

MODELING A COUNTER-FLOW PLATE HEAT EXCHANGER

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The paper presents the modeling of a counter-flow plate heat exchanger using the finite element method .

COSMOS/Flow program has been used to predict the fluid flow distribution. The temperature and velocity field a section of the plate heat exchanger are determinate and the overall heat transfer coefficient is calculate.

Keywords: plate heat exchanger, finite element method, fluid flow, heat transfer.

1. Introduction

Heat exchangers are very common equipments used in the field of thermal and electrical power generation. Plate heat exchangers have a lot of advantages, including a high heat exchange area per unit volume and good heat transfer performance [1] The correct operation of heat exchangers influences the energy efficiency of whole chain from production to distribution of electricity and heat.

An important number of numerical studies applying finite element method have been realized to research fluid flow and heat transfer into plate heat exchangers [2], [3].

The paper presents 3D – model realized using SolidWorks and COSMOS/Flow program has been used to analyze the fluid flow and heat transfer. The temperature and flow patterns inside of plate heat exchanger are observed and the overall heat transfer coefficient is calculated.

2. The studied plate heat exchanger

The studied heat exchanger is a pack of eight stainless steel plates with gaskets. The hot water flows are in one direction in alternating chambers while the cold water flows are in counter flow in the other alternating chambers. The number of passes is 1 and the number of channels is $N_c = 4$, like in figure 1.

The thermal agents are directed into their proper chambers either by a suitable gasket made from rubber EPDM. The width of channel is 0,004 m.

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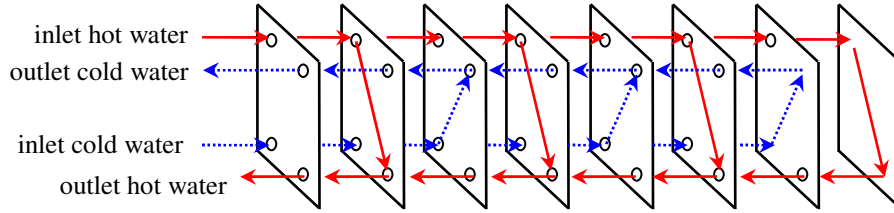


Fig.1. Schematic presentation of the hot and cold water flows

The geometric dimensions of the plate are represented in table 1.

Table 1

Geometric dimensions of the plate

Dimension	Notation	Value	Unit
Diameter of the inlet tube	d	0,002	m
Effective plate length, measured between ports	L	0,37	m
Effective plate width, measured between ports	l	0,09	m
Stainless steel plate thickness	δ_p	0,002	m

3. Mathematical modelling

Mathematical modelling includes assignation of governing equations. The partial differential equations (pdes) governing fluid flow and heat transfer include the continuity equation, the Navier-Stokes equations and the energy equation. These equations are intimately coupled and non-linear making a general analytic solution impossible except for a limited number of special problems, where the equations can be reduced to yield analytic solutions [4]. Because most practical problems of interest do not fall into this limited category, approximate methods are used to determine the solution to these equations.

Cosmos/Flow solves the mathematical equations which represent heat and momentum transfer in a moving fluid [5]. The finite element method is used to discretize the flow domain, thereby transforming the governing partial differential equations into a set of algebraic equations whose solution represents an approximation to the exact analytical solution.

A set of several simplified hypothesis must be introduced for realize the mathematical model:

- Hot water and cold water are Newtonian fluids;
- No phase change occurs, the fluids are unmixed;
- Turbulent flow is fully developed;
- Working fluids are incompressible;
- Steady state conditions.

The governing equations for fluid flow and heat can be written as [5]:

Continuity equation:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u}{\partial x} + \frac{\partial \rho v}{\partial y} + \frac{\partial \rho w}{\partial z} = 0 \quad , \quad (1)$$

x-, y-, z- momentum equations:

$$\begin{aligned} & \rho \frac{\partial u}{\partial t} + \rho u \frac{\partial u}{\partial x} + \rho v \frac{\partial u}{\partial y} + \rho w \frac{\partial u}{\partial z} = \\ & = \rho g_x - \frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left[2\eta \frac{\partial u}{\partial x} \right] + \frac{\partial}{\partial y} \left[\eta \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right) \right] + \frac{\partial}{\partial z} \left[\eta \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \right) \right] + S_\omega + S_{DR} \quad , \\ & \rho \frac{\partial v}{\partial t} + \rho u \frac{\partial v}{\partial x} + \rho v \frac{\partial v}{\partial y} + \rho w \frac{\partial v}{\partial z} = \\ & = \rho g_y - \frac{\partial p}{\partial y} + \frac{\partial}{\partial y} \left[2\eta \frac{\partial v}{\partial y} \right] + \frac{\partial}{\partial x} \left[\eta \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right) \right] + \frac{\partial}{\partial z} \left[\eta \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \right) \right] + S_\omega + S_{DR} \quad , \\ & \rho \frac{\partial w}{\partial t} + \rho u \frac{\partial w}{\partial x} + \rho v \frac{\partial w}{\partial y} + \rho w \frac{\partial w}{\partial z} = \\ & = \rho g_z - \frac{\partial p}{\partial z} + \frac{\partial}{\partial z} \left[2\eta \frac{\partial w}{\partial z} \right] + \frac{\partial}{\partial x} \left[\eta \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \right) \right] + \frac{\partial}{\partial y} \left[\eta \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \right) \right] + S_\omega + S_{DR} \quad . [2] \end{aligned}$$

The two source terms in the momentum equations, S_ω and S_{DR} , are for rotating coordinates and distributed resistances, respectively.

The distributed resistance term can be written in general as:

$$S_{DR} = - \left(K_i + \frac{f}{d} \right) \frac{\rho V_i^2}{2} - C \eta V_i, \quad [3]$$

where i refer to the global coordinate direction (u , v , w momentum equation), f - friction factor, d - hydraulic diameter, C – permeability and the other factors are described in table 2. Note that the K -factor term can operate on a single momentum equation at a time because each direction has its own unique K -factor. The other two resistance types operate equally on each momentum equation [5].

The other source term is for rotating flow. This term can be written in general as:

$$S_\omega = -2\rho \omega_i \times V_i - \rho \omega_i \times \omega_i \times r_i, \quad [4]$$

where i refer to the global coordinate direction, ω is the rotational speed and r is the distance from the axis of rotation.

For incompressible and subsonic compressible flow, the energy equation is written in terms of static temperature [5]:

$$\rho c_p \frac{\partial T}{\partial t} + \rho c_p u \frac{\partial T}{\partial x} + \rho c_p v \frac{\partial T}{\partial y} + \rho c_p w \frac{\partial T}{\partial z} = \frac{\partial}{\partial x} \left[U \frac{\partial T}{\partial x} \right] + \frac{\partial}{\partial y} \left[U \frac{\partial T}{\partial y} \right] + \frac{\partial}{\partial z} \left[U \frac{\partial T}{\partial z} \right] + q_V, [5]$$

The volumetric heat source term from equation (5) is considered zero for this model.

Table 2 presents the variable of the equations:

Table 2

Variables of the governing equations

Variable	Description
c_p	specific heat at constant pressure
k	thermal conductivity
p	pressure
q_V	volumetric heat source
T	temperature
t	time
u	velocity component in x-direction
v	velocity component in y-direction
w	velocity component in z-direction
ρ	density
η	dynamic viscosity

The equations describe the fluid flow and heat transfer under steady-state conditions for Cartesian geometries. For the turbulent flow, the solution of these equations would require a great deal of finite elements (on the order of $10^6 - 10^8$) even for a simple geometry as well as near infinitesimal time steps. COSMOS/Flow solves the *time-averaged* governing equations.

The time-averaged equations are obtained by assuming that the dependent variables can be represented as a superposition of a mean value and a fluctuating value, where the fluctuation is about the mean [5]. For example, the velocity component in y-direction can be written:

$$V = V + v', [m/s] \tag{6}$$

where V [m/s] – the mean velocity, v' [m/s] – the fluctuation about the mean. This representation is introduced into the governing equations and the equations themselves are averaged over time.

4. Numerical simulation

The 3D model realized in SolidWorks is presented in figure 2.



Fig.2. 3D model of counter-flow plate heat exchanger

The finite element method is used to discretize the flow domain, thereby transforming the governing partial differential equations into a set of algebraic equations whose solution represents an approximation to the exact analytical solution. The numerical formulation is derived from the SIMPLER solution scheme introduced by Patanker [6].

In COSMOS/Flow it is open the model and it is defined the case study, by specifying its name, analysis type, the disparagement mode. For this case, the study is fluid flow type and the analysis is thermal (result type: permanent or transient, the iterative resolving technique: Newton-Raphson scheme, initial temperature and calculus tolerance). It is defining material types for components. It is defining loading and boundary condition.

After those, it applied mesh, the disparagement mode, which is very important for the final results. It is very important to realize the mesh with great accurately. The mesh quality depends on the active mesh options. In this case, the quality of mesh is high (10-node tetrahedral). Mesh type is solid mesh. The number of fluid and solid material elements and nodes associated with the current geometry are given next:

- Fluid nodes: 42122;
- Solid nodes: 38915;
- Fluid elements: 164424;

- Solid elements: 281960.

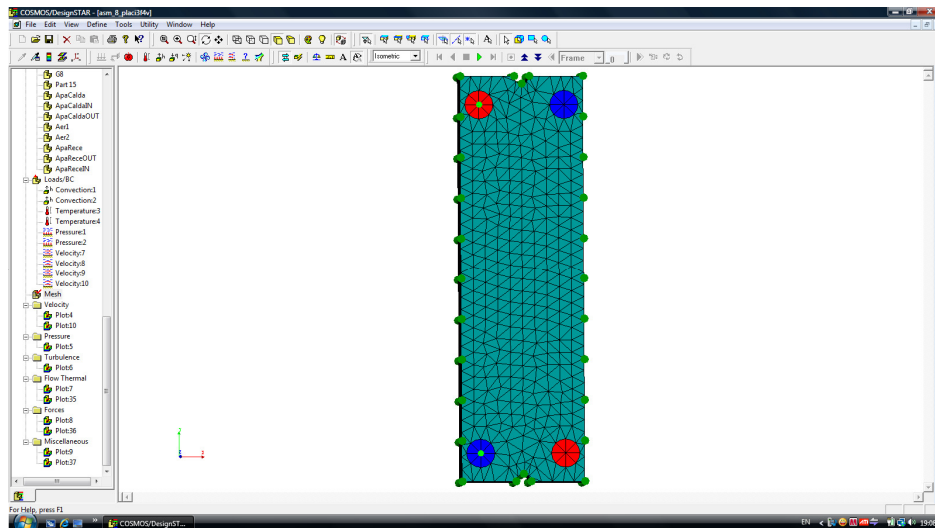


Fig.3. Mesh

The next boundary conditions are proposed:

Table 3

Inlet data				
No.	Fluid	Description	unit	Value
1	Hot Water inlet	Temperature – T_1'	$^{\circ}\text{C}$	53
2		Volume flowrate - \dot{V}_1	m^3/s	$9,524 \times 10^{-5}$
4	Cold Water inlet	Temperature – T_2'	$^{\circ}\text{C}$	8
5		Volume flowrate – \dot{V}_2	m^3/s	$8,163 \times 10^{-5}$
6	Hot water outlet	Static pressure	N/m^2	0
7	Cold water outlet	Static pressure	N/m^2	0

The numerical simulation of turbulent flow is modelled by K-epsilon turbulence model.

4. Results display

After the analysis was processed it can be visualized the results, under graphical form or numerical value. Analyze run for 100 iterations, in turbulence conditions. In figure 4 is presented the distribution of the nodal temperature on the heat exchange surface, after 50 iterations.

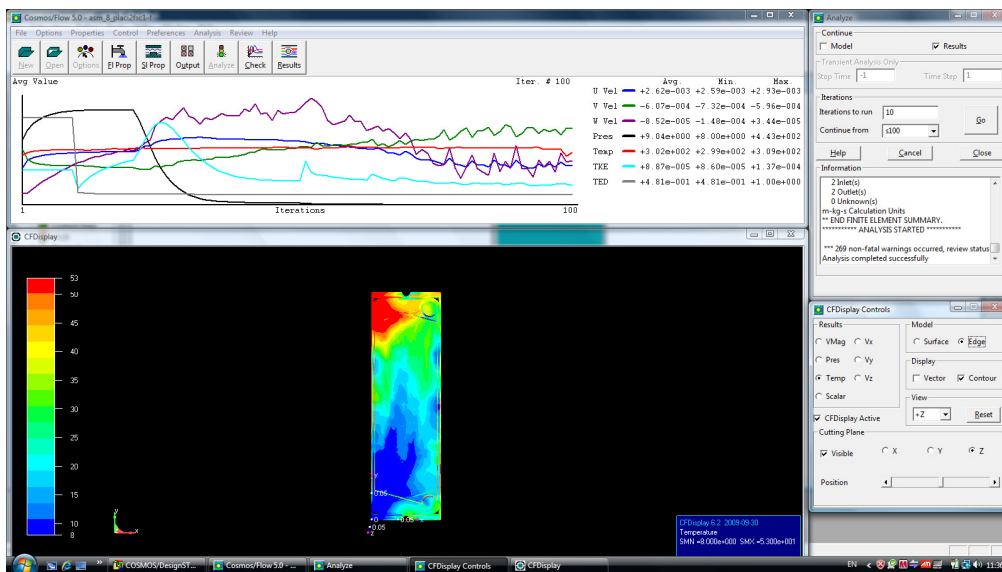


Fig.4.Distribution of temperature

The convergence of the average temperature, after 100 iterations, is shown in figure 5.

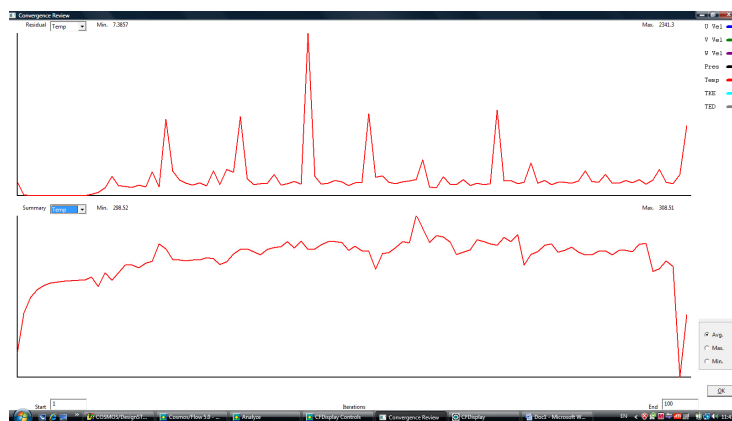


Fig.5. Convergence of average nodal temperature

The average temperature at the outlets of hot water and cold water are shown in table 4:

Table 4

Values of average temperatures determined				
No.	Fluid	Description	U.M.	Value
1	Hot Water outlet	Temperature - T_1	$^{\circ}\text{C}$	31,9
2	Cold Water outlet	Temperature - T_2	$^{\circ}\text{C}$	31,4

The overall heat transfer coefficient U is determined from heat transfer equation:

$$U = 1520 \text{ W/m}^2\text{K}. \quad (7)$$

5. Conclusions

Values obtained for the temperature, heat flux and overall heat transfer coefficient are in according with technical possibilities of studied heat exchanger.

The model developed using the program COSMOS/ Flow proves a very important tool for the study of heat transfer in a plate heat exchanger. In the sometime, the model is perfectible and offers good information about the behavior of counter-flow plate heat exchanger in different conditions of operation and it can be integrated in a more complex system.

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