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Design of Plate Heat Exchanger (PHE)

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ABSTRACT

This paper presents a part of a complex research program-baseddesign of plates for heat exchangers and shows the results of theoretical research conducted for designing plate heat exchanger. The main purposes of this research is to develop the suitable methodology for designing the plate type heat exchanger.

Key words:*PHE, Heat transfer.*

1. INTRODUCTION:

For being compact, easy to clean, efficient and very flexible, the gasketed plate heat exchanger (PHE) is widely employed in the chemical, food and pharmaceutical process industries. The PHE consists of a pack of gasketed corrugated metal plates, pressed together in a frame (see Fig. 1). The gaskets on the corners of the plates form a series of parallel flow channels, where the fluids flow alternately and exchange heat through the thin metal plates. The gasket design and the closed ports of the plates determine the fluid flow distribution, which can be parallel, series or any of their various possible combinations.(see Fig. 2) The number of plates, flow distribution, type of gaskets and the fluid feed locations characterize the exchanger configuration.

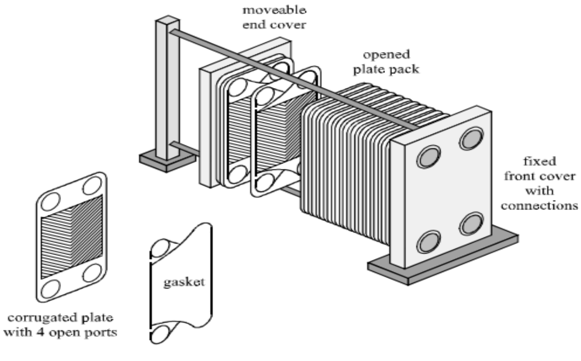

Figure.1

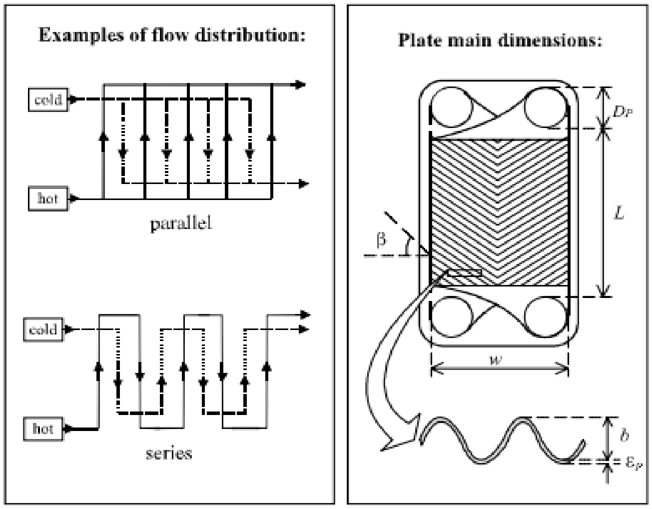

Figure.2

2. LITERATURE SURVEY AND FINDINGS

The simplified thermal modelling of a PHE in steady state yields a linear system of first order ordinary differential equations, comprising the energy balance for each channel and the required boundary conditions. The main assumptions are as follows: plug-flow inside the channels, constant overall heat transfer coefficient along the exchanger, uniform distribution of flow in the channels, no heat loss and no heat exchange in the flow direction. This basic model was presented by McKillop & Dunkley (1960) for 3 different configurations and also by Masubuchi& Ito (1977), where the dynamic responses of some usual configurations were studied. In both works, the Runge-Kutta-Gill integration method was used to solve the system of equations.

The integration is non-trivial because the boundary conditions are defined at different extremes of the channel. Approximate solutions were developed by Settari& Venart (1972) in polynomial form, and by Zaleski &Klepacka (1992) in exponential form. Both methods lead to good approximations of the exact solution, but they may not be reliable when there is a large difference between fluid heat capacities. Kandlikar& Shah (1989b) developed a method to calculate an approximated thermal effectiveness for large exchangers, where the effects of the end plates and of the changes of passes can be neglected. In this case, the exchanger is divided into a group of simpler exchangers that are interconnected, with known effectiveness. The analytical solution of the system of equations in matrix form was studied by Zaleski &Jarzebski (1973) and Zaleski (1984) for exchangers with series and parallel arrangements. This solution method may lead to numerical problems on the calculation of eigenvalues and eigenvectors, and it is not recommended for large sized exchangers. Kandlikar& Shah (1989a) and Georgiadis et al. (1998) used the finite difference method for the simulation of PHEs. Kandlikar& Shah (1989a) simulated and compared several configurations. It was verified that higher effectiveness is achieved when the exchanger is symmetrical, with the same numbers of passes for both streams, because the channels that are next to the changes of passes as well as the end channels have a lower effectiveness. However, when the fluids have very different flow rates or heat capacities, a non-symmetrical configuration must be used. In such cases, there is no rigorous design method to select the best configuration, which is made by comparison among the usual configurations from thermal effectiveness and pressure drop viewpoints. Georgiadis et al.

(1998) presented a detailed modeling of a PHE used for milk pasteurization that couples the dynamic thermal model with the protein-fouling model. Three different configurations were compared and the reduction of the overall heat transfer coefficient, caused by the protein adhesion on the plates, was studied. The model was solved with the finite difference method, implemented in the software gPROMS (Process System Enterprise, 2001).

3. DESIGN OF HEAT EXCHANGER:

3.1 Basic equations for the design of a plate heat exchanger

The methodology employed for the design of a PHE is the same as for the design of a tubular heat exchanger. The equations given in the present chapter are appropriate for the chevron type plates that are used in most industrial applications.

3.2. Parameters of a chevron plate

The main dimensions of a *chevron* plate are shown in Figure 3. The corrugation angle, β, usually varies between extremes of 25° and 65° and is largely responsible for the pressure drop and heat transfer in the channels.

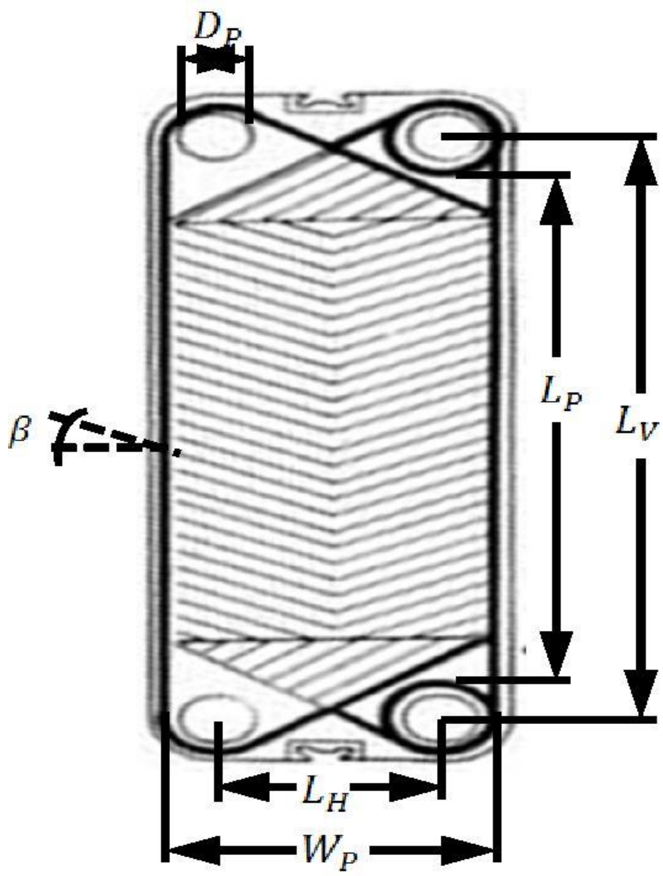

Figure 3 Chevron Plate.

Parameters of a chevron plate.

The corrugations must be taken into account in calculating the total heat transfer area of a plate (effective heat transfer area):

$$A_P = \Phi . W_P . L_P$$

Where,

A_P= plate effective heat transfer area

Φ = plate area enlargement factor (range between 1.15 and 1.25)

W_P = plate width

L_P= plate length

The enlargement factor of the plate is the ratio between the plate effective heat transfer area, A_P and the designed area (product of length and width WP*LP), and lies between 1.15 and 1.25. The plate length L_P and the plate width W_P can be estimated by the orifices distances. L_V, L_H, and the port diameter Dp are given by below equation.

$$L_P \approx L_V - Dp$$

$$W_P \approx L_H + Dp$$

For the effective heat transfer area, the hydraulic diameter of the channel is given by the equivalent diameter, De, which is given by:

$$De = 2b\Phi$$

where b is the channel average thickness.

3.3 Heat transfer in the plates

The heat transfer area is expressed as the global design equation:

$$Q = UA\Delta TM$$

where U is the overall heat transfer coefficient, A is the total area of heat transfer and ΔTM is the effective mean temperature difference, which is a function of the inlet and outlet fluid temperatures, the specific heat, and the configuration of the exchanger. The total area of heat transfer can be given by:

$$A=N_PA_P$$

where N_P is the number of plates. The end plates, which do not exchange heat, are not taken into account in determining the area. The inner plates are usually called thermal plates in order to distinguish them from the adiabatic end plates. The overall heat transfer coefficient can be determined by:

$$1/U=1/hhot+tP/kP+1/hcold$$

where

hhot = convective heat transfer coefficient of the hot fluid

hcold = convective heat transfer coefficient of the cold fluid

tP = plate thickness

kP= plate thermal conductivity

Rf,hot = fouling factor of the hot fluid

Rf,cold = fouling factor of the cold fluid

The convective heat transfer coefficient, h, depends on the fluid properties, fluid velocity, and plate geometry.

3.4 Design methods

There are two main approaches used in the design of PHEs, namely the log-mean temperature difference and the thermal effectiveness methods. For the first method, the rate of heat transfer is given by:

$$Q=UA(F\Delta Tlm)$$

where ΔTlm is the log-mean temperature difference, given by below formula and F is the log-mean temperature difference correction factor.

$$\Delta Tlm=\Delta T1-\Delta T2/\ln(\Delta T1/\Delta T2)$$

Where

$\Delta T1=\{Thot,in-Tcold,out\}$ if countercurrent $\{Thot,in-Tcold,in\}$ if concurrent

$\Delta T2=\{Thot,out-Tcold,in\}$ if countercurrent $\{Thot,out-Tcold,out\}$ if concurrent

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The correction factor is a function of the heat exchanger configuration and the dimensionless parameters Rand PC. For purely countercurrent or concurrent (single-pass) arrangements, the correction factor is equal to one, while for multi-pass arrangements, it is always less than one. However, because the end channels of the PHE only exchange heat with one adjacent channel, different to the inner channels that exchange heat with two adjacent channels, purely countercurrent or concurrent flow is only achieved in two extreme situations. These are:

1. when the PHE has only one thermal plate, so that only two channels are formed by the end plates and the thermal plate, with each stream flowing through one channel;
2. when the number of thermal plates is sufficiently large that the edge effect can be neglected.

The dimensional parameters R, PC are defined as:

R=Thot,in−Thot,outTcold,out−Tcold,in=(M˙cp)cold(M˙cp)hotR=Thot,in−Thot,outTcold,out−Tcold,in=(M˙cp)cold(M˙cp)hotE23

PC=Tcold,out−Tcold,inThot,in−Tcold,in=ΔTcoldΔTmaxPC=Tcold,out−Tcold,inThot,in−Tcold,in=ΔTcoldΔTmaxE24

The second method provides a definition of heat exchanger effectiveness in terms of the ratio between the actual heat transfer and the maximum possible heat transfer, as shown in below formula:

E=QQmax

The actual heat transfer can be achieved by an energy balance:

Q=(M˙cp)hot(Thot,in−Thot,out)

Q=(M˙cp)cold(Tcold,out−Tcold,in)

Thermodynamically, Qmax represents the heat transfer that would be obtained in a pure countercurrent heat exchanger with infinite area. This can be expressed by:

Qmax=(M˙cp)minΔTmax

Using above equation, the PHE effectiveness can be calculated as the ratio of temperatures:

E={ ΔThot/ΔTmax} if R>1 {ΔTcoldΔTmax} if R<1

The effectiveness depends on the PHE configuration, the heat capacity rate ratio (R), and the number of transfer units (NTU). The NTU is a dimensionless parameter that can be considered as a factor for the size of the heat exchanger, defined as:

$$NTU=UA(Mcp)min$$

3.6 Pressure drop in a plate heat exchanger

The pressure drop is an important parameter that needs to be considered in the design and optimization of a plate heat exchanger. In any process, it should be kept as close as possible to the design value, with a tolerance range established according to the available pumping power. In a PHE, the pressure drop is the sum of three contributions:

a. Pressure drop across the channels of the corrugated plates.

b. Pressure drop due to the elevation change (due to gravity).

c. Pressure drop associated with the distribution ducts.

The pressure drop in the manifolds and ports should be kept as low as possible, because it is a waste of energy, has no influence on the heat transfer process, and can decrease the uniformity of the flow distribution in the channels. It is recommended to keep this loss lower than 10% of the available pressure drop, although in some cases it can exceed 30%.

$$\Delta P=2fLVPGC_2\rho De+1/4GP22\rho+\rho gLV$$

where f is the Fanning factor, given by above equation, PP is the number of passes and GP is the fluid mass velocity in the port, given by the ratio of the mass flow, $M^{\cdot}$, and the flow cross-sectional area, $\pi DP2/4$.

$$GP=4M^{\cdot}\pi DP_2$$

$$f=KpRem$$

The values for Kp and m is function of the Reynolds number for some β values.

3.7. Experimental heat transfer and friction correlations for the chevron plate PHE

Due to the wide range of plate designs, there are various parameters and correlations available for calculations of heat transfer and pressure drop. Despite extensive research, there is still no generalized model. There are only certain specific correlations for features such as flow patterns, parameters of the plates, and fluid viscosity, with each correlation being limited to its application range. In this chapter, the correlation is used.

$$Nu=Ch(Re)n(Pr)1/3(\mu\mu w)0,17$$

where μw is the viscosity evaluated at the wall temperature and the dimensionless parameters Nusselt number (Nu), Reynolds number (Re) and Prandtl number (Pr) can be defined as:

Nu=hDek , Re=GCDeμ , Pr=cpμk

In Reynolds number equation, GC is the mass flow per channel and may be defined as the ratio between the mass velocity per channel m˙ and the cross sectional area of the flow channel (bWP):

GC=m˙bWP

The constants Ch and n, which depend on the flow characteristics and the chevron angle, are given in Table.

4. Design Calculation of Heat exchanger

The methodology employed for the design of a PHE is the same as for the design of a tubular heat exchanger. The equations given in the present chapter are appropriate for the chevron type plates that are used in most industrial applications.

Design conditions mentioned in the below table are based on the site conditions and requirements. As we know that we have 2 types of fluids between which the heat exchange will takes place in the heat exchanger, the properties, parameters of both the fluids are taken into consideration.

The below tabulated information is for the 2 fluids considered for which the heat transfer will take place. Standard material alloy-316 was selected referring to the present practice followed by manufacturer Alfa-laval.

4.1. Design Condition

Property	***Unit***	***Hot Fluid***	***Cold Fluid***
Density	*Kg/m^3*	*997.3*	*1000*
Specific Heat	*kJ/kgK*	*4.19*	*4.21*
Thermal Conductivity	*W/mK*	*0.604*	*0.583*
Volume flow rate	*l/s*	*10.6*	*27.8*
Inlet temperature	*C*	*35*	*4.6*
Outlet temperature	*C*	*12*	*13.2*
Heat Exchanged	*kW*	*1010*	
LMTD	*k*	*13.3*	
Fluid Flow direction		*Counter-current*	
Plate Material		*Alloy-316*	
Plate thickness	*mm*	*0.50*	

The inlet and Outlet temperature of hot water was considered as below:

Hot fluid inlet temperature: 35 C

Hot fluid outlet temperature: 12 C

Mass flow rate is considered: 10.6 l/s

The total heat to be rejected by hot fluid is:

$$Q=(\dot{M}\,cp)hot(Thot,in-Thot,out)$$

$$Q=10.6*\dot{}4187*(35-12)$$

$$Q= 1020\ kW$$

The total heat to be absorbed by cold fluid is:

$$Q=(\dot{M}\,cp)cold(T\ cold,\ out\ -T\ cold,\ in)$$

$$1010\ kW =27.8*\dot{}4187*\ (T\ cold,\ out\ -T\ cold,\ in)$$

$$(T\ cold,\ out\ -T\ cold,in)= 8.6\ C$$

Let's consider cold water inlet temperature is 4.6 C

T cold, out = 8.6 +4.6=13.2 C

4.2. Heat transfer in the plates

The heat transfer area is expressed as the global design equation:

Q=UA(LMTD)

where U is the overall heat transfer coefficient, A is the total area of heat transfer and LMTD is the effective logarithmic mean temperature difference.

$$\text{LMTD} = \frac{(Th1 - Tc2) - (Th2 - Tc1)}{ln\frac{(Th1-Tc2)}{(Th2-Tc1)}}$$

$$\text{LMTD} = \frac{(35 - 13.2) - (12 - 4.6)}{ln\frac{(35-13.2)}{(12-4.6)}}$$

ΔTM=13.3 C

The overall heat transfer coefficient can be determined by:

U=hhot+kP /tP +hcold

where

hhot = 2630 W/m^2K

hcold = 2693 W/m^2K

tP = 0.5 mm

kP= 16.3 W/mK

U=2630+16.3/0.0005+2693

U=60.11 W/m^2K

=1278.19 W/m^2K

The convective heat transfer coefficient, h, depends on the fluid properties, fluid velocity, and plate geometry.

Heat transfer Area :

$$Q=UA\ LMTD$$

$$1020kW=1278.19 *A*13.3$$

$$A=60\ m^2$$

The total area of heat transfer can be given by:

$$A=N_PA_P$$

$$N_P=94$$

where N_P is the number of plates. The end plates, which do not exchange heat, are not taken into account in determining the area. The inner plates are usually called thermal plates in order to distinguish them from the adiabatic end plates.

Parameters of a chevron plate.

The corrugations must be taken into account in calculating the total heat transfer area of a plate (effective heat transfer area):

$$A_P=\Phi.W_P.L_P$$

Φ = 1.25

W_P = 0.470 m

L_P= 1.084 m

$$A_P=1.25*0.470*1.084$$

$$A_P= 0.6368\ m^2$$

Number of plate required:

$$A=N_PA_P$$

A =60 m^2

A_P= 0.6368 m^2

$$60 = N_P * 0.6368$$

$$N_P = 94$$

Total number of plate required is 94.

5. Conclusions

In this work, simplified methodology for Plate heat exchanger design is presented.

- Chevron plate geometry is explained for better understanding and heat exchanger design with the two main approaches used in the design of PHEs, namely the log-mean temperature difference and the thermal effectiveness.
- lack of reliable design methods to quantify the potential benefit of applying plate heat exchanger limits the application of plate heat exchangers. The small minimum approach temperature of plate heat exchangers increases the energy saving, but the installation cost of plate heat exchangers is relatively high.
- Plan and improvement strategy for the PHE is exhibited which gives preferred arrangements over existing distributed techniques. It depends on scientific model representing the principle highlights deciding PHE warm and water driven execution. To get arrangement with insignificant warmth exchange zone for various process conditions is conceivable just for a sufficiently wide scope of plate writes and sizes. The advancement factors are: sort of plate, the quantities of goes for warm trading streams, the relative quantities of plates with various layering designs in one PHE.

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