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Numerical Investigation of Heat Transfer Enhancement of Double Pipe Heat Exchanger Using Metal Foam Fins

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ABSTRACT

The influence of adding metal foam fins on the heat transfer characteristics of an air to water double pipe heat exchanger is numerically investigated. The hot fluid is water which flows in the inner cylinder whereas the cold fluid is air which circulates in the annular gap in parallel flow with water. Ten fins of metal foam (Porosity = 0.93), are added in the gap between the two cylinder, and distributed periodically with the axial distance. Finite volume method is used to solve the governing equations in porous and non-porous regions. The numerical investigations cover three values for Reynolds number (1000 ,1500, 2000), and Darcy number (1 x10^{-1}, 1 x10^{-2}, 1x10^{-3}). The comparison between the two case with and without insertion the metal foam fins are examined in this study. Results show that the temperature of the inner pipe wall is affected by the Reynolds number, water inlet temperature, Darcy number variation. The resulting values and the behavior of the local and average coefficient of heat transfer are presented. The improvement in the mean coefficient of heat transfer (h_m) is found to be 129 % with the using of the metal foam fins.
Keywords: double pipe heat exchanger, convection heat transfer, metal foam fins, finite volume method.

الفحص العددي لتحسين انتقال الحرارة لمبادل حراري مزدوج الانبوب باستخدام زعانف من رغوة معدنية

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الخلاصة

لقد تم نظريا دراسة تأثير إضافة زعانف من رغوة معدنية في مبادل حراري مزدوج الانبوب على خصائص انتقال الحرارة داخل هذا المبادل مع استخدام الهواء والماء كمائع مشغل. حيث ان المائع الساخن والذي يجري في الأسطوانة الداخلية هو الماء بينما الهواء هو المائع البارد والذي يجري في الفتحة الحلقية بين الاسطوانتين في جريان متوازي مع جريان الماء. تم إضافة عشرة زعانف من الرغوة المعدنية في الفتحة الحلقية بين الاسطوانتين، وتم توزيعها بشكل دوري مع محور الانبوبين. تم استخدام طريقة الحجم المحكوم لحل المعادلات الحاكمة في منطقة وجود وعدم وجود الوسط المسامي. شملت الدراسة النظرية قيم رقم رينولدز (1000، 2000,1500) وقيم رقم دارسي (1×10^{-1}، 1×10^{-2}، 1×10^{-3}). أظهرت النتائج ان درجة حرارة سطح الأسطوانة

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الداخلية يتأثر بتغيير درجة حرارة دخول الماء للانبوب وكذلك بتغيير كل من رقم رينولدز وتغيير رقم دارسي مما يؤثر على عملية التبادل الحراري بين هذا السطح والمائع البارد (الهواء). نتائج تغير متوسط معامل أنتقال الحرارة في هذه الدراسة قد تم أظهارها وتحليلها. ووجد ان التحسين في متوسط معامل انتقال الحرارة (h_m) كان (129%) مع استخدام زعانف الرغوة المعدنية.

الكلمات الرئيسية: مبادل حراري مزدوج الانبوب، أنتقال الحرارة بالحمل، زعانف رغوه معدنية، طريقة الحجم المحكوم.

1. INTRODUCTION

Heat exchangers are the significant devices that used for numerous processes such as utilization, exchanging and transferring the thermal energy in various applications. Air to water double pipe heat exchanger can be selected as the significant type of heat exchangers that has various applications such as: air-conditioning, dehumidification, cooling in chemical processes, heating the fluid in manufacturing processes, and it is an important component in thermal power systems. A metal foam is one type of porous media with unique properties that can be used to enhance the heat transfer in various application. due to their unique thermal properties such as: high thermal conductivity, high mixing for the fluid flow and high porosity. Recently, due to the metal foam features, it is used with the application that exchanging the thermal energy to enhance the fluid flow and the heat transfer in such applications. the unique structure is the important feature of the metal foam. The selection of the foam structure depend on the application in which foam structures are used. Metal foams are produced as closed and open-cell metal foam. Closed cell metal foam has good mechanical properties so they are used in structural applications while Open-cell metal foam used for functional application. **Akpinar, 2006** experimentally investigated the influence of placing wires of helical shape inside the inner pipe of a double pipe heat exchanger, on the heat transfer performance. They found that the helical wires act as turbulators in the flow field, and they caused a remarkable increase in the pressure drop and friction factor. **Guerroudj and Kahalerras, 2010** numerically investigated the characteristics of heat that transfers in a channel subjected to constant heat flux and partially filled with blocks of metallic foam of different shapes. Results showed that the high rate of heat transfer can be achieved with triangular shape and the rectangular shape gives a higher pressure drop due to its volume comparison to the others shapes. **Hajeej, 2016,** carried out a numerical and experimental investigation to examine the enhancement of heat transfer inside a channel provided with blocks of copper foam. constant heat source inserted under each copper foam block. The enhancement in heat transfer can be achieved with different shapes of copper foam blocks. **Kurtabas and Celik, 2009,** analysed the characteristics and performance of heat transfer by using aluminium foam of different pore density they found that the local Nusselt number increased rapidly with the increasing of the Reynolds and Grashoof numbers. **Nima and Ali, 2017,** inserted eight blocks of metal foam inside an inclined channel of solar water collector, to investigate numerically the thermal performance enhancement under the climate conditions of Iraq. They found that the insertion of these blocks caused increase in the temperature of the collector exhaust, and the enhancement in the coefficient of heat transfer was more than 80% with the using of metal foam blocks. **Nima and Hajeej, 2016,** manufactured a horizontal channel and filled blocks of copper foam of different pores per unit of length (10, 40 PPI) inside the test section of this channel that heated by a heated section with a constant heat flux, to investigate the improvement in the mixed convection heat transfer experimentally. The working fluid was air that flew through each metal foam block which installed on each heated section. They found that at each heated section the temperature of wall was affected by the variation of the heat flux, Reynolds and Darcy number variation. Also, they found that in all cases the enhancement in heat transfer was more than 80% when inserting the blocks of copper foam. **Sheikholeslami, et al., 2015,** inserted agitators inside the heat exchanger of type air to water double pipe, to investigate experimentally the heat transfer improvement. they found that the inserting of agitator introducing swirl in to the fluid flow which results in more enhancement of heat transfer. **Sheikholeslami, et al., 2016** placing circular perforated rings in the annular pipe to investigate experimentally its effect on the pressure and heat transfer enhancement. They found that the Nusslet number increases with the increasing of the Reynolds number and increasing of perforated hole. Also they found that the friction factor improves with the increasing of Reynolds number, and decreases with increasing of pitch ratio and number of perforated holes. **Targui and Kahalerras, 2008,** found that the high thicknesses and low permeability of porous structures lead to appearance of recirculation zones (vortices) that contributed to the enhancement of heat transfer in a heat exchanger of type double pipe. **Xu, et al., 2014,** studied numerically the forced convection heat transfer in heat exchanger of type double pipe that fully filled with metal foam. Local thermal non-equilibrium model was established. Results showed that the pressure drop was decrease for higher porosity and lower PPI. They found that the increasing of heat exchanger effectiveness can be achieved by decreasing porosity or increasing in pore density.

In this study, convection heat transfers in an air to water double pipe heat exchanger provided with fins of metal foam (copper foam), is numerically examined. The effect of the process of heat transfer on the fluid and flow structure in such devices is studied. In this study the investigation and analysing of local heat transfer coefficient with the variation of Reynolds number and Darcy number is obtained. And the influence of the variation of these two numbers on the wall temperature distribution along the double pipe heat exchanger is analysed.

2. MATHEMATICAL FORMULATION

2.1 Geometry and Coordinate System

The geometry and coordinate system for mixed convection in an annular that provided with ten fins of copper foam installed periodically with the axial direction, is shown in **Fig. 1**. The system under investigation is a two-dimensional concentric cylinder with 10 cm outer diameter, 5 cm inner diameter, 190 cm length (L), [inlet section (li=30 cm), test section (100 cm), and exit section (le=60 cm)], wb=2 cm (width of metal foam fin), h= ro-ri (height of foam fin), and Sp=13 cm (space between fins) as shown in **Fig. 1**. The flow is parallel in which hot water and cold air entered the inner copper pipe and the gap between the two pipes, respectively with a constant temperature and a uniform velocity distribution.

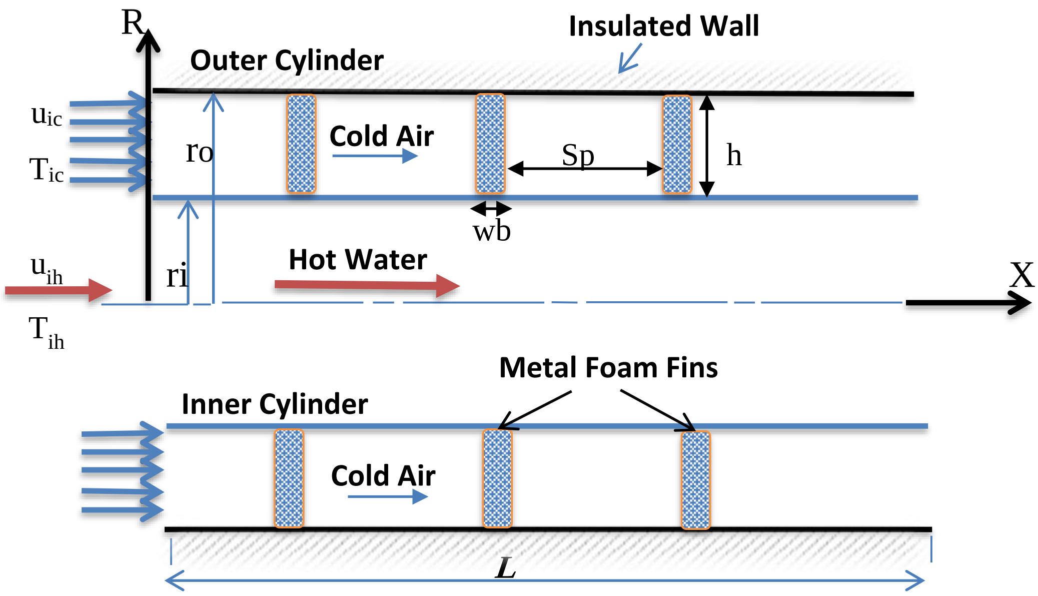

Figure 1. Schematic of the physical domain with metal foam fins.

2.2 Assumptions

To solve the governing equations (continuity, momentum and energy) the following assumptions should be taken into account:

1) Laminar flow.
2) Steady state.
3) Boussinesq approximation is invoked.
4) Incompressible fluid.
5) Axis-symmetric ($\frac{\partial}{\partial\theta} = 0$).
6) No heat generation
7) No viscous dissipation.
8) Homogeneous, isotropic porous medium.
9) The solid matrix of the porous medium is in local thermal equilibrium with the passing fluid.

2.3 Governing Equations

According to the above assumptions, the basic equations are reduced to the following equations, mass, momentum and energy equation for the flow inside the heat exchanger. In the present study the Darcy and Brinkman- Forchheimer model is used to model the flow in the porous regions, the Navier-Stokes equations in the fluid regions, and the thermal filed by the energy equation **Targui and Kahalerras, 2008**. The dimensionless governing equations is as follows;

1) Conservation of mass

$$\frac{\partial U}{\partial X}+\frac{1}{R}\frac{\partial (RV)}{\partial R}=0 \qquad (1)$$

2) Conservation of momentum

x-Momentum Equation:

- Annular gap

$$\{\lambda\left(\frac{1}{\varepsilon^2}-1\right)+1\}\left(\mathrm{U}\frac{\partial \mathrm{U}}{\partial \mathrm{X}}+\mathrm{V}\frac{\partial \mathrm{U}}{\partial \mathrm{R}}\right)=-\frac{\partial \mathrm{P}}{\partial \mathrm{X}}+\frac{1}{\mathrm{Re}}\{\lambda\left(\frac{1}{\mathrm{R}_{\mu}}-1\right)+1\}\left(\frac{\partial^2 \mathrm{U}}{\partial \mathrm{X}^2}+\frac{1}{R}\frac{\partial}{\partial \mathrm{R}}(\mathrm{R}\frac{\partial \mathrm{U}}{\partial \mathrm{R}})\right)-\frac{\lambda}{\mathrm{Re\,Da}}\mathrm{U}-\frac{\mathrm{CF}\,\lambda}{\mathrm{Re\,Da}}|\vec{\mathrm{V}}|\mathrm{U} \qquad (2)$$

- Inner cylinder:

$$\left(\mathrm{U}\frac{\partial \mathrm{U}}{\partial \mathrm{X}}+\mathrm{V}\frac{\partial \mathrm{U}}{\partial \mathrm{R}}\right)=-R_{\rho}\frac{\partial \mathrm{P}}{\partial \mathrm{X}}+\frac{1}{\mathrm{Re}}\left(\frac{\partial^2 \mathrm{U}}{\partial \mathrm{X}^2}+\frac{1}{R}\frac{\partial}{\partial \mathrm{R}}(\mathrm{R}\frac{\partial \mathrm{U}}{\partial \mathrm{R}})\right) \qquad (3)$$

y-Momentum Equation:

- Annular gap:

$$\{\lambda\left(\frac{1}{\varepsilon^2}-1\right)+1\}\left(\mathrm{U}\frac{\partial \mathrm{V}}{\partial \mathrm{X}}+\mathrm{V}\frac{\partial \mathrm{V}}{\partial \mathrm{R}}\right)=-\frac{\partial \mathrm{P}}{\partial \mathrm{R}}+\frac{1}{\mathrm{Re}}\{\lambda\left(\frac{1}{\mathrm{R}_{\mu}}-1\right)+1\}\left(\frac{\partial^2 \mathrm{V}}{\partial \mathrm{X}^2}+\frac{1}{R}\frac{\partial}{\partial \mathrm{R}}\left(\mathrm{R}\frac{\partial \mathrm{V}}{\partial \mathrm{R}}\right)-\frac{V}{Re^2}\right)-\frac{\lambda}{\mathrm{Re\,Da}}\mathrm{V}-\frac{\mathrm{CF}\,\lambda}{\mathrm{Re\,Da}}|\vec{\mathrm{V}}|\mathrm{V} \qquad (4)$$

- Inner cylinder

$$\left(\mathrm{U}\frac{\partial \mathrm{V}}{\partial \mathrm{X}}+\mathrm{V}\frac{\partial \mathrm{V}}{\partial \mathrm{R}}\right)=-R_{\rho}\frac{\partial \mathrm{P}}{\partial \mathrm{R}}+\frac{1}{\mathrm{Re}}\left(\frac{\partial^2 \mathrm{V}}{\partial \mathrm{X}^2}+\frac{1}{R}\frac{\partial}{\partial \mathrm{R}}\left(\mathrm{R}\frac{\partial \mathrm{V}}{\partial \mathrm{R}}\right)-\frac{V}{Re^2}\right) \qquad (5)$$

3) Conservation of energy

- Annular gap:

$$\mathrm{U}\frac{\partial \theta}{\partial \mathrm{X}}+\mathrm{V}\frac{\partial \theta}{\partial \mathrm{R}}=\frac{\lambda R_k(1-\lambda)}{Re\,Pr}\left(\frac{\partial^2 \theta}{\partial \mathrm{X}^2}+\frac{1}{R}\frac{\partial}{\partial \mathrm{R}}(\mathrm{R}\frac{\partial \theta}{\partial \mathrm{R}})\right) \qquad (6)$$

- Inner cylinder

$$\mathrm{U}\frac{\partial\theta}{\partial \mathrm{X}}+\mathrm{V}\frac{\partial\theta}{\partial \mathrm{R}}=\frac{\mathrm{R}_k}{\mathrm{Re\,Pr}}\left(\frac{\partial^2\theta}{\partial \mathrm{X}^2}+\frac{1}{R}\frac{\partial}{\partial \mathrm{R}}(\mathrm{R}\frac{\partial\theta}{\partial \mathrm{R}})\right) \qquad (7)$$

where $|\vec{V}|=\sqrt{u^2+v^2}$. The porosity(ε), the metal foam permeability (K), thermal conductivity and the effective viscosity are taken equal respectively to unity (ε = 1), infinity (K→ ∞) and thermal conductivity and fluid's viscosity in the case of without existence of metal foam fins. The term $\frac{\lambda}{\mathrm{Re\,Da}}\mathrm{V}-\frac{\mathrm{CF}\,\lambda}{\mathrm{Re\,Da}}|\vec{\mathrm{V}}|\mathrm{V}$ can be noticed as the resistance offered by the metal foam to the flow ($\mathrm{CF}=0.17,$ **Hajeej, 2016**). (λ) is a binary parameter which takes values of 0 in the fluid region and 1 in the porous region. The following dimensionless variables used to nondimensionalize the governing equations, **Targui and Kahalerras, 2008:**

$$X=\frac{x}{D_h};R=\frac{r}{D_h};U=\frac{u}{u_{ic}};V=\frac{v}{u_{ic}};P=\frac{p}{\rho_c u_{in}^2}\text{ and }\theta=\frac{T-T_{ic}}{T_{ih}-T_{ic}}$$

$$\mathrm{Re}=\frac{u_{ic}\rho_c\,D_h}{\mu_c};\ \mathrm{Da}=\frac{K}{{D_h}^2};Pr=\frac{\mu\mathrm{C_p}}{k};C=\varepsilon F;R_\mu=\frac{\mu_c}{\mu_{eff}}\ and\ R_k=\frac{k_{eff}}{k_c};$$

$$\mathrm{CF}=\frac{u_{ic}\,\varepsilon F\,\rho_c\sqrt{K}}{\mu_c}\ ;R_{keh}=\frac{k_e}{k_h}\ R_{kch}=\frac{k_c}{k_h};$$

In Eq. (2), (4), and (6) for annular gap, and Eq. (3), (5) and (7) for inner cylinder, the convective term on the left hand side of momentum and energy equations is involved in order to account for the development of the velocity profile. Whereas the right hand side of the same equations is the diffusive term which multiplied by the inverse Reynolds number. The diffusion in the flow results in the generation of the boundary layer.

2.4 The Boundary Conditions

The boundary conditions in dimensionless form can be written as:

- At the inlet: $X=0;\ 0<R<Ri:\ U=1;\ V=0\ and\ \theta=0$

  $X=0;\ Ri<R<Ro:\ U=1;\ V=0\ and\ \theta=0$
- At the exit: $X=L;\ 0<R<Ri:\frac{\partial U}{\partial X}=0;\ V=0\ and\ \frac{\partial\theta}{\partial X}=0$

  $X=L;\ Ri<R<Ro:\frac{\partial U}{\partial X}=0;\ V=0\ and\ \frac{\partial\theta}{\partial X}=0$
- At the symmetric axis:

  $R=0,\ 0<X<L:\frac{\partial U}{\partial R}=0;\ V=0\ and\ \frac{\partial\theta}{\partial R}=0$
- At the Inner cylinder wall:

$$R=Ri\,;\,0<X<L\begin{cases}U=0;\ V=0\\ \frac{\partial\theta}{\partial r}|_{hot\,fluid}=R_{kch}\frac{\partial\theta}{\partial r}|_{cold\,fluid}\\ \frac{\partial\theta}{\partial r}|_{hot\,fluid}=R_{keh}\frac{\partial\theta}{\partial r}|_{porous}\end{cases}$$

- At the outer cylinder wall: $R=Ro;\ 0<X<L:\ U=V=0$ and $\frac{\partial\theta}{\partial \mathrm{R}}=0$

3. FURTHER CALCULATIONS

3.1 The Nusselt Number

The local heat transfer coefficient (h_x) and Nusselt number (Nu_x) can be defined as, **Began, 2006**.

$$hx = \begin{cases} -\dfrac{k_e \frac{\partial T}{\partial r}|_{r=ri}}{Tw-Tm} & \text{porous } region \\ -\dfrac{k_c \frac{\partial T}{\partial r}|_{r=ri}}{Tw-Tm} & \text{fluid region} \end{cases} \tag{8}$$

$$Nu_x = \frac{hx\, D_h}{k_c} = \frac{k_e \frac{\partial T}{\partial r}}{k_c(T_w - T_m)} = \frac{R_k \frac{\partial \theta}{\partial R}}{\theta_w - \theta_b} \tag{9}$$

Where: T_w is the wall temperature, and θ_w is the dimensionless wall temperature, T_m is the bulk temperature, and θ_m is the dimensionless bulk temperature

$$\theta_m = \frac{\int_0^1 |U|\theta R dr}{\int_0^1 |U| R dr} \tag{10}$$

The average heat transfer coefficient (h_m) is calculated as follows:

$$h_m = \frac{1}{L}\int_0^L hx\, dX \tag{11}$$

The above integration is obtained over the physical domain, So, both L and dX are dimensionless.

3.2 Effective Thermal Conductivity

An important property of the porous medium is the effective thermal conductivity (k_e). for the open-cell metal foam, k_e, can be calculated from the following correlation that was proposed by, **Calmidi, et al., 1999** as;

$$k_e = \varepsilon k_{air} + 0.181(1-\varepsilon)^{0.763}\, k_s \tag{12}$$

where: k_{air} = 0.0263 W/m.K is the air thermal conductivity.
k_s =386 W/m.K is the Thermal conductivity of the copper foams, **Holman, 2010**.

4. NUMERICAL PROCEDURE

Numerical methods represent a useful and popular method to simulate the fluid flow and heat transfer characteristics. The problem of transferring heat by mixed convection through a double pipe heat exchanger is solved numerically by utilizing the finite volume method. The governing equations from (8) to (14) are discretized by finite volume-based finite difference method with the use of the power law scheme of, **Patenkar, 1980.** and solved by a tri-diagonal matrix algorithm to discretize the combined convective and conductive terms. The Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) is used to solve the coupling between the continuity equation and the momentum equations. A computer program was built in MATLAB R2015a language to perform the numerical solution formulated previously. The program consists of a main program and four-subroutines of TDMA solver for velocity, pressure and the temperature field as shown in **Fig. 2**.

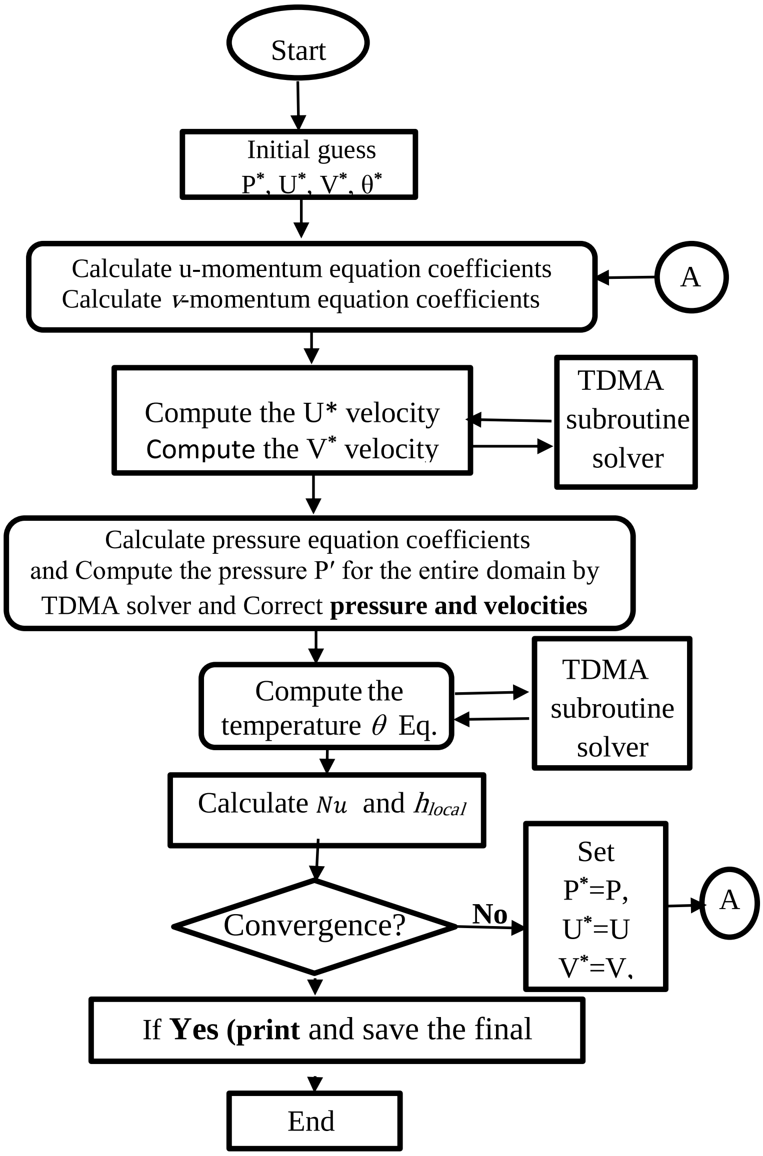

Figure 2. Flow chart for computer program.

First the grid independency tests are performed. Numerical tests three grid sizes 391 x 153, 691 x 153, and 791 x 121 to test the grid independent solution. It is observed from **Table 1** that the values of the local Nusselt number for a grid size larger than 691 x 153 does not vary more than 4.07%. Therefore, a grid size of 691 x 153 is adopted for further computations for consuming less computational time. In the X direction, a non-uniform structured grid system is provided with metal foam fins [inlet section (371), test section (250), and exit section (70)] as shown in **Fig. 3**.To achieve convergence solution, relative error must be 10^{-5} for both temperature and velocity and less than 10^{-7} for pressure, between successive iterations, and calculated from the following relation:

$$\bar{R} = \frac{\sum|a_E\phi_E + a_W\phi_W + a_N\phi_N + a_S\phi_S + b - a_P\phi_P|}{\sum|a_P\phi_P|} \leq 10^{-7} \tag{13}$$

Where: $a_{(E,W,N,S,P)}$ are the coefficients in the discretization equations on the neighbor nodes of the center point in the mesh, and $\phi_{(E,W,N,S,P)}$ are the variables (stand for the dependent variables U,V, θ) between the center of the mesh and that of the neighbor nodes .

Table 1. Grid sensitivity analysis.

m x n	Nu_x	Relative difference (%)
391 x 153	737.97	–
691 x 153	768.03	4.07
791 x 121	850.82	10.78

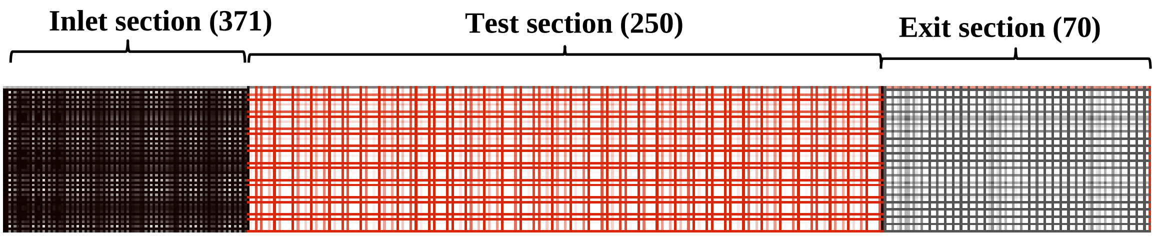

Figure 3. Non uniform grid generation.

5. CODES VALIDATION

The program codes of the present work are validated with the results that obtained by **Hadim, 1994**, who solved the governing equations by using finite volume method. He obtained the numerical results on a porous media with (ε =0.97, C=0.1, Pr=10, Re=250, Da=10^{-3}) fully and partially filled a horizontal channel. The validation obtained on the local Nusselt Number as shown in **Fig. 4**. This figure shows a good agreement between the results of the present work program and with **Hadim, 1994**.

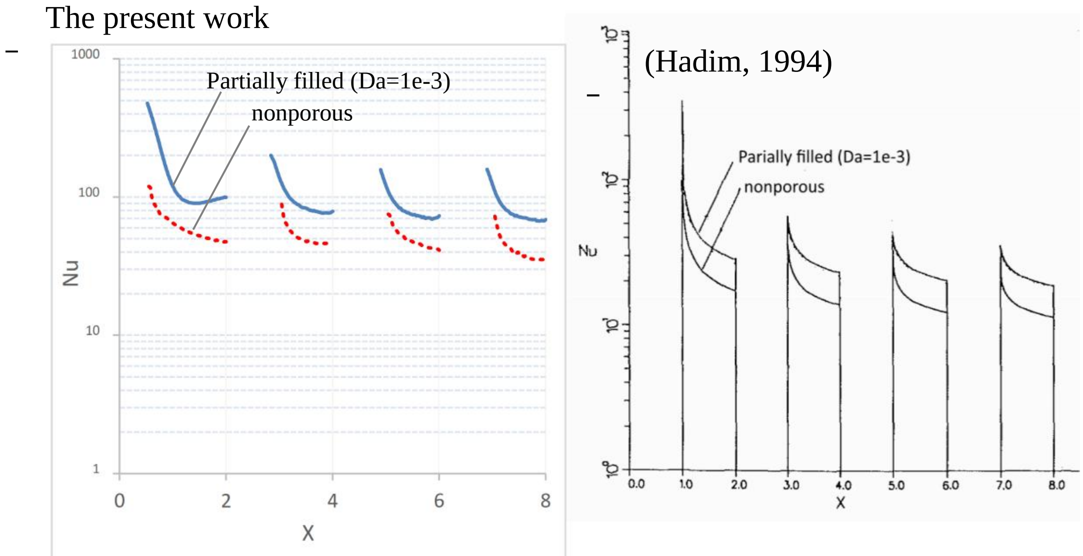

Figure 4. Local Nusselt number for porous and nonporous cases for Re=1000.

6. RESULTS AND DISCUSSION

In the present study, convection heat transfer inside the heat exchanger of type double pipe supplied with fins of coper foam is solved with the consideration of thermal equilibrium situation. The double pipe is provided with ten copper foam fins (inside the annular gap) of porosity (ε =0.903), inertial coefficient (C=0.17) , thermal conductivity ratio ($R_k = 449.86$), and viscosity ratio ($R_\mu = 1$,Brinkman assumption) with air (Pr=0.7) flows in the annular gap, and water (Pr=2.04248) with inlet temperature (T_{inw}= 85°C) in case of without insertion the copper foam fins and (Tin= 60, 80, 90°C) in case of with insertion of copper foam fins .

6.1 Velocity Vectors and Temperature Contours

Fig. 5 shows the influence of insertion fins of copper foam on the velocity vector for Da=1 x10^{-3} and Re=1000. it is observed from **Fig. 5** that the velocity vector with the copper foam fins case has slightly difference from that of without copper foam fins case, especially when 1 x$10^{-3} \geq$ Da $\leq$ 1 x10^{-1}. When the Darcy number increases, the permeability increase which allow the fluid flow with low velocity especially at low amount of the fluid flow. The existence of metal foam fins must consider as a disturbance and resistance to the fluid flow inside the annular gap. This depends on the permeability of the copper foam fins (the value of Darcy number), and on the amount of fluid that forced through the copper foam fins.

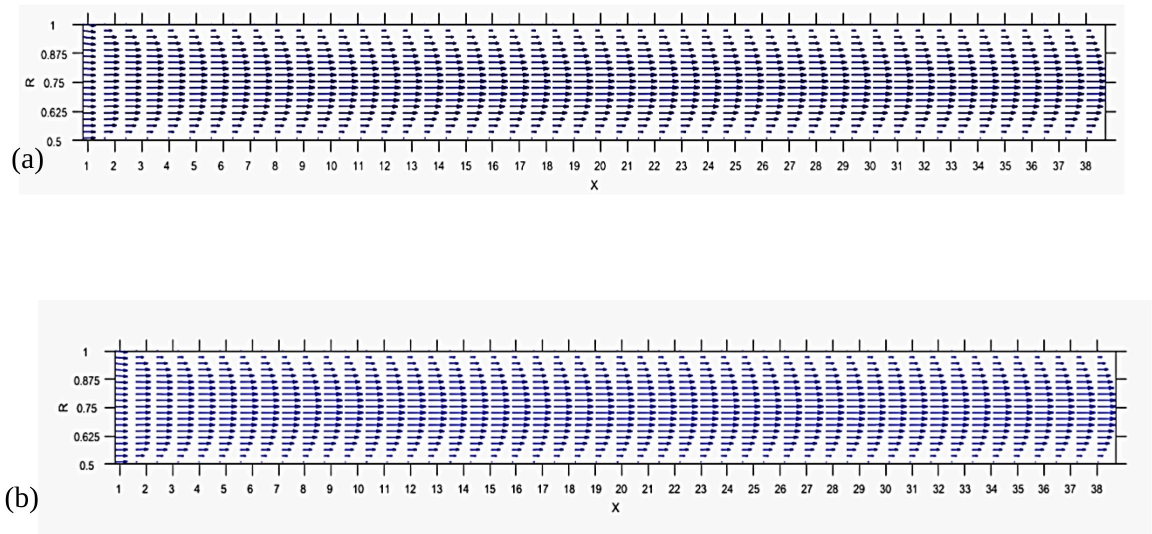

Figure 5. Velocity vectors for Re=1000 (a) fluid case (b) with metal foam (Da=1 x 10^{-3}).

Fig. 6 shows the influence of copper foam fins insertion on the temperature contours along the annular for Da=1 x10^{-3}, Re=1000. These contours show that the existence of the copper foam fins caused a great increased in the fluid temperature and the overall heat transfer process is enhanced, due to the mechanism of heat transfer with the existence of copper foam fins which provided a great heat diffusion that results in increase the air bulk temperature.

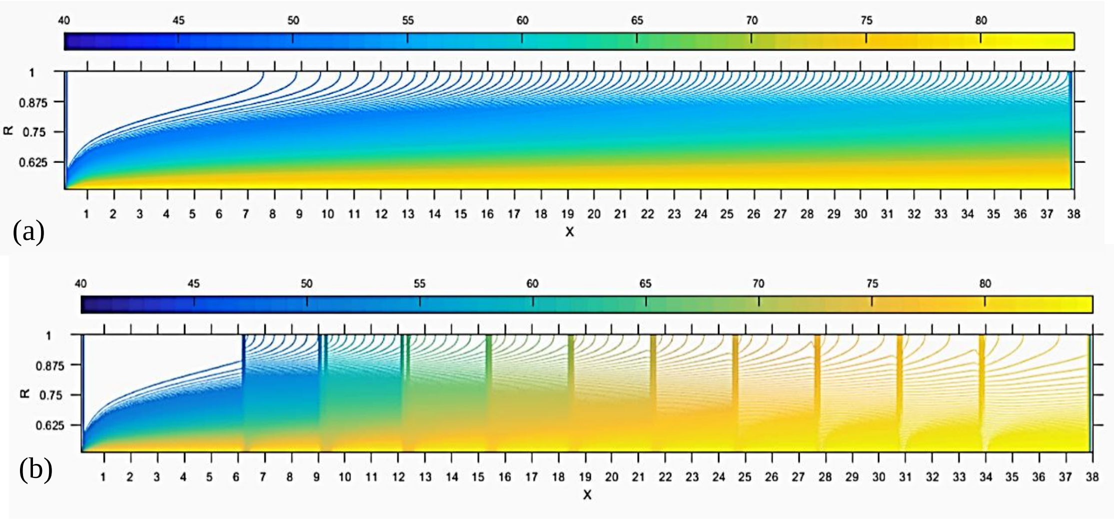

Figure 6. Temperature contours (°C) for Re=1000
(a) fluid case (b) with metal foam (Da=1 x 10^{-3}).

In **Fig. 7** the water inlet temperature variation influence on the temperature contours along the annular for Re=1000, and Da=1 x10^{-3} is shown. It is observed that the air bulk temperature is increased with the increasing of water inlet temperature. The buoyancy effect increased with the increasing of the temperature of the inner copper pipe wall which increased as the water inlet temperature increased. When the temperature of the inner copper pipe wall increases the thermal boundary layer along the copper pipe wall growths rapidly, and the existence of metal foam fins lead to diffuse the heat from the copper pipe wall and increase the air bulk temperature.

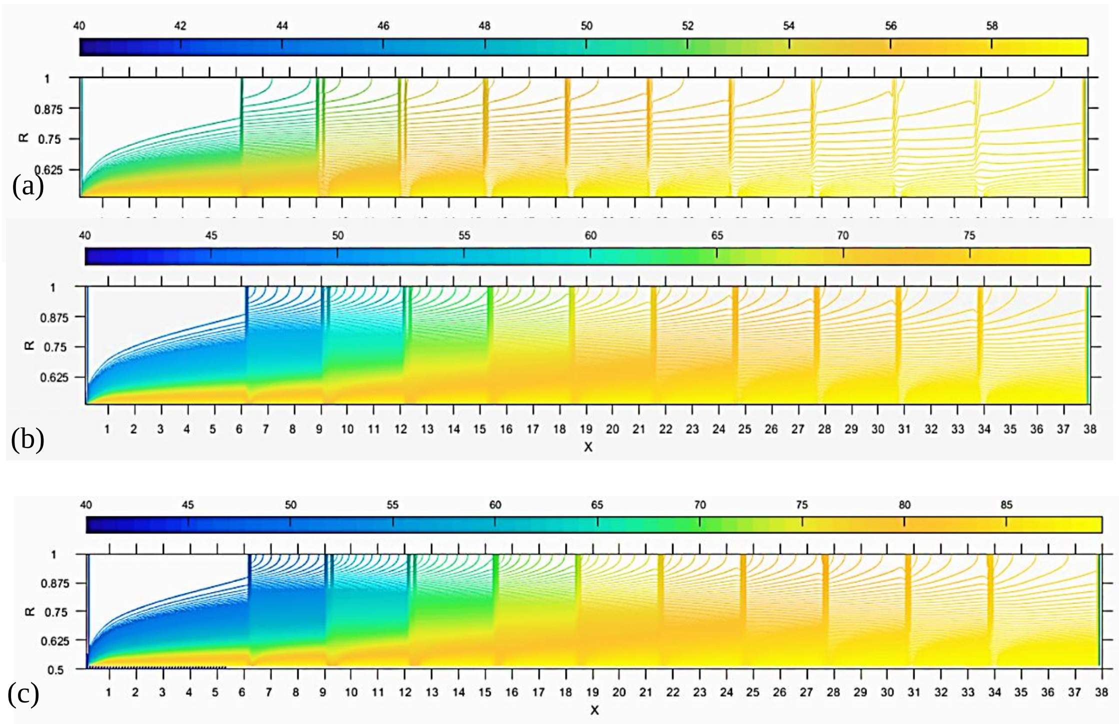

Figure 7. Temperature contours (°C) for Re=1000 and Da=1 x 10^{-3} (a) T_{w_in}=60 °C (b) T_{w_in} =80°C (c) T_{w_in} =90°C.

Fig. 8 shows the effect of increasing the Reynolds number values on the contours of temperature along the annular of heat exchanger.it is observed that when Reynolds number increases the wall temperature decreases especially at the metal foam fins positions. the thermal boundary layer is retreated along the copper pipe wall with the increasing of Reynolds number, due to the large amount of fluid that pass near the copper pipe wall and decrease its temperature with the increasing of the Reynolds number.

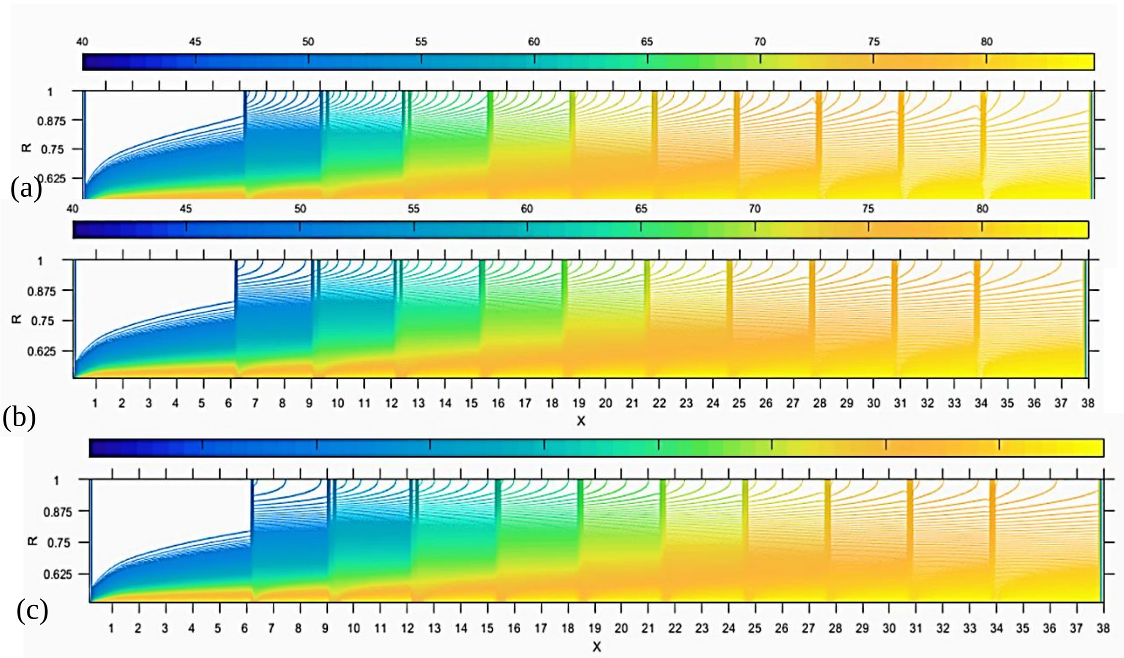

Figure 8. Temperature contours (°C) for T_{w_in}=85 °C and Da=1 x10^{-3} (a) Re=1000 (b) Re=1500 (c) Re=2000.

Fig. 9 shows the influence of adding metal foam fins on the temperature contours along the annular for Re=1000, for three cases (Da=1 x10^{-1}, Da=1 x10^{-2}, Da=1 x10^{-3}). The mechanism of heat transfer that works with the metal foam fins increases the heat transfer from the copper pipe wall and dissipate this heat to the entire domain which lead to increase the air bulk temperature as observed in **Fig. 9**. With the increasing of Darcy number, the permeability of the copper foam fins increases which lead to increase the heat convected by air.

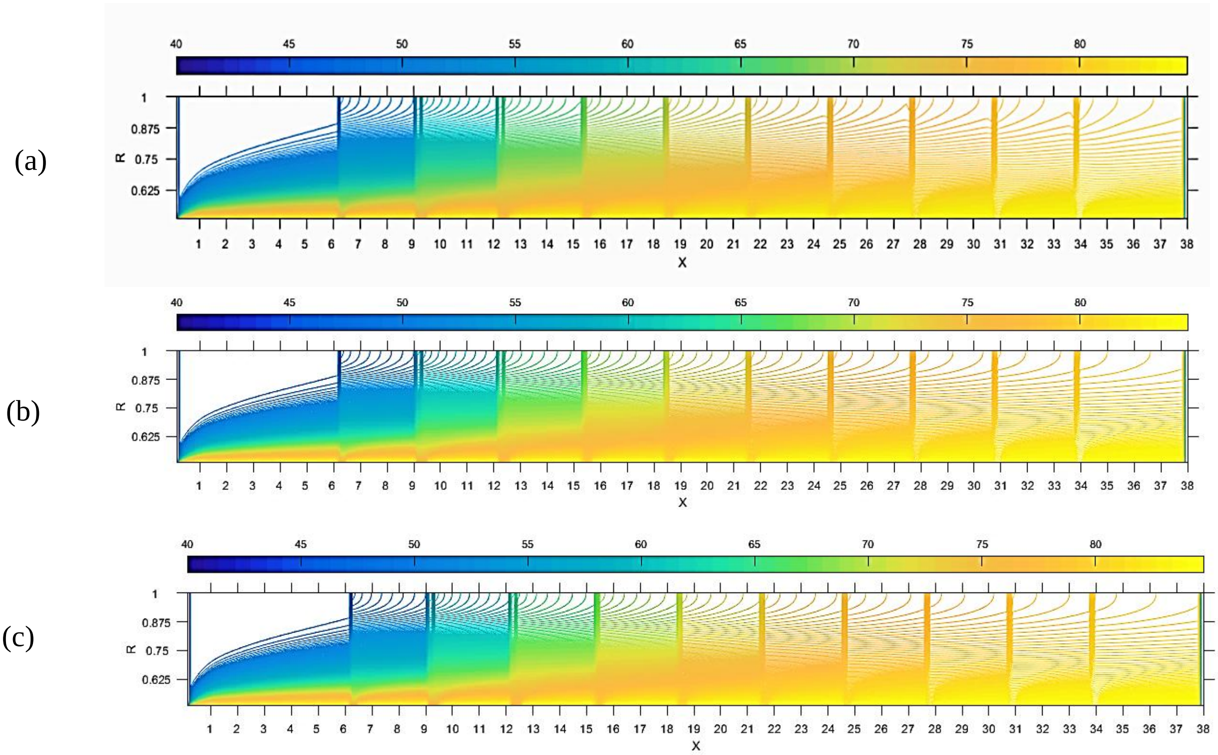

Figure 9. Temperature contours (°C) for T_{w_in}=85 °C and Re=1000

(a) Da=1 x10^{-1} (b) Da=1 x10^{-2} (c) Da=1 x10^{-3}.

6.2 Wall Temperature Distribution

Fig. 10 shows the influence of copper foam fins insertion on the temperature distribution along the inner copper pipe wall for Re=1000,1500 and 2000, and Da=1 x10^{-3}. It is observed that the temperature distribution in the case of copper foam fins insertion is very decreased at the copper foam fin position, compared with that in the case of without copper foam fins. Also, it can be noticed that there is a gradually distributed in a wall temperature along the annular gap. This is due to the boundary layer effect which is began to build at the entrance of the annular gap near the first copper foam fins and growing up through the annular gap. The presence of metal foam fins distorted this boundary layer gradually and caused higher heat transfers through the whole domain. The existence of the copper foam fins causes a higher fluid mixing inside the annular gap and increases the heat diffusion to the whole domain. The temperature of fluid flow increases more inside the domain with an increasing of Reynolds number, because of the increasing in the amount and mixing of the fluid flow inside the annular gap which leads to increase the heat disipation.

6.3 Local Heat Transfer Coefficient

Fig. 11 shows the general behavior of the local heat transfer coefficient. In all cases the local heat transfer coefficient decreases with increase in the axial distance. The highest rate of heat transfer happens at the entrance of the annular gap where the temperature difference has a maximum value between the wall of the hot copper pipe and the cold fluid. **Fig. 11** shows the increasing of the

Reynolds number that influence on the local coefficient of heat transfer for Da= 1 x10^{-3}. It is observed that the increasing of the Reynolds number lead to increase the local heat transfer coefficient. A large amount of fluid flow through the copper foam fins with the increasing of the Reynolds number, which lead to increase the temperature difference between both the inner copper pipe wall, the metal foam fins, and the incoming fluid. Also, it can be noticed from **Fig. 11** that the maximum heat transfer coefficient is obtained at the position of the metal foam fins on the copper pipe wall where a higher temperature difference is occurred. Also, this figure shows the influence of inserting fins of copper foam on the local coefficient of heat transfer. It is shown that the values of the local heat transfer coefficient with copper foam fins case, is higher than its values in case of without existence of metal foam fins. It is observed that the thermal boundary layer is vanished through the metal foam fins, which result in increased the rate of heat transfer.

In **Fig. 12** the effect of changing the value of Darcy number on the local coefficient of heat transfer for Re=1000, is shown. the local heat transfer coefficient increased when Darcy Number decreased (Da=1 x10^{-3}), due to the fluid flow that affected by the porous matrix of the metal foam and the local heat transfer coefficient decreased when Darcy Number increased (Da = 1 x10^{-1} or Da = 1 x10^{-2}). A higher fluid mixing, and thermal boundary layer disruption occur with the decreasing of the Darcy number, which results in enhancement of convection coefficient of heat transfer.

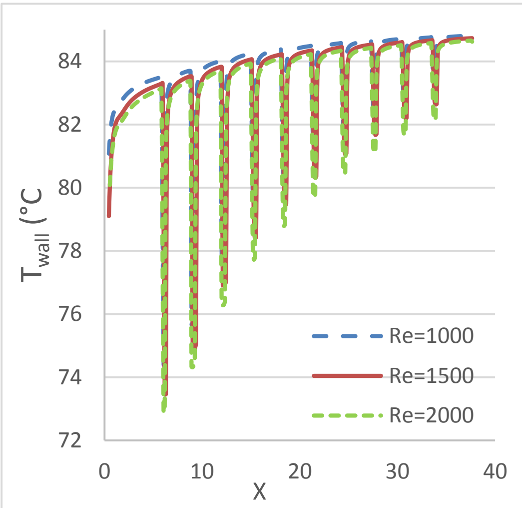

Figure 10. Variation of the wall Temperature with the axial distance for different Reynolds number,

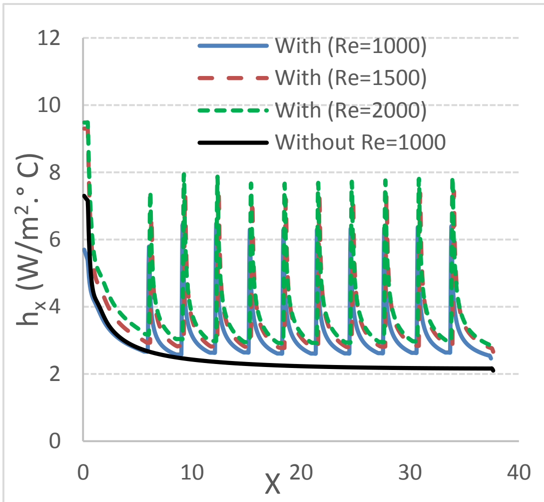

Figure 11. Local heat transfer coefficient with the axial distance for different Reynolds number.

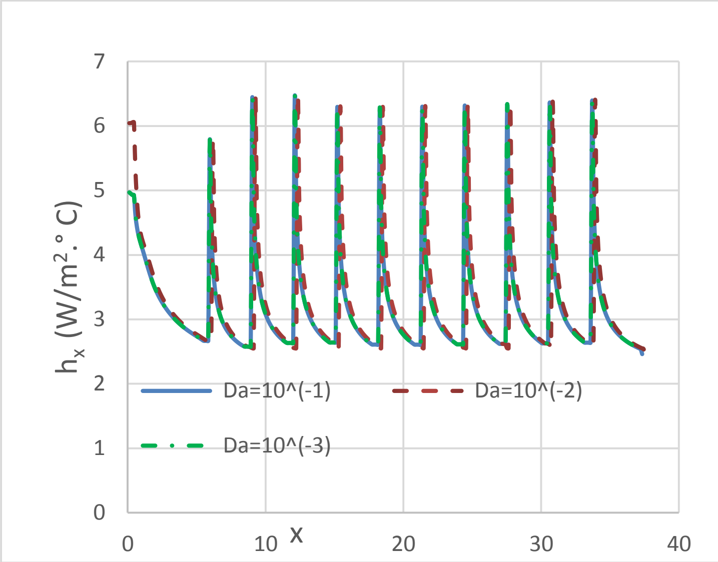

Figure 12. Local heat transfer coefficient with the axial distance for different Darcy number for Re=1000.

6.4 Average Convection Heat Transfer Coefficient

Fig. 13 shows the influence of increasing the Reynolds number, on the average coefficient of heat transfer. It is clear from this figure that with the increasing of Reynolds number the average coefficient of heat teansfer increases for both cases, but its values with the existence of copper foam fins is higher than its values in the case of without foam fins existence, due to the enhancement in heat transfer that happens with the insertion of copper foam fins inside the heat exchanger.

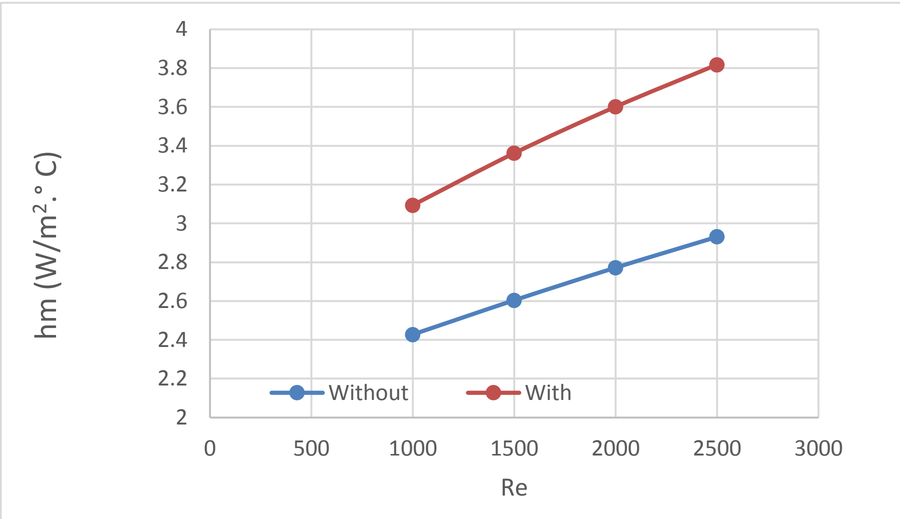

Figure 13. Mean convection heat transfer coefficient for different Reynolds number.

6.5 Local Nusselt Number

Fig. 14 Shows the general behavior of the local Nusselt number for two cases (with and without insertion of the copper foam fins) for the present work. For both cases it can be seen that the local Nusselt number has a behavior similar to that of the local heat transfer coefficient. Which means that the local Nusselt number decreases with the increasing of the axial distance, and have the largest values in the case of inserting the copper foam fins.

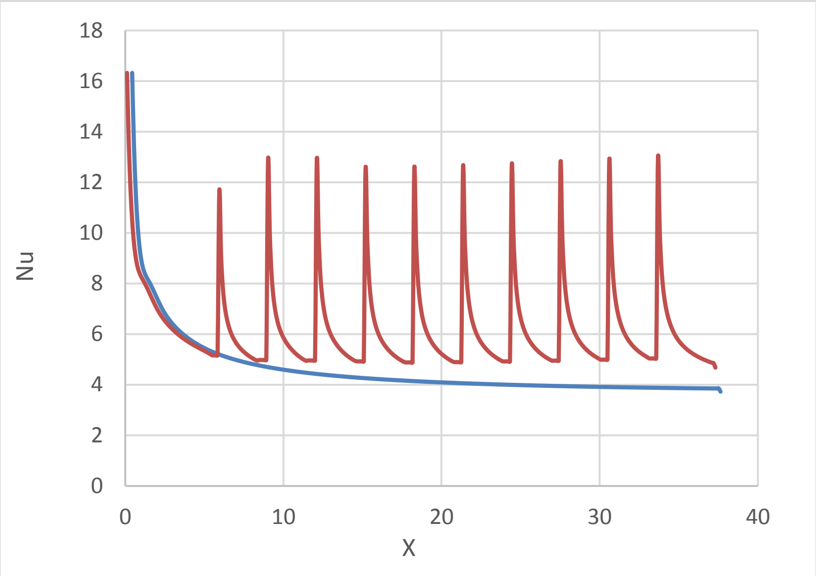

Figure 14. Local Nusselt number with the axial distance for Re=1000.

7. CONCLUSIONS

According to the investigation of the convection heat transfer and fluid flow inside the double pipe heat exchanger provided with fins of copper foam, the following conclusions can be written:

1- The temperature of the fluid with the inserting of the copper foam fins inside the annular gap have the greatest values than the fluid case.
2- The fluid temperature for both cases (with and without fins of copper foam) increases as the water inlet temperature increase.
3- The wall temperature at the metal foam fins positions decreases with the increasing of the Reynolds number.
4- The local and average coefficient of heat transfer increased with the increased of the Reynolds number.
5- The local and average coefficient of heat transfer increased with the decreased of the Darcy number.
6- The heat transfer improvement is increased for full coper foam fins when Darcy number decreased and Reynolds number increased.

9. NOMENCLATURE

C= inertia coefficient.
C_p=specific heat at constant pressure, J/kg.K.
Da= Darcy number.
F = Forchheimer coefficient.
h = local heat transfer coefficient, W/m^2. K.
k= thermal conductivity of fluid, W/m·K.
K = permeability of the porous medium, m^2.
k_e = effective thermal conductivity, W/m·K.
p = pressure, pa.
Re= Reynolds number.
R_k= thermal conductivity ratio.
R_μ= viscosity ratio.
$\overline{\mathrm{R}}$ = the relative error
sp = porous block spacing, m.
T = temperature, °C.
u = axial velocity, m/s.
v = transverse velocity, m/s.
Nu = Nusselt number.
Pr = Prandtl Number.
v = kinematic viscosity, m^2/s.
ε = porosity.
μ = dynamic viscosity, kg m/s.
ρ = density of fluid, $\mathrm{kg/m^3}$.

Subscript Meaning

b= bulk.
c = cold.
e= effective.
eff= effective.
h= hot.
i= inlet.
m = mean.
o= outlet.
w= wall.

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