ISSN: 2584-0231(Online)
International Journal of Multidisciplinary Research in Arts, Science and Technology
© IJMRAST | Vol. 2 | Issue 9 | September 2024
Available online at: https://ijmrast.com
DOI:https://doi.org/10.61778/ijmrast.v2i9.79

DESIGN OF MINIATURE VAPOUR COMPRESSION REFRIGERATION SYSTEM FOR ELECTRONICS COOLING

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ABSTRACT:

There is an increase in power consumption and thermal dissipation as the number of transistors in a semiconductor device increase. As chip power increases, existing passive heat dissipation methods become outmoded, necessitating the use of novel active cooling approaches. So, the removal of heat radiated from computer chips has become a critical difficulty in the design of portable and other space-limited electronic devices. The cooling capacity of widely used passive cooling methods (heat sink, heat pipe, and vapour chamber) is limited. Therefore, new active cooling methods are needed. Among numerous novel active chilling techniques, the Vapour Compression Refrigeration (VCR) system is the foremost technology. With the development of micro technologies, the refrigeration systems can be miniaturized and integrated to yield compactness, lightness, high performance and cost-effectiveness. This report presents a design and model development of miniature VCR system for electronic cooling. The miniaturized VCR system model was developed using CATIA V5 software. The model included a reciprocating compressor, condenser, capillary tube, accumulator, cold plate, and a hot plate (heat sink). The model of the small VCR system was simulated using DWSIM software. The dimensions and the cooling capacity of the miniaturized VCR system were found as 625×240×475 mm^3 and 1kW respectively. The VCR system showed coefficient of the performance (COP) as 2.2, the mass flow rate of refrigerant R134a as 0.007 kg/s, cold plate temperature (inlet temperature of the evaporator) as -26.25℃ and adiabatic efficiency of the compressor (AEC) equal to 75.09%. The calculated values of all the parameters were comparable to the simulated values. The present design of VCR is as ideal miniature cooling system is considered as cost effective and for its ability to transport heat away from its source.

Keywords: *Refrigeration, electronics, miniature, design, vapour and compression*

INTRODUCTION

The American Society of Heating, Air Conditioning and Refrigerating Engineers (ASHRAE) defines refrigeration as the science of achieving and sustaining temperatures lower than those of the immediate environment. It is, thus, the process of taking heat out of a medium. Stated differently, the phrase "refrigeration" refers to the preservation of an organism or system at a temperature below that of its environment. Applications for refrigeration are very varied and wide-ranging [1], including food preservation, the home and dairy sectors, ice making, ice cream production, the textile industry, etc. For high heat dissipation applications, it is one of the most promising cooling methods since it can maintain the junction temperature below the maximum operating temperature.

A refrigeration system is made up of several parts and pieces that are linked in a certain way to provide the effect of refrigeration. Refrigeration systems [2] operate on a set of sequential thermodynamic processes that constitute a cycle that restores the working material to the same condition. The main working fluid in a refrigeration system that absorbs and transfers heat is called a refrigerant. The Vapour Compression Cycle is commonly employed in the most prevalent refrigeration system in use today, which involves the input of work from a compressor [3]. To put it simply, mechanical VCR (one that uses a compressor and a volatile working fluid in a closed loop) has the highest COPs of all the refrigeration technologies tested in real life [4]. It is widely used in many industries, such as automotive, commercial, and household, but it has only been possible to imagine mesoscale applications [5] for it in the last ten years or so. This is mostly because of developments in material synthesis and manufacturing methods that make it possible to build pumps, compressors, and heat exchangers that are miniature while maintaining respectable levels of mechanical efficiency and dependability [5].

Because it can alter pressure to cover a suitable range of heating and cooling intended temperature and lower process temperature far below ambient, a VCR system is a valuable tool in the chemical process sector. The refrigeration system [6] operates as a closed system, circulating liquid refrigerant through four stages of alternating compression and expansion, transforming it from liquid to vapour. The temperature of the surrounding air that flows over the unit's components changes as a result of heat being either absorbed or released by the system. This cycle is employed by nearly all refrigeration systems in use today to provide cooling. The evaporator, condenser, compressor, and expansion valve comprise this refrigeration system [7]. The condenser and evaporator, both constructed of coils, serve to enhance the surface area available for the refrigerant's reaction. The mechanical components that regulate the amount of pressure and temperature change between the two stages are the compressor and expansion valve. The system's evaporator and condenser, which control the flow of heat into and out of the system, are located at its opposing ends. The VCR system offers numerous advantages, including the ability to dissipate substantial heat with minimal refrigerant mass flow, high efficiency, and arguably the highest efficiency among macroscale refrigeration systems, resulting in a high COP. Additionally, it can attain sub-ambient temperatures without the need for extra pumping energy at the cold junction, and it operates over an extensive temperature range. In contrast to air compression refrigeration systems, phase change refrigeration uses latent heat, which assures a high value of heat removal while requiring a smaller evaporator.

In the semiconductor industry, thermal control of semiconductor chips is becoming more important. The power consumption and thermal dissipation of a chip increase as the number of transistors increases [8]. The removal of heat generated by electronic chips has emerged as a significant difficulty in the design of portable and space-constrained electronic devices [9]. Since the cooling capacity of passive cooling methods is limited. Therefore, new active cooling methods are needed. With the development of micro technologies, refrigeration systems can be miniaturized and integrated to yield compactness, lightness, high performance, and cost-effectiveness. Miniatured VCR systems may provide how future high-performance chips can be maintained below predicted maximum temperature limits and improvement in performance parameters of the system may increase refrigerating capacity or reduce work done by the compressor of the system. VCR, one of the many active cooling techniques, is regarded as the perfect miniature cooling system because of its low mass flow rate, high COP (approximately 2 to 3), low cold plate temperatures, and capacity to transfer heat away from the source [10,11,12]. Additionally, because it operates at a temperature below ambient, CMOS (complementary-symmetry/metal-oxide semiconductor) transistors can switch on and off more quickly while removing extraordinarily large heat loads quickly. The present study was undertaken to design, develop a 3D model, simulate the working, and evaluate the performance of a Miniaturized VCR System for Electronics Cooling.

MATERIALS AND METHODS

Selection of the Right Refrigerant

There are several refrigerants to choose from when building a refrigeration system. The ones that are used most often are R-11, R-12, R-22, R-134a, and R-502.

Selecting a refrigerant required careful consideration of a number of factors: (i) Ozone depletion and global warming may occur because refrigerants containing chlorofluorocarbons (CFCs) thin the ozone layer in the atmosphere, increasing the amount of UV radiation that reaches the atmosphere and creating the greenhouse effect that causes global warming. International accords (such R-11, R-12, and R-115) prohibit the use of certain CFCs as a result. (ii) Combustibility: all hydrocarbon fuels, including propane. (iii) Detectability of leaks: the refrigerant saturated pressure at the evaporator must exceed P_{atm}. (iv) Thermal factors: (i) A high temperature is required for the refrigerant to vaporise, and as the h_{fg} increases, so does the refrigerating impact per kilogramme of fluid cycled. (ii) The lower the specific heat of the refrigerant, the greater the refrigerating action per kilogramme of refrigerant, and the less heat it will absorb during a given temperature change during throttling or in flow through the pipes. (iii) Because ambient temperatures control condenser and evaporation temperatures, the specific volume of the refrigerant should be low to decrease effort per kilogram of refrigerant cycled. (iv) The evaporator and condenser operating pressures have an impact on the selection. Other desirable properties of the refrigerant include its low cost, chemical stability, and nonflammability [13]. We selected R-134a as the system's refrigerant based on the aforementioned considerations.

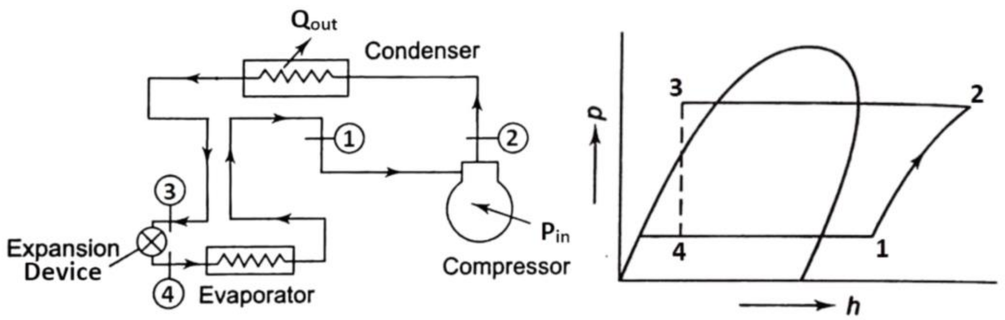
Selection of a Suitable Compressor

In order to find the specifications of a VCR system, we first selected a commercially available compressor with a smaller size, whose cooling capacity was greater than its power input required, such as Trucool Huayi HY113YZ [14]. The selected compressor was a Reciprocating compressor, which had a cooling capacity of 3414 Btu/Hr, i.e., 1000.54463298 W or 1kW (Since, 1 Btu/Hr = 0.29307107 Watts). The power input to the compressor was 455W or 0.455 kW. It had an Energy Efficiency Ratio (EER) of 7.50 Btu/W-Hr. It used R134a as a refrigerant.

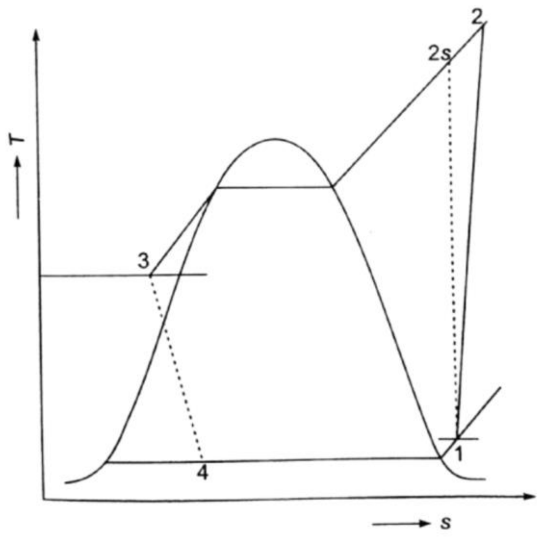
Numerical Method for the Design of VCR system

Determination of Inlet and Outlet Properties of the Evaporator

We assumed, the compressor increased the pressure from 1 bar to 10 bar, the temperature for the inlet of the evaporator was saturation temperature and that for outlet of the evaporator was superheated temperature (Figure I (c)).



(a) **(b)**



(c)

Figure I: (a) VCR system and its components, (b) Pressure-Enthalpy (*p-h*) diagram, and (c) Temperature-Entropy (*T-s*) diagram

Hence, at 1 bar inlet to the evaporator, the temperature was -26.25°C or 246.9 K (Using property tables for Saturated R-134a [15]). Assuming the vapour fraction at the inlet of the evaporator as 0.35, the specific enthalpy at the inlet of the evaporator (State Point 4 of Figure I (b)) using property tables for Saturated R-134a, is given as:

$$h_4 = (h_f)_4 + x(h_{fg})_4 \tag{1}$$

Similarly, the specific entropy at the inlet of the evaporator is given as:

$$s_4 = (s_f)_4 + x(s_{fg})_4 \tag{2}$$

In order to determine the temperature of outlet of the evaporator, we used the cooling capacity value of the compressor. Cooling capacity of a compressor or refrigeration effect being the measure of the amount of heat (in W), was removed from an evaporator or heating unit. Hence, the amount of heat provided by the evaporator was 1 kW (3414 Btu/Hr). Assuming mass flow rate of refrigerant as 0.007 kg/s, the specific enthalpy at outlet of the evaporator (State Point 1 of Figure I (b)) at 1 bar or 0.1 MPa was found using:

$$R.E.\, or\ Q_{in} = \dot{m}_{ref}(h_1 - h_4) \tag{3}$$

Using the specific enthalpy at State Point 1, we determined temperature and specific entropy at the outlet of evaporator with superheated R134a (vapour fraction of 1) at 1 bar or 0.1 MPa using property tables for Superheated R-134a.

Determination of inlet and outlet properties of Compressor

The compressor's intake and evaporator's outlet are coupled, thus the compressor's inlet and evaporator's outlet have the same characteristics. In order to establish the temperature of the outlet of the compressor, we utilised the power input value of the compressor to compute specific enthalpy at the exit of the compressor (State Point 2 of Figure I (b)), which is given as:

$$P_{in} = \dot{m}_{ref}(h_2 - h_1) \tag{4}$$

Using the specific enthalpy at State Point 2, the temperature and specific entropy of superheated R134a (vapour fraction of 1) was determined at 10 bar or 1 MPa using property tables for Superheated R-134a.

Determination of inlet and outlet properties of Condenser

The compressor's outlet was linked to the condenser's inlet, thus the condenser's inlet had the same qualities as the compressor outlet. Assuming that the refrigerant was subcooled (vapour fraction of 0) to 30°C or 303.15 K at 10 bar pressure, the values of specific enthalpy and specific entropy at the outlet of condenser (State Point 3 of Figure I (b) and Figure I (c)) given by the values of specific enthalpy of fluid (h_f) and specific entropy of fluid (s_f), respectively, at 30°C, were determined using property tables for Saturated R-134a.

Determination of inlet and outlet properties of expansion device (Capillary Tube/Throttle Valve)

The expansion device's inlet, which had the same characteristics as the condenser outlet, was linked to the condenser's outlet. The expansion process is a throttling process (isenthalpic process) in the VCR system, therefore, the specific enthalpy at the outlet of expansion device remained constant (State Point 4 of Figure I (b)). Also, the output of the expansion device was linked to the inlet of the evaporator at 1 bar pressure and -26.25°C (246.9 K) temperature (Saturated). In order to check the vapour fraction at outlet of expansion device, the following equation was used:

$$h_4 = h_3 = (h_f)_4 + x(h_{fg})_4 \quad (5)$$

Using property tables for Saturated R-134a, vapour fraction, x at the outlet of the expansion device was determined. The vapour fraction value at the outlet of the expansion device was the same as assumed value of 0.35 at the inlet of the evaporator, verifying that the calculations of the VCR system were correct. In order to calculate specific entropy at the outlet of the expansion device (State Point 4 of Figure I (c)), equation (2) was used. Using property tables for Saturated R-134a, specific entropy was determined at the outlet of the expansion device.

Determination of AEC

AEC is given by the following equation:

$$\eta_{Adiabatic} = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (6)$$

Under ideal conditions, the specific entropy at the outlet of the compressor (s_{2s}) is isentropic with specific entropy at the outlet of the evaporator (s_1). Therefore, h_{2s} was obtained from the corresponding value of s_{2s} using property tables for Superheated R-134a at 10 bar (1 MPa) pressure; and using the h_{2s} value along with the values of h_1 and h_2 calculated in the present study, the value of AEC was calculated.

Determination of COP of the VCR system

The COP of the VCR system was determined using the following methods:

(i) Compressor's EER:

The ratio of an individual cooling device's output cooling energy (measured in BTU/hr) to its input electrical energy (measured in Watts) at a certain operating point is known as its EER. The relation between EER and COP is given by the following equation:

$$EER = 3.412 \times COP \quad (7)$$

Using EER of the compressor, i.e., 7.50 Btu/W-Hr, we determined the COP of the VCR system.

(ii) Compressor's Cooling Capacity and Power Input:

COP is the ratio of Cooling Capacity (W) of the compressor to the power input (W) to the compressor. Therefore,

$$COP = \frac{Cooling\ Capacity}{Power\ Input} \quad (8)$$

Using Cooling Capacity (W) of the compressor as 1000 W, and power input (W) to the compressor as 455 W, we determined the COP of the VCR system.

(iii) Specific Enthalpy relations of the Compressor and the Evaporator:

COP was obtained by specific enthalpies of the evaporator and the compressor using following equation:

$$COP = \frac{h_1 - h_4}{h_2 - h_1} \quad (9)$$

Determination of heat rejected by the Condenser to the surroundings

Heat rejected to the surroundings by the condenser is given by the following equation:

$$Q_{out} = \dot{m}_{ref}(h_2 - h_3) \quad (10)$$

Determination of length and diameter of the Capillary Tube required for the expansion (if Throttle valve is not used)

We know that,

$$Head\ Loss\ due\ to\ Friction, h_f = \frac{4fLV^2}{2gd}\ (Darcy - Weisbach\ Formula) \tag{11}$$

Also,

$$P_2 - P_1 = \rho g h_f \tag{12}$$

Therefore, using equations (11) and (12), we get,

$$P_2 - P_1 = \frac{4\rho fLV^2}{2d} \tag{13}$$

Using property tables for Saturated R-134a at State Point 3 of Figure I (b) and Figure I (c), Specific Volume (v_f) at 30°C was determined in m^3/kg.

Since,

$$Discharge, Q\ (m^3/s) = \dot{m}_{ref} \times v_f \tag{14}$$

Therefore,

$$Velocity, V = \frac{Q}{A} = \frac{4Q}{\pi d^2} \tag{15}$$

Substituting equation (15) in (13), we get,

$$P_2 - P_1 = \frac{64\rho fLQ^2}{2\pi^2 d^5} = \frac{64fLQ^2}{2v_f\pi^2 d^5}\left(\because \rho = \frac{1}{v_f}\right) \tag{16}$$

In order to find Darcy Friction Factor (*f*) we used Colebrook-White equation:

$$\frac{1}{\sqrt{f}} = -2\log\left(\frac{\in}{3.7 \times d} + \frac{2.51}{Re\sqrt{f}}\right) \tag{17}$$

Where, Roughness Height ($\in$) and Inner Diameter of Capillary Tube (*d*) were assumed to be 100 µm and 1.5 mm, respectively. Using property tables for Saturated R-134a, Dynamic Viscosity at 30°C, $(\mu)_{30℃}$ = 0.1877 mPa-s = 0.0001877 Ns/m^2. Reynold's Number, $Re = \frac{\rho Vd}{\mu} = \frac{Vd}{v_f\mu}$.

Substituting $\in$, *d* and *Re* in equation (17), the Darcy Friction Factor (*f*) was calculated. Substituting the values determined above in equation (16), we determined the length of the Capillary Tube.

Analytical Method for the Design of VCR system using DWSIM simulation

The simulated system (Figure II) consisting of compressor, condenser (cooler), throttle valve and evaporator (heater) had material streams and energy streams. It had a recycle block in order to complete the cycle. Throttle valve was used in the simulated system instead of the designed capillary tube because capillary tube was tedious to design in the DWSIM software.

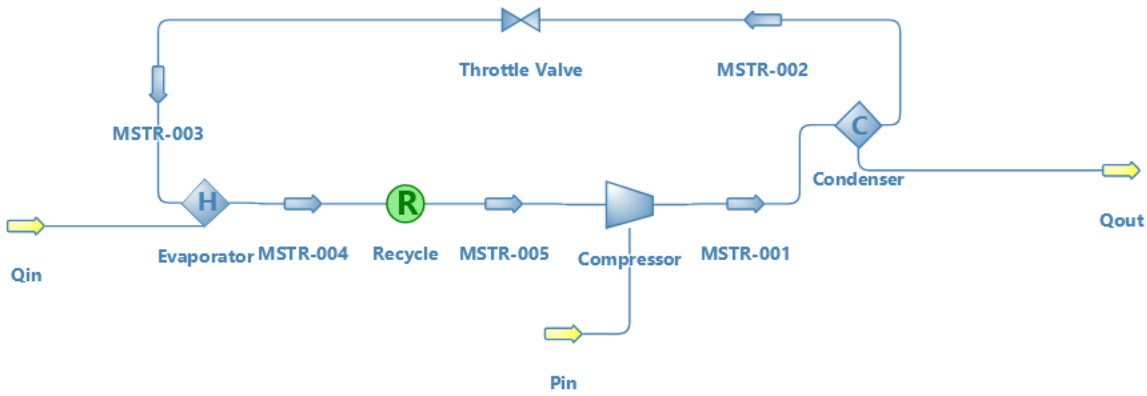


Figure II: Simulated system of the designed miniature VCR system

To perform a simulation in DWSIM, the process began by opening the software and starting a new steady-state simulation. Proceeded through the Configuration Wizard by selecting R134a as the compound, choosing CoolProp as the property package, and setting the system of units to SI. In the flowsheet, added a Heater, renamed it to "Evaporator," and input the required heat of 1 kW. Next, placed a Recycle Block, named it "Recycle," and set it to 50 iterations. Similarly, added a Compressor, renamed it, and input an outlet pressure of 1E+06 Pa. Then, placed a Cooler, renamed it "Condenser," and set the outlet temperature to 303.15 K. Afterwards, added a Valve, renamed it "Throttle Valve," and input an outlet pressure of 100,000 Pa. Adjusted the mass flow rate to 0.007 kg/s and pressed F5 to run the simulation. Upon successful execution, a message would confirm completion, while errors would cause the simulation model to be highlighted in red. To view the results, inserted a Master Property Table in the flowsheet and selected the desired components and properties. For a detailed simulation report, clicked on the Results tab, selected "Create Report," and included the relevant material and energy streams along with their properties.

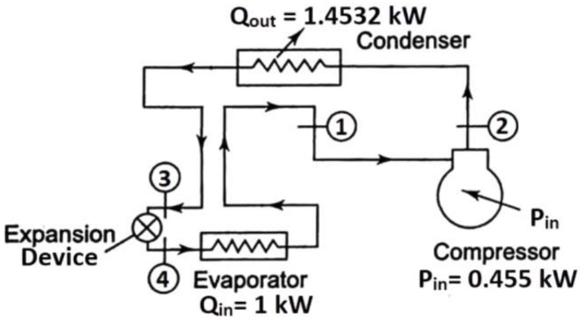
Determination of COP of the simulated VCR System

The COP of the simulated VCR system using DWSIM was calculated by the following relation:

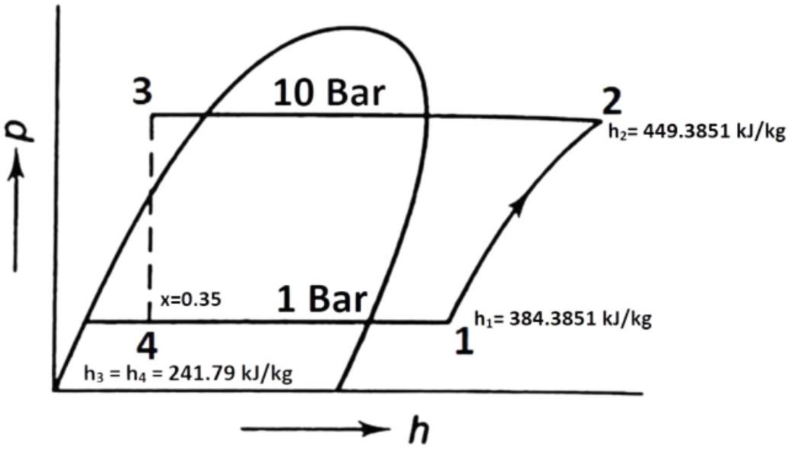
$$COP = \frac{Q_{in}}{P_{in}} \quad (18)$$

RESULTS AND DISCUSSION

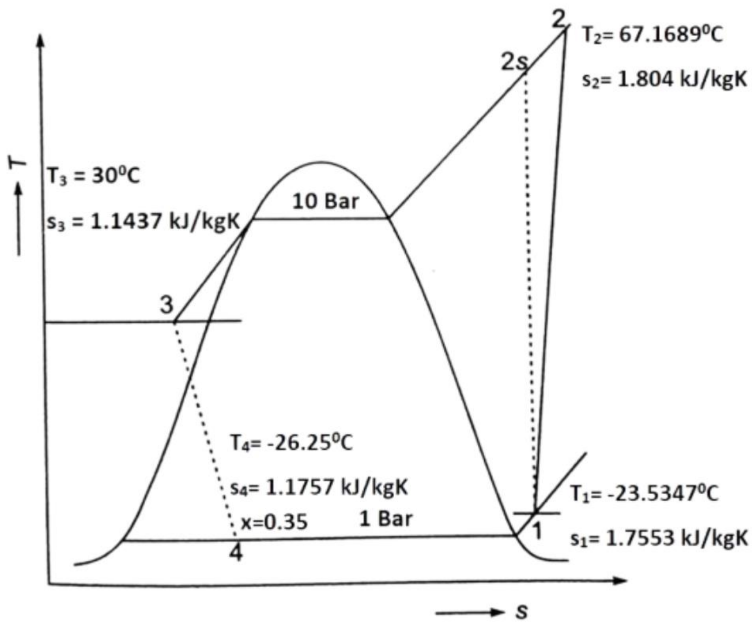
The results from the numerical method are shown in Figure III.



(a)



(b)

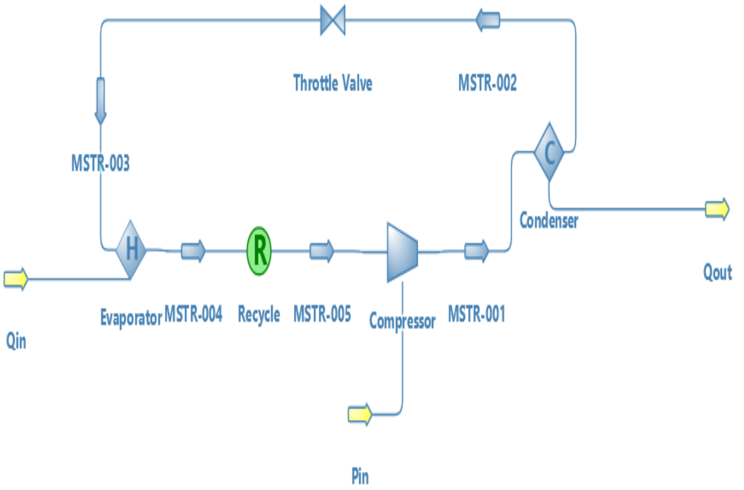


(c)

Figure III: (a) VCR system and its components with Energy Input/Output, (b) Pressure-Enthalpy (*p*-*h*) diagram with calculated values, and (c) Temperature-Entropy (*T*-*s*) diagram with calculated values

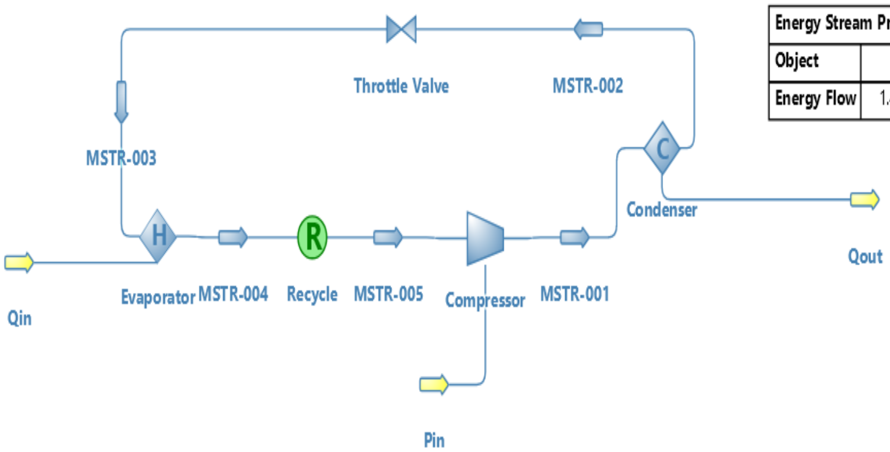
Using numerical method (Figure III), with a 1 kW heat load at the evaporator, the refrigerant R134a, flowing at a mass rate of 0.007 kg/s, heated up to -23.5347°C (249.6153 K) at 1 bar, reaching a superheated state (vapour fraction of 1). After passing through a selected compressor with a power input of 455 W and a calculated AEC of 75.09%, the refrigerant's pressure increased from 1 bar to 10 bar, raising the temperature to 67.1689°C (340.3189 K) while remaining in a superheated state (vapour fraction of 1). In the condenser, the refrigerant released 1.4532 kW of heat through the coils, cooling down to 30°C (303.15 K) at 10 bar, entering a subcooled state (vapour fraction of 0). In the final stage of the VCR system, the expansion device (a capillary tube with a calculated length of 0.6163 m and an inner diameter of 1.5 mm) reduced the pressure from 10 bar to 1 bar, and the temperature dropped to -26.25°C (246.9 K) at a vapour fraction of 0.35. The system's COP was calculated to be 2.2.

The results from the analytical method are as shown in Figures IV and V. Material Stream Property Table (Figure IV) provides results for the refrigerant properties at inlet/outlet and Energy Stream Property Table (Figure V) provides energy inlet/outlet of each component of the VCR system.



Material Stream Property Table						
Object	MSTR-005	MSTR-004	MSTR-003	MSTR-002	MSTR-001	
Temperature	249.277	249.277	246.789	303.15	340.827	K
Pressure	100000	100000	100000	1E+06	1E+06	Pa
Mass Flow	0.007	0.007	0.007	0.007	0.007	kg/s
Mixture Specific Enthalpy	384.573	384.573	241.716	241.716	449.577	kJ/kg
Mixture Specific Entropy	1.75545	1.75545	1.17672	1.14284	1.80449	kJ/[kg.K]
Vapor Phase Molar Fraction	1	1	0.35085	0	1	

Figure IV: Material Stream Property Table



Energy Stream Property Table				
Object	Qout	Qin	Pin	
Energy Flow	1.45503	1	0.45503	kW

Figure V: Energy Stream Property Table

Using the analytical method of DWSIM simulation (Figures IV and V), with a 1 kW heat load at the evaporator, refrigerant R134a, flowing at a mass rate of 0.007 kg/s, heated up to -23.873°C (249.277 K) at 1 bar, reaching a fully superheated state (vapour fraction of 1). After passing through a selected compressor with a power input of 455.03 W and an AEC of 74.88%, as determined by a detailed simulation report, the refrigerant's pressure increased from 1 bar to 10 bar, raising its temperature to 67.677°C (340.827 K) while remaining superheated (vapour fraction of 1). In the condenser, the refrigerant released 1.45503 kW of heat, cooling down to 30°C (303.15 K) at 10 bar, transitioning into a subcooled state (vapour fraction of 0). In the final stage of the VCR system, the expansion device reduced the pressure from 10 bar to 1 bar, dropping the temperature to -26.361°C (246.789 K) with a vapour fraction of 0.35085. The system's COP was calculated to be 2.2.

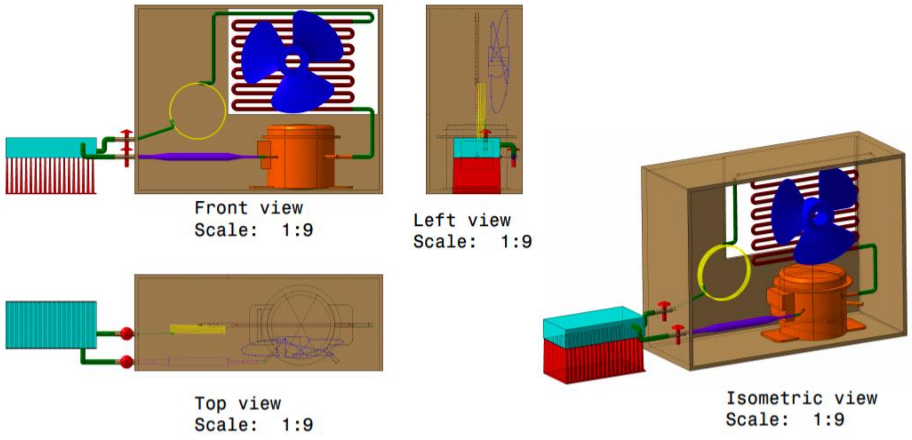
Table I provides a comparison of results from both numerical and analytical method:

Table I: Comparison of Simulation Results with Calculated Values

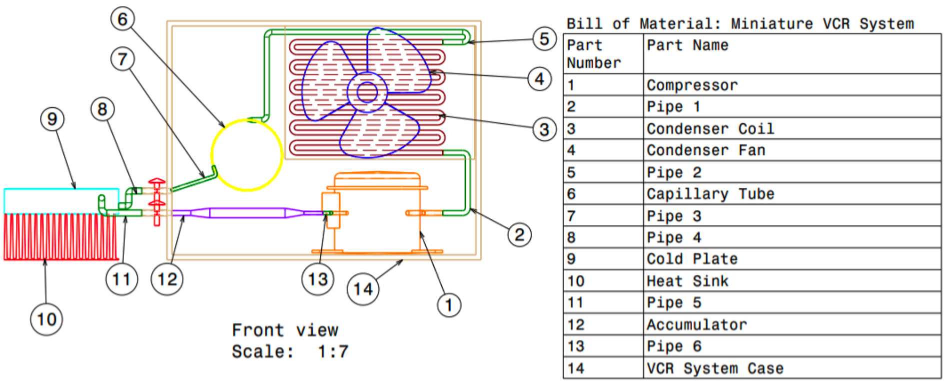
Parameter	Calculated Value	Simulated Value
COP	2.2	2.2
Refrigeration Effect or Cooling Capacity	1 kW	1 kW
Power Input	0.455 kW	0.45503 kW
Heat Rejected to the Surroundings by the Condenser	1.4532 kW	1.45503 kW
Refrigerant Mass Flow Rate	0.007 kg/s	0.007 kg/s
AEC	75.09%	74.88%
Evaporator Inlet Temperature	-26.25^{0}C (246.9 K)	-26.361^{0}C (246.789 K)
Evaporator Outlet Temperature	-23.5347^{0}C (249.6153 K)	-23.873^{0}C (249.277 K)
Compressor Inlet Temperature	-23.5347^{0}C (249.6153 K)	-23.873^{0}C (249.277 K)
Compressor Outlet Temperature	67.1689^{0}C (340.3189 K)	67.677^{0}C (340.827 K)
Compressor Inlet Pressure	1 bar (0.1 MPa)	1 bar (0.1 MPa)
Compressor Outlet Pressure	10 bar (1 MPa)	10 bar (1MPa)
Condenser Inlet Temperature	67.1689^{0}C (340.3189 K)	67.677^{0}C (340.827 K)
Condenser Outlet Temperature	30^{0}C (303.15 K)	30^{0}C (303.15 K)
Expansion Device Inlet Temperature	30^{0}C (303.15 K)	30^{0}C (303.15 K)
Expansion Device Outlet Temperature	-26.25^{0}C (246.9 K)	-26.361^{0}C (246.789 K)
Expansion Device Inlet Pressure	10 bar (1 MPa)	10 bar (1 MPa)
Expansion Device Outlet Pressure	1 bar (0.1 MPa)	1 bar (0.1 MPa)
Expansion Device Outlet Vapour Fraction	0.35	0.35085

The above table shows comparable results, thereby, verifying the practicality of the VCR system.

The VCR system was designed using CATIA V5 software based on the dimensions of its commercially available components, as shown in Figure VI.

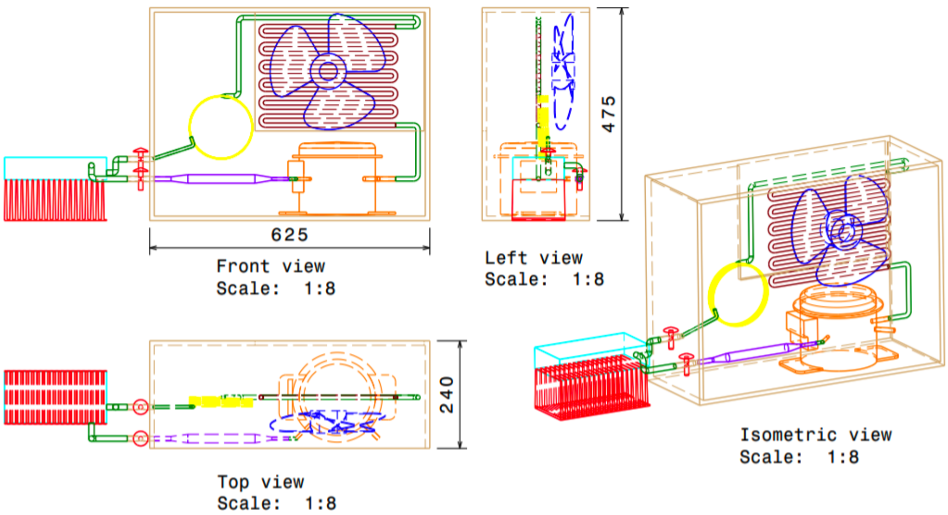


(a)



Part Number	Part Name
1	Compressor
2	Pipe 1
3	Condenser Coil
4	Condenser Fan
5	Pipe 2
6	Capillary Tube
7	Pipe 3
8	Pipe 4
9	Cold Plate
10	Heat Sink
11	Pipe 5
12	Accumulator
13	Pipe 6
14	VCR System Case

(b)



(c)

Figure VI: (a) Different Views of the designed VCR system, (b) Bill of Material for the VCR system, and (c) Size of the VCR system in mm

The analysis compares the performance of a VCR system using numerical and analytical methods. Both methods yielded consistent results, with similar values for key parameters such as COP (2.2), power input, and cooling capacity, confirming the system's efficiency and practicality. The thermodynamic behavior of the refrigerant (R134a) was well-represented in both the methods, showing minimal differences in temperature and pressure across components like the compressor, condenser, and expansion device. The close alignment between the two approaches validates the design and functionality of the system, making it suitable for practical applications. Additionally, the system was designed using CATIA V5 based on commercially available components, further supporting its feasibility.

CONCLUSION

In the present study, a miniature VCR system was designed and tested. Compressor, capillary tube, condenser, cold plate, accumulator, and heat sink were the components of the system. The system was designed for electronic cooling. The dimensions and cooling capacity of the miniaturized VCR system were found to be 625×240×475 mm^3 and 1kW respectively. The miniaturized VCR system showed a COP of 2.2, the mass flow rate of refrigerant R134a as 0.007 kg/s, cold plate temperature (inlet temperature of the evaporator) as -26.25°C, and the AEC as 75.09%. The calculated values of all the parameters were comparable to the simulated values. The current VCR design is an excellent little cooling system that is affordable and capable of removing heat from its source, according to the findings made above. As the COP of a VCR system depends on condenser and evaporator pressure, we suggest in the future one can design a VCR system that has more evaporator pressure and less condenser pressure to improve its performance.

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Cite this Article

Parankush Koul, Pritam Bhat, Aniket Mishra, Chirag Malhotra, Dhanraj Basavaraj Baskar, "DESIGN OF MINIATURE VAPOUR COMPRESSION REFRIGERATION SYSTEM FOR ELECTRONICS COOLING", *International Journal of Multidisciplinary Research in Arts, Science and Technology (IJMRAST), ISSN: 2584-0231, Volume 2, Issue 9, pp. 18-31, September 2024.*

Journal URL: *https://ijmrast.com/*

DOI: *https://doi.org/10.61778/ijmrast.v2i9.79*