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Research Article

Thermal design and CFD simulation of an internal heat exchanger with R1234yf as an alternate refrigerant to R134a

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ABSTRACT

The most widely utilized refrigerant in car air conditioners today is still R134a, particularly in developing nations, but its global warming potential is very high (1430). The use of high global warming potential working substances is progressively prohibited and restricted by the F-gas act and the Kigali Amendment in developing and developed nations. R1234yf has been the subject of thermodynamic research as a potential replacement for R134a in vehicle air conditioning systems. Compared to R134a, R1234yf performs somewhat inferior, that can be improved by incorporating an internal heat exchanger into the existing system. Refereeing to the literature available, computational fluid dynamics analysis of the internal heat exchanger for an automobile air conditioning system with refrigerant R1234yf is rarely observed. Hence the novel concept of computational fluid dynamics analysis of thermally designed internal heat exchanger is focused here. This study analyzes the thermal design of an internal heat exchanger and its impact on the coefficient of performance for R1234yf and R134a. And computational fluid dynamics analysis of the thermal designed internal heat exchanger is performed to finalize the dimension of the internal heat exchanger. For similar cooling capacity, the system with refrigerant R134a performance is not much affected by the application of an internal heat exchanger. Its COP increases from 3.636 to 3.676, i.e., only 1.09%. While the performance gap of the system with refrigerant R1234yf as compared to the system with R134a without an internal heat exchanger is 5.17%, while the gap is decreased up to 3.16% with an internal heat exchanger. Identical results are obtained in computational fluid dynamics analysis, with an increment in the internal heat exchanger length, heat transfer increases as well, and the outlet temperature meets the necessary level within a tolerable pressure drop.

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INTRODUCTION

Scientific study now prioritizes understanding how refrigerant affects the environment. Global warming potential (GWP) reports that the environmental emissions of several gases from industries are comparable to carbon dioxide. The amount of heat on Earth is greatly increased by high greenhouse gas concentrations. Third-generation HFC refrigerants, which have a high GWP and zero ODP, are now used in the automotive industry. Finding an R134a substitute in accordance with the Kyoto and Montreal protocols is therefore required to solve the issue of excessive GWP [1]. Environmentally acceptable alternative refrigerants are gradually replacing hazardous refrigerants as part of regulatory efforts to reduce their use. In addition, it preserves the ozone layer and lowers greenhouse gas emissions. Additionally, the substitute refrigerant for the current car air conditioner needs to be affordable and safe. Some nations concentrate on cutting emissions through the use of appropriate refrigerants and good equipment design. The Kigali Amendment [2], which aims to gradually phase out HFCs by reducing their production and usage, was accepted at the 28th conference of parties to the Montreal Protocol. By 2050, the Kigali Amendment calls for an over 85% reduction in the world's HFC usage. According to research, India's population demand prediction for passenger cars could rise by up to 9% between 2017 and 2038 [3]. According to the study, a car's lifespan is estimated to be 15 years, and there should only be 200 passenger automobiles for every 1,000 people. However, taking into account the advancements in cooling system technology, based on different system leak rates, the refrigerant demand scenario is evaluated. The refrigerant consumption by 2038 is estimated by the India Cooling Action Plan to be between 1900 and 24000 MT. A significant contributor to greenhouse gas emissions is the use of mobile air conditioning systems. According to Yang et al. [4], up to 20% of the fuel energy used to run the vehicle can be used by the air conditioning system. Shaikh et al. [5] state that the refrigerant R1234yf is a good substitute for R134a because both refrigerants have similar energy consumption, volumetric cooling capacity, COP and refrigerating effect. Satapathy [6] et al. performed comparative analysis using an ozone friendly mixture of refrigerant R23 and R507A for the auto cascade system and the cascade system to achieve better performance with environmental aspects. Table 1 summarizes the comparison of the thermophysical characteristics of refrigerants R134a and R1234yf [7-9].

Direk et al. [11] investigated the energy and exergy of refrigerant R1234yf in vehicle air conditioner. It was shown that the internal heat exchanger reduced energy destruction per cooling capacity by 13%–16% and raised COP by 4%–6%, respectively. An experimental investigation was conducted. And a two-pipe heat exchanger was chosen. Cooling capacity, COP and overall energy destruction with respect to cooling capacity were calculated and presented as functions of constant temperatures for condensation or evaporation based on experimental data. The results showed that the R1234yf with internal heat exchanger had a mean increase in cooling capacity of 11% to 12%, an improvement in COP value of 4% to 6%, an increase in compressor power of 5% to 8%, an increase in volumetric efficiency value of 4% to 5%, and an increase in isentropic efficiency value of 3% to 4%. The average reduction in total energy destruction per cooling capacity value was between 13% and 16%. And come to the conclusion that the system with R1234yf can benefit from an internal heat exchanger in terms of cooling and COP.

Hmood et al. [12] suggested that R1234yf is a suitable drop-in substitute for R134a in refrigeration systems. With few small adjustments to the AACs system, R1234yf can also be viewed as a viable long-term environmentally beneficial solution because it is non-flammable. Sharif et al. [13] concluded that R1234yf has somewhat lower heat transfer coefficients than R134a, but it makes up for it with smaller pressure drops. Furthermore, it is possible to replace R1234yf directly in current systems with the least amount of adjustment. There is potential for improving the efficiency of vapour compression systems with R1234yf refrigerant by integrating IHX and LSHX. Moreover, it has been discovered that employing IHX reduces energy usage and raises the COP of R1234yf systems.

Strong effects on heat transport were noted by Karademir et al. [14] using R1234yf refrigerant, which has minimal volatility and low GWP. It was proven that as the quality of the vapor increased, so did the heat transfer coefficient. Agrawal et al. [15] concluded that R1234yf's performance parameter values are lower than those of HFC-R134a. It can

Table 1. Comparison of property between R134a and R1234yf [Alkan et al [10], with permission from Yıldız Technical University Press]

Refrigerant	Chemical composition	Boiling point [°C]	Critical Temperature [°C]	Critical pressure [MPa]	ODP	GWP	Safety class (ASHRAE34)	Atmospheric lifetime (year)
R134a	CH_2FCF_3	-26.07	101.1	4.059	0	1430	A_1	14
R1234yf	$CF_3CF= CH_2$	-29.45	95	3.382	0	4	A_2L	<0.05 ys) 11 days

replace R134a quite effectively, though, as the differences are negligible and it has eco-friendly qualities.

Daviran et al. [16] performed simulated analysis on R134a and R1234yf. The lower discharge temperature and pressure ratio for R1234yf were observed. R1234yf experiences a smaller pressure drop during the condensation and evaporation process than R134a. So, a system of R1234yf with fewer thickness pipe, reduces price and improves performance. For R1234yf, COP is 1.3–5% less than that of R134a for similar cooling capacity, while it is 18% higher for constant mass flow rate. Lin et al. [17] recommend R1234yf and a blend of R1234yf and R125 as a very good substitute for R134a because of their comparable cooling capacities and COPs, which are only 5.5% lower than those of R134a. Similarities in the discharged temperature, capacity, and coefficient of performance of refrigerants R134a, R1234yf, and the combination of R134a and R1234yf were noted by Lee et al. [18] on a heat pump bench testing under varying weather conditions. Compared to R134a, the initial charge needed for R1234yf and combination is 11% less. When the percentage of R134a in the mixture reaches 10%, it becomes non-flammable.

Patel et al. [19] estimated the cooling load of an automobile air conditioning system for local climate conditions and performed energy and exergy analysis for R134a and R1234yf, concluded that as the evaporating temperature rises, COP and EDR rise as well. The percentage difference in EDR for the R134a and R1234yf ranges from 6.79% to 2.87 percent when the evaporation temperature is changed from -10 °C to 10 °C, and the difference for COP ranges from 2.44 percent to 4.78 percent. The exergy efficiency falls as the temperature of evaporation rises. At evaporating temperatures of -10 °C, the R1234yf exhibits a 4.78 % lower value of exergy efficiency than the R134a; the discrepancy narrows to 2.43 % once evaporating temperatures have reached 10 °C. At a fixed evaporating temperature of 0 °C, COP drops as the condensation temperature rises from 30 to 60 °C. At 30 °C, the COP of the system with the R1234yf system is 1.96 % less than the R134a system, and it increases to 9.2% at 60 °C.

Wantha [20] investigated the relationship between COP and the coefficient of heat transfer as influenced by annular space, the heat exchanger's length and efficiency, and pressure drop at 6.4 °C and -6.4 °C for evaporation and 46 °C for condensation. At the same operating temperatures of 46 °C for condensation and 0 °C for evaporation, while utilizing an internal heat exchanger, the COP of R134a increases by 2.11% and that of R1234yf increases by 3.78%. The tube in tube internal heat exchanger for R1234yf has a coefficient of heat transfer is 11–17% lower than that of R134a for an identical annular diameter ratio. Kwon et al. [21] made an experimental investigation of the internal heat exchanger's heat transfer properties in a CO_2 heat pump system. The longer the internal heat exchanger, the greater the effectiveness, capacity, and pressure decrease. Approximately twice as much capacity is available in micro-channel IHX as in coaxial IHX. Compared to the hot side, the cold side has a greater pressure reduction.

Alpaslan et al. [22] discovered that R1234yf produced a 9.3–22.3% higher refrigerant mass flow rate and a 1.1–3.5 °C higher temperature of conditioned air. However, in comparison to R134a, it also resulted in a 4.7–16.1 °C lower compressor outlet temperature, a 5.5–11.6% worse COP, and a 0.4–10.9% lower refrigeration capacity. Additionally, the R1234yf system's components often destroyed more energy, and overall, the R1234yf system's exergy destruction rate per unit refrigeration capacity was 4.1 -35.3% higher than that of the R134a system. Alpaslan et al. [23] examined the impact of compressor speed and the temperature range of the air stream at the evaporator and condenser inlet on the exergy and energy evaluation of the car air conditioning system. The condition for outside temperature is produced by 1.78 kW and 5.4 kW heaters in the condenser and evaporator ducts, respectively. Additionally, it is discovered that because R1234yf has a lower liquid density than R134a, its initial charge is 10% lower.

Direk et al. [24] compared R1234yf and R134a's performances by varied air stream temperatures and compressor speeds. According to the findings, the system incorporated with R1234yf had a lower cooling ability and COP of 17.1% and 12.4%, respectively, than the one with R134a. But with the IHX together with R1234yf system, the system's cooling capacity value and COP increased dramatically by 7.9% and 4.1%, respectively. The cooling capacity and coefficient of performance of the system using R1234yf were ontaied to be 13.9% to 20.4% and 7.5% to 16.5% lower, respectively, than those of the system using R134a. The experimental techniques were replicated using a double pipe heat exchanger. According to the results, at 27 °C and 35 °C, respectively, the cooling capacity of R1234yf was enhanced by 6.3% to 8.6% at 27 °C and 6.4% to 9.9% at 35 °C. Additionally, for air stream temperatures of 27 °C and 35 °C, respectively, the system's COP was raised by 2.8% to 7.4% and 2.4% to 4.8%. Direk et al. [25] examined the performance of R134a and R1234yf based on the internal heat exchanger efficiency is between -20 °C and 0 °C for evaporation and 40 °C and 50 °C for condensation. and discovered that when efficacy increases, so does the compressor discharge temperature. Ultimately, the critical effectiveness value to achieve the same COP with R1234yf was discovered to be 50%.

According to an experimental investigation conducted by Cho et al. [26], refrigerant R1234yf has a low COP and cooling capacity of 4.5% and 7%, respectively. When an IHX is used, performance decreases by 2.9% to 1.8%. It is possible to achieve satisfactory cooling capacity by operating under varied operating conditions and increasing the COP to 4.6%. Babiloni et al. [27] calculated that in the absence of IHX, R1234yf's volumetric efficiency is 4% lower, its cooling capacity is 9% lower, and its coefficient of performance is roughly 7% worse. They recommend that the R1234yf system's COP disparities be decreased by using an internal heat exchanger. In their numerical analysis of three fluid heat exchangers, in both parallel flow and counter flow systems, Mohapatra et al. [28] found that the total heat transfer coefficient and the efficiency of heat transfer via the helical tube side fluid to the outer side annulus fluid increase in tandem with the helix tube side fluid input temperature. It is further distinguished by the fact that TFHE is significantly more effective in the counter flow arrangement than in the parallel flow design with respect to fluctuation in the fluid inlet temperature on the helical tube side.

Navarro et al.[29] performed numerical analysis on automobile air conditioning systems with open piston compressors under different conditions, comparing R290, R134a, and HFO1234yf. Compared to R134a, the experimental investigation found that the volumetric efficiency of the R290 compressor improved significantly, while the efficiency of the R1234yf compressor improved as the pressure ratio increased. The system performance is even better with IHX: however, R1234yf's volumetric efficiency is 5% lower and its cooling capacity is 9% lower than R134a's. Additionally, the value deteriorates at high condenser temperatures. Kang et al. concluded that R1234yf is a recently synthesized artificial working medium: therefore, the production process is still in its infancy and its capacity is constrained. When choosing an automotive air conditioning refrigerant, one must take into account its high cost and patent limits.

This study aims to replace high GWP refrigerant with a less greenhouse gas-emitting alternative. An internal heat exchanger can be added to the current system to increase the performance of R1234yf, which is slightly lower than R134a. According to the literature, internal heat exchanger CFD analysis for car air conditioning systems using R1234yf refrigerant is not frequently seen. This work is therefore unique in that regard. This study's objective is to analyze the internal heat exchanger's (IHX) thermal design for R1234yf and how it affects COP. The flow parameters and temperature of the internal heat exchanger have been investigated numerically using CFD.

MATERIALS AND METHODS

Thermal Design of Internal Heat Exchanger

Internal heat exchangers, such as double-tube heat exchangers or tube-in-tube, can be used in air conditioning systems. A pair of concentric tubes with distinct inner diameters served as the models for this experiment Figure 1.

The tube-in-tube, counterflow internal heat exchanger was modelled using the energy balance and log means temperature difference (LMTD) methodology. For the thermal design of tube-in-tube type heat exchangers, the subsequent assumptions were made:

1. Kinetic energy and potential energy changes are neglected
2. No heat exchange with surroundings
3. Isenthalpic process considered for the expansion device
4. Steady-state operating conditions

The thermal design of the internal heat exchanger is studied for a 3.5 kW cooling capacity automobile air conditioning system that operates at 44 °C condensation temperature and 0 °C evaporator temperature. The thermodynamic cycle representation of an air conditioning system with R134a, R1234yf and R1234yf incorporated with an internal heat exchanger is represented in Figure 2-4 respectively. The property plot and overlay are generated in EES [30].

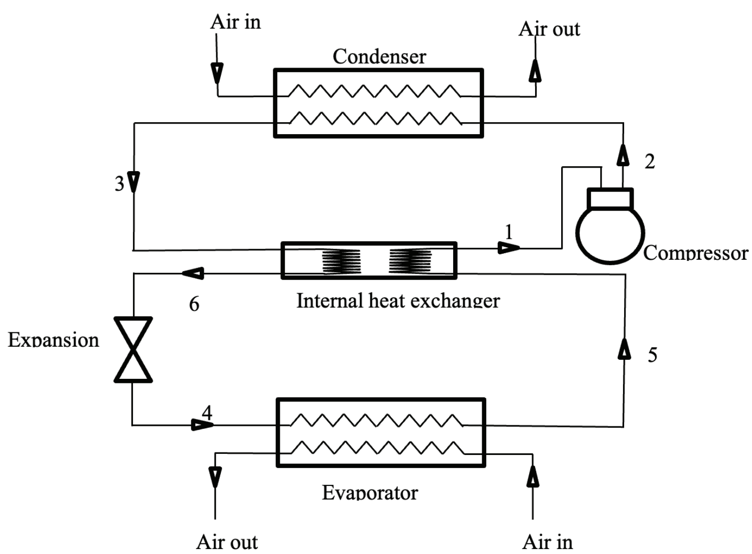

Figure 1. Internal heat exchanger (Tube in tube type).

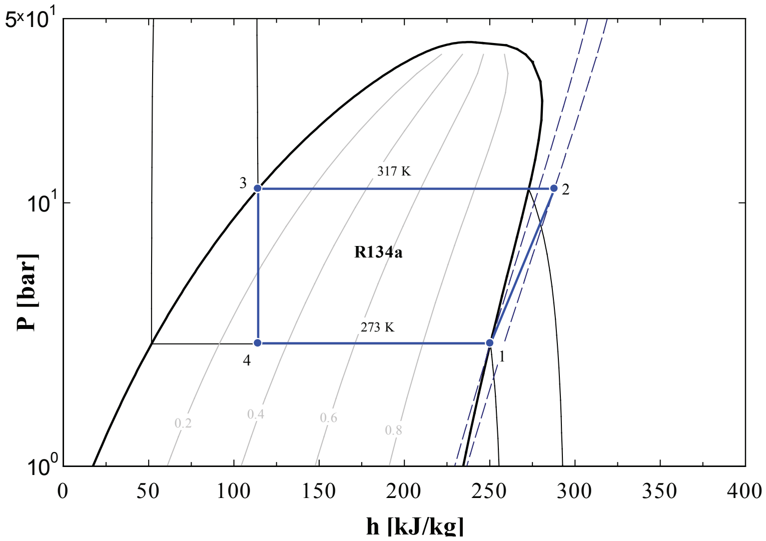

Figure 2. P-H diagram of R134a cycle.

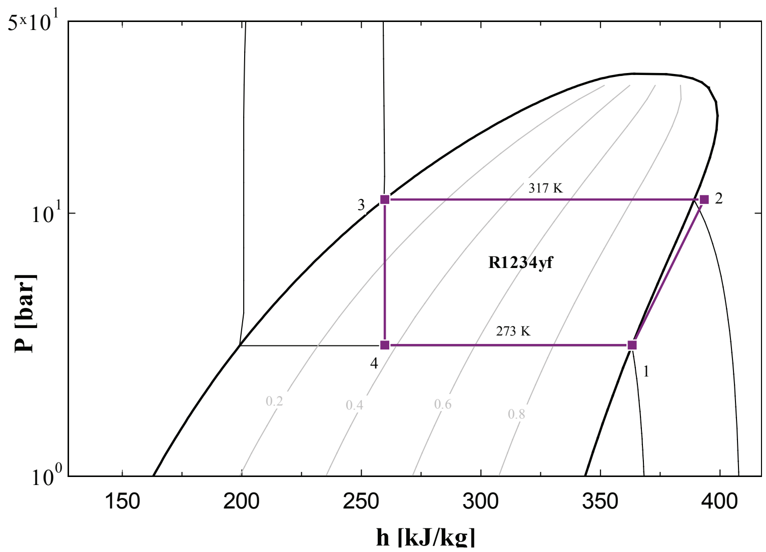

Figure 3. P-H diagram of R1234yf cycle.

The energy balance of the counterflow the tube in tube type heat exchanger can be established using the following equation from the fundamentals of the heat transfer.

The maximum amount of heat transfer rate that can occur is represented as,

$$Q_{\max} = c_{pc} \times (T_{hi} - T_{ci}) \tag{1}$$

The average specific heat of saturated vapor at evaporation temperature is c_{pc}, while the average specific heat of liquid at condenser pressure and condensation temperature is c_{ph}. The cold side of the internal heat exchanger's output temperature is 10 K, which is the degree of superheat taken into account in the original design.

$$Q_{CIHX} = c_{pc} \times (T_{co} - T_{ci}) \tag{2}$$

The effectiveness of the internal heat exchanger, ϵ, can be given by below mentioned equations. And the temperature outlets are calculated using the heat exchanger's effectiveness relation:

$$\in = \frac{Q_{CIHX}}{Q_{\max}} \tag{3}$$

$$Q_{hIHX} = \in \times Q_{\max} \tag{4}$$

where the coefficients of heat transmission within and outside the tube are hi and ho, respectively. In this study,

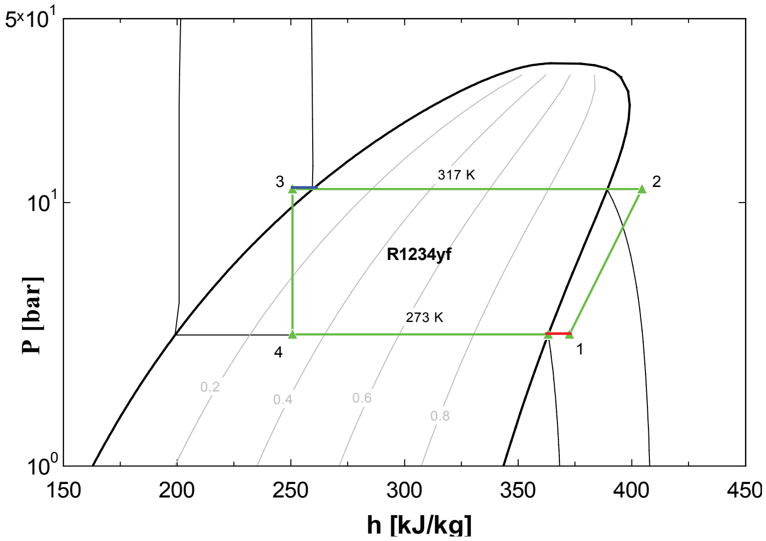

Figure 4. P-H diagram of R1234yf with IHX cycle**.**

the refrigerant vapour passes within the space between the heat exchanger's inner and outer tubes, while the refrigerant liquid passes via the inner tube. Consequently, for single-phase heat transfer the Dittus-Boelter correlation may be applied to calculate the heat transfer coefficient of the inner tube in turbulent fluid flows, which is given as [16].

$$N_u = 0.0238 \times \mathrm{Re}^{0.8} \times \mathrm{P_r}^{n} \tag{5}$$

When the vapor refrigerant is superheated, the Prandtl number exponent n is 0.4. The hydraulic diameter for the inner tube is Dh=Di, annular space hydraulic diameter is determined by the formula = Do-Di.: K and Re stand for the refrigerant's thermal conductivity and Reynolds number, respectively. If the Reynolds number in an annular space is less than 4000, the liquid flow is classified as laminar flow. For the fully developed flow, the Nusselt number expression is mentioned below.

$$h_h = 0.0238 \times \left(\frac{K_6}{D_h}\right) \times \mathrm{Re}^{0.8} \times \mathrm{P_{r\mathit{h}}}^{0.4} \tag{6}$$

$$hc = 0.0238 \times \left(\frac{K_5}{D_h}\right) \times \mathrm{Re}^{0.8} \times \mathrm{P_{r\mathit{h}}}^{0.4} \tag{7}$$

For a counterflow type tube-in-tube type heat exchanger, Tm, Rth and L are the LMTD, heat resistance and heat exchanger length respectively, calculated by the following equations [31].

$$Tm = \frac{(Thi - Tco) - (Tho - Tci)}{\ln \dfrac{(Thi - Tco)}{(Tho - Tci)}} \tag{8}$$

$$R_{th} = \frac{T_m}{Q_{\max}} \tag{9}$$

The heat transfer area, A is computed as follows, using the given relation:

$$A = \frac{\left(\dfrac{1}{h_h}\right) + \left(\dfrac{t}{K}\right) + \left(\dfrac{1}{h_c}\right)}{R_{th}} \tag{10}$$

$$A = L \times D_h \times \Pi \tag{11}$$

The pressure drop occurs in the internal heat exchanger on hot and cold sides can be calculated by considering the fluid properties at that state in the below equations.

$$Del\_\Pr = \frac{2 \times f \times L \times \mu^2 \times \mathrm{Re}^2}{D_h^{\,3} \times \rho} \tag{12}$$

Where,

$$f = 0.0079 \times \mathrm{Re}^{-0.22} \tag{13}$$

Without the internal heat exchanger cycle, the cycle's COP may be computed using

$$COP = \frac{RE}{Wc} \tag{14}$$

The COP of the air conditioning cycle with application of the internal heat exchanger can be calculated from

$$COP' = \frac{RE'}{Wc'} \tag{15}$$

An internal heat exchanger's efficiency and total heat transfer coefficient have a significant impact on how well refrigeration systems operate. The design of heat exchangers is determined after the heat transfer area of the heat exchanger. The software was utilized to extract the thermodynamic parameters of the refrigerant and write the mathematical model in Engineering Equation Solver (EES).

CFD Analysis of Internal Heat Exchanger

The current study looks into the flow physics inside the IHX that uses refrigerant R1234yf and is integrated into car air conditioning. A model of tube-in-tube heat exchanger with the internal diameter of 8.12 mm and thickness of 7mm, whereas the outer tube has a diameter 17.45 mm and thickness of 8 mm is studied. The material of the heat exchanger was selected as copper due to its high thermal conductivity. The cold liquid refrigerant temperature at the inlet is 273.2 K, the hot gaseous refrigerant temperature at the inlet is 317.2 K, and other boundary conditions are obtained from thermal design. The internal heat exchanger's temperature distribution and flow mechanics are predicted using the standard k-ε turbulence model. The geometry development Figure 5 and mesh generation Figure 6 are in Ansys 2021. For smoother transaction with better result, I had used program controlled linear mesh with element size of 0.5 mm. Here the best result outcome from three different length of internal heat exchanger is compared with theoretical thermal analysis results. The simulation study performed using the CFX solver in Ansys 2021 to explore the flow characteristics and flow parameters such as pressure, temperature

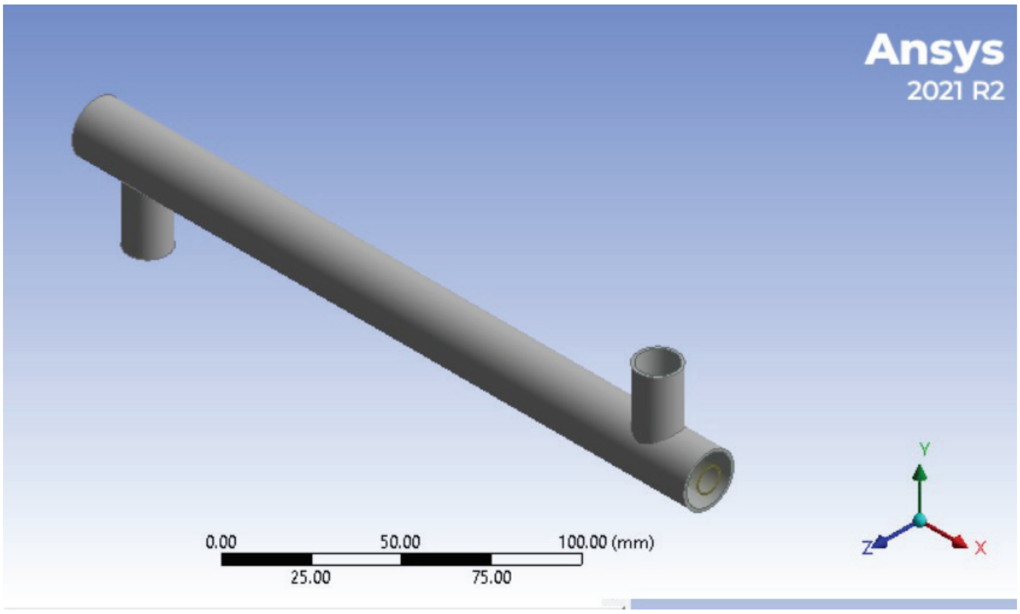

Figure 5. 3D Geometry of internal heat exchanger.

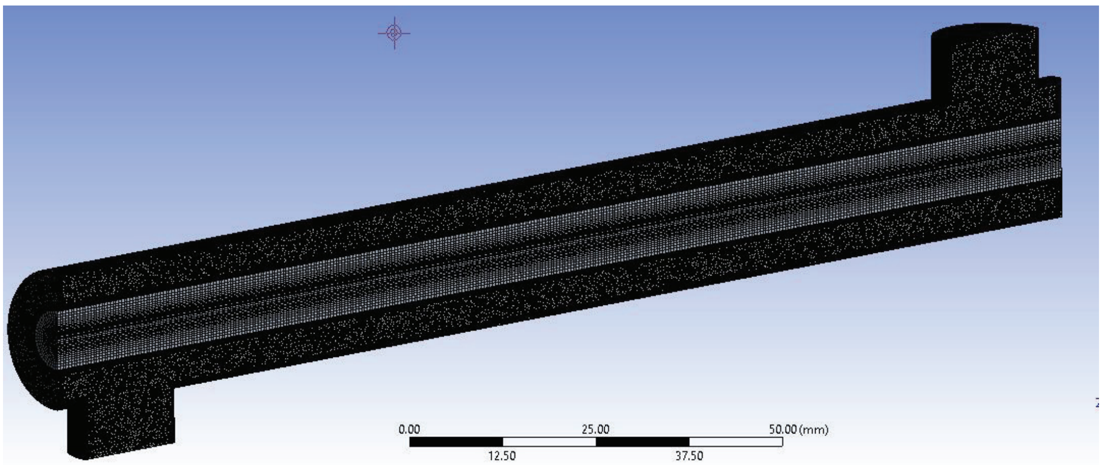

Figure 6. Meshing view of the internal heat exchanger.

and velocity is evaluated inside the tubes to obtain the data on the flow pattern.

To increase the accuracy and consistency of the numerical findings, the grid has been gradually improved by changing the element size and refinement level. Throughout the test, the number of elements was progressively raised from 827359 to 5934371. The temperature increase from intake to exit ($\Delta Tn = Tn2 - Tn1$) of the cold fluid on the outer annulus side, or R1234yf, is the parameter taken into account for the grid independence test. This specific parameter was selected over others since the outer annulus side was most affected by the manual increase in cell size and refinement levels (because of the bigger volume). Therefore, comparing the grid independence to a metric that is most impacted by the intervention becomes relevant. The mesh elements of 4646235 and 5934371 exhibit a 0.16 K shift in ΔTn as the grid is refined from 827359 elements to 5934371 elements with varying mesh sizes. Thus, for additional analysis and parametric examination, a grid of 4646235 components has been selected. Figure 7 illustrates the results of the grid independence test.

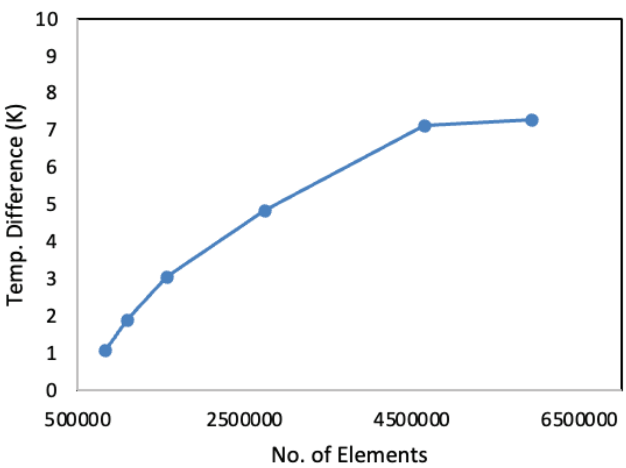

Figure 7. Grid independence test.

The finite volume approach has been used to iteratively solve a three-dimensional computational domain of the IHX using the CFX (ANSYS 21 R2). Before the governing equations are solved, suitable boundary conditions have been established. No-slip boundary conditions have been established at the inner and outer tube walls. The velocity inlet boundary condition has been applied to both hot and cold fluids at the intake. As a thermal boundary condition indicated in table 2, the temperature at the entrance for both hot and cold fluids have been provided. The mass flow outlet boundary condition has been established at the exit of both fluids. The outer tube's outside walls have been made adiabatic to lessen the shell's thermal contact with the environment; this is comparable to the insulation on the heat exchanger's outer tube's outer side. Better thermal interaction between the hot and cold fluids is achieved by specifying an interface boundary condition at the proper walls.

RESULTS AND DISCUSSION

The table No. 3 shows the heat exchanger dimensions and performance parameters. And table No. 4 compares the performance parameters of the air conditioning system with R134a with the system with R1234yf and R12324yf+IHX. For the below mentioned initial data, copper (ASTM B68 C 12200) is chosen for the internal heat exchanger material and the design methodology performed.

For similar cooling capacity, the performance of system with refrigerant R134a is not much affected by the application of an internal heat exchanger. Its COP increases from 3.636 to 3.676, only 1.09%, while the performance gap of the system with refrigerant R1234yf as compared to the system with R134a without an internal heat exchanger is 5.17%, while the gap is minimized by 3.16% with an internal heat exchanger. Figure 8 represents the percentage increase of refrigerant effect and compressor work with an increase

Table 2. Results of thermal design of internal heat exchanger

Cold Fluid (R1234yf)		
Boundary	**Parameter**	**Value**
Cold Fluid Inlet	Total Pressure	3.158 Bar
	Total Temperature	273.2 K
Cold Fluid Outlet	Mass Flow Rate	0.02873 kg/s
Hot Fluid (R1234yf)		
Boundary	**Parameter**	**Value**
Hot Fluid Inlet	Total Pressure	11.26 Bar
	Total Temperature	317.2 K
Hot Fluid Outlet	Mass Flow Rate	0.02873 kg/s

Table 3. Results of thermal design of internal heat exchanger

Inner tube diameter	8.12	mm
Inner tube thickness	0.7	mm
Outer tube diameter	17.45	mm
Outer tube thickness	0.8	mm
Length of internal heat exchanger	191.45	m
Heat transfer area	0.00561	m^2
Heat transfer coefficient on cold side	75.88	W/m^2K
Heat transfer coefficient on hot side	273.7	W/m^2K
Overall heat transfer coefficient	41.31	W/m^2K

in length of the internal heat exchanger. Whereas Figure 9 shows an increase in COP with variation in length of the internal heat exchanger. Figure 10 represents the change in degree of superheating and degree of subcooling with increase in length of the internal heat exchanger. Whereas Figure 11 shows pressure drops in hot side and cold side of the refrigerant in the internal heat exchanger with variation in length.

The flow pattern inside an internal heat exchanger expressed via velocity contours in Figures 12 and 13. The flow pattern inside an internal heat exchanger is expressed via temperature contours in Figures 14 and 15. The Figure 16 and Figure 17 represent the refrigerant temperature at the cold outlet and hot outlet junctions, respectively. Figure 18 and Figure 19 represent the cold gaseous refrigerant pressure contours and hot liquid pressure contours in tube in the tube internal heat exchanger, respectively.

The Table 5 represents the results of EES simulation and CFD analysis for different lengths of the internal heat exchanger as 191.4 mm, 220 mm and 240 mm and their comparison. Both the liquid and the vapor's temperature as they enter the internal heat exchanger determine how much heat is transferred there. The Figure 20 express the outlet temperature through the tube-in -tube internal heat exchanger. Additionally, it was observed that the heat exchange between the liquid and vapor in the heat exchanger caused a change in the temperature at which the liquid and vapor exit the tube-in-tube internal heat exchanger. The cold suction vapor refrigerant passing through the internal heat exchanger receives heat from the warm liquid refrigerant, which causes the vapor to become superheated and increase in temperature while the liquid subcools and decreases in temperature. Figure 21 shows the degree of superheating and degree of subcooling through the tube-in-tube internal heat exchanger. Figure 22 presents the pressure drop occurs through the tube-in-tube internal

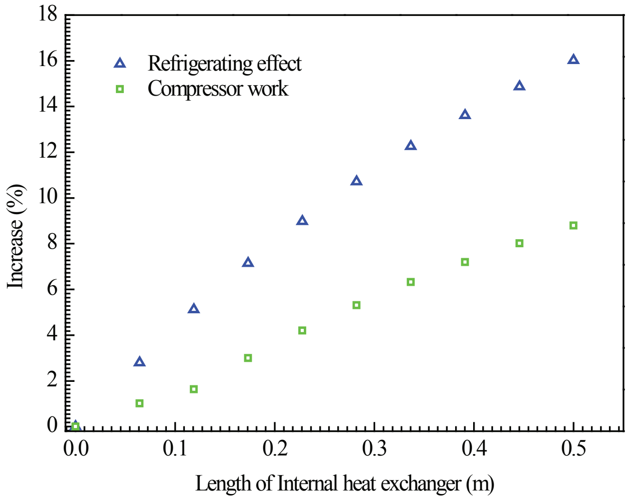

Figure 8. Effect of internal heat exchanger length on compressor work and refrigerating effect.

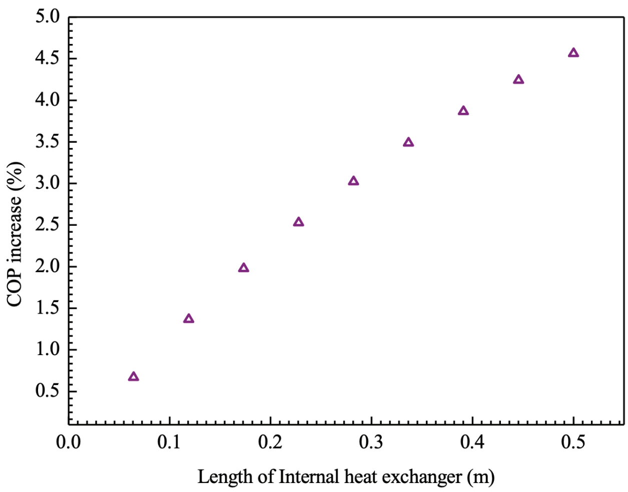

Figure 9. Effect of internal heat exchanger length on % COP increase.

Table 4. Result of thermal analysis

Parameter	R134a	R1234yf	R1234yf with IHX
m (kg/s)	0.0257	0.0338	0.03134
RE (kJ/kg)	136.2	103.6	111.7
Wc (kJ/kg)	39.89	30.07	31.75
COP	3.636	3.443	3.518

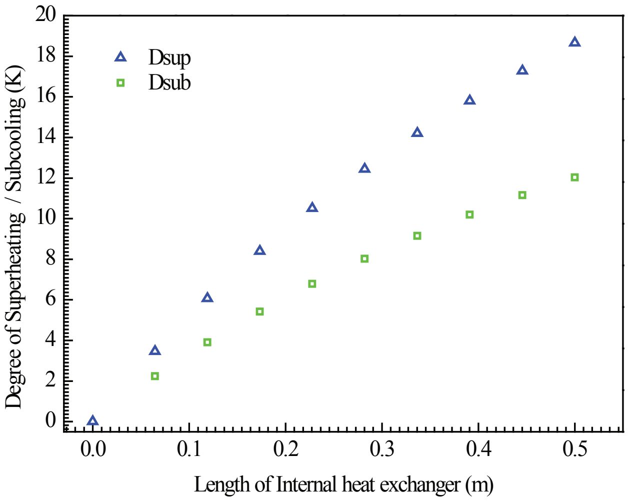

Figure 10. Effect of internal heat exchanger length on Degree of Superheating/Subcooling.

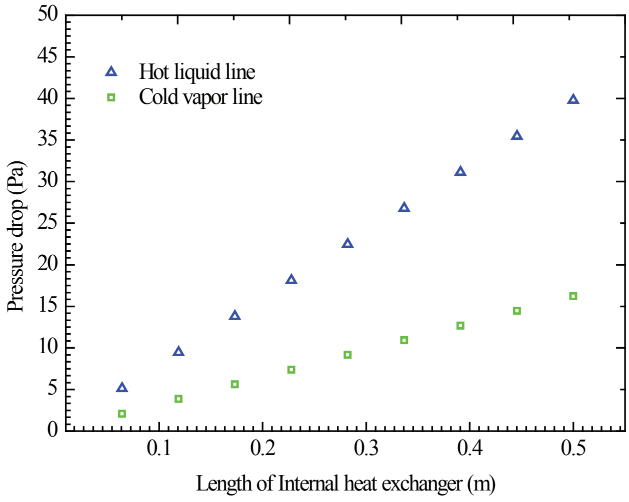

Figure 11. Effect of internal heat exchanger length on pressure drop in internal heat exchanger.

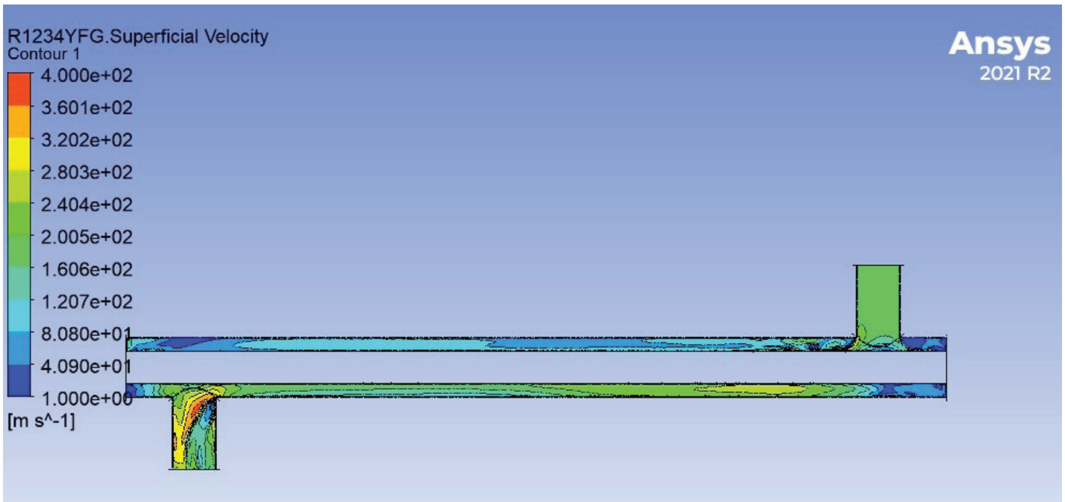

Figure 12. Velocity profile of gaseous R1234yf in outer tube of internal heat exchanger.

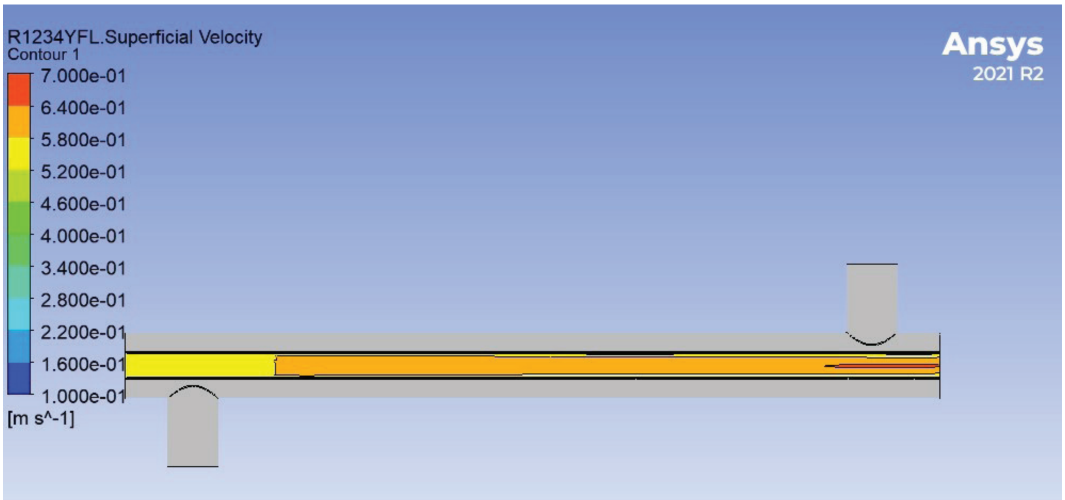

Figure 13. Velocity profile of liquid R1234yf in inner tube of internal heat exchanger.

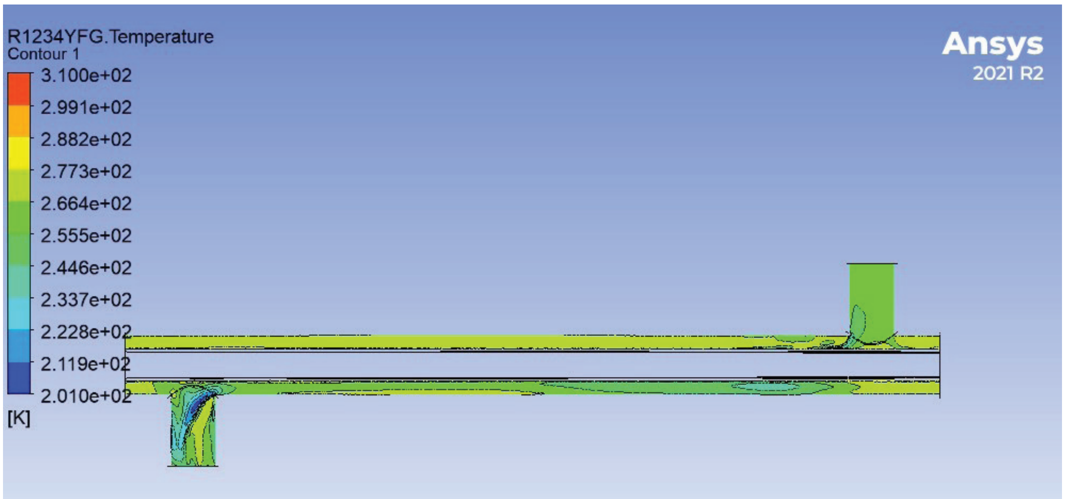

Figure 14. Temperature distribution vapour R1234yf in internal heat exchanger.

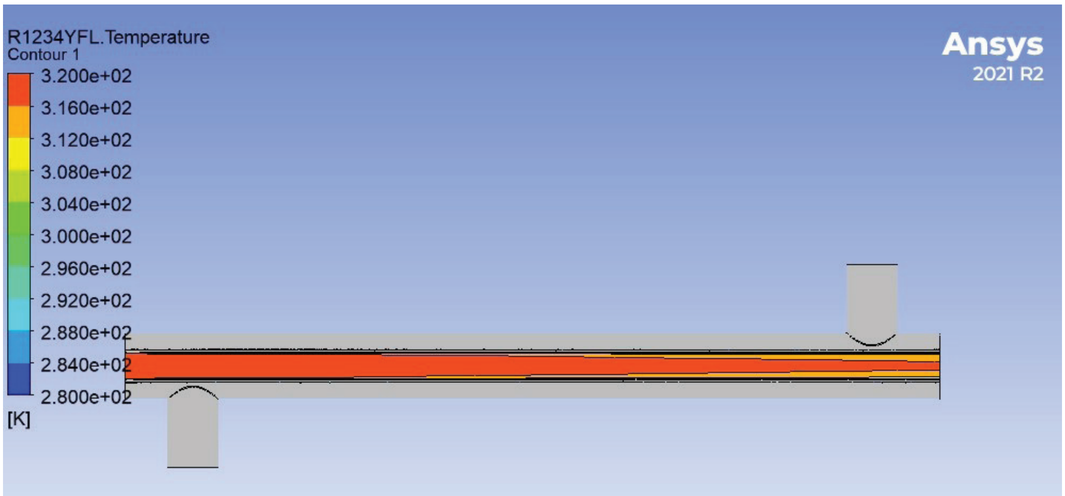

Figure 15. Temperature distribution liquid R1234yf in internal heat exchanger.

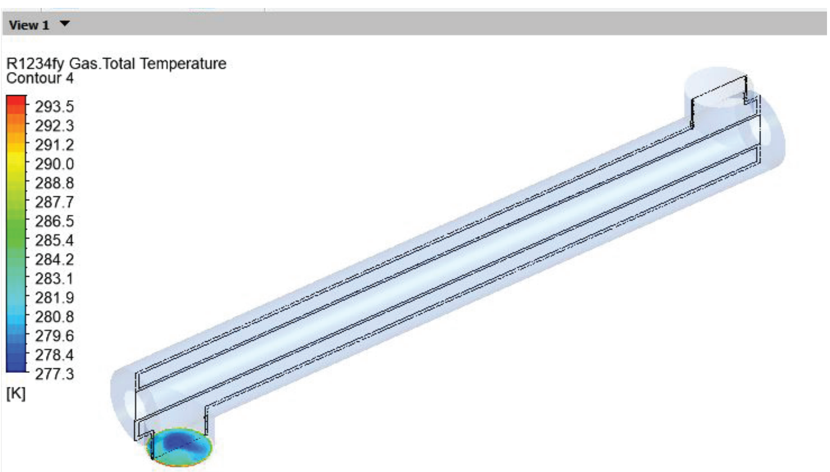

Figure 16. Cold outlet temperature of R1234yf.

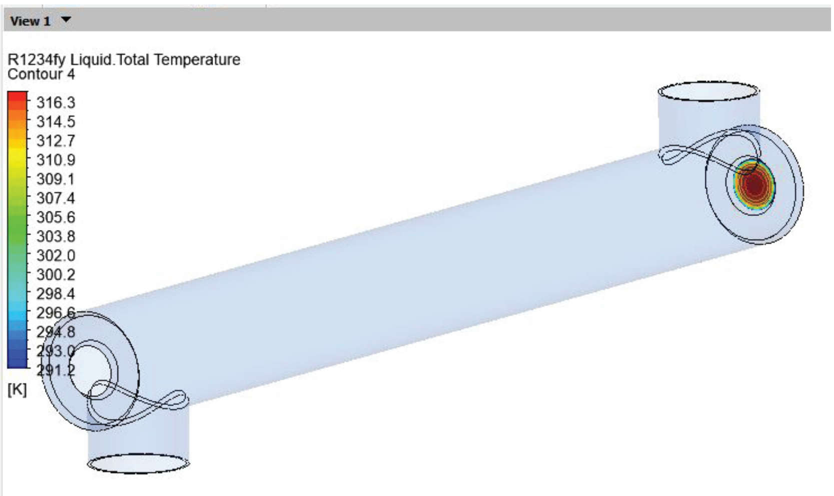

Figure 17. Hot outlet temperature of R1234yf.

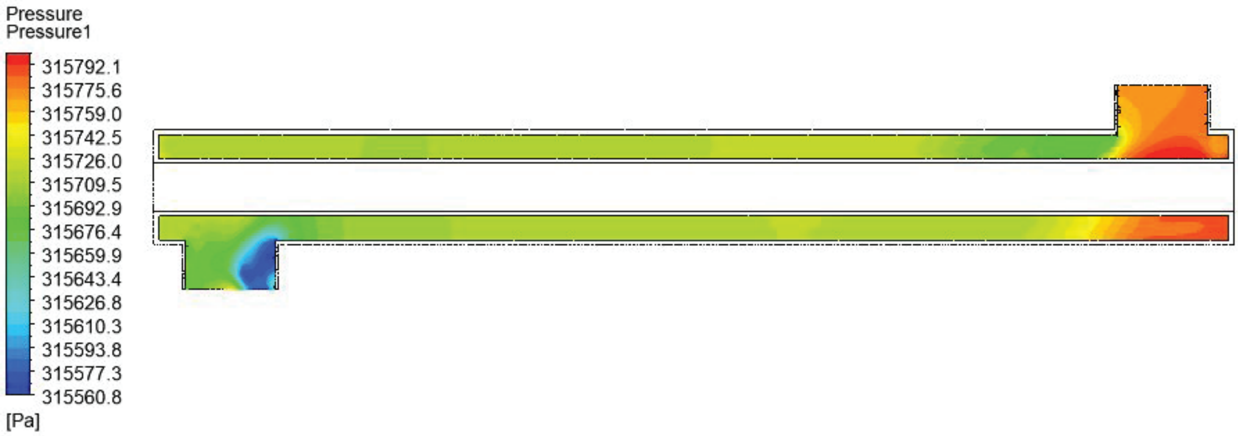

Figure 18. Pressure profile of gaseous R1234yf in outer tube of internal heat exchanger.

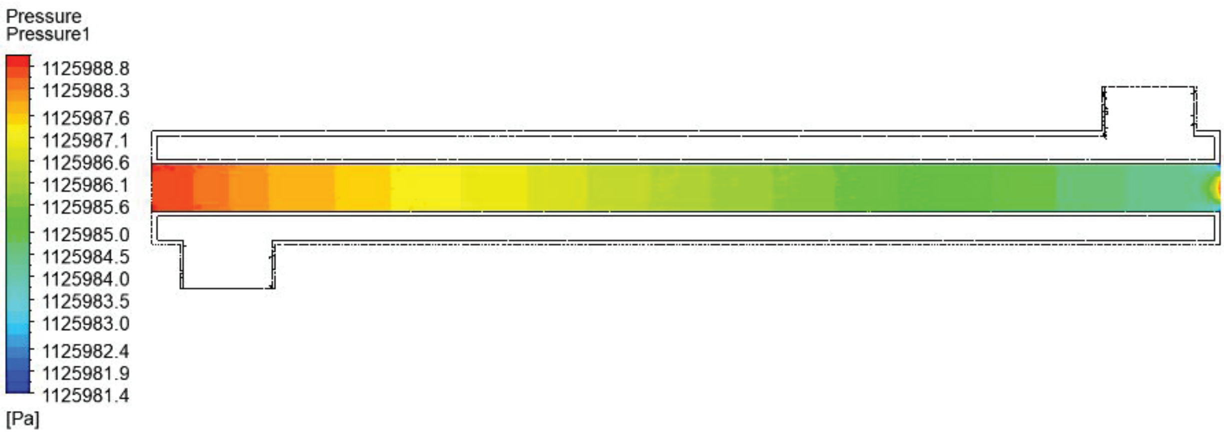

Figure 19. Pressure profile of liquid R1234yf in inner tube of internal heat exchanger.

Table 5. Result analysis table

Parameters	Theoretical	CFD Analysis		
Length of heat exchanger (mm)	**191.45**	**191.45**	**220**	**240**
Cold outlet temperature (K)	283.2	277.99	279.74	280.12
hot outlet temperature (K)	310.7	314.951	313.414	312.409
Degree of superheat (K)	10	4.99	6.74	7.12
Degree of subcooling (K)	6.5	2.249	3.786	4.791
Pressure drop hot side (Pa)	15.23	40	50	50
Pressure drop cold side (Pa)	6.207	14.86	24.43	25

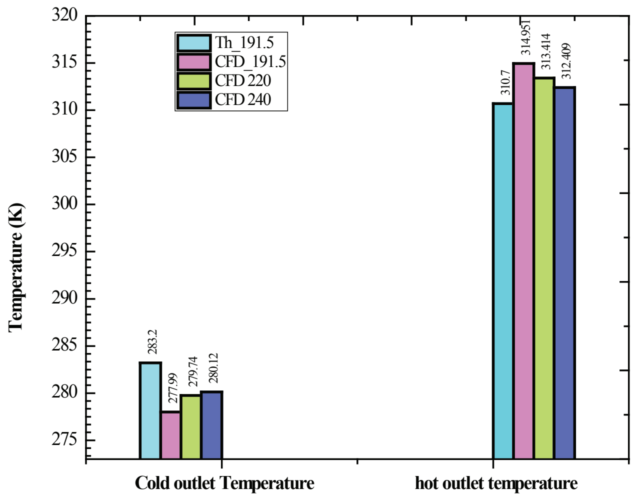

Figure 20. Internal heat exchanger outlet temperature.

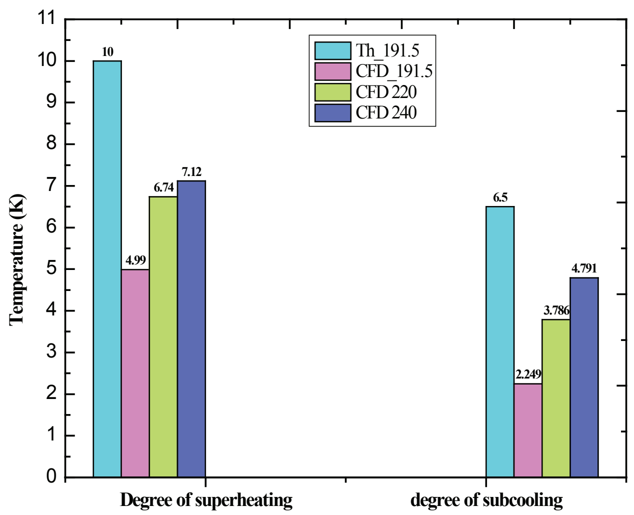

Figure 21. Degree of superheating and subcooling in internal heat exchanger.

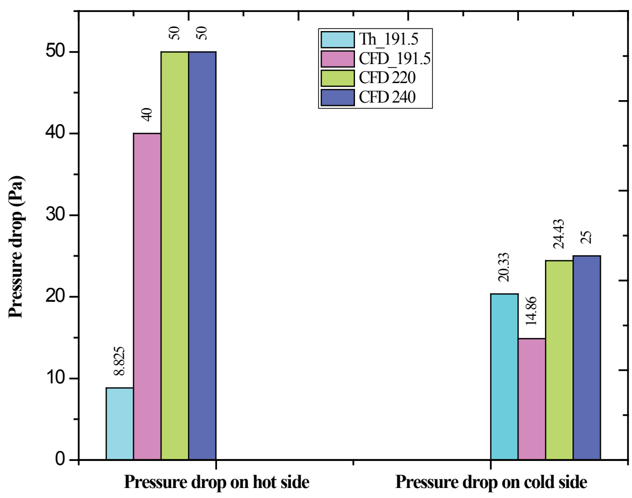

Figure 22. Pressure drop through in internal heat exchanger.

heat exchanger on hot side and cold side of refrigerant. Two variables affect how much the liquid and vapor temperatures vary when they exit the internal heat exchanger. The first is heat exchanger effectiveness, whereby suction vapor is progressively superheated as heat exchanger efficiency rises. The second is the fluid's specific heat, which is influenced by the working environment. However, the system's performance is hampered by a large amount of superheated vapor near the compressor entry. The suction vapor temperature in the R1234yf case increased to the superheated temperature through the internal heat exchanger, surpassing the superheated temperature of the cycle with R134a. Subcooling is less than superheating as a result of heat exchange between warm liquids and cold vapors in the internal heat exchanger. This is caused by the fluid's specific heat as well as the internal heat exchanger's heat transfer coefficient.

CONCLUSION

In the present study, the thermal design of an internal heat exchanger made for R1234yf and its effect on the coefficient of performance are analyzed with respect to R134a. Refereeing the literature available, CFD analysis of internal heat exchanger for automobile air conditioning system with refrigerant R1234yf is rarely observed. Hence the novel study of the numerical investigation has been conducted utilizing computational fluid dynamics analysis to explore the flow characteristics and temperature in the internal heat exchanger. Through the addition of the internal heat exchanger to the refrigeration cycle, the performance parameters of R1234yf were assessed for a range of evaporation and condensation temperatures in this theoretical investigation. The outcomes were then compared with the R134a basic cycle under varied internal heat exchanger effectiveness. The greatest impact of subcooling and superheating effects on the cycle efficiency of the R1234yf system was observed. And recommended 8.5 K degree of superheating for refrigerant R1234yf [32]. In this study, performed to achieve 10 K degree of superheating. The similar trend of results, i.e. COP, heat transfer coefficient, effectiveness and pressure drop, are observed in research by Wantha [20].

- The numerical analysis shows that the automobile air conditioning system with R134a has a higher coefficient of performance than the system with R1234yf, but the application of an internal heat exchanger with R1234yf improved the system coefficient of performance higher than R134a.
- As an effect of an internal heat exchanger, the suction gas is heated, thus decreasing the suction density: hence, the compressor discharge temperature rises. The system capacity and efficiency can be increased with an internal heat exchanger, but the compressor discharge temperature may rise to an intolerable level.
- The increasing suction-side pressure drop results in an increase in compressor work at some level. The pressure drop is in tolerable limit that justifies the application of an internal heat exchanger in an air conditioning system with R1234yf.
- According to the numerical model, the internal heat exchanger increase in heat transfer will always lead to improved performance of the system with R1234yf. An increase in enthalpy changes across the evaporator was observed along with degree of subcooling. Furthermore, the rates of increase for degree of superheat are higher than those for degree of subcooling, and these values rise with increasing efficiency independent of the operating temperatures.
- The computational fluid dynamics analysis shows that increasing length of the internal heat exchanger increases the heat transfer. Increasing the suction-side pressure drop results in increase in compressor work at some level. The pressure drop is within a tolerable limit that suggests the application of an internal heat exchanger with an air conditioning system with R1234yf.
- For the system with refrigerant R134a, performance is not much affected by the application of an internal heat exchanger. Its coefficient of performance increases from 3.636 to 3.676, only 1.09%, while the performance gap of the system with refrigerant R1234yf as compared to the system with R134a without an internal heat exchanger is 5.17%, while the gas is decreased up to 3.16% with an internal heat exchanger.
- The future research direction of the study is to conduct experimental analysis of automobile air conditioning system of R1234yf with IHX. And the ongoing attempt of manufacturer to increase the production of R1234yf, and decrease its cost that's the challenge.

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ABBRIVATION

GWP	Global Warming Potential
ODP	Ozone depleting potential
HFO	HFO Hydrofluoroolefin
MAC	MAC Mobile Air Conditioning
QL	Cooling capacity
IHX	Internal heat exchanger

NOMENCLATURE

m_r	Mass flow rate (kg/sec)
Q_{cIHX}	Heat transfer in cold line of IHX
Q_{hIHX}	Heat transfer in hot line of IHX
Q_{max}	Maximum heat transfer through IHX
T_m	Logarithmic mean temperature difference
R_{th}	Thermal resistance
$\in$	Effectiveness of internal heat exchanger
Nu	Nusselt number
Re	Reynold's number
Pr	Prandtl number
h	Heat transfer coefficient
A	Heat transfer area
K	Thermal conductivity
RE	Refrigerating effect

Subscript

a	air
c	condenser
Comp	compressor
e	evaporator
ex	Expansion valve
i	inlet
r	refrigerant
sub	subcooling

sup superheating
h hot side refrigerant
c cot side refrigerant

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

STATEMENT ON THE USE OF ARTIFICIAL INTELLIGENCE

Artificial intelligence was not used in the preparation of the article.

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