

Heat Transfer Characteristics for Natural Convection through Inline and Staggered Horizontal Banks of Tubes

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Abstract: In the present study, heat transfer characteristics of natural convection of air through 7x7 horizontal bank of tubes of 1cm diameter are studied numerically using Ansys-Fluent CFD code. Both the inline and staggered arrangements are considered. The study is carried out under constant heat generation in the tubes. The study is carried out at different spaces between the tubes in both the horizontal (longitudinal) and vertical (transverse) directions. The results of thermal and flow fields are obtained for each case. The effect of the longitudinal and transverse distances, S_L and S_T respectively, between the tubes on heat transfer is obtained for both the cases of inline and staggered arrangements. The studied range of both S_L and S_T is 1.2 to 4 cm. The results show that the heat transfer rate increases and the temperature decreases as a result of increasing the distance between the tubes in both the longitudinal and transverse directions. The effect of the longitudinal distance between the tubes on heat transfer is more pronounced than the corresponding effect of the transverse distance. Staggered arrangement has better heat transfer characteristics than the inline arrangement. Two correlations relating Nusselt number with tube spacing, one for the inline arrangement and the other for the staggered, were derived from the present results. Ideal tube spacing for the studied range of S_L and S_T was determined for the two arrangements.

Keywords: Heat transfer, horizontal tube bank, inline and staggered.

1. INTRODUCTION

Natural convection flow through horizontal tubes exists in many Engineering applications such as some types of heat exchangers, passive solar energy collectors and velocity measurements using the hot wire anemometer. A large amount of literature was presented for natural convection from horizontal cylinders. First, natural convection heat transfer from a horizontal cylinder has been studied and recently the research is directed to arrays of horizontal cylinders. Champagne et al. [1] showed that the temperature is uniform at the centre of a heated cylinder if $L/D > 200$. Kuehn T. and Goldstein R. [2] reported that the heat transfer from a horizontal cylinder behaves like heat transfer from a line heat source for low Rayleigh numbers. For larger Rayleigh numbers, i.e., $10^4 \leq Ra_D \leq 10^8$, the flow forms a laminar boundary layer around the cylinder. For larger Ra_D , the flow becomes turbulent. Many correlations for this problem were developed for the purpose of estimation of heat transfer coefficient. One set of these correlations was in the form: $Nu = a(GrPr)^b$, where the coefficient and exponent in the previous correlation depend on the level of $GrPr$. Also, other correlations of more complicated forms were presented. A summary of these correlations were given by Jagdish Chand and Dharam Vir [3]. In attempt to determine the appropriate constants, the applicability of some recommended correlations to experimental data on horizontal cylinders was checked.

The best coefficient is found to be 0.47 recommended by Fishenden and Saunders [4] with an exponent of 0.25 for the case of air. A comprehensive survey of the investigations done for this problem with the correlations derived analytically and experimentally for the earlier investigations and the ones derived by computational fluid dynamics for the recent investigations were listed by S.K.S. Boetcher [5]. Natural convective heat transfer from a horizontal array of heated cylinders to air was investigated experimentally by Kenzo Kitamura et al. [6]. They studied the effect of cylinder diameter, number of cylinders and horizontal spacing on the average Nusselt number (Nu). Optimal spacing for maximum Nu was determined. Toshiyuki Misumi et al. [7] studied experimentally the turbulent natural convective flows of air around large horizontal cylinders. They reported the conditions under which separations and turbulent transition occur. Claudio Cianfrini et al. [8] studied numerically the steady laminar free convection in air from an isothermal horizontal cylinder affected by a parallel cylinder of different diameter. They reported that there was two critical distance between the two cylinders. Reducing the space between the two cylinders below the first critical distance, the thermal state of the upper cylinder became effective on the thermal behaviour of the bottom cylinder. Smaller than the second critical distance, the heat transfer from the bottom cylinder degraded due to the existence of the upper cylinder. Natural convection heat transfer from a pair of vertically aligned horizontal cylinders was investigated by Olivier Reymond et al. [9]. Distributions of local heat transfer coefficients around the circumference of the cylinders were obtained for different spacing and different values of Raleigh number. They reported that if one cylinder is unheated, the other will not thermally affected by it. When both cylinders are heated, they found out that there was a plume rising from the lower cylinder which significantly affected the circumferential distribution of the heat transfer coefficient on the upper cylinder. Duli Yu et al. [10] studied numerically the mixed convection heat transfer through 3x3 tube bank placed between two vertical plates. The local data for different cylinder to cylinder spacing in the stream and transverse direction were obtained. They showed that the average Nu increases 20-30% by increasing the stream wise distance by 50%. Atmane et al. [11] studied the effect of vertical confinement on the surface heat transfer from a single heated cylinder. They showed that the heat transfer at the top of a vertically confined cylinder was enhanced due to the oscillation of the confined plume from one side to the other for H/D (height of the confinement to diameter) ratios of 0.5 and 1.0. Tokura et al. [12] studied the effect of a vertical confinement below a single heated horizontal cylinder. They showed that in the case where a cylinder is close to the bottom of the tank, the heat transfer would be decreased at the bottom of the cylinder. There is no effect of the bottom of the tank if the space between the bottom of the tank and the cylinder is greater than 0.1D. Many researches were directed to the problem of forced flow across bank of tubes. But few were done for natural convection across them. In the present work, the flow and heat transfer characteristics of natural convection through 7x7 horizontal bank of heated tubes of 1 cm diameter are studied numerically. The effect of tubes spacing in both the vertical and horizontal directions on the flow and heat transfer characteristics is presented. Comparison between the case of inline arrangement and that of staggered arrangement of the tubes is carried out.

Nomenclature

C	Specific heat at constant pressure, J/kg.K
D	Diameter of the tube, m
g	Gravitational acceleration m/s ²
Gr	Grashof number, dimensionless
H	Height of the vertical confinement, m
k	Thermal conductivity, W/mK
L	Length of the tube, m
Nu	Nusselt number, dimensionless
Pr	Prandtl number, dimensionless
P	Pressure, N/m ²
Ra	Rayleigh number, dimensionless

Ra _D	Rayleigh number based on tube diameter, dimensionless
S _L	Longitudinal pitch measured between tube centers in horizontal direction, cm
S _T	Transverse pitch measured between tube centers in vertical direction, cm
T	Temperature, K
T _o	Bulk temperature
T _{avg}	Average temperature, K
T _{Max}	Maximum temperature, K
u,v	Velocity components in X and Y directions, m/s
Q	Volumetric heat source, W/m ³

Greek Symbols

β	Thermal expansion coefficient, 1/K
ρ	Density, kg/m ³
μ	Dynamic viscosity, kg/ms

2. NUMERICAL MODELING

2.1 Physical model, governing equations and boundary conditions

The geometry of the physical model is illustrated in Fig.1. Both inline and staggered arrangements of heated tubes of 1 cm diameter are considered as shown in Fig.1 (a) and (b) respectively. The study is carried out for natural convection from these heated tubes in air of a bulk temperature of 300K. The thermal and flow fields are calculated numerically with commercial CFD software ANSYS FLUENT 19 R3. The flow is assumed at steady state, incompressible and turbulent. The effect of viscous dissipation and thermal radiation is neglected. The fluid and the solid properties are assumed to be constant. Buoyancy effects are modeled using Boussinesq approximation as the change in air density is within the applicable range of this assumption. The governing equations of the present model in two dimensions for constant properties are [13]:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad \text{Continuity eq.} \tag{1}$$

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = - \frac{1}{\rho} \frac{\partial p}{\partial x} + \frac{\mu}{\rho} \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \quad \text{(X momentum)} \tag{2}$$

$$u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = - \frac{1}{\rho} \frac{\partial p}{\partial y} + \frac{\mu}{\rho} \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) + g\beta(T - T_o) \quad \text{(Y momentum)} \tag{3}$$

$$\rho C_p \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) = k \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) + Q \quad \text{(energy)} \tag{4}$$

Where: *K*, *ρ*, *P*, and *T* are the fluid thermal conductivity, density, pressure, and temperature respectively. Whereas *V_i* and *V_j* indicate the velocity components in x and y directions respectively and *Q* is volumetric heat source. The SST (Shear-Stress-Transport) *k* - *ω* turbulence model has been adopted for turbulence modeling. The SST model includes both the *k* - *ω* turbulence model which is used near the wall and *k* - *ε* model used away from the wall as a unified two equation turbulence model. Table 1 shows the main Simulation domain parameters and boundary conditions.

Table 1 Simulation domain main parameters

Tube diameter D	1 cm					
Far field dimensions	100 cm x 100 cm					
S _L	1.2	1.6	1.8	2	3	4
S _T	1.2	1.6	1.8	2	3	4
Boundary conditions	Symmetry for the right boundary conditions & Outlet pressure for all other boundaries					
Tubes heat generation	40000 W/m ³					

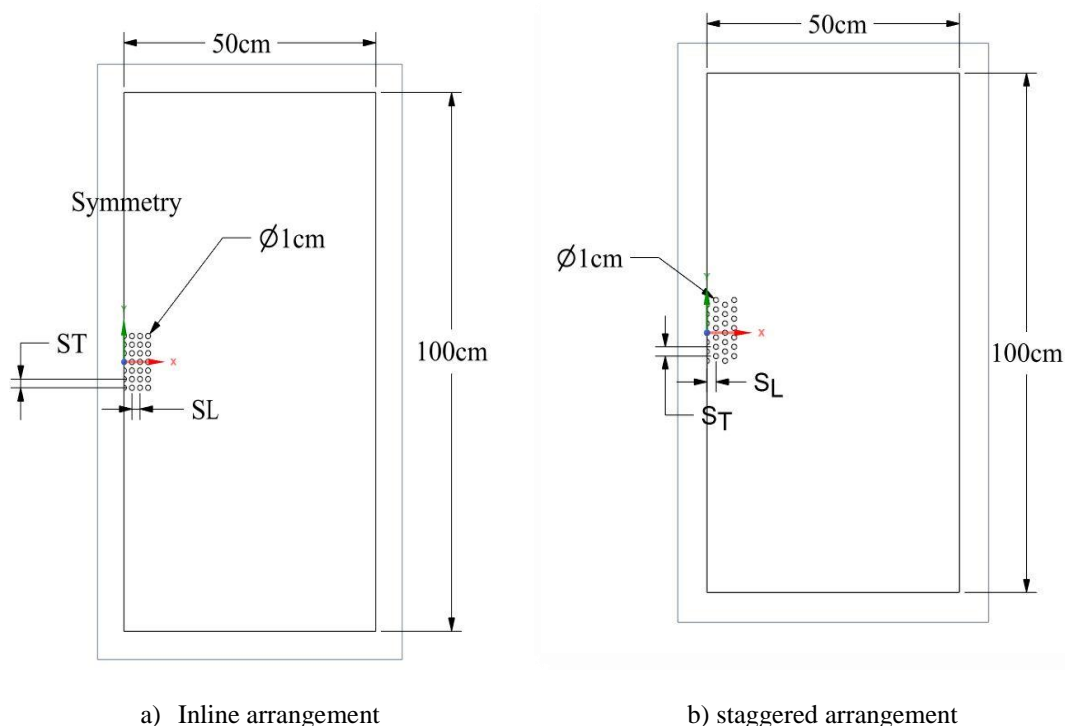


Fig.1 Dimensions and arrangements of the tubes

2.2. Grid test and model verification

In this work, grid independent test is carried out to verify that the results are independent from the used nodes number. To accomplish this purpose, the numerical study is conducted with different mesh densities. Starting from coarse structured mesh, the number of nodes is increased and the tube maximum temperature is checked. Then, the Y+ values on the tubes surfaces are checked. If Y+ values are greater than 1, a new fine mesh is generated and solved in ANSYS FLUENT and Y+ is checked again. This cycle is repeated until reaching a final acceptable mesh. The results show that at a nodes number of 326544, when the number of nodes increased from 326544 to 466647, a deviation of about 1 % in tubes maximum temperature is obtained. Whereas, when the number of nodes increases from 466647 to 544642, only 0.01% variation in tubes maximum temperatures is obtained. At the number of nodes of 544642, Y+ is verified to be in the order of 1 to match with small gradients of the flow and temperature in the region near the tubes surfaces. A structured mesh with inflation layers is generated on the tubes surfaces for all cases. For the purpose of verification of the numerical model, the flow and temperature fields around a single horizontal tube are calculated. The results are compared with available experimental correlations for single horizontal cylinder. Figure 2 shows a comparison of the present results of the variation of Nu with Ra for a single horizontal tube with the experimental results of Morgan [14] and Churchill and Chu [15]. As shown from this figure, the predicted results are in a very good agreement with the results of Churchill and Chu [15] and in a good agreement with these of Morgan [14] with a maximum deviation of about 5% at a Ra = 2E+7. Churchill and Chu correlation: $Nu = [0.825 + 0.387(GrPr)^{1/6} / \{1 + (0.492/Pr)^{9/16}\}^{8/27}]^2$, Morgan correlation:

$Nu = c Ra^n$, c and n are tabulated according to values of Ra.

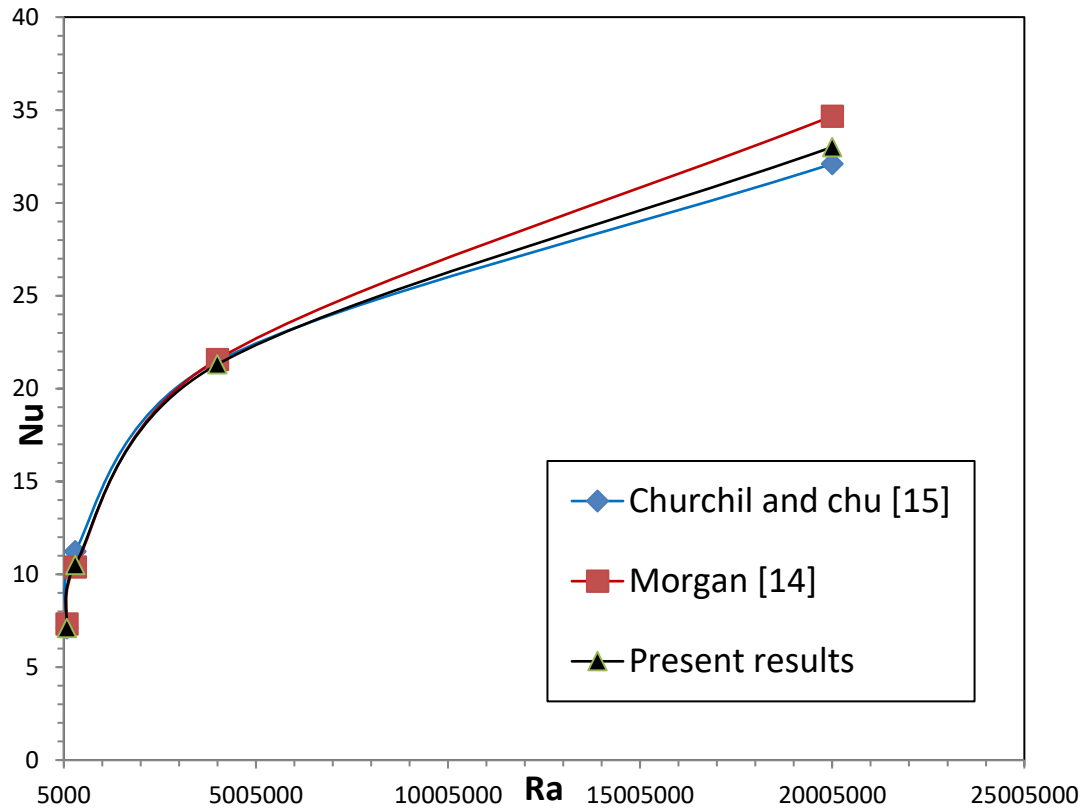


Fig.2 Comparison between the present results with results of Morgan [14] and Churchill and Chu [15]

3. RESULTS AND DISCUSSION

3.1 Thermal and flow fields for inline arrangement

Figures 3 and 4 show the temperature and velocity contours respectively in the studied domain at different values of S_T (distance between the tubes in the vertical direction) and a fixed value of $S_L = 1.2$ (distance between the tubes in the horizontal direction) for the case of inline arrangement. Due to the heated tubes, natural convection flow circulation is established through them due to the temperature difference between the tube surface and the outside fluid. So, the bulk outside cold fluid moves towards the tubes and in the upward direction. As shown from Fig.3, the temperature rises from the value corresponds to that of the outside bulk fluid (300K) at the region of the bottom row and the two vertical columns of tubes. Continue flowing upward due to natural convection, the temperature increases. This is because the fluid carries more heat from the hot pipes by constantly flowing upwards through them. The middle region of the pipes has higher temperatures than the sides due to the influence of the cold fluid adjacent to the side regions. Therefore, the region with the maximum temperature is at the top of the central region of the heated tubes. At the exit of the hot tubes zone, the maximum temperature of the plume is also in the middle and decreases towards the sides and by continue flowing upwards. This is due to the effect of mixing of the hot fluid coming out from the heated zone with the cold external fluid. Figure 4 shows that the velocity in the region between the pipes is very low at low values of S_T , i.e for $S_T = 1.2$ and 1.6 , due to the suffocation of the fluid in this area due to the lack of distances between the pipes. In these cases, the distance between the pipes is lower than the value of the velocity boundary layer thickness, and hence the fluid faces high resistance to flow. In these cases, the natural convection between the tubes is weak due to low temperature gradient between the tubes inside the bundle. Natural convection is better at the surfaces of the external tubes of the two external vertical columns and at the upper horizontal row as shown in Fig.4. This is due to the higher temperature gradient at this region. This leads to greater flow velocity near the surface of the external pipes. The maximum flow velocity is at the middle of the top end of the heated domain as shown in Fig.4. This occurs as the flow from the two vertical sides of the heated domain combines at the centre of the top surface going upward. Increasing the value of the distance S_T causes the resistance to flow to decrease as the distance between the

tubes becomes greater than the velocity boundary layer thickness. This causes the flow velocity to increase consequently as shown in Fig.4. This leads to decrease in the temperature values as shown in Fig.3 due to better convection through the tubes.

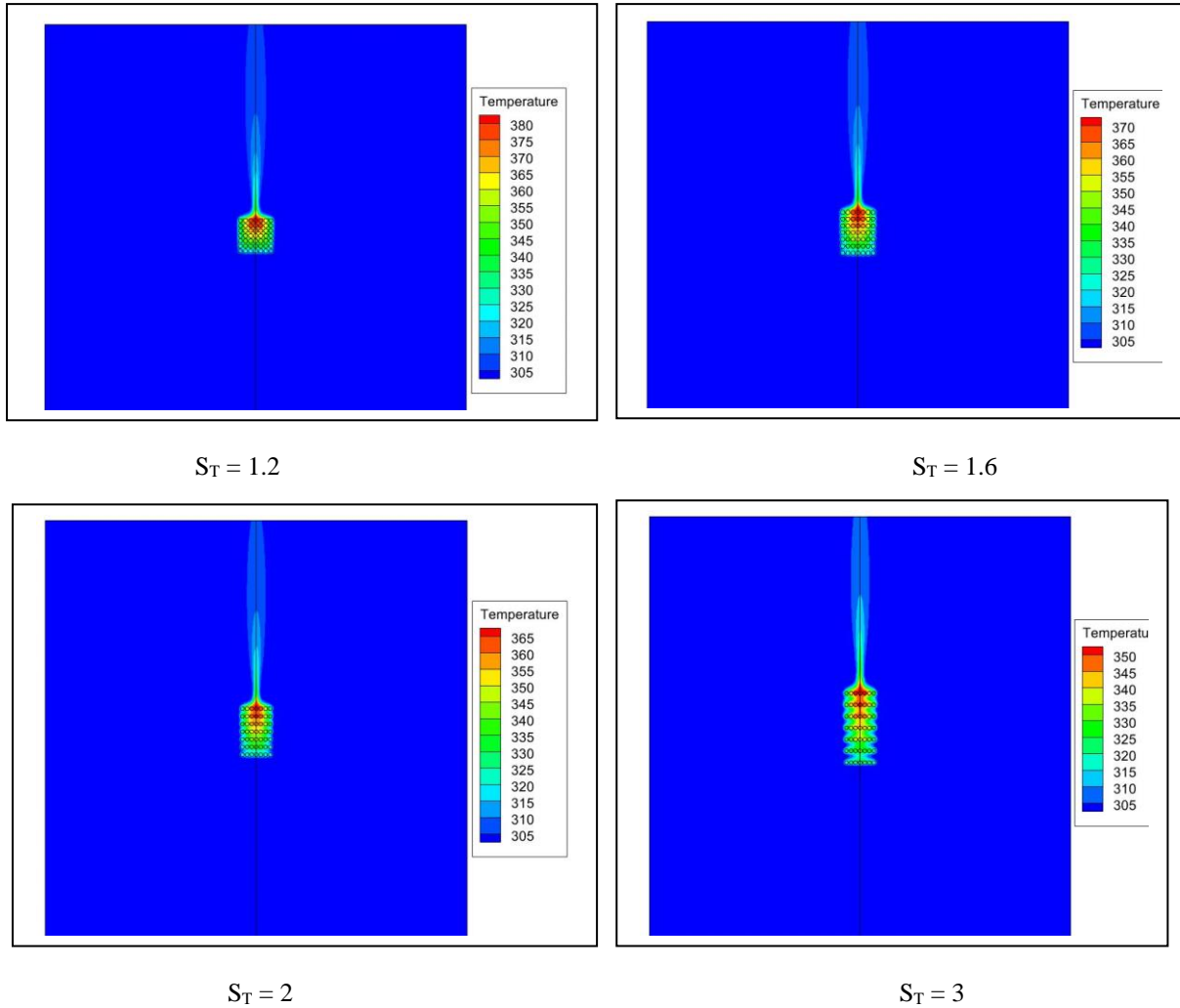
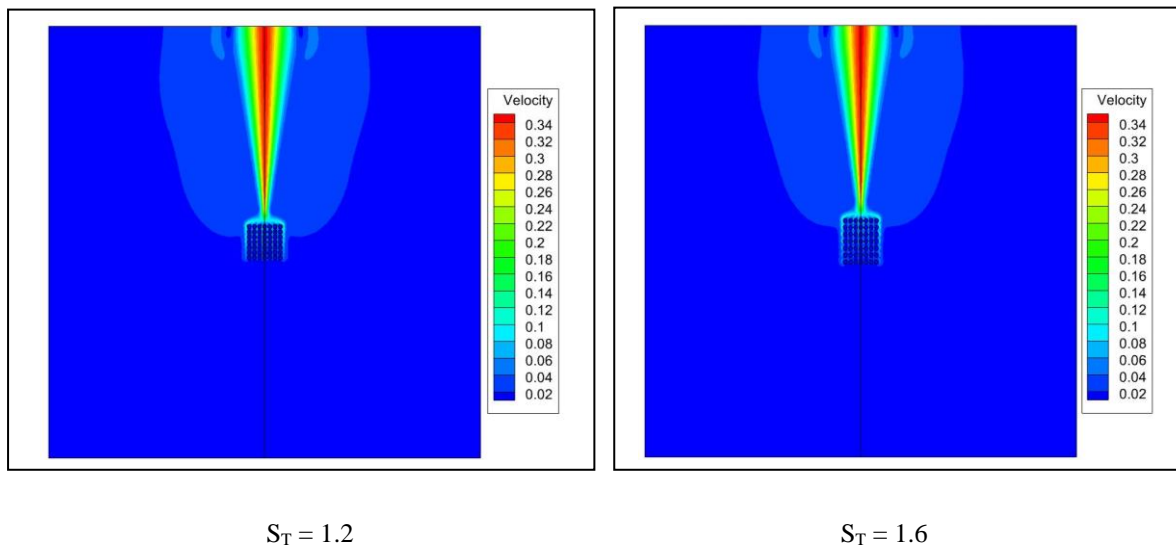


Fig.3 Temperature contours for inline arrangement at different S_T and $S_L = 1.2$



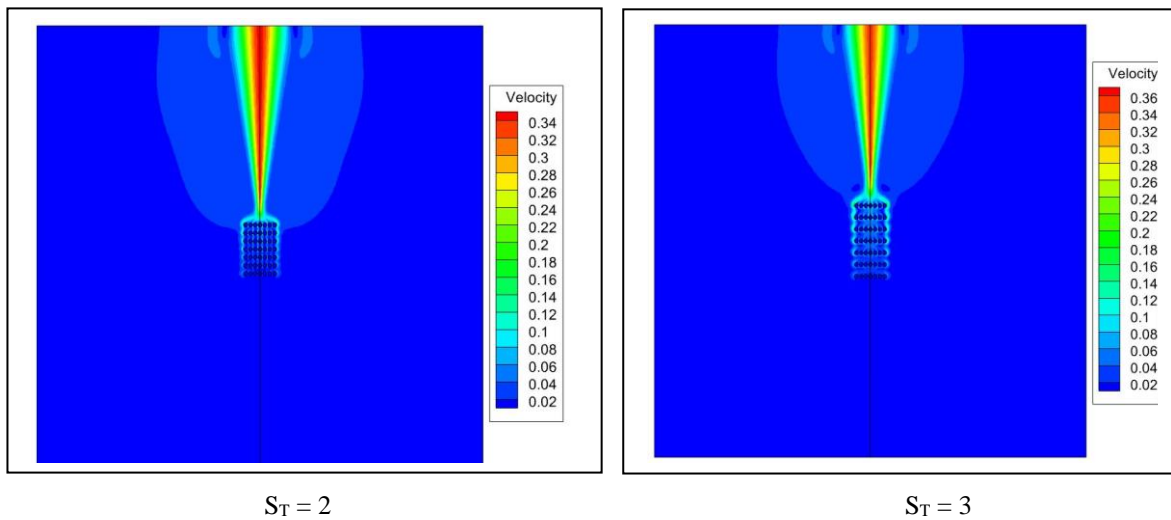
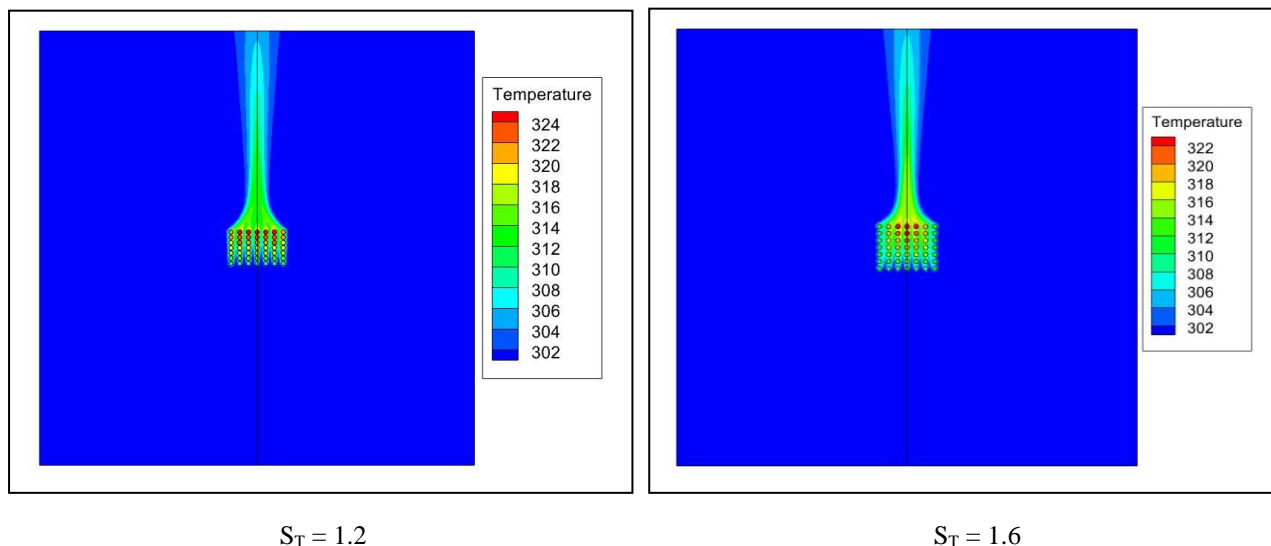
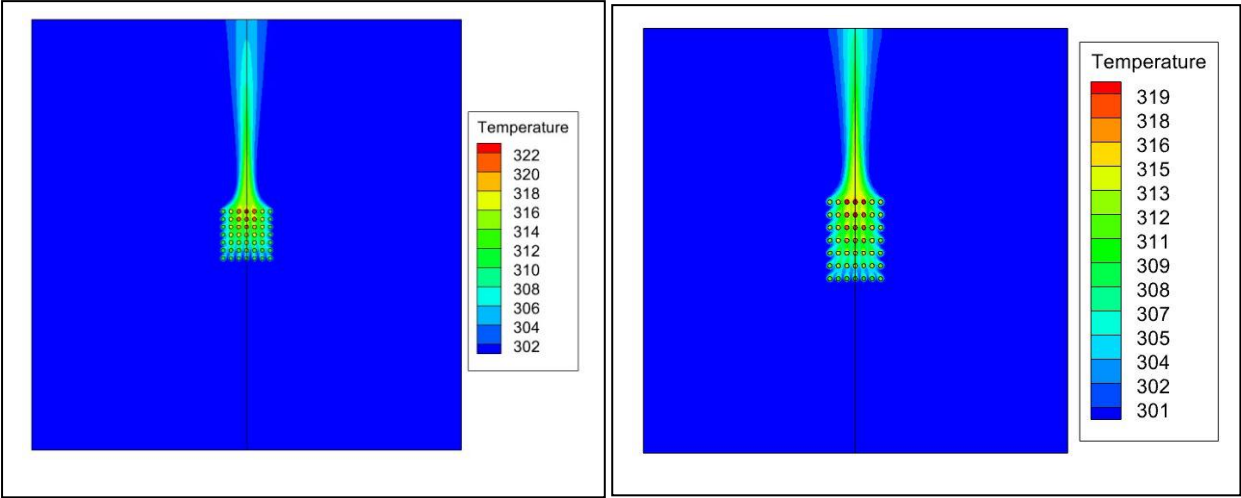


Fig.4 Velocity contours for inline arrangement at different S_T and $S_L = 1.2$

Figures 5 and 6 show the temperature and velocity contours respectively in the studied domain at different values of S_T and a fixed value of $S_L = 2$ for the case of inline arrangement. As shown from these figures, as S_T increases the velocity increases due to low flow resistance and hence the temperature values decrease. Individual plumes from the tubes and interaction between them increase with increasing S_T . At the same S_T , the flow velocity is greater due to lower flow resistance and hence the temperature values are lower in the case of $S_L = 2$ than for the previous case in which $S_L = 1.2$. This can be seen from the comparison between the figures 4 and 6 for the velocity and figures 3 and 5 for the temperature. Comparison between the previous figures shows also that the effect of increasing S_L from 1.2 to 2 on the reduction of the resistance to the flow is greater than the corresponding effect of increasing S_T from 1.2 to 2. The zone of maximum temperature is still at the top of the central region of the heated tubes as in the case of $S_L = 1.2$. The effect of increasing S_L on the temperature reduction is more than the corresponding effect of increasing S_T , as shown from the comparison between the temperature values at the same S_T and the comparison between the temperature values at the same S_L . For example, increasing S_T from 1.2 to 2 causes the maximum temperature to reduce from 380°C to 365°C at a fixed value of $S_L = 1.2$ and from 324°C to 322°C when $S_L = 2$. Whereas, increasing S_L from 1.2 to 2 causes the maximum temperature value to reduce from 380°C to 324°C at a fixed value of $S_T = 1.2$ and from 365°C to 322°C at $S_T = 2$. The same note can be reported for the velocity. For example, increasing S_T from 1.2 to 2 at a fixed value of $S_L = 1.2$, does not affect the maximum velocity value, and causes the maximum velocity to increase from 0.38 m/s to 0.40 m/s at $S_L = 2$. Whereas, increasing S_L from 1.2 to 2, causes the maximum velocity to increase from 0.34 m/s to 0.38 m/s when $S_T = 1.2$ and from 0.34 m/s to 0.40 m/s at $S_T = 2$. The above analysis shows also that the effect of both S_T and S_L on the temperature is more pronounced at lower S_L and S_T respectively.

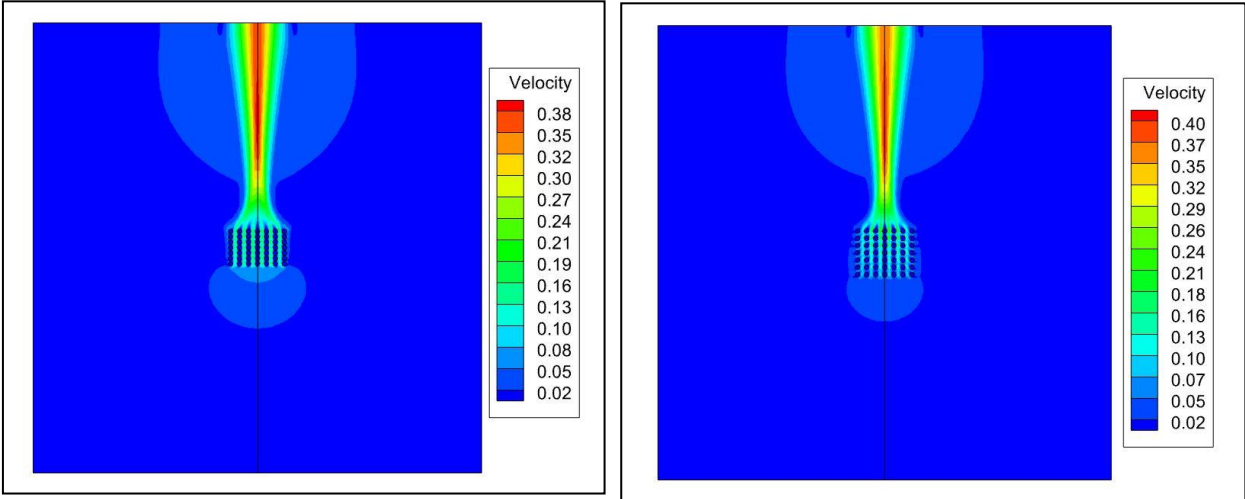




$S_T = 2$

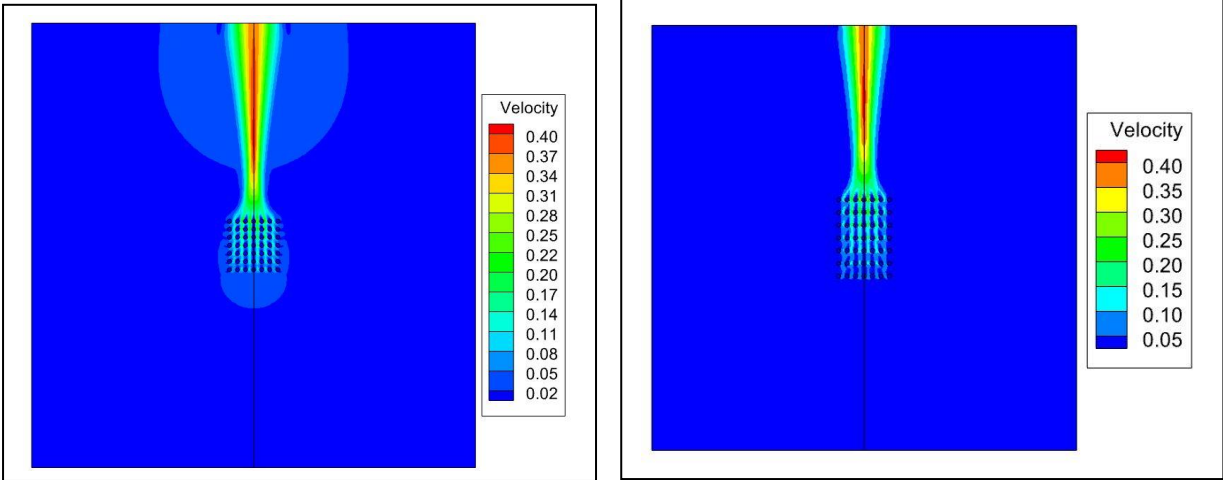
$S_T = 3$

Fig.5 Temperature contours for inline arrangement at different S_T and $S_L = 2$



$S_T = 1.2$

$S_T = 1.6$



$S_T = 2$

$S_T = 3$

Fig.6 Velocity contours for inline arrangement at different S_T and $S_L = 2$

3.2 Thermal and flow fields for staggered arrangement

Figures 7 and 8 show the temperature and velocity contours respectively in the studied domain at different values of S_T and a fixed value of $S_L = 1.2$ for the case of staggered arrangement. As shown from Fig.7, at the first row of tubes, the temperature rises from the value of the bulk fluid of 300 K by the effect of heating and continue to increase as the fluid continue to move upward due to accumulation of heat as discussed above. As S_T increases, plumes from the bottom rows of tubes combine with the plums from the higher rows to form a single plume moving upward. This plume becomes more intense as S_T increases as shown from Fig.7. As discussed above for the case of inline arrangement, the middle region of the pipes has higher temperatures than the sides and the region of maximum temperature is also at the top of the central region of the heated tubes. Temperature values decrease while velocity values increase with S_T as in the previous case of inline arrangement. Comparison between Fig.3 and Fig.7 shows that the temperature values for the case of staggered arrangement are lower than the corresponding ones for the case of inline arrangement. This can be attributed to the higher spaces between the tubes at the same values of S_T and S_L , due to the difference between the tubes arrangement. Higher spaces between the tubes lead to higher temperature gradients between them in addition to lower flow resistance which lead to better natural convection and higher flow velocity between the tubes. This can

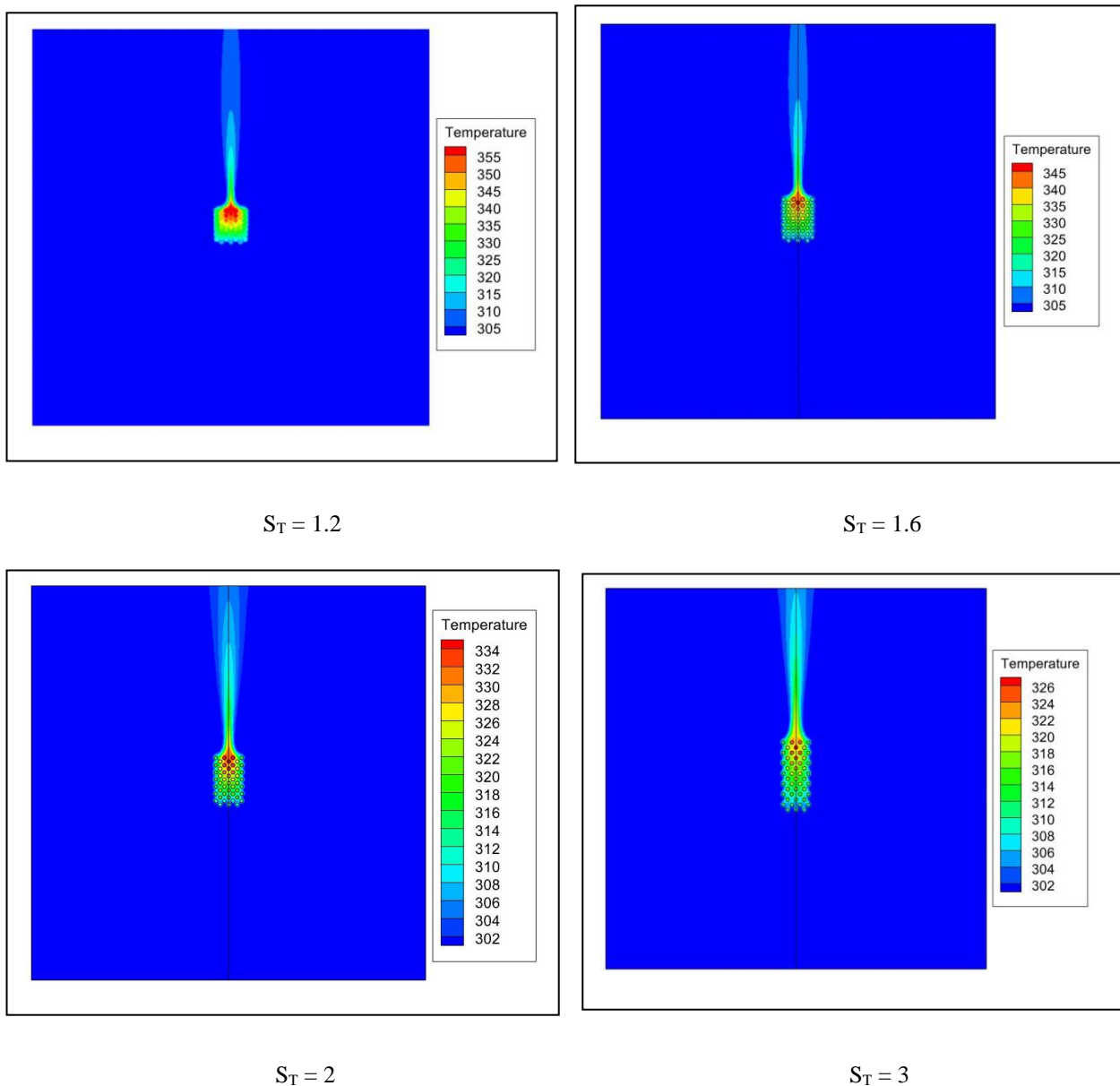


Fig.7 Temperature contours for staggered arrangement at different S_T and $S_L = 1.2$

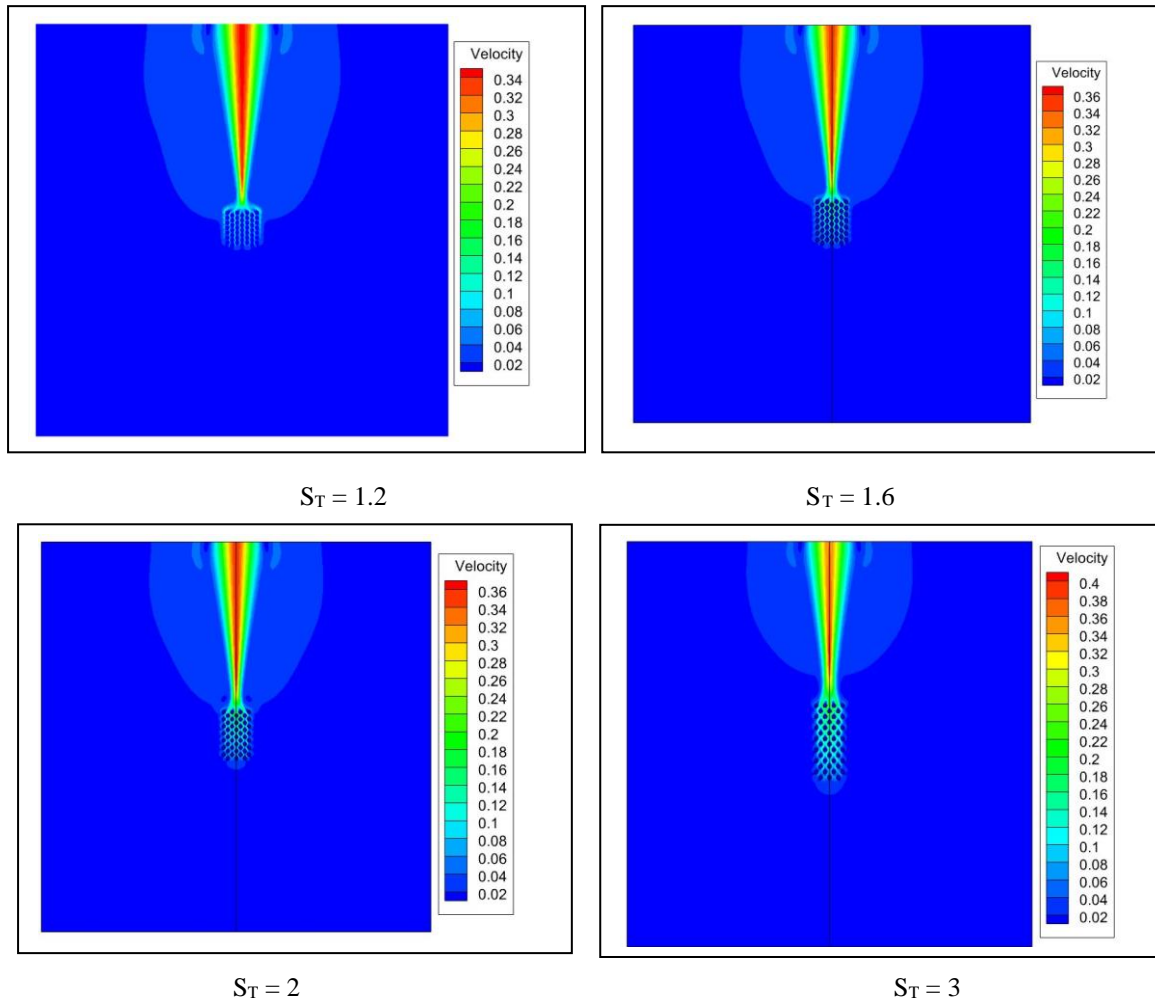
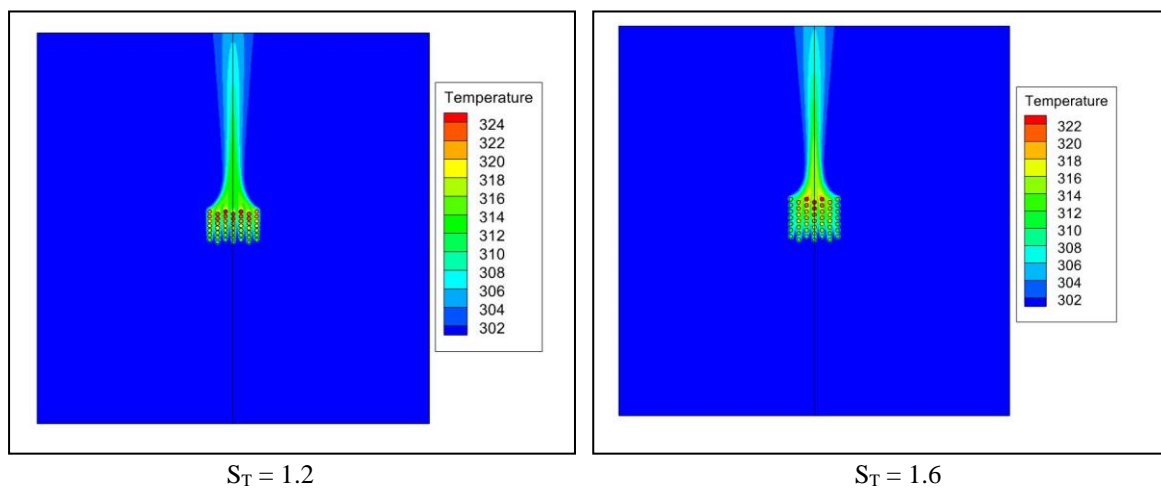
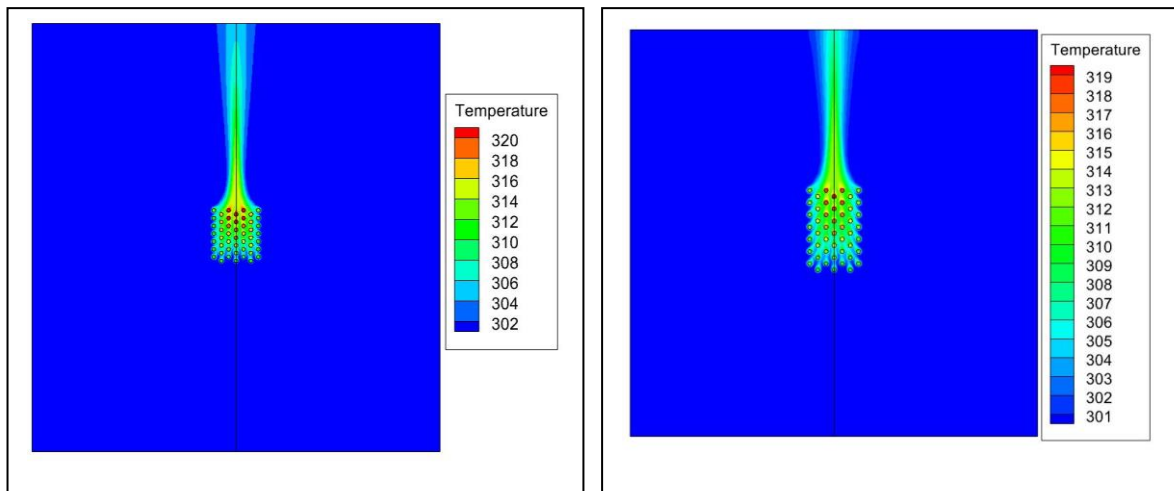


Fig.8 Velocity contours for staggered arrangement at different S_T and $S_L = 1.2$

be shown from the comparison of Fig.4 with Fig.8. Better natural convection leads to lower temperature values in the case of staggered arrangement. As in the case of inline arrangement, the effect of S_L on the temperature and velocity is more pronounced than the effect of S_T . This is deduced by comparison between figures 7 and 9 for temperature values and figures 8 and 10 for velocity. But the effect of S_L is more pronounced in the case of inline arrangement. This seems from the comparison between the reductions in the maximum temperature value for the inline arrangement from 380K to 324K by increasing S_L from 1.2 to 2 at constant S_T of 1.2 and corresponding reduction for the case of staggered arrangement. This reduction for the staggered arrangement is from 355K to 324K.

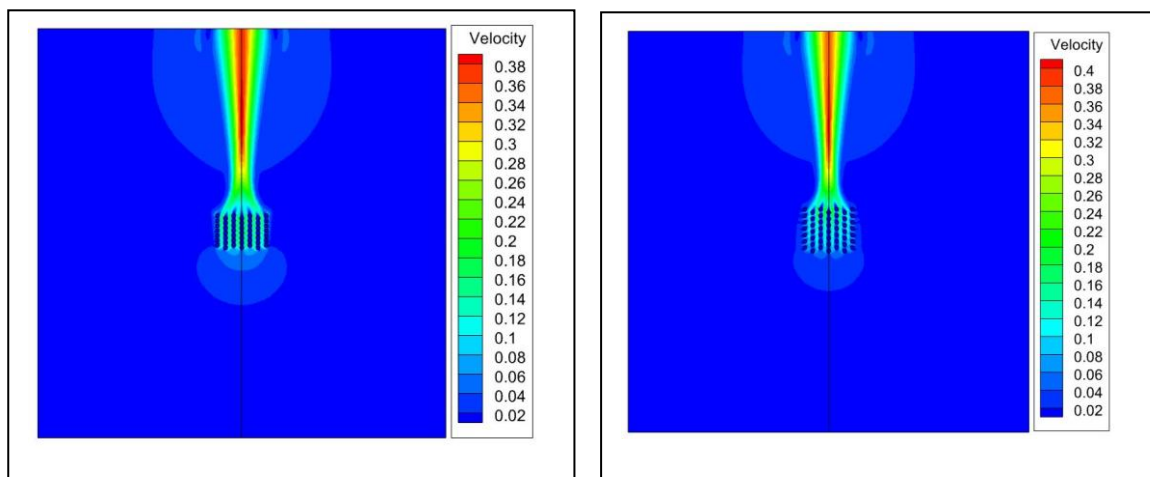




$S_T = 2$

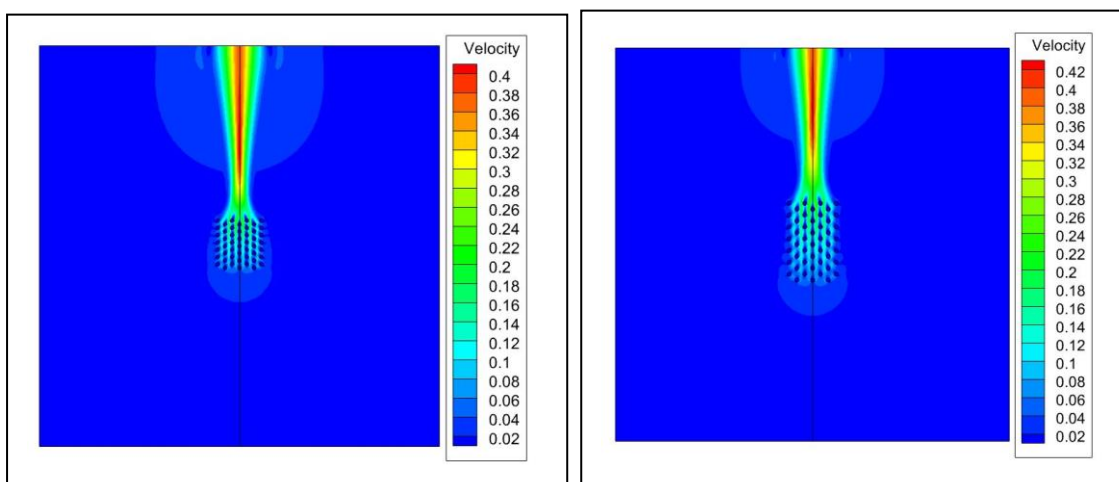
$S_T = 3$

Fig.9 Temperature contours for staggered arrangement at different S_T and $S_L = 2$



$S_T = 1.2$

$S_T = 1.6$



$S_T = 2$

$S_T = 3$

Fig.10 Velocity contours for staggered arrangement at different S_T and $S_L = 2$

3.3 Comparison between inline and staggered arrangements

Figure 11 show the variation of Nusselt number with S_L and different values of S_T for both the cases of inline and staggered arrangements for the purpose of comparison between them. Figures 12 and 13 show the corresponding results of the average and maximum temperatures respectively. Figure 11 shows that Nu increases with S_L and S_T for both the cases of inline and staggered arrangements. Increasing S_T at a fixed value of S_L or increasing S_L at a fixed value of S_T (except $S_T = 1.2$) causes Nu to increase for both inline and staggered arrangements. Nu has a maximum value for only the case of $S_T = 1.2$. This maximum value is between $S_L = 1.8$ and 2 for both arrangements. The trend of Nu at $S_T = 1.2$ can be explained by the interference between the velocity boundary layers of the adjacent tubes and the flow between them for low values of S_L . This leads to low temperature gradient between the tubes and hence poor heat transfer and low values of Nu. As S_L increases, the space between the tubes becomes close to or greater than the velocity boundary layer thickness and the flow through the tubes increases resulting in better heat transfer and higher values of Nu until reaching a maximum value. After reaching this maximum value, increasing S_L causes the tubes to be far apart such that there is no effect of the interaction between them on Nu. For $S_T > 1.2$, the rows between the tubes are far apart such that the interaction between them is weak, so there is no maximum value for Nu. From figures 11 to 13, it can be seen that for all values of S_L , the heat transfer expressed by Nusselt number increases with S_T at a fixed value of S_L and hence the maximum and average values of the temperature decrease. The increase in Nu with S_L is more pronounced at lower values of S_L (lower than 1.8) as shown from the slopes of the curves of Fig.11. This is true for all values of S_T except for the case of $S_T = 1.2$. For $S_T > 1.2$, as shown from figures 12 and 13, the maximum and average temperatures decrease with S_L with higher slopes at lower values of S_L (lower than 1.8) and continue to decrease with lower slope as S_L increase for both arrangements. At $S_L = 3$, they remain constant for the inline arrangement whereas they continue to decrease for the staggered one. With respect to S_T , the effect of the increase in S_T on Nu and the temperatures is more pronounced by increasing S_T from 1.2 to 1.6 as shown from figures 11 to 13. At higher values of S_T ($S_T > 1.6$), the tubes are far apart and there is no interference between their velocity boundary layers and the flow is not affected as before for lower values of S_L . So, both S_L and S_T have higher effect on heat transfer at low values. Staggered arrangement has better heat transfer than inline arrangement for low values of S_L (lower than 1.6). This can be shown from Fig. 11 for the values of Nu at $S_L < 1.6$ and figures 12 and 13 for the corresponding values of average and maximum temperature respectively. This is true for all values of S_T in this range. The reason is that at lower values of S_L the space between the tubes is lower than the velocity boundary layer in inline arrangement which hinders the flow resulting in low temperature gradient and weak natural convection. For the case of staggered arrangement, due to the nature of this arrangement, the space between the tubes at the same S_L approach the velocity boundary layer thickness resulting in higher temperature gradient between the tubes and hence better heat transfer compared with the inline arrangement. At values of $S_L > 1.6$, the heat transfer represented by the values of Nu and the average and maximum temperatures are nearly the same for both the inline and staggered arrangement. Effect of increasing S_T on the improvement of heat transfer for the staggered is greater than the corresponding effect for inline arrangement as evidenced by the more divergent distances between the curves of constant S_T in the figures from 11 to 13. For the case of inline arrangement, the effect of S_L on heat transfer is more pronounced than the effect of S_T at lower values of S_L (lower than 1.8).

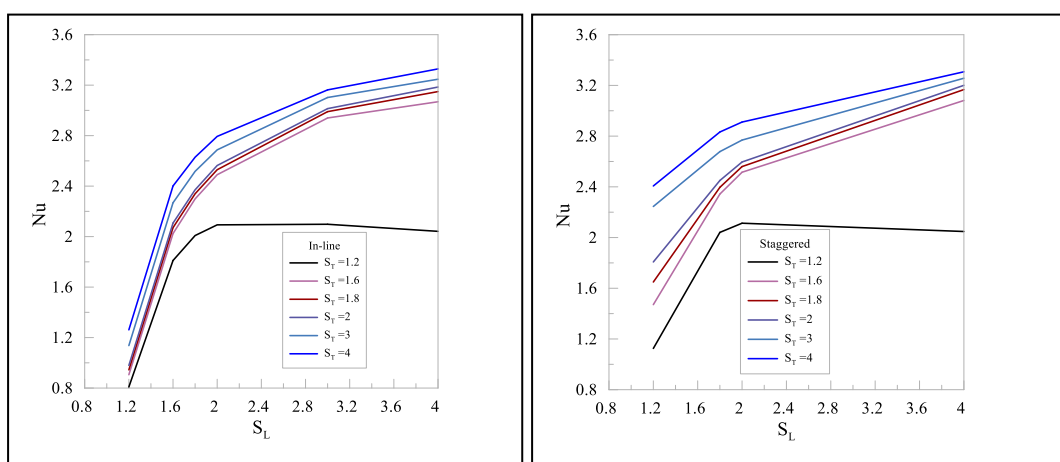


Fig.11 Variation of Nusselt number values with S_L and different values of S_T

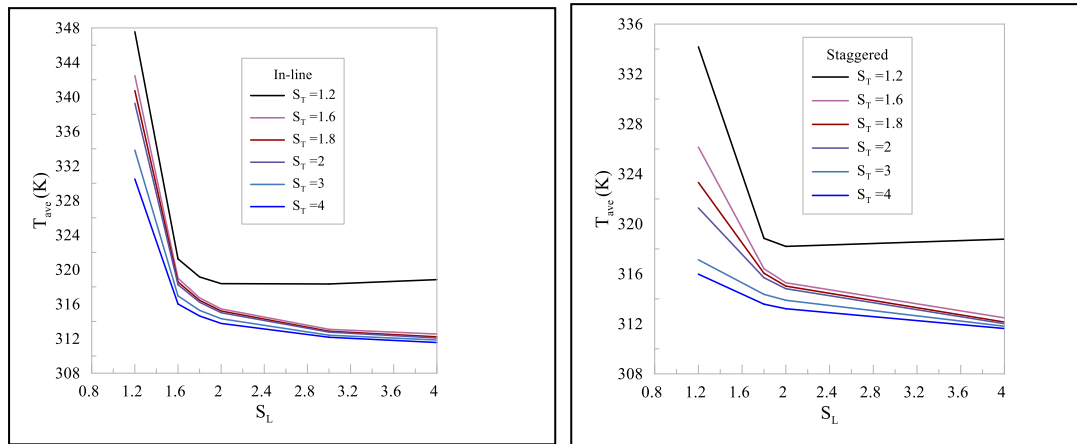


Fig.12 Variation of average temperature values with S_L and different values of S_T

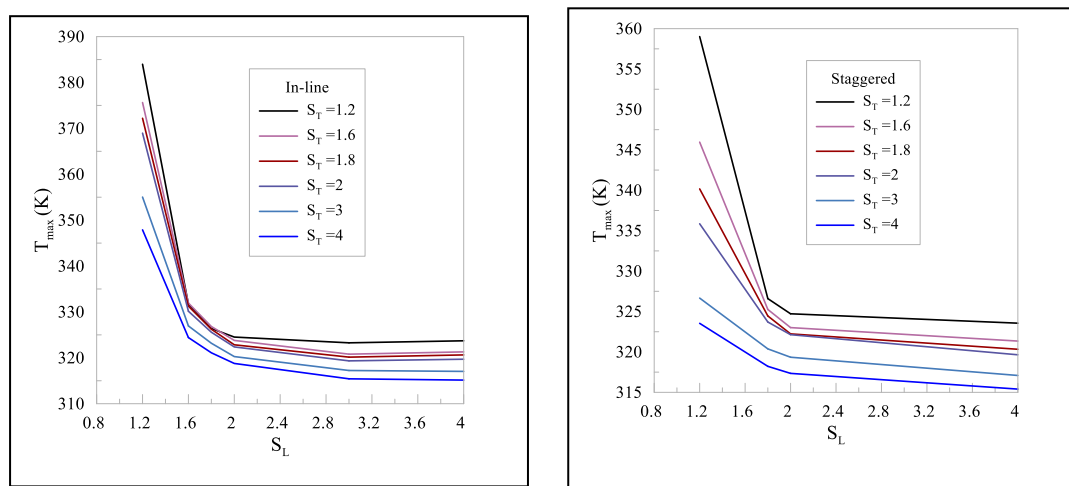


Fig.13 Variation of maximum temperature values with S_L and different values of S_T

3.4 Correlations and ideal tube spacing for the present results

From the above analysis in section 3.3, it can be shown that for the case of inline arrangement, at $S_L = 3$, the average and maximum temperatures remain constant in spite of increasing S_L as shown in figures 12 and 13. This is for all the values of S_T . At $S_L = 3$, increasing S_T causes both the average and maximum temperatures to decrease. So, it can be deduced that in the studied range of S_L and S_T , optimum tube spacing for the case of inline arrangement is at $S_L = 3$ and $S_T = 4$. As at these values the lowest possible average and maximum temperatures are obtained with the lowest volume. Whereas for the case of staggered arrangement the maximum temperature continues to decrease with both S_L and S_T (see Fig.13). So in the studied range the ideal tube spacing is at $S_L = S_T = 4$ as lowest temperatures are obtained under this condition. Data fit software program was used to make best fit for the present numerical results obtained in this work. So, two correlations relating Nusselt number with the tube spaces S_L and S_T were obtained. One correlation was for the inline arrangement whereas the other was for the staggered one. These correlations are:

$$Nu = 1.2669885 S_L^{0.5566} S_T^{0.22675} \quad (5) \text{ for inline arrangement.}$$

$$Nu = 1.51979846 S_L^{0.3553176} S_T^{0.2852526} \quad (6) \text{ for staggered arrangement.}$$

Not that the two previous correlations are only valid in the studied range:

$$1.2 \leq S_L \leq 4, \quad 1.2 \leq S_T \leq 4.$$

4. CONCLUSION

Natural convection flow through horizontal tubes is studied numerically using Ansys-Fluent CFD code. Both the inline and staggered arrangements are considered. The effect of the distance between the tubes in the direction of flow and normal to it on the heat transfer characteristics is presented. The following conclusions from the present work are:

- 1- The zone of maximum temperature is at the top of the central region of the heated tubes for all of the studied cases.
- 2- Heat transfer improves with the increase of the distance between the tubes in both the transverse and longitudinal directions S_T and S_L respectively for both the inline and staggered arrangements.
- 3- The effect of both the tube spaces S_L and S_T on heat transfer is more pronounced at their lower values.
- 4- Heat transfer is better in the case of staggered arrangement than that of inline arrangement at values of S_L lower than 1.6 and it is approximately the same for the two arrangements at higher values.
- 5- The effect of increase of S_T on the improvement of heat transfer is greater for the staggered than the corresponding effect for the inline arrangement.
- 6- The effect of the distance S_L is more pronounced on heat transfer than the corresponding effect of S_T for $S_L < 1.8$ for both the inline and staggered arrangement but this is more pronounced for the inline arrangement.
- 7- For the studied range of S_T and S_L , ideal tube spacing is at $S_L = 3$ and $S_T = 4$, for inline arrangement and at $S_L = S_T = 4$ for staggered arrangement.
- 8- Two correlations were derived from the present results relating Nusselt number with tube spacing, one for the inline and the other for the staggered arrangement (equations 5,6) respectively.

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