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Performance Analysis of Nanorefrigerants in Heated and Rotating Concentric and Eccentric Annulus Cylinders

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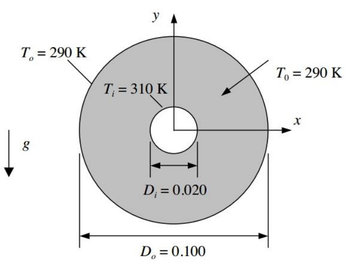
Graphical abstract



Abstract

The past decade has seen rapid development of nanofluids science in many aspects. In recent years, refrigerant-based nanofluids have been introduced as nanorefrigerants due to their significant effects over heat transfer performance. In this investigation, the Control Volume based Finite Element Method (CVFEM) is used to simulate the natural convection heat transfer of nanorefrigerant in cylindrical horizontal annuli with imposed temperatures in both surfaces. The Maxwell–Garnetts (MG) and Brinkman models are also employed to estimate the effect of thermal conductivity and viscosity of nanorefrigerant. The governing parameters are nanoparticles types, nanoparticles concentration and size, effect of Rayleigh numbers (Ra), eccentricity and rotation of inner cylinder. Results are presented in the form of isotherms and streamlines of nanorefrigerant temperature and velocity. The results indicate that Al_2O_3/R141b with concentration (2%) and nanoparticles size (20 nm) has the best heat transfer performances. Moreover, the heat transfer and fluid flow enhance by increasing the Rayleigh numbers (Ra). Eccentricity and rotation of inner cylinder also have effects on heat transfer characteristics.

Keywords: Natural convection, heat transfer, nanorefrigerant, annulus

1.0 INTRODUCTION

Refrigerant-based nanofluid is one kind of nanofluids, in which the host fluid is conventional pure refrigerant. Experimental studies showed that the refrigerant-based nanofluid has higher thermal conductivity than the host refrigerant [1], and the refrigeration system using refrigerant-based nanofluid has better performance than that using conventional pure refrigerant [2–4]. However, the aggregation and sedimentation of nanoparticles in the refrigerant-based nanofluid may reduce the stability of refrigerant-based nanofluid and limit the application of refrigerant-based nanofluid in the refrigeration system. Since then, numerous studies have been carried out on the thermophysical and heat transfer performance of nanorefrigerants. Literatures also show that thermal conductivity and heat transfer performance can be increased by suspending nanoparticles in refrigerants [5–6].

Refrigerants are used in refrigeration and air conditioning systems in industrial and commercial buildings. About 20–50% of total energy is used by these systems in industrial and commercial buildings. Moreover, energy is on the head of "Top Ten" global problems of humanity for the next fifty (50) years [7]. Nanorefrigerants have the potential to enhance heat transfer rate thus making a heat exchanger in air conditioning and refrigeration systems more compact. In addition, they decrease the amount of energy needed to operate those systems. This, consequently, will reduce energy consumption in the sectors along with reduction in emission, global warming potential and greenhouse gas effects. However, for accurate and reliable performance investigation (e.g. heat transfer, energy and lubricity) - determination of fundamental properties such as thermal conductivity, viscosity, density, surface tensions and heat capacity of the nanorefrigerant with varied concentrations - needs to be carried out.

Heat transfer within horizontal annuli has many engineering applications such as heat exchangers, solar collectors, thermal storage systems, and cooling of electronic components. Several applications use natural convection as the main heat transfer mechanism. Therefore, it is important to understand the thermal behaviour of such systems when only natural convection is in effect, so that methods to enhance heat transfer characteristics in such systems can be devised [8].

The geometric shape of the cylindrical annulus creates non-uniformity in heat transfer within the annulus. With a better understanding of the flow field, it is possible to devise methods for heat transfer enhancement. An innovative technique for improving heat transfer is using ultra fine solid particles in the base fluids, which has been used extensively in the past ten years. The term nanofluid refers to fluids in which nano-scale particles are suspended in the base fluid [9]. The particles are different from conventional particles (millimeter or micro-scale) in that they tend to remain suspended in the fluid and no sedimentation occur which causes no increase in pressure drop in the flow field [10].

The most widely studied horizontal annulus geometry is the concentric circular cylinders, with over 20 publications currently available that contain experiments and numerical data for the average heat transfer, through the enclosed region. These include the experimental studies of Koshmarov and Ivanov [11], Kuehn and Goldstein [12,13] and Collins *et al*. [14]. Numerical data for the concentric circular annulus are reported by Projahn *et al.* [15], Farouk and Guceri [16], Cho *et al*. [17], Prusa and Yao [18], Mahony *et al.* [19], Date [20], Yan-Fei *et al.* [21] and Yoo [22]. Teertstra *et al.* [23] presents a comprehensive review of all experimental and numerical data, along with a comparison of available correlations and analytical models.

The past decade has also witnessed extensive work on convective heat transfer using nanofluids. Studies on the enhancement of heat transfer characteristics in forced convection applications were conducted by a number of researchers. On the other hand, heat transfer enhancement in natural convection applications has received little attention. Examples of the work conducted on natural convection heat transfers include the work of Khanafer *et al.* [24] who studied Copper–Water nanofluids in a two dimensional rectangular enclosure. They found that the heat transfer rate increased by increasing the percentage of the suspended particles. Similar enhancement was achieved experimentally by Nnanna *et al.* [25] for Cu nanoparticles in ethylene glycol and by Nnanna and Routhu [26] for Alumina–Water nanofluids. However, different experimental findings were reported by Putra *et al.* [27] on the natural convection of Al_2O_3 and CuO–Water nanofluids inside a cylindrical enclosure heated from one side and cooled from the other. They found that the natural convection heat transfer coefficient was lower than that of pure water. Wen and Ding [28] found that the natural convection heat transfer coefficient in a vessel composed of two discs using TiO_2 nanoparticles decreases by increasing the volume fraction of nanoparticles. Jou and Tzeng [29] simulated natural convection heat transfer of Copper–Water nanofluids in a two dimensional enclosure. They reported an increase in heat transfer by the addition of nanoparticles.

Based on literature reviews, no work has focused on natural convection heat transfer enhancement within a concentric cylindrical annulus utilizing nano-refrigerant. Therefore, the goal of this work is to investigate heat transfer characteristics of natural convection in the annulus between horizontal concentric cylinders using different types of nanorefrigerants. The problem will be investigated numerically by solving the Navier–Stokes and energy equations (NSE) using the finite volume technique. Heat transfer characteristics will be analyzed using a wide range of volume fractions of nanoparticles at various Rayleigh numbers.

2.0 GEOMETRY DEFINITION AND BOUNDARY CONDITIONS

For this problem we modelled two-dimensional flow of nanorefrigerant due to natural convection in the annular space between two concentric cylinders as shown in Figure 1. The cylinders are assumed to be very long in the z-direction (into the paper), thus the flow can be modelled as two-dimensional. Initially, the modelled annulus is at uniform temperature T_o = 290 K. The temperature of the inner surface is set to T_i = 310 K while the outer surface is kept at T_o = 290 K. As the working fluid near the inner cylinder is heated its density decreases inducing an upwards flow. Due to continuity the cold fluid near the outer walls must go downwards and a circulating flow develops in the annular space.

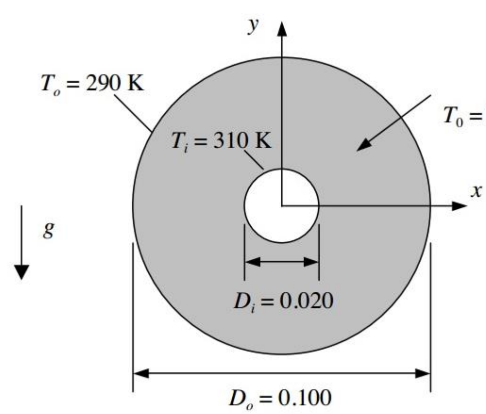


Figure 1 Sketch of the geometry of the problem (D_i and D_o in mm)

3.0 MATHEMATICAL MODELING AND NUMERICAL PROCEDURE

3.1 Problem Formulation

The continuity, momentum under Boussinesq approximation and energy equations for the laminar and steady-state natural convection in the two-dimensional annulus can be written in dimensional form as follows [30]:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \tag{1}$$

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho_{nf}}\frac{\partial P}{\partial x}+v_{nf}\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \tag{2}$$

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}=-\frac{1}{\rho_{nf}}\frac{\partial P}{\partial y}+v_{nf}\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+\beta_{nf}g(T-T_c) \tag{3}$$

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\alpha_{nf}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \tag{4}$$

where the effective density (ρ_{nf}) and heat capacitance $(\rho Cp)_{nf}$ of the nanofluid are defined as [30]:

$$\rho_{nf}=\rho_f(1-\phi)+\rho_s\phi \tag{5}$$

$$(\rho C_P)_{nf}=(\rho C_P)_f(1-\varphi)+(\rho C_P)_s\varphi \tag{6}$$

where φ is the solid volume fraction of nanoparticles. Thermal diffusivity of the nanofluids is

$$\alpha_{nf}=\frac{k_{nf}}{(\rho C_p)_{nf}} \tag{7}$$

and the thermal expansion coefficient of the nanofluid can be determined as

$$\beta_{nf}=\beta_f(1-\varphi)+\beta_s/ \tag{8}$$

The dynamic viscosity of the nanofluid given by Wang and Mujumdar [31] is

$$\mu_{nf}=\frac{\mu_f}{(1-\varphi)^{2.5}} \tag{9}$$

and the effective thermal conductivity of the nanofluid can be approximated by the Maxwell–Garnetts (MG) model as [8]:

$$\frac{k_{nf}}{k_f}=\frac{k_s+2k_f-2\varphi(k_f-k_s)}{k_s+2k_f-\varphi(k_f-k_s)}. \tag{10}$$

3.2 Numerical Procedure

Heat transfer enhancement in the annulus of the nanorefrigerant was investigated numerically by discretizing the governing equations using finite volume. The second order upwind scheme was applied to discretize the terms of the equations. The resultant algebraic equations were solved by iterations. The coupling and strong nonlinearity of the equations needs an under relaxation to ensure convergence. All the variables were calculated right up to the walls without using any wall function. On the wall surface, the boundary values for the stream functions were set to zero. The solution was converged when the relative error between the new and old values of vorticity became less than 10^{-4}, stream function became less than 10^{-4} and temperature fields become less than 10^{-6}. The range of the Rayleigh number Ra, volume fraction of the nanoparticles φ were $3.96\times10^9 \le Ra \le 1.19\times10^{10}$ and $0 \le \varphi \le 0.02$ respectively. The Prandtl number Pr is 5.2. Table 1 shows the thermophysical properties of the nanorefrigerant.

Table 1 thermophysical properties of pure refrigerant and various types of nanofluids at 300K

Thermophysical Properties	R-141b	Al2O3	CuO	ZnO	SiO2
Density, ρ(kg/m³)	1234	3970	6500	5600	2200
Dynamic Viscosity, μ (Ns/m²)	4.09E-04	-	-	-	-
Thermal Conductivity, k (W/m.K)	9.08E-02	36	17.65	13	1.4
Specific Heat, cp (J/kg.K)	1153.8	765	535	495.2	745
Coefficient of Thermal Expansion, β (1/K)	3.34E-03	5.8E-06	4.3E-06	4.3E-06	5.5E-06

4.0 RESULTS AND DISCUSSION

Figure 2 shows the results of the present code for the mean Nusselt number Nu versus the Rayleigh number Ra for the concentric annulus e = 0 are compared with the results obtained by Abu-Nada *et al.* [8] when φ = 0, RR = 2.5 and Pr = 6.2. Further, Figure 3 illustrates the comparison between the present results for Nu with numerical results reported by Shu *et al.* [32] for the eccentric annulus when φ = 0, RR = 2.36, Pr = 0.71 and $Ra=5.3\times10^3$. Experimental results by Kuehn and Goldstein [13] are shown in Figure 4 for comparison with the present results. The governing parameters for this comparison are RR = 2.6, e = 2.6, φ = 0 (regular or Newtonian fluid) and Pr = 0.71. It can be seen from Figures 4–6 that there is a very good agreement between the present results with those earlier reports.

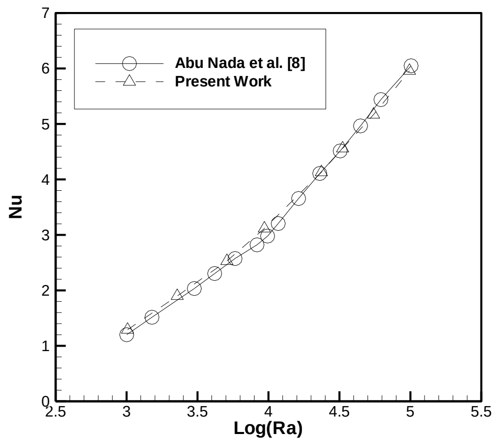


Figure 2 Comparison between the present results for Nu with the numerical results reported by Abu-Nada et al. [8] for the concentric annulus ε = 0 when φ = 0 (regular fluid), RR = 2.5 and Pr = 0.706

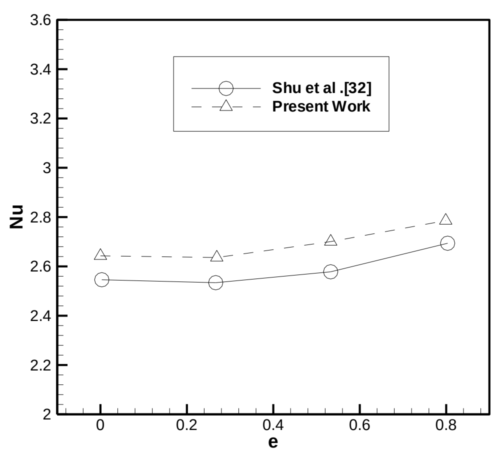


Figure 3 Comparison between the present results for Nu with numerical results reported by Shu et al. [32] for the eccentric annulus (ε = 0) when φ = 0 (regular fluid), RR = 2.36, Ra = 5.3×10^3 and Pr = 0.71

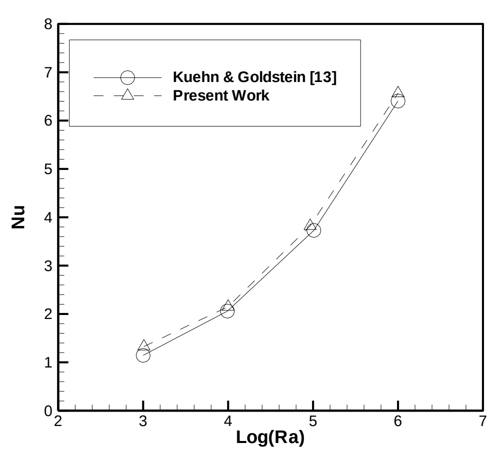


Figure 4 Comparison between the present results for Nu with the experimental results reported by Kuehn and Goldstein [13] for the concentric annulus (ε = 0) when φ = 0 (regular fluid), RR = 2.6 and Pr = 0.71

Nanorefrigerants are proven to enhance the heat transfer characteristics. However, there is no research done on finding which nanorefrigerant is the best among other nanorefrigerants. The effect of using various nanorefrigerants is presented in Figure 5. Al_2O_3 could transfer most energy compared to other nanofluids and refrigerant. Al_2O_3 has the best enhancement followed by ZnO, CuO and SiO_2. This is because Al_2O_3 has the highest thermal conductivity compared to other nanorefrigerants. Thermal conductivity has major role in determining the amount of the enhancement of heat transfer in a working fluid.

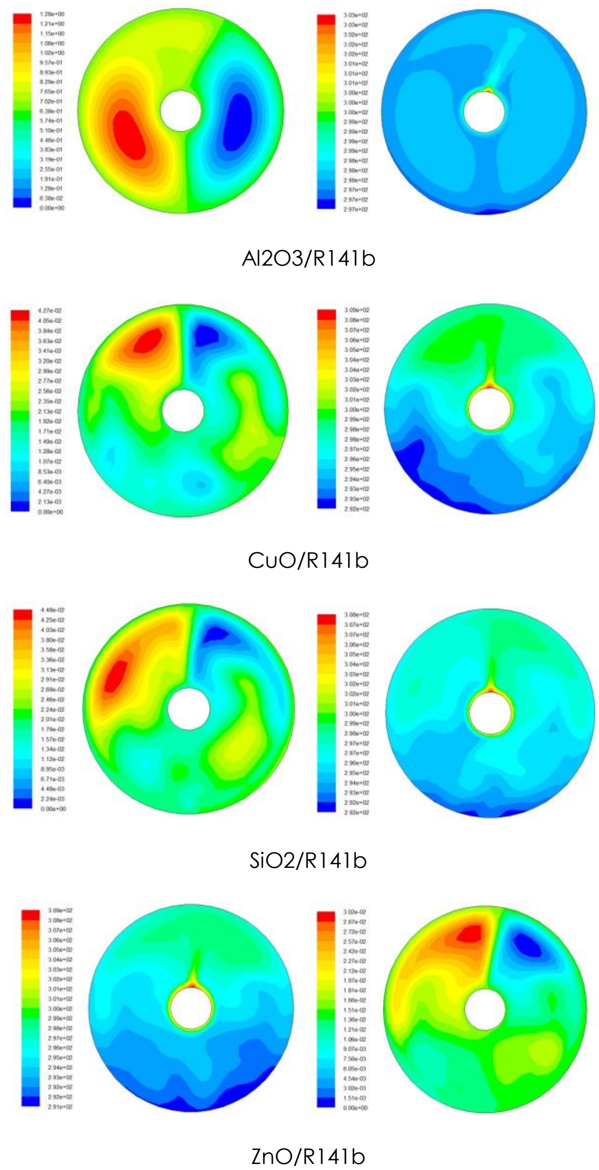
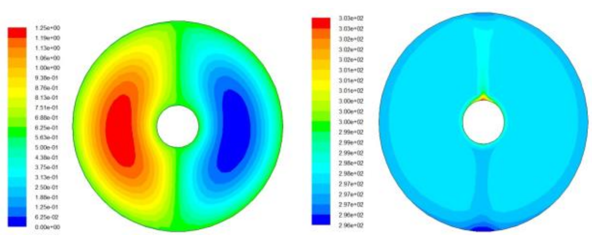
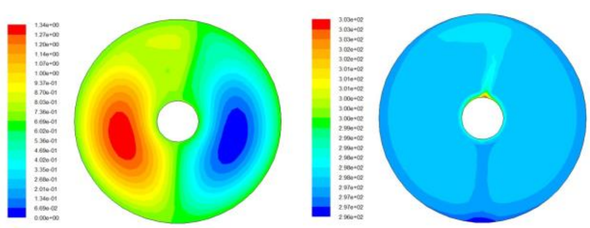


Figure 5 Stream functions and temperature contours of different nanorefrigerant types at φ = 1% and d_p = 20 nm

The effect of nanorefrigerant volume fraction, φ on the thermal field is shown in Figure 6. The range of volume fraction used is 1% ≤ φ ≤ 2%, because based on thermal conductivity and viscosity equations; maximum volume fraction to be used is 2. It can be seen from Figure 6, that the highest concentration of nanoparticles has the highest heat transfer profiles. This is due to the enhanced effective thermal conductivity of the nanofluid, which is accompanied by an increase in the thermal diffusivity. Heat transfer enhancement is increased when volume fraction is increased; it can be observed that 2% volume fraction has the highest heat transfer enhancement, while 1% concentration has the lowest enhancement as shown in Figure 6. This is because the physical properties of nanorefrigerant vary with the volume fraction. Thus, transfers more energy in the fluid, because of the momentum energy is much higher than the thermal energy in higher volume fraction.



φ = 1.5%



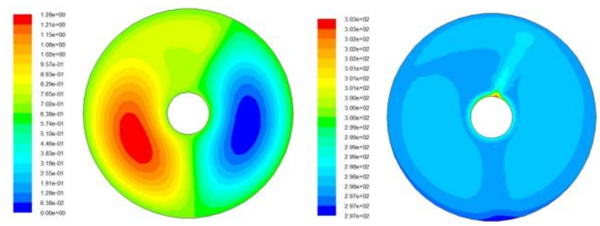
φ = 2%

Figure 6 Stream functions and temperature contours of different nanoparticles concentration

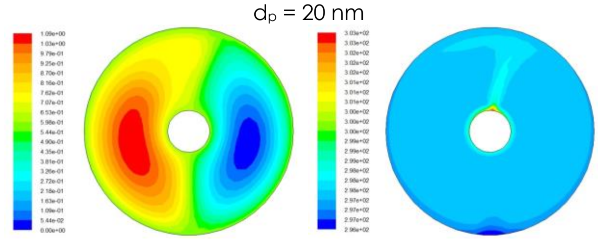
The effects of using different nanoparticles diameters are presented in Figure 7, with Al_2O_3 nanorefrigerant as the working fluid with a volume fraction of φ = 1%. It was observed that particles diameter of 20 nm resulted in the greatest heat transfer distributions, followed by 30 nm. This is because smaller diameter particles have more room for collision, and random movement of microscopic particles suspended in the base fluid allows more collisions with molecules of the surrounding medium, and reduces the wall temperature. Increasing the nanoparticle diameter causes non-uniformity of the particles distributions.

The effects of temperature difference between the hot and cold walls are presented in Figure 8. Three values of temperature difference were studied to examine the effects of Ra. The temperature differences are ΔT = 15, 20 and 25 degrees between the hot and cold walls of the annulus. It can be seen from the contours that the Ra increased when the temperature differences increased.

The effects of tubes eccentricity on the heat transfer performance is discussed in this section. Al_2O_3/R141b at φ = 1% and d_p = 20 nm were used in this study. The eccentricity of the tubes ($e = L/R_o$-R_i) was 0.5 and has four positions at (-X), (+X), (-Y) and (+Y). As shown in Figure 9, eccentricity at the position (+Y) has the best heat transfer characteristics.

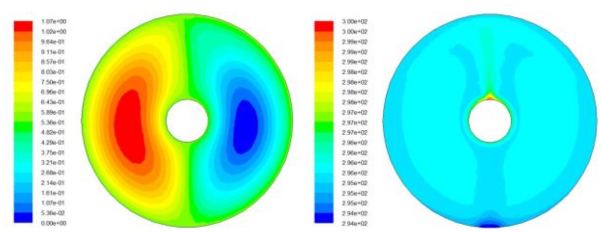


d_p = 20 nm

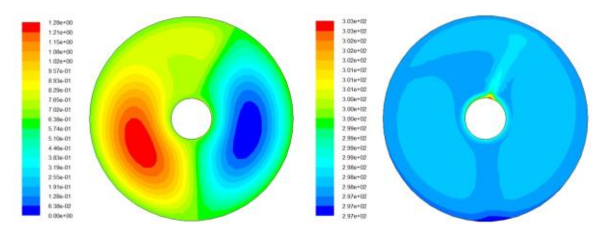


d_p = 30 nm

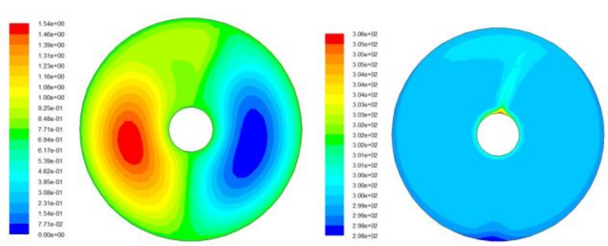
Figure 7 Stream functions and temperature contours of different nanoparticles sizes



ΔT = 15 K



ΔT = 20 K



ΔT = 25 K

Figure 8 Stream functions and temperature contours for various temperature difference for the hot and cold walls

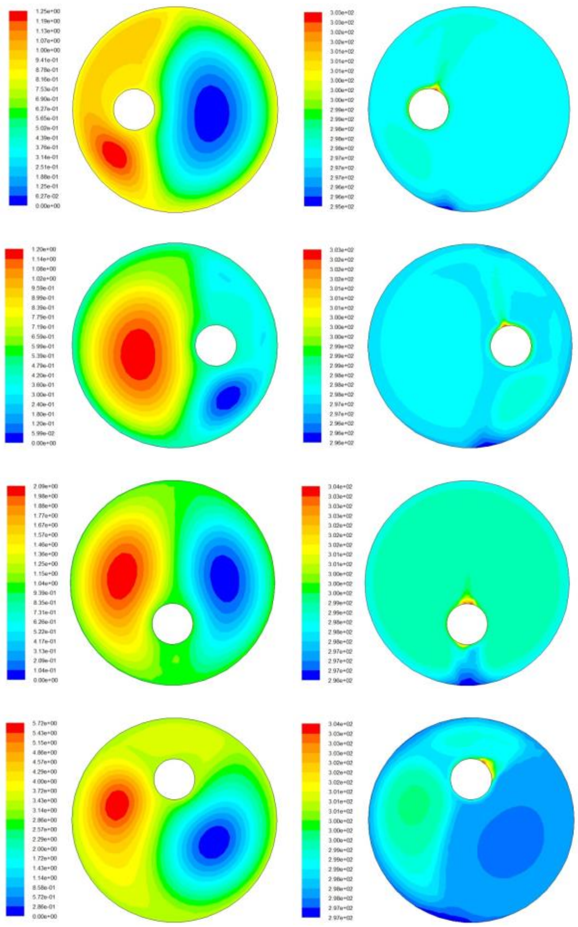


Figure 9 Stream functions and temperature contours for different tube eccentricity positions

The effect of inner cylinder rotation on the heat transfer of the nanorefrigerant was also examined. Al_2O_3/R141b at φ = 1% and d_p = 20 nm was adopted in this study. It can be seen from Figure 10 that the rotation of inner cylinder at 300 R.P.M gives high thermal performance and fluid flow, compared with the stationary case.

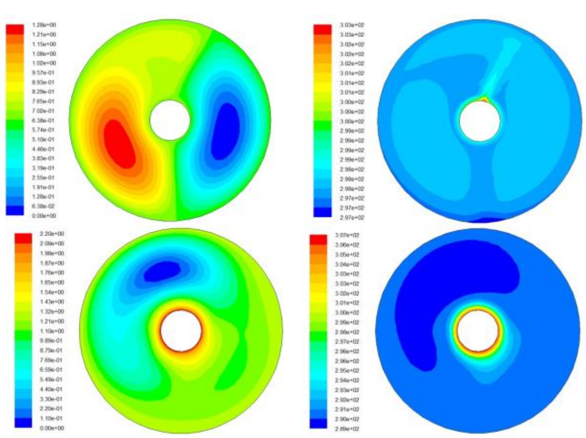


Figure 10 Stream functions and Temperature contours of stationary and rotating inner cylinder

5.0 CONCLUSIONS

Natural convection heat transfer from a heated horizontal concentric cylinder was investigated numerically. CFD analysis was performed as numerical solution in the study. Natural convection flow in annular spaces between horizontal concentric cylinders has been extended to nanoparticle suspensions by simply replacing the thermophysical properties appearing in it with the nano-refrigerant effective properties calculated at the reference average temperature. The main conclusions are summarized as follows:

1. Al_2O_3/R141b gives the best heat transfer enhancement among the types of nanoparticle fluids that were used in this study followed by ZnO, CuO and finally SiO_2. The optimal nanoparticles volume fraction is 2% with 20 nm as a nanoparticle diameter.
2. The heat transfer of the nanorefrigerant is enhanced by increasing the temperature difference between the inner and outer cylinders, according to the increase in Ra numbers.
3. The influence of pipe eccentricity on heat transfer characteristics at different positions is discussed with four positions. Eccentricity of 0.5 at (+Y) shows the best performance of utilizing nanorefrigerants.
4. The rotation of inner cylinder at fixed temperature to result in higher heat transfer and fluid flow profiles.

NOMENCLATURE

FVM	Finite Volume Method
CuO	Copper Oxide
ZnO	Zinc Oxide
NSE	Navier Stokes Equations
T	temperature (K)
Ra	Rayleigh Number, $g\beta\Delta TL^3/\nu a$
K	Thermal Conductivity, W/m.K
ε	Eccentricity
T_h	Inner Cylinder Temperature (K)
Pr	Prandtl number (ν/a)
Al_2O_3	Aluminium Oxide
SiO2	Silicon Oxide
E	Dimensionless eccentricity, e=ε/L
R_o	Outer Radius, mm
R_{in}	Inner Radius, mm
G	gravitational acceleration (m/s^2)
C_p	Specific Heat, kJ/kg.K
L	Distance Between Centers, mm
T_C	Outer Cylinder Temperature (K)

Greek symbols

μ	dynamic viscosity, pa.s
α	thermal diffusivity, m^2/s
ν	kinematic viscosity, m^2/s
ρ	density, kg/m^3
Φ	volume fraction of nanoparticles
β	coefficient of thermal expansion, 1/K

Subscripts

Bf	Base fluid
Nf	Nanofluid
S	Solid
f	Fluid
p	Particles

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