Experimental Study of Forced Convection Heat Transfer through Porous Media inside a Rectangular Duct of Fully Developed Region

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Abstract. This prospective study was designed to experimentally investigate the effects of inserting copper mesh in fully developed turbulent flow on the flow and heat transfer characteristics. Experimental work includes designing and manufacturing the test section of rectangular duct with dimensions of (30*3*40 cm) with aspect ratio and hydraulic diameter of 10 and 5.45 mm respectively. A constant heat flux ranged (1.5*10² – 1.8*10² W/m²) was applied to the lower surface of duct with Reynolds number range (3.3*10⁴ – 4.8*10⁴). The porosity range of (0.98 – 0.99). Also, the effects of porous height ratio (full and partial filling) were considered and considered air as working fluid. The results indicate that inserting mesh wire leads to enhance heat transfer coefficient and increase pressure drop. Taken together, these results show that the obtained values of performance enhancement criteria (PEC) ranged (1.01 -1.98), the enhancement in heat transfer coefficient is 315% due to wire mesh insert. Also, from obtained results empirical correlation for friction factor and Nusselt number were suggested.

Keywords: Forced convection, Porous media, wire mesh, friction coefficient

1. Introduction
In recent years considerable confirmation has been placed on the evaluation of heat transfer augmentation techniques. Heat transfer enhancement methods are divided in two categories: active and passive modes. In active method, heat transfer is improving by adding extra energy to the system. While, the passive mode can be acquired without any external energy [1]. The porous medium is one of the applications of the passive method. Important characteristic is represented by an extensive contact surface between solid and fluid surface. This lead to increase the internal heat exchange between the phases and consequently results in an increased thermal diffusivity. Using porous media (mesh wire) for heat transfer enhancement was found in many applications, e.g. heat exchangers, a solid matrix, enhanced recovery of petroleum reservoirs, electronic cooling and drying processes [2]. S.N.Sarada et al [3] experimentally and numerically investigated the enhancement of heat transfer in turbulent flow in horizontal tube by means of mesh inserts and it was compared with plane tube, the working fluid is the air with the porosity ranges of 99.73 to 99.98. The study carried out at constant heat flux and Reynolds ranged between 7000 to 14000. Through experimental investigation, the maximum increase in Nusselt number of 1.88 times was obtained for ratio of porous material $R_p$=0.8. While, CFD analysis showed that maximum Nusselt number can be obtained at $R_p$=0.9645. At these
conditions Nusselt number increased 2.15 times and pressure drop of 1.23 times compared to that of plane channel.

L. Varshney et al [4] concluded that geometrical parameters of wire mesh (wire diameter, pitch, porosity and number of layers) play a great role on the thermal efficiency of solar collector. The maximum thermal efficiency occurred at wire diameter 0.0585 mm, pitch 3.17 mm, number of layers 6 and porosity (0.945). Gazi I et al [5] studied flow and heat characteristics changing with the porous mesh screen having a sinusoidal shape normal to the flow direction inside a rectangular cross section air channel, Reynolds number range from 1360 to 3800. At 3100 < Re ≤ 3800, the fully developed $\text{Nu}/\text{Nu}_0$ is 2.2.5 and fully developed $f/f_0$ is 4.4. The study show that screen insert is only useful to increase the convection heat transfer in the channel over the range of transition Reynolds number tested.

Shokouhmand H, Emami S.M. [6] Experimentally investigated hydrodynamic and thermal behavior of air in rectangular duct fully filled with porous media with porosity of (0.95, 0.96, 0.98). Reynolds number changed from 500 to 2000 and a constant heat flux were applied to all walls of the duct. Commercial steel screens were used to manufacture the porous media. The results show that higher heat transfer rates are achieved when using porous inserts at the expense of a reasonable pressure drop. K.H. Hilal et al [7] studied experimentally the effect of using metallic wire mesh with porosity of 0.97 and 0.99 on forced convection heat transfer of air in rectangular duct. The experiments were carried out at Reynolds number (7682, 12497 and 17323) and at constant heat flux (192, 297 and 422 W/m$^2$). The results show that Nusselt number was increased with Reynolds number and heat flux but decreased with increasing pad porosity. The maximum increasing achieved in Nusselt number was (144%) at (porosity= 0.97) compared with plain duct. This enhancement of heat transfer was associated with pad weight addition. The ratio of increasing Nusselt number per 1 kg of pad is 47.

Jeng et al [8] studied experimentally the convection heat transfer in the heated rectangular duct filled with porous media. Air was used as the working fluid. Porous media was made of Brass beads with diameters 2, 4 and 6 mm. The duct width was constant at 60 mm, while Reynolds No. ranged (755-7921). The results show that the heat transfers in a packed duct depending on the bead diameter, rather than the hydraulic diameter. Kurtbas and Celik [9] studied the mixed convection heat transfer through a horizontal rectangular duct packed with metal foams. A constant heat flux was subjected on the bottom and top sides. Three aspect ratios were tested. New experimental correlations had been constructed to link the Nusselt number. The maximum increase in Nusselt number approximately is doubled with fully packed duct.

Although there is a large volume of published studies which describing the role of porous media on heat transfer enhancement, but there is relatively small body of literatures, that are concerned with effect of using mesh wire as porous media on heat and flow characteristics especially in fully developed region of turbulent flow. In the present experimental study, the hydrodynamic and thermal behavior of air flow are investigated in fully developed region of turbulent flow through rectangular duct fully and partially filled with porous media (mesh wire).

2. Experimental apparatus and procedure:
The experimental apparatus used in this study is shown photography and schematically in figure 1 and figure 2 respectively.
The description of experimental rig was explained in detail in previous published study [2]. All the measurement devices and experimental procedures are identical with [1]. The test section with cross section area (300*30 mm) of length 800 mm was positioned at (2000 mm) from duct inlet (in the fully developed region). This test section was divided into two parts. Each one is 400 mm length. The first of one operated as plain section (without porous media). While the second part is initiated with length 400 mm after the first one, in which porous media (wire mesh) inserted. The wire mesh was manufactured from commercial copper screen with a specification shown in table 1.

Figure 1. The experimental set up.

Figure 2. Schematic of experimental set up.
Table 1. Specification of wire mesh.

| Distance (mm) | Porosity | Density (kg m⁻³) | width (mm) |
|--------------|----------|-----------------|------------|
| 10           | 0.98     | 8964            | 300        |
| 15           | 0.985    |                 |            |
| 20           | 0.99     |                 |            |

3. Data reduction and operating procedure:
Air at ambient temperature was forced through the duct. The air velocity was varied by adjusting a gate at the exit of blower. After the required velocity was attained, the electric heater was switch on in order to choose the heat flux required. Once the study state condition was happened, all hydrodynamic and thermal measurements were recorded. The required parameters are calculated as following [10],
1. The mass flow rate for air,
   \[ \dot{m}_a = \rho_a \cdot u_a \cdot A_c \]  
2. The Reynolds number,
   \[ Re = \frac{D_h \cdot u_b}{v} \]  
3. The hydraulic diameter
   \[ D_h = \frac{4A_c}{P_w} = \frac{4(H \times W)}{2(H+W)} \]  
4. for plain surface the wall shear stress,
   \[ \tau_w = -\frac{H^2}{2} \cdot \frac{dp}{dx} \]  
5. The wall shear stress for porous surface utilizing Preston tube was shown in detail in previous published study [11]  
6. The friction coefficient (skin friction) for clear and porous cases,
   \[ C_f = \frac{\tau_w}{\frac{1}{2} \rho_a \cdot u_a^2} \]  
7. The heat gained by working fluid,
   \[ Q_a = \dot{m}_a \cdot C_{pa} \cdot (T_{a_{out}} - T_{a_{in}}) \]  
8. The electric heat input,
   \[ Q_e = I \cdot V \]  
9. The difference between air heat gained and electric heat input lays within 2% a according to estimated radiation and conduction losses through insulation materials. So, the heat obtained from eq.(6) and eq.(7) is nearly equal.
10. The local heat transfer coefficient,
    \[ h = \frac{q_{con}}{A_h \cdot (T_w - T_b)} \]  
11. The Nusselt number,
    \[ Nu = \frac{h \cdot D_h}{k_e} \]  
12. The Effective conductivity of a fluid filled porous media [7],
    \[ k_e = k_A \cdot k_p (1 - \varepsilon) \]  
13. The Performance evaluation criteria (PEC),
    \[ PEC = \frac{Nu_p/Nu_s}{(C_{fp}/C_{fs})^{1/3}} \]  
14. The uncertainty analysis was performed following the method suggested by [12] using the following expression:
\[ W_r = \left( \frac{\partial R}{\partial x_1} w_1 \right)^2 + \left( \frac{\partial R}{\partial x_2} w_2 \right)^2 + \ldots + \left( \frac{\partial R}{\partial x_n} w_n \right)^2 \]  

(12)

Where \( W_r \) denotes the uncertainty, and \( w_1, w_2, \ldots, w_n \) represent the uncertainty in the independent variables. The uncertainty in friction coefficient and local heat transfer coefficient was found to be 0.45% and 6.9% respectively.

In order to verify the results obtained in this study a comparison of the friction factor of a plane surface is performed with the modified Blasius equation (13), giving a maximum deviation 7.7-8.5%.

**Figure 3.** Comparison of experimental and estimated values of friction factor for smooth duct [13].

Also, the Nusselt number for smooth rectangular duct given by the Dittus –Boleter equation (13) is compared with the experimental results, the maximum deviation has been found to 18 -21% as shown in figure 4.

**Figure 4.** Comparison of experimental and estimated values of Nusselt number for smooth duct [9].
4. Results and discussion:
The obtained results were covered clear surface and all other surfaces manipulated with wire mesh (porous media). The effect of Reynolds number, porosity, ratio of porous media ($R_p$), and heat flux and on heat and flow characteristics in the fully developed region were experimentally investigated. The obtained results explain as follow:

4.1 Flow behaviors for smooth and modified surfaces:
Figure 5 show the variation of local skin friction for porous and smooth cases. For clear (smooth) case, skin friction coefficient as shown is nearly constant, while for porous case it increases in the stream wise direction. In this region with wire mesh, the flow is not yet periodically fully developed. This may attributed to the occurrence of flow separation behind the mesh and the subsequent boundary layer growth. Inspection of figure 5 reveals that decreasing porosity ($\varepsilon$) results in increasing the locale skin friction. The maximum value is 7.3 times that for smooth surface.

![Figure 5. Variation of local skin friction coefficient for smooth and porous surfaces in fully develop region at $R_p=1$ with Reynolds number 4.8x10^4.](image)

Figure 6 illustrates the effect of variation the porous media ratio ($R_p$) on skin friction coefficient, while keeping constant Re and $\varepsilon$. Increasing ($R_p$) tended to increase skin friction coefficient. This associated with increase surface area when $R_p$ increases and leads to increase pressure drop. These results agree with [14].

![Figure 6. Variation of local skin friction coefficient for smooth and porous media in fully develop region at $\varepsilon=98\%$ with Reynolds number 4.8x10^4.](image)
4.2 Heat transfer behaviors for smooth and modified surfaces:

Figure 7 depicts the changing of the bulk air temperature in the stream wise direction for clear and porous surfaces. The bulk temperature of air for porous surface is higher than that for clear surface. This can attribute to the contribution of porous media to generate more turbulence as the destruction of boundary layer and its rebuilding beyond the mesh. In addition, decreasing porosity (ε) caused greater increasing in the bulk air temperature; this is according to the same above reason as well as decreasing (ε) associated with increasing surface area of the wire mesh.

![Figure 7](image7.png)

**Figure 7.** Distribution of air temperature in stream wise direction for porous media in fully develop region at q=1833 W/m² and Reynolds number 4.8x10⁴.

Figure 8 depicts, the variation of the values of stream wise air bulk temperature with the ratio of porous media (Rp). It is clear that the higher bulk temperature obtained when using full filling porous media (Rp=1). This can be enough more hindering rise to air flow driving to more turbulence intensity. This is similar to that obtained by [15].

![Figure 8](image8.png)

**Figure 8.** Distribution of air temperature in stream wise direction for smooth and porous media in fully develop region at ε=98%, Re= 4.8x10⁴ and q=1833 W/m².
Figure 9 illustrates the variation of the bulk air temperature in the stream wise direction with the heat flux, the level of increasing bulk air temperature is sensitive to heat flux when keeping ($\varepsilon$) and (Rp) constants.

![Figure 9](image)

**Figure 9.** Distribution of air temperature in stream wise direction for smooth and porous media in fully develop at Rp=1, Reynolds number $4.8\times10^4$ and $\varepsilon=98\%$.

Figure 10 shows the variation of locale heat transfer coefficient with stream wise direction. As well known, the heat transfer coefficient at fully developed region at smooth surface (clear case) is nearly constant. While, inserting wire mesh (as porous media) leads to reconfiguration the turbulent boundary layer. Therefore, the heat transfer coefficient is still increasing in the stream wise direction. The higher increasing was attained at lower porosity ($\varepsilon$). The maximum increasing is nearly 315% when $\varepsilon=0.98$ and at $x/b = 158$.

![Figure 10](image)

**Figure 10.** Distribution of local heat transfer coefficient for smooth and porous media at Rp=1, Reynolds number $4.8\times10^4$ and $q=1833\;\text{W/m}^2$.

Figure 11 depicts the effect of the porous media ratio (Rp) on heat transfer coefficient. From the figure it is evident that the higher heat transfer coefficient appeared at (Rp=1). This attributed to the increasing of surface area, thermal conductivity of working fluid and turbulence intensity.
Figure 11. Distribution of local heat transfer coefficient for smooth and porous media at Reynolds number $4.8 \times 10^4$, $\varepsilon = 98\%$ and $q = 1833 \text{ W/m}^2$.

Figure 12 shows the variation of the heat transfer coefficient with the heat flux. It is clear that the level of heat transfer augmentation is sensitive to heat flux for porous surface.

Figure 12. Distribution of local heat transfer coefficient for smooth and porous media at $R_p = 1$, Reynolds number $4.8 \times 10^4$, and $\varepsilon = 98\%$.

Figure 13 depicts the effect of variation Reynolds number on the heat transfer coefficient. It is evident that the greater effect appeared at higher Reynolds number.
Figure 13. Distribution of local heat transfer coefficient for smooth and porous media at $q=1833 \text{ W/m}^2, \varepsilon=98\%$ and $R_p=1$.

Figure 14 and figure 15 show the variation of locale Nusselt number with Reynolds number at $R_p=1$ and 0.8, $\varepsilon=0.98$ and $q=1833 \text{ W/m}^2$. From these figures, it is clear that Nusselt number followed the same behavior of heat transfer coefficient. It revealed nearly constant for smooth surface in fully developed region, while for porous media it was increased in stream wise direction and give the higher value at large Reynolds number. While, for porous media it was increased in stream wise direction and the higher value obtained at higher value of Reynolds number and $R_p$ at the same heat flux and porosity.

Figure 14. Distribution of local Nusselt number for porous and smooth in fully develop region at different Reynolds number for ($R_p=1, \varepsilon=98\%$ and $q=1833 \text{ W/m}^2$).
Figure 15. Distribution of local Nusselt number for porous and clear in fully develop region at different Reynolds numbers for \((R_p=0.8, \varepsilon=98\% \text{ and } q=1833 \text{ W/m}^2)\).

All the above results for thermal characteristics can be attributed to the effect of porous media to create a complicated flow field containing recirculation, separation and reconfigure to boundary layer. As well as, porous media caused increasing surface area, thermal conductivity of working fluid and support temperature uniformity in the core flow when \((R_p<1)\) [11], part of these results was found by [14].

4.3 Performance evaluation criteria (PEC):
Figure 16 illustrates the variation of (PEC) with Reynolds number. It is clear that the increasing Reynolds number leads to decrease (PEC), and it is generally greater than unity. From this figure it is also shown that the higher value of (PEC) attained at the partial filling \((R_p=0.8)\) for all values of Reynolds number.

Figure 16. Distribution of thermal performance with different Reynolds number at \(\varepsilon=98\% \text{ and } q=1833 \text{ W/m}^2\).

The variation of (PEC) with the porosity \((\varepsilon)\) depicts in figure 17. It is clear that the higher value of (PEC) is obtained by lower porosity at specified values of \(q\) and \(R_p\) without consideration of Reynolds number.
4.4 Correlation relationships of the present work:
A statistical regression analysis was used to get relationships for all considered parameters according to [15, 16].

- Correlation for Nusselt number
  \[
  \text{Nu} = 12.81 \left(\text{Re}\right)^{0.25} \cdot \varepsilon^{30.25} \cdot R_p^{0.31}
  \]  

A comparison between experimental values of Nu and those predicted from above equations is illustrated in figure 18. The mean deviation is about ± 5.5%.

- Correlation for skin friction coefficient,
  \[
  C_f = 0.184 \left(\text{Re}\right)^{0.18} \cdot \varepsilon^{30.1} \cdot R_p^{1.6}
  \]  

A comparison between experimental value of skin friction coefficient and those predicted from above equations is illustrated in figure 19. The mean deviation is about ± 2.2%.
4.5 Comparison with previous work

The Comparison of the finding average Nusselt number and (PEC) with other studies [7, 14] was shown in figures 20 and 21. Previous studies have demonstrated the same behavior of the present results. The sensitive difference of the obtained results with previous work can be attributed to the absence of experimental parameters matching such as Re, Rp and \( \varepsilon \).

**Figure 19.** Variation of Predicted friction factor with measured friction factor.

**Figure 20.** Average Nusselt number as a function of Reynolds number for previous work.
5. Conclusions

1. Skin friction coefficient increases as a result of interesting wire mesh in the fully developed region. At $R_p=1$ and lower $\varepsilon$ the maximum increasing occurred, and it is about 7 times of the clear cases.

2. The augmenting of heat transfer coefficient due to wire mesh insert is about 315% compared with clear case.

3. Increasing Reynolds number leads to decrease the performance evaluation criteria and it is always larger than unity irrespective to $Re$, $R_p$, $\varepsilon$ and $q$.

4. The maximum values of (PEC) occurred at $R_p=0.8$ (partial filling) and it is about 1.98 depending on $Re$, $q$ and $\varepsilon$.

Nomenclature:

| Symbol | Description                  | Units   |
|--------|------------------------------|---------|
| $A_c$  | Cross-sectional area         | m$^2$   |
| $A_h$  | Heater area                  | m$^2$   |
| $b$    | Half duct height ($H/2$)     | m       |
| $C_p$  | Specific heat                | J/kg. K |
| $C_f$  | skin friction coefficient    | -       |
| $D_h$  | Hydraulic diameter           | m       |
| $H$    | Duct height                  | m       |
| $h$    | heat transfer coefficient    | W/ m$^2$.K |
| $I$    | Current                      | Amp.    |
| $k$    | Thermal conductivity         | W/m. K  |
| L      | Length                       | m       |
| Symbol | Description                                      | Unit       |
|--------|--------------------------------------------------|------------|
| m      | Mass flow rate                                    | kg/s       |
| P      | Pressure                                          | N/m²       |
| Q      | Rate of heat                                      | W          |
| Rp     | Porous radius ratio                               | -          |
| Te     | Wall temperature                                  | K          |
| Ta     | Air Temperature                                   | K          |
| u      | velocity                                          | m/s        |
| V      | Voltage                                           | Volt       |
| W      | Duct width                                        | m          |
| ε      | Porosity                                          | -          |
| τ      | Averaged Viscous Shear Stress                     | N/m²       |
| μ      | Dynamic viscosity                                 | Pa·s        |
| ν      | Kinematic Viscosity                               | m²/s       |
| ρ      | Density                                           | kg/m³      |
| Nuₐ₀   | Nusselt Number for smooth duct                    | -          |

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