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Natural Working Fluids in Retail Applications in Jordan & MENA Region

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ABSTRACT

The utilization of natural working fluids in commercial refrigeration and industrial cooling units worldwide can provide an energy efficient solution to combat the ozone depletion and the global warming. This has been proven over various past research, where natural refrigerants provided high energy efficiencies and savings compared to HFC units. It also showed that the total environmental impact has significantly improved after the increase in popularity of refrigeration and heat pump systems applying natural working fluids (NWF).

This report will give a solid proof of the sustainability of NWF systems by presenting a full thermodynamic analysis of a system charged with propane (R290) working on a DC inverter rotary compressor, presenting the performance of the system on three different ambient temperatures 25°C, 27.5°C and 30°C.

This analysis will allow to answer the question whether systems charged with natural refrigerants such as Propane are applicable in the middle eastern region.

Keywords: R290 (Propane), DC Invertor Compressor, NWF (Natural Working Fluids).

1. INTRODUCTION

1.1. General Introduction

Over the past couple of centuries, there was a huge rise in environmental consciousness around the world, and one of the bigger concerns was the use of HFC refrigerants, which are widely used in several industrial and commercial applications. The most commonly used refrigerants in the middle-easter region are HFCs, however, these refrigerants have some of the highest levels of global warming potential (GWP). Fortunately, there has been a rising interest in alternative refrigerants, namely natural refrigerants (Bolaji & Huan, 2013), which are naturally occurring substances that have virtually no impact on the environment, with an Ozone depletion potential (ODP) of 0 and very low GWP levels. For example, compared to R134a which has a GWP of 1430, R290 has a GWP of 3.3 (Yadav, Liu, & Kim, 2022).

1.2. Motivation and Objectives

- Natural refrigerants are more environmentally friendly, making them more desirable to environmentally conscious customers and manufacturers.
- Units applying natural refrigerants are more energy efficient, improving the overall product.
- Natural refrigerants are less expensive, predictable, and more readily available.

The objective of this study is to show the energy efficiency of natural working fluid system, which will raise awareness for NWF utilising systems and increase its popularity by proving that such systems are viable options through a thermodynamic analysis.

1.3. Previous Research

A paper by Choudhari et al., 2017 investigated the use of R290 as a substitute to R22 carried on a standard vapor compression cycle, which concluded that R290 had an insignificantly lower COP due to the fact that the system is designed for an R22 refrigerant. However, it suggested that with a refrigeration system optimized for an R290 refrigerant it will achieve a higher COP, moreover it is favourable for having better thermal properties and a very low GWP.

Another research compared R134a units and several refrigerants with a low GWP resulted in showing; R290 provides an average COP increase of 12.6 % and a cooling capacity increase of 54 % when used as a drop-in replacement for a R134a. However, it also increases power consumption by up to 44.8% due to the fact that the compressor was designed for R134a refrigerant (Sánchez, et al., 2017).

1.4. Research Question

To what extent is a system with an inverter driven compressor charged with R290 applicable in countries with higher mean ambient temperatures year-round.

2. BACKGROUND INFORMATION

2.1. General System Description

The general specifications of the system and the investigated display cabinet can be found in table 1.

Table 1. General specifications of the systems

Type	Closed cabinet
Net volume	1000 L
Length	133 cm
Width	80 cm
Height	206 cm
Insulation	40 mm of polyurethane
Compressor type	DC inverter rotary compressor Displacement: 10.0 cm^3
Motor type	Synchronous Motor (BLDC)
Controller type	Inverter

3. METHODOLOGY

3.1. System Setup

The cabinet temperature was set to a temperature of 2 °C and investigated experimentally in a controlled environment with a relative humidity of 50%, and at three different ambient temperatures of 25 °C, 27.5 °C, and 30 °C. Sensors were installed at all points of interest according to table 2. Noting that the sensors used are temperature and not pressure sensors.

Table 2: Sensor types and placement.

Sensor	Reading	Location
Temperature	Ambient temperature(°C) ±0.5°C	Ambient
	Air-off temperature(°C) ±0.3°C	Cabinet interior
	Condensation temperature (°C) ±0.3°C	Discharge line
	Compressor inlet temp. (°C) ±0.3°C	Suction line
	Evaporation temperature (°C) ±1.5%	Liquid line
Electronic Hygrometer	Relative Humidity ±5%	Ambient
Energy meter	Energy Demand (kWh) ±1.5%	compressor

3.2. Data Setup

The data collected from the system includes, the ambient temperature, ambient humidity, cabinet temperature, evaporation temperature, suction line temperature, condensation temperature, and the energy demand of the system. Each of the data were recorded at 10-minute intervals to increase the accuracy of the final answers.

To make the data easier to work with an average was taken for every hour of the 46 hours. From the data collected of 1-hour averages, the overall average of the 46 hours was found and used to calculate the COP, isentropic efficiency, shaft work (real work), and work loss due to compressor, process, expansion, and overheat.

3.3. Test Procedure

1. The refrigerator cabinets are filled with 66 test packages made of wood to represent a load of food.
2. The system is operated for 46 hours at the set ambient temperature.
3. Test data is recorded at 10 min intervals over the entire test period.
4. Steps 2 and 3 are repeated for the other two set ambient temperatures.

4. DATA ANALYSIS

4.1. Calculation Methodology

After collecting all the data recorded by the sensors, taking the averages as described earlier in section 3.2. the shaft work, COP, isentropic efficiency, and work losses were calculated as follow:

Starting with the shaft work, the energy meter records the energy demand/consumption of the system over the working hours. The system was operated and measured for 46 hours, after finding the average energy demand for the set ambient temperature the value is divided by 46 h to give the shaft work as shown below for ambient temp 25.3^{O}C.

$$Shaft\ Work = \frac{Energy\ Demand}{Number\ of\ working\ hours} = \frac{15.06(kWh)}{46(h)} = 0.33kW \qquad \text{eq.1}$$

The same method was repeated for the other two ambient temperatures.

The COP was found by dividing the cooling capacity by the shaft work:

$$COP = \frac{Q_o}{W_r} = \frac{0.75kW}{0.327kW} = 2.29 \qquad \text{eq.2}$$

The isentropic efficiency was found by first finding the mass flow of refrigerant in system then finding the isentropic work and dividing it by the real work. To find the mass flow of the refrigerant the enthalpy of the refrigerant at saturated gas and liquid (h_1 and h_4) at -10^{O}C are found:

$$\dot{m} = \frac{Q_o}{(h_1 - h_4)} = \frac{0.75}{(565.06 - 294.92)} = 0.0028kg/s \qquad \text{eq.3}$$

To find the isentropic work the enthalpy of superheated gas at discharge line (h_{2s}) is found then using h_1 and mass flow the isentropic work can be found:

$$W_{is} = (h_{2s} - h_1) * \dot{m} = 0.0028(624.0095 - 565.06) = 0.164 \qquad \text{eq.4}$$

Finally, the isentropic efficiency is found as follows:

$$\eta_i = \frac{W_{is}}{W_r} = \frac{0.164}{0.327} = 0.5 \qquad \text{eq.5}$$

The work losses of compressor, process, expansion and overheat are calculated as follows:

$$\Delta W_{comp} = W_r - W_{is} \quad \text{eq.6}$$

$$\Delta W_{process=}\left(W_{is} - W_{ca,min}\right) - \left(W_{ca,2} - W_{ca,min}\right) \quad \text{eq.7}$$

$$W_{ca,min} = Q_o * \frac{(T_{amb}+273.15)-(T_{room}+273.15)}{T_{room}+273.15} = 0.75 * \frac{(25+\quad)-(2.2+27\ )}{2.2+273} = 0.062kW \quad \text{eq.8}$$

$$W_{ca,2} = Q_o * \frac{T_c - T_o}{T_o} = 0.75 * \frac{(34.9+273)-(-9.96+273)}{-9.96+27} = 0.13kW \quad \text{eq.9}$$

$$\Delta W_{exp} = \frac{\dot{m} * T_c * (s_4 - s_3)}{1000} \quad \text{eq.10}$$

$$\Delta W_{overheat} = \Delta W_{process} - \Delta W_{exp} \quad \text{eq.11}$$

Using these equations all the data presented below were calculated.

4.2. Data Analysis

Table 3 shows all the calculated data and the boundary conditions, T_{amb}, Humidity and Temperature set point, as well as the evaporation and condensation temperatures for both systems under the three different ambient conditions. Cabinet temperature are the average cabinet temperature over the 46-hour period. The shaft work was found through the energy meters attached to the system. As well as the COP and isentropic efficiencies were found after finding the isentropic work of each system under each of the ambient temperatures.

Table 3: Results table for the system.

R290 system								
T_{amb} (°C)	Humidity	Temp. set point (°C)	T_o/T_c	Cabinet temp. (°C)	Shaft work (kW)	COP	Isentropic efficiency	Energy Demand (kWh)
25.3	50	2	-10/34.9	2.2	0.33	2.29	0.5	15.06
27.3	50	2	-9.9/35.0	2.2	0.33	2.29	0.5	15.08
30.9	48	2	-9.5/42.7	2.2	0.38	1.96	0.53	17.58

Figure 2 presents the COP of the system at the three different ambient temperatures. It can be seen as expected that as ambient temperature increases the COP of the system decreases. Where it was most prominent when the ambient temperature increases from 27.3 to 30.9°C.

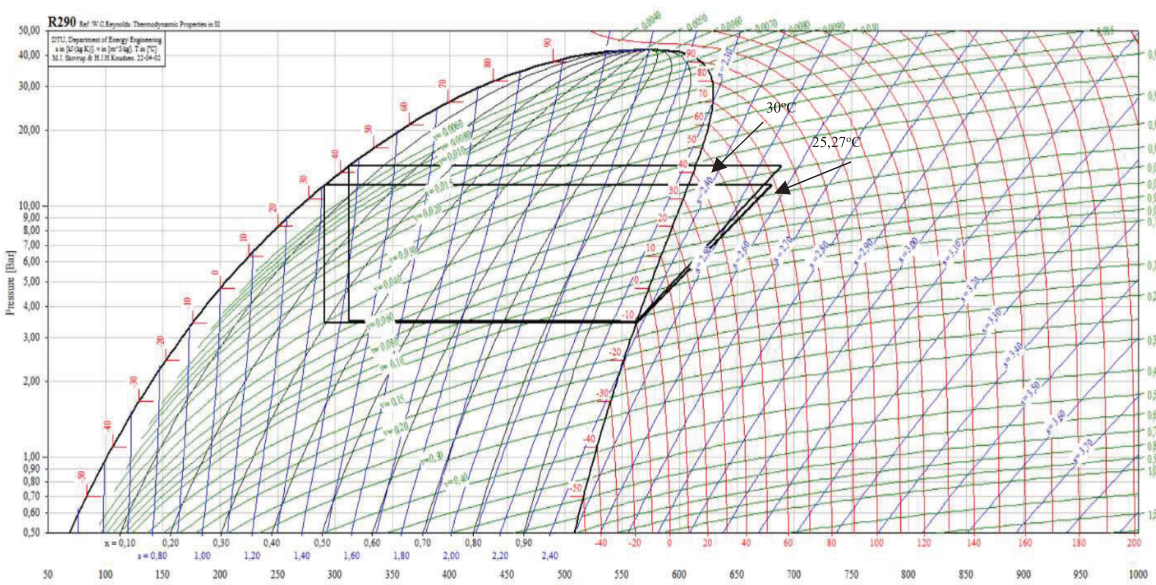


Figure 1: Log ph diagram for the system operated at three ambient temperatures.

Moreover, there is no decrease in COP of the R290 system when increasing the temperature from 25.3 to 27.3°C. However, a drop in COP for the R290 system occurred when increasing the temperature from 27.3 to 30.9°C showing 14.4% drop in COP from 2.29 to 1.96.

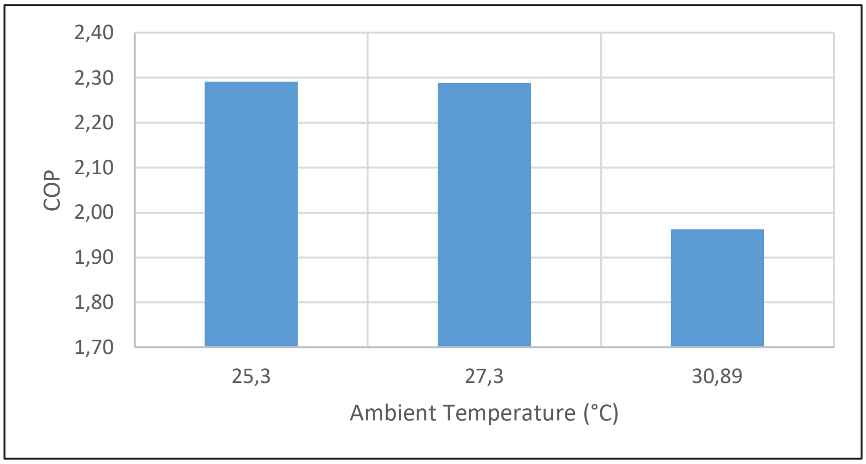


Figure 2: COP values of the system at three ambient temperatures.

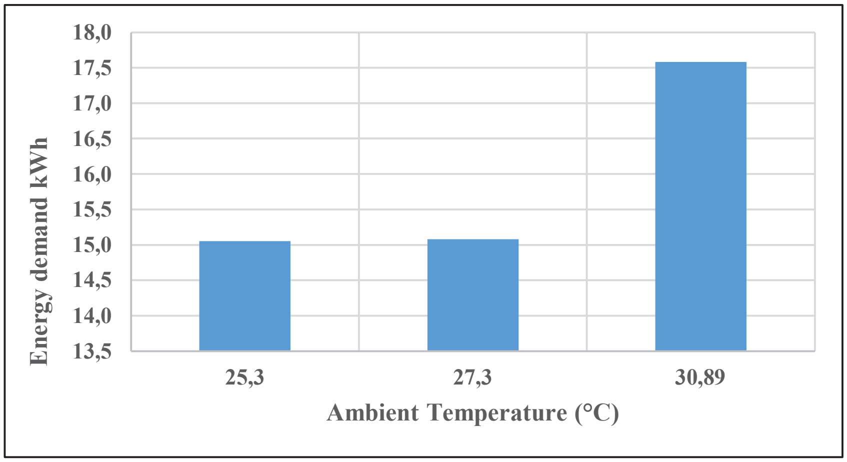


Figure 3: Energy demand of the system at three ambient temperatures for 46h of operation

Figure 3 shows how the energy demand increase for the system as the ambient temperature increased. Overall, the R290 system had a 16.7% increase in energy demand when increasing the temperature from 25.3 to 30.9°C.

Figure 4 shows that the isentropic efficiency of the systems increased with increasing the ambient temperature, where the R290 system has an average isentropic efficiency of 0.51 over the three ambient temperatures.

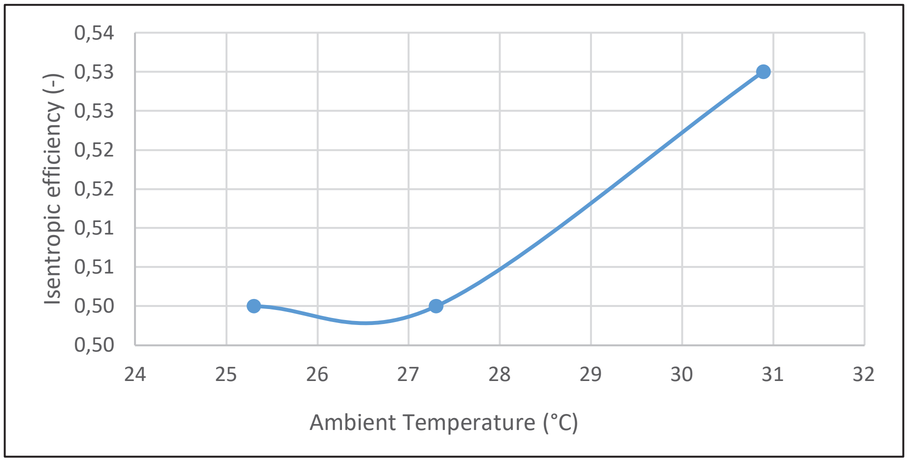


Figure 4: The isentropic efficiency at various ambient temperatures.

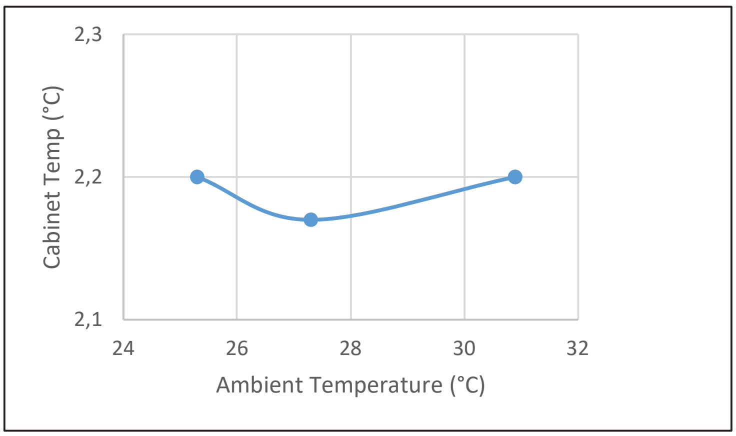
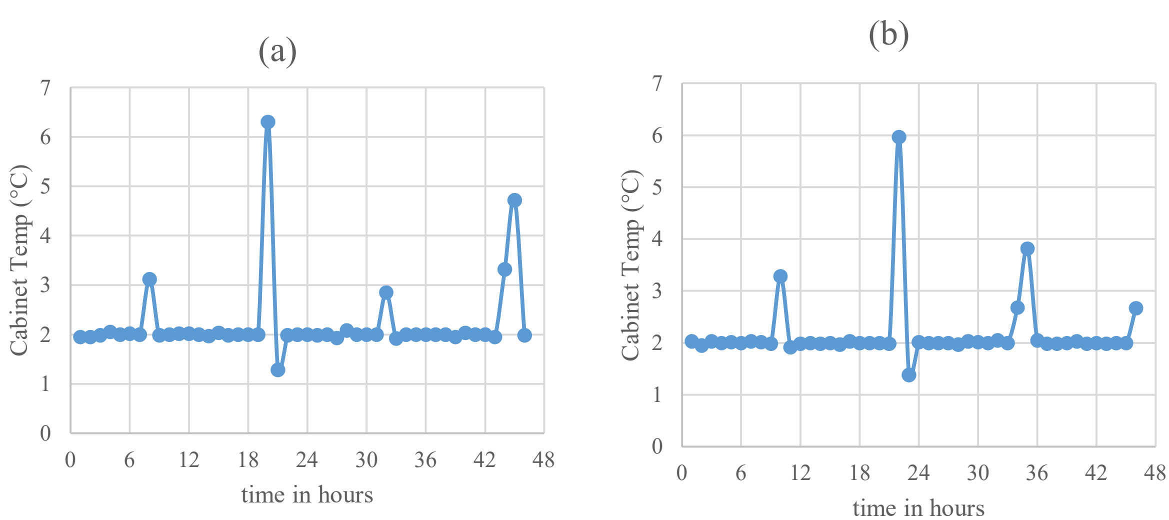


Figure 5: Average cabinet temperature against ambient temperature.

Figure 5 shows the average cabinet temperature as the ambient temperature is increased. The R290 system shows a very stable cabinet temperature. Where the R290 system was able to maintain an average cabinet temperature of 2.2 °C which is very close to the set point temperature as described in table 1.



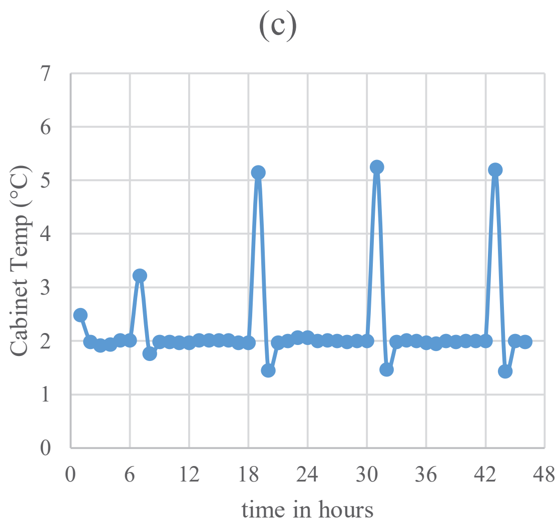


Figure 6: Cabinet temperature over the 46-hour period at 3 ambient temperatures: (a) 25.3 °C (b) 27.3 °C, (c) 30.9 °C

Similar to Figure 5, Figure 6 shows the cabinet temperature over the 46 hours for each ambient temperature. Overall, it can be seen that the R290 system provides a high level of stability for the three ambient temperatures. The temperature spikes occurred when the defrost cycles are on.

Table 4, on the other hand, shows the work losses due to compressor, process, expansion and overheat in kW. These values are calculated using minimum Carnot cycle of ambient temp, minimum Carnot cycle of the evaporation and condensation temperatures, isentropic and real work. With the total work loss at 25.3 ^{O}C being 0.24 kW for the R290 system, at 27.3 ^{O}C 0.24 kW for the system, finally at 30.9 ^{O}C being 0.29 kW for the R290 system. This is also presented in figure 7.

Table 4: Work losses due to compressor, process, expansion and overheat.

R290 system				
T_{amb} (°C)	DW comp. (kW)	DW proc. (kW)	DW exp. (kW)	DW overheat (kW)
25.3	0.16	0.036	0.036	0.00024
27.3	0.16	0.036	0.036	0.00024
30.9	0.18	0.054	0.054	0.00033

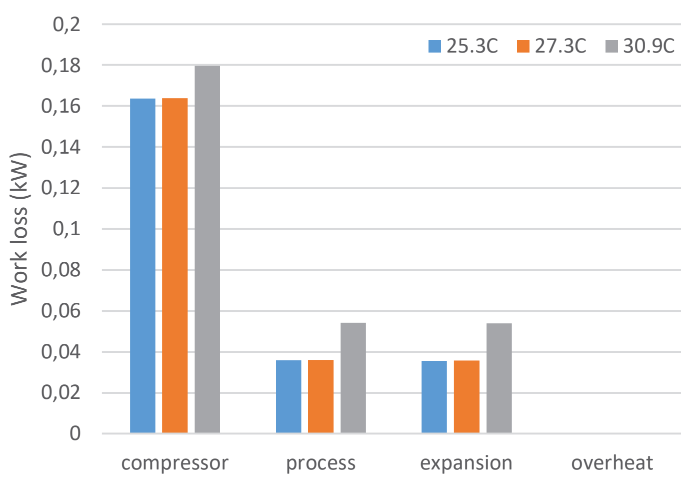


Figure 7: Work losses of the R290 system.

5. DISCUSSION

From analysing the figures and the data calculated, it can be seen from Figure 1 that the R290 system has an average COP of 2.2 over the range of the three ambient temperatures, with the lowest COP at the ambient temperature of 30 °C recording a COP of 1.96 which is a 14.4% decrease in COP when compared to the COP at 25.3 °C and 27.3 °C. From figure 2 it can be concluded that the average energy demand of the R290 system is 15.9 kWh. Moreover, a 6 % increase in isentropic efficiency was recorded when the temperature increased from 25 °C to 30 °C shown in Figure 3.

From Figures 4 and 5, it is clearly seen that the R290 system has a low and stable average cabinet temperature of 2.2°C which is only 0.2 K higher than the set point temperature of 2 °C. Finally, from Figures 6 and 7, most of the work loss was identified to be due to the compressor, by applying another compressor technology this loss could be reduced. The average work loss at the three ambient temperatures for the R290 system was 0.26kW respectively.

6. FURTHER WORK AND CONCLUSION

For further development, to investigate whether the system can perform efficiently when compared to alternative systems operating on HFC. Another possible work is researching the performance differences between two systems where the only variable is the type of refrigerant.

In conclusion, this research has proven that a system charged with R290 and working on an inverter compressor is applicable in warm areas with good performance. Hence, this confirms the fact that natural working fluid systems are very much valid in the commercial refrigeration industry.

7. NOMENCLATURE

W_r : Shaft Work (kW); Q_o : Cooling Capacity(kW); *COP:* Coefficient of Performance; $\dot{m}$: Mass flow rate (kg/s) W_{is}: Isentropic work (kW); *h:* enthalpy (kJ/kg); *s:* entropy (J/kgK); η: Isentropic efficiency; ΔW_{comp}: Compressor work loss (kW); $\Delta W_{process}$: Process work loss (kW); $\Delta W_{ca,min}$: Minimum carnot cycle (kW); ΔW_{ca}: Carnot Cycle at evap. and cond. temperatures (kW); ΔW_{exp}: Expansion work loss (kW); $\Delta W_{overheat}$: Overheat work loss (kW); T_o,T_c,T_{amb},T_{room}: Evaporation, Condensation, ambient and cabinet temperature.

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