DESIGN ANALYSIS OF GASKETED PLATE HEAT EXCHANGER

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ABSTRACT: Heat exchanger is a very important device used in many engineering systems related to heat transfer processes in many industrial applications. Gasketed Plate Heat Exchanger (GPHE) is a type heat exchanger which is used for condensing or evaporating system. Among many of factors which should be focused, the heat transfer and pressure drop is most important part for sizing and rating the performance of GPHE. Due to the complex corrugated surface design flow is highly turbulent. This makes design of plate heat exchanger using empirical correlations to deviate from actual. Sometimes flow medium cannot distribute uniformly which affects the performance of plate heat exchanger.

The optimization problem is formulated as the minimization of the heat transfer area, subject to constraints on the number of channels, pressure drop, flow velocity and thermal effectiveness, as well as the exchanger thermal and hydraulic model. Nowadays GPHE widely use in different industries such as chemical, food and pharmaceutical process and refrigeration. However in present work GPHE applied in the dairy milk pasteurizer plant model. Milk is pasteurized for killing the bacteria and preserving milk for long time. Present thesis emphasizes the CFD application to design optimization of pasteurizer plant. Thesis covers validation of CFD results with test and analytical results. Temperature distribution, flow combination and comparison of material thermal conductivity are studied. This thesis considers full assembly analysis of pasteurizer unit
KEYWORDS: Heat Exchanger, GPHE, milk pasteurizer, refrigeration.

I. INTRODUCTION

Plate fin heat exchangers are broadly utilized as a part of vehicle, aviation, cryogenic and synthetic enterprises. They are described by high adequacy, smallness (high surface range thickness), low weight and direct cost. Despite the fact that these exchangers have been widely utilized the world over for quite a few years, the advancements identified with their outline and fabricate stay restricted to a couple organizations in created nations. As of late endeavours are being made in India towards the advancement of little Plate fin heat exchangers for cryogenic and aviation applications.

II. ADVANTAGES AND DISADVANTAGES

The principal advantages are:
(1) High thermal effectiveness and close temperature approach. (Temperature approach as low as 3K between single part fluid streams and 1K between boiling and compression fluids is fairly common.)
(2) Giant heat transfer extent per unit volume (Typically one thousand m2/m3),
(3) Low weight,
(4) Multi-stream operation (Up to 10 method streams will exchange heat during a single device.),and
(5) True counter-flow operation (Unlike the shell and tube device, wherever the shell aspect flow is sometimes a combination of cross and counter flow).

The principal disadvantages are:

- Fouling problems Associated with the plates
- Problem in improvement of passages, that limits its application to wash and comparatively non-corrosive fluids,
- Problem of repair just in case of failure or outpouring between passages, and
- Restricted range of temperature and pressure.

III. MATERIALS

Plate fin heat exchangers may be created in an exceedingly style of materials. Aluminium is most well-liked in refrigerant and region applications thanks to its density, high thermal physical phenomenon and high strength at cold. The maximum design pressure for brazed aluminium plate fin heat exchangers is around ninety bars. At temperatures higher than close, most aluminium alloys lose mechanical strength. Stainless steels, nickel and copper alloys are used at temperatures up to 5000 C. The brazing material just in case of aluminium exchangers is an aluminium alloy of lower freezing point, whereas that employed in stainless-steel exchangers could be a nickel based alloy. however material used for the plates in this exchanger was SS304.

IV. MANUFACTURING DETAILS OF THE PLATE:

The basic element of the plate pack is the plate, a sheet of metal precision-pressed into a corrugated pattern, as shown in Figure 4.2. Single plate is of the order of 74 cm high × 15 cm wide. The heat transfer area for a single plate lies in the range 0.11–0.15 m2. The fluid should be equally distributed over the full width of the plate.

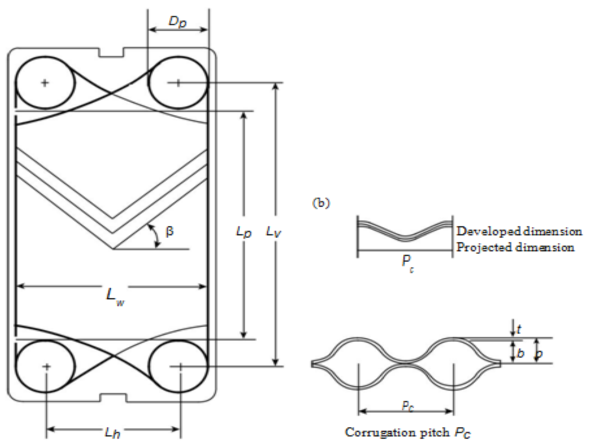

Fig:1 (a) Main dimensions of a chevron plate; (b) developed and projected dimensions of a chevron plate cross-section normal to the direction of troughs.

V. EXPERIMENTAL SETUP AND PROCEDURE

The test section consists of a counter flow plate fin heat exchanger with offset strip fin geometry. This Plate Fin heat exchanger was manufactured by Ved Engineering, Jaipur, for Kamdhenu Gauamrita dairy farm, Palsana(Raj.) Figure 1 shows the plate fin heat exchanger with all its dimensions and arrangements of inlet and Outlet ports. This plate fin heat exchanger consists of offset strip fins. And table 1 provides the details of core dimensions and thermal data respectively. This Project is largely an experimental set-up that is build up for the thermal performance testing of the plate fin heat exchanger for finding out its performance. The procured heat exchanger is a steel (SS 304) Plate Fin heat exchanger and that was manufactured at Ved Engineering, Jaipur For Kamdhenu Gauamrita dairy farm, Palsana(Raj.). As per the data gathered from the Amritsudha dairy farm, Palsana(Raj.)

Plate Material	SS 304
Plate thickness (mm)	0.6
Chevron angle (degrees)	45
Total number of plates	20
Enlargement factor, ϕ	1.33
Number of passes	One pass/one pass
Overall heat transfer coefficient (clean/fouled) W/m2.k	8,000/4,500
Total effective area (m2)	1.44
All port diameters (mm)	25
Compressed plate pack length, Lc, (m)	0.5
Vertical port distance, L_v, (m)	0.7
Horizontal port distance, L_h, (m)	12.5
Effective channel width, L_w, (m)	15
Thermal conductivity of the plate material (SS304),W/m.K	17.5

VI. RATING PROCEDURE

During my analysis I found that Cold brine solution was used to cool the fresh milk. The cold water with a flow rate of 0.42 kg/s enters the gasketed-plate heat exchanger at 1.5°C and will be heated to 11.34°C whereas milk has the maximum and minimum temperature as 35^0 C and 4^0 C. Milk has the flow rate of 0.14 kg/s. The maximum permissible pressure drop for each stream is 3.5 bar

6.1 Stepwise Performance Analysis

The required heat load can be calculated from the heat balance as

$Q_{rh} = 0.14 \times 0.931 \times (35\text{-}4) = 4.04$ W

$Q_{rc} = 0.14 \times 4.20501 \times (11.34\text{-}1.5) = 17.378$ W

The effective number of plates is

$N_e = N_t - 2 = 20 - 2 = 18$

The effective flow length between the vertical ports is $L_{eff} \approx L_v = 27.5$ cm

The plate pitch can be determined from Equation,

$$p = \frac{L_C}{N_t}$$

Where

N_t is the total no. of plates

$$= \frac{50}{20} = 2.5 \text{ cm}$$

Mean channel flow gap is given by,

$b = p - t = 25 - 6 = 19$ mm $= 1.9$ cm

The one channel flow area is, $A_{ch} = b \times L_w = 1.9 \times 17.5 = 33.25$ cm^2

And the single-plate heat transfer area is, $A_1 = \frac{A_e}{N_e} = \frac{1.44}{18} = 0.08$ m^2

The projected plate area A_{1p} from Equation is,

$A_{1p} = L_P.L_W = 70 \text{ x } 15 = 600 \text{ cm}^2 = 0.06 \text{ m}^2$

The enlargement factor has been specified by the manufacturer, but it can be verified from Equation

$$\Phi = \frac{A_1}{A_{1P}} = \frac{0.08}{0.06} = 1.33$$

The channel hydraulic/equivalent diameter from Equation

$$D_h = \frac{2b}{\Phi} = \frac{2 \text{ X } 0.019}{1.33} = 0.02857 \text{ m}$$

The number of channels per pass, N_{cp}, from Equation

$$= N_{cp} = \frac{N_t - 1}{2N_P} = \frac{19}{2 \text{ X } 1} = 10$$

6.2 Heat Transfer Analysis:

Mass flow rate per channel is given by $m_{ch} = \frac{0.14}{10} = 0.014$ Kg/S

And the mass velocity, G_{ch}:

$$G_{ch} = \frac{2.69 \text{ X } 10^4}{33.25} = 4.211 \text{ Kg/m}^2\text{s}$$

The hot fluid Reynolds number can be calculated as

$$Re_h = \frac{G_h D_h}{\mu_h} = \frac{4.211 \text{ X } 0.1425 \text{ X } 1000}{2.422} = 247.76$$

And the cold fluid Reynolds number can be calculated as

$$Re_c = \frac{G_c D_h}{\mu_c} = \frac{4.211 \text{ X } 0.1425 \text{ X } 1000}{1.37854} =$$

435.29

The hot fluid heat transfer coefficient, h_h, can be obtained by using Equation

$$\frac{hD_h}{k} = C_h\left(\frac{D_h G_c}{\mu}\right)^n \left(\frac{c_p \mu}{k}\right)^{1/3} \left(\frac{\mu}{\mu_w}\right)^{0.17}$$

From Table 3.1, C_h = 0.3 and n = 0.663, so the heat transfer coefficient from above Equation can be found as

$$Nu_h = \frac{h_h D_h}{k} = 0.3(Re)^{0.663}(Pr)^{1/3}\left(\frac{\mu_b}{\mu_h}\right)^{0.17}$$

Assuming $\mu_b \approx \mu_w$

$Nu_h = 30.01$

$$h_h = \frac{k\,Nu_h}{D_h} = \frac{30.01\ X 0.53578}{0.1425} = 113.085\ W/m^2K$$

Cold fluid heat transfer coefficient, h_c is determined similarly,

$$Nu_h = \frac{h_c D_h}{k} = 0.3(435.29)^{0.663}(9.419 \times 10^{-3})^{1/3}$$

= 3.615

$$hc = \frac{k\,Nu_{hc}}{D_h} = \frac{3.615\ X\ 0.61539}{0.1425} = 15.61\ W/m^2K$$

For the overall heat transfer coefficient, the clean overall heat transfer coefficient can be found by the Equation is

$$\frac{1}{U_c} = \frac{1}{h_c} + \frac{1}{h_h} + \frac{0.006}{17.5} = \frac{1}{15.61} + \frac{1}{113.085} + \frac{0.006}{17.5}$$

$$U_c = 13.71\ W/m^2K$$

The fouled (or service) overall heat transfer coefficient is calculated from Equation

$$\frac{1}{U_f} = \frac{1}{U_c} + 0.0000907$$

$$U_f = 13.693\ W/m^2K$$

The corresponding cleanliness factor is

$$CF = \frac{U_f}{U_c} = \frac{13.693}{13.71} = 0.99$$

Which is rather low because of the high fouling factor.

The actual heat duties for clean and fouled surfaces are

$Q_c = U_c A_e \Delta T_m$= 13.71 X 1.44 X 9.415 = 185.87 W

$Q_f = U_f A_e \Delta T_m$= 13.693 X 1.44 X 9.415 = 185.644 W

The safety factor is

$$C_s = \frac{Q_f}{Q_c} = \frac{185.644}{185.87} = 0.998$$

The percent over surface design, from Equation is

$OS = 100 U_c R_{ft}$ = 100 x 13.71 x 0.0000907

Where R_{ft} = higher fouling resistance

OS = 12.43 %

VII. TEST RESULT & DISCUSSION

All results are listed below in a table:-

S.No.	Physical Quantities	Experimental Result
1	heat load for cold fluid, Qrh	4.04 w
2	heat load, Qrc	17.38 W
3	Logarithmic mean temperature difference(LMTD),ΔTm	9.415°C
4	Plate pitch ,p	2.5 cm
5	Mean channel flow gap, b	19 mm
6	Single plate heat transfer area ,A1	0.08 m^2
7	Projected plate area ,A1p	0.06 m^2
8	Enlarrgement factor, Φ	1.33
9	No. Of channel per pass,Ncp	10
10	Mass flow rate per channel , mch	0.014 Kg/S
11	Mass velocity,Gch	4.211 Kg/m^2s
12	Hot fluid reynold's no, Reh	247.76
13	Cold fluid Reynold's no,Rec	435.29
14	C_h	0.3
15	h_h	113.085 W/m^2K
16	h_c	15.61 W/m^2K
17	clean overall heat transfer co-efficient	13.71 W/m^2K
18	fouled overall heat transfer co-efficient	13.693 W/m^2K
19	corresponding cleanliness factor	0.99
20	Actual heat duties for clean surfaces,Qc	185.87 W
21	Actual heat duties for clean surfaces,Qh	185.644 W
22	safety factor, Cs	0.998
23	percent over design	12.43%

This is a rather small heat exchanger, which could be enlarged to satisfy the process specifications with less frequent cleaning. It may be preferable to use a 30% over surface design to decrease the cleaning cost, and then the cleaning scheduling could be arranged accordingly. Therefore, the proposed design may be modified and rerated. As introduced Earlier, the phenomenon of fouling is complex and very little information is available on the subject. It is often necessary to carry out prolonged experimental studies under actual operating conditions to incorporate the influence of fouling on the heat transfer rates. A careful choice of fouling factors while designing exchangers is very necessary in order to minimize the chances of over- or under-design. Besides adding to the costs, overdesign of a plate heat exchanger often reduces the flow velocities, resulting in increased fouling rates.

Fouling is much less in a gasketed-plate heat exchanger than in a tubular unit for the following reasons:

- High turbulence maintains solids in suspension
- Velocity profiles across a plate are uniform with zones of low velocities absent
- The plate surfaces are generally smooth and can be further electropolished
- Deposits of corrosion products, to which fouling can adhere, are absent because of low corrosion rates
- In cooling duties, the high film coefficients maintain a moderately low metal wall temperature; this helps prevent crystallization growth of the inverse solubility compounds
- Because of the ease with which plate units can be cleaned in place, deposits that grow with time can be kept to a minimum by frequent cleaning.

A gasketed-plate heat exchanger can be easily opened for inspection, mechanical cleaning, gasket replacement, extension or reduction to the number of plates, or other modifications of the duties. Since percent over surface design is only 12.43 %. Due to which fouling on the plate surface increases which the Lower heat transfer and increases pressure drop resulted from fouling and hence decrease the effectiveness of a heat exchanger. As it is a small heat exchanger the cleaning scheduling Should be arranged accordingly. Therefore, the proposed design may be modified and rerated.

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