FLOW ANALYSIS OF ISOBUTANE (R-600A) INSIDE AN ADIABATIC CAPILLARY TUBE

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Abstract

Capillary tubes are simple narrow tubes but the phase change which occurs inside the capillary tubes is complex to analyze. In the present investigation, an attempt is made to analyze the flow of Isobutane (R-600a) inside the coiled capillary tubes for different load conditions by Homogeneous Equilibrium Model. The length and diameter of the capillary tube not only depend on the pressure and temperature of the condenser and evaporator but also on the cooling load. The present paper investigates the change in dimensions of the coil capillary tube with respect to the change in cooling load on the system for the constant condenser and evaporator conditions. ANSYS CFX (Central Florida Expressway) software is used to study the flow characteristics of the refrigerant. Appropriate helical coil is selected for this analysis.

KEYWORDS: Homogeneous equilibrium model, Capillary tube, Adiabatic flow, R600a.

1. INTRODUCTION

Capillary tubes are used as expansion device in vapor compression refrigeration cycle to bring down the pressure and temperature of the refrigerants so that the refrigerant entering the evaporator coil has minimum temperature. Experimental and numerical studies show that the flow within the capillary tubes can be divided into two regions, a single phase region, in which only liquid flows and a two-phase flow length in which liquid and vapor flow simultaneously (Fig 1). The transition point is called the bubble or the flash point.

Fig.1: Adiabatic Capillary tube

Sub cooled Region: In adiabatic capillary tubes heat transfer from the tube to the surrounding is assumed to be zero. The refrigerant enters inside the capillary tube as high pressure and high temperature liquid. The pressure of the refrigerant starts to decrease gradually as soon as it enters the capillary due to friction offered by the inner walls (sub-cooled region in Fig.1). The pressure of the liquid refrigerent in this region is above the saturation pressure corresponding to the inlet temp of the refrigerant. The length of sub-cooled region depends on the degree of the sub-cooling of the refrigerant at the inlet as well as the sand grain roughness of the inner wall.
Metastable Region: The tendency of a fluid to stay in its liquid state even if the fluid pressure falls below its saturation pressure is termed as metastable state. This is a non-equilibrium state of the fluid in which the liquid exists in a superheated state (Fig.1). Thus, the phenomenon of metastability exists whenever a fluid undergoes a transition from liquid phase to vapor phase. The Pressure of the refrigerant falls below the saturation pressure of the refrigerant corresponding to the temperature of the refrigerant at the inlet. Superheated liquid exists due to delay in vaporization even though the saturation has reached.

Two-Phase Region: This is the region where the flashing of the refrigerant takes place. The single phase liquid refrigerant comes into two phases due to which the overall density of the refrigerant decreases. The decrease in density is followed by the increase in kinetic energy of the refrigerant (Fig. 2).

Cooper et al. (1957), reported experimental and analytical design of capillary tube based on Fanno flow (adiabatic flow through constant cross section) of refrigerant with choked flow condition [1]. Prajapati et al., reported numerical simulation of refrigerant flow through adiabatic capillary tube using R134a (Tetrafluoroethane) as a working fluid [2]. They used ANSYS FLUENT (Version 12) based on finite volume method using k-ω as a turbulent model. Ingle et al., reported on a homogeneous equilibrium approach to model the flashing phenomenon along with the cavitation model (based on mechanism of transfer of mass through capillary tubes) [3]. They analyzed R12 (Dichlorodifluoromethane) and R134a. Bansal and Rupasinghe [4] introduced CAPIL model for homogeneous two-phase flow to be solved numerically through adiabatic capillary tube. Numerical investigation of adiabatic spiral capillary tube and its geometrical effect on mass flow rate of refrigerant R-134a has been reported by Khan et al. [5]. Li et al. [6] experimentally investigated the metastable phenomenon and under-pressure of vaporization in an adiabatic capillary tube. Chingulpitak and Wongwises compared pressure drop and mass flow rate obtained numerically for straight and helical capillary tube [7]. Mikol et al. studied single phase and two-phase flow in narrow tube under adiabatic condition [8]. Shiva et al., did the numerical simulation of R12 using ANSYS CFX for straight capillary tube for different condenser pressure and sand grain roughness [9]. Park et al. (2007), performed a numerical simulation as well as experimental investigation for three type refrigerants flowing through straight as well as coiled capillary tube [10]. They reported a mass flux correlation based on the equivalent tube length by using the dimensionless parameters. They also found, the mass flux of the coiled capillary tubes would fall by nearly 5-16% in comparison with the straight capillary tubes under similar test conditions.

The effects of sub-cooling on R22 (Chlorodifluoromethane) flowing through the coiled capillary tube was investigated by Valladares (2007) [11]. They reported that the mass flux of through capillary
increases with increases in the degree of sub cooling as well as condensing pressure. A separated two-phase flow model was employed for refrigerant flow simulation through coiled tube. Valladares also considered metastability while reporting numerical simulation results for refrigerant flowing through adiabatic and diabatic straight capillary tube. Zhou et al. (2010) reported on numerical simulation and experimental validation for R-22 refrigerant flowing through coiled capillary tubes [12]. They used a homogenous two-phase flow model while building mathematical model and three different friction factor correlations to retrofit R290 (Propane) in place of R22. They found cooling capacity and input power decrease when R290 is used. Wong and Ooi made a numerical investigation to compare the performance of two refrigerants R12 and R134a [13]. This numerical model is used for comparison of flow characteristic of various refrigerants namely extended CAPIL model, which is developed by Bansal and Rupasinghe (1998) [4]. Marcy developed an analytical method to calculate capillary tube length and mass flow rate using graphical integration technique [14]. Sahoo and Das proposed a theoretical way of calculating adiabatic capillary tube length for 0.1T refrigerator using R600a [15].

Among various hydrocarbons that have been proposed as a replacement to chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) isobutane (R600a) is a promising one. In addition to their environmental benefits like zero ozone depletion potential (ODP), hydrocarbons are cheaper and easily available refrigerants suitable for regular refrigerators and air conditioners [12]. Therefore the present work is focused on simulation of R600a considering all important parameters like wall roughness, metastability, turbulence and flashing.

2. CFD SIMULATION

There are various physical models already developed to predict the length and diameter of the capillary tubes such as Drift Phase model (DPM), Homogeneous Equilibrium models (HEM), and Non equilibrium models. In the drift phase model, the conservation equation is formulated by considering the entire mixture and the model is formulated in terms of four field separations: three for the mixture (continuity, momentum and energy) plus the drift velocity for one of the phases. HEM assumes thermal equilibrium with the phases. In the homogeneous flow model, the two-phase mixture can be simulated as a single-phase fluid possessing mean fluid properties. This model is widely used in analyzing the flow in capillary tubes. Here, the mixture is homogeneous in phase composition; vapor and liquid velocities are equal. Two-phase mixture is in thermodynamic equilibrium and vapor and liquid exists at same temperature. NEM (Non-Equilibrium model) assumes unequal phase velocities but considers non-homogeneous flow characteristics.

Here the numerical investigation is done first on R134a using ANSYS CFX with the help of HEM and the results are validated comparing with other reported data in the literature [2, 16]. Then numerical investigation of the isobutane (R600a) inside capillary tube is done using same method for different load conditions. Geometric model of the capillary tube was prepared in Autodesk inventor and the file was converted suitably for numerical simulation. A suitable timescale factor was selected along with the convergence criteria of 1e-06.

2.1 Turbulence model

Standard k-ω model, a two-transport-equation model [17], is used for turbulent flow condition. It includes two transport equations to represent the turbulent properties of the flow and to account for convection and diffusion of turbulent energy. The first transported variable is turbulent kinetic energy, k (determines the energy in the turbulence; Equation 1). The second transported variable is the specific dissipation, ω (the scale of the turbulence; Equation 2). This model accommodates a more accurate near wall treatment with an automatic switch from a wall function to a low Reynolds number formulation based on grid spacing. It might yield superior performance for wall-bounded and low Reynolds number flows to predicting transition. Two transport equations are as follows:

\[
\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left( \Gamma \frac{\partial k}{\partial x_j} \right) + G_k - Y_k + S_k
\]  \hspace{1cm} \text{(Equation 1)}

\[
\frac{\partial (\rho \omega)}{\partial t} + \frac{\partial (\rho \omega u_i)}{\partial x_i} = \frac{\partial}{\partial x_j} \left( \Gamma \frac{\partial \omega}{\partial x_j} \right) + G_\omega - Y_\omega + S_\omega
\]  \hspace{1cm} \text{(Equation 2)}
In these equations, $G_k$ represents the generation of turbulence kinetic energy due to mean velocity gradients. $G_\omega$ represents the generation of $\omega$. $\Gamma_k$ and $\Gamma_\omega$ represent the effective diffusivity of $k$ and $\omega$, respectively. $Y_k$ and $Y_\omega$ represent the dissipation of $k$ and $\omega$ due to turbulence.

2.2 Boundary Conditions

Boundary conditions are of two type here; 1) Inlet/outlet conditions (Pressure values) and 2) Wall condition (Sand grain roughness) [18, 19]. All thermo-physical properties of the refrigerants have been taken from literature. No slip conditions for walls with sand grain roughness 0.00198 mm (for validation) have been chosen. Sand grain roughness is the degree of smoothness of any surface. Sand grain roughness affects the flashing point of the refrigerant inside the capillary tube because it depicts the friction factor [9].

3. RESULTS AND DISCUSSION

3.1 Tetrafluoroethane or R134a

R134a is the most popular refrigerant for refrigerator now hence simulation data is widely available in literature [2, 16]. Present study hence focuses on validating R134a results first. The simulation results are as represented in Fig. 3 to Fig. 6. Prajapati et al., used ANSYS Fluent with user defined function to induce mass transfer from liquid phase to vapor phase to study the flow of R134a. The length of the capillary tube is 1.5 m and diameter 0.66 mm. The flashing point for their work is reported ∼1.1 m from the inlet in the axial direction at ~24°C temperature and 8.37 bar inlet pressure.
From the results we can see that the pressure distribution and total temperature distribution are almost same as compared to the pressure and total temp distribution of Prajapati et al. [2]. (Fig. 6 and Fig. 9). The flashing point noted in present simulation is 1.104 m from the inlet of the capillary but the flashing point noted in Prajapati et al. simulation is 1.1 m from the inlet. The difference in flashing point is negligible. There is only marginal difference in mass flow rate in present work (3.685608 kg/h) and result showed by Prajapati et al. (3.726 kg/h). This marginal difference might be due to the different approach to model the two-phase problem.

Propainop and Suen reported that R600a is best replacement for R134a [20]. Therefore here a helical coil (saves space in a practical refrigerator) is selected with tube dia of 1.25 mm (commercially available). If R134a is used in 0.1 TR refrigerator with condenser temperature 307.5 K and evaporator temperature of 268 K then total length required is 4.85 m (from simulation). Refrigeration effect (RE) can be calculated as product of mass flow rate, x₁ and h₉₆, where x₁ is liquid mass fraction of the refrigerant at the inlet of the evaporator and h₉₆ is latent heat of vaporization of refrigerant liquid at evaporator pressure [21, 22]. From Simulation we obtain a mass flow rate of 0.00261733 kg/s (liq. mass fraction =0.7294) for R134a and latent heat corresponding to evaporator pressure of 2.401 bar is 202.45 kJ/kg [23]. Thus RE =0.11TR matches cooling load of 0.1 TR.

### 3.2 Isobutane or R600a

Now the same homogeneous equilibrium model is used to simulate R600a for different load conditions. Cooling load condition (Ton of Refrigeration, TR) is also an important factor to decide evaporator temperature. The mass flow rate of the capillary tube should be such that it shouldn’t overflow or underflow the evaporator which means it should be proportional with respect to the cooling load condition.

#### Table 1: Dimensions and boundary conditions (Isobutane)

| Load   | Coil dia (mm) | Tube dia (mm) | Pitch (mm) | No of Turns (N) | Total Length (m) | Condenser pressure (Pa) | Condenser Temp (K) | Mass flow rate (kg/h) |
|--------|---------------|---------------|------------|----------------|------------------|-------------------------|---------------------|-----------------------|
| 0.1 TR | 60            | 1.25          | 4          | 14             | 2.642            | 501830                  | 307.5               | 5.0964                |
| 0.2 TR | 80            | 1.6           | 4          | 15             | 3.8455           | 501830                  | 307.5               | 9.108                 |
| 0.3 TR | 102           | 2.2           | 4          | 18             | 5.8924           | 501830                  | 307.5               | 15.0192               |

Wall roughness= 0.0032 mm
Degree of Sub-cooling= 3.5 K
Temperature and pressure variation along axial direction for 0.1TR, 0.2TR and 0.3TR are shown in Fig. 7, Fig 9 and Fig 11 respectively. In Fig 8, Fig 10 and Fig 12 vapor mass fraction contour is shown for 0.1TR, 0.2TR and 0.3TR systems respectively. In all pressure and temperature plots its clearly visible that the temperature remain stable till flashing and later there is a sharp decline in two phase region. Compared to pressure, temperature drops more sharply therefore the purpose of attaining low temperature is easily achievable. In mass fraction contor towards the end minimum density is expected (two phase region). As visible in the Fig. 7, for a load of 0.1 TR flashing happens at 1.323 m and total capillary tube length needed is 2.642 m (Table 1). Therefore using R600a saves us the capillary tube length when compared to R134a. Theoretical length of capillary tube for a 0.1 TR refrigerator using R600a as reported by Sahoo and Das (2014) is 2.29 m (1 mm dia) when condenser temperature is 313K and evaporator temperature is 253 K with 10 K of subcooling [15]. However in present work boundary condition differs. The mass flow rate for R600a is ~45% less than the mass flow rate for R134a, i.e; R600a will be required 45% less charge as compared to R134a for same capacity.

Using similar procedure capillary tube required for 0.2TR and 0.3TR refrigerator is also simulated and results are shown in Fig. 9-Fig.12. For similar condenser temperature and condenser pressure capillary tube length required is 3.8455 m and 5.8924 m for 0.2TR and 0.3TR refrigerator respectively.

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**Fig. 7** Temp., Pressure variation along the axial length  
**Fig. 8** Vapor mass fraction contour

**Fig. 9** Temp., Pressure variation along the axial direction  
**Fig. 10** Vapor mass fraction contour
**Table 2: Calculation of actual Refrigeration Effect**

| For Isobutane (R 600a) | 0.1 TR | 0.2 TR | 0.3 TR |
|-------------------------|--------|--------|--------|
| Pressure at the outlet  | 131560Pa | 131299Pa | 131397 Pa |
| Temperature at the outlet| 268.254 K | 268.211 K | 268.231 K |
| Liquid mass fraction at the outlet | 0.7572 | 0.7570 | 0.7571 |
| Latent heat of vaporization \(hfg\) [23] | 358.12 kJ/kg | 358.12 kJ/kg | 358.12kJ/kg |
| Refrigeration Effect | \(0.001415 \times 0.7572 \times 358.12\) \(= 0.1096\) TR | \(0.002530 \times 0.7570 \times 358.12\) \(= 0.196\) TR | \(0.00417 \times 0.7571 \times 358.12\) \(= 0.32\) TR |

To calculate the RE, mass flow rate is obtained from simulation and latent heat for given evaporator pressure is obtained from literature as represented in Table 2. At same condenser and evaporator conditions refrigeration effect (cooling load) is directly proportional to the mass flow rate. Therefore the mass flow rate through the capillary tube changes with respect to the change in load condition. In all the present numerical simulation evaporator and condenser conditions are kept constant and the parameters will vary with the load on the evaporator side. Presumably load on the domestic refrigerator varies from 0.1 TR to 0.3 TR for different capacities of the refrigerator. R600a can hence be used in domestic refrigeration system without much modification.

### 4. CONCLUSION

The flow pattern of two phase refrigerants in adiabatic capillary tube has been investigated numerically using ANSYS CFX for different cooling load capacities. First the simulation is done on R134a using homogeneous equilibrium model to validate the method of present work. When compared with R134a, R600a will require lesser length capillary tube. The numerical investigation showed that refrigerant charge required will be higher for higher tonnage capacity of refrigerator. The mass flow rate through the capillary tube should always be proportional to the cooling load on the evaporator. The design of the capillary tube not only depends on the pressure and temperature of the refrigerant at the condenser and evaporator but also on the wall roughness, degree of sub-cooling etc.

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