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Experimental Investigation of Forced Convective Heat Transfer Coefficient & Pressure Drop in Bi-metallic Plate Heat Exchanger

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Abstract: The present work reports on heat transfer enhancement and the corresponding pressure drop over a flat surface equipped with copper plate fins in a rectangular channel with forced air cooling. The channel had a cross-sectional area of 100 mm x 250 mm. The experiments covered the following range: Heat input corresponding to the voltage (Q) 40 Volt, 60 Volt and 80 Volt, number of plate fins (N_P) 4, 6 and 8 and number of slots on plate (N_S). Forced convective heat transfer coefficient and pressure drop are considered as performance parameters. The full factorial method of design of experiment (DOE) with 3 factors and 2 levels is used for the sequence of experimental trials. The analysis is done with regression method and 'D'optimal method is used for response optimization. The experimental results are plotted on main effect plots, plots of the standardized effect and analyzed for influence of each input parameters. The selection of optimum parameters for heat transfer enhancement is investigated to maximize convective heat transfer coefficient and minimize the pressure drop.

Keywords: Heat Transfer Enhancement, Heat Transfer Coefficient, Full Factorial Method, Performance Parameters, Forced Air Cooling.

I. INTRODUCTION

The most of the industrial era having heat dissipation is a drastic issue to tackle due to continued integration, miniaturization, compacting and lightening of equipment. Heat dissipaters are not only chosen for their thermal performance but also for other design parameters that includes weight, cost and reliability, depending on application. For example, weight and reliability are important for space applications. The thermal systems must be designed and sized to generate, dissipate the appropriate amount of unwanted heat with the required demand. The development of high performance thermal systems has been stimulated in many fields of new technologies. Conventional heat transfer devices have to be substantially improved to answer the needs of systems from the micro scale to large power plants. In this perspective, convective heat transfer can be enhanced in several ways, by using either active or passive techniques. In the latter case, it is made possible by changing the structure of the heat exchanger or the properties of the heat exchange surface. In a conventional heat exchanger heat is transferred from one fluid to another through a metallic wall. The rate of heat transfer is directly proportional to the extent of the wall surface, the heat transfer coefficient and to the temperature difference between one fluid and the adjacent surface. The ability of an object to reject excess heat is a required task to ensure operability, and if not accomplished sufficiently, can result in device malfunction or even failure. In industrial processes, internal heat-generation may cause overheating and hence sometimes even system failure. So an effective means of removing this heat is often required.

II. EXPERIMENTAL SET-UP

The experimental set-up consisting of the main duct, blower, diffuser, differential manometer, heating unit, test section and tested bi-metallic plate fins, temperature indicator with sensors, voltmeter, ammeter and variac as shown in figure 3.1. The components of test rig are designed or selected according to some reference specifications of the components. It is installed according to the basic laws or standard used for a specific component installation. The main duct or tunnel constructed of wood of 16 mm thickness, has an internal cross-section of 250 mm width and 100 mm height (channel aspect ratio 2.5:1, hydraulic diameter, D_h = 142.86 mm). The total length of the channel is 1100 mm. It is operated in suction mode and positioned horizontally. The experimental set up uses a centrifugal blower; it uses the kinetic energy of the impellers or the rotating blade to increase the pressure of the air stream. A diffuser or bell mouth section is the mechanical device that is designed to control the characteristics of a fluid at the entrance to a thermodynamic open system. Differential manometer is an instrument for the determination of pressure drop along the test section.

Heating unit mainly consist of plate type mica heater has a cross-section of 200 mm x 250 mm and placed on wooden block having the same dimension of base plate. The test section has a cross-section of 250 mm x 100 mm square, and a length of 450 mm. The steady state temperature of the base plate is measured by five PT100 flexible sensors for temperature 0 to 250℃, pipe washer type with spring, length 1.5 meter and 3 cores with tef/fg/ss wire inserted into different grooves in the base plate. The readings of the each sensor are recorded using a single point digital temperature indicator of size 96 mm x 48 mm, and the average of all these readings is taken to be the steady state temperature of the test surface.

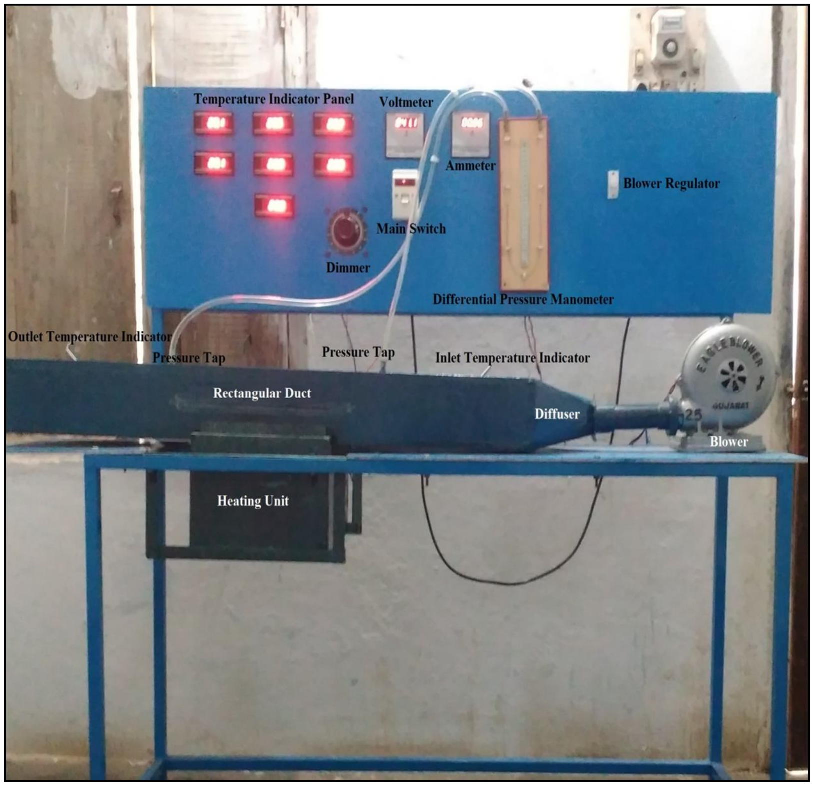

Figure 1: Photographic View of Experimental Set-up

A voltmeter is an instrument used for measuring electrical potential difference between two points in an electric circuit. Ammeter is a measuring instrument used to measure the current in a circuit. In present experimental set up open type hand operated 5 ampere single phase with input voltage 230V and output 0 to270V variable transformer is installed.

Table 1: Operating Conditions Tested in Experimentation

Sr. No	Parameters	Values		
		Low Level	Medium Level	High Level
1	Q	40 Volt	60 Volt	80 Volt
2	N_P	4	6	8
3	N_S	4	6	8

III. DATA PROCESSING

A. Heat Transfer & Pressure Drop

The data obtained from experiments are processed for further analysis with the help of various formulae. The convective heat transfer rate Q_{conv} from electrically heated test surface is calculated by following formula:

$$Q_{conv} = Q_{elect} - Q_{cond} - Q_{rad} \tag{1}$$

Where, Q indicates the rate of heat transfer in which subscripts conv, elect, cond and rad denote convection, electrical, conduction and radiation, respectively. The electrical heat input is obtained from the electrical voltage and current supplied to the surface. In similar studies, investigators reported that total radiative heat loss from a similar test surface would be about 0.5% of the total electrical heat input. The conductive heat losses through the sidewalls can be neglected in comparison to those through the bottom surface of the test section. Using these findings together with the fact that the two sidewalls and the top wall of the test section is well insulated therefore one could assume with some confidence that the last two terms of Eq. (1) may be ignored. The heat transfer from test section by convection can be evaluated as:

$$Q_{conv} = h_{av} A_s [T_s - (\frac{Tout+Tin}{2})] \tag{2}$$

Hence, the average convective heat transfer coefficient can be expressed as:

$$h_{av} = \frac{Q_{conv}}{A_s[T_s - (\frac{T_{out} + T_{in}}{2})} \tag{3}$$

Either the projected or the total area of the test section can be treated as the heat transfer area in the calculations. The projected area is considered for present investigation for calculation of heat transfer coefficient. The projected area is calculated by using following formula:

$$Projected\ area = W \times L \tag{4}$$

The dimensionless groups are calculated as follows:

1)Reynolds Number: It is ratio of inertia force to viscous force. It is computed by using the equation:

$$\mathrm{Re} = \frac{D_h U}{v} \tag{5}$$

2) Pressure Drop: It is defined as the difference in total pressure between two points of a fluid carrying network. Pressure drop occurs when frictional forces, caused by the resistance to flow, act on a fluid as it flows through the tube or duct. The pressure drop can be calculated as follows:

$$\Delta P = \rho g H \tag{6}$$

In all calculations, the values of thermo-physical properties of air were obtained at the bulk mean temperature, which is

$$T_m = \frac{T_{in} + T_{out}}{2} \tag{7}$$

The system of bi-metallic plate heat exchanger will be tested under forced convection for different combinations of number of plates with different number of slots over the plates. The number of sets of experiment obtained by using factorial design of experiment and evaluation of response variable will be done by using above equations. Further analysis for maximum convective heat transfer coefficient and minimum pressure drop and optimization will be done by using analysis of variance techniques.

B. Uncertainty Analysis

All the parameters that are computed in order to investigate the forced convective heat transfer coefficient and pressure drop are subjected to uncertainties due to errors in the measuring instrument. The uncertainty analysis is carried out using the estimation method of Kline and McClintock considering the measurement errors. The uncertainty calculations based on experimental results for forced convective heat transfer coefficient and pressure drop are presented in this section. The uncertainties in variables are summarized in table 2.

Table 2: Uncertainties in Values of the Relevant Variables for Heat Transfer Coefficient

Sr. No.	Variables	Uncertainty (%)
1	Temperature (T)	1.62
2	Voltage (V)	1
3	Current (I)	1
4	Projected area of plate (A_s)	2
Total Uncertainty		5.62

Table 3: Uncertainties in the Values of the Relevant Variables for Pressure Drop

Sr. No.	Variables	Uncertainty (%)
1	Temperature (T)	1.62
2	Pressure (P)	1
3	Projected area of plate (A_s)	2
4	Length of Test Section (L_t)	1
Total Uncertainty		5.62

IV. RESULTS AND DISCUSSIONS

A. Validation of Experimental Set-up

In order to have a basis for the evaluation of the effects of the fins, some experiments were carried out without any fins attached to the plate. Using the data obtained from these test, the Nusselt number (Nus) for the smooth surface (without pin fins) was correlated as function of Re and Pr as follows (B. Sahin et al., 2008).

$$Nu_s = 0.0077\ \mathrm{Re}^{0.716}\ Pr^{1/3} \qquad (8)$$

This equation gives the theoretical Nusselt Number and the experimental results for a smooth channel were compared with the results obtained from above equations and the average percentage error obtained between theoretical and experimental Nusselt number for smooth plate without pin-fins is 6.12 %. The deviation between theoretical and experimental results in acceptable range and there are good agreements between theoretical and experimental results as shown in figure 2. Hence experimental setup is validated.

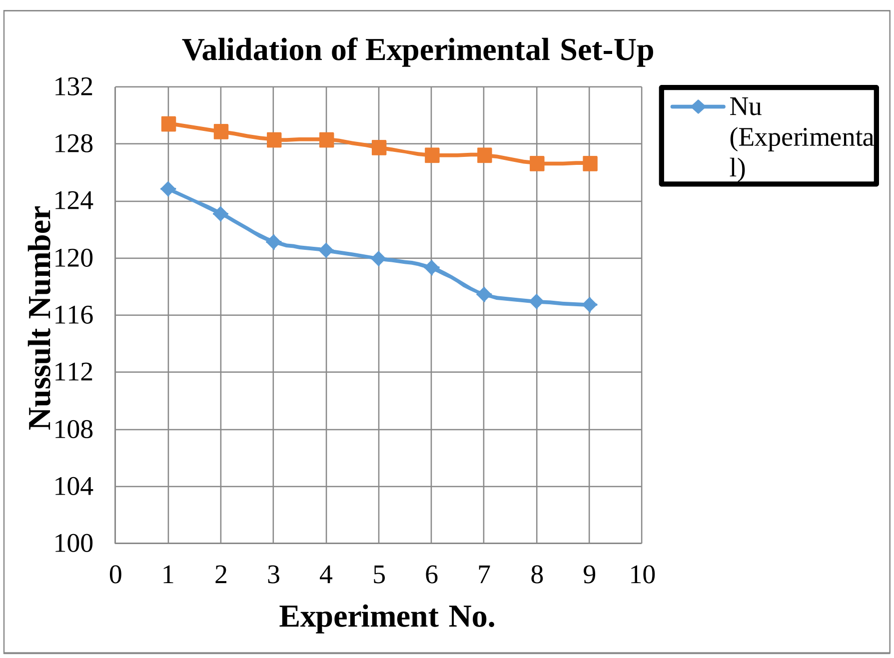

Figure 2: Comparison of Theoretical and Experimental Results for Nusselt Number

B. Forced Convective Heat Transfer Coefficient Analysis

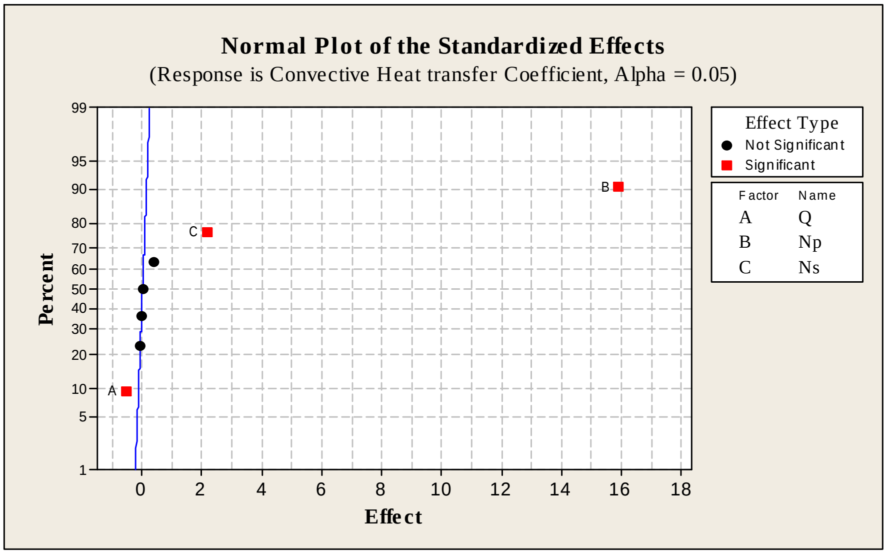

Figure 3: Plot Showing Effect of Each Parameter on Convective Heat Transfer Coefficient

Figure 3 shows normal plots indicates the percent effect of each parameter on convective heat transfer coefficient, there are three significant effects ($\alpha \leq 0.05$). These significant effects includes all three main effects, heat input (Q), number of plate fins (N_P) and number of slots on plate fin (N_S). In which, number of plate fin (N_P) shows 90.54% effect, number of slots on plate fin (N_S) shows 77.03% effect and heat input (Q) shows 9.45 % effect. In addition, the plot indicates the direction of the effect. Out of three parameters, two have positive effects because they reside to the right of the line. This means when shift from the low level to the high level of the parameters, the convective heat transfer coefficient increases. One parameter resides to the left of the line, this means when shift from the low level to high level of the parameters, the convective heat transfer coefficient decreases.

C. Pressure Drop Analysis

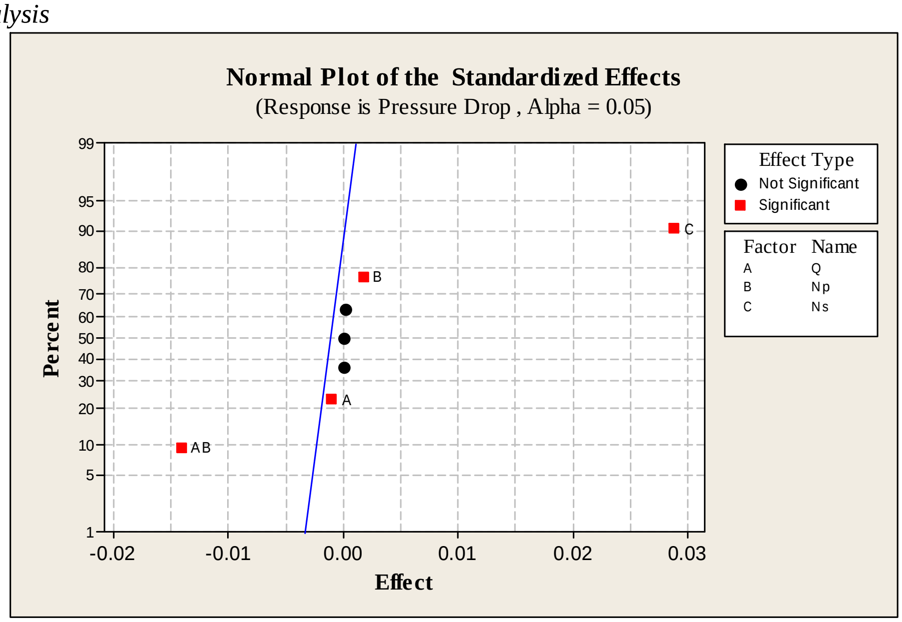

Figure 4: Plot Showing Effect of Each Parameter on Pressure Drop

Figure 4 shows normal plots indicates the percent effect of each parameter on pressure drop, there are four significant effects ($\alpha \leq 0.05$). These significant effects includes all three main effects, heat input (Q), number of fin plate (N_P) and number of slots on plate fin (N_S). All four parameters have largest effect because it lies furthest from the line. Out of four there is one significant effect of parameter included in 2-way interaction.

In which heat input (Q), shows 22.97% effect, number of fin plate (N_P) shows 76.94% effect, number of slots on plate fin (N_S) shows 90.34% effect and in 2-way interaction ($Q*N_P$) shows 9.84% effect. In addition, the plot indicates the direction of the effect. Out of four parameters, three parameter i.e. heat input (Q), number of fin plate (N_P), number of slots on plate fin (N_S) shows positive effects because they reside to the right of the line, this means when shift from low level to high level of parameter, the pressure drop increases and remaining one parameters i.e. ($Q*N_P$) shows negative effect because they reside to the left of the line. This means when shift from the low level to the high level of the parameters, the pressure drop decreases.

V. CONCLUSIONS

In this study, the effect of the various design parameters on the convective heat transfer coefficient and pressure drop for the heat exchanger equipped with copper plate fins mounted on aluminium base plate were investigated experimentally. The effects of the heat input, number of fin plate and number of slots on plate on the convective heat transfer coefficient and pressure drop characteristics were determined and regression equations have been obtained. By using the full factorial experimental design method optimum parameters for convective heat transfer coefficient and pressure drop were determined. The following conclusions are drawn for effective heat transfer and summarized as:

A. Development of Experimental Set-up

In this work design and development of experimental test rig are carried out for desired investigation of forced convective heat transfer coefficient and pressure drop. The components of test rig are designed or selected according to some reference specifications of the components and ISO 5167 standard. It is installed according to the basic laws or standard used for a specific component installation.

B. Validation of Experimental Set-up

Validation of experimental set up is carried out with well known equations for Nusselt number. The average percentage error of experimental and theoretical Nusselt number is 6.12 %. The experimental results are deviated from theoretical results in acceptable range and there are good agreements between theoretical and experimental results. Hence experimental set-up is validated.

C. Convective Heat Transfer Coefficient Investigation

The higher values of input parameters such as number of fin plate (N_P = 8 Nos.), number of slots on plate (N_S = 8 Nos.) and lower value of heat input (Q = 40 Volt) provided the maximum value of convective heat transfer coefficient which is equals to 43.7533 $W/m^2 °C$. The analysis is carried out to check the influence of each input parameter which gives the statistically significant effect on convective heat transfer coefficient.

D. Pressure Drop Investigation

The lower values of input parameters such as heat input (Q = 40 Volt), number of fin plate (N_P = 4 Nos.) and number of slots on plate (N_S = 4 Nos.), provided the minimum value of pressure drop which is equals to 4.42 N/m^2.

E. Regression Equation

The following mathematical models are obtained for convective heat transfer coefficient and pressure drop from regression results.

Convective Heat Transfer Coefficient (h_{conv}) = 14.6 – 0.170(Q) + 2.54(N_P) + 1.96(N_S).

Pressure Drop (ΔP) = 27.6 + 0.0521 (Q) + 2.36 (Np) + 4.20 (Ns).

F. Response Optimization

It is carried out to optimize two responses at the same time. To maximize convective heart transfer coefficient and minimize pressure drop in one set of input variables. It is observed that convective heat transfer coefficient and minimum pressure drop are obtained simultaneously by employing Q = 44.66 Volt, N_P = 5 Nos. and N_S = 8 Nos.

G. Validation of Experimental Results

The confirmation experiment must be performed to verify their validity. The independent variable values with inclusion of medium level values are selected for the confirmation experiment must lie within the ranges for which equations were derived. The mean absolute error between the experimental and predicted value is found to be 5.56 and 6.60 for the convective heat transfer coefficient and pressure drop respectively. Hence this confirms excellent reproducibility of the experimental conclusions.

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