Thermal Hydraulic Behavior of Research Reactor during Natural Convection Cooling Mode

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ABSTRACT

In the present investigation, an experimental apparatus is designed and constructed to simulate the cooling process through the channel in the core of research reactor during natural cooling. The measurements included the effect of the pool temperature, heat flux and the height of the chimney (extension ratio), on the coolant velocity, heat transfer coefficient, coolant temperature and surface temperature. The cooling channel was made from two heated vertical parallel plates of aluminum alloy acting as a vertical rectangular channel where water was used as a cooling fluid. Experiment was carried out under the atmospheric conditions. The measurements were done for free convection from vertical isoflux and parallel-walled channels to explore the heat transfer enhancement due to adding the adiabatic extensions of different heights (15, 35 and 55 cm) to the heated channel. The coolant velocity and the heat transfer coefficient increases with the increase in the heat flux at constant coolant inlet temperature. For constant heat flux the coolant velocity increases and the heat transfer coefficient is enhanced by the increase of the coolant inlet temperature, i.e. the reactor operates more efficiently in the summer conditions in natural circulation cooling mode.

Key Words: Heat Transfer/ Natural Convection / Thermal Hydraulic.

INTRODUCTION

For a reactor to operate in a steady state conditions with an internal temperature distribution, all of the heat released in the system must be removed as fast as it is produced. Liquid or gaseous coolant should pass through the core and through other regions where heat is generated. The nature and operation of this coolant system is one of the most important considerations in the design of the nuclear reactor. For an efficient natural cooling process, it is necessary to fully understand the mechanism of heat dissipation under different operating conditions. In present study, common free convection geometry involves channels formed by vertical parallel plates which are open at the opposite ends, is adopted. This geometry is considered here to simulate the cooling process through the channel formed by fuel elements in the core of Plate Type Research Reactor. The heat channel configurations formed by vertical parallel plates used in reactors cooling are found in many technological and engineering applications, which include the cooling of electronic equipment in computers, heat exchangers and solar chimneys. However, most investigations have treated severely idealized situations for channels such as vertical or inclined parallel walls, which are isothermal or heated with a uniform heat flux. Many experiments have been conducted to measure the heat transfer coefficients for natural convection in fuel elements for both rod bundles and plate type including isothermal and iso-flux conditions.
Vertical channel formed by parallel and vertical surfaces has received great attention in the past. Elenbaas 1942 first has open edges that introduced the problem of natural convection between vertical plates as mentioned by Wirtz and Stutzman (1). A number of experimental studies have been carried out (e.g. Wirtz and Haag, 1985; Azevedo and Sparrow, 1985; Miyamoto et al., 1986; Sparrow and Ruiz, 1988; and Krishnan et al, 2004). Wirtz and Haag (2) have obtained experimental results for isothermal symmetrical heated plates with an unheated entry channel portion. It was found that the flow was quite insensitive to the presence of the unheated entry section for large channel spacing, while it was severely affected when the gap spacing is small. Azevedo and Sparrow (3) have performed experiments on an inclined, isothermal channel using water as the convecting fluid. Three modes of heating were investigated; heating both walls, top wall only heated, and bottom wall only heated. Correlations describing this modification were obtained. Miyamoto et al. (4) have determined experimentally the heat transfer coefficient in case of turbulent natural convection flow in an asymmetrically heated vertical channel where two vertical parallel plates formed the channel. One plate was heated by imposing a uniform heat flux along the plate and the opposite plate was adiabatic. The channel was open at the bottom and top. Experiments were performed using channel widths of 50, 100 and 200 mm.

Sparrow and Ruiz (5) have obtained useful heat transfer data for natural convection in divergent vertical channels to visualize the flow in the channels, and to develop a universal Nusselt number correlation for divergent, convergent, and parallel-walled channels. The experiments are performed with water as a working fluid (Pr=5). The measured Nusselt numbers for the divergent channel are brought into tight correlation with those for the convergent and parallel walled channels by using correlating parameters based on the maximum inter wall spacing as the characteristic dimension. Krishnan et al (6) have investigated experimentally natural convection and surface radiation between three parallel vertical plates, symmetrically spaced, with air as the intervening medium. The analysis consists of heating the central plate at different levels and recording the temperatures of both the central and the side plates at steady state conditions. The measurements were performed for a range of emissive 0.05=ε=0.85, aspect ratio 2.38=A=17, and total heat flux 32=q=1590 W/m². A number of numerical studies have been carried out (e.g. Sparrow, et al., 1984; Naito and Nagano, 1989; Naylor et al., 1991; Straatman et al., 1993; Lee, 1994; Straatman et al., 1994; Floryan and Novak, 1995; Roberts and Floryan, 1998; Campo et al., 1999; and Shahin and Floryan, 1999).Sparrow, et al (7) obtained some numerical results for the heat transfer in a vertical channel using fluids with Prandtl numbers ranging from 0.7 to 10.0. The results showed that increasing the Prandtl number enhances the heat transfer rate through the channel. Naito and Nagano (8) have investigated the effect of buoyancy on the hydrodynamic and thermal characteristics of downward-flow laminar convection in the entrance region between inclined parallel plates. Numerical solutions are given for three thermal conditions of parallel plates with uniform wall temperature or insulation. The values of local Nusselt number became smaller than those of a horizontal channel with increasing the inclination angles.

straatman et al. (9) have carried out a numerical and experimental investigation of free convection from vertical isothermal, parallel-walled channels with adiabatic extensions of various sizes and shapes. The experiments were performed with ambient air using a Mach-Zehnder interferometer. In all cases, the adiabatic extensions showed an increase in heat transfer. The increase varied from 2.5 at low Rayleigh number to 1.5 at high Rayleigh number. Floryan and Novak (10) have investigated numerically the free convection heat transfer in multiple parallel vertical channels with isothermal walls. Different systems consisting of two, three and an infinite number of channels located side by side and with aspect ratios ranging from 5 to 20 and for Grashof numbers up to 105 were considered. The results show that the heat transfer in a channel is affected by its interaction with the neighboring channels. Shahin and Floryan (11) analyzed numerically the chimney effect in a system of isothermal multiple vertical channels. Each channel had an adiabatic extension. It was observed that the
interaction between multiple vertical channels increases the induced flow rate and that the associated chimney effect is stronger than in a single channel with adiabatic extensions.

**DESCRIPTION OF THE EXPERIMENTAL SETUP**

The aim of this work is to investigate experimentally and analytically the heat transfer characteristics occurring by natural convection cooling thorough the channel formed by fuel elements of plate type in the core of the Plate Type Research Reactor. For this reason, an experimental setup is being designed and constructed here as shown in Fig. (1). This geometry is considered to simulate the cooling process thought the channel formed by fuel elements. As shown in Fig. (1), the coolant (water) enters the heated channel (1) from the bottom and flows in upward direction. The channel is heated by two plate type electric heaters (2) which are placed vertically. The power generated by the electric heaters is consumed to raise the temperature of the clad (3) and the heat added to the clad is removed by the circulation of cooling water in the channel. The heated channel is thermally insulated from the surrounded coolant in the pool (4) by foam insulation (5). The hot water is moved upward through the channel and chimney (6) due to buoyancy effect. After exiting from the channel the water is mixed with the cold water in the pool then enters the plenum (7) under atmospheric conditions.

**Fig. (1): Experimental setup.**
MEASURING DEVICES AND INSTRUMENTATION

Forty-five calibrated copper-constantan thermo-couples were used to measure the local temperatures of the clad surface. The thermocouples junctions were embedded inside grooves of 3 mm depth created on the rear surface of the clad. Nine thermocouples were used to measure the coolant local temperatures through the heated channel. Two additional thermocouples were used; the first was fixed at inlet of the channel while the second was located at exit of the chimney. The electric circuit built here consists of a control panel which has two Selector switches each of which is provided with eleven selecting points, heaters and indicating lamps. The circuit is also provided with a DC variac and wiring. Measurement of the surface temperatures of the test section and coolant temperatures were carried out using the thermocouples. The output millivolts of these thermocouples were recorded through a sensitive digital multi-meter of 0.01-millivolt accuracy.

DATA REDUCTION

The heat transfer coefficient, $h$ is calculated from Newton’s law of cooling

$$h = \frac{Q_{seg}}{2(L_h \times w)\Delta T_{seg}}$$

Where

$$Q_{seg} = \frac{Q_{net}}{n}$$

$$Q_{net} = I \times V - Q_{con} - Q_{rad}$$

$Q_{con}$ and $Q_{rad}$ are small and are neglected.

ii) Local Nusselt number, $Nu$.

$$Nu = \frac{h \times b}{k}$$

iii) Flow velocity, $u$ 

$$u = \frac{m^*}{\rho \times b \times w}$$

iv) Rayleigh number, $Ra$ and modified Rayleigh $Ra^*$

$$Ra = \frac{g \beta \Delta T b^3}{\alpha \nu}$$

$$Ra^* = Ra \times b$$

All properties of coolant are calculated at the mean temperature, $T_m$

$$T_m = \frac{T_{co} + T_{ci}}{2}$$

RESULTS AND DISCUSSION

Natural convection cooling through vertical rectangular channel with uniform heat flux is calculated here using RELAP5 code to simulate the cooling process-taking place through the coolant channel of Plate Type Research Reactor. The study included the effects of chimney height (extensions of the cooling channel), operating power and pool temperature on the thermal-hydraulic behaviour of the coolant channel. The main interesting parameters include velocity, heat transfer coefficient and the local temperatures distribution of coolant and clad surface.
The RELAP5 code [14] is based on formulation of the field equations, the conservation of mass, momentum and energy equations in addition to the appropriate boundary and initial conditions. The input file to the Relap5 code contains essentially the description of the problem to be executed and the geometrical description of the system under analysis. The present analytical model is based on the assumptions: vertical rectangular channel, uniform heat flux, steady state one-dimensional analysis and incompressible fluid.

**EXPERIMENTAL MEASUREMENTS:**

**Local Distribution of the Coolant Temperatures**

Figure (2) shows the distribution of the local temperature of the coolant along the axial distance of the channel for different values of coolant inlet temperature (Tin) and constant values of the heat flux (q"=10084 W/m²) and the same extension ratio (L*=1.2). While Fig. (3) illustrates the effect of heat flux on the distribution of the local coolant temperature, for constant coolant inlet temperature (Tin=40 °C) and extension ratio (L*=1.2). Also, Fig. (4) shows the effect of the extension ratio (L*) on the local distribution of the coolant temperature in the channel, for constant value of heat flux (q"=10084 W/m²) and inlet coolant temperature (Tin) of 40 °C. From Figures (2), (3) and (4) it is noticed that:

a) The local coolant temperature increases with the increase in the coolant inlet temperature. In addition, it is clear that the increase in the local coolant temperature along channel is higher close to the outlet of the channel.

b) The effect of heat flux is obvious close to the outlet of the channel. The increase in the local coolant temperature is due the increase of heat flux.

c) The decrease in the local coolant temperature is due to the increase of the extension ratio from 1.2 to 1.62. In addition, it is observed that there is an increase in the local coolant temperature through the channel, towards its outlet.

**Local Distribution of the Surface Temperatures**

Figure (5) indicates the effect of variation of coolant inlet temperature (Tin) on the local distribution of the clad surface temperature (Ts) for constant value of the heat flux (q"=10084 W/m²) and same extension ratio (L*=1.2). It is clear that as the inlet temperature of the coolant increases the local coolant temperature increases. Figure (6) shows the distribution of the clad surface temperature (Ts) along the axial distance of the channel for different values of the heat flux and same extension ratio (L*=1.2) and coolant inlet temperature (Tin) of 40 °C. Also, Fig. (7) Shows the effect of variation of extension ratio on the local surface temperature for constant value of heat flux (q"=10084 W/m²) and inlet coolant temperature Tin of 40 °C. From Figures. (5), (6) and (7) it is obvious that:

a) The local temperature of clad surface increases toward the channel outlet, with the increase of coolant inlet temperature.

b) The local surface temperature increases with the increase in the heat flux. In addition, it is noticed that the local surface temperature shows an increase towards the outlet of the channel.

c) The local surface temperature increases from inlet to outlet through the Channel, with the increase in the value of the extension ratio.
Local Distribution of the Heat Transfer Coefficient

Figure (8) illustrates the distribution of the local heat transfer coefficient along the axial distance of the channel for different values of the heat flux, for constant coolant inlet temperature \(T_{in}=40 \, ^\circ\text{C}\) and same extension ratio \(L^*=1.2\). While Fig. (9) presents the effect of variation of the coolant inlet temperature on the local heat transfer coefficient for constant value of the heat flux \(q''=10084 \, \text{W/m}^2\) and same extension ratio \(L^*=1.2\). Also, Fig. (10) illustrates the effect of variation of the extension ratio, on the local of heat transfer coefficient for constant value of coolant inlet temperature \(T_{in}=40 \, ^\circ\text{C}\) and constant value of the heat flux \(q''=10084 \, \text{W/m}^2\). It can be concluded from Figures. (8), (9) and (10) that:-

a) The local heat transfer coefficient increases with the increase in the heat flux. Also it is clear that the local heat transfer coefficient increases along channel with its highest value close to the outlet of the channel.

b) The local heat transfer coefficient increases from inlet to outlet through the channel, with the increase in the coolant inlet temperature.

c) The heat transfer coefficient increases through the channel and is enhanced with the increase of the extension ratio.

Coolant Velocity

Figure (11) presents the variation of the average coolant velocity through the channel with the coolant inlet temperature for different values of the extension ratio \(L^*=1.2, 1.41\) and 1.62), for the same heat flux \(q''=10084 \, \text{W/m}^2\). While Fig. (12) presents the variation of average coolant velocity through the channel with the heat flux for different values of the extension ratio \(L^*=1.2, 1.41\) and 1.62), at same coolant inlet temperature \(T_{in}=40 \, ^\circ\text{C}\). Both figures show that:-

a) The coolant velocity increases with the increase of the coolant inlet temperatures for all values of the extension ratio. The rate of increase in the coolant velocity is high for larger value of the extension ratio.

b) The coolant velocity increases with the increase in the extension ratio for all values of the coolant inlet temperatures. The variation of the coolant velocity with the extension ratio is almost linear for the tested range. The rate of increase in coolant velocity is relatively high for higher values of coolant inlet temperature.

c) The variation of coolant velocity with the heat flux is almost linear. The coolant velocity increases with the increase in the heat flux for all values of the extension ratio.

d) The coolant velocity increases with the increase in the extension ratio for all values of the heat flux. It is also clear that the rate of increase in coolant velocity is high for larger values of heat flux. This means that the increase in extension ratio increases the value of buoyancy force, which in turn increases the coolant velocity.

Coolant Temperature Difference

Figure (13) illustrates the variation of the coolant temperature difference \(\Delta T_c\) with the coolant inlet temperature for different extension ratios at constant value of heat flux. Figure (14) illustrates the variation of the coolant temperature difference \(\Delta T_c\) with the heat flux for different extension ratios at
constant value of coolant inlet temperature. It is observed that the coolant temperature difference (ΔTc) decreases with the increase of the coolant inlet temperatures for all values of the extension ratio.

**Average Nusselt Number**

Figure (15) shows the variation of the average Nusselt number with the coolant inlet temperature \(T_{in}\) for same value of heat flux \((q''=10084 \text{ W/m}^2)\) and different values of extension ratio \((L^*=1.2, 1.41\) and \(1.62)\). Figure (16) shows the variation of the average Nusselt number with the heat flux for constant value of coolant inlet temperature \((T_{in}=40^\circ\text{C})\) and different values of extension ratio \((L^*=1.2, 1.41\) and \(1.62)\). It is observed from these figures that:

a) The average Nusselt number increases with the increase of the coolant inlet temperature.

b) The average Nusselt number increases with the increase in the extension ratio for all values of the coolant inlet temperature.

c) The average Nusselt number increases with the increase in the extension ratio for all values of heat flux.

Figure (17) shows the variation of the average Nusselt number, \(\text{Nu}\), with the modified Rayleigh number, \(\text{Ra}^*\). An empirical correlation equation is deduced to fit the present experimental measurements, which is given by \(\text{Nu}=1.778 \text{ Ra}^* 0.2017\) (with maximum deviation of 15 %).

**COMPARISONS OF THE RESULTS**

Comparisons between the present experimental measurements with calculation made by Relap5 code and the published data are performed in this study. This comparison is illustrated in the following section Figure (20) gives a comparison between the present experimental measurements for the average heat transfer coefficient and Relap5 code calculations for different values of the heat flux and constant coolant inlet temperature. The values of experimental measurements of the average heat transfer coefficient are lower than those calculated by Relap5 code and the deviation between them decreases with the increase of heat flux. Figure (21) demonstrates the comparison between the present experimental measurements for the local distribution of the local clad surface temperature through the channel length and those calculated from Relap5 code for different values of the coolant inlet temperature. From Fig. (21), there is a good agreement (small deviation) between the experimental measurements and calculated results particularly at the first section from the inlet of the channel higher values of inlet temperatures. The values of experimental measurements of the average surface temperature are higher than those calculated by Relap5 code and the deviation between them decreases with the increase of heat flux from 2.9 % to 0.9 %. Figure (22) shows a comparison between the present experimental measurements for the heat transfer coefficient through the channel length and those calculated from the Relap5 code. The calculations are made for constant coolant inlet temperature and different values of the extension ratio. The values of the experimental measurements lie down those obtained by Relap5 code. The difference between them decreases with the increase of the heat flux; the deviation decreases by lower values of the heat flux to the higher value (\(\text{he} - \text{ht}/\text{he}\)) by 15.5% to 1.1%. 4.3.

**Comparison between the Experimental Measurements with the Published Data**

Figure (23) shows a comparison between the present experimental measurements and the published data obtained by Fedorov and Viskanta \(^{(12)}\) and Auletta, et al. \(^{(13)}\). This figure indicates the relation between the Nusselt number and the dimensionless quantity \((\text{Gr.Pr.b}^*)\). It is obvious from the figure that the present measurements compare favorably with the published data.
Fig. (6) Variation of the average Nusselt number with the heat flux for different values of the extension ratios ($L_e = 60$).

Fig. (7) Variation of the average Nusselt number with the heat flux for different $R_e$ values.

Fig. (8) Variation of coolant temperature difference with the heat flux for different values of the extension ratios ($L_e = 60$).

Fig. (9) Variation of the average Nusselt number with the coolant inlet temperature for different values of the extension ratios ($L_e = 60$).
CONCLUSIONS

An experimental study is performed for natural convection cooling through vertical channel to simulate the natural convection heat transfer occurring in the Plate Type fuel of Research Reactor. Three extension ratios (different heights of chimney), using different values of heat flux and different pool temperatures are considered. The effect of these parameters on the Thermal-hydurlic behavior of the reactor coolant during natural circulation mode is studied. Based on the present results, significant features can be drawn as follows:

1) The flow rate and heat transfer coefficient increase by increasing the extension ratio for the same heat flux.

2) The coolant and clad surface temperatures increase with the increasing the pool temperature for constant height of chimney.

3) The heat transfer coefficient is enhanced by adding an adiabatic extension acts as chimney to the cooling channel for constant heat flux and constant pool temperature.

4) It is found that increasing the heat flux increases the mass flow rate and enhances the heat transfer rate.

5) The coolant temperature difference decreases with increase of the pool temperature.

6) There is reasonable agreement between the experimental measurements and analytical results obtained here.

7) The values of experimental measurements of the average surface temperature are higher than those calculated by Relap5 code and the deviation between them decreases with the increase of heat flux from 2.9 % to 0.9 %.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>b</td>
<td>Width of channel</td>
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<tr>
<td>b*</td>
<td>Aspect ratio</td>
</tr>
<tr>
<td>w</td>
<td>Depth of channel</td>
</tr>
<tr>
<td>h</td>
<td>Heat transfer coefficient</td>
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<tr>
<td>h₁</td>
<td>Heat transfer coefficient per segment</td>
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<tr>
<td>I</td>
<td>Current through the heater</td>
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<tr>
<td>k</td>
<td>Thermal conductivity of the coolant</td>
</tr>
<tr>
<td>L</td>
<td>Total length</td>
</tr>
<tr>
<td>L_chim</td>
<td>Height of the chimney</td>
</tr>
<tr>
<td>L*</td>
<td>Extension ratio</td>
</tr>
<tr>
<td>m</td>
<td>Mass flow rate through the coolant channel</td>
</tr>
<tr>
<td>n</td>
<td>Number of segments of the channel</td>
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<tr>
<td>Q_{seg}</td>
<td>Power per segment</td>
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<tr>
<td>Q_{con}</td>
<td>Heat losses due to conduction</td>
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<tr>
<td>Q_{gen}</td>
<td>Heat generated in the electric heater</td>
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<tr>
<td>Q_{net}</td>
<td>Net heat exchange</td>
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<tr>
<td>Q_{rad}</td>
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<td>Q_t</td>
<td>Total thermal power</td>
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<td>q</td>
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REFERENCES