RESEARCH PAPER | Engineering | Volume : 3 | Issue : 8 | Aug 2013 | ISSN - 2249-555X

Force Convection Heat Transfer Analysis through Different Channel (Review Of Work)

KEYWORDS: Force convection, experiment, convective heat transfer coefficient

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ABSTRACT *The heat transfer through convection mode is very important in the thermal engineering and industrial application. In the present paper force convection heat transfer through different channel has been summarized/reviewed. The detailed and objective of the experimental set-up has been studied. It is studied that experiment were conducted on the flate plates in horizontal and vertical direction and circular pipes. In studied it is found that both laminar and turbulent/zig-zag flow were considered. The factors effecting the force convection heat transfer rate has been studied in the present review work.The effects of non-dimensional numbers and mass flow rate in the convective heat transfer analysis is predominant factor. With the help of present review work tries to find research gap and implementation in the experimental set-up that is use full in my future investigation.*

1. Introduction: Work done in the field of convective heat transfer analysis through different channel has been studied in this paper. The heat transfer in forced convection in an asymetrically heated flat plate channel with grooves located on the heated wall has been analysied [1] The temperature fields and local Nusselt numbers were measured by means of holographic interferometry analysied. A simple general method has been developed to predict force convection heat transfer from isothermal body shapes such as flat plates, infinite circular cylinders and spheres for a wide range of both Reynolds number and Prandtl number [2].The radial heat conduction in insulated pipes under external convection has been analyzed. In the the significance of different non dimensional number such as prandtl and reynold and their graphical representation of the dimensionless characteristic parameters, will allow a better understanding of the phenomenon convective heat transfer[3]. The friction factor and flow analsis has been done intwisted tapes for heating and cooling of Servotherm oil (medium grade) under uniform wall temperature. In the isothermal friction factors found to be 3.13 to 9.71 times the plain tube values for the laminar flow and mass flow rate for the were obtained 0.25 for heating and 0.12 for cooling, for the plain tube having a range of reynold number 0.12 for $Re_s \leq 250$ and 0.06 for $Re_s \geq 250$, for both heating and cooling sides [4]. The forced convection in a saturated porous medium subjected to heating with a permeable wall perpendicular to the flow direction is investigated analytically. To verify the analytical solution, experiments were carried out in a porous structure consisting of glass beads heated by a finned surface. The analytical solution is shown to be reasonable agreement with the experimental data [5].

2. Experimental analysis:
The experimental results of mixed convection heat transfer in a vertical packed channel with asymmetric heating of opposing wallshas been analyzed . The experiments were carried out in the range of 2<Pe<2200 and 700<Ra<1500. The measured temperature distribution indicates the existence of a secondary convective cell inside the vertical packed channel in the mixed convection regime .The swirling motion of the air was produced by a radial guide vane swirl generator. The vanes of the swirl generator were designed to be adjustable to obtain different swirl intensities. Different guide vane angles (15, 30, 45, 60 and 75^0) were used for the swirling flow experiments. The results were correlated in the form of Nusselt number as a function of Reynolds number, Prandtl number and the vane angle as $Nu=0.133Re^{0.65}Pr^{0.4}(1+\tan y)^{0.406}$ [6].

The methods used to determine convective heat transfer on interior surfaces of building envelope, which is the key linkage between ES with CFD programs. The study found that the size of the first grid near a wall in CFD is crucial for the correct prediction of the convective heat. A finer grid resolution in CFD does not always lead to a more accurate solution when using zero-equation turbulence models [7]. Experiments were carried out for both parallel and counter flow models of the fluids at Reynolds numbers between 8500–17 500. Heat transfer, friction factor and exergy analyses were made for the conditions with and without swirl generators and compared to each other. Some empirical correlations expressing the results were also derived and discussed. It was observed that the Nusselt number could increase up to 130% at a value of about 2.9 times increase in the friction factor by giving rotation to the air with the help of the swirl elements. The increase the dimensionless energy loss was about 1.25 times in comparison with that for the inner pipe without swirl generators [8].Using the conical nozzle and the snail it is found that, increase considerably the heat transfer rate over that of the plain tube by about 278% and 206%, respectively. The use of the conical nozzle in common with the snail leads to a maximum heat transfer rate that is up by 316%. Correlation equations for Nusselt number, friction factor and performance evaluation criteria to assess the real benefits in using the turbulators and swirl generator of the enhanced tube are determined [9].The experiments, hot (air) and cold (water) fluids flowed through the inner pipe and annulus, respectively. The experiments were performed for both parallel and counter current flow modes of the fluids at Reynolds numbers between 6500 and 13,000. An augmentation of up to 2.64 times in Nusselt number compared to the empty pipe was obtained in the helical system [10]. The forced convection heat transfer phenomena around spheri-

Volume : 3 | Issue : 8 | Aug 2013 | ISSN - 2249-555X

cal particles packed in fluid flow, we numerically analyzed the heat transfer and flow pattern of the air using a single sphere and then the closest packed structure arrangement of spherical particles. We used 3-dimensional thermo fluid computation code "STAR-CCM+". We calculated the forced convection heat transfer coefficient for spheres of 10 mm diameter with Reynolds number 63 – 6340.The average heat transfer coefficient for a single sphere agree with the correlation equation. Local heat transfer coefficient is high at portions where local flow impinges to the surface of spheres for packed spherical particles. Our calculation results of the average heat transfer coefficient for packed spherical particles are close to the correlation equation [11].Experimental and numerical study of forced convection heat transfer from an inclined heated plate placed beneath a porous medium carried out for plate with or without porous medium and the same inclination angle, the experimental results showed that the heat transfer coefficient was increased with increasing the Reynolds number. Also, it was found that the heat transfer coefficient was increased with increasing the inclination angle, and was reached the maximum enhancement ratio at inclination angle of 20° and with further inclination of the heating plate ($20° < \alpha \leq 30°$), the heat transfer coefficient was slightly decreased[12].

3. Experimental detailed.

The experiments were carried out for counter current flow models of the fluids at different Reynolds numbers. The heat transfer rates increased with decreasing diameter and with increasing number of the injectors used in the experiments [13]. Heat transfer, friction factor and enhancement efficiency characteristics in a circular tube fitted with conical-ring turbulators and a twisted-tape swirl generator have been investigated experimentally. The heat transfer test section is heated electrically imposing axially and circumferentially constant wall heat flux boundary conditions. For all the devices used, the enhancement efficiency tends to decrease with the rise of Reynolds number and to be nearly uniform for Reynolds number over 16,000. In addition, correlations for Nusselt number, friction factor and performance evaluation criteria to assess the real benefits in using the turbulator and swirl generator of the enhanced tube are determined [14].The mean Nusselt number, friction factor and enhancement efficiency characteristics in a round tube with short-length twisted tape insert under uniform wall heat flux boundary conditions. In addition, it is apparent that the enhancement efficiency of the tube with the short-length tape insert is found to be lower than that with the full-length one. The mean deviation between measured and correlated values of the Nusselt number is in the order of ±7% in the range of Reynolds numbers from 4000 to 20,000 [15].

The experimental were set up for the opposite plane wall is kept at ambient temperature. Both walls are maintained at uniform temperature. The working fluid is air. The thermal boundary layer develops along the test section under laminar flow conditions. The aspect ratio of the grooves, w/h, is 4:1 and the ratio of channel width to groove depth is 2:1. Experiments were performed for different Reynolds numbers. The flow is fully developed with parabolic velocity profiles at the entrance of the test section. The Two-Dimensional temperature fields were visualized and reconstructed using holographic interferometry, and local and average heat transfer data were obtained [1].

Experiments were conducted in two double-pipe heat exchangers--one for heating and the other for cooling, used in series. The test liquid was flowing on the tube-side (copper tube, D = 25.0 mm, L_h = 2245 mm and Lp = 2590 ram) and the heat transfer medium was passed, in counter-flow, through the annulus. Dry and saturated steam at a pressure of about 0.2 MN m -2 (g) was used as the heating medium in the heater. Chilled water at a constant flow rate of 0.692 kg s^{-1} (temperature = 16°C) was used as the cooling medium in the cooler. The pressure drops across the test-section were measured by using inclined U-tube manometers (0 = 5-90°). The manometers were mounted on a specially designed aluminium frame which can be adjusted easily to give any angle of inclination to the horizontal. Mercury was used as the manometric liquid. A mild steel storage tank (capacity 0.50 m 3) equipped with a cooling coil was used as a storage tank for the test liquid [4]

The geometric structure of a turbulator constructed by twisting galvanized iron strips 60 mm wide (H) and 2 mm thick (t) is given. Their total lengths are 1100 mm. The distance obtained on the strip after a complete revolution about the rotational axis is defined as one step or pitch, as shown in this figure. Two kinds of turbulator, having 100 or 170 mm pitches, were prepared for the experiments. The concentric double-pipe heat exchanger used in the experiments and its accessories are shown in flow diagram. The inside diameters of the 1100 mm length inner and outer pipes were 50 and 70 mm, respectively[16].

A forced convection of R-113 loop was used in performing the experiments. The test section is vertically oriented with R-113 flowing against gravity. It consisted of four vertical walls: two aluminum plates serving as the heat source and the heat sink, and two acrylic plates serving as adiabatic side walls. Chrome steel beads (dp=6.35 mm) were used as porous media. The overall dimensions of the test section were 66.04 cm in length, 20.32 cm in width, and 30.48 cm in depth. The cross-sectional flow area of the test section was 5.08 x 15.24 cm. The heated and cooler plates, 45.72 cm in length (L) and facing opposite from each other, were separated by a distance (H) of 5.08 cm [6].

In order to verify the above analyses, experiments were performed in the test sections. The vertically oriented test section, 31.5 mm in height, 99 mm in width, and 28 mm in depth, was enclosed with four vertical walls: three Teflon plates located in the both lateral and the back sides and one transparent Pyrex glass plate located in the front side [5].

The experiments were conducted in an open loop experimental facility the loop consisted of a 2.2 kW blower, orifice meter to measure the flow rate and the heat transfer test section. The copper test tube has a length of L = 1250 mm, with 47.5 mm inner diameter (D), 50.5 mm outer diameter (Do) and 1.5 mm copper tube thickness (t) as depicted in Fig. 2a and b. Fig. 2a represents the conical-nozzle arrangement used in the present experiment. The conical nozzle was made of aluminum with 47.5 mm (1.0 D) length, and its end diameters were 46 mm and 26 mm, respectively. The conical nozzles were placed with three different pitch lengths, having L = 95 mm (PR = 2.0), L = 190 mm (PR = 4.0) and L = 332.5 mm (PR = 7.0), for each experiment [9].

The experiments were carried out using an experimental facility. The test tube is made of copper and has inner diameter of 19 mm (D), outside diameter of 21 mm (Do), wall thickness of 1.5 mm (t) and length of 1000 mm (L). In the experiments, the single and twin-counter/co-twisted tapes were inserted at the core tube along the test section. The test loop consists of a water pump, data logger, pressure transmitter, thermocouple rotameter and heat transfer test section [17].

4. Application experimental set-up:

1. Heat transfer measurements and in the visualization of temperature fields.
2. Determining the flow friction and heat transfer results in a plain empty tube and comparing them with the available correlations.
3. Steady-state values of isothermal and non isothermal friction factors and mean Nusselt numbers for uniform wall temperature heating and cooling of Servotherm oil were then determined with each of the twisted tape insert.
4. Determining local and average convective heat transfer

coefficient.

5. Determine local skin friction coefficient.
6. Validation and evaluation the various non dimensional numbers such as Re, Pr, Pe, Nu etc.

5. Scope of analysis:

The work can be extended in following direction

1. Acquired with increasing pitch size.
2. Considering the cause a increase in pressure drop, but the heat equivalent of this energy loss is negligible in comparison with the heat gained by the tabulators.
3. The detailed effects of pitch size and on applications in flue-gas tube boilers involving very hot gases.
4. Convective heat transfer rate in the circular pipe can be increase using the resistant (insert rib or another geometry) in the direction of flow.

Conclusion:

The present review work is use full to develop a new experimental set-up as well find the research gap/directions in which proposed work can be extended, that will be appear in the further analysis.

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