# HEAT AND MASS TRANSFER STUDIES IN A NOVEL DEVICE COILED FLOW INVERTER

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by

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# DEDICATION

*To my beloved mother, brother and sisters,*

*To my dear wife,*

*And to the soul of my dear father.*

# CERTIFICATE

This is to certify that the thesis entitled, '***Heat and Mass Transfer Studies in a Novel Device Coiled Flow Inverter'*** being submitted by ***Mr. Jogender Singh*** to the Indian Institute of Technology, Delhi for award of Doctor of Philosophy is a record of bonafide research work carried out by him under my guidance and supervision in conformity with the rules and regulations of Indian Institute of Technology, Delhi.

The research report and results presented in this thesis have not been submitted, in part or full, to any other university or institute for the award of any degree or diploma.

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# ABSTRACT

The process intensification of mixing, heat and mass transfer process is the subject of much intensive research work. The term process intensification refers to a set of innovative principles in process and equipment design, which can bring significant advancement in terms of process efficiency, safety and cost. Process equipment designs must satisfy several criteria such as; the equipment must be compact in size and have high efficiency with low energy losses. It is becoming increasingly evident that the complex nature of the primary and secondary flow patterns due to centrifugal forces acting on fluid flowing in coiled tube gives some advantages over straight tubes. Some characteristic features of the flow in curved tube are lower axial dispersion, better radial mixing, heat transfer and mass transfer. Because of the above advantages, coiled tubes are frequently encountered in practical applications, such as heat exchangers, chemical reactors and piping systems. The different curved geometries including T-bend, U-bend, spiral coil and helical coil have been considered, to utilize lower axial dispersion, better radial mixing and augmented heat and mass transfer coefficients.

A novel device coiled flow inverter (CFI) offers multiple flow inversion, in addition to the secondary flow regime and hence significantly augments the radial mixing and heat and mass transfer as compared to the other curved geometries. Kumar et al. (2007b) shows that, 8-banks coiled flow inverter heat exchanger (CFIHE) augments the heat transfer coefficient by 30% as compared to the helical coil tube. In the present study, the heat transfer area of CFIHE has been optimized experimentally for the possible intensification and more compact size of the CFIHE. The new experimental facility of 4-banks ($A_t = 0.88\ m^2$), 2-banks ($A_t = 0.44\ m^2$) and 1-bank ($A_t = 0.22\ m^2$) CFIHE's have been installed within the existing pilot plant of 8 banks ($A_t = 1.76\ m^2$) CFIHE. The experiments were carried-out in counter-flow configurations using hot fluid in the tube and cold fluid in shell side of the CFIHE. The tube side flow rate of water and air were varied over extensive ranges of 50 – 300 kg/h and 30 – 100 kg/h, respectively. The shell side flow rate of water and air were varied from 100 – 2000 kg/h and 400 – 750 kg/h, respectively. The performance of CFIHE was also compared with shell and tube heat exchanger (SHE) and plate type heat exchanger (PHE) under identical heat transfer area and process conditions.

The intensification in heat exchangers can be achieved, either by improving hydrodynamics or thermal conductivity of the fluid itself. The geometry perturbation is well established technology to improve the fluid phenomena. However, the lower thermal conductivity of conventional fluids viz. water, oil, and ethylene glycol mixture is a serious limitation for the performance and compactness of heat exchangers. A careful literature review reveals that, only a few studies have considered nanofluid flow inside complex geometries. The present study is the first to investigate the hydrodynamics and heat transfer characteristics of nanofluid flow in coiled flow inverter. The present study has been performed for a wide range of Reynolds numbers and nanoparticle volume fractions. It was noted that, the CFI significantly augments the heat transfer rate as compared to helical coil and straight tube of same dimension. The nanofluids considerable improves the heat transfer coefficients at a little higher value of friction factor as compared to the base fluid. The new correlations of the Nusselt number and friction factor have been proposed for the nanofluids flow inside CFI. The merit coefficient of CFI to that of straight tube has been also calculated.

The traditional heat exchangers technology is approaching to its limit, even though they employ conventional tubes with various cross-sections, orientations, and even the enhanced surface textures. Issues as higher energy demands, space limitations, and material savings have highlighted the necessity for miniaturized light-weight heat exchangers, which provide high heat transfer for a given heat duty. Microchannels (broadly ≤ 1 mm) represent the next step in heat exchanger development. The development of the microchannel heat exchanger has become of great importance in microelectronics cooling, chemical, biomedical, robotics, aerospace and automotive industries. In the present study, the hydrodynamics and heat transfer performance of the CFI and helical coil have been investigated at micro-scale, for the first time. The micro coiled flow inverter (MCFI) offers a significant improvement in heat transfer coefficients as compared to the micro helical coil (MHC). The effect of fluid properties on heat-transfer coefficient was also investigated by varying Prandtl number. The results show that the Prandtl number significantly affects the heat transfer coefficient.

A careful literature survey reveals that practically little attention has been paid to study the hydrodynamics of two phase gas-non-Newtonian fluid flow in complex geometries. In the present study, flow characteristics of single and two-phase non-Newtonian fluids have been investigated in three different CFI geometries. The CFI not only provides an academically challenging problem to study the complex flow behavior but also finds the potential applications in food and bio-processing industries due to the significant improvement in heat and mass transfer. The mass transfer performance of CFI has been also investigated under the laminar flow regime. The velocity and mass fraction profiles have been analyzed as they elucidate the basic physics behind the enhancement of mass transfer in coiled modules. The CFI offer a great improvement in Sherwood number as compared to the helical coil of same design parameters, for both the oxygenation and carbonation processes.

The present study is a successful attempt to investigate the CFI as a next-generation novel compact heat exchanger for thermal load removal application in industries such as power plants, production and chemical processes. The study takes a closer look at process intensification of coiled tube heat exchanger and must motivate to study and design truly three-dimensional microstructures devices for the different engineering applications.

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# NOTATIONS

| | |
|---|---|
| a | cross-section area, $m^2$ |
| $A_t$ | heat transfer area, $m^2$ |
| $A_s$ | Shell side surface area, $m^2$ |
| $C_1$ & $C_2$ | constants |
| Cp | specific heat, kJ/(kg.K) |
| $d_t$ & $d_h$ | tube diameter, m |
| $d_b$ | baffle diameter, m |
| $d_c$ | coil diameter, m |
| $D_{AB}$ | molecular diffusion coefficient, $m^2/s$ |
| $d_e$ | equivalent diameter, m |
| $d_S$ | shell diameter, m |
| $f$ | Fanning friction factor for coil ($= \frac{\Delta P_c}{2\rho V^2} \cdot \frac{d}{L_t}$), [–] |
| k | thermal conductivity, W/(m.K) |
| $K_B$ | Boltzmann constant, ($m^2$ kg/$s^2$ K) |
| $L_b$ | length of baffle, m |
| $L_S$ | length of shell, m |
| $L_t$ | length of heat exchanger, m |
| l | arc length along the centerline of the curved tube, m |
| h | heat-transfer coefficient, W/$m^2$ K |
| $\dot{m}$ | mass flow rate, kg/h |
| $m_t$ | mass flow rate in tube side of CFIHE, kg/h |
| $m_s$ | mass flow rate in shell side of CFIHE, kg/h |
| N | number of bank, [–] |
| $N_b$ | number of bends, [–] |
| n | power law index, [–] |
| P | pressure, N/$m^2$ |

| | |
|---|---|
| $P_t$ | pitch of coil, m |
| $q''$ | heat flux, $W/m^2$ |
| q | heat-transfer rate, W |
| Q | volumetric flow rate, L/h |
| $r_c$ | radius of the coil, m |
| $r_h$ | hydraulic radius, m |
| T | temperature, K |
| $t_i$ | hot fluid inlet temperature, K |
| $T_i$ | cold fluid inlet temperature, K |
| $t_o$ | hot fluid outlet temperature, K |
| $T_o$ | cold fluid outlet temperature, K |
| $T_\infty$ | free steam temperature, K |
| $\Delta T_1$ | temperature difference at inlet, K |
| $\Delta T_2$ | temperature difference at outlet, K |
| $U_o$ | overall heat transfer coefficient, $W/m^2$. °C |
| u* | frictional velocity ($=\sqrt{\tau_w/\rho}$), m/s |
| $u_i$ | velocity components, m/s |
| $u_0$, v, V | inlet velocity, m/s |
| $V_{dr,k}$ | drift velocity |
| $V_{SG}$ | gas superficial velocity, m/s |
| $V_{SL}$ | Liquid superficial velocity, m/s |
| x, y, z | master Cartesian coordinates |

**Greek letters**

| | |
|---|---|
| $\alpha_{bf}$ | thermal diffusivity, $m^2/s$ |
| $\delta$ | inverse of curvature ratio ($d_t/d_c$), [–] |
| $\delta_{ij}$ | Kronecker delta function [–] |
| $\varepsilon$ | turbulent energy dissipation, |
| $\varepsilon'$ | heat transfer efficiency [–] |

| | |
|---|---|
| $k$ | turbulent kinetic energy, $m^2/s^2$ |
| $\lambda$ | curvature ratio ($d_c/d_t$), [–] |
| $\mu$ | viscosity of the fluid, kg/m.s |
| $\rho$ | density of the fluid, $kg/m^3$ |
| $\Phi$ | wall flux, $W/m^2$ |
| $\varphi$ | axial distance, ($^0$) |
| $\phi$ | dependent variable |
| $\Theta$ | dimensionless temperature, $(T - T_w)/(T_b - T_w)$ |
| $\nu$ | kinematic viscosity, $m^2/s$ |
| $\sigma_\tau$ | turbulent Prandtl number in energy equation, [–] |
| $\sigma_k$ | diffusion Prandtl number for $k$, [–] |
| $\sigma_\varepsilon$ | turbulent Prandtl number for $\varepsilon$, [–] |
| $\tau_w$ | wall shear stress, $N/m^2$ |
| $\tau$ | torsion $(=p/(R^2+p^2))$, [–] |

**Dimensionless numbers**

| | |
|---|---|
| De | Dean number ($Re/\sqrt{\lambda}$), [–] |
| Eu | Euler number $\left(= \dfrac{d^{0.85} D_e^{0.85}}{L}\right)$, [–] |
| He | Helical number $\left(= Re\left[\dfrac{(d_t/d_c)}{1+(p/\pi d_c)^2}\right]^{1/2}\right)$, [–] |
| Pr | Prandtl number ($C_p.\mu/k$), [–] |
| St | Stanton number ($h/(C_p.\rho.V)$), [–] |
| Re | Reynolds number ($\rho.V.d/\mu$), [–] |
| $Re_{crit}$ | critical Reynolds number, [–] |
| $Pr_w$ | Turbulent wall Prandtl number, [–] |
| Nu | Nusselt number $(= \dfrac{hd}{k})$, [–] |

| | |
|---|---|
| Ra | Rayleigh number ( $= \frac{g\beta}{v\alpha}(T_s - T_\infty)L_t$ ), [ - ] |
| Sc | Schmidt number ($\mu/(\rho.D_{AB})$) , [–] |
| Tn | Torsion number (=2τRe), [–] |

**Subscripts and superscripts**

| | |
|---|---|
| 0 | inlet condition |
| 1 | first bend |
| 2 | second bend |
| b | bulk quantity |
| c | coiled tube |
| m | mean value |
| cp | value with constant property |
| s | straight tube |
| inner | inner tube |
| outer | outer tube |
| i | inside/inner |
| o | outside/outer |
| w | wall condition |
| + | standard wall coordinates. |
| θ | local quantity |

**Abbreviations**

| | |
|---|---|
| AA | Alternate Axis |
| ADI | Alternative Direction Implicit |
| BC | Boundary Condition |
| CA | Constant Axis |
| CFI | coiled flow inverter |
| CFIHE | coiled flow inverter heat exchanger |
| CVFDM | Control Volume Finite Difference Method |

| | |
|---|---|
| HE | heat exchanger |
| LMTD | log-mean temperature difference |
| MHD | Magneto Hydro Dynamics |
| MF | Micro Filtration |
| NF | Nano Filtration |
| RTD | Residence Time Distribution |
| UF | Ultra Filtration |