularly under high ambient temperature conditions.

Materials and Methods

The experiential work was done by using a home refrigerator that works with a vapor compression refrigeration cycle. The study was conducted under controlled conditions to study the comparison of the two refrigerants R600 and R134a. The experiments were carried out on the same refrigerator and under the same conditions to ensure accurate results. Three different wire condensers were used to study the effect of condenser surface area on the system's cooling cycle performance. A condenser with no fins, another with half fins, and a third with full fins (traditional condenser) were used to cover three different areas of the study. The experiments were carried out in a insulation closed test chamber to maintain the ambient temperature of the condenser accurately, where electric heaters were used to raise the temperature inside the chamber, and an air conditioning system was used to cool the chamber. Three ambient temperatures (20°C, 30°C, and 40°C) were evaluated as moderate to hotter conditions for the experiments. The top-loading, high-accuracy digital balance (±0.00002 g) was used to obtain the precise mass of charged refrigerants (R600a and R134a) in the experimental system so that each charge quantity could be repeated. The charges were adjusted for each refrigerant to achieve steady operations and best performance. Data were recorded only after steady-state operation of the system was achieved to insure measurement accuracy. Measurements of temperature and pressure were recorded at t the main points along the refrigeration cycle, including before and after the compressor, condenser inlet and outlet, expansion device, and evaporator exit. K-type thermocouples were directly attached to the copper tube walls with thermal grease to minimize thermal contact resistance. Tape was used to insulate the sensors in order to mitigate the effect of ambient air convection. Two Bordon pressure gauges with an accuracy of ±0.05 bar were installed to measure the high- and low-side pressures, respectively. The electrical energy consumption of the compressor was measured by using A power meter. In this experiment, all the measurements were performed at every 10 min interval in order to get the transient or steady-state behavior. This to insure an accurate monitoring of the system performance and stability for each test condition.

Table 1. Charge Mass of Refrigerants.

Refrigerant	Charge Mass
R134a	300 g
R600a	80 g

Condenser Specifications

The designed and structural details of the three wire-tube condensers used in experimental work are recorded in Table 1.

Table 2. Specifications of Wire-and-Tube Condensers.

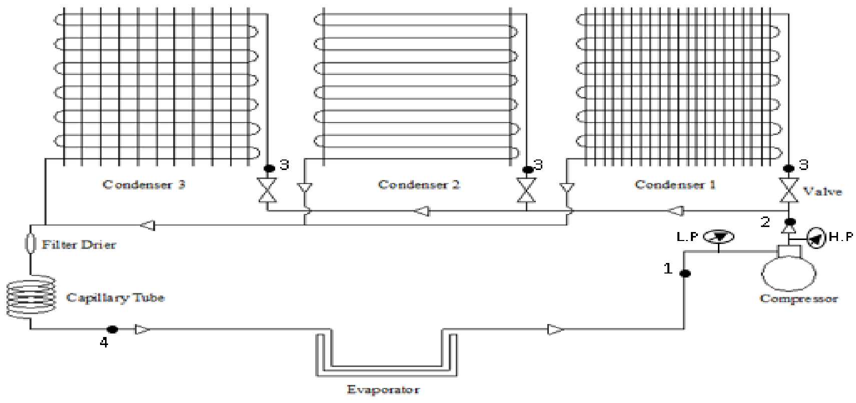
Description	**Wire- cond. C1**	**Wire- cond. C2**	**Wire- cond. C3**
Outer Diameter of Tube	4.8 mm	4.8 mm	4.8 mm
Inter Diameter of Tube	3.2 mm	3.2 mm	3.2 mm
No. of Tube	16	20	20
Space of tube	50 mm	50 mm	50 mm
No. of wires	---	54*2	27*2
Space of Wire	---	7 mm	14 mm
Wire diameter	1.2 mm	1.2 mm	1.2 mm
Length of wire	950 mm	950 mm	950 mm

Experimental Cases

Practical experiments were recorded to represent each case, including the type of fluid, ambient temperature, type of condenser, and a symbolic representation for each experiment, as recorded in Table 2.

Table 3. Experimental Test Matrix.

Refrigerant	Ambient Temperature	Condenser Type	Case Symbol
R134a	20°C	Standard	CaseA_T1C1
R134a	20°C	Medium Fins	CaseA_T1C2
R134a	20°C	No Fins	CaseA_T1C3
R134a	30°C	Standard	CaseA_T2C1
R134a	30°C	Medium Fins	CaseA_T2C2
R134a	30°C	No Fins	CaseA_T2C3
R134a	40°C	Standard	CaseA_T3C1
R134a	40°C	Medium Fins	CaseA_T3C2
R134a	40°C	No Fins	CaseA_T3C3
R600a	20°C	Standard	CaseB_T1C1
R600a	20°C	Medium Fins	CaseB_T1C2
R600a	20°C	No Fins	CaseB_T1C3
R600a	30°C	Standard	CaseB_T2C1
R600a	30°C	Medium Fins	CaseB_T2C2
R600a	30°C	No Fins	CaseB_T2C3
R600a	40°C	Standard	CaseB_T3C1
R600a	40°C	Medium Fins	CaseB_T3C2
R600a	40°C	No Fins	CaseB_T3C3



Stats		Process	
1	$P_1,\ h_1\ ,\ S_1$	State 1 – State 2	Compression work
2	$P_2\ ,\ h_2\ ,\ S_1 = S_2$	State 2 – State 3	Condensation
3	$P_3 = P_2, h_3 = h_4$	State 3 – State 4	Throttling
4	$P_4 = P_1\ , h_4$	State 4 – State 1	evaporation

Performance Evaluation

Thermal performance calculation of a refrigeration system is done using conventional thermodynamic relations. Electrical power for the compressor was determined from voltage and current measurements during tests, while the mechanical work of the compressor was typically computed as a change in enthalpy across the compressor. From the difference in enthalpy through the evaporator and condenser, the required heat rejection and cooling load. By defining the ratio of cooling effect to compressor work as the Coefficient of Performance COP;, the pressure ratio was defined as the ratio of discharge pressure and suction pressure; again, check that value from 0 & 3. These parameters were used to compare the performance of both refrigerants in different condenser configurations and ambient conditions.

Data Analysis

The experimental data were subsequently analyzed to quantify the effect of different refrigerants, ambient temperatures, and condenser fin densities on the performance of the system. Over a period of time, it was ensured that the system remained stable, and therefore average values were used for comparisons. The analysis was conducted to find the best operational configuration that provides both high thermal efficiency and performance at ambient temperature conditions of 38 °C

Performance Equations

The thermodynamic performance of the refrigeration system was evaluated using standard vapor compression cycle relations. The compressor specific work was calculated from the enthalpy difference across the compressor, while the refrigeration effect and condenser heat rejection were determined from the enthalpy differences across the evaporator and condenser, respectively. The compressor specific work is given by:**[14] [15] [16]:**

Table 4. Thermodynamic State Points of the Refrigeration System.

State Point	Description
1	Compressor inlet
2	Compressor outlet
3	Condenser outlet
4	Expansion device outlet

Compressor work (per unit mass):

$$W_c = h_2 - h_1 \qquad (1)$$

Refrigeration effect (cooling capacity per unit mass):

$$RE = h_1 - h_4 \qquad (4)$$

Heat rejected in the condenser:

$$qc = h_2 - h_3 \qquad (3)$$

Coefficient of Performance (COP):

$$COPR = \frac{h_1 - h_4}{h_2 - h_1} \qquad (5)$$

Pressure ratio:

$$Pr = \frac{P_h}{P_L} \qquad (6)$$

The thermodynamic properties of the refrigerants were obtained using standard refrigerant property tables/software based on measured pressure and temperature values.

Table 5. Data collection.

Symbol	Units	Calculation	Description
WIC	W	-	Electrical work energy of compressor
Wc	kj/kg	-	Mechanical work of compressor
RE	kj/kg	-	heat absorption from evaporator
qc	kj/kg	-	heat rejection from the condenser
C.O.PR	Unitless	-	Coefficient of performance Ref.
Pr	Unitless	-	Pressure Ratio

Figure 1. Room test & Divese test.

Result and Discussion

The influence of ambient temperature on the performance of vapor-compression refrigeration systems is an important consideration in system design and thermal performance evaluation. With increasing ambient temperatures, particularly in hot-climate regions such as the Middle East, evaluating the thermal performance of alternative refrigerants has become increasingly important. This includes experimentally comparing their thermal behavior and energy performance with the conventional refrigerant R134a under controlled operating conditions. The experiment was conducted at three ambient temperatures to gain a comprehensive understanding of thermal performance under different external conditions. This approach establishes a basis for comparison and evaluates the influence of condenser fins on refrigerator performance using two refrigerants: R134a and R600a. Experiments were initially conducted at ambient temperatures of 20°C, 30°C, and 40°C using the standard wire-and-tube condenser configuration (CaseA1T1C1). The temperature was recorded after every 10 minutes of operation for a total operation time of 80 min, giving enough time for the system to reach steady-state operating conditions. The refrigerator was then switched off, and the cycle was changed to a second wire-tube condenser (CaseA1T1C2). The same ambient temperature conditions were maintained for data collection, and the second experimental case was performed. Based on these results, the refrigeration system was then altered to add a third configuration of wire-tube condenser (CaseA1T1C3), and data were recorded. The experiments were carried out using R134a and R600A as refrigerants. Refrigerant enthalpy values were obtained from the pressure-enthalpy (P-h) diagram using measured pressure and temperature data, based on our previously mentioned temperatures and conditions for R134a and R600a. Values for the inlet and outlet temperatures of the compressor, condenser, evaporator, and high and low pressure were recorded. Figure 3 presents the P-h diagram of the refrigeration cycle used for the thermodynamic analysis. Experimental uncertainties were minimized by maintaining controlled ambient conditions throughout all test cases. The temperature sensors used were calibrated prior to the experiments, with an accuracy of ±0.5°C.

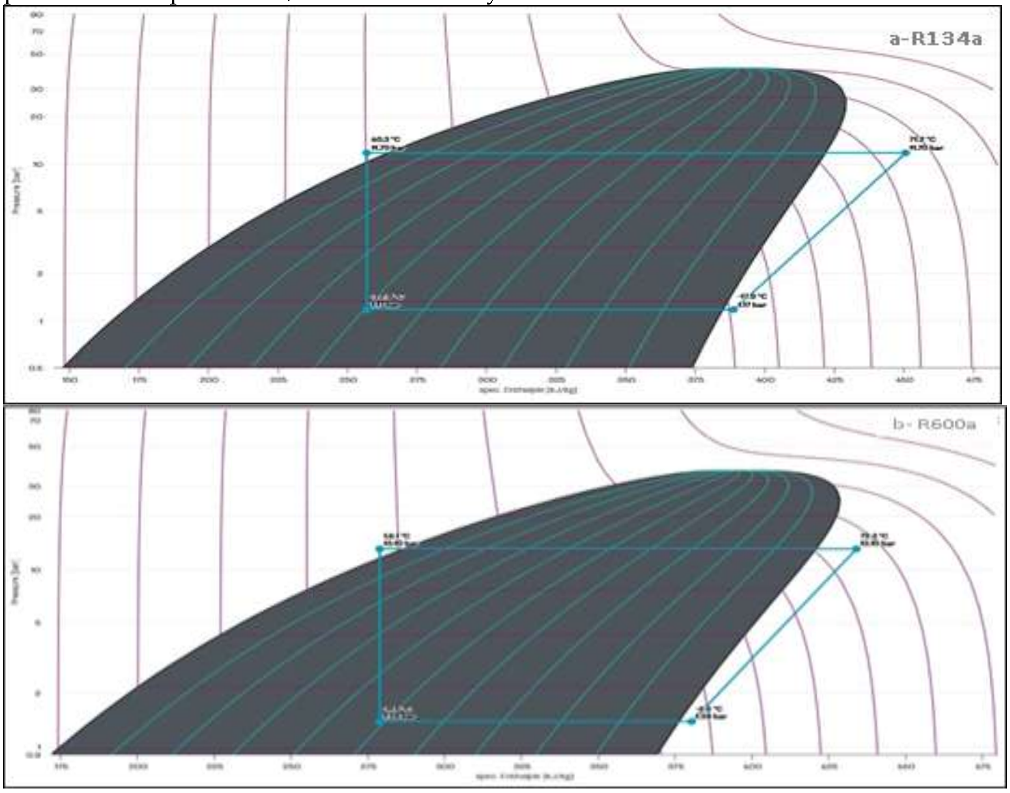


Figure 2. The P-h diagram of Vapor compression Refrigeration Cycle (a-R134a & b-R600a).

The thermal performance of the modified condenser configurations was compared with that of the conventional wire-and-tube condenser. In this context, Wc refers to the compressor work, RE represents the heat absorbed in the evaporator, and qc indicates the heat rejected in the condenser. The thermodynamic states corresponding to all experimental configurations are illustrated on the P-h diagram.

freezing and fresh food temperatures

As the ambient temperature increases, the refrigeration system performance decreases, and all condensers configurations (C1, C2, and C3) require longer operating times to approach stable operating conditions. Figure 4 illustrates the effect of ambient temperature on the freezing temperature behavior of the three condenser configurations using R134a refrigerant, In this configuration, C1 represents the standard finned condenser, C2 represents the medium-fin condenser, and C3 represents the finless condenser. The results show that cooling performance reduces significantly with ambient temperature, as shown by the reduced rate of decrease in temperatures or high final freezing temperatures. The most significant influence of ambient temperature is during the first 20 minutes of operation. This is, of course a time when the effectiveness of the condenser heat rejection process is primarily influenced by the surrounding air temperature, resulting in large variances in system performance. Cooling rates and freezing temperatures also differed most at the lowest ambient temperature of 20°C. While similar patterns were reported with higher ambient temperatures (30°C and 40°C), which led to slower cooling rates during operation and thus a higher final temperature.

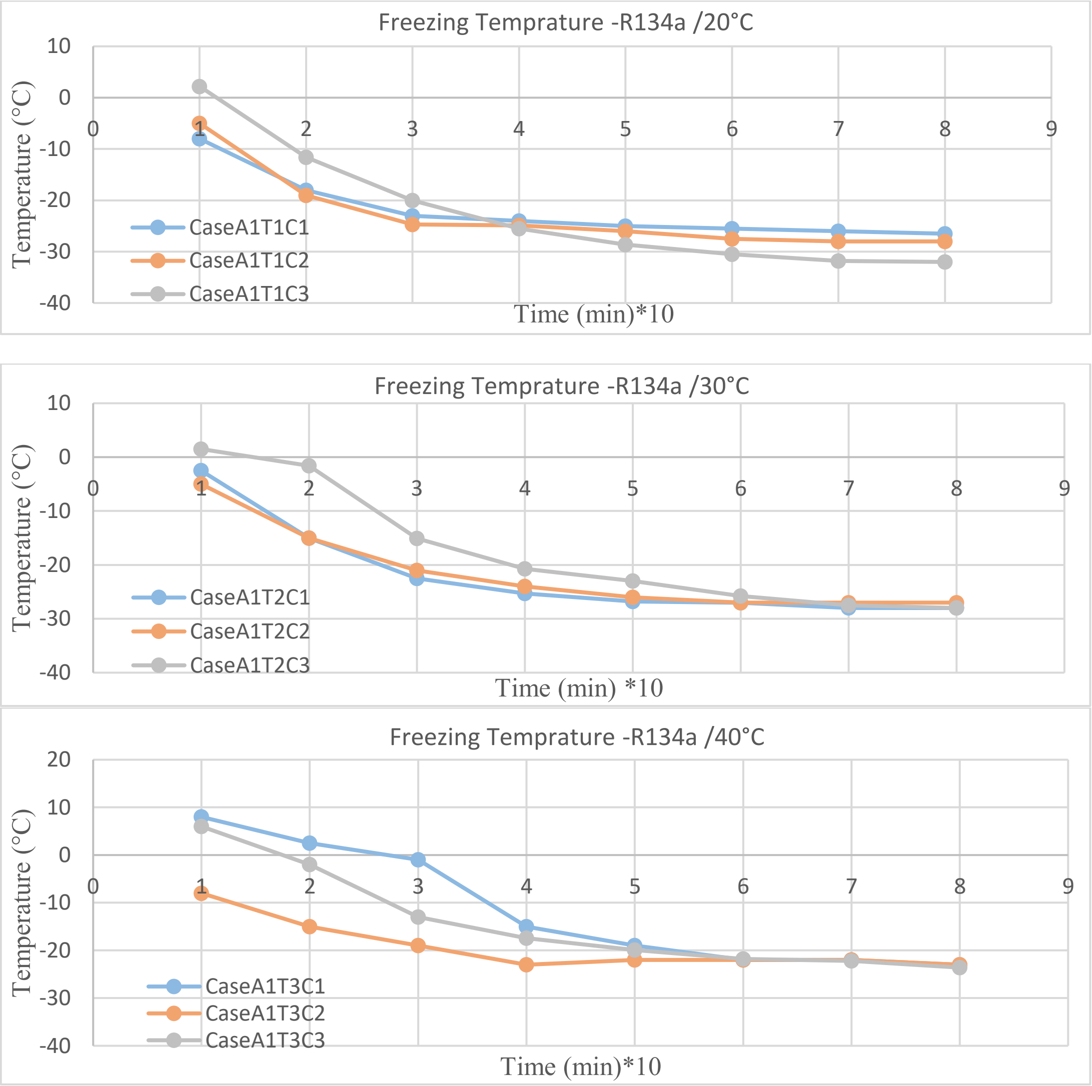
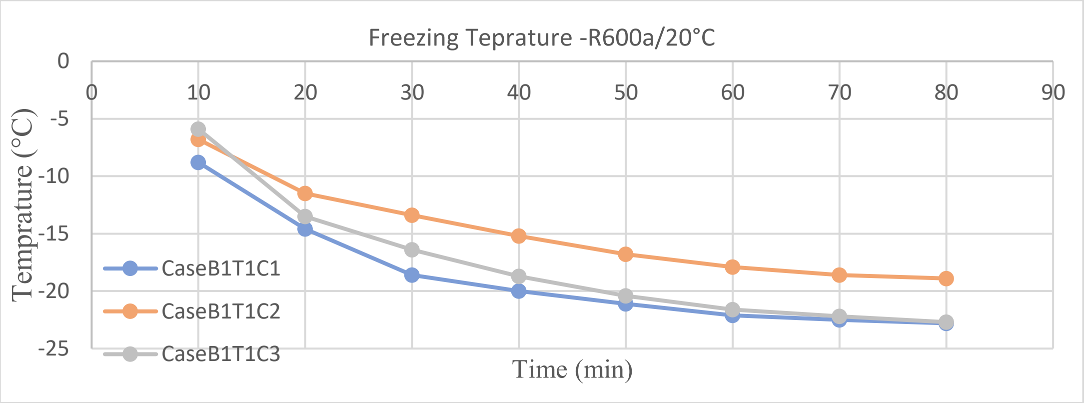
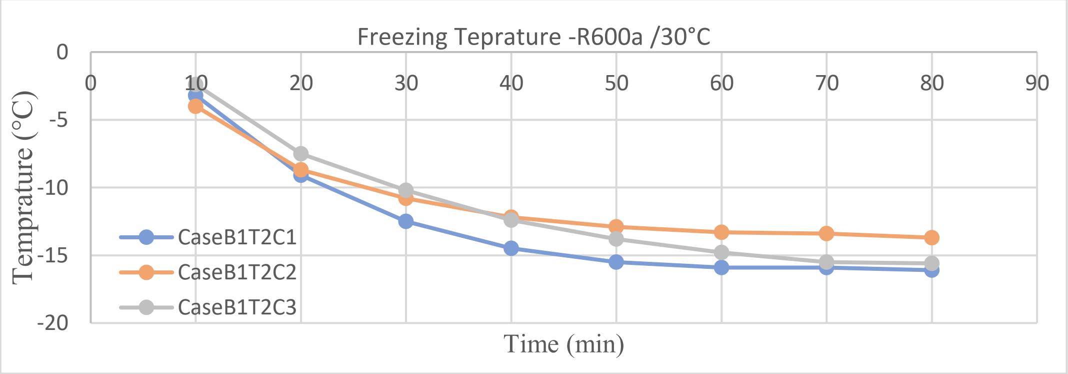


Figure 3. Freezing Temperature with time for R134a for 3mode of ambient temperatures.

Figure 5 show after the After a first 30 mints of operation, the freezing temp drop is relatively fast regardless of the condenser. This behavior indicates that the refrigeration system has a high initial capacity for cooling. The temperature curves began to stabilize after 60 min, showing that the refrigeration system was entering a steady state. R600a has the best cooling effect at moderate ambient temperatures (20°C); however, when under a high ambient temperature of 40°C, R600a achieved poor thermal performance owing to higher condensing pressure increasing the condensing temperature and compressor load, where condensation temperature curves corresponding to condenser configurations (C1, C2 and C3) diverge accordingly. This behaviour reflects lower thermal stability and poor cooling performance in high ambient temperature scenarios.

The ambient temperature has a considerable impact on the fresh food temperature. Figure 6 shows the fresh food compartment temperature changes with ambient temperature using the R134a refrigerant. At a temperature of 20°C, it dropped to 9°C, at 30°C, the exhaled air reached 13°C and then cooled up to 24° C. This behavior can be explained by the time lag of cooling in the fresh food compartment and affects the refrigerant flow directions between freezer and fresh food evaporator sections, as a result of which it takes longer to lower its temperature during the first 30minit. This effect was particularly noticeable at 30°C and 40°C. Figure (7) further confirms the significant influence of ambient temperature on fresh food compartment cooling performance when using R600a refrigerant. The best cooling performance was achieved at 20°C, whereas increasing ambient temperature reduced the rate of heat rejection in the condenser and degraded the cooling performance, with the temperature curves approaching stable conditions after approximately 60–80 minutes of operation. The lowest compartment temperatures were obtained at 20°C ambient temperature, while higher final temperatures of approximately 8°C and 13°C were recorded at ambient temperatures of 30°C and 40°C, respectively.





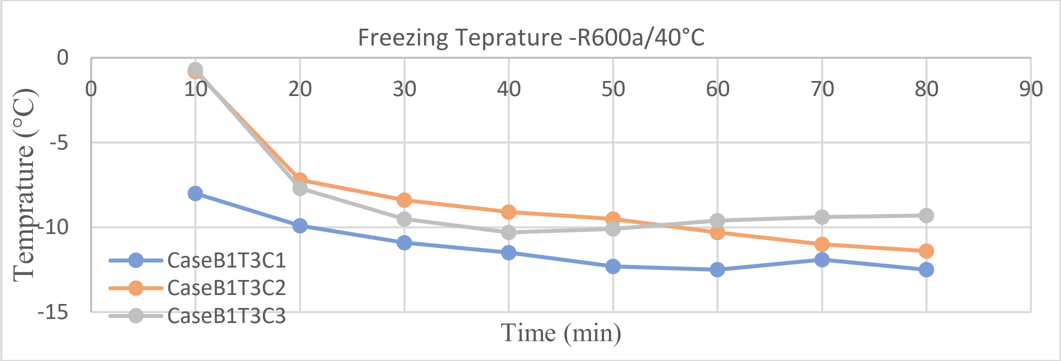


Figure 4. Freezing Temperature with time for R600a for 3mode of ambient temperatures.

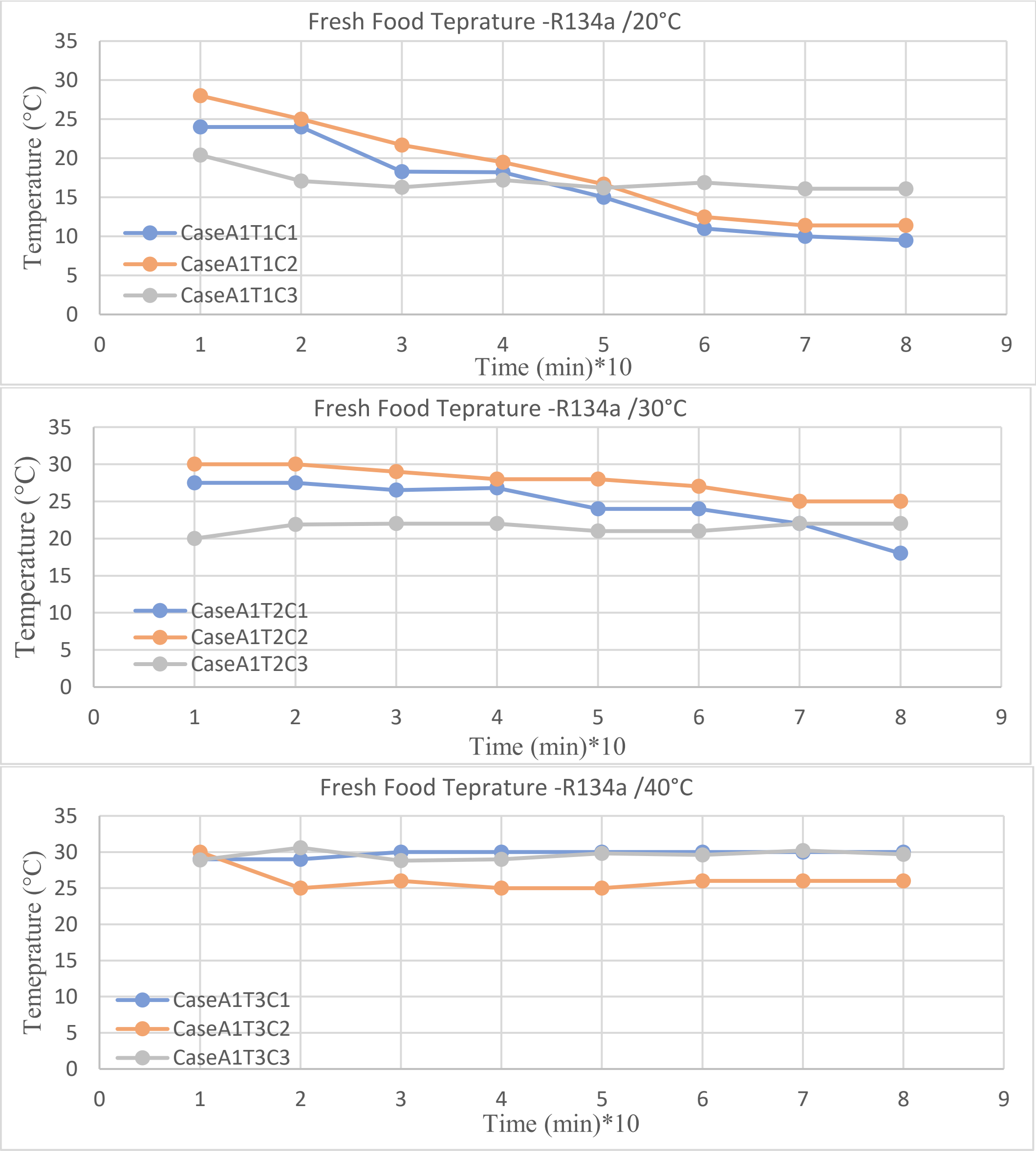


Figure 5. Fresh Food Temperature with time for R134a for 3mode of ambient temperatures.

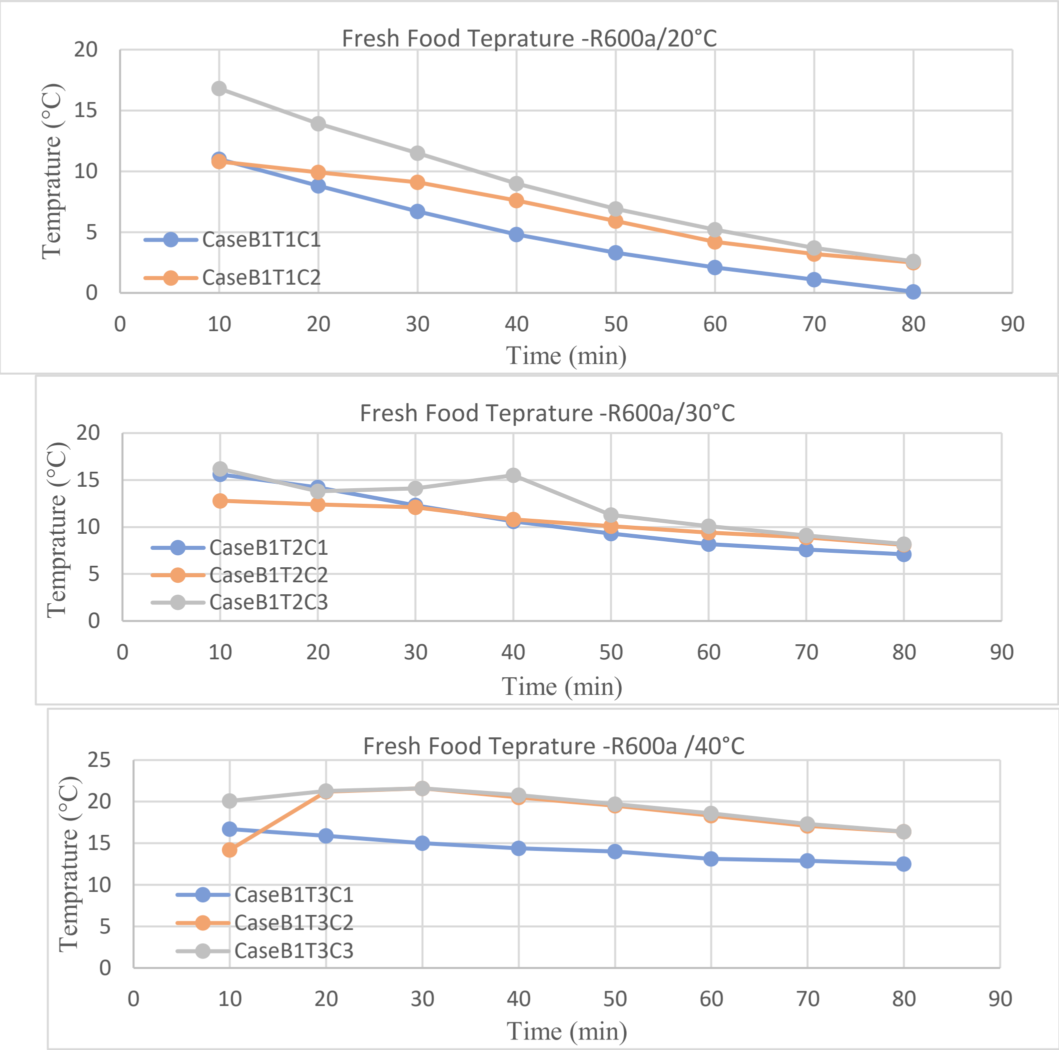


Figure 6. Fresh Food Temperature with time for R600a for 3mode of ambient temperatures.

Refrigeration effect and condensing duty

Figure 8, illustrates the variation of the refrigeration effect (RE) for the three condenser configurations (C1, C2, and C3) using R134a refrigerant. At an ambient temperature of 20°C, the refrigeration effect remained relatively stable around 170 kJ/kg for all condenser configurations throughout most of the operating period, At higher ambient temperatures, noticeable differences between the condenser configurations began to appear after approximately 40 minutes of operation, particularly in the refrigeration effect values. The reduction in refrigeration effect at elevated ambient temperatures is mainly associated with the increase in condensing pressure and compressor load, which reduces the evaporator cooling capacity.

Figure 9 presents the refrigeration effect behavior of R600a for the three condenser configurations. Condenser configuration C1 generally exhibited slightly higher refrigeration effect values compared with configurations C2 and C3 under different ambient temperature conditions. The results show that R600a is more sensitive to ambient temperature changes compared to R134a, especially in the initial period. This behavior is due to the thermodynamic properties of R600a, with changes in condensing temperature under hot ambient conditions having a greater effect. Figure 10 shows the effect of ambient temperature on the condenser heat rejection rate for R134a refrigerant, The condenser heat rejection rate remains relatively stable throughout the operating period, Minor variations were observed between condenser configurations C2 and C3, particularly at elevated ambient temperatures. A larger surface area allows the heat in a vaporized refrigerant to be removed quickly into the air surrounding it, allowing for a much more stable performance of the condenser. As illustrated in Fig. 11, the condenser heat rejection behaviour of R600a is fairly constant at the ambient temperatures of 20°C and 30°C.

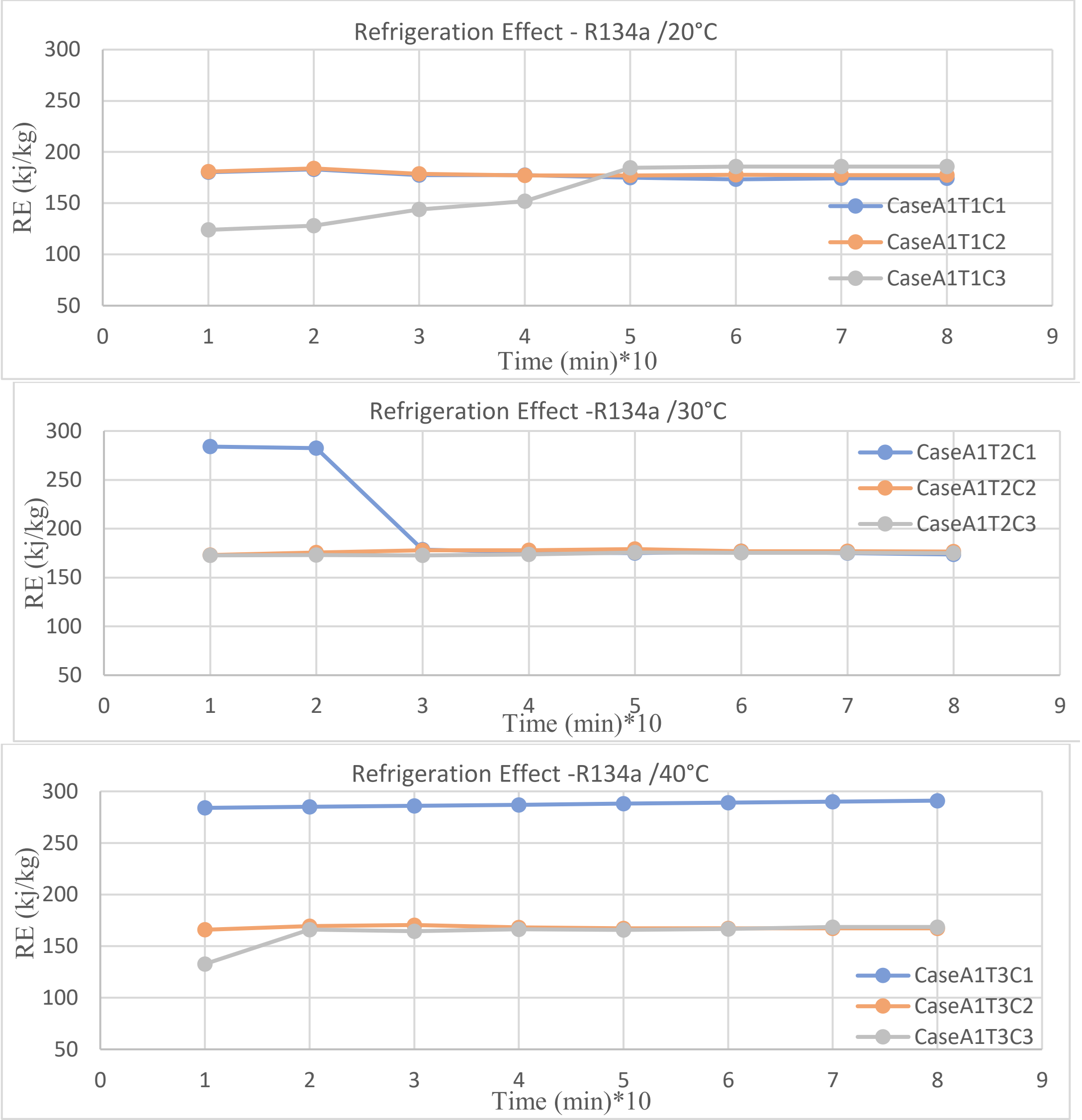
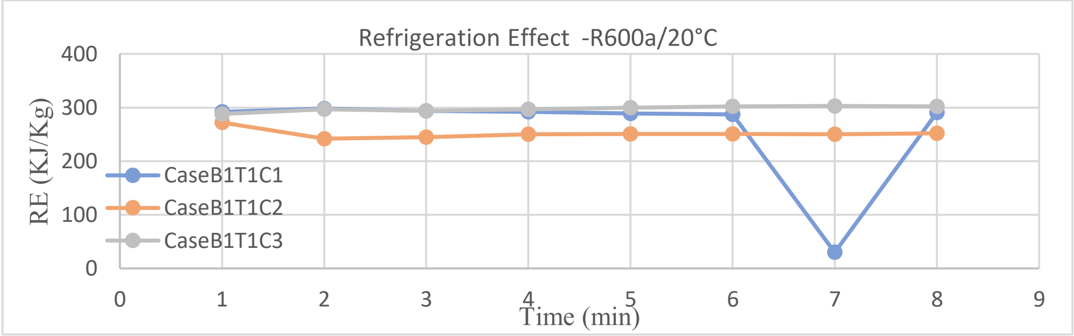


Figure 7. Refrigeration effect (RE) with time for R600a for 3mode of ambient temperatures.



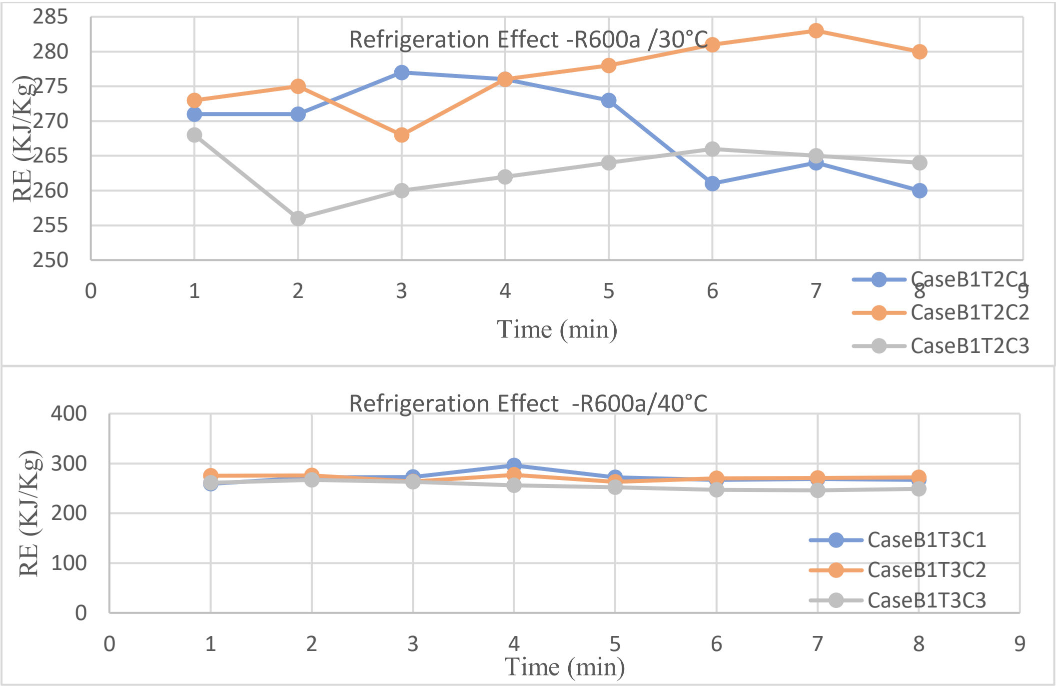


Figure 8. Refrigeration effect (RE) with time for R600a for 3mode of ambient temperatures.

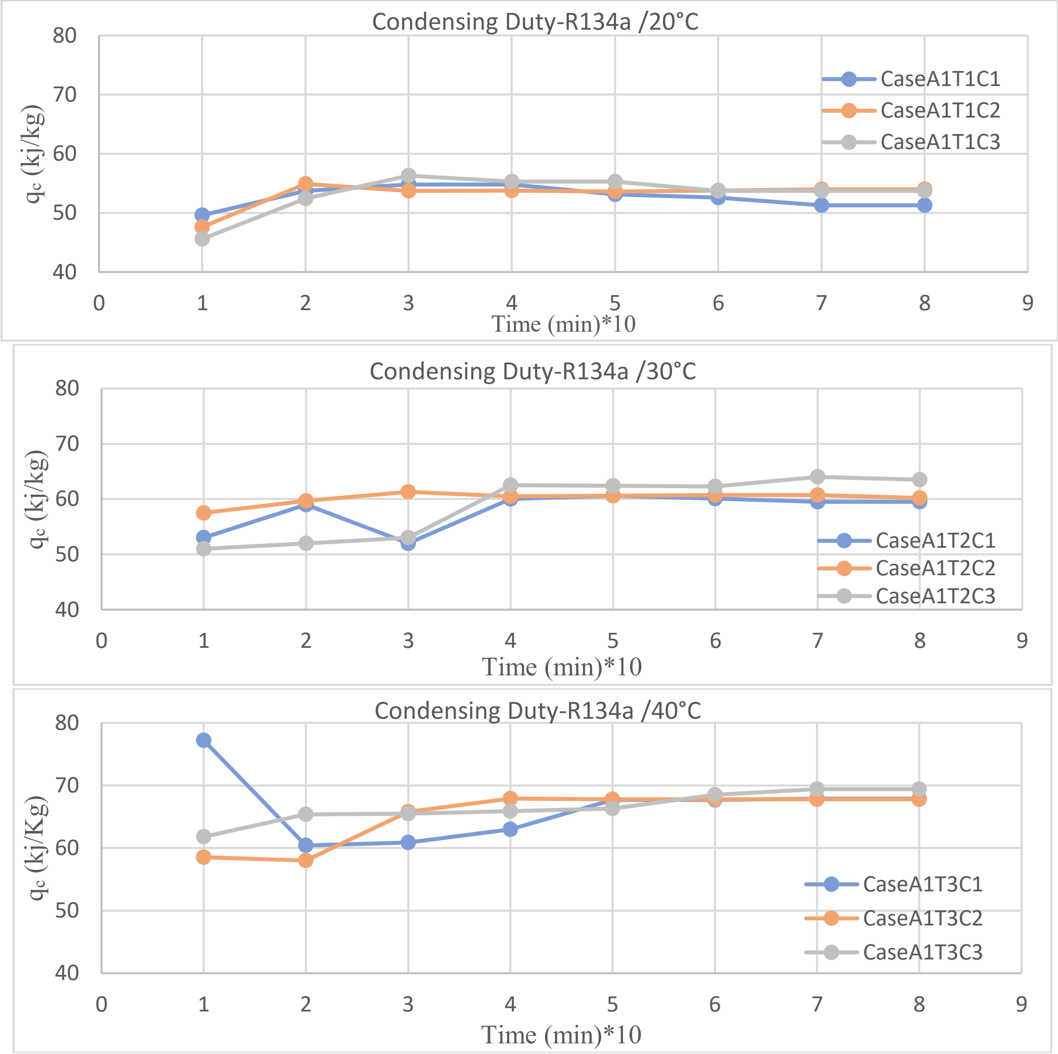


Figure 9. 3mode of ambient temperatures Condensing Duty (qc) with time for R134a.

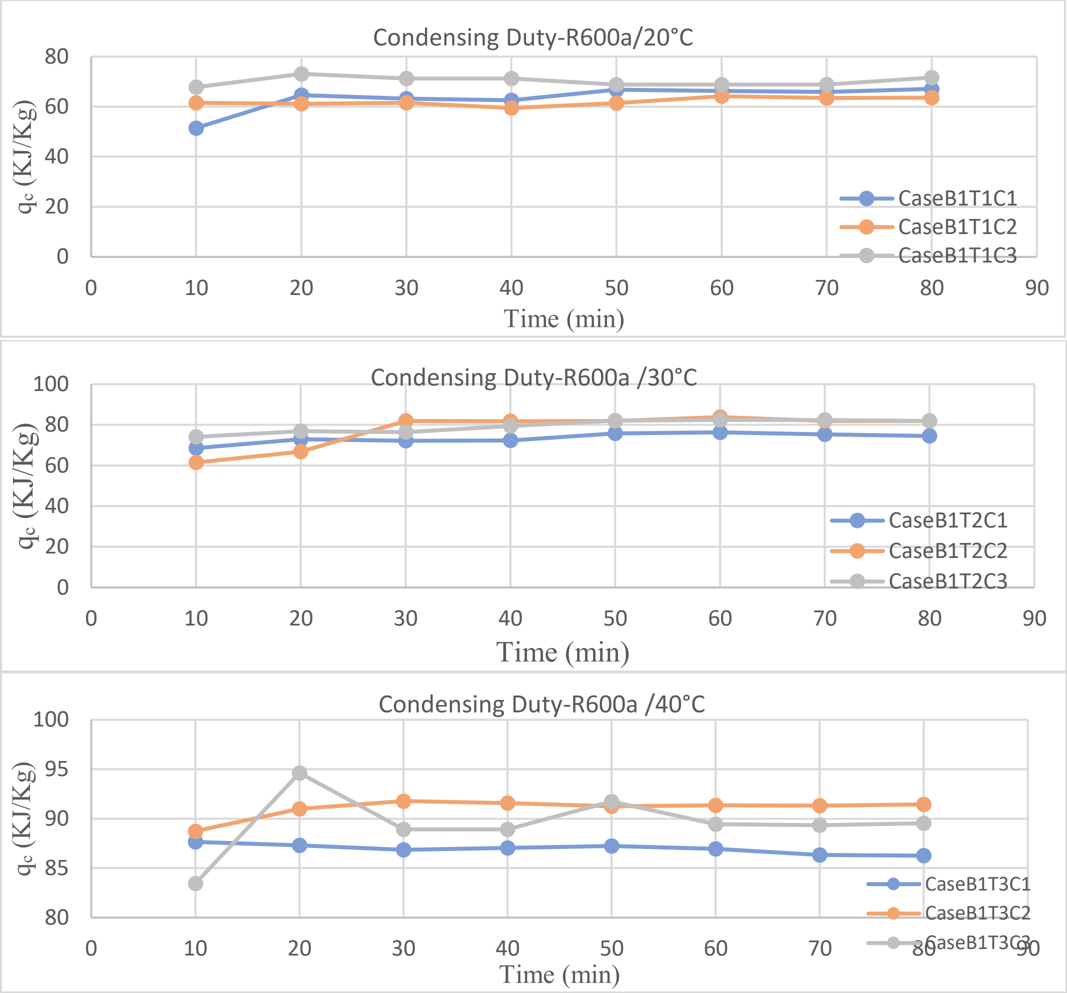
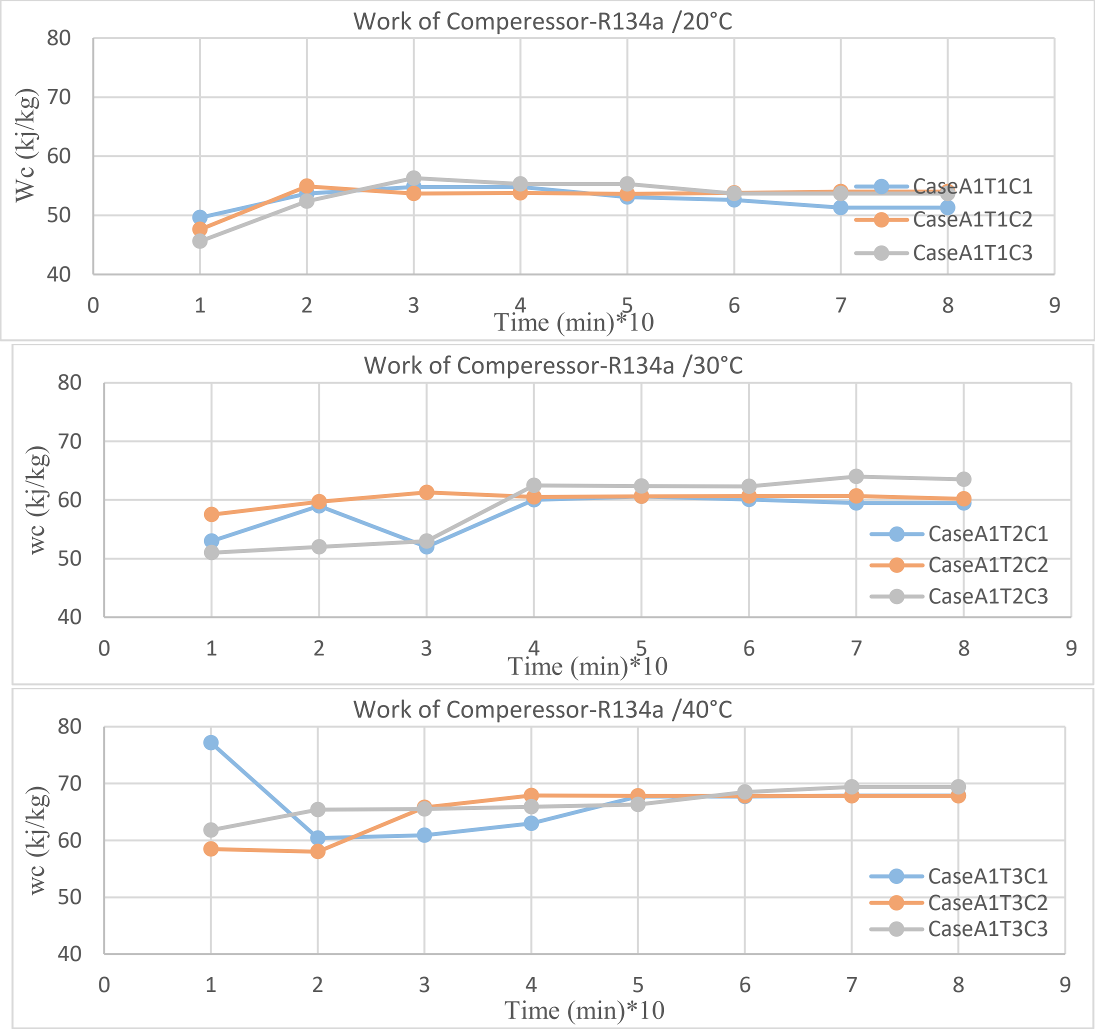


Figure 10. 3mode of ambient temperatures Condensing Duty (qc) with time for R600a.

Compressor Pressure ratio and compressor work

In Figure 12, the variation of compressor work (Wc) with operating time at ambient temperatures of 20°C, 30°C, and 40°C is shown, revealing that as ambient temperature increases, Wc also increases because condensation temperature and condensing pressure both rise when ambient temperature increases, thereby raising the compression load on the compressor. Compressor work values are much more stable at 20°C and 30°C, but at 40°C it is significantly increased because the thermal load on the refrigeration system is higher. Figure 13 shows the effect of time on Wc for refrigerant R600a in a similar manner. The effectiveness increased significantly with the ambient temperature. The maximum values were observed at 40°C, which indicates more work for the compressor to maintain its function due to high condensing pressure.

All three condenser configurations follow the same directions when R134a refrigerant is applied, as Pr approaches relatively constant values during the last 40minutes of operation (Figure14). A similar trend is observed for R600a refrigerant in Figure 15, however, the pressure ratio values differ between the two refrigerants due to their different thermodynamic characteristics and sensitivity to ambient temperature. Increasing ambient temperature leads to higher condensing pressure because of the reduced heat rejection capability of the condenser, in addition to the heat energy added by the compressor. As the refrigeration cycle approaches stable operating conditions, the variations in compressor inlet temperature and pressure ratio become less significant with time. The increase in compressor work and pressure ratio at elevated ambient temperatures is directly associated with the higher condensing temperature, which reduces the overall thermal efficiency of the refrigeration cycle.

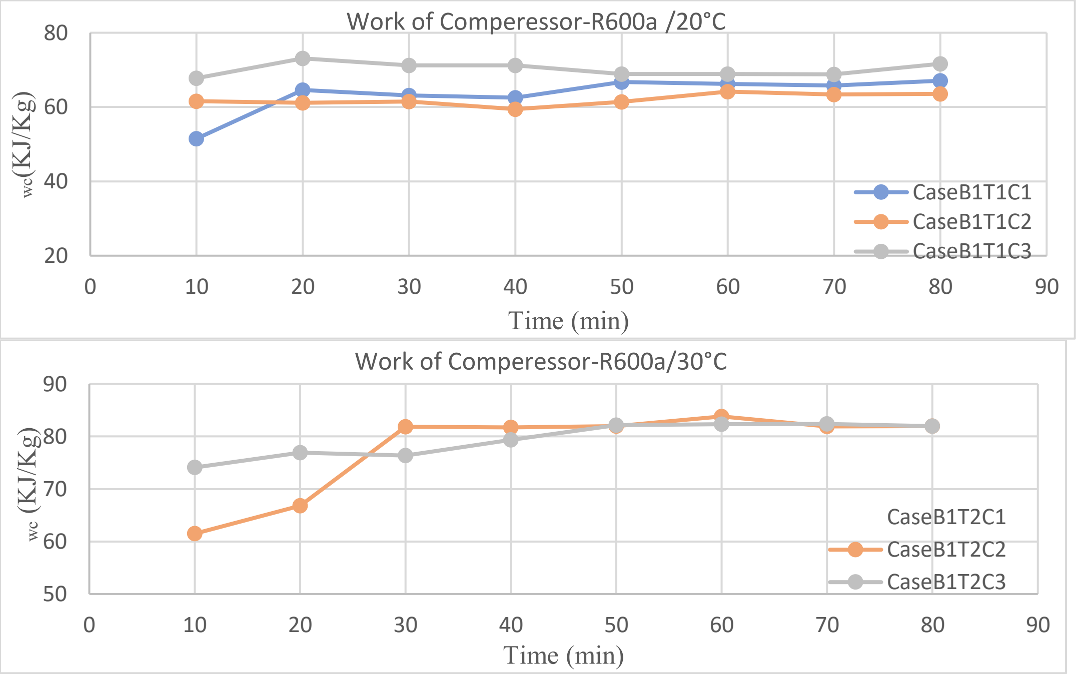


Work of Comperessor-R134a /20°C

Work of Comperessor-R134a /30°C

Work of Comperessor-R134a /40°C

Figure 11. Work of Compressor (wc) with time for R134a for 3mode of ambient temperatures.



Work of Comperessor-R600a /20°C

Work of Comperessor-R600a/30°C

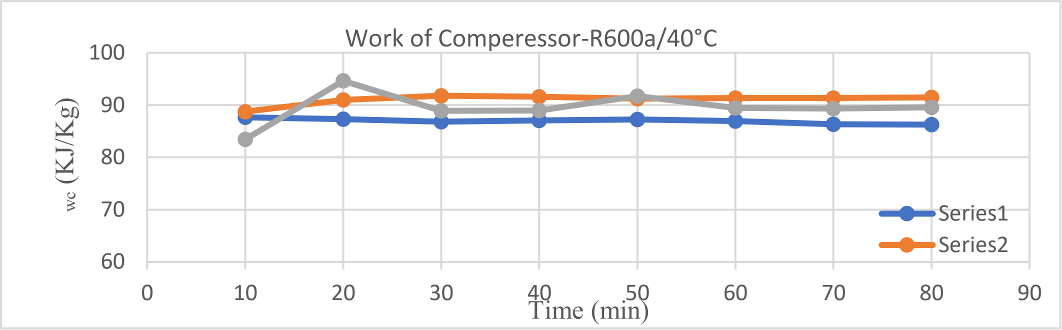


Figure 12. Work of Compressor (WC) with time for R600a for 3mode of ambient.

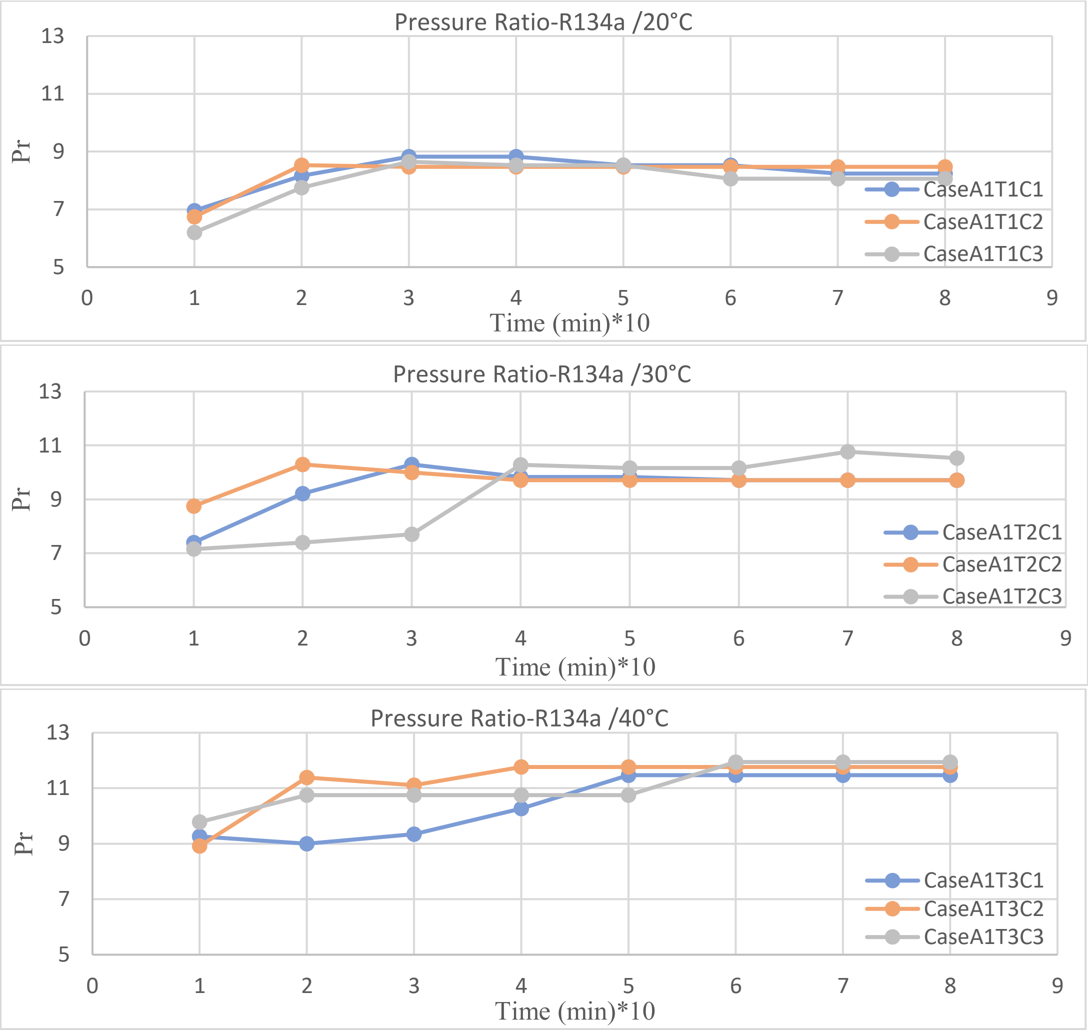
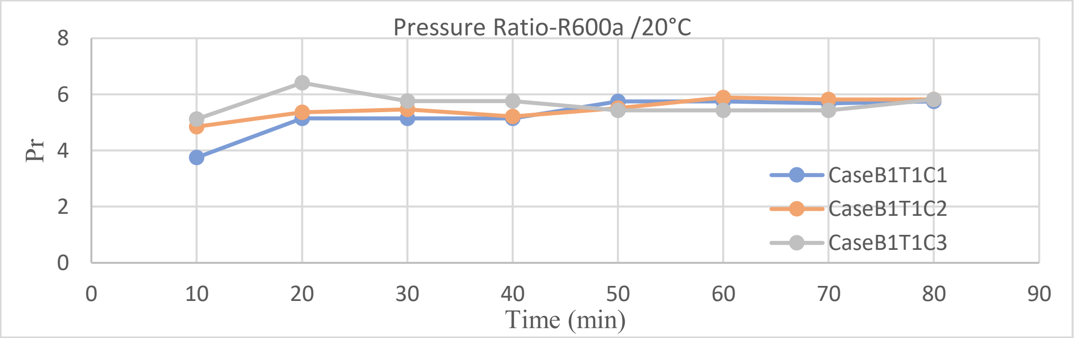


Figure 13. Pressure ratio (Pr) with time for R134a for 3mode of ambient temperatures.



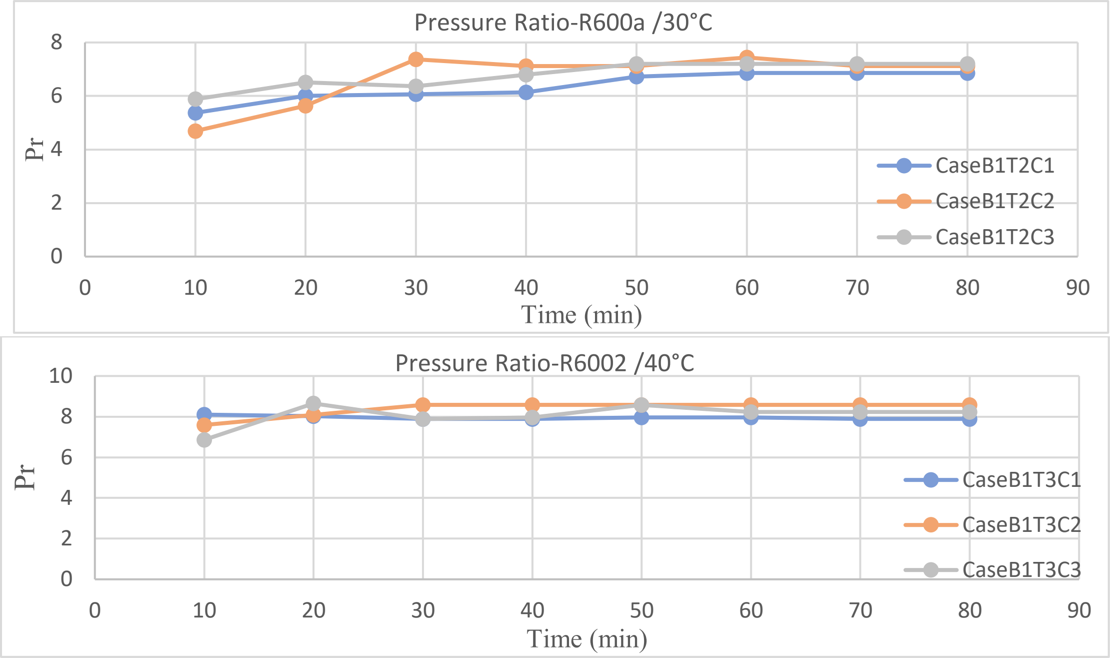


Figure 14. Pressure ratio (Pr) with time for R600a for 3mode of ambient temperatures.

Coefficient of performance

Figure 16 illustrates the variation of the coefficient of performance (COP) with operating time for condenser configurations C1, C2, and C3 under different ambient temperature conditions using R134a refrigerant. It can be seen that the COP curves have obvious oscillations during the initial operating period, while the COP values gradually arrive stable operating condition after approximately 20 to 30 min. Due to its extended effective heat transfer area and higher heat removal capacity, the conventional finned condenser (C1) shows better COP values than both configurations C2 and C3. C2 and C3 are similar in value, with very little difference. When the ambient temperature increases, COP values of all three condensers of the system decrease. This drop is a consequence of increased compressor work and condensing pressure at high ambient temperatures. The COP shows a decrease with respect to increasing ambient temperature, which is 3–3.5 at 20 °C, 2.7–2.8 at 30 °C, and 2.4–2.5 at 40 °C, respectively, based on selected condenser configurations where the conventional finned condenser had superior thermal performance with a maximum COP value of 4.2. This behavior indicates that more condenser fins are effective in increasing the heat transfer area, leading to better heat transfer from the refrigerant to the surrounding air, thus allowing a low temperature difference and improved thermal performance of condensers at high ambient temperatures (up to 40 °C). Figure 17 shows variations in COP for R600a refrigerant using three configurations of condenser coils working at different ambient temperatures (20°C, 30°C, and 40°C). At 40°C, the lowest values were 2.8 to 3.2 from the condensers. R600a had a higher COP than R134a when the ambient temperature was 20°C whereas R134a performed better at an ambient temperature of 40°C compared to R600a. This means that R600a is much more susceptible to high ambient temperatures, while R134a has better thermal stability at critical temperature ranges.

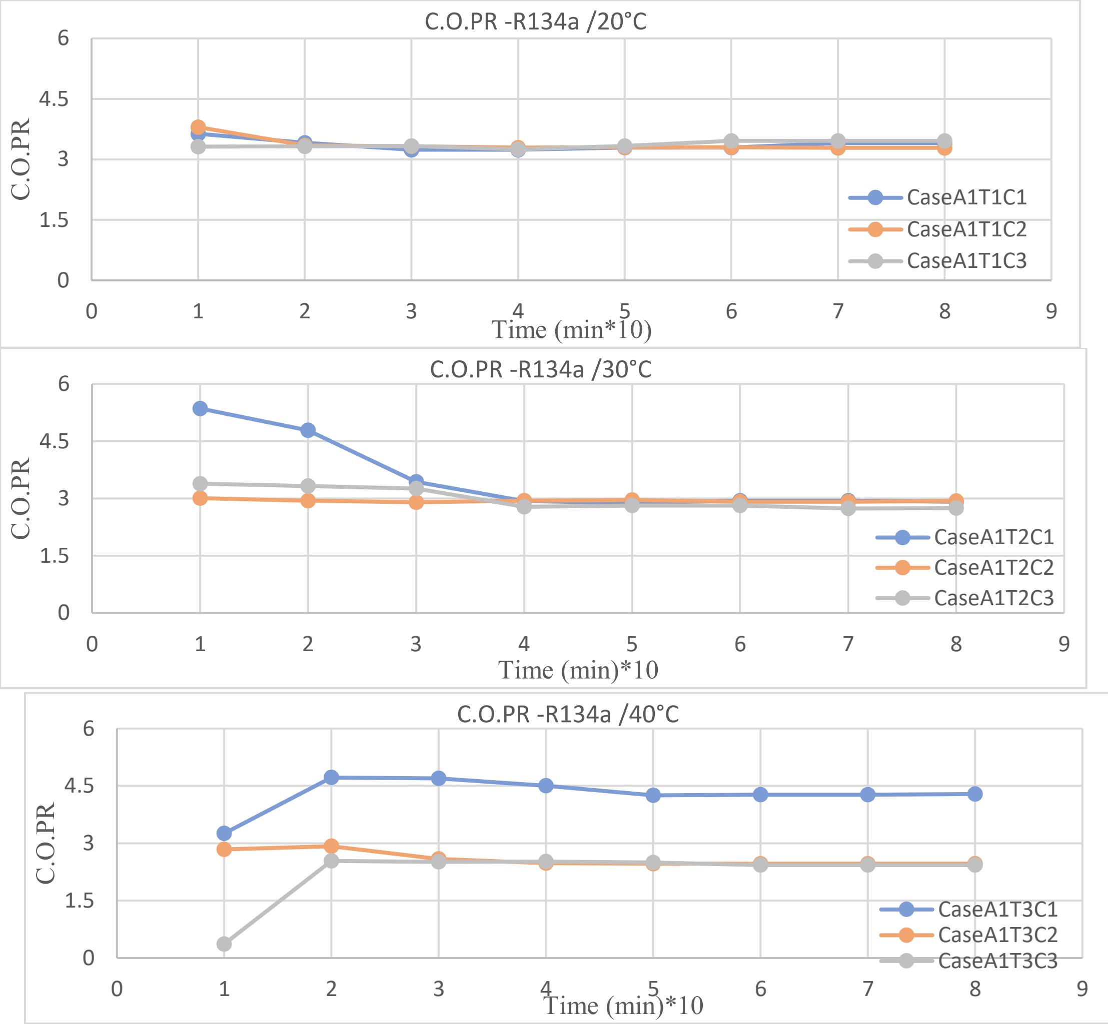
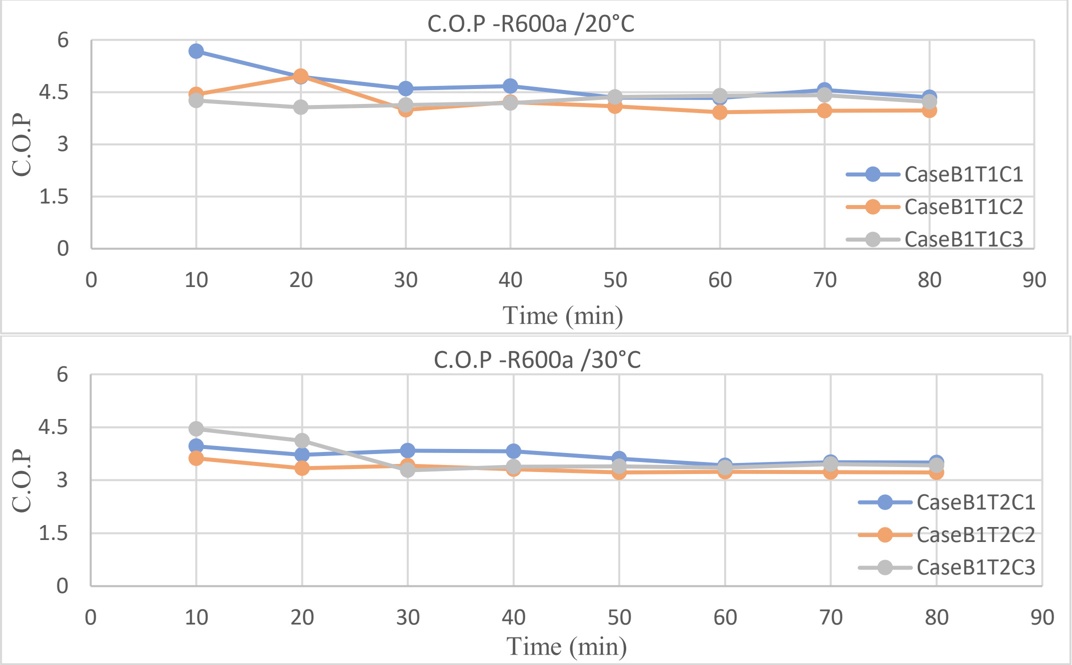


Figure 15. 3mode of ambient temperatures Coefficient of performance (C.O.PR) with time for R134a.



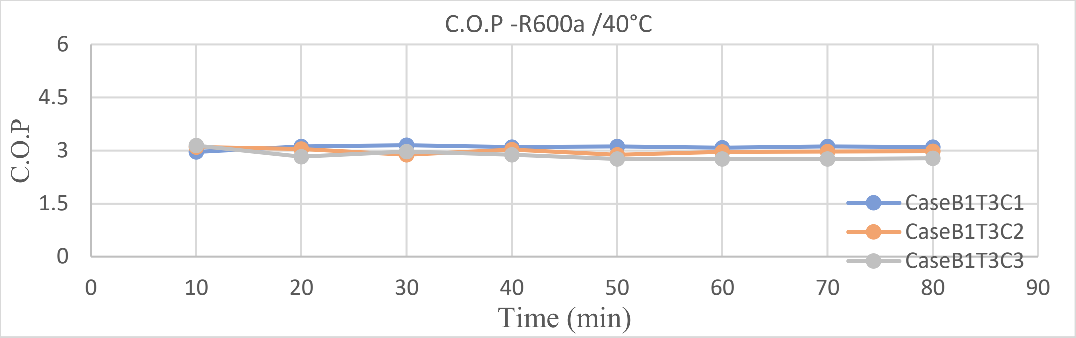


Figure 16. 3mode of ambient temperatures Coefficient of performance (C.O.PR) with time for R600a.

Conclusions

1. At an ambient temperature of 20 °C, the performance of condensers was optimal for both refrigerants (R134a and R600a). The lowest temperatures reached in ambient conditions of 40 °C are -24°C and -18°C, both achieved with R134a and R600a, respectively

2. Both refrigerants show a comparatively stable refrigeration effect (RE) of 170 kJ/kg in the various tested condenser configurations.

3. For both refrigerants, compressor work increased with increasing ambient temperature, and the pressure ratio displayed a stable trend after the system approached quasi-steady-state operation.

4. The maximum value of COPR for R134a was 3.4, 2.75, and 2.5 at temperatures of 20, 30, and 40 °C, respectively. Among all the investigated condenser configurations, a conventional finned condenser showed a significantly higher COP of 4.2 under optimal operating conditions. The COP values for R600a were between about 2.6 and 3, depending on the ambient temperature and condenser configuration. In conclusion, R600a was found to have a higher COP than R134a under moderate ambient temperatures, while the refrigeration systems using R134a were found to achieve better thermal performance under high ambient temperatures.

5. Generally, the conventional configuration of a finned condenser achieved higher thermal performance than the medium-fin and the finless cases for R134a and R600a refrigerants.

6. The results indicate that increasing the effective condenser surface area has a positive impact on improving refrigerator thermal performance and energy efficiency, especially under high ambient temperature conditions, which are experienced in tropical hot climates.

Nomenclature

COPR	*coefficient of performance Refrigeration*
WIC	*The electrical work of the compressor (W)*
Wc	*work of compressor (kj/kg)*
h	*enthalpy (kj/kg)*
I	*electric current (A)*
V	*electric voltage (V)*
RE	*refrigeration effect (kj/kg)*
Qe	*cooling capacity (W)*
Qc	*condenser capacity (kj/kg)*
Pr	*Pressure Ratio*

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