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An Investigation about using Nanorefrigerants in Air Conditioning Systems According to the Theoretical, CFD and Experimental Review of the Recent Literature

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Greenhouse gases (GHG) causing global warming and climate change. In the year 2014, 32.3 billion tones CO2 emitted to the atmosphere as the most important greenhouse gas. According to the statistics, a significant portion of this amount is related to electricity demand of air conditioning systems, for producing a one ton of refrigeration in HVAC air cooled or water cooled systems respectively 1026 and 764 grams GHG emitted in the atmosphere. Therefore, air conditioning systems have an important role in the global warming and climate change. By increasing the COP of air conditioning systems the electricity demand of them reduced. One strategy for increasing the COP of air conditioning systems is using nanorefrigerants. In the present study, a comprehensive information is given regarding to use nanorefrigerants in air conditioning systems according to the theoretical, CFD and experimental review of the recent literature. This paper gives assistance to designers of air conditioning systems in their future efforts for selecting refrigerant for their systems.

Keywords: *air conditioning systems, global warming, climate change, Greenhouse gas, Nanorefrigerants, COP*

1. Introduction.

NASA has just reported that global temperature increased 1.4°F since 1880, the arctic ice decreased 13.3 percent per decade and the land ice diminished 287 billion metric tons per year. The alarming report that makes heating,ventilation,air conditioning and refrigeration (HVAC&R) systems researchers more aware about coefficient of performance (COP) of HVAC&R systems because their energy consuming issue. Some researchers are working on increasing COP of air conditioning systems by using nanorefrigerants as the first or secondary working fluid. The nanorefrigerants obtain by mixing less than one percent of nano particle same as Al_2O_3, CuO, CNT, SiO_2…with ordinary refrigerants same as R134a, R410, R22….

According to the thermodynamics equations the COP of air conditioning systems depend on compressor work consumption and refrigeration effect. The nano particles when added to ordinary refrigerants improve thermophysical properties and heat transfer characteristics of base fluid due to their high thermal conductivity when suspended to base fluid. In the present paper, the comprehensive information about using nanorefrigerants in air conditioning system is given. According to the literature review, some researchers just focused on experimental aspect and some focused on theoretical aspect and others focused on computational fluid dynamics (CFD) aspect of using nanorefrigerants in air conditioning systems but, the present study is comprised of all aspects, therefore can help more to designers of air conditioning systems in their future effort about choosing the best refrigerants.

2. Literature review of recent years

2.1.Theoretical aspect

In this section theoretical aspect of using nanorefrigerants in air conditioning systems is given.

Mahbubul et al. [1] theoretically investigated about thermophysical properties and heat transfer performance of Al_2O_3 /R134a nanorefrigerant. They showed that the thermal conductivity of nanorefrigerant was calculated by Sitprasert et al.[2].

$$k_{r,n} = \frac{(k_p-k_l)\phi k_l[2\beta_1^3-\beta^3-1]+(k_p+2k_l)\beta_1^3[\phi\beta^3(k_l-k_r)+k_r]}{\beta_1^3(k_p+2k_l)-(k_p-k_l)\phi[\beta_1^3+\beta^3-1]} \tag{1}$$

Where $k_{r,n}$, k_p, k_r are the thermal conductivity of nanorefrigerants, solid particles and pure refrigerants, respectively $\beta=1+\frac{t}{r_p}$, $\beta_1=1+\frac{t}{2r_p}$; the thickness of interfacial layer, t depend on temperature, where $t=0.01(T-73)r_p^{0.35}$ and the thermal conductivity of the interfacial layer can be found from $k_l=C\frac{t}{r_p}k_r$; where C=30, a constant for Al_2O_3 nanoparticles, T is the temperature in Kelvin and r_p is the radius of nanoparticles and ϕ is particle volume fraction.

Brinkman model [19] stated as following have been used to investigate the viscosity of nanorefrigerant,

$$\mu_m=\mu_r\frac{1}{(1-\phi)^{2.5}} \tag{2}$$

Where, μ_m is the effective viscosity of nanorefrigerant and μ_r is the viscosity of pure refrigerant.

Peng et al.[20] correlation was used to investigate the effect of nanoparticles on frictional pressure drop of nanorefrigerant and it is shown below:

$$\Delta P_{r,n,\,frict} = F_{PD}.\Delta P_{r,\,frict} \tag{3}$$

Where, F_{PD} is the nanoparticle impact factor and $\Delta P_{r,frict}$ is the frictional pressure drop of pure refrigerant.

To find the pumping power required to pump the nanorefrigerant throughout the system, an equation from Routbort et al. [21] can be used:

$$P_{Pumping} = Q\Delta P_{r,n,\,frict} \tag{4}$$

Where, Q is the volumetric flow rate which is a function of velocity.

The convective heat transfer coefficient of nanorefrigerant is:

$$h_{c,r,n} = \frac{Nu \times k_{r,n}}{D_i} \tag{5}$$

Where the Nusselt number, Nu for turbulent flow can be obtained from Dittus-Boelter equation [22]:

$$Nu = 0.023\,\mathrm{Re}_{r,n}^{0.8}\,\mathrm{Pr}_{r,n}^{0.4} \tag{6}$$

Reynolds number and Prandlt number can be calculated from Eq.(7) and (8) [23].

$$\mathrm{Re}_{r,n} = \frac{G \times D_i}{\mu_{r,n}} \tag{7}$$

$$\mathrm{Pr}_{r,n} = \frac{C_{P-r,n}\mu_{r,n}}{k_{r,n}} \tag{8}$$

Where $C_{p-r,n}$ is the specific heat transfer of nanorefrigerant and it can be calculated by [22]

$$C_{P-r,n} = (1-\phi)C_{P-r,L} + \phi C_{P-P} \tag{9}$$

Where Ø is particle volume fraction.

Mahbubul et al. [1] concluded that thermal conductivity of the nanorefrigernat increases with the increase of nanoparticle volume fraction and temperature. The thermal conductivity of nanorefrigerant decreases by increasing the particle size.

The viscosity of nanorefrigerant also increases with augmentation of nanoparticles concentration and the viscosity enhancement due to some particles concentration is differed for different types of refrigerants.

The frictional pressure drop shows rapid increment of more than 3 vol% with particle volume fraction and pumping power increases with particle concentration similar to pressure drop increment. The convective heat transfer coefficient increase significantly by using nanoparticle.

A. Alawi et al. [15] concluded that carbon nanotube (CNT) can be considred as a promising passive heat transfer enhancement additive in comparison with the spherical nanoparticles of Al, Si, Ti, Cu, diamond and their oxide versions.

2.2. Experimental aspect

Balaji et al. [5] investigated experimentally about improving air conditioning system with nanofluid based intercooler. They used split air conditioner with cooling capacity 18000 btu/hr as shown in Fig.1 and Fig.2.

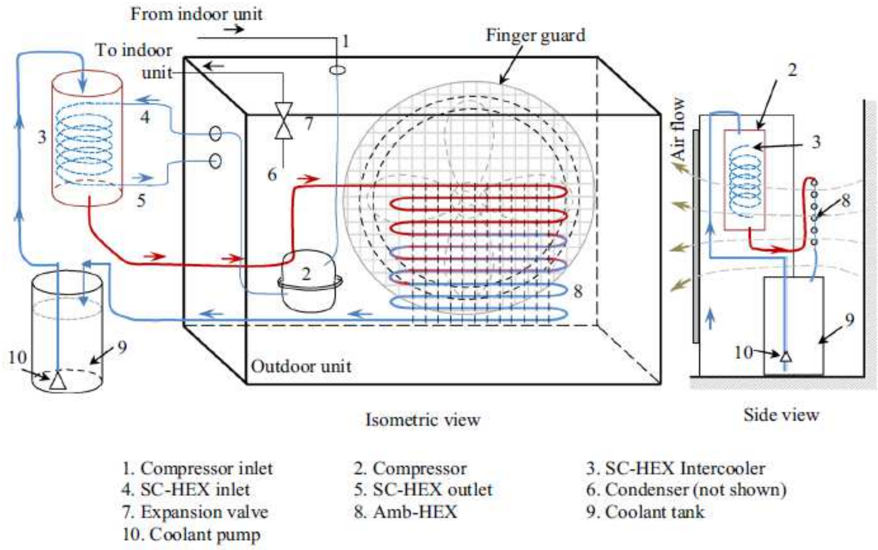

Figure1. Schematic arrangement of the experimental set-up

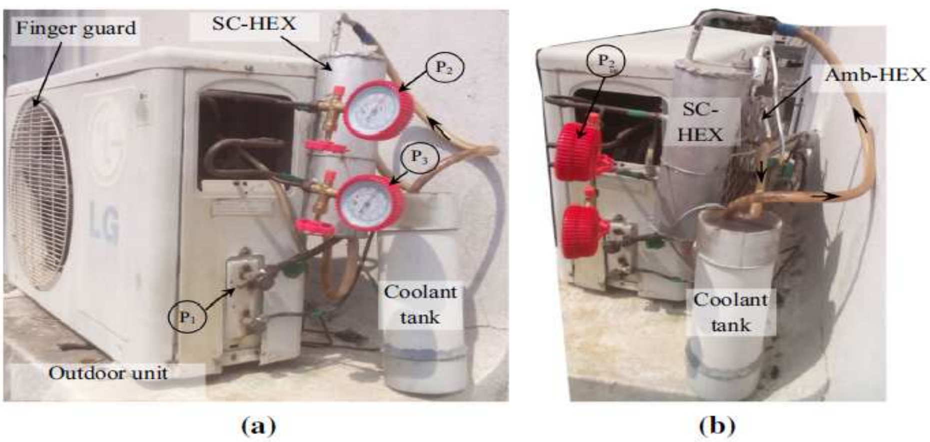

Figure 2. Photographic view of the experimental set-up, a. Front side, b. Rear side

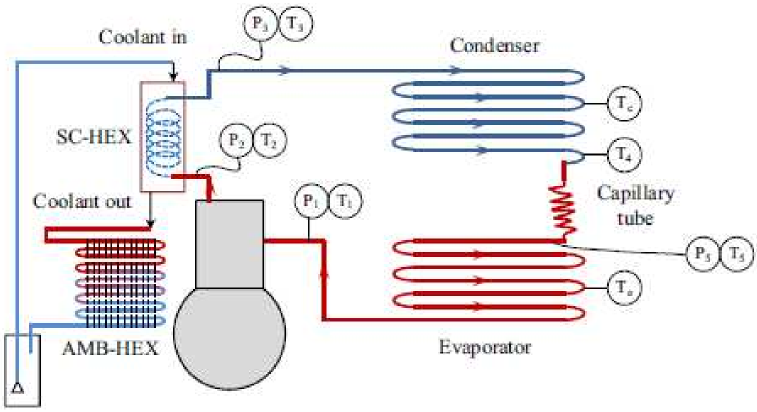

Figure 3. Location of temperature and pressure measurement in the experiment

In their test, Al_2O_3 nanoparticles with different volume concentration were added to the base fluid, Fig.3 (shows SC-HEX intercoole). The hot refrigerant from the compressor outlet flows through the helical coil in the SC-HEX. The coolant nanofluid passes through the shell side and refrigerant through tube side.

Nanoparicles used in that work were spherical shaped powders with size ranges of 20-40 nm for Al2O3 and prepared at three different concentrations 0.25, 0.5, 0.75% by volume fraction.

They experimentally concluded the COP of modified air conditioning system with the intercooler improved significantly compared with the existing air cooled condenser.

As shown in Fig.4, by increasing the volume concentration of nanoparticle in the coolant and by increasing the flow rate of nanofluid in the intercooler, the COP was found to increase. Power consumption reduced significantly when the condenser temperature reduced and also when the pressure ratio between the compressor and condenser reduced. The highest increment in the COP value of around 31% was observed for 30:70 EG:W with flow rate of 2 LPM and increment of 49.32% was observed for the nanofluid with 0.75% nanoparticle volume concentration with a flow rate of 2LPM.

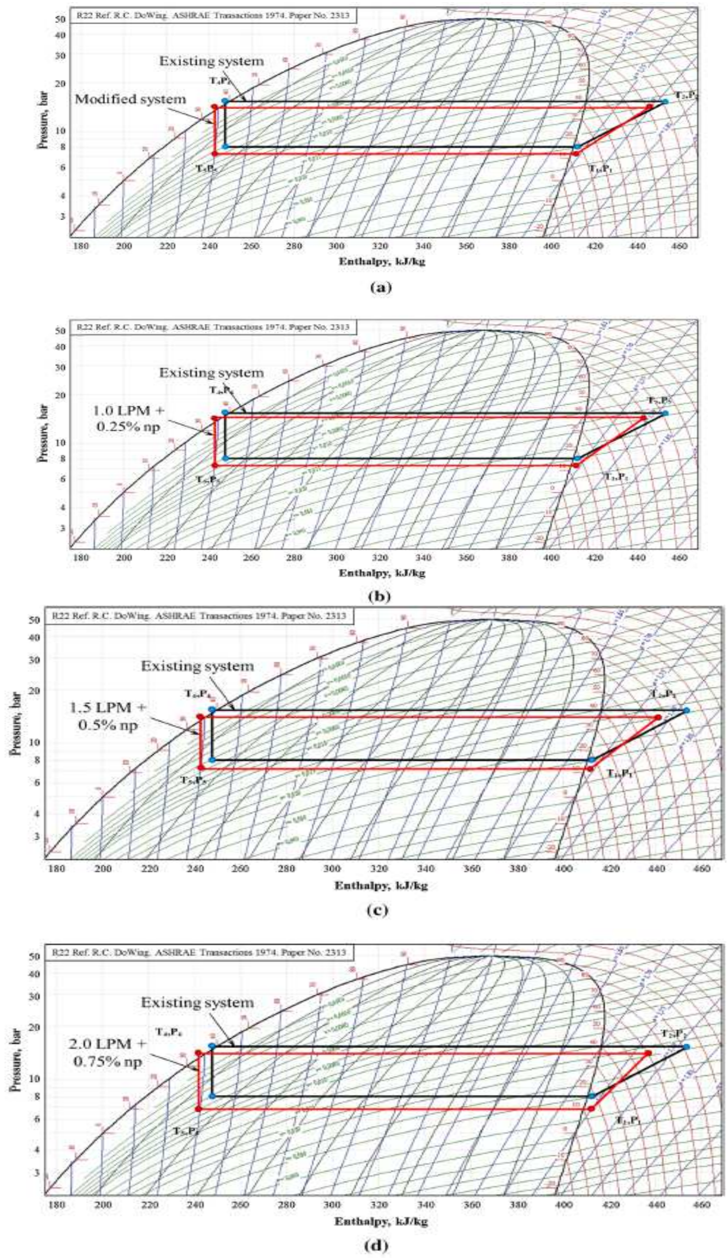

Figure 4. P–h diagram of air conditioning systemswith and without intercooler. A. Existing system (without IC) versus modified system (with IC–BF)—1 LPM. B. Existing system (without IC) versus modified system (with—NF + 0.25% v/v)—1 LPM. C. Existing system (withoutIC) versus modified system (with IC–NF + 0.5% v/v)—1.5 LPM. D.Existing system (without IC) versus modified system (with IC—NF +0.75% v/v)—2 LPM.

2.3. Numerical aspect

N.A. Che Sidik et al.[11] built 3D model by using gambit V 2.4.6 for laminar flow and ansys for turbulent flow due to its increased complexity and near wall treatment. As shown in Fig.5 (an evaporator annuls) was studied for that work. The inner cylinder with diameter D=20mm, the thickness of cylinder wall t=5mm and the hydraulic diameter is D_h=10mm, the computational length of annulus L=400mm. The outer surface of pipe of the annular space is maintained under constant heat flux. Whereas the inner surface of the pipe is kept iso thermally at a constant temperature T_i. They numerically concluded that SiO_2 has the greatest Nusselt number followed by Al_2O_3, ZnO, CuO and the lowest value for pure refrigerant. SiO2/R141b has the highest Nusselt number comparing to other types of nanorefrigerants according to Fig.6 and they obtained from figure 7 that SiO2/R-141b has the highest Nusselt number comparing to other types of nanorefrigerants. This is because the R141b base fluid has highest viscosity which leads to increase the velocity for the nanorefrigerants since the velocity proportional directly with the viscosity of the base fluid. However, second base fluid is R-134a then R-12 and finally R-22 according to their viscosity.

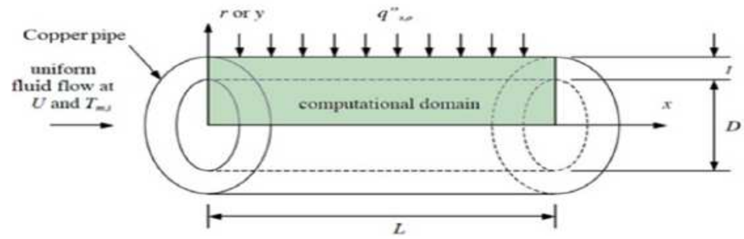

Figure 5. Evaporator test section

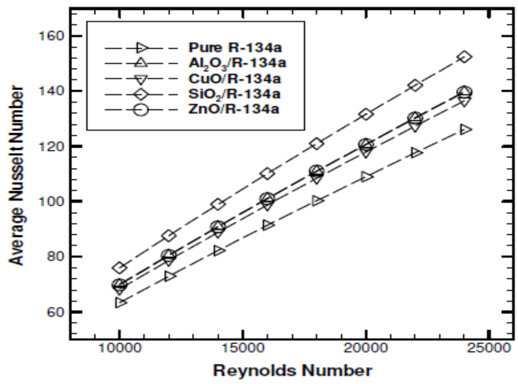

Figure 6. The effect of nanorefrigerant types on the average Nusselt number (Using φ = 4%,dp = 20 nm and qw = 5000 W/m2).

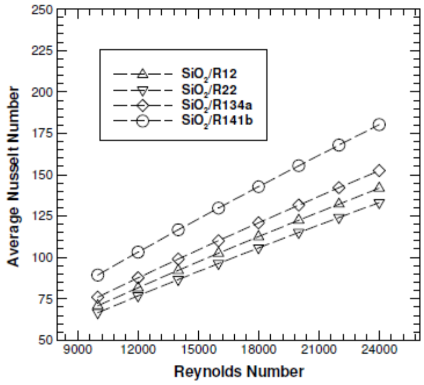

Figure 7. The effect of different types of pure refrigerants on the average Nusselt number (Using SiO2, dp = 20 nm, φ = 4% and qw = 5000 W/m2)

Kamil Arslan [12] numerically investigated the heat transfer and fluid friction of R134a based TiO_2 nanorefrigerants for different particle volume fractions. The hydro dynamically and thermally developing three dimensional steady laminar flow conditions in a horizontal circular cross-sectioned duct were carried out as shown in Fig.8, Reynolds number was changing from 8×10^2 to 2.2×10^3.

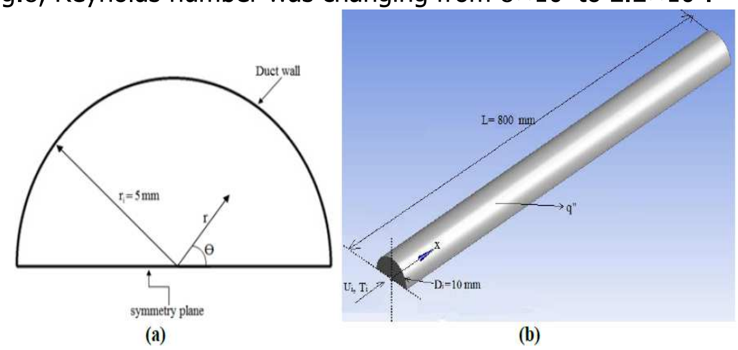

Figure 8.a) Cross-section of the duct, b). computational domain of the duct with boundary.

In their study, a general purpose finite-volume based commercial CFD software package Ansys Fluent 14.5 has been used to carry out the numerical study. The software provides mesh flexibility by structured and unstructured meshes as shown in Fig.9.

The results of numerical computations were presented in terms of average heat transfer coefficient and average Darcy friction factors in Fig.10. On the other hand, average Darcy friction factor decreases with increasing Reynolds number, also average heat transfer coefficient increases with increasing particle volume fractions.

The velocity and temperature distribution for different number of Reynolds were presented graphically in Fig.11 and Fig.12.

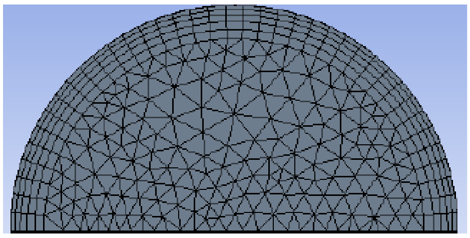

Figure 9. Mesh distribution of the duct

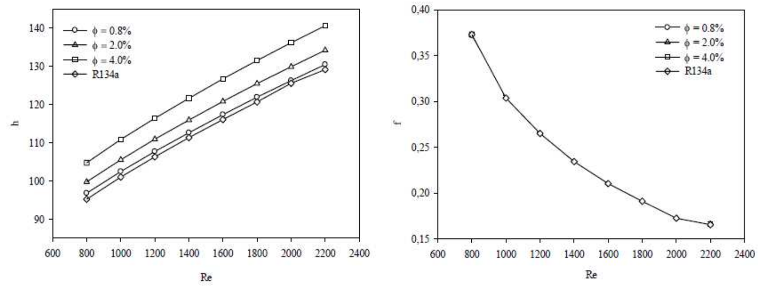

Figure 10. The changing of (a) average heat transfer coefficient and (b) average Darcy friction factor; with Reynolds number for different particle volume fractions

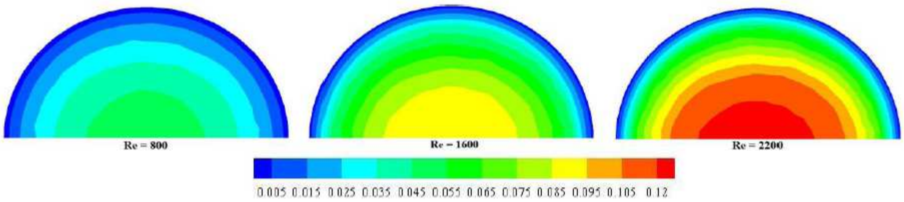

Figure 11. Velocity distributions at the outlet of the duct for different Reynolds numbers.

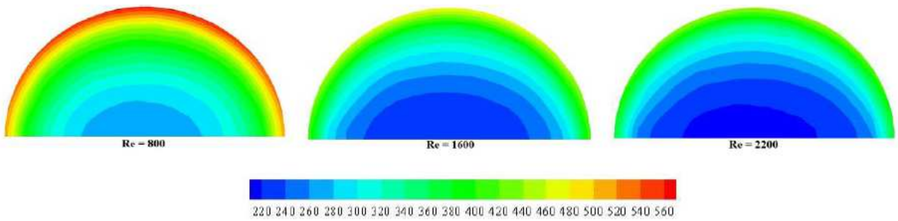

Figure 12. Temperature distributions at the outlet of the duct for different Reynolds numbers

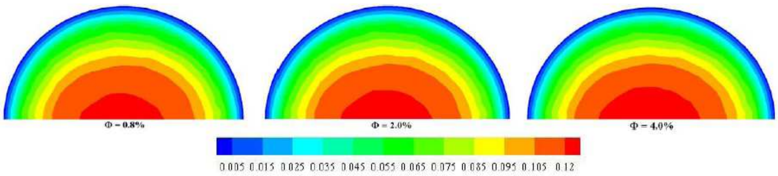

Figure 13. Velocity distributions at the outlet of the duct for different particle volume fraction

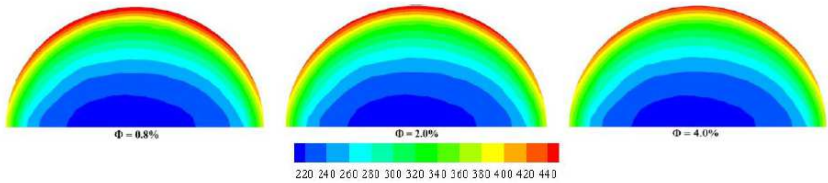

Figure 14 Temperature distributions at the outlet of the duct for different particle volume fraction

Velocity and temperature distribution at the outlet of the duct for Re=2.2×103 and different particle volume fraction were presented in Fig.13 and Fig.14, respectively. It is obtained in the figures that increasing the particle volume fraction does not have significant effect on the velocity and temperature profiles. While the velocity magnitude decreases towards the duct wall, the temperature magnitude increases.

3. Conclusion

In the present paper the most of the studies about using nanoreffrigerants according to the recent literature are summarized. Due to more COP of air conditioning systems which are used nanorefrigerants, obviously, it is expected using nanorefrigerant instead of common refrigerants increase in near future.

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Nomenclature

C	alumina constant = 30
C_p	specific heat of the fluid, J/kg K
d_p	nanoparticle diameter (nm)
D_i	inner diameter, m
G	mass flux (kg/m2 s)
h	heat transfer coefficient (W/m2 K)
K	thermal conductivity (W/m K)
Nu	Nusselt number
Q	volumetric flow rate (m3/s)
P	pressure (Pa)
Pr	Prandlt number

Subscripts

Frict	Frictional
i	inner wall
l	interfacial layer/nanolayer
n,r	nanorefrigerant
p	particle
PD	pressure drop
r	pure refrigerant

ΔP frictional pressure drop (Pa)
Re Reynolds number
T temperature (K)
t interfacial layer thickness (m)

Greek letters

 volume fraction of nanoparticle
μ fluid dynamic viscosity, $Kg/m.s$

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