

A CASE STUDY OF A LOW POWER VAPOR COMPRESSION REFRIGERATION SYSTEM

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Abstract

Reported in this paper is a case study on a normal vapor compression refrigeration system which is expected to be run by photovoltaic panels to utilize minimum grid power. A small 120 W refrigerator is fabricated out of commercially available components and run by an inverter and battery connected to solar photovoltaic panel as well as grid. Temperature at several points was measured and the performance was evaluated. The Coefficient of performance (COP) to run such refrigerator is estimated after numerical simulation of major components namely, evaporator, condenser and a capillary tube. The simulation was done to obtain an effective cooling temperature and the results were compared with measured temperatures. Calculation proves to be in conformity with the actual model.

Keywords: Vapor compression refrigeration, Photovoltaic panels, CFD simulation

1. INTRODUCTION:

Refrigeration is the process of achieving temperatures below that of the local environment. The main purpose of refrigeration is thermal conditioning (e.g. food, medicine, vaccine, blood etc.), and the basic apparatus is a refrigerator. Ramani *et al.* reported a consolidated review on solar refrigeration technology [1]. Most of the Solar refrigeration is focused on Ammonia-water system. They reported that the generator inlet temperature of the chiller is the most important parameter in the design and fabrication of a solar powered refrigeration and air conditioning system. Optimized thermal collector, system design and arrangement are other important parameters for the system operation. Inconsistent heating in the thermal collector is the biggest challenge in such system.

A vapor compression refrigeration system can also be run by photovoltaic (PV) panel. Ewart *et al.*, [2] reported the results of field testing on photovoltaic direct drive, battery free solar refrigerator. Solar refrigeration system studied by Klein and Reindl, members of ASHRAE, emphasizes on minimizing environmental impacts associated with refrigeration system operation [3]. They concluded that it is reasonable to evaluate the prospects of a clean source of energy vis-à-vis solar refrigeration. Inan, *et al.*, [4], gave an insight on various methods of analysis performed on the refrigerated space and on evaporator such as heat and mass transfer through a domestic refrigerator and evaluation of evaporator under frosted conditions. Critoph reported that a PV operated refrigerator would require standard mechanical vapor compression cycle, requiring an electrical input to a hermetically sealed compressor [5]. The electricity is generated by photovoltaic panels. This has the advantage of using off the-shelf technology, but the disadvantages of high cost and the probable need for a battery.



A vapor compression refrigeration system consists of compressor, evaporator, condenser and a capillary tube. Because of expansion in the capillary tube the cooling effect is experienced at the evaporator. Analyzing the performance of a capillary tube is of great interest for thermal engineers and researchers. The CFD simulation on different types of capillary tubes was performed by Kumar and Sri [6]. They replaced a straight capillary tube by a helical tube of 40 turns and a pitch of 3 mm. A comparison of capillary tube simulation using C++ code and ANSYS CFX was done by Shivkumar and Hebbal [7] where a conclusion was made that there was a considerable agreement in experimental and simulation results for straight capillary tube. A simulation on the heat transfer analysis on Evaporator was done with influx of CuO nanoparticle by Coumaressin and Palaniradja [8] and proved that evaporating heat transfer coefficient increases with amount of CuO. A numerical simulation of the liquid overfeed system has been done by Valladeres *et.al* [9]. This has been done by considering general conservation equations for capillary tube and lumped models for evaporator and condenser. This study considers Refrigerant R134a which is commercially known as Floron. R134a used in this system satisfies the Montreal Protocol [10].

2. MATERIALS AND METHODS:

A small 120 W refrigerator is fabricated out of commercially available components and run by an inverter (Sukam Brainy 850VA) and battery (Exide Invatubular 150Ah) connected to solar photovoltaic panel (250Wp Multi crystalline) as well as grid so that the power requirement remain minimum. Thermocouples were used to measure the temperature at evaporator, condenser and capillary tube entry as well as at exit. For a low temperature producing device like a refrigerator it is important to estimate the internal energy from the following equation:

$$mC_v \Delta T = Q + E - \int p dV \quad (\text{Eqn. 1})$$

In absence of any colder system to accept thermal energy to (i.e. with $Q=0$), and knowing that energy dissipated by friction is $E \geq 0$ in any process, lower the initial temperature of a system (i.e. achieve $\Delta T < 0$) by forcing a fast expansion in a compressible substance ($dV > 0$). This adiabatic cooling may be used to cool (by normal heat dissipation) the space to be refrigerated (evaporator in Figure 1). The expansion must end sooner or later, and to restart the cycle compress the refrigerant by a mechanical vapour compressor, which also increases its temperature. Refrigerant has to get rid of the thermal energy through a condenser to heat sink (surrounding atmosphere). A highly compressible refrigerant (R134a) with minimum friction dissipation is to be preferred. R134a has zero Ozone Depletion Potential (ODP) and it's widely used [11].

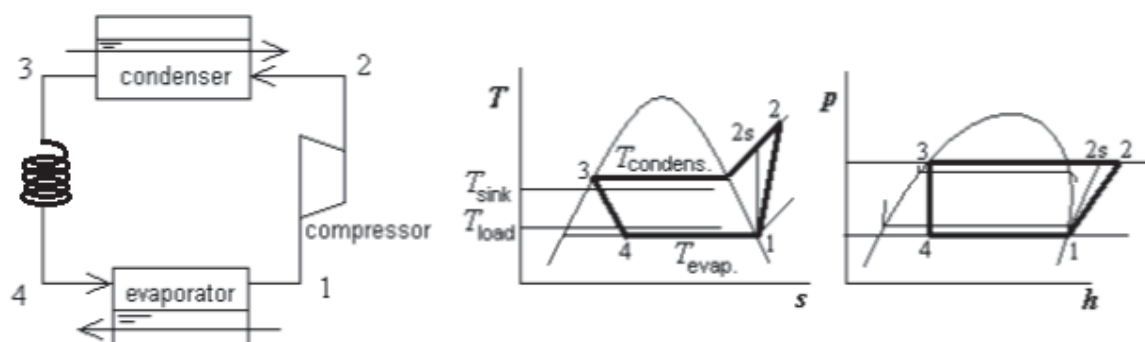


Fig 1: Schematic of a simple vapour compression refrigeration system and its Temperature (T) Enthalpy (s) diagram and pressure (p) enthalpy (h) diagram [12]

3. EQUATIONS AND MATHEMATICS:

The numerical analysis was done following the paper of Valladeres *et.al*, [9] considering the following general conservation laws: 1) Conservation of mass, 2) Conservation of momentum and 3)

Conservation of energy. These give rise to a set of equations from which the heat transfer amount may be calculated [13].

- **EVAPORATOR:**

The following equation (Eq. 2) gives the heat flux from the evaporator considering the inlet conditions as mentioned above (process 4-1 in T-s and p-h diagram). These give the general boundary conditions for the simulation to be run.

$$\dot{Q}_e = \dot{m}_e C_p (T_e - T_i) \quad (\text{Eqn. 2})$$

Where the mass flow rate and C_p of the refrigerant is known and the total heat flux can be calculated.

- **CONDENSER:**

The heat transfer for the condenser is calculated by using the following equation (Eq. 3). With the inlet conditions of mass flow rate input C_p of R-134a at the superheated vapor conditions (process 2-3 in T-s and p-h diagram).

$$\dot{Q}_c = \dot{m}_c C_p (T_{c,i} - T_{c,o}) \quad (\text{Eqn. 3})$$

- **CAPILARY TUBE:**

The Expansion device or the capillary tube is considered to undergo a throttling process. Length of the helical tube can be obtained from Eqn. 4; where p is coil pitch and N is no of turns respectively (process 3-4 in T-s and p-h diagram).

$$L = \{ (\pi D N)^2 + (p N)^2 \}^{1/2} \quad (\text{Eqn. 4})$$

Power of the compressor can be calculated using the formula as mentioned below. The work done by compressor is given by formula

$$W_{\text{Compressor}} = \frac{n}{n-1} m R (T_{c,in} - T_{e,out}) \quad (\text{Eqn. 5})$$

In the Eqn. 5, m is the molecular mass of the refrigerant and R is the universal gas constant = 287 J/kgK. The compression process is assumed to be polytropic and $n=1.52$ in liquid conditions and $n=1.18$ in vapour conditions. The Coefficient of Performance or C.O.P is estimated by the following equation.

$$C.O.P_{\text{estimated}} = \frac{T_{e,o} - T_{e,i}}{\left(\frac{n}{n-1}\right) \left(\frac{\gamma-1}{\gamma}\right) (T_{c,i} - T_{e,o} + T_{e,i} - T_{c,o})} \quad (\text{Eqn. 6})$$

4. COMPUTATIONAL ANALYSIS:

The geometric model creation was done using CATIA V5R20 (shown in Fig 2, Fig 3 and Fig 4) with the measurements taken from the setup and given in table 1. Dimensions of the parts are as per commercially available components.

Table 1: Three major components of vapor compression refrigerator

COMPONENT	DIMENSIONS
Capillary Tube	D= 2mm; p=2mm; N=6
Condenser	D=6.3mm, l=43.5mm (half loop)
Evaporator	D=5mm, l=85mm

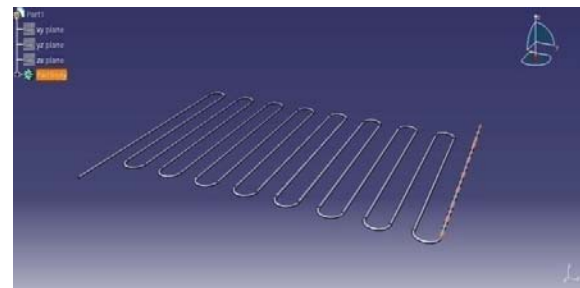


Fig 2: Condenser of the vapor compression refrigerator

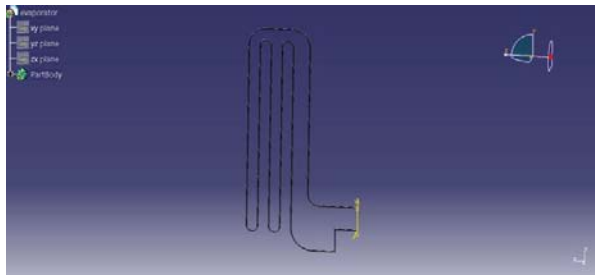


Fig 3: Evaporator of the vapor compression refrigerator



Fig 4: Capillary tube of the vapor compression refrigerator

For the computational analysis [14] we used ANSYS Fluent [15, 16]. The boundary conditions were considered as mentioned above. CFD analysis was carried out with the conditions as follows (Table 2).

Table 2: Boundary conditions

Conditions	Condenser	Evaporator	Capillary Tube
Mass Flow rate	0.05 kg/s	0.05 kg/s	0.05 kg/s
Pressure	100kPa	88kPa	10kPa
Inlet Temperature	318 K	263 K	313 K

The model that was chosen was turbulent flow [17, 18] with $k-\omega$ viscous turbulence model considering the Reynolds number of the flow from the mass flow rate and that the $k-\omega$ gives a faster convergence when compared to other turbulent models. Spalart-allmaras was not chosen as it is primarily used for external flows in application of aerodynamics with an extra viscosity term [16], in addition the $k-\omega$ model takes into account turbulent kinetic energy (k) and specific rate of dissipation of kinetic energy (ω) while considering wall functions.

Table 3: Properties of the refrigerant R134a is obtained from International Institute of Refrigeration [10]

Properties at 0°C (at saturation)	Unit (SI)	Liquid	Vapour		
Pressure	MPa	0.29	0.29	Chemical Formula	CF ₃ CH ₂ F
Volume mass	dm ³ /kg	0.77	69.31	Molar mass	0.102
Specific heat capacity				Boiling Temperature	246.6 K
• at constant pressure	kJ/(kg K)	1.34	0.90	Critical Temperature	374 K
• at constant volume	kJ/(kg K)	0.88	0.76		
Viscosity	10 ⁻⁶ Pa s	271.08	10.73	Critical pressure	4.07kPa
Thermal conductivity	W/(m K)	0.092	0.012	Liquid Density at 298 K	1370 kg/m ³
Surface Tension	N/m	0.012	-	Vapor Pressure at 298 K	107kPa
Heat of Vaporization	kJ/kg	18.6	-		

5. RESULTS AND DISCUSSION

Variation of Reynolds number (Re) of the analyzed components is shown in Figure 5, 6 and 7 respectively. Every run converged within our expected number of iterations for given accuracy. Reynolds number is given in each contour. Viscosity is almost kept constant at 271×10^{-6} Pa s for capillary tube [19] and the entry viscosity varies between 10.73×10^{-6} and 271×10^{-6} Pa s for the Condenser and evaporator since they host multiphase flow.

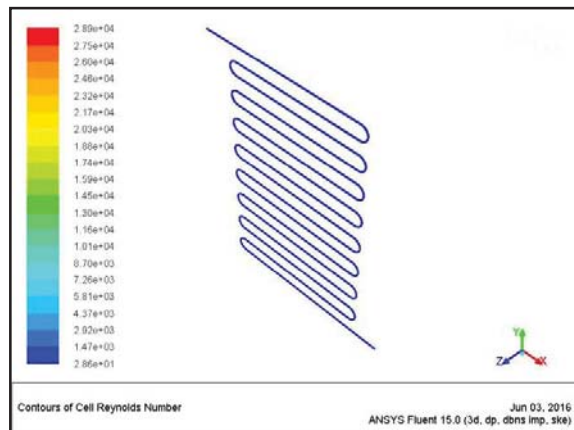


Figure 5: Variation of Re through Condenser

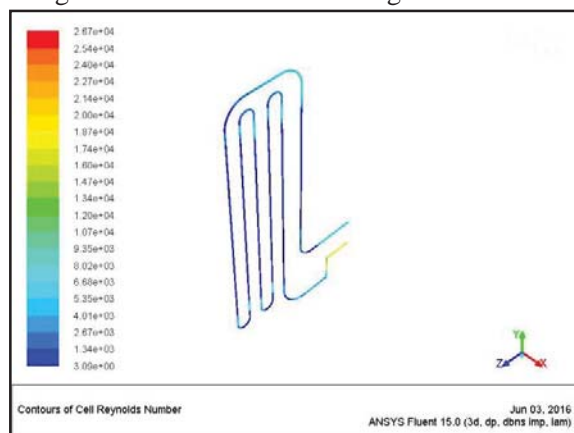


Figure 6: Variation of Re through Evaporator

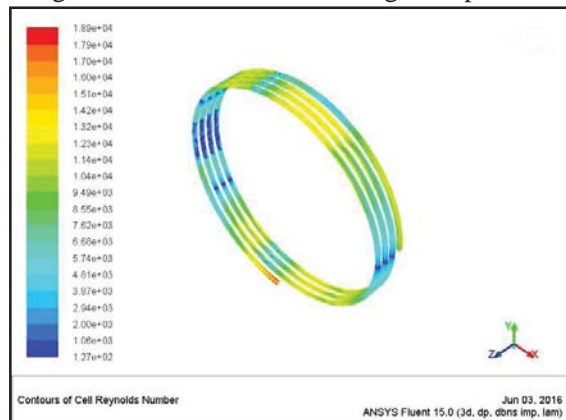


Figure 7: Figure 5: Variation of Re through Capillary Tube

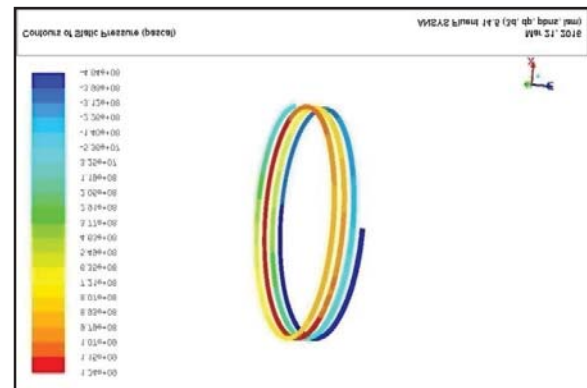


Figure 8: Pressure distribution through Capillary Tube

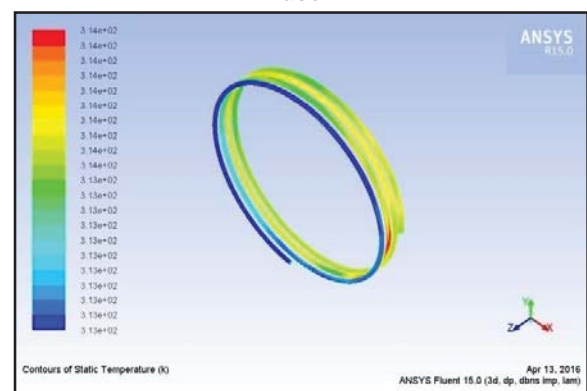


Figure 9: Temperature distribution through capillary tube

Table 4: Summary of simulation results

COMPONENTS	TEMP. (K)		PRESSURE (kPa)	
	IN	OUT	IN	OUT
Condenser	318	313	100	88
Capillary Tube	313	261	88	10
Evaporator	261	271	10	16.2

Almost similar temperatures were observed when measured using a thermocouple in the working model. Therefore applying Eqn. 6 estimated COP is ~3, however, we may assume a more realistic value of 2 for further calculation (section 5.1).

From Table 4 we get the approximate values of temperature. Average temperature of evaporator is important here, -7 °C. Now we develop a Cuboid box similar to our refrigerator box using CATIA and run analysis using ANSYS. To check the insulation effectiveness, heat dissipation is analyzed [20, 21]. The following contour shows the temperature profile of the refrigerator when a body having temperature of maximum 80 °C is kept in the refrigeration space. The evaporator is present on the lateral side of the refrigerated space with dimensions as 42cm×18cm (Figure 10). The following temperature profile shows

the refrigeration effect and the heat transfer when a sudden change in temperature is observed. The radiation from walls is assumed to be 100W/mK according to calculations done by heat resistance method.

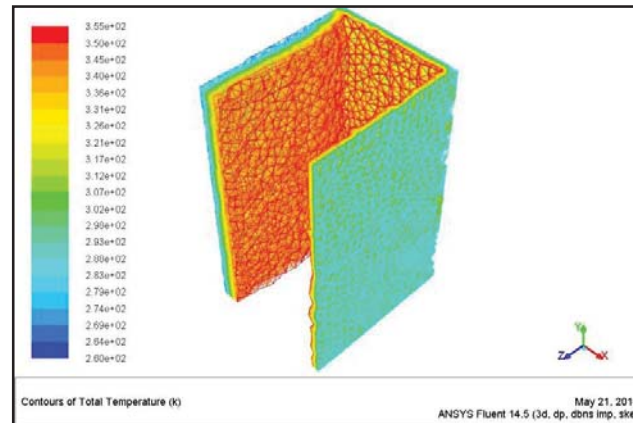


Figure 10: Temperature distribution through refrigerator wall when a hot body is kept inside

5.1. Calculations for refrigeration load

Though applying Eqn. 5 estimated power consumption of compressor is ~4.42 W only, the refrigerator will consume much more power than compressor alone (rated power of the refrigerator is 120 W). A model calculation with basic heat equations is given as follows:

Say, refrigerator is required to store 20 kg of fish (a common domestic food item). The fish is supplied at a temperature, $t_2 = 30^\circ\text{C}$. The specific heat of fish above freezing point is 2.93 kJ/kg K [20]. The specific heat of fish below freezing point is 1.26 kJ/kg K. The fish that can be stored at $t_1 = -7^\circ\text{C}$ (average of inlet and outlet temperature of evaporator; Table 4). The approximate freezing point of regular fish, $t_3 = -4^\circ\text{C}$ [22]. The latent heat of fish is 235 kJ/kg. We try to find time required to achieve cooling.

Assume actual COP = 2; Heat removed by the plant = Actual COP \times Work required = $2 \times 0.12 = 0.24$ kJ/s

We know that heat removed from the fish above freezing point, $Q_1 = m C_{AF} (T_2 - T_3) = 1992$ kJ

Similarly, heat removed from the fish below freezing point, $Q_2 = m C_{BF} (T_3 - T_1) = 75.6$ kJ

Total latent heat of fish, $Q_3 = m \times h_{fg \text{ fish}} = 20 \times 235 = 4700$ kJ

Total heat to be removed by the plant = $Q_1 + Q_2 + Q_3 = 1992 + 75.6 + 4700 = 6793$ kJ

Time taken to achieve cooling = $6793 / 0.24 / 3600 \text{ h} = 7.83 \text{ h}$

Therefore, running the refrigerator more than 7.86 h is needed to achieve the required cooling.

6. CONCLUSION:

The current work is an attempt to find the suitability of simulation of regular components of a refrigerator system. The simulation was done to obtain an effective cooling temperature and compare it with measured temperatures. The influence of the boundary conditions like heat source and heat sink temperatures has been studied. The refrigerated space was constructed in accordance to the heat transfer concepts such as conduction and convection. Simulation was also carried out to find the suitability of the given insulation. According to the results obtained insulation is found sufficient.

COP is calculated based on measured temperatures which are almost similar to predicted temperatures. A test case is developed and basic calculations were shown to find the suitability of given refrigerator for a predetermined cooling load and its relationship with COP.

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