Eulerian–Eulerian multi-phase RPI modeling of turbulent forced convective of boiling flow inside the tube with porous medium

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Abstract

Purpose – The purpose of this study is simulation the flow boiling inside a tube in the turbulent flow regime for investigating the effect of using a porous medium in the boiling procedure.

Design/methodology/approach – To ensure the accuracy of the obtained numerical results, the presented results have been compared with the experimental results, and proper coincidence has been achieved. In this study, the phase change phenomenon of boiling has been modeled by using the Eulerian–Eulerian multi-phase Rensselaer Polytechnic Institute (RPI) wall boiling model.

Findings – The obtained results indicate using a porous medium in boiling process is very effective in a way that by using a porous medium inside the tub, the location of changing the liquid to the vapor and the creation of bubbles, changes. By increasing the thermal conductivity of porous medium, the onset of phase changing postpones, which causes the enhancement of heat transfer from the wall to the fluid. Generally, it can be said that using a porous medium in boiling flows, especially in flow with high Reynolds numbers, has a positive effect on heat transfer enhancement. Also, the obtained results revealed that by increasing Reynolds number, the created vapor phase along the tube decreases and by increasing Reynolds number, the Nusselt number enhances.

Originality/value – In present research, by using the computational fluid dynamics, the effect of using a porous medium in the forced boiling of water flow inside a tube has been investigated. The fluid boiling inside the tube has been simulated by using the multi-phase Eulerian RPI wall boiling model, and the effect of thermal conductivity of a porous medium and the Reynolds number on the flow properties, heat transfer and boiling procedure have been investigated.

Keywords Turbulent flow, Porous medium, Nusselt number, Bubble, Boiling

Paper type Research paper

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Nomenclature

A_b = portion of wall surface covered by nucleating bubble
A_i = interfacial area
C_{lv}^D = drag coefficient
C_{lv}^L = lift coefficient
C_{wt} = coefficient for bubble waiting time
$C_{p.1}, C_{p.v}$ = liquid and vapor heat capacity
D_b, D_d = bubble and dispersed-phase diameter
D_{bw} = bubble departure diameter
f_{bw} = bubble departure frequency
$\vec{F}$ = inter-phase momentum forces
g = gravity
h = heat transfer coefficient
H_{lv} = latent heat
$\dot{m}$ = rate of mass transfer
H_q = specific enthalpy
k_l = liquid thermal conductivity
$\dot{m}$ = mass flow rate
$\dot{q}$ = heat flux
R_e, R_c = evaporation and condensation rate
Re = phase Reynolds number
T_{sat} = liquid saturation temperature
VoF = volume of Fraction
ΔT_{sub} = liquid subcool

Greek symbols
α = phase volume fraction
β = contact angle
γ = fluid/phase heat diffusivity
ρ = fluid/phase density
μ = dynamic viscosity
σ = surface tension coefficient
Γ = diffusion coefficient

Subscripts
l, v, c, d = liquid, vapor, continuous and dispersed phases
Q = quenching
E = evaporation
L = liquid–vapor interface
W = wall

1. Introduction

The boiling phenomenon is one of the complicated physical phenomena in which its simulation is very challenging. The heat transfer coefficient for a fluid in boiling phenomenon has a maximum rate; therefore, the investigation of heat transfer coefficient in boiling flows has attracted many researchers. Also, today, owing to the especial importance of energy, the engineers and researchers in energy fields are trying to prevent wasting of energy by optimizing and enhancing the amount of heat transfer. Faulkner *et al.* (2003) and Lee and Mudawar (2007) investigated the heat transfer in the forced boiling of water flow, and to enhance the heat transfer, they used

Al_2O_3 and ceramic nanoparticles. Their results showed that by adding nanoparticles to the base fluid with the existence of boiling phenomenon, the heat transfer was enhanced. The forced boiling of water/CuO nanofluid has been studied by Peng *et al.* (2009a), (2009b) and Boudouh *et al.* (2010). Their results revealed that using nanofluid in boiling phenomenon causes heat transfer enhancement but adding nanoparticles culminates in the enhancement of pressure drop. Also, Kim and his coworkers, by studying the forced boiling flow of water/Al_2O_3 nanofluid, concluded that, the increase of the amount of nanoparticles to the base fluid enhances the critical flux to 70 per cent (Kim *et al.*, 2010b, 2010a). Tang *et al.* (2016) experimentally studied the physics of bubbling in an enclosure and concluded their study by increasing the destruction frequency in a subcooled liquid, the heat flux enhances. Bang and Chang (2005) and Kwark *et al.* (2010) investigated the pool boiling in water/Al_2O_3 nanofluid in a flat plate under the atmosphere pressure. According to their reports, the heat transfer characteristic in the nuclear boiling, by using nanofluid, was reduced and the critical heat flux, comparing to water, was increased. This issue is due to the changing of properties of heat transfer surface, because of the sedimentation of nanoparticles and creation of the thin film layer on the heater surface. Rudemiller (1989), in a research about boiling phenomenon, studied the phase changing in a porous medium. In the mentioned study, the porous surface has been considered to be made of ceramics. Their results revealed that, the properties of porous medium have great effect on heat transfer enhancement of the boiling phenomenon. Mori and Okuyama (2009) studied the critical heat flux in the pool boiling with the existence of porous medium. Rao (1997) studied the boiling phenomenon in a porous medium by using an empirical code. In his research, the pressure drop inside the porous area has been calculated by using Darcy flow. The slope of vapor pressure figure has been revised by using Clapeyron equation. The portion of heat flux caused by the external single-phase convection has been considered when the porous medium is small. The pressure drop results and the created flux on the wall have been obtained. Lioumbas and Karapantsios (2015) empirically studied the bubble dynamics caused by oil boiling in a porous medium. A relation between gravity and heat transfer coefficient which is non-linear in the porous medium was reported. Their results showed that, by increasing the initial temperature of oil, boiling happens quicker and the temperature profile is created with high rates. These properties also cause enhancement of heat transfer rate at the beginning and during the boiling. Taherzadeh and Saidi (2015) studied the phase changing in the porous medium. Two geometrics including the flat and porous surfaces have been studied. Their results evidenced that using porous surface had great effect on heat transfer. Gong and Cheng (2015) numerically simulated the pool boiling phenomenon by using lattice Boltzmann method. In the mentioned study, the rate of surface wettability on the bubble dynamics and the created flux during the boiling phenomenon have been investigated. The presented results indicate that the amount of surface hydrophobic has a great influence on the boiling cycle and the created flux. Jun *et al.* (2016) experimentally studied the pool boiling of the saturated water in the atmosphere pressure on the copper surface with temperature, high thermal conductivity and the microporous copper cover. They studied the effect of particles size and the thickness of cover on the pool boiling properties. According to their results, the nuclear boiling and the critical heat flux enhance significantly by using the micro-porous cover and increasing the dimensions of particles. Kim *et al.* (2015) empirically studied the enhancement of heat transfer rate in the pool boiling by using micro-porous cover. Experiments have been carried out on a 1×1cm horizontal heater in the atmosphere pressure. Their results reported enhancement of heat transfer rate and critical heat flux in both fluids, by using the micro-porous cover.

In present research, by using the computational fluid dynamics, the effect of using a porous medium in the forced boiling of water flow inside a tube has been investigated. The fluid boiling inside the tube has been simulated by using the multi phase Eulerian RPI model (Rensselaer Polytechnic Institute).

2. Problem statement

In this research, the effect of porous medium in the fluid flow (water) inside a vertical tube under the constant heat flux with the existence of boiling phenomenon has been investigated numerically. Flow and heat transfer have been simulated at the Reynolds numbers of 36,000, 73,000 and 110,000 inside the tube. In Figure 1, the geometry and boundary conditions are illustrated. As it can be seen, velocity and the pressure outlet are used as boundary condition, and walls of tube are under the constant heat flux. The inlet temperature of fluid has been considered 60 K lower than the saturated temperature ($T_{sat} - 60\,K$) in the pressure of 45 bar.

3. Governing equations

In present research, the multi-phase Eulerian–Eulerian RPI wall boiling model for simulating the boiling has been used. The governing equations are as follows:

3.1 Equations of the fluid and boiling model

Momentum equation:

$$\frac{\partial\left(\alpha_q \rho_q \vec{V}_q\right)}{\partial t} + \nabla\cdot\left(\alpha_q \rho_q \vec{V}_q \vec{V}_q\right) = -\alpha_q \nabla P + \nabla\cdot\left(\overline{\overline{\tau}}_q\right) + \alpha_q \rho_q \overrightarrow{\beta_f} + \sum_{r=1}^{n}\left(\vec{F}^{D}_{rq} + \vec{F}^{TD}_{rq} + \dot{m}_{rq}\vec{V}_{rq} - \dot{m}_{qr}\vec{V}_{qr}\right) + \left(\vec{F}_q + \vec{F}^{L}_q + \vec{F}^{vm}_q\right) \quad (1)$$

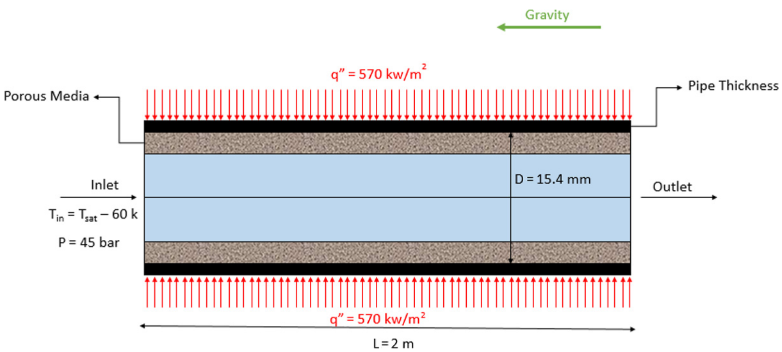

Figure 1.
The geometry of the tube

where, P is pressure; $\overline{\overline{\tau}}_q$ is the shear stress tensor in q^{th} phase; β_f is the body force; $\overrightarrow{F}_{rq}^{D}$ is the interaction drag force between two phases; $\overrightarrow{F}_{rq}^{TD}$ is the turbulent dispersed force; $\overrightarrow{V}_{qr}$ and $\overrightarrow{V}_{rq}$ are the relative velocity of two phases relative to each other; and $\overrightarrow{F}_q$, $\overrightarrow{F}_q^{L}$ and $\overrightarrow{F}_q^{vm}$ are the external forces, lift and the virtual mass exchange forces, respectively.

Energy equation:

$$\frac{\partial(\alpha_q \rho_q H_q)}{\partial t} + \nabla \cdot \left(\alpha_q \rho_q \overrightarrow{V}_q H_q\right) = \overline{\overline{\tau}}_q : \nabla \cdot \overrightarrow{V}_q + \alpha_q \frac{\partial p}{\partial t} - \nabla \cdot \overrightarrow{\dot{q}} + S_{H.q} + \sum_{r=1}^{n} \left(\dot{q}_{rq} + \dot{m}_{rq} H_{rq} - \dot{m}_{qr} H_{qr}\right) \tag{2}$$

where H_q is the specific enthalpy of each phase, $\overrightarrow{\dot{q}}$ is heat flux, $S_{H.q}$ is the external energy source term, $\dot{q}_{rq}$ is the rate of heat exchange between two phases and H_{rq} and H_{qr} are the enthalpy of each phase.

In present numerical solving procedure, for discretizing governing equations on momentum and pressure, the second-order discretizing and for coupling velocity and pressure, simple algorithms are used. Also, for discretizing energy equation, the second-order method is used and the convergence criterion is considered 10^{-6}. Depending on the situation, the convergence of the solution is very difficult. The relaxation factors are changed during the solution to obtain convergence.

3.1.1 Rensselaer Polytechnic Institute boiling model. The most common and known model for numerical simulation of boiling is RPI wall boiling model (Cole, 19606). According to RPI model, the total heat flux from the wall to the fluid is divided into three parts: $\dot{q}_C$ is the liquid phase heat flux caused by the vicinity of wall, $\dot{q}_Q$ is the quenching heat flux related to the convection of bubbles and $\dot{q}_E$ is the evaporation heat flux:

$$\dot{q}_W = \dot{q}_C + \dot{q}_Q + \dot{q}_E \tag{3}$$

The RPI model, by considering the wall surface to the area covered by bubble $(1 - A_b)$ and the other area covered by liquid, for three terms of the mentioned heat flux, following equations are governed:

The convection of liquid phase heat flux:

$$\dot{q}_C = h_C(T_w - T_l)(1 - A_b) \tag{4}$$

where h_c is the convection heat transfer coefficient of liquid phase, T_w and T_l are the wall temperature and the liquid phase temperature neighbor to the wall, respectively. $\dot{q}_Q$ part of the total heat flux which is periodically filled with f_{bw} frequency by the liquid after separating the bubble from the surface is given as:

$$\dot{q}_Q = C_{wt} \frac{2k_l}{\sqrt{\pi \gamma_l / f_{bw}}} (T_w - T_{l.q}) A_b \tag{5}$$

where, k_l is the thermal conductivity of liquid phase, f_{bw} is the frequency of bubble creation and $C_{wt} = 1$. In RPI model, $T_{l.q}$ is the mean temperature of liquid from the cross section.

The evaporation heat flux is calculated from following equations:

$$\dot{q}_E = \frac{\pi}{6} d_{bw}^3 f_{bw} N_w \rho_v H_{lv} \tag{6}$$

where, d_{bw} is the departure diameter of bubble, N_w is the active nuclear density, ρ_v is the vapor density and H_{lv} is the evaporation latent heat. The above equation is determined by the following parameters:

The influence area is divided according to the diameter of the created bubble and the core density:

$$A_b = min\left(1, \eta \frac{\pi}{4} d_{bw}^2 N_w\right) \tag{7}$$

Where η is the obtained coefficient from the empirical experiments which are suggested by Del Valle for its calculation:

$$\eta = 4.8\, exp\left(-\frac{\mathrm{Ja}}{80}\right) \tag{8}$$

Here, Ja is the Jacob number. Usually, the frequency of bubble creation can be calculated by the equation presented in Abu-Hijleh *et al.*'s (2004) study as follow:

$$f_{bw} = \sqrt{\frac{4g(\rho_l - \rho_v)}{3\rho_l d_{bw}}} \tag{9}$$

Where, g is the gravity acceleration.

3.2 Porous media equation

Porous media are modeled by the addition of a momentum source term to the standard fluid flow equations. The source term is composed of two parts: a viscous loss term (Darcy, the first term on the right-hand side of equation (10) www.sharcnet.ca/Software/Fluent6/html/ug/node272.htm-pormedia-ctn-eq and an inertial loss term [the second term on the right-hand side of equation (10)]:

$$S_i = -\left(\sum_{j=1}^{3} D_{ij}\mu u_j + \sum_{j=1}^{3} C_{ij}\frac{1}{2}\rho|u|u_j\right) \tag{10}$$

To recover the case of simple homogeneous porous media:

$$S_i = -\left(\frac{\mu}{K}u_i + C_1\frac{1}{2}\rho|u|u_j\right) \tag{11}$$

μ and ρ are viscosity and fluid density, K is permeability and C_1 is the inertial resistance factor of the porous medium. In fact, the matrices C and D are two diagonal matrices whose elements are C_1 and $1/K$, respectively. This model has been developed to calculate the pressure drop inside the porous medium, namely, the Darcy–Forshimer equation, which is expressed as an experimental relation between permeability and inertial coefficient. This model has a wide application in numerical and experimental modeling of porous media with low porosity and low permeability (Abu-Hijleh *et al.*, 2004).

3.2.1 Energy equation in porous media. The standard energy transport [equation (12)] in porous media regions with modifications to the conduction flux and the transient terms only. In the porous medium, the conduction flux uses an effective conductivity, and the transient term includes the thermal inertia of the solid region on the medium:

$$\frac{\partial}{\partial t}\left(\varepsilon \rho_f E_f + (1-\varepsilon)\rho_s E_s\right) + \nabla \cdot \left(\vec{u}\left(\rho_f E_f + p\right)\right)$$

$$= \nabla \cdot \left[k_{eff} \nabla T - \left(\sum_i h_i\, j_i\right) + \left(\overline{\overline{\tau}} \cdot \vec{u}\right)\right] + S_f^{\,h} \tag{12}$$

where, E_f is total solid medium energy, ε is porosity of the medium, $S_f^{\,h}$ is fluid enthalpy source term and k_{eff} is effective thermal conductivity of the medium:

$$(\mathrm{k}_{eff} = (1-\varepsilon)k_s + \varphi k_f)$$

where k_f is the fluid thermal conductivity and k_S is the solid thermal conductivity (Bartolemei and Chanturiya, 1969).

3.3 Turbulence model

The general equation of turbulence model is as follows (Rezaei *et al.*, 2017; Pourfattah *et al.*, 2017; Toghraie, 2016; Hosseinnezhad *et al.*, 2018; Parsaiemehr *et al.*, 2018; Ali Rahimi Gheynani *et al.*, 2018; Erfan *et al.*, 2019; Toghraie *et al.*, 2019; Pourdel *et al.*, 2019):

$$\frac{\partial\left(\alpha_q \rho_q \phi_q\right)}{\partial t} + \nabla \cdot \left(\alpha_q \rho_q \vec{V}_q \phi_q\right) = \nabla \cdot \left(\alpha_q\ \Gamma_{\phi.q} \nabla \phi_q\right) + \alpha_q S_{\phi.q} \tag{13}$$

where, $\Gamma_{\phi.q}$ is the diffusion coefficient, $S_{\phi.q}$ is the source term including the generation, dissipation and the term of source created from the interaction of bubbles.

4. Grid independency

To investigate the grid independency of the obtained numerical, different grid numbers have been used and the obtained results are presented in Figure (2). According to this figure, it can be observed that if the grid number is more than 38,000, the Nusselt number does not have considerable change. So, in all simulations, this grid number has been used. In the mentioned meshing, the size of all elements along the tube has been considered as 0.0014 m.

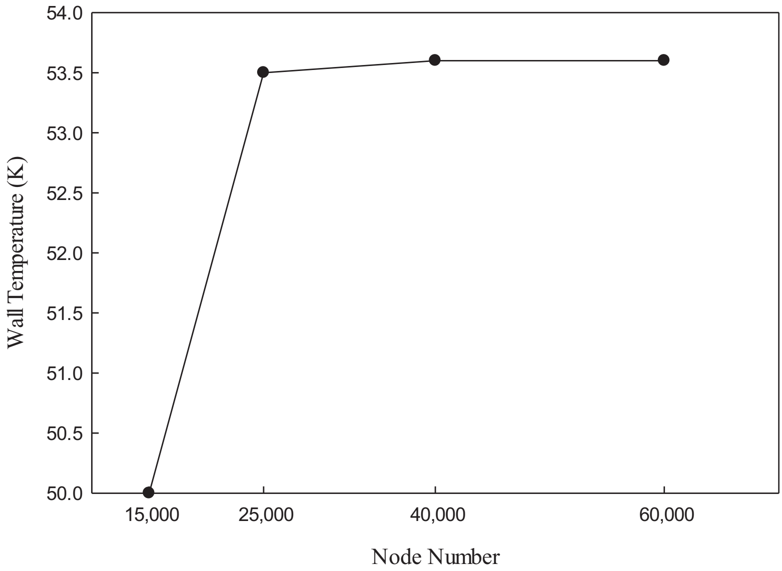

Figure 2.
The grid independency study

5. Validation

Before simulation, it is necessary to ensure the accuracy of the obtained results. For this reason, after reviewing some studies Bartolemei and Chanturiya's (1969) study has been selected as a valid reference. The mentioned reference has experimentally studied the water flow inside a vertical tube with the existence of boiling phenomenon. In this reference, the diameter of tube is 15.4 mm and the walls of tube are made of stainless steel, which is under the constant heat flux. In Figure 3, the temperature of the liquid phase obtained from the numerical simulation at the center of tube has been compared with the presented results presented in Bartolemei and Chanturiya (1969). As it can be seen, the obtained results have good agreement with the experimental results, indicating that the numerical procedure has sufficient accuracy.

6. Results and discussions

6.1 Effect of the Reynolds number on the flow boiling

In this section, the effect of Reynolds number on boiling process has been studied. In Figure 4, the temperature distribution on surface has been illustrated. As seen, by increasing the Reynolds number, surface temperature reduces; therefore, at the Reynolds numbers of 36,000, 73,000 and 110,000, the maximum temperature of tube are, respectively, equal to 568 K, 549 K and 540 K which indicates the increase of heat transfer from surface with the increase of Reynolds number and is compatible with the empirical and theoretical results governing on the flow about the proportionality of the Reynolds number and the Nusselt number. By investigating the distribution of volume of fraction (VOF) on tube wall illustrated in Figure 5, it is specified that by increasing the Reynolds number, the beginning of phase changing postpones and the volume fraction of created vapor in tube reduces. In a way that flow at Re = 36,000 fills with the created vapor phase with the maximum volume fraction is 0.95, at Re = 73,000 the

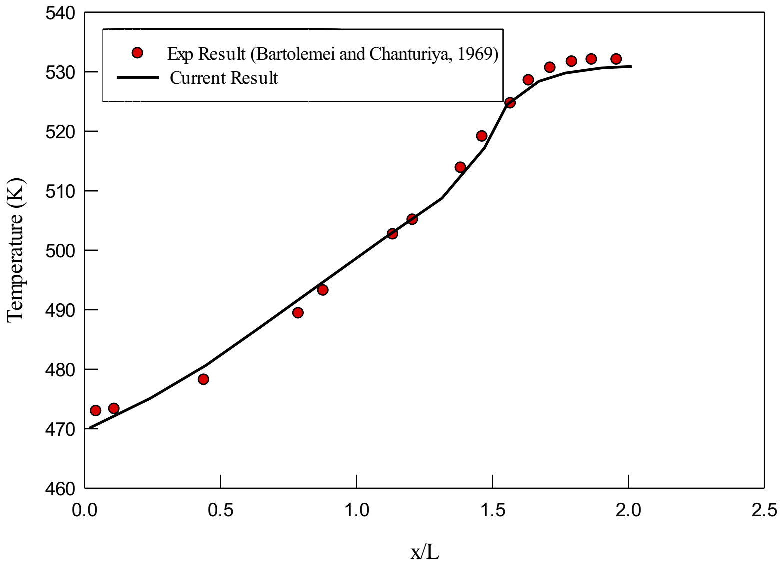

Figure 3.
Validation of the present results with (Abu-Hijleh *et al.*, 2004)

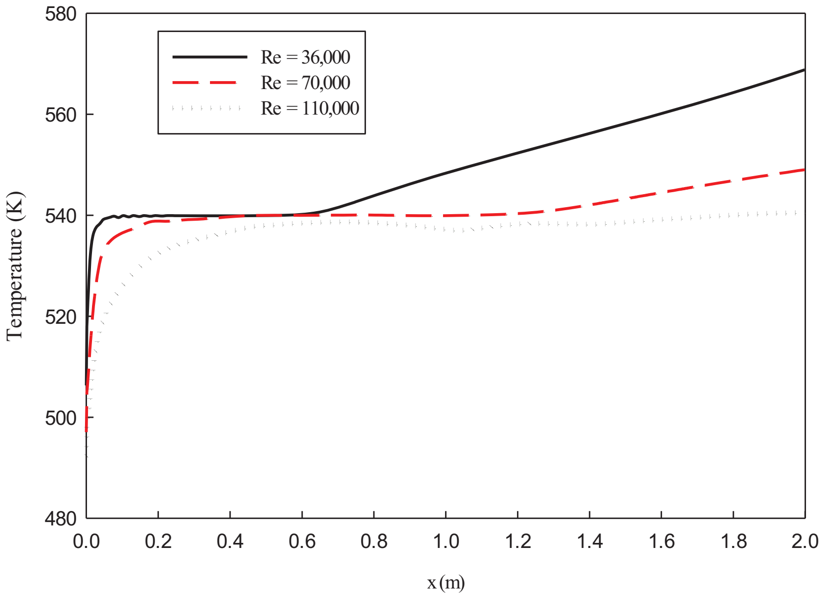

Figure 4.
Temperature distribution of tube wall in flow with three different Reynolds numbers

maximum amount of volume fraction is 0.8 and at $Re = 110{,}000$, the maximum amount of vapor phase volume fraction arises to 0.45. According to the obtained results, it can be said that the increase in the Reynolds number accompanied with the enhancement of fluid flow momentum in the uniform heat flux leads to the reduction of boiling rate.

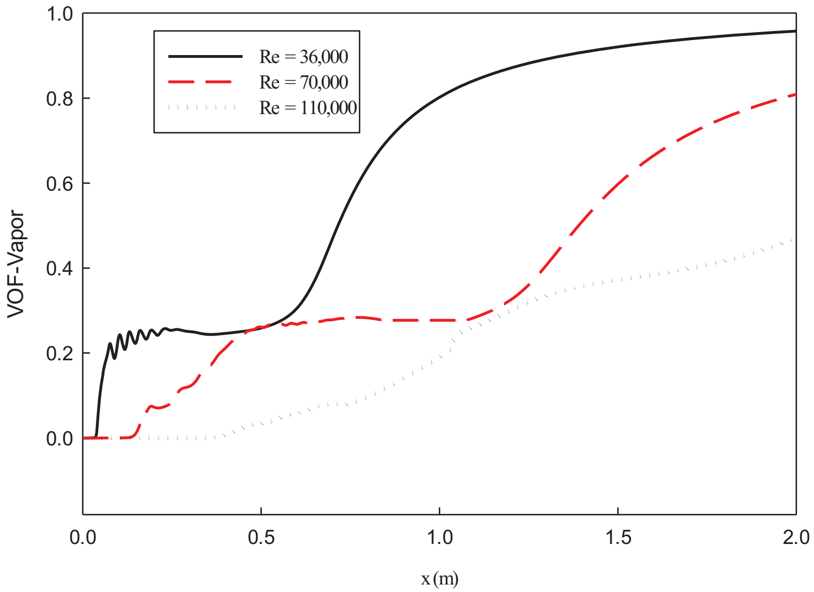

Figure 5.
Volume fraction distribution of vapor phase in flow with three different Reynolds numbers

6.2 Effect of porous medium at Re = 36,000

In this section, the effect of using a porous medium with two different thermal conductivities and the same porosity of 0.5 inside the tube have been investigated. In Figure 6, the temperature distribution on the wall in three different conditions including flow inside the tube without porous medium and with the existence of porous medium with two thermal conductivities of 15 W/m-K and 30 W/m-K has been illustrated. As it can be observed, by using porous medium, the wall temperature in the inlet region reduces (the region in which the prevailed phase is liquid and the two-phased flow has not yet begun) and the temperature increases with lower slope. The reason of this issue is the improvement of heat transfer from the wall to the fluid inside the tube by porous medium. In Figure 7, the volume fraction distribution of vapor phase has been demonstrated. As it can be seen, by increasing the thermal conductivity of porous medium, the beginning of bubbling postpones and flow remains a single phase liquid in most of the initial parts, which according to the high thermal conductivity of liquid is compared to the vapor, the wall temperature in sections of tube with liquid phase is low. In the following, by studying the effect of temperature and volume fraction distribution of the vapor phase on the wall at higher Reynolds numbers and presenting the mean temperature of wall, convection heat transfer coefficient and the Nusselt number, the thermal performance will be investigated.

6.3 Effect of porous medium at Re = 73,000

In previous sections, by studying the effect of Reynolds number, it is specified that, by increasing Reynolds number, bubbling delays and in fact, with constant heat flux applied to the wall, by increasing fluid velocity, the rate of boiling reduces and the maximum quality of the created vapor decreases and this issue causes the reduction of mean temperature of wall. In this section, the effect of using a porous medium in the

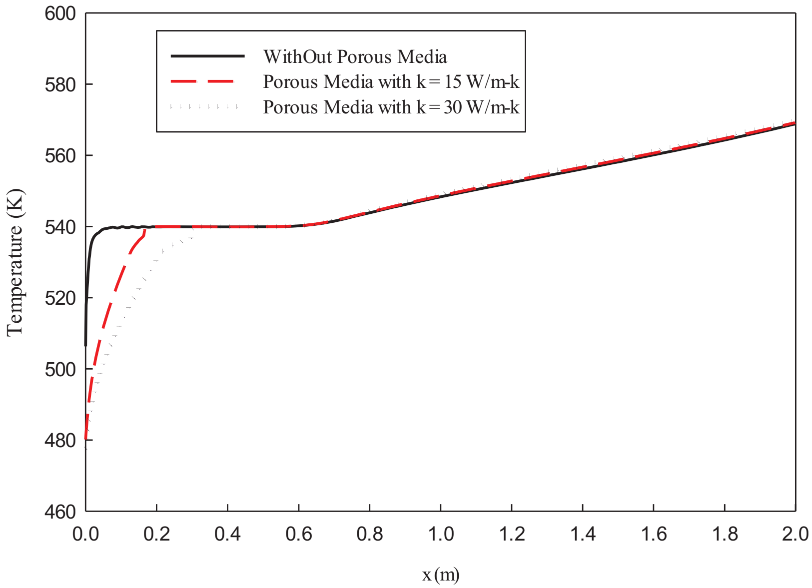

Figure 6.
Distribution of wall temperature at Re = 36,000 with the existence of porous medium

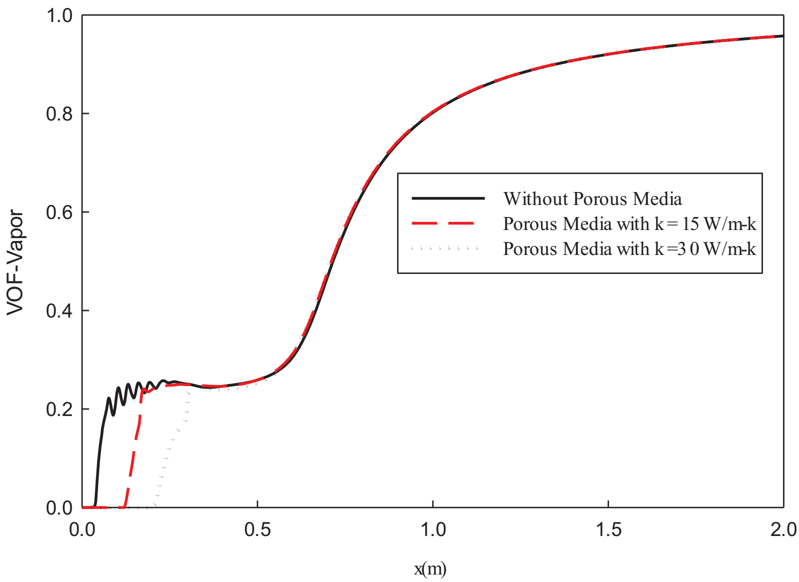

Figure 7.
Volume fraction distribution at Re = 36,000 with the existence of porous medium

arrangement with two thermal conductivities of 15 W/mK and 30 W/mK indicated in Figure 1 is investigated. In Figure 8, the temperature distribution on the wall in three different studied conditions has been shown. As it can be seen, by using of porous medium inside the tube, the tube temperature at the inlet section of tube grows with

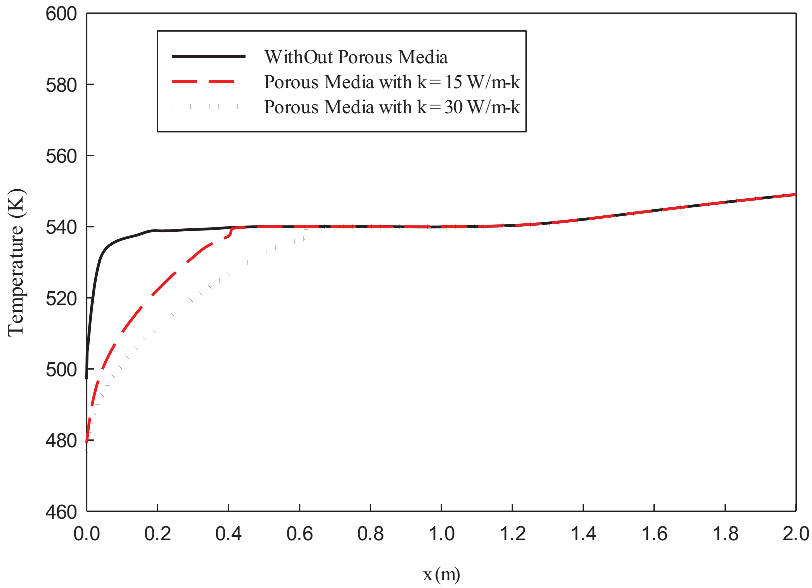

Figure 8. Distribution of wall temperature at $Re = 73{,}000$ with the existence of porous medium

lower slope, and by increasing thermal conductivity of porous medium, this slope reduces more. Also, using a porous medium postpones the phase change procedure. In Figure 9, the volume fraction distribution indicates this issue.

6.4 The effect of porous medium at Re = 110,000

In this section, the distribution of local temperature and distribution of volume fraction of vapor phase on the tube in flow with $Re = 110{,}000$ have been presented. By studying the effect of porous medium in boiling procedure with lower Reynolds number, it is specified that, by using a porous medium, bubbling delays and the two-phase flow with the abeyance in tube temperature at the initial sections grows with lower slope. The temperature distribution and the volume fraction distribution of vapor phase at $Re = 110{,}000$ have been demonstrated in Figures 10 and 11, respectively. By studying these Figures 11 and 12, it is specified that, at $Re = 110{,}000$, the governing procedure on the effect of porous medium on the boiling is like flow with $Re = 36{,}000$ and $Re = 730{,}000$. With increasing Reynolds number, the onset of bubbling postpones and by using a porous medium with $k = 30$ W/mK, the beginning of boiling procedure, in 1/3 of tube, postpones.

6.4.1 Investigation of flow physics. In previous sections, by presenting the quantitative results, the effect of Reynolds number and porous medium on the temperature distribution and volume of fraction distribution of vapor phase have been studied. In this section, by presenting contour of vapor volume of fraction and liquid temperature distribution at $Re = 36{,}000$ and $Re = 73{,}000$, the physics of flow have been quantitatively investigated. By studying the vapor volume fraction and liquid temperature distribution inside the tube (Figures 12 and 13), it can be observed that by entering the flow at $60\,K$ less than the saturated temperature to the tube under the constant heat flux, at the inlet area, at first there is a single-phase flow. By continuing

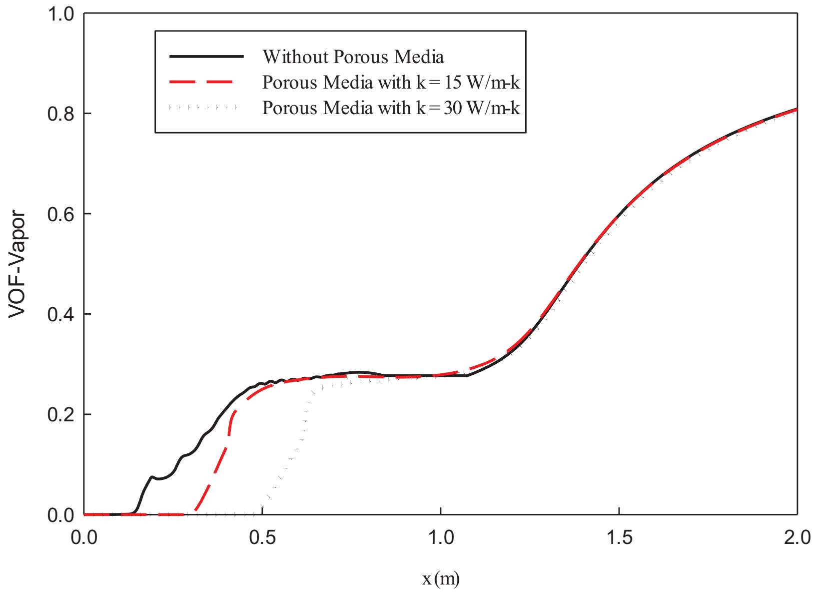

Figure 9.
Volume fraction distribution at *Re* = 73,000 with the existence of porous medium

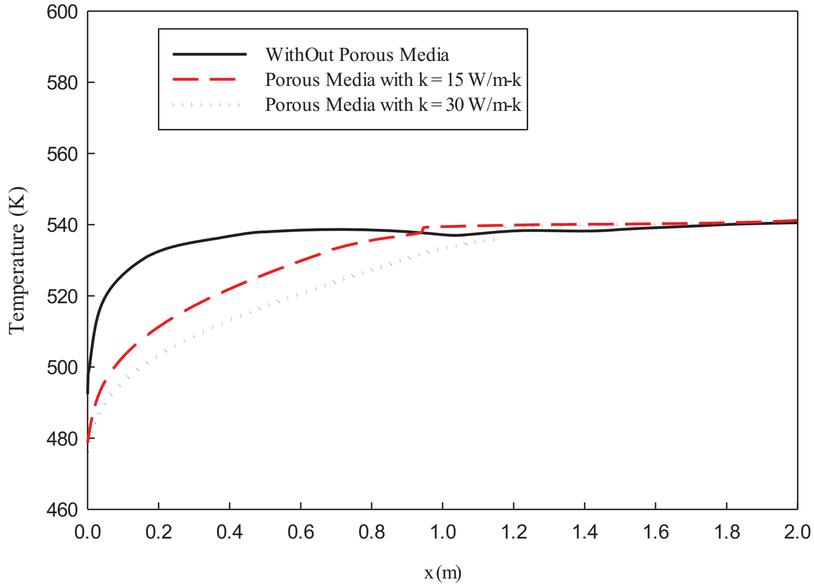

Figure 10.
Temperature distribution of wall in flow with *Re* = 11,000 with the existence of porous medium

the flow along the tube, fluid temperature increases [Figure 12(b) and Figure 13(b)] and after reaching the saturated temperature, the bubbles are generated. With an increase in number of bubbles, the volume fraction of vapor phase increases along the tube.

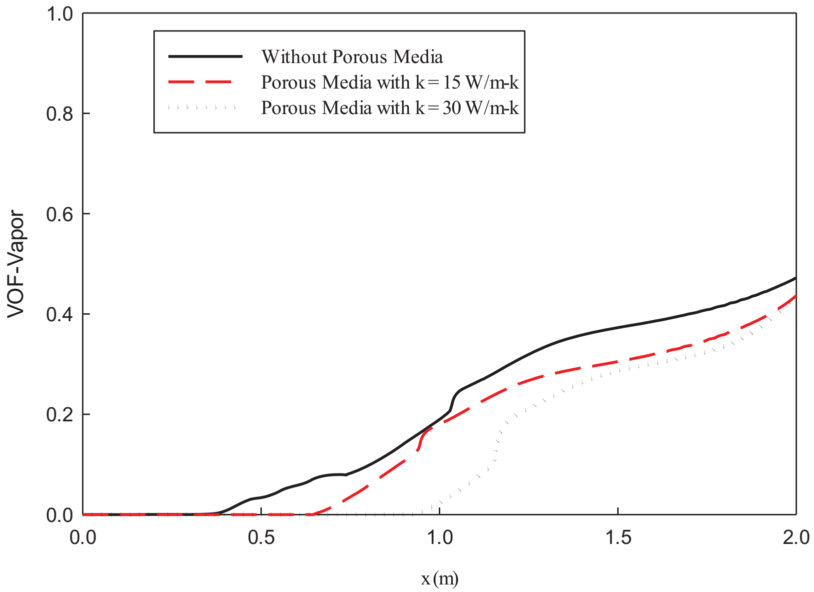

Figure 11. Volume fraction distribution of vapor phase in flow with *Re* = 11,000 with the existence of porous medium

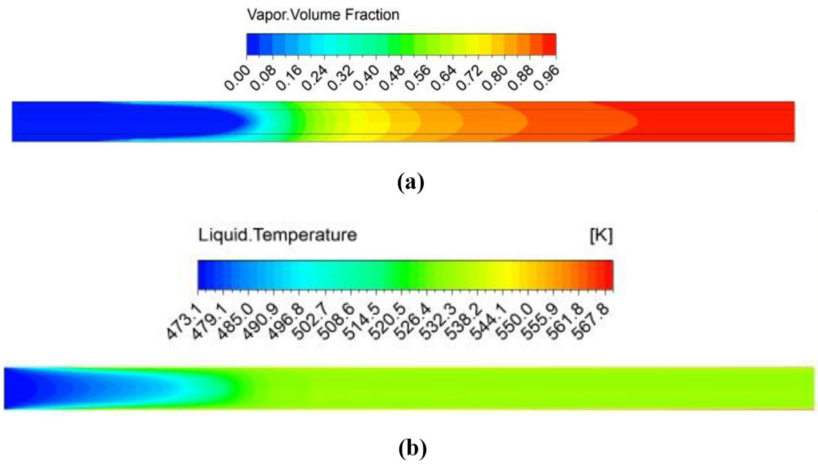

Figure 12. (a) Vapor volume of fraction, (b) Liquid temperature distribution – *Re* = 36,000 in porous medium with *k* = 30W/m-K

6.5 Thermal performance

In this section, the effect of Reynolds number of flow and porous medium specifications on the thermal performance is investigated. In Figure 14, the mean temperature of wall in different conditions has been demonstrated. As it can be seen, by increasing Reynolds number, the mean temperature of wall reduces, indicating enhancement of heat transfer from the surface to the fluid. By studying the effect of thermal conductivity of porous medium, it is observed that by using a porous medium with $k = 15$ W/m-K, the mean temperature of wall reduces. It is notable that by increasing the Reynolds number, thermal conductivity of porous medium has more effect on the reduction of mean temperature.

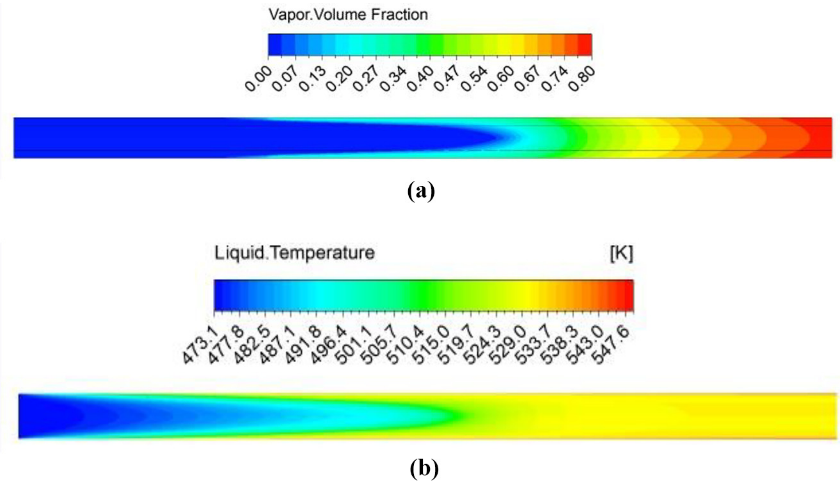

Figure 13. (a) Vapor volume of fraction, (b) Liquid temperature distribution – $Re = 73000$ in porous medium with $k = 30$W/m-K

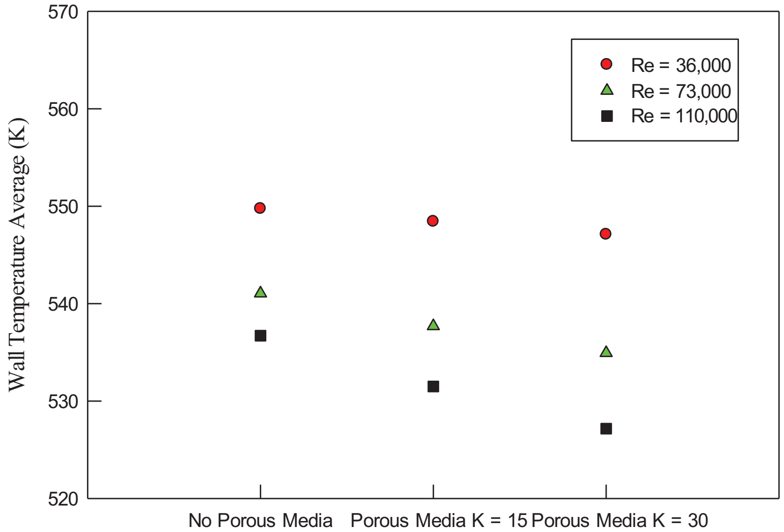

Figure 14. The mean temperature of wall in different conditions

Heat transfer coefficient and the Nusselt number are defined as:

$$h = \frac{q''}{(T_w - T_b)}$$

$$Nu = \frac{hd}{k}$$

where, q'' is the applied heat flux, T_w is the channel wall temperature, T_b is bulk temperature, d is tube diameter and k is the thermal conductivity of the water.

In Figure 15, the convection heat transfer coefficient in the studied conditions has been shown. According to this figure, with increase of the Reynolds number, heat transfer coefficient increases and by increasing the thermal conductivity of porous medium, this coefficient enhances. By considering the Nusselt number, the amount of this quantity in different studied conditions has been demonstrated in Figure 16. According to the mentioned figure, by increasing Reynolds number, the amount of this dimensionless quantity increases and by enhancing the thermal conductivity of porous medium, the Nusselt number increases. At $Re = 36{,}000$ and with $k = 30$ W/mK, the Nusselt number enhances from 55 to 56.87. In other words, because of the existence of porous medium, the Nusselt number enhances to 3.3 per cent, whereas flow at $Re = 110{,}000$, because of the use of porous medium with $k = 30$W/m-K, the Nusselt number enhances from 55.7 to 60.4. In other words, the existence of porous medium enhances the amount of heat transfer to 8.5 per cent. According to the obtained results, it can be said that because of the existence of boiling inside the tube and by using a porous medium to increase the heat transfer, the efficiency of this method in flow at high Reynolds numbers is more.

7. Conclusions

In present research, by using of computational fluid dynamics, the effect of using a porous medium in flow domain with boiling has been investigated. For this reason, the two-phase flow, by using Eulerian–Eulerian multi-phase RPI wall boiling model, has been simulated inside a vertical tube, and the effect of using a porous medium (water) has been studied. The obtained numerical results have been compared with the experimental results and proper coincidence has been arisen. The obtained results revealed that:

- By increasing the Reynolds number, the amount of volume fraction of vapor phase reduces along the tube.
- By increasing Reynolds number, the position of bubbling delays.

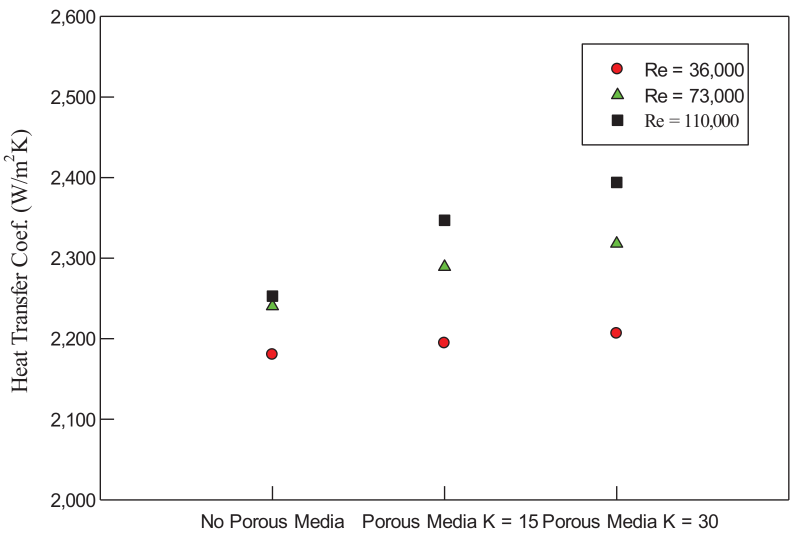

Figure 15.
Heat transfer coefficient in different conditions

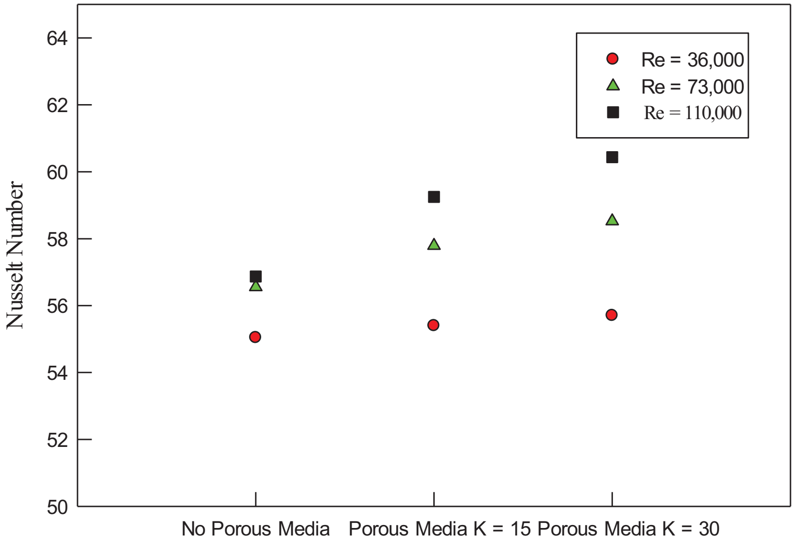

Figure 16.
Nusselt number in different conditions

- By using a porous medium, the mean temperature of wall reduces, and the heat transfer from wall to the fluid enhances.
- By increasing thermal conductivity of porous medium, the mean temperature of wall reduces and the effect of thermal conductivity enhancement in flow with higher Reynolds numbers is more observable.
- By using porous medium at higher Reynolds numbers flow, the heat transfer increases up to 8.5 per cent.

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