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# Refrigeration by using R134-a and Nanofluids with Capillary Tube Optimization

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**ABSTRACT**

**The use of nano-fluid along with the conventional refrigerants in vapour compression cycle is a relatively a new idea, where nano-refrigerants, so obtained are found to have their improved thermal, physical properties over the conventional refrigerants. Nanoparticles can be used along with refrigerant in order to improve the performance of vapour compression refrigeration system. In this study, alumina ($Al_2O_3$) nanoparticles of 20 nm diameter are dispersed in refrigerant R134-a to improve its heat transfer performance.**

***Keywords:*** **Nano-fluid, Nano-refrigerants, Heat transfer, Mixture-3(75%R134a, 25% $Al_2O_3$)**

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## I. INTRODUCTION

Refrigeration may be defined as the process of achieving and maintaining a temperature below that of the surroundings, the aim being to cool some product or space to the required temperature. One of the most important applications of refrigeration has been the preservation of perishable food products by storing them at low temperatures. Refrigeration systems are also used extensively for providing thermal comfort to human beings by means of air conditioning.

Most of the domestic refrigerators manufactured uses hermetically sealed compressors with R134a as refrigerant. In India the use of CFC refrigerants in new systems was stopped since the year 2002. The rapid industrialization has led to unprecedented growth, development and technological advancement across the globe. Today global warming and ozone layer depletion on the one hand and spiraling oil prices on the other hand have become main challenges. Excessive use of fossil fuels is leading to their sharp diminution and nuclear energy is not out of harm's way. In the face of imminent energy resource crunch there is need for developing thermal systems which are energy efficient. Thermal systems like refrigerators and air conditioners consume large amount of electric power. It is essential to developing energy efficient refrigeration and air conditioning systems with nature friendly refrigerants. By addition of nanoparticles to the refrigerant results in improvements in the thermo physical properties and heat transfer characteristics of the refrigerant, thereby improving the performance of the refrigeration system. In a vapour compression refrigeration system the nanoparticles can be added to the lubricant.

### Problem Statement

Evaporating heat transfer is very important in the refrigeration and air-conditioning systems. HFC 134a is the mostly widely used alternative refrigerant in refrigeration equipment such as domestic refrigerators and air conditioners. Though the global warming up potential of HFC134a is relatively high, it is affirmed that it is a long term alternative refrigerants in lots of countries. By addition of nanoparticles to the refrigerant results in improvements in the thermo physical properties and heat transfer characteristics of the refrigerant, thereby improving the performance of the refrigeration system. In this experiments the effect of using $Al_2O_3$ -R134a in the vapour compression system on the evaporating heat transfer coefficient will be investigated. An experimental apparatus will be build according to the national standards of India. Heat transfer coefficients will be evaluated using nano $Al_2O_3$ concentrations ranged from 0.05 to 1% and particle size from 10 to 70 nm.

## II. LITERATURE SURVEY

 

A large number of researchers have carried out research in the field of vapor absorption refrigeration using different working pairs and the most common working pairs are LiBr-H2O and NH3-H2O. Alizadeh et al [1] carried out theoretical studyon design and optimization of water lithium bromide refrigeration cycle. They concluded that for a given refrigerating capacity higher generator temperature causes high cooling ratio with smaller heat exchange surface and low cost. There is a limiting factor for water lithium bromide cycles because of the problem of crystallization.

Anand and Kumar [2] carried out availability analysis and calculation of irreversibility in system components of single and double effect series flow water lithium bromide absorption 19 systems. The assumed parameters for computation of results were condenser and absorber temperature equal to 87.8 oC and 140.6o C for single effect and double effect systems respectively.

Tyagi [3] carried out the detailed study on aqua-ammonia VAR system and plotted the coefficient of performance, mass flow rates as a function of operating parameters i.e. absorber, evaporator and generator temperatures. He showed that COP and work done are the function of evaporator, absorber, and condenser and generator temperature and also depends on the properties of binary solution.

Ercan and Gogus [4] showed the irreversibility's in components of aqua-ammonia absorption refrigeration system by second law analysis. They calculated the dimensionless energy loss of each component, energetic coefficient of performance, coefficient of performance and circulation ratio for different generator, absorber evaporator and condenser temperature. They concluded that aqua-ammonia system needs a rectifier for high ammonia concentrations but it will lead to additional energy loss in the system. They observed the highest energy loss in evaporator followed by absorber. I was also concluded that the dimensional less total energy loss depends on generator temperature.

Oh et al [5] investigated a gas fired, air cooled LiBr/H20 double effect parallel flow type absorption heat pump of 2TR being used as an air conditioner. They investigated the performance of the absorption heat pump in the cooling mode through cycle simulation. They obtained the system characteristics depending on the inlet temperature of air to the absorber, the working solution concentration, the solution distribution ratio of the mass of the solution in to the first generator to he total mass of 20 the solution from the absorber, and the leaving temperature differences of the heat exchanging components. They concluded that there exists a critical value of the solution distribution ratio that maximizes the cooling performance of the system.

Aphornratana and Eames [6] investigated single effect water lithium bromide system using exergy analysis approach. It was shown that the irreversibility in generator was highest followed by absorber and evaporator.

Bell et al [7] developed a LiBr-H20 experimental absorption cooling system driven by heat generated by solar energy. The components of the system are housed in evacuated glass cylinders to observe all the processes. They determined the Thermo dynamic performance of the system by applying mass and energy balance for all the components. Their work was based on the assumption that the working fluids are in equilibrium and the temperature of the working fluid leaving the generator and absorber is equal to the temperature of generator and absorber respectively. They concluded that the COP of the system depends on generator temperature and there is optimum value of generator temperature at which COPis maximum. They also concluded that by operating the system at low condenser and absorber temperatures a satisfactory COP is obtained at a generator temp.as low as 68o

C. Horuz [8] explained the fundamental vapour absorption refrigeration system and carried out comparative study of such system based on ammonia-water and water lithium bromide working pairs. The comparison of two systems is presented in respect of COP, cooling capacity and maximum and minimum pressures. He concluded that VAR system based on water-lithium bromide is better than ammonia water. However, problem of crystallization lies with water-lithium bromide system.

## III. EXPERIMENTAL SETUP

Fig 1. Experimental setup

## IV. CAPILLARY TUBE OPTIMIZATION

Instead of an orifice, a small diameter tube can be used for the expansion of the flowing fluid. This small diameter tube is known as the capillary tube expansion device and it produces the same effect as produced by the orifice. The term „capillary tube" means „hair-like". It is so called because of its very small bore diameter. The inside diameter of the capillary tube used for the purpose of refrigeration ranges from about 0.5 mm to 2.30 mm. longer the capillary tube and/or smaller the inside diameter of the capillary tube, greater is the pressure drop it can create in the refrigerant flow. In other words, greater will be the pressure difference needed between the high pressure side and the low pressure side to establish a given flow rate of the refrigerant.

Various literature sources are focused towards finding out the influence of the geometrical configurations of a capillary tube on the performance of the refrigeration system. The accurate size of the capillary tube and its configuration can be predicted with the help of the calculations for the refrigeration effect, coefficient of performance (cop) of the system and mass flow rate of the system. The effects of

 

different geometries of capillary tubes have been studied by many researchers. Since the capillary tube can be straight, helical coiled and also serpentine coiled and all three configurations have their own distinct effect on the system performance, thus the literature review here is focused to give a brief introduction of the effects of the various configurations of the capillary tube on the system performance.

Hirendra Kumar Paliwal and Keshav Kant (2006) developed a flow model for designing and studying the performance of helical coiled capillary tubes and to mathematically simulate a situation closer to one existing in real practice. Homogeneous flow of two phase fluid was assumed through the adiabatic capillary tube. The model included the second law restrictions. The effect of the variation of different parameters like condenser and evaporator pressures, refrigerant flow rate, degree of sub cooling, tube diameter, internal roughness of the tube, pitch and the diameter of the helix and the length of the capillary tube were included in the model. Theoretically predicted lengths of helical coiled capillary tube for R-134a are compared with the length of the capillary tube actually required under similar experimental conditions and majority of predictions were found to be within around 10% of the experimental value.

M.Y.Taib et al. (2010) studied the performance of a domestic refrigerator and developed a test rig from refrigerator model NRB33TA. The main objective of the performance analysis was to obtain the performance of the system in terms of refrigeration capacity, coefficient of performance (cop), and compressor work by determining three important parameters which are temperature, pressure and refrigerant flow rate. The analysis of the collected data gave the cop of the system as 2.75 while the refrigeration capacity was ranging from 150 watt to 205 watt.

Sanggoon Park et al. (2008) simulated the effects of a non adiabatic capillary tube on refrigeration cycle. The simulation focused on the effect of capillary tube- suction line heat exchangers (CT-SLHX). The simulation of steady state were based on the fundamental conservation equations of mass and energy and these equations were solved simultaneously through iterative process. The non – adiabatic capillary tube model was based on homogenous two phase model. The length and location of soldering region between capillary tube and suction line tube were changed and performance of the refrigeration cycle was compared in terms of condenser pressure, evaporator pressure, refrigeration effect, compressor work and cop. The simulation results showed that both the location and length of the heat exchange section influence the cop of the system.

M.M.Tayde et al. (2013) compared the performance of a miniature vapour compression refrigeration system with four different refrigerants namely NH3, R12, R22 and R134a. The study revealed that NH3 gives maximum value of cop for the system. Next highest cop for the system came out for R12 and then for R134a. Refrigerant R22 gave the least cop for the miniature system.

Nishant P. Tekade et al. (2012) reviewed the investigation about the coiling effect of spiral capillary tubes on the refrigerant mass flow rate for the same cooling load. The work also reviewed the effects of changes in the parameters such as capillary tube dimension i.e. capillary tube diameter, capillary tube length, coil pitch and inlet conditions of the refrigerant to the capillary tube i.e. degree of subcooling and inlet pressure of the refrigerant charge.

Ankush Sharma and Jagdev Singh (2013) experimentally investigated about the effects simple and twisted spirally coiled adiabatic capillary tubes on the refrigerant flow rate. Several capillary tubes with different bore diameters, lengths and pitches were taken as test sections. LPG was used as an alternative for R134a. mass flow rates for different capillary tubes were measured for different degrees of subcooling with constant inlet pressure of the capillary tube. Experiments were conducted on straight capillary tubes as well so as to facilitate proper comparison. The test results showed that mass flow rate is greater in straight capillary tube and least in twisted spirally coiled capillary tube.

Akash Deep Singh (2009) developed a mathematical model for adiabatic capillary tube. The mathematical model was developed using equations of conservation of mass, momentum and energy and was used for predicting the length of adiabatic capillary tube. Moody (1944) correlation was used to calculate the friction factor. McAdams et al. (1942) viscosity correlation was used to evaluate the two phase viscosity of the refrigerant. Input parameters were taken from the data of Mendoca et al. (1998).A geometric model was developed in Pro-E and the mesh was created in Ansys ICEM CFD and analysis is carried out in Ansys CFX which has three modules CFX-Pre, Solver and CFX-Post.

## V. RESULT

| Refrig - Errant | Charge in grams | Lcap In meters | Calorimeter Temperature in $^0$C | Power in watts | Refrigeration effect in watts | COP |
|---|---|---|---|---|---|---|
| [illegible] | 110 | 3.3 | 2 | 192 | 234 | 1.22 |
| | | | 5 | 196 | 276 | 1.41 |
| | | | 8 | 199 | 336 | 1.69 |
| | | 5.4 | 2 | 170 | 224 | 1.32 |
| | | | 5 | 172 | 263 | 1.53 |
| | | | 8 | 175 | 318 | 1.82 |
| | | 6.3 | 2 | 177 | 214 | 1.21 |
| | | | 5 | 180 | 252 | 1.4 |
| | | | 8 | 184 | 306 | 1.66 |
| | 120 | 3.3 | 2 | 185 | 263 | 1.42 |
| | | | 5 | 187 | 295 | 1.58 |
| | | | 8 | 190 | 361 | 1.9 |
| | | 5.4 | 2 | 163 | 241 | 1.48 |
| | | | 5 | 166 | 279 | 1.68 |
| | | | 8 | 168 | 334 | 1.99 |
| | | 6.3 | 2 | 166 | 229 | 1.38 |
| | | | 5 | 170 | 262 | 1.54 |
| | | | 8 | 173 | 321 | 1.86 |
| | 130 | 3.3 | 2 | 200 | 250 | 1.25 |
| | | | 5 | 203 | 294 | 1.45 |
| | | | 8 | 207 | 353 | 1.71 |
| | | 5.4 | 2 | 173 | 227 | 1.31 |
| | | | 5 | 176 | 269 | 1.53 |
| | | | 8 | 179 | 328 | 1.83 |
| | | 6.3 | 2 | 179 | 222 | 1.24 |
| | | | 5 | 182 | 260 | 1.43 |
| | | | 8 | 186 | 313 | 1.68 |

Table 1. Experimental results of mixture-3

| S.No. | Description | R134a | Mixture-3 |
|---|---|---|---|
| 1 | Power-Theoretical | 155 | 162 |
| 2 | Power-Experimental | 162 | 168 |
| 3 | COP- Theoretical | 1.89 | 2.11 |
| 4 | COP- Experimental | 1.81 | 1.99 |
| 5 | mass flow rate(kg/sec) | 0.00246 | 0.00161 |

Table 2. Performance comparison of R134a with mixture-3 at 32$^0$C ambient temperature

 

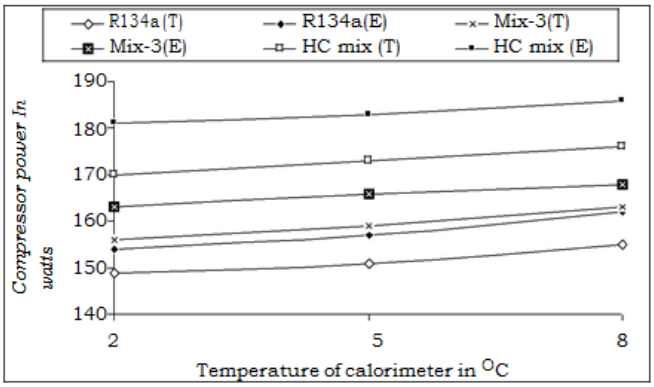


Fig 2. Comparison of predicted and experimental values of compressor power at different calorimeter temperatures for R134a, Al2O3and mixture-3

## VI. CONCLUSION

**1. This is the Optimize Mixture-3(75%R134a, 25% $Al_2O_3$)**

The cop for the optimize For mixture-3 the theoretical values are deviating from the experimental values by 7.6% to 8.67%, whereas it is 5.9% to 6.2% for R134a and for $Al_2O_3$deviates 8.1% to 9.0%. This proves the validity of the present model.

**2. Cost Analysis of the Proposed Mixture-3**

The cost of the proposed alternative mixture-3 is 3.43% lower than the HC mix, which is presently used as an alternative to R134a.

The cost of the refrigerant. Cost comparison of the mixture-3 with the base refrigerants

| Refrigerant | Charge Quantity in grams | Cost in Rupees |
|---|---|---|
| R134a | 240 | 192 |
| HC mix | 104 | 233 |
| Mixture-3 | 120 | 225 |

**3)** Table 3 Optimization of capillary length and refrigerant charge for the selected refrigerants

| Mixture | Optimum capillary length in meters | Optimum refrigerant charge in grams |
|---|---|---|
| Mixture-1 | 6.3 | 106 |
| Mixture-2 | 5.4 | 113 |
| **Mixture-3** | **5.4** | **120** |
| Mixture-4 | 5.4 | 129 |
| Mixture-5 | 5.4 | 139 |
| R134a | 3.3 | 240 |

Mixture 3 is having optimize capillary length of 5.4 m

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