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A Study Of Concentric Tube Heat Exchanger With Different Porous Particles Using Wilson Plot Analysis

K. Senthilkumar[1], P. Palanisamy[2]

[1]Department of Mechanical Engineering, Adhiyamaan College of Engineering, Hosur-635109, Tamil Nadu, India
[2] Annai Mathammal Sheela Engineering College, Erumapatty, Namakkal- 637 013, Tamil Nadu, India

Abstract: This paper deals with an experimental investigation of the tube side heat transfer co-efficient in concentric tube heat exchanger with inner tube,arrangement as inserting of different porous material on the outer surface of the inner tube wall of concentric tube heat exchanger using Wilson plot analysis. The exhaust gas of twin cylinder diesel engine is used as coolant and water as heating fluid. The flow circulation develops turbulent motion which results in increase the heat transfer co-efficient values. The experimental tests are conducted on developed concentric tube heat exchanger with four different porous particles selected as cast iron, mild steel, copper and aluminum. The heat transfer co-efficient is analyzed and verified by Wilson plot method and Nusselt's number correlations are developed. The results with Wilson plot method and the experimental results are found with good agreements.
Keywords: Wilson plot method, Enhanced porous medium, heat exchanger, turbulent flow.

Introduction

The heat exchangers are devices frequently used in process industries to transfer the heat between two or more fluids of different temperatures. Thermal performance of heat exchanger could improve by adopting the heat transfer enhancement techniques. Heat transfer enhancement has significant results on energy conservation and environmental damage reduction. Most of the heat transfer processes in industries belong to turbulent convective heat transfer. Internal inserts in tubes are found economical for turbulent convective heat transfer enhancement; therefore, internally inserted tubes are currently taking part in a very important role in commercial applications, for instance, heat recovery processes, air-con and refrigeration systems, chemical reactors, and food and farm processes. Many research works were carried out this area and summarized in this section. Tarawneh et al [1] had studied Experimental investigation of two-phase laminar forced convection in a single porous tube heat exchanger and predicted the combined effect of the Reynolds number, Darcynumber, porosity, and Prandtl number on the heat transfer and pressure drop of carbondioxide during the condensation process in a porous medium. D. Srinivasacharya and O. Surender[2] investigated the influence of Dufour and Soret effects on mixed convection heat and mass transfer over a vertical plate in a doubly stratified fluid-saturated porous medium. The plate is maintained at a uniform and constant wall heat and mass fluxes. The Darcy–Forchheimer model is employed to describe the flow in porous medium. Smith Eiamsa-ard *et.al* [3] studied Experimentally investigated the turbulent flow heat transfer and friction characteristic in a concentric pipe heat exchanger with louvered strip inserts.Shokouhmand *et al* [4] had performance studied of shell and helically coiled tube heat exchangers. Three heat exchangers with different coil pitches and curvature ratios were tested for both parallel-flow and counter-flow. Nebbali and Bouhadef [5] numerically studied the Non-

Newtonian fluid flow in plane channels and enhanced the porous blocks. The heat transfer can be improved without increase the pressure drop by inserts the porous because of higher thermal conductivity. Nimvari *et al*[6] numerically studied the turbulent and heat transfer in a channel partially filled with a porous media, proposed optimum porous layer thickness is 0.55, 0.60 and .65 respectively, achieved a higher Nusselt Number and lower pressure drop. Lopez Penha *et al* [7] Numerical studied the heat transfer coefficient of fully developed conjugated heat transfer in porous media with uniform heated solid. Sepideh Esmaeili Rad et.al [8] A numerically studied the heat transfer enhancement in a shell-and-tube heat exchanger using a porous medium inside its shell and tubes, separately. A three-dimensional geometry with k-ε turbulent model was used to predict the heat transfer and pressure drop characteristics of the flow. The effects of porosity and dimensions of these media on the heat exchanger's thermal performance and pressure drop were analyzed. Wamei Lin et.al[9] had been studied performance and CFD analysis of the graphite foam and the aluminum heat exchanger under countercurrent flow condition, the comparative study was conducted for the thermal performance (heat transfer coefficient) and the pressure loss. Yang and Nakayama[10] had been investigated analytically synthesis of porosity and dispersion in effective thermal conductivity of porous media and expressed for the effective stagnant thermal conductivity had been derived using a unit cell model, which consists of rectangular solids with connecting arms in an in-line arrangement. The validity of the expression for the stagnant thermal conductivity had been confirmed, compared the present results with available experimental and theoretical data for packed beds, porous foams and wire screens. The resulting expressions for the longitudinal and transverse thermal dispersion conductivities agree well with available experimental data and empirical correlations. Bogdan and Abdulmajeed [11] had been investigated numerical and experiment compared with and without metallic porous materials, inserted in a pipe, on the rate of heat transfer, the pipe is subjected to a constant and uniform heat flux. The effects of porosity, porous material diameter and thermal conductivity as well as Reynolds number on the heat transfer rate and pressure drop, constant diameter of the porous medium improvement can be attained by using a porous insert with a smaller porosity and higher thermal conductivity. Rp and Ɛ had a positive influence upon heat transfer and a negative impact on pressure drop Li ,H.Y *et al* [12] had been numerically studied fluid flow and heat transfer characteristics in a channel with staggered porous blocks, the effects of the thermal conductivity ratio between the porous blocks and the fluid on the local heat transfer were analyzed. The pressure drops across the channel for different cases were discussed. Increased the thermal conductivity ratio between the porous blocks and fluid, the heat transfer at the locations of the porous blocks can be greatly increased. Xuelei Chen and William[13] had been numerically studied the Enhancement of heat transfer with Combined convection and radiation in the entrance region of circular ducts with porous insert, compared with and without porous inserts Increasing Darcy number of the porous insert will also increase this enhancement effect.. The objective of this study is to investigate the tube side heat transfer co-efficient for four different porous particles inserts inner tube of the outer surface of the heat exchanger using Wilson plot analysis and compare with experimental results also to estimate R^2 value with Wilson plot method results.

Development of Inner Tube Geometry

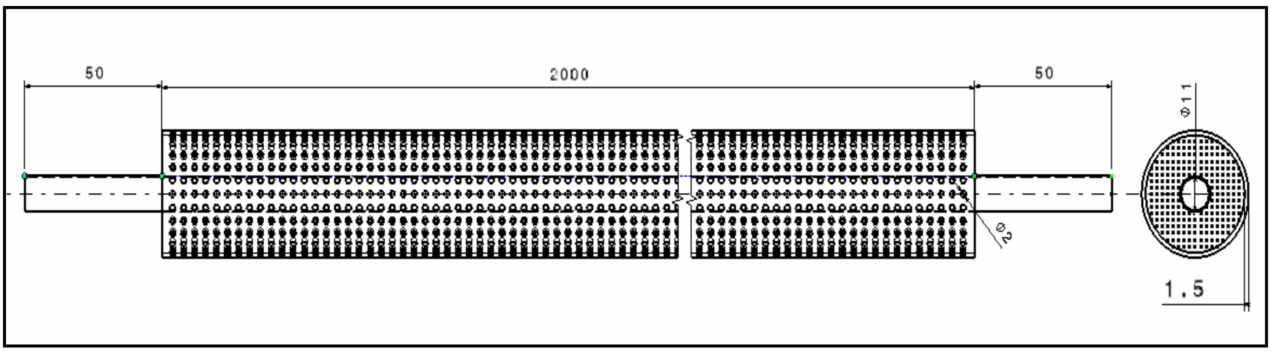

Figure 1: Geometric view of inner porous tube

The schematic diagram of porous structures used in this experiments is shown in fig. 1 and are prepared from commercially available 1.5mm thick perforated aluminum sheets having stamped holes 2mm diameter. Perforated sheet having dimensions is 2000mm x 116mm x 1.5mm. Inner tube outer diameter is 11mm and length of the tube is 2100mm. Aluminum sheet made up of cylindrical shape and four different porous particles are packed in the inner tube of outer surface is covered with aluminum sheet. Each porous medium made up of one tube geometry.

Experimental Arrangement

The schematic diagram of experimental set-up is shown in fig.2. The set-up is a well instrumented heat exchanging system in which a cold water stream flowing inside the porous tube is heated by a hot flue gas flowing in the annular side. The main parts of the cycle are heat exchanger (1), pump (2), storage tank (3), and hot flue gas from engine exhaust (4). The heat exchanger is made up of galvanized iron and porous tubes, annular aspect fluid as exhaust gas and tube aspect fluid as water. The total surface area of porous packed inner tube is $0.2324m^2$ whereas plain tube surface area is $0.0691m^2$. Table.1 shows the specifications of concentric tube heat exchanger. Data logger is used to record the temperatures at various locations, model CT708U; resolution is 1°*C* for thermocouples, and accuracy is ±1°*C*±1 least significant digital for thermocouples (type *K*). Thermocouples are placed at the inlet and outlet of the heat exchanger. A pump maintains the circulation of water in this setup. The mass flow rate of the fluid is measured by Rota meter.

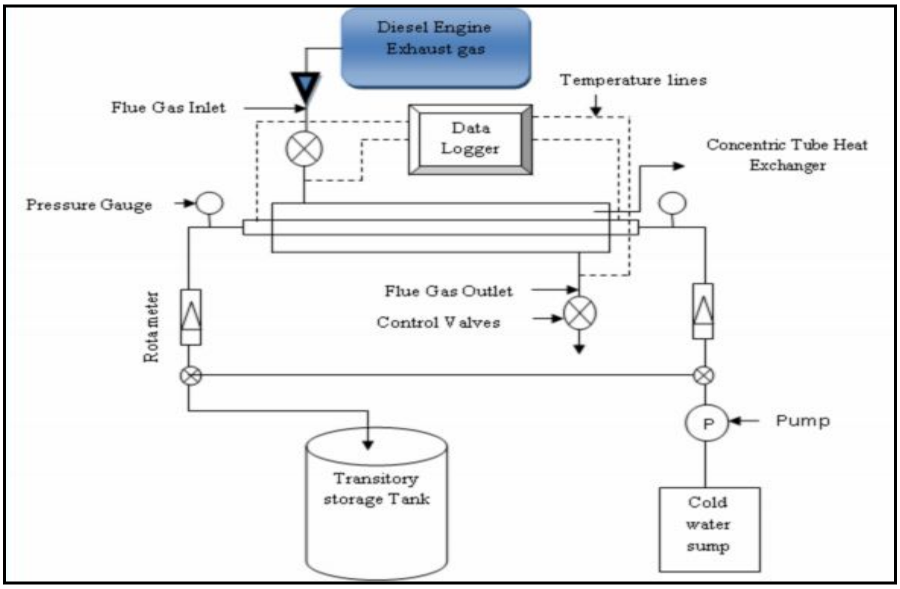

Figure 2: Schematic diagram of porous tube heat exchanger

Table 1: Specification of the porous tube heat exchanger

S. No	Name of the Materials	Specifications Details
1	Annulus material	Galvanized Iron
2	Outer diameter	76.2 mm
3	Inner diameter	72.2 mm
4	Annulus thickness	4 mm
5	Thermal conductivity of galvanized iron	72.7 w/mk
6	Tube material	Copper
7	Inner diameter	10 mm
8	Outer diameter	11 mm
9	Outer Diameter of the porous cylinder	37 mm
10	Tube thickness	1 mm
11	Length of the tube	2000 mm
12	Thermal conductivity of copper	386 w/mk
13	Type of insulation used	Glass Wool

Table 2: Experimental setup parameters range

S. No	Name of the Parameters	Range Value
1	Tube side water flow rate	0.01388 -0.0277 kg/s
2	Annular side flue gas flow rate	0.02611- 0.02629 kg/s
3	Tube side inlet temperature	23.6 -27.8˚C
4	Tube side outlet temperature	30.5 – 80.5˚C
5	Annular side inlet temperature	65 - 400˚C
6	Annular side outlet temperature	50 - 120˚C

Experimental Procedure

Experiments are conducted by performing the Wilson plot test to evaluate the tube side heat transfer coefficient. This is done with annular flow rate held constant and the inner tube flow varied through a sort of flow rates. The flows on both sides are kept in the turbulent section while the total heat flux through the system is held constant. After adequate time is allowed for steady state condition to recognized, the inlet and outlet temperatures are recorded by means of the data logger. The information on the experimental ranges of parameters used for analysis purpose is given in the table 2.

Data Collection and Analysis

The overall heat transfer coefficient U_o is calculated from the temperature data and the flow rates using the following equations Eq.1:

$$U_o = \frac{Q}{A_o LMTD} \quad (1)$$

Where, Q is the average heat rate of the annulus and tube: Eq.2:

$$Q = \left(\frac{(\dot{m} C_p \Delta T)_i + (\dot{m} C_p \Delta T)_o}{2} \right) \quad (2)$$

Where A_o is the outside surface area of porous packed tube and LMTD is the log mean temperature difference, based on the inlet temperature deviation, ΔT_1 , and the outlet temperature deviation, ΔT_2, using the following equation:

$$LMTD = \frac{(\Delta T_1 - \Delta T_2)}{\ln(\Delta T_1 / \Delta T_2)} \quad (3)$$

$$\Delta T_1 = T_{i,in} - T_{o,out} \quad (4)$$

$$\Delta T_2 = T_{i,outn} - T_{o,in} \quad (5)$$

Where, $T_{i,in}$ and $T_{i,out}$ are inlet and outlet of water in the inner tube, and $T_{o,in}$ and $T_{o,out}$ denotes the inlet and outlet temperature of flue gas in the annulus. At first stage, the data are analyzed by the Wilson plot method and can be described as follows.

The experimental determined overall heat transfer resistance $1/UA$ of the test tube is related to individual heat transfer resistance, Eq.6:

$$\frac{1}{U_o A_o} = \frac{1}{h_o A_o} + R_{wall} + \frac{1}{h_i A_i} \quad (6)$$

Where, h_i and h_o represent the average outside and inside heat transfer coefficient and R_{wall} denotes wall resistance and given by $R_{wall} = \frac{(\ln(d_o/d_i)}{2\pi k_{eff} l}$. In the present calculation, the overall resistance is based on the outer surface area, which is evaluated as $\pi d_o L$, where d_o is the outer diameter of the tube. Note that the inside heat transfer coefficient is based on nominal inside surface area $\pi d_i L$. The properties for both streams are calculated using average of the inlet and outlet bulk fluid temperatures. The inner tube side heat transfer coefficient h_i is given by Eq.7:

$$h_i = C_i \frac{k_i}{d_i} (\mathrm{Re})_i^M (\mathrm{Pr})^{.333} \quad (7)$$

$$K_{eff} = \varepsilon_v k_f + (1 - \varepsilon_v) k_s \quad (8)$$

Where d_i, d_o are inner and outer diameters of the tube respectively; k is the thermal conductivity of the wall; and L is the length of the heat exchanger. After determine the overall heat transfer coefficients, the only unknown variables in Eq. (6) are the heat transfer coefficients. By keeping the constant mass flow rate in the annular side, the inner heat transfer coefficient is assumed to behave in the following manner with the fluid velocity in the tube side.

$$\left(\frac{1}{U_o A_o} - R_{wall}\right) = \frac{1}{C_i (k/d_i) \mathrm{Re}^M \mathrm{Pr}^{333}} + \frac{1}{h_o A_o} \quad (9)$$

Equation 9 has the linear form of:

$$Y = mX + b \quad (11)$$

$$Y = \frac{1}{U_o A_o} \quad (12)$$

$$m = \frac{1}{C_i} \quad (13)$$

$$X = 1*[A_i*(k/d_i)*(\mathrm{Re}^8)*(\mathrm{Pr}^{333})]^{-1} \quad (14)$$

$$b = \frac{1}{h_o A_o} + R_{wall} \quad (15)$$

Hence, a simple linear regression equation, the slope of the resulting straight line is equal to1/C_i. The Eq.9 has the linear form, if R_{wall} and h_o are constant and it is used to generate a straight line graph which describes the overall heat transfer process the concentric tube when the cold water temperature and mass flow rates changes but quantity of heat transfer is constant. The outcome of this, the internal heat transfer coefficients are balanced at different values so that while the overall heat transfer coefficient varies, the heat transfer coefficients and annular side heat transfer coefficients remain unchanged. Given such a test sequence, a line is plotted as shown in fig. 3.

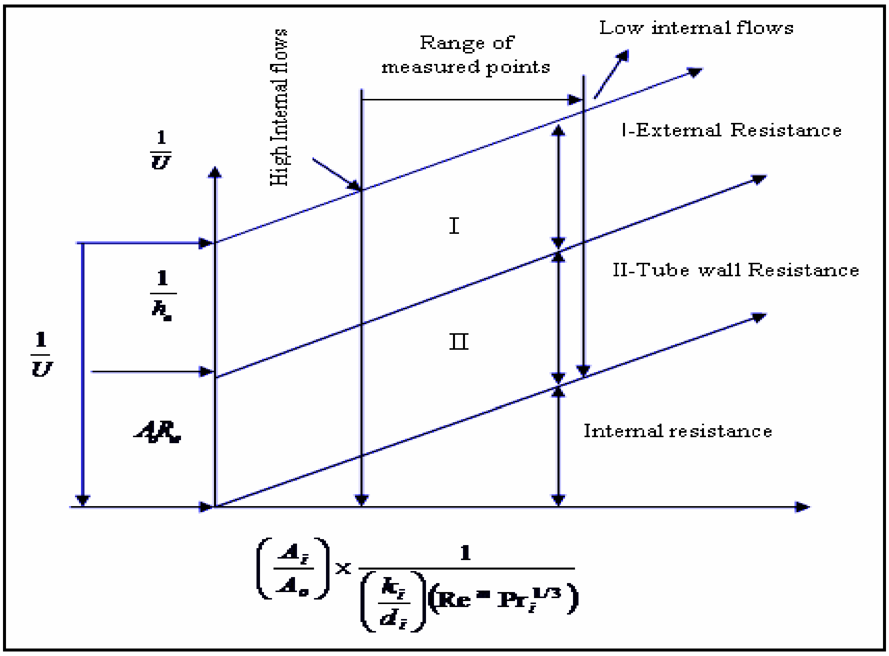

Figure 3: Wilson plot method – General Features (shah, 1990)

Results and Discussion

In order to validate the experimental investigation and testing methods using difference porous medium and porous particles inserts inner tube of the outer surface and with smooth inner surfaces. The Fig.4-7 shows the relationship of X and Y for the validation test for four different porous particles tube for various flow rates of hot flue gases. The linear regression result of the plain tube yields $C_i = 0.0228$ which rounded up very close to the well known Dittus- Boelter correlation constant of 0.023. The authors have chosen *Re* exponent of 0.8 for confirmation reason as such value is used briefly by previous studies, as in Shah[14]. Note that the exponent 0.8 is not mandatory a constant 0.8 as shown in Fig.5 referred from Tiruselvam Ramahlingam [15]. As addressed by shah [14], the *Re* exponent it is a function of the Prantl number and Reynolds number. It varies from 0.59 at *Pr* =2.92 to 4.24 at *Pr* = 100 for *Re* = 50,000 for circular tube. The author has adapted an approach where the *Re* exponent of the Nusselt correlation is plus 1 of the *Re* exponent of the fanning friction factor. The relevant data extracted from a test are given as follows:

Aluminum (porous medium) tube, turbulent, *Re* > 8500:

$$f = 0.0624 \ \mathrm{Re}^{-0.333} \quad (16)$$

Copper (porous medium) tube, turbulent, Re > 8500:

$$f = 0.0617 \ \mathrm{Re}^{-0.333} \quad (17)$$

Mild steel (porous medium) tube, turbulent, Re > 8500:

$$f = 0.0593 \ \mathrm{Re}^{-0.333} \quad (18)$$

Cast iorn (porous medium) tube, turbulent, Re > 8500:

$$f = 0.0599 \ \mathrm{Re}^{-0.333} \quad (19)$$

Based on the *Re* exponents from the fanning friction factor correlations, the Wilson plot test was conducted and the data are analyzed for the four different porous particles (aluminum, copper, mild steel and cast iron). The corresponding Wilson plots are shown in Fig.4-7 for four different porous particles tube respectively. R^2 values are having good agreement for all the tests. Corresponding to the turbulent flow region of the individual test section, the side Nusselt numbers for heat transfer is written as:

Aluminum (porous medium) tube, turbulent, *Re* > 8500:

$$Nu = 0.1402 \ \mathrm{Re}^{0.467} \ \mathrm{Pr}^{-0.333} \quad (20)$$

Copper (porous medium) tube, turbulent, *Re* > 8500:

$$Nu = 0.1391\,\mathrm{Re}^{0.467}\,\mathrm{Pr}^{-0.333} \quad (21)$$

Mild steel(porous medium) tube, turbulent, $Re > 8500$:

$$Nu = 0.1335\,\mathrm{Re}^{0.467}\,\mathrm{Pr}^{-0.333} \quad (22)$$

Cast iron (porous medium) tube, turbulent, $Re > 8500$:

$$Nu = 0.1348\,\mathrm{Re}^{0.467}\,\mathrm{Pr}^{-0.333} \quad (23)$$

The Fig. 4- 7 shows the comparative results of Wilson plot analysis for porous tube. The regression analysis shows the good agreement with Wilson plot analysis value.

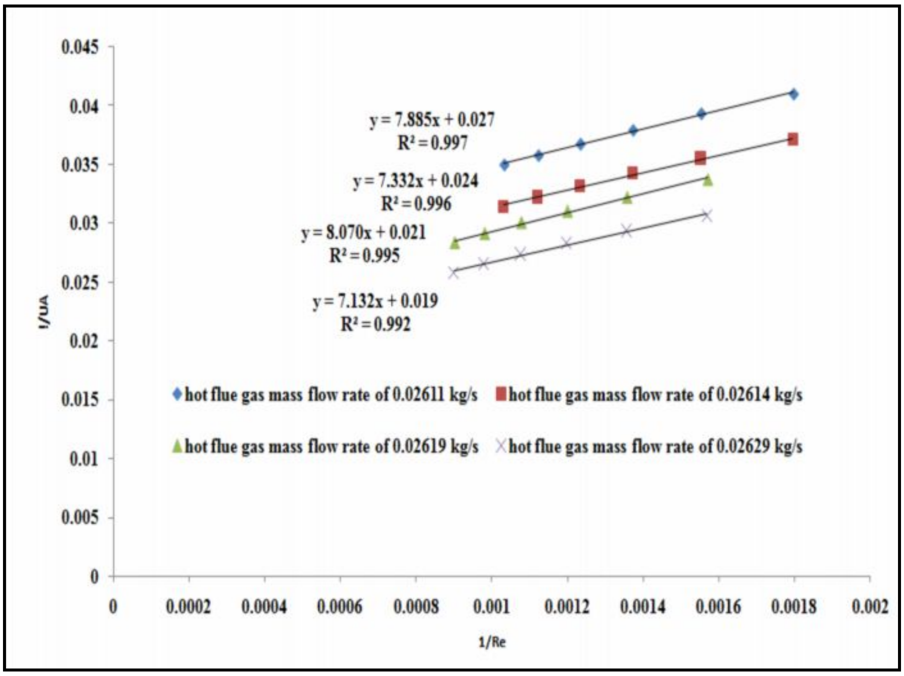

Figure 4: Wilson plot analysis of Aluminum porous tube

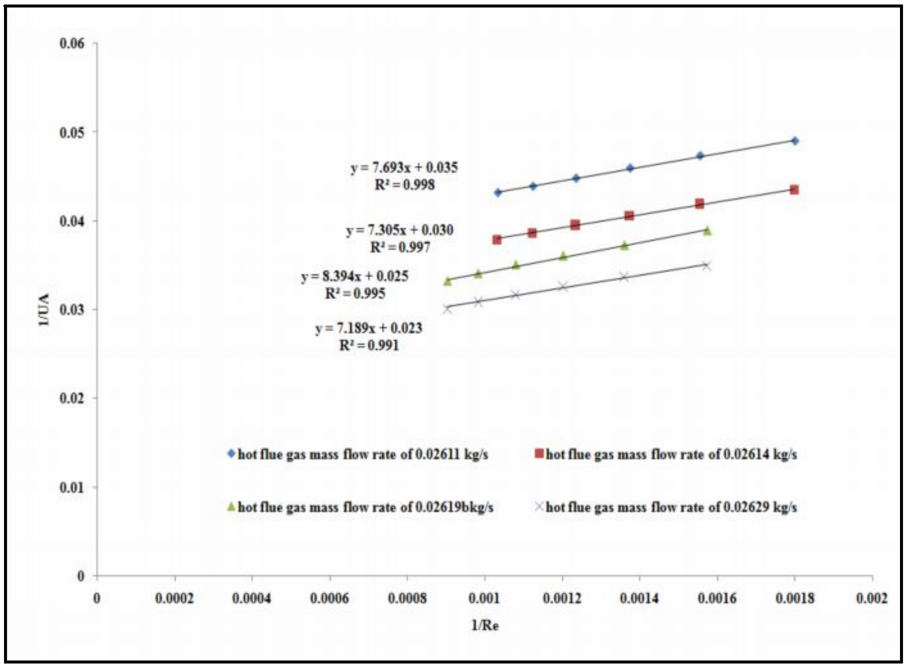

Figure 5: Wilson plot analysis of Copper porous tube

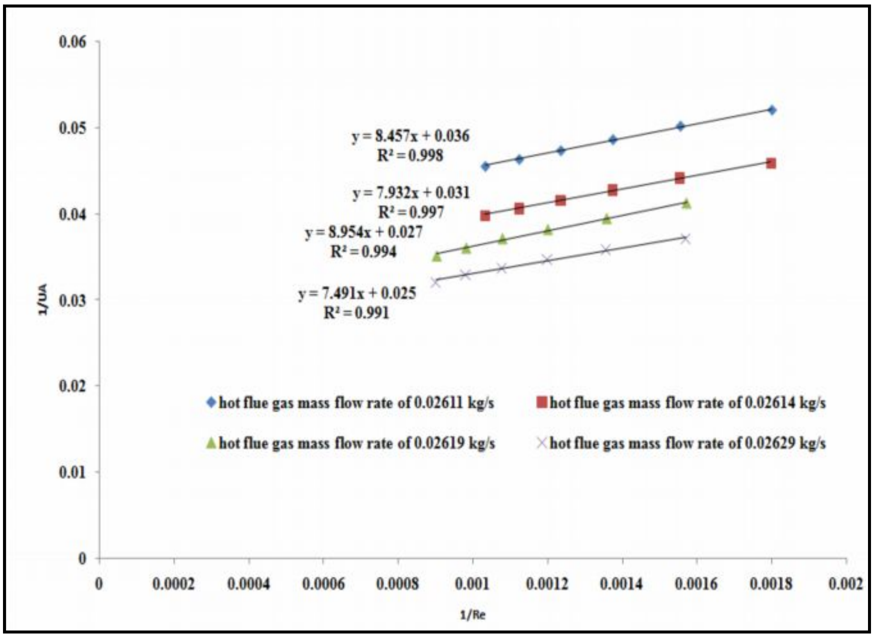

Figure 6: Wilson plot analysis of Mild Steel porous tube

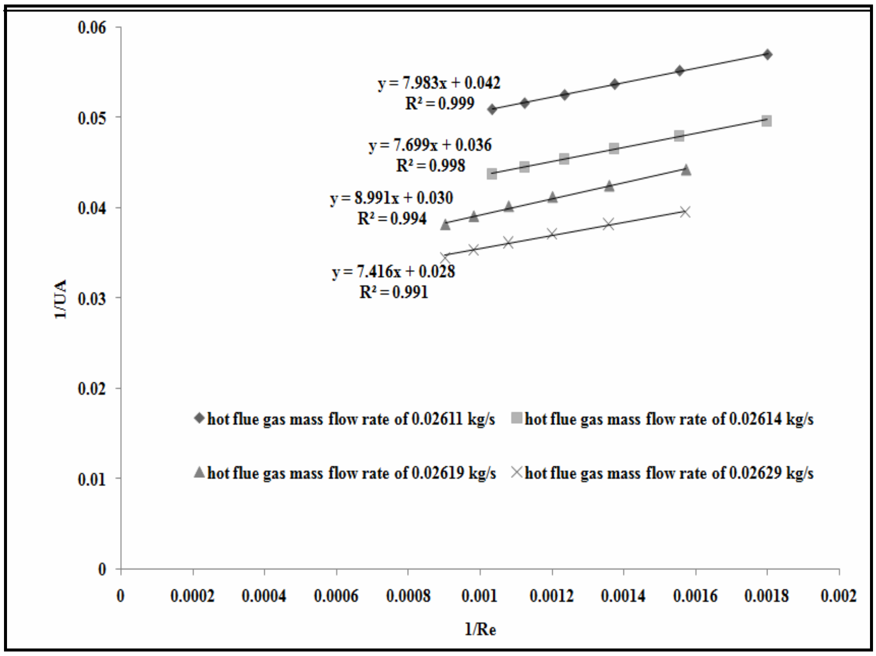

Figure 7: Wilson plot analysis of cast iron porous tube

Conclusions

Convective heat transfer and pressure drop characteristic for four different porous medium types of enhanced concentric tubes were reported in the present investigation. Experiments were conducted in a concentric tube heat exchanger with hot flue gas of twin cylinder diesel engine exhaust as test fluid in annulus and water as the tube side fluid. The four annulus namely (aluminum, copper, mild steel and cast iron) porous particles enhanced tube was investigated. The heat transfer coefficients of the inner tube side of the concentric tube test section were obtained using the standard Wilson plot technique. The Wilson plot test was conducted for turbulent flow region. An initial test was conducted for four different porous particles annulus to validate the testing method and procedure using *Re* exponent value of 0.8. After attaining the experimental validation, the test was conducted on the four different porous particles annulus using *Re* exponent from the fanning friction characteristic correlation. The Nusselt inside heat transfer correlations, Eq. (20-23), obtained in this investigation for the tube side will be used for future study.

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Nomenclature

A_i - Inside heat transfer area of the tube (m^2)
A_o-Outside heat transfer area of the tube (m^2)
b -Intercept on leader with ordinate (K/W)
C_i-Constant for inside heat transfer correlation
C_p-Specific heat of water and flue gas (J/kgk)
d_i -Inside diameter of the tube (m)
d_o- Outside diameter of the tube (m)
h_o-Heat transfer coefficient on the on the annulus side (w/m^2k)
h_i- Heat transfer coefficient on the on the inside tube (w/m^2k)
k- Thermal conductivity of water (W/m^2k)
k_{eff}- Effective thermal conductivity of porous medium (w/m^2k)
k_s- Thermal conductivity of solid (wm^2k)
k_f- Thermal conductivity of fluid (w/m^2k)
L- Length of the tube (m)
LMTD -Logarithm Mean Temperature Difference (k)
m- Slope of the least-square deviation line, dimensionless
$\dot{m}$ - Mass flow rate of coolant water (kg/s)
M- Reynolds number exponent, dimensionless
Nu- Nusselt number (hidi/k),
Dimensionless
Pr- Prandtl number
Re- Reynolds number
R_{wall}- Wall Resistance (k/w)
ΔT - Temperature rise on the cold water (k)
V_v - Void Fraction
X- Wilson plot function (k/w)
Y- Wilson plot function (k/w)
Greek symbols
μ- Dynamic viscosity of water (Ns/m^2)
ρ -Density of water (kg/m^3)

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