A Practical Study of the Positions of the Offsite-Strip Fins [OSF] Heat Transfer Performance to the Turbulent Water Flow

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Abstract: Off-site strip rectangular fin arrays are commonly used, for example, in the marine engineering: wind and wave loading, sloshing, propulsion, mechanical engineering: pumps, fans, heat exchangers, chemical and petrochemical engs. Process: static mixing, separation, reactions, electrical eng.: equipment cooling and environmental eng.: pollutant and effluent control, fire management, shore protection, fans, heat exchangers. In designing any kind of heat sink or heat exchanger, the need to save energy and reduce costs always sets the following requirements for the design: the total heat transfer rate should be high while the volume and weight of the system should be low.

The aim of this study is to investigate experimentally the fin heat transfer performance of an offset-strip fins [OSF] at vertical and horizontal positions in the water fluid flow as cooling system. This is done at different water volumetric flow rates (Qv) from 0.5 to 2.0 L/s at constant hydraulic diameter (Dh, 0.42 m). In the present study, experiments are conducted at the range of water system temperature from 26 to 30 ºC. The computations are conducted by assuming that the flow in the offset-strip fin channels is steady and turbulent at the range of Reynolds numbers from 4300 to 6000. Can be deduced from this research work, that horizontal OFS position has a high fin thermal performance than other.

Keywords: (Experimental Fluid Flow and Heat Transfer, Forced Water Flow Convection, Heat Transfer Performance and Fins Position)

1. Introduction

Among the engineering, production industries is the use of liquid or gas via pipelines or ducts in heating, cooling, and fluid distribution network applications. The fluid in such an application is usually forced to flow by a fan or pump through the flow section. We pay special attention to the heat transfer across the fin surface and the base of the fins, which is directly related to the temperature drop and the extent to which the special coefficients affect the loss of the amount of heat during the flow through the tubes and channels. The fluid velocity in a pipe changes from zero at the surface because of the no slip condition to a maximum at the pipe center.

At Reynolds number (Re no.) the inertial force, which is proportional to the fluid density and the square of the fluid velocity, is largely relative to the viscous force, and thus the viscous force cannot prevent the random and rapid fluctuation of the fluid at small Reynolds number; however, the viscous forces are large enough to overcome the inertial force and to keep the fluid in line thus the flow is turbulent in the first case and laminar in second.

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The Reynolds number at which the flow becomes turbulent is called the critical Reynolds number $Re_{cr.}$ The value of the critical Reynolds number is different for different geometries and flow condition. For internal flow in a circular pipe, and generally accepted value of the critical Reynolds number is Re =2300. For flow through noncircular pipes, the Reynolds number is based on the hydraulic diameter D_h defined (Cengel, 2004), and (Incropera, DeWitt, Bergman, Lavine, et al., 1996)

Hydraulic diameter $D_h = \frac{4\, A_c}{P}$

$$Re. = \frac{V \times \rho \times D_h}{\mu} \quad [1]$$

The hydraulic diameter is defined such that it reduces to ordinary diameter Dh for rectangular duct.

$$D_h = \frac{4\, A_c}{P} = \frac{2ab}{(a+b)} \quad [2]$$

In convection studies, it is common practice to non-dimension the governing equations and combine the variables, which group together into dimensionless numbers in order to reduce the number of total variables. It is also common practice to non-dimension the heat transfer coefficient h with the Nusselt number, defined as (Incropera & DeWitt, 1996):

$$Nu = \frac{hL_c}{k} \quad [3]$$

The Nusselt number represents the enhancement of heat transfer through a fluid layer as a result of convection relative to conduction across the same fluid layer the larger the Nusselt number, the more effective the convection. A Nusselt number of Nu = 1 for a fluid layer represents heat transfer across the layer by pure conduction. For practical applications, a reliable and appropriate correlation based on the constant-property assumption can be modified or corrected so that it may be used when the variable –property effect become important. Two correction methods for constant –property correlations for the variable –property effect have been employed: namely, the reference temperature method and the property ration method. In the former, a characteristic temperature is chosen at which the properties appearing in no dimensional groups are evaluated so that the constant –property results at that temperature may be used to account for the variable –property behavior in the latter, all properties are taken at the bulk temperature and then all variable-property effect are lumped into a function of the ration of one property evaluated at wall (surface) temperature to that property evaluated at the average bulk temperature. Some correlations may involve a modification or combination of these two methods.

We shall present only the results for thus more complex situation. The reader is referred to Ref.1 for details on the mathematical methods used to obtain the solution in reality, however, the temperature of the fin will drop along the fin and thus the heat transfer from the fin will be less because of the decreasing temperature difference T(x) – T (∞) toward the fin tip to account for the effect of this decrease in temperature on heat transfer, we defined a fin efficiency as. (Incropera & DeWitt, 1996), and (streeter1975solutions).

$$Fin\ efficiency(\eta_F) \quad \frac{Q_{fin}}{Q_{fin\ max}} \quad [4]$$

$$Fin\ efficiency(\eta_F) = \frac{\text{actual heat transfered}}{\text{heat which would be transfered if entire fin area were at base}}$$

Fins are used to enhance heat transfer, and the use of fins on a surface cannot be recommended unless the enhancement in heat transfer justifies the added cost and complexity associated with the fins. In fact, there is no assurance that adding fins on a surface will enhance heat transfer. The performance of the fins judged on the basis of the enhancement in heat transfer relative to the no-fin case. The performance of fins expressed in term of the fin effectiveness ϵfin is defined as:

$$Fin\ effectiveness\ (\varepsilon)\ = \frac{Q_{fin}}{Q_{no\ fin}}$$

$$\varepsilon\ = \frac{h\,A_t\,(T_s - T_\infty\ )}{h\,A_B\ (T_b - T_\infty\ ))} \qquad [5]$$

Here, AB is the cross-sectional area of the fin at the base and (Q no fin) represents the rate of heat transfer from this area if no fins are attached to the surface. An effectiveness of ∈fin =1 indicates that the addition of fins to the surface does not affect heat transfer at all. That is, heat conducted to the fin through the base area AB is equal to the heat transferred from the same area AB to the surrounding medium. An effectiveness of ∈fin < 1 indicates that the fin actually acts as insulation, slowing down, down the heat transfer from the surface. This situation can occur when fins made of low thermal conductivity are enhancing heat transfer from the surface, as they should. However, the use of fins cannot be justified unless ∈fin is sufficiently larger than1. Finned surfaces are designed on the basis of maximizing effectiveness for a specified cost or minimizing cost for a desired effectiveness (Cengel, 2004; Al-Jewaree & Alhamil, 2015).

Over the past few decades, a large amount of study has been conducted to analyze the heat transfer and pressure drop characteristics of compact heat exchangers. Although various types of interrupted fin surfaces have been performed in the past, this study focuses on the offset-strip fin (OSF) type compact heat exchanger. Various similar studies about this type of fin are available in the literature and they will be summarized in this chapter.

Patankar and Prakash (Patankar & Prakash, 1981) presented a two-dimensional numerical analysis for the flow and heat transfer in an interrupted plate passage, which is an idealization of the OSF heat exchanger. The main aim of the study is investigating the effect of plate thickness as a non-dimensional form t/H on heat transfer and pressure drop in OSF channels because the impingement region resulting from thick plate on the leading edge and recalculating region behind the trailing edge are absent if the plate thickness is neglected. Their calculation method was based on the periodically fully developed flow through one periodic module because the flow in OSF channels attains a periodic fully developed behavior after a short entrance region, which may extend to about 5 (at the most 10) ranks of plates (Sparrow, Baliga, & Patankar, 1977). In the periodic regime the flow repeats itself in an identical manner for successive geometrical modules. The existence of this fully developed periodic regime was first identified in the study of Sparrow, Baliga, and Patankar (1977). They assumed the flow steady and laminar in the Reynolds number ranges from 100 to 2000. Patankar and Prakash (1981) argue that in this range of Reynolds number the real flow is expected to be mostly laminar, although it is possible that the transition to turbulence may occur somewhat before the Reynolds number is almost 2000 especially for the higher values of t/H. They also used a constant heat flow boundary condition with additional specification that each row of fins was at a fixed temperature. They made some analysis for different fin thickness ratios t/H= 0.1, 0.2, 0.3 at the same fin length given by L/H=1, and also they fixed the Prandtl number of fluid as 0.7 for the lower Reynolds number such as 100 and 500, they observed an impingement flow on the leading edge of the fin and a small recirculation zone behind the trailing edge for the higher Reynolds numbers such as 1000 and 2000, they also observed the recalculating flow fills the space between the trailing edge of one plate and the leading edge of next plate, and the impingement flow disappears. As a result of their investigation, it was concluded that the thick plate situation leads to significantly higher pressure drop while the heat transfer does not sufficiently improve despite the increased surface area and increased mean velocity.

Of above mentioned there are too many correlations for heat transfer and pressure drop characteristics obtained by lots of researchers. The reason of developing the predictive equations is that generally in most heat exchanger problems the working fluid, the heat flow rate and mass flow rate are usually known. If certain correlations between geometry and fin performance is also known, then the problem can be greatly simplified. For that purpose, developing the two correlation factors called fanning friction factor f and Colbourn j factor are important for heat exchanger designers. Prof. Shiriramshasri and Mr. Nalkande S.A developed a heat exchanger by made some different in the design of rectangular fin by added some holes inside the fins and give a significant difference in heat transfer rate at forced convection (Shiriramshasri et al., n.d.).

Chabane, Moummi, and Benramache (2014) conducted study of experimentally fin heat transfer performance with longitudinal fins of solar air heater. This work was done by test the effect of mass flow rate of air on the outlet temperatures, the heat transfer in the thickness of the solar collector and the thermal efficiency was studied. They found a substantial increasing in efficiency solar collector with and without fins.

Ganesha and Naveen Prakash (n.d.) used different voltage for three fins shapes [triangular rectangular, circular] at different air velocity. They found significant effects to the air velocity and the fin geometry to the heat transfer rate. Recently, Sadeq and Shehab (2020) did a research work on forced convection heat transfer rate by determining the influence of two different types of the inner finned tube (with rectangular interrupted fins or with trapezoidal interrupted fins) utilized. There results shown that the local Nusselt number for the case of trapezoidal interrupted fins is higher than that in the case of rectangular interrupted fins. As well as the forced-convection performance of the inner finned tube with trapezoidal interrupted fins in terms of average Nusselt number is more significant about 11% to 20.3% than that the rectangular interrupted fins.

2. Experimental Methods

In order to experimentally measure the thermal performance of the finned heat sink, it is essential that the rate of heat transfer between the heat sink and the flowing water be accurately measured at different position. For the present experimental studies of heat sink performance, electrical patch heaters were used to heat the fin base.

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Figure 1: The rig of experimental work

It would be highly desirable to be able to indirectly obtain an accurate measurement of the rate of heat transfer from the heat sink to the flowing of water, by simply measuring the temperature to the patch heaters. In order to experimentally measure the thermal performance of the finned heat sink, it is essential that the rate of heat transfer between the heat sink and the flowing of water be accurately measured. For the present experimental studies of heat sink performance, thermocouple type (K) was used to heat the fin base and the surface. It would be highly desirable to be able to indirectly obtain an accurate measurement of the rate of heat transfer from the heat sink to the flowing of water. Equating these two energy rates assumes a negligible heat loss of the backside and edges of the patch heater assembly, these two energy rates assume a negligible heat loss of the backside and edges of the patch heater assembly. Geometrical parameters of rectangular aluminum fins with code name at thermal conductivity equal to 330w\m K. There are 12 fins fixed at rectangular aluminum base with an electrical heater with heat supply up to 1250 w.

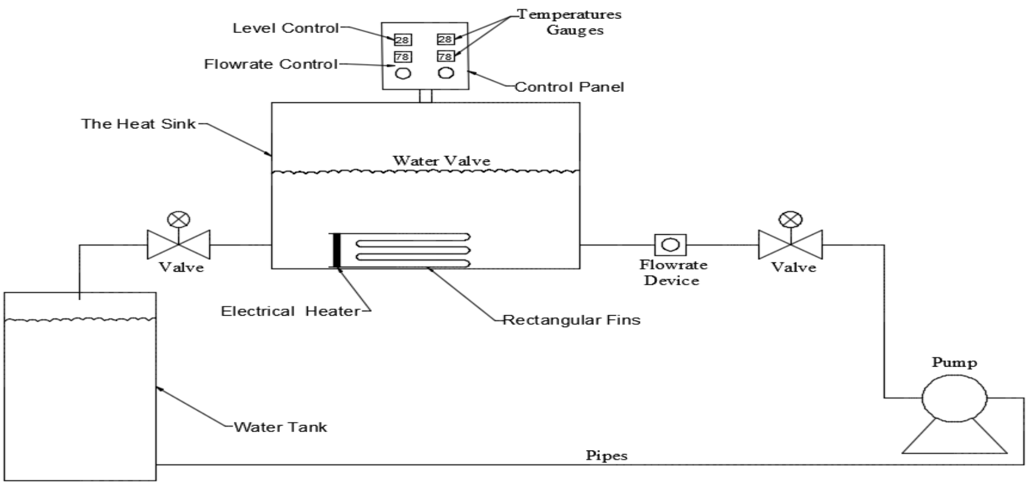

Figure 2: The schematic diagram of the rig experimental work

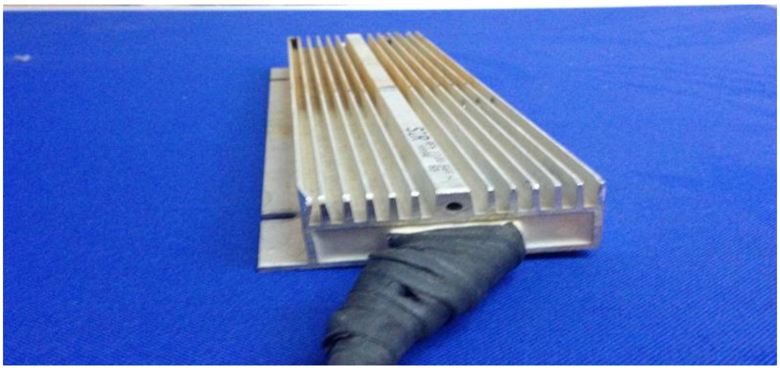

Figure 3: The experimental fins used

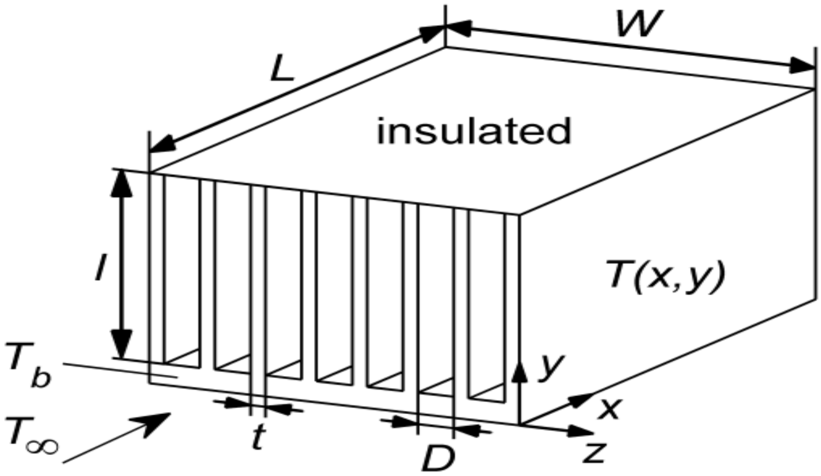

Figure 4: Rectangular fin parameters

Table 1: Fins properties

Geometrical fin Parameters	Dimensions (mm)
Fin Length (wf)	320
Fin Height (L_F)	10
Fin Thickness (t)	1
Fin Spacing (s)	5
Channel Length (L_c)	30
Fin Parameter (P)	688

Rectangular duct made of galvanized iron and Geometrical parameter of rectangular duct is used to examine the effects of water flow rate to the fin heat transfer performance for two positions to the array of fins.

Table 2: Duct dimensional

Duct Geometrical	Dimensional (mm)
Thickness (t)	1.5
Length of duct (L)	1500
High of duct (b)	400
Width of duct (a)	200

3. Results and Discussions

The aim of the present study is to show that it is possible to record the change on a heat transfer performance in the change Off-site strip fins positioned in the water system. Forced convection investigates turbulent water flow at of values (Re) from 4300 to 6000. The range of the fin base temperature (Tb) is from 130 to 150 oC for examining all the above parameters. The array of rectangular twelve fins heats transfer performance in the water phase at galvanized iron duct in two directions, examine different water volumetric flow rates from (0.5 to 2.8 L/s.) to the range of water temperatures from 26 to 30 Co. The results of constant hydraulic diameter (Dh, 0.42m) water phase to roughness galvanized iron illustrated are summarized in the next figures. We used a matlab language program for calculation and drawn all the following figures to the vertical and horizontal fin base position.

The relation of the Nusselt number (Nu) with Reynolds number (Re) is shown in figure 5. The range of (Nu) is from 4 to 6, this mean there is heat transfer by convection bigger than heat transfer by conduction of both fins base positions in the range of turbulent flow used in this work.

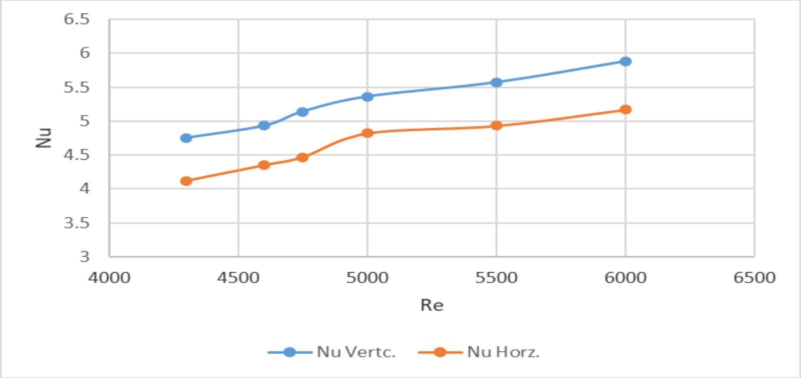

Figure 5: The relation of Re no. with Nu no. for vertical position (green curve) and horizontal position (blue curve) of array fin base at constant hydraulic diameter (Dh)

The heat transfer coefficient has a high value for this work due a good cooling system and there is a small difference in the result of changing the fin base position. The horizontal fins direction has a less value than the other for all ranges of turbulent water flow.

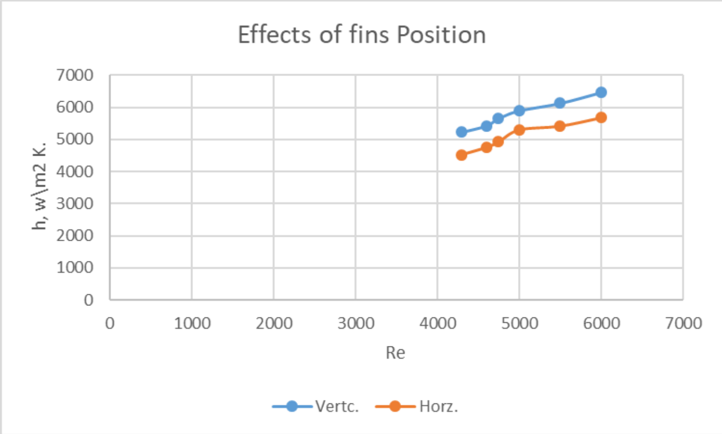

Figure 6: The relation of Re no. with a heat transfer coefficient for both directions of strip fins base in the case of turbulent flow. Where the vertical position (green curve) and horizontal position (blue curve)

The relation of (Re no.) with (Nu no.) for different volumetric flow rates and constant hydraulic diameter (Dh) is linear as illustrated in figure 4, for the range of flow rate from 0.5 to 2 L/s. A linear relation obtained between the (Re) and the heat transfer coefficients (h) as shown in figure 6. The range of (h) is from 4500 to 6500 in this experimental work. The heat flux of horizontal direction is bigger than vertical. This because of, the decreasing the heat transfer coefficients at different volumetric flow rate of turbulent flow. The heat transfer ratio (Qwater \ Q fin) with (Re) is first represented in the literature survey shown as a linear relation. It is clear from figure 7, the big difference happens in the value heat transfer ratio at the beginner of turbulent flow in water cooling system.

Performance of heat transfer for both locations of off-site strip fins measured by efficiency, effectiveness and heat transfer coefficient of these fins. This is illustrated in figures 8 and 9. The range of efficiencies of rectangular fins is from 94 to 99 % and the effectiveness of fins has a range from 20 to 21 is too high, because the heat supply is low 1250w with enough number of fins (12 fins) and suitable hydraulic diameter in this range of turbulent flow. The effectiveness is larger than 20 that is mean the fins used are being considered justifiable. Also, the efficiency of fins is too higher than 94%, this means an excellent cooling down by the water flow system.

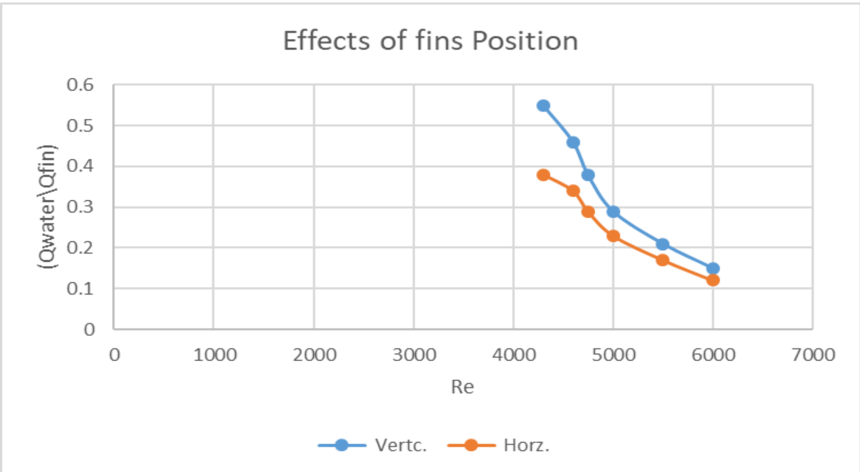

Figure 7: The relation between the fin heat transfer ratio [Q water\ Q fins] with range of [Re] to the turbulent flow at vertical position (green curve) and horizontal position (blue curve)

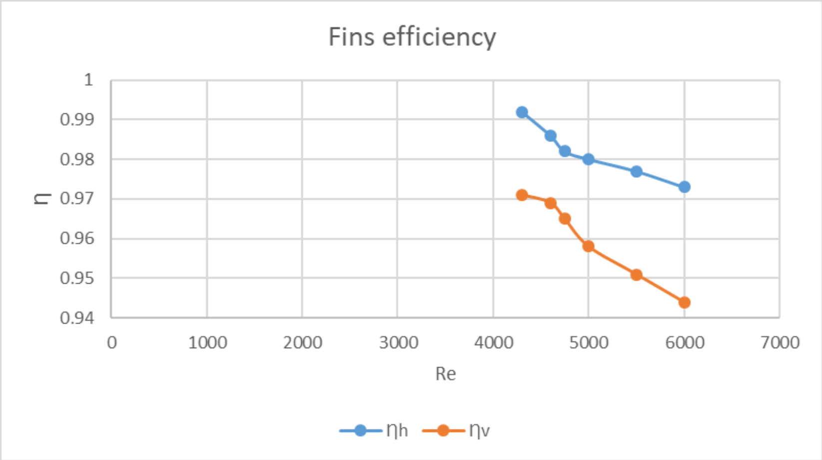

Figure 8: The relation between the fin efficiency (Ŋ) with range of turbulent flow for vertical position (green curve) and horizontal position (blue curve) of array fins base

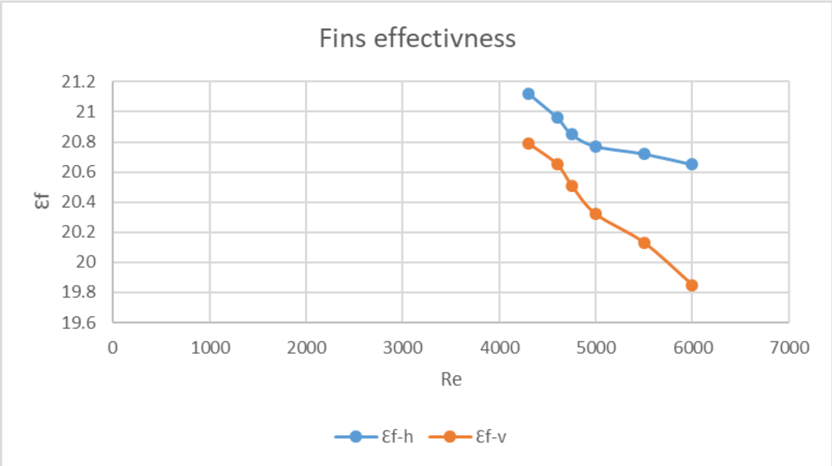

Figure 9: The relation between the fin effectiveness (Ɛ f) with range of turbulent flow for vertical position and horizontal position of array fins base

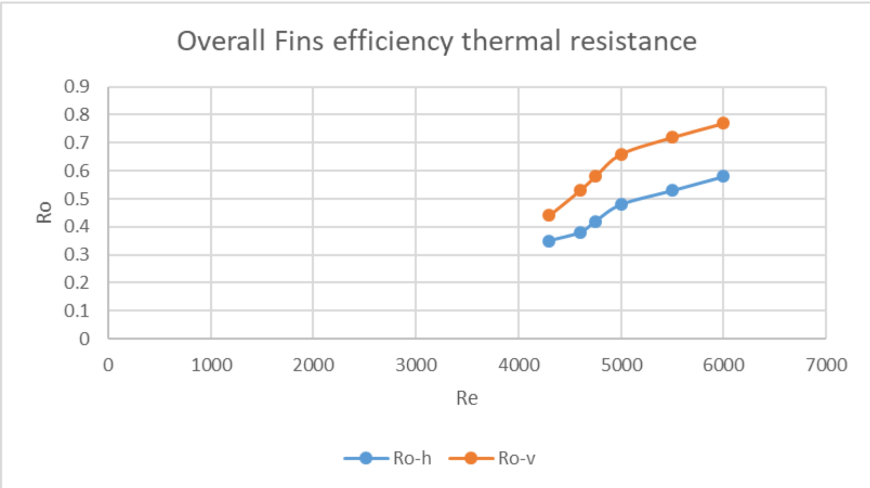

Figure 10: The relation between the overall fins Resistances (Ro) with a range of turbulent flow for vertical position and horizontal position of array fin's base

It is clear from the above figures that the fin heat transfer performance decrease with increasing the Re. number at constant hydraulic diameter (Dh) for both positions of rectangular array fin. The effectiveness and the efficiency of fins have a big difference with increasing the Re. number for both positions of array of rectangular fins.

The Stanton number, St, is a dimensionless number that measures the ratio of heat transferred into a fluid to the thermal capacity of fluid. It is used to characterize the heat transfer in forced convection flows:

$$St = \frac{h}{G \times C_p} = \frac{h}{\rho \times u \times C_p} \qquad [5]$$

where:

h = convection heat transfer coefficient

ρ = density of the fluid

Cp = specific heat of the fluid u = speed of the fluid

It can also be represented in terms of the fluid's Nusselt, Reynolds, and Prandtl numbers: A more formal definition is.

$$St = Nu \,/\, (Re \times Pr) \qquad [6]$$

Where Nu is the Nusselt number, Re the Reynolds number and Pr the Prandtl number. The Prandtl number of waters at a range of temperature from 25 to 30oC is about 6.

So, the range of Stanton number from 0.00015 to 0.0002 for this research work to both positions of off-site fine stripe. The following figure represents the relation of Re.no. with Stanton no.

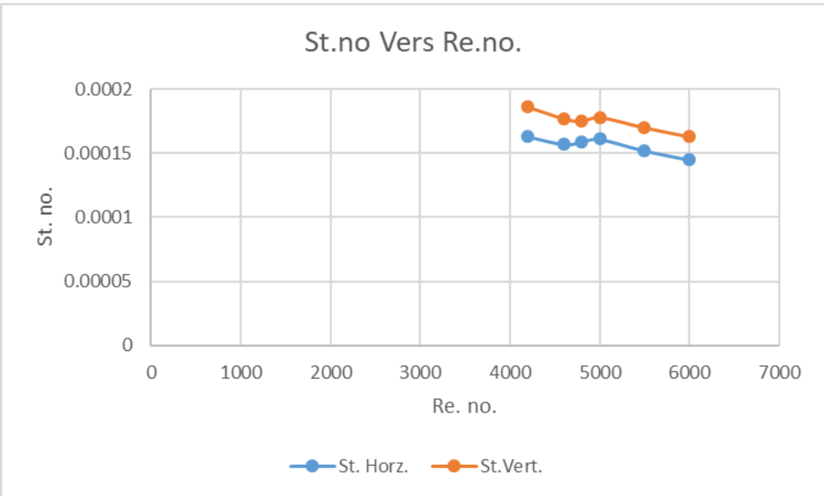

Figure 11: The relation of St.no. with Re. no. to the both fin strips position

4. Conclusion

The following conclusions can be drawn from this research work:

1. The heat transfer performance for vertical position is less than the horizontal position in the range of turbulent water flow.
2. The fin heat transfer performance decrease with increasing the water turbulent flow at constant hydraulic diameter (Dh) for both positions.
3. Nusselt number increase with increasing the Reynolds for off-site strip of fins in both positions.
4. Stanton number decrease with increasing the Reynolds for off-site strip of fins in both positions.

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