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EXPERIMENTAL INVESTIGATION OF NATURAL CONVECTION HEAT TRANSFER FROM FIN ARRAYS FOR DIFFERENT TIP-TO-BASE FIN SPACING RATIOS

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Abstract : Natural convection heat transfer from fin arrays was investigated experimentally at a wide range of Rayleigh numbers for fin height and different tip-to-base fin spacing ratios *($C=S_t/S_b$)* . Base fin spacing, fin length and fin thickness were kept fixed at 12, 100 and 3 mm respectively. The ratio *C* was varied from 0.25 to 1 and fin height was also varied from 15 to 40 mm. It was found that the convection heat transfer rate from fin array takes on a maximum value as a function of the *C* and fin height for a given temperature difference between fin arrays and surrounding. The results obtained from experimental study showed that the optimum *C* which yields the maximum heat transfer rate is between *C*=0.50 and *C*=0.75. The optimum ratio also depends strongly on the fin height and Rayleigh number. It was observed that fin arrays with an optimum ratio gave higher Nusselt numbers compared to rectangular shaped fin arrays. The increase in the Nusselt number was determined to be about 33%. An empirical correlation for the Nusselt number was derived in the result of experimental study.
Keywords: Natural convection, Fin arrays, Heat transfer, Fin design.

KANATÇIK UÇ VE TABAN NOKTALARI ARASINDAKİ FARKLI BOŞLUK ORANLARI İÇİN KANATÇIK DİZİLERİNDEN DOĞAL TAŞINIM İLE ISI TRANSFERİNİN DENEYSEL ARAŞTIRILMASI

Özet : Kanatçık dizilerinden doğal taşınım ile ısı transferi kanatçık yüksekliği ve kanatçık uç ve taban noktaları arasındaki farklı boşluk oranları için geniş bir Rayleigh sayısı aralığında deneysel olarak araştırılmıştır. Taban kanatçıklar arası mesafe, kanatçık yüksekliği ve kanatçık kalınlığı sırasıyla 12, 100 ve 3 mm olarak sabit tutulmuştur. Kanatçık uç ve taban noktaları arasındaki boşluk oranı 0.25-1 aralığında ve kanat yüksekliği de 15-40 mm arasında değiştirilmiştir. Kanatçık dizisi ve çevre ortam arasındaki verilen bir sıcaklık farkı için kanatçık dizilerinden olan ısı transferinin, kanatçıkların uç ve taban noktaları arasındaki boşluk oranının ve kanatçık yüksekliğinin bir fonksiyonu olarak maksimum bir değer aldığı bulunmuştur. Deneysel çalışmadan elde edilen sonuçlar, maksimum ısı transferini veren kanatçıkların uç ve taban noktaları arasındaki optimum boşluk oranının *C*=0.50 ile *C*=0.75 arasında olduğunu göstermiştir. Maksimum ısı transferini sağlayan bu optimum oran özellikle kanatçık yüksekliğine ve Rayleigh sayısına bağlıdır. Dikdörtgen kesitli kanatçık dizileri ile karşılaştırıldığında optimum orana sahip kanatçık dizilerinden daha yüksek Nusselt sayıları elde edilmiştir. Nusselt sayısındaki bu artışın 33%'e kadar olduğu belirlenmiştir. Yapılan bu çalışma sonucunda Nusselt sayısı için bir ampirik bağıntı türetilmiştir.
Anahtar kelimeler: Doğal taşınım, Kanatçık dizileri, Isı transferi, Kanatçık tasarımı.

INTRODUCTION

Additional surface areas which are referred to as fins are applied in a wide range of engineering applications like heat exchangers, cooling of electronic equipment and similar industrial applications to increase heat transfer. Finned surfaces are extensively used for natural convection to cool electronic devices. When an array of fin is used to enhance the heat transfer under natural convection conditions, fin´s optimum geometry which corresponds to a maximum heat transfer rate should be used, provided this is compatible with available space and financial limitations. Although a finned surface increases the heat transfer area, it reduces the flow rate compared to a base plate. Hence, if not properly designed, it is possible that no improvement is achieved in terms of overall heat transfer. Therefore, it is important to perform a study on the geometry of fin array to have a design with considerable heat transfer enhancement. The heat transfer to the external ambient atmosphere by the heat dissipating apparatus can be obtained mainly using the mechanisms of the heat transfer which are forced convection, natural convection and radiative heat transfer. This paper deals with those issues related to the heat transfer obtained by natural convection. Available literature in this field has shown that various experimental and numerical studies have been performed on fin arrays. The most common technique for enhancing natural convection is using parallel plate channels and various fin configurations. A great number of experimental and analytical work have been carried out on this problem since Elenbaas (1942). He was first, who introduced the problem of natural convection between parallel plates. Starner and McManus (1963), who measured the average heat transfer coefficient not only in horizontal but also in 45^0 and vertical base positions, have performed the first work on horizontal rectangular fin arrays. They showed that incorrect application of fins to a surface actually might reduce the total heat transfer to a value below that of the base alone. An extensive review and discussion of work done on the convective heat transfer in electronic cooling was presented by Incropera (1988), summarizing various convection cooling options. Welling and Wooldridge (1965) conducted a similar experimental study on rectangular vertical fins. They reported optimum values of the ratio of fin height to spacing. Harahap and McManus (1967) observed the flow patterns in two series of horizontal rectangular fin arrays. From the observations, they concluded that the single chimney flow pattern yielded higher rates of heat transfer. Jones and Smith (1970) investigated the effects of fin height, and fin spacing on heat transfer coefficient. They concluded that fin spacing was the main geometrical parameter, and it should to be chosen as characteristic length. There are numerous experimental investigations in the literature on natural convection heat transfer from rectangular fin arrays placed on either horizontal or on vertical plates and annular circular fins (Leung et al., 1985; Leung and Probert, 1989; Yüncü and Anbar, 1998; Chen and Hsu, 2007; Nada, 2007; Yazıcıoğlu and Yüncü, 2009). In these studies, the effects of fin height, fin spacing and the difference between the base surface temperature and ambient air temperature were investigated. As a result of these studies, some correlation equations were proposed as a function of fin height and spacing to obtain the maximum heat transfer in the case of natural convection with fin arrays for a certain temperature difference between the fin base and ambient air. Also, there are several numerical studies investigating the natural convection heat transfer from vertical plate fins protruding from a vertical base or from a horizontal base and for diferrent fin geometries (Baskaya et al., 2000; Mobedi and Yüncü, 2003; Arquis and Rady, 2008; Tari and Mehrtash, 2013; Wong and Huang, 2013; Turkyilmazoglu M., 2015; Fabbri, 1999; and Doğan et al., 2012). The natural convective cooling of horizontally based vertical rectangular fins, in the presence of a horizontal shroud situated adjacent to and above the horizontal fin-tips, was investigated experimentally by Naik et al. (1987) and numerically by Yalcin et al. (2008). In these studies, optimum fin spacing which corresponds to the maximum heat transfer rates were determined for various combinations of fin heights and shroud clearance to fin height ratios for a constant fin-base temperature. In general, for optimum fin spacing, the higher fins and the shroud clearance to fin height ratio yielded the higher heat transfer rates but smaller fin spacings. A model has been developed analytically to carry out the performance and optimum design analysis of fin arrays by Kundu and Das (2009). Their analysis suggests that conduction through the supporting structure and convection from the interfin spacing had a pronounced effect on the performance of a fin array. They also offered that the optimum fin dimensions in a fin assembly have been determined by consideration of the constant total height of the fin assembly and interfin spacing. A theoretical and experimental study was carried out on the thermal performance of a pin-fin heat sink by Kobus and Oshio (2005a). They developed a theoretical model that has the capability of predicting the influence of various geometrical, thermal, and flow parameters on the effective thermal resistance of the heat sink. Then, based on the exact analytical correlations, the simplified approximate correlations for calculating the gray body view factor from a diffuse and gray plate fin heat sink were presented by Kobus and Oshio (2005b). Thermal optimization of plate-fin heat sinks and performance of square pin fin heat sinks were studied respectively by Huang et al. (2008) and Kim (2012).

Radiation plays an important role in the heat transfer from fin arrays. Some studies exist in literature, which take into account the effects of radiation on convective heat transfer from fin arrays N. Ellison (1979). Khor et al. (2010) concluded that the practice of neglecting the radiation view factor in the thermal analysis of fin arrays should be prohibited based on the fact the errors generated are noticeably larger than those of solely neglecting thermal radiation. Rammohan and Venkateshan (1996) made an interferometric study of free convection and radiation heat transfer from a horizontal fin array. The authors stressed the importance of the mutual interaction between free convection and radiation. Dharma et al. (2006) carried out a conjugate analysis in which the heat transfer from a horizontal fin array by natural convection and radiation were determined numerically. The problem was theoretically tackled by treating the adjacent internal fins as two-fin enclosure. Numerical results were obtained to study the effectiveness for different values of fin heights, emissivities, number of fins on a fin base, fin base temperature, and fin spacing. Sparrow and Vemuri (1985) carried out an experimental study on the combined mode natural convection-radiation heat transfer from highly populated pin fin arrays. In which they investigated the effect of various parameters on the heat transfer. They concluded that the heat transfer performance increased with fin length. The contribution of radiation was determined to be substantial and was the best for more populous arrays and longer fins at the smaller base plate-to-ambient temperature differences. Sparrow and Vemuri (1986) later extended their study to different orientations.

From the above literature review, one can see that the additional area provided by extended surfaces increases heat transfer, and the natural convection heat transfer from fin arrays on a base surface depends greatly on the geometrical parameters such as fin spacing, fin length and fin height. Although the heat transfer from fins on a horizontal surface has been the subject to numerous experimental and theoretical investigations, almost all of them are on rectangular fin arrays. Because such fins are simple and cheap to manufacture. When the aim is to improve the performance of heat exchangers, one particular area of interest lies in using different fin shapes that are able to give higher heat transfer rates. Nevertheless, today the developments in manufacturing methods have made the production of different fin shapes easier and cheaper. Therefore, it is important to perform a study on various shaped fin arrays to have a design with considerable heat transfer enhancement. The geometrical parameters affect the natural convection heat transfer from rectangular fin arrays since the flow patterns and temperature gradients in fin arrays change with respect to these parameters. This suggests that it might be possible to have greater heat transfer rates than those of rectangular fin arrays, using different fin shapes, which have better flow characteristics and temperature distributions for the fluid in the fin channel. Because of the manufacturing difficulties of fins with different shapes other than rectangular profile, their usage should not has been limited. Nevertheless, today the developments in manufacturing methods have made easier and cheaper the production of different shaped fins.

Heat transfer rate from the fin arrays can be enhanced by increasing the velocity of fluid flowing through fin arrays. The increase in the velocity of fluid causes the heat transfer coefficient and cold fluid flow rate to increase. Accordingly, the outlet area of the fin array was gradually narrowed (like a converging nozzle) to increase outlet velocity of the hot fluid which flows toward to the tip of the fin arrays and therefore these fin profiles was suggested. The ultimate purpose of this research was to determine an optimum fin profile, to obtain better convective heat transfer rate from fin array compared with rectangular fin profile.

EXPERIMENTAL SET-UP AND DATA REDUCTION

A summary of information on the experimental setup is presented below.

Experimental Set-Up

A schematic drawing of the experimental set-up is shown in Fig.1. The experimental set-up consists mainly of fin array, heater plate, insulation material, wooden frame, supporting frame, variac, regulator, data logger and PC.

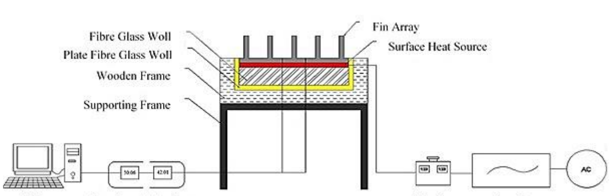

Figure 1. Schematic diagram of the experimental set-up.

A sheet heater plate was placed at the bottom surface of the fin array, with a size equal to the size of base of the fin array having dimensions of 250×100 mm. The electrical power was provided to the heater plate via a regulator which maintained a constant voltage. The output of the regulator is fed to a variable transformer (variac) so that providing a heat flux boundary condition specified for a decided experimental case. Although the current flow and voltage drop can be read from an ammeter and voltmeter which are mounted on the variac, a digital multimeter with resolution of 0.01 V, 0.001 A and accurancy of 1% and 1.5% for voltage and current, respectively is used to measure the electrical power input accurately. The base plate surface temperatures were measured at 22 point by T copper-constant thermocouples, whose distribution is shown in Figure 2. To facilitate thermocouple installation, a hole drilled through the base of the fin array from bottom surface at each thermocouple location. The thermocouples were inserted into these holes and fixed with thermal adhesive (Artic Silver 5). To estimate conduction heat losses across the wooden frame, two thermocouples were fixed on the inner and outer surface of each side of the wooden frame.

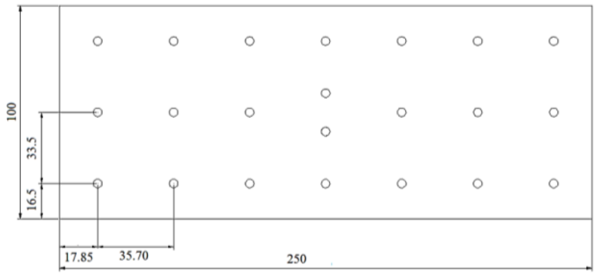

Figure 2. Locations of the thermocouples (measures are in mm).

Temperatures were also measured for ambient air temperature by separate thermocouples. Each thermocouple is manufactured using welding machine (electric arc welding). All thermocouples were calibrated in a constant temperature bath and a measurement accuracy of ±0.10 ^{0}C was obtained. The temperature signals were transferred to two 32-channel data acquisition unit, and finally, sent to a PC for further processing. Temperatures measured for 15 seconds time intervals were collected, stored and analyzed in this PC. It was observed that experimental conditions reach a steady-state condition about approximately 5 to 6 h. Steady state condition was considered to achieve when differences in temperatures between two intervals become negligible (ΔT=0.2 ^{0}C). Once the steady state condition has been established, readings of all thermocouples, ambient air temperature and electrical power input were recorded.

Fig. 3a and b show the generally observed flow patterns for rectangular fin arrays. For an array of vertical rectangular fins protruding vertically upwards from a horizontal rectangular base, the surrounding fluid enters to the channel from the two open ends and develops a vertical component of velocity as the air is heated (single chimney type flow pattern). However, if the fins are long, these air streams will leave the open-topped

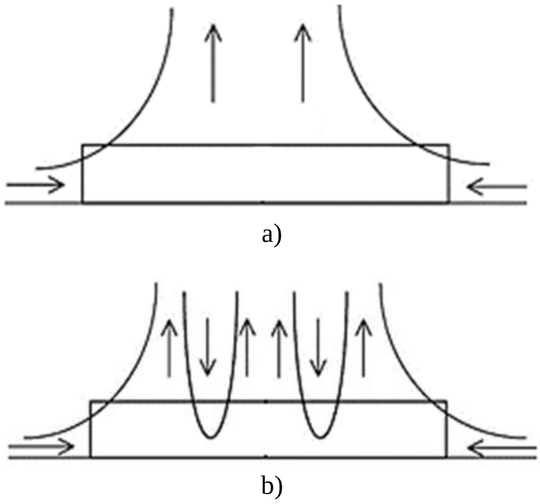

Figure 3. Types of flow pattern a) single chimney b) multiple chimney.

channel well before they reach the mid-length regions of the fins. In these regions some air will be sucked downwards, become heated, and eventually move upwards (multiple chimney type flow pattern). These types of flow patterns particularly show that the natural convection from fin arrays on a horizontal base depends on the fin geometry.

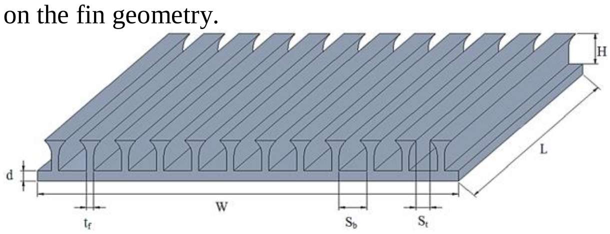

Figure 4. Fin array geometry.

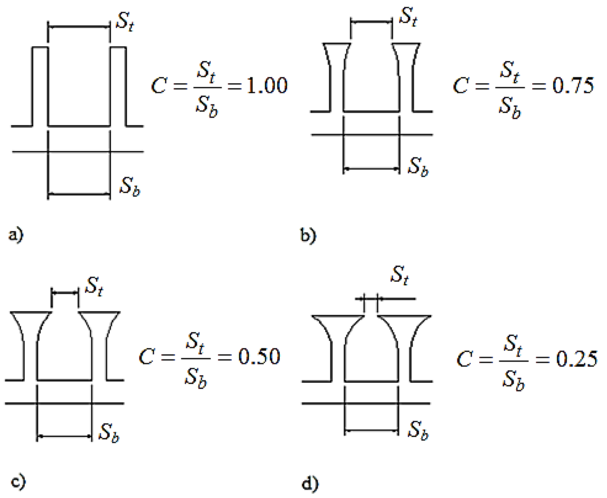

Figure 5. Tested types of the fin.

A schematic drawing of the fin array and types of the fin tested in this study are presented in Fig. 4 and Fig. 5 respectively. The fin arrays with the dimensions of 255×105×25 mm, 255×105×35 mm and 255×105×45 mm are made of 5083 series aluminum alloy block with a thermal conductivitiy of 121 W/mK. Aluminum was chosen as the fin array material due to its high thermal conductivity, low radiative emissivity (about 0.05), easy machinability and its common use in many technological applications. The dimensions of the aluminium blocks have been selected a little larger. In this way, fin arrays were obtained by milling and turning on the lathe to have more flat and smooth surface. Finally, all surfaces were carefully cleaned and polished.

Table 1. Fin-arrays configurations.

Fin-array set	Fin height H, mm	Base fin spacing S_b, mm	Tip fin spacing S_t, mm	Ratio $C=S_t/S_b$
1	15	12	12	1
2	25	12	12	1
3	40	12	12	1
4	15	12	9	0.75
5	25	12	9	0.75
6	40	12	9	0.75
7	15	12	6	0.5
8	25	12	6	0.5
9	40	12	6	0.5
10	15	12	3	0.25
11	25	12	3	0.25
12	40	12	3	0.25

Table 1 lists base fin spacing S_b , tip fin spacing S_t , the fin height H, tip-to-base fin spacing ratio $C= S_t / S_b$ and number of fins of tested all fin-arrays. The base of the fin array width W=250 [mm], the fin length L=100 [mm], the fin thickness t=3 [mm] and fin array base thickness d=5 [mm] were kept fixed.

Processing of the Experimental Data

The data obtained during the experiments are temperature, voltage drop across the heater, and electric current. Using these measurements, the average Nusselt number based on the fin height is defined as

$$Nu_H = \frac{h_{av} H}{k_{air}} \quad (1)$$

where h_{av}, H and k_{air} are the average heat transfer coefficient, the fin height, and thermal conductivity of air, respectively. The average heat transfer coefficient based on total surface area h_{av} is defined as follows,

$$h_{av} = \frac{\dot{Q}_{convection}}{A_t (T_w - T_{in})} \quad (2)$$

T_w and T_{in} are the average fin base surface temperature and the inlet air temperature (ambient air temperature), respectively (Leung et al., 1985; Yüncü and Anbar, 1998; Naik et al. 1987). A_t [m^2] is the total fin and base area. Convection heat transfer from both the fin base and fins ($\dot{Q}_{convection}$) was determined from an energy balance

$$\dot{Q}_{total} = \dot{Q}_{convection} + \dot{Q}_{conduction} + \dot{Q}_{radiation} \quad (3)$$

where $\dot{Q}_{total}$ is the total dissipated $Q_{convection}$ energy from the surface heater source, $\dot{Q}_{conduction}$ is the total conduction heat loss through the wooden frame, and $\dot{Q}_{radiation}$ is the total radiation heat loss from the fin base and fin surfaces. The total dissipated energy was determined from Ohm's law, $\dot{Q}_{total} = VI$ at the heater source. The voltage drop V, and current I, were measured during the experiment. The heat loss through the hornbeam frame was calculated from

$$\dot{Q}_{conduction} = k_{wood} A_{wood} \frac{\Delta T_{wood}}{L_{wood}} \tag{4}$$

where k_{wood} is thermal conductivity of made of hornbeam, [$Wm^{-1}K^{-1}$]. A_{wood}, L_{wood} and ΔT_{wood} are area thickness and the difference between internal and external surface temperatures of wood, respectively. Radiation loss was determined from the assumption of isothermal, gray, diffuse and opaque surfaces. It was evaluated by using the procedure outlined by N. Ellison (1979) and the following equation:

$$\dot{Q}_{radiation} = F A_R \sigma (T_w^4 - T_a^4) \tag{5}$$

where F is the gray body shape factor that varies between 0.037 and 0.034 according to the fin height. T_a is ambient air temperature. For radiation heat transfer, A_R is the surface area that is equal to total heat transfer surface area, defined as A_t and σ is Stephan-Boltzman constant. The radiation losses were approximately 5% of the total power dissipated. The total heat losses were also calculated to be less than 8% of the total power dissipated. The Rayleigh number based on the fin height Ra_H :

$$Ra_H = \frac{g\beta(T_w - T_{in})H_f^3}{\upsilon_{air}^2} Pr \tag{6}$$

The fluid properties used in these definitions were determined at a reference temperature equal to $T_f = (T_w + T_a)/2$. Pr is the Prandtl number of the air, β is the thermal expansion coefficient, g is the gravitational acceleration, and υ_{air} is the kinematic viscosity.

In order to test the reliability of the experimental results, an uncertainty analysis was conducted on all measured quantities as well as the quantities calculated from the measurement results. Uncertainty analysis was estimated according to the standard procedures reported in the literature (Moffat, 1982; Moffat, 1985). On the overall, the uncertainty in the convective heat transfer rate is around ±5.9% and it is around ±7.26% for the Nusselt number.

RESULTS AND DISCUSSIONS

The natural convection heat transfer from various fin arrays on a horizontal base has been investigated experimentally for constant heat flux condition. By adjusting the voltage drop and electrical current, experiments were conducted under various temperature differences between average fin base and surroundings. Measurements were taken with non-dimensionalized fin height ranging between $0.15 \le H/L \le 0.40$ and for the tip-to-base fin spacing ratios of $0.25 \le C = S_t / S_b \le 1$.The variation of convection heat transfer rate per unit base area with base to inlet temperature difference are presented in Figure 6 for the $C = S_t / S_b = 0.25$, 0.50, 0.75 and 1 and for fin heights of $H/L = 0.15$, 0.25 and 0.40. From these figures it can be seen that the convection heat transfer rate from a fin array is dependent on C , fin height, temperature difference between fin base and surroundings. The convection heat transfer rate increases with increase in the temperature difference for each fin height. The air enters from open ends, moves along the fin length and develops a vertical component of velocity as it is heated. It can be defined as the buoyancy driven secondary flows. The buoyancy driven secondary flows which augment the heat transfer, cannot develop, if the fin height is not sufficiently large. The increase in fin height provided that other parameters remain the same, provides the pressure drop in the region bounded by plate- fins. Consequently, this condition leads to the development of the secondary flows. Then, the high velocity regions become present at the entrance and exit regions of air flowing through the fin channel. As a result of increase in fin height, the convection heat transfer rate from the fin array increases. Moreover, increasing fin height causes an increase in the heat transfer surface area. Consequently, surface of fins and base-plate temperatures decrease, which results with an increase in the convection heat transfer rate.

The rate of increase in the convection heat transfer rate from the fin array with small fin height is smaller than the fin array which has a large fin height. This is due to the fact that the buoyancy driven forces become strong enough with an increase in fin height and temperature of air in fin array. Therefore, buoyancy driven secondary flows have more important effects on the rate of heat transfer.

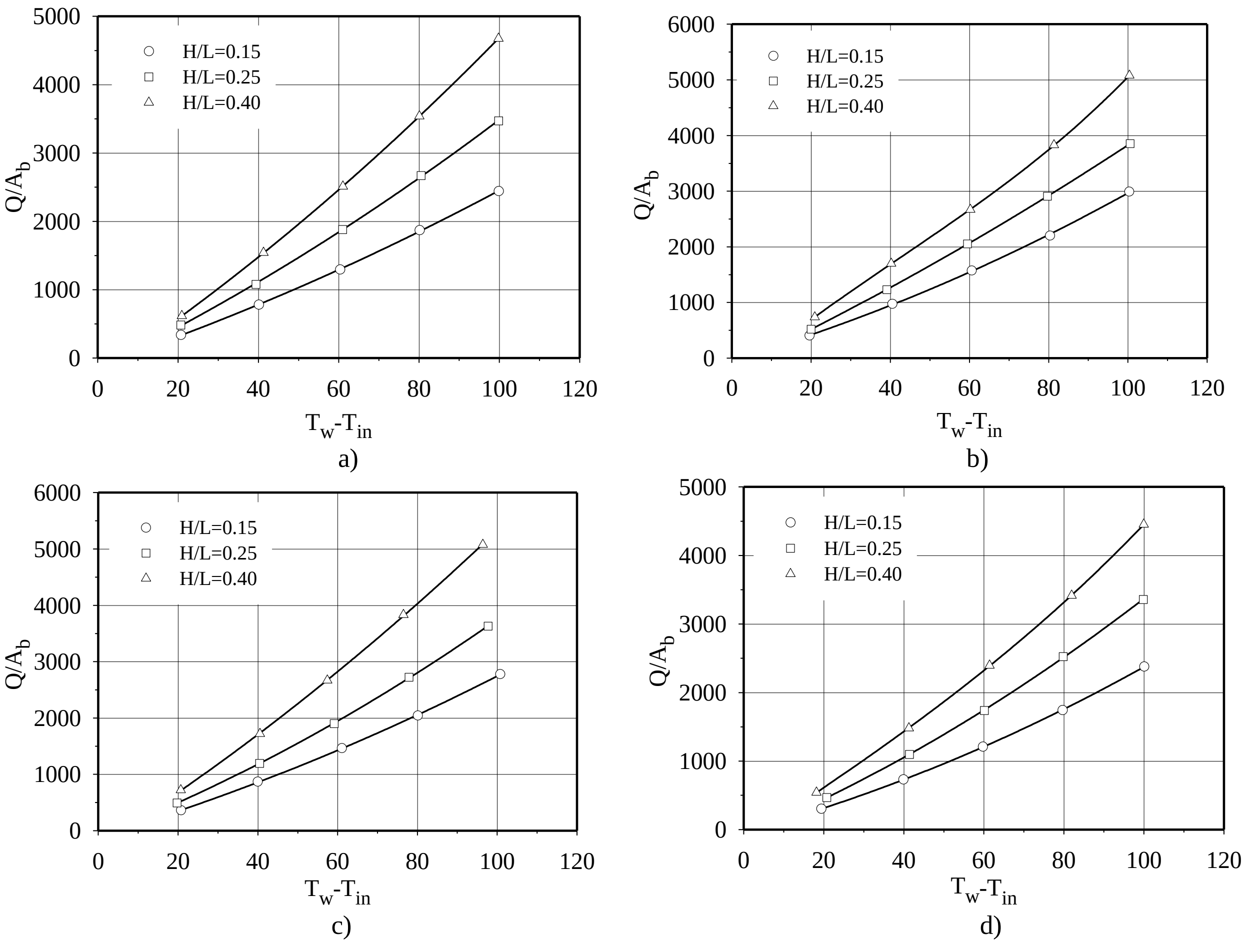

Figure 6. Variation of convection heat transfer rate per unit base area with base to inlet temperature difference for a) C=0.25, b) C=0.50, c) C=0.75 and d) C=1.00.

The effect of C on convective heat transfer rate per unit base area is presented in Fig. 7 for different fin height to length ratio values. As can be seen from these figures, increase or decrease in the value of C after a fixed value does not cause an enhancement on heat transferred per unit base area. On the contrary, it causes a reduction in heat transfer. The convective heat transfer rate first increases with C up to a maximum value and then it decreases with an increase in the value of C. The value of the tip-to-base fin spacing ratio at which the heat transfer takes its maximum value can be defined as optimum ratio, C_{opt}.

These graphs can be examined by separating them into three regions which are before the optimum ratio region, the optimum ratio region and after the optimum ratio region.

- Firstly, until the optimum ratio: When decrease in outlet area of fluid or decrease in the ratio C (from optimum ratio to zero), the convective heat transfer decreases. Because, the further reduction in the outlet area of fluid causes velocity of fluid flowing through fin arrays to decrease, preventing cold fluid to enter fin grooves and thus, hot fluid stays much longer between fin arrays. The decrease in fluid velocity results in a deficiency in removing heat energy from heated surfaces. Therefore, large part of the duct is occupied by heated fluid and this also prevents the heat transfer to be achieved effectively.

- Secondly, the region of optimum ratio: Obviously, better heat transfer is achieved at the optimum ratio region. This is due to variations in the flow patterns. In the region of optimum ratio, the channel formed by fins may act like a converging nozzle. Thus, the velocity of fluid flowing through fin arrays increases locally towards the outlet of fins array (for subsonic flow). As a result, the local heat transfer coefficient increases. Therefore, the convective heat transfer rate from fin array increases. Furthermore, the increase in the total heat transfer surface area also contribute to the increase in the heat transfer compared with flat rectangular fins.

- Thirdly, after the optimum ratio: As the value of C increases, the convective heat transfer from fin array decreases due to reduction in the total heat transfer surface area

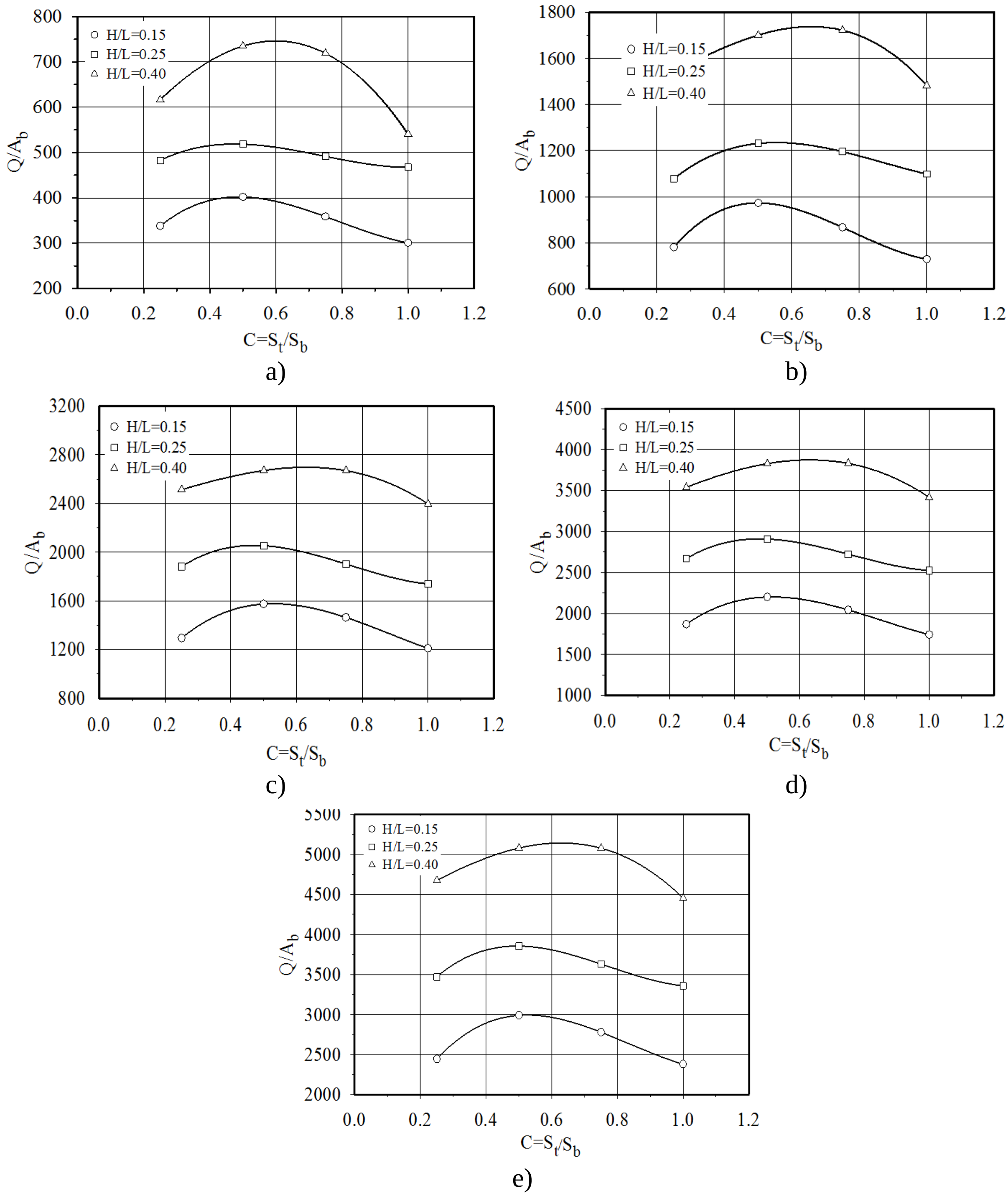

Figure 7. Effect of C on heat transferred per unit base area for $(T_w\text{-}T_{in})$=20, 40, 60, 80, 100 °C, respectively.

It can be seen that from these figures, optimum ratio is between C=0.50 and C=0.75 which yields maksimum convective heat transfer rate that is dependent on fin height. In this case, the following observations can be made: Firstly, we can say that the optimum ratios are about C_{opt}=0.50 and C_{opt}=0.60 for the fin height H/L=0.15 and 0.25 respectively. Secondly, for the fin height H/L=0.40, the optimum ratio is about C_{opt}=0.75. Therefore, to achieve maximum amount of heat transfer in the design of fin arrays on a horizontal surface for a given fin height, the tip-to-base fin spacing the ratio should have an optimum value.

The variation of Nusselt number with C for different Rayleigh number and fin heights to length ratio, H/L=0.15, 0.25 and 0.40 is shown in Fig 8. As seen from these figures, for each of the Rayleigh number, the Nusselt number increases with C up to the optimum value and then it decreases. The maximum Nusselt number was obtained at around C=0.5 for the fin heights H/L=0.15 and H/L=0.25. But, as it can be seen from Fig 8c, the maximum Nusselt number was obtained for low Rayleigh number at around C=0.6 and for high Rayleigh number at around C=0.75. As a result of the increase in Rayleigh number, the optimum ratio was increased.

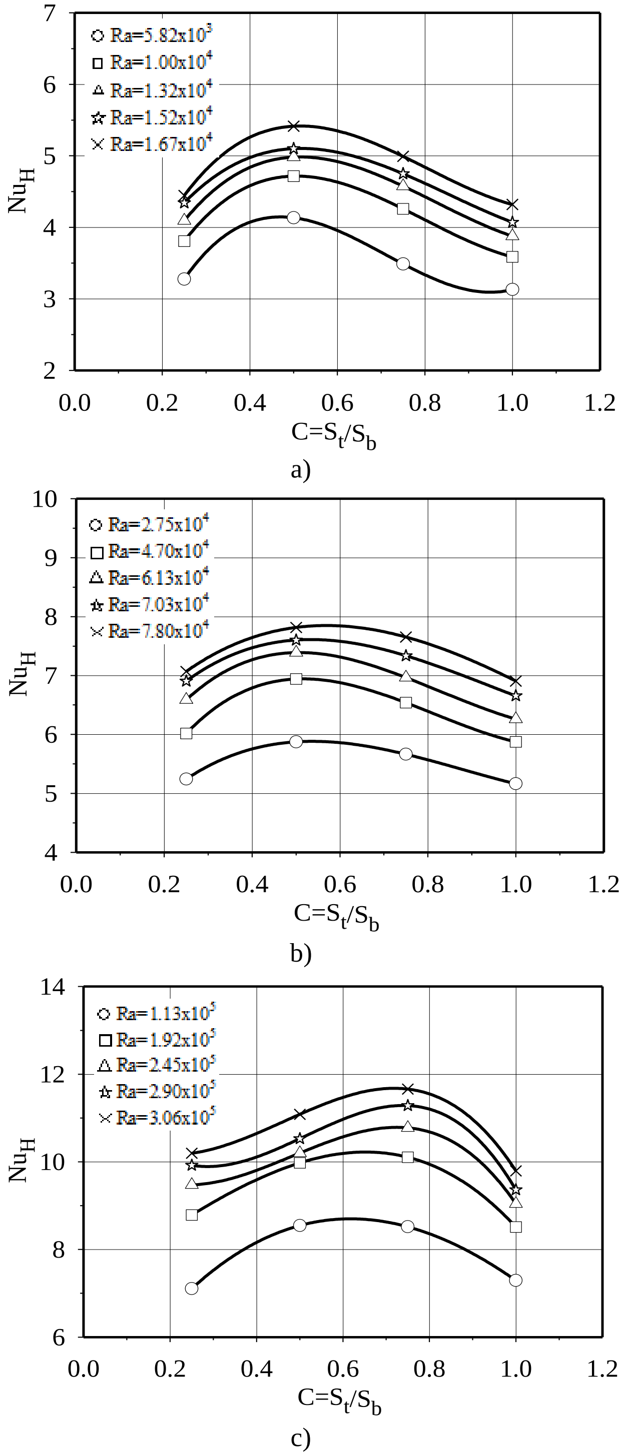

Figure 8. Variation of Nusselt number with *C* at fin heights a) *H/L*=0.15 b) *H/L*=0.25 c) *H/L*=0.40.

Fig. 9 shows the relation of Nu_H/Nu_{H0} and *C* for three fin heights. It can be seen that *C* can significantly enhance heat transfer. The average Nusselt number increases with the ratio and *C*=0.5 shows the highest enhancement in the average Nusselt number in the range of present study for fin heights *H/L*=0.15 and *H/L*=0.25. For fin height *H/L*=0.40, the highest enhancement in the average Nusselt number was obtained at *C*=0.75.

In this study, the suggested fin configuration provides an increase up to 5-33% in Nu number, compared with rectangular fin array. The reason of the increase in heat transfer is briefly explained as follow. As a result of decrease in the outlet sectional area of the fin array, the local heat transfer coefficient increases due to increase of the outlet velocity of the fluid flowing through the fin array and thus, convective heat transfer rate increases.

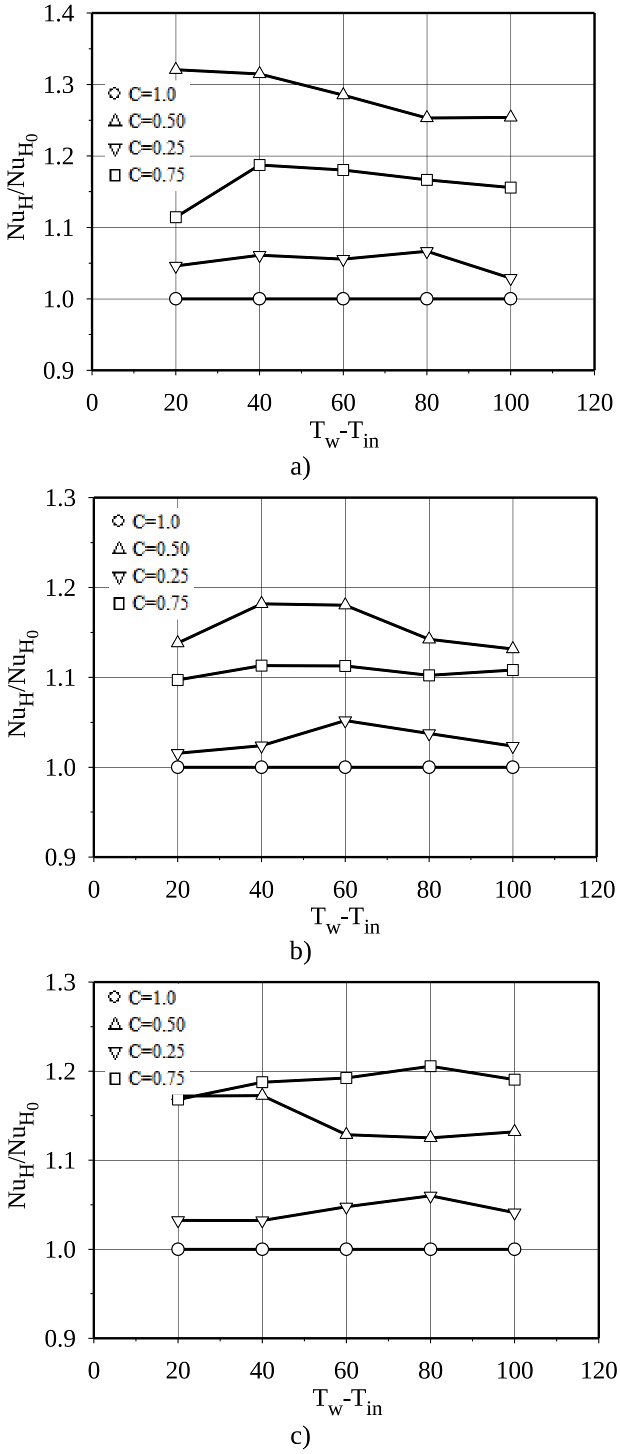

Figure 9. Nusselt number ratio Nu_H/Nu_{H0} at fin heights a) *H/L*=0.15 b) *H/L*=0.25 c) *H/L*=0.40.

This study was carried out for the base fin spacing *S*=12 mm and the Nusselt number was obtained higher compared to rectangular fin array. The higher Nusselt numbers are obtainable also for the different base fin spacing values. It is shown in Figure 10. Tari and Mehrtash (2013) suggested a Nu_s correlation for rectangular fin arrays on a horizontal base, as shown in Eq. (7), using fin spacing as the characterisric length.

$$\overline{Nu_S} = 0.0929\left(Gr' \Pr\right)^{1/2} \text{ for } Gr' \Pr\langle 250 \tag{7a}$$

$$\overline{Nu_S} = 0.2413\left(Gr' \Pr\right)^{1/3} \text{ for } 250\langle Gr' \Pr\langle 10^6 \tag{7b}$$

Similarly, Wong and Huang (2013) proposed another correlation for the same rectangular fin arrays on a horizontal base, as shown in Eq. (8). Fin spacing S is again selected as the characteristic length.

$$Nu_S = 0.069\left(Gr' \Pr\right)^{1/2} \text{ for } Gr' \Pr \le 250 \tag{8a}$$

$$Nu_S = 0.228\left(Gr' \Pr\right)^{1/3} \text{ for } 10^6 \rangle Gr' \Pr\rangle 500 \tag{8b}$$

The comparison with our experimental results is shown in Fig. 10. The experimental results of Nu_S appaer to be higher about 25% as a consequence of narrowing flow path near the fin tips than those by Eq. (7) and Eq. (8).

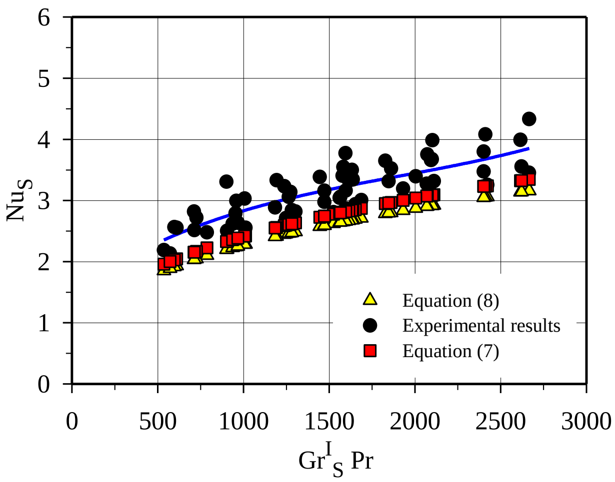

Figure 10. Comparison of present experimental results with the Nu_S correlations of Tari and Mehrtash (2013) and Wong and Huang (2013).

In the present work, it has been shown that the average Nusselt number depends on the Grashof number, fin spacing, fin height, and fin length. Considering the effects of above parameters, an empirical equation has been derived to correlate the average Nusselt number. The correlation could be expressed as follows;

Grashof number based on fin height H is defined as

$$Gr_H = \frac{g\beta(T_w - T_{in})H^3}{\nu^2} \tag{9}$$

And we can define a modified Grashof number based on fin height H is defined as

$$Gr_H^I = Gr_H\left(\frac{H}{L}\right)^{1/2}\left(\frac{S_t}{S_b}\right)^{1/3} \tag{10}$$

The average Nusselt number based on fin height H,

$$Nu_H = \frac{hH}{k} = a\left(Gr_H^I \Pr\right)^b \tag{11}$$

Using the form, Eq. (11), we obtained a constant coefficient of 0.4162 and b constant coefficient of 0.2599. The squared correlation coefficient R^2 of the fit is 0.9616, that is, the suggested form fits very close to experimental results. Consequently, we proposed the following correlation equation and it is shown in Fig 11.

$$Nu_H = 0.4162\left(Gr_H^I \Pr\right)^{0.2599} \tag{12}$$

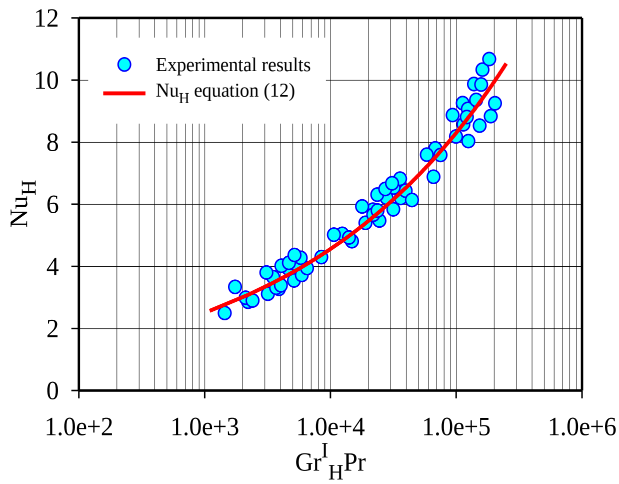

Figure 11. Correlation obtained from the present experimental results.

The plot of the Nu, predicted by Eqs. (12) and experimental Nu are depicted in Fig. 12. In the figure, the majority of the experimental Nu falls within ±10% for, the predicted Nu.

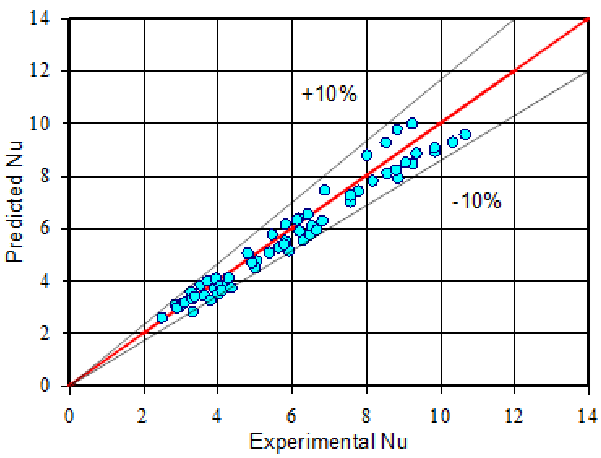

Figure 12. Comparison between predicted Nu and experimental Nu.

CONCLUSIONS

The natural convection heat transfer from vertical fin array on a horizontal base was studied experimentally. Experimental results were presented for tip-to-base fin spacing ratio, fin heights, fin spacing, Rayleigh numbers as well as temperature difference between fin base and surroundings. It has been determined that the rate of heat transfer from a fin array depends on the geometry of the fin array, fin height, fin spacing and temperature difference between fin base and surroundings.

The effects of the tip-to-base fin spacing ratio on heat transfer was investigated by conducted experiments at four different ratios (C=1, 0.75, 0.50 and 0.25). Firstly, the Nusselt number increases with C and then, it takes its maximum value after which, it starts to decrease with the increase in the value of C. To achieve the maximum amount of heat transfer, the value of C should be an optimum. Its optimum value depends mainly on fin height. The optimum ratio increased with an increasing fin height and it was obtained around C_{opt}=0.50 and C_{opt}=0.70 for small fin height and for high fin height, respectively. The fin array that has higher fin height acts as a converging nozzle in the region of the optimum ratio. As a result, the heat transfer performamce of the tested fin array configuration was obtained higher compared to the performance of tradional rectangular fin array.

An ampirical correlation for average Nusselt number was presented to relate the heat transfer from fin arrays with dimensionless, Grashof number and other experimental parameters such as fin spacing, fin height and fin length.

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