Thermophysical and Heat transfer Characteristics of R134a-TiO_2 Nanorefrigerant: A Numerical Investigation

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Abstract. Nanorefrigerants are new class of nanofluids in which nanoparticles are suspended in the refrigerant host fluid. They have superior thermophysical and heat transfer characteristics than conventional refrigerants. Present study investigates the thermophysical, heat transfer and pressure drop characteristics of TiO_2-R134a nanorefrigerants flow boiling in a mini tube. Studies have been conducted at various heat fluxes, vapour qualities and particle concentrations. From the studies it is observed that the presence of nanoparticles in refrigerant can enhance the thermophysical and heat transfer characteristics. Studies show that with TiO_2-R134a nanorefrigerant, the heat transfer coefficient increases by 30.2% when particle concentration is 1.5 %.

1. Introduction

Nanorefrigerants are a new research frontier described by the application of nanoparticles as additives into various refrigerants. The studies of nanorefrigerants are still in the embryonic stage. Recently scientists used nanoparticles in refrigeration and air conditioning systems because of its remarkable improvement in thermo-physical, and heat transfer capabilities. M. Akbari et al. [1] conducted a comparative study on heat transfer of nanofluids. CFD predictions of laminar mixed convection of Al_2O_3-water nanofluid in a horizontal tube with uniform heat flux by, single-phase and different two-phase models were compared. The numerical predictions of the convective heat transfer coefficient are compared with experimental data. Based on the study the authors reported that, the predictions by the three two-phase models are essentially the same and the less expensive model was VOF. Jacqueline et al. [2] conducted experimental studies on flow boiling heat transfer and pressure drop of pure R134a in a mini tube. The results show that in the low quality region, there is a significant influence of heat flux on heat transfer coefficient.The COP and freezing capacity of vapour compression system enhances by using refrigerant-nanolubricant mixtures [3]. Experimental studies on flow condensation heat transfer studies show an increased heat transfer coefficient for nanorefrigerant [4]. However, research works in the area of nanorefrigerants is scarce due to complexity of the phenomenon of flow boiling heat transfer. This necessitates more studies regarding thermophysical and heat transfer characteristics of nanorefrigerants through mini tubes. Present study includes Modelling of thermal conductivity of R134a based nanorefrigerants and numerical investigations by simulation of flow boiling of R134a-TiO2 nanorefrigerant through a mini channel using ANSYS FLUENT. Parametric studies also been performed to evaluate the effect of addition of nanoparticles to the refrigerants.

2. Modelling of Thermal conductivity of nanorefrigerant

Various theoretical models were developed by researchers to explain the unusual behaviour in thermophysical properties of nanofluids. In order to investigate the effect of nanoparticle on the thermal conductivity of TiO2-R134a nanorefrigerant, a Modelling study has been conducted with available classical models.

Thermal conductivity models		
Model	Correlation	Description
Maxwell	$\frac{k_{eff}}{k_{bf}} = \left[\frac{k_p + 2k_{bf} - 2\phi(k_{bf} - k_p)}{k_p + 2k_{bf} - \phi(k_{bf} - k_{pbf})}\right]$	k_{eff} and k_{bf} -thermal conductivities of nanofluids and base fluid. k_f and k_p thermal conductivity of the liquid and the particle. ϕ is the particle volume fraction.
Hamilton and crosser	$\frac{k_{eff}}{k_{bf}} = \left[\frac{k_p + (n-1)k_{bf} - (n-1)\phi(k_{bf} - k_p)}{k_p + (n-1)k_{bf} - \phi(k_{bf} - k_{bf})}\right]$	Modified Maxwell model. Introduced the shape factor (n)
Yu and Choi	$\frac{k_{eff}}{k_{bf}} = \left[\frac{k_p + 2k_{bf} + 2(k_p - k_{bf})\phi(1+\beta)^3}{k_p + 2k_{bf} - \phi(k_p - k_{bf})(1+\beta)^3}\right]$	Introduced the effect of interfacial layering.
Timofeeva	$\frac{k_{eff}}{k_{bf}} = 1 + 3\phi$	Suggested the effective medium theory
Koo and Kleinstreuer	$5X\, 10^4 \beta\phi\, \rho_{bf} C_p \sqrt{\frac{k_{bf}\, T(-6.04\phi + 0.4705)T + (1722.3\phi - 134.63)}{d_p \rho_p}}$	Considered Brownian motion enhanced heat transfer

3. Modelling and simulation of flow boiling.

The computational fluid dynamic code FLUENT is employed for the present study. For nanofluids, the number of particles in the computational domain, even for a very small particle volume fraction, is extremely large due to the very small size of the particles. This makes Lagrangian-Eulerian approach difficult to solve the nanofluid flow problems. Therefore, in the present study Eulerian-Eulerian VOF model has been selected. All the physical properties are calculated by taking weighted average of different phases based on their volume fraction throughout each control volume. The single set of momentum equations is solved to find the velocity components which are shared by all the phases. In the same manner, a shared temperature is calculated from a single energy equation. Tracking of the interfaces between the phases is accomplished by the solution of a continuity equation for the volume fraction of one (or more) of the phases. The k-ε turbulence model is used to simulate turbulent flow. This model provides general description of turbulence which allows for the effects of transport of turbulent properties by the mean flow and diffusion for the production and destruction of turbulence. Turbulent kinetic energy (K) and its dissipation rate (ε) are coupled to the governing equations via the turbulent viscosity relation. Second order upwind scheme is applied for solving the convective and diffusive terms. PISCO discretization was adopted for pressure velocity factor.

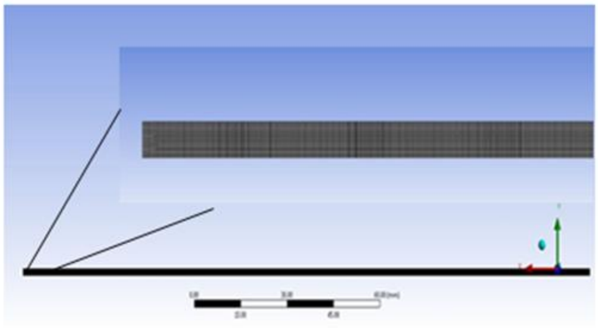

Fig. 1 Geometry of computational domain with mesh

A 2- D axisymmetric geometry is considered. Geometry of the computational domain is shown in Fig 1. Flow through the mini tube is assumed to be a steady and incompressible. Constant heat flux boundary condition is applied at the outer surface of the tube. The mesh used is of rectangular type. In order to solve the governing equations, appropriate boundary conditions were applied. Operating pressure is set as 6.0178 bar and accordingly the saturation temperature of the refrigerant was 295 K. The vapour quality is varied from 0.1 to 0.9 with an interval of 0.1. Studies have been conducted for a mass flux of 440 kg/m^{2}s. Constant heat flux wall boundary conditions are applied at the outer surface varying from 5000 to 47000 W/m^2. No slip boundary conditions are imposed at the wall.

4. Results and discussion

4.1 Thermal conductivity

Modelling of thermal conductivity of TiO_2/R134a nanorefrigerant with theoretical correlations proposed by various researchers has been conducted and which is presented here. Broad range of particle concentrations from dilute level has been considered for the study.

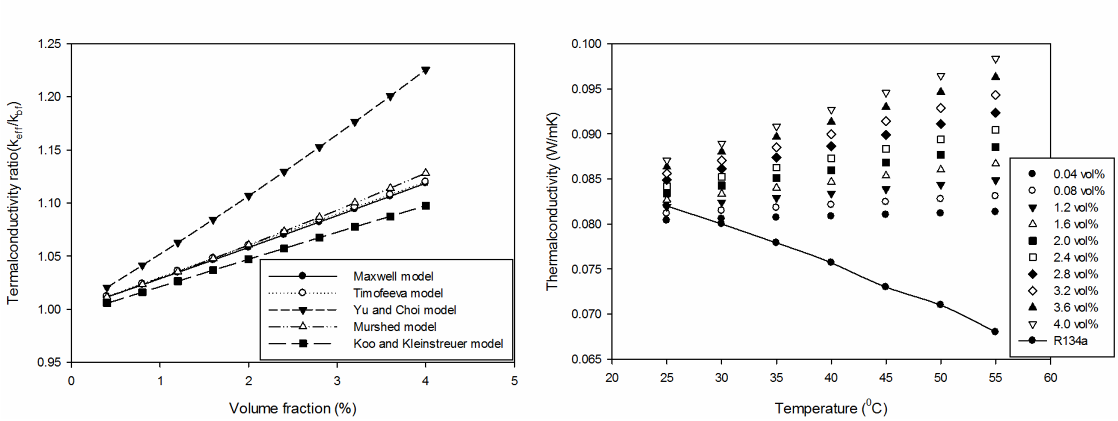

Fig. 2 Variation of thermal conductivity and thermal conductivity ratio of nanorefrigerant as a function of particle volume fraction and temperature.

From Fig. 2 it is evident that the thermal conductivity of TiO_2-R134a nanorefrigerant increases linearly with nanoparticle volume concentration of (0.4-4 vol. %). It can be concluded that the thermal conductivity of nanorefrigerant increases with the increase of particle volume fractions. Fig. 3 shows the variation of thermal conductivity of pure refrigerant and nanorefrigerant with temperature ranging from 25^{0}C to 55^{0}C. It can be seen that, the thermal conductivity of TiO_2/ R-134a nanorefrigerant increases linearly with temperature, while for pure refrigerant, thermal conductivity moderately decreases with increase in temperature. Besides, with the rise of temperature, the Brownian motion of nanoparticles will intensify and the contribution of micro convection in heat transport will increase, which results in the augmentation of thermal conductivity. For pure refrigerant, thermal conductivity was decreased with increasing temperature. This is due to the fact that when temperature increases, the liquid is evaporated, which causes the atoms to be positively charged and vibrate with greater amplitude. Results show that, the thermal conductivity enhancement is about 44% at a temperature of 55^0C with higher nanoparticle concentration (4 vol. %) compared to pure R134a refrigerant. At lower particle concentration and temperature (0.4 vol. % and 25^0C), the obtained result shows lowest increase in thermal conductivity value i.e. 2%.

4.2 Flow boiling heat transfer coefficient

A grid independency test is conducted to guarantee the accuracy of the numerical results. Twelve different sets of uniform grids have been used to check for grid independency. Tests were carried out with pure refrigerant at different heat flux conditions for each grid. It can be seen from Fig. 3 that, the value of the heat transfer coefficient converges as the number of grid cells increases. Based on this test optimum grid was selected as 24360.

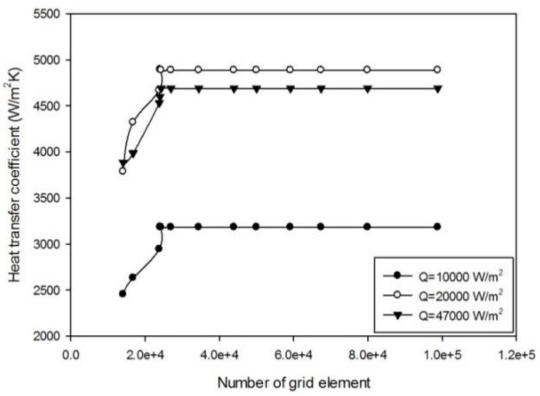

Fig. 3 Grid independent study.

Before conducting the numerical studies with nanorefrigerants, the validation of the numerical scheme is carried out by comparing the result with experimental results reported in literature (Jacqueline et al., 2011). Results show good agreement with experimental data. Subsequently, studies were extended with TiO_2-R134a nanorefrigerants in order to find out effect of nanoparticles on heat transfer coefficient and pressure drop characteristics of R134a flow boiling through horizontal mini channel. Fig. 4 shows the comparison of boiling heat transfer coefficient of pure R134a obtained from experimental results (Jaqciline.et al., 2011) and numerical predictions for a mass flux of440kg/m^{2}s at heat fluxes in the range of 10000W/m2 to 47000 W/m^2.The obtained heat transfer coefficient from the simulation shows good agreement with experimental results. From figures, it is clear that heat transfer coefficient increases with an increase in heat flux from 10000 W/m^2 to 47000W/m2. Both experimental and modelling results exhibit the similar trends.

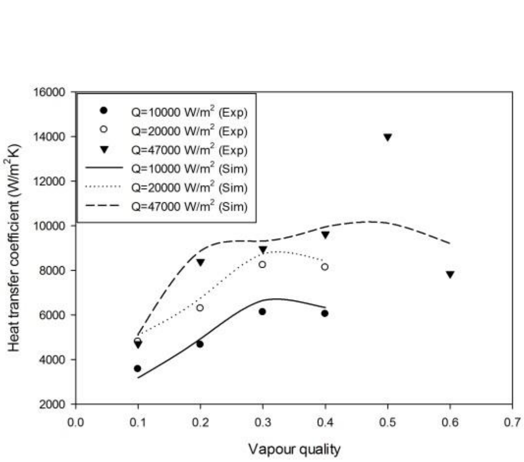

Fig. 4 Comparison of hear transfer coefficients of pure R134a from experiment and simulation at a mass flux of $440kg/m^2s$.

Fig. 5(a) shows the comparison of heat transfer coefficient of pure R134a and TiO_2-R134a nanorefrigerant respectively at a particle concentration of 0.5 % at a mass flux of $440kg/m^2$ and three different heat fluxes. From figure it is found that heat transfer coefficient increases with increase in inlet vapour quality up to about 0.4 and thereafter decreases with increase in vapour quality. The similar trend is observed for all heat flux conditions. There is an increase in heat transfer coefficient by 19% for TiO2 –R134a nanorefrigerant (0.5% volume fraction) compared to pure refrigerant at a heat flux 47000 W/m2 and mass flux of 440kg/m2s. At higher vapour quality region nanorefrigerant and pure refrigerant show similar trends, that is decrease in the heat transfer coefficient. This is due to the formation of the vapour barrier between the bulk fluid and the wall of the tube
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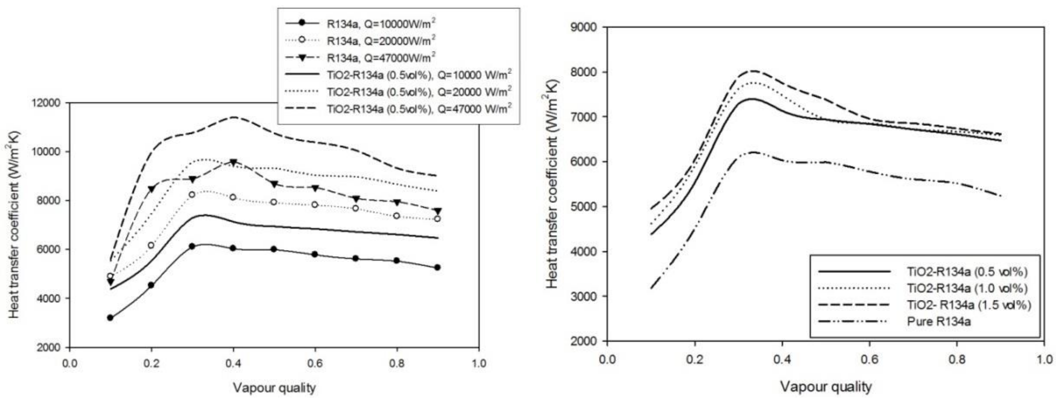

Fig. 5 Variation of heat transfer coefficient with (a) Vapour quality (b) particle concentrations.

Fig. 5(b) shows the variation of heat transfer coefficient at 3 particle concentrations of TiO_2 nanoparticles. While increasing the volume concentration of the nanoparticle the heat transfer coefficient increases in a noteworthy manner. In the case of TiO_2-R134a nanorefrigerant, heat transfer coefficient increases from 19% to 30.2% when particle concentration is increased from 0.5 to 1.5 vol. %. This anomalous enhancement in heat transfer coefficient is due to the improved thermophysical and boiling heat transfer properties of nanorefrigerant.

4.3 Pressure drop characteristics:

The local pressure drop dp/dz, calculated by assuming pressure as the linear function of tube length. Fig. 6 shows the effect of particle concentrations on pressure drop. It is evident that, for nanorefrigerant, irrespective of the heat flux, the pressure drop was increased with increase in particle concentration. As the volume fraction increase from 0 .5 % to 1.5%, the pressure drop increases by 21.4 % 27.5% compared to pure refrigerant.

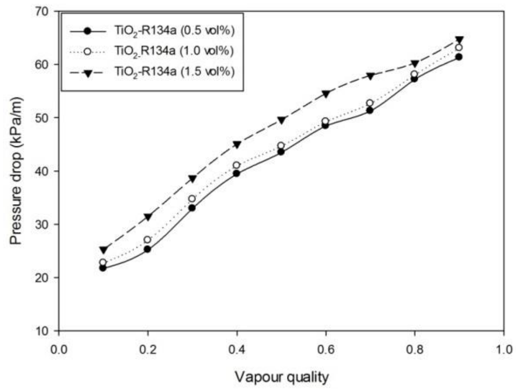

Fig 6 pressure drop of nanorefrigerants at different particle concentrations at a heat flux of 47000w/m^2.

5. Conclusions

Numerical investigations on heat transfer characteristics of nanorefrigerants flowing in a mini channel have been carried out. An increase in thermal conductivity by 44% for nanorefrigerant compared to pure refrigerant was manifested. The heat transfer coefficient increases with increase in heat flux and particle concentration. The heat transfer coefficient increases with increase in inlet vapour quality up to about 0.4 and thereafter decreases. The maximum increase in heat transfer coefficient of TiO_2–R134a nanorefrigerant compared to pure refrigerant was by 30.2% ,at a volume fraction 1.5%.. The maximum increase in pressure drop was observed as 27.5% at a volume fraction of 1.5 % .TiO2 based nanorefrigerant shows superior heat transfer performance than pure refrigerant.

References

[1] M.Akbari, N.Galanis and A.Behzadmehr.,Comparative analysis of single and two- phase models for CFD studies of nanofluid heat transfer, Int .J thermal science, 50, 1343-1354.

[2] Jacqueline B.Copetti, Fiavia Zinaniand Nicole L.F.Kunsler., Flow boiling heat transfer and pressure drop of R-134a in a mini tube: An experimental investigation, Int. J. Thermal and fluid science, 35, 636-644.

[3] Subramani, and N., Prakash, M.J., (2011). Experimental studies on a vapour compression system using nanorefrigerants, Int. J. Eng. Science and technology, 3, 95–102

[4] M.A. Akhavan-Behabadi, M.K. Sadoughi , Milad Darzi and M. Fakoor-Pakdaman (2015) Experimental study on heat transfer characteristics of R600a/POE/CuO nanorefrigerant flow condensation, J. Thermal and fluid Science, 66,46-52