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Enhancement of Heat Transfer in Shell and Tube Heat Exchanger using Different Porous Medium: A CFD-based Study

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***Abstract*—*The present study is to investigate the heat transfer enhancement in a cylindrical heat exchanger using porous media. The heat exchanger is modelled by a cylindrical cavity (Shell) with inlet and outlet thermally insulated ports and five tubes which contain hot water and cold water flows in shell. The effect of porosity on heat transfer enhancement is studied at different mass flow rate 0.15, 0.2, 0.25 and 0.30 Kg/sec. The study about effect of porosity on heat transfer enhancement is done by both experimentally and CFD based and the results are compared with simple heat exchanger. In present study, two different types of porous materials are used and Porosity is taken as 80%. The effect of varying mass flow rate on outlet temperature, heat transfer coefficient, Reynolds number and Nusselt number has been investigated.*

Keywords*—*Porous medium, Heat exchanger, CFD, porosity, mass flow rate

I. INTRODUCTION

The heat exchanger applications in industrial and engineering purposes are quite popular. In today scenario, energy providing devices such as heat exchangers have become more and more efficient. The efficiency can be improved using the twisted tape, coiled wire, helical screw-tape insert and porous medium.**Zhang-Jing Zheng et al. [1]** explained the numerical investigation of the heat transfer improvement for convection heat transfer of turbulent fluid flow in a central receiver tube filled with the porous medium under non-uniform circumferential heat flux was carried out. The results are shown that the enhanced receiver tube (ERT) with down-filling porous medium inserts and in-filling porous medium inserts have good thermal enhancement when the ratio of thermal conductivity (K) of the porous medium to working fluid medium (λ_s/λ_f) is less than 1,000. **Ya-Ling He et al. [2]** investigated the numerical investigation on augmentation for convection heat transfer of fluid flow in a solar central receiver tube filled with porous medium and non-uniform circumferential heat flux was carried out. Enhanced receiver tubes (ERTs) of four kinds with different porous insert designs were modelled to optimize the performance of ERT.**N. Targui and H. Kahalerras [3]** numerically studied a double pipe heat exchanger with porous structures inserted in the annular gap is made in two configurations: (A) on the inner cylinder and (B) on both the cylinders in a staggered fashion. The flow field is modelled by the Darcy–Brinkman–Forchheimer model for the porous medium. **Bo-wen Hu et al. [4]** studied the GAHP-TSU presented for the potential solar application like air conditioning and refrigeration systems. In this composite granular solid–liquid PCMs compounded by RT100 and high-density polyethylene with phase change temperature of 100^0C are piled up as a porous PCMs medium layer. **MajidSiavashi et al. [5]** explained the simultaneous application of nanoparticles and porous media in the pipe to enhance heat transfer inside an annulus is investigated numerically. Two-phase mixture model along with Darcye -Brinkmane -Forchheimer relation has been implemented for nanofluid flow simulation in porous media. The results show that for design with high permeabilities (Da = 0.1, 0.01), PN has an increasing trend with porous element thickness, while for design with low permeabilities (Da = 0.0001), PN is found a decreasing trend with porous element thickness and for design with a moderate permeability (Da = 0.001), an optimum thickness is found corresponds to PN.**Yasser Mahmoudi and Nader Karimi [6]** studied the effect of different parameters like Darcy number (Da), inertia (F), porosity, conductivity ratio and particle diameter on account of local thermal equilibrium (LTE) are studied. The results showed that for a given model and for (Da < 10^{-3}), the Nusselt number is found independent of Friction factor however, for (Da > 10^{-3}) as F increases the computed Nu number increases. **MahboobeMahdavi et al. [7]** studied the convective heat transfer and entropy generation through a pipe partially filled with a porous material has been studied numerically for two different cases. In the first one, the cylindrical porous material is placed at the core, while in another case it attaches to the inner wall, which has been subjected to constant heat flux. It is concluded that the thermal conductivity ratio has the significant effect on increasing enhanced heat transfer in first case. For example for Da = 10^{-4} and R_p = 0.9, multiplying thermal conductivity ratio by ten leads to 641 % enhancement of Nusselt number

II. DESCRIPTION OF GEOMETRY SELECTION

For the present study, experiment and simulation are performed on various designs by varying the mass flow rates from 0.15 Kg/sec to 0.30 Kg/sec. The length of shell heat exchanger is 750 mm with 200 mm diameter. The thickness of the shell is 5 mm. Internal tubes are made of GS pipe with total five tubes are installed. Three baffles are used in this study for more heat transfer analysis as shown in fig. 1.

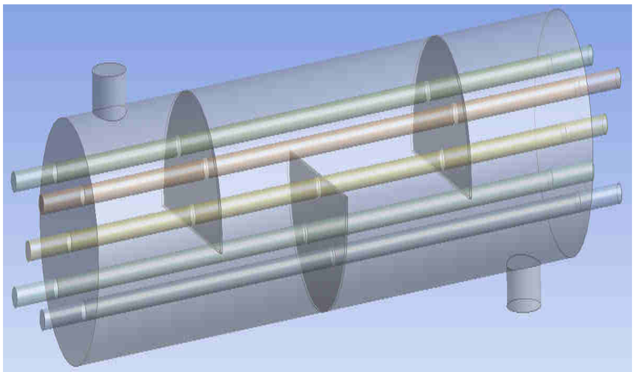

Fig.1: Isometric view of heat-exchanger

III. METHODOLOGY

The ANSYS FLUENT is used for simulation of three-dimensional calculations for all designs. The steady state mass, momentum, and energy equation are solved. In present work, the whole domain consists with 1 mm sizing which shows grid independency and meshing element almost 80000 to 400000 depending on different designs. Computational fluid dynamics (CFD) technique is used for computing the data along with k-omega turbulence model because this model is widely used by many researchers. The inlet of the cold fluid flow is designed as mass flow inlet to the normal direction of wall surface boundary with variation from 0.15 to 0.30 kg/s at the temperature of 300K and hot fluid enters at 323K. The outlet is described as the pressure outlet.

IV. DATA REDUCTION

The heat transferred from heated wall to fluid is calculated as the amount of heat which is gained by fluid that is

$$Q = m.Cp(T_o - T_i) \quad (1)$$

Where m. is the mass flow rate, Cp is the specific heat, To is outlet temperature, and Ti is inlet temperature of fluid

The average temperature of inlet and outlet temperature of fluid for the calculation of heat transfer coefficient

$$T_{pm} = \frac{T_o - T_i}{2} \quad (2)$$

And heat flux is

$$q = \frac{Q}{A_h} \quad (3)$$

$$A_h = \pi DL \quad (4)$$

where A_h is the area of the heated wall of the tube.

The heat transfer coefficient is determined by using

$$h = \frac{q}{T_h - T_{pm}} \quad (5)$$

where, T_h is the heated wall temperature

Flow area of fluid

$$A_f = \frac{\pi D^2}{4} \quad (6)$$

The Nusselt number, Reynolds number and Prandtl number is calculated by using

$$Re = \frac{\rho V_m D_h}{\mu} \quad (7)$$

$$Nu = \frac{h.D_h}{k} \quad (8)$$

$$Pr = \frac{\mu Cp}{k} \quad (9)$$

V. RESULT AND DISCUSSION

In present study, effect of porous medium material is studied in simple shell and tube heat exchanger. All simulationare performed on ANSYS FLUENT software.

Field experiments are performed for experimental validation. Setup is made of simple M.S plates.

In present study two types of porous mediums are considered for simulation, first medium is simple spherical metal balls (design 1) which is made of Cast Iron and second porous medium is cylindrical metal chips (design 2) which is made of mild steel. Porosity is considered as 80% level for all experiments. Counter flow heat exchanger analysis is performed for current study

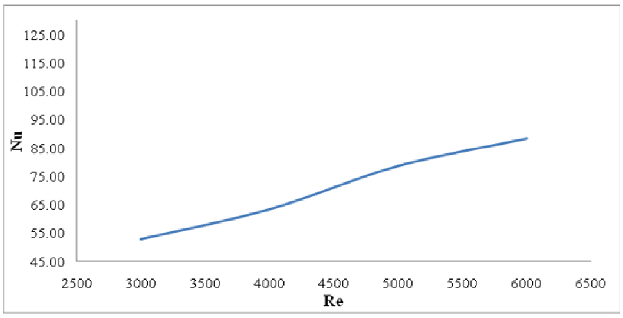

Fig.2: Variation in Nusselt number with Reynolds number for simple heat exchanger

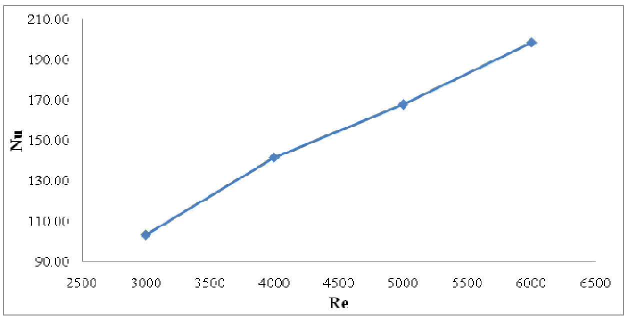

Fig.3: Variation in Nusselt number with Reynolds number for design-1

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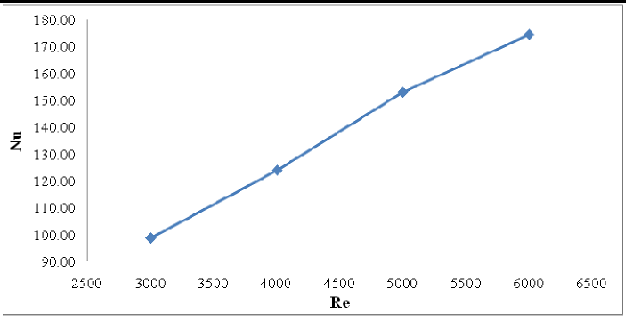

Fig.4: Variation in Nusselt number with Reynolds number for design-2

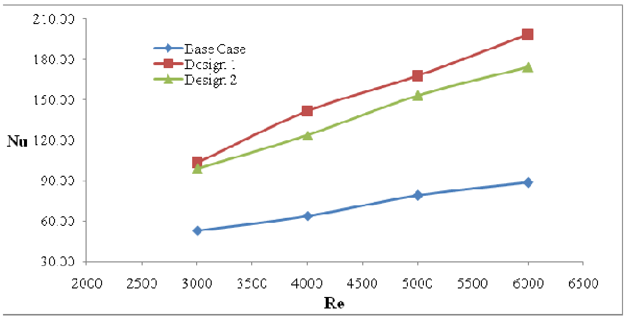

Fig.5: Variation in Nusselt number with Reynolds number for all designs

Figure 2 to figure 4 show the variation in Nusselt number with Reynolds number for simple shell and tube heat exchanger without porous medium. It is seen from figure that Nusselt number is continuously increasing by increasing of Reynolds number and it helps in better heat transfer in the heat exchanger. Nusselt number is found in the range of 50 to 90, 100 to 205 and 95 to 175 for heat exchanger without porous medium, design-1 and design-2 respectively.

VI. CONCLUSION

Figure 5 shows the comparison graph for Nusselt number with Reynolds number for all three designs. In this graph, simple heat exchanger(base case) is compared with the design 1 and design 2. As we know that if Nusselt number is found to be maximum for any design then this design gives the better heat transfer rate. So it is clearly seen that the design 1 has the highest value of Nusselt number as compared to design 2 and base case. It is also found that use of porous medium in the working fluid can enhance the heat transfer rate.

a. Nusselt number is continuously increased with increase of Reynolds number in shell and tube heat exchanger with and without porous medium. Nusselt number is found maximum for design-1.
b. When porous medium are used in the heat exchanger, which increases the heat transfer rate as compare to simple heat exchanger.

Nomenclature

A_h	Area of heated wall of tube (m^2)
A_f	Area of flow region (m^2)
d	Diameter of the tube (m)
h	Heat transfer coefficient ($W/m^2.K$)
K	Thermal conductivity of fluid (W/m-K)
L	Effective tube length for heat transfer (m)
Q	Heat transfer rate (W)
q	Heat flux (W/m^2)
T_h	Temperature of heated wall of tube (K)
T_i	Inlet temperature of fluid (K)
T_o	Outlet temperature of fluid (K)
T_{pm}	Average temperature of fluid (K)
V_m	Mean velocity (m/s)
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number

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