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(Research Article)

Calculation of exergy losses, exergy efficiency and cooling coefficient of refrigerant R22

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Abstract

This study presents a detailed analysis of the exergy losses, exergy efficiency, and cooling coefficient of performance (COP) for the refrigerant R22 in refrigeration systems. Exergy analysis, which considers both the quantity and quality of energy, is used to identify and quantify the irreversibilities within the system. The objective of this analysis is to provide insights into the areas of inefficiency and potential improvements for enhanced energy performance. Results indicate that the exergy losses in R22 systems are primarily associated with the compressor and expansion valve, highlighting these components as key targets for efficiency improvements. The exergy efficiency was found to vary significantly with operating conditions, suggesting that system performance could be optimized through appropriate design and control strategies. Additionally, the cooling COP of R22 was evaluated, demonstrating its effectiveness in converting input energy into cooling capacity under various conditions. This analysis contributes to the broader understanding of R22's thermodynamic performance and offers a foundation for developing more sustainable and efficient refrigeration technologies.

Keywords: Cooling coefficient; Exergy losses; Efficiency; R22; Refrigerant

1. Introduction

Refrigeration and air conditioning systems are fundamental in numerous industrial, commercial, and domestic applications. The efficient operation of these systems is crucial to reducing energy consumption and minimizing environmental impact. Among the various refrigerants used, R22 (Chlorodifluoromethane) has been one of the most widely employed due to its favorable thermodynamic properties and performance characteristics [1]. However, concerns regarding its environmental impact, specifically its ozone depletion potential and global warming potential, have led to increasing scrutiny and the search for more sustainable alternatives [2].

Exergy analysis is a valuable tool for assessing the performance of thermodynamic systems. Unlike traditional energy analysis, which only accounts for the quantity of energy, exergy analysis evaluates both the quantity and quality of energy, providing a clearer understanding of system inefficiencies and irreversibilities [3]. By identifying the sources of exergy losses, engineers can target specific areas for improvement, leading to more effective design and operational strategies [4]. Exergy analysis has been extensively applied in the study of refrigeration systems to quantify losses and enhance their overall efficiency [5].

In this study, we focus on the calculation of exergy losses, exergy efficiency, and the cooling coefficient of performance (COP) for the refrigerant R22. These parameters are crucial for optimizing refrigeration systems, as they help identify inefficiencies and suggest ways to enhance energy utilization. Previous research has highlighted the importance of these metrics in the context of R22, demonstrating that improvements in exergy efficiency directly contribute to energy

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savings and reduced environmental impact [6]. Furthermore, understanding the cooling COP provides insights into the system's effectiveness in providing cooling relative to its energy input [7].

The objective of this paper is to conduct a detailed thermodynamic analysis of R22-based refrigeration systems using exergy principles. By analyzing the exergy losses and efficiencies, as well as the cooling coefficient, we aim to offer a comprehensive understanding of R22's performance in refrigeration applications. This study intends to provide valuable insights that can guide future improvements in refrigerant selection and system design for enhanced sustainability and efficiency.

1.1. Study objects and methods

Apply the theory of energy method and exergy analysis to calculate the cooling coefficient (COP), olfactory and exergy efficiency of the two-stage compression refrigeration system using refrigerant R22 as shown in the working principle diagram in Figure 1 and the working cycle of the refrigerant in Figure 2.

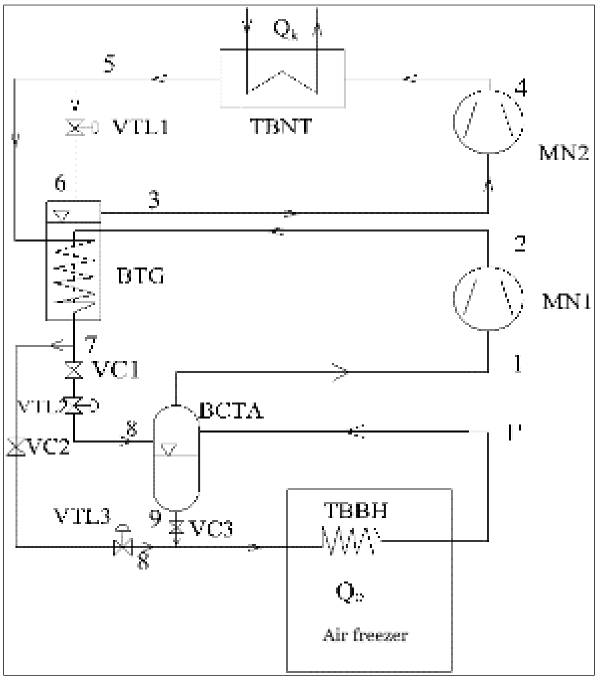

Figure 1 Schematic diagram of the refrigeration system

To supply fluid directly, close the stop valve (VC1), stop valve (VC3) and open the stop valve (VC2). Then the refrigerant after being overcooled at the intercooler does not enter the low pressure tank but enters the throttle directly through the throttle valve (VTL3) to supply fluid to the air freezer. To supply liquid indirectly through the low pressure tank (BCTA), close the stoxp valve (VC2) and open the stop valve (VC1), stop valve (VC3). Then the refrigerant after being supercooled at the intermediate cooler goes through the throttle valve (VTL2) into the low pressure tank. Here the liquid is separated and supplied to the evaporator of the air freezer.

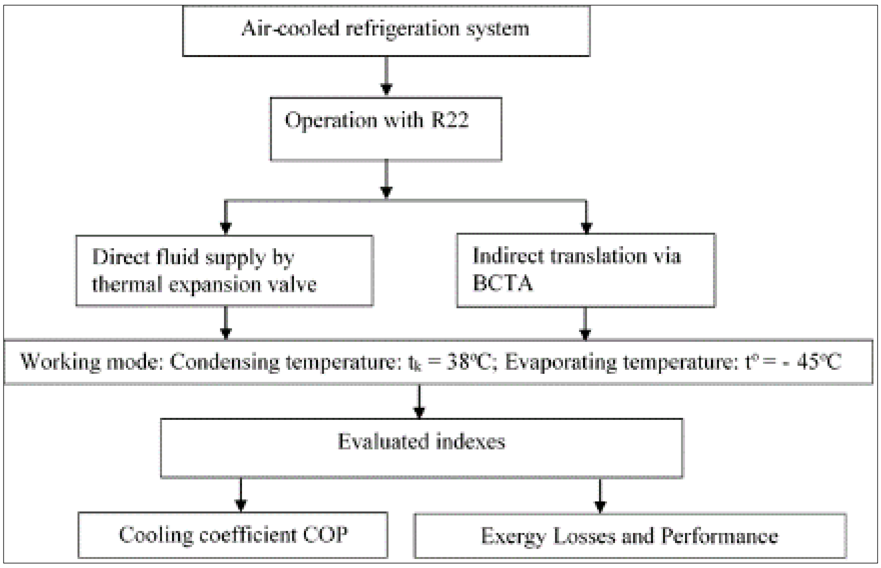

Figure 2 Theoretical research layout

2. Theoretical and experimental research methods

From the working principle diagram of the theoretical refrigeration system in Figure 1, proceed to install the actual refrigeration system with temperature probes, pressure gauges, power consumption, freezer temperature... shown in Figure 3. To study the implementation of the process of supplying liquid to the evaporator of the freezer by adjusting the opening and closing of the solenoid valves $(\frac{SV}{1})$ and $(\frac{SV}{2})$. With the direct fluid supply method, power is supplied to the solenoid valve $(\frac{SV}{1})$ and stop supplying power to the solenoid valve $(\frac{SV}{2})$. With the indirect fluid supply method through BCTA, power is supplied to the solenoid valve $(\frac{SV}{2})$ and stop supplying power to the solenoid valve $(\frac{SV}{1})$.

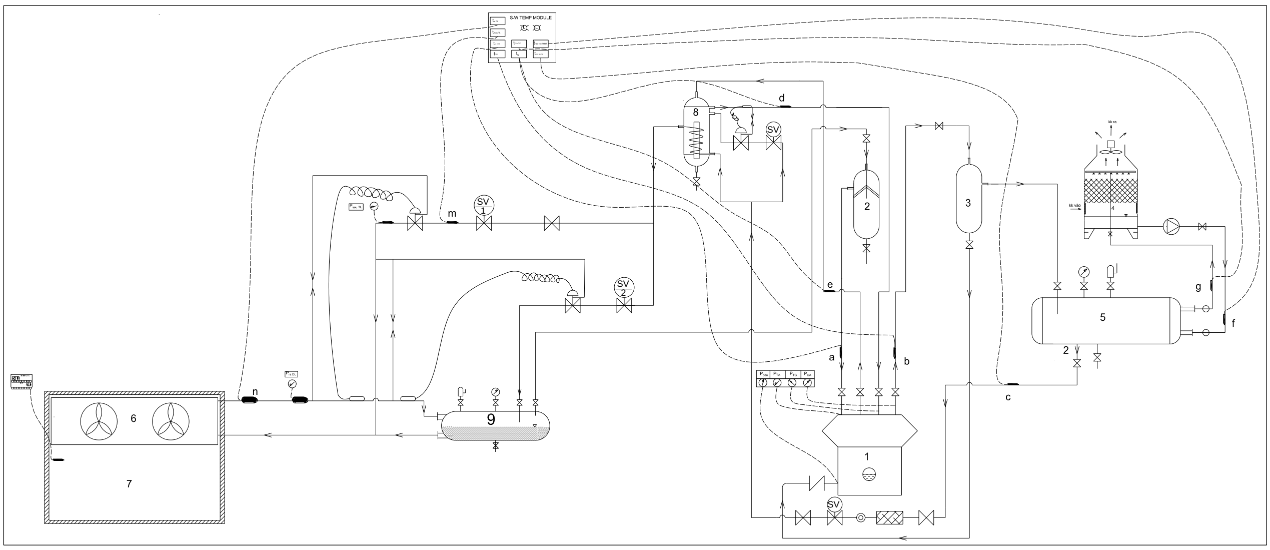

Figure 3 Experimental layout of the air-cooled refrigeration system

Notes:

1. Compressor
2. Liquid separator
3. Oil separator
4. Cooling tower
5. Condenser
6. Evaporator
7. Freezer
8. Intermediate tank
9. Low pressure tank

2.1. Temperature probe placement points.

- Low-stage compressor suction air temperature sensor
- High-stage compressor compression air temperature sensor
- Refrigerant temperature sensor exiting TBNT
- High-stage compressor suction air temperature sensor
- Low-stage compressor compression air temperature sensor
- TBNT inlet water temperature sensor
- TBNT outlet water temperature sensor
- m. Refrigerant temperature sensor before throttle valve
- n. Refrigerant temperature sensor exiting TBBH

2.2. The symbol on the meter "S.W TEM MODULE" reads the temperature

T_{outDL} – Temperature of the refrigerant leaving the TBBH

$T_{before\ TL}$ – Refrigerant temperature before throttle valve

$T_{suck\ MN}$ – Temperature of the refrigerant drawn to MNC1

$T_{pressure}$ – Temperature of the refrigerant drawn to MNC2

$t_{H2O\ out\ TBNT}$- Cooling water temperature leaving the TBNT

t_{tg}- Temperature at BTG position, suction to high-level compressor

$t_{H2O\ in\ TBNT}$- Water temperature in TBNT

$t_{after\ stop}$ – MC temperature out of TBNT

$t_{pressure\ MNC1}$ – Temperature of compressed refrigerant leaving the first stage compressor

2.3. When operating the refrigeration system using R22 refrigerant.

Vacuum and charge R22 gas for the refrigeration system. Carry out direct and indirect fluid supply methods by controlling solenoid valves $(\frac{SV}{1})$ and $(\frac{SV}{2})$. During machine operation, record data every 15 minutes for machine running with and without load.

The machine runs without load (no material in the freezer) when the air temperature in the freezer reaches -45°C and the machine runs with load when the product core temperature reaches -18°C, the freezing process ends. For the machine running with load, the amount of material in the freezer is 100 kg of water.

2.3.1. Evaluation methods

- **Exergy analysis method**

Apply exegry theory to determine the losses of first stage, second stage compressor, condenser, evaporator, throttle device 1, throttle device 2, BCTA, BTG and from there determine the total loss and exergy efficiency.

- **Energy balance method to determine cooling coefficient**

Applying the refrigeration engineering theory of energy balance to determine the refrigerant flow rate circulating through the first stage (low stage) and second stage (high stage) compressors, the cooling capacity, the capacity of the first and second stage compressors, the energy efficiency coefficient COP (COP = ε) or also known as the cooling coefficient.

2.3.2. Equipment and software used in research

Using a two-stage air-cooled freezing device, a Bitzer piston compressor with a capacity of 7.5HP equivalent to 5.59 kW, a semi-hermetic compressor. Horizontal shell and tube condenser, centrifugal pump to pump cooling water with a capacity of 0.65 kW, cooling tower fan is an axial fan with a capacity of 0.12 kW. The freezer is installed with a panel with a thickness of 200 mm and dimensions: length 2.5 m, width: 2.5 m, height 2 m. The condenser is installed with 2 fans with a capacity of 0.216 kW each, capacity 100 to 150 kg/batch. The wind speed in the freezer is 6 m/s. The refrigeration system uses an external thermal balancing expansion valve suitable for the compressor capacity for two methods of direct and indirect liquid supply. The evaporator is an air-type evaporator with a capacity suitable for the compressor capacity.

Use the "VSE3T" electronic meter to measure the power consumption of the freezing process: The machine runs without load when the air temperature in the freezer reaches -45^{o}C. The machine runs with load when the product core temperature reaches -18^{o}C. Use the "EXTECH MT500" digital thermometer to measure the product core temperature. Use the "Danfoss" pressure gauge to measure the pressure of the refrigeration system. Use the "S.W TEM MODULE" sensor to measure the temperature at the compression head, suction head, liquid refrigerant after condensation, before throttling, refrigerant leaving the evaporator, water entering and leaving the TBNT…. Use the Dixell digital thermometer to measure the air temperature in the freezer. Use the "EES" software to determine the thermodynamic parameters of the R22 refrigerant: Enthalpy, entropy… based on the basic parameters of pressure and working temperature of the refrigeration system.

2.3.3. Data processing and graphing methods

Experimental data were processed using mathematical statistics methods, graphs were drawn and processed using Excel software (Office Professional 2010). Each experiment was performed 3 times and the experimental results were the average of the results of the 3 experiments.

3. Results and discussion

From the schematic diagram of the refrigeration system and the theoretical research layout, based on the EES (Engineering Equation Solver) software, the thermodynamic parameters of the refrigerant R22 at special points of the cycle are determined. Calculate the losses and exergy efficiency, the cooling coefficient of the refrigeration system as follows.

Refrigerant flow through stage 1 and stage 2 compressors

The capacity of the compressor is: W_C=7,5 HP≅ 5,59 kW

Compressor performance:

$$\eta_c = \frac{W_{LT}}{W_{TT}} \Rightarrow W_{TT} = \frac{W_{LT}}{\eta_c} \quad (3.1)$$

Actual capacity of the compressor

(3.2)

$$W_{TT} = \frac{W_{c1} + W_{c2}}{\eta_c} \Rightarrow$$

$$w_{c1} + w_{c2} = 4.749$$

$\Rightarrow m_1.(h_2-h_1) + m_2.(h_4-h_3) = 4.749 \Rightarrow m_1.(425.6-385.6) + m_2.(442.3 - 400.9) = 4.749 \Rightarrow 40m_1 + 42.3m_2 = 4.749$ (*)

Heat balance equation at intermediate device,

$$\sum Q_{come} = \sum Q_{go}$$

$m_1h_2 + m_1h_5 + (m_2-m_1).h_6 = m_1h_7 + m_2h_3 \Rightarrow m_2.(h_3-h_6) = m_1.(h_2+h_5-h_6-h_7)$

$$m_2 = \frac{h_2 - h_7}{h_3 - h_6} = m_1 x \frac{426.5 - 209.6}{400.9 - 247.2} = 1.4m_1$$

From there we have the following system of equations:

$$\begin{cases} 40m_1 + 42.3m_2 = 4.749 \\ 1.4m_1 - m_2 = 0 \end{cases}$$

Solve the system of equations to find: m_1 = 0.047 kg/s and m_2 = 0.067 kg/s.

In which:

+ m_1: Refrigerant flow through the first stage compressor, kg/s

+ m_2: Refrigerant flow through the second stage compressor, kg/s

+ W_c: Compressor capacity, kw

+$n_c = 0.85$: Engine performance

Compressor cooling capacity:

$Q_o = m_1.(h_1-h_8)=0.047.(385.6-209.6) = 8.40$ kW (3.4)

Capacity of stage 1 compressor

$W_{c1} = m_1.(h_2-h_1) = 0.047.(425.6-385.6) = 1.91$ kW (3.5)

Capacity of stage 2 compressor

$W_{c2} = m_2(h_4-h_3) = 0.067.(443.2-400.9) = 2.84$ kW (3.6)

Cooling coefficient

$$\varepsilon = COP = \frac{Q_o}{W_{c1}+W_{c2}} = \frac{8.4}{1.91+2.84} = 1.77 \quad (3.7)$$

- Calculating exergy losses

Stage 1 compressor losses: E_{LMNC1}

$E_{LMNC1} = m_1T_a(s_2-s_1) = 0.047 \times 299.5 \times (1.855 - 1.834) = 0.3$ kW

Stage 2 compressor loss: E_{LMNC2}

$E_{LMNC2} = m_2T_a(s_4\text{-}s_3) = 0{,}067.299{,}5.(1{,}785 - 1{,}766) = 0{,}48$ kW

Condenser loss: E_{LTBNT}

$E_{LTNNT} = (E_4+E_9) - (E_5\text{-}E_{10})$ (3.8)

$$= m_2[(h_4\text{-}h_5)+T_a(s_5\text{-}s_4)]+m_{H20}[(c_p.T_v\text{-}c_p.T_{ra})+.c_p.\ln\frac{T_{ra}}{T_v})]+W_p+W_q$$

Water flow required for TBNT.

Capacity of condenser

$Q_k = m_{H20}.c_p.(t_{ra}\text{-}t_v) = W_{c1}+W_{c2}+Q_{o=}1.91 + 2.84 + 8.4=13.1$ kW (3.9)

$$=> m_{H20} = \frac{13.1}{4.186\ x\ (31.5-26.5)} = 0.63\ kg/s$$

Condenser exergy loss:

E_{LTBNT}=0.067 x [(443.2 – 247.2) + 299.5 x (1.159 – 1.785)] + 0.63 x 4.186 x [(299.5 – 304.5) + 299.5 x $\ln\frac{304.5}{299.5}$)] + 0.65 + 0.12 = 1.14 kW (3.9)

Throttle valve exergy loss 1:

$E_{LVTL1}=m_{tg}.T_a(s_6\text{-}s_5)$ = 0.01934 x 299.5 x (1.181 – 1.159) = 0.13 kW (3.12)

In which mtg is calculated according to the following formula

$m_{tg}=m_2\text{-}m_1$=0.067 – 0.047 = 0.02 kg/s

Throttle valve exergy loss 2:

$E_{LVTL2}= m_{tg}.T_a.(s_8\text{-}s_7)$=0.02 x 299.5 x (1.063 – 1.032) = 0.23 kW

Evaporator exergy loss:

$$E_{LTBBH} = (E_8\text{-}E_1)+Q_0\cdot\left(1-\frac{T_a}{T_f}\right)) = m_1.[(h_8\text{-}h_1) +T_a.(s_1\text{-}s_8)+Q_0\cdot\left(1-\frac{T_a}{T_f}\right)+2.W_f \quad (3.13)$$

= 0.047 x [(209.6 – 385.6) + 299.5 x (1.834 – 1.063)] + 8.4 x (1 - $\frac{299.5}{-45+273}$) + 2 x 0.216 = 0.3 kW

Intermediate tank exergy loss:

$E_{LBTG}=(m_1.h_2\text{-}m_1.T_a.s_2+m_1.h_5-m_1.T_a.s_5+m_{tg}.h_6\text{-}m_{tg}.T_a.s_6) - (m_2.h_3\text{-}m_2.T_a.s_3+m_1.h_7\text{-}m_1.T_a.s_7)$ = 0.301 kW

Total exergy loss of the refrigeration system:

$E_{LTOT} = E_{LMNC1} + E_{LMNC2} + E_{LTBNT} + E_{LVTL1} + E_{LVTL2} + E_{LTBH} + E_{LBTG}$ (3.14)

E_{LTOT} = 2.88 kW

Total exergy into the refrigeration system:

$E_{in}=W_{c1}+W_{c2}+W_p+W_q+2.W_{f1}=1.91+2.84+0.65 + 0.12 + 2 \times 0.216 = 6.79$ kW (3.15)

Exergy Performance

$$\eta_E = \frac{E_{in} - E_{Ltot}}{E_{in}} = \frac{6.79-2.88}{6.79} = 0.58 = 58\% \quad (3.16)$$

In the case of indirect fluid supply through low-pressure tanks, in addition to the above losses, there are also losses at the low-pressure tank determined by the following formula.

$$E_{LBCTA} = m_1.\left[h_8 - h_9 + T_a.(s_9 - s_8) + (h_{1'} - h_1) + T_a.(s_1 - s_{1'})\right] \quad (3.17)$$

With point 9 in a completely liquid state and supercooled in a low pressure tank, point 1' is in a saturated steam state with a dryness x ranging from 75 to 85%. From the initial data of the R22 refrigerant, using the software "Engineering Equation Solver" (EES) to determine the enthalpy and entropy parameters of points 9. Substituting the calculated values, the BCTA loss result is:

$E_{LBCTA}=0.047 \times [(209.6-149) + 299.5 \times (0.7976 - 1.063) + (314.6 - 385.6) + 299.5 \times (1.834 - 1.523)] = 0.041$ kw

Total exergy loss of the system is fed through BCTA

$E_{LTOT} = E_{LMNC1}+E_{LMNC2}+E_{LTBNT}+E_{LVT1}+E_{LVTL2}+E_{LTBBH}+E_{LBTG}+E_{BCTA} = 3.29$ kW (3.18)

=> $\eta_E = \frac{6.79-3.29}{6.79} = 0.52 = 52\%$

From the above calculation results, the exergy loss and exergy efficiency of the direct and indirect refrigeration system are summarized in Table 1.

Table 1 Loss, exergy efficiency and cooling coefficient of MCL R22

STT	Thông số	Direct translation	Indirect translation
1	Compressor capacity, kW	5.59	5.59
2	Compressor cooling capacity, kW	8.40	11.29
3	Compressor capacity, kW	1.91	1.91
4	Compressor capacity, kW	2.84	2.84
5	Coefficient of cooling (COP)	1.77	2.38
6	Compressor exergy loss, kW	0.30	0.30
7	Compressor exergy loss, kW	0.48	0.48
8	Condenser exergy loss, kW	1.14	1.14
9	Throttle valve 1 exergy loss, kW	0.13	0.13
10	Throttle valve 2 exergy loss, kW	0.23	0.23
11	Evaporator exergy loss, kW	0.30	0.41
12	Intermediate tank exergy loss, kW	0.30	0.30
13	Low pressure tank loss, kW		0.30
14	Total exergy loss, kW	2.88	3.29

15	Exegry efficiency, %	58.0	52.0

The variation in exergy loss of the equipment in the refrigeration system is shown.

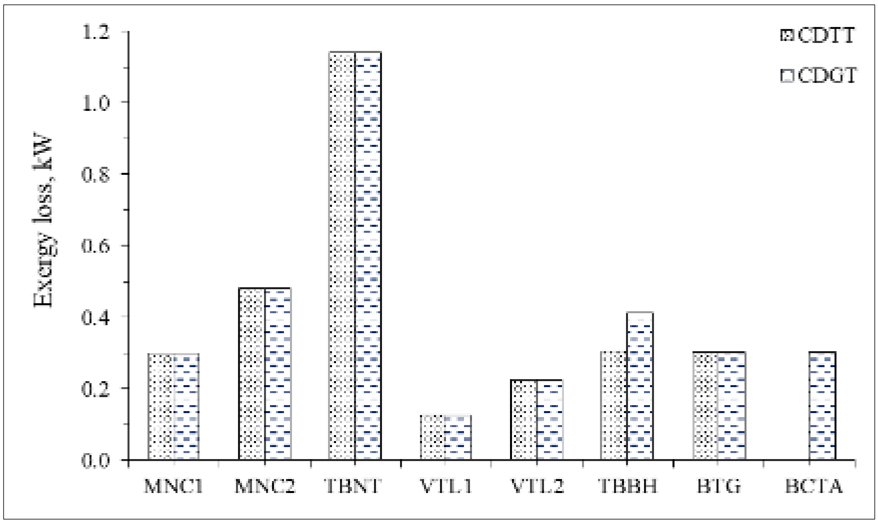

Figure 4 Exergy losses of MCL R22 appliances

The exergy loss of the condenser (TBNT) is the largest at 1.14 kW and the least at the throttling device 1 (VTL1) is 0.013 kW. This shows that it is necessary to design the condenser, choose a suitable fan water pump, perform regular maintenance as well as be able to reuse the waste heat in the condenser to limit the exergy loss for the refrigeration system. The exergy loss of the 2nd stage (high stage) compressor of 0.48 kW is larger than that of the 1st stage (low stage) compressor with a loss of 0.3 kW. The 2nd stage is 37.5% higher than the 1st stage compressor. The reason is that the mass flow rate of the second-stage compressor is larger than that of the first-stage compressor. The total loss and exergy efficiency of the refrigeration system according to the fluid supply methods shown. The total exergy loss of the direct fluid supply method is 2.88 kW, which is less than that of the indirect fluid supply method, which has a total exergy loss of 3.26 kW. The exergy efficiency of the direct fluid supply method is 58%, which is larger than that of the CDGT method, which has an exergy efficiency of only 52%. The exergy efficiency of the CDTT method is 6% higher than that of the CDGT method. The above difference is due to the indirect fluid supply method having an additional BCTA.

From the above calculation results, it shows that the total loss of the indirect fluid supply method is greater than the total loss of the direct fluid supply method. Thus, based on theoretical calculations, when installing, the direct fluid supply method will be chosen to reduce losses during operation and also reduce investment costs.

4. Conclusions

The analysis of exergy losses, exergy efficiency, and cooling coefficient of performance (COP) for the refrigerant R22 in refrigeration systems has provided valuable insights into the thermodynamic behavior of this widely used refrigerant. The study reveals that significant exergy losses occur primarily in the compressor and expansion valve, which are identified as the main contributors to the overall system inefficiencies. Improving these components could lead to notable gains in system performance. The findings also indicate that exergy efficiency varies with changes in operating conditions, emphasizing the importance of optimal design and operational strategies to enhance system effectiveness. Understanding the cooling COP of R22 further highlights its capability to deliver adequate cooling relative to its energy input, despite its known environmental concerns. While R22 has historically been a popular choice due to its favorable properties, the need for more environmentally friendly and energy-efficient refrigerants is increasingly evident. This study lays the groundwork for future research into alternative refrigerants and advanced refrigeration technologies, aiming to reduce energy consumption and mitigate environmental impact. The results from this exergy analysis offer a pathway for optimizing refrigeration systems and support the transition toward sustainable cooling solutions.

Compliance with ethical standards

Disclosure of conflict of interest

No conflict of interest to be disclosed.

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