I, Madhura Shreeram Karve, hereby submit this original work as part of the requirements for the degree of Master of Science in Mechanical Engineering.

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Numerical Analysis of Heat Transfer Enhancement and Pressure Drop Reduction for an A-frame Air Cooled Steam Condenser

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ABSTRACT

Air cooled steam condensers (ACSC) consist of finned-tube arrays bundled in an A-frame structure. Inefficient performance under extreme temperature operating conditions is a common problem in ACSCs. The purpose of this study was to improve the heat transfer characteristics of an annular finned-tube system for better performance in extreme climatic conditions. Perforations were created on the surface of the annular fins to attain maximum increase in heat transfer coefficient (h) and minimum increase in pressure drop (ΔP). Numerical computations were performed using finite volume method with RNG k-ε turbulent model to calculate h and ΔP. Solid (no perforations) finned-tubes were simulated with free stream velocity ranging between 1 m/s – 5 m/s and validated with the published data. Computations were performed for perforated (perforations at ±60° to ±180° with 30° interval) finned-tubes, followed by five cases of a single perforation and three cases with multiple perforations. Later, the spacing of the fins along the arms of A-frame ACSC was altered to decrease ΔP across the finned-tube array. The increase in h and ΔP values for perforated fins was 7% and 12% respectively as compared to the solid fins. Perforation located in the wake area, at 120°, gave favorable results with 2.2% increase of h and only 1.4% increase in ΔP. Multiple perforations (at 120°, 150°, 180°) resulted in 4.9% increase in h and 3.9% increase in ΔP. Perforations in the downstream region, at 120°, 150° and 180°, resulted in better heat transfer rates as compared to the upstream region. Fin spacing in the A-frame structure with sparsely spaced fins in the center resulted in 1.8% reduction in ΔP. Results showed that perforation at 120° had balanced results with maximum h and minimum ΔP. Multiple perforations can be used to obtain higher h values, provided the increase
in $\Delta P$ is allowable. Also, additional reduction in $\Delta P$ can be achieved by using clustered fin spacing at the inlet and top ends of A-frame structure and large spacing at the center.
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## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Total surface area (m$^2$)</td>
</tr>
<tr>
<td>$A_c$</td>
<td>Frontal area of the</td>
</tr>
<tr>
<td>D</td>
<td>Outer diameter (m)</td>
</tr>
<tr>
<td>E</td>
<td>Total Internal energy (m$^2$/s$^2$)</td>
</tr>
<tr>
<td>Eu</td>
<td>Euler number</td>
</tr>
<tr>
<td>f</td>
<td>Friction factor</td>
</tr>
<tr>
<td>G</td>
<td>Mass flow rate (kg/m$^2$-s)</td>
</tr>
<tr>
<td>g</td>
<td>Gravitational force (kg/s$^2$)</td>
</tr>
<tr>
<td>h</td>
<td>Heat transfer coefficient (W/m$^2$-K)</td>
</tr>
<tr>
<td>k</td>
<td>Thermal conductivity (W/m-K)</td>
</tr>
<tr>
<td>k_{k}</td>
<td>Turbulent kinetic energy (m$^2$/s$^2$)</td>
</tr>
<tr>
<td>l</td>
<td>Fin length (m)</td>
</tr>
<tr>
<td>n</td>
<td>Number of tube rows</td>
</tr>
<tr>
<td>P</td>
<td>Static pressure of air (Pa)</td>
</tr>
<tr>
<td>s</td>
<td>Fin pitch (m)</td>
</tr>
<tr>
<td>T</td>
<td>Temperature (K)</td>
</tr>
<tr>
<td>t</td>
<td>Fin thickness (m)</td>
</tr>
<tr>
<td>V</td>
<td>Axial velocity of air (m/s)</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt number = $h \times D / k$</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl number = $\mu \times C_p / k$</td>
</tr>
<tr>
<td>Re_{in}</td>
<td>Reynolds number = $\rho \times V \times D / \mu$</td>
</tr>
</tbody>
</table>
$S_L$  Longitudinal pitch of finned tubes

$S_T$  Transverse pitch of finned tubes

**Greek symbols**

$\varepsilon$  Turbulent energy dissipation (m$^2$/s$^3$)

$\upsilon$  Specific volume (m$^3$/kg)

$\omega$  Specific dissipation rate (1/s)

$\rho$  Density of air (kg/m$^3$)

$\sigma = \left( \frac{A}{A_C} \right)$

$\mu$  Viscosity of air (kg/m-s)

$\theta$  Angular location, degrees

$\tau$  Shear tensor

$\nabla = \frac{d}{dx} i + \frac{d}{dy} j + \frac{d}{dz} k$

**Subscripts**

$\text{eff}$  Effective

$max$  Maximum

$t$  turbulent
CHAPTER 1
INTRODUCTION

1.1 Background

Air Cooled Steam Condensers (ACSCs) have gained increasing popularity over water cooled condensers in the areas where water resources are limited. Water cooled condensers are reportedly more efficient and have less capital cost than the ACSCs. However, the operational cost for the working life of an ACSC is less than the water cooled condensers [1].

ACSCs are commonly employed in power plants to condense the steam and recycle the water. The exhaust steam from the steam turbine passes through the A-frame condenser and the condensed water is supplied back to the boilers. The schematic of A-frame structure of ACSCs is shown in Figure 1 [2]. ACSC consists of finned-tube bundles stacked along the two inclined arms of A-frame structure. Exhaust steam is distributed into these finned-tubes through a large manifold located at the top center of the A-frame structure. The structure also has a large fan at the base to force the air over the finned-tube array; thus enhancing the cooling effect. The condensate ducts at the base collects the condensed water and is supplied back to boilers.

ACSCs are located close to the steam turbine outlet to minimize the pressure losses [1]. The reduction in volume of fluid due to the condensation of steam into water creates vacuum in the condenser. This causes reduction in pressure at the outlet of the turbine and assists in maintaining the backpressure of the steam turbine in the prescribed range suggested by the manufac-
turer [3]. Operating at the backpressure greater the recommended range reduces the efficiency of steam turbines. Thus for the efficient power generation in steam turbines, condenser should be also be efficient in all weather conditions.

Maximizing heat transfer from finned-tube array is an important factor to be considered in the performance of condenser. The design of such an array has been investigated by previous researchers due to the complexities involved in the cross flow of air over the tube bundles. The efficiency of finned-tubes depends on geometric parameters such as fin and tube spacing, operating conditions and material properties. Efficient fins help to reduce the number of tube rows, resulting in compactness of design as well as saving of energy.
Inefficient performance in extreme operating conditions is a common problem of ACSCs. Enhancing heat transfer coefficient (h) of the fins can help in efficient steam condensation and minimizing the problems associated with extreme climatic conditions. Researchers have explored several approaches of reducing air side resistance in order to increase heat transfer from finned-tube surfaces.

Figure 2. A-frame ACSC by Hudson Products Corporation, Texas. [4]
1.2 Literature review

Webb [5] surveyed developments in plate-fin and circular-fin heat exchangers and suggested the use of slotted and punched fins for increasing the heat transfer. Webb [5] recommended the use of the correlation for heat transfer presented by Briggs and Young [6] and correlation for pressure drop presented by Robinson and Briggs [7]. Žukauskas [8] introduced correlations for Nusselt number (Nu) and pressure drop (ΔP) for an in-line as well as a staggered arrangement of finned-tube bundles. Flow through the tube bundles was reported to have three circumferential regions, namely laminar, turbulent and separated flow. Žukauskas [8] also mentioned the need to artificially disrupt the boundary layer to attain better performance of the heat exchanger.

Heat transfer enhancement techniques were classified as active and passive techniques by Berger et al. [9]. Vibration, magnetic field and electrical field which required application of external power were classified as active techniques. Passive techniques, which included rough surfaces, extended surfaces and additives for liquid or gases, did not require use of external power [9]. Several studies investigating these methods and reporting a noteworthy increase in $h$ were available. The passive technique of extended surface had many alterations including wavy fins, slotted fins and perforated fins. Wang et al. [10] studied wavy fins and presented an empirical correlation for herringbone plate-fin-tube heat exchangers. Tao et al. [11] studied plate fins with slotted ‘X’ arrangement of strips commonly used in air conditioning. This arrangement resulted in 97% increase in heat transfer coefficient ($h$) and 63% increase in pressure drop (ΔP). Cheng et al. [12] studied three different configurations of ‘X’ protruding strips, positioned along the
flow direction according to the rule of ‘front coarse and rear dense’. Recent studies of pin fins by Sahin et al. [13] and of rectangular fins by Shaeri et al. [14] increased surface contact area of fluids by perforations. This assisted in improving turbulence and mixing and in reducing the friction factor.

Jang et al. [15] was the first to study annular finned-tubes numerically. Simulations of staggered arrangement of annular finned-tubes, under dry and wet operating conditions, were conducted and validated experimentally. They showed that isothermal fin approximation over-estimates the heat transfer coefficient by 5–35%. Mon et al. [16] performed numerical calculations of annular finned-tubes to investigate the effect of variation of fin spacing on the rate of heat transfer. Mon et al. [16] accounted for heat conduction and convection through the fin thickness and applied constant temperature to the tube walls. They reported that the boundary layer development on fin and tube surfaces mainly depends on the ratio of fin spacing to fin height.

**Losses in ACSC.** Different types of losses such as pressure loss across the tube bundle and secondary losses such as inlet loss, fan loss and jetting loss were reported by the researchers in the A-frame structures. Kroger [17] studied various secondary losses in A-frame structures and reported that secondary losses were of the same order as heat exchanger losses (pressure losses). Meyer et al. [18] investigated plenum losses and found that heat exchanger inlet geometry has an influence on plenum losses. Meyer et al. [19] conducted experiments with four types of eight blade axial flow fans. They used both elliptical and rectangular cross sections of tubes and fins to study inlet flow losses. The cross sectional profile of finned-tubes was found to be a
primary factor influencing inlet flow losses. Also, losses were observed to increase with a decrease in angle of incidence of inlet air. Rooyen et al. [4] conducted three-dimensional fluid flow analysis of the A-frame ACSC under windy conditions. Distorted air flow was reported resulting in significantly reduced performance of fan and the ACSC. They recommended adding a cost effective skirting around the ACSC to improve performance under these conditions.

1.3 Problem statement

Abundant literature regarding annular fins had been published, but the idea of perforated annular fins has not been studied in detail. Perforations disrupt the boundary layer and enhance mixing of the flow. This motivated the authors to evaluate performance of the annular finned-tubes with perforations. The main objective of the present study was to assess the increase in heat transfer and pressure drop (ΔP) due to the circular perforations. Annular finned-tubes, assumed to carry steam inside, were cooled with forced air flow. This research employed three-dimensional numerical calculations of four row finned-tubes. Comparison of both the heat transfer rates and changes in ΔP was performed, for solid (no perforations) and perforated (perforations at ±60° to ±180° with 30° interval) fins. Variations in ΔP were evaluated for various combinations of perforations.

A-frame structure of ACSCs was studied by researchers for various types of losses. However, to the authors’ knowledge, heat exchanger loss due to the finned-tube bundle has not been studied. This study altered the spacing of the fins along the arms of A-frame structure to reduce the pressure drop across the tube bundle.
CHAPTER 2

METHODOLOGY

2.1 Finned-tube analysis

Numerical calculations were conducted by simulating three-dimensional air flow and heat transfer over solid (no perforations) and perforated (perforations at 60° to ±180° with 30° interval) finned-tube configurations.

2.1.1 Computational domain

The domain with four finned-tubes in staggered arrangement and symmetric boundaries was constructed, as shown in Figure 3a. This type of arrangement is typical for ACSCs. Geometrical dimensions were similar to those by Jang et al. [15], as shown in Table 1. Thickness of the tubes was considered with inner diameter of the tube as the boundary of the domain. Circular perforations, 4 mm in diameter, were located at a pitch interval of 30° around the annular fin, starting at 60° to -60° in the clockwise direction, as shown in Figure 3b. Since the domain showed geometrical symmetry, only the top part from 0° to 180° of finned-tube was constructed. Thus, a perforated finned-tube with a symmetric surface had four perforations at 60°, 90°, 120°, 150° and a half perforation at 180°. Perforations resulted in 10.8% reduction in the surface area of the annular finned-tube. Domain boundaries were extended to minimize the effect of uniform flow boundary condition at inlet on the flow velocity near the finned-tube walls. The inlet plane
of the flow domain was located 1.5 fin diameters upstream from the first finned-tube while the outlet plane was 5 fin diameters downstream from the last finned-tube.

**Table 1.** Dimensions of finned-tube array for computational and analytical calculations

(by Jang et al. [15])

<table>
<thead>
<tr>
<th>Geometrical parameter</th>
<th>Dimensions in mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inside tube diameter (D&lt;sub&gt;i&lt;/sub&gt;)</td>
<td>23.2</td>
</tr>
<tr>
<td>Outside tube diameter (D)</td>
<td>27</td>
</tr>
<tr>
<td>Fin outer diameter (D&lt;sub&gt;f&lt;/sub&gt;)</td>
<td>41</td>
</tr>
<tr>
<td>Fin thickness (t)</td>
<td>0.5</td>
</tr>
<tr>
<td>Fin spacing (s)</td>
<td>3.5</td>
</tr>
<tr>
<td>Transverse tube pitch (S&lt;sub&gt;T&lt;/sub&gt;)</td>
<td>21.5</td>
</tr>
<tr>
<td>Longitudinal tube pitch (S&lt;sub&gt;L&lt;/sub&gt;)</td>
<td>73</td>
</tr>
<tr>
<td>Tube material</td>
<td>Steel</td>
</tr>
<tr>
<td>Fin material</td>
<td>Aluminum</td>
</tr>
</tbody>
</table>
Figure 3. Geometry of finned-tube array. (a) Staggered arrangement of finned-tube labeled fin-1, fin-2, fin-3 and fin-4 from left to right and with perforated finned surface. (b) Angular locations of perforations on any fin surface and (c) 3-Dimensional view of finned-tube array. All dimensions are in mm.
2.1.2 Governing equations

Conjugate fluid flow and heat transfer calculations were carried out to solve conduction in the tube and fin walls and forced convection over the finned-tubes.

Three-dimensional equations of continuity and momentum equations for the air side flow were solved as follows.

\[ \nabla \cdot (\rho v) = 0 \quad (1) \]

\[ \nabla \cdot (\rho vv) = -\nabla p + \nabla \cdot \tau \quad (2) \]

where, \( \tau = \mu (\nabla v + \nabla^T v) \) is the sheer tensor. (3)

Reynolds number (Re) was calculated using the maximum velocity in the domain and outer diameter of tube as the length parameter (\( Re = \rho D \times v_{max} / \mu \)). Re was found to be in lower turbulent scale within the range of 4,000 to 24,000. Steady state numerical calculations were carried out using RNG k-\( \varepsilon \) turbulence model. Ansys-Fluent (version 6.3) was used for fluid flow and heat transfer analysis.

This is a two equation RANS based turbulence model. It consists of two additional transport equations to represent the turbulent properties of the flow. The two transport variables considered are turbulent energy (k) and turbulent dissipation (\( \varepsilon \)). RNG k-\( \varepsilon \) model is derived from instantaneous Navier-Stokes equations using statistical technique called ‘renormalization
group’ (RNG) methods [20]. This model accounts for small scale vortices and the effects of swirl on turbulence. It also gives analytically-derived differential formula for effective viscosity that accounts for low-Reynolds-number effects. Thus RNG model is more responsive to the effects of rapid strain and streamline curvature than the standard model.

The equations solved for RNG model are similar to k-ε model with change in epsilon equation as [20]

\[ \nabla \cdot (\rho k \mathbf{v}) = \nabla \cdot (\alpha_k \mu_{\text{eff}} \nabla \cdot k) + \rho \varepsilon + G_k \tag{4} \]

\[ \nabla \cdot (\rho \varepsilon \mathbf{v}) = \nabla \cdot (\alpha_\varepsilon \mu_{\text{eff}} \nabla \cdot \varepsilon) + C_{1\varepsilon} \left[ \frac{\varepsilon}{k} \right] \mu_t S^2 + C_{2\varepsilon} \rho \left[ \frac{\varepsilon^2}{k} \right] - R_\varepsilon \tag{5} \]

where, \( \alpha_k = \alpha_\varepsilon = 1.393 \) are the inverse effective Prandtl numbers for k and \( \varepsilon \), respectively, \( C_{1\varepsilon} = 1.42 \) and \( C_{2\varepsilon} = 1.68 \).

\[ G_k = -\rho u_i u_j \frac{\partial u_j}{\partial x_i} \] represents production of turbulence kinetic energy. \( \tag{6} \)

\[ \mu_{\text{eff}} = \mu + \mu_t \] is effective viscosity. \( \tag{7} \)

\( \mu_t \) is turbulent viscosity and is defined based on the turbulence model being used.
For RNG k-ε turbulence model [20],

\[
\mu_t = \rho \times c_\mu \times k^2 / \varepsilon
\]  
(8)

where, \( c_\mu = 0.0845 \)

\[
R = \frac{c_\mu \rho \eta^3 \left(1 - \frac{\eta}{\eta_0}\right)}{1 + \beta \eta^3} \left(\varepsilon^2 \right) / k
\]  
(9)

where, \( \beta = 0.012, \eta = \frac{S k}{\varepsilon} \) and \( \eta_0 = 4.38 \)

where, \( S \) is strain rate magnitude.

Energy equation [20] for the convection over the finned-tubes was solved as

\[
\nabla \cdot [v(\rho E + p)] = -\nabla \cdot (k_{\text{eff}} \nabla T)
\]  
(10)

where, \( E = h + \frac{p}{\rho} + \frac{v^2}{2} \)  
(11)

where, \( h = \) specific enthalpy defined as \( h = \int_{T_{\text{ref}}}^{T} c_p \partial T \)  
(12)
\( k_{\text{eff}} = \alpha_{c} \frac{\rho}{\mu_{\text{eff}}} \) is effective conductivity and is defined based on turbulence model.

In solid, energy equation solved has following form [20],

\[
\nabla \cdot (\nu \rho h) = \nabla \cdot (k \nabla T) \quad (13)
\]

where,
\[\rho = \text{density of solids}\]
\[k = \text{conductivity of solids.}\]
\[h = \text{sensible enthalpy defined in equation (12)}\]

### 2.1.3 Boundary conditions

Solid (no perforations) and perforated (perforations at \(\pm 60^\circ\) to \(\pm 180^\circ\) with \(30^\circ\) interval) cases were simulated for inlet free stream air velocities of 1, 3 and 5 m/s for validation of Nusselt number (\(\text{Nu}\)) and friction factor (\(f\)). Other combinations of perforated cases were simulated for free stream velocity of 3 m/s \((v_{x} = 3, v_{y} = 0, v_{z} = 0)\). Pressure outlet boundary conditions was imposed on the outlet of the domain (gauge pressure = 0). No slip condition was applied to the outer tube and fin walls. Symmetric planes, as shown in Figure 1a, were assumed to have zero diffusion flux. All the normal gradients of flow variables were set to zero at the plane of symmetry. Inlet temperature of the free stream air was 300 K. The inner tube wall was assumed to have a constant temperature of the condensing steam (350 K). This model considered conduction within the fin and tube thickness. Temperature at outlet, outer tube wall and the fins was calculated by the simulations.
The turbulent boundary conditions were specified for inlet and the outlet boundaries. These conditions assist in providing an initial guess for the turbulence parameters at the given boundaries. In this case, initial guess of turbulent kinetic energy \((k)\) and its dissipation rate \((\varepsilon)\) were specified at inlet boundary and outlet boundary was specified with turbulent intensity and hydraulic diameter. Turbulent intensity is defined as the ratio of root-mean-square of velocity fluctuations \((u')\) to the mean flow velocity \((u_{avg})\) [20]. It is also calculated based on Reynolds number at the inlet as [20],

\[
I = \frac{u'}{u_{avg}} = 0.16 \left( \frac{Re}{Dh} \right)^{-0.125} 
\]  
(14)

where, \(Re_{Dh}\) = Reynolds number based on hydraulic diameter.

\[
k = 1.5 \left( \frac{u_{avg}}{I} \right)^2 
\]  
(15)

\[
\varepsilon = C_{\mu}^{3/4} \frac{k^{3/2}}{\mu} \frac{1}{l} 
\]  
(16)

where, \(C_{\mu}\) = turbulence model constant.

Double precision implicit solver was used for the analysis. Equations were solved sequentially as segregated based solver was used. Momentum, energy and turbulence equations were solved with second order discretization schemes. SIMPLE scheme was used for pressure-velocity coupling. Solution was initialized from the inlet boundary of the domain with velocity, \(v_x = 3 \, \text{m/s}\), gauge pressure = 0 and turbulence parameters calculated in equation (15) and (16).
Free stream air properties were specified as viscosity ($\mu$) = $1.798 \times 10^{-5}$ kg/m-s, specific heat ($C_p$) = 1006.43 J/kg-k and thermal conductivity ($k$) = 0.0242 W/m-k. Fins and tubes are usually made up of copper, aluminum or steel. In this case, fins were specified as steel while the fins were specified as aluminum.

Incompressible ideal gas density scheme was employed for the inlet air flow at a temperature of 300 K. According to incompressible ideal gas law, the density depends only on the operating pressure and not on the local relative pressure field [20]. Density in the computational domain is calculated as [20]

$$\rho = \frac{P_{op}}{\left( \frac{R}{M_w} \right) T}$$  (17)

where, $P_{op}$ = operating pressure.

$R$ = universal gas constant.

$M_w$ = molecular weight of the gas.

Finite volume mesh with hexahedral elements, as shown in Figure 4, was generated in the flow domain. Total element count in the domain was 426,500 and maximum skewness was 0.76. Mesh with finer element size was used near the fins and outer tube walls to capture the rapid change in the flow accurately. The size of mesh elements increased and became coarser away from the walls.
Figure 4. Hexagonal mesh. (a) Mesh across the domain with fine mesh elements near the finned-tube walls and relatively coarser mesh elements away from the walls and (b) Mesh over the fin and tube surface.
2.2 A-frame structure analysis

High pressure drop results in higher pumping cost for the fan, which is used as induced draft in ACSCs. A-frame structure of the ACSC was simulated to reduce the overall pressure drop in the structure by altering the fin spacing across the tube.

2.2.1 Computational domain.

A two-dimensional representative geometry of the A-frame was constructed, as shown in Figure 5. Fins representing four row finned-tube arrays along the arms of the A-frame were constructed. Perforations on the finned-tubes were not considered in this representative fin array. Outlet boundary of the domain was constructed at a distance of 25 times fin length. As the A-frame structure has geometrical symmetry, only of the right symmetric half side of the A-frame was constructed as shown in Figure 6. Four cases, with change in spacing of the fins over the entire arms of the A-frame, were simulated to assess the reduction in ∆P, as described in Table 2 and Figure 6. Total number of fins in each of the cases was the same.
2.2.2 Governing equations

Continuity and momentum equations were solved, as described in equations (1) and (2) respectively, to calculate the pressure drop ($\Delta P$) across the fins. $\Delta P$ in the system was calculated at three locations, marked as 1, 2 and 3 in Figure 3. Steady state simulations were performed using SST $k-\omega$ turbulent model.
Figure 6. Locations of pressure measurement and uneven distribution of fins.

1-1: Location 1 for $\Delta P$ calculation
2-2: Location 2 for $\Delta P$ calculation
3-3: Location 3 for $\Delta P$ calculation
Table 2. Distribution of fins over the tube length

<table>
<thead>
<tr>
<th>Case</th>
<th>% length from the inlet side</th>
<th>Fin spacing in mm</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>100 %</td>
<td>3.5</td>
<td>Base case</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Evenly spaced fins</td>
</tr>
<tr>
<td>B</td>
<td>0 – 73.34 %</td>
<td>3.4</td>
<td>Clustered near the inlet.</td>
</tr>
<tr>
<td></td>
<td>73.34 – 90.53 %</td>
<td>3.7</td>
<td>Largely spaced towards the outlet</td>
</tr>
<tr>
<td></td>
<td>90.53 – 100 %</td>
<td>4</td>
<td></td>
</tr>
<tr>
<td>C</td>
<td>0 – 80.96%</td>
<td>3.4</td>
<td>Clustered near the inlet</td>
</tr>
<tr>
<td></td>
<td>80.96 – 90.53%</td>
<td>3.8</td>
<td>Largely spaced towards the outlet</td>
</tr>
<tr>
<td></td>
<td>90.53 – 100%</td>
<td>4.2</td>
<td></td>
</tr>
<tr>
<td>D</td>
<td>0 – 40.46%</td>
<td>3.4</td>
<td>Clustered near the inlet</td>
</tr>
<tr>
<td></td>
<td>40.46 – 45.22%</td>
<td>3.8</td>
<td>Largely spaced in center</td>
</tr>
<tr>
<td></td>
<td>45.22 – 54.69%</td>
<td>4.2</td>
<td>Clustered near the outlet</td>
</tr>
<tr>
<td></td>
<td>54.69 – 59.50%</td>
<td>3.8</td>
<td></td>
</tr>
<tr>
<td></td>
<td>59.50 – 100%</td>
<td>3.4</td>
<td></td>
</tr>
</tbody>
</table>
Reynolds number (Re) was calculated using velocity and length at the inlet and was found to be 580,700. SST k-ω turbulence model was used to conduct steady state numerical calculations using Ansys-Fluent (version 6.3).

k-ω turbulence model is also a two equation turbulence model with two additional transport equations for turbulent kinetic energy (k) and specific dissipation rate (ω), which can also be defined as the ratio of ε to k [20]. It is a modified k-ε turbulent model with which include cross diffusion term in ω equation. The definition of the turbulent viscosity is modified to account for the transport of the turbulent shear stress. The two extra transport equations are similar to standard k-ω model with an additional term of cross diffusion in ω equations [20].

\[
\nabla \cdot (\rho k v) = \nabla \cdot (\Gamma_k \nabla k) + G_k + Y_k
\]

\[
\nabla \cdot (\rho \varepsilon v) = \nabla \cdot (\Gamma_\omega \nabla \omega) + G_\omega + Y_\omega + 2(1 - F_1) \rho \sigma_\omega \frac{1}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j}
\]

where, \( \Gamma_k = \mu + \frac{\mu}{\sigma_k} \) is effective diffusivity of k

\[
\Gamma_\omega = \mu + \frac{\mu}{\sigma_\omega} \text{ is effective diffusivity of } \omega
\]
where \( \mu_t = \frac{\rho k}{\omega} \max\left(\alpha, \omega, SF_2\right) \) is turbulent viscosity. \( (22) \)

where, 
\[
F_2 = \tanh\left[\max\left(2 \frac{\sqrt{k}}{0.09 \omega y}, \frac{500 \mu}{\rho y^2 \omega}\right)\right]
\]
is a blending function. \( (23) \)

\[\sigma_k \quad \text{and} \quad \sigma_\omega \]
are the turbulent Prandtl numbers for \( k \) and \( \omega \), respectively.

\( S \) is strain rate magnitude

\( Y_k \) and \( Y_\omega \) represent the effective dissipation of \( k \) and \( \omega \), respectively.

\( G_k \) represents the generation of turbulence kinetic energy due to mean velocity gradients and is defined as given by equation (6).

\[
G_\omega = \frac{\omega}{k} G_k \quad \text{is the generation of} \quad \omega. \quad (24)
\]

\( Y_k = \rho \beta^* k \omega \) is dissipation of \( k \) due to the turbulence. \( (25) \)

\( Y_\omega = \rho \beta \omega^2 \) is dissipation of \( \omega \) due to the turbulence. \( (26) \)

\[
F_1 = \tanh\left[\min\left\{\max\left(\frac{\sqrt{k}}{0.09 \omega y}, \frac{500 \mu}{\rho y^2 \omega}, \frac{4\rho k}{\sigma_\omega^2 D_\omega^y y^2}\right)\right\}^4\right]
\]
is a blending function. \( (27) \)
where, 
\[ D^+_{\omega} = \max \left[ 2 \rho \frac{1}{\sigma} \frac{1}{\omega_{2}} \frac{\partial k}{\partial x} \frac{\partial \omega}{\partial x_j}, 10^{-10} \right] \]
is the positive part of the cross diffusion function (last term on left side of equation 19).

SST k-\( \omega \)turbulence model blends together the properties of standard k-\( \omega \)and k-\( \varepsilon \) turbulence models. Blending functions F1 and F2, described in equations (23) and (27), are used to accurately merge the two turbulence models and achieve accurate results in near-wall and far-field regions. Other model constants are used in the calculation of all the listed turbulence variables are as follows,

\[ \alpha_1 = 0.31, \quad \beta_{i,1} = 0.075, \quad \beta_{i,2} = 0.0828 \]

\[ \sigma_{k1} = 1.42, \sigma_{\varepsilon 1} = 1.68, \quad \sigma_{k2} = 1.42 \quad \text{and} \quad \sigma_{\varepsilon 2} = 1.68 \]

2.2.3 Boundary conditions

Inlet boundary (Figure-5) represented forced air velocity of 3 m/s from the fan \( (v_x = 0, \ v_y = 3, \ v_z = 0) \). Pressure outlet boundary condition was used at the outlet boundary of the domain\( (\text{gauge pressure} = 0) \). No slip/no penetration wall boundary condition was enforced on the fin array. Wall boundary condition was also specified at semi-circular boundary, representing the steam inlet duct and other construction walls. Symmetry condition was imposed on the left side boundary by specifying all normal gradients as zero. Turbulent boundary conditions at the inlet were calculated as discussed earlier (equations 14–16). Solution was initialized from the inlet boundary with velocity, \( v_y = 3, \ v_x = v_z = 0 \) and gauge pressure = 0.

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Geometry was finely meshed with quad elements between the fins and with relatively coarse quad elements in the remaining domain. The interface type of boundary was used to account for this transition in size of mesh elements. The use of interface boundary type assisted to reduce the size of the mesh and also to generating fine mesh for the fin array. Total element count was 100,000 and had maximum skewness of 0.75.

2.3 Validation with analytical results

Computational results of solid fins were validated for Nusselt number (Nu) with Briggs and Young correlation [6] and for friction factor (f) with Robinson and Briggs correlation [7] as mentioned in equations (28) and (29), respectively.

\[
Nu = 0.134 \times Re^{0.681} \times Pr^{(1/3)} \times \left[ \frac{s}{l} \right]^{0.2} \times \left[ \frac{s}{t} \right]^{0.1134}
\] (28)

\[
f = 18.93 \times Re^{-0.361} \times \left[ \frac{S_T}{D} \right]^{-0.927} \times \left[ \frac{S_T}{S_L} \right]^{-0.515}
\] (29)

Analytical formula to calculate maximum velocity, in non-dimensionless parameters such as Re, was given by Žukauskas [8] as follows

\[
v_{\text{max}} = \frac{S_T v}{S_T - D}
\] (30)
Friction factor \( f \) for the numerical simulations was calculated as

\[
f = \frac{\Delta P}{n \rho v^2}
\]  

Comparison of analytical and computational results of \( \text{Nu} \) and \( f \) is shown in Figure 7. Computational results for \( \text{Nu} \) number were within 15.7\% of the analytical correlation (Figure 7a). The \( f \) values (Figure 7b) were close to the analytical results for the lower velocities, within 24.6\%. However, they resulted in a 43.7\% deviation for a velocity of 5 m/s. Though the results for \( f \) are not close to the analytical results, they are consistent with the data reported by Mon et al. [16]. Mon et al. [16] showed that numerical results for \( \text{Nu} \) and \( f \) were within ± 25\% and ± 40\%, respectively, with those given by analytical results. Average percentage difference and maximum percentage difference are summarized in Table 3. Other perforation combinations, as shown in Table 4, were also simulated in order to study the variation of \( \Delta P \) with the location of perforation.

Table 3. Comparison of analytical and computational results

\[
\left[ \frac{|\text{Analytical} - \text{computational}| \times 100}{\text{analytical}} \right]
\]

<table>
<thead>
<tr>
<th></th>
<th>Average difference</th>
<th>Maximum difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nusselt number (( \text{Nu} ))</td>
<td>13.7 %</td>
<td>15.7 %</td>
</tr>
<tr>
<td>Friction factor (( f ))</td>
<td>33.6 %</td>
<td>43.7 %</td>
</tr>
</tbody>
</table>
Figure 7. Comparison of computational and analytical results for solid case. (a) Comparison of Nusselt number with Briggs and Young correlation and (b) Comparison of friction factor with Robinson and Briggs correlation.
CHAPTER 3
RESULTS

3.1 Finned-tube Analysis

Numerical calculations were conducted for the solid fins (no perforations) to assess the variation of velocity and temperature distributions across the domain. The velocity and temperature distribution on the surface of solid fins were then compared with those of perforated-fins (perforations at ±60° to ±180° with 30° interval). Other cases with single, double and triple perforations, as described in Table 4, were simulated to assess the effect of perforation location on velocity and temperature profiles. The case of solid fins was used as the base case to compare subsequent cases involving various combinations of perforations.

3.1.1 Comparison of solid and perforated cases.

Contours of velocity vectors and temperature of the solid finned-tubes, for the free stream velocity of 3m/s, are shown in Figure 7 and 8. It was observed that free stream air flow approaching the finned-tube array encountered stagnation at the angular location (θ) of 0°, at the first row (Figure 7a). At this point, flow velocity dropped to zero and total pressure, being inversely proportional to velocity, was the maximum in the domain. Pressure started decreasing as the flow advanced, creating favorable flow conditions \([dp/dx < 0]\) in the upstream region of all fins. Maximum velocity was observed at \(θ = 90°\) (point E in Figure 9).
Figure 8. Computational results for Solid fins for free stream velocity of 3 m/s. a) Distribution of velocity in the domain consisting of four finned-tube array. b) Contours of temperature in the domain.
Figure 9. Boundary layer separation and vortex formation on a circular cylinder.

$S$ is the point of separation [22].

Till this point the boundary layer region consists of large frictional forces. Thus fluid particles lost most of their kinetic energy in overcoming these forces. Thereafter, velocity started decreasing and pressure started increasing, creating a positive pressure gradient $[dp/dx > 0]$. Fluid particle in this region did not have energy to overcome pressure forces. This resulted in flow separation from the fin wall (point $S$ in Figure 9), creating wake areas. Point $S$ is defined as velocity gradient normal to the wall is zero and hence shear stress, $(\tau)$ is also zero [22]. Fluid particles did not move away from the wall region thus forming a recirculation zone.
Wake area was observed in the downstream regions of the first, second and third finned-tube rows followed by a large wake area at the downstream region of the fourth finned-tube row (Figure 8a). Because of slow air flow, high temperatures persisted in wake areas (Figure 8b). Thus, the upstream half of the region experienced higher heat removal than the downstream region.

Subsequent to the solid fins, perforations were introduced in the downstream region with an attempt to improve the flow characteristics and increase the heat transfer efficiency. Four and one half perforations were created on a symmetric model of the fin surface (equivalent to nine perforations on a complete fin surface), as discussed earlier (Figure 3b). Solid and perforated fins were compared at a free stream velocity of 3m/s, using both temperature and velocity contours, as shown in Figure 10 and 11. Temperature contours of solid fins (Figure 10a) and perforated fins (Figure 10b) showed reduction in size of the high temperature zones, in the downstream region of the tubes. The average temperature of the fins and tube walls in the perforated case was 4 K less than the solid case. Significant reduction in temperature was observed in the downstream region of the fourth row of finned-tubes. Comparison of velocity contours showed that perforations reduced the velocity gradients in the domain. Recirculation zones in the downstream region of the first and subsequent finned-tubes were disrupted by the perforations. As a result, these recirculation zones shrunk in size. Perforations in this region also increased the mixing in the downstream areas. An overall enhancement in the values of heat transfer coefficient (h) was observed for perforated fins as compared to solid fins.
Figure 10. Comparison of temperature contours for inlet velocity of 3 m/s. (a) Solid finned-tubes and (b) Perforated finned-tubes.
Figure 11. Comparison of X-velocity contours for inlet velocity of 3 m/s. (a) Solid finned-tubes and (b) Perforated finned-tubes.
Nusselt number (Nu) is directly proportional to h and is defined as \( Nu = \frac{hD}{k} \). Comparison of Nu number along the fin surface for \( \theta \) ranging from 0° to 180° is plotted in Figure 12, for solid and perforated fins. As shown in Figure 10a, the first row of finned-tube was directly exposed to free stream air flow and was generally unaffected by perforations. However, the value of Nu number for the perforated finned-tubes was observed to increase gradually as the flow advanced from fin-1 to fin-4. As discussed earlier, the flow in the solid fin was detached from the wall at about 90° resulting in a recirculation zone in downstream region. Nu number at \( \theta = 90° \) (near the point of detachment) was observed to have almost the same value for perforated and solid fins (refer to Figure 12a and 12b). As angular locations from 120° to 180° were in the recirculation areas, perforations at these locations improved mixing due to enhanced turbulence. Thus, Nu numbers were observed to increase for every perforated finned-tube for \( \theta \) values from 90° to 180°. The flow in the upstream region of the inner finned-tubes (rows 2 and 3) was influenced by the increased upstream mixing and turbulence. Hence, considerable rise in Nu was observed at the upstream locations for the third (Figure 12c) and the fourth (Figure 12d) finned-tubes. It was evident that downstream finned-tubes benefited from the perforations. The increase in the area averaged Nusselt number (\( \bar{Nu} \)) for the third and the fourth perforated finned-tube was 6.7% and 9.5%, respectively. Percentage increase of Nu number for perforated fins was defined with respect to the solid fins as \( \left( \frac{Nu_{\text{perforated}} - Nu_{\text{solid}}}{Nu_{\text{solid}}} \right) \times 100 / Nu_{\text{solid}} \). Increase in Nu for the entire array of finned-tubes was observed to be 7%.
Figure 12. Plot of Nusselt number vs. $\theta$ (theta in degrees) along fin surface shows enhanced heat transfer. (a) Fin-1, the first finned-tube facing the free stream air flow of 3m/s. (b) Fin-2. (c) Fin-3 and (d) Fin-4, the last finned-tube in the direction of free stream.

Žukauskas [8] mentioned that hydrodynamic forces arising due to the turbulent fluctuations of the fluid pressure was one of the causes of flow-induced vibrations. Fluid flow over the cylindrical tube produces lift forces which in turn results in vibrations in the direction perpendicular to the flow direction [23]. Hydrodynamic forces alternate in equal and opposite magnitude after the flow separation from the tube surface. Thus, limiting the pressure drop ($\Delta P$) becomes
one of the primary challenges in the design of finned-tubes. Perforations, which increased $h$ and Nu number, also increased the pressure drop by about 12%.

$$\left(\frac{\Delta P_{\text{perforated}} - \Delta P_{\text{solid}}}{\Delta P_{\text{solid}}}\right) \times 100$$

Thus, for better optimization, perforations should be placed only at the locations which result in the minimum $\Delta P$ value and the maximum Nu number (or $h$ value).

### 3.1.2 Fin factor.

Symmetric fin combinations were configured to find location of perforation resulting in minimum $\Delta P$, as described in Table 4. Angular location for each combination was shifted by 30°. Four combinations of single perforation at 60° (case-1), 90° (case-2), 120° (case-3) and 150° (case-4) were constructed. Due to the geometric symmetry half perforation was created at $\theta = 180°$ (case-5).

Figure 13–15 show the results for single perforation cases. Velocity vectors, plotted in Figure 13, show that the distinct reduction in recirculation area for perforation at 120°. Reduction in the size of wake area was also seen in temperature contours in Figure 14. High temperature patches resulting due to the wake area at the rear of the finned-tubes showed drastic reduction for perforation at 120°.
Figure 13. Velocity vectors for single perforation cases. (a) Perforation at 60°. (b) Perforation at 90°. (c) Perforation at 120°. (d) Perforation at 150° and (e) Perforation at 180°.
Figure 14. Temperature contours for single perforation cases. (a) Perforation at 60°. (b) Perforation at 90°. (c) Perforation at 120°. (d) Perforation at 150° and (e) Perforation at 180°.
The efficiency of perforations was studied to find optimum case with the least ΔP as well as the maximum h. Calculations, as shown in Table 4, were conducted to find the fin factor, a dimensionless measure representing the performance of different combinations of perforation and defined as

\[
\text{Fin factor} = \frac{\% \text{ Increase in } h}{\% \text{ Increase in } \Delta P}
\]

The high fin factor implies a combination of maximum increase in h values in conjunction with minimum increase in ΔP values. Thus, a fin factor value greater than unity indicates a better-performing fin configuration.

Amongst single perforated cases, a perforation at θ = 120° (case-3) provided the best result with 2.2% increase in h value and 1.4% increase in ΔP value. Fin factor for case-3, case-4 and case-5 was calculated as 1.6, 1.2 and 1.0, respectively. Relative increase in h values was less for the perforations at θ = 60° (case-1) and θ = 90° (case-2). These combinations resulted in a significant increase in ΔP resulting in a low fin factor of 0.2. Hence, these perforations were of no practical use. It was evident that perforations placed in the downstream half portion (between θ = 90° to 180°) were more advantageous than in the upstream half portion (between θ = 0° to 90°). Hence for better fin factor, it was recommended to have perforations only in the downstream region, beyond the point of detachment (θ = 90°). As discussed earlier, perforated case (perforations at ±60° to ±180° with 30° interval) had high pressure drop and hence also resulted in a fin factor < 1.
Table 4. Fin factor for single perforation cases. Angular location (theta in degrees) of perforation along the fin surface, heat transfer coefficient and pressure drop details

*: indicates favorable Fin factor values

<table>
<thead>
<tr>
<th>Cases</th>
<th>Angular location of perforation</th>
<th>Heat transfer coefficient, $h$ (W/m$^2$-k)</th>
<th>Pressure drop, $\Delta P$ (Pa)</th>
<th>Fin factor (B/D)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Average (A)</td>
<td>% increase (B)</td>
<td>$\Delta P_c$ (C)</td>
</tr>
<tr>
<td>Solid</td>
<td>No perforations</td>
<td>76.4</td>
<td>–</td>
<td>115.4</td>
</tr>
<tr>
<td>Perforated</td>
<td>4.5 perforations</td>
<td>81.8</td>
<td>7.0</td>
<td>129.1</td>
</tr>
<tr>
<td>Case-1</td>
<td>1 perforation at 60°</td>
<td>77.3</td>
<td>1.1</td>
<td>120.9</td>
</tr>
<tr>
<td>Case-2</td>
<td>1 perforation at 90°</td>
<td>77.4</td>
<td>1.2</td>
<td>122.0</td>
</tr>
<tr>
<td>Case-3</td>
<td>1 perforation at 120°</td>
<td>78.1</td>
<td>2.2</td>
<td>117.0</td>
</tr>
<tr>
<td>Case-4</td>
<td>1 perforation at 150°</td>
<td>78.0</td>
<td>2.1</td>
<td>117.4</td>
</tr>
<tr>
<td>Case-5</td>
<td>0.5 perforation at 180°</td>
<td>77.5</td>
<td>1.3</td>
<td>116.9</td>
</tr>
</tbody>
</table>

Percentage increase in $h$ and $\Delta P$ values for single and half perforation cases with reference to the solid case is shown in Figure 15. Significant increase pressure drop caused by the perforation at $\theta = 60^\circ$ (case-1) and $90^\circ$ (case-2) was clearly observed. Case-3 resulted in the least increase in $\Delta P$ and maximum increase in $h$ values. The results for case-4 and case-5 also
showed that increase in $h$ was greater than the increase in $\Delta P$. Thus perforation at 120° was the best location amongst all cases followed by 180° and 150°.

![Graph showing percentage increase in heat transfer coefficient ($h$) and pressure drop ($\Delta P$) vs. change in location for single perforation.](image)

**Figure 15.** Plot of percentage increase in heat transfer coefficient ($h$) and pressure drop ($\Delta P$) vs. the change in location for single perforation.

Based on the results of the best single perforation location, additional cases with multiple perforations, 120°-150° (case-6), 120°-180° (case-7) and 120°-150°-180° (case-8), were solved to obtain a combination that has a better balance between enhanced heat transfer coefficient ($h$) and minimal increase in pressure drop ($\Delta P$).
Figures 16–18 show the results for multiple perforations. Case-7 with perforations at 120° and 180° is observed to be the better than the rest of the cases. Similar to case-3, case-7 shows significant reduction in recirculation area, in Figure-16, at the rear of finned-tubes. Case-6 and case-8 resulted in descent increase in h values however wake areas persisted due to high ∆P. Thus, it was observed that perforation at 150° was not beneficial as compared to those at 120° and 180°. Temperature contours, in Figure 17, also show reduced high temperatures areas at the rear of fined-tubes.

**Figure 16.** Velocity vectors for multiple perforation cases. (a) Perforation at 120°–150°. (b) Perforation at 120°–180° and (c) Perforation at 120°–150°–180°.
Figure 17. Temperature contours for single perforation cases. (a) Perforation at 120°–150°. (b) Perforation at 120°–180° and (c) Perforation at 120°–150°–180°.

Table 5 shows the fin factor calculations for multiple perforation cases. Percentage increase in both ΔP and h were plotted for cases with multiple perforations in Figure 18. Case-8 showed maximum increase in h (4.9 %) as well as ΔP (3.9 %) among all the combinations. Since the fin factor > 1 for case-8 (1.3), it was more favorable than the perforated case (with 7% increase in h). Case-7 (fin factor = 1.1) and case-8 (fin factor = 1.3) can be incorporated if higher heat transfer rates are desired and increase in pressure drop is allowable for the structure. However, it must be noted that from the perspective of fin factor, case-3 (perforation at θ = 120°; fin factor = 1.6) is better than case-8 (perforations at θ = 120°, 150° and 180°; fin factor = 1.1).
Table 5. Fin factor for multiple perforation cases. Angular location (theta in degrees) of perforation along the fin surface, heat transfer coefficient and pressure drop details

*: indicates favorable Fin factor values

<table>
<thead>
<tr>
<th>Cases</th>
<th>Angular location of perforation</th>
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<th>Pressure drop, $\Delta P$ (Pa)</th>
<th>Fin factor (B/D)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solid</td>
<td>No perforations</td>
<td>76.4</td>
<td>115.4</td>
<td>–</td>
</tr>
<tr>
<td>Perforated</td>
<td>4.5 perforations</td>
<td>81.8</td>
<td>129.1</td>
<td>0.6</td>
</tr>
<tr>
<td>Case-6</td>
<td>2 perforations at 120° and 150°</td>
<td>78.9</td>
<td>119.8</td>
<td>0.9</td>
</tr>
<tr>
<td>Case-7</td>
<td>2 perforations at 120° and 180°</td>
<td>79.1</td>
<td>119.1</td>
<td>1.1*</td>
</tr>
<tr>
<td>Case-8</td>
<td>2.5 perforations at 120°, 150° and 180°</td>
<td>80.1</td>
<td>119.8</td>
<td>1.3*</td>
</tr>
</tbody>
</table>
Figure 18. Plot of percentage increase in heat transfer coefficient (h) and pressure drop (ΔP) vs. the change in location for multiple perforations.

3.2 Fin spacing in A-frame structure

Four cases of fin spacing were configured as discussed in Table 2. Analysis was started with case-A, the base case, with evenly spaced fins at a pitch of 3.5 mm. Contours of velocity and pressure were plotted to study the flow field in the domain. Pressure drop was measured across all three locations and tabulated in Table 6. It was observed that velocity at the upstream side of the fins increased from inlet end towards the top end. Case-B and case-C were configured such that fins were clustered at the inlet end and the fin spacing was increased towards the top end. However, ΔP across the fins for case-B increased by 2.1% (location 2). Case-C showed a
favorable change with decrease in $\Delta P$ by 1% (location 2). Case-C showed that more air flow occurred through the central region of the fin array. Based on this observation, case-D with clustered fins at both inlet and top ends and sparsely spaced fins at the center was constructed.

**Table 6.** Pressure drop ($\Delta P$) characteristics for combinations of fin spacing

*: indicates favorable change in $\Delta P$

<table>
<thead>
<tr>
<th>Case</th>
<th>Location-1</th>
<th>Location-2</th>
<th>Location-3</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$\Delta P$ (Pa)</td>
<td>% change in $\Delta P$</td>
<td>$\Delta P$ (Pa)</td>
</tr>
<tr>
<td>A</td>
<td>2.85</td>
<td>–</td>
<td>2.87</td>
</tr>
<tr>
<td>B</td>
<td>2.91</td>
<td>2.11</td>
<td>2.93</td>
</tr>
<tr>
<td>C</td>
<td>2.82</td>
<td>-1.16</td>
<td>2.84</td>
</tr>
<tr>
<td>D</td>
<td>2.80</td>
<td>-1.75</td>
<td>2.82</td>
</tr>
</tbody>
</table>

Comparison of velocity contours between case-A and case-D is shown in Figure 19a. The sparsely placed fins at the center facilitated greater air flow with reduced air resistance. Pressure magnitude (Figure 19b) was observed to be reduced on the upstream side of the fin array. Thus, $\Delta P$ across the fins was decreased by 1.9% (location 2). This additional reduction in $\Delta P$, provided by the use of unequal fin spacing along the arms of A-frame, can facilitate the use of multiple perforations to yield higher heat transfer.
Figure 19. Comparison of velocity contours for the case-A and case-D.
(a) Contours of velocity and (b) Contours of pressure.
CHAPTER 4

DISCUSSION

Heat transfer coefficient (h) and pressure drop (∆P) are important parameters in the design of an annular finned-tube array. Heat transfer enhancement with perforations on the annular fin surface, was studied numerically using various combinations of perforations. These arrays, commonly used in air cooled steam condensers (ACSC), are known for high pressure losses. The fin spacing along the arms of A-frame was changed to reduce the pressure drop across the fin bundle. Fin spacing is one of the important geometric factor influencing the fluid flow and heat transfer. Unequal spacing reduced the pressure loss in A-frame structure but its effects in annular finned-tube array are unknown. Finned-tube array with unequal fin spacing needs to be studied to assess the change in heat transfer and pressure drop characteristics.

Elliptical shaped finned-tubes, such as studies by Rocha et al. [17], will provide more aerodynamic properties than the circular fins. Perforated finned-tube array with elliptical cross section can be studied in future to evaluate h and ∆P characteristics. Heat transfer and pressure drop in three-dimensional A-frame structure can be evaluated using optimum number of perforations on annular fins and fin spacing along the arms of A-frame.
Annular finned-tube array with perforations was studied numerically to enhance heat transfer and minimize pressure drop across the domain. To calculate the efficiency of the location of a perforation, a fin factor was defined as the ratio of percentage increase in $h$ and percentage increase in $\Delta P$. A perforation in the wake area (120°, 150° and 180°) beyond the point of detachment was found to be more efficient. Multiple perforations were recommended to obtain higher $h$ values and the number of perforations was limited by the allowable $\Delta P$ in the system.

Unequal spacing along the A-frame structure of ACSCs could decrease the pressure drop across the finned-tube bundle. Clustered spacing at the inlet and top ends and large spacing at the center assisted in reducing the upstream side pressure of the air and $\Delta P$ in the system.
REFERENCES


