PREDICTION OF THE GLOBAL PERFORMANCE OF A FINNED TUBE AIR COOLLER AND EFFECT OF RADIAL SLITS ON ITS FIN SURFACE

S. RAVI ANNAPRAGADA
VIJAY R. RAGHAVAN

13TH INTERNATIONAL CONFERENCE
ON HEAT TRANSFER
13 – 18 AUGUST 2006
SYDNEY, AUSTRALIA

PREDICTION OF THE GLOBAL PERFORMANCE OF A FINNED TUBE AIR COOLER AND EFFECT OF RADIAL SLITS ON ITS FIN SURFACE

S. Ravi Annapragada[1], Vijay R. Raghavan[2]
[1]Purdue University, USA
[2] College University Technology Tun Hussein Onn, Malaysia

Abstract

In the present work, numerical modeling was used to study the local behavior of flow and energy variables to understand the underlying physics in a Finned Tube Cooler. The cooler under consideration is a single circular-fin air cooler tube confined in a duct. The optimizations of fin pitch and fin clearance as predicted by Rangan *et al.* (2003) have been elucidated based on the underlying physical principles involved. The model was further applied to study the augmentation of heat transfer and suppression of pumping power due to symmetric radial slits on the fin surface. Reversal of heat transfer direction is seen in the wake of the tube in the case of an unserrated fin. Radial slits reduce this region by aiding mixing of the flow. The heat transfer performance of the fin is enhanced by 5% by placing symmetric radial slits at the onset of flow separation in comparison against an unserrated heat exchanger. But the pumping power requirements also increase at a similar rate.

Nomenclature

β_{ang}	Angle measured from forward stagnation point	m_f	Mass flow rate in fin space
τ	Non-dimensional time	Nu	Nusselt number (hD_T/k)
D_f	Outer diameter of fin	Re	Reynolds number ($\rho U_\infty D_T/\mu$)
D_T	Tube diameter	U_∞	Approach velocity
m_c	Mass flow rate in clearance	δ_f	Fin space
		δ_c	Clearance space between fin and duct

1. Introduction

Heat transfer and the design of heat transfer equipment continue to be a centrally important issue in engineering. In recent decades, there has been increased emphasis on the optimal use of energy and, with increased energy cost, efficient heat transfer has become of vital importance. Moreover, with the recent global promotion of energy efficiency and protection of the environment, heat transfer in efficient utilization of energy plays a major part particularly for energy intensive industries, such as electric power generation, petrochemical, air conditioning, refrigeration, cryogenics, food, and manufacturing. Almost all chemical, process and power plants depend upon the application of heat to produce physical and chemical changes, or to do work. An understanding of the mechanisms of heat transfer and an ability to predict heat transfer rates are essential to the task of building modern heat exchangers. Heat exchangers which have finned tube structure are called finned tube coolers.

Finned tube coolers have been studied for a long period of time. But most work has been experimental and for simplified cases, analytical. Experiments to obtain local behaviour have to be non-intrusive as any probe inserted in the narrow space would change the actual behavior of the flow being studied resulting in incorrect predictions. Hence in most studies, only global predictions are made without much detail about the local behavior. With the increase of computational capabilities of current day computers, a few groups took up the numerical analysis of finned tube

10 m/s

coolers to understand the local phenomena. The effects of a fin on the flow characteristics and heat transfer mechanism related to the vortical structure (horseshoe shaped) were studied numerically by Lee *et al.* (2004) using Fourier-Chebyshev spectral simulations. Krikkis and Razelous (2004) proposed a methodology for design of an optimal fin for a tubular radiator using numerical optimization techniques. The local behaviour, however, was not captured. Shew and Tsai (1999) studied the effect of modifying the fin surface on heat transfer and pumping power requirements of a plate fin tube cooler in a staggered configuration based on the local behavior using a finite volume discritization method and SIMPLE-based solution algorithm. The influence of Reynolds number on the conjugate heat transfer behavior of a finned tube cooler and the augmentation of heat transfer by using vortex generators was investigated by Fiebig *et al.* (1995) numerically. There is a need to understand a single circular finned tube heat exchanger and the influence of a radial serration on the local heat transfer behavior near the fin surface. Such an analysis would help us in designing efficient heat exchangers based on the underlying local phenomena.

The present study makes an effort to understand the heat transfer behavior and pumping power requirements of such a single circular-finned cylindrical tube at a fundamental level. Heat transfer in this case is more complicated than heat transfer from smooth tubes. This is due to the conjugate nature of the fluid flow over the fin surface and the core tube, the effect of non-uniform temperature distribution over the fin, the effect of the complex geometry of finned tubes, and other factors. These factors interact closely with one another and make it difficult to analytically solve the heat transfer from finned surfaces. Therefore a numerical formulation has to be applied. Rangan *et al.* (2003) optimized the fin spacing and fin clearance based on global predictions using finite volume discritization with PISO method of velocity-pressure coupling. In this work, we seek to explain these predictions based on the underlying heat transfer principles and understand the local behaviour in each case. We further extend the model to investigate the heat transfer performance and pumping power requirements due to the introduction of slits/serrations on the fin surface.

Figure 1: Schematic diagram of the computational domain.

2. Modeling

The computational domain considered in this analysis is shown in Figure 1. The dimensions and the initial conditions chosen are taken to be typical of that of commercial applications such as cross flow finned air-coolers. The velocity of air along the major flow direction (positive x-direction as seen in Figure 2) is taken to be 2.5m/s. The velocity in the other directions at inlet is taken to be 0, representing a plug flow. The inlet temperature of air is taken as 293K and the temperature of the inner wall of the tube is taken to be a constant 323K. This corresponds to a case where there is liquid/vapor phase change of the fluid inside the tube. Heat flow through the inner tube, the inner face of the fins and the fin tips are studied. All the dimensions, except the fin spacing and fin clearance, are kept constant. The relevant dimensions are marked in the Figure 2 and their values are given in Table 1.

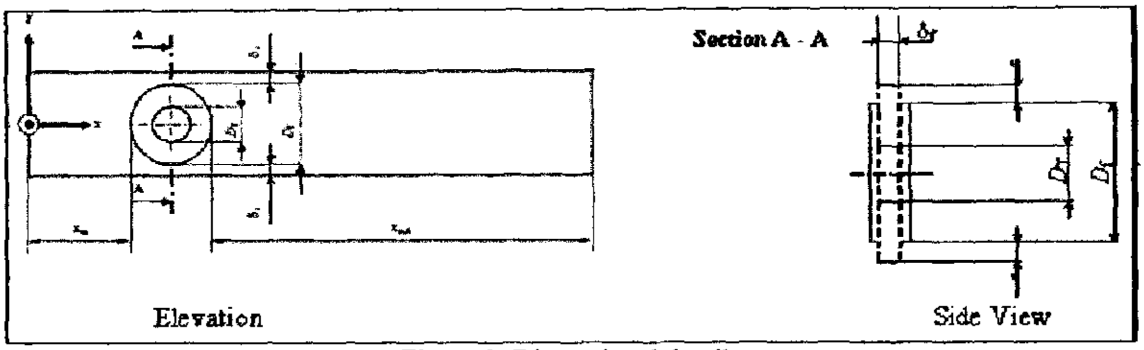


Figure 2: Dimensional details

Table 1: Domain Dimensions

Parameter	Value (mm)
D_T	25
D_f	57
X_{in}	$\approx 1.25.D_f = 80$
X_{out}	$\approx 6.5.D_f = 400$
δ_f	variable (1.5, 2, 2.5)
δc	variable (1, 2, 3, 4, 5)

The fluid faces on the sides in the z-direction are made periodic such that the conditions on the left wall are reflected on the right wall since the model is one of a series of fins. As the computational domain passes through the middle of the fin, the fin mid-plane can be considered insulated on account of symmetry. The entire duct is taken into account, as the duct mid-plane cannot be assumed to have symmetry, due to vortex shedding. The upper and lower walls are considered as real walls and the no-slip condition is applied to them.

The variation in the fin clearance and the fin spacing leads to fifteen different configurations for which models were created and analyzed. Block structured grids were used to discretize the domain. The details of the geometry and grid independence study are discussed in Rangan *et al.* (2003).

The governing mass, momentum and energy equations are:

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho\vec{v}) = 0 \tag{1}$$

$$\frac{\partial}{\partial t}(\rho\vec{v}) + \nabla.(\rho\vec{v}\vec{v}) = -\nabla p + \rho\vec{g} + \nabla.(\bar{\bar{\tau}}) \tag{2}$$

$$\frac{\partial(\rho h)}{\partial t} + \nabla.(\rho \vec{v} h) = \nabla.(k \nabla T) \tag{3}$$

The stress tensor is given by $\bar{\tau} = \mu\left[(\nabla\vec{v} + \nabla\vec{v}^{T}) - \frac{2}{3}\nabla.\vec{v}I\right]$ where I is a unit tensor. Fluent Version 6.0, a commercial CFD software package based on Finite Volume Method, is used for solving the above equations. The laminar model is chosen since the Reynolds number of this problem (Re = 3500) is in the sub-critical range as shown in Jalaiah and Raghavan, (2002)]. The characteristic length is chosen to be the tube diameter. As the flow is bound to shed vortices, unsteady formulation is used. The segregated scheme is used which solves the momentum and energy equations individually. The Pressure-Implicit Splitting of Operators (PISO) method, which is the most suitable for time dependent problems, is used for the pressure-velocity coupling. For better accuracy, the quadratic upwind interpolation scheme is used in the solver. The standard properties for the fluid medium (air) and the solid medium (aluminum fin) are taken.

3. Fin Spacing and Fin Clearance Optimization Study

The effect of varying fin spacing and fin clearance was studied by varying the fin spacing from 1mm to 5 mm in steps of 1 mm and fin clearance of 1.5 mm to 2.5 mm in steps of 0.5 mm. Fin clearance does not affect the performance of the heat exchanger within the range of values analysed. The averaged results of Rangan *et al.* (2003) show that fin spacing between 2mm and 3mm gives the best heat exchanger performance. Figure 3 shows the same for different fin clearances. These results are best understood by analysing the local velocity, temperature, Nusselt number behavior, as we vary the fin spacing.

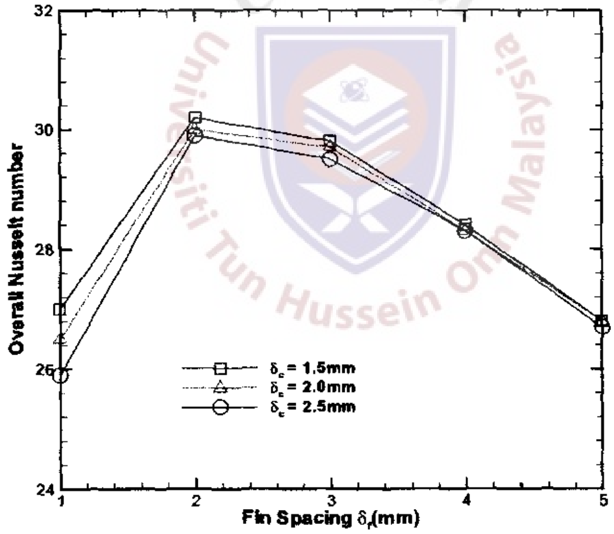


Figure 3: Nusselt number variation as a function fin spacing for various fin clearance values

3.1 Flow Behavior

The fin spacing has a strong impact on the flow characteristics in the heat exchanger. Ideally we would like high velocities of fluid over the fins which will lead to better heat removal. At a low fin spacing of 1 mm, the flow between the fin spacing downstream of the tube is stagnant causing a loss of performance due to low velocities as seen in Figure 4. The recirculation region is outside the fin spacing. This also implies that a fin spacing lower than 2mm would lead to a large pumping power requirement due to the stagnation flow downstream of the tube. As little flow penetrates the fin space, this can be visualized as the flow treating the entire fin-tube as a bluff body. As we increase the fin spacing, the recirculation zone moves closer to the tube surface and into the fin space region. For better heat transfer from the fin to the bulk fluid, the heated fluid near the fin needs to be replaced by cooler bulk fluid. A recirculating zone in between the fins would increase the velocities in the region of importance for heat transfer. This would thence result in better heat removal from the fin and tube assembly.

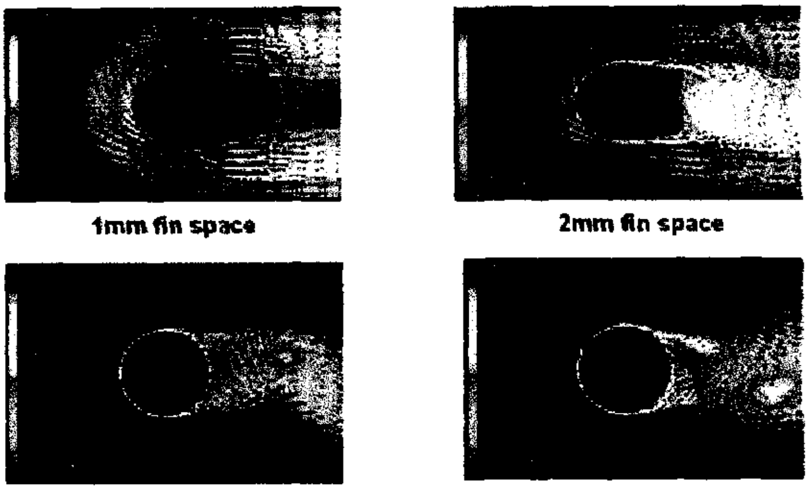
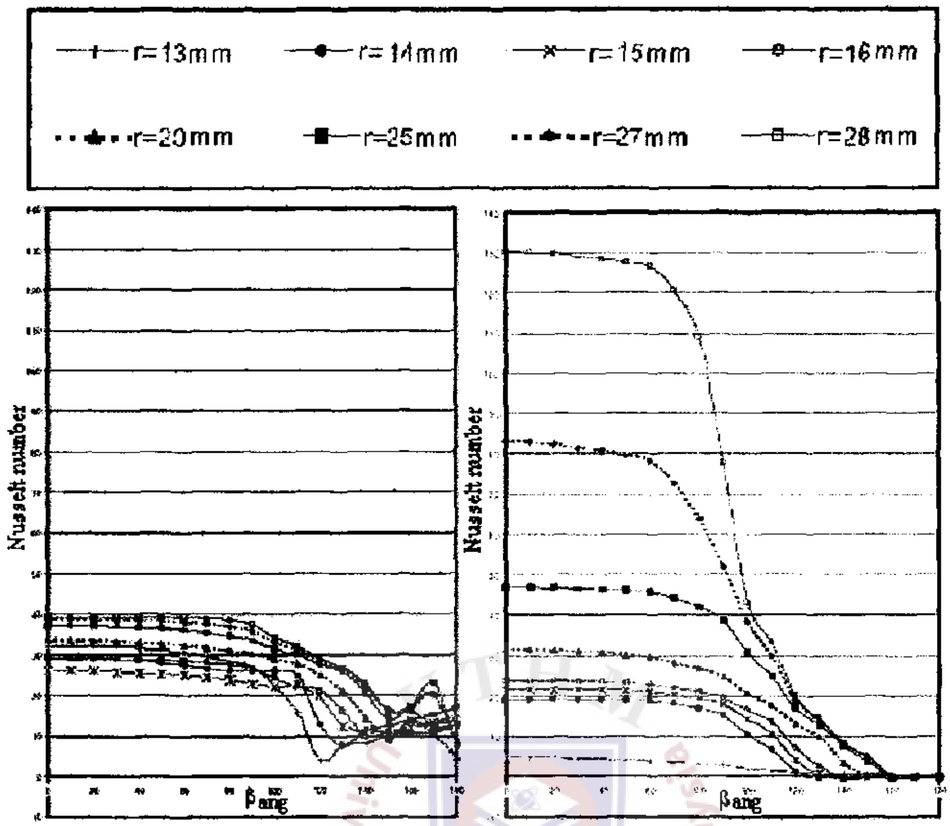


Figure 4: Velocity vectors and temperature contours at the mid-plane for various fin space and 2mm clearance spacing

3.2 Heat Transfer Behavior

The heat transfer characteristics are best understood by looking at the local temperatures of the fluid between the fin spacing and the local Nusselt numbers near the fin surface. The temperatures on a plane through the centre of the fin spacing give us indirect information about the heat transfer between the fluid and the fin structure. A lower temperature of fluid within the fin spacing is sought as it leads to better heat transfer performance for the same Nusselt numbers. But at the same time, a lower temperature means less heat has been removed from the fin surface upstream of that point. So there are two competing factors governing the heat transfer behavior of the heat exchanger based on temperatures. Hence in a good heat exchanger we would see a low fluid temperature in the fin spacing region, upstream of the tube. Also the bulk fluid downstream of the heat exchanger would have high temperatures signifying high heat removal from the fin-tube assembly. From Figure 4 it is evident that decreasing fin spacing causes an increase in the overall temperature of the fluid within the fin spacing. The upstream temperatures are not affected much unless we go below a fin spacing of 2mm. The downstream temperatures show that higher temperatures are seen as we reduce the fin spacing and that 2mm -3 mm fin spacing seems to be the ideal range for fin spacing.

The plots of local Nusselt number as a function of radius and angle over the fin spacing paint a similar picture. The distribution of local instantaneous Nusselt numbers are presented in Figure 5(a) and Figure 5(b) for the maximum and minimum values of fin space, viz., 5 mm and 1 mm, as a function of angle β_{ang}, measured from the forward stagnation point. In general, it is seen that Nusselt numbers are highest at the forward stagnation point and decrease downstream of the cylinder. At small β_{ang} values the Nusselt numbers are a maximum at the outermost periphery of the fin, begin to decrease towards the tube surface, reach a minimum at a certain radius and again increase towards the tube surface. It can be stated in general that at all radii, a minimum occurs beyond β_{ang}=80°, at the rear of the tube. This is true for fin spacing of 2 mm and above. For fin spacing of 1 mm, a distinct departure in behaviour is observed, wherein the instantaneous Nusselt numbers decrease monotonically with increasing β_{ang}. This confirms the trends observed in temperature contours (Figure 4) and Nusselt number plots (Figure 5) for this case.



(a) (b)

Figure 5: Nusselt number plots for various radii for 2.5mm clearance and (a) 5mm fin space (b) 1mm fin spacing as a function of angle β_{ang}

Some other important characteristics need to be taken note of. For all fin spaces, the outer radius r=28 mm in the range of β_{ang}=0° to 90°, is clearly affected by the incipience of the boundary layers on the fins. While the highest local Nusselt numbers occur for 1mm fin space, typically at the forward edge of the fin, these high values occur over a much smaller fin area as compared to the 5mm fin spacing. Consequently, the average Nusselt number is highest for fin spacings of 2 mm and 3 mm as predicted in Rangan *et al.* (2003).

4. Performance Enhancement due to Introduction of Radial Slits on Fin Surface

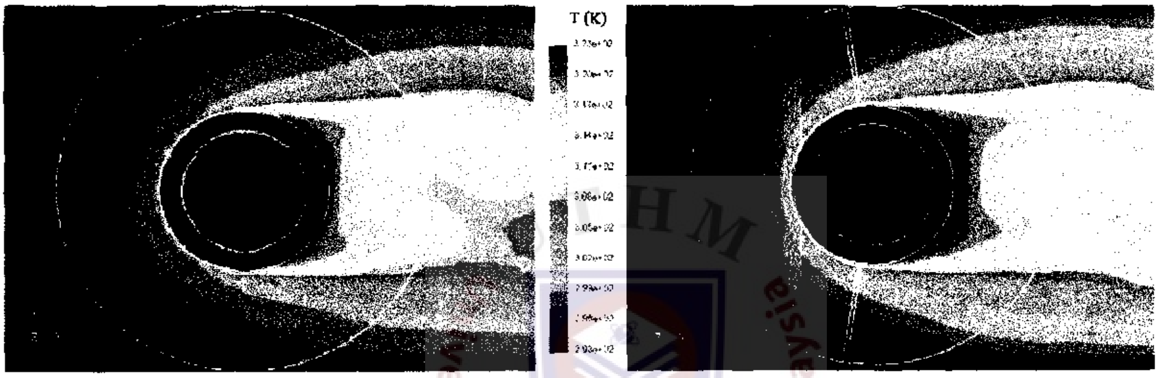
Once the geometry is optimized to provide the best heat transfer performance with minimal pumping power requirement there is a need to develop innovative ways to increase the heat exchanger performance. One such modification is introduction of radial slits on the fin surface. Slits could have a positive impact of increasing heat transfer performance by acting as flow trippers which would restart the boundary layer and increase mixing. Among the various kinds of slits, radial slits are easy to fabricate as such slits could be established on a sheet metal strip which is wound on the tube as helical (radial) fins. For sub-critical flow of Re = 3500 in the examined problem, the separation angle will be around β_{ang} = 80°. This, as seen from Figure 5 also corresponds to the onset of decreasing Nusselt numbers. Hence making a symmetric slit at around

β_{ang} = 80° seems logical. The geometry of the finned cooler with the slit is shown in Figure 6. Direct Numerical Simulations were run also for a β_{ang} = 60°. The rationale for this is presented later.

Symmetric slits are introduced on the top and bottom of the fins of a base case - the geometry which gives the best performance. The base case has a fin spacing of 2mm and a clearance of 2.5mm from the wall. The serrations are symmetric about the XZ-plane and are at an angle β_{ang} = 80° on the top and the bottom. Numerical Simulations were run with the same boundary conditions as the base case to simulate the above cases with slits. The effect of introducing these radial slits on the performance of the heat exchanger is measured against the base case. To have a realistic comparison all snapshots are shown below are at non-dimensional time (τ) = 650. The non-dimensional time is defined as:

$$\tau = \frac{t_{actual}}{\frac{D_{oT}}{U_\infty}} \tag{4}$$

where t_{actual} is the time in seconds and D_{oT} and U_∞ are as defined in the nomenclature.

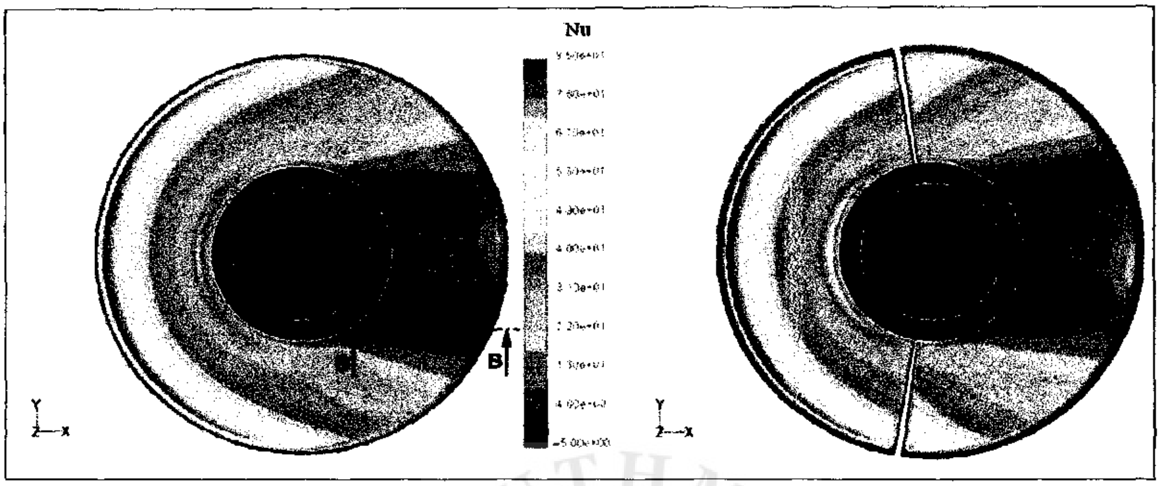


(a) (b)

Figure 6: Temperature contours on fin surface (a) unserrated and (b) with serration at β_{ang} = 80°

A comparison of the mid-plane temperatures between the cases with slits and no slits, as seen from Figure 6, show that the slits do not have an influence away from the fin surface. The snapshots are taken at times close to each other so that the comparison is meaningful. The fin boundary is shown in white. The serration has an influence only on the local properties. Any minor local changes of the fin contour do not affect the mid-plane flow and temperature profiles. The flow with increased velocities in the core, sweeps away the local changes due to the irregularities near the fin surface and hence the influence fades out after a particular distance from the fin surface. This distance is seen to be 0.375mm in this case.

As the mid-plane temperature contours show that the serrations only affect the local characteristics near the plane, it is logical to look at the local behavior close to the fin surface. Local Nusselt number contours are shown for the base case and the case with serrations at β_{ang} = 80° in Figure 7. The fin diameter has been used as the reference length in calculating the Nusselt numbers. Figures are shown on a common scale to allow better comparison. Since the flow and heat transfer characteristics are symmetric about the mid-plane, results from only one fin surface are presented. Figure 7(a) shows a large blue area signifying a region of negative heat transfer coefficient. That is, there is reversal of temperature difference. The temperature of the bulk fluid is higher than the temperature of the fin. This means that the heat is transferred from the fluid to the plate. This is highly undesirable. The reason for such a phenomenon is that the fluid which is in contact with the tube is heated to a higher temperature. Simultaneously, the temperature of the fin drops from the value at the base of the fin along the radius. Due to these combined effects, at certain regions in the wake of the tube, the unusual situation of a local reversal of temperature difference is witnessed and hence reversal in direction of heat transfer occurs, i.e., from the bulk fluid to the fin as observed in Figure 7(a). The temperature at the section B-B marked in Figure 7(a) corroborates the same as seen in Figure 8. The core temperatures are higher as compared to the temperature at the fin surface and hence the heat transfer direction is from the fluid to the fin. Such regions of reversed heat transfer have been discovered before and sited in the literature. Fiebig, M. *et al.* (1995) termed these regions as a 'dead water zone' as the working fluid in their case was water.



(a) (b)

Figure 7: Local Nu Contours on fin surface of (a) unserrated fin (b) fin with serration at $\beta_{ang} = 80°$

Figure 8: Temperature contour at section B-B of Figure 7(a).

This region of reversed heat transfer direction, also called 'deadzone', is seen to be reduced by the introduction of serrations. The serrations trip the boundary layer before the flow reaches the region of low and reversed heat transfer. This makes the temperature of the bulk fluid more uniform over the width of the flow and hence reduces the region of reversed heat transfer.

Figure 9 shows the radially averaged Nusselt number as a function of β_{ang} for unserrated case and cases with serrations at $\beta_{ang} = 80°$ and $\beta_{ang} = 60°$. Radially averaged Nusselt number depicts the variation of the heat transfer characteristics obtained from the finned cooler. The case with a serration at $\beta_{ang} = 80°$ shows that the fall in Nusselt number has already set in and the serration does not salvage much. Hence a serration at $\beta_{ang} = 60°$ has also been experimented with. There is improvement near the serration but further away the values drop drastically as seen from Figure 9.

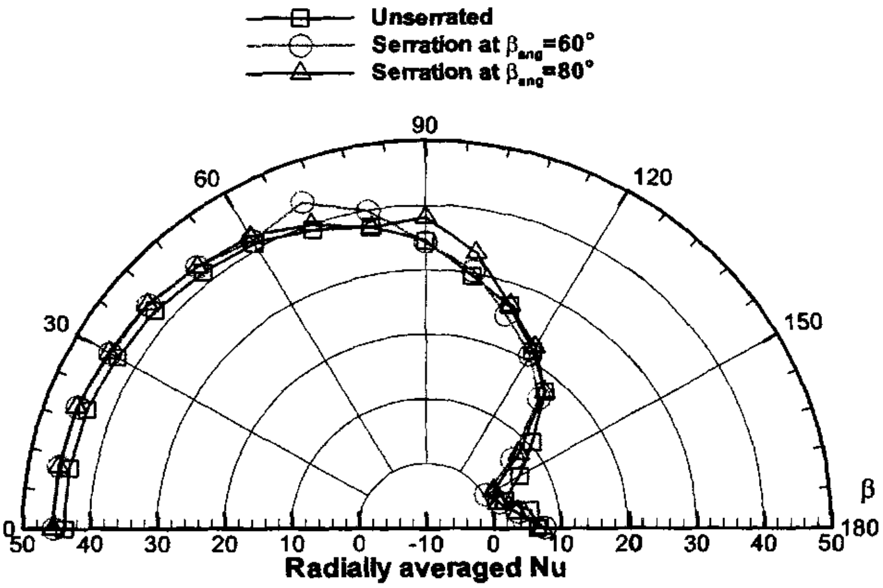


Figure 9: Radially averaged Nu on Unserrated fin, fin with serrations at $\beta_{ang} = 60°$ and $\beta_{ang} = 80°$

To understand the global performance of the finned tube cooler, space averaged values need to be compared. The ideal configuration should maximize the Nusselt number and minimize the Skin friction coefficient. Both of these are incorporated into a single value by taking the ratio of Nusselt number and skin friction coefficient (f). These space averaged results are shown in Figure 10.

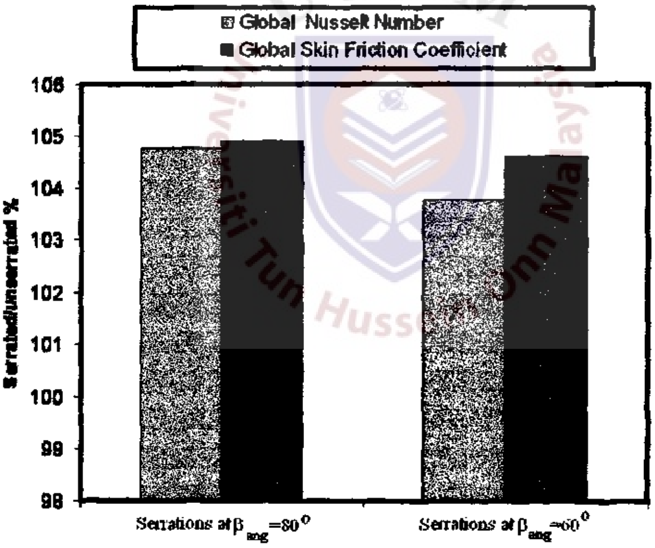


Figure 10: Space averaged Nusselt Number and Skin friction Coefficient for fins with serrations at $\beta_{ang} = 60°$ and $\beta_{ang} = 80°$ as a percentage of values of unserrated fin

In the present cases experimented with, the performance with serrated fins with serrations at β_{ang}=80° seems to enhance heat transfer performance but also at the cost of slightly increased pumping power. The serrated fin with serrations at $\beta_{ang} = 60°$, though seemingly promising, loses out on performance and requires high pumping power.

5. Conclusions

The local distribution of temperature and Nusselt numbers confirm the general observations made in Rangan *et al.* (2003). The fin clearance has little effect on the heat transfer characteristics. The fin spacing has been found to be the driving parameter. The vortex shedding region changes as we reduce the fin spacing to 1mm.

Reversal of heat transfer direction takes place in the wake of the tube in the case of optimized unserrated case. Serrations reduce this area of reversed heat transfer direction by aiding in the mixing the flow property and hence making the temperature more uniform at a cross-section. Among the two serrations, the 80° serration seems to be the better one. One explanation for this is that the drop in heat transfer coefficient starts taking place from 80° and hence it is advisable to place a serration to interrupt the boundary layer at that particular angle.

The present study took into consideration only 2 slits. It would be interesting to see what additional slits would do to the performance of the fin tubed cooler. The geometry of the serration itself can be varied in terms of length, width, shape of edge and projection height. It is seen that some areas downstream of the tube have fluid temperatures higher than that near the fin surface and that there is reversal in the direction of heat transfer. One way to eliminate the negative heat transfer in these areas is to remove the fin surface in these areas. The combined benefits of higher heat transfer coefficients in the forward part of the fin and elimination of reversed heat transfer could yield considerable monetary savings. Longitudinal vortex generators have been used to tackle this problem of 'dead zones'. The vortex generators help in diverting the flow into the tube wake region and hence reduce these 'dead zones'. The area of fin which experiences this 'dead zone' phenomenon can be physically removed. This would also reduce the weight and hence cost of the fin tube cooler. But additional studies need to be performed on fin stability and vibrations due to reduction in fin tube stiffness consequent upon the partial removal of fin area, before such an approach is considered.

6. References

Fiebig, M., Grosse-Gorgemann, A., Chen, Y. and Mitra, N.K., 1995, Conjugate heat transfer of a finned tube Part A: heat transfer behavior and occurrence of heat transfer reversal, Numerical Heat Transfer; Part A: Applications, 28, n 2, 133-146

Fiebig, M., Chen, Y., Grosse-Gorgemann, A. and Mitra, N.K., 1995, Conjugate heat transfer of a finned tube Part B: heat transfer augmentation and avoidance of heat transfer reversal by longitudinal vortex generators, Numerical Heat Transfer; Part A: Applications, 28, n 2, 147-155

Jalaiah, N., and Raghavan, V.R., 2002, Effects of blockage on flow and heat transfer over a tube in cross flow, Proc., 12th International Heat Transfer Conference, Grenoble, France, 2002, 711-716.

Krikkis R. N. and Razelos P., 2004, On the Optimization of Circular Radiating Fins with Fin-to-Fin and Fin-to-Base Radiant Interaction, J. Heat Transfer, 126, 134-137

Lee, D.H., Ha, M.Y., S. Balachandar and Lee, S., 2004, Numerical Simulations of Flow and Heat Transfer past a Circular Cylinder with a Periodic Array of Fins, Physics of Fluids, 16, n 5, 1273-1286

Rangan, B.K., Krishnamurthy, A. and Raghavan, V.R., 2003, Analysis of flow and heat transfer at a finned tube in crossflow, Proc. ASME Summer Heat Transfer Conference, 2003, 917-922

Shew, T.W.H. and Tsai, S.F., 1999, A Comparison Study on Fin Surfaces in Finned-Tube Heat Exchanger, Intl. J. Numerical Methods for Heat & Fluid Flow, 9, n 1, 92-106