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# Analysis of Heat Transfer in Rectangular microchannel for Nanofluids flow 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 Rectangular microchannel [6]. The test piece was made of copper and experiments were conducted with deionized water. The experimental work which was performed by Lee and Mudawar, (2007) [1]on the test rigis simulated in the present study. Fluid is flowing through a rectangular micro channel embedded in a test module. There are 21 parallel rectangular micro-channels in themodule. The dimension of each micro channel is 215 μmwidth, 821 μm depth and4.48 cm length. The inlet velocity is u (m/s). The micro channel is made of oxygen free copper. The heating is provided by 12 cartridge heaters that are embedded in underside oftest module. The top surface of micro channel is subjected to adiabatic conditions. Operating conditions for the study are as follows: The operating range of number based on the hydraulic diameter of the channel, $Re_{Dh}$=140−941, the power input range to the channel, Q = 100 − 300W, the inlet temperature of fluid to the channel, $T_{in}=30^0C$, the range of inlet pressure, $P_{in}$= 1.17 −1.36bar, and the output pressure, $P_{out}$= 1.12 bar. 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 (Al2O3) are used as the coolant fluid in the microchannel heat sink.[5] 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 k is 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 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 xdirection [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]
$\Delta p$ = 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**

$\mu$= Dynamic viscosity,[kg/m-s]
$\rho$ = Density of the object, [$kg/m^3$]
$\alpha$ =Thermal diffusivity, [$m^2/s$]
$\upsilon$=Kinematic viscosity, [$m^2/s$]
$\theta$ = 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 nanosize, 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.

**Energy Balance for Nanofluids flow in Rectangular Microchannel:**

Consider a circular tube of length L, radius and

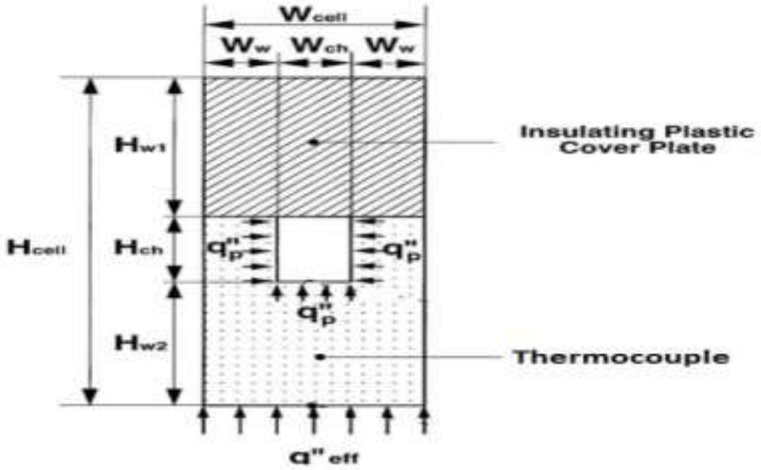


**Fig .1 Problem Diagram**

coordinates ( x,y,z) as shown in above Figure .Now consider the control volume (shaded region) shown in Figure.1 through which fluid flows and heat is convected through the pipe walls. The energy balance of this control volume is given by

$$\dot{m}(C_V T_m + pU_x) + dq_{conv} = [\dot{m}(C_V\, T_m + pU_x) + \dot{m}\frac{d}{dx}\,(C_V\, T_m + pU_x)\, dx\,] \quad \ldots\ldots\ldots (1)$$

Where $dq_{conv}=\dot{m}c_p d\ \ T_m$for both ideal gases and incompressible liquids.Eq. (1) reduces to

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

Through integration over the length of the tube, the heat applied to the control volume through convection is found to be

$$q_{conv}=\dot{m}c_p(T_{out}-T_{in}) \ \ldots\ldots\ldots\ldots(3)$$

Note also that through the use of Newton's law of cooling, $q'' = h\,(T_s-T_m)$ and the fact that $dq_{conv}=q_s''Pdx$, Eq. (3) can be converted to

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

### Constant Surface Heat Flux

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

$$T_m(x) = T_{mi} + \frac{q_s''P}{\dot{m}c_p}x\ldots\ldots\ldots\ldots\ldots. (5)$$

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

$$q_S'' = h\,(T_s – T_m)\ldots\ldots\ldots.................... (6)$$

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

$$q_S'' = h(T_s – \frac{T_{mi} + T_{mo}}{2}) \ \ldots\ldots\ldots.(7).$$

It is important to note that the surface heat flux in a channel with a constant heatflux onthe wall does not have a constant surface temperature. From Eq. (7), the varying Nusseltnumber 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

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

Thus, the temperature of the wall varies linearly and parallel to the mean fluid temperature in the fully developed region, and in the developing region the wall temperature varies according to Eq.(8) (This means that it is nonlinear in this region)A single phase micro channel friction factor is determined from the pressure drop acrossthe channel[1][2]

$$\Delta P_{chf} = \frac{\Delta P_{ch}}{2Lv^2}\frac{D_h}{\rho_f}$$

The heat flux for unit cell is given equation ( Qu and Mudawar 2007)[1]

$$q''_{eff} = \frac{P_w}{NA_{bot}}$$

Where N is number of micro channel. $A_{bot} = W_{Cell} \times L$ is the bottom area of unit cell microChannel.The heat flux to channel is given by (Qu and Mudawar 2004)[1]

$$q''_p = \frac{q''_{eff} W_{cell}}{W_{ch} + 2H_{ch}}$$

Single phase heat transfer coefficient along micro channel defined as:

$$h_{sp} = \frac{q''_p}{T_w - T_m}$$

Where $T_w$is the wall temperature, and mean fluid temperature is evaluated by energy balance around the channel.

## BOUNDARY CONDITIONS

A no slip boundary condition was assigned for the surfaces, where both velocity components were set to zero at that boundary i.e. u = υ = 0. A uniform velocity inlet and a constant inlet temperature were assigned at the channel inlet. At the exit, Pressure was specified. At all wall surfaces heat flux was assigned. All the sink surfaces subjected to adiabatic conditions (heat flux is zero except bottom sink). Effective heat flux is assignedto channel bottom, channel left and channel right .Channel top is also subjected adiabatic conditions. The Continuum is assigned as solid in heat sink and fluid in channel.

## RESULTS AND DISCUSSIONS

The variation of velocity of water and its nanofluid (1% alumina and 2% alumina) with axial position (x) at Re=140 is shown in Fig. 2 Velocity is at center line as indicted in figure 2. It is observed almost at the entrance velocity of all types of fluids got fully developed. The entrance length of all the fluids also found to be same.

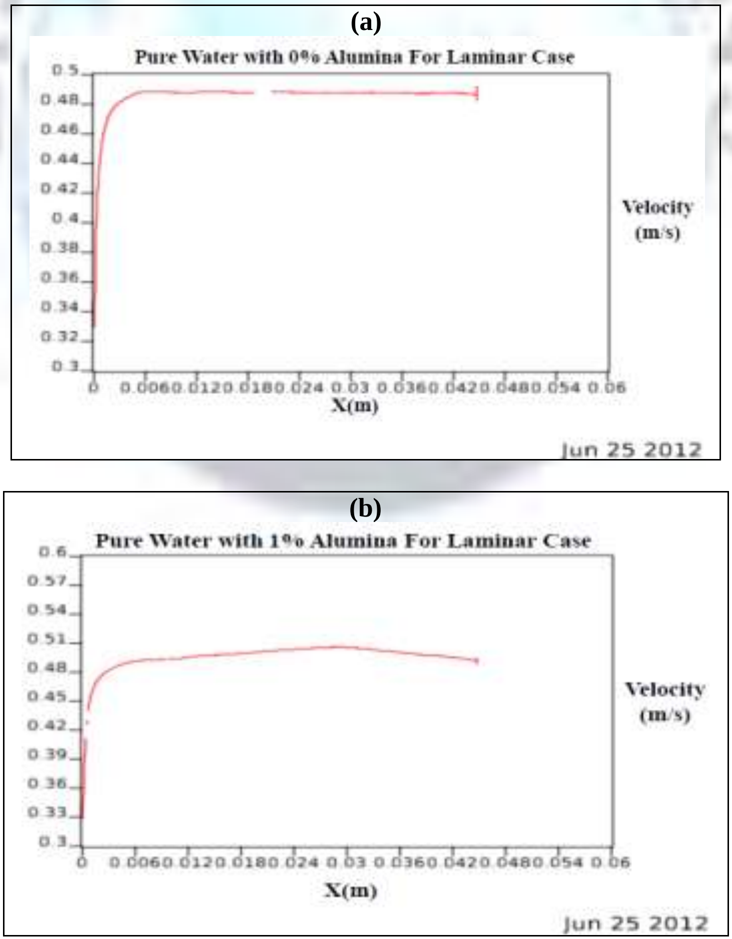


**Fig.2: Velocity profile for (a) Pure Water (b) Nanofluid with 1% Alumina Re=140**

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The hydrodynamic and thermal behaviour are also studied in terms of contours of velocity, pressure and temperature on a surface passes through the centerline in the flow direction of the channel. The velocity contours at different Re using water as the coolant are shown in Figs. 8.8 to 8.10. The figure shows that the entrance length increases with increases in Re.

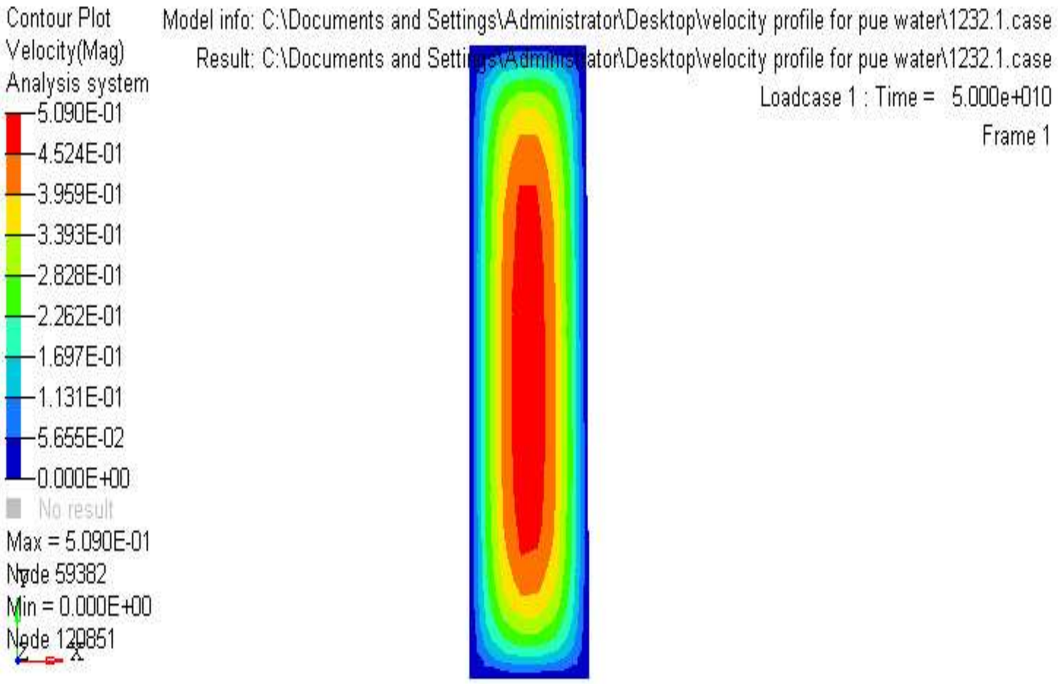


**Fig. 3: Velocity Contour Plot of water at Re = 140**

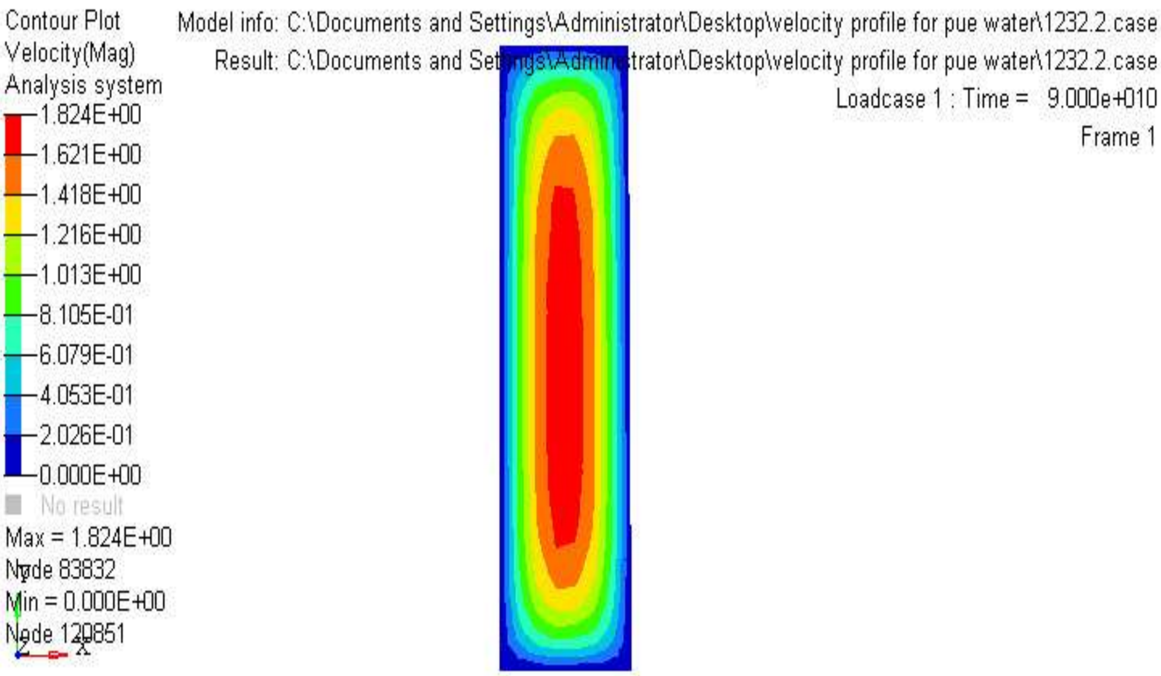


**Fig. 4: Velocity Contour Plot of water at Re = 500**

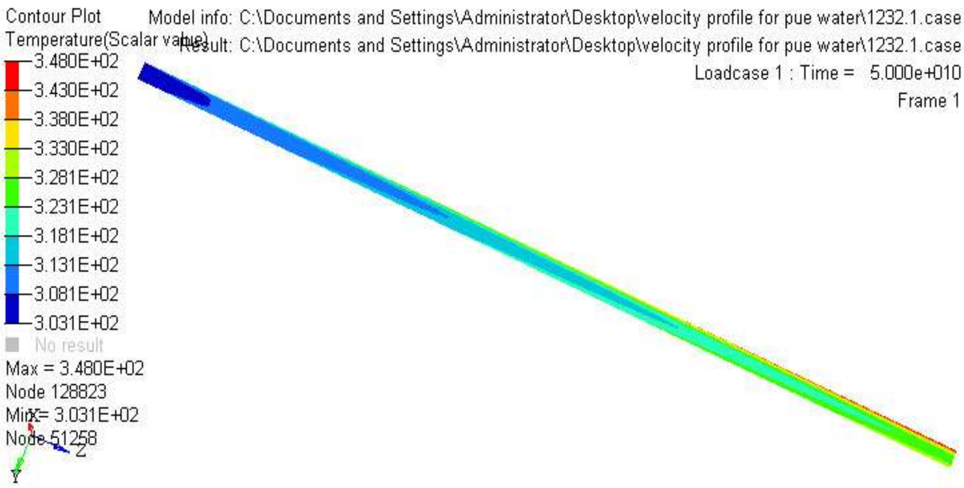


**Fig. 5: Temp. Contour plot for water at Re = 140**

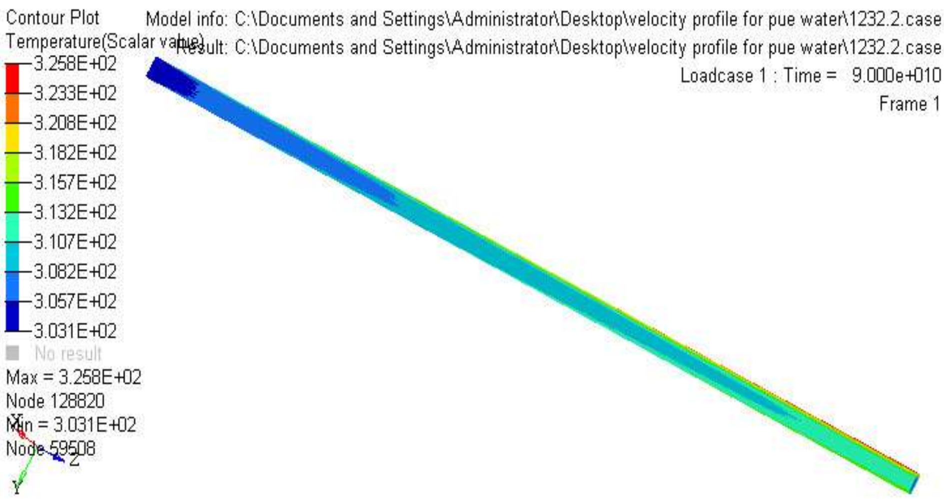


**Fig. 6: Temp. Contour plot for water at Re = 500**

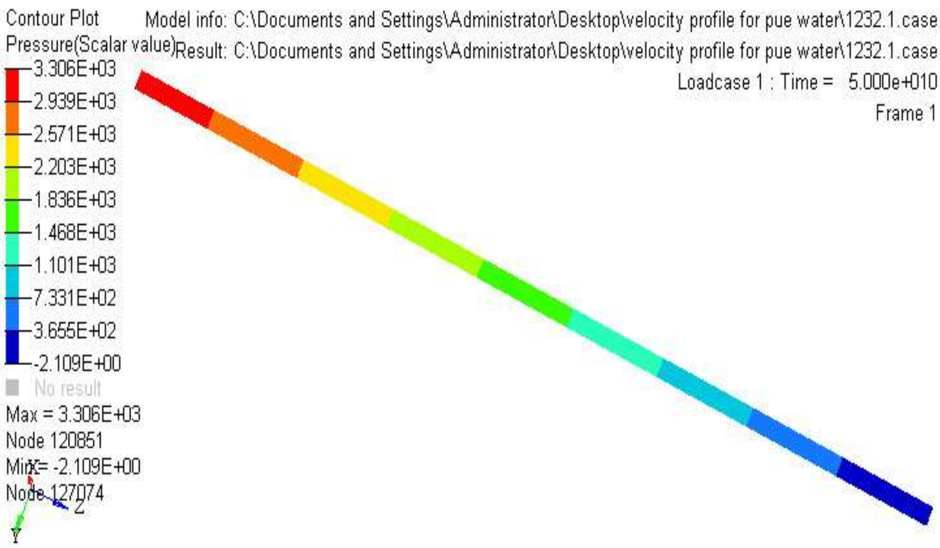


**Fig. 7: Pressure Contour Plot of water at Re = 140**

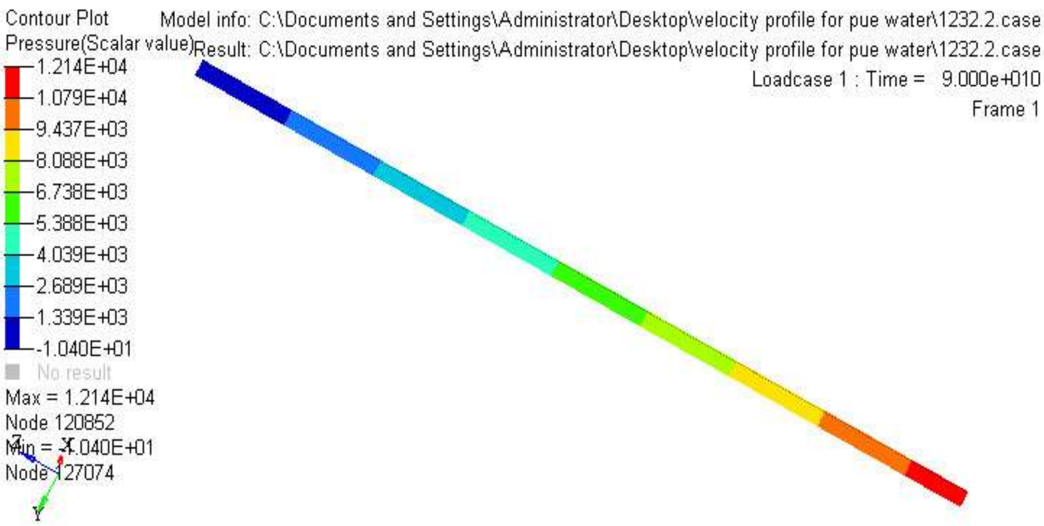


**Fig. 8: Pressure Contour Plot of water at Re = 500**

## CONCLUSION

Computational result successfully validated the test data in terms of wall temperature distributions, pressure drop of the channel and friction factor. Pressure drop increases as Reynolds number increases. Increasing nanoparticle concentration increases single- phase pressure drop compared to pure fluids at the same Reynolds number Greater heat transfer coefficient is obtained at micro channel entrance. Wall temperature increases from entry region of micro channel to exit region.

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