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Performance Analysis of Microchannel Heat Exchanger in Vapour Compression Refrigeration System

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ABSTRACT

The purpose of the present study is to develop a setup of a vapor compression refrigeration system operating on low GWP refrigerant i.e. Refrigerant R134-a. Also the important part evaporator construction for maintaining future high- performance microprocessor chips within an acceptable operating temperature range. This requires removing a large quantity of heat, from smaller areas. The present study reports on an experimental evaluation of flow boiling of R134a inside a rectangular microchannel heat sink. The copper test section of parallel rectangular channels is used for the study. The experiments are to be performed with a direct circulation of refrigerant through the microchannel. The wall heat flux can be increased from 12 kW m^{-2} until reaching critical heat flux.

Keywords: *microchannel, heat flux, compressor, condenser, capillary tube*

1. INTRODUCTION

The high heat transfer rate is the need of many systems in today's world. In order to improve their performance, the operating temperatures can be maintained below acceptable levels as in case of high performance computer chips and microelectronic equipment. Microchannels have become very popular in applications where very high heat transfer rates are necessary. Microchannel trend increased as electronics equipment becomes more advanced and smaller in size with continuing innovations. It faces thermal engineering challenges from the high level of heat generation and the reduction of available surface area for heat removal. In the absence of sufficient heat removal, the working temperature of this component may exceed a desired temperature level which then increases the critical failure rate of equipment. Therefore, advanced electronic equipment with high heat generation requires an efficient and compact device to provide proper cooling operation.

In parallel with the rapidly evolving technology, the electronics in every field is increasing and the use of high power electronic systems is becoming widespread. Higher amounts of heat that are generated during operation of such devices are accompanied by a demand for powerful cooling systems because of the fact that traditional cooling systems are inadequate. Microchannel cooling systems are seen as the most promising system to meet this challenge. Due to its very small size and ability to remove excess heat, microchannels are compatible with electronic systems, and it has a wide range of applications such as computer, aviation and space technology. For these reasons, many researchers have been studying the heat transfer characteristics of microchannels for recent years.

There are numerous experimental studies which have concentrated use of microchannel in heat transfer in electronics application. Advanced cooling techniques are essential for further improvement of the gas turbine cooling, compact heat exchangers etc. Boiling heat transfer in microchannel heat sinks has attracted significant interest due to its capability for dissipating high heat flux. High heat flux dissipating microchannels are need of small space applications in microelectronics. Recent advances in micro fabrication technology have resulted in a surge of microchannel cooling.

The main problem in cooling of micro-devices is their high heat generation rate in a limited space. Microchannel heat sink has special consideration due to its capabilities such as high capacity of heat removal. The standard microchannel heat sink consists of a set of microchannel conventionally machined or micro-machined into a conducting block. The performance of microchannel heat exchanger has been the focus on many investigations in recent years, and the subject is treated analytically, numerically, and experimentally. Both the industrial and academic people have taken interest in this area.

The purpose of the present study is to develop a setup of a vapour compression refrigeration system operating on low GWP refrigerant. Also The important part i.e. evaporator construction for Maintaining future high performance microprocessor chips within an acceptable operating temperature range is likely to involve the removal of large quantities of heat, from small areas. Figure 1 shows the proposed layout of system.

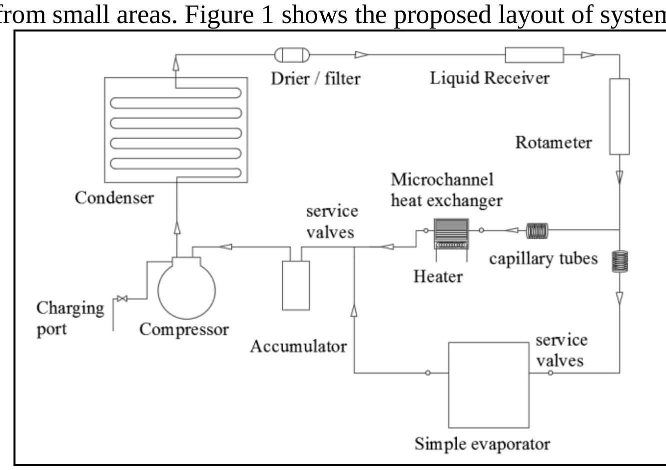


Figure Error! No text of specified style in document.**: Proposed System layout of vapour compression refrigeration system**

3. Design and Selection of System Components

Selection of components is one of the most critical activities in the design cycle for any System. The parameters to be considered for component selection can be broadly classified as technical parameters and non-technical parameters. The selection process in many times is a fine balancing act between the technical and non-technical aspects. The subsequent sections dwell upon these parameters.

The first and foremost question to be asked by the designer is whether the component under consideration meets all the functions desired for the application. The design starts with the technical parameters like functioning ability of components, power consumption, reliability data and scope for enhancement. The non-technical parameters of components selection depends on physical size and cost. Scrutinizing all the data obtained for components selection final design is made for development of experimental setup.

The design of system will consist of following steps:

A. Selection of compressor.
B. Selection of condenser.
C. Design of capillary.
D. Design of evaporator.
E. Selection of accessories.

3.1 Selection of compressor:

A refrigerant compressor, as the name indicates, is a part used to compress vapour refrigerant from the evaporator and raise its pressure so that the corresponding saturation temperature is higher than that of cooling medium. Its function is to compress refrigerant at high pressure and temperature. It also continuously circulates the refrigerant through the refrigerating system [1].

A hermetic or sealed compressor is one in which both compressor and motor are confined in a single outer welded steel shell. The motor and compressor are directly coupled on the same shaft, with the motor inside the refrigeration circuit. Thus the need for a shaft seal with the consequent refrigerant leakage problem was eliminated. All the refrigerant pipeline connections to the outer steel shell are done by welding or brazing. It makes the whole compressor and the motor a single compact and portable unit that can be handled easily.

3.1.1 Advantages

a) More silent in operations and very low in vibration.
b) More compact and less space required
c) The leakage of refrigerant is completely prevented.
d) The lubrication is simple as the motor and compressor operate in a sealed space with the lubricating oil.

3.1.2 Selection Criteria.

a) Evaporating temperature and suction temperature:

Evaporating temperature of compressor affects the volumetric efficiency of compressor. The volumetric efficiency of compressor decreases with decrease in evaporating temperature. This reduces the mass flow of refrigerant through system and hence work of compression increases i.e. Compressor requires more power. For economic performance the evaporator temperature must be kept in limit.

b) Refrigerant:

Hermetic compressors are mostly used for refrigerants like R12, R-22 and R-134a. Hermetic reciprocating compressors are developed with small displacement and relatively high condensing pressure. For the present work to the meet requirement hermetic compressor of 1/3 TR capacity with refrigerant R-134a was selected for attaining low temperature. The specifications of compressor are shown in Table 1.

3.1.3 Specifications of compressor:

Table 1: Specifications of compressor

Sr. No.	Property	Description
1	Manufacturer	Emerson Climate Technologies (India) Limited
2	Refrigerant	R-134a
3	Evaporating temperature	-6.7 ˚C
4	Condensing temperature	54.5 ˚C
5	Number of Cylinders	One
6	Net Weight	11.8 kg
7	Oil Charge	310 cc
8	Oil Type	Refrigeration grade Polyolester (POE)
9	Displacement	12.05 cc/rev
10	Rated cooling capacity	551 W
11	Power consumption	339 W
12	Compressor cooling	Fan, 350 ft3 / minute
13	Electrical rating	230V; 50Hz; 1PH

3.2 Selection of Condenser:

Condenser is heat transfer surface like evaporator. In condenser copper tube transfers heat of refrigerant to the aluminum fins/foils which are cooled with the help of fan .The condenser load is the total heat rejected at the condenser. It includes both the heat absorbed and the energy equivalent and the energy equivalent of the work of compression in compressor. Since the work of compression per unit refrigerating capacity depends on compression ratio, the quantity of heat rejected at the condenser per unit refrigerating capacity varies with the operating condition of the system [1]. Air cooled condenser used in this system for heat rejection through refrigerant.

3.2.1 Advantages of Air Cooled Condenser

A. Air-cooled condensers are cooled by ambient air and no water is required.
B. There is also no need for a cooling tower and condenser water pump.
C. There is no handling problem with air cooled condenser
D. No problem with disposing used air as like water cooled condenser
E. Fouling factor has less effect.

Firstly face area is calculated for condenser design. By considering tube side dimensions that are available in market. The outside, inside and mean areas are calculated for the tube side. The fin effectiveness is calculated from fin efficiency diagram by converting rectangular fin section to annular fin. The prime area for air flow and overall extended fin area is calculated. The Air side heat transfer coefficient is calculated for normal fan velocity.

As by required mass flow rate obtained from theoretical cycle the properties of refrigerant at condensing temperature are obtained from charts. From these properties velocities of refrigerant flow is calculated. The dimensionless parameters like Reynolds number and Prandtl number are calculated, they shows turbulence in their nature. Then modified Reynolds no is calculated from liquid and vapor phase of refrigerant and used for Nussle number calculation given by Akers, Dean and Crosser Correlation [2].

$$Re_m = Re_f \left[1 + \left(\frac{\rho_f}{\rho_g}\right)^{0.5}\right] \quad (1)$$

$$N_u = 0.0265 \times Re_m^{0.8} \times P_{rf}^{1/3} \quad (2)$$

Heat transfer coefficient at refrigerant side by considering thermal conductivity of liquid W/m^2 K

$$hi_f = \frac{N_u\, k_l}{d_i} \quad (3)$$

After getting inside heat transfer coefficient of refrigerant flowing overall heat transfer coefficient is obtained by,

$$\frac{1}{U_o\, A_o} = \frac{1}{h_a\left(A_p + \eta A_e\right)} + \frac{A_o r_i \ln\frac{r_o}{r_i}}{A_i k_t} + \frac{1}{A_i h_i} \quad (4)$$

The log mean temperature difference is calculated by considering temperature increment of 5ºC air as cooling medium,

$$\Delta T = \frac{(T_{heated\ air} - T_{ambient})}{\ln\left(\frac{T_{condenser} - T_{ambient}}{T_{condenser} - T_{heated\ air}}\right)} \quad (5)$$

Finally condenser capacity is obtained which is greater than the required capacity.

$$Q = U_o A_o \Delta T \quad (6)$$

3.3 Design of Capillary tube:

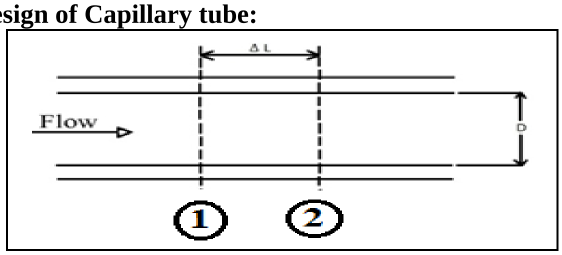


Figure 2: Control volume of capillary tube

The mass flow rate through corresponding evaporator is w = 0.0017 kg/sec. The diameter of capillary tube used is 0.000797 m. i.e. (1/64th inch). The fundamental equations [3] applicable to the control volume bounded by points 1 and 2 in Figure 2 are

a) Conservation of mass.
b) Conservation of energy.
c) Conservation of momentum.

The equation for conservation of mass states that,

$$w = \frac{V_1 \times A}{v_1} = \frac{V_2 \times A}{v_2} \quad (7)$$

$$\frac{w}{A} = \frac{V_1}{v_1} = \frac{V_2}{v_2} \quad (8)$$

$\frac{w}{A}$ Will be constant throughout the length of capillary tube

The Statement of conservation of energy is,

$$1000\, h_1 + \frac{V_1^{\,2}}{2} = 1000\, h_2 + \frac{V_2^{\,2}}{2} \quad (9)$$

Assuming negligible heat transfer in and out of the tube.

The momentum equation in words states that the difference in forces applied to the element because of drag and pressure difference on opposite ends of the element equals that needed to accelerate the fluid.

$$\left[(P_1 - P_2) - f\frac{\Delta L}{D}\frac{V^2}{2v}\right] = \frac{w}{A}\,(V_2 - V_1) \quad (10)$$

As the refrigerant flow passes through the capillary tube, its pressure and saturation temperature progressively drop and fraction of vapour x continuously Increases at any point.

$$h = h_f(1 - x) + h_g\, x_o \quad (11)$$

$$v = v_f(1 - x) + v_g x \quad (12)$$

As the refrigerant flows through the capillary tube from point 1 to point 2, V/v is constant so that,

$$\left[f\frac{\Delta L}{D}\frac{V^2}{2v}\right] = \left[f\frac{\Delta L}{D}\frac{V^2}{2}\frac{w}{A}\right] \quad (13)$$

Now, mean velocity used for calculation being,

$$V_m = \frac{V_1 + V_2}{2} \quad (14)$$

For Reynolds numbers in the lower range of turbulent region an applicable equation for friction factor 'f' is obtained from Hopkins 1950 is,

$$f = \frac{0.33}{R_e^{\,0.25}} = \frac{0.33}{\left(\frac{V\,D}{\mu\, v}\right)^{0.25}}$$

Viscosity of the two phase refrigerant at a given position in the tubes is a function of the vapour fraction x,

$$\mu = \mu_f(1 - x) + \mu_g\, x$$

The mean friction factor f_m applicable to increment of length 1-2 is,

$$f_m = \frac{f_1 + f_2}{2} \quad (17)$$

3.3.1 Calculating length of increment:

Combining above equations (7) to (17) one quadratic equation is obtained as,

$$1000\, h_{f_2} + 1000(h_{g_2} - h_{f_2})x + \left[\frac{v_{f_2} + (v_{g2} - v_{f_2})x}{2}\right]\left(\frac{w}{A}\right)^2 = 1000\, h_1 + \frac{V_1^{\,2}}{2} \quad (18)$$

Everything in that equation is known except x which can be solved as,

$$x = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} \quad (19)$$

Where,

$$a = 0.5 \times (v_{g2} - v_{f_2})^2\left(\frac{w}{A}\right)^2 \quad (20)$$

$$b = 1000\left(h_{g2} - h_{f_2}\right) + v_{f_2}(v_{g2} - v_{f_2})\left(\frac{w}{A}\right)^2 \quad (21)$$

$$c = 1000\,(h_{f2} - h_1) + 0.5 \times \left(\frac{w}{A}\right)^2 \times v_{f_2}^{\ 2} - \left(\frac{V_1^{\,2}}{2}\right)^2 \quad (22)$$

With the value of x known h_2, v_2, V_2 can be calculated. Computation equations for properties of saturated refrigerant R 134-a applicable to a temperature range -10 to 50 ˚C can be used, conditions at entrance to capillary tube at point 1. The entering refrigerant at 40˚C and with x=0, then calculation done for Velocity, Reynolds no, and friction factor as shown in Eq. (7) to (22) and using excel computing program then calculating the length of segment having temperature drop of 1˚C. Then addition of each increment up to -5˚C gives the total capillary length is equal to 1.75 m as shown in Table 2.

Table 2: Iterations performed on excel program

Posistion	Temperature °C	Pressure kPa	x	Specific volume	Enthalpy kJ/kg	Velocity m/s	Increment length m	Cumulative length
1	40	1016.6	0	0.0009	256.41	2.42	0	0
2	39	9898.7	0.009	0.0010	256.41	2.91	0.000164	0.44
On further iterations								
40	-4	252.68	0.29	0.02	254.38	67.49	0.0007	1.75
41	-5	243.48	0.30	0.02	254.17	70.96	0.0005	1.75

3.4 Selection of Accessories:

Receiver-Drier-Filter assembly is also an important part to refrigeration system [1]. It is installed in the refrigeration loop on the high-pressure side, downstream of the condenser. It uses several types of desiccants, the most common of which is spherical molecular sieves; silica gel is occasionally used. The receiver-drier,

a) Serves as a reservoir for refrigerant from part- to full-load operating conditions.
b) Removes moisture from the system.
c) Filters out debris headed for the capillary, and
d) Only allows liquid refrigerant to enter the capillary tube

Rotameter is used to measure the mass flow rate of the refrigerant and to determine visually whether or not system has sufficient charge of refrigerant. It works on the simple principle that as the area of the flow passage increases as the float (a shaped weight, made either of anodized aluminium or a ceramic) moves up the tube, the scale is approximately linear for the flow rate range of 6.5 to 65 LPH.

Compressors are extremely susceptible to damage from liquid refrigerant. Excessive liquid refrigerant return may cause not only oil dilution but complete loss of the compressor oil charge which results in equipment damage due to lack of proper lubrication. Accumulator in the line, as close as possible to the compressor, is a lifesaver for the equipment, assuring adequate oil return and that refrigerant return only, is returning to the compressor. An accumulator assures control and protection of the compressor. A suction-line accumulator is required with an orifice tube to ensure uniform return of refrigerant and oil to the compressor, to prevent slugging, and to cool the compressor. It also stores excess refrigerant.

3.5 Design of microchannel heat exchanger

The microchannels contains 29 channels with the cross section of 1.1×0.28 mm^2 and is made by joining two such microchannel of cross section 0.55×0.28 mm^2 on one over another, channels are separated by ribs of 0.28 mm width. Two parts of microchannels are both 40 mm long in the flow direction. The heat sink case is fabricated in copper, with the overall height of 29 mm, length of 160 mm and width of 50 mm. This consists of combination of lower and upper manifold. The upper as well as lower manifolds are joined with each other with the help of brazing to make total height of channel 1.1 mm in the middle position, with the purpose of enhancing loading capacity and preventing the heat sink surface from deformation under high-pressure operation. Taking into consideration the microchannel heat sink and heat source are also brazed together, with a heat transfer area (20×40 mm^2) at the bottom of microchannel in the manifold.

In manifold, there were 7 holes both sides of manifold with an inner diameter of 2 mm, used for inserting the thermocouples to measure the temperature of heat sink bottom surface, inlet and outlet fluid. These thermocouples were sealed with thermal insulation adhesive. Due to the limitation of machining technology, the insertions of thermocouples were not deep enough to measure the central temperature of heat sink. In order to optimize the mass flux distribution in the microchannel heat sink, two holes with an inside diameter of 8 mm and length of 10 mm are drilled to the inlet and outlet of heat sink manifold, so that refrigerant can flow into these holes and microchannels[4]. The simulation result obtained from the Ansys fluent shows the coolant refrigerant R134-a flowing in the microchannel heat exchanger without heat transfer as displayed in Figure 3. The flow velocity distribution can be considered uniform, because the maximum deviation from average velocity was found below 12%. Although there were small dead zones or vortex can occur in the non-heat transfer sections, it had little effect on the heat transfer. As the flow boiling occurs in microchannels, the mutual intervention exists between gas-liquid phases, which can improve flow uniformity to some extent.

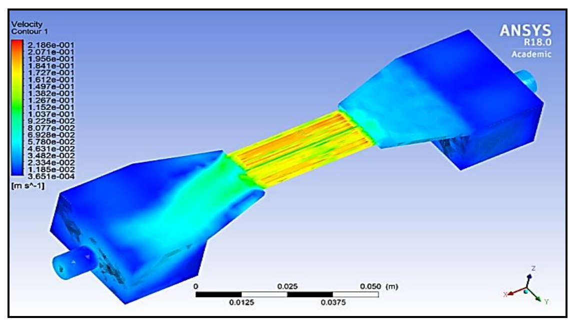
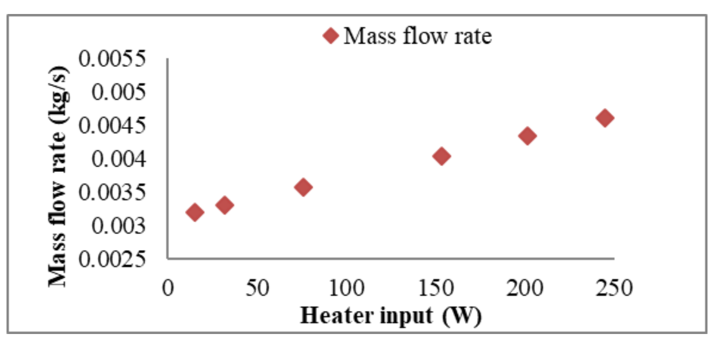


Figure 3: Velocity contour in microchannel fluid domain

4. RESULTS AND DISCUSSIONS

The observation in Table 3 shows that the mass flow rate of refrigerant through the microchannel evaporator needs to be increased with the increased wattage of heater. The heater wattage is increased from the 15.03W to 244.77W to reach the critical heat flux value. The temperature readings in ºC at various thermocouple points are recorded for theoretical calculations.



Graph 1: Heater input vs. mass flow rate

Table 3: Observations for increased heater Watt

Heater W	C_i	C_o	E_i	E_o	Mf_i	$Plenum_i$	Ch_i	C_h near heater	Ch_o	$Plenum_o$	inner cyl. Glycol	coil bath
15.03	58	34	0	1	5	4	4	6	4	4	4	2
32.076	57	36	0	2	5	7	5	8	5	4	6	2
75.9	53	31	0	3	5	9	6	12	6	6	10	4
153.26	57	32	1	1	6	11	5	15	8	5	3	2
201.6	56	31	1	1	4	11	4	14	8	3	3	2
244.77	58	31	2	2	6	14	6	17	10	5	2	2

The Graph 1 shows that as the heater input is increased then the mass flow rate of refrigerant required to be increased. This is done manually with the help of service valves in order to get same COP.

5. CONCLUSIONS

Experimentation on microchannel heat exchanger is done. The results showed that as the wattage of heater input is increased at the time of experimentation the mass flow rate of refrigerant is required to increase. This increased mass flow rate takes care of the increasing heat flux. This provides that the mass flow of refrigerant accommodates the heater input for the stabilization of the refrigeration system in order to remove high heat flux from the substrate or source.

Based on the results obtained following conclusions are made:

1) It was an attempt to develop a setup of vapour compression refrigeration system with individual expansion valves for microchannel evaporator as well as the simple evaporator.
2) With the present system 5°C temperature in microchannel evaporator is obtained, while some refrigerant discharge is circulated through the simple evaporator.
3) High heat flux from microchannel evaporator is removed by accommodating the required charge distribution in the system.
4) The microchannel assembly is also analyzed with the help of simulation software ANSYS Fluent that showed the behavior of refrigerant flow in microchannel.

6. Acknowledgement

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