Influence of Tube Arrangement on the Thermal Performance of Indirect Water Bath Heaters

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Abstract

Natural convection heat transfer from a tube bundle in the indirect water bath heaters is investigated. A computer-code is used for the solution of the governing equations of mass, momentum and energy transfer based on the SIMPLE-C algorithm. Simulations are carried out for the gas pressure station heater of Kermanshah city with various tube bundle arrangements. In order to validate the numerical code, results of the simulation compared with experimental data which measured from this heater. Effects of the tube bundle arrangement on heat transfer are presented. It is observed that changing the tube bundle arrangement (horizontal and vertical pitch) can affect the rate of heat transfer. In other word it can lead to increase the thermal performance of the indirect water bath heater. Finally, based on this framework it is suggested that the optimum arrangement of tube bundle can lead to the maximum heat transfer. Hence the performance enhances to 5.27%.

Keywords: Tube arrangement, Natural convection, Indirect water bath heater, City gas station, Heater performance

Introduction

Indirect water bath heaters have been used in the oil and gas production, gas processing and gas distribution industry. The high-pressure natural gas and oil stream heating are the most common applications of the indirect water bath heaters.

As it can be seen in Figure 1, indirect heaters are comprised of three basic components: shell, fire tube and gas tubes. Natural convection heat is transferred from the fire tube to the bath fluid and from the bath to the gas tube bundle.

Natural convection heat transfer from a tube bundle of horizontal cylinders has received much attention for different applications, some of which are heat exchangers in the refrigeration industry, the cooling electronic devices, space heating, etc.

Many heat exchangers commonly use multiple rows of tubes while a cross flow of a fluid is passing through outside. It is studied that turbulent cross flow dramatically increase heat transfer efficiency in comparison with parallel flow for a same fluid velocity. One should note that the velocity distribution of the flow around a tube is not uniform. Depending on application and design criteria, there are many possible tube layouts in a tube bundle.

Due to the wide engineering applications in heat exchangers and passive solar energy collectors, researching on natural convection from multiple cylinders has become a great concern in recent years comparing to the single horizontal cylinders which have been studied for several decades. Morgan [1], Churchill and Chu [2] and many other researchers have found empirical relations that focus on time-averaged Nusselt number.

Collis and William [3] proposed the following correlations in a single tube, for determining Nusselt numbers for a range of Rayleigh numbers between 10^{-10} - 10^{10} in a single tube:

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$$\begin{array}{l|l} \text{Ra} & \text{Nu} \\ \hline 10^{-10}-10^{-2} & 0.675\,\text{Ra}_D^{0.058} \\ 10^{-2}-10^{2} & 1.02\,\text{Ra}_D^{0.148} \\ 10^{2}-10^{4} & 0.85\,\text{Ra}_D^{0.188} \\ 10^{4}-10^{7} & 0.48\text{Ra}_D^{0.25} \\ 10^{7}-10^{10} & 0.125\text{Ra}_D^{0.333} \end{array} \quad (1)$$

Free convection heat transfer from an array of two vertically separated cylinders, was studied by Sparrow and Neithammer [4]. They found a direct relation between average Nusselt and the space distance between two cylinders for a range of Rayleigh number from 2×10^4 to 2×10^5. They also showed that the maximum heat transfer occurs when the distance is between 7 to 9 diameter apart while wall temperature is assumed to be uniform. Tokura [5] worked on vertical arrays of 2, 3 and 5 horizontal cylinders at a uniform temperature and within a range of Rayleigh numbers from 2.8×10^4 to 2.8×10^5. Investigating heat transfer profile around a cylinder showed the space effect on their free convection heat transfer. These studies showed that the spacing between the cylinders has direct relation with the heat transfer at the bottom of upper cylinder.

Based on some experimental studies on natural convection of two horizontal cylinders in air Chouikh [6], referred to a fact that at higher Rayleigh numbers (10^4 compared to 10^2), the most of the fluid flow through the sides rather than the bottom. Analysis of the effect of spacing and Rayleigh number on heat transfer by Lieberman and Gebhart [7] and Marsters [8] showed a different behavior of upper and lower cylinder while Yousefi and Ashjaee [9] studied natural convection from a vertically aligned array of horizontal elliptic cylinders. Marsters [8], Ashjaee and Yousefi [10] have also shown that spacing just influence on the heat transfer of upper cylinder while the lower ones act as single cylinders. Accordingly, Marsters [8] describes that the amount of heat transfer of the upper cylinder could be significantly less than that of the lower one for small spacing, while for large spacing the heat transfer could approximately increase by 30%. Lieberman and Gebhart [7] related this difference of heat transfer to lower cylinder plume rising by explaining two simultaneous effects with opposite results. The plume temperature is higher than that of the bulk fluid and this leads to the reduction of local temperature difference and consequently the surface heat transfer from the upper cylinder. On the other hand, the plume velocity increases heat transfer of the upper cylinder by imposing a forced convection flow condition.

In this study, the influence of tube bundle arrangement on the global Nusselt number, heat transfer and thermal performance were investigated. The numerical solutions were performed for laminar flow that is performed by natural convention from fire tubes, placed in the lower part of heaters. Also the numerical code has been developed to calculate two-dimensional buoyancy-driven convection in differential arrangements of tube bundle.

As a first step and in order to validate the numerical code, heat transfer and Nusselt number are calculated for different rows of tube bundle arrangement for 150,000 SCMH indirect water bath gas heaters which is located in the gas pressure station of Kermanshah city. The calculated results are compared with those of the experimental that are measured from this heater and at international standard.

In the second step, the amount of heat transfer of tube bundle for different arrangements were obtained and finally offered the optimum arrangement of tube bundle that can lead to a maximum heat transfer.

2. Theoretical approach

2.1. Mathematical model

The flow of water through the tube bundle is simulated mathematically. For this purpose, a computer-code is used for the solution of the differential equations related to the conservation of mass, momentum and energy.

The viscous dissipation term in the energy equation is neglected. The Boussinesq approximation is applied for the properties of fluid to relate density to temperature, and hence to couple the fields of temperature to that of the flow.

$$\nabla.V=0 \tag{2}$$

$$(\nabla.V)V = -\nabla p + \nabla^2 V - \frac{Ra}{\Pr} T \frac{g'}{g} = 0 \tag{3}$$

$$(\nabla.V)T = \frac{1}{\Pr} \nabla^2 T \tag{4}$$

Where Ra is Rayleigh number that is defined as $g\beta(T_W-T_\infty)D3/\upsilon\alpha$ and is based on the cylinder-diameter; V is the velocity vector with dimensionless velocity components U and V and is normalized with υ/D; T is the dimensionless temperature that is normalized with temperature difference (T_W-T_∞); P is the dimensionless pressure which is normalized with $\rho_\infty\upsilon^2/d^2$; g is the gravity vector and $Pr=\upsilon/\alpha$ is the Prandtl number. Also ρ, β, υ, α and g are the fluid density, the volumetric expansion coefficient, the kinematic viscosity, the thermal diffusivity and the gravitational acceleration, respectively. The boundary conditions for the present study are later discussed.

The temperature of tube bundle surface T_w is assumed to be constant, at 24˚C (that is the average value between outlet (38˚C) and inlet gas temperature (10˚C) according to indirect water bath gas heater 150,000 SCMH in the gas pressure station of Kermanshah city).

The dimensions of the heater are shown in Figure 1. The shell is insulated and the temperature of the heater is maintained below the boiling point of water. The average temperature of water bath T_∞ is 60˚C that is obtained from the gas pressure station of Kermanshah city. The temperature of the fire tubes is kept constant and the temperature of their surfaces is calculated by trial and error method. For this purpose, the temperature of these tubes is first guessed, and then the temperature of water around them is calculated (see Table 1). According to the experimental values, the average of water temperature should be around of 60˚C (this value elicited from present heater), so as much as the temperature of water bath is closer to this value, then the assumed temperature will be closer to the right value of the fire tubes. Finally based on the performed calculation (table 1), the surface temperature of the fire pipes was obtained to be 127˚C (400 K).

Table 1: Fire tube temperature computed by trial and error method

Assumed fire tube temperature (˚C)	Average temperature of water bath (˚C)
75	40
100	55
127	61
227	92
327	100
427	147

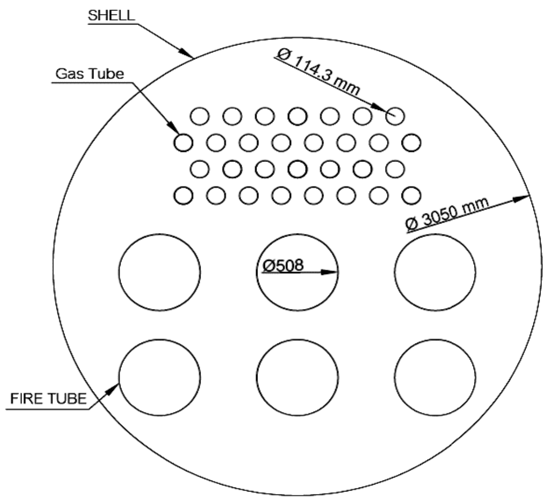

Figure 1: Schematic diagram of indirect water bath heater

2.2. Solution procedure

The governing equations (2)–(4) is solved with the boundary conditions, through a control-volume formulation of the finite-difference method. Van Doormaal and Raithby[11] handled the pressure-velocity coupling by using the SIMPLE-C algorithm, which is essentially a further implicit variant of the SIMPLE algorithm by Patankar and Spalding[12]. Leonard [13] evaluated the convective fluxes across the surfaces by using the QUICK discretization scheme. Details on the SIMPLE procedure may be found in Patankar[14].

Both Polar and Cartesian grid regions discretized by using fine uniform mesh spacing. Starting from specified first approximation distributions of the dependent variables across the integration domain; iterative solution through a line-by-line application of Thomas algorithm for governing equations is done, while under-relaxation is used to ensure the convergence of the iterative procedure. The solution is assumed to be fully converged when the maximum absolute values of both mass source and percent variations of the dependent variables at any grid-node; are smaller than prescribed values, 10^{-3} and 10^{-5}, respectively.

By achieving convergence, the average Nusselt numbers Nu_{ij} of any ij_{th} cylinder in the tube bundle are calculated (i and j represent row and column number):

$$Nu_{ij} = \frac{Q_{ij}}{\pi k (T_W - T_\infty)} \tag{5}$$

Where Q_{ij} is the heat flux.

Tests on the dependence of the results acquired on the mesh-spacing of the Cartesian grids, as well as on the thickness and mesh-spacing of the polar grid regions around the gas tube and fire tube, and on the extension of the entire computational field, have been carried out for a wide range of geometrical configurations and Nusselt numbers. In particular, the optimal values of grid-size and the positions of the interface between polar and Cartesian grids; represent a good compromise between time required and computational accuracy for solution process. They are assumed as those over which more refinements or displacements do not create considerable modifications in both flow field and heat transfer rates.

In order to validate the numerical code and the composite-grid discretization scheme specifically developed for the present study; the average Nusselt numbers obtained for a single cylinder at Rayleigh numbers 6.19×10^9 have been compared with the Eq (1). The computed values of Ra, using thermal conditions of the present heater were mentioned in the boundary conditions. The numerical result of the average Nusselt number is 232 and the corresponding calculated value is 229.5.

$$(Nu-Nu_{eq.1})\times100/Nu_{eq.1}=1.08\%$$

In particular, it may be seen that the local results are well within 1% of the benchmark data.

3. Results and discussion

3.1. Heater with current arrangement tube bundle

The general dimension of 150000 SCMH heaters is shown in Figure 1. Gas tube arrangement of this heater is in such a way that $S_p/d=1.5$, $S_n/d=1.75$ (S_p and S_n are shown in Figure 2) and the length of each tube is 9 meter. The results obtained for this step is mentioned in table 2.

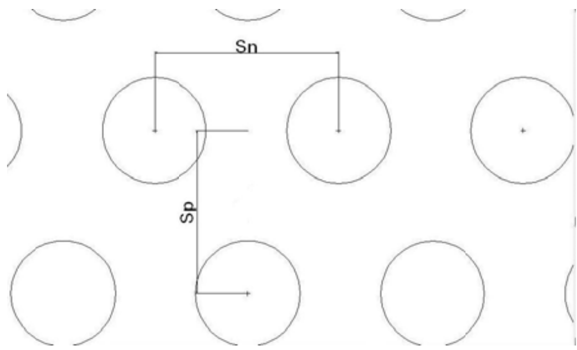

Figure 2: Schematic diagram of Sp and Sn

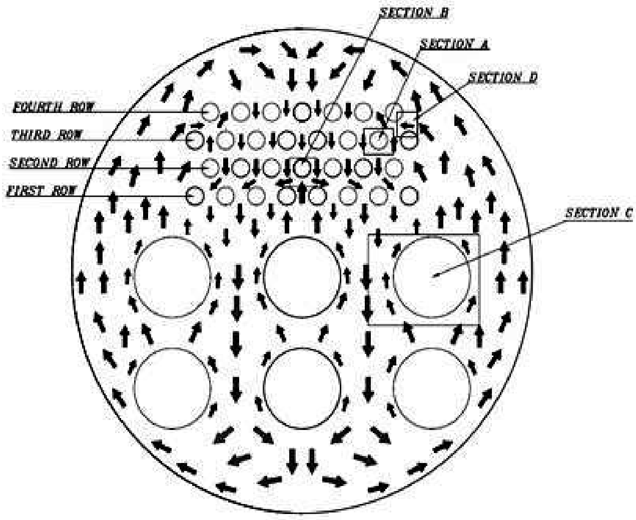

Figure 3: Layout of velocity vector for current heater.

Table 2: Results of modeling the indirect heater with present tube bundle arrangement.

Nu for first row	328
Nu for 2th row	346
Nu for 3th row	315
Nu for 4th row	320
Heat flux (W/m^2)	30,076
Average water bath temperature (K)	333.5

Figure 3 shows the layout of velocity vectors for this heater and indicates how natural convection causes water to flow through the heater. The generated water plume flowing at the bottom of fire tube, try to flow through the upper part of the shell. The water flow impinges to the tube bundle and a major portion flows toward the surrounding of the shell and goes down with colder water to the center of shell. Also some portions of water go ahead inside the tube bundle.

Section A, B, C and D (Figure 4 to 7) give more useful information about flow vectors around gas tubes in the different rows. These figures indicate that the directions of flow around each tube are much different than the other tubes. For example, section A (Figure 4) shows that the flow direction at the right side of the section is opposite of the other side. According to Table 1, maximum heat transfer occurs in the second row of the tube bundle, this can be interpreted by considering the section B (Figure 5).

Natural convection plume from the fire tube, contact directly to these tubes and also the plume temperature is higher than the average of fluid temperature in the other rows.

The flow going down the upper cylinder, contact the plume that is rising form the fire tubes and make vortex flow near this row, and hence increases the heat transfer.

The overall heat flux from all tube bundle equals to 30,076 (W/m^2) which is a normal value for this type of heaters with given API SPECIFICATION 12K [15] (for Indirect Type Oilfield Heaters).

Furthermore, basing on current dimensions of the heater, (9m length, 30 gas tubes, 114.3mm diameter) the total surface area becomes 96.9 m^2. Therefore, the amount of energy received by gas is equal to 2.914 MW. This energy leads to 30 degree increase in the gas temperature for 150,000 SCMH heaters. Thus, when the entrance temperature of gas equals to 10 degree centigrade, the outflow temperature will be 40 degree centigrade which is approximately equal to the outflow temperature mentioned at boundary conditions.

These calculations demonstrate validity of numerical solution too.

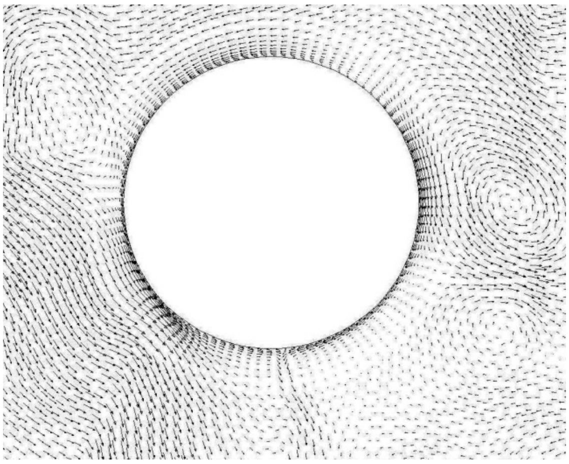

Figure 4: Velocity vector diagram for section A

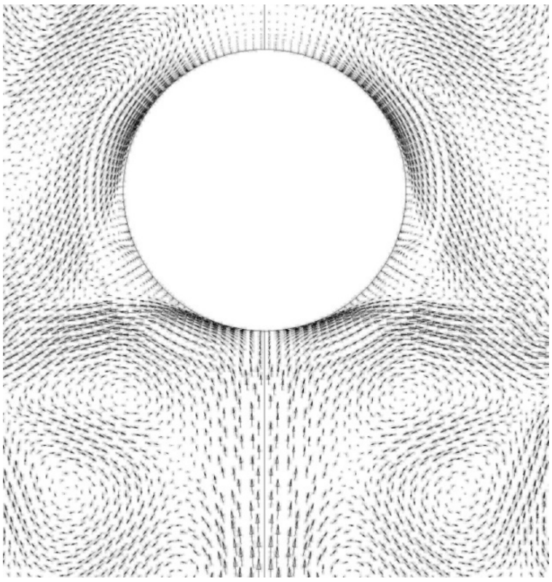

Figure 5: Velocity vector diagram for section B

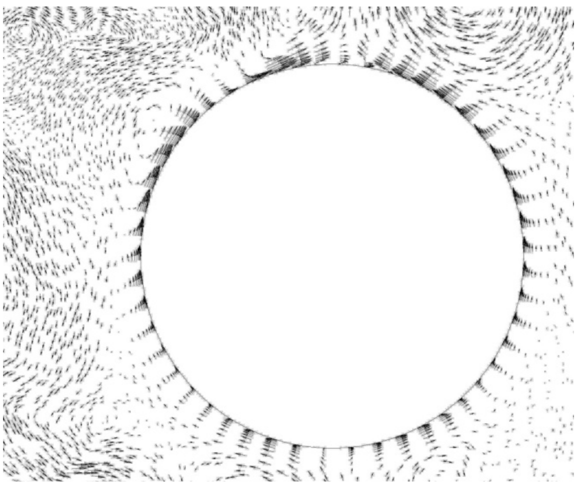

Figure 6: Velocity vector diagram for section C

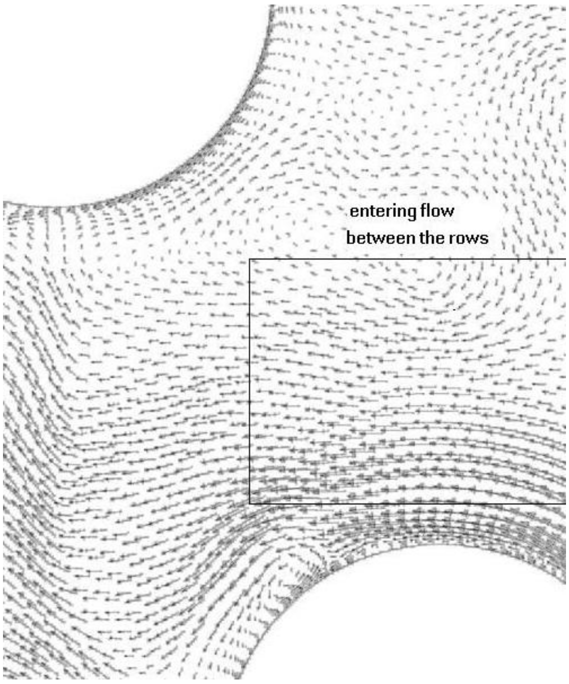

Figure 7: Velocity vector diagram for section D

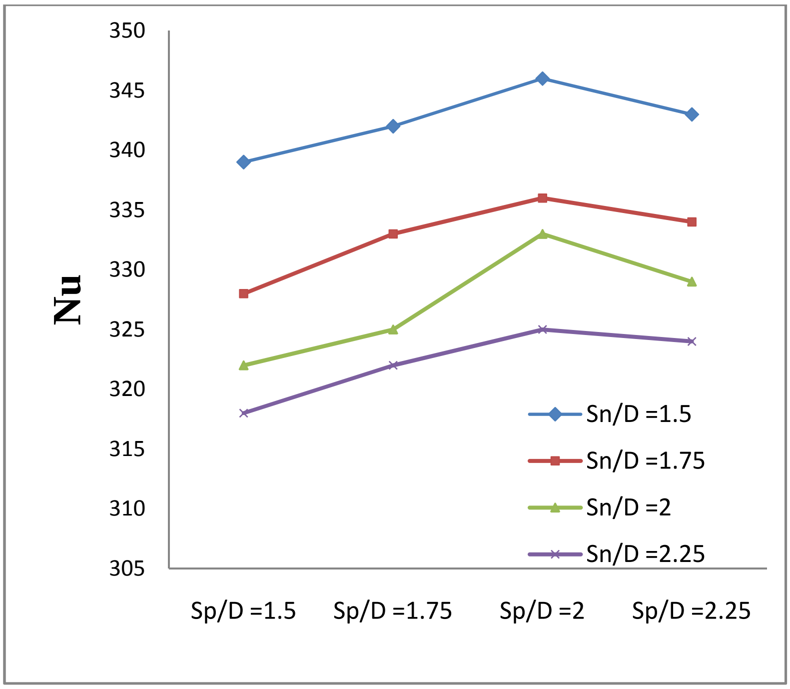

Figure 8: Nusselt number of first row

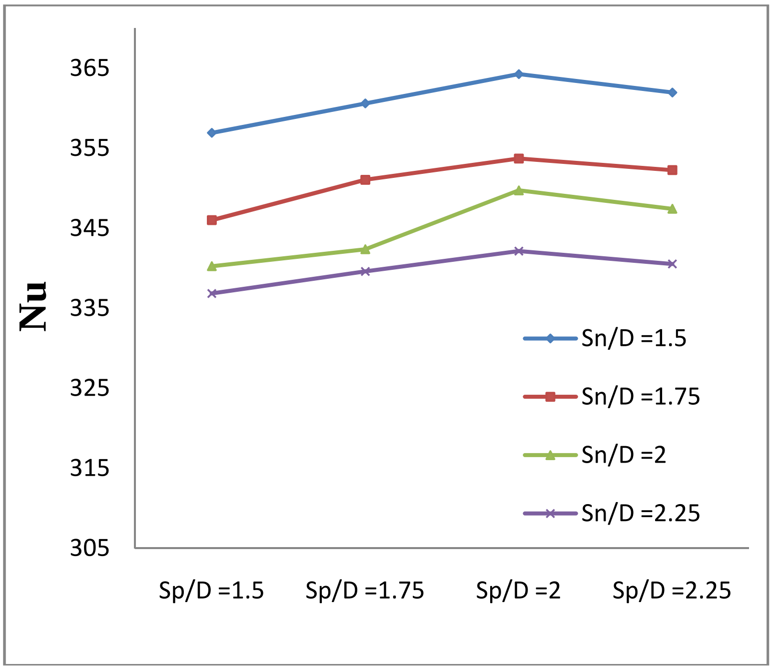

Figure 9: Nusselt number of second row

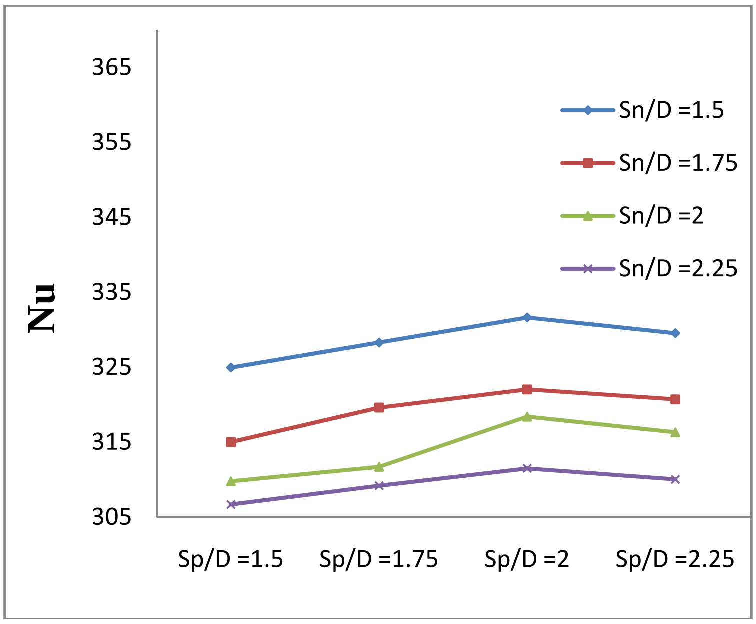

Figure 10: Nusselt number of third row

3.2. Heater 150,000 SCMH with different tube bundle arrangements

The numerical results obtained for the effects of tube bundle arrangement on the Nusselt number for different rows of this heater with average magnitude of heat transfer are expressed in table 3-8 and figure 8-13. These show the perpendicular pitch (S_n/D) and longitudinal pitch (S_p/D) to have great effects on the heat transfer. Results are summarized by as follows:

It is clear from figures 12 and 13 that, an increase in S_n/D leads to a decline in the Nusselt number and heat transfer. That is due to the nozzle effect between gas tubes. Increasing S_n leads to a minimum flow area increase. Therefore, the velocity of impingement flow on the upstream row of tubes drops and finally contributes a decrease in heat transfer. Similar researches done by Wilson and Khalil [16] also confirm these results.

Table 3: Nusselt number of first row

S_n/D	S_p/D			
	1.5	1.75	2	2.25
1.5	339	342	346	343
1.75	328	333	336	334
2	322	325	333	329
2.25	318	322	325	324

Table 4: Nusselt number of second row

S_n/D	S_p/D			
	1.5	1.75	2	2.25
1.5	357	360	363	361
1.75	346	352	355	353
2	340	343	351	346
2.25	336	341	343	341

It could clearly be seen that when $S_P/D \leq 2$, any increase in S_p leads to enhance the Nusselt numbers and heat transfer. This is because, an increase of S_P/D gives the chance to the flow behind the last row, to approach the reattachment point and enhance heat transfer coefficient. On the other hand, any climb in S_p/D leads to increase in the rate of entering flow through the rows (shown in section D Figure 7). Also, these flows are hotter than those inside the spacing of the tube bundle. This phenomenon causes an increase in the average of heat transfer. Further increase in S_p/D (exceeding 2), leads to decline in Nu. This is because of the upstream row of the tubes, results from fom the flow separation of the last row.

The comparison between the contour plots in Figures 14 and 15, shows the average fluid temperature surrounding the tube bundle in Figure 14 (with $S_n/D=1.5$ and $S_P/D=2$) is higher than the fluid temperature in Figure 15 (with $S_n/D=1.75$ and $S_P/D=1.5$). In addition, Figure 14 shows offering arrangement ($S_n/D=1.5$ and $S_P/D=2$) for tube bundle. The offering arrangement leads to improve the circuit of fluid and consequently contributes an increase in the heat transfer.

The performance of these heaters calculated by the following equation:

$$\eta = Q_{out}/Q_{in} \quad (6)$$

Where Q_{out} is the amount of energy that is absorbed by tube bundle and Q_{in} is the energy that is partially entering into the heater and the remaining is absorbed by tube bundle.

According to figure 13, the maximum heat transfer occurs at $S_n/D=1.5$ and $S_P/D=2$, using the equation (6) with consistent Q_{in}. the increase of the heater performance equals to:

$(Q_{out2}-Q_{out1})\times100/Q_{out1}=5.27$ %

Where Q_{out2} equals to the amount of heat transfer by the optimum arrangement of the tube bundle ($S_n/D=1.5$ and $S_P/D=2$) and Q_{out1} is the heat transfer by the present arrangement of the tube bundle.

As it can be seen, we can improve the performance of theses heaters to 5.27% by using the optimum arrangement which is offered in this study.

Table 5: Nusselt number of third row

S_n/D	S_p/D			
	1.5	1.75	2	2.25
1.5	326	328	333	330
1.75	315	320	323	321
2	309	312	319	315
2.25	306	309	312	310

Table 6: Nusselt number of fourth row

S_n/D	S_p/D			
	1.5	1.75	2	2.25
1.5	331	334	340	336
1.75	320	325	328	326
2	314	318	323	320
2.25	311	314	317	315

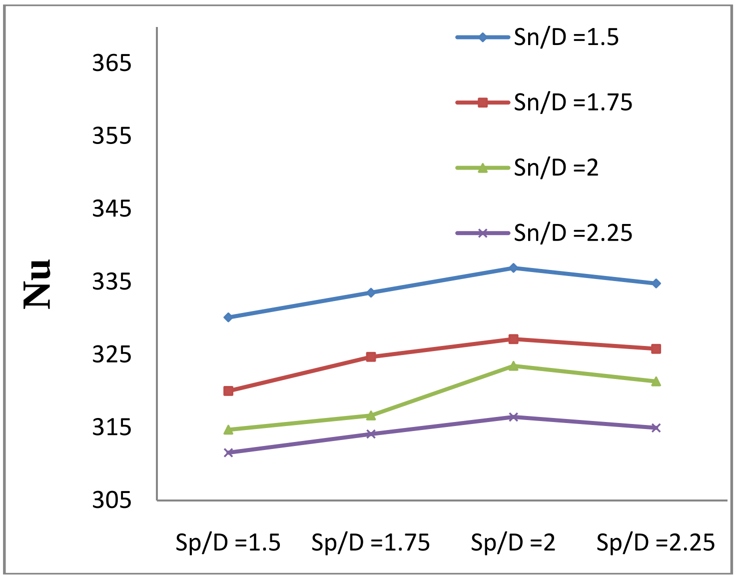

Figure 11: Nusselt number of fourth row

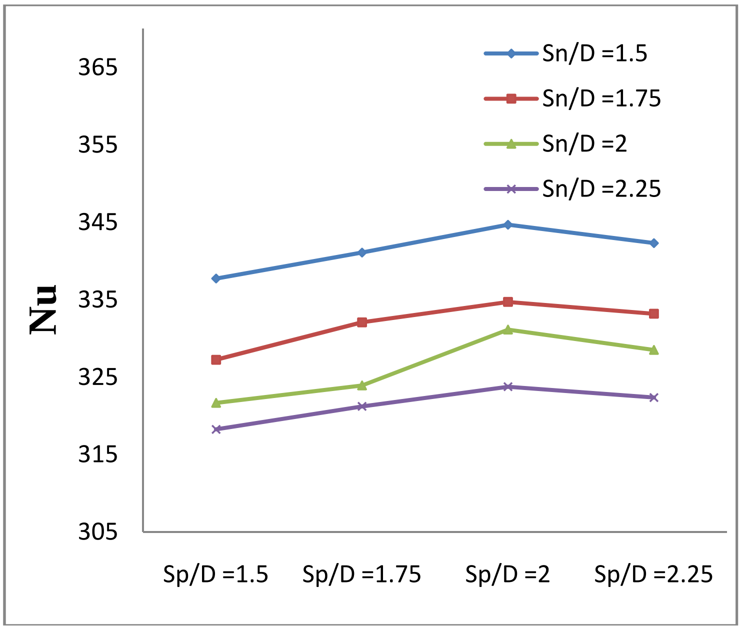

Figure 12: Average Nusselt number for all tubes in the bundle for different arrangement

Table 7: Average Nusselt number for all tubes in the bundle for different arrangements

S_n/D	S_p/D			
	1.5	1.75	2	2.25
1.5	338	341	345	342
1.75	327	332	335	333
2	322	324	331	329
2.25	319	321	324	322

Table 8: The amount of heat flux (W/m^2) obtained for different arrangements

S_n/D	S_p/D			
	1.5	1.75	2	2.25
1.5	31,025	31,344	31,663	31,463
1.75	30,076	30,515	30,746	30,620
2	29,576	29,760	30,400	30,200
2.25	29,280	29,520	29,740	29,600

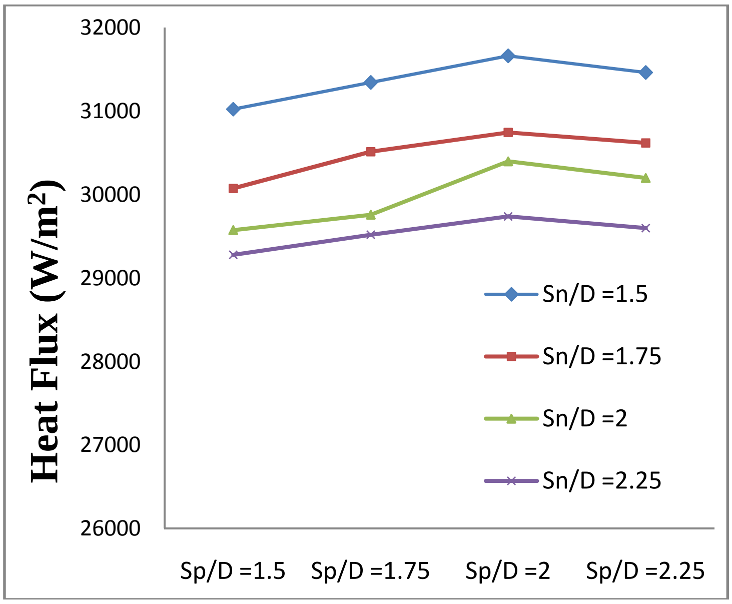

Figure 13: Calculated heat flux (W/m^2) for different arrangements

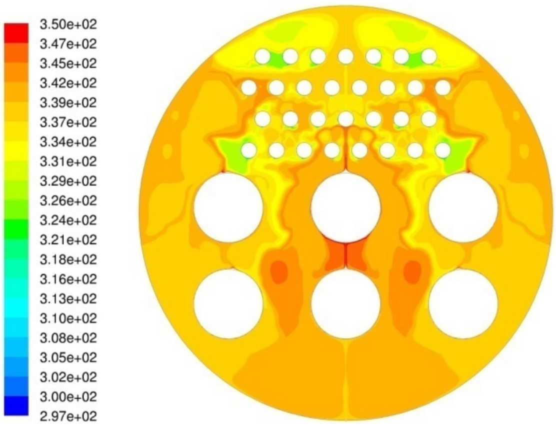

Figure 14: Contour plots of the fluid temperature for optimum arrangement (S_n/D=1.5 and S_P/D=2)

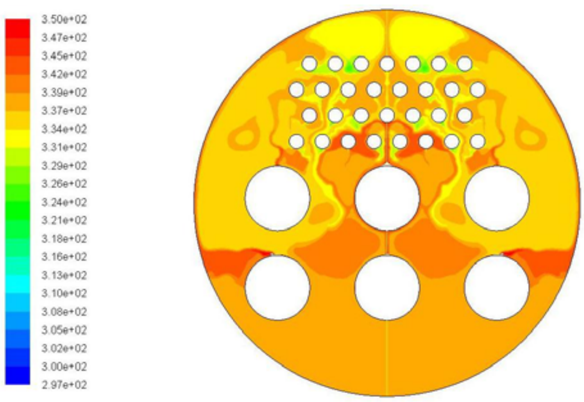

Figure 15: Contour plots of the fluid temperature for present arrangement (S_n/D=1.75 and S_P/D=1. 5)

4. Conclusion

Steady laminar natural convection from fire tube to tube bundle in an indirect water bath heater has been studied numerically through a developed computer-code based on the SIMPLE-C algorithm. Simulations have been performed for 14 different arrangements of tube bundle. The results were compared with current heater in the gas pressure station at Kermanshah city.

It was observed that by using the optimum tube bundle arrangement ($S_n/D=1.5$ and $S_P/D=2$), the average heat transfer of is increased and consequently improve the performance by 5.27%.

Nomenclature

D	Diameter of gas tube [m]
g	Acceleration due to gravity, [m/s^2]
g'	Gravity vector
h	Convective heat transfer coefficient, [W/m^{2}K]
k	Thermal conductivity, [W/mK]
Nu	Nusselt number, [-]
Nu_{ij}	Tube Nusselt number for the i row and j column, [-]
p	Dimensionless pressure
Pr	Prandtl number, [-]
Q	Heat transfer rate [W/s.m^2]
r	Dimensionless radial coordinate
Ra	Rayleigh number, [-]
S_P	Longitudinal pitch [m]
S_n	Perpendicular pitch [m]
SCMH	Standard Cubic Meters per Hour [m^3/h]
T	Temperature [K]
Greek symbols	
α	Thermal diffusivity of the fluid [m^2/s]
β	Temperature coefficient of volume expansion, [K^{-1}]
υ	Kinematic viscosity of the fluid [m^2 /s]
θ	Dimensionless polar coordinate [-]
ρ	Density, [kg/m^3]
η	Performance
Subscripts	
in	Inlet
out	Outlet
w	Referred to the tube surface
∞	Referred to the undisturbed fluid

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