International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:6, No:1, 2012

Improvement of Energy Efficiency using Porous Fins in heat Exchangers

Hadi Niknami Esfahani ,Hossein Shokouhmand, Fahim Faraji

***Abstract*—**The forced convection heat transfer in high porosity metal-foam filled tube heat exchangers are studied in this paper. The Brinkman Darcy momentum model and two energy equations for both solid and fluid phases in porous media are employed .The study shows that using metal-foams can significantly improve the heat transfer in heat exchangers.

***Keywords*—**metal foam, Nusselt number, heat exchanger, heat flux

I. Introduction

RECENTLY, the use of high porosity metal foams have spread to include applications, such as aircraft wing structures for the aerospace industry, catalytic surfaces for chemical reactions, core structures for high strength panels, and containment matrices and burn rate enhancers for solid propellants. Due to the high surface-area density and strong mixing capability for the fluid, opencell metal foams are now regarded as one of the most promising materials for the manufacture of efficient compact heat exchangers (Fig. 1).

Due to high prices, in previous years this technology was rarely used. But nowadays with decreasing in manufacturing costs, using metal foams has become more prevalent. Therefore recently some scientific researches were investigated to study the phenomenon of heat transfer in fluid flow inside the metal foam. V. V. Calmidi and R. L. Mahajan [1-2] conducted a comprehensive experimental and numerical study of forced convection in high porosity aluminum metal foam with variety of porosities and pore densities and using air as the fluid. K. Boomsma and D. Poulikakos [3] developed a geometrical effective thermal conductivity model of metal foam based on the idealized three dimensional basic cell geometry of a foam. Shadi Mahjoob and Kambiz Vafai [4] studied the effects of micro structural metal foam properties, such as porosity, pore and fiber diameters, pore density, and relative density, on the heat exchanger performance. W. Lu a and C.Y. Zhao and S.A. Tassou [5-6] presented an analytical study of the forced convection heat transfer characteristics in high porosity open-cell metal foam filled pipes. L. Tadrist and his colleagues [7] studied the effect of metal foams on heat transfer and pressure drop in a compact heat exchanger.

Hadi Niknami Esfahani is with the School of Mechanical Engineering, College of Engineering, University of Tehran, Tehran, Iran (h_niknami@yahoo.com)
Hossein Shokouhmand is with the School of Mechanical Engineering, College of Engineering, University of Tehran (hshokoh@ut.ac.ir)
Fahim Faraji is with School of Mechanical Engineering, College ofEngineering, University of Tehran, Tehran, Iran (fahimfaraji@ut.ac.ir)

Practically, high porosity metal foams can be considered as a porous medium. For modeling all the heat transfer problems, It requires to solve momentum equation to obtain the velocity distribution. The most simple relation presented for modeling the velocity field in porous media is Darcy [8] equation. He discovered that velocity inside a porous material is proportional to the pressure drop and inversely proportional to the viscosity. After Darcy, and unlike his equation, some other relations (including Brinkman-Darcy [9]) were presented which considered the effect of boundary conditions. Energy equations are derived from the First law of thermodynamics, and they are solved to find out the temperature distribution with the help of velocity distribution that are just acquired by solving momentum equations.

TABLE I
NOMENCLATURE

Symbol	Quantity	
T	temperature	
t	time	
ε	porosity	
Pr	Prandtl number	
d	pipe diameter	m
P	pressure	Pa
C_f	friction factor	m
h	heat transfer coefficient	w/m^2K
h_{sf}	heat transfer interfacial coefficient of metal foams	w/m^2K
k	thermal conductivity	W/m K
K	Permeability	m^2
Nu_s	Nusselt number	hd/k_s
q	heat flux	w/m^2
q''	Internal heat generation	w/m^3
R	pipe radius	m
u	velocity	m/s
u_m	mean fluid velocity	m/s
Re_d	local Reynolds number	$\rho u_m d/\mu$
ã	surface area density	m^{-1}
Br	Brinkman number	
C	heat capacity of fluid	J/kg K
μ	dynamic viscosity	kg/m s
ρ	density	kg/m^3

II. Mathematical modeling

Mass conservation equation in integral form is expressed as follow:

International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:6, No:1, 2012

$$\pi R^2 U_m = 2\pi \int u.r.dr \quad (1)$$

An alternative form of Darcy's equation is what is commonly known as Brinkman's [10] equation that have two viscous terms.

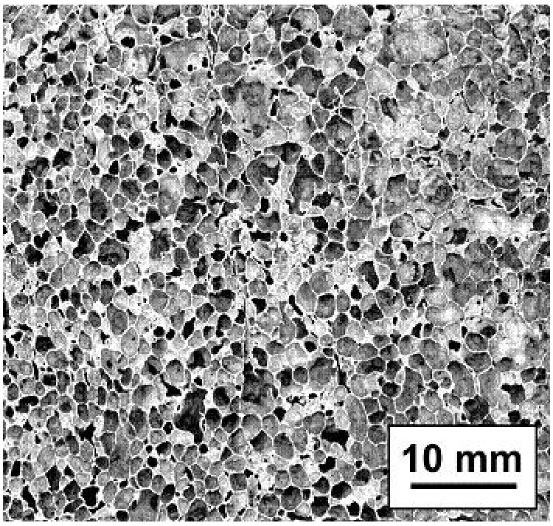


Fig. 1 Photo of a 10 PPI aluminum foam

$$\nabla P = -\frac{\mu}{K}u + \frac{\mu}{\varepsilon}\nabla^2 u \quad (2)$$

Energy equation for the solid phase [10],

$$(1-\varepsilon)(\rho c)_s \frac{\partial T_s}{\partial t} = \nabla.(k_s \nabla T_s) + q_s'' + a_{sf}h_{fs}(T_f - T_s) \quad (3)$$

and for the fluid phase,

$$\varepsilon(\rho c)_f \frac{\partial T_f}{\partial t} + (\rho c)_f u.\nabla T_f = \nabla.(k_f \nabla T_f) + q_f'' + a_{sf}h_{fs}(T_s - T_f) \quad (4)$$

In above equations, $q''[W/m3]$ is the heat production per unit volume. The equations have been simplified as follow:

$$k_s \nabla^2 T_s + q_s'' + a_{sf}h_{fs}(T_f - T_s) = 0 \quad (5)$$

$$(\rho c)_f u.\nabla T_f = k_f \nabla^2 T_f + q_f'' + a_{sf}h_{fs}(T_s - T_f) \quad (6)$$

Energy and momentum equations in cylindrical coordinates are in the following form:

$$\frac{dP}{dx} = -\frac{\mu}{K}u + \frac{\mu}{\varepsilon}\left(\frac{\partial^2 u}{\partial r^2} + \frac{1}{r}\frac{\partial u}{\partial r}\right) \quad (7)$$

$$k_s\left(\frac{\partial^2 T_s}{\partial r^2} + \frac{1}{r}\frac{\partial T_s}{\partial r}\right) + q_s'' + a_{sf}h_{fs}(T_f - T_s) = 0 \quad (8)$$

$$(\rho c)_f u.\frac{dT_f}{dx} = k_f\left(\frac{\partial^2 T_f}{\partial r^2} + \frac{1}{r}\frac{\partial T_f}{\partial r}\right) + q_f'' + a_{sf}h_{fs}(T_s - T_f) \quad (9)$$

The boundary conditions can be specified as follows:

$$r = 0 \qquad \frac{\partial u}{\partial r} = \frac{\partial T_S}{\partial r} = \frac{\partial T_f}{\partial r} = 0 \quad (10)$$

$$r = R \qquad u = 0,\ T_s = T_f = T_w \quad (11)$$

For the pipes directly heated from outside surface with constant heat flux, The following equation is obtained for dT_f/dx [11],

$$\frac{dT_f}{dx} = 2\frac{q_w}{R(\rho c)_f u_m} \quad (12)$$

A. Effective thermal conductivities of the solid and fluid (K_s and K_f)

To determine the effective thermal conductivity of open cell metal foams, K_s and K_f, the following correlation, which is based on the three-dimensional cellular morphology, was proposed by Boomsma and Poulikakos [3].

$$\mathrm{K_e} = \frac{\sqrt{2}}{2(\mathrm{R_A} + \mathrm{R_B} + \mathrm{R_C} + \mathrm{R_D})}. \quad (13)$$

Where:

$$\mathrm{R_A} = \frac{4\mathrm{d}}{(2\mathrm{e}^2 + \pi\mathrm{d}(1-\mathrm{e})\mathrm{K_s} + (4 - 2\mathrm{e}^2 - \pi\mathrm{d}(1-\mathrm{e})\mathrm{K_f}} \quad (14)$$

$$\mathrm{R_B} = \frac{(\mathrm{e} - 2\mathrm{d})^2}{(\mathrm{e} - 2\mathrm{d})\mathrm{e}^2\mathrm{K_s} + (2\mathrm{e} - 4\mathrm{d} - (\mathrm{e} - 2\mathrm{d})\mathrm{e}^2)\mathrm{K_f}} \quad (15)$$

$$\mathrm{R_C} = \frac{(\sqrt{2} - 2\mathrm{e})^2}{2\pi\mathrm{d}^2(1 - 2\mathrm{e}\sqrt{2})\mathrm{K_s} + 2(\sqrt{2} - 2\mathrm{e} - \pi\mathrm{d}^2(1 - 2\mathrm{e}\sqrt{2}))\mathrm{K_f}} \quad (16)$$

$$\mathrm{R_D} = \frac{2\mathrm{e}}{\mathrm{e}^2\mathrm{K_s} + (4 - \mathrm{e}^2)\mathrm{K_f}} \quad (17)$$

Where $e = 0.339\ and$:

$$d = \sqrt{\frac{\sqrt{2}(2 - \left(\frac{5}{8}\right)e^3\sqrt{2} - 2\varepsilon)}{\pi(3 - 4e\sqrt{2} - e)}} \quad (18)$$

B. Surface area density (a_{sf})

The solid–fluid interfacial surface area density for an array of parallel cylinders intersecting in three mutually perpendicular directions, whose cylinder diameter is d and interval is a, is $\frac{3\pi d}{a^2}$. However, the topology of metal-foams is different from the cross-cylinder. Furthermore, the cross-section of the fiber is not circular when the porosity of metal-foams is higher than 0.85. Shape factors must be introduced when the formula of cross-cylinder is used to simplify the structure of metal-foams, which are a = 0.59dp, and $d = (1 - \exp\left(-\left(\frac{1-\varepsilon}{0.04}\right)\right)).d_f$ [1]. Then the surface area density of metal-foams becomes:

$$a_{sf} = \frac{3\pi d_f (1 - e^{-\left(\frac{1-\varepsilon}{0.04}\right)})}{(0.59\text{dp})^2} \tag{19}$$

III. Numerical results

Numerical solutions for dimensionless energy and momentum equations are obtained by finite difference method. Numerical integration, was carried out based on the Romberg method [12]. To confirm the validity of the numerical results, we have compared them with analytical relations in section 4.

A. Momentum equation

Fig. 2 shows that velocity distribution for $r^* < 0.95$ is uniform.This result is a useful assumption for simplification the heat transfer equations.

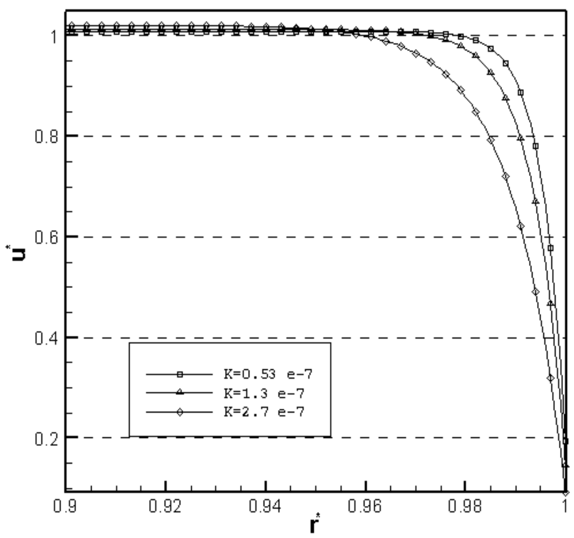


Fig. 2 Velocity distribution in a metal foam filled pipe

B. Energy equation

As it can be seen, energy equations depend on various parameters. Therefore, it is impossible to summarize the numerical results in a simple diagram. So in any of the following diagrams, it has been tried to examine the effect of tow parameters on the heat transfer phenomenon. The effect of Re_d, K_s and K_f on the Nusselt number is shown In Fig. 3,4. Fig. 3 shows that with increasing k_s , Nusselt number reduces, but it doesn't show a decrease the heat transfer coefficient. Practically one of the main methods of increasing heat transfer coefficient is increasing k_s. It also can be understood that the effect of Reynolds number on the Nusselt number is negligible when $Re_d > 10^5$.

In Fig. 4 the effect of fluid thermal conductivity has been investigated. It shows that the thermal conductivity of gases has a little impact on Nusselt number and it is a useful result for simplification of heat transfer equations.

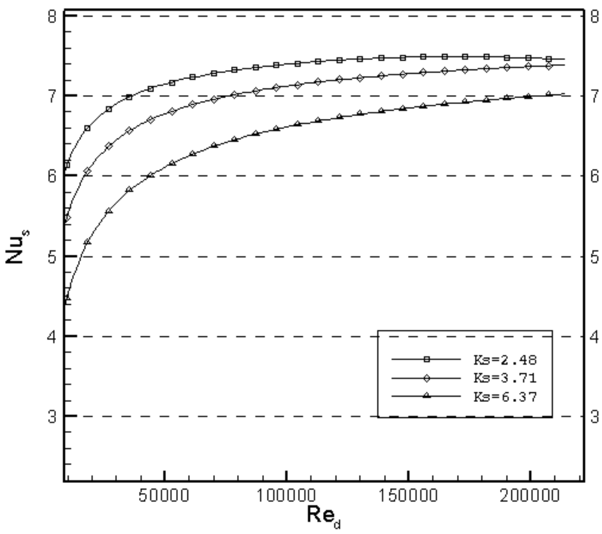


Fig. 3 The effect of k_s on Nusselt number

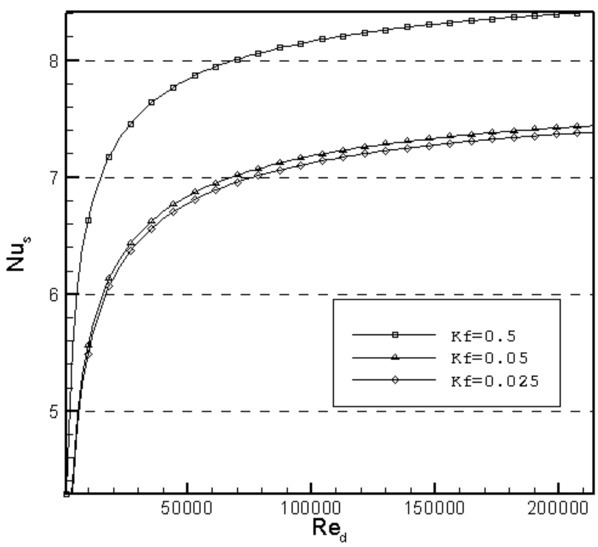


Fig. 4 The effect of k_f on Nusselt number

IV. Analytical result

With the help of numerical results, we can assume that the velocity distribution is uniform. Also when we have gas flow in the metal foam, we can neglect the conduction in the gas phase. Therefore we can write the equations as follows:

$$k_s\left(\frac{\partial^2 T_s}{\partial r^2} + \frac{1}{r}\frac{\partial T_s}{\partial r}\right) + q_{fr} + a_{sf}h_{fs}(T_f - T_s) = 0 \tag{20}$$

$$(\rho c)_f u.\frac{dT_f}{dx} = q_{fr} + a_{sf}h_{fs}(T_s - T_f) \tag{21}$$

$$\frac{dT_f}{dx} = 2\frac{q_w}{R(\rho c)_f u} \quad (22)$$

Variable F is defined as follows:

$$F = \left(\frac{2q_w}{R\,k_s} - \frac{2q_{fr}}{k_s}\right) \quad (23)$$

the temperature profiles can be obtained from 18,19.

$$T_s = T_w - \frac{F}{4}(R^2 - r^2) \quad (24)$$

$$T_f = T_w - \frac{F}{4}(R^2 - r^2) - \frac{F\,k_s + q_{fr}}{a_{sf}h_{fs}} \quad (25)$$

T_{mf} is the bulk mean fluid temperature averaged along the cross section of the channel, given by:

$$T_{mf} = \frac{\int_0^{2\pi}\int_0^r uT_f r dr d\theta}{\int_0^{2\pi}\int_0^r u r dr d\theta} = T_w - \frac{F}{8}R^2 - \frac{F\,k_s + q_{fr}}{a_{sf}h_{fs}} \quad (26)$$

And finally we have heat transfer coefficient and Nusselt number relations.

$$\frac{1}{h} = \frac{T_w - T_{mf}}{q_w} = \frac{d}{8\,k_s} - \frac{d.Br}{8\,k_s} + \frac{4}{d\,a_{sf}h_{fs}} - \frac{2Br}{d\,a_{sf}h_{fs}} \quad (27)$$

$$\frac{1}{Nu_s} = \frac{k_s}{hd} = \frac{1}{8} - \frac{Br}{8} + \frac{4k_s}{a_{sf}h_{fs}d^2} - \frac{2k_s.Br}{a_{sf}h_{fs}\,d^2} \quad (28)$$

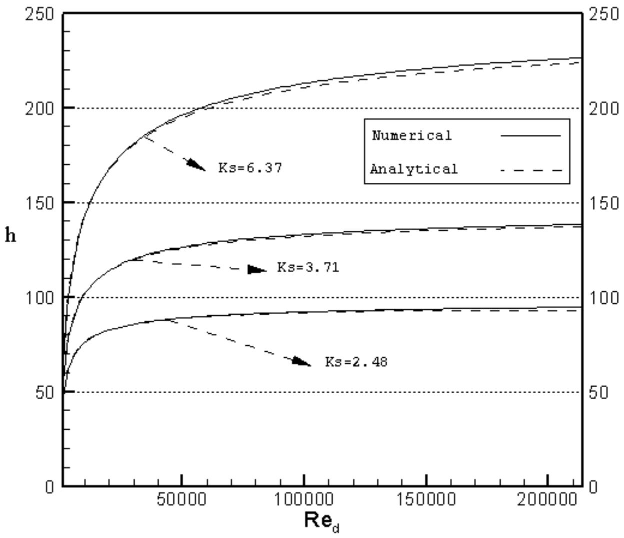


Fig. 5 Comparison of numerical and analytical results

We have compared the heat transfer coefficient obtained from analytical and numerical results in Fig. 5. It shows that simplified analytical relations predict the numerical results very well in gas flow.

TABLE II
THE EFFECT OF METAL FOAMS ON HEAT EXCHANGERS (RE=31629)

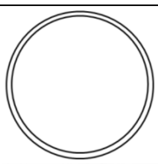
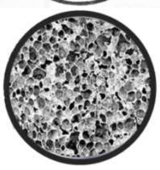
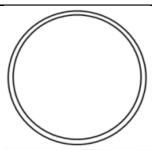
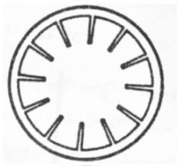
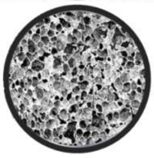
Foam Type	Pressure Drop	Heat Transfer Coefficient
	f= 0.0061 $\frac{dp}{dx} = 3.58$	$Nu = 0.021\,Re^{0.8}\,Pr^{0.4}$ h=19.05 $A_i h_i = 5.9847$
	$Re_h = \frac{\rho u_m d_h}{\mu} = 6058.5$ f=0.0093 $\frac{dp}{dx} = 28.49$	$Nu_{equal} = 0.021\,Re_h{}^{0.8}\,Pr^{0.4}$ h=26.31 $A_i h_i = 39.83$
	Re=31629 $\frac{dp}{dx} = 724.31$	$Nu_s = \frac{hd}{k_s} = 5.28$ h=353.86 $A_i h_i = 114.74$

TABLE III
THE EFFECT OF METAL FOAMS ON HEAT EXCHANGERS (RE=1800)

Foam Type	Pressure Drop	Heat Transfer Coefficient
	$f = \frac{16}{Re}$ $\frac{dp}{dx} = 0.0169$	$Nu = 4.36$ h=1.09 $A_i h_i = 0.342$
	$Re_h = \frac{\rho u_m d_h}{\mu} = 345$ $f = \frac{16}{Re_h}$ $\frac{dp}{dx} = 0.46$	$Nu_{equal} = 4.36$ h=5.67 $A_i h_i = 8.59$
	Re=1800 $\frac{dp}{dx} = 41.32$	$Nu_s = \frac{hd}{k_s} = 2.81$ h= 188.27 $A_i h_i = 59.11$

We have compared some of heat exchangers in tables 2 and 3. It shows that using metal foam can significantly enhance heat transfer in heat exchangers.

V. COMPARISON WITH EXPERIMENTAL DATA

The numerical results have been compared with the results presented in [2] in order to verify them. The main problem with this comparison is the difference between the geometries. Therefore it has been tried to compare the dimensionless numbers in order to eliminate of the difference between two geometries. In Fig. 9, the experimental results of V. V. Calmidi and R. L. Mahajan [2] have been compared with numerical results that have been obtained before.It must be considered that V. V. Calmidi and R. L. Mahajan[2] defined Reynolds number as $Re_k = u_m\sqrt{k}/\nu$ and in comparison with $Re_d = u_m d/\nu$, we should use the following relation:

$$Re_d = (d/\sqrt{k})\, Re_k \tag{29}$$

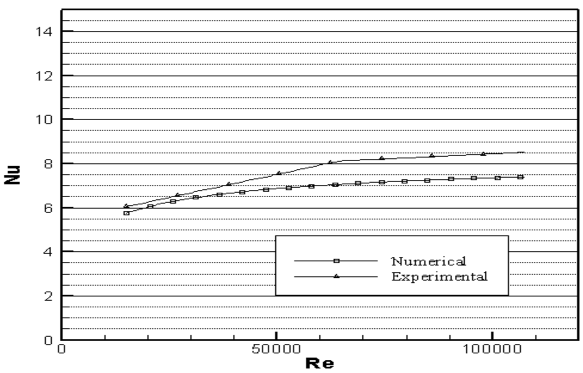


Fig. 6 Nusselt number for metal foam filled pipe: numerical results versus experimental data

VI. CONCLUSION

Temperature and velocity distributions have been obtained by solving the equations numerically with constant heat flux boundary condition. After inspecting the effects of different parameters on the heat transfer, the governing equations have been simplified with neglecting the parameters with lesser impacts. Simplified equations were solved analytically which had predicted the numerical results accurately in a gas flow. The results show that using metal foams can enhance heat transfer performance significantly.

ACKNOWLEDGEMENTS

The support of University of Tehran through the Project is gratefully acknowledged.

REFERENCES

[1] V.V. Calmidi, R.L. Mahajan, Forced convection in high porosity metal foams, Journal of Heat Transfer vol. 122, pp 557–565, 2000
[2] V.V. Calmidi, R.L. Mahajan, The effective thermal conductivity of high porosity fibrous metal foams, ASME. J. Heat. Transf.vol. 121, pp 466–471,1999
[3] K. Boomsma, D. Poulikakos, On the effective thermal conductivity of a three-dimensionally structured fluid-saturated metal foam, International Journal of Heat and Mass Transfer.vol 44, pp 827–836, 2001
[4] Shadi Mahjoob, Kambiz Vafai, A synthesis of fluid and thermal transport models for metal foam heat exchangers, International Journal of Heat and Mass Transfer.vol 51,pp 3701–3711,2008
[5] W. Lu, C.Y. Zhao, S.A. Tassou, Thermal analysis on metal-foam filled heat exchangers. Part I: Metal-foam filled pipes, International Journal of Heat and Mass Transfer.vol 49,pp 2751–2761,2006
[6] C.Y. Zhao, W. Lu, S.A. Tassou, Thermal analysis on metal-foam filled heat exchangers. Part II: Tube heat exchangers, International Journal of Heat and Mass Transfer.vol 49,pp 2762-2770, 2006
[7] L. Tadrist ,M. Miscevic, O. Rahli, F. Topin, About the use of fibrous materials in compact heat exchangers, Experimental thermal and fluid science .vol 28,pp 193–199,2004

[8] A. Bejan, Convection Heat Transfer, second ed., Wiley, New York, 1995.
[9] K. Vafai, C.L. Tien, Boundary and inertia effects on flow and heat transfer in porous media, International Journal of Heat and Mass Transfer,vol 24,pp 195–203.1981
[10] A. Bejan, Convection in Porous Media, Third ed., springer, New York, 2006
[11] W M Kays, M E Crawford, Bernhard Weigand, Convective Heat and Mass Transfer, Mc Graw-Hill, fourth ed, New York,2004
[12] L.Burden, J.Douglas Faires, Numerical analysis, Pws-Kent publishing company, fifth ed, Boston, pp 199-205,1993.