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Correlation of Convective Heat Transfer for Laminar Conditions on Hexagonal Sub-Channel in Bandung TRIGA Research Reactor using Computational Fluid Dynamics

Efrizon Umar[1], Nathanael Panagung Tandian[2], Anggoro Septilarso[3], Azmairit Aziz[1], Anwar Ilmar Ramadhan[4,*], Wan Hamzah Azmi[5]

[1] National Research and Innovation Agency, Bandung 40132, Indonesia
[2] Bandung Institute of Technology, Bandung 40132, Indonesia
[3] Nuclear Energy Regulation Agency, Jakarta 11120, Indonesia
[4] Department of Mechanical Engineering, Faculty of Engineering, Universitas Muhammadiyah Jakarta, Jakarta 10510, Indonesia
[5] Faculty of Mechanical and Automotive Engineering Technology, Universiti Malaysia Pahang Al-Sultan Abdullah, 26600 Pekan, Pahang, Malaysia

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ABSTRACT

The forced convective heat transfer in hexagonal vertical sub-channels has been carried out using computational fluid dynamics (CFD). The size and geometry of the hexagonal used in the research specifically replicate the fuel element arrangement of the Bandung TRIGA research reactor. The main objective of this research is to develop new correlations to predict convective heat transfer coefficients in developing regimes or laminar flow regimes in sub-channels, especially for Reynolds numbers ranging from 260 to 1,500. The correlation equation from this research has been compared to the correlation equation for the square sub-channel resulting from previous research. There are small differences, which are predicted to be caused by differences in model geometry. The simulation was done for various heat fluxes in the heater surface. It varied from 100 W/m^2 to 25,000 W/m^2 with an increase of (500-5,000) W/m^2 intervals. Based on the analysis results, a new correlation equation is proposed in the form of $Nu = 1.73(Gz)^{0.339}$ and applies in the range for Graetz numbers $75 \leq Gz \leq 15{,}400$.

1. Introduction

Convective heat transfer for upward flow is often found in practice, for example in heat exchangers or nuclear fuel rods arranged in square or hexagonal geometries. The activities of designing, operating and analysing a heat exchanger [1] or analysing heat transfer in a nuclear reactor core, both from research reactors and power reactors [2-7] require accuracy in predicting the convection heat transfer coefficient between the heat exchanger pipe or fuel element rod and working fluid [8-11].

To predict the thermal-hydraulic parameters in the TRIGA reactor core, many researchers have carried out a lot of research, both theoretically [12-16] and experimentally [17]. For the TRIGA

* *Corresponding author.*
E-mail address: anwar.ilmar@umj.ac.id (Anwar Ilmar Ramadhan)

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Bandung research reactor, to study the thermohydraulic parameters of this reactor, the STAT and RELAP/SCDAPSIM/MOD3.4 computer codes have been used [6,18]. STAT calculates the natural convection flow through a vertical water coolant channel bounded by cylindrical heat sources [19]. The analysis results using STAT code show significant deviations compared to experimental data, especially related to the sub-channel exit temperatures [6]. To improve the analysis using STAT computer code, it is necessary to use another method, one of which is using three-dimensional analysis with computational fluid dynamics (CFD) which has been widely applied in various reactor [20-24].

In previous study, analysis using CFD has also been carried out, both for the initial core design [25] and for the modified core by opening the bottom cover [26]. The results of both studies show that the flow of the primary cooling system has a significant effect on the sub-channel inlet velocity of the reactor core [27]. The inlet velocity provides a forced convection effect on the sub-channel even though it is still in the laminar flow and causes the flow in the sub-channel to still be in the developing regime.

Forced convection heat transfer research in sub-channels for turbulent flow and for fully developed regimes have been widely discussed in the literature, both for a triangular sub-channel [28] and a square sub-channel [29-34]. However, for hexagonal sub-channels with laminar flow regime and for developing regime it is still limited.

In this research, simulation of forced convection heat transfer in the hexagonal sub-channel has been carried out by using CFD code. The new correlations for the convective heat transfer coefficients in developing regimes or laminar flow regimes, especially for Reynolds numbers ranging from 260 to 1,500 has been also proposed. The new correlation that has been obtained can be used to improve existing computer code for thermohydraulic analysis of research reactors.

2. Methodology

2.1 Model Geometry

The arrangement of several vertical cylinders in a hexagonal configuration can simulate a set of nuclear reactor fuel rods. So, in this case, the cylinder replicates the fuel element of the Bandung TRIGA research reactor with a pitch per diameter ratio (*P/D*) of 1.16. This reactor has 5 fuel elements equipped with temperature-measuring instrumentation and 5 control rods equipped with fuel elements at the bottom (fuel follower control rod). Meanwhile, the total number of fuel elements that can be placed in the reactor core is 116. The reactor core configuration of the Bandung TRIGA research reactor is shown in Figure 1.

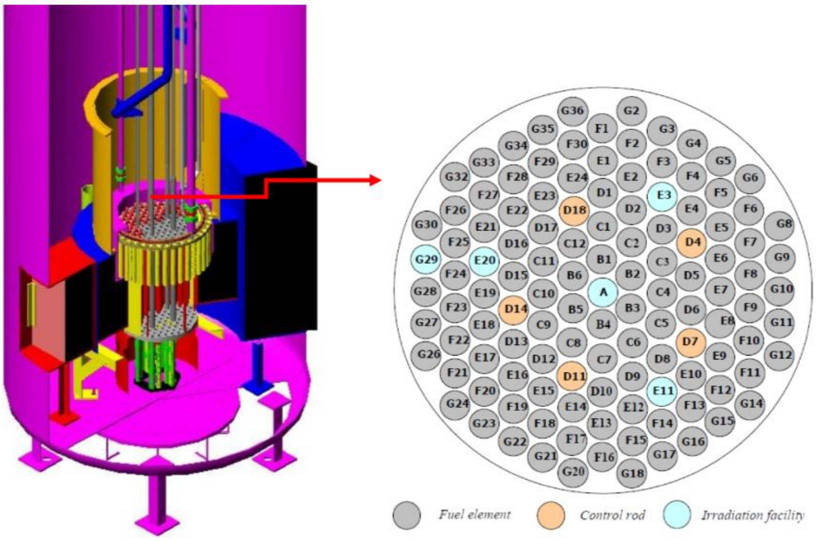


Fig. 1. The reactor core of the Bandung TRIGA research reactor [26]

While the schematic of the simulation test of the reactor core is shown in Figure 2.

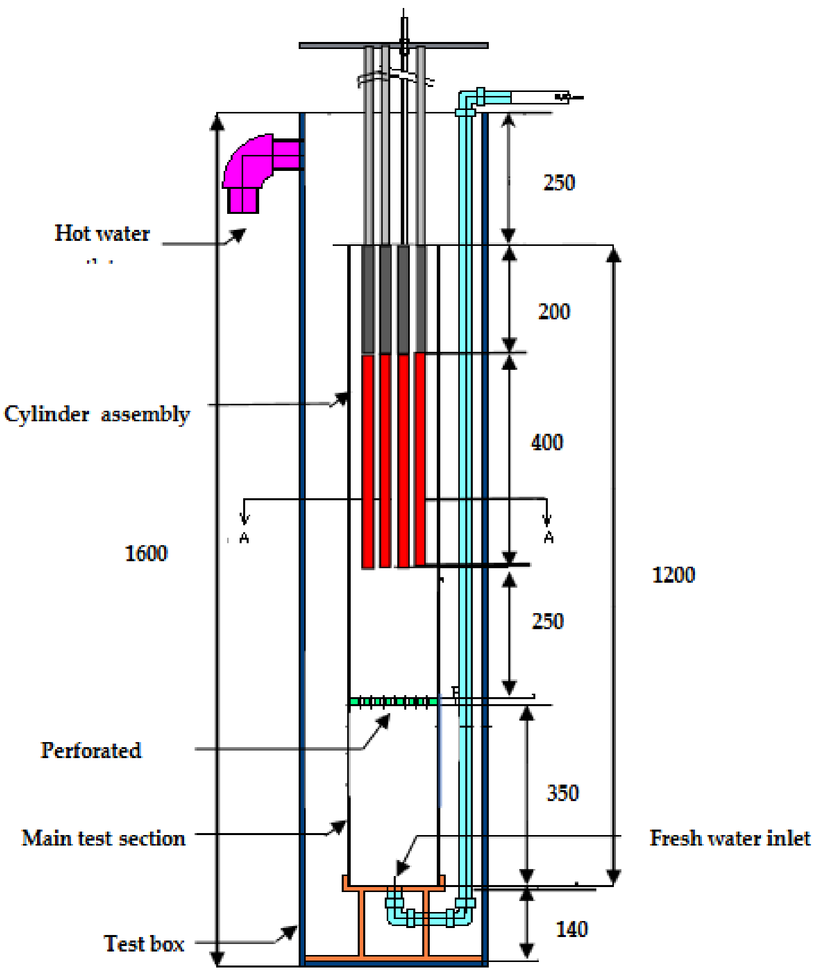


Fig. 2. A schematic of the simulation test section

The horizontal cross-section of the main simulation test section consisting of cylinder assembly and hexagonal test box is shown in Figure 3.

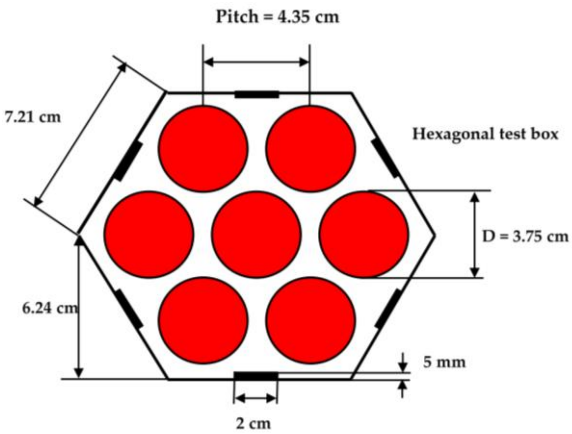


Fig. 3. Cross-section of simulated test section

During the forced convection simulation, the cooling water enters the inlet section, then flows downwards towards the bottom of the simulation test section, flowing upwards through the distributor plate, test chamber and hexagonal sub-channel. Finally, it comes out through the outlet.

The CFD software package is used for heat transfer analysis with the computational domain covering the volume of the simulated test section filled with cooling the water. Figure 4 illustrates the model from the vertical hexagonal sub channel, then the meshing done and the boundary conditions given in the numerical process using CFD.

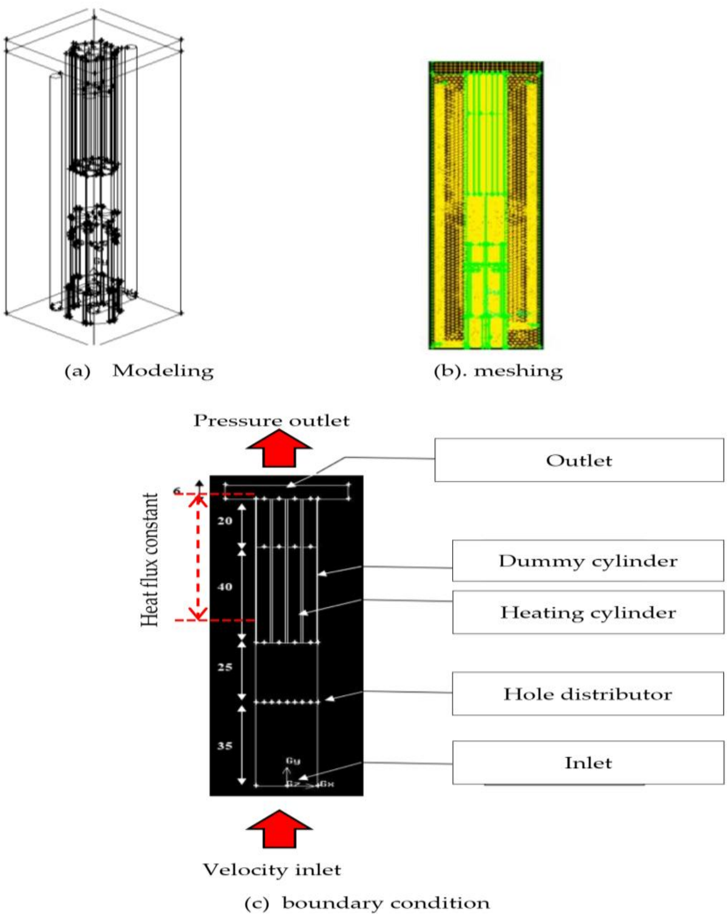


Fig. 4. CFD model and boundary conditions of hexagonal sub-channel

2.2 Boundary Conditions

The boundary conditions in this research are carried out with the following conditions:

i. The simulation is carried out after the steady state conditions have been achieved.
ii. The heat flux on the cylindrical surface is assumed to be constant and uniform.
iii. The pressure on the water surface in the simulation test is 1 bar.
iv. The water temperature entering the simulation test section is 300 K.
v. Gravitational acceleration is 9.8 m/s^2.
vi. The physical properties of water are obtained from the literature and its values depend on the temperature.

2.3 Heat Transfer Coefficient

The Newton's cooling law is used to calculate the local heat transfer coefficient on the surface of the heating cylinders model as shown in Eq. (1) [35].

$$q_c = h_c A (T_{surface} - T_{fluid}) \tag{1}$$

Based on dimensional analysis, the experimental data from forced convection heat transfer experiments in long channels can be correlated in the following Eq. (2):

$$Nu = \phi (Re)\, \Theta (Pr) \tag{2}$$

The convection heat transfer coefficient for forced convection flow in a circular tube can be determined using the Dittus-Boelter correlation [29,36,37].

$$Nu = 0.023\, Re^{0.8}\, Pr^{0.4} \tag{3}$$

Where,

Nu = Nusselt number = $\frac{h.D}{k}$
Re = Reynolds number = $\frac{\rho.V.D}{\mu}$
Pr = Prandtl number = $\frac{Cp.\mu}{k}$

For the channel through which fluid is not circular, the correlation of convection heat transfer is based on the hydraulic diameter. The hydraulic diameter (D_h) is defined as:

$$\boldsymbol{D_h = 4\frac{A}{P}} \tag{4}$$

Where, *A* is the flow cross section and *P* is the wetted perimeter. Meanwhile, Nusselt number for non-circular tube is shown in Eq. (5) [9,37].

$$Nu = \Psi(Nu)_{ct} \tag{5}$$

The value of the correction factor Ψ is determined by the *L/D* ratio. For flow in short channels, particularly in laminar flow, Eq. (2) is modified to include D_h/x or aspect ratio:

$$Nu = \phi\ (Re)\ \Theta\ (Pr)\ f\ (\frac{D_h}{x}) \quad (6)$$

Meanwhile, for laminar flow in sub-channels, the forced convection heat transfer correlation can be formulated as follows:

$$Nu = f\ (Re, Pr, Dh/x) \quad (7)$$

$$Nu = f\ (Gz). \quad (8)$$

The Graetz number (*Gz*) indicates the characteristics of laminar flow in the sub-channel.

3. Results

3.1 Grid-Independent Study

To determine the best mesh size, grid-independent tests have been carried out for mesh sizes of 0.1, 0.2, 0.3, 0.4 and 0.5 mm. After 2,286,436 element numbers, the difference between Nusselt numbers is determined to be less than 2.5 %. Therefore, the element number of 2,286,436 has been used to perform all the simulations. Variation of element number and Nusselt number is given in Table 1 and Figure 5. The same process for determining the grid independence test has been discussed by Umar *et al.,* [8] and other literatures [38,39].

Table 1
The results of grid independence study

Mesh size (mm)	Grid Element	*Nu*	Error *(%)*
0.1	1,811,804	114.063	50.542
0.2	1,918,680	107.413	41.771
0.3	2,052,379	97.075	28.121
0.4	2,123,097	87.805	15.887
0.5	2,286,436	77.352	2.091

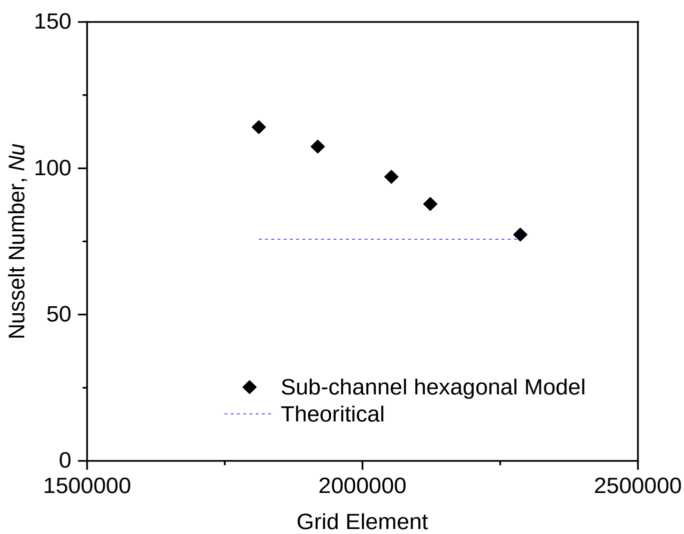


Fig. 5. Results of the grid independent testing

3.2 Validity of Numerical Solution

To ensure that the results obtained are correct when using a model with an optimal mesh size, a validation process is required. This validation process can be seen in Figure 6 where the Nusselt number increases with increasing Reynolds number (*Re*). In Figure 6, the theoretical results are also shown as black dotted lines. There seems to be a good agreement between the CFD results and their similarity to the equations.

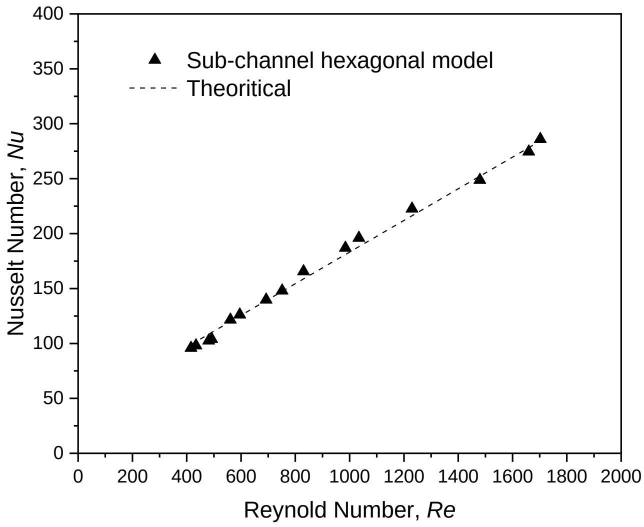


Fig. 6. Validation results of Nusselt number (Nu)

3.3 Temperature Distribution

By using computational fluid dynamics, a simulation to obtain the heater surface temperature has been carried out. The typical distribution of heater surface temperatures for a heat flux of 1,000 W/m^2 is shown in Figure 7. The colour bar on this figure shows the surface temperature values of heaters in K and at higher elevations, the heater surface temperature is also higher. The water temperature is also higher at higher elevations because when it flows upwards, the water receives heat from the heater.

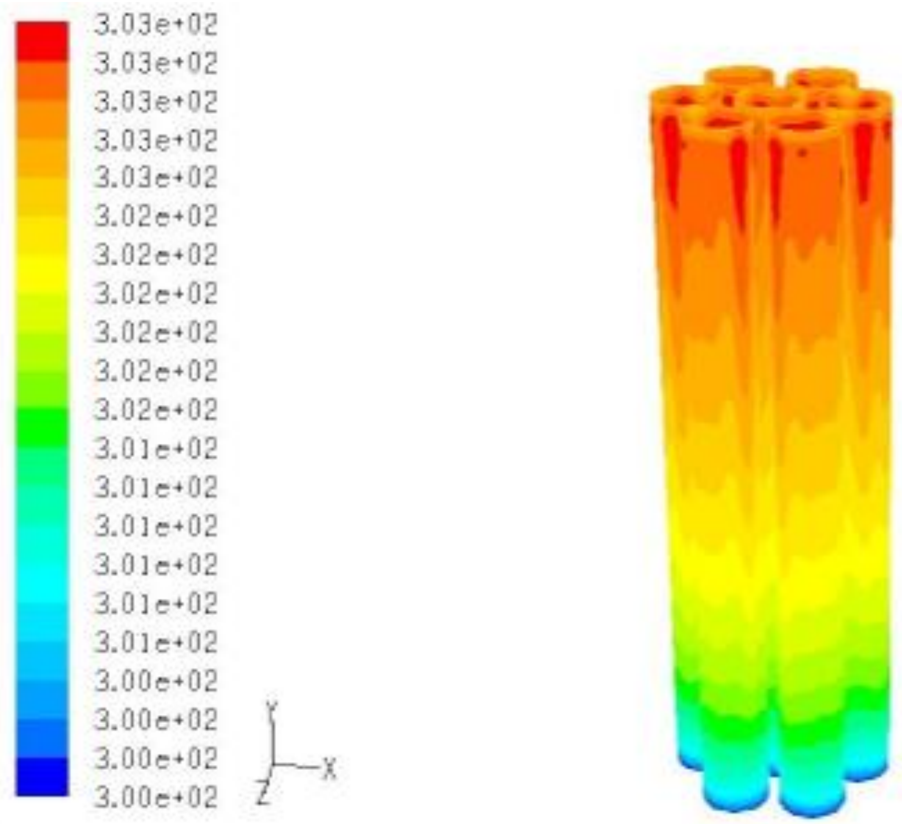


Fig. 7. The distribution of heater surface temperatures for heat flux of 1,000 W/m^2

Figure 8 shows the growth of the thermal boundary layer near the heating surface and the thickness of the thermal boundary layer increases with increasing elevation. This condition can be interpreted as the convective heat transfer coefficient also tends to be smaller at higher elevations.

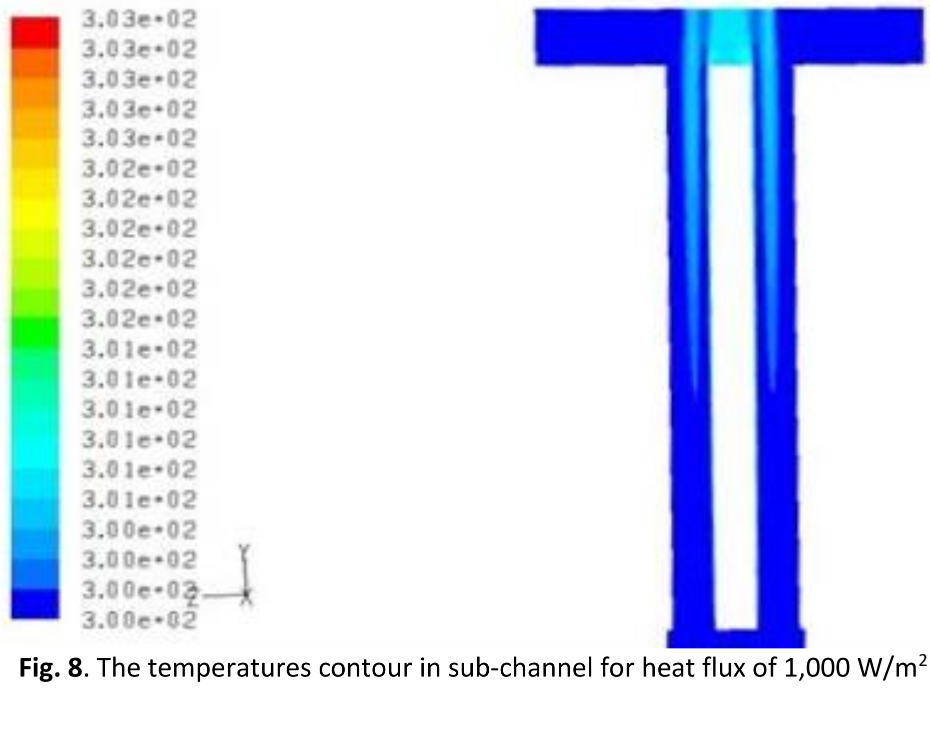
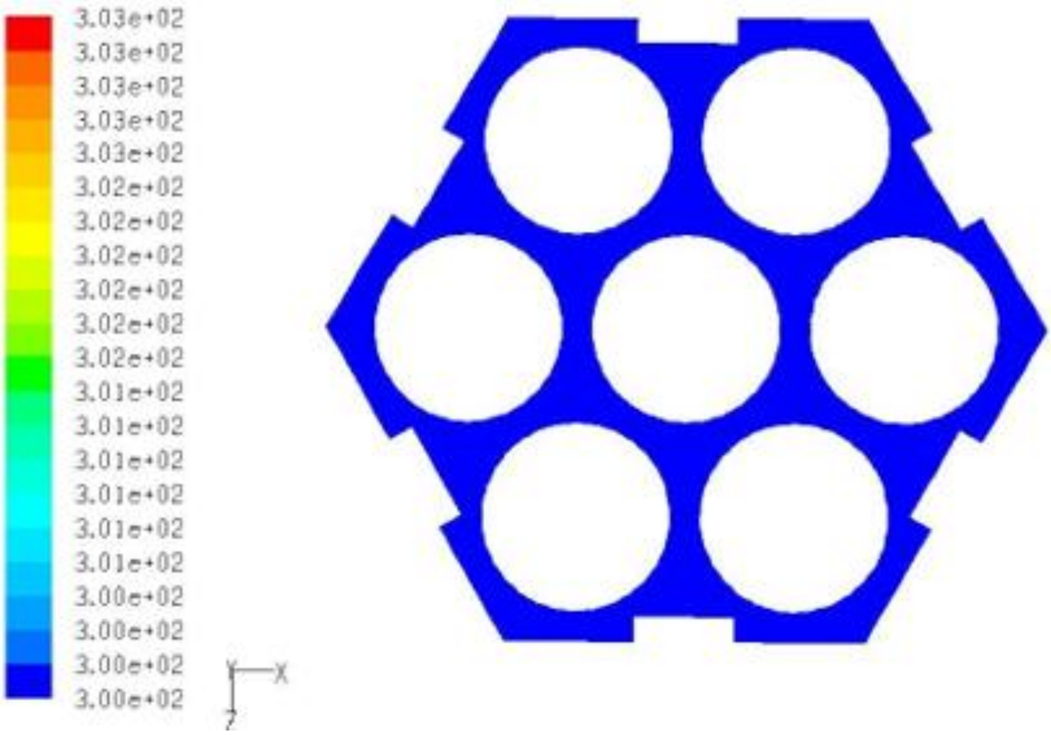
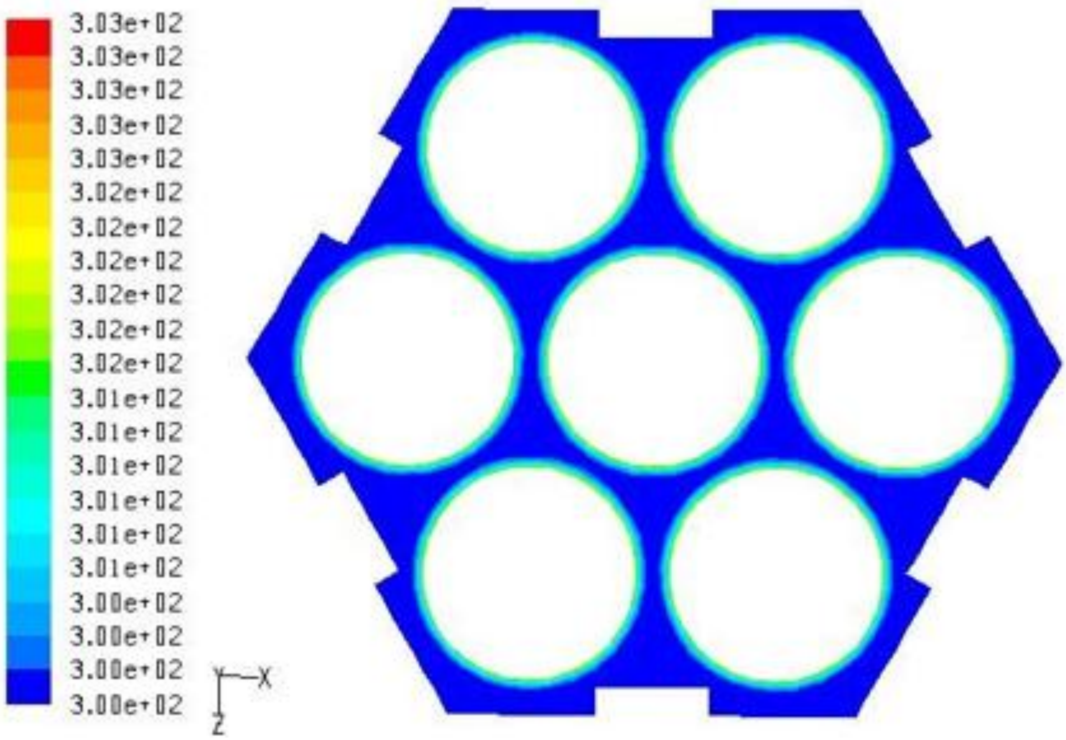


Fig. 8. The temperatures contour in sub-channel for heat flux of 1,000 W/m^2

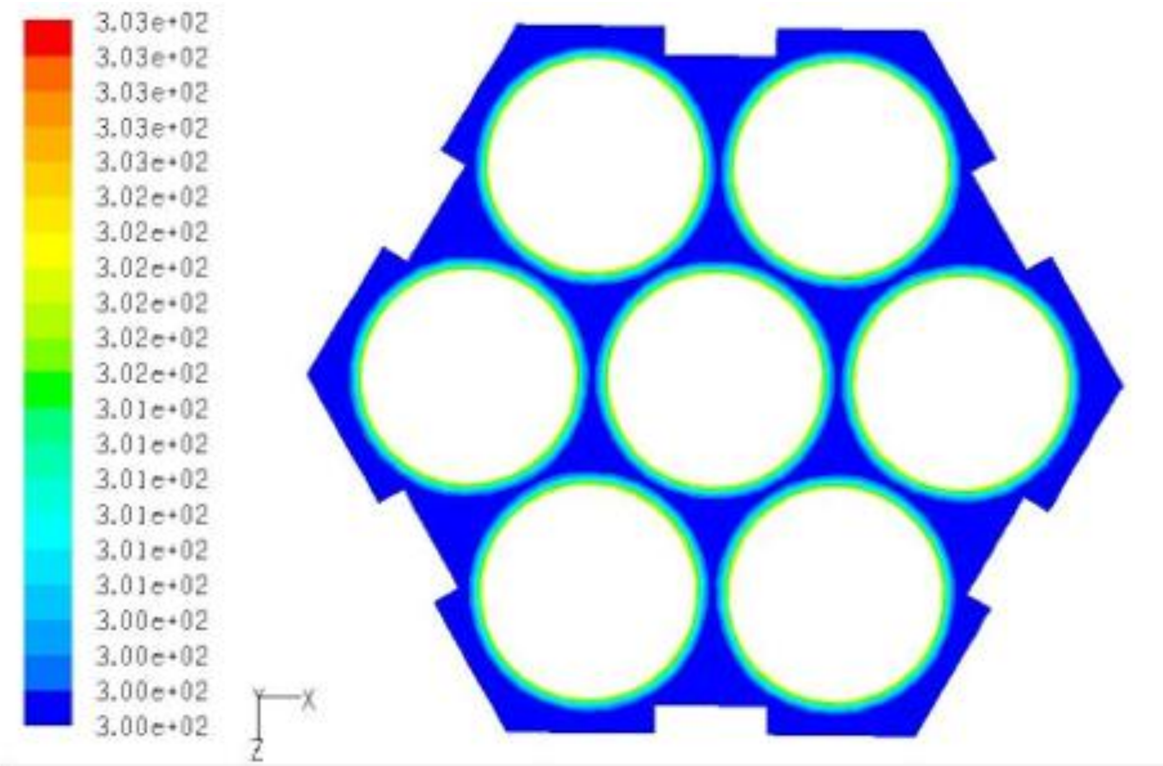
Figure 9(a) to 9(e) show the water temperature according to its tangential location and the water located between the two heaters has the highest temperature. This condition is caused by the waters in the region being less free because the water gap in the region is thinner than the water gap in other regions.



(a) y = 0.6 m



(b) y = 0.7 m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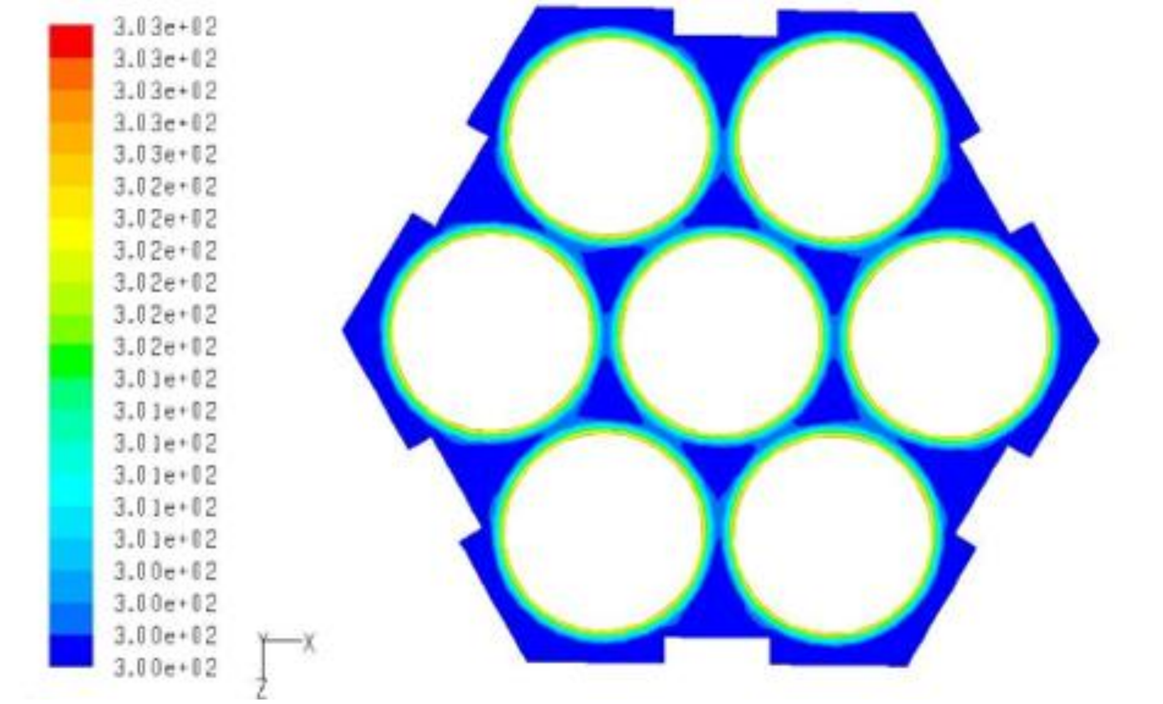
(c) y = 0.8 m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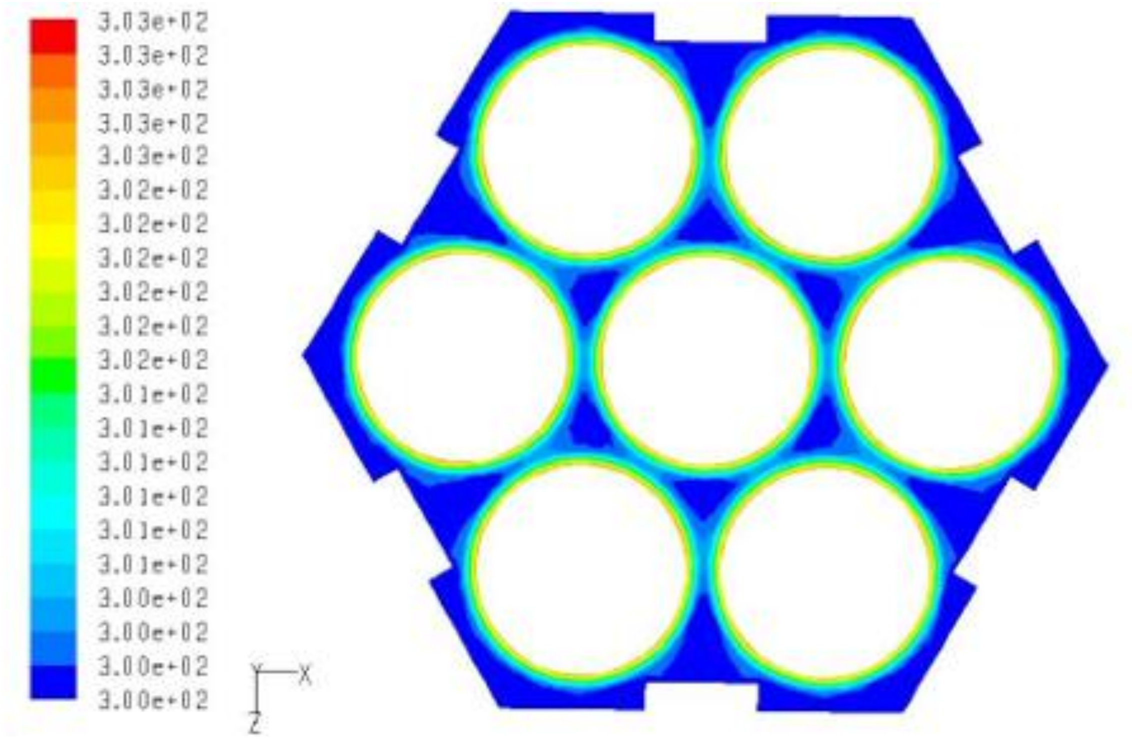
(d) y = 0.9 m



(e) y = 1 m

Fig. 9. The water temperature according to its tangential location for various elevations

For varying surface heat fluxes, the average water temperature at various sub-channel elevations is shown in Figure 10 and the water temperature tends to increase as the elevation increases. Besides that, Figure 10 also shows that the temperature gradient in the axial direction of the sub-channel decreases with elevation. This is because at higher elevation, process of heat transfer from the heating surface to the water penetrates deeper than at lower elevation so that heat is transferred and distributed to the water in greater quantities.

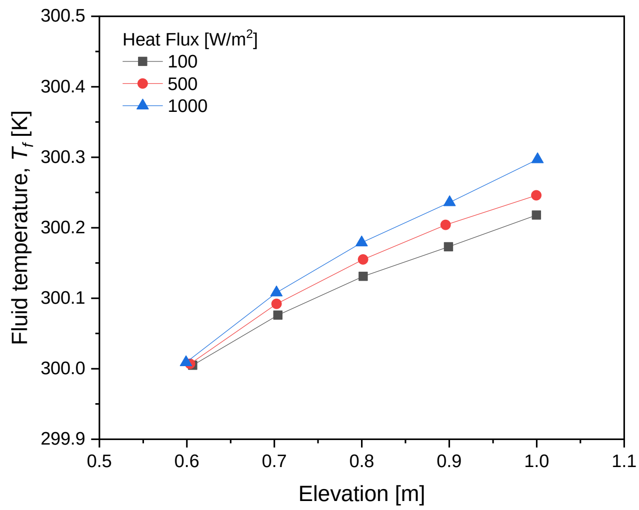


Fig. 10. The water temperature distribution at various heat fluxes

The surface temperature of the heaters, both for various heat fluxes and for various water inlet velocity are shown in Figure 11 and Figure 12, respectively. In both figures it can be seen that the surface temperature of heaters tends to increase with increasing elevation and heat flux but decrease with increasing water inlet velocity.

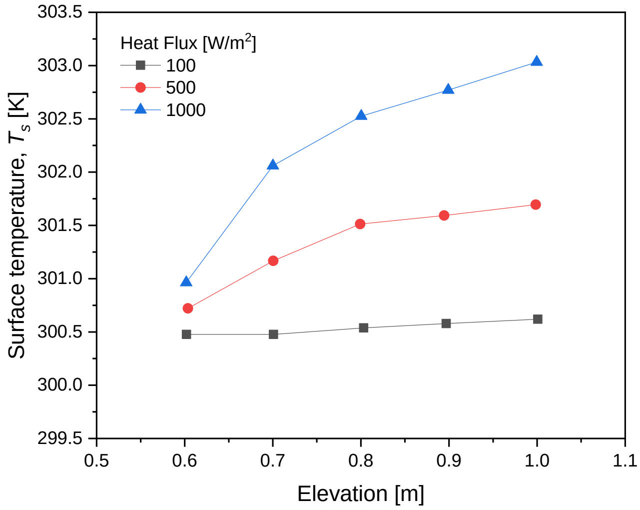


Fig. 11. The surface temperature of the heaters for various heat fluxes

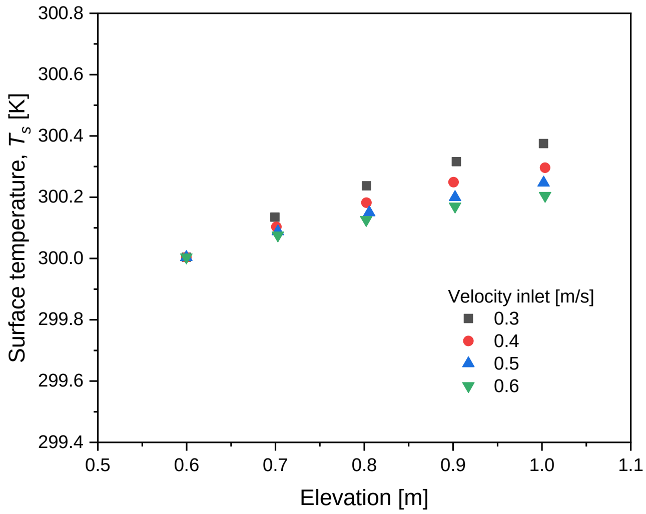


Fig. 12. The surface temperature of the heaters for various water inlet velocity

3.4 Heat Transfer Coefficient

For the Reynolds number range 260<*Re*<1,500, the convective heat transfer coefficient (h) of laminar flow is presented in Figure 13. In this figure, it can be seen that the difference in the convective heat transfer coefficients for various elevations is very large. Therefore, in this research, the laminar convective heat transfer coefficient used in the development of the correlation is the local heat transfer coefficient. This condition is an indication that the simulation range is still in the developing flow region.

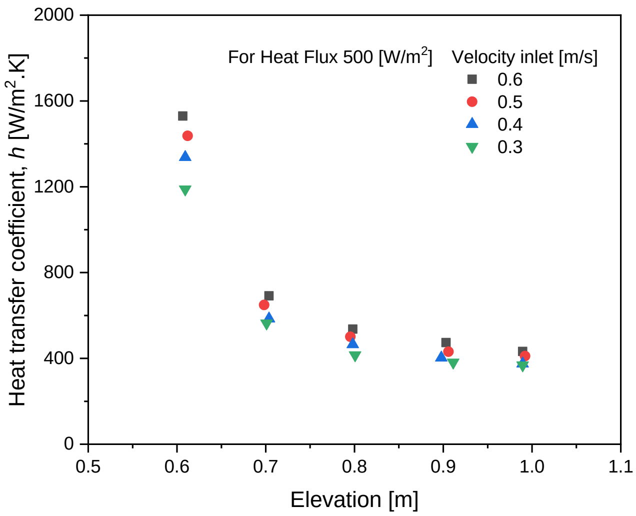


Fig. 13. The laminar convective heat transfer coefficient for varying water inlet velocity

3.5 Development of Heat Transfer Correlation

Eq. (1) is used to determine the local heat transfer coefficient on the heaters surface, while heat flux is one of the inputs in the simulation. Meanwhile, the heater surface temperature and water temperature for each elevation are observed and its values can be retrieved from the CFD software. The Nusselt number can be determined using Eq. (9), while the Graetz number uses Eq. (10). The physical properties of the water are evaluated at its bulk temperature.

$$\mathrm{Nu} = \frac{\mathrm{hD_h}}{\mathrm{k}} \tag{9}$$

$$\mathrm{Gz} = \mathrm{Re.\,Pr.}\frac{\mathrm{D_h}}{\mathrm{x}} \tag{10}$$

The correlation equation for the hexagonal sub-channel can be determined by plotting the simulation data as shown in Figure 14, then determining the correlation using linear regression.

$$\log(\mathrm{Nu}) = 0.339 \log(Gz) + 0.2389 \tag{11}$$

$$\mathrm{Nu} = 1.73\,(\mathrm{Gz})^{\mathbf{0.339}} \tag{12}$$

For applies in the range for Graetz numbers $75 \leq Gz \leq 15{,}400$.

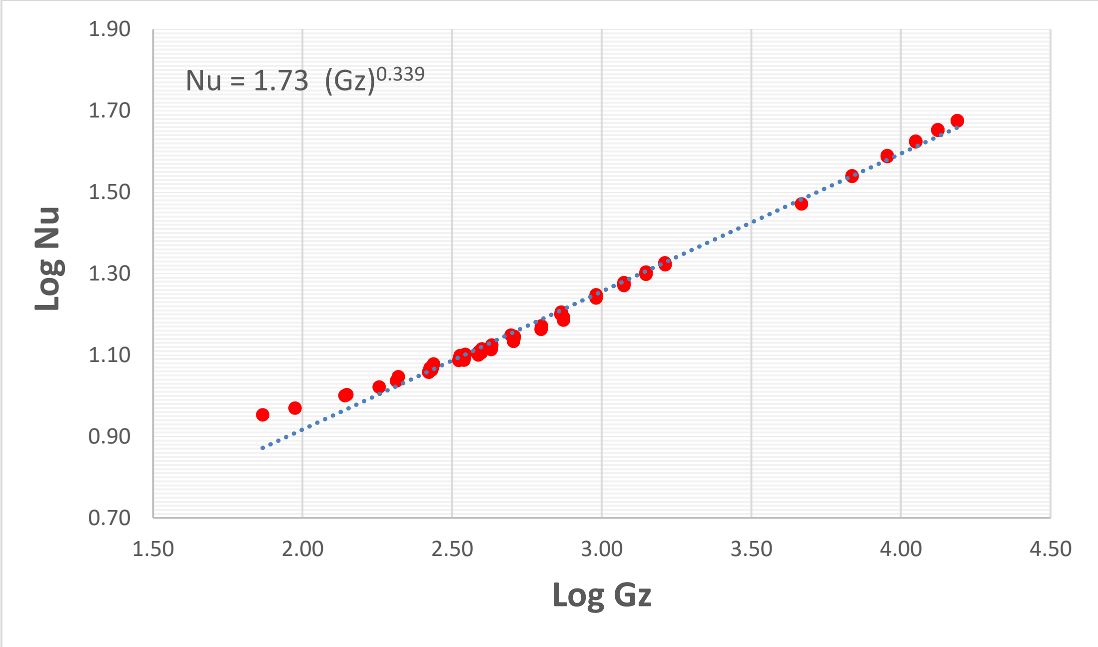


Fig. 14. The new forced convective heat transfer correlation for hexagonal sub-channel

Figure 14 shows the new forced convective heat transfer correlation in hexagonal sub-channel for laminar flow regime in the form of Nu = 1.73 (Gz)$^{0.339}$ and applies in the range for Graetz number range 75 < Gz < 15,400 and Reynolds number range 260<Re<1,700. The correlation equation resulting from this research has been compared to the correlation equation for the square sub-channel resulting from previous research [8]. There are small differences, which are predicted to be caused by differences in model geometry as shown in Figure 15.

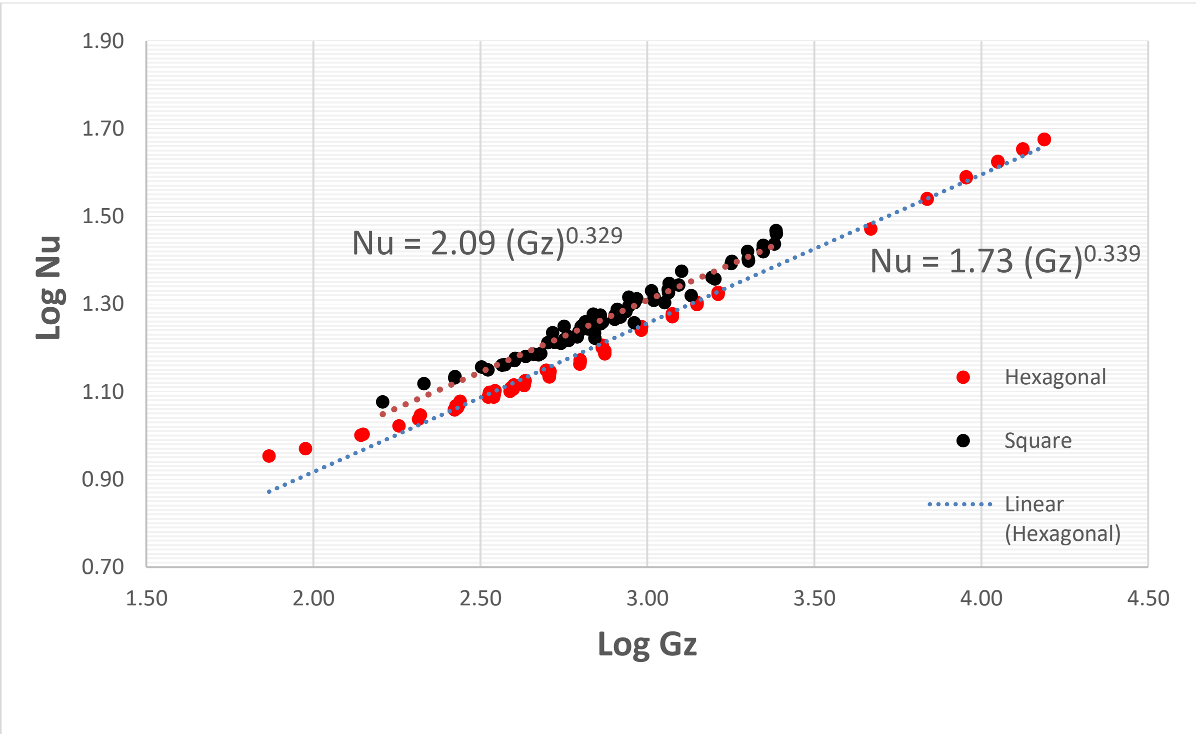


Fig. 15. Comparison of forced convective heat transfer correlation for hexagonal sub-channel and square sub-channel

4. Conclusions

The forced convective heat transfer simulation for the laminar flow regime has been performed in uniform heat fluxes. Based on the simulation and analysis result, it can be concluded that the flow simulated in this research is in a developing region or entrance region. In this study it also is proposed the new forced convective heat transfer correlation for the hexagonal sub-channel in the form of $Nu = 1.73\,(Gz)^{0.339}$ and applies in the range for Graetz number range $75<Gz<15{,}400$ and Reynolds number range $260 < Re<1{,}700$. The new correlation is hoped to be inserted into existing computer code as an alternative correlation in the thermohydraulic analysis of TRIGA-type research reactor.

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