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ORIGINAL ARTICLE

Numerical investigation to analyse the effect of fin shape on performance of finned tube heat exchanger

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ABSTRACT – Numerical investigations are carried out to determine the effect of change in geometrical parameter of annular fin on thermal and hydraulic characteristics. Fin shape is changed by varying horizontal diameter of fin. Three different cases of annular fin are investigated corresponding to fin diameter ratio (X) of 0.65, 0.85 and 1. Turbulence is modelled using k-ω shear stress transport model. Relative assessment of tube with annular fin of different horizontal diameter (D_{01}) is done for 2500 < *Re* < 13000. The comparison of Nusselt number reveals that circular tube with annular fin enhances heat transfer rate. Nusselt number augments by 35% and 10% in case of annular fin with diameter ratio of 0.65 and 0.85 as compared to that of annular fin with diameter ratio of 1.00 for 2500 < *Re*< 13000. Pressure drop augments with increase in horizontal diameter of fin and Reynolds number. Annular fin with diameter ratio of 0.65 performs better in terms of heat transfer and pressure penalty as compared to other cases for range of reported Reynolds number.

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INTRODUCTION

Annular finned tube heat exchangers are designed according to their intended use and rate of heat transfer. While designing the annular fin tube heat exchanger, several important parameters must be taken into account, such as volume of the equipment, pressure drop, heat transfer requirement, mass flow rate and thermal efficiency of heat exchanger. Among all these parameters, pressure drop across a finned tube surface is fatal for many industrial applications because fluid must be pumped across a finned tube arrangement [1–5]. The flow behaviour across a different fin-tube arrangement has been studied by many researchers.

Thermal and hydraulic characteristics across a finned tube vary with variation in flow characteristics. Hu and Jacobi [6] and Watel et al. [7] examined the effect of variation in Reynolds number on thermal characteristics across a finned tube surface. Hu and Jacobi [6] did experiments to analyse the flow characteristics across a single row annular fin-tube heat exchanger. The flow patterns and variation in coefficient of heat transfer near to tube surface were observed and the fin efficiency was calculated for 13700 < *Re* < 49800. Lower fin efficiency has been reported when it is calculated from the results of mass transfer compared to when it is calculated by taking a constant coefficient of heat transfer across a surface.

The fin efficiency is affected due to variation in local heat transfer characteristics across a tube surface [7–9]. Watel et al. [7] analysed the temperature variation on fin surface to examine the variation in local heat transfer characteristics and fin efficiency for 2550 < *Re* < 42000. It has been observed that the influence of fin spacing on overall heat transfer rate becomes weak when the fin spacing exceeds about twice the thickness of the boundary layer at the edge of the tail of the fin. Hofmann [10] also analysed two different geometry of fin numerically. Heat transfer and pressure drop difference were calculated at 4500 to 35000 range of Reynolds number. It was concluded that the rate of heat transfer and pressure drop difference increases with increase in Reynolds number. Scuz [11] also investigated the variation in thermal and hydraulic characteristics. Boundary layer over a leading edge was captured experimentally and the effect of pressure gradient in stagnant zone was analysed. Higher heat transfer rate was reported in stagnant zone in upstream of cylinder. Kundu and Das [12–14] carried out investigations to analyse the effect of change in fin eccentricity and other geometrical parameters on heat transfer performance. Semi analytical method was used to analyse the performance of elliptical fin [13]. Relative assessment was carried out for different axis ratio of fin. Efficiency and effectiveness of fin were calculated by predicting fin volume and rate of heat transfer, which is unreasonable. Webb [15] reviewed many methods for enhancement of heat transfer. He concluded that enhanced heat transfer rate can also be achieved by escalating the turbulence in free stream, but at a disadvantageous cost in terms of the higher operating pressure. Moore et al. [16] performed a set of experiment to examine the effect of turbulence on heat transfer characteristics in case of single row fin-tube heat exchanger. Experimental investigations were carried out for 25000 < *Re* < 250000. Increase in Nusselt number was reported due to increase in turbulence intensity. Huisseune et al. [17] experimentally investigated the flow physics across a single row fin-tube heat exchanger. Correlations were developed for coefficient of heat transfer and friction factor. Higher rate of heat transfer achieved using concept of turbulence.

Benmachiche et al. [18] studied the flow behaviour across a finned tube heat exchanger. The eccentricity of fin over a circular tube was analysed for 5500 to 29700 Reynolds number range. The results obtained from an experimental investigation were verified with numerical one. It was observed that increasing value of Reynolds number increases the

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overall rate of heat transfer but decreases the efficiency of annular fin. Mon and Gross [19] carried out 3D numerical investigations to observe the effect of fin spacing on heat and flow characteristics across a finned tube heat exchanger. Turbulence was captured using k-ε RNG model. Effect of change in Reynolds number and fin pitch on Nusselt number was investigated. Improvement in heat transfer coefficient and pressure penalty was reported with de-crease in fin spacing and increase in Reynolds number in case of staggered and inline arrangement of annular fin tube heat exchanger. Chen and Hsu [20] numerically analysed the average heat transfer coefficient across a finned tube heat exchanger. Investigations were carried out for 0.005 m to 0.018 m fin spacing and 1550 to 7760 Reynolds number. It was analysed that the average heat transfer coefficient enhances with increase in Reynolds number and decrease in spacing between two fins.

Other design parameters like fin spacing, fin height and fin thickness also affect the overall performance of heat exchanger [21–23]. Hofmann [10] and Bilrigen et al. [24] carried out numerical investigation to analyse the effect of fin spacing, fin height, fin thickness and fin material on heat transfer characteristics and pressure penalty. Bilrigen et al. [24] did relative assessment for $10000 < Re < 45000$. For Numerical study turbulence was modelled using k-ε RNG model. The heat transfer rate and pressure penalty increase with increase in Reynolds number and decrease in spacing between two fins. The negligible effect on heat transfer characteristics was captured due to variation in fin thickness.

The superior performance of circular fin has been reported by many researchers. However, the higher pressure penalty across a circular finned tube is an enormous problem in the thermal design of heat exchangers, it is still needed to verify the effect of change in geometrical parameters of annular fin on heat transfer characteristics. Hence, the goal of the present numerical investigation is to improve the performance of cross flow annular fin-tube heat exchanger by reducing a pressure penalty across a tube bank by reducing horizontal diameter of fin.

NUMERICAL FORMULATION AND MODELING

Modeling in Flow Domain

A 3-D numerical investigations are carried out to investigate the effect of change in fin shape on heat transfer and flow characteristics of fin-tube heat exchanger. Figure 1 represents the top view of single tube finned heat exchanger. Tube of diameter 0.025 m and length of 0.003 m is considered for present study. The enlarge view of computational domain is shown in Figure 2. Top boundary has 6D distance from the center (C) of the tube. Upstream and downstream boundaries are at 5D and 10D distance from the center of the tube respectively. Relative assessment is carried out to investigate the effect of change in design parameters of annular fin on heat transfer and pressure penalty. In first case, thermal and hydraulic characteristics are studied for a 0.052 m diameter annular fin. Fin base is maintained at 373 K and performance of fin is compared with that of smooth surface tube. In second and third case, the effect of change in fin diameter ratio (X) on thermal and hydraulic characteristics is investigated for different Reynolds number. Fin diameter ratio, X, is defined as ratio of horizontal fin diameter (D_{01}) to vertical fin diameter (D_{02}). The schematic front view of different geometrical conditions of annular fin are shown in Figure 3. Table 1 indicates value of D_{01}, D_{02}, and X for respective cases.

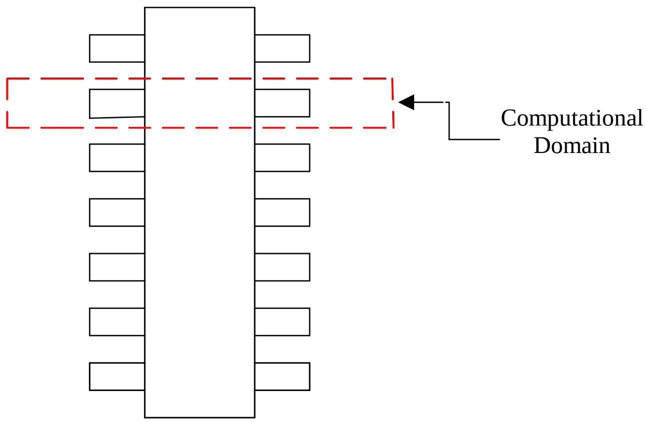

Figure 1. Top view of single tube finned heat exchanger

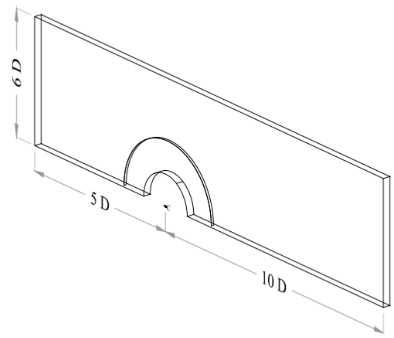

Figure 2. Flow domain for numerical investigation

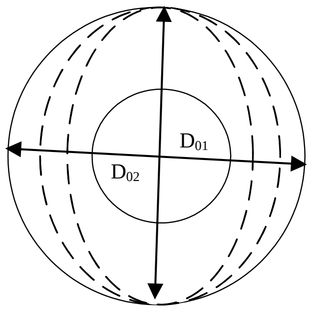

Figure 3. Front view of different geometrical cases of annular fin

Table 1. Different geometrical cases of annular fin

Cases	Horizontal Diameter of fin (D_{01})	Vertical Diameter of fin (D_{02})	X (D_{01}/D_{02})
1	0.052 m	0.052 m	1
2	0.044 m	0.052 m	0.85
3	0.034 m	0.052 m	0.65

Governing Equations

The flow behaviour and temperature variation in a flow domain can be numerically predicted by solving governing equations based on theory of mass, momentum and energy conservation. The mass conservation equation (equation of continuity) for steady, incompressible fluid in three dimensional form is represented as:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}= 0 \tag{1}$$

For 3-D, steady incompressible flow the momentum equations are:
Equation for x - momentum,

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial x}+\upsilon(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2})+(-\frac{\partial \overline{u'^2}}{\partial x}-\frac{\partial \overline{u'v'}}{\partial y}-\frac{\partial \overline{u'w'}}{\partial z})+f_x \tag{2}$$

Equation for y - momentum,

$$u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial y}+\upsilon(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2})+(-\frac{\partial \overline{u'v'}}{\partial x}-\frac{\partial \overline{v'^2}}{\partial y}-\frac{\partial \overline{v'w'}}{\partial z})+f_y \tag{3}$$

Equation for z - momentum,

$$u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}=-\frac{1}{\rho}\frac{\partial p}{\partial z}+\upsilon(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2})+(-\frac{\partial \overline{u'w'}}{\partial x}-\frac{\partial \overline{v'w'}}{\partial y}-\frac{\partial \overline{w^{2'}}}{\partial z})+f_z \tag{4}$$

where, υ = kinematic viscosity, p = pressure and u, v, w are the mean velocity component in x, y and z direction respectively.

The variation thermal characteristics in surrounding of tube surface is encapsulated by solving energy equation, and the energy equations comes from the first law of thermodynamics.

$$(\rho C_p)(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}) = k_f(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}) \tag{5}$$

where $k_f = k + k_{tb}$, ρ = Density of fluid, C_p = Specific heat.

Turbulent kinetic energy and turbulance dissipation rate are the essential parameters to capture turbulence in a flow rigime. Following equations are used to solve them.
Equation for turbulent kinetic energy (k) :

$$u\frac{\partial k}{\partial x}+v\frac{\partial k}{\partial y}+w\frac{\partial k}{\partial z}=P_x\text{-}\ \beta_1\rho k\omega+\frac{\partial}{\partial x}[(\mu+\frac{\mu_{tb}}{\sigma_k})\frac{\partial k}{\partial x}]+\frac{\partial}{\partial y}[(\mu+\frac{\mu_{tb}}{\sigma_k})\frac{\partial k}{\partial y}]+\frac{\partial}{\partial z}[(\mu+\frac{\mu_{tb}}{\sigma_k})\frac{\partial k}{\partial z}] \tag{6}$$

Equation for turbulence dissipation rate (ω) :

$$\begin{gathered}u\frac{\partial \omega}{\partial x}+v\frac{\partial \omega}{\partial y}+w\frac{\partial \omega}{\partial z}=\mathrm{A}\rho S^2\text{-}\ \beta_2\rho\omega^2+\frac{\partial}{\partial x}[(\mu+\frac{\mu_{tb}}{\sigma_\omega})\frac{\partial \omega}{\partial x}]+\frac{\partial}{\partial y}[(\mu+\frac{\mu_{tb}}{\sigma_\omega})\frac{\partial \omega}{\partial y}]+\frac{\partial}{\partial z}[(\mu+\frac{\mu_{tb}}{\sigma_\omega})\frac{\partial \omega}{\partial z}]\\ +2(1\text{-}F_1)\frac{\rho}{\sigma_\omega\omega}(\frac{\partial^2(k\omega)}{\partial x^2}+\frac{\partial^2(k\omega)}{\partial y^2}+\frac{\partial^2(k\omega)}{\partial z^2})\end{gathered} \tag{7}$$

where β_1, β_2 and A are SST model constants, S is absolute value of shear strain rate and σ is SST diffusion coefficient.

Boundary Condition

Numerical investigations are carried out to find the optimum size and shape of annular fin geometry. Air is considered as a working fluid. Air enters from upstream boundary of cylinder at uniform velocity and temperature of 300 K with turbulence intensity of 5%. Velocity component in Y and Z direction is considered to be zero. Cylinder and fin surface are considered as a solid region in computational domain. No slip boundary condition is applied on cylinder and fin surfaces. Heat is transferred from cylinder to fin through conduction mode and from fin to air through convection mode. Cylinder surface is maintained at constant temperature 373 K. Pressure outlet boundary condition is assigned at domain outlet.

Meshing and Flow Modeling

In order to capture a complexity of current flow physics, the flow domain is split into number of subdomains. The governing equations are solved for each element of the entire domain. As shown in Figure 4, structural grid is generated for solving the thermal and hydraulic characteristics across a finned tube surface. To capture the accurate flow characteristics near to tube surface, fine grid is generated. The grid density varies in a flow domain using successive ratio as shown in Figure 4.

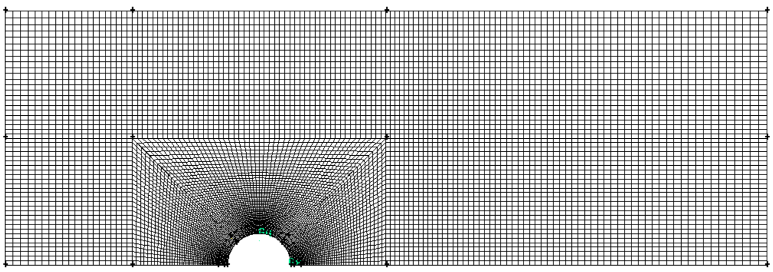

Figure 4. Meshing in flow domain

Solution of mass, momentum and energy is essential to analyse the actual flow physics. Fluent is used as a solver to analyse thermal and hydraulic structure near to tube surface. In present numerical investigation, *k-ω* SST closer model is used for turbulence [25]. *k-ω* SST model includes two additional transport equations to represent turbulent properties of flow to account convection and diffusion effects of turbulent energy. It is enough capable to capture turbulence in region with large normal strain; stagnation region and region with strong acceleration. SIMPLEC algorithm is adopted for coupling of pressure and velocity. QUICK scheme used to discretise governing equations. The criteria for convergence of mass and momentum conservation equations are set to 10^{-4} and 10^{-6} for energy conservation equation.

RESULTS AND DISCUSSION

Grid Independence Study

Grid generation is an indispensable part of numerical investigation. A grid independence study is essential to analyse the size of grid for computational that does not allow the solution to change substantially. To determine the optimum number of elements, relative error in average Nusselt number is evaluated for four different grid sizes. Table 2 represents the number of elements used for each case in grid independence study. The effect of change in grid size on average Nusselt number for the case of annular fin at 7065 Reynolds number is shown in Figure 5.

Table 2. Details of grids used for grid independence study

Grid Types	Number of Elements
Grid A	1320200
Grid B	1510000
Grid C	1600344
Grid D	1684890

It is observed that Nusselt number varies substantially for Grid A and Grid B but it is almost comparable for Grid C and Grid D. It is concluded that the change in grid size beyond Grid C doesn't affect results significantly. Thus Grid C is considered as grid independent mesh.

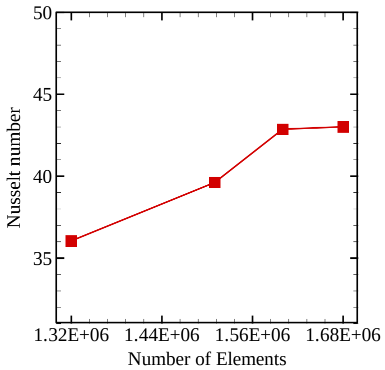

Figure 5. Grid independence study

Apart from that, the wall y^+ is one of the suitable parameters to determine the precision in the selection of the number of elements, when the problem is associated with convection phenomena. Turbulent flows are significantly affected by the presence of a wall. Therefore, mesh accuracy is essential to predict the nature of wall bounded turbulent flow. Therefore, the grid near the wall surface is generated in such a way so that wall y^+ is maintained within 1.03 to 0.05 in case of fin diameter ratio 1 (X=1), 1.074 to 0.06 in case of X = 0.85 and 1.04 to 0.055 in case of X = 0.65 for the range of Reynolds number considered in present work.

Validation

Numerical investigations are carried out to investigate the effect of change in design parameters of fin surface on thermal and hydraulic characteristics. The overall heat transfer between surface of fin and air is evaluated from numerical simulation.

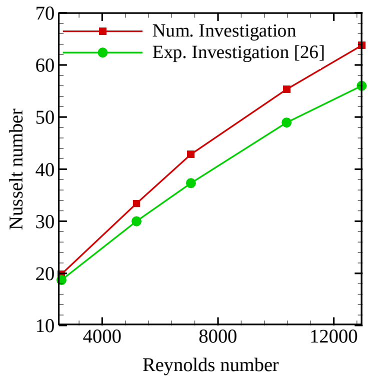

Figure 6. Validation of work

The average value of coefficient of heat transfer (h_{avg}) is calculated as,

$$h_{\text{avg}} = \frac{Q}{((A_t + \eta A_f)\theta_{\text{mean}})} \tag{8}$$

where,

$$\theta_{\text{mean}} = \frac{(T_{\text{inlet}}\text{-}T_{\text{outlet}})}{\ln\left(\frac{(T_{\text{inlet}}\text{-}T_{\text{wall}})}{(T_{\text{outlet}}\text{-}T_{\text{wall}})} \right)} \tag{9}$$

Where, Q is heat transfer from surface of fin. The difference in average heat transfer coefficient on surface of tube and fin is considered to be marginal.

Average Nusselt number for case of diameter ratio X = 1 of an annular fin-tube is investigated numerically and compared with experimental work done by Gianolio and Cuti [26]. Figure 6 represents the comparison of numerically calculated average Nusselt number (Nu_{avg}) with that obtained from experiments [26]. Numerical study over predicts the Nusselt number. Numerically investigated Nusselt number is deviating 3% from experimentally investigated Nusselt number at low Reynolds number. Deviation in result increases with increase in Reynolds number. There is a deviation of 10% in numerically calculated Nusselt number as compared to that of the experimental one at 12965 Reynolds number. It is difficult to measure the temperature distribution on surface of fin without disturbing the flow characteristics through experiments. Thermocouples create disturbance in flow field and change the heat transfer characteristics within fin surface as well as over a fin surface. This may be reason for deviation between experimental and numerical results.

Overall Flow Behaviour

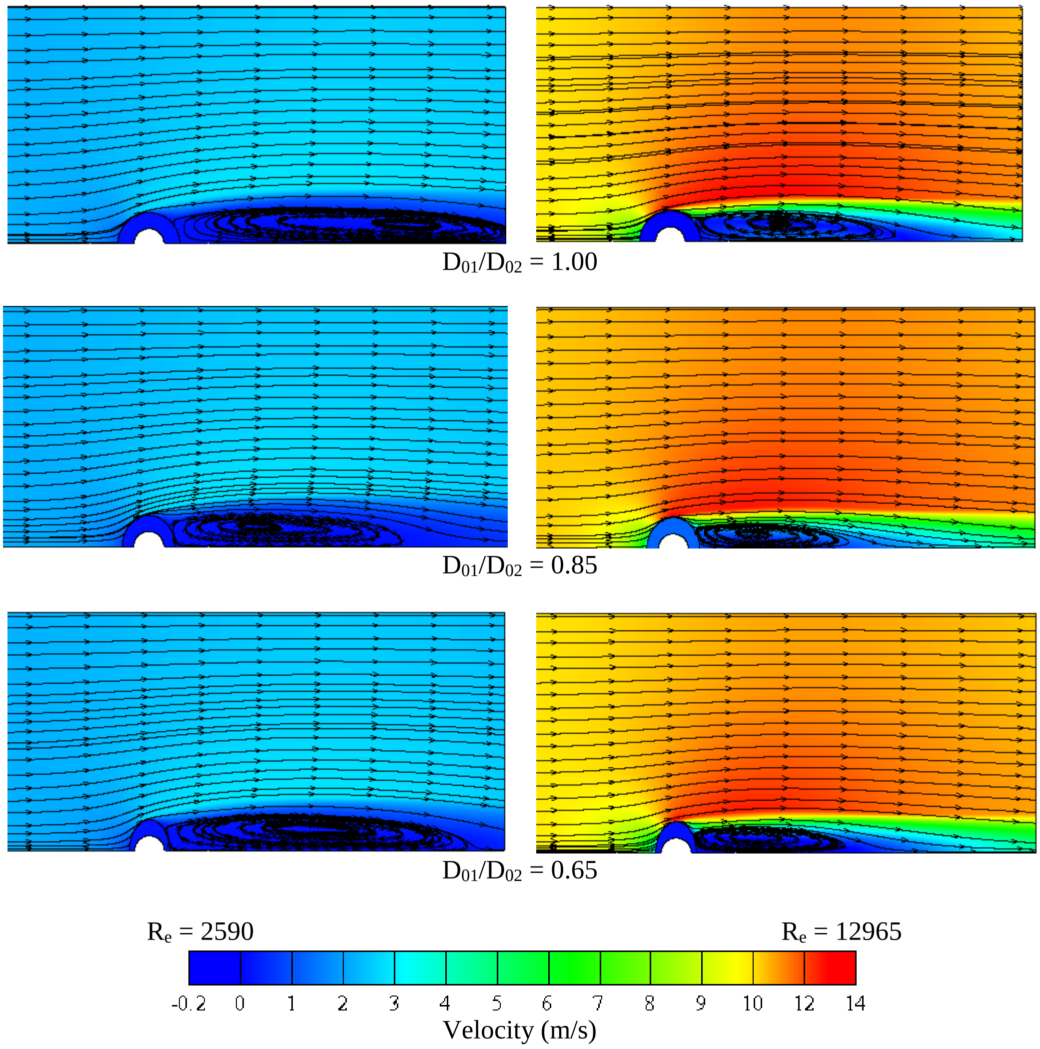

Figure 7. Velocity contours with streamline plot

The effect of change in fin design on flow characteristics near tube surface is observed for 2500 < Re < 13000. For visualization of flow behaviour, velocity contours on x-y plane at 2590 and 12965 Reynolds number are plotted and shown in Figure 7. As shown in Figure 7, cylinder offers obstruction to flow. The fluid stream has to adjust its flow pattern according to shape of obstruction. Near tube surface in upstream, fluid stream cannot change its direction due to sudden obstruction. Hence there is formation of stagnant zone in upstream. In downstream fluid particles detach from the cylinder surface. Due to this separation and negative velocity of fluid particles, wake region is formed. The region of negative velocity and stagnant point in flow region varies with fluid freestream velocity and shape of the fin.

As shown in Figure 7, annular fin with diameter ratio of 1 offers longer wake zone at reported Reynolds number as compared to other two cases of annular fin. As the value of approaching free stream velocity increases, the length of wake zone decreases in particular fin-tube case. The variation in wake zone at downstream and stagnant zone at upstream affect the heat transfer form tube and fin surface to surrounding. Figure 8 represents the temperature contours on x-y plane for different fin shape at 2590 and 12965 Reynolds number to understand the heat transfer phenomena in effective manner. As shown in Figure 8, air enters in the domain at 300 K and strikes to tube surface, which is maintained at 373 K. The variation in air temperature due to convection is easily identified near the tube surface in all the reported cases.

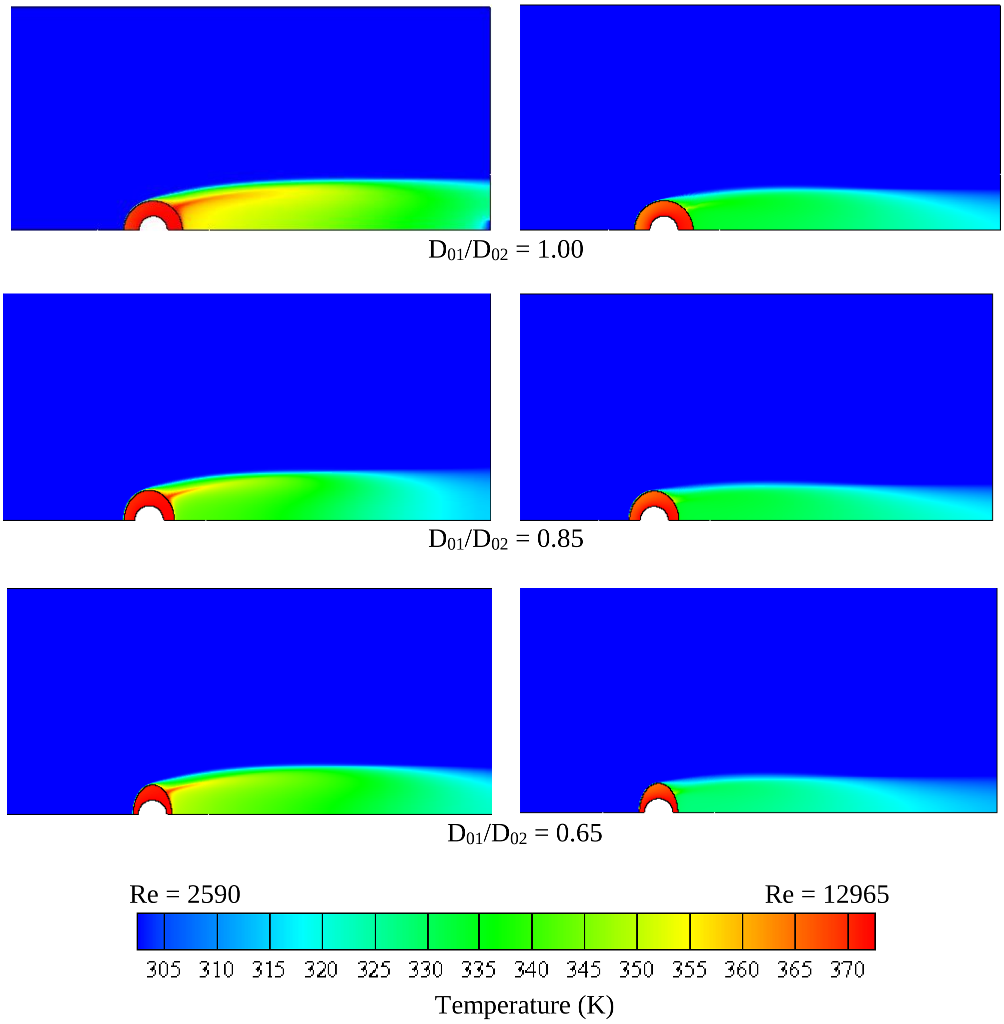

Figure 8. Temperature contours for flow domain

Stagnant and separated zones near to tube surface are captured in Figure 7. Due to wake, temperature of air in separated region increases. Stronger separated zone is captured in case of annular fin shape and because of that lower temperature difference between air and tube surface is captured in that region in Figure 8. Intensity of wake zone decreases with decrease in horizontal radius of annular fin at particular Reynolds number and that reduces the length of high temperature zone near to the tube surface. Heat transfer enhances with increase in temperature difference between fin surface and surrounding and uniformity in temperature distribution on fin surface is essential for this.

Figure 8 represents the temperature contours on *x-y* plane for different fin shape at 2590 and 12965 Reynolds number to understand the heat transfer phenomena in effective manner. As shown in figure 8, air enters in the domain at 300 K and strikes to tube surface, which is maintained at 373 K. The variation in air temperature due to convection is easily identified near the tube surface in all the reported cases. Stagnant and wake zones near to tube surface are captured in Figure 7. Due to separation phenomena, temperature of air in that region increases. Stronger wake zone is captured in case of annular fin shape and because of that lower temperature difference between air and tube surface is captured in that region in Figure 8. Intensity of wake zone decreases with decrease in horizontal radius of annular fin at particular Reynolds number and that reduces the length of high temperature zone near to the tube surface. Heat transfer enhances with increase in temperature difference between fin surface and surrounding and uniformity in temperature distribution on fin surface is essential for this.

Figure 9 represents the temperature variation on fin surface at 2590 and 12965 Reynolds number for different geometrical cases. As shown in Figure 9, maximum temperature on surface of fin is captured at lower base surface in all the cases, which is connected to surface of tube. Temperature on surface of fin decreases with increase in fin radius. The difference in temperature gradient increases as the diameter ratio increases from 0.65 to 1.0. In case of 1.0 diameter ratio, temperature on tip of fin is much lower as compared to other geometrical cases. Due to lower temperature on tip of fin, temperature difference between surface of fin and air decreases, which also decreases the ability of heat transfer from surface of tube to surrounding. Similar observation was also made by Benmachiche et al [18] in their experimental work. In case of 0.85 fin diameter ratio, temperature on tip of fin is higher and it is maximum in case of 0.65 fin diameter ratio. The higher value of temperature on fin tip increases the temperature gradient and enhances the rate of heat transfer between fin surface and atmosphere.

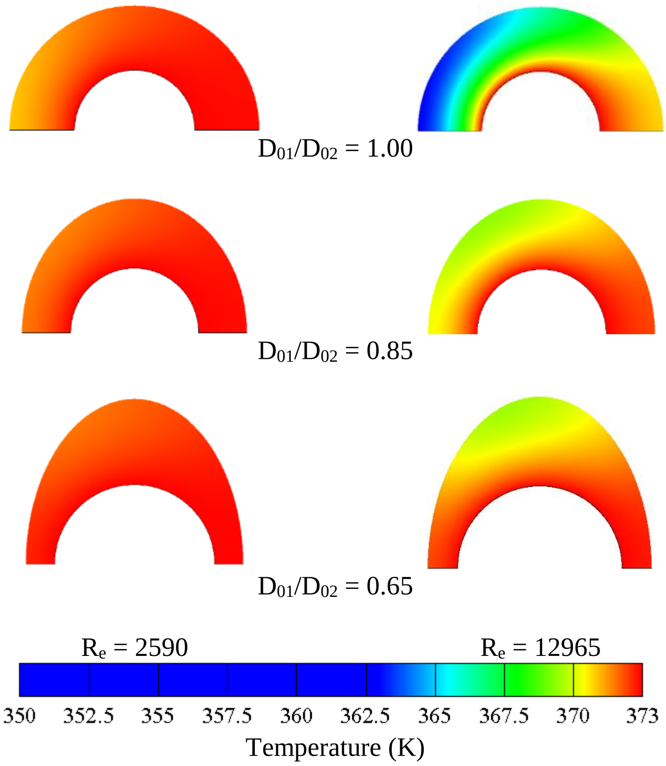

Figure 9. Temperature contours on fin surface

Calculation of Heat Transfer Coefficient and Pressure Penalty

The effect of change in flow behaviour near tube surface on heat transfer characteristics is analysed for 2500 < *Re* < 13000. Figure 10 represents the variation in average Nusselt number over fin surface due to change in fin shape and change in Reynolds number. Comparison of average Nusselt number for different diameter ratio at same Reynolds number indicates that average Nusselt number decreases with increase in horizontal radius of fin. Highest value of average Nusselt number in the range of Reynolds number investigated is achieved with diameter ratio of 0.65 at particular Reynolds number. Average Nusselt number decreases by 13.5% and 36.2% when fin diameter ratio is increased to 0.85 and 1.00 at 2590 Reynolds number. Same trend of change in Nusselt number is observed for higher Reynolds number. Nusselt number decreases by 4% and 34% when annular fin diameter ratio is increased to 0.85 and 1.00 at 12965 Reynolds number.

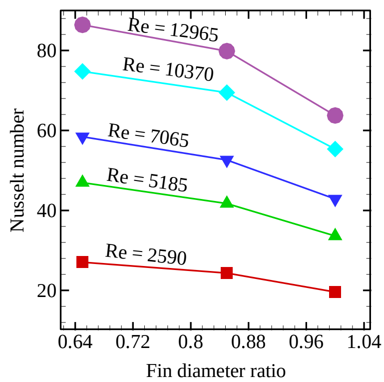

Figure 10. Variation in Nusselt number

Reduction in Nusselt number with increase in diameter ration at same *Re* can be attributed to change in flow regime in downstream of fin. With increase in diameter ratio, the length and width of wake zone in downstream increase. The same can be observed by comparing stream line pattern and velocity contours presented in Figure 7. This larger shape of wake zone acts as a barrier for heat transfer. Also, larger shape of wake will bring more heat from main stream flow back to the source and this in turn will reduce the temperature gradient near fin surface in downstream. This can be observed by comparing temperature distribution in downstream near fin surface in wake zone as shown in Figure 8. Thus, enhancement in shape of wake region with increase in diameter ratio reduces the heat transfer rate from source which is present in upstream of wake zone.

Operating pressure is one of the important parameters in designing process of annular fin tube heat exchanger. Obstruction in flow always require additional pressure penalty to overcome flow obstruction for maintaining constant fluid flow rate. Figure 11 represents the Euler number ($dp/(\rho V^2)$) for different diameter ratio of fin Reynolds number. Pressure penalty depends on shape, size and intensity of wake zone. The most intense wake zone is captured in case of annular fin with diameter ratio of 1.00 as shown in Figure 7. Due to that highly intense wake zone, highest pressure penalty is predicted in case of annular fin with diameter ratio of 1.00.

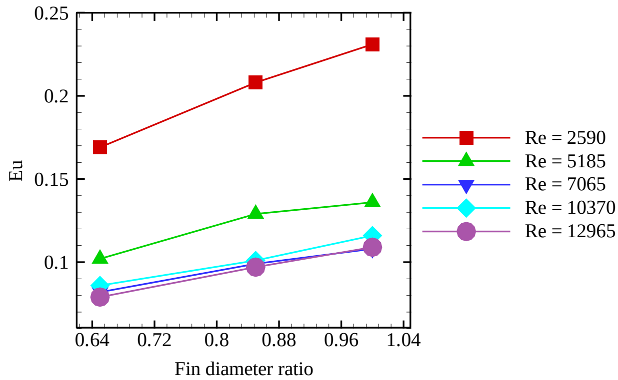

Figure 11. Variation in pressure penalty

Euler number decreases by 27% and 10% in case of annular fin with diameter ratio of 0.65 and 0.85 as compared to that of annular fin with diameter ratio of 1.00 at 2590 Reynolds number respectively. Same trend of change in Pressure penalty is observed for higher Reynolds number. Euler number decreases by 27% and 11% in case of annular fin with diameter ratio of 0.65 and 0.85 as compared to that of annular fin with diameter ratio of 1.00 at 12965 Reynolds number respectively.

Calculation of performance index is essential for evaluating the overall performance of heat exchangers. Performance index is (*Pi*) defined as the ratio of heat flux (*q*) to pumping power for unit cross-sectional area for a particular case. Figure 12 represent the ratio of performance index (*Pi/P*i*) for different fin design, where *P*i* is the performance index calculated for annular finned tube with diameter ratio of 1.00 at particular Reynolds. Higher heat transfer rate is reported in case of annular fin with diameter ratio of 1.00 but the overall performance decreases due to high operating pressure penalty.

As shown in Figure 12, Performance index increases by 85%, and 37% in case of annular fin with diameter ratio of 0.65 and 0.85 respectively as compared to that of annular fin with diameter ratio of 1.00 at 2590 Reynolds number. The trend of performance index is almost similar at higher freestream velocity. It increases by 88%, and 41% in case of annular fin with diameter ratio of 0.65 and 0.85 respectively as compared to that of annular fin with diameter ratio of 1.00 at 12965 Reynolds number. The overall performance of annular fin with diameter ratio of 0.65 is overwhelming for reported Reynolds number range.

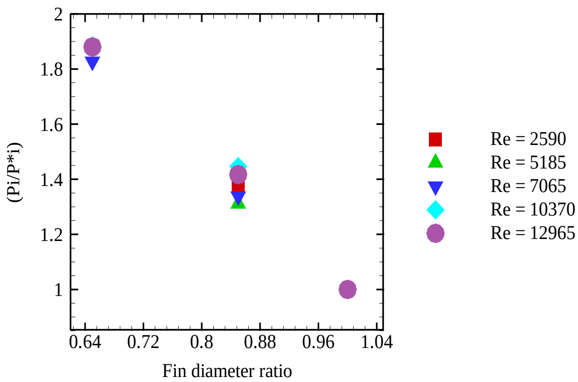

Figure 12. Variation in performance index

CONCLUSION

Numerical investigations are carried out to predict effect of annular fin shapes on heat transfer and pressure penalty. Three different fin diameter ratio; 1.00, 0.85 and 0.65 are investigated for 2500 to 13000 Reynolds number. Based on present study major conclusions are drawn as follows:

- Horizontal diameter of annular fin affects the shape and size of wake zone formed in downstream of fin and tube surface. Wake zone act as heat trap for the heat source present in upstream and offers resistance to heat transfer. Optimum level of wake zone is observed at diameter ratio 0.65 for 2500 < *Re* < 13000.
- Annular fin with horizontal diameter ratio 0.65 is performing better in terms of pressure penalty and heat flux for 2500< *Re* <13000. Highest Nusselt number is reported in case of annular fin with diameter ratio of 0.65. Optimum pressure penalty is observed in case of annular fin with diameter ratio of 0.65.
- Optimum value of performance index is achieved at diameter ratio 0.65 for 2500< *Re* <13000. Performance index increases by 37%, and 85% in case of annular fin with diameter ratio of 0.85 and 0.65 as compared to that of annular fin with diameter ratio of 1.00 at 2590 Reynolds number respectively. It increases by 41%, and 88% in case of annular fin with diameter ratio of 0.85 and 0.65 as compared to that of annular fin with diameter ratio of 1.00 at 12965 Reynolds number respectively.

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