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A NUMERICAL PROCEDURE FOR TUBE COUNT DETERMINATION IN TUBULAR HEAT EXCHANGERS

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Abstract. *The development of a computational procedure that allows the precise determination of important thermal and mechanical design parameters of tubular heat exchangers, such as the tube count and the tube bundle outside diameter, is discussed in the present work. Equipments for industrial applications are usually designed using empirical expressions and data tables for determination of the tube bundle parameters. The importance of the proposed procedure resides in addressing cases for which empirical expressions are inapplicable or data table are unavailable. Initially, the shell positions in which tubes can be placed are determined based on a specified tube pitch, angle of the arrangement, inlet and outlet nozzle diameters and tube bundle-to-shell clearance. The maximum number of tubes for a given configuration is obtained from the previously described searching procedure. A sorting algorithm, based on the tube distance to the shell center, is used to appropriately place a specified number of tubes within the heat exchanger cross section. Results for single-pass fixed-tubesheet heat exchangers are presented and compared with tube count tables available in the literature.*

Keywords: *Heat Exchangers, Industrial Equipments, Mechanical Design, Pressure Vessels*

1. INTRODUCTION

Shell-and-tube heat exchangers are process equipment within which energy is transfered between two convective fluid streams at different temperatures and separated by a solid wall. The wide variety of practical interest applications, ranging from steam production and condensation in power plants to process heating and cooling of products in chemical industries, serve as motivation for the design optimization of these equipments (Fraas, 1989).

Heat exchangers are usually designed having a desired heat load and an allowable pressure drop across the device as the primary constraints (Kern, 1965). Therefore, a thermal and hydrodynamical analysis of the equipment becomes an important part of the design effort. Shell-and-tube heat exchangers are usually built from circular tubes, mounted in a predefined arrangement

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forming the tube bundle, and an involving cylindrical shell. Tube and shell axes are usually parallel. The knowledge of the geometrical characteristic parameters of the tube bundles, such as the tube count, which corresponds to the maximum number of tubes that can be accommodated within a specified shell, are important design data usually obtained from empirical correlations or tables (Saunders, 1988).

Applications of practical interest where tables of the tube bundle geometrical parameters are not available or that fall outside the applicability range of the empirical correlations lead to the need of hand or computer drafting the tube arrangement within the shell. Due to the increasing need for compact heat exchangers, situations where traditional design tools are not available are becoming common, leading to the need of new Engineering design tools. Besides, it is usually not possible to provide exact tube count tables because of the large number of depending variables. Therefore, tube count tables found in the literature are usually applied only during early design stages (Saunders, 1988).

A computational procedure for the determination of tube bundle geometric parameters is presented in the present work and applied to single tube pass tubular heat exchangers. Initially, a searching procedure is used, allowing the determination of the possible tube center positions. Through the validation of the possible tube positions, the tube count for the given configuration is determined. A sorting procedure is used to appropriately place the tubes according to the tube distance to the shell center. The proposed computational procedure accounts for the effects of inlet and outlet nozzle diameters and relative nozzle position on the tube bundle distribution and for the presence of impingement plates. Tube bundle layout drawing is also provided for both the maximum and an user defined number of points. The parameters calculated by the proposed procedures can be used for thermal and mechanical design of heat exchangers and in performance analysis of operating equipment.

2. TUBE POSITION SEARCH PROCEDURE

A sketch of the tube searching procedure is depicted in Fig. 1, which also shows principal dimensions and the Cartesian system of coordinates used in the present work.

Possible tube center positions are defined by the intersection of two straight lines which are used to scan the heat exchanger cross section by varying the straight line linear coefficients. Writing the straight line equations as

$$y = ax + b; \qquad a > 0, \qquad l_{max} > b > l_{min} \tag{1}$$

$$y = cx + d; \qquad c < 0, \qquad l_{max} > d > l_{min} \tag{2}$$

a possible tube position of order k is represented by

$$x_k = \frac{d_i - b_j}{a - c}; \tag{3}$$

$$y_k = a\left(\frac{d_i - b_j}{a - c}\right) + b_j \tag{4}$$

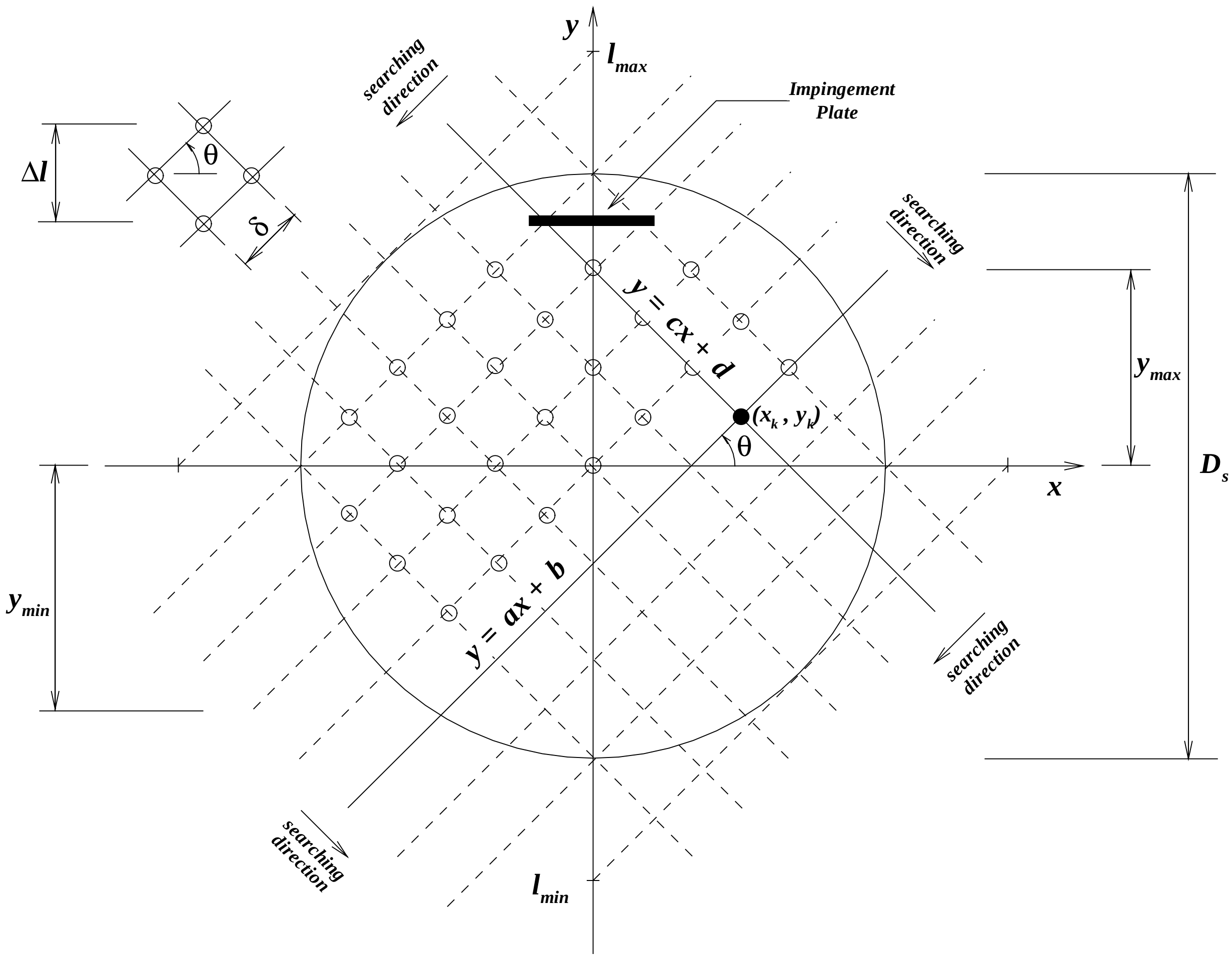


Figure 1: Sketch of the tube searching algorithm showing current (solid) and possible (dashed) searching line positions, previously (○) and lastly (●) found tube positions and principle dimensions.

The tube-center to shell-center distance, which is used in the searching procedure, is defined by

$$\Delta_k = \sqrt{x_k^2 + y_k^2} \tag{5}$$

In order to cover the heat exchanger cross section, the limiting values of the straight line linear coefficients are defined as

$$l_{max} = \frac{D_s}{2 \cdot cos(\theta)} \tag{6}$$

and

$$l_{min} = -\frac{D_s}{2 \cdot cos(\theta)} \tag{7}$$

where D_s is the heat exchanger shell inside diameter. For the tube searching procedure, the linear coefficient increment, which corresponds to two times the distance between tube rows, is written as

$$\Delta l = 2 \cdot \delta \cdot sin(\theta) \tag{8}$$

where δ is the tube pitch. The angular coefficients of the searching lines a and c are defined as

$$a = tan(\theta) \tag{9}$$

and

$$b = -tan(\theta) \tag{10}$$

It is worthy mentioning that the angle θ is the complementary of the usually defined angle of the arrangement α.

Once found by the geometrical procedure described above, a possible tube position must be validated to guarantee that it can be used by a tube. At this stage of the searching algorithm, different criteria can be introduced, based on the heat exchanger configurations being addressed. In the present work, two restrictions will be imposed on the initially found tube position. A radial condition for a valid tube position can be written as

$$\Delta_k + \frac{D_t}{2} \leq \frac{D_s}{2} - \epsilon_r \tag{11}$$

where D_t is the tube outside diameter and ϵ_r corresponds to the radial clearance between tube bundle and the inner shell wall. The radial clearance may be defined by fouling constrains or the presence of sealing strips on the shell side.

Tube vibration and erosion also impose conditions on the tube placement. A reduction on the number of tube rows that effectively can be defined for a heat exchanger configuration can be expected due to inlet and outlet nozzles and the use of impingement plates. The use of externally fitted impingement plates and distributors avoid the effective row number reduction, but those devices are not considered in the present work. The plate position is chosen in order to guarantee an annular escape area at least equal to the nozzle cross-section. Therefore, the position of the last row of tubes (y_{max}) is given as

$$y_{max} = \frac{D_s}{2} - \frac{\phi_{in}}{4} - t_p - \epsilon_{tp} - \frac{D_t}{2} \tag{12}$$

where t_p is the impingement plate thickness and ϵ_{tp} is the clearance between the impingement plate and the tube walls. The impingement plates are about $6mm$ thick with dimensions slightly wider than the inlet nozzle internal diameter (ϕ_{in}). A fluid escape area must also be provided at the outlet nozzle and the position of the first row of tubes (y_{min}) is given as

$$y_{min} = -\frac{D_s}{2} + \frac{\phi_{out}}{4} + \frac{D_t}{2} \tag{13}$$

where ϕ_{out} is the outlet nozzle internal diameter, since impingement plates are usually not necessary at the shell outlet. A similar expression to Eq. 13 is applied for y_{max} if the use of an impingement plate is unnecessary at the inlet nozzle. Therefore, a possible tube position will be validated if

$$y_{min} \leq y_k \leq y_{max} \tag{14}$$

Once finished, the searching procedure allows de determination of the heat exchanger tube count, tube bundle layout and the maximum outer tube limit for the given configuration. Nevertheless, further calculations are necessary if a given number of tubes is specified and the other tube bundle parameters are to be determined.

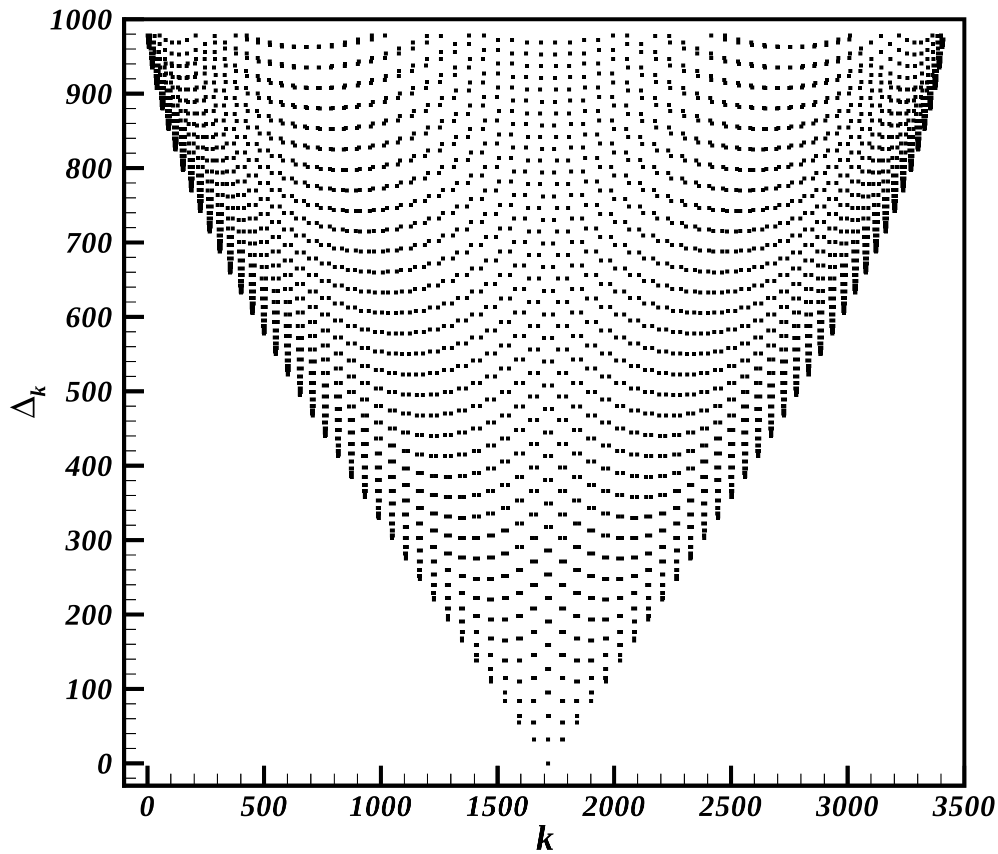
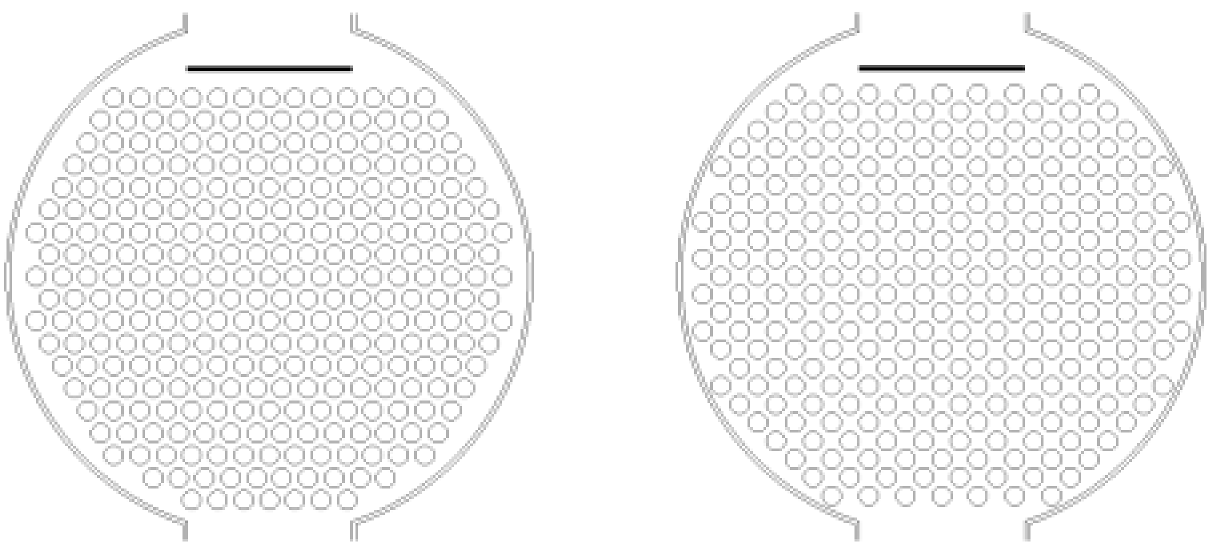


Figure 2: Typical tube distance (Δ_k) distribution obtained from searching procedure – $D_s = 2000mm$, $D_t = 25.4mm$, $\alpha = 60^o$, $\phi_{in} = 250mm$, $\phi_{out} = 250mm$, with impingement plate.

3. TUBE DISTANCE SORT PROCEDURE

The tube searching procedure leads to a very particular kind of distribution of tube center distances Δ_k. As shown in Fig. 2 for a 3410-tube heat exchanger, tubes located closer to the shell center are found halfway into the searching procedure. If a given number of tubes is to be placed within the heat exchanger cross section, validated tube positions must be occupied in crescent order of tube center distances (Δ_k). Therefore, an efficient sorting procedure must be used in order to adapt the initially found tube distribution to the given requirements.

The natural choice of a sequential sorting procedure, in which points of order k and $k+1$ are switched if $\Delta_k > \Delta_{k+1}$, is used in the present study. However, the sequential sorting procedure can become computationally inefficient, specially when a large number of tubes is considered (Knuth, 1998). Developments are currently being introduced in order to improve the pure sequential sorting algorithm using the characteristics of the tube distance distribution obtained from the searching algorithm. Nevertheless, a CPU time requirement of less than $2s$ was observed for each of the individual results shown in the current study. A Pentium Pro 200MHz was used for the calculations.



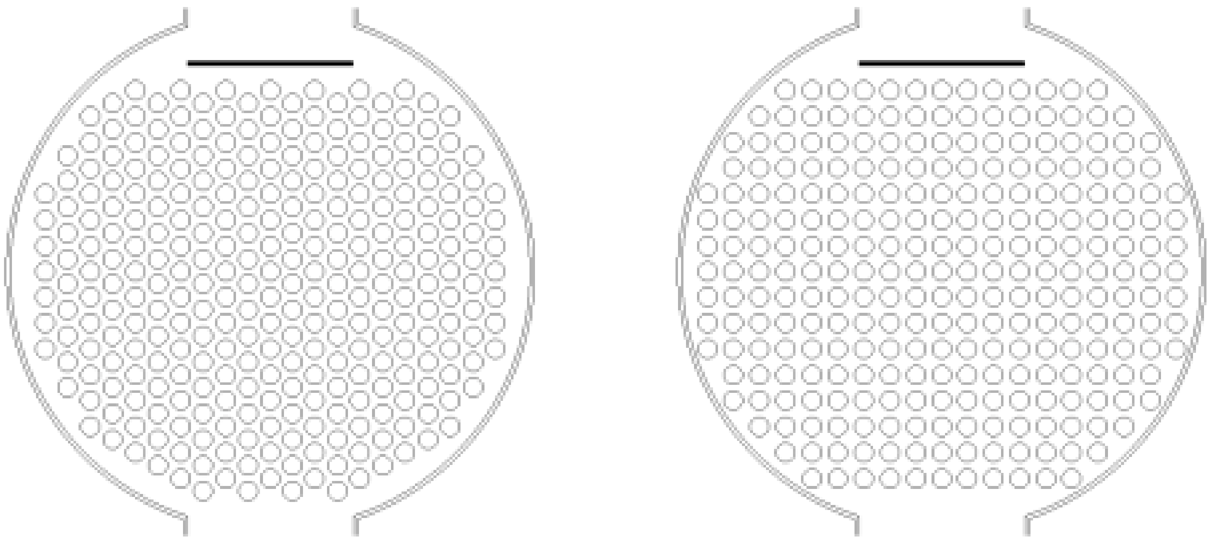
(a) (b)

Figure 3: Results for (a) $\alpha = 30^o$ and for (b) $\alpha = 45^o$ – $D_s = 635mm$, $D_t = 25.4mm$, $\delta = 31.75mm$, $\epsilon_r = 3.175mm$ and $\phi_{in} = \phi_{out} = 203mm$.

4. RESULTS

The implementation of the discussed searching and sorting procedures was performed using the Delphi 3 Integrated Development Environment (IDE) (Cantù, 1997), object oriented programming and a simple and practical Human Computer Interface (HCI) (Pressman, 1995). The choice of the IDE made available the Unit Math, which contains subroutines specially compiled for the Floating Point Unit (FPU) of Pentium processors (Taylor, 1999). Besides, the Delphi 3

IDE is recognized as having the best interacting performance with the Graphics Display Interface (GDI) of the Microsoft Windows Operating System.



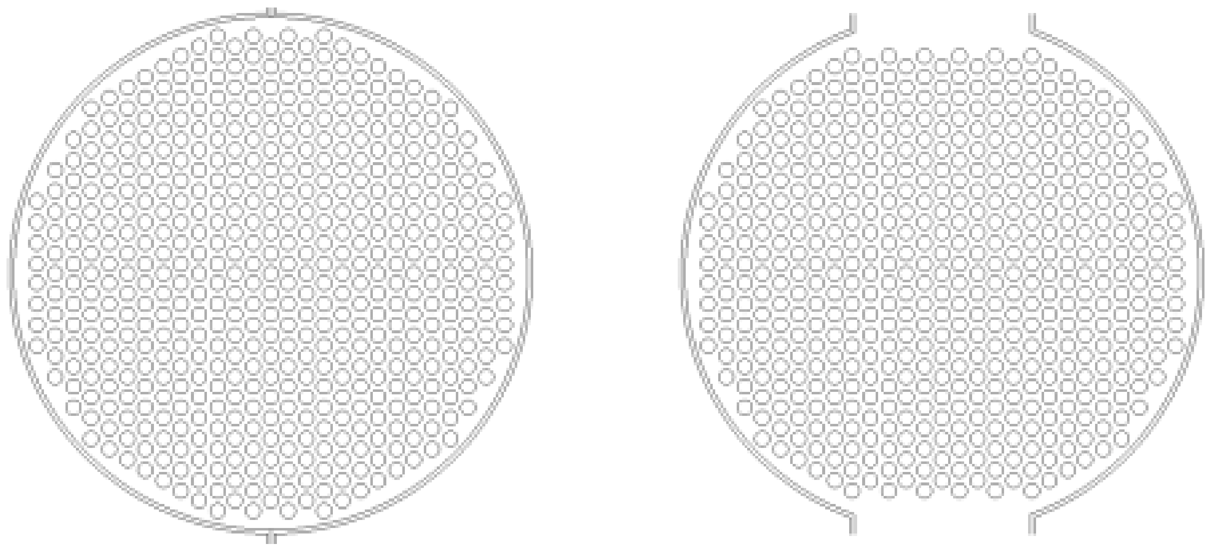
(a) (b)

Figure 4: Results for (a) $\alpha = 60^o$ and (b) $\alpha = 90^o$ – $D_s = 635mm$, $D_t = 25.4mm$, $\delta = 31.75mm$, $\epsilon_r = 3.175mm$ and $\phi_{in} = \phi_{out} = 203mm$.

Results for different angles of arrangements are shown in Figs. 3 and 4 for a base configuration of $D_s = 635mm$, $D_t = 25.4mm$, $\delta = 31.75mm$, $\epsilon_r = 3.175mm$ and $\phi_{in} = \phi_{out} = 203mm$. For the given configuration, 296, 270, 292 and 268 tubes can be used for $\alpha = 30^o$, 45^o, 60^o and 90^o, respectively. Tube count tables for $\alpha = 30^o$ (Sauders, 1988) indicate that, for a similar configuration, a maximum of 311 tubes, corresponding to a 4.8% deviation from the calculated value. It is noteworthy the influence of radial clearance on the results. As $\epsilon_r \rightarrow 0$, the difference between calculated and tube count table values tends to vanish. Nevertheless, the value of ϵ_r used by Saunders (1988) in developing the data tables is unclear.

Different results of the implemented algorithm are also shown in Figs. 5-7 for a base configuration of $D_s = 591mm$, $D_t = 19.05mm$, $\delta = 23.81mm$, $\alpha = 60^o$ and $\epsilon_r = 3.175mm$. Considering $\phi_{in} = \phi_{out} = 0$, a tube count of 511 is obtained, while data table indicate a maximum of 516 tubes (1% deviation) can be accommodated within the heat exchanger shell. A sketch of the tube bundle layout is shown Fig. 5a for the 511-tube configuration. Considering inlet and outlet nozzle diameters equal to $203mm$, calculations show that 493 tubes can be accommodated and the corresponding tube layout is shown in Fig. 5b. If an impingement plate is placed at the inlet nozzle position and $\phi_{in} = \phi_{out} = 203mm$, the maximum number of tubes is reduced to 472, as shown in Fig. 6a. Placing the inlet and outlet nozzle at the same side of the heat exchanger shell, as depicted in Fig. 6b, allows the addition of 9 tubes to the previous bundle.

Cases where the number of tubes to be accommodated is smaller than the calculated maximum shell capacity of 472 tubes are illustrated in Fig. 7. In Fig. 7a, 408 tubes are used, leading to a calculated outer tube limit (OTL) of $534.2mm$. The outer tube limit corresponds to the diameter of the largest circle drawn about the tubesheet center, which is tangent to the most externally placed tube. Reducing the number of tubes to 350, the outer tube limit is reduced to $448.0mm$, and a sketch of the tube bundle layout is depicted in Fig. 7b.

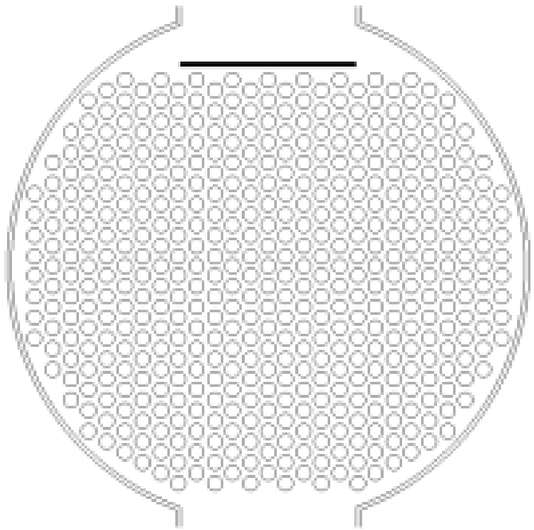


(a) (b)

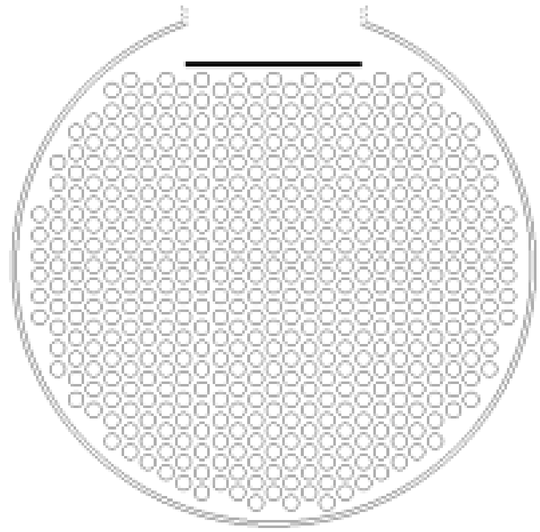
Figure 5: Results for (a) maximum number of points and for (b) $\phi_{in} = \phi_{out} = 203mm$ – $D_s = 591mm$, $D_t = 19.05mm$, $\delta = 23.81mm$, $\alpha = 60^o$ and $\epsilon_r = 3.175mm$.

5. CONCLUSIONS

A numerical procedure for the determination of tube bundle parameters for tubular heat exchangers is presented. Initially, a tube position searching algorithm is discussed and the validation criteria for the found positions defined for a given equipment configuration. A sorting algorithm is also introduced in order to address design situations where the number of tubes to be used is smaller than the shell tube count. Results for single tube pass heat exchangers are presented showing characteristics of the implementation and the precision of the calculations, when compared to tube count table data. The extension of the discussed methodology to multiple tube pass heat exchangers and improvements on the sorting procedure are currently being developed.



(a)



(b)

Figure 6: Results for $\phi_{in} = \phi_{out} = 203mm$ and considering (a) impingement plate and (b) nozzles at the same shell side – $D_s = 591mm$, $D_t = 19.05mm$, $\delta = 23.81mm$, $\alpha = 60^o$ and $\epsilon_r = 3.175mm$.

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REFERENCES

Cantù, M., 1997, Dominando o Delphi 3, Makron Books, São Paulo.

Fraas, A.P., 1989, Heat Exchanger Design, John Wiley, New York.

Kern, D.Q., 1965, Process Heat Transfer, McGraw Hill, New York.

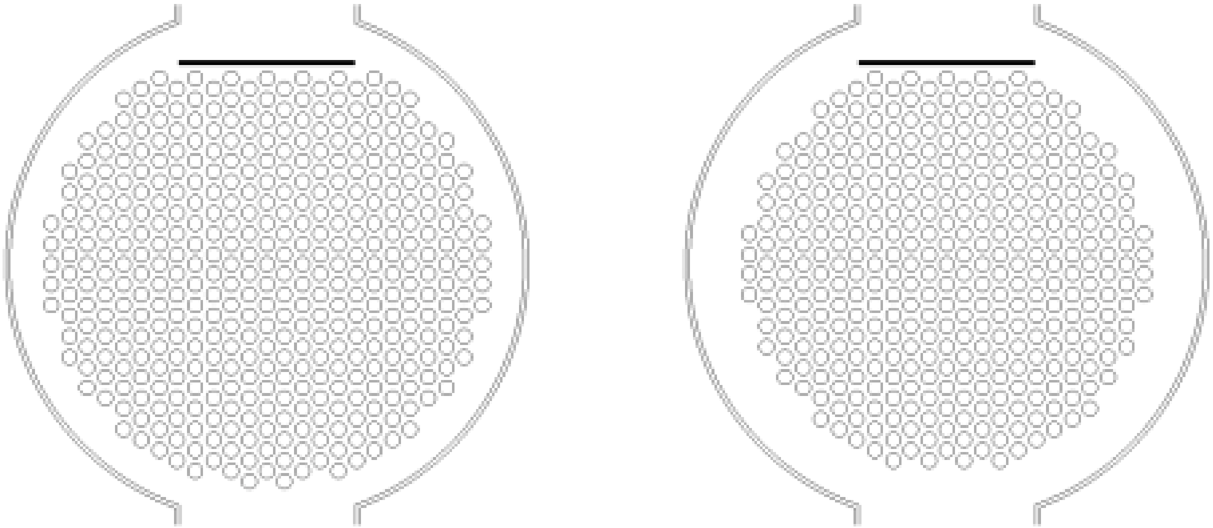
Knuth, D.E., 1998, The Art of Computer Programming: Sort and Searching, Vol.3, Addison-Wesley, New York

Pressman, R., 1995, Engenharia de Software, Makron Books, São Paulo.

Saunders, E.A.D., 1988, Heat Exchangers Selection, Design and Construction, Longman Scientific & Technical, Harlow, Essex, England.

Taylor, D., 1999, Programando Delphi 3 - Técnicas Avançadas, Makron Books, São Paulo.

Walker, G., 1990, Industrial Heat Exchangers: A Basic Guide, Hemisphere, New York.



(a) (b)

Figure 7: Results for (a) 408 tubes and (b) 350 tubes – $D_s = 591mm$, $D_t = 19.05mm$, $\delta = 23.81mm$, $\alpha = 60^o$ and $\epsilon_r = 3.175mm$, $\phi_{in} = \phi_{out} = 203mm$.