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Plate Heat Exchanger Performance Optimization Using CFD Simulation

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***Abstract*— Earlier than this study Plate Heat Exchanger (PHE) is not widely used because of their inability to seal large gaskets between each of the plates. This problem has limited their use to small and low pressure applications. Computational Fluid Dynamics (CFD) software Gambit (pre-processor) and Fluent (solver) were used to modify the existing geometry to further improve their performance. The simulation result has clearly shown how the modification of the plate heat exchanger geometry affects its performance to an optimum level.**

Keywords*— *CFD, Performance, Geometry, optimum

I. INTRODUCTION

A heat exchanger is a device which is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. They are one of the most important and frequently used processes in engineering. The heat exchange mechanism depends on many other variables such as the heat transfer area, temperature difference, flow rates of the fluids, flow pattern, etc. Heat exchangers are key devices used in a wide variety of thermal applications in industries, including petroleum refining and petrochemical processing; in the food industry, for example, for pasteurization of milk; in the generation of steam for production of power and electricity; in air conditioning houses and vehicles; aircraft and space vehicles.[1] They are the workhorses of the entire field of heating, ventilating, air-conditioning and refrigeration. There are several different types of heat exchangers including shell-and-tube plate heat exchangers, double pipe heat exchangers and spiral tube heat exchangers. [2]

Several types of plate heat exchangers are available as described by [3], they can be distinguished on the basis of their specific structure and how the plates are attached together; the most common type is gasket plate heat exchanger, in which they consist of a pack of gaskets and corrugated metal plates pressed together with a frame. A gasket that seals around the plate prevents fluid mixing. It can also be used to create flow configurations such as series, parallel, and multi-pass arrangements by closing and opening ports at the four plate corners. The number of plates, their perforation, the type and position of the gaskets and the location of the inlet and outlet connections at the covers characterize the configuration of plate heat exchanger.[3]

The challenge has always been to find the optimum design with maximum thermal efficiency, lower pressure drop and fouling. By using CFD simulation, the performance of plate heat exchanger can be optimized.

II. DESIGN OF THE PLATE

The materials used in designing the PHE are contained in the tables 1 and 2:

Table 1: Plate Materials used in PHE

Material	Thermal conductivity (W/m^2)
Stainless steel (316)	16.5
Aluminum	205
Inconel 600	16
Incolay 825	12
Hastelloy C-276	10.6
Monel 400	66
Nickel 200	66
9/10 Cupronickel	52
70/30 Cupronickel	35

Table 2: Gasket Materials used in PHE

Gasket Material	Maximum operating temperature (ºC)	Application
Natural Rubber	70	Oxygenated solvent, acids, alcohols
SBR (styrene butadiene)	80	General-purpose aqueous, alkalies, acids and oxygenated solvent
Neoprene	70	Alcohols, alkalies, acids, aliphatic hydrocarbon solvents
Nitrile	100-140	Dairy, fruit juices, pharmaceutical and biochemical applications, oil, gasoline, animal and vegetable oils, alkalies.
Butyl	120-150	Alkalies, acids, animal and vegetable oils, phenols, and some esters.
Ethylene propylene rubber	140	Alkalies, oxygenated solvents
Silicon Rubber	140	General low temperature use, alcohols, sodium hypochlorite.
Flurinated Rubber	175	High temperature aqueous solutions, mineral oils and gasoline, organic solvents, animal and vegetable oils

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PLATE HEAT EXCHANGER PERFORMANCE

The six most important parameters as given by [4] are:

1. The amount of heat to be transferred (heat load).
2. The inlet and outlet temperatures on the primary and secondary sides.
3. The maximum allowable pressure drop on the primary and secondary sides.
4. The maximum operating temperature.
5. The maximum operating pressure.
6. The flow rate on the primary and secondary sides.

The amount of heat to be transferred is given in eqn 2.0:

$$Q = mc_p(T_1 - T_2) \quad --- \quad 2.0$$

Where:

Q is the heat transfer rate (W),
m is the mass flow rate (kg/s),

$$P = m\ x\ c_p\ x\ dT \quad --- \quad 2.1$$

$$P = U * A * LMTD --- \quad 2.2$$

Which are derived from the following equations:

$$m = \frac{P}{C_p\ x\ dT} \quad -- 2.3$$

$$dT = \frac{P}{m\ x\ c_p} \quad ----- \quad 2.4$$

Where;

P is the heat load (kW),
m is the mass flow (kg/s),
C_p is the Specific heat (KJ/kg-K) or (KJ/kg-K),
dT is the Difference between inlet and outlet temperatures on one side (K),
U is the total overall heat transfer coefficient (W/m^2 ^{0}C),
A is the heat transfer area (m^2),
LMTD is the log mean temperature difference (^{0}C).
Logarithmic Mean Temperature Difference:
Logarithmic mean temperature difference (LMTD) is the effective driving force in the heat exchanger.

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{ln \frac{\Delta T_1}{\Delta T_2}} --- 2.5$$

Here $\Delta T_1 = T_{h1} - T_{c2}$ & $\Delta T_2 =$

$$Pr = \frac{C_p \mu}{k} \quad --- \quad 2.6$$

I. Nusselt number:

The Nusselt number is important dimensionless parameters that represent the temperature gradient at a surface where heat transfer by convection is taking place. It is a function of Reynolds number, the Prandlt number, and if the fluid is still at hydrodynamic entrance region, the Nusselt number will also serve as a function of the length of the tube. However, if the flow is fully developed, the Nusselt number becomes constant. [5]

If the value of the Nusselt number is unity, then it is pure conduction. Higher values means that heat transfer is enhance by convection. A large Nusselt number means very efficient convection. For example a cold fluid flows over a hot surface; the first layer of the fluid (which sticks to the surface) gets heat from the surface by pure conduction. It then gives this newly acquired energy to all of the fluid molecules that it comes in contact with as they pass by it (this is convection).[6]

$$Nu = 0.26(Re)^{0.65} Pr^{0.4} \quad --- \quad 2.7$$

II. The heat transfer coefficient between the fluid and the heat transfer surface is given by:

$$h_p = \frac{Nu(k)}{D_e} \quad ----- \quad 2.8$$

III. The friction factor is defined by the following equation

$$f = \frac{k_p}{R_e^m} \quad --------- \quad 2.9$$

IV. Pressure Drop:

Pressure drop (Δp) is in direct relationship to the size of the plate heat exchanger. If it is possible to increase the allowable pressure drop, and incidentally accept higher pumping costs, then the heat exchanger will be smaller and less expensive. [7]. Total pressure drop for both sides is the sum of the channel pressure drop and port pressure drop.

$$\Delta P_t = \Delta P_c + \Delta P_p \quad ----$$

The pressure drop of channel is given by;

$$\Delta P_C = 4f \frac{L_v N_p}{D_e} * \frac{G_c^2}{2\rho} \quad ---- \quad 2.9.2$$

The pressure drop in port ducts is given by 2.9.3;

$$\Delta P_p = 1.4 N_p \frac{G_c^2}{2\rho} \quad --------- 2.9.3$$

Where:

is the friction factor,
L$_v$ is the vertical distance between centers of ports (m),
N$_p$ is the number of passes,
D$_e$ is the channel equivalent diameter (m).

Overall heat transfer coefficient (U)

Overall heat transfer coefficient **(U)** is a measure of the resistance to heat flow, made up of the resistances caused by the plate material, amount of fouling, nature of the fluids and type of exchanger used. Overall heat transfer coefficient is expressed as W/m^2 °C or kcal/h, m^2 °C.

$$\frac{1}{U} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{t}{k} \quad ----2.9.4$$

Where:

h$_h$ is the heat transfer coefficient between the warm medium and the heat transfer surface (W/m$^{2\,0}$C),
h$_c$ is the heat transfer coefficient between the heat transfer surface and the cold medium (W/m$^{2\,0}$C),
t is the thickness of the heat transfer surface (m),
k is the thermal conductivity (W/m^{0}C).

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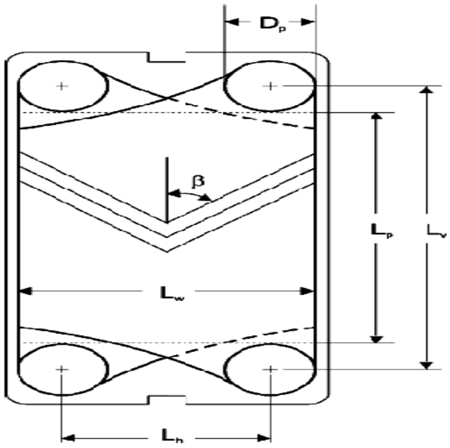

Fig 1; Basic geometry of Plate

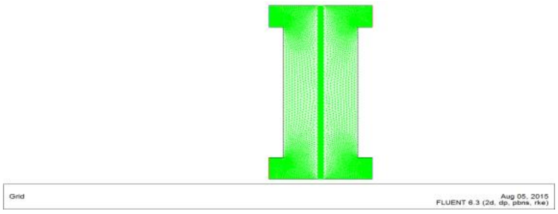

Fig 2: Grids showing three meshed plates carrying hot and cold fluid without mixing the two streams

III. MESHING THE PLATE GEOMETRY USING GAMBIT

Table 3: Specifying boundary type's gambit

EDGES POSITION	NAME	TYPE
EDGE 2	COLD FLUID INLET	VELOCITY INLET
EDGE 20	COLD FLUID OUTLET	OUTFLOW
EDGE 4	HOT FLUID INLET	VELOCITY INLET
EDGE 21	HOT FLUID OUTLET	OUTFLOW
EDGES 17, 18, 19	WALL 1: WALL AT THE COOL PLATE	WALL
EDGES 22, 23, 24	WALL 2: WALL AT THE HOT PLATE	WALL
EDGES 3, 25, 27	WALL 3: WALL AT THE OTHER END OF COOL PLATE	WALL
EDGES 1, 26, 28	WALL 4: WALL AT THE OTHER END OF HOT PLATE	WALL
EDGES 26, 27, 29, 30	WALL 5: INTERMEDIATE WALL	WALL

Table 4.Specifying continuum boundary types in gambits

FACES POSITION	NAME	TYPE
FACE 1	HOT FLUID FLOW	FLUID
FACE 3	COLD FLUID FLOW	FLUID
FACE 4	MIDDLE PLATE	SOLID

IV. SETTING THE BOUNDARY CONDITIONS

Table 5: Boundary condition for the velocity inlet at cold stream

Velocity specification method	**Magnitude, Normal to boundary**
Reference frame	Absolute
Velocity magnitude	0.5 m/s
Specification method	K and Epsilon
Turbulent kinetic energy (m^2/s^2)	1
Turbulent dissipation rate (m^2/s^3)	1
Temperature	298

Table 6: Boundary condition for the velocity inlet for hot stream

Velocity specification	**Magnitude, Normal to boundary**
Reference frame	Absolute
Velocity magnitude	0.5 m/s
Specification method	K and Epsilon
Turbulent kinetic energy	1
Turbulent dissipation rate	1
Temperature	373

Effective area of plate is given by:

$$A = L * W \quad ---------- \; 2.6$$

Where:
L is the length of the plate
W is the width of the plate.

$$Number\ of\ plate = \frac{total\ heat\ transfer\ area}{effective\ area} \quad --- \; 2.7$$

I. Number of channels is given by:

$$N_C = \frac{N-1}{2} \quad -------- \; -2.8$$

Where:
N is the number of plate.
II. The channel cross sectional area is:

$$A_c = b * W \quad -------- \; 2.9$$

Where:
A_c is the channel cross sectional area
b is the plate spacing.
III. Channel equivalent diameter is:

$$D_e = 2 * b \quad --------- \; 3.0$$

Where: D_e is equivalent diameter

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V. RESULTS AND DISCUSSION
SIMULATION RESULTS

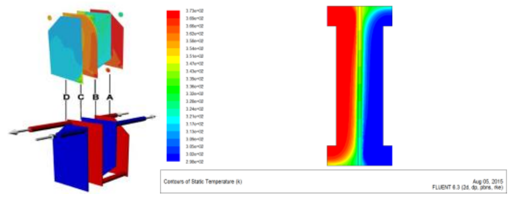

Fig 4: Contours of Static Temperature (K) showing midplane temperature distribution for the channels of a series flow arrangement: 2D model results; Contours of temperature distribution within the plates showing the simulation results at which heat will be transferred from one medium to another, where the minimum temperature for cold stream was 298k and the maximum temperature was 321k and for the hot stream minimum temperature was 354k and the maximum was 373k respectively.

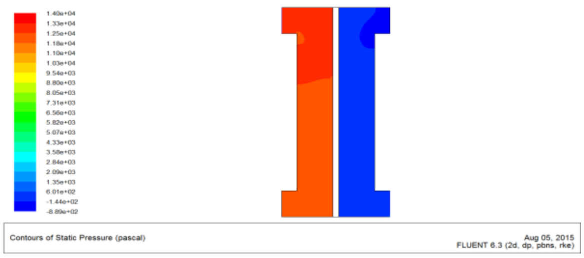

Fig 5: Contours of Static Pressure.

Contours of static pressure showing the simulation results for pressure drop within Midplane for the channels of a series flow arrangement: 2D model results of the plate from inlet to the outlet port, the results shows variation of pressure within the plates, the pressure decreases due to head loss. The pressure of the fluid decreases due to energy losses and the magnitude of velocity near the walls are smaller compared to areas far from the walls due to shear stresses exerted from the wall to the fluid.

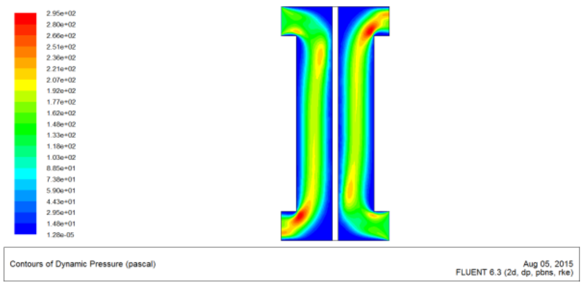

Fig 6: Contours of Dynamic Pressure

Contours of dynamic pressure distribution showing the variation at the entrance of both the hot fluid and cold fluid and the maximum dynamic pressure occurs along the plate at about 295 Pascal while the minimum dynamic pressure is about 128 Pascal

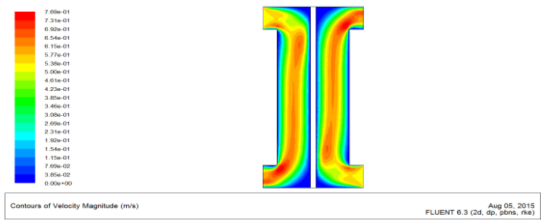

Fig 7: Contours of Velocity Magnitude in (m/s)

Contours of velocity distribution showing the velocity variation at the entrance and within the Midplane for the channels of a series flow arrangement: 2D model results of the plates near the walls showing velocity magnitudes are low due to shear stresses

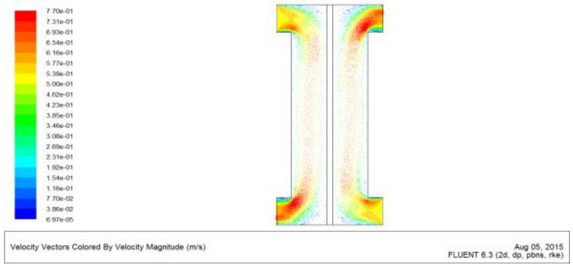

Fig 8: Contours of Velocity Vectors in (m/s)

Contours of velocity vectors distribution showing the velocity variation at the entrance and within the plates, near the walls velocity magnitudes are low in the series arrangement as clearly shown in the figure; this is also similar to the behavior in fig 7 influence by shear stresses.

VI. CONCLUSSION

Using a CFD tool it was possible to build a virtual prototype of a PHE with four channels and flat plates. The simulation results included outlet temperatures, heat load, as well as the 2D temperature and velocity distribution.

The plate heat exchanger was successfully modeled using Gambit and Fluent. The simulation was carried out by meshing the channels of the plate heat exchanger with counter flow configuration, the simulation setup was performed to study water-to-water heat transfer in the plate heat exchanger, After convergence the results show that the heat rejection rate of water to water in the plate heat exchanger were obtained where the maximum temperature is 373k and the minimum is 298k, the pressure of the fluid decreases due to energy losses and the magnitude of velocity near the walls is smaller compared to areas far from the walls due to shear stresses exerted from the wall to the fluid. The detailed results from the CFD model allow the analysis of velocity and temperature distribution inside the PHE. High temperature regions and stagnation areas could be observed, indicating regions susceptible to fouling. Examples of temperature and velocity distribution were presented in Figs. 4, 7 and 8 for a series arrangement. Heat transfer of the plate heat exchanger was also studied using CFD software i.e. Gambit and Fluent to produce numerical results. By using CFD method we can design a shape for the plate heat exchanger that has less pressure drop and also we can design a more effective plate that will give high rate of heat transfer, therefore FLUENT software is suitable to analyze the behavior of the flow in a process.

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