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CFD Analysis of Nanorefrigerant through Adiabatic Capillary Tube of Vapour Compression Refrigeration System

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Abstract

Over time attempts have been made to understand the flow characteristics of refrigerants through capillary tubes as well as to seek more thermally efficient working fluids for refrigeration systems. This study investigated the flow of nanorefrigerants through adiabatic capillary tubes of vapour compression refrigeration systems; and afterwards creates numerical models that will account for solution of refrigerant side pressure drop and mass flow rate. Also in this study, a CFD flow analysis was carried out using a CFD simulation/solver such that the results of the simulations obtained were discussed so as to establish a distinction between the conventional and nano-refrigerants. Upon comparison of the CFD results of nanorefrigerants (CuR134a, CuR600a) and the conventional refrigerants (R134a, R600a), the conventional refrigerants were noticed to have more isothermal regions implying that heat was not being transferred quickly enough to raise the temperature of the adjoining region thus proving that the addition of nanoparticles improves the thermophysical properties of the base fluid. Also, based on the results of the study of the flow patterns of both working fluids, the density of pressure contours in the conventional refrigerants was far larger than that of the nanorefrigerant implying that more compressor work and ultimately greater power will be required. The findings from this study were validated with experimental results showing that a CFD analysis tool/method can be employed to understudy the phenomenal changes that take place in nano-refrigerant movement through capillary tubes without recourse to experimentation.

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1. Introduction

In today's world, energy crisis is a major issue and it is therefore necessary to manage and reduce the consumption of energy. Refrigeration plays a major role in energy consumption and therefore to manage its consumption, it is important to have efficient systems that operate as designed. To this end, it is necessary that some sort of alteration be made to the system or its working fluid. The life force of every refrigeration plant is the refrigerant as it is solely responsible for the heat exchange process. Several studies have been carried out to improve the properties of refrigerants with the aim of improving the performance of systems [1-7]. It has however, been halted by several factors. One being that the refrigerants with the best thermophysical properties happens to be halogenated hydro carbons which are detrimental to the environment [8-10]. Another being that the inorganic refrigerants which are environmentally safe have poorer thermophysical properties while ammonia causes corrosion to the pipelines of the refrigeration plant. Also conventional refrigerants consume a lot of electric power [11]. These issues have given rise to the idea of nanotechnology in refrigeration [1]. In addition to this, over the past decade, researches have focused on understanding and improving capillary tube flow and in recent times, increased efforts have been made to integrate previous experimental results into computer codes for tube simulation. With the increasing awareness of our environment and the gradual phasing out of conventional refrigerants harmful to the atmosphere, researchers are now interested in modelling the characteristics of flow through capillary tubes using alternative refrigerants. Therefore, an ideal model that will be valid for alternative refrigerants (like nanorefrigerants) and provide accurate predictions with minimal laboratory experiments is inevitable [2]. This study aims to develop such a model with specific focus on the need to understand nanorefrigerant flow through small diameter (capillary) channel.

2. Model Geometry Development

Determining the model geometry is the first step to modelling and solving the multi-phase flow problems. The geometry of the model was characterized by a narrow pipe of uniform cross section as shown in Fig. 1.

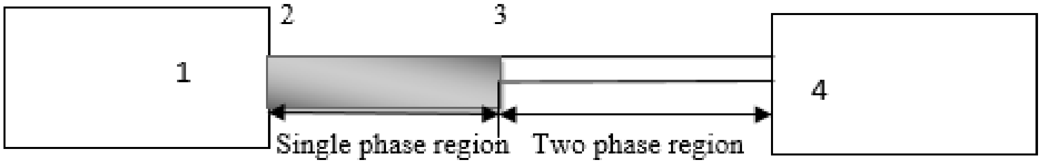

Fig. 1: Schematic diagram of capillary tube between condenser and evaporator [12]

The modelling of the flow characteristics in the straight capillary tube of a refrigeration system was carried out by applying the laws of conservation of mass, energy and momentum on the required geometry of the tube. However, in order to simplify flow conditions without compromising the flow characteristics, some assumptions were made for the adiabatic region. These include: that the volume fractions of all phases sum to unity, that the heat exchange with the ambient is negligible, and that there is a steady state nanorefrigerant one dimensional flow. Other assumptions were that there is homogenous equilibrium two phase flow, critical conditions reached when the Mach number of the homogenous mixture at the exit section equals 1.0, negligible effect of nanoparticle size on the flow characteristics of the nanorefrigerant, and also that there is no planar difference between condenser and evaporator. Worthy of note is the fact that the homogenous two phase model assumes a thermodynamic and hydrodynamic equilibrium between the two phases such that they both have equal temperatures and velocities and that there is no delay in vaporization.

Conservation of Mass

The application of continuity can be expressed as [3]:

$$\dot{m} = \rho A U$$

Or

$$G = \frac{\dot{m}}{A} = \rho U$$

The conservation of mass equation is given as $\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} = 0$. Since it is a steady and incompressible flow, therefore

$$\frac{\partial \rho}{\partial t} = 0$$

Conservation of Momentum

$$\nabla.\left(\rho_{nf}.U.m\right) = -\nabla P + \nabla.\left(\mu_{nf}.\nabla U\right)$$

Considering a fluid element in the tube, due to shear force inside the capillary tube and pressure difference on opposite ends of the element being considered, this equation is expressed as:

$$P.A - (P + dP).A - \tau_w(\pi d)dx = mdU$$

where τ_w is shear stress at tube wall and dx is small change in length of the elemental fluid being considered.

Conservation of Energy

On applying the Steady Flow Energy Equation for adiabatic flow with no external work and ignoring any variation in elevation, thus:

$$h + \frac{U^2}{2} = constant$$

2.1. Modelling the flow at the single phase region

This region is defined by points 2-3 in Fig. 1. It is characteristically liquid and defined as the region between the inlet of the capillary tube to the point in space where the pressure decreases such that it becomes saturated and vapour begins to emerge.

Applying Bernoulli's principle of incompressible flow:

$$\frac{P_2}{\rho_2 g} + \frac{{U_2}^2}{2g} + Z_2 = \frac{P_3}{\rho_3 g} + \frac{{U_3}^2}{2g} + Z_3 + h_T \tag{1}$$

where h_T = total head loss = major head loss + minor head loss and given as:

$$h_T = f_{sp}\frac{L_{sp}U^2}{Dg} + \frac{kU^2}{2g} \tag{2}$$

where f_{sp} = friction factor of single phase, L_{sp} = length of single phase region, D = internal diameter of capillary tube, k = coefficient of entrance loss

Applying the continuity equation and assume that the tube is perfectly horizontal, and also that the fluid is incompressible:

$$\rho\left[\frac{P_2 - P_3}{\rho^2 U^2}\right] = \frac{f_{sp}L_{sp}}{D} + \frac{k}{2} \tag{3}$$

Let the mass flux be given as ρV, and the friction factor f_{sp} computed from [4]:

$$\frac{1}{\sqrt{f_{sp}}} = 1.14 - 2\,l\,n\left[\frac{e}{D} + \frac{9.3}{Re\,\sqrt{f_{sp}}}\right]$$

where Re = Reynolds' number = $\frac{\rho U D}{\mu}$, e = surface roughness of tube wall and μ = refrigerant viscosity

$$L_{sp} = \frac{D}{f_{sp}}\left[\frac{2\rho}{G^2}(P_2 - P_3) - k\right] \qquad (4)$$

At the end of the single phase region, flashing begins, so that pressure increase favours the vapourisation of the liquid refrigerant. Thus, Eq. (3) becomes:

$$\therefore\ P_2 - P_3 = \rho U^2\left(\frac{f_{sp} l_{sp}}{D} + \frac{k}{2}\right)$$

2.2. Modelling the flow at the two phase region

The basic equations governing this region can be derived from the continuity and momentum equations (Eqs. 5 and 6):

$$\nabla.\left(\rho_{nf}.U\right) = 0 \qquad (5)$$

$$\nabla.\left(\rho_{nf}.U.m\right) = -\nabla P + \nabla.\left(\mu_{nf}.\nabla U\right) \qquad (6)$$

Where the density of the nanofluid, ρ_{nf}, can be obtained from the results of Arslan's study:

$$\rho_{nf} = (1 - \emptyset)\rho_f + \emptyset\rho_f$$

Applying continuity equation between 3 and 4 and the Steady Flow Energy Equation for adiabatic flow with no external work, and ignoring any elevation difference, the vapour quality, x can be expressed as:

$$x = \frac{-h_{fg} - G^2 v_f v_g + \sqrt{\left(G^2 v_f v_g + h_{fg}\right)^2 - 2G^2 v_{fg}^2\left(\frac{G^2 v_f^2}{2} - h_3 - \frac{U_{nf}^2}{2} + h_f\right)}}{G^2 v_{fg}^2} \quad \text{(Lin et al 1999)} \qquad (7)$$

Where the U_{nf} is the velocity model of the nanofluids and given as:

$$U_{nf} = \frac{1}{d_p}\sqrt{\frac{18 K_b T}{\pi \rho_p d_p}} \qquad (8)$$

Applying Hagen Poiseulle's law of flow of viscous fluid in circular pipes, a small concentric circle (of fluid element) of radius r and length dx can be considered as a free body. Therefore, the forces acting on the fluid element will be given by Fig. 2 as:

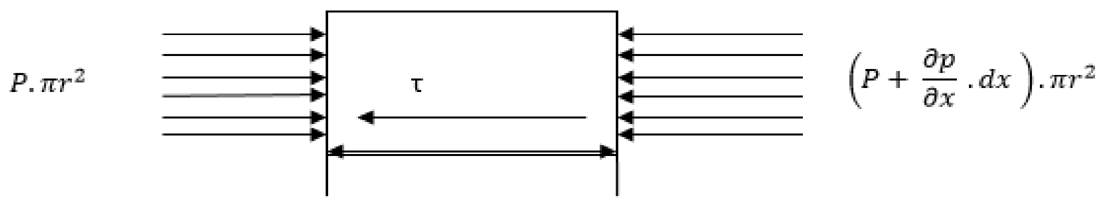

Fig. 2: Action of flow Stresses on fluid element

Shear force, $\tau \times 2\pi r \times dx$ on the surface of the fluid element
where τ is the shear stress

Balancing the forces on the fluid element becomes:

$$\tau = -\frac{\partial p}{\partial x} \cdot \frac{r}{2} \quad (9)$$

Consideration at the wall of the tube where pressure is maximum, and applying two phase flow friction factor:

$$\tau_w = \frac{f_{tp} \rho U^2}{8} \quad (10)$$

$$dL = \frac{-2D}{f_{tp}} \left[\frac{PdP}{G^2} + \frac{dU}{U} \right] \quad (11)$$

where f_{tp} = friction factor of the two phase region, and can be expressed thus:

$$f_{tp} = 8\left[\left(\frac{8}{Re_{tp}}\right)^{12} + \left(\frac{1}{(A+B)^{1.5}}\right)^{\frac{1}{12}}\right]$$

where $A = 2.4571 \, \mathrm{n}\left[\frac{1}{\left(\frac{7}{Re_{tp}}\right)^{0.9} + 0.27\left(\frac{e}{D}\right)}\right]^{16}$, $B = \left(\frac{37530}{Re_{tp}}\right)^{16}$, e/D = capillary tube's relative roughness, $Re_{tp} = \frac{Vd}{\mu_{tp} v_{tp}}$
where $\mu_{nf} = (1-\emptyset)^{2.5}\mu_f$ is the Brinkman viscosity model for nanofluids and ϕ = volume fraction of nanoparticles.

The length of the two phase region, can be evaluated from:

$$\therefore \; L_{tp} = \frac{2D}{f_{tp}}\left(\frac{1}{G^2}\int_{P_i}^{P_n} \rho dP + \int_{P_i}^{P_n} \frac{d\rho}{\rho}\right)$$

Therefore, the total optimum capillary tube length that will facilitate the refrigerant side pressure drop required to give optimum pumping power for each refrigeration system is evaluated from the addition of L_{sp} and L_{tp}.

The velocity distribution of the nanorefrigerant through the capillary tube can be describe using the Newton's law of viscosity, where r represents any point in the cross section of the tube (measured from the center of the tube) from which the value of the stress is given as:

$$\tau = \mu_{nf} \cdot \frac{dU}{dr} \quad (9)$$

Thus after mathematical evaluation, the velocity of the nanorefrigerant through the tube at any point r is:

$$U = -\frac{1}{4\mu_{nf}} \cdot \frac{\partial p}{\partial x}(R^2 - r^2) \quad (9a)$$

More so, Eq. 9a shows that, at the centre of the tube where r = 0, the velocity is maximum. Therefore, fouling at the tube's wall and the impact of particle size on the wall-surface (or relative) roughness becomes an issue of further research.

3. **Results and Discussion**

The dynamic equations developed were simulated using the computational fluid dynamics (CFD) approach and the results were compared with the experimental results reported by Attabo [5]. Further to this, CFD analysis of the thermodynamic and flow characteristics of R134a, R600a, 0.1%vol. copper nano-R134a and 0.1%vol. cupper nano-R600a through adiabatic capillary tube were carried out. Fig. 3 presents the results of comparisons between that obtained from the simulation and those obtained by Attabo [5]. It shows that the two results are appreciably close to with few exceptions. Moreover, the results of Fig. 3 were extended to include other refrigerant variants of bimetallic copper (Cu)-aluminum (Al) nanorefrigerant and monometallic Al-nanorefrigerant variants of R134a and R600a respectively. It was further employed to include the variants for R12 to test for wider suitability and application. The results showed that while the simulation predicted slightly higher values for the variants of R12 and 0. 1% (Cu-R12 & Al-R12), others had very close values. Also, Figs. 4 to 6 show the results of the CFD analysis of the flow of the refrigerants and nano-refrigerants through the capillary tube.

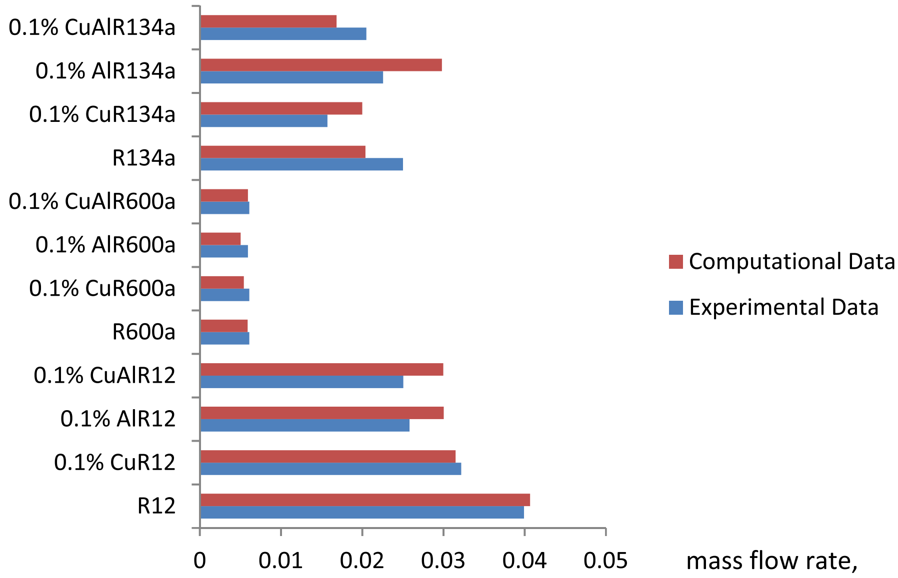

Fig. 3: Comparison of mass flow rate results of Attabo [5] and those of simulation.

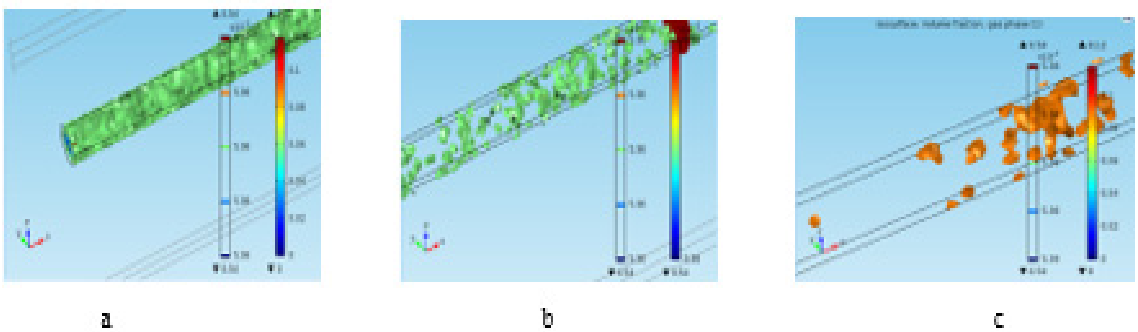

Fig. 4: Flow characteristics from tube inlet to outlet for all the refrigerant variants (a) into towards the midpoint of tube, (b) from the midpoint towards the outlet, (c) points very close to the outlet

Fig. 4 shows that at the beginning of the tube, the fluid is purely liquid and then changes phase as it flows across the length of the tube towards the outlet. As the fluids flows along the tube, flashing begins and then vapourises. Moreover, Fig. 4 demonstrates that nanorefrigerant and the conventional refrigerants have similar behaviours and functions in the same manner and therefore can be used as ready replacements without modification of the refrigeration cycle or system's parts. However, the results of Fig. 4 further demonstrate that there are some areas that need further studies. This include the characterisation of the flow boiling along the tube and the impact of particle size and nanorefrigerant viscosity on the flow characteristics and pressure drop.

Fig. 5 presents the isothermal analysis of the refrigerants movement through the capillary tube.

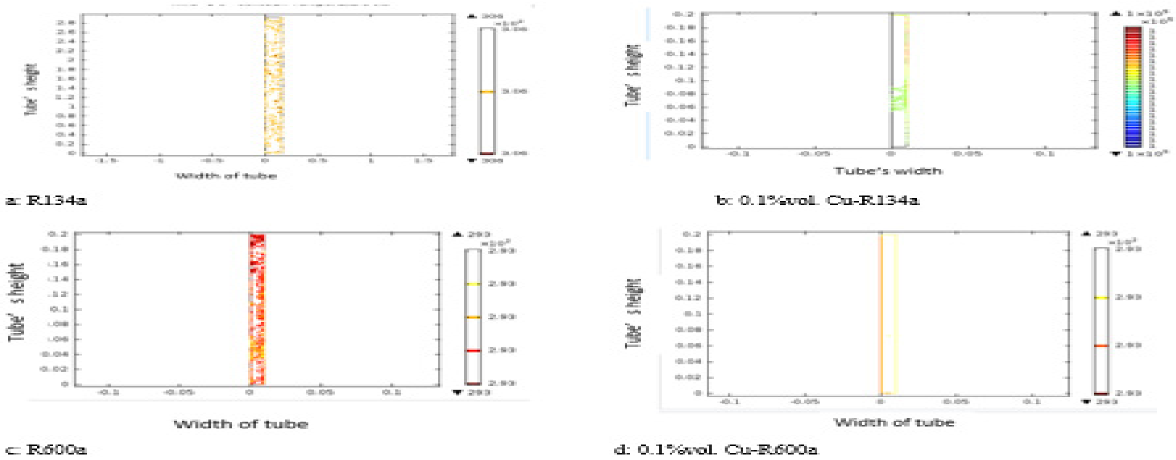

Fig. 5: Isotherm regions of the refrigerant variants and they move through the tube

The isotherm regions are regions where steady state temperature occurs at particular instant of time. It demonstrates the rate at which thermal heat transfer occurs in order to raise or drop the temperature of the adjoining regions. Fig. 4 shows that R134a and R600a are relatively thermodynamically not as efficient when compared with their nanorefrigerant variants. Moreover, comparison between R134a and R600a shows that though R600a performs poorly as against the performance of R134a, in terms of heat transfer properties, its nanorefrigerant variant (Cu-R600a) performs best. The results obtained corresponds to that of Attabo [5].

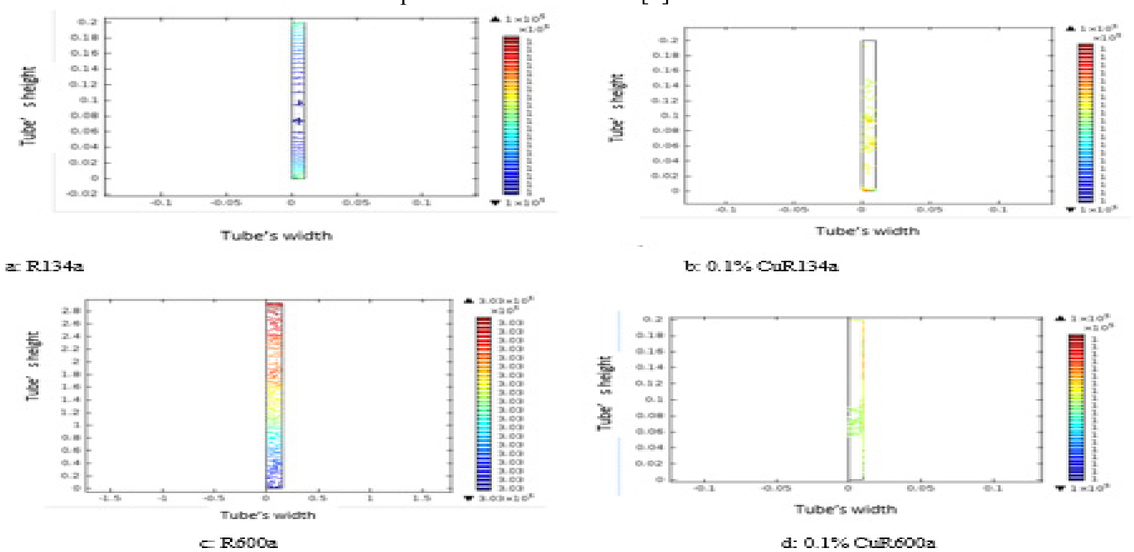

Fig. 6: Pressure Contours of Tube with the refrigerant variants

Fig. 6 shows that the pressure contours within the tube as the refrigerants flows through. The greater the density of the pressure contours, the greater the pressure inside the tube. From previous studies, it was noted that, the larger the refrigerant side pressure, the greater the compressor work and ultimately the greater the refrigeration power requirement. Therefore, both R134a and R600a possess greater pressure contours than the nanorefrigerants indicating that there is more pressure in the tube, thus increasing the compressor work. It can also be interpreted that the pressure in the nanorefrigerant is dissipated more quickly than in the conventional refrigerants R134a and R600a.

4. Conclusion

The study demonstrated that the use of nanorefrigerants will result in a significant reduction in the power requirements of refrigerators and refrigeration plants. This result validates the experimental results obtained by Attabo [5]. R600a which is considered as a replacement for R134a due to its low global warming potential and zero ozone depletion has been proven to be relatively less thermally inefficient than R134a. However, upon the introduction of copper nanoparticles, its thermophysical properties have been improved so greatly that it can readily replace R134a. This study has also been able to prove that without experimentation and financial commitment, based on knowledge of boundary conditions, material properties and relevant physics, CFD models can be used to determine the effect that the infusion of conductor based nanoparticles has on the performance of nanorefrigerants as working fluids in refrigeration systems.

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