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ENERGY PERFORMANCE ANALYSIS OF R32 AND R134A REFRIGERANT FOR SPRING POOL WATER HEATER

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Syafria Wildan Hadi[1], Yulius Tomy Wijaya[1], Intan Regina Elysabeth[1], Nyayu Aisyah[2], Robertus Dhimas D. Putra[1], Hifni Mukhtar Ariyadi[1]*

[1]Department of Mechanical and Industrial Engineering, Faculty of Engineering, Universitas Gadjah Mada, Indonesia
[2]Department of Mechanical Engineering, Vocational College, Universitas Gadjah Mada, Indonesia

Abstract:
For a country with a tropical climate, such as Indonesia, heat pump technologies are usually necessary in high-altitude regions, where the temperatures tend to be cooler, such as a pool heater for an artificial hot spring swimming pool. A vapor compression heat pump's optimal performance relies on its working fluid's characteristics. This paper aimed to assess the performance and compare the time required to heat a hot spring pool using heat pumps with two different refrigerants: R32 and R134a. A thermodynamic modeling approach was employed to determine the better refrigerant choice between R32 and R134a for the heat pump, aside from other important factors that are considered when selecting the refrigerant, such as safety and environmental impacts. Comparing the results of R134a and R32 refrigerants, R134a has a higher GWP value than R32, indicating that R32 is more environmentally friendly. Regarding safety, R134a has low toxicity and flammability, while R32 exhibits low toxicity and mild flammability. Thus, R134a is considered safer than R32. Furthermore, the COP value of R134a was found to be higher than that of R32, indicating that R134a offers greater efficiency for the heat pump compared to R32.

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1. INTRODUCTION

The advancements in vapor compression heat pump system technologies have profoundly impacted our daily needs. These systems can be broadly categorized into two types: those used for air conditioning and refrigeration and those used for heating [1, 2], commonly referred to as heat pumps. Heat pumps find particular application in regions with highlands or areas experiencing extreme weather changes, such as in Indonesia, where they are frequently used as a primary or secondary source to fulfill hot water requirements.

A heat pump is a thermal machine that transfers heat from one location to another [3]. While heat pumps operate on the same principle as air conditioning units, they have the added capability of providing heat. In the heating mode, a heat pump extracts heat from the outside air and transports it to the desired location. This process allows heat pumps to efficiently generate heat by utilizing the heat in the outside air as their primary heat source [4-6].

One notable advantage of heat pumps is their eco-friendliness and cost-effectiveness compared to conventional heating and cooling systems that rely on fossil fuels [7]. By utilizing the ambient heat from the environment, heat pumps minimize the need for direct combustion of fossil fuels [8], thereby reducing costs, greenhouse gas emissions

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*CONTACT: H.M. Ariyadi, e-mail: hifni.m.a@ugm.ac.id

[9] and contributing to environmental sustainability [10, 11]. Additionally, since heat pumps primarily rely on electricity for their operation, they offer the potential for utilizing renewable energy sources, further enhancing their environmental benefits.

Heat pumps also excel in terms of energy efficiency. They can achieve high levels of efficiency by employing advanced technologies and optimizing heat transfer. This efficiency translates into reduced energy consumption and lower operating costs, making heat pumps an attractive long-term investment for residential, commercial, and industrial applications [10, 12, 13].

A prediction by CEMAC [14] suggests a 4.5-times increase in air conditioning demand for non-Organization of Economic Coordination and Development (OECD) countries and a 1.3-times increase for OECD countries. Considering the need to reduce greenhouse gas emissions while maintaining performance, selecting the refrigerant for heat pump applications is an essential stage in the design process [15-17]. For example, a R32 refrigerant in a Heat Pump Water Heater offers advantages such as low Global Warming Potential (GWP) and near-zero ozone depletion potential. Compared to other refrigerants like R410A [18], R32 also demonstrates higher energy efficiency. Another factor in favor of choosing R32 is its significantly more negligible GWP, which is only one-third that of R410A [19].

Similarly, R134a is a suitable alternative due to its absence of chlorine atoms, making it environmentally benign without harming the ozone layer. It also exhibits good performance and safety levels [20]. Additionally, R134a refrigerant boasts a higher Coefficient of Performance (COP) than other refrigerants [21, 22].

For a country with a tropical climate such as Indonesia, heat pump technologies are usually necessary in high-altitude regions, where the temperatures tend to be cooler compared to lowland areas. Using a heat pump as a pool heater can benefit the artificial hot spring swimming pool, and the pool water can be maintained at a comfortable and consistent temperature throughout the year [23]. This ensures the pool remains usable even in cooler weather conditions, allowing for a more enjoyable swimming experience. A heat pump extends the usability of the pool by efficiently heating the water, enabling swimming and recreational activities during months when the water would otherwise be too cold for comfortable use.

This paper aimed to assess the performance and compare the time required to heat a 48 m^3 of hot spring pool using heat pumps with two different refrigerants, namely R32 and R134a. This becomes the novelty of this paper because there are still few heat pump applications for hot spring pools in Indonesia. The comparison includes the characteristics of the working fluids in terms of their properties and impact on the environment and the energetics performance regarding the heating capabilities applied as a spring pool water heater.

2. METHODOLOGY

2.1 System Description and Calculation Model

The heat pump water heater operates by extracting heat from the environment and utilizing it to raise the water temperature. The refrigerant cycle involves several components, such as a compressor, condenser, expansion valve, and evaporator [24]. On the other hand, the water cycle requires a control valve, hot water storage tank, water filter, and cold-water supply tank. The primary objective of this system is to provide hot water for a swimming pool, as depicted in Fig. 1.

The first stage of the refrigerant cycle is compression. The gas refrigerant enters the compressor, where it is compressed, increasing temperature and pressure. The second stage is condensation. The high-pressure gas refrigerant enters the condenser located in the water tank, releasing heat and elevating the water temperature. During this process, the gas refrigerant undergoes a phase change and transforms into a liquid. The third stage is expansion. The liquid refrigerant enters the expansion valve, where its pressure is reduced, enabling it to enter the evaporator pipe. In the fourth stage, evaporation occurs. The refrigerant within the evaporator pipe absorbs heat from the surroundings at an ambient temperature of 10°C, causing it to evaporate and change into a gaseous phase.

In the water cycle, the water temperature in the tank with the immersed condenser increases to 40°C. The hot water is then transferred to the hot water storage tank and directed to the swimming pool. As the water temperature in the pool decreases due to the low ambient temperature, the water passes through the filtration system. After the filtration process, the water enters the cold-water supply tank and finally flows back to the water tank with the immersed condenser. The control valve ensures that the pressure and flow of water remain consistent and in the same direction.

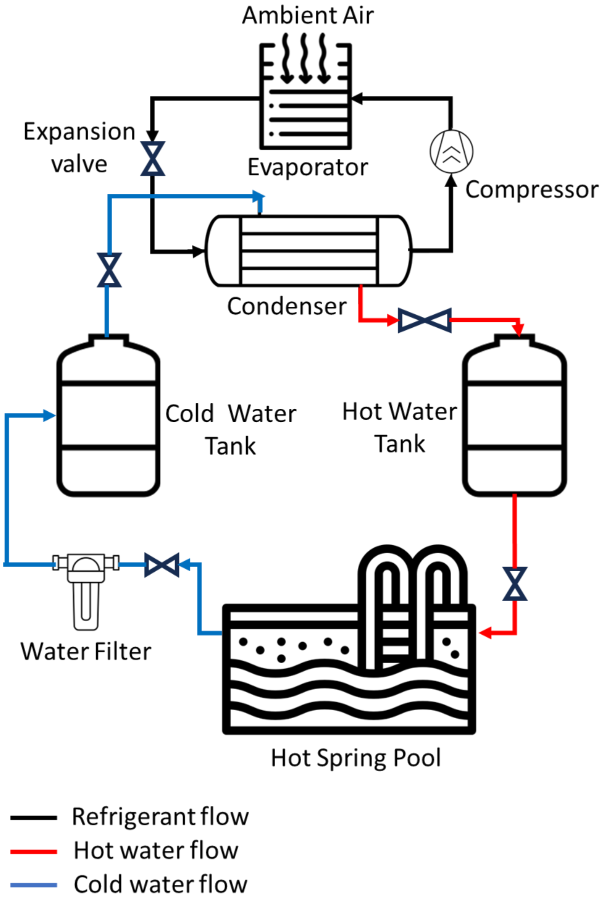

Fig. 1. Schematic of the heat pump system and heat water supply cycle

Several parameters were considered in this cycle. Firstly, the ambient temperature plays a significant role. Higher ambient temperatures facilitate heat absorption by the evaporator. Secondly, the properties of the refrigerant affect each stage of the process. Thirdly, the amount of water to be heated, whether large or small, impacts the heating time. Fourthly, the COP is crucial, as higher COP values indicate greater efficiency of the heat pump. Lastly, electrical power is utilized to operate the compressor. Increased electrical power results in a higher amount of transferred heat.

The calculation model and conditions of the system described above are conducted under specific assumptions:

a. The system is assumed to be steady, meaning that all variables remain constant over time.
b. The compressor is assumed to have an isentropic efficiency of 90%. This efficiency represents the compression process.
c. The expansion valve is assumed to operate under isenthalpic conditions, meaning no heat transfer occurs during the expansion process.
d. It is assumed that the condenser has a heat loss of 10%. This accounts for any energy losses during heat transfer.
e. The system operates above atmospheric pressure, indicating that the pressure within the system is higher than the surroundings.
f. The heat pump system is designed to operate within specific temperature ranges, which are specified in the given information.

The Equations (1-10) were used to evaluate the performance of the heat pump system, considering various parameters and variables to quantify the system's performance [25]. It considers factors such as the refrigerant mass flow rate, compressor work, and heating time. These parameters significantly determine the system's efficiency and effectiveness in heating the pool water.

The theoretical enthalpy difference in the compressor refers to the ideal change in enthalpy that occurs as the refrigerant passes through the compressor in an isentropic process. Theoretical enthalpy difference in the compressor can be calculated using Eq. (1):

$$\boldsymbol{W_{comp} = h_{2s} - h_1} \tag{1}$$

Where W_{comp} is the ideal specific work of the compressor (kJ/s), h_{2s} is the specific enthalpy at the compressor outlet (kJ/kg), and h_1 is the specific enthalpy at compressor inlet (kJ/kg). The actual specific enthalpy difference in the compressor represents the fundamental change in enthalpy that occurs during the compression process, considering isentropic efficiency. Actual specific enthalpy difference (h_2-h_1) in the compressor is calculated using Eq. (2), representing the actual specific work of the compressor, $W_{comp,act}$ (kJ/s):

$$\boldsymbol{W_{comp,act} = h_2 - h_1} \tag{2}$$

Where the actual specific enthalpy at the exit of compressor, h_2 (in KJ/kg) can be calculated using Eq. (3):

$$\boldsymbol{h_2 = h_1 + (h_{2s} - h_1)/\eta_{comp}} \tag{3}$$

Where $\boldsymbol{\eta_{comp}}$ is isentropic efficiency of compressor. The actual compressor power, $\boldsymbol{\dot{W}_{comp}}$ in kW, can be calculated using Eq (4) by defining the refrigerant mass flow rate ($\dot{m}_{ref}$, in kg/s):

$$\boldsymbol{\dot{W}_{comp} = \dot{m}_{ref} \cdot W_{comp}} \tag{4}$$

The heat rate to be transferred from the heat pump system refers to the amount of heat transferred from the system to heat the water using the condenser. Eq. (5) is used to find the actual heat rate to be transferred from the heat pump system to heat the water using a condenser ($\dot{Q}_{cond}$, kW):

$$\begin{aligned} \dot{Q}_{cond} &= \dot{m}_{ref} \cdot Q_{cond} \\ &= \dot{m}_{ref}\,(h_2 - h_3) \end{aligned} \qquad (5)$$

Where, h_3 is the specific enthalpy at condenser outlet (kJ/kg). The heat transfer from the source refers to the amount of heat absorbed from the surrounding air by the evaporator. Heat transfer from the source (air surrounding) that is absorbed by the evaporator ($\dot{Q}_{evap}$, kW) is calculated using Eq. (6):

$$\begin{aligned} \dot{Q}_{evap} &= \dot{m}_{ref} \cdot Q_{evap} \\ &= \dot{m}_{ref}\,(h_1 - h_4) \end{aligned} \qquad (6)$$

Where h_4 is the specific enthalpy at the condenser inlet (kJ/kg), which is the same value as h_3 as the expansion process is isenthalpic. The heat that needs to be fulfilled to achieve the desired temperature increase from an ambient temperature 10°C to 40°C is calculated using Eq (7). The mass flow rate of water ($\dot{m}_{water}$, kg/s) is determined by the density of water (ρ_{water}, kg/m^3) and the volume of water (V_{water}, m^3). Multiplying these values by the specific heat capacity (Cp_{water}, kJ/kg K) and the temperature difference (ΔT, K) yields the heat required to raise the water temperature.

$$\begin{aligned} Q_{water} &= m_{water}\, Cp_{water}\, \Delta T \\ &= \rho_{water} v_{water} Cp_{water} \Delta T \end{aligned} \qquad (7)$$

Next, the heat transferred from the condenser to the water ($\dot{Q}_{water}$, kW) is calculated, considering a 10% heat loss in the condenser and pool. This can be calculated using Eq (8):

$$\dot{Q}_{water} = 90\% \cdot \dot{Q}_{cond} \qquad (8)$$

To find the time required (t, hours) to increase the water temperature, Eq. (9) is applied:

$$t = Q_{water}/(\dot{Q}_{water} \times 3600) \qquad (9)$$

The COP of the refrigerants used in the heat pump will be calculated using Eq. (10):

$$COP = \dot{Q}_{cond}/\dot{W}_{comp} \qquad (10)$$

2.2 Refrigerant Properties

Table 1 presents a comparison of the properties of the refrigerants used in this evaluation. The refrigerant evaluation was conducted based on [21, 26, 27]. This paper evaluated two refrigerants, namely refrigerant R32 and R134a.

Table 1. Refrigerant Properties [21, 26, 27]

	Refrigerant	
	R32	**R134a**
IUPAC Name	CH_2F_2 (Difluoro-methane)	CH_2FCF_3 (tetrafluoro-ethane)
Molar Mass	52.02 g/mol	102.03 g/mol
Critical Pressure	5.782 MPa	4.059 MPa
Critical Temperature	351.26 K	374.21 K
Freezing Point	137.15 K	170.15 K
Normal Boiling Point	221.45 K	247.08 K
ASHRAE Classification	A2L (*low toxic* dan *mild flammability*)	A1 (*low toxic* dan *no flame propagation*)
Ozone Depletion Potential (ODP)	0	0
Global Warming Potential (GWP)	675	1600

The evaluation considers the conditions of two types of refrigerants considering the ambient temperature, T_{amb}, the target water temperature, T_{water}, and pool water volume, V_{water}. Temperature pinch points at the condenser (TPP_{cond}) and at the evaporator (TPP_{evap}) were also defined. Details of the operation condition are explained in Table 2.

Table 2. Operating Condition of the System

Parameter	**Unit**	**Value**
T_{amb}	°C	10
T_{water}	°C	40
V_{water}	m^3	48
TPP_{cond}	°C	5
TPP_{evap}	°C	10

3. RESULTS AND DISCUSSIONS

3.1 Refrigerant Comparison

Two refrigerants R32 and R134a were compared to determine the suitable refrigerant for the heat pump system in terms of properties, environmental aspects, and performance. By collecting the data for each refrigerant's properties, as illustrated in Fig. 2, the evaluation for both refrigerants can be conducted.

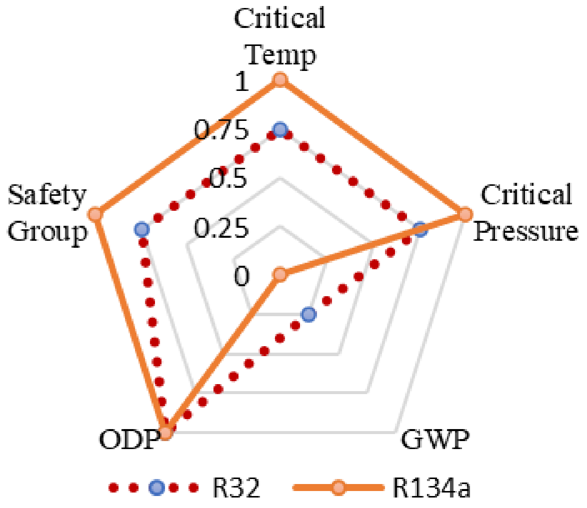

Fig. 2. Spider plot to compare R32 and R134a

From Fig. 2, it can be revealed that R134a has a wider spider plot compared to R32. It means that R134a is better than R32 regarding the refrigerant's physical properties. R134a has a higher critical temperature, lower critical pressure, A1 in the safety group, and similar ODP (zero ODP). However, R134a has a relatively high GWP (about 1430, while R32 is 675), which makes R32 better in terms of environmental aspects.

3.2 Performance Evaluation

A performance evaluation is conducted after evaluating the properties and environmental aspects of both refrigerants. The results of these simulations are presented in Table 3, which displays the data based on T_{evap} (evaporator temperature) and $T_{c,out}$ (condenser outlet temperature) for the different refrigerants discussed earlier. Additionally, these results can be visualized on the temperature and entropy (*T-s)* diagram, as shown in Fig. 3 (a) and (b).

The COP obtained for refrigerant R134a is greater than the COP for refrigerant R32. The COP for refrigerant R134a is 5.98, and R32 is 5.77.

Once the simulation data was collected for each refrigerant, calculations were performed by substituting the data into Equations (1) to (9), while varying the mass flow rate of the refrigerant. The mass flow rate of water was kept constant at 0.554 kg/s. By employing Equation (6), the power requirement of the compressor was determined for different mass flow rates. Furthermore, the analysis aimed to determine the time required to heat the 48 m^3 water tank from 10°C to 40°C using Equation (9) as illustrated in Fig. 3 (a) and (b).

Table 3. Simulation results of refrigerant R32 and R134a

Stage	*T* (K)	*P* (MPa)	*h* (kJ/kg)	*s* (kJ/kg K)
	R32			
1	283.15	1.102	516.65	2.119
2	351.56	3.130	558.05	2.119
3	323.15	3.130	297.14	1.317
4	283.15	1.102	297.14	-
	R134a			
Stage	***T* (K)**	***P* (MPa)**	***h* (kJ/kg)**	***s* (kJ/kg K)**
1	283	0.413	404.23	1.722
2	326.59	1.313	428.21	1.722
3	323	1.313	271.39	1.237
4	283	0.413	271.39	-

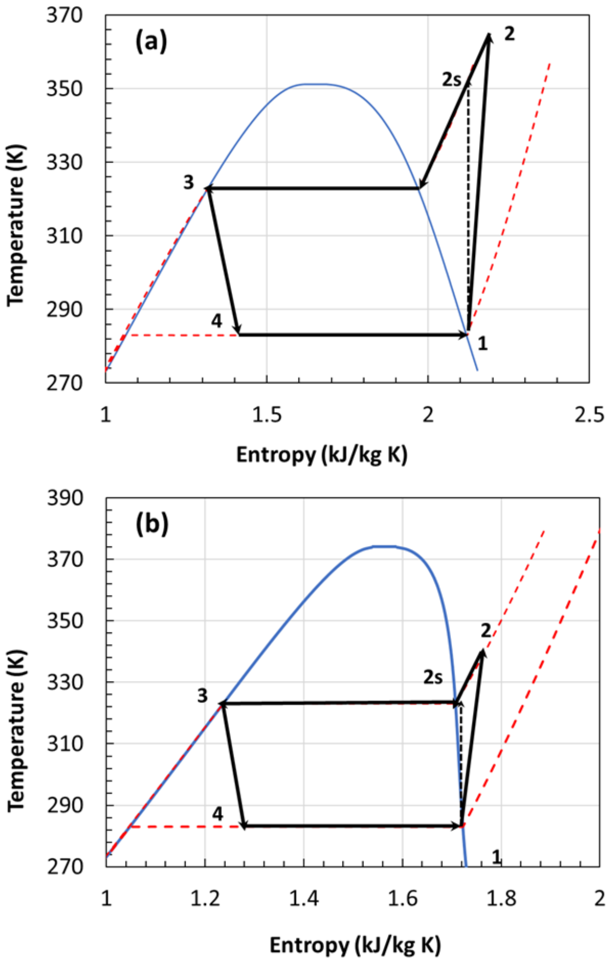

Fig. 3. Cycle plot in T-s diagram for (a) R32, (b) R134a

While Fig. 4 reveals several important insights. Firstly, it shows that increasing the mass flow rate of the refrigerant leads to higher compressor work. This relationship makes sense since a higher compressor work requires more refrigerant to be circulated through the system to meet the heat transfer demands. On the other hand, a lower compressor work would result in a lower mass flow rate of the refrigerant.

Furthermore, the analysis indicates that the time required to heat the cold-water supply decreases as the compressor work increases. This relationship implies that a more powerful compressor can transfer heat more efficiently, resulting in faster water heating. Conversely, a lower compressor work would lead to a longer heating time.

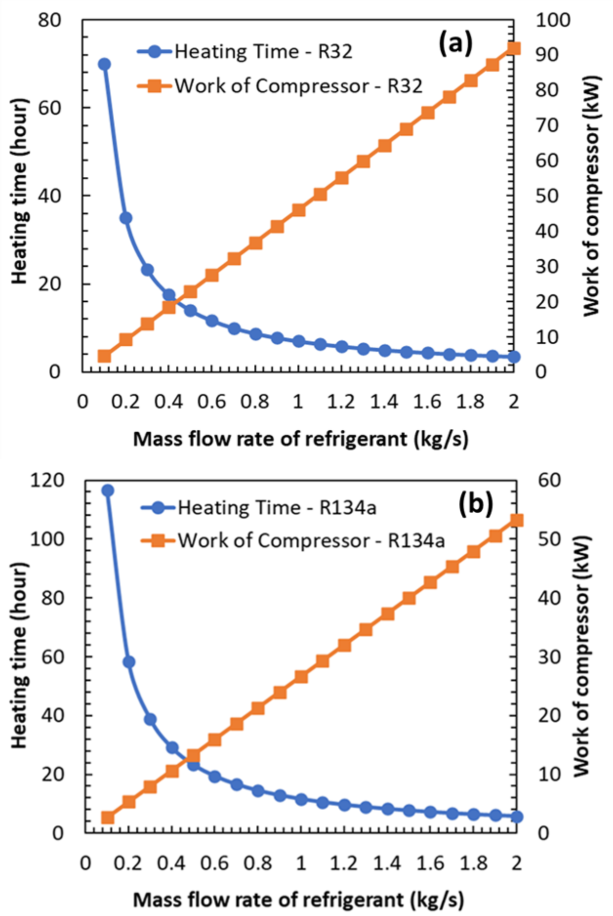

Fig. 4. (a) Compressor works, (b) Time consumption to heat the 48 m^3 water

As mentioned previously, in Fig. 4 (a), an increase in 4 kW compressor work could lead to a significant decrease in heating time, as seen in the beginning trend of the heating time curve with blue color. However, after the blue and yellow curves met at the intersection point, the trend of the heating time graph began to slope, which means that even if the compressor work continues to increase, it does not provide a significant change in heating time. This point is then referred to be an optimum point. In addition, it is clearly stated that when the compressor worked 60 kW and continued to increase to 90 kW, the heating time barely experienced a significant change in value and was almost constant at 4 hours.

The same trend also occurs in Fig. 4 (b), which uses a different refrigerant, R134a. When the blue and yellow curves met at the intersection, the compressor works noticed as 12 kW while the heating time is 24 hours. The trend of the blue curve begins to slope down, causing very little or even no significant change in heating time, as shown in compressor work increases from 30 kW to 54 kW, heating time is almost the same, which is around 4 hours.

The analysis also highlights a comparison between refrigerant R32 and R134a. It shows that, at the same mass flow rate, refrigerant R32 requires more compressor work than R134a. This means that R32 demands a higher electrical power input to operate the compressor and generate the necessary work. However, despite requiring more compressor work, R32 has a shorter time to heat the cold-water supply than R134a. This finding suggests that R32 has a higher performance, enabling it to heat the water despite the higher compressor work requirement rapidly.

4. CONCLUSION

Several conclusions can be drawn based on the modelling comparison between refrigerant R32 and R134a. Firstly, increasing the mass flow rate of the refrigerant results in a shorter time required to heat the water. However, it is important to note that a higher mass flow rate also leads to an increased workload for the compressor, requiring greater electrical power.

Furthermore, the calculation of COP reveals that refrigerant R134a has a higher COP (5.98) compared to refrigerant R32 (5.77). This finding is in accordance with the research conducted by Nie et al. [21]. It is evident that refrigerant R32 is more environmentally friendly, while R134a is safer in terms of its impact on safety.

Ultimately, the choice of refrigerant for the heat pump system depends on several factors, including the COP value, safety considerations, and the environmental impact. Balancing these factors is crucial in selecting the most suitable refrigerant for a given application.

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