Heat transfer to Newtonian and non-Newtonian fluids in cross-corrugated chevron-type plate heat exchangers: Numerical approach.

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Abstract

In this numerical work we study the laminar flow and heat transfer to Newtonian and power-law fluids, during their processing in cross-corrugated chevron-type plate heat exchangers passages. The flow index behaviour varied in a broad range (0.25 - 2) and the area enlargement factor from the plates was 1.17 for all the corrugation angles (31º, 35º, 40º, 45º, 50º, 55º and 60º). The corrugation angle was defined relatively to the horizontal and the results suggested that the Nusselt number is not inversely proportional to the corrugation angle. Simulations with different power law-fluids and Newtonian fluids with different Prandtl numbers suggested that the Nusselt number reaches a maximum for corrugation angles between 40º and 50º, approximately. It is also shown that the shear thinning effects greatly affect the thermal-hydraulic performance of the plate heat exchangers passages. Local convective heat transfer coefficients were studied using repeated unitary cells due to the periodicity of the flow in the complete plate heat exchangers passages. In thermal and hydraulic fully developed flows it was found that the average convective heat transfer coefficient from the first half of a unitary cell differs from that of the second half. In a unitary cell, the maximum/minimum local heat transfer coefficients and interstitial velocities had the same location along the main flow direction.

Introduction

Most of the food products processed in plate heat exchangers (PHEs) are non-Newtonian fluids and little is known about their flow and heat transfer behaviour (Kim et al., 1999). In addition, many of these food fluids have a high viscosity and, therefore, the flow regime is surely not always the turbulent flow regime (Leuliet et al., 1987, 1988; Rene et al., 1991; Delplace and Leuliet, 1995; Metwally and Manglik, 2000, 2002; Fernandes et al., 2005, 2006; Fernandes et al., 2007, 2008). The low Reynolds number range can also be found in micro PHEs (Gschwind et al., 2001).

Due to the periodicity observed in a cross corrugated chevron-type PHE passage, these ducts can be studied using simplified geometrical domains (Ciofalo et al., 1996; Mehrabian and Poulter, 2000; Fernandes et al., 2007, 2008), i.e., unitary cells (Fig. 1).

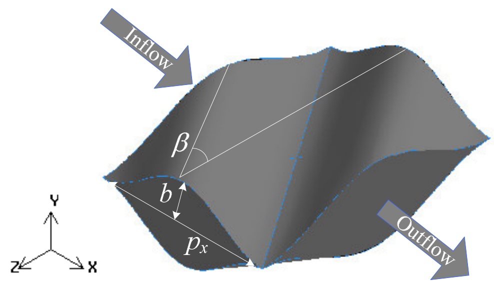


Fig.1: Unitary cell.

In Fig. 1, b, p_x and β represent the inter-plates distance, pitch corrugation in the main flow direction and corrugation angle, respectively. The area enlargement factor (the ratio between developed and projected area), ϕ, can be calculated with precision by (Martin, 1996; Fernandes et al., 2007):

$$\phi = \frac{1}{6}\left\{1+\left[1+\left(\frac{\pi}{2\cos(\beta)}\right)^2\gamma^2\right]^{0.5}+4\left[1+\left(\frac{\pi}{2\sqrt{2}\cos(\beta)}\right)^2\gamma^2\right]^{0.5}\right\}, \tag{1}$$

γ representing the channel aspect ratio:

$$\gamma = \frac{2b}{p_x}. \tag{2}$$

The thermal-hydraulic performance from the PHEs passages is closely connected to the fluid physical properties and to the geometrical properties mentioned above (Ayub, 2003; Kakaç and Liu, 2002; Palm and Claesson, 2006; Reppich, 1999).

The aim of the present work is to compare the thermal performance, under laminar regime, of different chevron-type PHEs passages (β = 31º, 35º, 40º, 45º, 50º, 55º and 60º), with an area enlargement factor equal to 1.17, a value widely used in this type of equipments (García-Cascales et al., 2007; Kumar, 1984; Martin, 1996). Different power-law fluids are studied, as well as Newtonian fluids with different Prandtl numbers.

Numerical calculations

The numerical calculations were performed using the commercial finite element software package POLYFLOW®. The equations solved were the conservation of mass, momentum and energy equations for laminar incompressible flow of Newtonian and power-law fluids and the used constitutive model was the power-law model:

$$\eta = \eta_0 \dot{\gamma}^{n-1}, \tag{3}$$

$\dot{\gamma}$ representing the shear-rate, η_0 the consistency index, n the flow index behaviour and η the apparent viscosity. In the present investigation, the flow index behaviour varied between 0.25 and 2.

The simulations were performed using channels containing seven consecutive unitary cells (Fig. 2), since thermal and hydraulic fully developed flows were achieved in the fifth or sixth consecutive cell, as described in previous works (Fernandes et al., 2007, 2008).

The inter-plates distance was 2.5 mm for the different angles (β = 31º, 35º, 40º, 45º, 50º, 55º and 60º). Since ϕ assumed the value of 1.17 in all tested geometries, the value of p_x needed in order to obtain this area enlargement factor was calculated resorting to Eqs. (1) and (2).

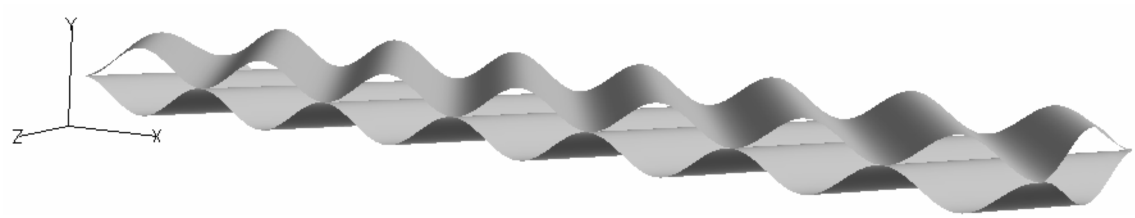


Fig. 2: Geometrical domain containing seven consecutive unitary cells.

Due to the complexity of the computational domain, an unstructured mesh constituted by tetrahedral, hexahedral and pyramidal elements was used (Fernandes et al., 2005, 2006, ; Fernandes et al., 2007, 2008) and the size of the used elements was fixed after a grid independence test (detailed information about this test can be found in previous works (Fernandes et al., 2007, 2008)).

The accuracy of the numerical model used in this study was extensively tested in previous works (Fernandes et al., 2005, 2006; Fernandes et al., 2007, 2008). Using a numerical approach, Fernandes et al. (2005, 2006) studied the cooling of stirred yoghurt (n = 0.42) in a chevron-type PHE with a corrugation angle of 30º and an area enlargement factor close to 1.1. The numerical thermal data from Fernandes et al. (2005, 2006) compared well with experimental data from Afonso et al. (2003). Fernandes et al. (2007) studied Newtonian fully developed laminar flows in chevron-type PHEs passages. The numerical results of pressure drop from Fernandes et al. (2007) were in good agreement with the semi-theoretical correlation from Wanniarachchi et al. (1995) (recommended by Ayub (2003)) and experimental data from Kumar (1984). Fernandes et al. (2008) studied the pressure drop of power-law fluids (n varied between 0.25 and 1) in chevron-type PHEs. The numerical results of pressure drop from Fernandes et al. (2008) were in good agreement with the experimental data from Rene et al. (1991).

Results

The knowledge of the local convective heat transfer coefficients, h, can provide information in order to improve the performance of chevron-type PHEs (Heggs et al., 1999). For the analysis of the local behaviour of the flow and heat transfer, a dimensionless length was defined: $x^* = x / p_x$. In Fig. 3 the local values of h along the main flow direction, i.e., in the first six consecutive unitary cells, are shown. As already mentioned, thermal fully developed flows were obtained in the fifth (x^* = 4 to 5) / sixth (x^* = 5 to 6) consecutive unitary cell.

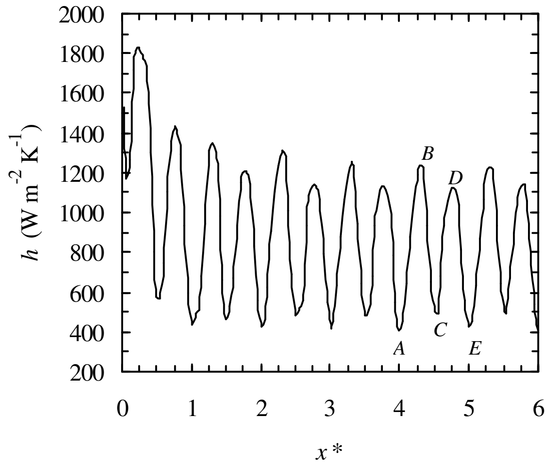


Fig. 3: Local convective heat transfer coefficients for $\beta = 31°$.

In Fig. 4 the temperature profile ($\beta = 31°$) obtained in fully developed flow (fifth unitary cell) is shown. The lower values of h (see also Fig. 3) were obtained in the planes A, C and E ($x*$ = 4, 4.5 and 5, respectively) while the higher values of h were obtained in the planes B and D ($x*$ = 4.25 and 4.75, respectively). This was observed in all studied corrugation angles ($\beta = 31°$, 35°, 40°, 45°, 50°, 55° and 60°).

Plane C includes one contact point between the plates, located in the centre of the unitary cell. The unitary cell includes four additional contact points, located in the corners of the referred cell (planes A and E).

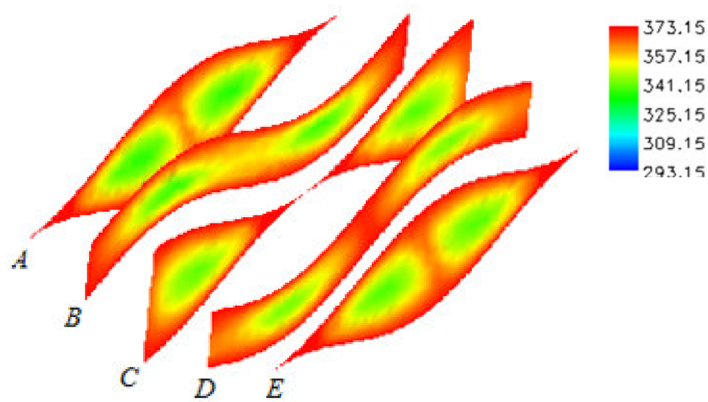


Fig. 4: Temperature profile for $\beta = 31°$ and fifth consecutive unitary cell.

As can be seen in Fig. 3, the average convective heat transfer coefficient from the first half of a unitary cell is superior to that from the second half. In thermal fully developed flow, the average convective heat transfer coefficient from the first half of a unitary cell was 5 to 10% superior when compared with that from the second half, for all the studied values of β.

In Fig. 5 the ratio between the interstitial, u_i, (Fernandes et al., 2007, 2008) and average velocity, u (Fernandes et al., 2007, 2008) is shown. This ratio (u_i/u) is known as tortuosity coefficient (Fernandes et al., 2007, 2008; Dias et al., 2007, 2008). The highest and lowest values of u_i and h were obtained for the same values of $x*$ (see Figs. 3, 4 and 5).

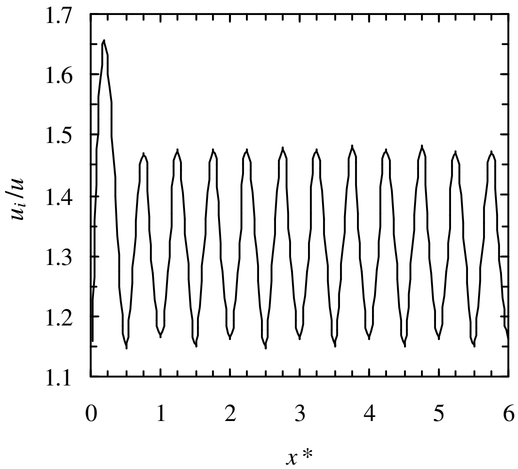
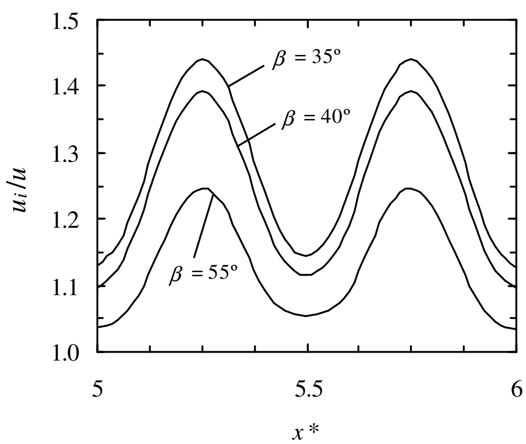
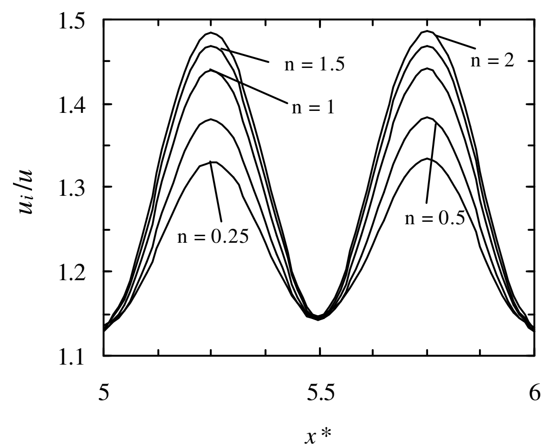


Fig. 5: Local tortuosity coefficient, u_i/u, for $\beta = 31º$.

The tortuosity coefficient decreases with the increase of the value of β (Fig 6(a)) (Fernandes et al. 2007, 2008). Chevron-type plate heat exchangers with low corrugation angle are less prone to particulate fouling, the asymptotic fouling resistance being inversely proportional to the velocity u squared (Thonon et al., 1999). For the same velocity u, lower corrugation angles present higher interstitial velocities u_i (Fig 6(a)). This fact may help to understand why chevron-type PHEs with low corrugation angles are less prone to particulate fouling. With the decrease of the flow index behaviour (see Eq. (3)) the tortuosity coefficient (u_i/u) also decreases, Fig 6(b).



(a)



(b)

Fig. 6: Local tortuosity coefficient. (a) Newtonian fluid and different corrugation angles. (b) Different power-law fluids and $\beta = 35º$.

The average values of the local convective heat transfer coefficients (Fig. 3) in fully developed flow (fifth/sixth unitary cell) allowed the calculation of the average Nusselt number (referred here simply as Nusselt number).

Using Reynolds numbers, Re, superior to 400, Martin (1996) found that the Nusselt number reaches a maximum for corrugation angles located in the range 10º-20º, Fig. 7(a). In the present work (laminar regime), for different power-law fluids, Nusselt number reached a maximum for corrugation angles close to 50º, Fig.7(b). In Figs. 7 and 8, Nu* represents a normalized Nusselt number, i.e., the ratio between the Nusselt number for a given β and the Nusselt number for $\beta = 45°$ (Martin, 1996).

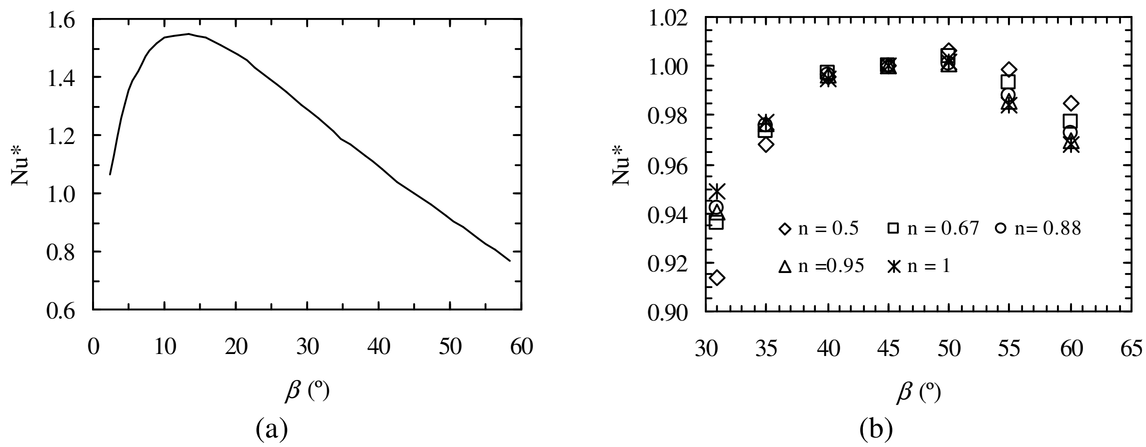


Fig. 7: Normalized Nusselt number. (a) Model from Martin (1996) for Re = 1500. (b) Present work, for Re = 10, different power-law fluids and fully developed flow.

Further simulations with Newtonian fluids allowed to conclude that the value of β in which the maximum Nusselt number is reached depends from the Prandtl number, Fig 8. With the decrease of the Prandtl number this maximum is reached for lower corrugation angles.

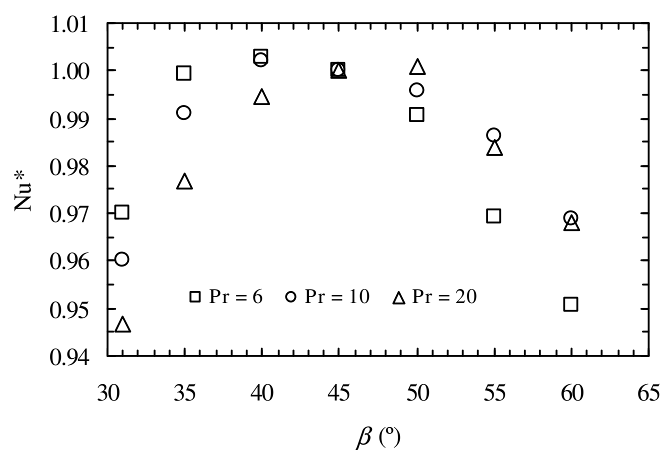


Fig. 8: Normalized Nusselt number in fully developed flow for Newtonian fluids with different Prandtl numbers and Re = 10.

The thermal-hydraulic performance of different PHEs passages can be compared using the area goodness factor, i.e., the ratio between the Colburn factor, j, and Fanning friction factor, f (Metwally and Manglik, 2004). In Fig. 9 it is possible to observe that, due to the shear-thinning effects, the area goodness factor increases with the decrease of n.

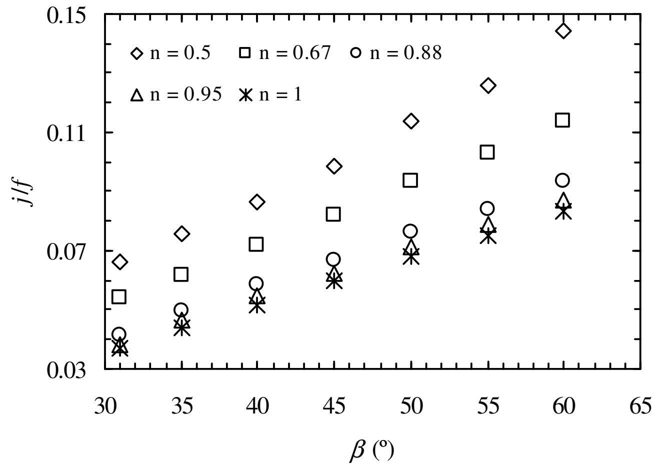


Fig. 9: Area goodness factor in fully developed flow for different power-law fluids and Re = 10.

For a fixed *n*, the area goodness factor increases with the increase of β. This happens since the Fanning friction factor decreases with the increase of β (Fernandes et al. 2007, 2008). It is important to note that the viscosity used in the Reynolds number calculation was the apparent viscosity - Eq.(3) - (Fernandes et al. 2005, 2006).

Conclusions

Using computer fluid dynamics techniques it was studied the local and average characteristics of the flow and heat transfer of Newtonian and non-Newtonian fluids in chevron-type plate heat exchangers passages. The chevron-type plate heat exchangers passages had different corrugation angles but the same area enlargement factor (1.17), this value being commonly used by industrials.

The knowledge of the local properties of the flow and heat transfer in chevron-type plate heat exchangers passages can provide information in order to improve the performance of these equipments.

Our results suggest that the Nusselt number is not inversely proportional to the corrugation angle (defined relatively to the horizontal). For the Newtonian and non-Newtonian fluids used in the present investigation the Nusselt number reached a maximum for corrugation angles located in the approximate range 40º-50º. Shear thinning effects enhance the thermal-hydraulic performance of the plate heat exchangers.

References

Afonso, I. M., Hes, L., Maia, J. M., & Melo, L. F. (2003). Heat transfer and rheology of stirred yoghurt during cooling in plate heat exchangers. *Journal of Food Engineering*, 57, 179-187.

Ayub, Z. H. (2003). Plate heat exchanger survey and new heat transfer and pressure drop correlations for refrigerant evaporators. *Heat Transfer Engineering*, 24, 3-16.

Ciofalo, M., Stasiek, J., & Collins, M.W. (1996). Investigation of flow and heat transfer in corrugated passages – II. Numerical simulation. *International Journal of. Heat and Mass Transfer*, 39, 165-192.

Delplace, F., & Leuliet, J. C. (1995). Generalized Reynolds number for the flow of power law fluids in cylindrical ducts of arbitrary cross-section. *The Chemical Engineering Journal*, 56, 33-37.

Dias, R. P., Fernandes, C. S., Mota, M., Teixeira, J. A., & Yelshin, A. (2007). Permeability and effective thermal conductivity of bisized porous media. *International Journal of Heat and Mass Transfer*, 50, 1295-1301.

Dias, R. P., Fernandes, C. S., Teixeira, J. A., Mota, M., & Yelshin, A. (2008) Permeability analysis in bisized porous media: Wall effect between particles of different size. *Journal of Hydrology*, 349, 470-474.

Fernandes, C. S., Dias, R., Nóbrega, J. M., Afonso, I. M., Melo, L. F., & Maia, J. M. (2005). Simulation of stirred yoghurt processing in plate heat exchangers. *Journal of Food Engineering*, 69, 281-290.

Fernandes, C. S., Dias, R. P., Nóbrega, J. M., Afonso, I. M., Melo, L. F., & Maia, J. M. (2006). Thermal behaviour of stirred yoghurt during cooling in plate heat exchangers. *Journal of Food Engineering*, 76, 433-439.

Fernandes, C. S., Dias, R. P., Nóbrega, & Maia, J. M. (2007). Laminar flow in chevron-type plate heat exchangers: CFD analysis of tortuosity, shape factor and friction factor. *Chemical Engineering and Processing: Process Intensification*, 46, 825-833.

Fernandes, C. S., Dias, R. P., Nóbrega, & Maia, J. M. (2008). Friction factors of power-law fluids in chevron-type plate heat exchangers. *Journal of Food Engineering*, doi: 10.1016/j.jfoodeng.2008.05.022.

García-Cascales, J. R., Vera-García, F., Corberán-Salvador, J. M., & Gonzálves-Maciá, J. (2007). Assessment of boiling and condensation heat transfer correlations in the modelling of plate heat exchangers. *International Journal of Refrigeration*, 30, 1029-1041.

Gschwind, P., Gaiser, G., Zimmerer, C., & Kottke, V. (2001). Transport phenomena in micro heat exchangers with corrugated walls. *Microscale Thermophysical Engineering*, 5, 285-292.

Heggs, P. J., Sandham, P., Hallam, R. A., & Walton C. (1997). Local transfer coefficients in corrugated plate heat exchanger channels. *Transactions of the Institution of Chemical Engineers*, 75, 641-645.

Kakaç, S., & Liu, H. (2002). *Heat exchangers selection, rating, and thermal design* (2nd, ed., pp. 131-136, 373-412). CRC Press, Boca Raton.

Kim, B. K., Tadini, C. C., & Singh, R. K. (1999). Heat transfer in a plate heat exchanger during pasteurization of orange juice. *Journal of Food Engineering*, 42, 79-84.

Kumar, H. (1984). The plate heat exchanger: construction and design. In: Proceedings *First UK National Conference on Heat Transfer*, University of Leeds, Inst. Chem. Symp. Series No. 86, pp. 1275-1288.

Leuliet, J. C., Maigonnat, J. F., & Lalande, M. (1987). Etude de la perte de charge dans de échangeurs de chaleur à plaques traitant des produits non-newtoniens. *Revue Générale de Thermique*, 308-309, 445-450.

Leuliet, J. C., Maigonnat, J. F., & Lalande, M. (1988). Thermal behaviour of plate heat exchangers with Newtonian and non-Newtonian fluids. In: Proceedings *Congress Eurotherm 5* et *Colloque TIFAN*, Compiègne, pp. 1-15.

Martin, H. (1996). A theoretical approach to predict the performance of chevron-type plate heat exchangers. *Chemical Engineering and Processing*, 35, 301-310.

Mehrabian, M. A., & Poulter, R. (2000). Hydrodynamics and thermal characteristics of corrugated channels: computacional approach. *Applied Mathematical Modelling*, 24, 343-364.

Metwally, H. M., & Manglik, R. M. (2000). A computational study of enhanced laminar flow heat transfer in laminar flows of non-Newtonian fluids in corrugated-plate channels. In:

Proceedings ASME *Advances in Enhanced Heat Transfer* , HTD-Vol.365/PID-Vol.4, pp. 41-48.
Metwally, H. M., & Manglik, R. M. (2002). Computational modelling of enhanced laminar flow heat transfer in viscoplastic fluids in corrugated-plate channels. In: Proceedings ASME *International Mechanical Engineering Congress & Exposition*, New Orleans, Louisiana, pp. 1-8.
Metwally, H. M., & Manglick, R. M. (2004). Enhanced heat transfer due to curvature-induced lateral vortices in laminar flows in sinusoidal corrugated-plate channels. *International Journal of Heat and Mass Transfer*, 47, 2283-2292.
Palm, B., & Claesson, J. (2006). Plate heat exchangers: Calculation methods for single- and two-phase flow. *Heat Transfer Engineering*, 27, 88-89.
Rene, F., Leuliet, J. C., & Lalande, M. (1991). Heat transfer to Newtonian and non-Newtonian food fluids in plate heat exchangers: Experimental and numerical approaches. *Transactions of the Institution of Chemical Engineers*, 69, 115-126.
Reppich, M. (1999). Use of high performance plate heat exchangers in chemical and process industries, *International Journal of Thermal Sciences*, 38, 999-1008
Thonon, B., Grandgeorge, S., & Jallut, C. (1999). Effect of geometry and flow conditions on particulate fouling in plate heat exchangers. *Heat Transfer Engineering*, 20, 12-24.
Wanniarachchi, A.S., Ratnam, U., Tilton, B.E., & Dutta-Roy, K. (1995). Approximate correlations for chevron-type plate heat exchangers. In: Proceedings ASME HTD - vol. 314, 1995 *National Heat Transfer Conference*, vol.12, pp. 145-151.