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THREE-DIMENSIONAL CONJUGATE NUMERICAL ANALYSIS OF FIN AND TUBE HEAT EXCHANGERS WITH VARIOUS FIN THERMAL CONDUCTIVITY VALUES AND GEOMETRIC PARAMETERS

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Abstract: It is obvious that heat exchanger performance is highly related with fin material thermal conductivity. In this paper 3-D numerical simulations were performed for conjugate heat transfer and fluid flow characteristics of fin and tube heat exchanger. The effects of the fin materials interacting with five factors: Reynolds (Re) number, fin pitch (FP), fin thickness (FT), tube diameter (TD) and fin length (FL) were examined. However, high thermal conductivity materials may not be suitable under some operating conditions such as food processing. It is found that thermal conductivity of the material slightly increases the effect of the Re number on Nusselt (Nu) number and the thermal conductivity is becoming very important parameter while investigating the effect of tube diameter on Nu number. Similarly the effect of fin length and fin thickness is also affected by fin material. A new correlation is proposed to predict Nu number includes the effect of all these parameters.
Keywords: 3-D Numerical Simulation, Fin and Tube, Heat Exchanger, Fin Materials

FARKLI KANAT ISI İLETİM KATSAYILARI VE GEOMETRİK PARAMETRELER İÇİN KANATLI BORULU ISI DEĞİŞTİRİCİLERİN ÜÇ BOYUTLU BÜTÜNLEŞİK SAYISAL ANALİZİ

Özet: Isı değiştirici performansının, kanatçık malzemesinin ısıl iletkenliği ile yüksek bir alakası olduğu açıktır. Bu çalışmada, kanatlı borulu ısı değiştiricinin bütünleşik ısı transferi ve akış karakteristiklerini belirlemek için üç boyutlu modelleme ile sayısal hesaplamalar yapılmıştır. Kanatçık malzemesinin etkisi şu beş faktör ile birlikte incelenmiştir: Reynolds sayısı (Re), kanatçıklar arası mesafe (FP), kanatçık kalınlığı (FT), boru çapı (TD) ve kanatçık uzunluğu (FL). Bununla birlikte yüksek ısı iletim katsayılarına sahip malzemelerin gıda işleme gibi bazı proseslerde kullanılması uygun olmayabilmektedir. Malzemenin ısı iletim katsayısı, Re sayısının Nu sayısı üzerindeki etkisinde ufak bir artışa sebep olmakla birlikte boru çapının Nu sayısı üzerindeki etkisi incelendiğinde önemli bir parametre olduğu bulunmuştur. Benzer olarak kanatçık uzunluğunun ve kanatçık kalınlığının etkileri de kanatçık malzemesinden etkilenmektedir. Nu sayısının bulunması için bütün bu parametrelerin etkilerini içeren yeni bir korelasyon önerilmiştir.
Anahtar Kelimeler: 3-B sayısal modelleme, Kanatlı borulu, Isı değiştirici, Kanatçık malzemesi

NOMENCLATURE

A	area, m^2
A_c	cross sectional area of the fin
A_f	fin surface area, m^2
A_l	lateral area of the fin
c_p	specific heat at constant pressure, J/kgK
D	tube diameter, mm
FC	friction coefficient
FL	fin length, mm
FL^*	non-dimensional fin length
FP	fin pitch, mm
FP^*	non-dimensional fin pitch
FT	fin thickness, mm
FT^*	non-dimensional fin thicknes
h	heat transfer coefficient, W/m^2K
k	thermal conductivity, W/mK
L	fin length, mm
LMTD	logarithmic mean temperature difference, K
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number
P	pressure, Pa
Re	Reynolds number based on tube diameter
T	temperature, K
TD	Tube Diameter, mm

u, v, w	x, y, z velocity components, m/s
W	wetted perimeter of the fin
U_c	maximum velocity at minimum cross-sectional area, m/s
U_{in}	air inlet velocity, m/s
Greek symbols	
η	fin surface efficiency
μ	dynamic viscosity of air, kg/ms
υ	kinematic viscosity of air, m^2/s
ρ	density, kg/m^3
Subscripts	
A	ambiant
in	inlet
out	outlet
tw	tube walls
f	fin surface
Superscripts	
*	normalized or non-dimensional

INTRODUCTION

Fin and tube heat exchangers are widely used in many engineering applications such as automotive radiators, air conditioning evaporators and condensers and liquid or oil coolers. Due to the optimization and design problems, it is important to determine the Nusselt number (Nu), heat transfer rate and pressure drop in fin and tube heat exchangers. Several researchers made experiments to identify these parameters. Kays and London (1984) made experiments for compact heat exchangers and determined j factors for various heat exchangers. Plain fin-tube heat exchangers were also studied and many geometrical parameters were investigated. Wang et al. (2000a, 2000b) were made an experimental study on plain fin-tube heat exchangers and gave a correlation for heat transfer. Kim and Kim (2005) investigated heat exchangers with large fin pitch and obtained a correlation from experimental data. Choi et al. (2010) studied Coulborn j factor and friction factor (f) in large fin pitch heat exchangers with various geometric parameters. Halici and Taymaz (2006) examined effects of the distance between tubes according to fin surface is wet or dry. They showed that j and f factors are higher in wet surfaces than that of dry surfaces. Decreasing the distance between tubes increases both heat transfer and pressure loss in heat exchanger. Yan and Sheen (2000) compared plain fins characteristics with different fin types. They showed that f and j factors were higher in louvre fins. Wang et. al. (2015), compared different types of fin geometry experimentally and found that different fin types have different heat transfer characteristics. A louvered fin and elliptical tube heat exchanger were investigated both experimentally and numerically by Karthik et. al. (2015) and Du et. al. (2013) investigated the effect of the delta winglet pairs on wavy fins. It was shown, that air inlet angles to the heat exchanger have a significant effect on pressure drops. Different methods can be used to determine the heat transfer characteristics. Ay et al. (2002), Hueng et al. (2009) and Du et. al. (2014) calculated the local heat transfer coefficients on fin surfaces by infrared thermography. The hot-wire anemometer and the liquid-crystal technique are also used for the turbulence and mean flow and the heat transfer measurements, respectively (Kuvvet and Yavuz, 2012). Besides experimental studies, numerical methods are also an important tool to determine the characteristics of this type of heat exchangers. Sheu and Tsai (1999) showed that slit fins had better thermal characteristics than plain fins using 3-Dimensional Numerical analysis. Plain fins are also investigated with 3-dimensional numerical analysis of Tao et al. (2007) with the field synergy principle. The effect of the air inlet flow maldistribution was shown using 3D CFD analysis by Yaici et. al. (2014). Romero-Méndez et al. (2000) examined the effect of fin pitch on heat transfer both numerical and experimental. They found that increasing the fin pitch had a similar effect with increasing Re number. He et al. (2005) examined the parameters which affect the thermal characteristics of heat exchangers using the field synergy principle. They focused on Re number, fin pitch, tube number and tube alignment. They pointed out that there was an optimum fin pitch value where Nu maximum is. Peláez et al. (2010) studied air side and water side together in their simulations and acquire more accurate results for plain fin heat exchangers. Many heat exchangers with different fin types and vortex generators were examined using the CFD tool (Karthik and Khan, 2015; Wais, 2014; Gong et al., 2013; Özen and Altınışık, 2012). Juan and Qin (2014) optimized the geometric parameters of a plain fin and tube heat exchanger in order to obtain maximum heat transfer rate and minimum pressure drop.

Where high conductivity materials are often utilized in heat exchangers, this kind of materials may not be suitable under some operating conditions. Pressure loss, temperature and corrosion are the main parameters in material choices (Webb, 1983; Cevallos et al., 2012). Especially temperature resistance and corrosion properties cannot be in desired values and usage of low thermal conductivity materials are mandatory. Thermal conductivity of fin became important in plastic heat exchangers and affects the heat exchangers performance (Chen et al., 2009). Meng and Jacobi (2011) optimized a polymer tube-bundle heat exchanger in aspect of cost effectiveness using genetic algorithm. Chiu and Chen (2002) studied the thermal conductivity effect and the optimum fin length with various thermal conductivity values of fin using the Adomian decomposition method to solve nonlinear equations. Lizardi et al. (2004) examined the effect of thermal conductivity on a vertical fin with condensation process. They showed that thermal conductivity of fin affected the mass flow rate of condensed liquid. Xie et al. (2009) compared the heat transfer rates of Al fins with Cu fins. While thermal conductivity values of these materials are closing each other, thermal characteristics are also similar.

Many researchers focused on the effect of the fin and tube geometry however, thermal properties of the fin change the heat transfer performance of the heat exchanger dramatically. The aim of this paper is to investigate the effect of thermal conductivity values of a fin material together with the several geometry parameters which is not combined and studied in the present literature as far as the authors' knowledge. In this study, the effect of fin thermal conductivity on thermal performance of plain-fin tube heat exchangers is investigated. It is found that the effects of geometrical parameters are influenced substantially by the fin material thermal conductivity. To understand the effect of fin material thermal conductivity value as well as other geometric parameters may be helpful for the design and analysis of heat exchangers.

MODEL DESCRIPTION

Physical Model

A two-row plain fin-and-tube heat exchanger was employed in the simulations. Due to symmetry conditions in the z and y-direction, only half of the fin and the half of the air between fins are selected as a solution domain. Figure 1 shows the selected computational domain of the heat exchanger.

The dimensions and the zones of the selected portion of the heat exchanger are illustrated in Fig 2. Two additional zones are considered in the computational domain: an inlet zone to study the incidence of the air flow over the fin and an outlet one, to minimize the backflow during the simulations.

Heat exchanger dimensions were chosen according to model, which was studied by Pelaez et al. (2010) and He et al. (2005) due to validation and comparison of the results. Temperature distribution is to be determined in both solid fin and fluid domain in order to examine the thermal conductivity effect.

Flow Considerations

Some assumptions were made to simplify the flow as follows: The fluid is considered to be incompressible, homogeneous constant properties in both fluid and solid zones and steady state condition. The Reynolds number of the simulations in this work varies between 500 and 5000 based on the tube diameter. There is not a certain criterion to determine whether the flow is laminar or turbulent. If the flow is regarded as a flow between parallel plates, transition to laminar flow takes place for Re number 2300 (Incropera, 2002). Characteristic length is defined as the double distance between surfaces. According to this approach the maximum Re number occurred 2190 in this work. The flow can be considered as a cross flow past a cylinder, the turbulent flow occurs at Re number 2 x 10^5, based on the tube diameter. According to the literature, laminar flow consideration is acceptable for the simulations. Previous works on similar 3-D simulations justified the use of laminar steady models (He et al., 2005; Pelaez et al., 2010; He et al., 2013; Xie et al., 2009; Iniguez et al., 2015; Zhou et al., 2011).

Governing Equations

Simulations are performed using Ansys 12 Fluent on this paper. The code is based on the finite volume method consisting in the discrete approximation of the volume and surface integrals of the Navier-Stokes equations in steady state. The governing equations for continuity, momentum and energy can be expressed as follows:

Continuity equation:

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{1}$$

Momentum equation:

$$\frac{\partial}{\partial x_i}(u_i u_k) = \frac{\mu}{\rho}\frac{\partial}{\partial x_i}\left(\frac{\partial u_k}{\partial x_i}\right) - \frac{1}{\rho}\frac{\partial p}{\partial x_k} \tag{2}$$

Energy equation:

$$\frac{\partial}{\partial x_i}(u_i T) = \frac{k}{\rho c_p}\frac{\partial}{\partial x_i}\left(\frac{\partial T}{\partial x_i}\right) \tag{3}$$

Because of the conjugated type of the problem, the fin zone is considered as a part of the solution domain. Similar treatments can be found in references (He et al., 2005; Xie et al., 2009).

Boundary Conditions

The velocity and temperature boundary conditions are as follows:

Velocity:

Inlet boundary: $u = U_{in}$, $v = w = 0$

Outlet boundary: $P_{out} = 0$

Tube and fin walls: $u = v = w = 0$

The right, upper and lower sides of the air side solution domain is treated as symmetric:

Temperature:

Inlet boundary: $T=T_{in}$

Tube walls: $T=T_{tw}$

The right, upper and lower sides of the air side solution domain is treated as symmetric and the left, upper, and lower sides of the fin side solution domain are treated as adiabatic: $\partial T/\partial y = \partial T/\partial z = 0$

The numerical values of the boundary conditions are as follows: U_{in}= 1.4 m/s, T_{in}=300 K and T_{tw}=280 K. The governing equations were discretized by the finite volume method and the second order scheme was used to discretize the convection-diffusion terms. The coupling of the pressure and velocity is implemented by SIMPLEC algorithm (Vandoormaal and Raithby, 1984).

The heat conduction equation for a fin is given in Eq. (4).

$$\frac{d}{dz}\left(A_c\frac{dT}{dz}\right)-\frac{h}{k}\frac{dA_l}{dz}\left(T-T_a\right) \qquad (4)$$

In this equation A_c and A_l are the cross sectional and lateral area of the fin, respectively. T and T_a is the temperature of the fin and ambient air respectively

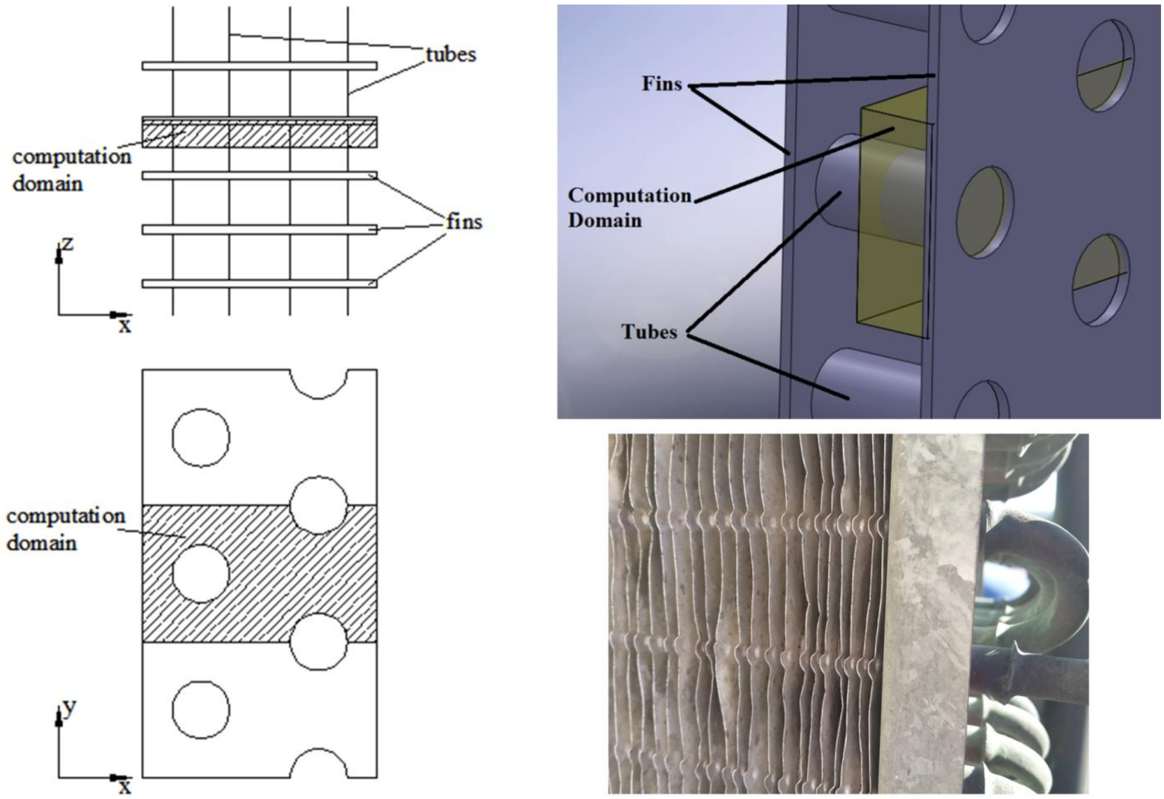

Figure 1. Computation domain of heat exchanger

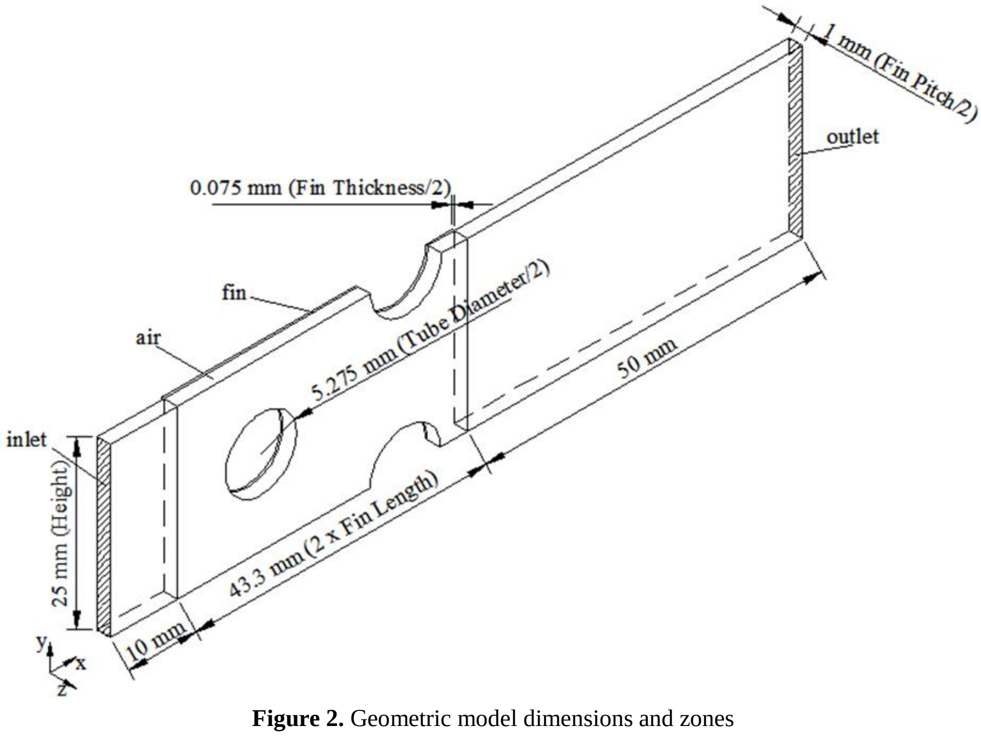

Figure 2. Geometric model dimensions and zones

NON-DIMENSIONAL PARAMETERS

The Reynolds number is based on the air properties and is defined as

$$\mathrm{Re} = \frac{U_c D}{\upsilon} \tag{5}$$

Where U_c is the velocity at the minimum cross sectional area.
Friction coefficient (FC) was calculated by Eq. (6).

$$FC = \frac{\Delta P \times D}{\rho U_c^2 (FL)} \tag{6}$$

Where $\Delta P = P_{in} - P_{out}$, is the pressure drop across the heat exchanger and average pressure is evaluated by Eq. (7).

$$P = \frac{\int P dA}{A} \tag{7}$$

From the numerical results the averaged Nusselt number was defined in Eq. (8).

$$Nu = \frac{hD}{k} \tag{8}$$

Where D is the tube diameter, h is the convective heat transfer coefficient for air, k is the heat conduction coefficient of the air. h was calculated using Eq. (9).

$$h = \frac{\dot{m} c_p (T_{in} - T_{out})}{A \Delta T_{\ln}} \tag{9}$$

Where $\dot{m}$ is air flow rate, c_p is specific heat, A is the heat transfer surface area, $\Delta T_{\ln}$ is the log-mean temperature difference. T_{in} and T_{out} was calculated by $\int T\rho \left|\vec{\upsilon} \cdot d\vec{A}\right| \Big/ \int \rho \left|\vec{\upsilon} \cdot d\vec{A}\right|$ on the inlet and outlet surfaces.

Nu number was normalized by dividing the Nu number derived from minimum value of each parameter. Normalized Nu numbers (Nu*) was calculated by Eq. (10-14).

$$Nu^*_{\mathrm{Re}=500} = \frac{Nu}{Nu_{\mathrm{Re}=500}} \tag{10}$$

$$Nu^*_{TD=5} = \frac{Nu}{Nu_{TD=5}} \tag{11}$$

$$Nu^*_{FL=12.5} = \frac{Nu}{Nu_{FL=12.5}} \tag{12}$$

$$Nu^*_{FT=0.1} = \frac{Nu}{Nu_{FT=0.1}} \tag{13}$$

$$Nu^*_{FP=0.75} = \frac{Nu}{Nu_{FP=0.75}} \tag{14}$$

The surface efficiency η, is defined as the actual heat transfer rate for the fin and base when the fin is at same base temperature T_{tw} and can be described by Eq. (15) (Xie et al., 2009).

$$\eta = \frac{\tanh(m(FL))}{m(FL)} \tag{15}$$

Where

$$m = \sqrt{\frac{hW}{kA_b}} \tag{16}$$

In Eq. (16), W is the wetted perimeter and A_b is the base area of the fin. Actual heat transfer rates were computed from the results of conjugate simulations. All cases were recalculated to evaluate the heat transfer rate when the fin is at same base temperature.

MODEL VALIDATION

A grid system of 186x50x25 is used in computation domain. A mesh convergence analysis of the numerical solutions was made to ensure the validity of the numerical results. For this purpose, four different grid systems, 206x56x27, 186x50x25, 168x45x23 and 130x35x18 were tested. Despite Nu number in 206x56x27 grid system is higher than that of 186x50x25 the difference of Nu number between two grids is smaller than 0.1%. Thus the 186x50x25 grid system is used in computations to save computer resources (Figure 3). Nu numbers versus Re numbers were compared with the studies of Peláez et al. (2010) and He et al. (2005). Nu numbers are also presented that obtained from correlations given by Wang and Chi (2000b). Peláez et al. (2010) and He et al. (2005) studied the same geometry with same dimensions in their works while the same model was examined in this study. Wang and Chi (2000a; 2000b) used considerable amount of experimental data to develop a correlation. The ranges of the correlation are as follows: number of tube row: 1 – 6, TD: 6.35 – 12.7 mm, FT: 1.19 – 8.7 mm, FP: 17.7 – 31.75 mm, FL: 12.4 – 27.5 mm. As seen in Figure 4., a good agreement was obtained for the variation of Nu with Re for a wide range of Re numbers. The variation of friction coefficient (FC) with Re are also close to experimental and numerical data in the literature.

RESULTS AND DISCUSSION

After this section, simulations were performed in conjugate type to analyze the effect of thermal conductivity with various parameters: Re number, fin pitch, tube diameter, fin length and fin thickness. Parameters are analyzed individually. It is obvious that, high thermal conductivity of a fin means smaller thermal resistance thus; heat transfer performance can be improved by increasing fin thermal conductivity. For this reason, Nu increases while increasing k, in all cases.

Fin efficiency is highly related to thermal conductivity due to the definition of η (Eq. (15)). Because of the characteristics of the hyperbolic tangent function, fin efficiency will rise while thermal conductivity is increasing.

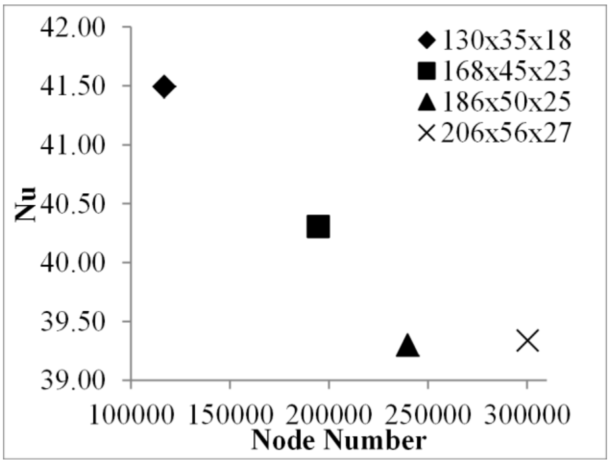

Figure 3. Results of grid independence tests

It is also clear that, increase of Re causes a decrease of fin efficiency where h is enhanced by velocity.

The Effect of Thermal conductivity with Re number

The performed simulations were for the values of Re number between 500 and 5000 and the air inlet velocity between 0.4 m/s and 4.0 m/s, tube diameter 10.55 mm, fin length 21.65 mm, fin thickness 0.15 mm and fin pitch 2.0 mm. As a known phenomenon Nu is highly related with Re where Nu increases with increase in Re (He et al., 2005; Pelaez et al., 2010). This fact was shown by many researchers by experimentally (Wang and Chi, 2000a; Wang et al., 2015; Du et al., 2014). However, Re number has a greater effect in case of using low thermal conductivity materials. The dimensionless Nusselt number of the lower fin thermal conductivity is higher than that of high ones as seen in Figure 5a, 5b. This effect is pointed out more notable when considering surface efficiencies as seen in Figure5c. It is also known that FC is highly related to Reynolds number, FC decreases with increasing Re, this can be seen in Figure 5d .

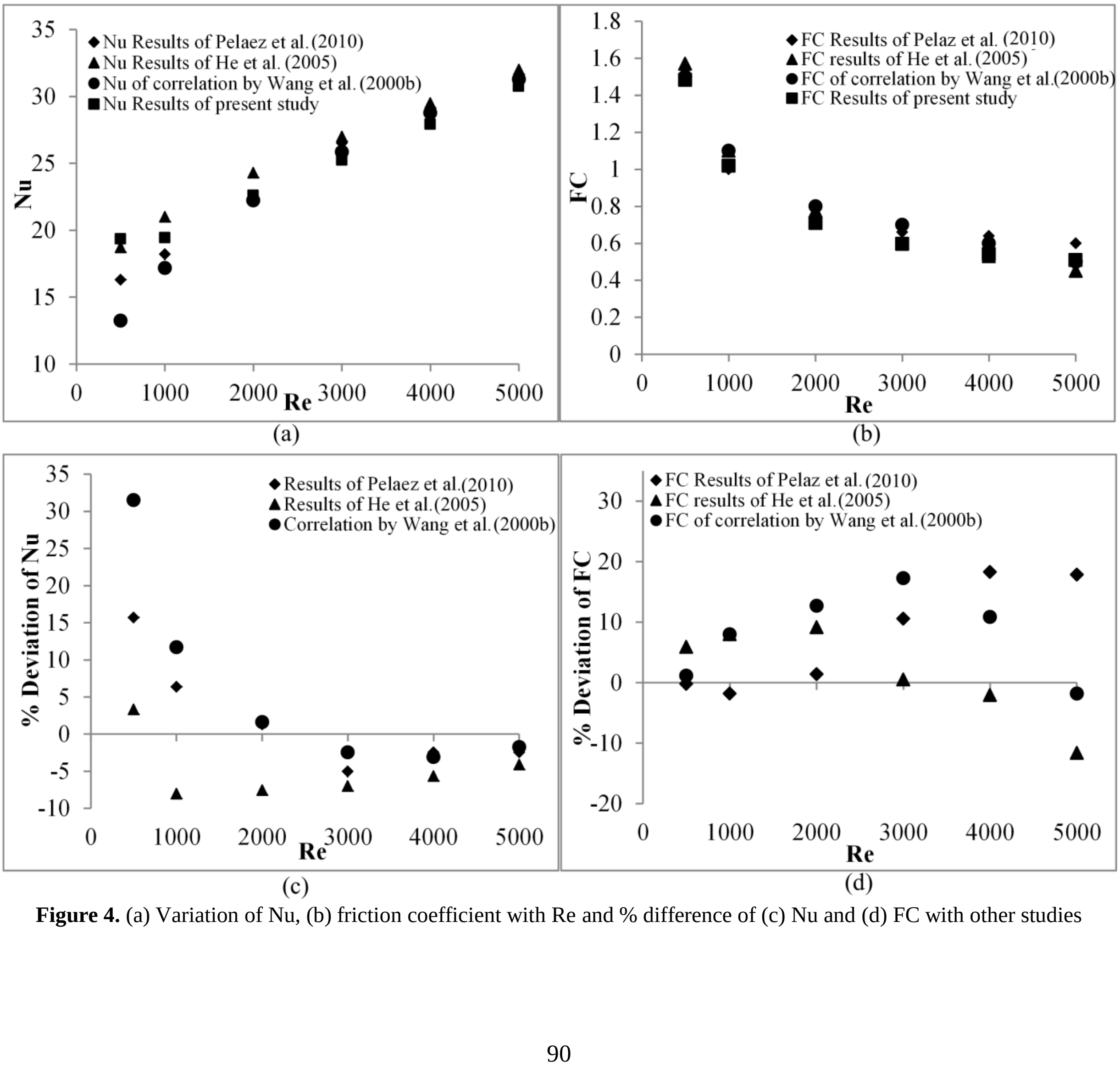

Figure 4. (a) Variation of Nu, (b) friction coefficient with Re and % difference of (c) Nu and (d) FC with other studies

While surface efficiencies in high fin thermal conductivity become close each other, Re number alters the surface efficiency at high fin thermal conductivity. The thermography results of Ay et. al. (2002) shows that the maximum temperature difference on fins occurs in the entrance region of the fin while the tube walls are kept at constant temperature 60 °C. Similarly, examining the temperature distribution on fin surface, it is seen that temperature of near tube region is colder than that of other parts of the fin (Figure 6a). Increase in fin thermal conductivity causes an increment in the cold region area and at k = 200 W/m^2K, almost all fin surfaces is becoming at tube wall temperature (Figure 6f). Therefore fin surface efficiency close to 1.0 at high thermal conductivity values. Thus the heat transfer rates, Nu numbers and fin efficiency is enhanced by increasing the thermal conductivity of fin material

The Effect of Thermal Conductivity with Tube Diameter

Tube diameter varies between 5.0 and 15.0 mm, Re = 3000, fin length is 21.65 mm, fin thickness is 0.15 mm and fin pitch is 2.0 mm. Previous studies showed that there is a strong dependence of the Nu number with the tube diameter (Pelaez et al., 2010; Xie et al., 2009). As the tube diameter decreases, fin conductivity tends to be less effective on $Nu^*_{TD=5}$ and thermal conductivity becomes insignificant (Fig 7a, 7b). It is clear that Nu changes with tube diameter due to definition of Nu where specific length is tube diameter. There are big differences in heat transfer coefficients (h) between various tube diameters due to temperature gradients on low conductivity fin surface material and these differences in h will be increased by the effect of tube diameter. Therefore $Nu^*_{TD=5}$ shows big differences in low conductivity values of fin material. Since thermal conductivity is increased, the differences in h will be decreased hence variation in $Nu^*_{TD=5}$ will be depended on only tube diameter. As seen from Figure 7c, surface efficiencies show wide differences in low thermal conductivity values and large tube diameters have high surface efficiencies. It is obvious that, FC is highly related to tube diameter since pressure drop increases with increasing the tube diameter (Figure 7d).

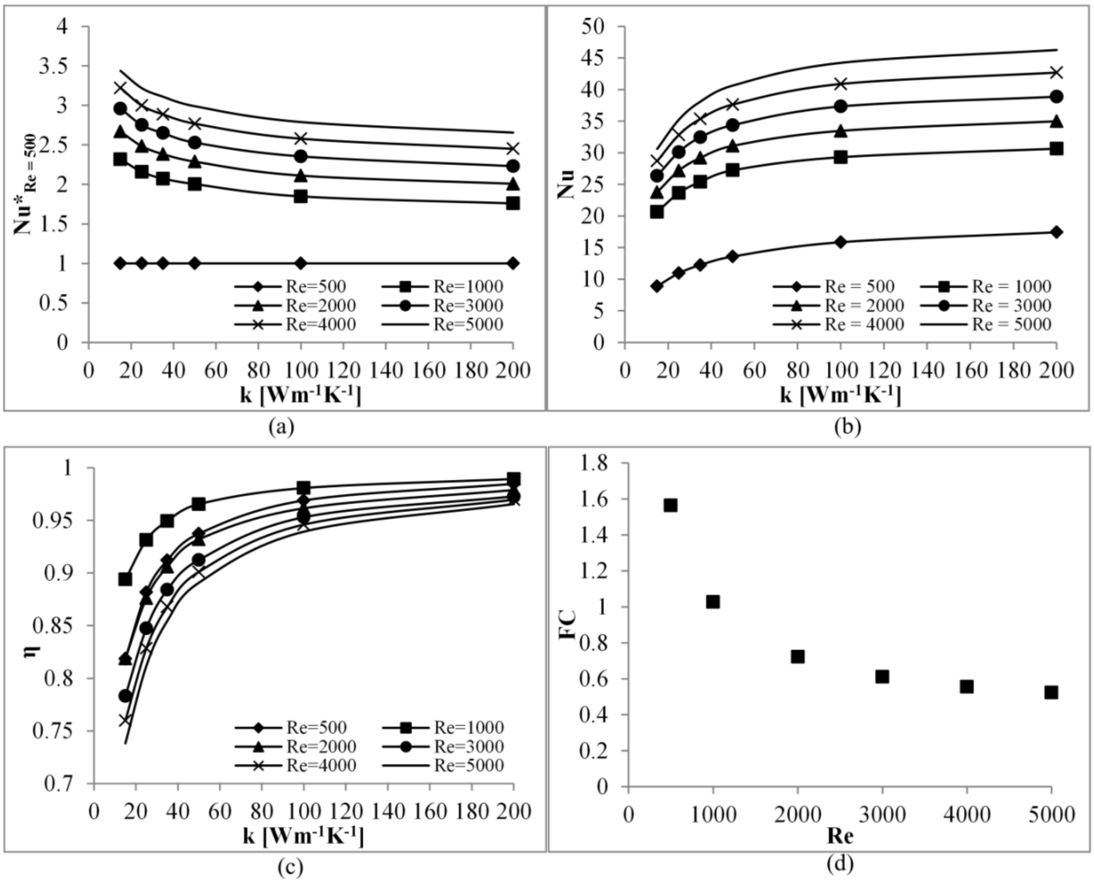

Figure 5. The effects of thermal conduction with Re number on (a) Nu*Re = 500, (b) Nu, (c) fin efficiency and (d) variation of the FC with Re

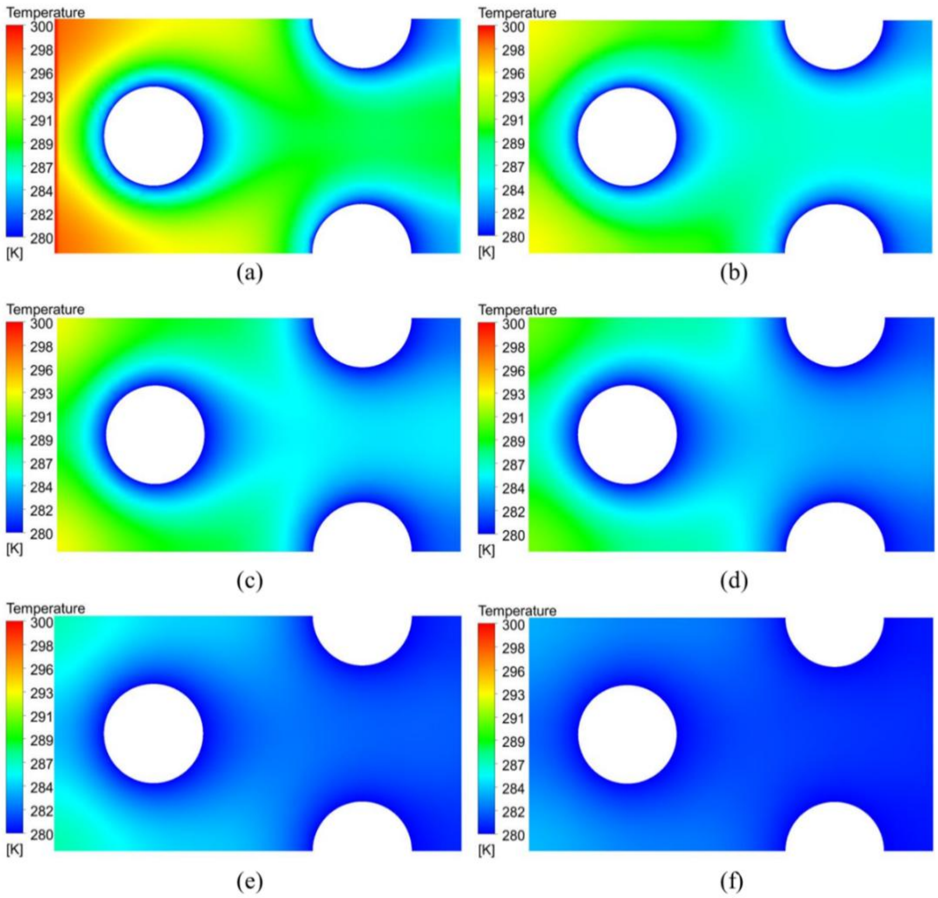

Figure 6. Temperature distribution on fin surface for the thermal conductivity value of (a) 15W/mK, (b) 25 W/mK, (c) 35 W/mK, (d) 50 W/mK, (e) 100 W/mK and (f) 200 W/mK for Re = 2000

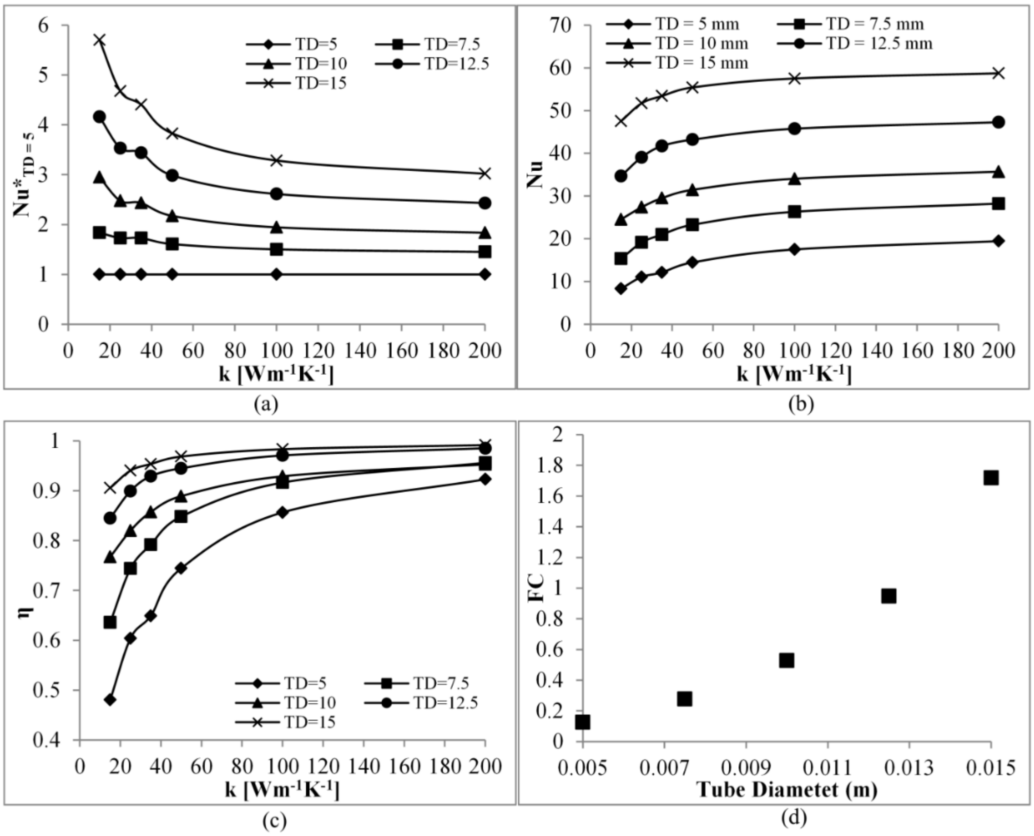

Figure 7. The effects of thermal conduction with tube diameter on (a) $Nu^*_{TD=5}$, (b) Nu, (c) fin efficiency and (d) variation of FC with tube diameter

The Effect of Thermal Conductivity with Fin Length

The range selected for the fin length varies between 12.5 and 30.0 mm, Re number is 3000, tube diameter is 10.55 mm, fin thickness is 0.15 mm and fin pitch is 2.0 mm. The tubes maintain their position, centered in the fin. Previous studies showed that decreasing the fin length increases the Nu number (He et al., 2005; Pelaez et al., 2010; Xie et al., 2009). Additionally the thermal conductivity effect tends to disappear while decreasing fin length. Heat transfer rates are improved with increasing heat transfer surface area. However, heat transfer coefficient is decreased since the surface area has greater effect on the heat transfer coefficient. The differences disappear in $Nu^*_{FL=12.5}$ values while the heat transfer coefficient increases due to increase in thermal conductivity (Figure 8a, 8b). Using long fin lengths also has a negative effect on surface efficiencies. As seen in Figure8c surface efficiencies do not vary significantly with the fin length in high fin conductivities. Pressure drop rises with increasing the fin length on the other hand, FC is inversely proportional with FL (Eq. (6)) as a result, FC decreases (Figure 8d) with increasing fin length.

The Effect of Thermal Conductivity with Fin Thickness

The fin thickness varies between 0.1 and 0.3 mm, Re number is 3000, tube diameter is 10.55 mm, fin length is 21.65 mm, fin pitch is 2.0 mm. The increase in the fin thickness causes an increase in Nu number and fin efficiency in low fin conductivities in low fin conductivities (Figure 9b, 9c). Fin cross-section area is increased by using thicker fins and heat transfer from the fin surface will be enhanced. While it does not show a significant change in Nu numbers for high thermal conductivity materials, thicker fins causes an increase in heat transfer rates and heat transfer coefficients for low thermal conductivity materials as seen in Figure 9. However, previous studies (Pelaez et al., 2010) showed that there is not a significant relation between the fin thickness and Nu number and Tao et. al. (2007) pointed out the thermal resistance of the fin which causes a decrease in heat transfer capacity. Fin thickness has a considerable effect on Nu number and it is also valid for surface efficiencies in low conductivities. Approximately 40% increment in Nu numbers can be obtained by using thicker fins. It can be seen in Figure 9d that, fin thickness has a minor effect on FC.

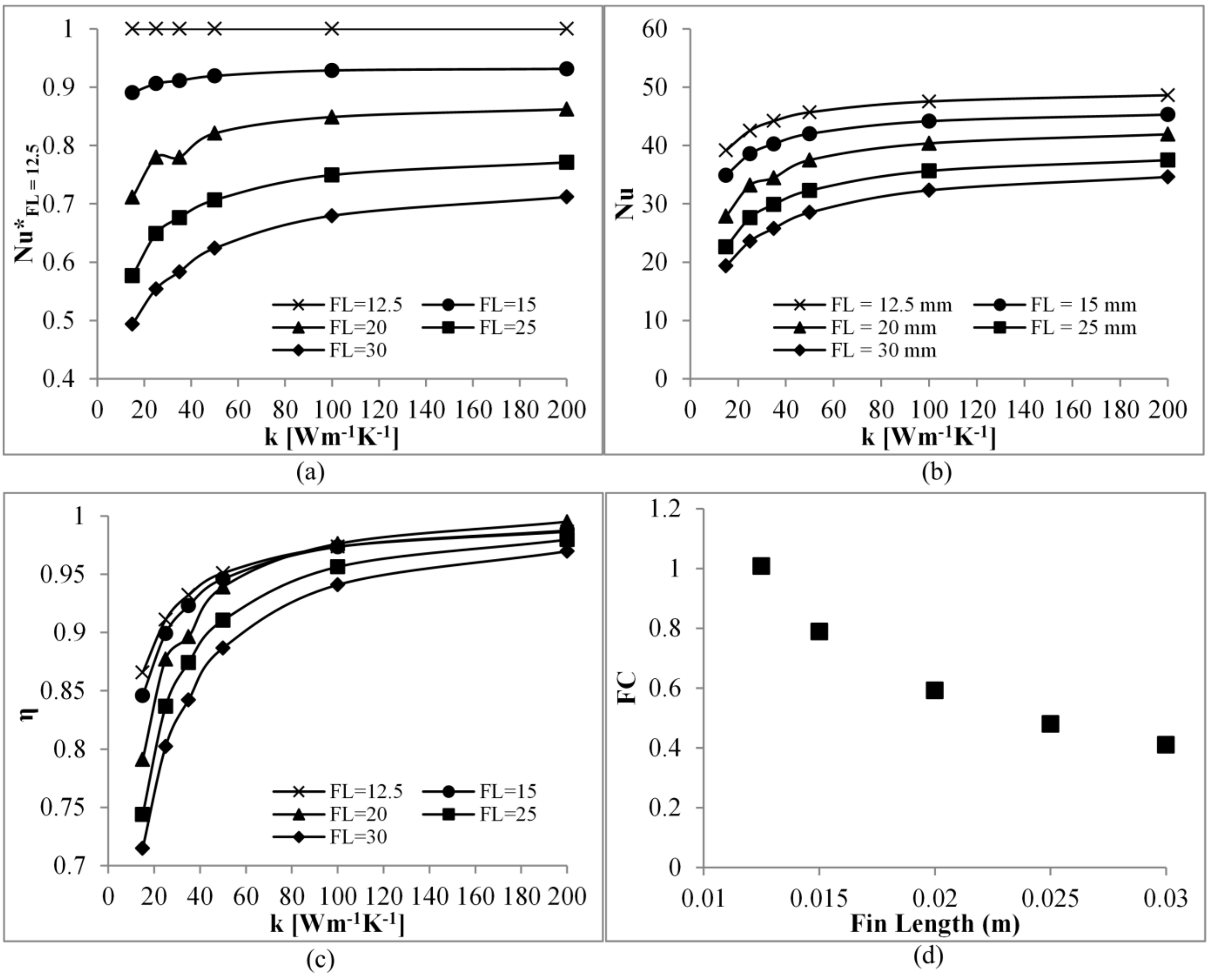

Figure 8. The effects of thermal conduction with fin length on (a) $Nu^*_{FL=12.5}$, (b) Nu, (c) fin efficiency (d) and variation of FC with fin length

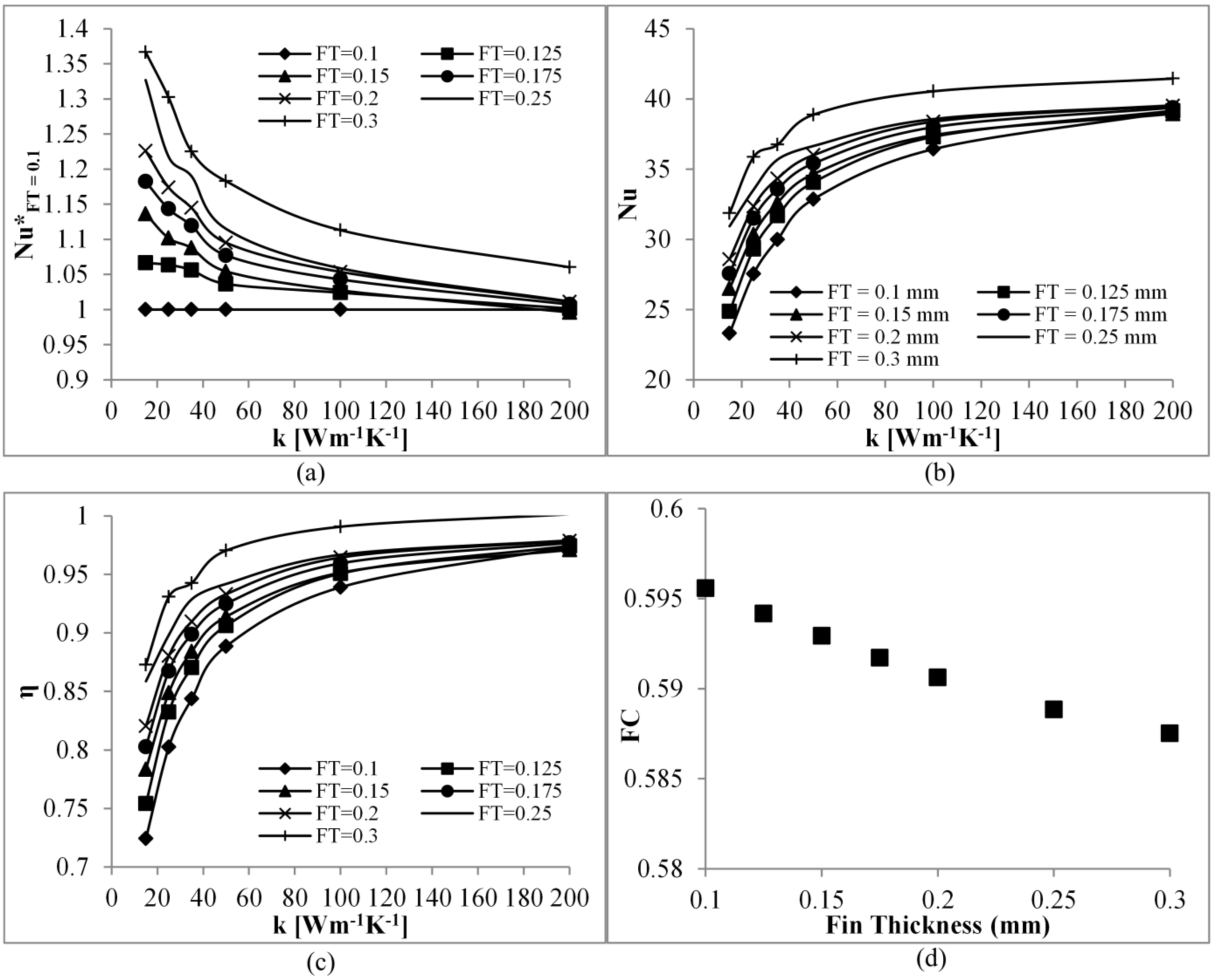

Figure 9. The effects of thermal conduction with fin thickness on (a) $Nu^*_{FT=0.1}$, (b) Nu, (c) fin efficiency, and (d) variation FC with fin thickness

The Effect of Thermal Conductivity with Fin Pitch

The fin pitch varies between 0.75 and 4.0 mm, Re number is 3000, tube diameter is 10.55 mm, fin length is 21.65 mm and fin thickness is 0.15 mm. The Nusselt number increases as the fin pitch is reduced (Kim and Kim, 2005; He et al., 2005; Pelaez et al., 2010; Xie et al., 2009). The mass flow rate is greatly influenced by increasing or decreasing fin pitch where cross flow section changes. Any increase in mass flow rate causes a drop in temperature difference between inlet and outlet of the heat exchanger. It is not seen any significant drop in Nu numbers since the heat transfer rates decrease in case of using low thermal conductivity material. Temperature difference increases in high thermal conductivity materials due to heat transfer rates higher than that of low thermal conductivity materials, even mass flow rate increases and fin pitch becomes important for high thermal conductivity materials. There are significant differences between various fin thermal conductivity values (Figure 10a, 10b). The effect of fin pitch can be neglected in low fin thermal conductivity materials. As seen in Figure 10c fin efficiency and Nu number are not influenced by fin pitch larger than 1.5 mm. Fin thermal conductivity is the major factor which affects the fin efficiency. The effect of the FP on FC is given in Figure 10d and it is seen that, FC increases in case of reducing the distance between the fins while the boundary layer can be corrupted thus pressure drops increases in small FP values.

The effect of the FT/FP ratio on Nu was given in Figure 11. It can be seen in Figure 11 that Nu number is increasing remarkably with the rising FT/FP ratio for the case of the low thermal conductivity (k=15 W/mK). However, Nu number increase with the FT/FP ratio is reducing when the thermal conductivity of the fin material having the high values. And, FT/FP ratio has no significant effect while thermal conductivity has higher values (> 100 W/mK). It can be concluded that the FT/FP ratio is an important parameter for the design of the fin and tube heat exchangers when the fin material has the low thermal conductivity values.

Correlation for Nu Number

A correlation for Nu number was developed that includes low thermal conductivity fin material effect using dimensionless parameters Re, $FL^* = FL/TD$, $FT^* = FT/TD$, $FP^* = FP/TD$ and $k^* = k/k_a$ (Eq. 17). k_a is the thermal conductivity of the air (0.0263 W/mK) and assumed constant. Computed results from 174 cases are used to obtain a correlation in this study. The ranges of geometric parameters are given in Table 1.

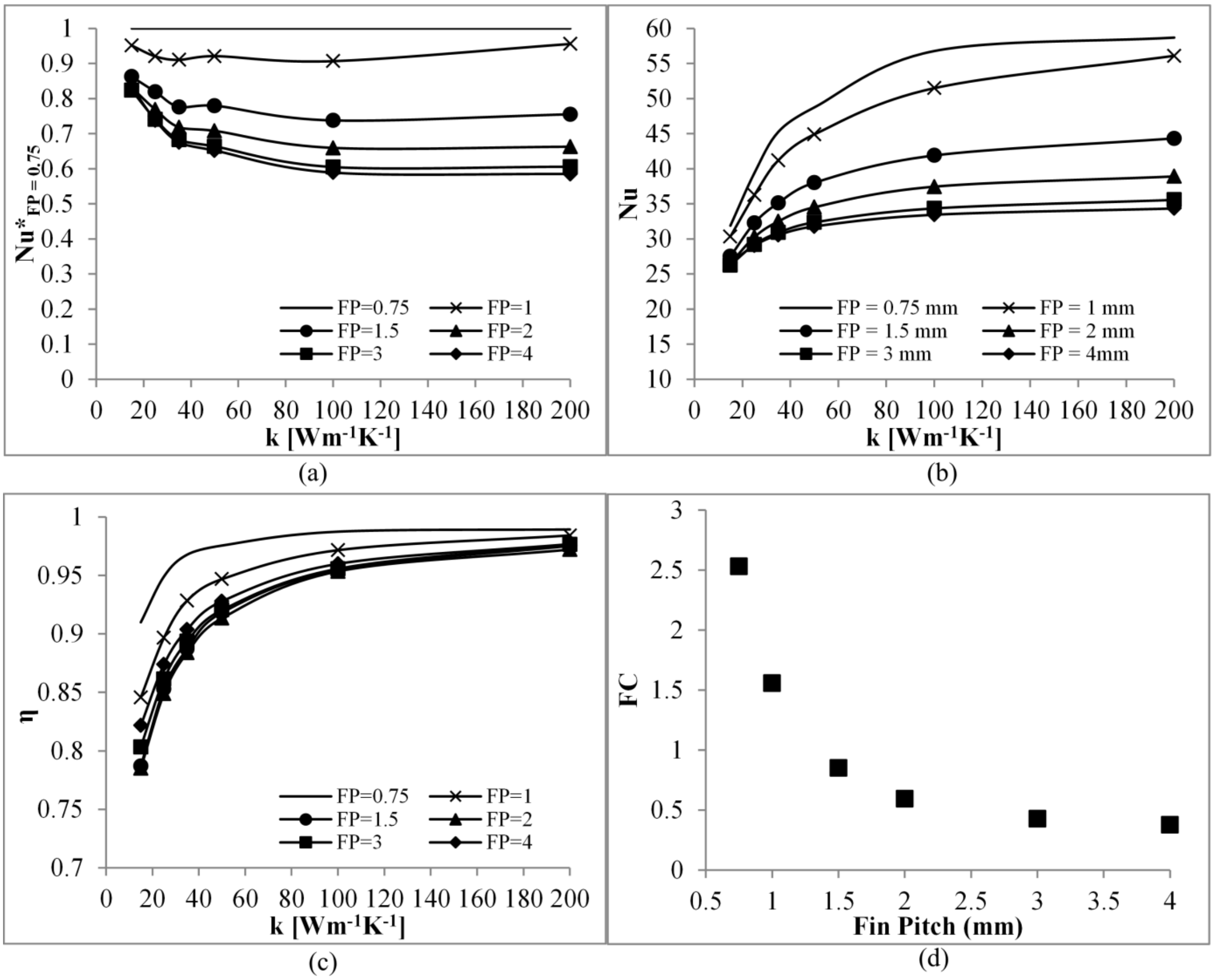

Figure 10. The effects of thermal conduction with fin pitch on (a) $Nu^*_{FP\,=\,0.75}$, (b) Nu, (c) fin efficiency and (d) variation of FC with fin pitch

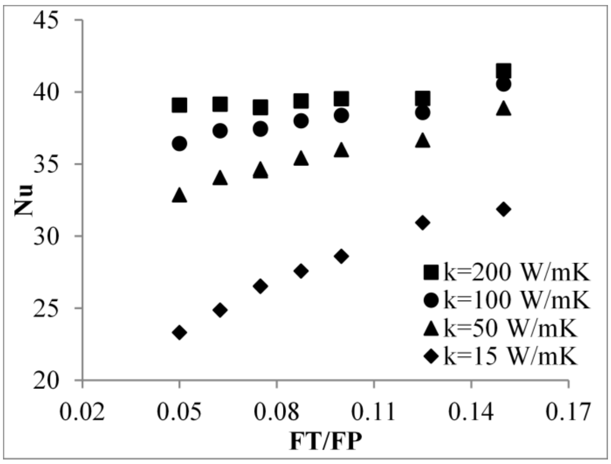

Figure 11. The effect of FT/FP radio on Nu number.

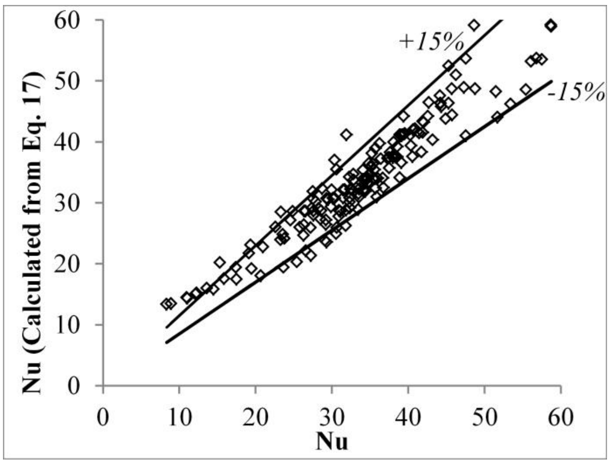

Figure 12. Comparison of the numerical and calculated data using the present correlation

This correlation can be used with the average deviation of 7.33. 87.4% of calculated data from correlation are placed within the ± 15% region of Nu derived from solutions (Figure 12).

Correlation includes both all geometric parameters and heat conduction coefficient of fin material.

$$Nu = 0.37\,\mathrm{Re}^{0.42}(FP^*)^{-0.37}(FT^*)^{0.01}(FL^*)^{-0.66}(k^*)^{0.14} \quad (17)$$

Table 1. Ranges of geometric parameters

Parameters	Interval
Re Number (Re)	500 - 5000
Thermal Conductivity (k) (W/mK)	15 - 200
Tube Diameter (m)	0.005 - 0.015
Fin Length (m)	0.0125 - 0.030
Nondimesional Fin Length- FL*	0.83 - 6
Fin Pitch (m)	0.00075 - 0.004
Nondimesional Fin Pitch- FP*	0.05 – 0.8
Fin Thickness (m)	0.0001 - 0.0003
Nondimesional Fin Thickness-FT*	0.0067 – 0.06

CONCLUSIONS

Geometrical parameters of fin and tube heat exchangers are main criteria to obtain an optimal design where thermal conductivity of fin material is neglected in most cases. However, high thermal conductivity fin materials may not be suitable under some processes and operating conditions.

Especially hygienic conditions, temperature resistance and corrosion properties cannot be in desired range and usage of the low thermal conductivity materials can be mandatory. In this case, the effect of the thermal conductivity on the heat exchanger thermal performance must be considered in the design. In this study, it is shown that heat conduction coefficient is also one of the major parameter related to the heat exchanger's design and its thermal performance. A 3-D numerical conjugate simulations are conducted to study the influence of the fin thermal conductivity with Re number, fin pitch, tube diameter, fin length and fin thickness on the Nu number and fin efficiency of a fin and tube heat exchanger. Fin thermal conductivity values are varied between 15 and 200 W/mK. It is observed that the employing high thermal conductivity materials enhance the thermal performance of the heat exchanger. However, using the low thermal conductivity materials cause a resistance to transfer heat through the fin and temperature gradient is occurred in the fin. As a result of this Nu number and fin efficiencies decreases compare to higher thermal conductivity values of the fin materials. It is also shown that the tube and fin heat exchangers performance and heat transfer rate is highly affected by the geometric parameters when the low thermal conductivity fin materials are used. The characteristics of some parameters in low thermal conductivity fin materials show different characteristics than high thermal conductivity fin materials. It should be careful to determine the values of FT and FP as well as the ratio of FT/FP when the low thermal conductivity fin materials are used in fin and tube heat exchangers.

A correlation is proposed for the Nu number as a function of the non-dimensional geometric parameters and the non-dimensional thermal conductivity. Nu number can be obtained by the proposed correlation for this kind of heat exchangers.

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