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Numerical Investigation of the Cooling of Shear Thinning Fluids in Cylindrical Horizontal Ducts

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ABSTRACT

The present paper is an investigation of the cooling of hot shear thinning fluids flowing through cylindrical pipes. The study is achieved via numerical simulations with the help of the computer code CFX, which is based on the finite volume method to solve the governing equations. The efficiency of two techniques for achieving the cooling process is investigated, namely: the counter flow and the baffling techniques. In the first part and for the first strategy, the hot fluids are cooled by an external turbulent counter flow of a Newtonian liquid. In the second part and in an attempt to enhance the energy efficiency of the heat exchanger system, semi-circular baffles are inserted. We note that two strategies are used in combination in the second part of study. Effects of the flow rates and the pitch ratio of the inserted baffles on the flow and thermal fields are explored. The obtained results show a great enhancement of heat transfer rates when using both strategies in combination.

1. Introduction

In many industries such as food, polymer, chemical, and petrochemical industries, non-Newtonian fluids are subjected to some thermal processing. Many fluids whose viscosities are

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dependent on the shear rates are described as non-Newtonians, such as emulsions, slurries, polymer melts, foams, blood and paint. The physical properties of these fluids are generally very sensitive to the temperature, which has an influence on the flow fields, pressure drop and the heat transfer rate.

Thorough analyses are required to well understand the flow and thermal behavior of such fluids. Some studies have been achieved to determine the thermal characteristics in different situations such as natural convection [1–3], forced convection [4,5] mixed convection [6,7]. Chiba et al. [8] studied analytically the heat transfer in a pipe.

Sheela-Francisca et al. [9] studied the heat transfer of power-law fluids inside parallel-plates. For parallel-plates and circular micro-channels, Shojaeian and Kosar [10] explored the heat transfer of power law fluids with variable thermo-physical properties.

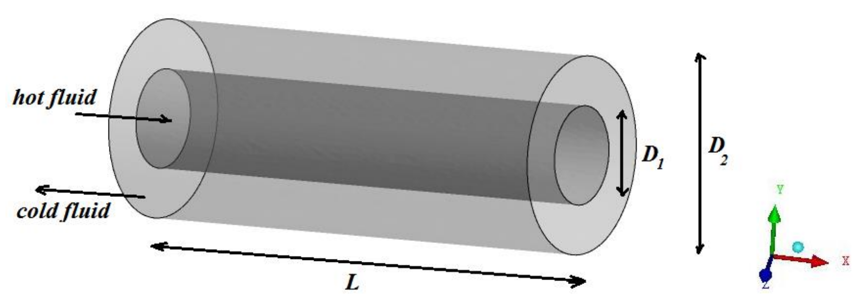
In the present paper, the cooling of complex shear thinning fluids in horizontal cylindrical pipes is studied. We focus on the efficiency of two strategies: the counter flow technique and the baffling technique.

2. Problem description

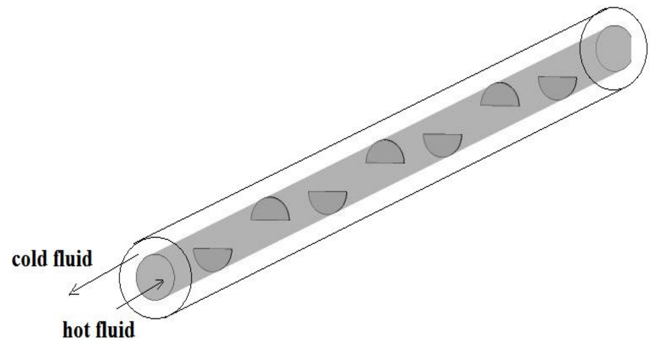
The geometry of the system studied is shown in Fig. 1. The counter flow technique is schematized in Fig. 1a which consists of two concentric horizontal pipes. The hot shear thinning fluid flows in the inner pipe which has a diameter D1 = 0.034 mm and a length L = 106*D1. The external cooling liquid (a mixture of water and ethyl glycol) flows in an annular section (D11 = 35 mm, D2 = 72 mm) surrounding the tube. Because of the chemical composition, the mean inlet temperature of the cooling fluid may be as low as -10 °C.

The inlet temperature of the cold fluid is fixed at -5 °C for a flow rate Q0 = 2 m3/h, and that for the hot fluid is fixed at 308 K. The Reynolds number for the shear thinning fluid is varied from 0.1 to 100.

The baffling technique is schematized in Fig. 1b which consists of semi-circular baffles inserted in the cylindrical pipe. We note that the geometrical parameters in Fig. 1b (Diameter and length) are the same of the internal cylindrical pipe used in the counter flow technique.



(a) Counter flow technique



(b) Combination of the counter flow technique with the baffling technique

Fig. 1. Geometry simulated.

3. Mathematical equations

Water solution of Carboxy Methyl Cellulose (CMC) with a concentration of 4 percent in weight is used as a working media (its flow behavior index (n) is 0.68). The Oswald law is used to describe the shear thinning behavior of such fluids:

$$\tau = k \dot{\gamma}^{n} \tag{1}$$

where k is the consistency index. The fluid is incompressible and thermo-dependent. The exchange coefficient for the shear thinning fluid $h = h(x)$ is defined by:

$$h(x) = \varphi(x) / [T_m(x) - T_w(x)] \tag{2}$$

where $\varphi(x)$ is the local heat flux density for the inner surface of the internal tube. $T_m(x)$, $T_w(x)$ are the mean and the wall temperatures, respectively, with:

$$T_m(x) = T_e + \frac{1}{\rho . C_p . Q} \int_0^x \varphi(x') . \pi . D . dx' \tag{3}$$

For the external cooling fluid:

$$h_0 = \varphi'(x) / [T_m(x) - T_w(x)] \tag{4}$$

where $\varphi'(x)$ is the local heat flux density for the outer surface of the internal tube. Since the cooling fluid flows in the turbulent regime and at a constant temperature, the coefficient h_0 is considered as constant along the tube:

$$\psi = \int_0^L \varphi'(x) . \pi . D_1 . dx \tag{5}$$

With $\psi = \rho . C_{p0} . Q_0 \; T_{0,0} - T_{i,0}$ obtained by a heat balance on the cooling water.

The results are represented by means of the local Nusselt number, define as:

$$Nu(x) = h(x) . D / \lambda \tag{6}$$

where λ is the thermal conductivity.

The generalized Reynolds number (Re_g) is defined as:

$$Re_g = \frac{1}{\left(3n + \frac{1}{4n}\right)^n} \rho U_d^{2-n} D^n / 8^{n-1} K \tag{7}$$

Where ρ is the fluid density and U is the velocity.

4. Numerical modeling

To perform calculations, the computer code (CFX 16.0) developed by AEA Technology, UK, was used. This computer program is based on the finite volume method. The grid-generation tool, ICEM CFD 16.0 was used to create the geometry and to mesh the computational domain by tetrahedral grids (Fig. 2). Mesh tests were achieved by checking that additional grid density did not change the Nusselt number by more than 2%. The selected meshes for all geometrical models studied have about 800,000 to 900,000 elements. For a computer machine with Core i7 CPU 2.20 GHz and 8.0 GB of RAM, calculations are performed with residual targets of 10^{-7} for temperature and velocities. Convergence was achieved after about 800-1000 iterations and 3-4 hours in CPU time.

5. Results and discussions

5.1. Validation

Before any investigation, we have checked the validity of the computer program and the accuracy of the predicted results. We have referred to the experimental work performed by Azevedo et al. [11] and have realized the same geometrical model. Variations of the temperature wall along the pipe length ($X^* = X/D$) are presented on Fig. 2. The comparison between our predicted results and the experimental data of Azevedo and his co-workers shows a satisfactory agreement.

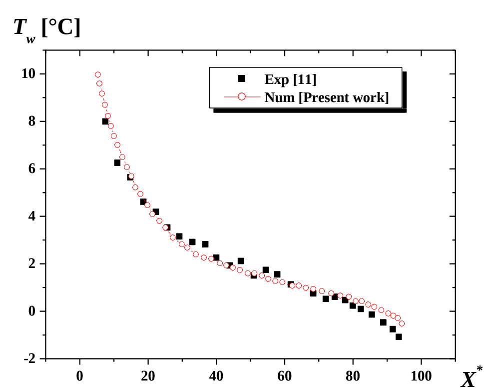


Fig. 2. Variation of the wall temperatures for $n = 0.68$.

5.2. Effect of Reynolds number

In the first part of our investigation, just the counter flow technique is used. Fig. 3 shows an example of the evolution of the working fluid temperature at $Re_g = 1$. As observed, the flow of two fluids with large different temperatures into opposite directions makes the system of heat exchange very efficient. The figure shows obviously the gradient of temperature between the middle of tube and the wall due to the formation of thermal boundary layer on the tube wall.

Fig. 4 presents variations of the axial velocity (U^* = Velocity u/U_{inlet}) at the outlet part of pipe (X^* = 90). It is clear that the maximum peak is reached for the larger value of Re_g. The chaotic motions observed are intensified with the increase of Re_g. Where, the highest value observed on the highest value of Reynolds number due to the augmentation of shear stress on the tube wall.

Fig. 5 presents the variations of Nusselt number (Nu) along the tube for different Re_g. as remarked, Nu decreases along the tube and increases proportionally to Re_g. These augmentations due to the increases of the velocity fluctuations near the wall where its augment the heat transfer execution.The thermal fields are illustrated on Fig. 6 under vertical planes. The hot shear thinning fluid is rapidly cooled for lower Reg. However, the required length for efficient cooling is increased with the increase of Reg. We note that the fall in the wall temperature causes an increase in the apparent viscosity near the wall.

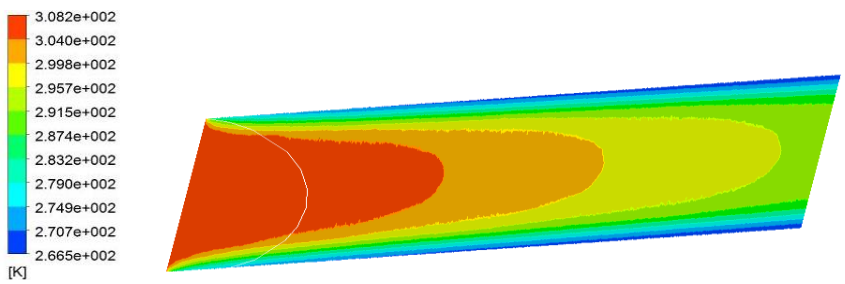


Fig. 3. Evolution of the working fluid temperature at $Re_g = 1$.

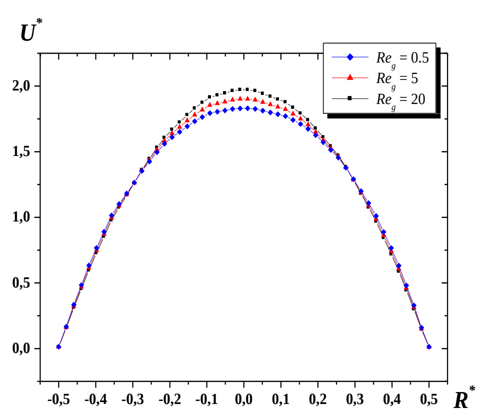


Fig. 4. Axial velocity vs. pipe radius at $X^* = 90$.

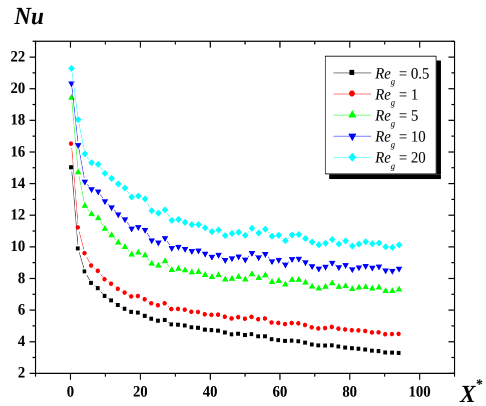
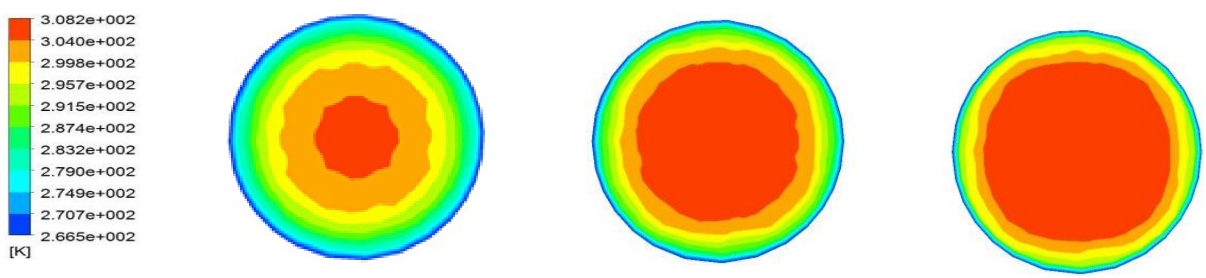


Fig. 5. Evolution of the Nusselt number along the pipe.

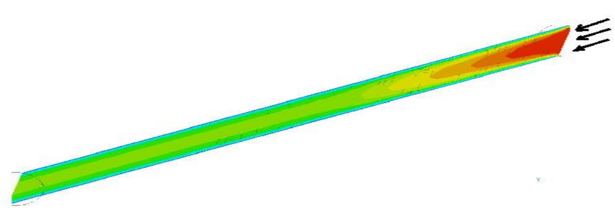


$Re_g = 0.2$ $Re_g = 3.5$ $Re_g = 12$

Fig. 6. Temperature contours for an unbaffled tube, at Z* = 0.5.

5.3. Effect of the presence of baffles

In this section, we try to compare the efficiency of the counter flow technique used alone and then combined with the baffling technique. The counter flow technique is found to be efficient to accelerate the cooling process. However, and when compared to the combined techniques, the last one (i.e. combination) seems more effective since it provides a shorter distance for achieving the required temperature of the CMC solution (Fig. 7). As a description, the presence of baffles inside the tubes helps to stop the apparition of the thermal and hydro-dynamic boundary layers and its intense the turbulence near the walls where it added a supplementary advantage for the phenomenon of heat transfer.



(a)

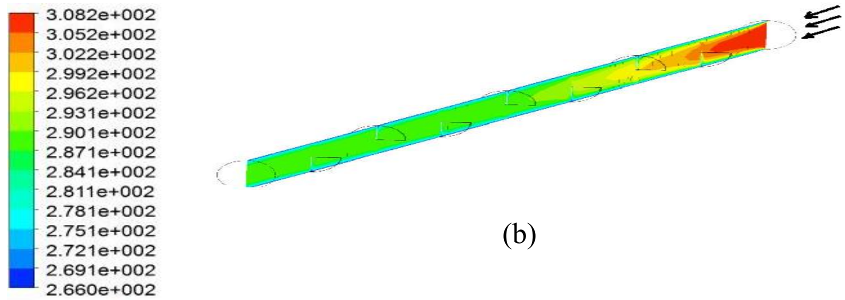


Fig. 7. Temperature contours [K] for a baffled tube ($P^* = 0.125$) and a tube without baffles, at $Re_g = 12.1$.

5.4. Effect of the pitch ratio

In this section, effects of the space between baffles are explored. Five geometries are realized with different values of the pitch ratio ($P^* = P/L$), which are: $P^* = 0.075$, 0.125, 0.187, 0.25 and 0.375.

The flow fields generated by the presence of semi-circular baffles are shown on Fig. 8. This type of baffles creates a vortex behind the baffle and intensifies the kinetic energy of flows, even at low Reynolds number. The fluid particles are then well-mixed and the heat transfer will be enhanced. The size of vortices increases with the increase of the distance between baffles, i.e. the pitch ratio ($P^* = P/L$), and this is due to the strong interaction of fluid particles when the space between baffles is small. This factor is responsible for the fast decrease of Nusselt number along the tube length and the great enhancement of cooling process, as observed on Fig. 9.

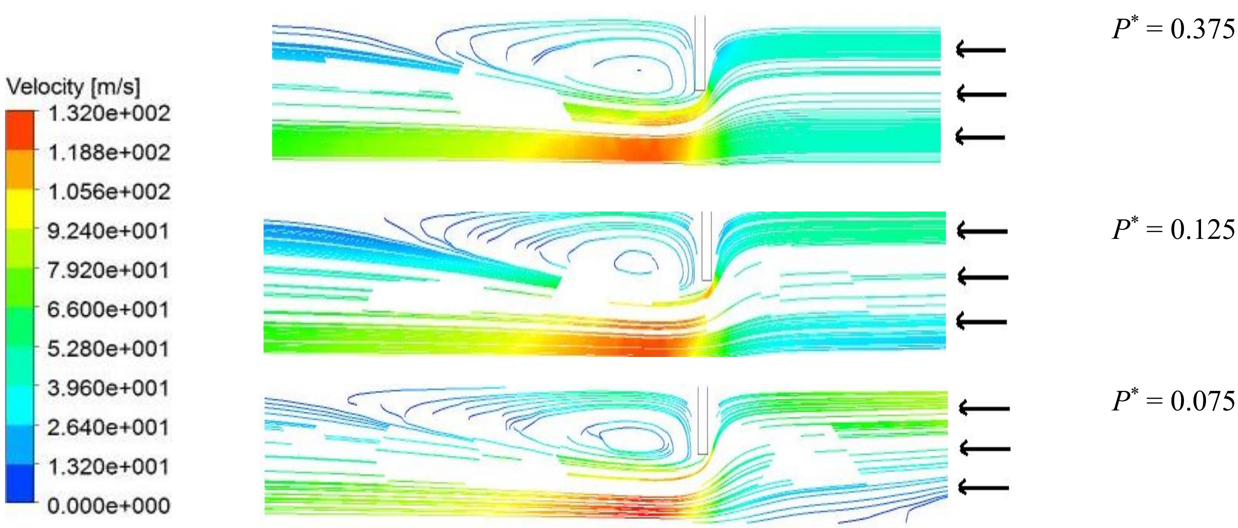


Fig. 8. Flow fields for a tube with baffles at different pitch ratio, $Re_g = 100$.

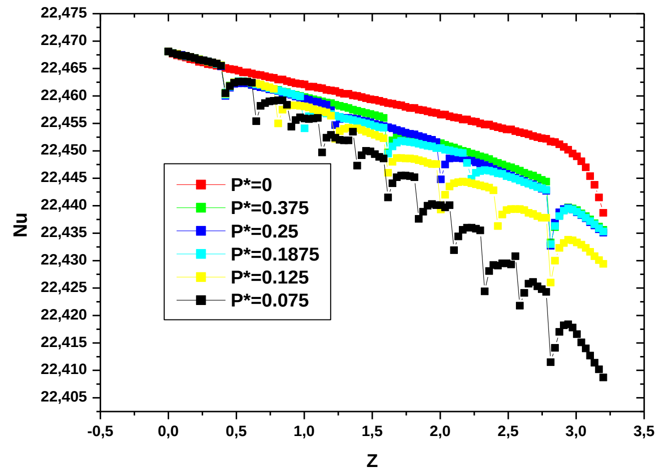


Fig. 9. Variation of Nusselt number for $Re_g = 100$.

6. Conclusion

A numerical investigation of the cooling process a shear thinning fluid in a cylindrical pipe was presented. The Two techniques were used to enhance the heat transfer in the tube, namely: the counter flow technique and baffling technique. The first technique was used alone in the first part of study and then combined with the second technique in the second part. Effects of Reynolds number and space between baffles were also described.

The counter flow technique was found to be very effective for the cooling of hot shear thinning fluids. The presence of baffles inside the pipe may create a vortex behind each baffle, which intensifies the local mixing and enhance the heat transfer rates. The decrease of the space between baffles yields powerful interaction between fluid particles and gives a better enhancement of thermal transfer. The increase of Reynolds number lengthens the distance to obtain the required temperature of the cooled fluid. Finally, the combination of both techniques seems more promising in term of heat transfer enhancement.

To better understand the hydrodynamic and thermodynamic behavior of such complex fluids, further works are needed on the combination of two techniques with taking into account the thermo-dependence on temperature, pressure drop and energy consumption.

Nomenclature

C_p, C_{p0} Specific heats (working fluid and cold fluid)

D, D_1, D_2 Diameters

$h(x)$, h_0 Heat exchange coefficients

k Fluid consistency

L Length of the exchange zone

n Flow behavior index

Nu Nusselt number

Q, Q_0 Flow rates (working fluid and cold fluid)

R^*, X^* Dimensionless radial and axial abscises

Re_g Generalized Reynolds number

TTemperature

U, U_0 Local, mean axial velocities

Greek letters

ηApparent viscosity

ρ Fluid density

φ, φ' Heat flux densities

τ Shear stress

λ Thermal conductivity

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