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Study of Nanofluids flow for Heat Transfer in circular micro-channel using Altair® Acusolve

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ABSTRACT: An experimental investigation was conducted to explore the validity of classical correlations based on conventional sized channels for predicting the thermal behavior in single-phase flow through circular microchannels. The test piece was made of copper and experiments were conducted with deionized water. The flow is laminar and turbulent for 0.4 gm/s and 4.0 gm/s inlet mass flow rates respectively with the Reynolds number ranging from approximately 1278 to 12780.The diameter and length of circular micro channel are 0.0005m and 0.1m respectively. The inlet velocity is u (m/s), which is constant over the inlet cross section. The fluid exhausts into the ambient atmosphere which is at a pressure of 1 atm. The computational fluid dynamics (CFD) model equations are solved to predict the hydrodynamic and thermal behaviour of the exchanger. The geometry of the problem and meshing of it have been made in Altair HyperWorks® Workbench. The models have been solved by AcuSolve solver. The utility of Nanomaterial as a heat enhancer has been justified by studying a circular microchannel thermal behaviour. Water and its nanofluids with alumina (Al_2O_3) are used as the coolant fluid in the microchannel heat sink. The present CFD calculated heat transfer coefficient values have compared with the analytical values and very close agreement is observed. The result shows that Nanofluids help to increase the heat transfer coefficient by 15% and 10% respectively in laminar and turbulent zone. Thus use of Nanofluids has been found beneficial both in laminar and turbulent zone. The relation between heat transfer coefficient and thermal conductivity of the fluid i.e. h α kis proved in the present study.

The entrance length for the fully developed velocities depends on Reynolds number. The temperature rise between outlet and inlet depends on the Reynolds number, Re and Peclet number, Pe. Temperature distribution is found to be independent of radial position even for Pe<<1.0. The hydrodynamic and thermal behaviour of the system have been studied in terms of velocity, pressure and temperature contours. The velocity contours at the exit show that wall effect penetrates more towards the center and the thickness of the zone with maximum velocity shrinks with increase in Re. The pressure drop across the channel increases with increase in Re. In all the cases, a closed form solution is obtained between temperature, Nusselt number, thermal behaviour and sensitive parameter. The result of the present analysis has been computationally compared with earlier numerical and analytical results. A good agreement has been obtained between the present prediction and the available results. The experimental work done by Lee and Mudawar (2007) [1] has been predicted by the present CFD results. The hydrodynamics and thermal behaviour of a circular microchannel are studied here. The variation wall temperature, pressure drop in the channel and the friction factors calculated using Altair HyperWorks® can well predict the experimental data. The effect of Re on the behaviour the channel are also studied. Its behaviour also has been analyzed with the help of temperature, pressure and velocity contours.

Keywords: micro-channels, heat exchangers, nanoparticles, nanofluids, CFD, hetransfer coefficient, pressure drop, friction factor.

NOMENCLATURE

General Terms

p = Perimeter, [μm]
w = Width of the microchannel, [μm]
L = Length of the microchannel, [μm]
h = Conv. heat transfer coeff., [W/m^2-k]
k = Thermal conductivity, [W/m-k]
Q = Heat transfer rate, [W]
R= Radius of microchannel, [μm]
m= Mass of the object, [kg]
v = Volume of the object, [m^3]

D = Diameter of the microchannel, [μm]
$\dot{m}$ = Mass flow rate, [kg/s]
f = friction factor
V = Velocity of nenofluid, [m/s]
dT = Temperature difference,[k]
dx = Axial difference, [mm]
q″ = Heat flux, [W/mm^2]
Nu= Nusselt Number
Pr= Prandtl Number
Re= Reynold`s Number
C_p= Specific heat, [kJ/kg-s]
V_x= velocity in x direction [m/s]
V_r = velocity in radialdirection, [m/s]
T_m = Mean temperature, [k]
T_{mi} = Mean temperature at inlet, [k]
T_{mo}= Mean temperature at outlet, [k]
T_s = Surface temperature, [k]
Δp= Pressure difference,[bar]
C_v = Specific heat volume, [kJ/kg-s]
Pe.= Peclet number,
U = dimensionless velocity in x-coordinate
V = dimensionless velocity in y-coordinate
(x , r, ø) = Coordinate

Greek Letters
μ = Dynamic viscosity,[kg/m-s]
ρ = Density of the object, [kg/m^3]
α = Thermal diffusivity, [m^2/s]
υ = Kinematic viscosity, [m^2/s]
θ = Temperature difference, [k]

Subscripts
conv = convection
ch = channel
sp = single phase
bot = bottom
f = fluid
i = inlet
o = outlet

Micro-channel Heat transfer has the very potential of wide applications in cooling high power density microchips in the CPU system, the micro power systems and even many other large scale thermal systems requiring effective cooling capacity. This is a result of the micro size of the cooling system which not only significantly reduces the weight load, but also enhances the capability to remove much greater amount of heat than any of large scale cooling systems. It has been recognized that for flow in a large scale channel, the heat transfer Nusselt number, which is defined as hD/k, is a constant in the thermally developed region where h is the convective heat transfer coefficient, k is thermal conductivity of the fluid and D is the diameter of the channel. One can expect that as the size of the channel decrease, the value of convective heat transfer coefficient, h, becomes increasing in order to maintain a constant value of the Nusselt number. As the size of the channel reduces to micron or nano size, the heat transfer coefficient can increase thousand or million times the original value. This can drastically increase the heat transfer and has generated much of the interest to study micro-channel heat transfer both experimentally and theoretically

Kandlikar et al. [3] as well as Mehendale et al. [4] propose the following classification of different heat exchangers, based on empirical studies, Conventional channels D_h > 3mm, Mini-channels 200μm<D_h<3mm, Micro-channels 10μm < D_h < 200μm. A simpler classification was proposed by Obot (2003) [5] based on the hydraulic diameter rather than the smallest channel dimension. Obot classified channels of hydraulic diameter under 1 mm (D_h< 1 mm) as micro-channels, which was also adopted by many other researchers such as Bahrami and Jovanovich (2006) [6], Bahrami et al. (2006) [7] and Bayraktar and Pidugu (2006) [8].This definition is considered to be more appropriate for the purposes of this thesis. The higher volumetric heat transfer densities require advanced manufacturing techniques and lead to more complex manifold designs (Kandlikar et al. 2006) [9]. Maxwell [10] initiated a novel concept of dispersing solid particles in base fluids to break the fundamental limit of HTFs having low thermal conductivities. Most of these earlier studies on this concept used millimeter or micrometer solid particles, which led to major problems such as rapid settling of the solid spherical particles in the fluids, clogging in micro-channels and surface abrasion. In addition, the high pressure drop caused by these particles limited their practical applications.

SPECIFICATION OF PROBLEM

Consider a steady state fluid flowing through a circular micro channel of constant cross-section as shown in Fig.4.1 (Lee and Mudawar, 2007). The diameter and length of circular micro channel are 0.0005 m and 0.1 m respectively. The inlet velocity is u (m/s), which is constant over the inlet cross-section. The fluid exhausts into the ambient atmosphere which is at a pressure of 1 atm. The temperature at inlet is 30^0C and mass flow is 0.4 g/s for (laminar flow) and 4.0 g/s for (turbulent flow).There is constant heat flux over the surface of circular tube is 100 W/m^2 .

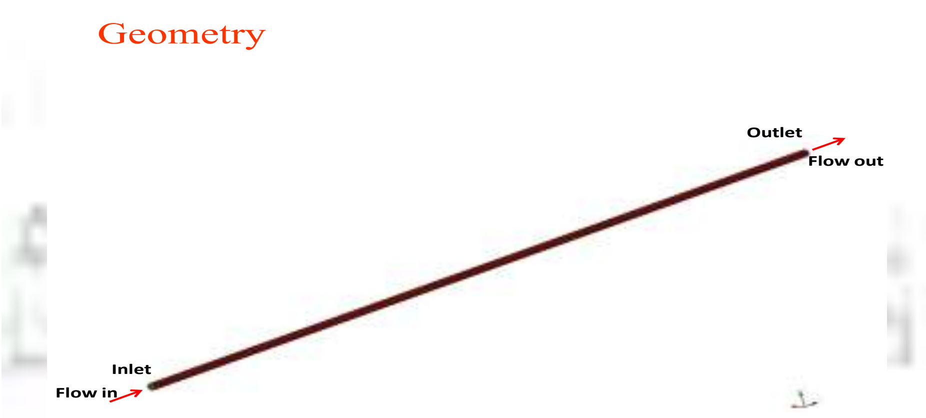

Fig 1: Problem Geometry

BOUNDARY CONDITIONS

A fluid of uniform velocity entering a tube is retarded near the wall and a boundary layer being to develop. The thickness of the boundary layer is limited to the pipe radius because of the flow being within a confined passage. Boundary layer from the pipe wall meet at the center of the pipe and entire flow acquires the characteristics of a boundary layer. A no slip boundary condition was assigned for the non porous wall surfaces, where bothvelocity components were set to zero at that boundary i.e. $\upsilon_x = \upsilon_r = 0$. A constant heatflux (100 W/m^2) is applied on the channel wall. Axis symmetry was assigned atcenterline. A uniform mass flow inlet and a constant inlet temperature were assigned atthe channel inlet. At the exit, pressure was specified.

Tube Flow Energy Balance

Consider a circular tube of length L, radius R (diameter D), and coordinates (r, φ, x) as shown in Figure 1 Now consider the control volume (shaded region) shown in Figure 2.1 through which fluid flows and heat is convected through the pipe walls. The energy balance of this control volume is given by

$$\dot{m}(C_V T_m + p U_x) + dq_{conv} = [\dot{m}(C_V T_m + p U_x) + \dot{m}\frac{d}{dx}(C_V T_m + p U_x)\, dx \quad \text{……. (1)}$$

Where $dq_{conv} = \dot{m} c_p d\ T_m$ for both ideal gases and incompressible liquids.

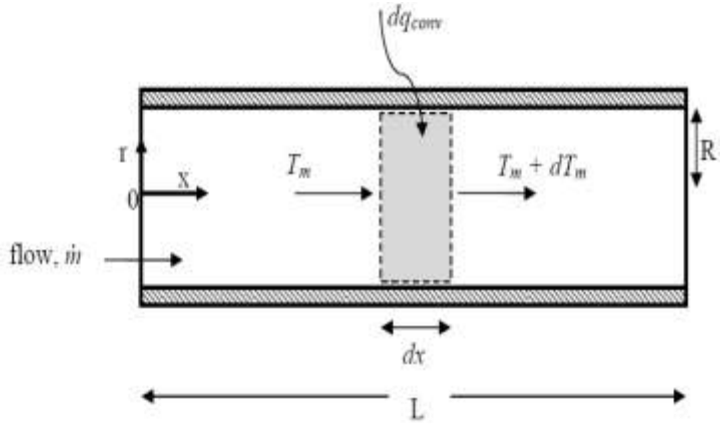

Figure 2: Tube flow control volume

Eq. (1) reduces to

$\frac{d}{dx}(q_{conv}) = \dot{m}\frac{d}{dx}(c_v T_m)$ ....(2)

Through integration over the length of the tube, the heat applied to the control volume through convection is found to be $q_{conv}=\dot{m}c_p\,(T_{out}-T_{in})$Eq. (3)
Note also that through the use of Newton's law of cooling,
$q'' = h\,(T_s-T_m)$ and the fact that

$dq_{conv}= q_s''\,Pdx$, ...(3)
can be converted to

$\frac{dT_m}{dx} = \frac{P}{\dot{m}c_p}(T_S - T_m)$ .... (4)

Constant Surface Heat Flux

For the constant heat flux condition, Eq. (4) can be integrated to an arbitrary distance x using $q_{conv} = q_s''PL$ to obtain

$T_m(x)=T_{mi}+\frac{q_s''P}{\dot{m}c_p}x$ ....(5) This implies that the mean fluid temperature varies linearly with axial distance in a tube. Also, the heat transfer coefficient for the constant surface heat flux condition is given by Newton's lawof cooling:

$q_s'' = h\,(T_s-T_m)$ ....(6)

This can be further adapted to the tube situation by using the linearity of the mean fluid Temperature along the tube length to obtain

$q_s'' = h\,(T_s - \frac{T_{mi}+T_{mo}}{2})$...........(7)

It is important to note that the surface heat flux in a channel with a constant heat flux on the wall does not have a constant surface temperature. From Eq. (7), the varying Nusselt number can be substituted to obtain

$T_S(x) - T_m(x) = \frac{q_s''}{Nu_x}\times\frac{D}{K}$ ....(8)

If Nu_x is constant (i.e. fully developed conditions), then the following can be shown

$\frac{dT_m}{dx} = \frac{dT_x}{dx}$..........................(9)

Thus, the temperature of the wall varies linearly and parallel to the mean fluid temperature in thefully developed region, and in the developing region the wall temperature varies according to Eq.(8) (This means that it is nonlinear in this region)

MATERIAL PROPERTIES

Pure water is used as base working fluid and Alumina (Al_2O_3) is taken as nanoparticles. The density, heat capacity and thermal conductivity of alumina are 3,600 kg/m^3, 765 J/kgK and 36 W/mK respectively. The properties of nanofluids (nf) are given in Table 1 at 30^{o}C temperature and 100 kPa pressure. **Table 1:** Water base fluid properties with different concentration of alumina nanoparticles (Lee and Mudawar, 2007)[1] given below:

	Ø = 0 %	Ø=1 %	Ø =2%	Ø= 3%	Ø =4%	Ø =5%
k_{nf}	**0.603**	**0.620**	**0.638**	**0.656**	**0.675**	**0.693**
ρ_{nf}	**995.7**	**1021.7**	**1047.7**	**1073.8**	**1099.8**	**1125.9**
μ_{nf}	**79700**	**81700**	**83760**	**85760**	**87750**	**89740**
C_{pnf}	**4.183**	**4.149**	**4.115**	**4.081**	**4.046**	**4.012**

RESULTS AND DISCUSSIONS

The hydrodynamics behaviour of the channel can be studied in terms velocity distribution within the channel. The value of Re is 1278.0 for inlet mass flow rate 0.4 gm/s and it becomes 12780.0 for 4.0 gm/s inlet mass flow rate. Thus the flow is laminar and turbulent for 0.4 gm/s and 4.0 gm/s inlet mass flow rates respectively. The variation of centerline velocity at laminar state with axial position (X) for water and its nanofluid are displayed in Fig. 3. The figure shows that for all the fluids the entrance lengths i.e. the length required to reach fully developed state are the same. The density and viscosity of nanofluids increases with increase in the nanoparticles concentration in water. Thus, the velocity at any axial position decreases with increase in the nanoparticles concentration as found in Fig. 3. It is also observed here that the axial velocity decreases with increase in nanofluid concentrations. Unlike laminar flow, the entrance lengths are found different in turbulent flow condition.

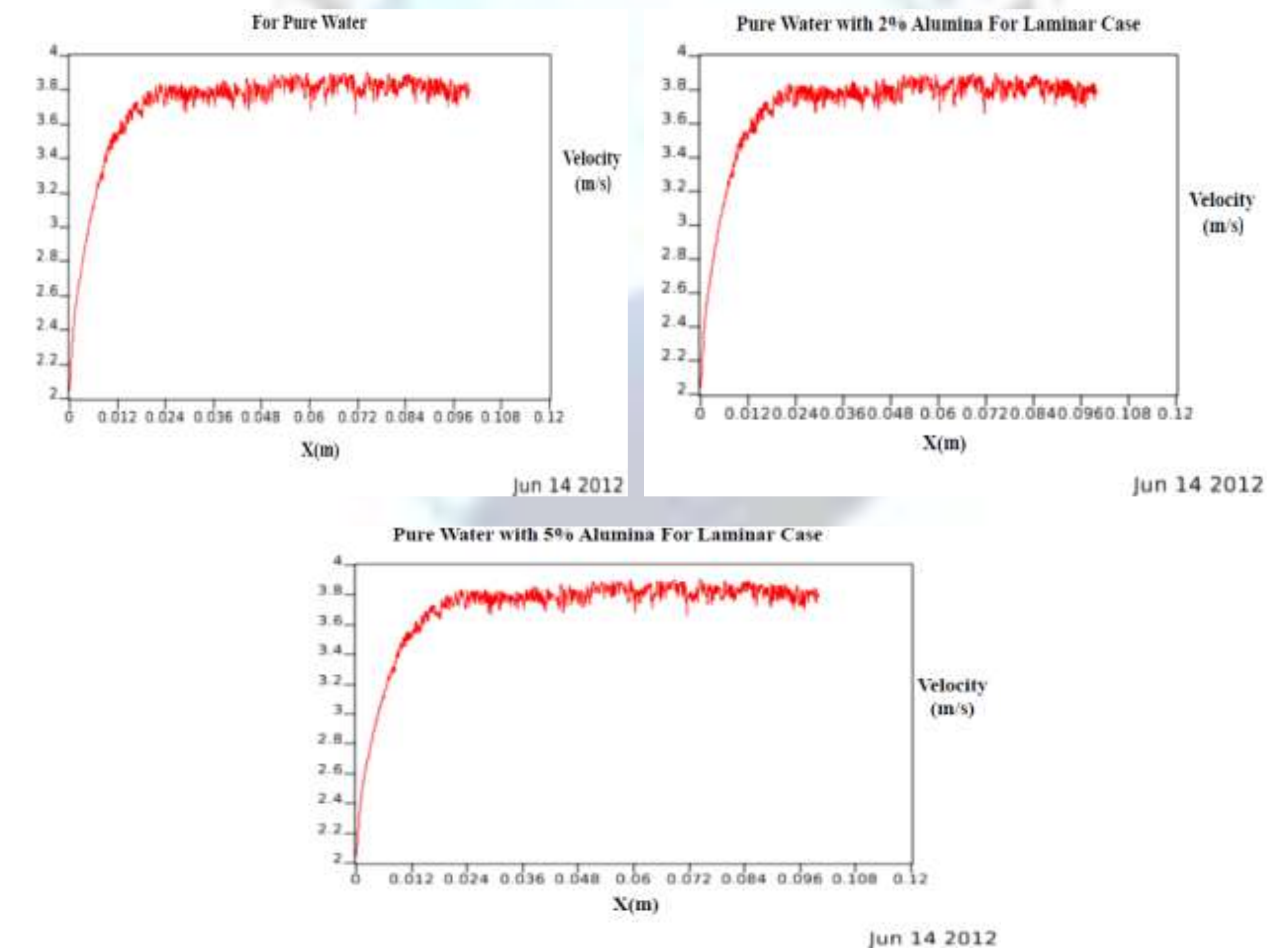

Figure 3: Velocity profile at centerline in the circular micro channel at Re =1278. Pure Water =5.53, $Pr_{2\%}$ = 5.40, $Pr_{5\%}$ = 5.20

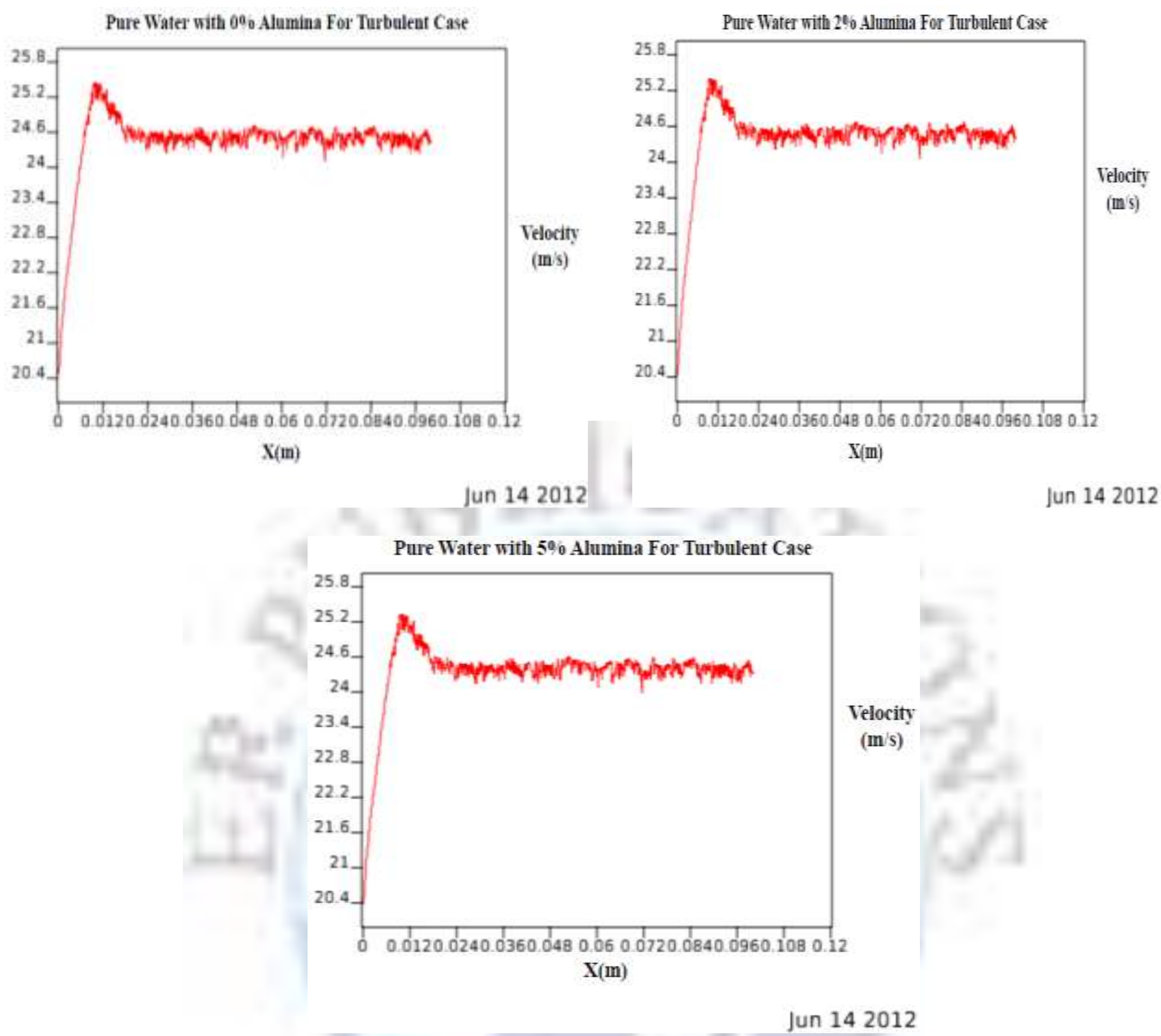

Figure 4: Velocity profile at centerline in the circular micro channel at Re =12780.Pr_{Water} =5.53, $Pr_{2\%}$ =5.40, $Pr_{5\%}$ =5.20

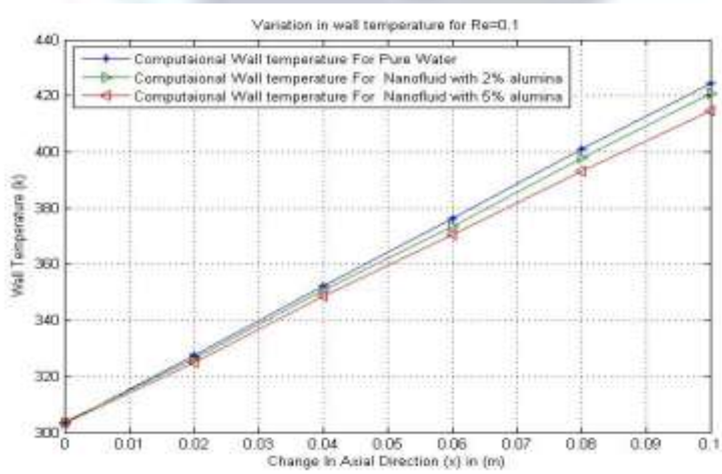

Figure 5.1: Variation wall temperature for Re = 0.1 in circular micro channel for water and its nanofluid

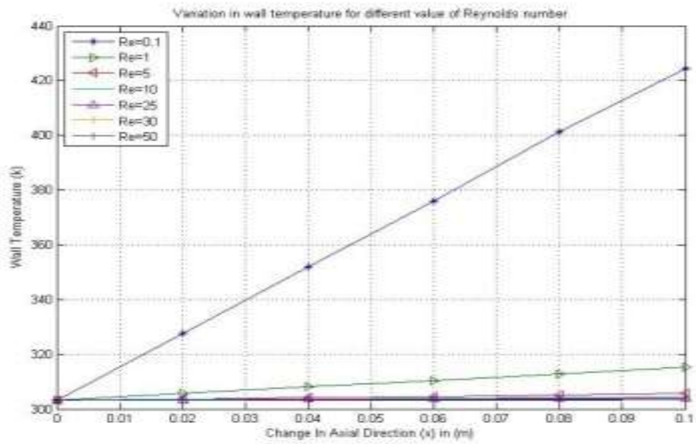

Figure 5.2: Variation of wall temperature with axial distant for different Reynolds number and Peclet number. Water is used as the fluid in the heat exchanger

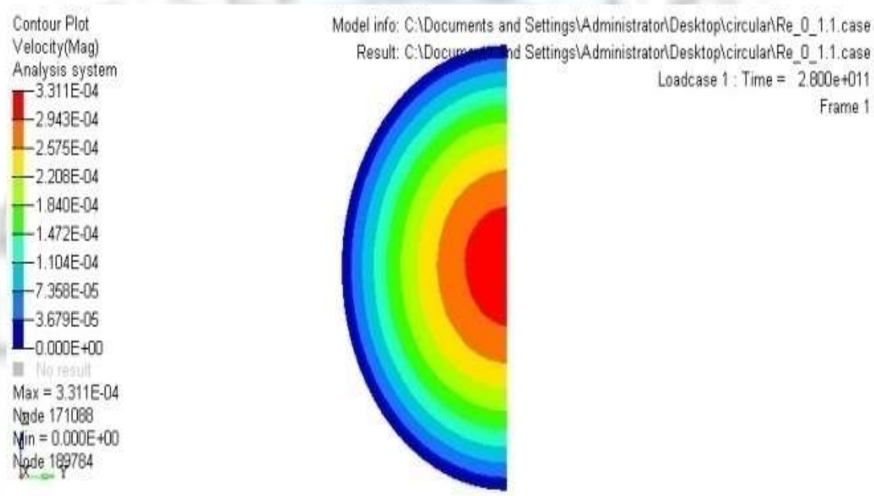

Fig. 6: Water velocity contour plot at Re = 0.1 for water

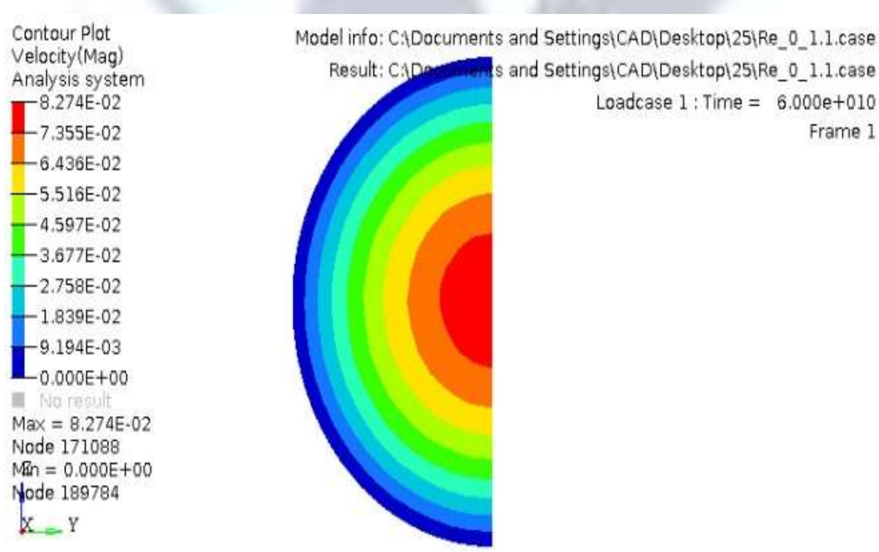

Fig. 7: Water velocity contour plot at Re = 25 for water

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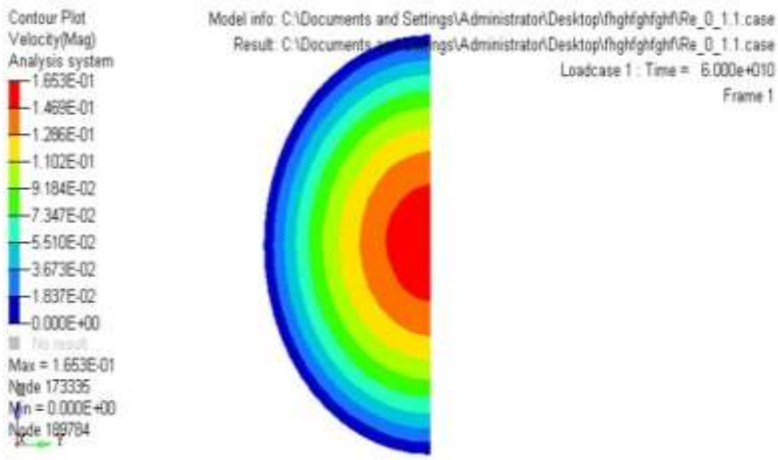

Fig. 8: Water velocity contour plot at Re = 50 for water

Temperature contour plots are shown in Figs. 9 to 11 for Re equal to 0.1 and 50 respectively. The contours support the different temperature profiles as discussed before. These also show that there is no variation of temperature in the radial direction.

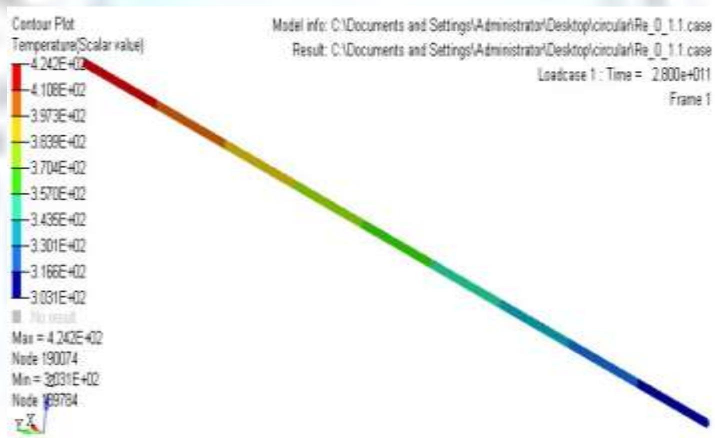

Fig.9: Temperature contour plot at Re = 0.1 for water

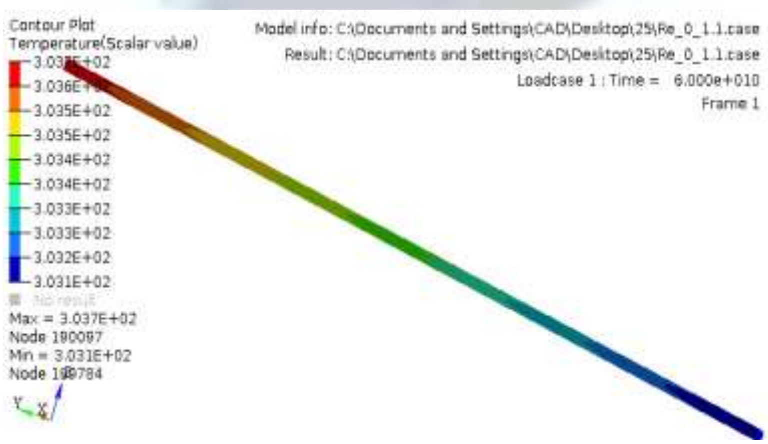

Fig. 10: Temperature contour plot at Re= 25 for water

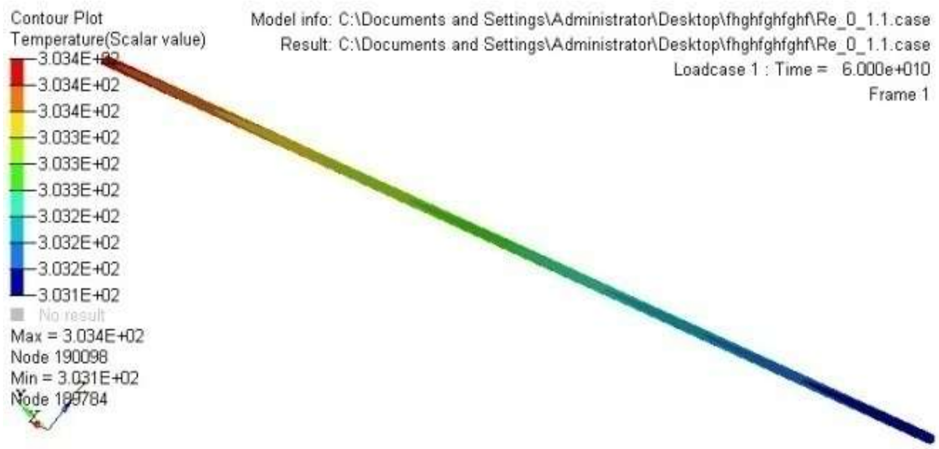

Fig. 11: Temperature contour plot at Re = 50 for water

Pressure contour plots at different Reynolds no. for water are shown Figs. 12 to 14. From pressure contour it is clear that as Reynolds number. Increases pressure drop increases. A linear variation of pressure is with axial distance at all Re. The same is observed with all nanofluids.

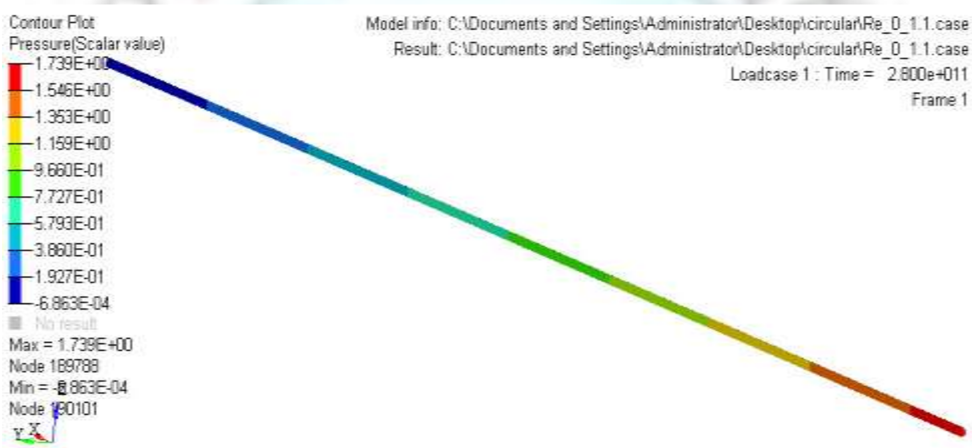

Fig. 12: Pressure contour plots Re = 0.1 for water

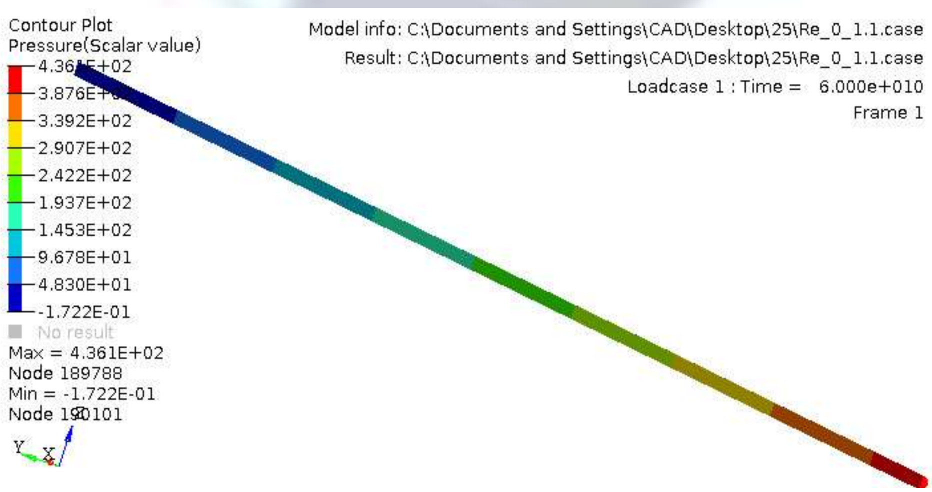

Fig. 13: Pressure contour plots Re = 25 for water

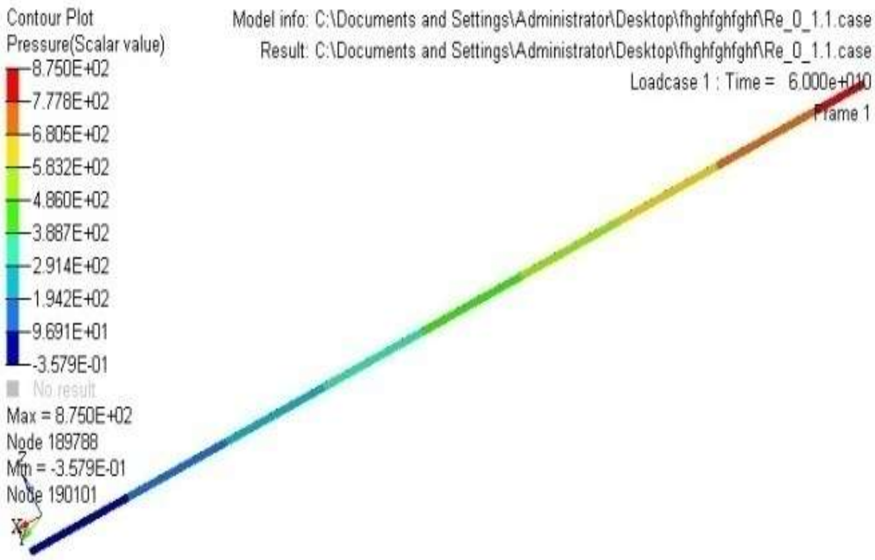

Fig. 14: Pressure contour plots Re = 50 for water

CONCLUSIONS

The computational results successfully validated the analytical data for circular micro channel.Heat transfer coefficient is constant throughout the circular micro channel due to its fully developed conditions.As the concentration of nanoparticle increases heat transfer coefficient also increases.The enhancement of heat transfer in laminar flow is greater as compared to turbulent flow and use of nanoparticles both in laminar and turbulent conditions are found beneficial.Wall temperature increase within the flow direction of circular micro channel at very low Re.Wall temperature has negligible variation for higher Reynolds due to greater value of Peclet no.Pressure and temperature contours represent successfully the hydrodynamic and thermal behaviour of the system.

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