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Thermal Designing of Plate Fin Heat Exchanger: A Review

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ABSTRACT
Thermal designing of plate fin heat exchanger explains the heat transfer coefficient, hydraulic diameter, Colburn factor, friction factor, Reynolds number of plate fin heat exchanger. The present review explains the various correlations used in the thermal designing of plate fin heat exchanger. Colburn factor, hydraulic diameter and Friction factor are the major parameters in the design correlations.

Introduction

Plate fin heat exchanger (PFHE) is consists of alternate flat plates and corrugated fins, both are brazed together as a box. The flat plates are called parting sheets. Between the parting sheets, the streams exchange heat. Separating plates act as the primary heat transfer surfaces and fins act as the secondary heat transfer surfaces. The commonly used material for fin is aluminum and stainless steel employed in high pressure and high temperature applications.

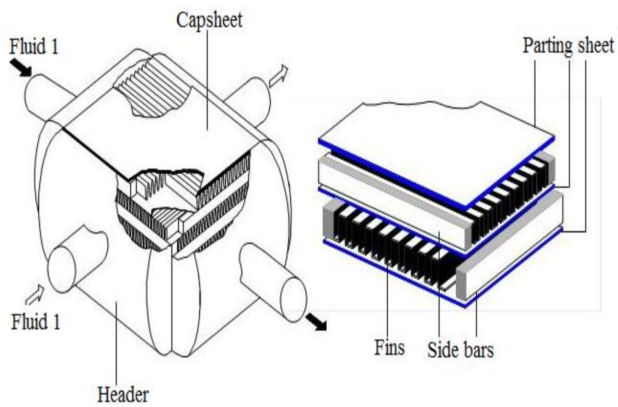

Figure 1. Plate fin heat exchanger assembly[26]

Advantages of Plate Fin Heat Exchangers

i)Compactness: Large heat transfer surface area per unit volume is usually provided by plate fin heat exchangers. Small passage size produces a high overall heat transfer coefficient.

ii)Effectiveness: Thermal effectiveness more than 98% can be obtained.

iii)Temperature control: The plate fin heat exchanger can operate with small temperature differences.

iv)Flexibility: Changes can be made to heat exchanger performance by utilizing a various range of fluids and conditions that can be done to adapt to various design specifications. Multi stream operation is possible up to 10 streams.

v) Counter flow: True counter-flow operation is possible

Disadvantages of Plate Fin Heat Exchangers

i) The rectangular geometry used puts a limit on operating range of pressure and temperatures

ii) Difficulty to cleaning of passages.

iii)Difficulty of repair in case of failure or leakage between passages.

iv)High pressure drop due to narrow passages

Applications of Plate Fin Heat Exchangers

The plate-fin heat exchangers are used for gas to gas, and gas to liquid activities. They are used for many applications. They are mainly used in the field of cryogenic engineering, air liquefaction, processing of natural gas, petrochemical production and refrigeration field. The plate fin heat exchangers that are used for cryogenic air separation and its application and LPG fractionation are the main important applications. Other important applications of brazed aluminium plate fin exchangers are widely used in the aerospace engineering field, because of compactness and weight to volume ratio is less. For the environment control system of the aircraft machines, cooling of avionics, cooling of hydraulics and fuel heating are the other field of plate fin applications. In automobile industries, they demanded that the making of heat exchangers as compact as possible and it has been an everlasting one. But air conditioning industries are space conscious. In the automobile field they are also used for making the radiators.

Plate Fin Heat Transfer Surfaces

The plate fin exchangers are used for liquid-to-gas and gas-to-gas applications. Special surface geometries provide high heat transfer coefficients than extended surfaces, but at the same time, the pressure drop are also high. A variety of extended the surfaces like the plain trapezoidal, plain rectangular [26]shown in figure 2 can perform such function.

The rectangular offset strip fin geometry is included in the present study. In order to improve the gas side heat transfer coefficients, surface features are needed to be provided on the gas side. These features may be divided into two. The first, in which the surface remains continuous (wavy and herring-bone

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fins) and the second is un continuous (offset, louvered)[26]. In a wavy type fins, the corrugations cause the gas to make sudden direction changes so that the velocity and temperatures are increased.

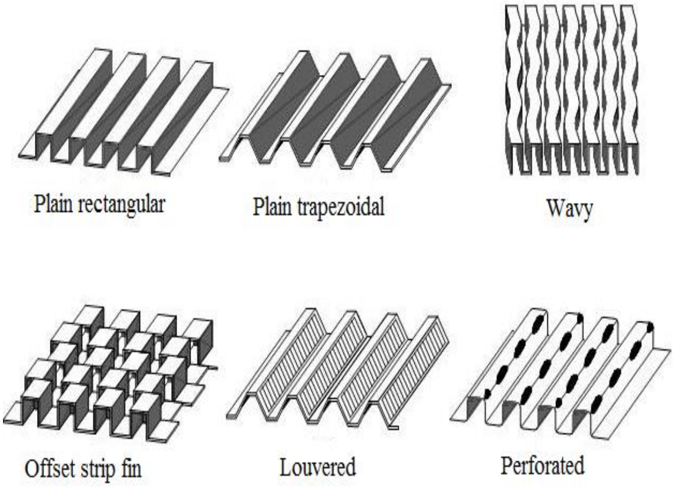

Figure 2. Types of plate fin surfaces [26]

Heat Transfer and Flow Friction Characteristics

The Colburn factor and fanning friction characteristics of a heat exchanger surfaces are commonly expressed in non-dimensional form and are referred to as the basic characteristic factors of the surface. Various correlations are available in literatures are the Colburn factor, j and Friction factor, f as functions of Reynolds number and other geometrical properties**.** The heat transfer coefficient and the flow resistance are expressed in non-dimensional form as Colburn factor, j and the friction factor, f. Accurate prediction of the heat transfer coefficient and friction factor is essential for proper design of heat exchangers.

The Colburn factor, j and the friction factor, f are expressed as functions of Reynolds number and other geometrical parameters. j and f factors can be determined by numerical modeling of the flow field through CFD. In spite of the progress in computing power, it is not possible to predict j and f data by numerical solution. This is because the models are usually based on certain simplifying assumptions. Numerical solution along with flow visualization, however, helps in understanding the flow physics associated with heat transfer enhancement. It is also possible to carry out a parametric study on the effect of geometrical parameters on the performance of finned surfaces. Fundamental relations describing various types of heat transfer correlations and heat exchanger design correlations have been discussed in well-known text books [2-8]. The heat exchangers design text books provides an excellent introduction to the analysis of plate fin heat exchangers, and contains the heat transfer and flow friction characteristics of several fin geometries. The recent work of shah [7-8] provides the most important information on the subject, particularly on compact plate fin heat exchangers. Several specialized monographs and conference proceedings, covering basic heat transfer, heat transfer augmentation and design and simulation methodologies have further enriched the literature [9-13].

Analytical and Numerical Studies of Flow Friction Characteristics

The performance of a plate and fin heat exchanger is not uniquely determined by the hydraulic diameter like simpler geometry. Other geometrical parameters such as fin spacing, fin height, fin thickness, offset strip length and wave amplitude etc. play significant roles. It will be very expensive and time consuming to fabricate heat exchanger and conduct experiments over various ranges of all the geometric variables and Reynolds numbers. In contrast, it is nearly easy and effectively to carry out a parametric study through numerical simulation and derive acceptable correlations for use by the heat exchanger industry. With the development of more powerful computational tools, numerical prediction of j and f factors are now feasible by solving the continuity, momentum and energy equations.

Patankar [14] provides a comprehensive summary of CFD equations relevant to compact heat exchanger passages and techniques employed for their solution**.** Levent Bilir et al[15] used CFD program fluent to analyze the effect of three different types of vortex exchangers on the basis of performance. They found that the three vortex generators when placed suitably will increase the heat transfer with moderate increase in pressure drop. Numerical studies, supplemented by flow visualization, can definitely be a means for the understanding of the heat transfer enhancement mechanism.

Offset strip fin surfaces

Sparrow, Patankar and coworkers [16] were the first to use numerical techniques for prediction of j and f data in offset strip fin heat exchangers. Patankar and Prakash [17] extended their work further and compared their numerical results for a two dimensional heat transfer matrix having offset strip fins with the experimental results of London and Shah [18]. The results indicated reasonable agreement for the f factors. But the predicted j factors were about twice as large as the experimental data.

Suzuki et al [19] took a different numerical approach by solving elliptic differential equations of momentum and energy to study the thermal performance of a staggered array of vertical flat plates at low Reynolds number. The validation of their numerical model was done by carrying out experiments on a two dimensional system. The experimental results explained the computed values in the Reynolds number range Re < 800. Zhang et al [20] has attempted solving the unsteady Navier-stokes and energy equations on a massively parallel computer. Their study shows that the flow unsteadiness is an important role in accurate prediction of j and f factors. One of the earliest method of experimental j and f data on plate and fin surfaces is the monograph of compact heat exchangers by Kays and London [2].

Kays and London conducted experiments on different types of plate and fin surfaces and observed from experiments that the heat transfer coefficient and friction factor f of surfaces having the same effective diameter differed mutually according to the fin geometrical properties like h/s, l/s and t/s etc. Therefore, it is imperative that the j and f factors are obtained experimentally as functions of Reynolds number and other geometrical properties. The expression for j and f data is obtained separately for each surface type. j and f so presented are applicable to surfaces of any hydraulic diameter, and geometric similarity introduced different heat transfer correlations for offset strip fins are given as below:

a) Wieting Correlations

Wieting [21] developed an empirical correlation from experimental heat transfer and flow friction data on 22 offset strip fin surfaces of Kays and London[2], London and Shah[18], Walters[22] etc over two Reynolds number ranges:

Re< 1000 and Re >2000

For Re<1000

$j=0.483(l/D_h)^{-0.162}(s/h)^{-0.184}(Re)^{-0.536}$

$f=7.661(l/D_h)^{-0.384}(s/h)^{-0.092}(Re)^{-0.712}$

For Re>2000

$j=0.242(l/D_h)^{-0.322}(t/d_h)^{-0.08}(Re)^{-0.368}$

$f=0.483(l/D_h)^{-0.384}(s/h)^{-0.092}(Re)^{-0.712}$

For predicting j and f in the transition zone, extrapolating the equations up to their respective transition zone boundaries was suggested. Although 85% of all available data were correlated within 15% for friction factor and 10% for heat transfer, a few points showed discrepancy as high as 40%.Wieting's correlation can be successfully used for the design of practical heat exchangers, but care should be taken in extrapolating the data to fins with geometrical parameters outside the recommended range.

b) Joshi and Webb Correlation

Joshi and Webb [23] conducted flow visualization experiments to identify the transition from laminar flow. As the flow rate increases, velocities developed is, leading to vortex shedding with further increase in Re. The on set flow and the change in the wake structure were found to correspond approximately to laminar base of j and f. A width based equation was introduced to finding the critical Reynolds number. They developed an analytical model in the laminar zone based on the numerical solution done by Sparrow and Liu [24] and a semi empirical method has been used for the turbulent region.

for laminar range Re< Re'

$j=0.53(re)^{-0.5}(l/d_e)^{-0.15}(s/h)^{-0.14}$

$f=8.12(re)^{-0.74}(l/d_e)^{-0.41}(s/h)^{-0.02}$

for turbulent range Re>Re'

$j=0.21(re)^{-0.4}(l/d_e)^{-0.24}(t/d_h)^{-0.02}$

$f=1.12(re)^{-0.36}(l/d_e)^{-0.65}(t/d_h)^{-0.7}$

The empirical correlation for j and f factors proposed by the authors were verified with experimental data on 21 heat exchanger geometries and their own observations on scaled up geometries. They were able to correlate 82% of the f data and 91% of the j data within 15%.

c) Manglik and Bergles Correlation

Manglik and Bergles [25] examined the heat transfer and friction data for 18 offset strip fin surfaces given by Kays and London [2], Walters [22] & London and Shah [18], and analyzed the effect of various geometrical attributes of offset strip fins. The equations that describe the behavior of the data in the deep laminar and fully turbulent zones. . The equation for j and f which are valid for laminar, turbulent and transition zones.

$j=0.6522(Re)^{-0.5403}(s/h)^{-0.1541}(t/l)^{-0.1499}(t/s)^{-0.0678}\,x[1+5.269\,x\,10^{-5}\,Re^{1.340}(s/h)^{-0.504}(t/l)^{-0.546}(t/s)^{-1.055}]^{0.1}$

$f=9.6243(Re)^{-0.7422}(s/h)^{-0.1856}(t/l)^{-0.3053}(t/s)^{-0.2659}\,x[1+7.669\,x\,10^{-8}\,Re^{4.429}(s/h)^{0.920}(t/l)^{3.767}(t/s)^{0.236}]^{0.1}$

These equations predict all of the heat transfer data and approximately 90% of the friction data within 20%.

d) Maiti and Sarangi Correlation

Maiti and Sarangi [26] used CFD as numerical tool for computing velocity, pressure and temperature fields in plate and fin passages. The correlations for the non-dimensional heat transfer coefficient, j and pressure drop characteristic ,f in terms of Reynolds number and other geometrical parameters using both computed and experimental results . Some of the constants in the correlation are found by multiple regression from the numerically computed results and the rest of the constants from experimental data on the same geometry by another worker in the laboratory. They thus combined both the experimental and computational methods.

They also obtained the expression for the transition Reynolds number.

For laminar range Re< Re'

$j=0.36(Re)^{-0.51}(h/s)^{0.275}(l/s)^{-0.27}(t/s)^{0.063}$

$f=4.67(Re)^{-0.70}(h/s)^{0.196}(l/s)^{-0.181}(t/s)^{-0.104}$

for turbulent range Re>Re'

$j=0.18(Re)^{-0.42}(h/s)^{0.288}(l/s)^{-0.184}(t/s)^{-0.05}$

$f=0.32(Re)^{-0.286}(h/s)^{0.221}(l/s)^{-0.185}(t/s)^{-0.023}$

e) ε –NTU Method

In the ε -ntu method [7], the heat transfer rate from the hot fluid to the cold fluid in the exchanger is expressed as

$q = \varepsilon\, c_{min}\,(Tt_{hi}-t_{ci})$

Where **ε** is the heat exchanger effectiveness, Sometimes referred to in the literature as the thermal efficiency. The effectiveness can be shown that in general it is dependent on NTU, rate of heat capacity and direct flow transfer type heat exchanger. Effectiveness **ε** is the thermal performance of a heat exchanger. For a given heat exchanger of any flow arrangement, the heat transfer rate from the hot fluid to the cold fluid to the possible heat transfer q_{max} is known as effectiveness.[7

Heat transfer characteristics of an exchanger surface are presented in terms of the Nu, St, or Colburn factor vs. the Reynolds number. Fanning friction characteristics are presented in terms of the fanning friction factor vs. Re [7]. The Colburn factor is a modified as St number to take into account the variations in the fluid Prandtle number. It is defined as

$j = st.pr^{(2/3)}$

As the Stanton number is dependent on the fluid Prandtle number, the colburn factor j is nearly independent of the flowing fluid for 0.5< Pr <10 from laminar to turbulent flow conditions. Thus the j vs. Re data for a given heat exchanger for air can be used for water under certain flow conditions. For a plate fin heat exchanger the geometrical parameters such as fin spacing fin height fin thickness and offset strip length etc. plays significant roles in the designing. Colburn factor and friction factor defined as

$$j=0.6522(re)^{-0.5403}\left(\frac{s}{h'}\right)^{-0.1541}\left(\frac{\delta}{l}\right)^{0.1499}\left(\frac{\delta}{s}\right)^{-0.0678})$$

$$[1+(0.00005269\,(re)^{1.34}\left(\frac{s}{h'}\right)^{0.504}\left(\frac{\delta}{l}\right)^{0.456}\left(\frac{\delta}{s}\right)^{-1.055}]^{0.1}$$

$$f=9.6243(re)^{-0.7422}\left(\frac{s}{h'}\right)^{-0.1856}\left(\frac{\delta}{l}\right)^{0.3053}\left(\frac{\delta}{s}\right)^{-0.2653})[1+(0.00000007669\,(re)^{4.429}\left(\frac{s}{h'}\right)^{0.920}\left(\frac{\delta}{l}\right)^{3.767}\left(\frac{\delta}{s}\right)^{0.236}]^{0.1}$$

Mochizuki et al [27] correlations are once again a reworking of the Wieting [21] equations, with the coefficients and exponents modified to fit their own experimental data for offset strip fin surfaces. Only fully laminar flow and fully turbulent flow are considered, with an abrupt change of flow regime at re=2000. Muzychka and Yovanovich [28] developed a new model to predict the heat transfer performance and fanning friction coefficient of offset strip fin geometries. They considered the offset strip fins as an array of short channels or straight ducts. They developed simple analytical models for the laminar or turbulent wake regions and suitably combined the resulting asymptotic relations to create expressions for the turbulent zone.

Hydraulic Diameter

Hydraulic diameter is may be defined[7] as,

$$D_h = \frac{4\ X\ A_c}{P}$$

A_c is the Free flow area and P is the total heat transfer area. So there are different expressions for hydraulic diameter in the literature. At least three different expressions can be identified in the literature, which are as given below:

Manglik and Bergles [25]

Free flow area $A_c = s \times h$

Heat transfer area $A = 2(sl + hl + ht) + ts$

Therefore hydraulic diameter is given by the formula:

$$D_h = \frac{4\ X\ s\ X\ h\ X l}{2(sl + hl + ht) + ts}$$

Joshi and Webb [23], **and Maiti and Sarangi** [26]

Free flow area and heat transfer area are given as

Free flow area, $A_c = (s - t)\ h$

Heat transfer area, $A = 2(sl + hl + ht)$

Therefore hydraulic diameter is given by the formula:

$$D_h = \frac{2\ X\ (s - t)h}{sl + hl + ht}$$

Wieting [21] **and Kays and London [2]**

Considering a rectangular channel of cross section, hydraulic diameter,

$$D_h = \frac{2\ X\ s\ X\ h}{s + h}$$

Conclusion

This paper explains the important considerations to be given for the thermal design of plate fin heat exchanger. It describes the significance of Colburn factor ,friction factor and hydraulic diameter. The present review explains the Wieting correlations, Maiti and Sarangi correlations, Joshi and Webb correlations, ε –Ntu Method, Manglik and Bergles Correlations.

Nomenclature

A	heat transfer area (mm^2)
A_c	free flow area (mm^2)
C*	heat capacity rate
D_h	hydraulic diameter (mm)
f	friction factor
h	fin height (mm)
h’	plate height (mm)
j	colburn factor
l	fin length (mm)
NTU	number of transfer units
P	total heat transfer area (mm^2)
Pr	Prandtl number
q_{max}	heat transfer rate
Re	Reynold’s number
S	fin spacing (mm)
St	Stanton number
T_{hi}	temperature of hot fluid inlet(K)
T_{ci}	temperature of cold fluid inlet(K)
t	plate thickness (mm)
δ	fin thickness (mm)
ε	Effectiveness

References

[1] Barron, R.F. Cryogenic systems, Oxford University Press (1985)

[2] Kays, W.M. and London, A.L. Compact Heat exchangers, McGraw-Hill, New York (1984)

[3] Kern, D.Q., Kraus, A.D. Extended Surface Heat Transfer, McGraw-Hill, New York (1972)

[4] Frass,A.P., Heat Exchanger Design, 2nd ed., Wiley-Inter science (1989)

[5] Ozisik, M.N., Heat Transfer-A Basic Approach, McGraw-Hill (1985)

[6] Incropera, F.P., Dewitt, D.P. Fundamentals of Heat Transfer 2nd ed. John Wiley, New York, 1985

[7] Shah, R.K., Dusan,P., Sekulic,D.P. Fundamentals of Heat Exchanger Design, John Wiley & Sons, (2003)

[8] Shah, R.K. Compact Heat Exchangers & Enhancement Technology, Begell House Publication, (1999)

[9] Kakac,S., Bergles, A.E., Mayinger, F. Heat Exchangers, Thermal-Hydraulic Fundamentals and Design, by Hemisphere Publication (1980)

[10] Shah, R.K., Subbarao, E.C. Mashelkar, R.A. Heat Transfer Equipment Design edited by R.K.Shah, Publishing Corporation, Washington, DC, (1988)

[11] Taborek, J., Hewitt, G.F. and Afgtan,N Heat Exchangers-Theory and Practice(Eds.) Hemisphere Publishing, New York (1983)

[12] Taylor, M.A. Plate Fin Heat Exchangers: Guide to their Specification and Use HTFS, Oxon, UK (1987)

[13] Sunden, B., Faghiri, M. Computer Simulations in Compact Heat Exchangers 1st ed. by Computational Mechanics Publications (1998)

[14] Patankar, S.V. Numerical prediction of flow and heat transfer in compact heat exchanger passages, in Compact Heat Exchangers edited by R.K.Shah, A.D.Kraus and D.Metzger, Hemisphere Publishing Corp New York (1990) 191-204.

[15] Bilir, L., B. Ozerdem, A. Erek and Z. Ilken. Heat Transfer and Pressure Drop Characteristics of Fin-Tube Heat Exchangers with Different Types of Vortex Generator Configurations, Journal of Enhanced Heat Transfer, 17(3), 243-256 (2010)

[16] Patankar, S.V., Liu, C.H., and Sparrow, E.M. Fully Developed Flow and Heat Transfer in Ducts Having Stream wise-Periodic Variations of Cross-Sectional Area, Transactions of the ASME Journal of Heat Transfer (1977) 99 180-186.

[17] Patankar, S.V., and Prakash, C. An Analysis of the Effect of Plate Thickness on Laminar Flow and Heat Transfer in Interrupted–plate passages, International Journal of Heat and Mass Transfer, (1981) 24(11)1801-1810.

[18] London, A.L., and Shah,R.K. Offset Rectangular Plate –fin surfaces-Heat Transfer and Flow Friction Characteristics, Transactions of the ASME,Journal of Engineering for Power (1968) 90 218-228.

[19] Suzuki, K., Hirai, E., Miyake, T. Numerical and Experimental Studies on a Two- dimensional Model of an Offset-strip-fin Type Compact Heat Exchanger Used at Low Reynolds Number, International Journal of Heat and Mass Transfer, (1985) 28(4)823-836

[20] Zhang, L.W., Tafti, D.K., Najjar, F.M., and Balachandar, S. Computations of Flow and Heat Transfer in Parallel-Plate Fin Heat Exchangers on the CM-5: Effects of Flow Unsteadiness and Three-Dimensionality, International Journal of Heat and Mass Transfer (1997) 40 (6) 1325-1341

[21] Wieting, A. R. Empirical Correlations for Heat Transfer and Flow Friction Characteristics of Rectangular Offset-fin Plate-fin Heat Exchangers, ASME Journal of Heat Transfer (1975) 97 488-490.

[22] Walters, F.M., Hypersonic Research Engine Project –phase II A, category I Test report on fin Heat transfer and

pressure drop testing, Data item No. 63.02, Ai Research Manufacturing co., Doc, AP-69-5348, 1969
[23] Joshi, H.M. and Webb, R.L. Heat Transfer and Friction in the Offset Strip-fin Heat Exchanger, International Journal of Heat and Mass Transfer, (1987) 30 (1) 69-84.
[24] Sparrow, L.U. and Liu, C.H. Heat Transfer, Pressure Drop and Performance relations for in-line, Staggered and Continuous Plate Heat Exchangers, International Journal of Heat and Mass Transfer, (1979) 22 1613-1625
[25] Manglik, R. M., and Bergles, A.E. Heat Transfer and Pressure Drop Correlations For the Rectangular Offset strip Fin Compact Heat Exchanger, Experimental Thermal and Fluid Science (1995) 10 171-180
[26] Maiti, D.K.,and Sarangi.S.K. Heat Transfer and Flow Friction Characteristics of Plate Fin Heat Exchanger Surfaces-A Numerical Study PhD Dissertation, Indian Institute of Technology, Kharagpur (2002)
[27] Mochizuki, S., Yagi, Y., and Yang, W.J., Transport phenomena in stacks of Interrupted parallel plate surfaces, Experimental Heat Transfer 1, 127-140, 1987
[28] Muzychka, Y.S. and Yovanovich, M.M. Modeling the f and j Characteristics for transverse flow through an offset strip fin at low Reynolds number, Journal of Enhanced Heat Transfer (2001) 8 261-277.