NUMERICAL STUDY OF FLOW OVER PLATES AND GASKET HEAT EXCHANGER

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Abstract Plate heat exchangers achieve higher compactness than multitubular exchangers. The plate design optimizes heat transfer by providing a large and total compact surface in which heat can be transferred from one fluid to another. We seek to improve the heat transfer between these plates however reducing the pressure drop during flow of fluid in the device. Numerical simulation makes possible to calculate the thermal flow of fluid in the heat exchanger; pressure drop and performance of the heat exchanger while varying the flow velocities as well as the space between plates. Results have led to saying that an increase in the heat flux exchanged between two plates for different thicknesses of gasket is observed when plates are tightened, they allow a wider heat exchange by increasing space between plates, this increase is accompanied by increased pressure drop.

*Key Words***:** Heat Transfer, plate an gasket heat exchanger, flow rate between plate, heat flow, pressure drop.

1. Introduction

Today with a permanent growth in energy prices, energy management has become a major issue in all areas of activity. For energy professionals, the first challenge is to design energy systems and processes with better efficiencies.

One of the most common modes of energy exchange is heat transfer. Which is a phenomenon found in many sectors of the industry. Engineers and technicians are faced with this kind of problem and try to maximize or minimize this phenomenon according to the needs of the industry and in order to save this expensive energy. As a result, heat transfers have an often essential role, both in field of pure and technological sciences applications. Example: (exchanger, heat engines, thermal insulation, thermal insulation, etc.). Knowledge of the physical laws that govern these modes of heat transfer is essential and very important, because they allow us to control the way and the quality of this heat flow.

Heat exchangers are indeed the obligatory passage point for all energy flows, and any effort to control this energy must be based on well-designed and well-used heat exchangers.

Heat exchanger is a thermal device of great importance in thermal and energy installations. At least one heat exchanger is encountered in a thermal installation. These include energy production; food, chemical industries; electronics field; environmental technology; industry; air conditioning and refrigeration [1].

Unlike other thermal devices, the heat exchanger contains no moving mechanical parts. The calculation of this device is very complex, we must know exactly: its geometry (exchange surface and cross-section of fluids), its thermophysical characteristics, flow rates of fluids, inlet temperatures of fluids, etc.

2. Mathematical model

During general studies relating to heat exchanger installations, the need to assess often arises [2]:

- The heat exchange coefficients,
- Exchange areas,
- Pressure losses,
- Temperature differences between the heat transfer fluids.

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2.1. Load losses

In general, it is considered that dimensions of pipes are large enough so that the pressure drops are negligible compared to those of the plates. Pressure drop of an isothermal flowing fluid in a plate (ΔP) between inlet and outlet manifolds, depends on many parameters such as density, viscosity, fluid speed, as well as geometry of the plates .

$$\Delta P = \Delta P_{collecteur} + 4.f.\frac{\rho U^2}{2}\frac{L}{D_h} \quad (1)$$

Friction coefficient (f) is generally expressed as follows

$$f = \frac{k}{R_e{}^x} \quad (2)$$

K and x depend on the flow regime and the geometry of plate. The exponent x of the Reynolds number varies from 0.1 to 0.4 [3].

2.2. Number of NUT transfer units

The dimensionless grouping h.S / C_{min} represents what is called the number of transfer units denoted NUT.

$$NUT = \frac{H.A}{(m.C)_{min}} = \frac{1}{R_{eq}.(m.C)_{min}} \quad (3)$$

2.3. Efficiency of a heat exchanger

The following relations give the relation relating the exchange efficiency to the number of transfer units for two different operating modes of the exchangers for counter-current operation:

$$E = \frac{1-e^{[-(1-R_f)NUT]}}{1-R_f e^{[-(1-R_f)NUT]}} \quad (4)$$

2.4. Determination of the thermal convection coefficient h [4].

The problem of convection is in fact to determine according to the conditions of flow of fluid, the geometrical characteristics of the walls and the possible changes of state of the fluid. We therefore define three dimensionless numbers:

- Nusselt number given in the form: $N_U = \frac{h.D_h}{\lambda}$
- Reynolds number given in the form: $R_e = \frac{\rho.U.D_h}{\mu}$
- Prandt number given in the form: $P_r = \frac{\mu\ C_p}{\lambda}$

2.5. Heat flow exchanged between plates

In the thermal study of different types of heat exchangers, we often use the type equation

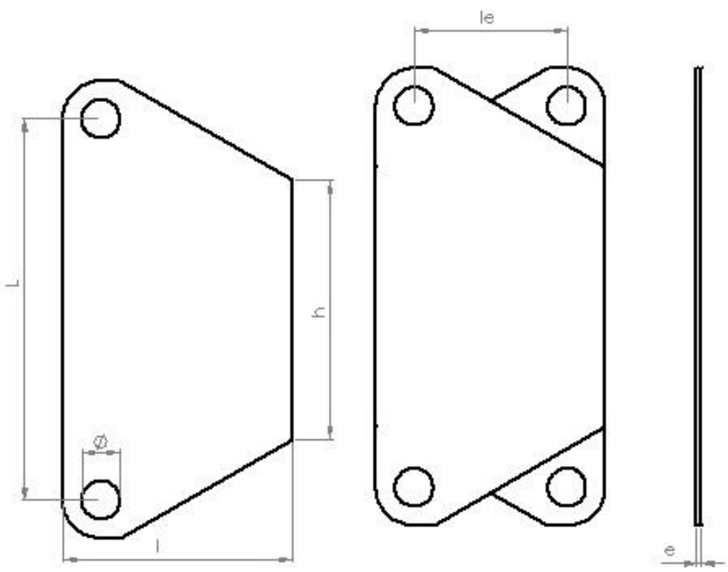
$$\Phi = h.A(T_1 - T_2) \quad (5)$$

The heat flux exchanged between plates is expressed as [5].

$$\phi = h.A.\Delta TLM \quad (6)$$

3. Results and discussions

The present study concerns the study of heat transfer through plates of a plate and gaskets heat exchanger. The flow is assumed to be laminar three-dimensional, compressible (water), cooled between the plates by changing the thicknesses (distance between plates) according to the table below.



L (mm)	l (mm)	h (mm)	le (mm)	∅ (mm)
1000	600 mm	680	400	100

e (mm)	1	3	4	6	8	10

Figure 1 Detailed diagram of geometry of the heat exchanger plate.

3.1. Structure of flow between two adjacent plates of a heat exchanger

The plate design optimizes heat transfer by providing a large and compact total area through which heat can be extracted from one liquid or gas to another.

The heat transfer area of the plates is between two superimposed plates with opposite herringbone patterns, the helical flow and high turbulence result in high transfer coefficients and efficient self-cleaning.

The plate distribution area ensures uniform fluid flow over the entire plate to maximize heat transfer capacity. Optimized flow distribution also reduces fouling and uneven temperature zones, while keeping performance levels high, without unnecessary energy loss, maintenance costs or unplanned downtime.

Gaskets are key components in the performance of heat exchangers. The gasket and plate are designed to ensure optimum sealing. Each of them is adapted to the mission of the heat exchanger. The right profile, width, thickness, type of polymer and compound make all the difference to avoid the risk of premature leakage or damage to seals or plates.

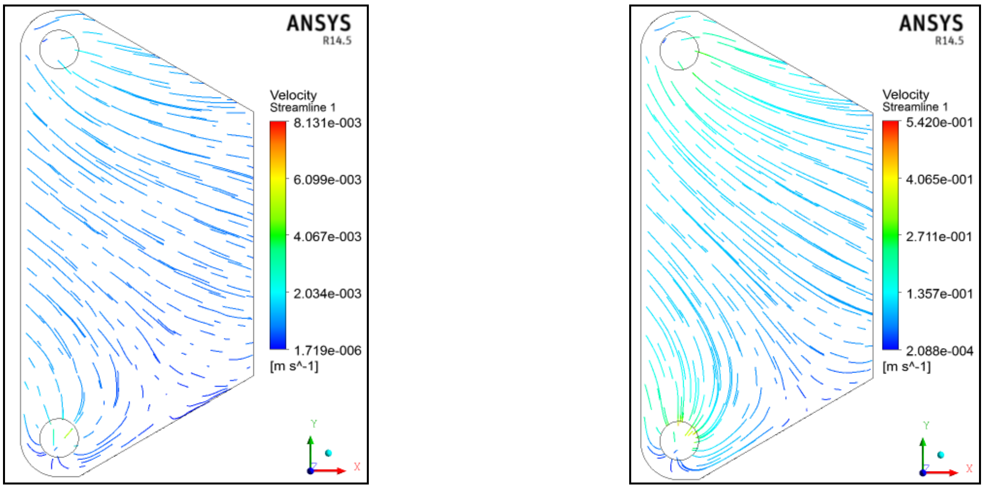


Figure 2Structure of the flow on a simple plate of a heat exchanger

3.2. Initial conditions

The CFX 14.0 software provides us with a subroutine in which it is possible to introduce an initial distribution for each of the flow variables.

- The fluid is water.
- The density: $\rho = 999.7$ [kg.m^{-3}].
- The dynamic viscosity: $\mu = 8.899 * 10^{-4}$ [kg / m / s].
- The reference pressure: 0 [atm].
- The physical and transport characteristics of water are defined by the calculation code.

3.3. Boundary conditions

Boundary conditions are defined and must be applied to all the the ends of the computation field namely:

- Fluid inlet (INLET): speed at the inlet of the plate is varied from 0.0006, 0.001,0.01, 0.05 and 0.1 [m/s]
- Fluid outlet (OUTLET) : $\frac{\partial \varphi}{\ } = 0$; p = 0 [atm]
- Walls: A non-adherence condition is considered for the front and rear plate or the overall heat transfer coefficient is defined as h = 379.44 [W/m^2 K]

$$u = v = w = 0$$

- Adiabatic wall: The gasket surfaces are defined as adiabatic walls.

3.4. Influence of flow velocity

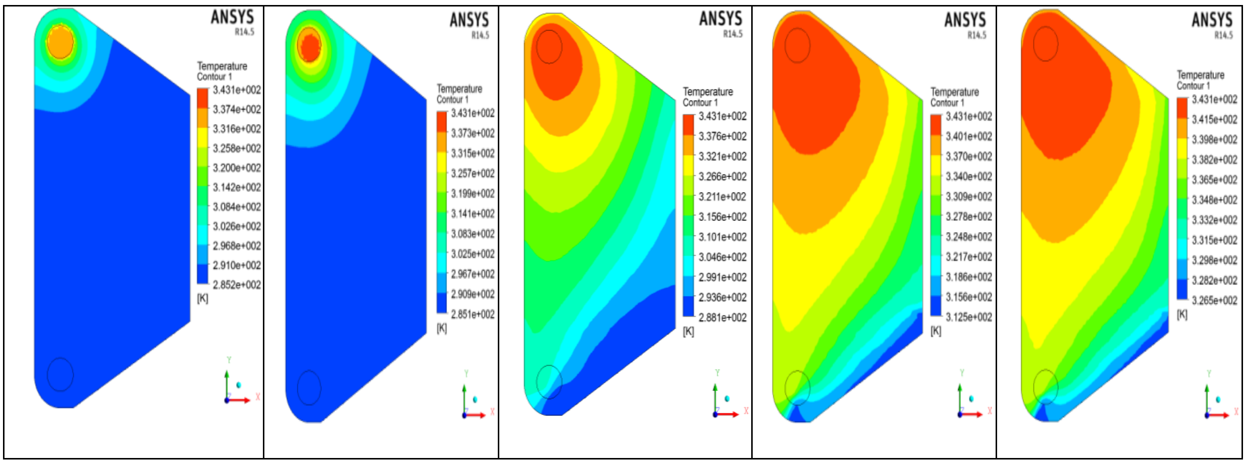


Figure 3 Variation of temperature profile between two plates of 3 mm of thickness for different speeds

The evolution of the flow velocity profile between two plates of 3 mm thickness varies significantly depending on the velocity in m /s. This development calls for a temperature gradient varying between 343 K at the inlet to 326 K at the outlet which allows good cooling of the fluid between plates.
This finding is reflected in an increase in the heat flux exchanged between the two plates for different thicknesses of the gasket, when the plates are tightened, they allow a wider heat exchange than by increasing the inter-plate space.

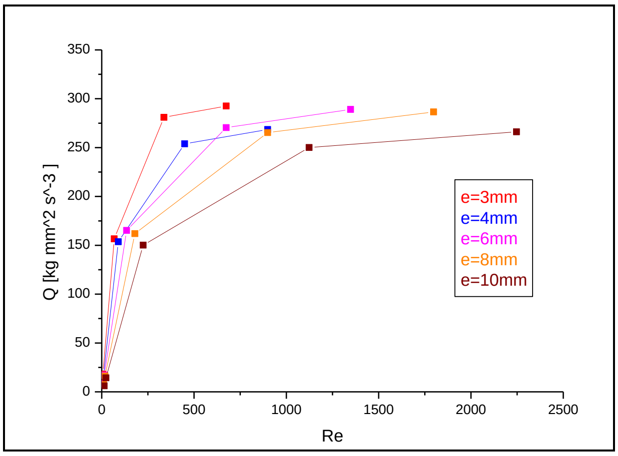


Figure 4 Variation of heat flow with Reynolds nubmer for different thicknesses of gasket.

The heat exchange between clamped plates is clearly recorded on the different temperature profiles by varying the thickness of the gasket for the same speed. The temperature gradient varies from 343 K at the entry of the plate up to 323 K at the exit for a thickness of 3 mm of gasket when the difference is from 343 K at the entry to 326 K at the exit for thicknesses varying from 6 to 10 mm.

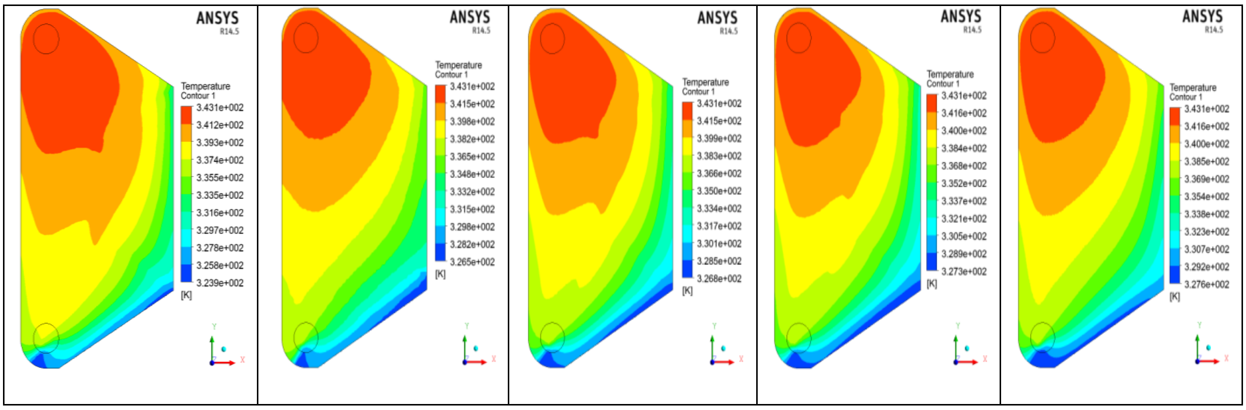


Figure 5Variation of heat flux with Re for different thicknesses of gasket

3.5. Influence of the thickness of gasket

Figure 6 shows that the pressure drop for a large thickness of gasket takes maximum values and it is slightly lower for smaller values of the space between the plates for the same flow speed. This is mainly due to the shear of the flow between surfaces of the plates

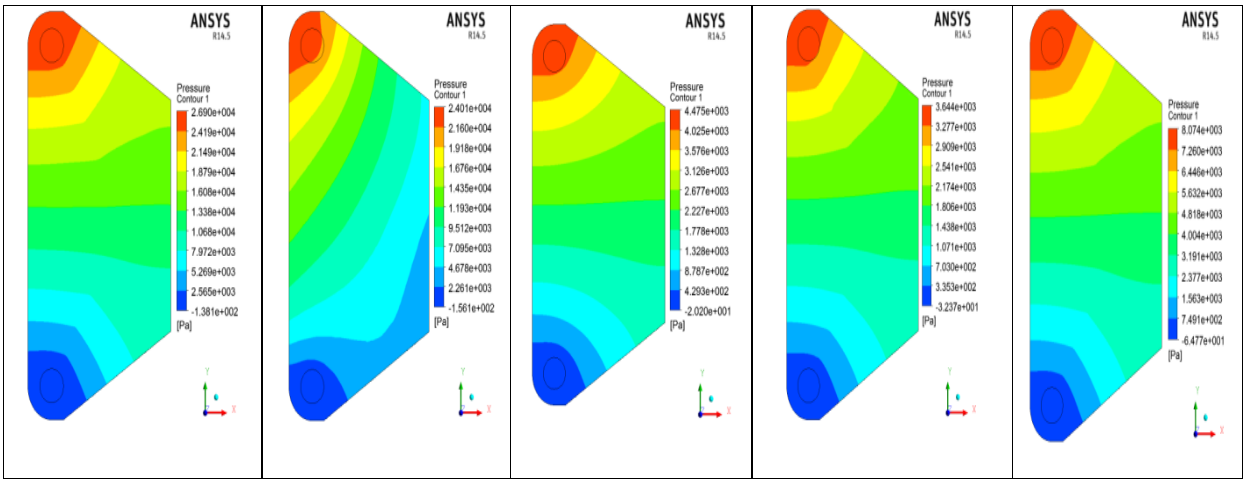
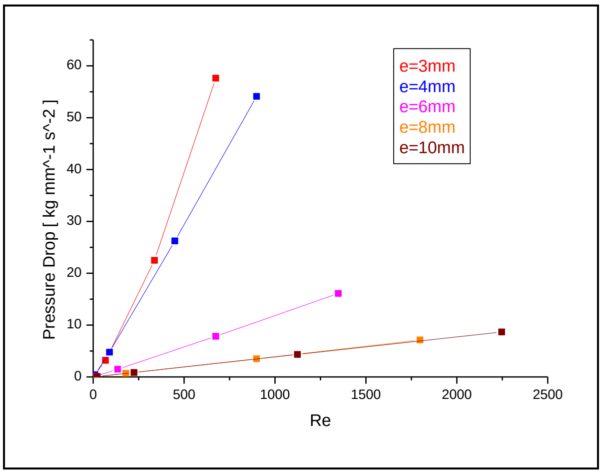


Figure 6Variation of the pressure profile between two plates of different thicknesses of gasket for a speed of 0.05 m/s.

Usually, increased heat transfer between plates of a heat exchanger generates pressure losses. The evolution of the pressure drop along the plate is shown in figure 7.



.Figure 7 Variation of pressure drop with Re for different thicknesses of gasket.

The variation in pressure drop tends to increase with increasing Reynolds number. Each time the joint thickness decreases, the pressure drop value increases.

3.6. Heat exchanger performance

In compact surface terminologies, the heat transfer coefficient and the pressure drop are generally expressed in terms of dimensionless factors. Two well-known dimensionless parameters, the Colburn factor, j, and the Fanning friction factor, f, were used to describe the heat transfer performance and pressure drop characteristics,

$$j = \frac{h}{\rho u C_p} Pr^{2/3} \quad (7)$$

$$f = \frac{2\Delta P}{\rho u^2} \frac{D_h}{L} \quad (8)$$

where Dh = 2 * e and e represents the thickness of gasket.

The variation of thermal performance for the fluid circulating in a plate and gasket heat exchanger is shown in figure 8. In this figure, the performance of the exchanger tends to increase by decreasing the thickness of gasket for different speeds corresponding to the different numbers of Re.

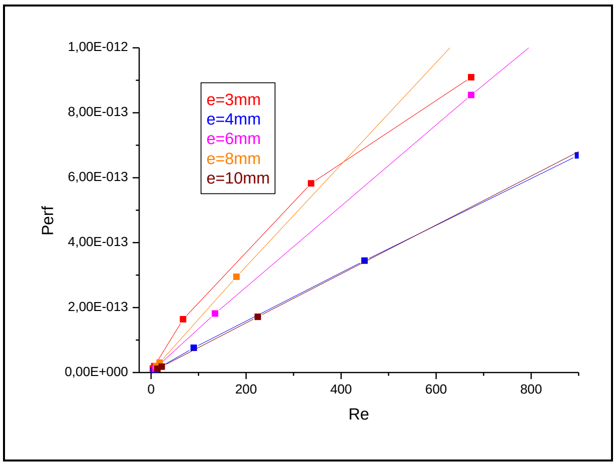


Figure 8Variation of performance of the heat exchanger with Re for different thicknesses

Conclusions

The heat exchanger is defined according to the role played (heating, cooling) of the nature of the fluids (vapor or liquid, risk of corrosion) and required flow rates.

We are looking to optimize investment and operating costs. It is therefore necessary to study together the purely thermal aspects (transfer coefficient, exchange surface) and the aspects related to fluid mechanics.

In this study, unlike the classic method of calculating and optimizing heat exchangers, we presented results obtained using CFX 14.0 software. These results interpret in a simple way the various variations that fluid carried inside plates undergoes as well as its heat exchange with the second fluid at different speeds and different thicknesses of gasket.

In fact, the heat fluxes on the plates of a plate and gasket heat exchanger are stronger when flow velocities are higher. So if the fluids have high flow rates, it is possible to be satisfied with a smaller exchange surface which reduces investment. On the other hand, pressure drops will be higher, which will require more powerful pumps (increase in investment cost) and higher energy expenditure (increase in operating cost).

The case-by-case optimization study will provide the answer to determine the most economical choice.

Nomenclature

Symbols

A or S exchange area, m²

Cp : Specific heat at constant pressure J/kg K

Dh Average hydraulic diameter m

*Δ*TLM Average log temperature difference °C

ΔP Pressure drop Pa

E Efficiency

h Heat transfer coefficient W/ m^2 K

L Plate length m

ṁ.Cp Heat flow rate of the fluid kJ/h °C

NUT Number of transfer units

P Pressure, Pa

T Temperature, K

u, v w : Speed components along the three axes m/s

Dimensionless Numbers

J: Colburn factor

f : friction factor

Re Reynolds number

Nu Nusselt number

Pr Number of Prandtl

Greek letters

Φ Thermal power kW

ρ Density, kgm^{-3}

λ Thermal conductivity $W\cdot m^{-1}\cdot K^{-1}$

μ Dynamic viscosity kg /m /s

Clues

1 temperature at the inlet of the exchanger

2 temperature at the outlet of the

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