The Optimal Spacing between Finned Tubes Cooled by Free Convection Using Constructal Theory

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Abstract

The optimal spacing between finned tubes cooled by free convection is studied numerically. A row of isothermal finned tubes are installed in a fixed volume and the spacing between them is selected according to the constructal theory (Bejan's theory). In this theory the spacing between the tubes is chosen such that the heat transfer density is maximized. A finite volume method is employed to solve the governing equations; SIMPLE algorithm with collocated grid is utilized for coupling between velocity and pressure. The range of Rayleigh number is ($10^3 \leq Ra \leq 10^5$), the range of the tube position is ($0.25 \leq \delta \leq 0.75$), and the working fluid is air (Pr =0.71). The results show that the optimal spacing decreases as Rayleigh number increases for all tube positions, and the maximum density of heat transfer increases as the Raleigh number increases for all tube positions and for Ra=10^5 the highest value of heat transfer density occurs at tube position (δ =0.75) while the lowest value occurs at tube position (δ =0.25). The results also show that the optimal spacing remains constant with change of the tube position at constant Rayleigh number.

Keywords: Constructal theory, optimal spacing, finned tubes, natural convection

Nomenclature

b	Position of the tube (m)
d	diameter of the tube = 2a(m)
D	Non-dimensional diameter of the tube
g	Gravity acceleration (m/s^2)
h	Total height of fin and tube (m)
H_d	Dimensionless downstream extension
H_u	Dimensionless upstream extension
k	Thermal conductivity (W/m.k)
L	Total length of the domain (m)
p	Pressure (N/m^2)
P	Non-dimensional pressure
Pr	Prandtl number
q	Heat transfer rate (W)
Q	Dimensionless heat transfer density
Ra	Rayleigh number
s	Spacing between the tubes (m)
S	Dimensionless Spacing
t	Temperature (^{o}C)
T	Dimensionless temperature
T_w	Wall temperature (^{o}C)
T_∞	Ambient temperature (^{o}C)
u	Horizontal velocity (m/s)
U	Dimensionless horizontal velocity
v	Vertical velocity (m/s)
V	Dimensionless vertical velocity
V	Volume (m^3)
w	Width (m)
x	Horizontal Coordinate (m)
X	Dimensionless Horizontal Coordinate
y	Vertical coordinate (m)
Y	Dimensionless vertical coordinate

Greek Symbols

α	Thermal diffusivity (m^2/s)
β	Coefficient of thermal expansion (K^{-1})
δ	Dimensionless position of tube
ρ	Density (Kg/m^3)
ν	Kinematic viscosity (Pa.s)

Subscripts

Max	Maximum value
Opt	Optimum value

1. Introduction

In heat transfer, constructal theory (Bejan's theory) is used to generate the flow configuration by optimizing the heat transfer density under (space) volume constraint. Constructal theory states that the flow configuration is free to morph in the follow-up of maximal global performance (objective function) under global constraints, Bejan A. and Lorente S., (2008), [1]. By depending on constructal theory, the optimal spacing between plates and cylinders cooled by natural convection can be found, in each geometry, the total volume is fixed and the objective is to maximize the overall thermal conductance between the tubes. Bejan A., (1984), [2] found the optimal spacing between vertical plates installed in a fixed volume by using the intersecting of asymptotes method. The study was employed for isothermal vertical plates cooled by natural convection. He found that the optimal spacing was proportional to the Rayliegh number to the power of (-1/4). Bejan

A. et al. (1995), [3] carried out a numerical and experimental study of how to choose the spacing among horizontal cylinders installed in a fixed volume cooled by laminar free convection. They maximized the total density of heat transfer between the assembly and the ambient. The Numerical and experimental simulations cover the Rayleigh number range of $10^4 \leq Ra \leq 10^7$and Pr = 0.72. Ledezma G. A. and Bejan A., (1997), [4] investigated numerically and experimentally the free convection from staggered vertical plates installed in fixed space. They maximized the density of heat transfer and they considered three degrees of freedom; the horizontal spacing between adjacent columns, the stagger between columns and the plate dimensions. Numerical and experimental simulations cover the Rayleigh number range of $10^3 \leq Ra \leq 10^6$, and the working fluid was air with Pr=0.72. The conclusion demonstrated numerically and experimentally that it was possible to optimize geometrically the internal architecture of a fixed volume such that its global thermal resistance was minimized. Da Silva and A. Bejan,(2004),[5] studied numerically the free convection in vertical converging or diverging channel with optimized for density of heat transfer. They considered three degrees of freedom: the distribution of heat on the wall, wall to wall spacing, and the angle between the two walls. The optimization was performed in the range of $10^5 \leq Ra \leq 10^7$and Pr=0.7. The walls were partially heated either at top of the channel or at the bottom of the channel. They proved that the density of heat transfer increased by putting the unheated part at the upper sections. They also showed that the best angle among the walls was almost zero when Ra number was high. Da Silva A. K. and Bejan A., (2005), [6] designed numerically a multi-scale plates geometry cooled by free convection by using constructal theory. They maximized the density of heat transfer rate. They put small plates in the unused heat transfer area between the large plates. They used finite element method to discretize the governing equation in the range of Rayleigh number of $10^5 \leq Ra \leq 10^8$, and Pr= 0.7. They showed that the density of heat transfer increased by putting the small plates between the large plates. Da Silva A.K. et al. (2005), [7] studied the free convection from discrete heat sources placed in vertical open channel with the constructal theory. They considered two cases, the first was single heat source under variable size, and the second was heat sources with fixed size. They applied the constructal theory to maximize the thermal conductance between the cold air and the discrete heat sources or to minimize the hot spot on the hot sources. Rayleigh number was in the range of ($10^2 \leq Ra \leq 10^4$) and Pr = 0.7. They showed that for case one the thermal performance can be maximized as the heat source not covering the entire wall at Ra $=10^2$ and 10^3. Bello-Ochende T. and Bejan A., (2005), [8] designed numerically a multi-scale cylinders geometry cooled by free convection by using constructal theory. They maximized the density of heat transfer rate. They put small cylinders in the unused heat transfer area between the large cylinders. They used finite element method to discretize the governing equation in the range of Rayleigh number of $10^5 \leq Ra \leq 10^8$, and Pr= 0.7. They showed that the density of heat transfer increased by putting the small cylinders between the large cylinders. Page L. et al., (2011), [9] investigated numerically the free convection from single scale rotating cylinders. They used the constructal theory to maximize the density of heat transfer rate. The range of Rayleigh number was ($10^1 \leq Ra \leq 10^4$), the range of rotating speed was ($0 \leq \tilde{\omega}_o \leq 10$), and the fluid was air (Pr=0.7). They found that the optimized spacing decreases as Rayleigh number increases and the heat transfer density increases. Page L. et al. (2013), [10] investigated numerically the free convection from multi-scales rotating cylinders. They used constructal theory in order to find the optimal arrangement of the geometry. The range of Rayleigh number was ($10^2 \leq Ra \leq 10^4$), the range of rotating speed was ($0 \leq \tilde{\omega}_o \leq 10$), and the fluid was air (Pr=0.7). Small cylinders were put in the unused regions of heat transfer. They found that there were no effects of the rotating cylinders on heat transfer density in compare with the stationary cylinders except at high speeds of rotation. It is obvious from the literature that there is no attempt to find the optimal spacing between finned tubes cooled by free convection with constructal theory, so that the present study uses the constructal theory to find the optimal spacing numerically.

2. Mathematical Model

Consider a row of finned tubes installed in a fixed volume per unit depth (*h L*) as shown in figure (1). Longitudinal fins are attached to the tubes and the total height of the tube and fin is (h), the diameter of the tubes is half of the total height ($d=h/2$). Three different vertical positions of the tube with respect to the fin (b) are considered as ($b=0.25h$, $0.5h$, and $0.75h$). Fin thickness is negligible in compare with the diameter of the tube. The position of the tube as a ratio is defined as ($\delta = b/h$). The tubes and the fins are maintained at constant wall (hot) temperature of (T_w), and the ambient temperature is maintained at constant temperature of (T_∞). The objective is to find the number of tubes or the tube – to – tube spacing (s) for different tube positions (δ) in order to maximize the density of heat transfer. In this geometry there are two degrees of freedom, the first is the spacing between the tubes (s) and the second is the tube position (δ). The dimensionless governing equations for steady, laminar, two dimensional and incompressible flow with Boussinesq approximation for the density in the buoyancy term can be written as; Zhang Z. et al. (1991), [11]

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \quad (1)$$

$$\left(U\frac{\partial U}{\partial X} + V\frac{\partial U}{\partial Y}\right) = -\frac{\partial P}{\partial X} + \left(\frac{Pr}{Ra}\right)^{1/2}\left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2}\right) \quad (2)$$

$$\left(U\frac{\partial V}{\partial X} + V\frac{\partial V}{\partial Y}\right) = -\frac{\partial P}{\partial X} + \left(\frac{Pr}{Ra}\right)^{1/2}\left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2}\right) + T \quad (3)$$

$$\left(U\frac{\partial T}{\partial X} + V\frac{\partial T}{\partial Y}\right) = \frac{1}{(Ra\, Pr)^{\frac{1}{2}}}\left(\frac{\partial^2 T}{\partial X^2} + \frac{\partial^2 T}{\partial Y^2}\right) \quad (4)$$

The non-dimensional variables and groups used are;

$$X = \frac{x}{h}, \quad Y = \frac{y}{h}, \quad U = \frac{u}{\left(\frac{\propto}{h}\right)(Ra\, Pr)^{\frac{1}{2}}},$$

$$V = \frac{v}{\left(\frac{\propto}{h}\right)(Ra\, Pr)^{\frac{1}{2}}}, \quad P = \frac{Pd^2}{\propto^2 \rho\, Ra\, Pr}, \quad T = \frac{t - T_\infty}{T_w - T_\infty},$$

$$Pr = \frac{\upsilon}{\propto},$$

$$Ra = \frac{g\beta h^3 (T_w - T_\infty)}{\propto \upsilon} \quad (5)$$

Since the flow is symmetrical between the tubes, only half of the flow channel between two tubes can be used to find the spacing in the numerical solution. Half of the flow channel is shown in figure (2). The total height of the channel is (H_u+H+ H_d), the upstream height (H_u) and downstream (H_d) are added to avoid the applying of incorrect velocity and temperature at the inlet and outlet of the channel, these extension (H_u, H_d) are selected according to accuracy tests as shown later.

The flow and thermal dimensionless boundary conditions on the half channel are shown in figure (2) and can be summarized as;

Tube and fin surfaces ($0 \le Y \le H$) (no slip and no penetration and constant wall temperature $U = V = 0, T = 1$)

Channel inlet $\left(0 \le X \le \left(\frac{S+D}{2}\right)\right)$ $\left(U = \frac{\partial V}{\partial Y} = 0, T = 0, P = 0\right)$

Channel exit $\left(0 \le X \le \left(\frac{S+D}{2}\right)\right)$ $\left(\frac{\partial(U,V,T)}{\partial Y} = 0, P = 0\right)$

Left and right sides of the upstream section ($-H_u \le Y \le 0$) (free slip and no penetration $U = \frac{\partial V}{\partial X} = 0, \frac{\partial P}{\partial X} = 0, \frac{\partial T}{\partial X} = 0$)

Left side of the downstream section ($H \le Y \le H + H_d$) (free slip and no penetration $U = \frac{\partial V}{\partial X} = 0, \frac{\partial P}{\partial X} = 0, \frac{\partial T}{\partial X} = 0$)

Right side of the downstream section

($H \le Y \le H + H_d$) (zero stress $\frac{\partial(V,U)}{\partial X} = 0, \frac{\partial P}{\partial X} = 0, \frac{\partial T}{\partial X} = 0$)

The right side of the downstream boundary condition is applied to permit fluid to enter the domain horizontally in order to avoid the vertical acceleration which generated by chimney effects, Bello-Ochende T. and Bejan A.,(2005), [8].

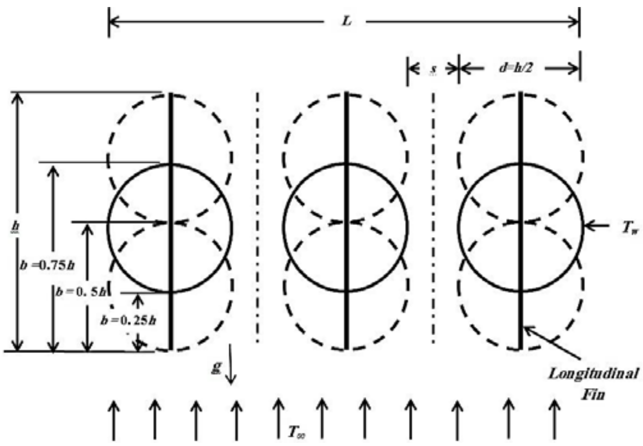

Figure (1): Physical Geometry of the Present Problem

3. Optimization of Heat Transfer (Maximum Heat Transfer Density) Based on Constructal Theory

The spacing between the tubes is to be chosen such that the heat transfer density (objective function) is maximized. The heat transfer density is the heat transfer rate per unit volume and given as;

$$q''' = \frac{q}{V} = \frac{q}{(s+d)hw} = \frac{q'}{(s+d)h} \quad (6)$$

Where q′ = Total heat transfer rate from one tube per unit width.

The heat transfer density can be written in non-dimensional form as;

$$Q = \frac{q' h^2}{k\,(T_w - T_\infty)(s+d)h} \quad (7)$$

$$Q = \frac{-\left(\int_0^h k\frac{\partial t}{\partial x}dy\right)h}{k(T_w-T_\infty)(s+d)} = \frac{-\int_0^1 \frac{\partial T}{\partial X}dY}{(S+0.5)} \quad (8)$$

The objective function (heat transfer density) subjected to the constraint that the total volume per unit width is fixed. (This is based on constructal law)

$$\therefore (h\,L) = \text{Constant} \quad (9)$$

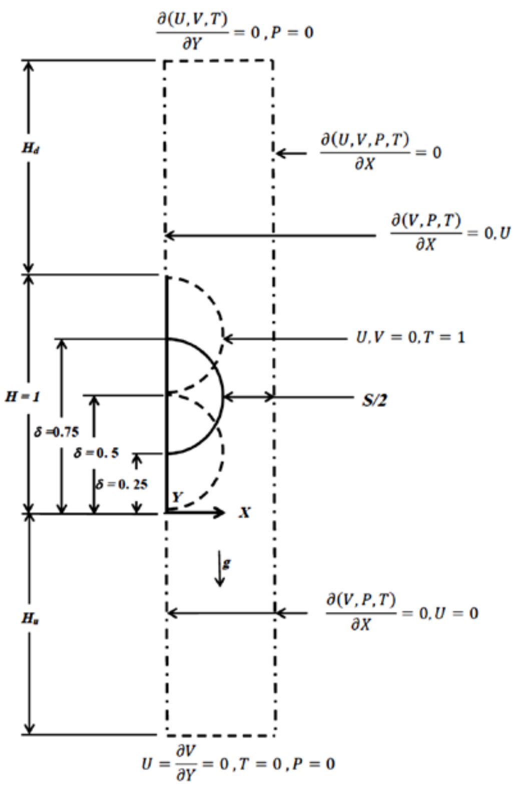

Figure (2): Dimensionless Boundary Conditions on the Flow Channel

4. Numerical Procedure, Grid Independence Test, and Validation

A FORTRAN program is written to solve the algebraic equations which obtained by the finite volume method. The general transport equation is firstly transformed to curvilinear coordinates and the convective term is discretized by hybrid scheme while the diffusion term is discretized by second order central scheme. For coupling between the pressure and velocity SIMPLE algorithm is employed. To prevent the oscillation in the pressure field the interpolation method of Rhie, C. M., and Chow, W. L., (1983), [12], is used. The solution algorithm can be summarized as;

1- Solve the discretized momentum equations to find the velocity field.
2- Solve the pressure correction equation to find the corrected pressure.
3- Correct the velocity field by using the corrected pressure.
4- Solve the discretized energy equation to find the temperature.
5- Repeat the steps (1-4) until convergence attained.
6- Find the heat transfer density from equation 8.

The grid independence test is performed for three grids for configuration at which (Ra = 10^4, δ =0.25, and S = 0.3). The grid independence test showed that the increasing of the grid size decreases the error percentage, and the minimum error occurs at 50×50 control volumes per (H). So this grid size is used and adopted in all the numerical results. Gird independence test is illustrated in table (1). A generated grid for control volumes of (175x50) in the whole domain is illustrated in figure (3). To apply the correct velocity and temperature at the inlet and outlet of the channel, the upstream extension (H_u) is added at the inlet of the channel and downstream extension (H_d) is added at the outlet of the channel. It is observed from the table (2) for (Ra = 10^5, δ =0.75, and S = 0.2) that the increasing in downstream extension to (H_d =3.5) and keeping the upstream at (H_u =0.5) leads to reduce the error in the heat transfer density to 1.4%. Based on this test the value of (H_u=0.5) and (H_d =3.5) have been depended in all numerical results.The numerical results are validated by comparing the results of (S_{opt}) with the numerical results of Da Silva and Bejan,(2004)**,** [6] for natural convection between vertical isothermal plates and with Bello-Ochende and Bejan, (2005), [8] for natural convection between isothermal cylinders. Both comparisons are carried out at (Ra =10^5). Good agreement can be shown in table (3) for both cases.

Table 1 Grid Independence Test for the Case (Ra = 10^4, δ=0.25, and S =0.3)

Number of Control Volumes Per H	Q	Error%
30 x 30	13.925020	---------
40 x40	14.071420	1.04
50 x 50	14.149130	0.554

Table 2 Downstream Extension Test for the Case (Ra = 10^5, δ=0.75, H_u =0.5 and S =0.2)

H_d	Q	Error %
2	23.406150	----------
2.5	23.877630	1.9
3	24.271650	1.6
3.5	24.630340	1.4

Table 3 Comparison of the Numerical Results for (S_{opt}) with the Previous Results for Case Ra =10^5 for flat plate and circular tube.

Flat Plate	
Da Silva A.K., and Bejan A., (2004) [6]	**Present**
0.129	0.13
Circular Tube	
Bello- Ochende T., and Bejan A., (2005) [8]	**Present**
0.104	0.12

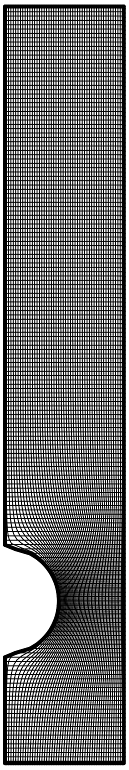

Figure (3) Generated Grid for CVs (175X50)

5. Results and Discussion

The numerical results are presented in this section for, temperature contours, optimal spacing, and density of heat transfer for different values of tube position ($0.25 \leq \delta \leq 0.75$)**.** The range of Rayleigh number is ($10^3 \leq Ra \leq 10^5$) and the working fluid is air with (Pr =0.71).

Figure (4) shows the temperature contour as a function of the spacing between the tubes (S) for ($Ra = 10^3$) and tube position ($\delta = 0.25$). For small spacing ($S < 0.25$) the downstream region is occupied by hot fluid at temperature same as the wall temperature (red region), this is due to that the small spacing between the tubes prevents the cold air to flow downstream and the air there still hot (overworked fluid). As the spacing between the tubes increases ($S > 0.25$) the downstream temperature begins to decrease and become less than the wall temperature and this is clear from the appearance of the (orange, yellow and green) regions. At some spacing the thermal boundary layers from both sides are merged at the downstream region (the channel is fitted with the convective flow body) , at this spacing the heat transfer density becomes maximum and the spacing represents the optimal spacing, in this case (S_{opt} = 0.35). Further increasing in spacing between the tubes leads to a cold fluid region to appear in the downstream as seen in the blue region (underworked fluid) for ($S \geq 1$), this large spacing permits the ambient (cold) fluid to flow downstream and leads to decrease the heat transfer density since the thermal conductance between the tubes decreased.

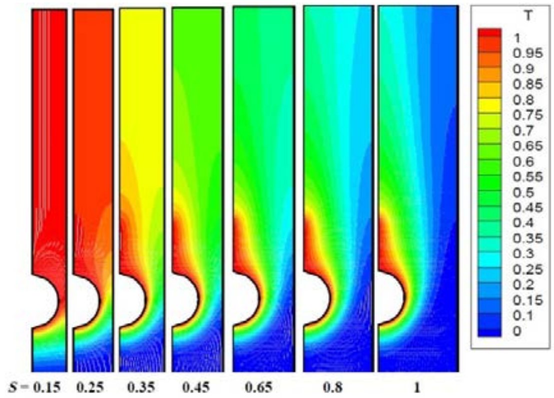

Figure (4) Temperature contour with various spacing between the tube for (Ra=10^3, Pr =0.7, and tube position δ=0.25)

As Rayleigh number increases to ($Ra = 10^5$) same behavior of the temperature contour to that of ($Ra = 10^3$) can be observed in figure (5) except that the optimal spacing here becomes smaller, note that (S_{opt} = 0.35 at Ra=10^3) while (S_{opt} = 0.1 at $Ra = 10^5$), so as Rayleigh number increases the optimal spacing decreases because the thermal boundary layer thickness decreases with increasing of Rayleigh number.

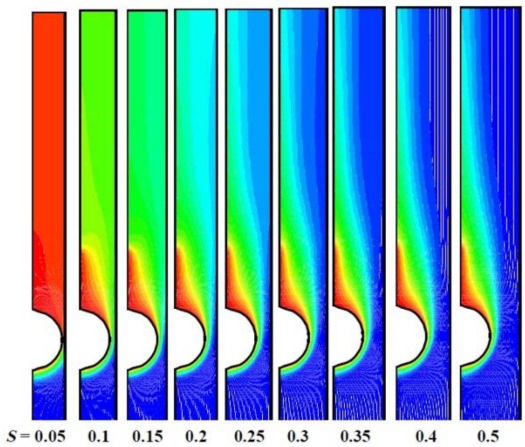

Figure (5) Temperature contour with various spacing between the tube for (Ra=10^5, Pr =0.7, and tube position δ=0.25)

Figures (6, and 7) illustrate the temperature contours at (δ = 0. 5) for Rayleigh numbers (10^3, and 10^5), respectively. In can be seen from both figures that the thermal boundary layer thickness on the lower fin is thinner than the thermal boundary layer on the upper fin due to the presence on the tube in the mid position (δ = 0. 5).

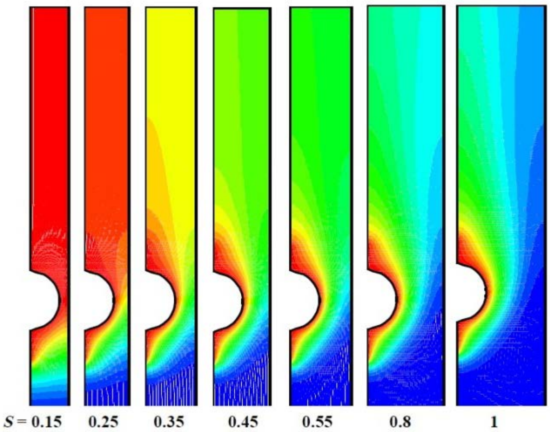

Figure (6) Temperature contour with various spacing between the tube for (Ra=10^3, Pr =0.7,

and tube position δ=0.5)

Figures (8, and 9) illustrate the temperature contours at (δ = 0. 75) for Rayleigh numbers (10^3, and 10^5), respectively. It is interesting to note that as the tube moves from the position (δ= 0.25) to the position (δ= 0.75) the thermal boundary layer thickness on the fin surface becomes thicker as shown in figures (8, and 9) in compare with figures (5, and 6).

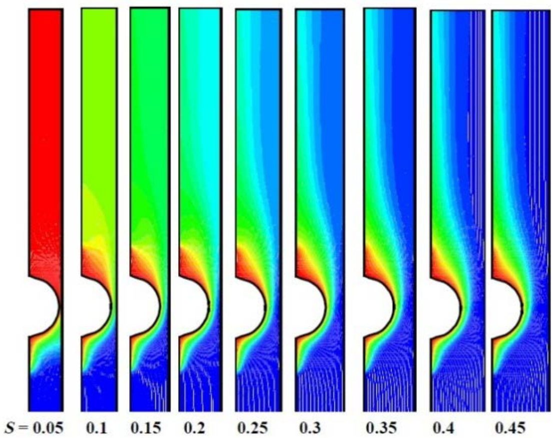

Figure (7) Temperature contour with various spacing between the tube for (Ra=10^5, Pr =0.7, and tube position δ=0.25)

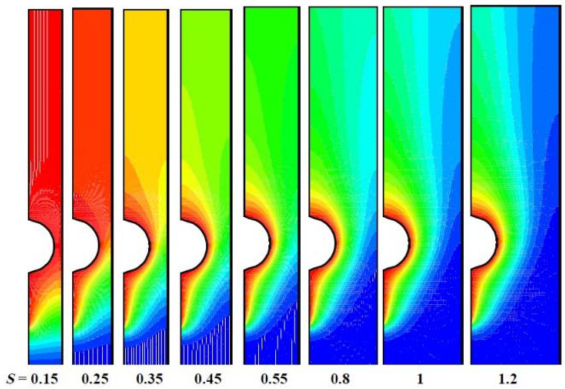

Figure (8) Temperature contour with various spacing between the tube for (Ra=10^3, Pr =0.7, and tube position δ=0.75)

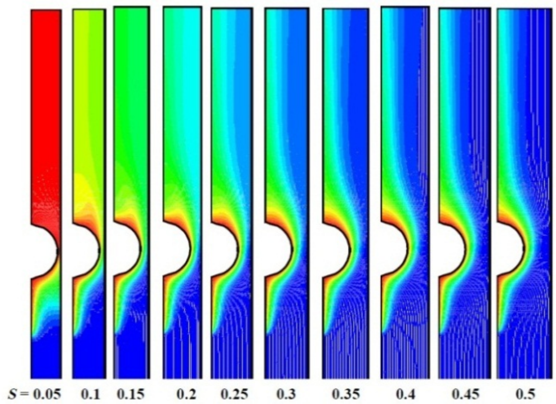

Figure (9) Temperature contour with various spacing between the tube for (Ra=10^5, Pr =0.7, and tube position δ=0.75)

Figures (10 and 11) show the dimensionless heat transfer density as a function of the spacing at different Rayleigh numbers and for tube positions (δ =0.25, and 0.75) respectively. These figures show that there is optimal value of spacing for each Rayleigh number. At this value of spacing the heat transfer density reaches its maximum value (tops of the curves).

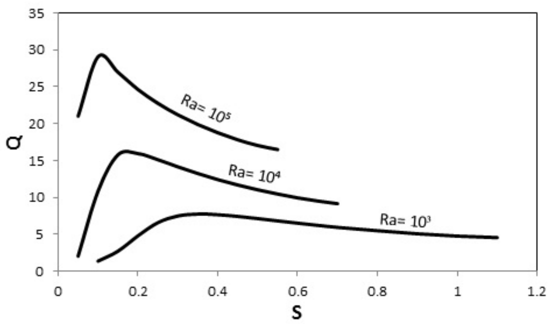

Figure (10) Heat Transfer Density with spacing at different Rayleigh numbers for tube position (δ =0.25)

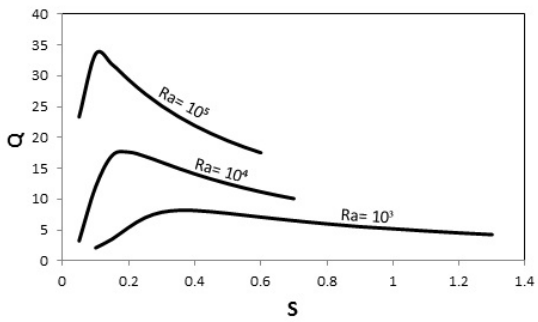

Figure (11) Heat Transfer Density with spacing at different Rayleigh numbers for tube position (δ =0.75)

Figure (12) shows the optimal spacing (S_{opt}) versus Rayleigh number at tube position (δ = 0,25), it is interesting to note that the optimal spacing decreases as Rayleigh number increases, as mentioned above the increasing of Rayleigh number reduces the thermal boundary layer thickness and thus the optimal spacing decreased.

Figure (13) shows the maximum heat transfer density versus Rayleigh number at various tube position (δ), it can be noted that the maximum heat transfer density increases as Rayleigh number increases for all values of (δ), the increasing of Rayleigh number leads to increase the buoyancy force and thus increase the maximum heat transfer density. It also can be seen that at ($Ra=10^5$) the highest value of the maximum heat transfer density occurs at (δ = 0.75) and the lowest value occurs at (δ =0.25). This can be explained as the tube moves upward to (δ = 0.75) the temperature gradient near the lower fin increases and thus the maximum heat transfer density increases.

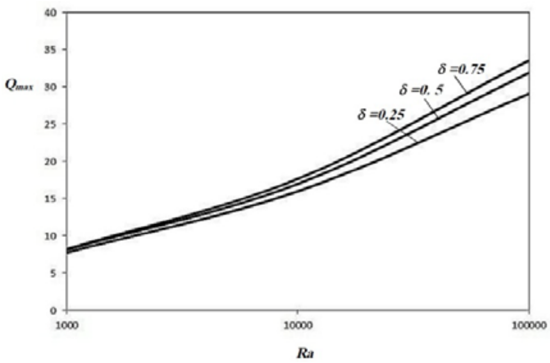

Figure (12) Optimal spacing with Rayleigh number for for tube position (δ =0.25)

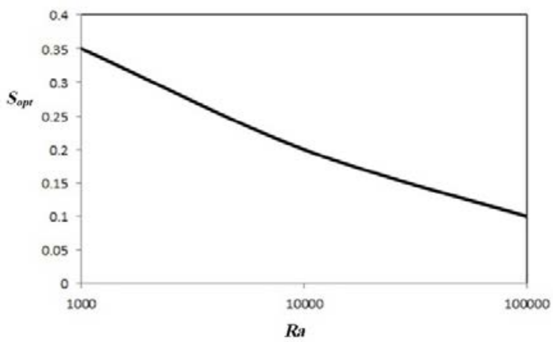

Figure (13) Maximum heat transfer density with Rayleigh number for different tube positions

Figure (14) shows the optimal spacing versus the tube position of the tube at different Rayleigh numbers. The optimal spacing is constant for all values of the tube position. Since the optimal spacing is constant for all (δ), the number of tubes installed in a fixed volume is the same for all tube positions (δ).

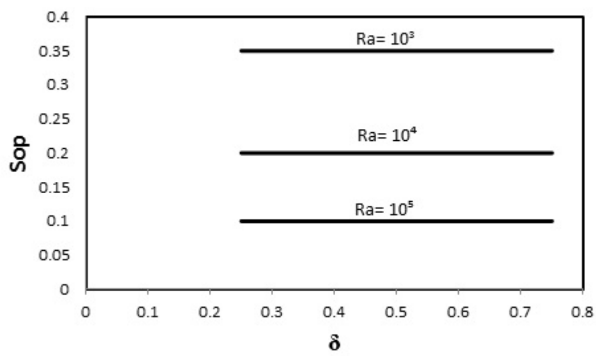

Figure (14) Optimal spacing with different axis ratios for Rayleigh number

6. Conclusions

The conclusions for optimal spacing between finned tubes cooled by free convection can be summarized as:-

1- The optimal spacing decreases as Rayleigh number increases for all tube positions.
2- The maximum heat transfer density increases as Rayleigh number increases for all tube positions.
3- At $Ra=10^5$, the highest value of the maximum heat transfer density occurs at tube position (δ = 0.75) and lowest value occurs at tube position (δ = 0.5).
4- The optimal spacing remains constant as the tube position increases at constant Rayleigh number.
5- The number of finned tubes installed in a fixed volume is the same for all tube positions.

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البعد الامثل بين انابيب مزعنفة مبردة بالحمل الحر باستخدام نظرية التشييد

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الخلاصة

البعد الامثل بين انابيب مزعنفة مبردة بالحمل الحر درس عدديا. صف من الانابيب المزعنفة ثابتة درجة الحرارة نصبت في حجم محدد والبعد بينهم اختير بموجب نظرية التشييد (نظرية بيجان). في هذة النظرية البعد اختير بحيث تكون كثافة انتقال الحرارة اقصى ما يمكن. طريقة الحجم المحدد استخدمت لحل المعادلات الحاكمة, خوارزمية SIMPLE مع شبكة متحدة الموقع استخدمت للربط بين السرعة والضغط. مدى رقم رايلي ($10^3 \le Ra \le 10^5$) , مدى موقع الانبوب($0.25 \le \delta \le 0.75$) ومائع التشغيل هو الهواء ($Pr=0.7$). بينت النتائج ان البعد الامثل يقل مع زيادة رقم رايلي لكل مواقع الانبوب وعند $Ra=10^5$ القيمة العليا لكثافة انتقال الحرارة العظمى تحدث عند موقع الانبوب ($\delta=0.75$) ,وقيمتها السفلى تحدث عند موقع الانبوب ($\delta=0.25$).بينت النتائج ايضا ان البعد الامثل يبقى ثابت مع تغير موقع الانبوب عند رقم رايلي ثابت.