COMSOL CONFERENCE EUROPE 2012

Numerical Investigation on The Convective Heat Transfer Enhancement in Coiled Tubes

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Heat transfer enhancement techniques

- Primary aim: study the influence of the curvature profile on the convective heat transfer mechanism

- Heat Transfer enhancement techniques:

    - Active techniques

    - Passive techniques

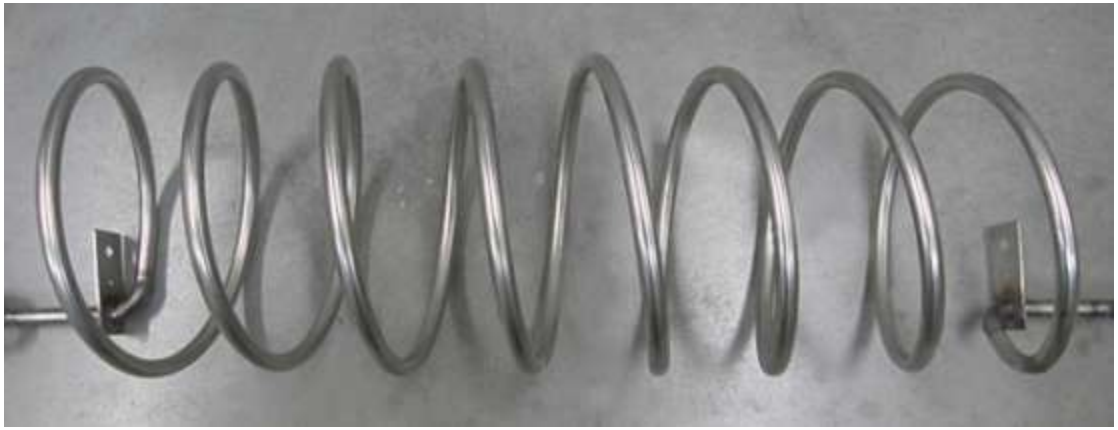
Passive techniques: curved tubes

- Curved tubes:
    - centrifugal force that cause a distortion of the velocity profile
    - increase of the temperature gradients at the wall
    - maximization of the heat transfer
    - secondary flow that promotes the mixing of the fluid in the boundary layer

Experimental Study

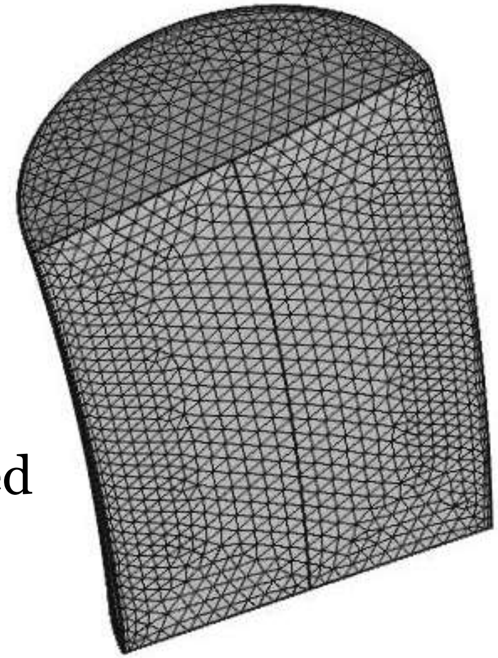
- The Department of Industrial Engineering of the University of Parma is carrying out a study on curved tubes

- In order to better investigate the phenomenon a numerical model has been realised



Geometry

- Toroidal geometry
- Tube diameter D: 14 mm
- Curvature ratio δ: 0.06
- Only one half of the tube has been simulated
- A portion of 10 degrees was considered
- Mesh : 600000 elements



Model implementation

- The study was performed by integrating the continuity, momentum and energy equations within Comsol Multiphysics 4.2a environment

- Assumption of :
    - constant properties fluid
    - incompressible Newtonian fluid
    - periodically fully developed laminar flow for the hydrodynamic and the thermal problem

- By considering negligible viscous dissipation the formulation of these equations is the following:

$$\rho(u\cdot\nabla)u=\nabla\cdot\left[-\rho l+\mu\left(\nabla u+(\nabla u)^{T}\right)\right]+F$$

$$\rho\nabla\cdot u=0$$

$$\rho C_p\frac{\partial T}{\partial t}+\rho C_p u\cdot\nabla T=\nabla\cdot(k\nabla T)$$

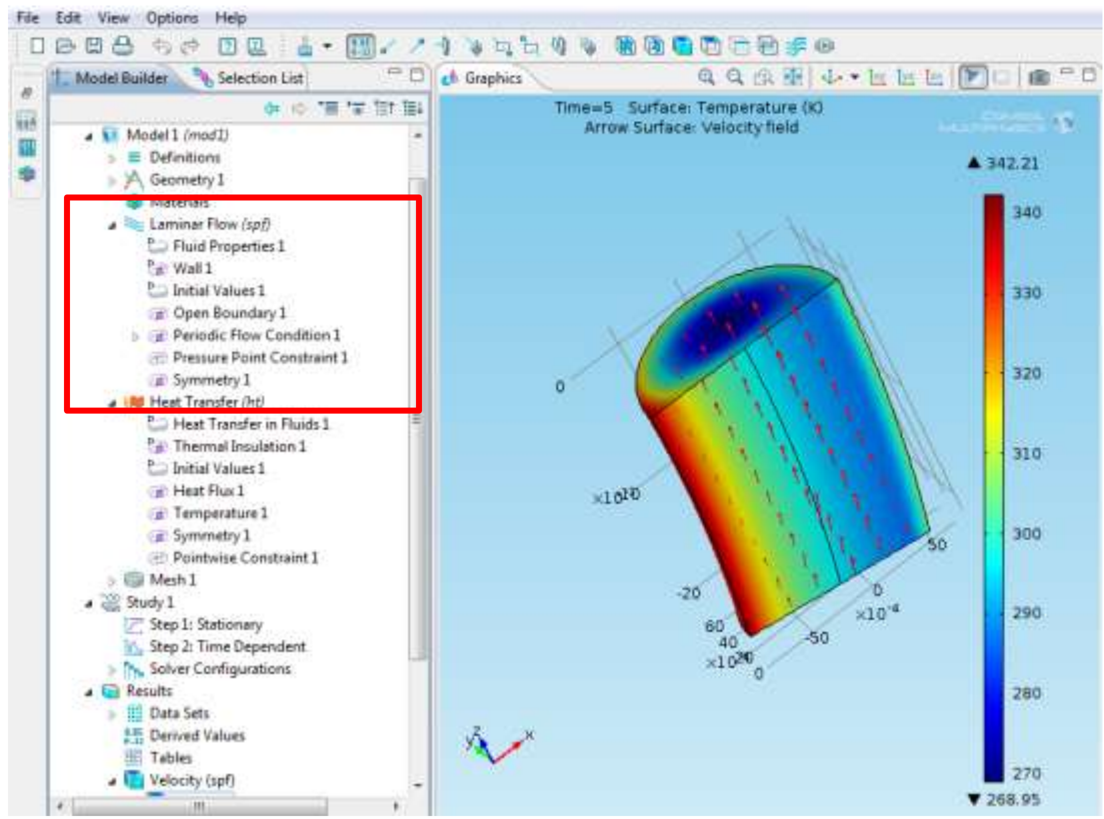
Model implementation:hydrodynamic problem

- Pressure condition:

$$p_{entrance} - p_{exit} - \Delta p = 0$$

- No slip condition

- *Pointwise* constraint for the pressure in one point of the entrance region



Model implementation: thermal problem

- Uniform wall heat flux boundary condition

- As a consequence the temperature distribution has a profile which repeats periodically as follows:

$$T(x,y,z,t)_{entrance} - T(x,y,z,t)_{exit} + \Delta T = 0$$

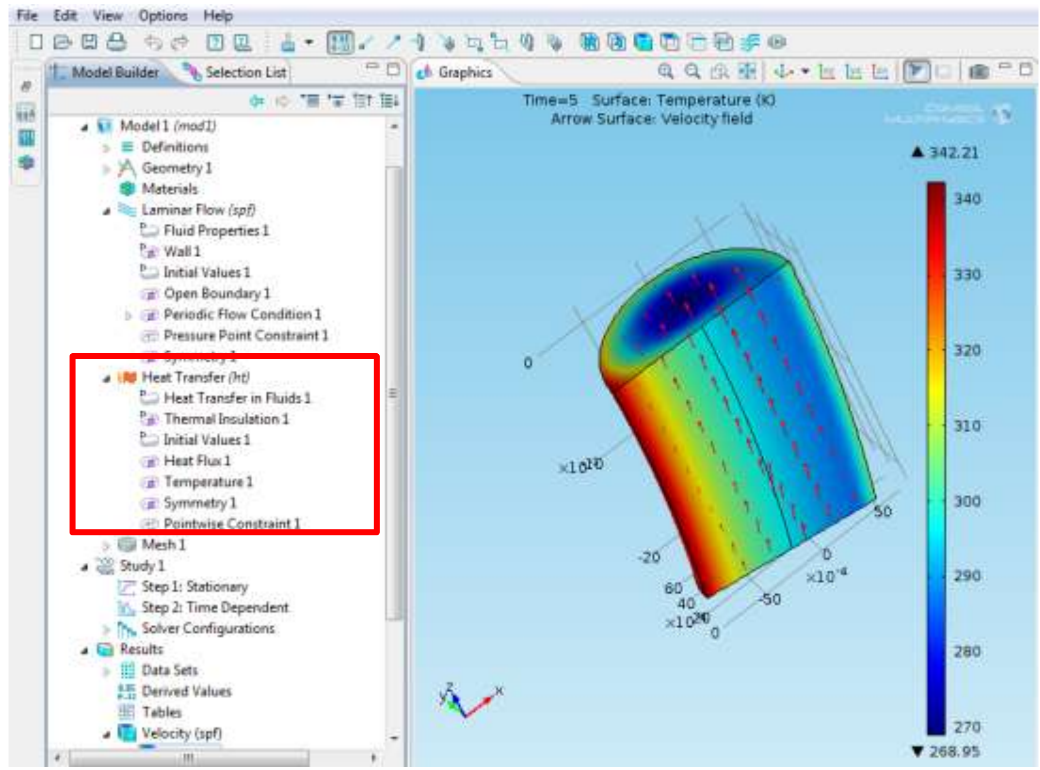
- This condition in Comsol Multiphysics 4.2a environment is obtained by :

$$linext\,(T_{entrance}) - T_{exit} + \Delta T = 0$$

- *Pointwise* constraint for temperature in one point of the entrance section

- Considering the energy balance, ΔT is related to the wall heat flux density as follows:

$$\Delta T = \frac{q \cdot S}{m \cdot c_p}$$



Data Processing

- Nusselt number: $$Nu = \frac{\bar{h} \cdot D}{\lambda}$$

The average convective heat transfer coefficient

$$\bar{h} = \frac{q}{\left(T_{w,av} - T_{b,m}\right)}$$

The surface averaged wall temperature

$$T_{w,av} = \frac{1}{A}\int_A T dA$$

The arithmetic mean of the inlet and of the outlet bulk temperature

$$T_{b,m} = \frac{1}{2}\left(\left(\frac{\int_{A_c} Tu \cdot ndA}{\int_{A_c} Tu \cdot ndA}\right)_{inlet} + \left(\frac{\int_{A_c} Tu \cdot ndA}{\int_{A_c} Tu \cdot ndA}\right)_{outlet}\right)$$

- Friction factor $$f = \frac{\Delta p}{\rho} \cdot \frac{D}{l} \cdot \frac{2}{v_{av}^{2}}$$

The average axial velocity

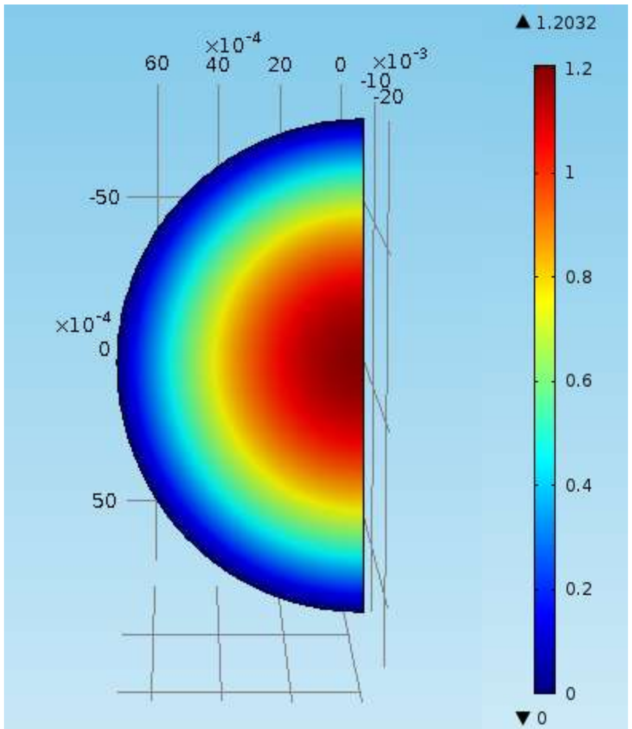
$$v_{av} = \frac{1}{A_c}\int_{A_c} v dA$$

The Reynolds number

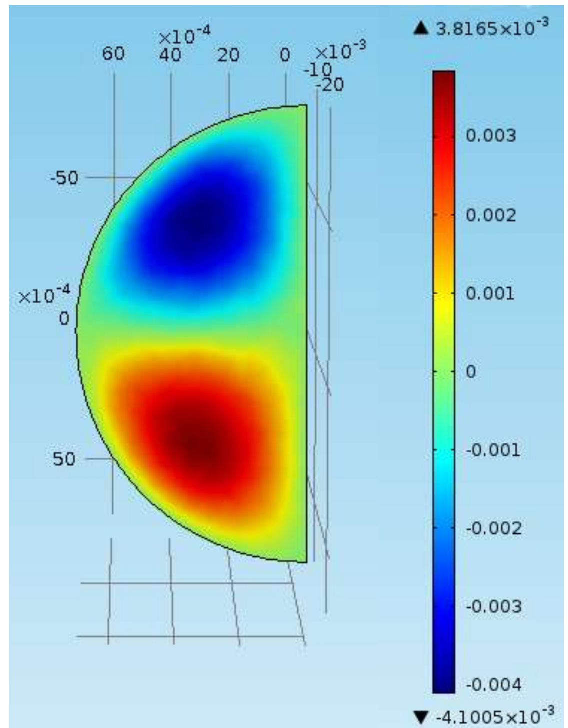
$$\mathrm{Re} = \frac{\rho \cdot v_{av} \cdot D}{\mu}$$

Results

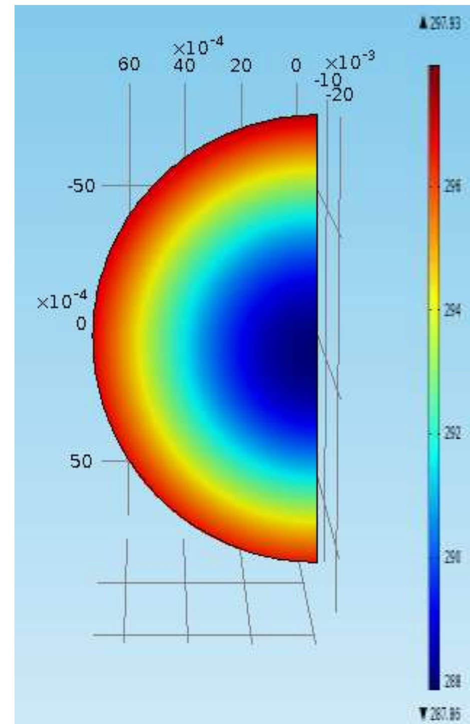
- Re number range: 2-100



Axial velocity: Re=2.9



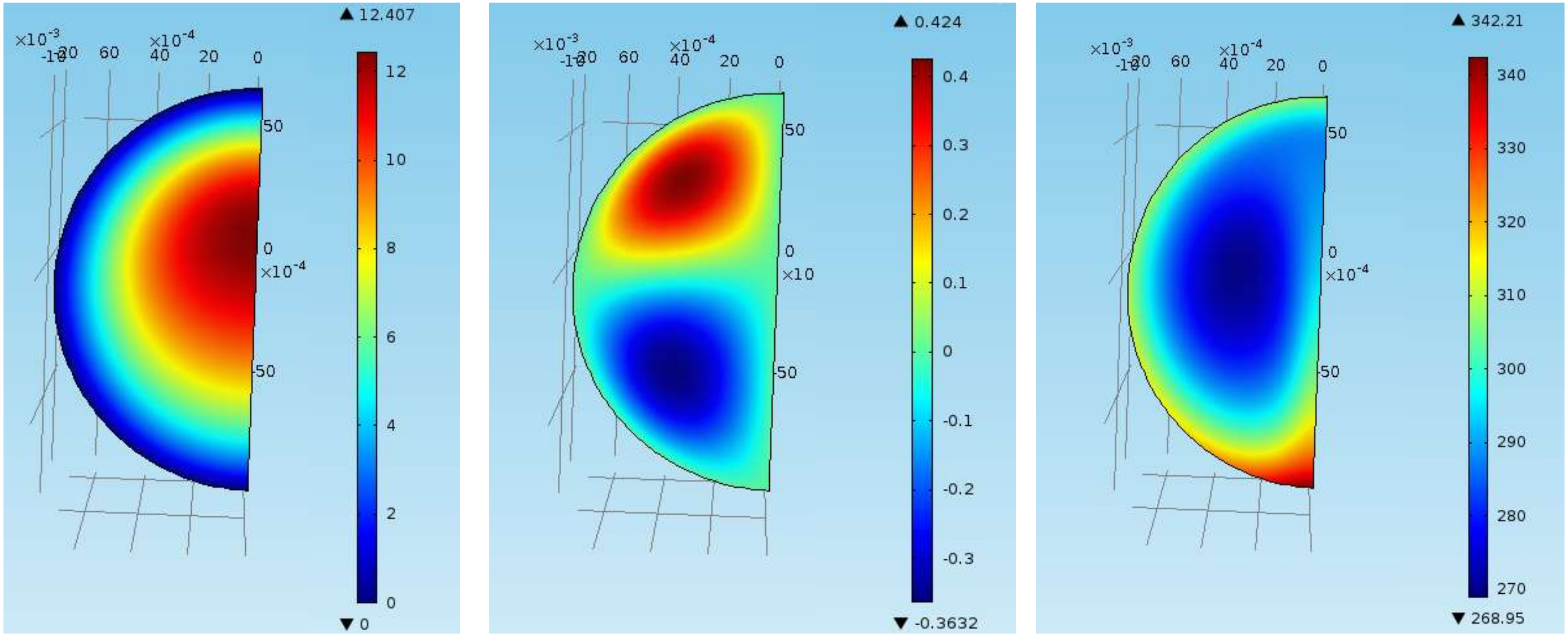
Radial velocity: Re=2.9



Temperature distribution: Re=2.9

- The distortion in the axial velocity distribution is still negligible
- The presence of a secondary flow: its intensity is still not significant

Results



Axial velocity: Re=97

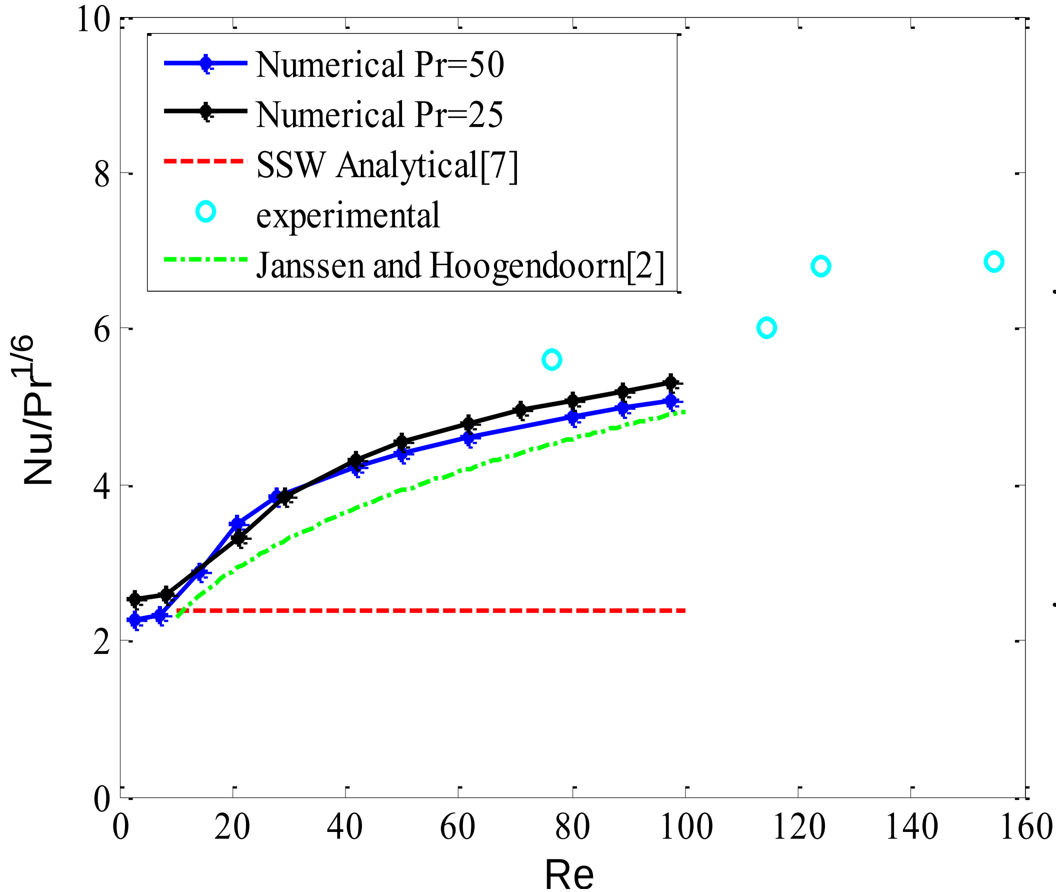
Radial velocity: Re=97

Temperature distribution: Re=97

- The secondary flow became more significant and the distortion of the velocity profile is not still negligible

- Also the temperature profile is distorted

Results



- The values obtained within Comsol Multiphysics 4.2a environment are supported by the experimental results: the Nusselt number reaches values higher than the ones expected for straight pipes
- The augmentation effect was due to the wall curvature
- The numerical results are in good agreement with the correlations proposed by Janssen and Hoogendoorn

- Janssen and Hoogendoorn (1978) correlations:

$$Nu = 1.7(De^2 \Pr)^{1/6}$$

for De<20 and $(De^2 \Pr)^{1/2} < 100$

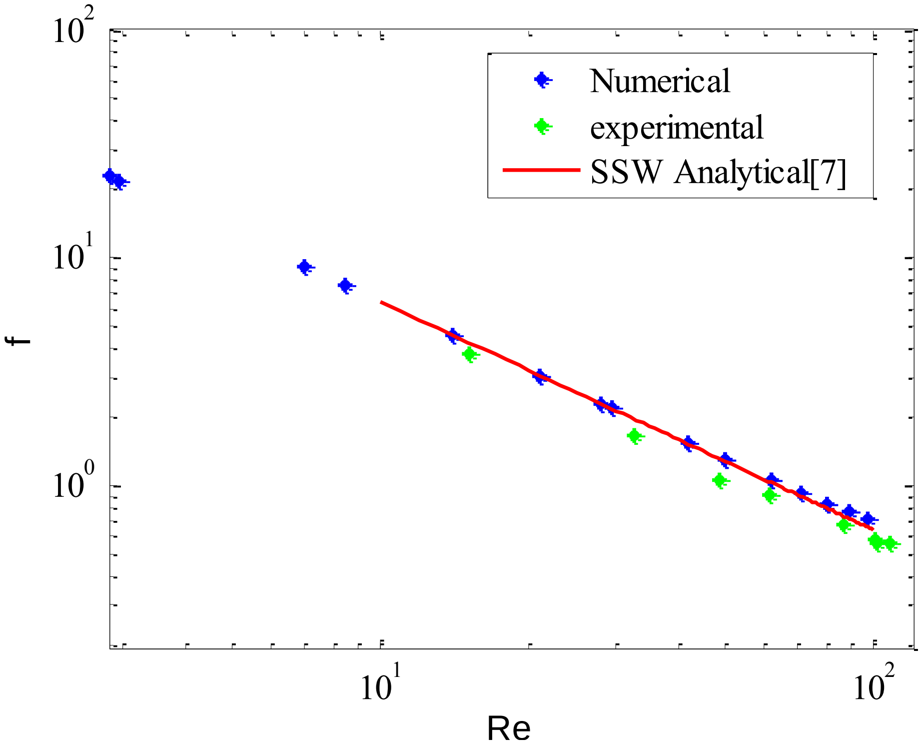
$$Nu = 0.9(\mathrm{Re}^2 \Pr)^{1/6}$$

for 20 < De < 100, 20 < Pr < 450 and 0.01 < δ < 0.083

$$Nu = 0.7 \cdot \mathrm{Re}^{0.43} \Pr^{1/6} \delta^{0.07}$$

for 100 < De < 830 and 20< Pr < 450 and 0.01 < δ < 0.083

Results



- The results support the values obtained experimentally

- The increasing over SSW analytical solution of the friction factor due to the wall curvature profile is negligible for this flow regime

Conclusions

- The curvature of the wall tube represents an efficient solution for enhancing the heat transfer in case of laminar flow regime

- The curvature profile of the tube induce the origin of a centrifugal force that cause a distortion of the velocity profile, inducing local maxima in the velocity distribution

- Increase of the temperature gradients at the wall and maximization of the heat transfer

- Processes in which highly viscous fluids are involved: food, chemical and pharmaceutics industries

- The results are in good agreement with the correlation proposed by Janssen and Hoogendoorn for helical coiled tubes