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THREE-DIMENSIONAL NUMERICAL STUDY OF STAGGERED FINNED CIRCULAR AND ELLIPTIC TUBES IN FORCED CONVECTION

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Abstract. *In this study, a three-dimensional (3-D) numerical heat transfer investigation was performed in a bundle of finned or non-finned tubes for a given volume and a given external laminar forced flow. Both circular and elliptic tubes geometry, for a general staggered configuration in laminar regime, were investigated. The optimization procedure started by recognizing the design limited space availability as a fixed volume constraint. The finite volume method was used to discretize the fluid flow and heat transfer governing equations. OpenFOAM, an open-source computational fluid dynamics (CFD) code, was used in all simulations. Results identified a numerical geometric optimization for maximum heat transfer. A relative heat transfer gain and a pressure drop reduction were also observed in the optimal elliptic arrangement in comparison to the circular one. All obtained results were validated by direct comparison with previous numerical and experimental studies found in literature.*

Keywords: *heat exchanger; numerical analysis; CFD; optimization.*

1. INTRODUCTION

The increase in energy demand in all sectors of the human society requires an increasingly more intelligent use of available energy. Many industrial applications require the use of heat exchangers with finned tube arrangements to be used in air conditioning systems, refrigeration, heaters, radiators, etc. Clear examples are the design rules for determining geometric characteristics of a package with fixed volume such that the overall thermal conductance between package and coolant is maximized. Optimal spacing have been already reported for several geometries. Heat exchangers with finned elliptical tubes were studied by Brauer [1], Bordalo and Saboya [2], Saboya and Saboya [3], and Jang and Yang [4], Matos et al [5, 6] and Manairdes et al [7] showing a relative heat transfer gain observed in the elliptical arrangements, as compared to the circular ones. Nusselt number correlations for heat transfer was established as a function of Reynolds number and quantity of tubes in bundle in difference arrangement (in line and staggered) by Colburn [8], Zukauskas [9], Grimison [10] and Khan [11].

In this study, a numerical investigation on the geometric parameters for staggered arrangement with circular and elliptic tubes in a fixed volume to maximized the heat transfer.

2. PROBLEM DESCRIPTION

A typical row tube and plate fin heat exchanger with a staggered configuration is shown in Fig. 1. The problem was reduced to a unit cell also shown in Fig. 1. Fowler and Bejan [12] showed that in the laminar regime, the flow through a large bank of cylinders could be simulated accurately by calculating the flow through a single channel. Due to the geometric symmetries, there is no fluid exchange or heat transfer between adjacent channels, or at the top and side surfaces. At the bottom of each unit cell, no heat transfer is expected across the plate fin mid-plane. In Fig. 1, L, H and W are the length, height and width (tube length) of the array, respectively. The fins are identical, where t is the thickness and d is the fin-to-fin spacing.

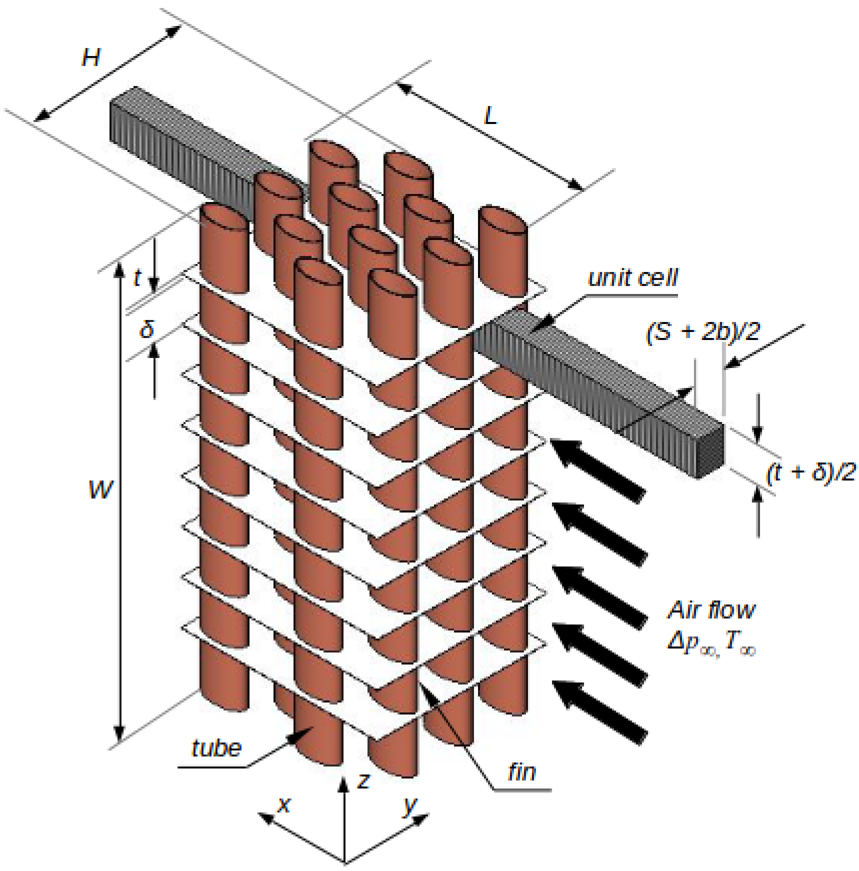

Figure 1: Arrangement of finned elliptic tubes, and the three-dimensional computational domain.

3. GOVERNING EQUATIONS

Governing equations for the problem are the mass, momentum and energy equations which were simplified in accordance with the assumptions of three-dimensional incompressible steady-state laminar flow with constant transport properties, for a Newtonian fluid:

Continuity equation:

$$\nabla \cdot \vec{V} = 0 \tag{1}$$

Navier-Stokes:

$$\rho \frac{D\vec{V}}{Dt} = -\nabla p + \nabla . \overline{\tau} + \rho \vec{g} \tag{2}$$

Energy equation:

$$\rho c_p \frac{DT}{Dt} = k \nabla^2 T \tag{3}$$

Problem formulation is closed by application of the following boundary conditions for the extended three-dimensional computational domain of Fig. 2:

inlet (blue) $p=10\,Pa \quad ; T=300\,K$ (4)

back (green) and front (cyan) $v=w=0;\quad \frac{\partial u}{\partial Z}=\frac{\partial T}{\partial Z}=0$ (5)

where *u*, *v* and *w* are Cartesian components of the velocity vector.

topAndBottom (gray) $v=w=0;\quad \frac{\partial u}{\partial Y}=\frac{\partial T}{\partial Y}=0$ (6)

outlet (red) $p=0\,Pa;\quad \frac{\partial T}{\partial X}=0$ (7)

tubes (yellow) $u=v=w=0;\quad T=310\,\mathrm{K}$ (8)

fin (pink) $u=v=w=0;\quad \frac{\partial T}{\partial Z}=0$ (9)

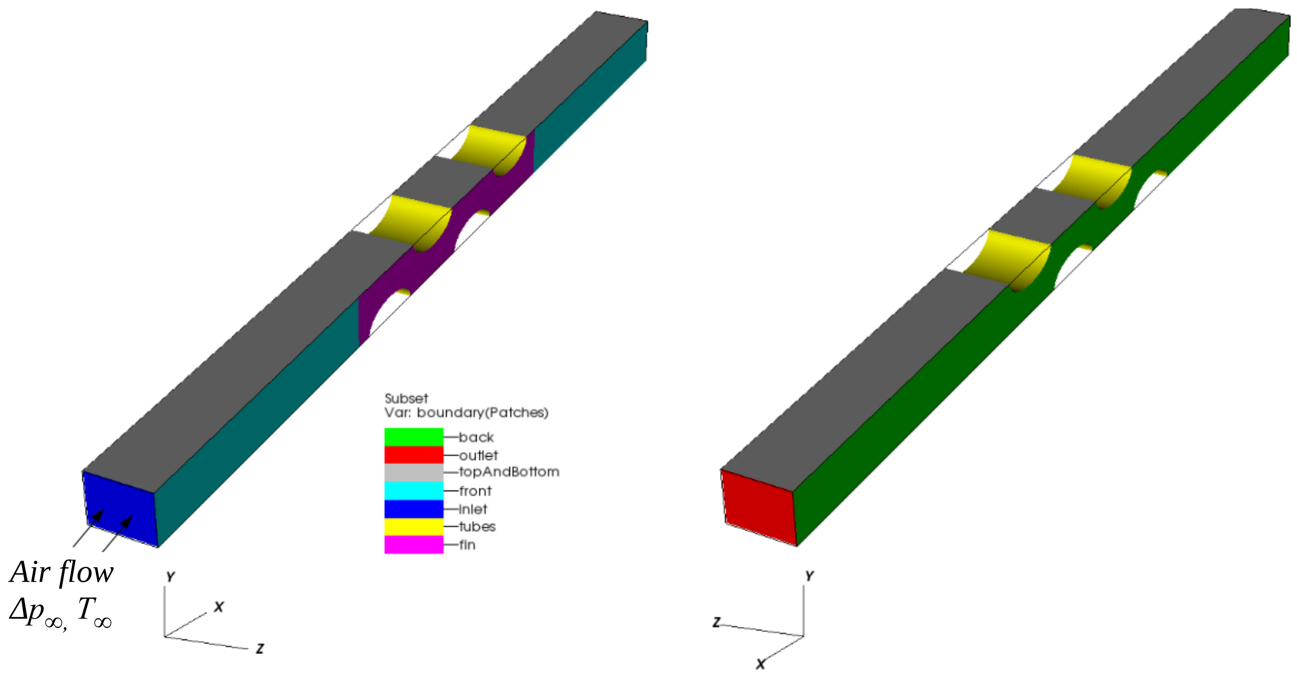

Figure 2: The boundary condition of the 3-D computational domain.

The Eq. (8) represent boundary condition for tube walls that are modeled as isothermal, Tw = constant, this in order to account for the presence of a well mixed fluid inside the tubes. In order to represent the actual flow with boundary conditions (inlet) and (outlet), two extensions need to be added to the computational domain, upstream and downstream, as shown in Fig. 2. The actual dimensions of these extra lengths need to be determined by an iterative numerical procedure, with convergence obtained according to a specified tolerance.

For all cases, flow regime is driven by the pressure difference Δp, which is maintained across the assembly.

$$\Delta p \approx \frac{1}{2} \rho U_\infty^2 \qquad (10)$$

The dimensionless parameters used in the numerical simulation are given by:

$$\mathrm{Re}_D = \frac{U_\infty D}{\nu} \qquad (11)$$

where υ is kinematic viscosity (m^2/s) and ρ is density (kg/m^3)

$$\overline{Nu}_D = \frac{hD}{k} \qquad (12)$$

where h is the heat transfer coefficient (W/m^2 K) and k is the thermal conductivity of the fluid (W/mK).

4. NUMERICAL SOLUTION

Numerical solution of Eqs. (1)-(3) was performed with the finite volume method (FVM) [13], resulting the velocity and temperature fields in the unit cell of Fig. 1. Problem symmetries allow computational domain to be reduced to one unit cell represented by the extended domain (height = $S/2+b$ and width = $\delta/2 + t/2$) shown in Figs. 1 and 2. Fluid flow simulation was carried out using OpenFOAM [14] with the *buoyantBoussinesqPimpleFoam* solver. Geometry and mesh were created using Gmsh [15]. For post-processing, Visit visualization tool [16] has been used. All simulations are performed using computers with 6 Intel Core i7-6800K of 3.40 GHz clock and 16.0 Gb of RAM.

Initial steps on the simulation procedure were to evaluate mesh quality, solver precision and the computational model. For that, an established example taken from literature was choose. The case of the flow around a circular cylinder was solved to test mesh independence and the accuracy of solver software.

Figure 3 showed some details of chosen mesh and boundary conditions for a domain of 7D and 0,7D, for x and y axes respectively, with follow parameters: D = 0.1 m, T_∞ = 300 K, Tw = 320 K, Re_D = 100 and Pr = 0,71, where T_∞ and Tw are the temperatures of free stream and cylinder wall, respectively, and is D the cylinder diameter.

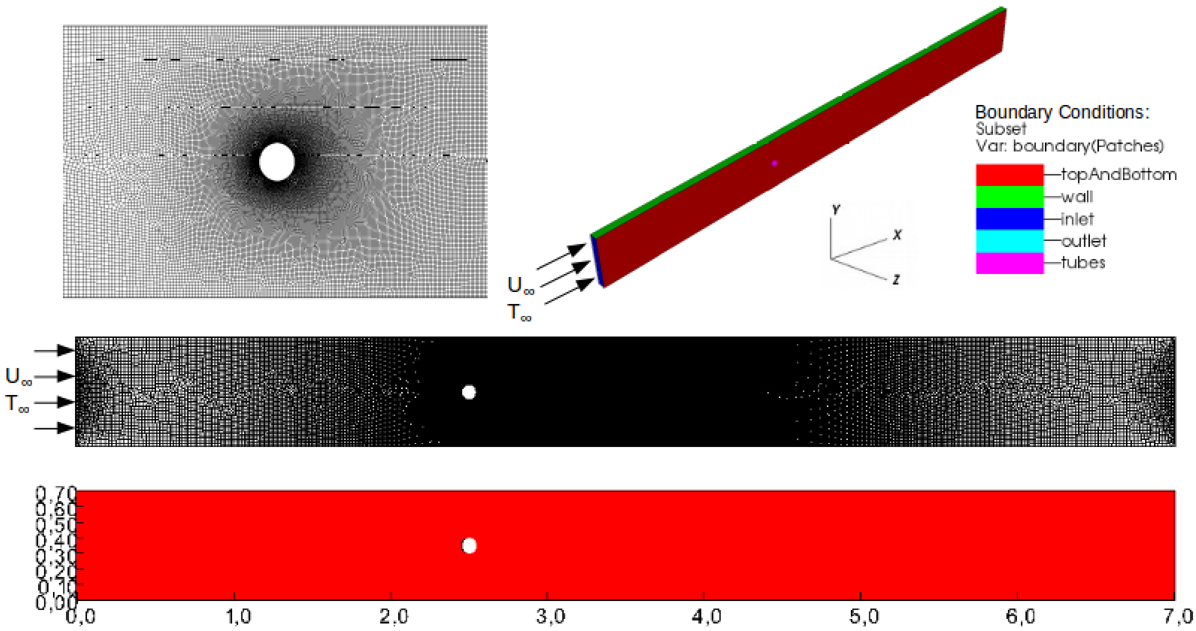

Figure 3: Independent mesh (80 divisions on cylinder perimeter)

Independence mesh test started with a coarse grid (20 divisions on cylinder perimeter, according to Fig. 4) and for each new simulation the number of divisions was increased up to the limit where difference in results of two sequential grids was smaller than 1%. For each step, average Nusselt number ($\overline{Nu}$) was calculate and used was verification parameter. Four simulations were carried out for 20, 40, 80 and 160 divisions. The third simulation with 80 divisions was considered independent.

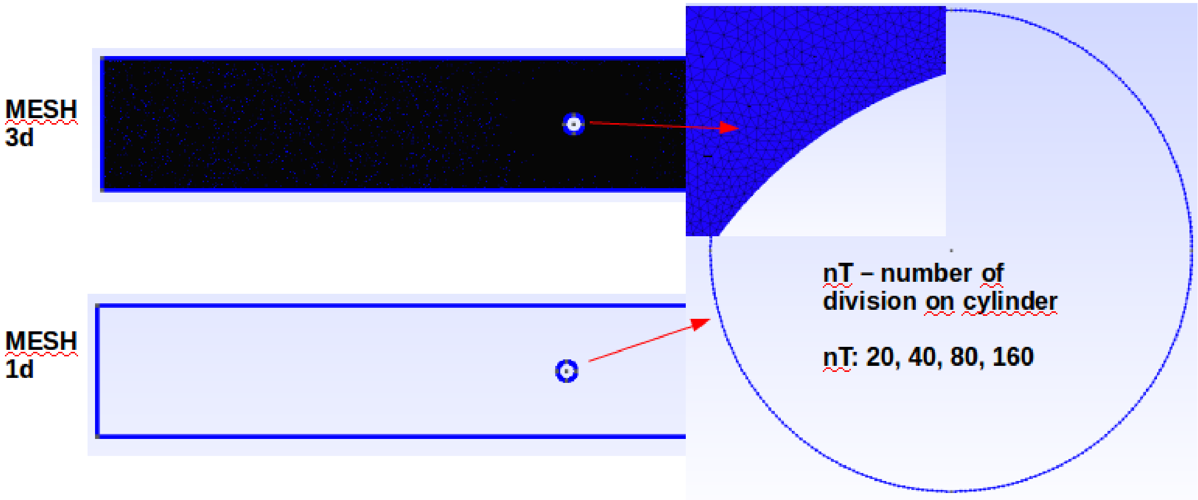

Figure 4: Details of refined mesh around cylinder perimeter.

For laminar cases with low Reynolds (Re_D = 100), the correlation of Average Nusselt ($\overline{Nu}$) is given by [17]:

$$\overline{Nu}_{cyl} = \frac{hD}{k} = CRe_D^m Pr^n \qquad Re_D = 40-4000, \quad C = 0.683, m = 0.466 \tag{13}$$

Solution of Eq. (13) for Re = 100 and Pr = 0,71 returns $\overline{Nu}$= 5.21, while numerical solution obtained was $\overline{Nu}$= 5.23, with 0.3% of relative error. In the Fig. 5 is showed temperature and velocity fields and velocity vectors for the independent mesh.

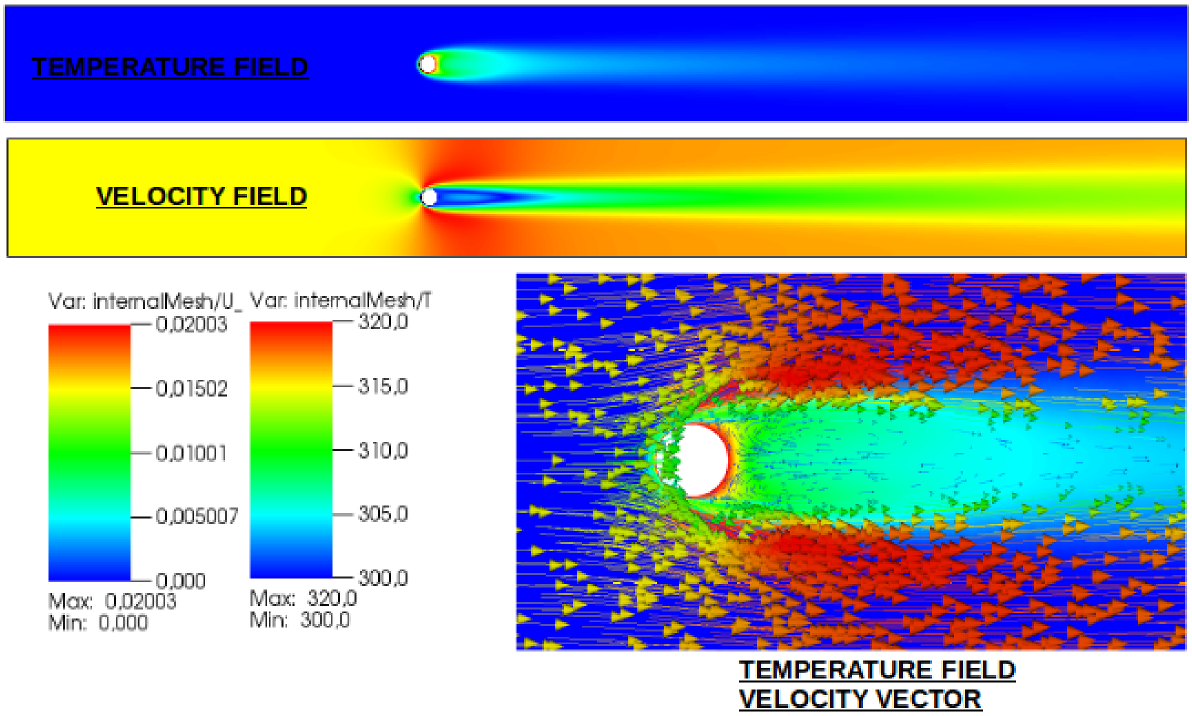

Figure 5: Temperature and velocity fields and velocity vectors. Steady-state converged solution.

5. RESULTS AND DISCUSSION

Based on mesh independent test presented in section 4, 3D independent mesh is showed in Fig. 6.

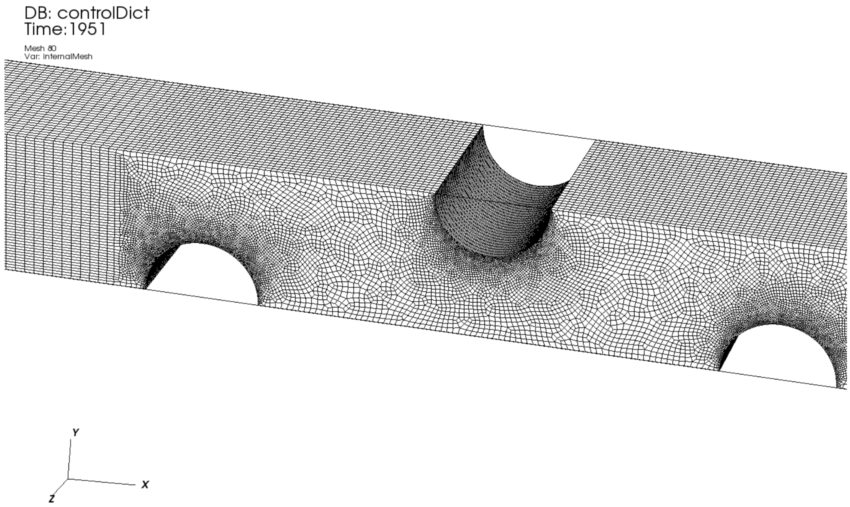

Figure 6: Details of 3D mesh for a “unit cell” with 04 elliptical tubes, eccentricity 0,5.

Results obtained in this study for numerical heat transfer of a staggered array of tubes with isothermal fins were compared with other studies presented in literature. Reference parameters are: numerical calculated average Nusselt and Reynolds number. As can be observed in Fig. 7, a good agreement with analytical and experimental correlation for Reynolds above 600 (2% to 16%) and a greater variation (~30%) for low Reynolds.

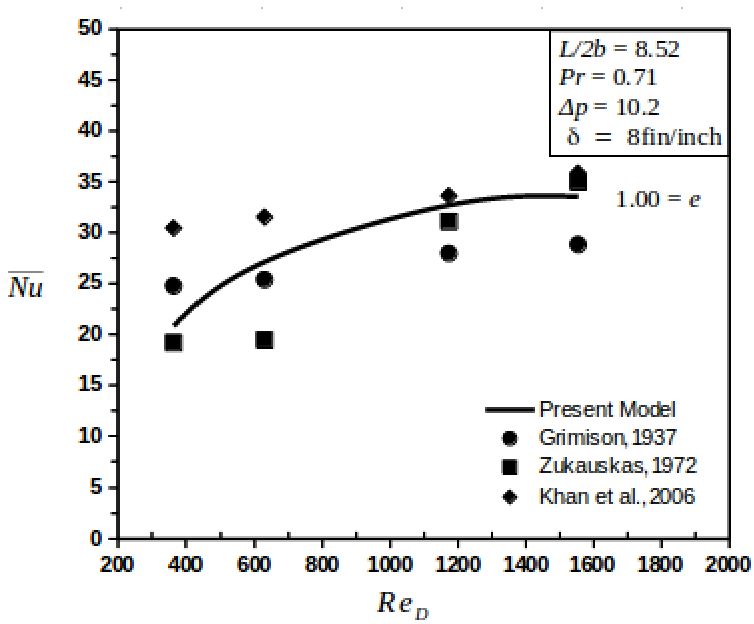

Figure 7: Average Nusselt number for finned circular staggered arrangement.

Results were obtained for laminar flow regime. Velocity field was driven by a fixed pressure drop of $\Delta p = 10$ Pa.

The aim of optimization is to find optimal tube-to-tube spacing, $S/2b$, for eccentricities e = 1, 0.75, 0.5 and 0.4, respectively.

Figs. 8, 9 and 10 show the optimization results for different finned arrangement (δ = 0.5 fin/inch, 8 fin/inch and 22 fin/inch). The results of global optimization with respect to the two degrees of freedom, S/2b and eccentricities. And shown that the elliptical arrangement is more efficient than circular one.

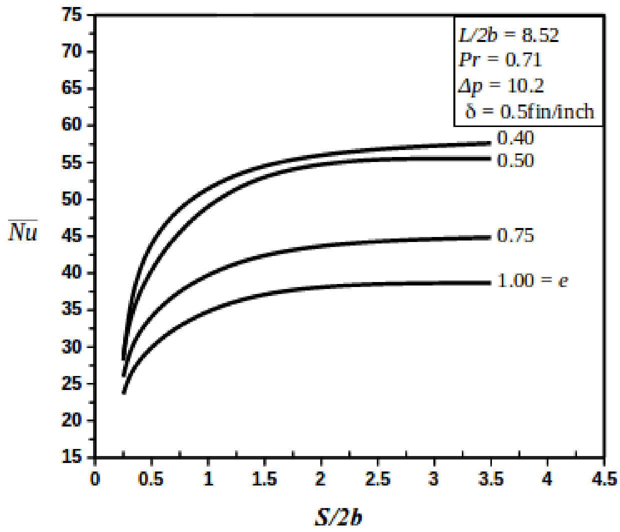

Figure 8: Numerical results of $\overline{Nu}$ for finned circular and elliptic arrangements (δ = 0.5 fin/inch).

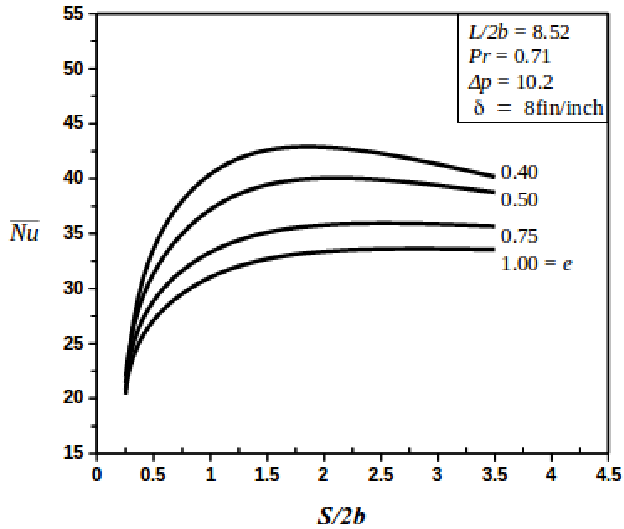

Figure 9: Numerical results of $\overline{Nu}$ for finned circular and elliptic arrangements (δ = 8 fin/inch).

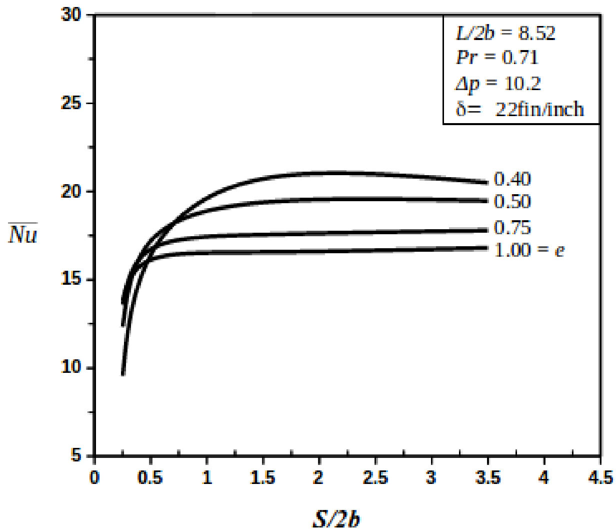

Figure 10: Numerical results of $\overline{Nu}$ for finned circular and elliptic arrangements (δ = 22 fin/inch).

In the Fig. 11 is showed an example of 3-D elemental channel (computational domain) with fields of temperature for a follow configuration: *S/2b*= 0.5; *e*= 0.5 and δ = 0.5 fin/inch.

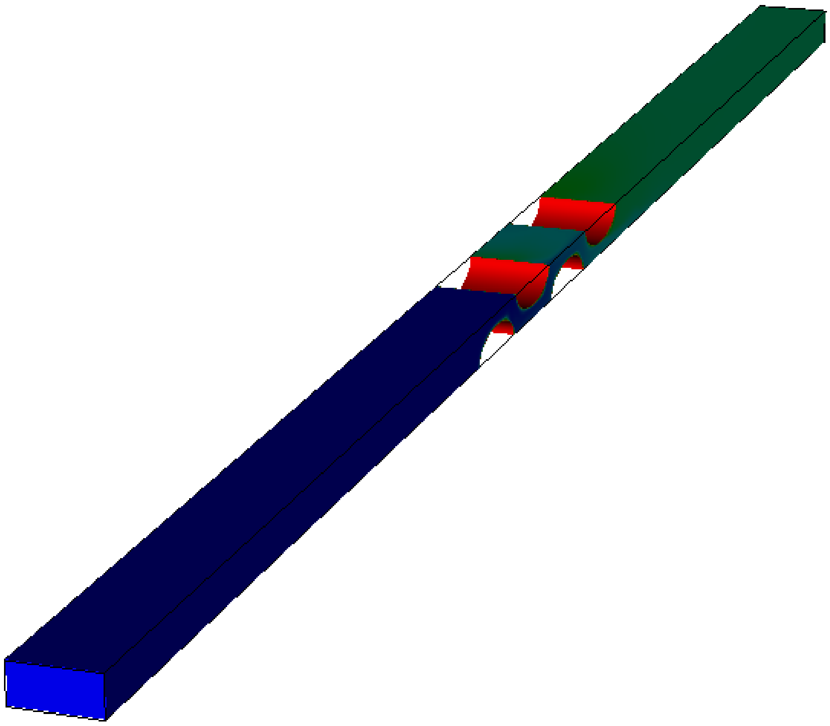

Figure 11: Temperature fields (T) in computational domain.

Fig. 12shown the fields of temperature and velocity for different finned-tube arrangement.

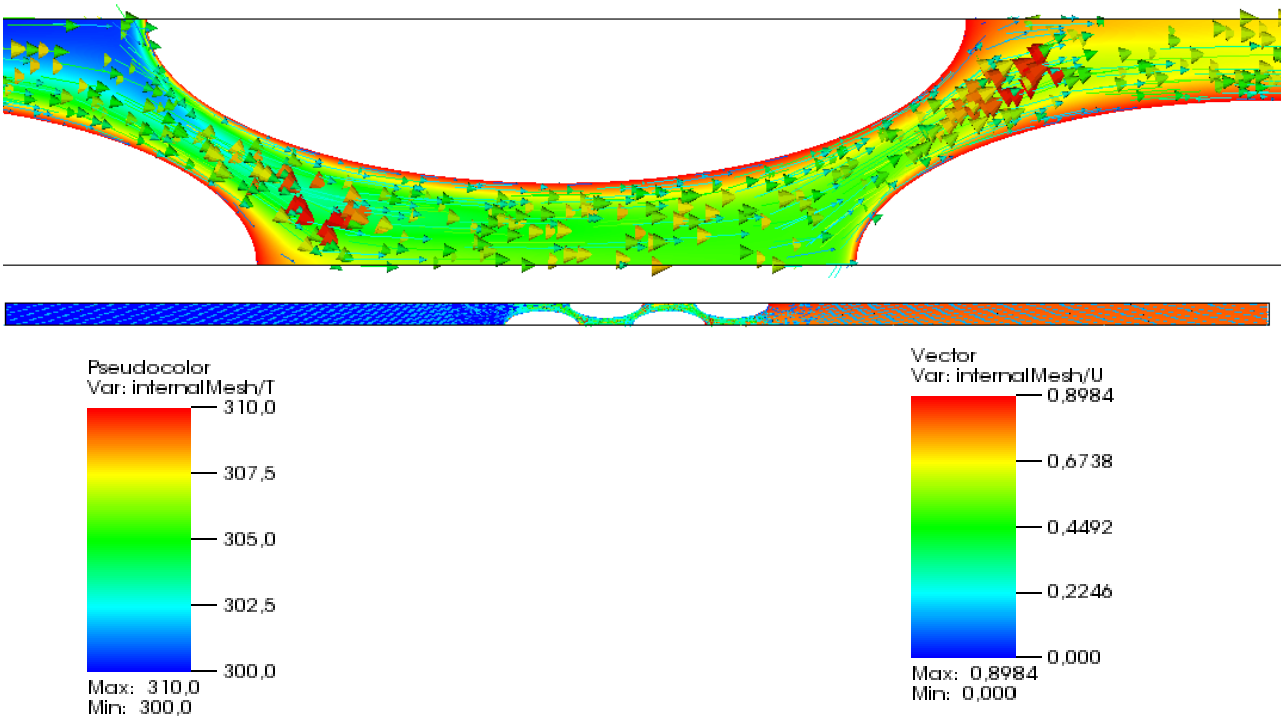

Figure 12: Temperature fields (T) and velocity vectors ($\vec{V}$) for: *S/2b*= 0.5; *e*= 0.4 and δ = 22 fin/inch

The Fig. 13 is represent some difference tube-to-tube spacing, $S/2b$ for finned circular tube arrangement.

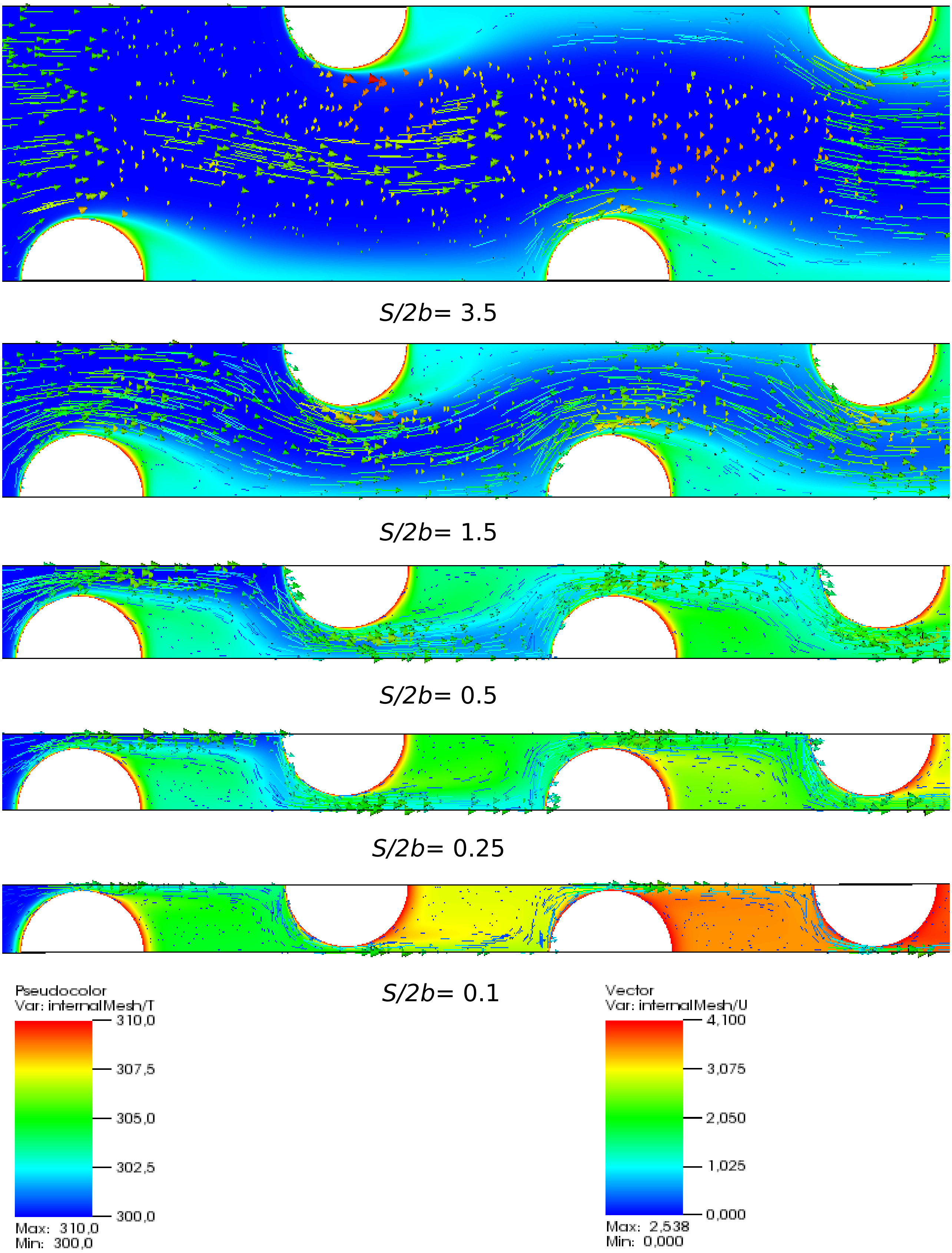

Figure 13: Temperature fields (T) and velocity vectors ($\vec{V}$) for: e= 1.0 and δ = 8 fin/inch

6. CONCLUSIONS

In this paper, a numerical study was conducted to demonstrate that non-finned and finned circular and elliptic tubes heat exchangers can be optimized for maximum heat transfer, under a fixed volume constraint. The internal geometric structure of the arrangements was optimized for maximum heat transfer. Such globally optimized configurations are expected to be of great importance for actual heat exchangers engineering design, and for the generation of optimal flow structures.

7. ACKNOWLEDGEMENTS

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