Natural Convection Cooling for Spent Fuel of Nuclear Reactors

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ABSTRACT

Experiments were carried out to evaluate the thermal behavior of nuclear spent fuel stored in the auxiliary pool of Egypt Test Research Reactor No.2 (ETRR-2) during natural convection cooling condition (pump failure). An experimental test rig was designed and constructed to simulate this case. The test rig was equipped with two vertical and parallel electrically heated aluminum plates with a uniform heat flux. The test rig was designed to investigate the effect of the extension ratio ($L^* = L_t/L_h$), the aspect ratio ($B^* = W/b$) and the inlet coolant temperature (T_i) on the natural convection cooling behavior. Four different extension ratios (1.44, 1.94, 2.44 and 2.94) and four different channel aspect ratios of 2, 2.67, 4 and 8 were studied. The inlet coolant temperature was varied from 30 up to 45 ^oC. The present experiments were conducted within a range of modified Rayleigh number Ra^* from 1.2 $x10^5$ to 2.4 $x10^8$. The results showed that the heat transfer coefficient increases with the increase in either the extension ratio or the inlet coolant temperature, while it decreases with the increase in the aspect ratio. New correlation was obtained for the average Nusselt number as a function of the channel extension ratio L*, the channel aspect ratio B* and the modified Rayleigh number Ra*.

KEYWORDS: Natural convection, vertical channel, parallel plates, extension ratio, aspect ratio, spent fuel, nuclear fuel cooling.

1. INTRODUCTION

Natural convection phenomenon in a vertical rectangular channel is of practical importance and widely considered in electronics industry and nuclear power applications, especially in research reactors where heat dissipation in different modes of operation is an important task for safety requirements. The present study focuses on natural convection between iso-flux heated plates. The auxiliary spent fuel storage tank of the Egyptian Second Research Reactor (ETRR-2) is taken as a case study for the application of this phenomenon, under variation of some design parameters such as aspect and extension ratios as well as condition parameters like inlet coolant temperature.

Various investigators have been studied the heat transfer by natural convection from vertical plates with uniform heat flux. Wirtz and Statzman [1] investigated the case of two-dimensional natural convection between vertical plates subjected to uniform heat flux. They obtained experimentally the effect of changing of the heat flux (from 50 to 150 W/m^2) and space between plates (in such manner 7.94, 9.53, 12.7 and 17.78 mm). An empirical correlation relating Nusselt number and the modified Rayleigh number, Ra* was deduced. Haalland and Sparrow [2] investigated the case of natural convection flow in a vertical channel with a point heat source or distributed heat sources situated at the channel inlet. They also used a numerical solution for this case. Their results showed that a higher flow rate of fluid through a confined open-ended enclosure could be induced by the chimney effect. Oosthuizen [3] investigated numerically natural convection between isothermal parallel-walled channel by the addition of straight adiabatic extensions at the exit. The results indicated that there is a substantial increase in the heat transfer rates, and this increase could be achieved to 50 % using very long adiabatic sections. Sparrow, et al. [4] studied numerically the natural convection in a vertical channel using fluids with Prandtl numbers ranging from 0.7 to 10. Their results indicated that the increasing in the Prandtl number leads to the enhancement of the heat transfer rate through the channel. Wirtz and Haag [5] obtained experimentally results for isothermal symmetrical heated plates with an unheated entry channel portion. Their results showed 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. Miyamoto et al. [6] determined experimentally the heat transfer coefficient in case of turbulent natural convection flow in an asymmetrically heated vertical channel formed by two parallel plates. One plate was heated by imposing a uniform heat flux along the plate and the opposite plate was adiabatic. Experiments were performed using channel widths of 50, 100 and 200 mm. Their results showed that the decrease in channel widths leads to the enhancement of the heat transfer. Naito and Nagano [7] studied the case of combined forced and free convection for downward flow in the entrance region between inclined parallel plates with uniform wall temperature. The inclination angle was varied from 0^{o} to 90^{o}. It was found that the values of the local Nusselt number became smaller than those in the case of horizontal channel with increasing the inclination angles. Nelson and Wood [8] investigated theoretically the case of fully developed combined heat and mass transfer for natural convection between parallel plates with asymmetric boundary conditions. Their results showed that, the wall temperature and the velocity were independent of Prandtl and Grashof numbers. Naylor, et. al [9], studied numerically the case of free convection between isothermal vertical plates including entrance flow effects. Prandtl number was fixed at 0.7, Grashof number range was: $50 \leq Gr \leq 5x10^{4}$. A new phenomena, such as inlet flow separation, was observed and its effects on local Nusselt number were determined. Straatman, et. al [10], studied by using numerical and experimental methods the case of free convection in isothermal vertical channels. They investigated the case with the addition of adiabatic extensions of various sizes and shapes. They found that the adiabatic extensions increase the heat transfer process. The increase was varied from 2.5 at low Rayleigh number to 1.5 at high Rayleigh number. A new correlation in terms of the channel Rayleigh number and the geometric parameters as the aspect ratio, the expansion ratio and the heated length ratio was obtained. Fathalla and Amin [11] developed a theoretical model to predict the thermo hydraulic parameters for the First Egypt Test Research Reactor (ETRR-1) during natural convection mode. The model was verified experimentally by core measurements and the results agreed well for power up to 50 kW, while at higher reactor power, e.g. 100 kW the model over predicts the experimental results within 10 %. Tarasuk, et. al [12] demonstrated the case of free convection between inclined isothermal plates. The channel was investigated for single aspect ratio of 24, over the range $2.917 \leq Ra \leq 291.7$ and $0^{\circ} \leq \theta \leq 30^{\circ}$. The results showed that the overall heat transfer from the channel was reduced as the inclination angle increased. Also the overall channel flow rate decreases with the increase of the inclination angle. Campo et al. [13] presented numerical solutions to the wall temperature distribution and the thermal and fluid-dynamic fields in a channel with partially iso-flux heated parallel plates. The results showed that a reduction in the maximum wall temperature was observed when an insulated extension was placed downstream of the heated part. Shahin and Floryan [14] investigated numerically the case of upward flow natural convection between vertical parallel plates arranged periodically in the horizontal direction. The results showed that the increase in chimney extensions enhances the heat transfer and induces higher flow rate. The results showed also that the addition of a straight extension increases the mean Nusselt number by about 10% and the addition of an expanded extension increases the mean Nusselt number by about 20%. Talha [15] studied numerically the natural circulation mode in the ETRR-2 at 400 kW power. Three main reactor parameters; power, pool temperature and core outlet temperature were implemented in the study. The pool temperature and the core flow rate were also implemented. The results have been validated with experimental data from the commissioning stage and good agreements were found. Auletta, et al. [16] investigated experimentally the effect of adding adiabatic extensions downstream of a vertical iso-flux symmetrically heated channel. The results showed that the increase in the average channel Nusselt number was of the order 10–20 % depending on the channel aspect ratio and the imposed wall heat flux. Empirical correlations relating the average Nusselt number, the maximum dimensionless wall temperatures and the modified Rayleigh number, Ra* were deduced. Abd El-Latef, et. al [17] investigated experimentally the effect of changing the extension ratio (from 1.2 to 1.62), the heat flux (from 8.4 to 12.6 kW/m^2) and the inlet coolant temperature (from 30 ^oC to 50 ^oC) on the natural convection performance in a vertical channel between two parallel plates. An empirical correlation relating Nusselt number and the modified Rayleigh number, Ra* was deduced.

On view of the survey of the previous work, little researches investigated the effects of changing both the aspect and extension ratios with a wide range on the natural convection performance of a vertical channel between two parallel plates under uniform heat flux. Therefore, the objectives of the present study are to investigate the heat transfer by natural convection in a vertical channel between two electrically heated parallel plates and to establish the effects of changing the aspect and extension ratios and the inlet coolant temperature on the thermal performance.

2. EXPERIMENTAL APARATUS

A special test rig was built up for natural convection heat transfer measurements in a vertical channel as shown in Fig.(1-a). The test rig consists of a cooling water tank, a piping system, a test section and measuring instruments. Figure (1-b) shows a photograph for the experimental set up. The test rig was designed to perform the following capabilities:

i- Variable distances between the heating plates to investigate the effect of changing of the aspect ratio, ($B^* = W/b$).

ii- Variable water head above the heating plates to investigate the effect of changing of the extension ratio, ($L^* = L_t/L_h$)

iii- Variable inlet coolant temperature.

Details of the experimental apparatus are described in the following sections:

2.1. Cooling water tank and piping system:

The cooling water tank consists of a PVC cylindrical tank with dimensions of 3000 mm height and 205 mm diameter. It contains eight holes. Fresh coolant was feeding to the tank from the upper hole and the lower one was used for coolant inlet to the test channel. The over flow exits from one hole. Five holes were used for changing the height of the coolant above the test channel. The lower nozzle was a conical section varying from 205 mm to 25 mm and it was used for minimizing vortices in the inlet to the test channel. The tank was thermally insulated from surroundings. The level of the coolant above the test channel was changed by using piping system consists of six segments of pipes with diameter of 25 mm.

One segment leads the coolant to the test channel and five segments were used for changing the level of water above the test channel. Each one of the five segments has a valve used for governing the flow. Special rubber connections were used to connect the test channel with the coolant pipes. The piping system was thermally insulated from the environment.

2.2. Test section:

The test section was made of aluminum alloy ducts with rectangular cross-section. Four ducts were used with the same length of 1100 mm and depth (W) of 80 mm; while the width (b) of the ducts was changed as 40, 30, 20, 10 mm respectively. The ducts dimension leads to channel aspect ratios (W/b) of 2, 2.67, 4 and 8 respectively. Heat was supplied to the test section through only the two vertical side walls while the other two walls were insulated. The heat transferring walls were made of an aluminum alloy plates with dimensions of 900 mm length, 110 mm width and 10 mm thickness. A groove with dimensions of 800 mm length, 64 mm width and 6 mm depth was made in the back surface of each plate for fixation of the electrical heaters. Two main electrical heaters were used to heat the test section, one for each vertical plate. The electrical heaters were made of nickel chromium tape of 3 mm width and 1 mm thickness. The tape was turned around a sheet of mica with dimensions of 800 mm length, 64 mm width and 0.5 mm thickness with uniform pitches of 5 mm. The electrical heaters were sandwiched between another two mica sheets and then inserted in the grooves which were made at the back side of each aluminum plate followed by a sheet of asbestos with 4 mm thickness. Two similar heaters, which worked as guard heaters, were used to minimize the heat loss from the backside of the heated plates. A cover plate with dimensions of 900 mm length, 110 mm width and 1.5 mm thickness was used to cover and protect the heaters parts. The whole system was tightly fastened with through bolts to insure good contact. Figure (2-a) shows a schematic diagram for the test section.

2.3. Instrumentations:

The test section was instrumented with calibrated Chromel-Alumel thermocouples equally distributed along the mid-plane of the heating plates. Eighteen thermocouples were used for measuring the surface temperature of the heating plates equally distributed in the axial direction as shown in Fig.(2-b). In addition, nine thermocouples which could be traversed horizontally were used for measuring the coolant temperature equally distributed in the axial direction. Also, two thermocouples were used for measuring the inlet and outlet test channel temperatures. These thermocouples were made from wires of 0.3 mm diameter and connected to a multi channel digital thermometer accurate to 0.1 ^{o}C. Electric power was supplied via voltage regulators to provide uniform heat fluxes for the heating plates. Also, a digital multimeter was used to measure the electric resistances, currents and voltages of the main and guard heaters, respectively.

3. DATA ANALYSIS

The coolant at ambient temperature ($T_{f,i}$) enters the test channel from the bottom, moving upward due to buoyancy effect and leaves the channel at an outlet temperature ($T_{f,o}$). The hot coolant at the outlet of the test channel was mixed with the cold coolant in the cooling tank. The inlet coolant temperature was controlled by governing the amount of the cooling water circulated. The inlet coolant temperature was changed from 30 up to 45 ^{o}C. The extension ratio (total coolant height/heated length) was varied as 1.44, 1.94, 2.44 and 2.94 while the values of the channel aspect ratio (depth of channel/width of channel) were 2, 2.67, 4 and 8. The test channel was divided into nine axial test locations. Each test location was located by a thermocouple position. After reaching steady state, the coolant and the heating surface temperatures were recorded. The power input to the heaters was computed from the measurements of the voltage and resistance of the main heaters and then the applied heat flux was calculated. It was found that the heat lost by radiation to the surroundings and by conduction through the insulation layers is small enough to be neglected within the tested range of parameters. Knowledge of the heat flux q" and the local surface temperatures of the heating plates allowed for the computation of the local heat transfer coefficient h_x as:

$$h_x = \frac{q''}{T_{W,x} - T_{f,x}}$$

Where, q" is the net heat flux

$T_{w,x}$ is the local surface temperature of the test plates

$T_{f,x}$ is the local flow bulk temperature and it was calculated by applying the heat balance for the test channel segment as;

$$T_{f,x} = T_{f,x\text{-}\Delta x} + \frac{q''(W.\Delta x)}{m.c_P}$$

Where, Δx is the test channel segment length. The value of the measured inlet flow temperature was assigned to $T_{f,i}$ for the first channel segment.

The local Nusselt number based on the heating channel width (b) was calculated simply by, $Nu_x = h_x\ b/k$ and then the average Nusselt number (Nu) was obtained. Rayleigh number (Ra) was calculated based on the heat flux, Incropera and David [18], for each segment and then the average value was obtained by:

$$Ra = \frac{g \times \beta \times q'' \times b^4}{k \times \alpha \times \nu}$$

Where, α is the thermal diffusivity of the coolant [$\alpha = k / (\rho \times C_p)$]

β is the volumetric thermal expansion coefficient.

The modified Rayleigh number (Ra*) was calculated as: $Ra^* = Ra \times b/L_h$

Where, L_h is the heated length.

The flow mean temperature $(T_{f,i}+T_{f,o})/2$ was taken as a reference at which the coolant properties were obtained.

An uncertainty analysis was conducted to determine the overall error in the calculated heat transfer coefficient due to uncertainties in the measurements of the voltage drop, electric resistance, heating surface area, surface and flow mean temperatures. The differential method was accomplished and a total of 2% average uncertainty in the heat transfer coefficient was found.

4. RESULTS AND DISCUSIONS

The present measurements were carried out to study the effects of the extension ratio, the aspect ratio and the inlet coolant temperature on the natural convection heat transfer between two vertical parallel plates heated with a uniform heat flux.

4.1. Effects of the extension ratio, L^*

The effects of the extension ratio for different aspect ratios, (B^* =2, 2.67, 4 and 8), at fixed inlet coolant temperature (T_i = 40 °C) and the effects of the extension ratio for different inlet coolant temperatures, (T_i = 30, 35, 40 and 45 °C), for an aspect ratio of (B^*=8) are shown in Figs.(3 and 4) respectively. The results show that the average heat transfer coefficient almost increases with the extension ratio for the different aspect ratios at different inlet coolant temperatures. The increase in the extension ratio leads to an increase in the height of the unheated coolant length which enhances the buoyancy effect and increases the coolant velocity. This leads to an enhancement in the heat transfer coefficient. The effects of the increase in the extension ratio (L^*) on the average Nusselt number Nu are shown in Figures (9 and 10). It is observed that the average Nusselt number increases with the increase in the extension ratio for all values of the aspect ratios and the inlet coolant temperatures.

4.2. Effects of the aspect ratio, B^*

The effect of the aspect ratio on the thermal performance at different extension ratios, (L^* = 1.44, 1.94, 2.44 and 2.94) and at T_i = 40 °C is shown in Fig.(5). Also the effect of the aspect ratio at different inlet coolant temperatures (T_i = 30, 35, 40 and 45 °C) for extension ratio of (L^* = 1.44) is illustrated in Fig. (6). The results indicate that the average heat transfer coefficient decreases with the increase in the aspect ratio for different extension ratios at different inlet coolant temperatures. The increase in the aspect ratio is due to the decrease in the channel width (B^* = W/b), this decrease the flow passage and increases the flow friction. This increase in the flow friction decreases the flow velocity for the limited buoyancy effect. Thus leads to an unexpected reduction in the heat transfer coefficient as shown in Figs.(5,6). The average Nusselt number versus the aspect ratio at different extension ratios is shown in Fig.(11) while Fig.(12) shows this effect at different inlet coolant temperatures.

4.3. Effects of the inlet coolant temperature, (T_i)

Figure (7) shows the influence of the inlet coolant temperature on the thermal performance for different extension ratios at a fixed aspect ratio of (B^* = 8). The results for different aspect ratios at an extension ratio of (L^* = 1.44) are shown in Fig.(8). It is clear that the heat transfer coefficient increases with the increase in the inlet coolant temperature. This is due to increase in the average coolant temperature, which tends to a decrease in the coolant viscosity and then increase the average heat transfer coefficient. The same trend was observed for the variation of the average Nusselt number versus the inlet coolant temperature as shown in Figs.(13 an 14) for different extension and aspect ratios. It is observed that the average Nusselt number slightly increases with the increase in the inlet coolant temperature for different extension and aspect ratios.

The Nusselt number was correlated as a function of the modified Rayleigh number Ra^*, the extension ratio L^* and the aspect ratio B^* and the following correlation was obtained:

$$\mathbf{Nu = 1.675\ Ra^{*0.209}\ L^{*0.0821}\ B^{*-0.0107}}$$

Such correlation may be helpful for the design of related devices such as electronic cooling systems and nuclear power reactors. Figure (15) illustrates the correlated Nusselt number versus the experimental data. It is shown that the calculated results exhibit a relatively good agreement with experimental results. The correlation is valid within ± 9 % maximum deviations with the present experimental data for the investigated ranges of different parameters, ($1.44 \leq L^* \leq 2.94$), ($2 \leq B^* \leq 8$), (30 °C $\leq T_i \leq$ 45 °C) and ($1.2 \times 10^5 \leq Ra^* \leq 2.4 \times 10^8$).

Figure (16) shows the comparison for the average Nusselt number Nu calculated from the present correlation at $L^* = 1.44$ and $B^* = 8$ and that given by Wirtz et. al [1] and Auletta et. al [16]. The comparison shows a good agreement between the present correlation and that previously published work.

5. CONCLUSIONS

The experiments conducted in the present work lead to the following conclusions:

1. The increase in the extension ratio leads to an increase in the buoyancy effect and enhances the heat transfer coefficient within the experimental range ($1.44 \leq L^* \leq 2.94$).
2. The increase in the aspect ratio increases the coolant velocity and the flow friction, while the heat transfer coefficient was decreased within the experimental range ($2 \leq B^* \leq 8$).
3. The increase in the inlet coolant temperature decreases the coolant viscosity and slightly enhances the heat transfer coefficient within the experimental range ($30\ ^oC \leq T_i \leq 45\ ^oC$).
4. An empirical correlation has been obtained for the average Nusselt number Nu as a function of the modified Rayleigh number Ra*, the extension ratio L* and the aspect ratio B*.

Nomenclature

SI units were used for the whole parameters within this paper.

b	heating channel width
c_p	specific heat at constant pressure
h	
k	
L	test-section length
m	
q"	
T	temperature
W	heating channel depth
x	

Subscripts:

f	for fluid
h	heating
m	mean value
i	inlet
t	total
w	wall, surface
x	local value
x-Δx	test segment inlet

Greek letters:

α	thermal diffusivity
β	coefficient of thermal expansion
Δ	
ν	kinematic viscosity
ρ	
θ	inclination angle

Dimensionless Terms:

B^*	aspect ratio (W/b)
L^*	extension ratio (L_t/L_h)
Gr	Grashof number
Nu	Nusselt number
Ra	Rayleigh number [eq. (3)]
Ra^*	modified Rayleigh number

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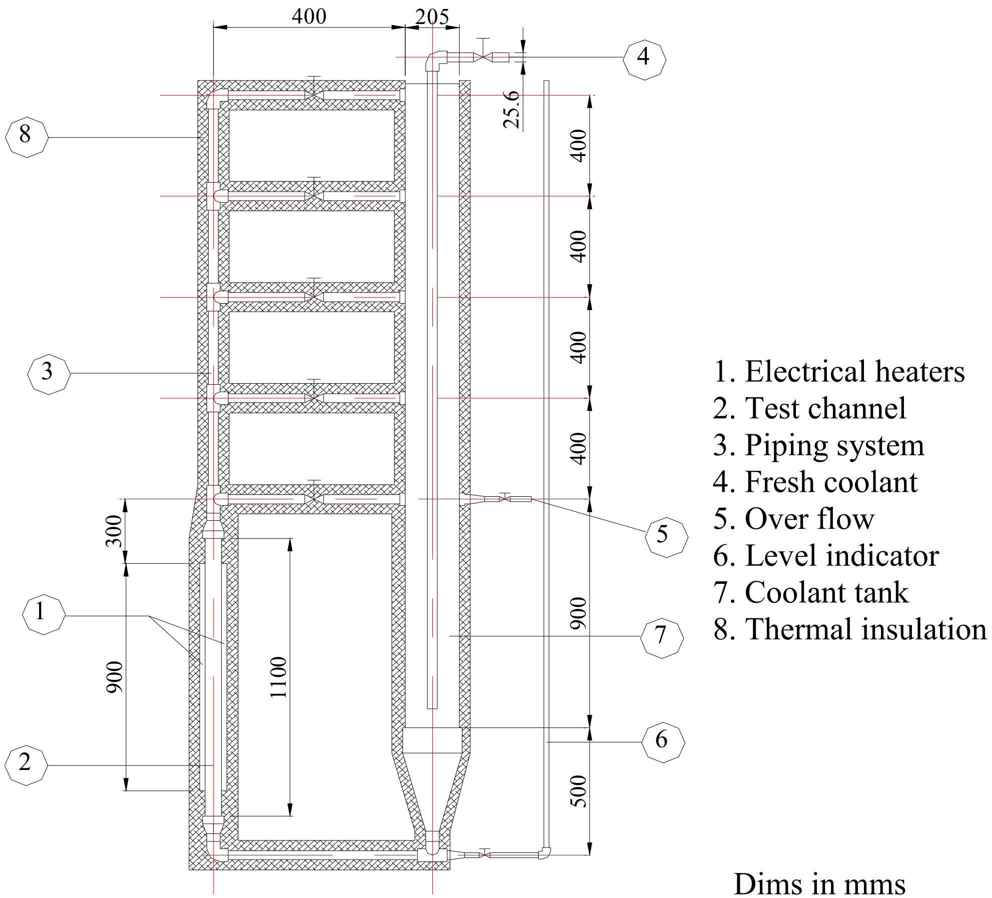
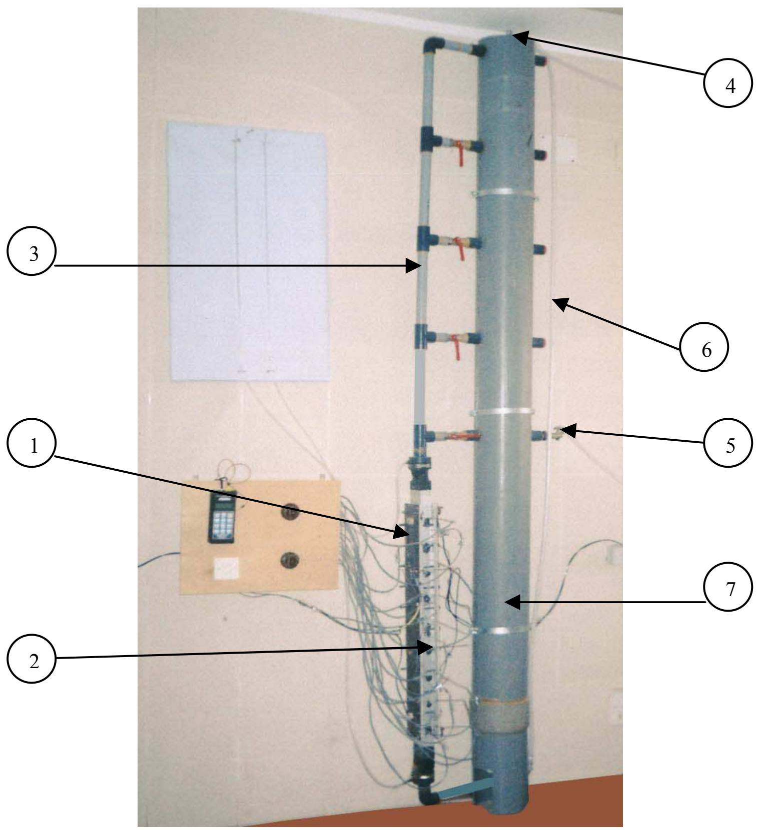
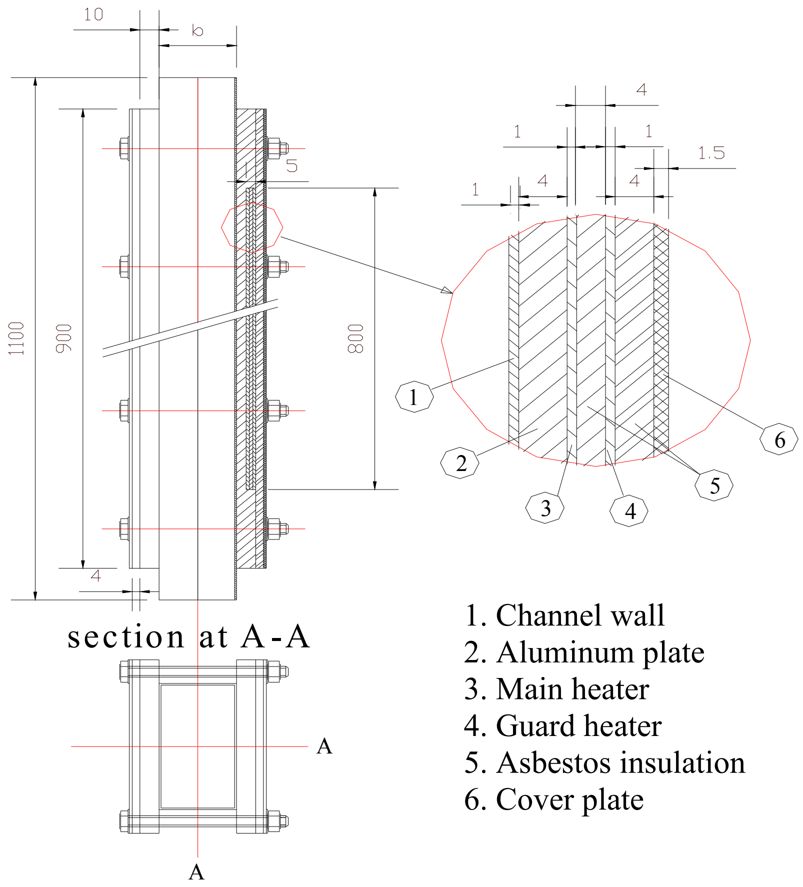


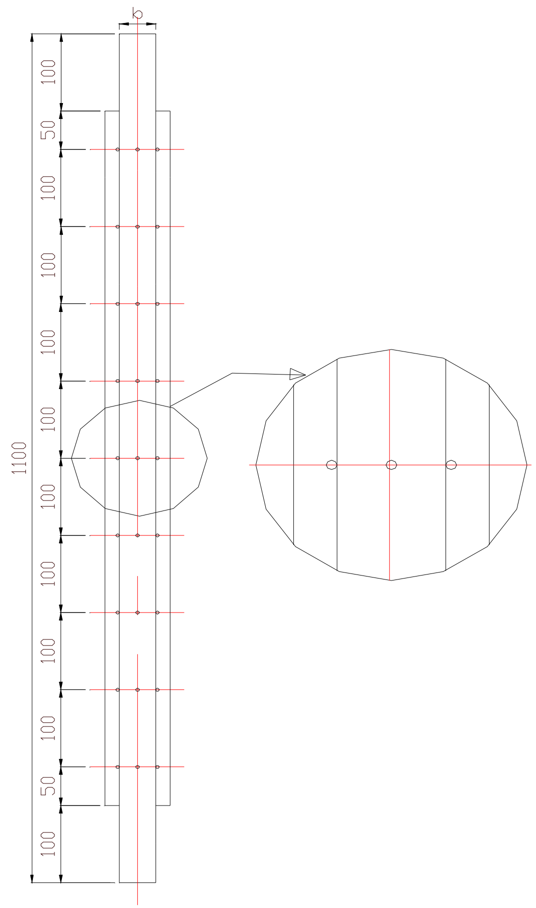
(a): Schematic diagram of the experimental setup



(b): Photograph for the experimental setup

Fig.(1): The experimental setup





Dims in mms

(a): Details of the test section

(b): Thermocouples distributions

Fig.(2): The test section

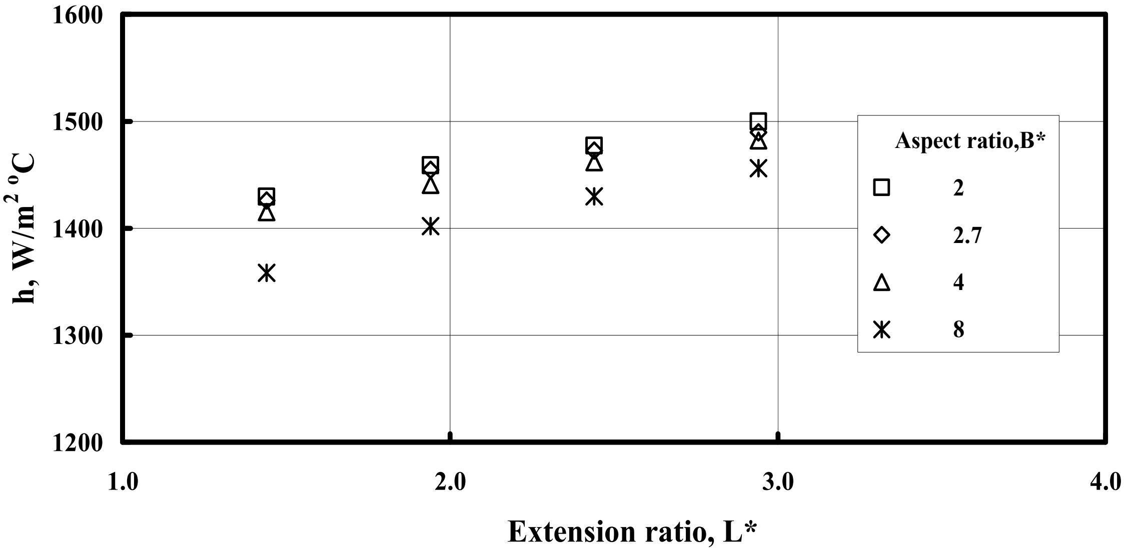


Fig. (3): The average heat transfer coefficient versus the extension ratio for different aspect ratios at inlet coolant temperature, $T_0 = 40\ ^0C$

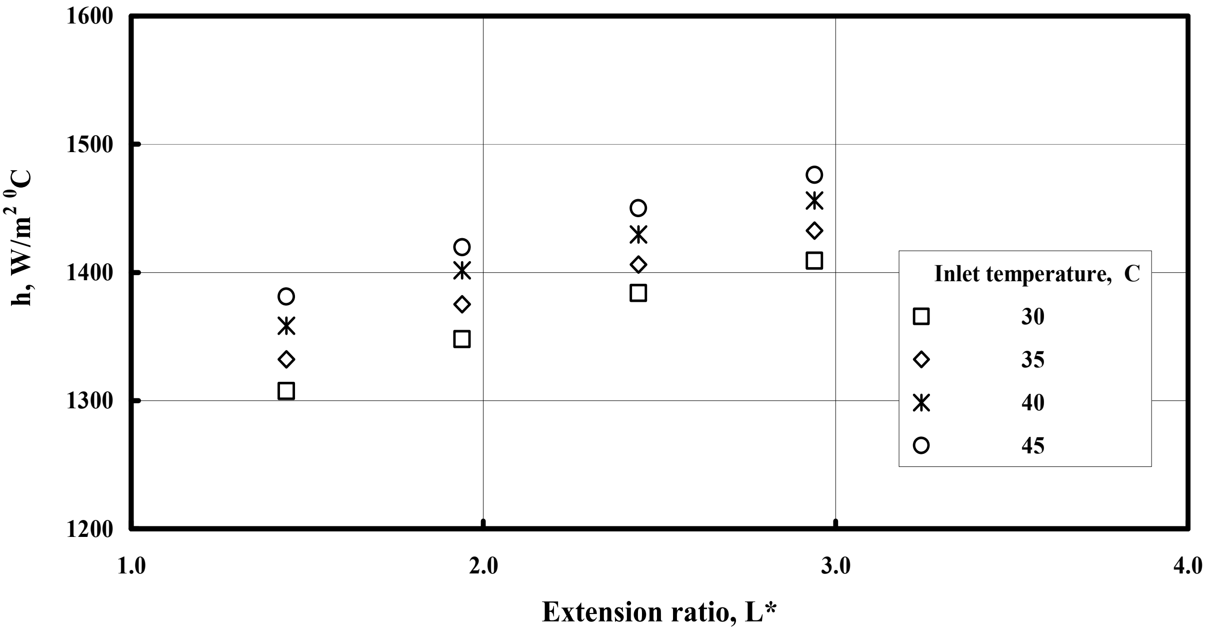


Fig. (4): The average heat transfer coefficient versus the extension ratio for different inlet coolant temperatures at aspect ratio, B* = 8

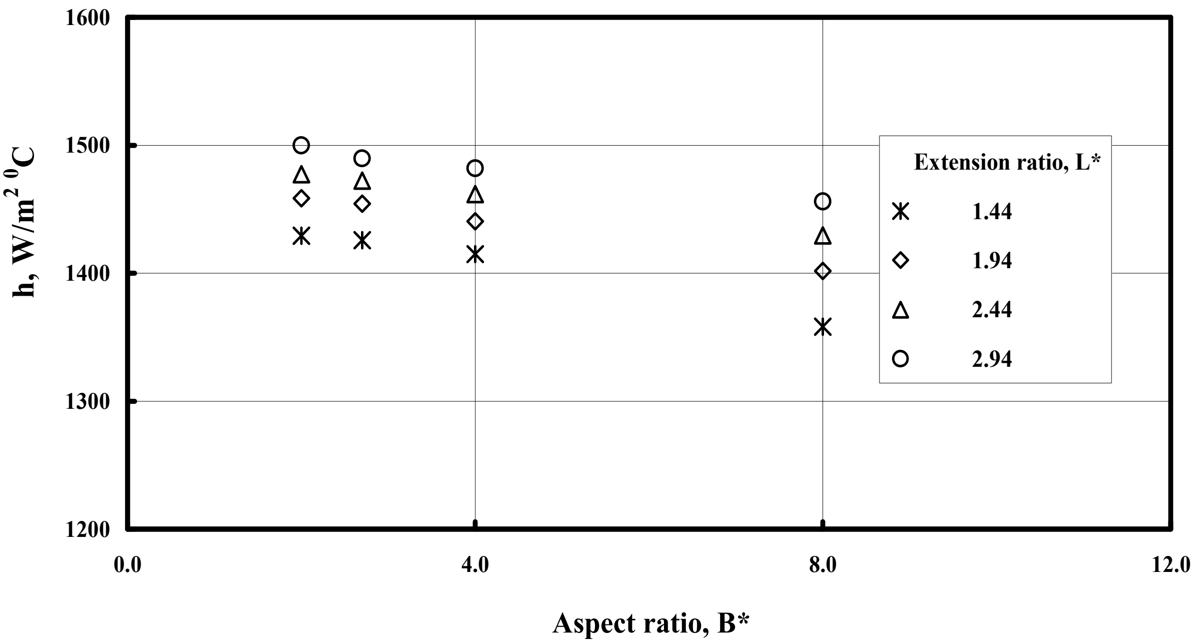


Fig. (5): The average heat transfer coefficient versus the aspect ratio for different extension ratios at inlet coolant temperature, $T_0 = 40\ ^0C$

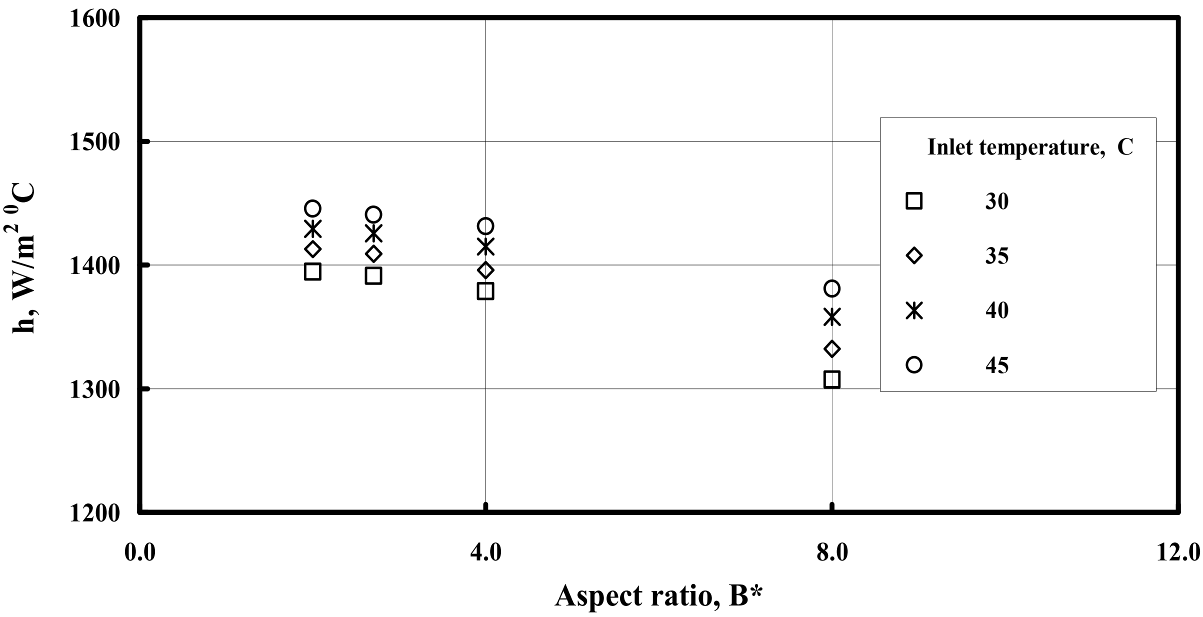


Fig. (6): The average heat transfer coefficient versus the aspect ratio for different inlet coolant temperatures at extension ratio, $L^* = 1.44$

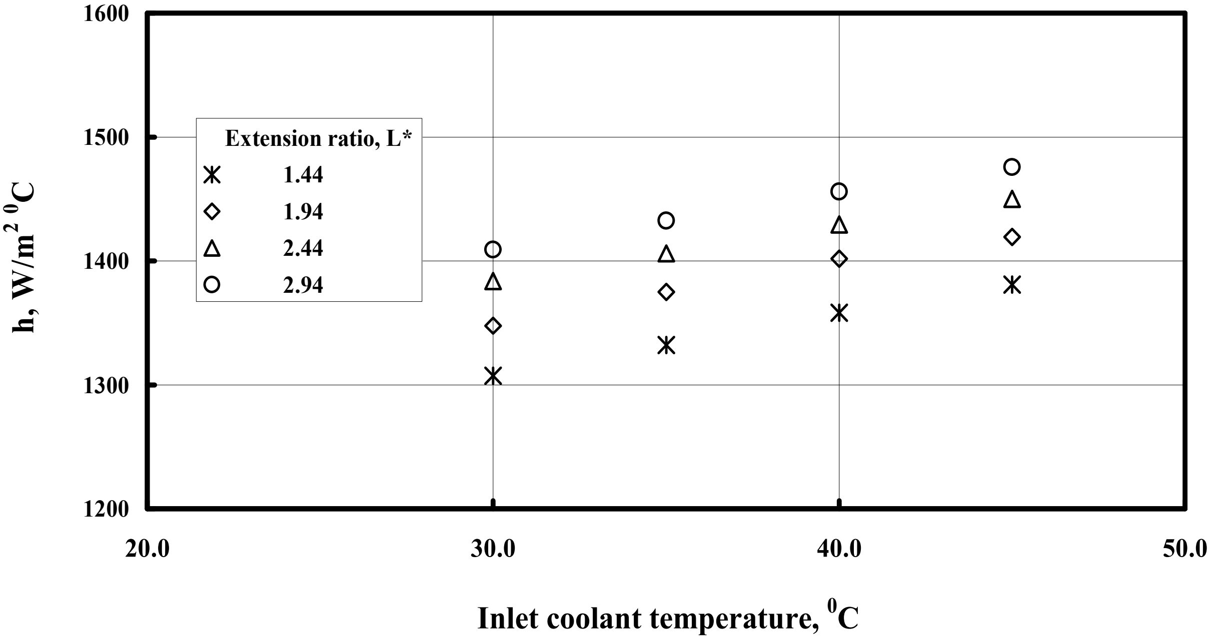


Fig. (7): The average heat transfer coefficient versus the inlet coolant temperature for different extension ratios at aspect ratio, B* = 8

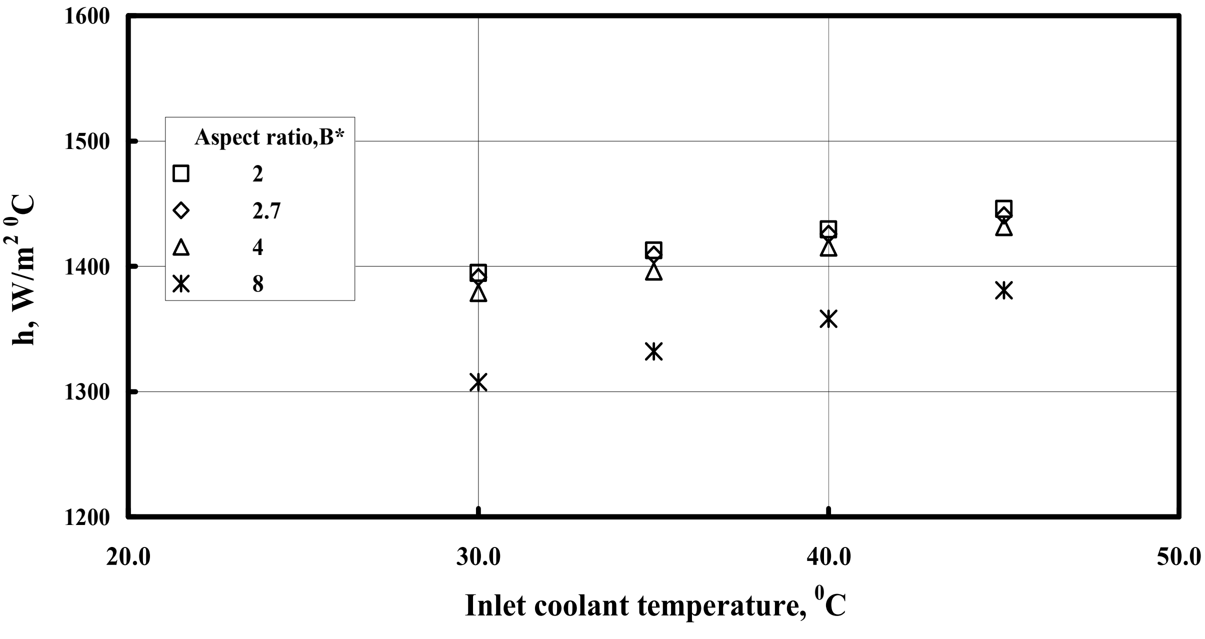


Fig. (8): The average heat transfer coefficient versus the inlet coolant temperature for different aspect ratios at extension ratio, L* = 1.44

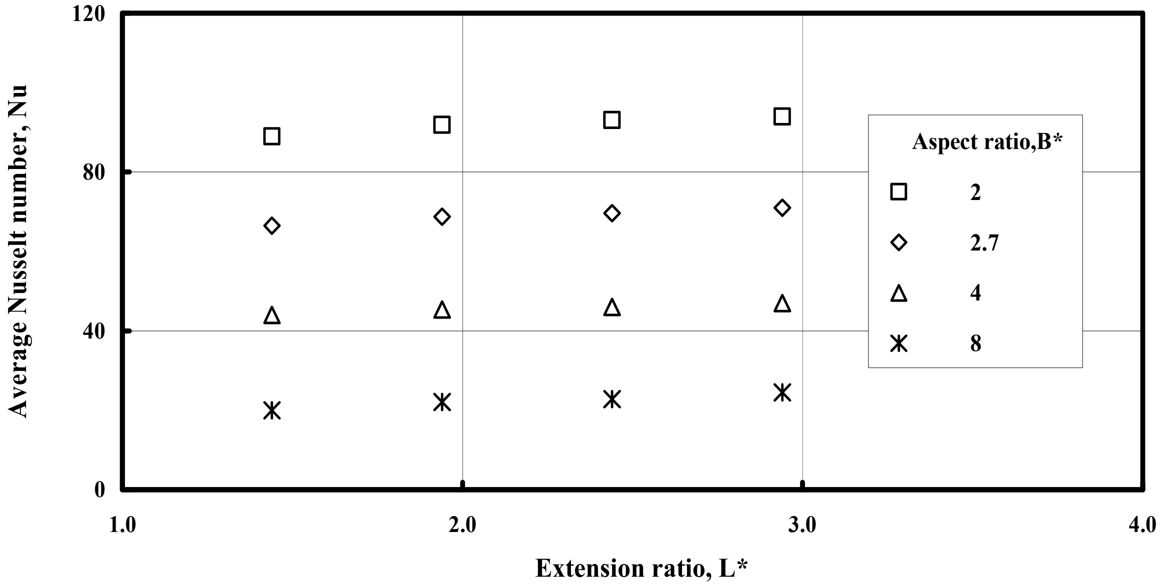


Fig. (9): The average Nusselt number versus the extension ratio for different aspect ratios at inlet coolant temperature, $T_0 = 40\ ^0C$

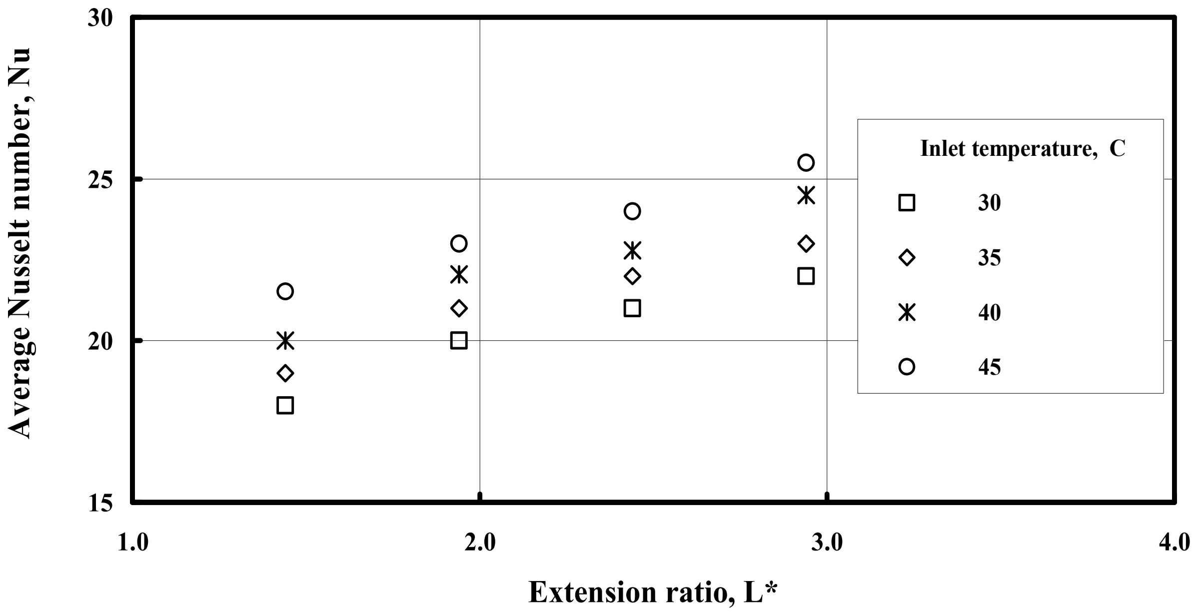


Fig. (10): The average Nusselt number versus the extension ratio for different inlet coolant temperatures at aspect ratio, B* = 8

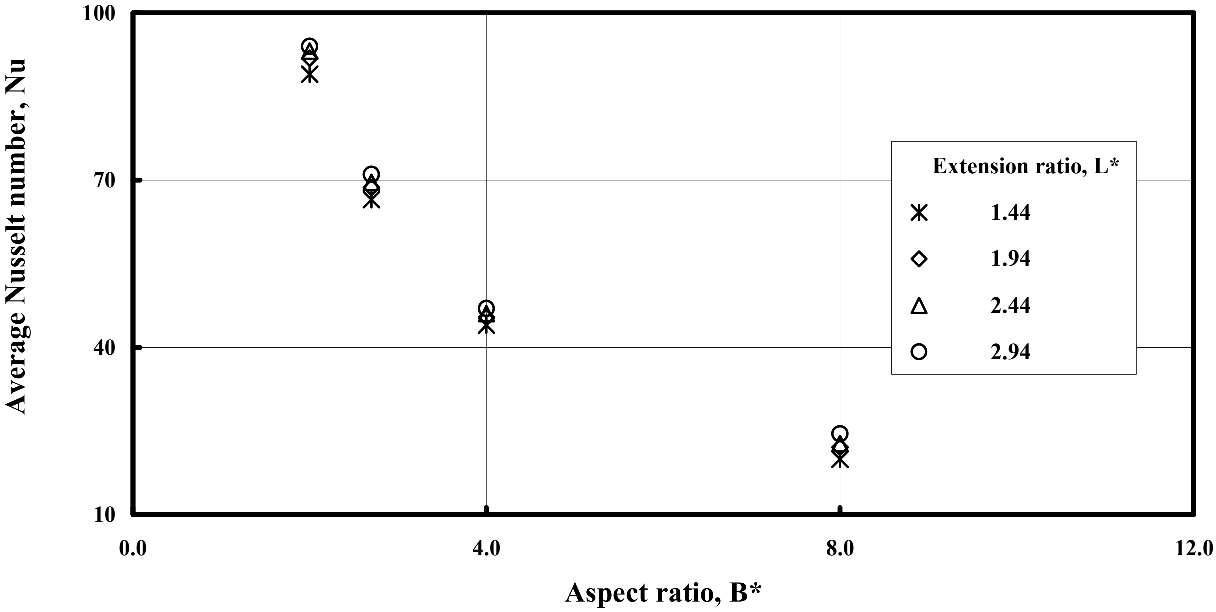


Fig. (11): The average Nusselt number versus the aspect ratio for different extension ratios at inlet coolant temperature, $T_0 = 40\ ^0C$

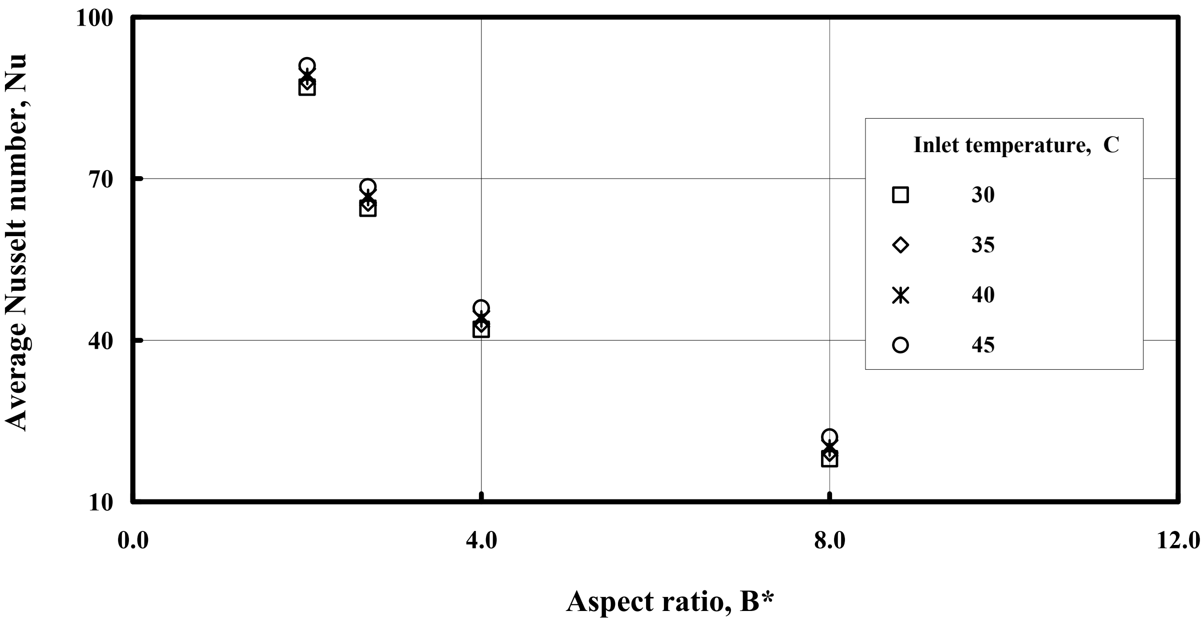


Fig. (12): The average Nusselt number versus the aspect ratio for different inlet coolant temperatures at extension ratio, L* = 1.44

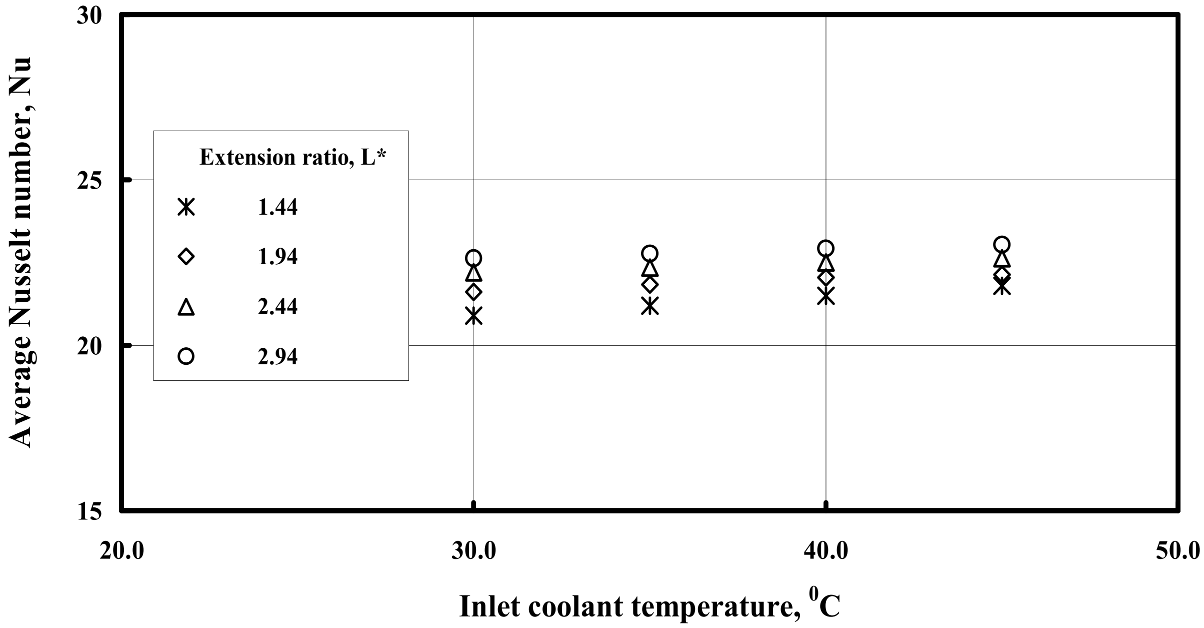


Fig. (13): The average Nusselt number versus the inlet coolant temperature for different extension ratios at aspect ratio, B* = 8

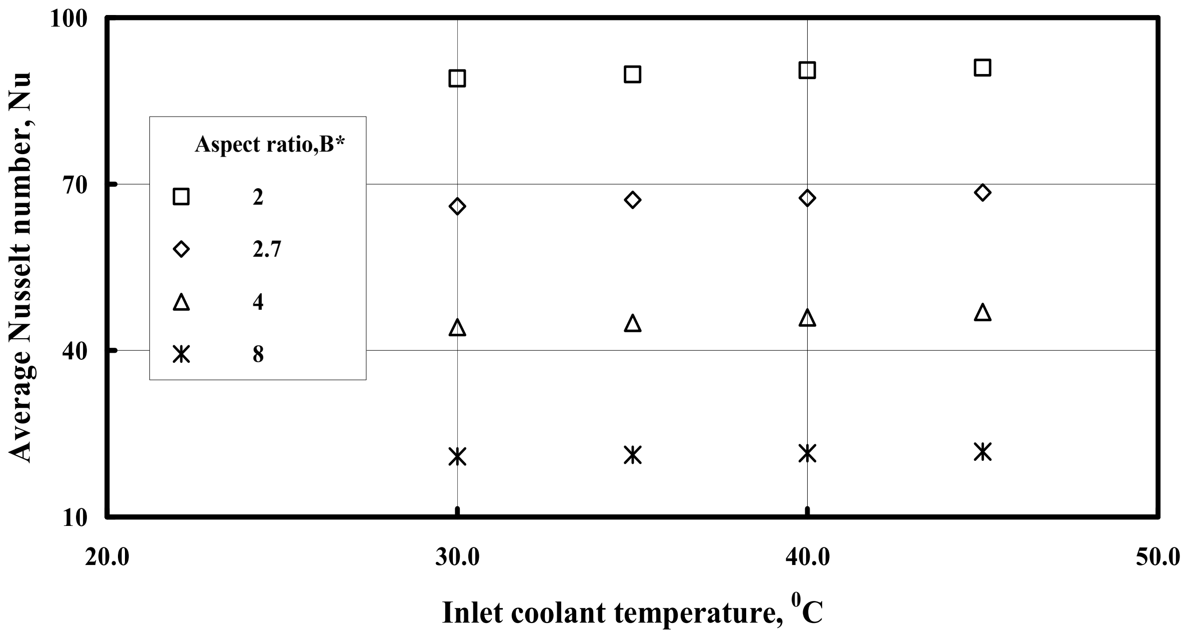


Fig. (14): The average Nusselt number versus the inlet coolant temperature for different aspect ratios at extension ratio, L* = 1.44

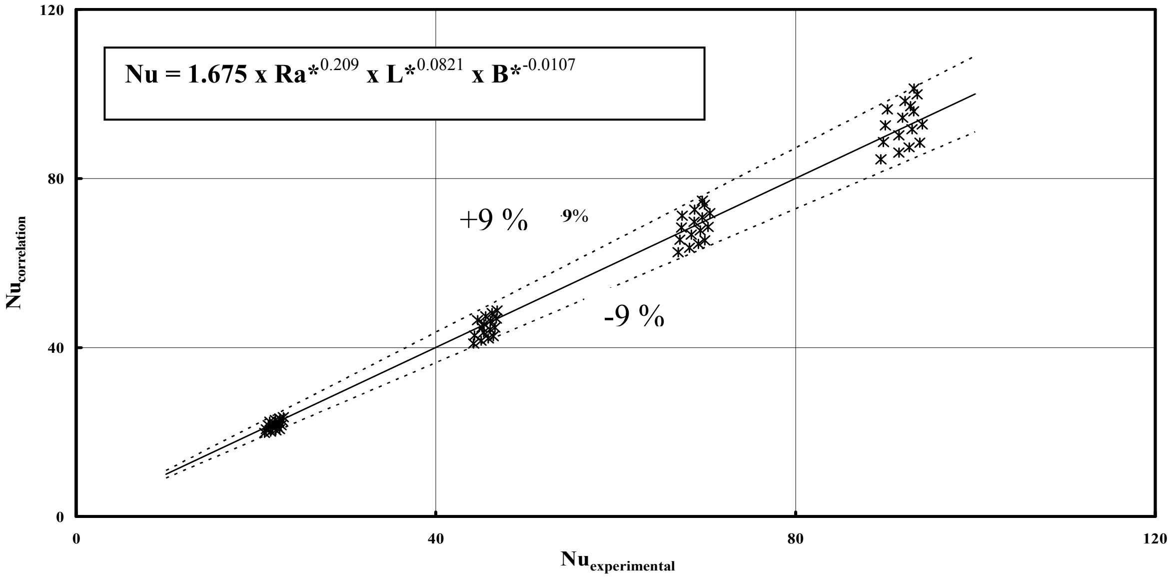


Fig.(15): Present Correlated Nusselt number versus experimental data

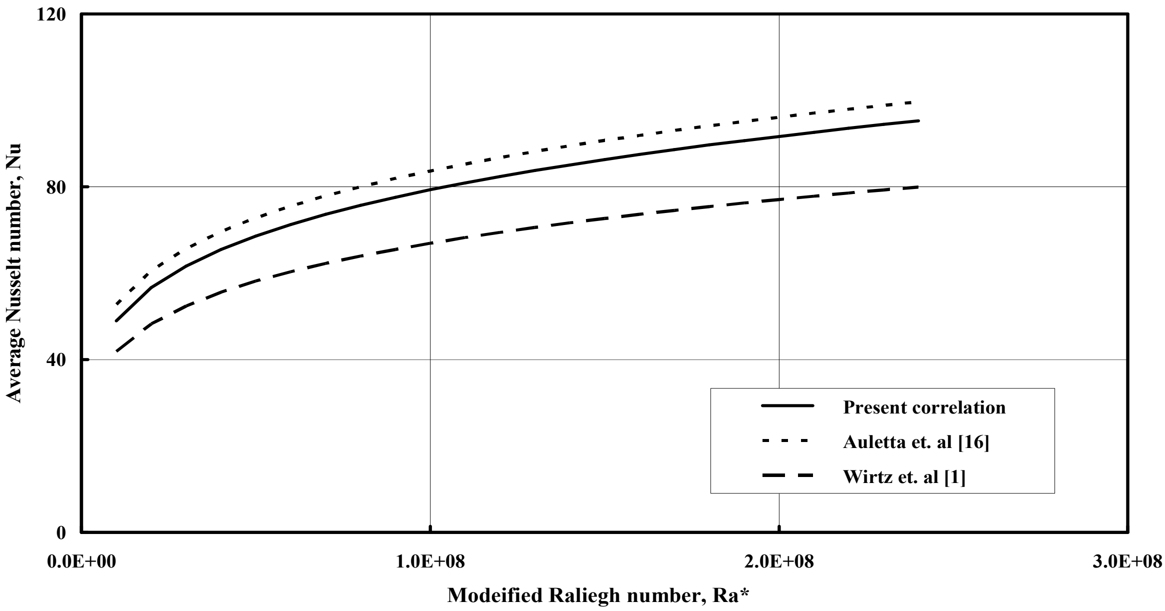


Fig.(16): Comparison between the present correlated Nusselt number and that previously published at L^* =1.44 and B^* = 8