J u r n a l P o l i m e s i n
Department of Mechanical Engineering
State Polytechnic of Lhokseumawe
http://e-jurnal.pnl.ac.id/polimesin

e-ISSN : 2549-1999 No. : 5 Month : October
p-ISSN : 1693-5462 Volume : 21 Year : 2023

Article Processing Dates: Received on 2023-07-18, Reviewed on 2023-08-17, Revised on 2023-09-09, Accepted on 2023-09-22 and Available online on 2023-10-29

Designing of Evaporator Length in Very Low Temperature Chest Freezer by using Environmentally Friendly Refrigerant R290

Kasni Sumeru[1], Deden Kahfi[1], Apip Badarudin[1*], Nani Yuningsih[1], Luga Martin Simbolon[1], Mohamad Firdaus bin Sukri[2].
[1]Department of Refrigeration& Air Conditioning, PoliteknikNegeri Bandung, Bandung 40559, Indonesia
[2]Faculty of Mechanical Engineering UniversitiTeknikalMalaysia Melaka, Hang Tuah Jaya,Durian Tunggal Melaka 76100, Malaysia.
*Corresponding Author: apipbdr@polban.ac.id

Abstract

Chest freezers generally use R600a or R134a as working fluids. When using R600a, the minimum cabin temperature is only -10°C, whereas when using R134a, it can reach -25°C. The purpose of this study is to calculate the evaporator length of a chest freezer that uses R290 as a refrigerant so that its cabin temperature can reach below 35 °C, lower than the cabin temperature of a typical chest freezer. Calculation of the evaporator pipe length is done using the forced convection heat transfer equation to calculate the heat transfer coefficient inside the evaporator pipe and natural heat transfer to calculate the heat transfer coefficient outside the evaporator pipe. Based on the calculations, the chest freezer has a compressor capacity of 200 W, an evaporator length of 3.57 m, and a diameter of 3/8 inch or 9.52 mm. The test results show that the temperature of the chest freezer cabin can reach -36oC in the 36th minute with a cooling capacity of 289 W, while the input power and COP are 198 watts and 1.46, respectively. Compared to R134a, the use of R290 is more advantageous. In addition to lower cabin temperatures, it is also much more environmentally friendly, because the GWP (global warming potential) value of R134a is much higher than that of R290. It means that the use of R290 as a working fluid in the chest freezer will significantly reduce emissions of gases that cause global warming.

Keywords:
chest freezer, GWP, evaporator length, R134a, R290.

1. Introduction

A chest freezer is designed to store food products at freezing temperatures. The refrigeration system in the chest freezer uses a vapor compression cycle, which has four main components, namely the compressor, condenser, expansion valve, and evaporator[1][3]. To produce the desired design temperature, the capacity of the four components must match. If not, then the desired cold temperature is not achieved and the cooling capacity is also not optimal.

To produce very low temperatures, namely below 30°C in the chest freezer cabin, the refrigerant used must have a normal boiling-point temperature (NBP) below -30°C[4]. Nowadays chest freezers on the market use R134a and R600a. However, because both NBPs are above -30°C, as a result, the temperature of the chest freezer cabin cannot reach -30°C, because the temperatures of NBP R134a and R600a are -26.1°C and -11.8°C[5], [6], respectively. Several refrigerants that have NBP temperatures below 30°C are HCFC (hydrochlorocarbon), namely R22, HFCs (hydrofluorocarbon),i.e., R404A, R410A and R407C, and HC (hydrocarbon), namely R290[5], [7], [8]. Some of the refrigerants mentioned have been banned for use, namely R22, and will be banned in the near future, i.e., R404A, R410A and 407C[9][12]. Therefore, the best refrigerant alternative for very low temperature chest freezers (below 30°C) is R290. This working fluid is environmentally friendly, because it has a GWP value of only 3 and an ODP (ozone depletion potential) value of zero, and it can be self-produced in Indonesia from natural gas refineries. However, one of the drawbacks of R290 is its flammability, which is at a concentration of 2.1% to 9.5% by volume or approximately 38 gr/m^3 to 172 gr/m^3[13], [14].The filling mass of R290 forsmall-capacity chest freezers is generally no more than 150 gr. If the freezer is placed in a room with a size of (3x4) m^2, with a height of about 3.5 m, then if there is a total leakage of R290, the maximum concentration in the room is only 3.6 gr/m^3.This means that the minimum concentration is not reached, in other words, the potential for fire will not occur. The characteristics of the working fluid mentioned above are shown in Table 1. The table shows that the GWP values of HCFCs and HFCs are still very high, above one thousand, while R290 only has a GWP value of 3[15][18].

Table 1. Characteristics of various refrigerants [12], [15]–[17]

Refrigerant	NBP(°C)	GWP	ODP
R134a	-26.1	1300	0
R404A	-46.5	3922	0
R410A	-50.5	2088	0
R407C	-43.6	1530	0
R22	-40.8	1810	0.055
R600a	-11.8	3	0
R290	-42.1	3	0

There are not many articles that discuss the design of long evaporator pipes in refrigeration systems, because generally, the refrigeration industries do not publish their research results. This is because the optimal length of the evaporator will produce optimal performances, so the secrecy of the maximum length of the evaporator pipe is the property of the manufacturer. Based on reference searches, the author obtained an article that discussed calculating the length of the evaporator in a refrigeration system [19][16]. For this reason, the method used in this study will follow the article, but by adding equations that are adjusted to the chest freezer to be designed.

The purpose of this research is to design the length of the evaporator pipe in the chest freezer so that the cabin temperature can reach below -35°C, using R290 as the working fluid.Very low temperatures (below 30°C) are generally used to store seafood, ice cream and some types of vaccines[20]. The use of R290 as a substitute for R134a will reduce the potential for emissions of gases that cause global warming from the refrigeration sector because the GWP of R290 is much lower than R134a.According to Lebrun and Ziegler [21]the refrigeration sector contributes 7.8%. Contribution to global warming can be direct, which is through refrigerant leakage, or indirect, which is from the input power source of the refrigeration system. The percentage of direct contribution is 37% and indirectly 63% [22].

In addition, because R290 can be produced in Indonesia, the use of R290 in the refrigeration system will reduce refrigerant imports, because currently all refrigerants (CFCs, HCFCs, and HFCs) used in Indonesia are still imported.

2. Materials and Methods

This study uses a condensing unit with a compressor capacity of 200 W. Before calculating the length of the evaporator pipe, the first step is to determine the design data that will be used as a reference for calculations. The design data is shown in Table 2. Based on these data, the refrigeration cycle is plotted on the P-h (pressure-enthalpy) diagram R290, as shown in Fig. 1. Since the input power of the compressor (P) is known, based on Fig.1 it can be calculated the mass flow rate of the refrigerant in the system using Eq. (*1*).

Table 2. The designdata of chest freezer

No	Parameter	Designed
1.	Condensing temperature	40°C
2.	Evaporating temperature	-40°C
3.	Degree of subcooling	5 K
4.	Degree of superheating	7 K
5.	Input power of compressor	200 W
6.	Isentropic efficiency	0.6

$$\dot{m} = \frac{P}{(h_2 - h_1)} \qquad (1)$$

After the mass flow rate can be determined, using Fig. 1, the cooling capacity (Q) and COP are expressed by Eqs. (*2*) and (*3*), namely,

$$Q = \dot{m} \cdot (h_1 - h_3) \qquad (2)$$

$$COP = \frac{Q}{P} \qquad (3)$$

Where $Q$is cooling capacity, W. $\dot{m}$ is mass flow rate of refrigerant, kg/s. $h_1$is enthalpy at point 1, kJ/kg. $h_2$isenthalpy at point 2, kJ/kg. $h_3$isenthalpy at point 3, kJ/kg. Pisinput power of compressor, W

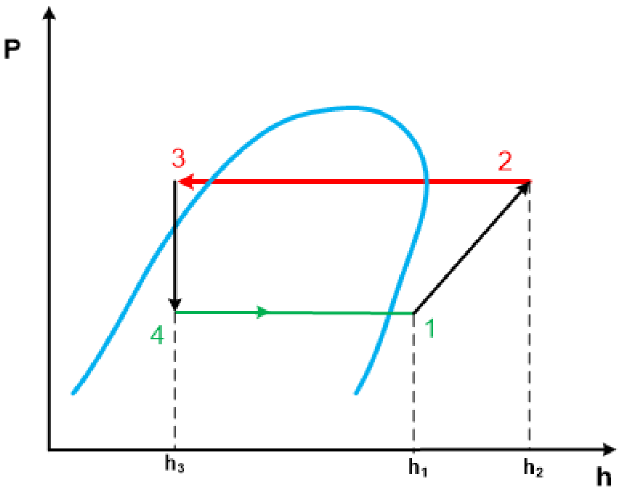

Fig. 1. The chest freezer refrigeration cycle on the P-h diagram is based on design data

In this study, the length of the evaporator pipe was calculated by calculating the heat transfer from the air in the cabin to the outer surface of the evaporator pipe, namely by Eq. (*4*)[23].

$$Q = \frac{2\pi \cdot L_{evap} \cdot \ (T_o - T_i)}{\left(\frac{1}{ri \cdot hi}\right) + \frac{\ln(\frac{ro}{ri})}{k} + (\left(\frac{1}{ro \cdot ho}\right)} \qquad (4)$$

Where$Q$iscooling capacity, Eq. (*1*), $L_{evap.}$islength of evaporator pipe, h_i isheat transfer coefficient inside pipe, $h_o$is heat transfer coefficient outside pipe, $r_i$is radius of inside pipa, $r_o$is radius of outside pipe, $T_o$istemperature of cabin before cooling, $T_i$istemperature of refrigerant inside pipe, $k$isconduction coefficient of pipe.

To solve Eq. (*4*), the values of "h_i" and "h_o" must be known. The value of "*hi*" is calculated by the forced convection equation, as shown in Eq. (*5*)[4].

$$h_i = z^2 \frac{G^2 \cdot \Delta T}{D_i} \qquad (5)$$

Where $z$is constant, 0,99 for R290[4], G ismass velocity, kg/s, ΔTisrefrigerant temp. difference with cabin, °C.$D_i$is diameter of inner pipe, m

Meanwhile, the heat transfer between the outer surface of the evaporator pipe and the cabin air is natural convection, as a result, the equation used to calculate "h_o" is Eq. (*6*)[23].

$$NU_f = \frac{h_o \cdot D}{k_f} = \mathrm{C}(Gr_f Pr_f)^m \qquad (6)$$

Where Gr_fisGrashoff number at film temperature, Pr_f = Prandlt numberat film temperature

The *Pr* value can be seen from the table of properties of the R290 refrigerant, while the *Gr* value is calculated by the equation,

$$Gr = \frac{g.\beta\ .(T_W - T_\infty).D^3}{v^2} \qquad (7)$$

Where g = gravity, 9.8 m/s^2.$\beta = \frac{1}{T_f}$ = coefficient of expansion volume, $v$is kinematic viscosity of refrigerant, m^2/s

By using Eqs. (*4*), (*5*), (*6*), and (*7*), the length of the evaporator can be calculated. Based on the length of this evaporator, it is installed into the chest freezer cabin.

In collecting data, several instruments were used as shown in Table 3. Thermocouples are used to measure temperature at several locations, such as at the suction, discharge, condenser and evaporator inlets and outlets. Fig. 2 depicts thermocouples measuring the compressor suction and discharge temperatures, as well as the condenser temperature. The compressor is positioned at the bottom of the chest freezer while the suction and discharge pressure gauges are shown in Fig. 3. The figure also shows thermocouples to measure the evaporator temperature.Low-pressure and high-pressure gauges are used to measure condensing and evaporating pressures. Based on the condensing and evaporating pressures, the condensing and evaporating temperatures can be determined.The input power consumed by the chest freezer is known by measuring the electric current and voltage in the system.

Table 3. The accuracies of measuring instruments

No.	Instrument	Accuracy
1.	Thermocouple K-type	± 0,1°C
2.	High pressure gauge	± 0,5 bar
3.	Low pressure gauge	± 0,1 bar
4.	Ampere meter	± 0,1 A
5.	Volt meter	± 1 V

3. Results and Discussion

3.1. Calculation Results

The calculation of the length of the evaporator pipe iscarried out manually, using Eq. (*4*). The parameters in the equation that are not yet known are the values of "h_i" and "h_o". To calculate the value of "hi", Eq. (5) is used, while the value of "h_o" is calculated using Eqs. (6) and (7). Using design data in Table 2, the values of "h_i" and "h_o" are:

$$h_i = 31.178,76 \text{ J/m}^2.\text{s.}^{\circ}\text{C}$$
$$h_o = 24.38 \text{ J/m}^2.\text{s.}^{\circ}\text{C}$$

By using a 3/8-inch diameter copper pipe, the inner and outer radius of the copper pipe are:

$$r_i = 0{,}0043825 \ m$$
$$r_o = 0{,}0047625 \ m$$

The next step to calculate Eq. (4) is to determine the cooling capacity of the chest freezer. The amount of cooling capacity is calculated using Eq. (3). Based on the design data in Table 2, the *COP*can be determined by depicting the refrigeration cycle of the chest freezer in the P-h diagram in Fig. 1. The *COP* is,

$$COP = 1.21$$

After the COP is known, and the input power is also known, the cooling capacity (Q) can be calculated by Eq. (3). As mentioned above, this research uses a compressor with an input power of 200 W. Furthermore, the cooling capacity is,

$$Q = COP \text{ x } P = 1.21 \text{ x } 200 = 242 \text{ W}$$

By substituting the above values into Eq. (4), the length of the evaporator pipe is,

$$L_{evap} = 3.57 \text{ m}$$

After knowing the length of the evaporator pipe, the next step is to install it in the chest freezer cabin, as shown in Fig. 3. In the figure it can be seen that the pipe used is a copper pipe with a diameter of 3/8 inch or 9.52 mm. Because this is still a prototype, the position of the evaporator pipe is still outside the chest freezer wall. When optimal results have been obtained, the position of the evaporator pipe is installed inside the wall of the chest freezer and it will not be visible in the cabin.

Fig. 2. Thermocouples to measure compressor suction and discharge temperatures as well as condenser temperature.

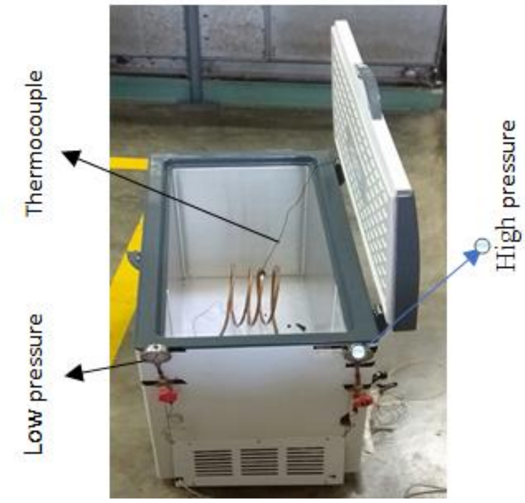

Fig. 3. The position of the evaporator pipe in the chest freezer cabin.

3.2. Performance Test

This section will discuss the differences between designed data and measured results. The results of the cabin temperature test are presented in Fig. 4. The figure shows that the cabin temperature can reach -36°C in 36 minutes and is stable for up to 120 minutes. In the picture, it can be seen that from the time the chest freezer was operated up to 36 minutes, the cabin temperature dropped drastically. After that the cabin temperature stabilized around -36°C.

Cabin temperature depends on the evaporating temperature in the evaporator [4]. Based on the test, the evaporation temperature of the chest freezer is -38°C, as shown in Fig. 5.From the figure it can be seen that the designed evaporating temperature is slightly lower, which is 2°C, than the measured evaporating temperature.Because the evaporating temperature is -38°C, the minimum cabin temperature will be several degrees above the evaporating temperature.In this study, there was a difference of 2°C between the evaporating temperature and the cabin temperature. Besides being affected by the length of the evaporator tube, the evaporating temperature is also strongly influenced by the length of the capillary tube.For this reason, further research by varying the length of the capillary tube to obtain an evaporating temperature of -40°C still needs to be carried out.

From this research it can be seen that the replacement of R134a with R290 can be applied, it can even lower the cabin temperature. When using R134a, the minimum cabin temperature is only around -25°C, while using R290, the cabin temperature can reach around -36°C.In addition, the replacement of R134a with R290 will also reduce the contribution of the refrigeration sector to global warming, as has been reported by several researchers [7], [21].

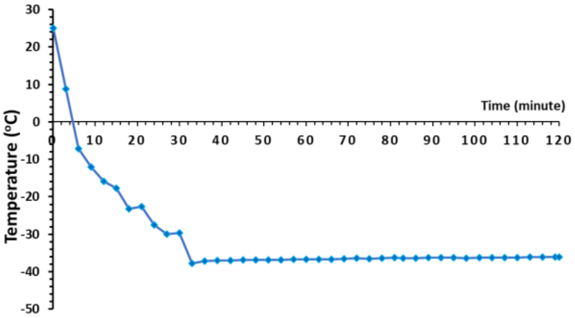

Fig. 4. Results of measured temperature in the chest freezer cabin

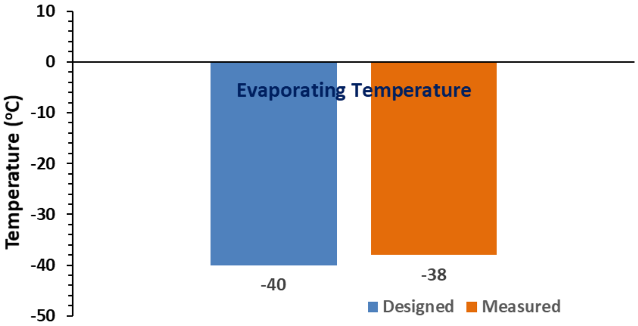

Fig. 5. The evaporating temperatures: Designed versusMeasured results

It has been mentioned above that the superheating designed temperature in this study is 7 K, as shown in Table 2. If the evaporator length calculation is accurate, the superheating test results are the same or slightly different from the design. However, if the superheating measured results are much lower than that of the design, the length of the evaporator pipe is shorter than the ideal length.Conversely, if the superheating measured results are higher than that of the designed, the length of the evaporator pipe that is designed is longer than the ideal length[4].

Fig. 6 shows that the superheating temperature measured results are lower than that of the designed, namely 7 K and 1.8 K, respectively.As previously explained, this is probably due to the length of the calculated evaporator pipe being shorter than the ideal length. For this reason, further research is needed by varying the length of the evaporator pipe in order to obtain optimal chest freezer performance. If the refrigeration system in the chest freezer works optimally, the designed temperature can be reached more quickly.

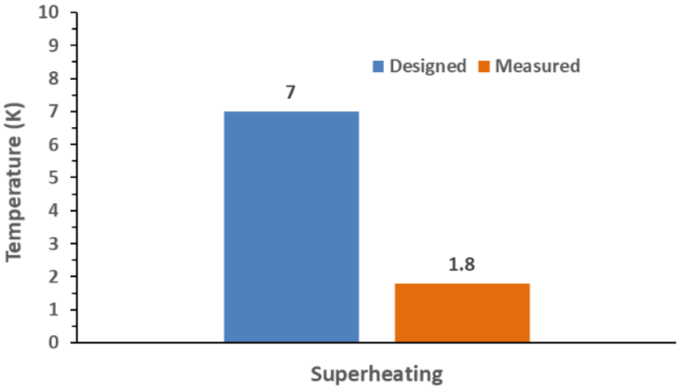

Fig. 6. Superheating comparison: Designed versus Measured results

For superheating, the main influencing factor is the evaporator dimension, while for subcooling, the main influencing parameter is the condenser dimension[4], [19]. This study uses a condensing unit, which consists of a compressor and a condenser, where the dimensions of the condenser have been designed according to the capacity of the compressor by the manufacturer. Therefore, the difference in design subcooling is not much different from the measurement results, which is only 0.1°C, as shown in Fig. 7. This means that the capacity of the condenser used is in accordance with the capacity of the installed compressor.

In general, if the dimensions of the condenser are smaller than the ideal length, it will result in smaller subcooling.Conversely, if the condenser dimensions are larger than the ideal length, the subcooling will be higher.

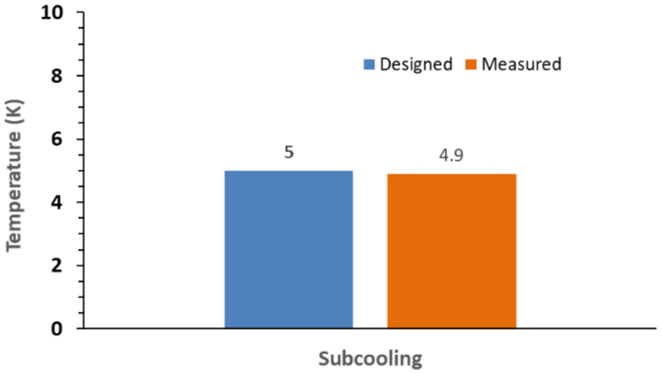

Fig. 7. Subcooling comparison: Designed versus Measures results

Figure 8 shows a comparison of the designed input power with the measured. From the figure, it can be seen that the designed input power is slightly different from the measured, which is only 2 W. This is one of the indicators that the evaporator pipe length calculation method in this study is relatively valid. If the length of the evaporator pipe that is designed is very different from the ideal length, it will produce a measured input power that is also very different from what was designed.This is because if the length of the evaporator is not suitable, it will cause the refrigerant charging mass to be incorrect too.The mass of charging the refrigerant in the refrigeration system affects the input power consumed by the compressor to circulate the refrigerant in the system [19].

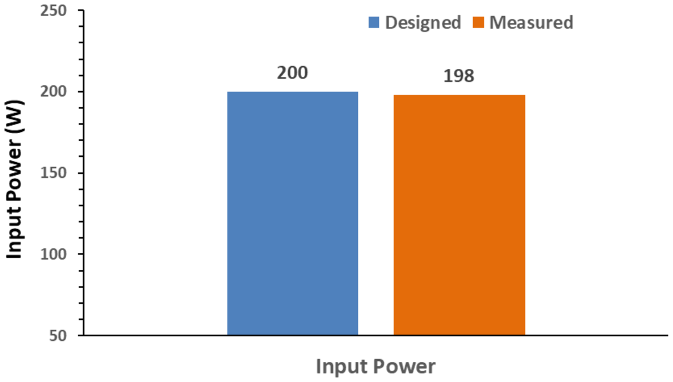

Fig. 8. The input power comparison: Designed versus Measured results

In a refrigeration system, cooling capacity is the most important parameter. Because the cooling capacity is a measure of how large the quantity of heat is drawn from inside the cabin or product by the evaporator.For the same compressor capacity, a good refrigeration system is a system that has a high cooling capacity.Figure 9 displays a comparison of the designed cooling capacity with the measured results. It can be seen that the designed cooling capacity is smaller than that of the measured. It means that the ability of the installed evaporator to produce cold temperatures faster in the cabin compared to the design.The difference between the designed cooling capacity and the measured one is 47 W.When viewed from the cooling capacity side, the measured results are better than what was designed.This is because the measured evaporating temperature is higher than that of the design.In general, a higher evaporating temperature will result in a greater cooling capacity for the same condensing temperature [4], [24].

After knowing the input power and cooling capacity, the next step is to determine the COP of the refrigeration system. Fig. 10 illustrates a comparison of the designed COP with the measured. It can be seen that the measured COP is slightly higher than designed, namely 1.46 compared to 1.21. In other words, the measured COP is 0.25 higher than that of the designed.

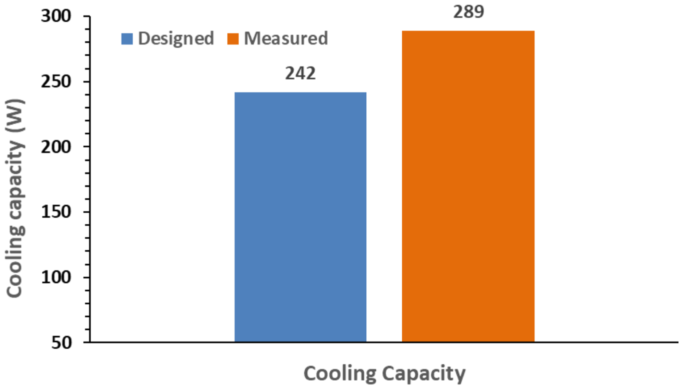

Fig. 9. The cooling capacity comparison: Designed versus Measured results

The main factors that affect the COP of refrigeration systems are evaporating and condensing temperatures [24]. The higher the evaporating temperature, the higher the COP. Conversely, the lower the condensing temperature, the higher the COP.

As explained in Fig. 5, the measured evaporating temperature of the chest freezer is higher than that of the designed, namely -38oC and -40^{o}C. This means that if the system has the same condensing temperature, the measured COP of the chest freezer will have a higher COP than what was designed.

Another factor that affects COP is the condensing temperature [7], [24]. The evaporating temperature is determined based on the desired cabin temperature, while the condensing temperature is greatly influenced by the ambient temperature.

In general, the condensing temperature is about 8^{o}C to 12^{o}C higher than the ambient temperature. The higher the ambient temperature, the lower the COP. In this design, it is assumed that the condensing temperature is 40^{o}C because it is assumed that the ambient temperature is around 30^{o}C.Whereas during the test, the ambient temperature was only around 26^{o}C, so the condensing temperature was only 33.1^{o}C, as shown in Figure 11. Because the designed condensing temperature is higher than that of the measured, the measured COP will be higher than that of the design.

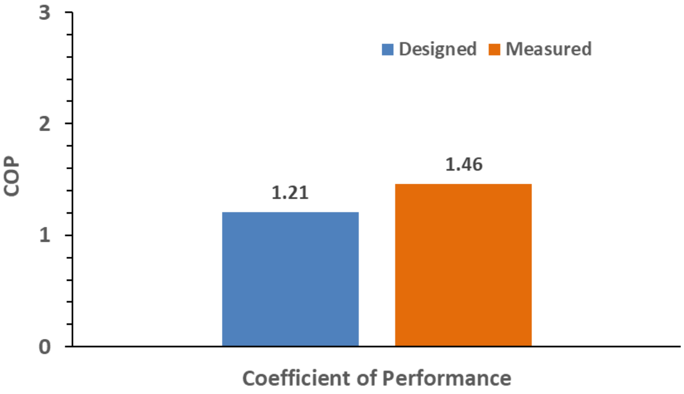

Fig. 10. The COP comparison: Designed versus Measured results

Based on the comparison of the parameters discussed above, it can be seen that the difference in the measured results is not much different from the design.This means that the evaporator pipe length calculated by the method described above is relatively valid. To obtain more optimal results, further research is still needed.

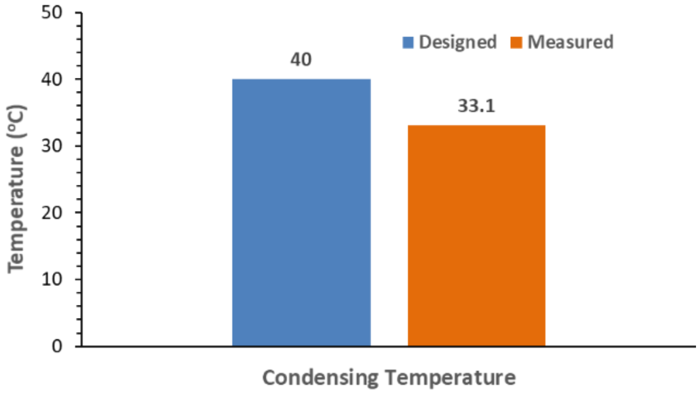

Fig. 11. The condensing temperature: Designed versus Measured results

Replacing R134a with R290 in chest freezers, apart from being able to produce lower cabin temperatures, will also greatly reduce emissions of gases that cause the greenhouse effect from the refrigeration sector. This is because the GWPof R134a is much higher than that of R290, which is 1300 compared to 3, as shown in Table 1 [12], [15].The smaller the GWP value of the refrigerant used in the refrigeration system, the less emission of gases that cause global warming from the global refrigeration sector[21], [23].

4. Conclusions.

Based on the test, it shows that the calculation method for determining the length of the evaporator pipe in the chest freezer using R290 as the working fluid is quite valid. This is indicated by the designed evaporating temperature which is almost the same as the test results, which is -38^{o}C. One of the parameters that can be used as an initial indication for lowering the evaporating temperature which is measured to be the same as that designed is data on superheating. Based on superheating data, it shows that the pipe length used is shorter than the ideal length.For this reason, further research is still needed so that the performance of the refrigeration system can be further improved.

Acknowledgments

We would like to thank Politeknik Negeri Bandung for funding through the Applied Research scheme Number: B/92.48/PL1.R7/PG.00.03/2023. We would also like to extend our gratitude to the Refrigeration & Air Conditioning Engineering Department for allowing us to use the facilities for this research.

References

[1] S. Sawalha, S. Piscopiello, M. Karampour, L. Manickam, and J. Rogstam, “Field measurements of supermarket refrigeration systems. Part II: Analysis of HFC refrigeration systems and comparison to CO2 trans-critical,” *Appl Therm Eng*, vol. 111, pp. 170–182, Jan. 2017, doi: 10.1016/J.APPLTHERMALENG.2016.09.073.

[2] Y. Cheng, J. Yu, G. Yan, and S. Qian, “Improving temperature uniformity in a large frost-free chest freezer,” *International Journal of Refrigeration*, vol. 141, pp. 12–20, Sep. 2022, doi: 10.1016/J.IJREFRIG.2022.05.021.

[3] Á. R. Gardenghi, M. M. Campanini, J. Ferreira Lacerda, C. B. Tibiriçá, and L. Cabezas-Gómez, “A detailed study of the transient behavior of dual-skin chest-freezer with R290,” *International Journal of Refrigeration*, vol. 131, pp. 300–311, Nov. 2021, doi: 10.1016/J.IJREFRIG.2021.07.014.

[4] C. P. Arora, *Refrigeration and Air Conditioning*, 3rd ed. Singapore: McGraw-Hill, 2009.

[5] Y. H. Yau and Y. C. Foo, “Comparative study on evaporator heat transfer characteristics of revolving heat pipes filled with R134a, R22 and R410A,” *International Communications in Heat and Mass Transfer*, vol. 38, no. 2, pp. 202–211, Feb. 2011, doi: 10.1016/J.ICHEATMASSTRANSFER.2010.12.011.

[6] A. G. Devecioılu and V. Oruç, “Characteristics of Some New Generation Refrigerants with Low GWP,” *Energy Procedia*, vol. 75, pp. 1452–1457, Aug. 2015, doi: 10.1016/J.EGYPRO.2015.07.258.

[7] D. Sánchez, R. Cabello, R. Llopis, I. Arauzo, J. Catalán-Gil, and E. Torrella, “Energy performance evaluation of R1234yf, R1234ze(E), R600a, R290 and R152a as low-GWP R134a alternatives,” *International Journal of Refrigeration*, vol. 74, pp. 269–282, Feb. 2017, doi: 10.1016/J.IJREFRIG.2016.09.020.

[8] A. Sethi, E. Vera Becerra, S. F. Yana Motta, and M. W. Spatz, “Low GWP R22 replacement for air conditioning in high ambient conditions,” *International Journal of Refrigeration*, vol. 57, pp. 26–34, Sep. 2015, doi: 10.1016/J.IJREFRIG.2015.05.013.

[9] R. Cabello, D. Sánchez, R. Llopis, I. Arauzo, and E. Torrella, “Experimental comparison between R152a and R134a working in a refrigeration facility equipped with a hermetic compressor,” *International Journal of Refrigeration*, vol. 60, pp. 92–105, Dec. 2015, doi: 10.1016/J.IJREFRIG.2015.06.021.

[10] Y. Heredia-Aricapa, J. M. Belman-Flores, A. Mota-Babiloni, J. Serrano-Arellano, and J. J. García-Pabón, “Overview of low GWP mixtures for the replacement of HFC refrigerants: R134a, R404A and R410A,” *International Journal of Refrigeration*, vol. 111, pp. 113–123, Mar. 2020, doi: 10.1016/J.IJREFRIG.2019.11.012.

[11] L. Zhang, Z. Yang, R. Zhai, Z. Lv, Y. Zhang, and Q. Deng, “Flammable performance and experimental evaluation of a new blend as R404A lower-GWP alternative,” *International Journal of Refrigeration*, vol. 135, pp. 113–120, Mar. 2022, doi: 10.1016/J.IJREFRIG.2021.12.019.

[12] S. Kasera and S. C. Bhaduri, “Performance of R407C as an Alternate to R22: A Review,” *Energy Procedia*, vol. 109, pp. 4–10, Mar. 2017, doi: 10.1016/J.EGYPRO.2017.03.032.

[13] Q. Ning, G. He, M. Fan, J. Xiong, and X. Li, “Flammable refrigerant leakage hazards control for split-type household air conditioners,” *International Journal of Refrigeration*, vol. 144, pp. 188–201, Dec. 2022, doi: 10.1016/J.IJREFRIG.2022.08.009.

[14] W. Tang, G. He, W. Sun, S. Zhou, D. Cai, and Y. Zhu, “Assessment of leakage and risk reduction of R290 in a split type household air conditioner,” *International Journal of Refrigeration*, vol. 89, pp. 70–82, May 2018, doi: 10.1016/J.IJREFRIG.2018.03.012.

[15] B. H. Shon, C. W. Jung, O. J. Kwon, C. K. Choi, and Y. Tae Kang, “Characteristics on condensation heat transfer and pressure drop for a low GWP refrigerant in brazed plate heat exchanger,” *Int J Heat Mass Transf*, vol. 122, pp. 1272–1282, Jul. 2018, doi: 10.1016/J.IJHEATMASSTRANSFER.2018.02.077.

[16] R. Llopis, E. Torrella, R. Cabello, and D. Sánchez, “HCFC-22 replacement with drop-in and retrofit HFC refrigerants in a two-stage refrigeration plant for low temperature,” *International Journal of Refrigeration*, vol. 35, no. 4, pp. 810–816, Jun. 2012, doi: 10.1016/J.IJREFRIG.2012.01.001.

[17] B. Nowak, P. Życzkowski, and R. Łuczak, “Functional Dependence of Thermodynamic and Thermokinetic Parameters of Refrigerants Used in Mine Air Refrigerators. Part 1 – Refrigerant R407C,” *Archives of Mining Sciences*, no. No 1, 2017, doi: 10.1515/amsc-2017-0005.

[18] M. Petersen and S. Kujak, “Review of Lower GWP Refrigerants For Retrofitting R-410A Applications Review of Lower GWP Refrigerants For Retrofitting R-410A Applications,” in *The 19th International Refrigeration and Air Conditioning*, purdue: Purdue University, 2022, pp. 2115,page1-2115,page10.

[19] K. Sumeru, M. Markus, N. F. Pratama, R. Muliawan, and Y. P. Hikmat, “Metode Praktis Perancangan Panjang Pipa Evaporator pada Mini Freezer Bertemperatur Rendah Menggunakan Refrigeran R404A,” *Jurnal Rekayasa Mesin*, vol. 18, no. 1, p. 9, 2023, doi: 10.32497/jrm.v18i1.3878.

[20] J. Sun *et al.*, “COVID 19 vaccine distribution solution to the last mile challenge: Experimental and simulation studies of ultra-low temperature refrigeration system,” *International Journal of Refrigeration*, vol. 133, pp. 313–325, Jan. 2022, doi: 10.1016/J.IJREFRIG.2021.11.005.

[21] A. Coulomb, D., Dupon, J.L., Pichard, “The Role of Refrigeration in the Global Economy,” *38th Note on Refrigeration Technologies.*, no. November, 2019, doi: 10.18462/IIF.NITEC38.06.2019.

[22] R. Cabello, D. Sánchez, R. Llopis, A. Andreu-Nacher, and D. Calleja-Anta, “Energy impact of the Internal Heat Exchanger in a horizontal freezing cabinet. Experimental evaluation with the R404A low-GWP alternatives R454C, R455A, R468A, R290 and R1270,” *International Journal of Refrigeration*, vol. 137, pp. 22–33, May 2022, doi: 10.1016/J.IJREFRIG.2022.02.007.

[23] J. P. Holman, *Heat Transfer*, 9th ed. Boston: McGraw-Hill, 2002.

[24] E. W. Lemmon, M. L. Huber, and M. O. McLinden, “NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.1.” 2013. Accessed: Jul. 16, 2023. [Online]. Available: https://www.nist.gov/publications/nist-standard-reference-database-23-reference-fluid-thermodynamic-and-transport