Numerical Investigation on Effect of Operating Parameters on Plate Fin Heat Exchanger

Nilesh K. Patil and Manish K. Rathod

Abstract—Compact heat exchangers (CHE) are very well known for their special design which includes high heat transfer coefficient and maximum temperature driving force between the hot and cold fluids. The accurate prediction of the thermal performance of a compact heat exchanger in the design stage is highly desirable for most aerospace applications. In the present study, the numerical investigation for effect of operating parameters on plate fin heat exchanger is carried out. Plate fin heat exchanger is analyzed for offset strip fins, having rectangular cross section. A steady state model for the core dimensions of a plate fin cross flow heat exchanger is developed using MATLAB. Design variables such as surface areas, free flow areas, exchanger core size are calculated for the given operating parameters of cross flow CHE. The effect of effectiveness of the CHE on size of the heat exchanger is also established by varying effectiveness from 0.8 to 0.9 with different operating parameters.

***Index Terms*—**Plate fin heat exchanger, Offset fin, Thermo-fluid analysis, parametric analysis.

I. Introduction

Today compactness is the most important criteria of the world. The demand for high performance heat exchange devices having small spatial dimensions is increased due to their requirement in applications such as aerospace and automobile vehicles, cooling of electronic equipment, and artificial organs. Compact heat exchangers compared to shell-and-tube exchangers, are characterized by a large heat transfer surface area per unit volume of the exchangers. Compact heat exchanger reduces space, weight, support structure and foot print, energy requirements and cost. It improves process design, plant layout and processing conditions, together with low fluid inventory.

Plate fin heat exchangers (PFHE) form one of the main categories of compact heat exchangers designed to pack a high heat transfer capacity into small volume. In the last eight decades various models for designing plate fin heat exchanger based on pressure drop and heat transfer have been represented on open literature. Kays and London [1] have given different design data for the selected fin surface for plate fin heat exchangers.

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The variation of friction factor and colburn factor with different Reynolds number for different surface configuration is given in graphical form and tabular form as well. They used exchanger heat transfer and flow friction performance theory in conjunction with the basic design data to size the heat exchanger core for a specified heat transfer duty and pressure drop.

London and Shah [2] discussed performance of strip-fin core due to the following four non-dimensional geometrical parameters: dimensionless fin thickness δ, aspect ratio of flow passage in one fin pitch, fin surface area to total surface area β on the fin side, and dimensionless strip length ls of offset strip fin geometry. Higher δ, β tend to make higher j and f factors, and when ls is higher, both the j and f factors will tend to be lower. Shah and London [3] provided laminar flow analytical results (Nu and f vs. Re for various aspect ratios) for rectangular ducts. Joshi and Webb [4] explained the goodness factor comparison for different fin surfaces particularly for plate fin surfaces. They show goodness factor of offset strip fin is higher than that of triangular and rectangular plain surface operation in same conditions. Later, they presented analytical models to predict the heat transfer coefficients and friction factors of an offset strip-fin heat exchanger by idealizing a unit cell model. The model neglected the possible burrs on the fin ends and also the roughness on the top and bottom of the channel.

Shah and Sekulic [5] present in depth thermo dynamic and fluid dynamic design theory of two fluid single-phase heat exchangers for steady state operations. They have designed the heat exchanger for both the rating and sizing problem with very fundamental steps involved. They have presented ε-NTU method for compact cross flow heat exchangers for plate fin and tube fin configurations by assuming the temperatures, mass flow rates, and geometrical characteristics. Commercially for available fin surfaces and effectiveness, core dimensions is found out by the heat transfer and pressure drops on both sides for a cross flow exchanger, when no constraints are imposed on the dimensions.

Smith [6] has studied the performance of particular rectangular and triangular ducts in order to assess their relative merits and found that the thermal performance of the rectangular duct is 14.8% better and the exchanger is 14.25% smaller, and the LMTD reduction factor is 13.4% smaller. Manson [7] developed correlation equations using a database of different geometries: offset strip fins, louvered fins, and tube fins. For rectangular offset strip fin surfaces, Weiting [8]

Nomenclature-

- A Total heat transfer surface area, m^2
- b Distance between plates in plate-fin heat exchanger, m
- C Flow stream heat capacity rate, W/K
- c_p Specific heat of fluid at constant pressure, J/kg K
- D_h Hydraulic diameter of flow passages, m
- f Fanning friction factor
- G Fluid mass velocity based on the free flow area, kg/m^2
- g_c Proportionality constant in Newton's second law of motion
- h Heat transfer coefficient, W/m^2K
- j Colburn factor
- k Fluid thermal conductivity, W/mK
- L Fluid flow (core) length on one side of an exchanger, m
- L_1 Flow (core) length for fluid 1 of a two-fluid heat exchanger, m
- L_2 Flow (core) length for fluid 2 of a two-fluid heat exchanger, m
- L_3 No-flow height (stack height) of a two-fluid heat exchanger, m
- m Fluid mass flow rate, kg/s
- Nu Nusselt number
- Pr Prandtl number
- p Fluid static pressure, Pa
- Δp Fluid static pressure drop of a heat exchanger core, Pa
- R Heat capacity rate ratio
- T Fluid temperature, ^oC
- U Overall heat transfer coefficient, W/m^2K
- α Ratio of total heat transfer area on one fluid side of an exchanger to the total volume of an exchanger, m^2/m^3
- β Ratio of fin area to total surface area
- δ Fin thickness, m
- ε Heat exchanger effectiveness
- η Extended surface efficiency on one fluid side
- ρ Fluid density, kg/m^3

Subscripts-

- a Air side
- c Cold-fluid side
- f Fouling
- g Gas side
- h Hot-fluid side
- i Inlet to the exchanger
- liq Liquid
- m Mean or bulk mean
- max Maximum
- min Minimum
- w Wall or properties at the wall temperature
- 1 Fluid 1; one section (inlet or outlet) of the exchanger
- 2 Fluid 2; other section (outlet or inlet) of the exchanger

obtained j and f correlations for the laminar and turbulent flow regions (but not for the transition region). Later on, Joshi and Webb [4] modified Weiting's [8] correlations by identifying the transition region. Mochizuki and Yagi [9] reexamined Weiting's [8] correlations, and made some modifications to the coefficients and exponents to fit their experimental data. Manglik and Bergles [10] reviewed the literature and provided correlations of the f and j factors in the laminar, transition, and turbulent regions. Manglik and Bergles [10] also indicated that further analyses should be conducted to extend the applicability of these results to liquid media. This technique is also described by Churchill and Usagi [11]. Bhowmik and Lee [12] studied the heat transfer and pressure drop characteristics of an offset strip fin heat exchanger using a steady-state three-dimensional numerical model. They observed the variations in the Fanning friction factor f and the Colburn heat transfer factor j relative to Re_{dh}. General correlations for the f and j factors were derived, which was used to analyze fluid flow and heat transfer characteristics of offset strip fins in the laminar, transition, and turbulent regions. Nuntaphan et al. [13] studied the effect of the fin spacing on the air side performance at low Reynolds number for both staggered and inline arrangements. They proposed correlations which give fairly good predictive ability against the used test data.

However, it is observed that less literature is available in which effect of operating parameters on the size of heat exchanger is to be analyzed. The objective of the present work is to investigate the physical size of an exchanger to meet the specified heat transfer and pressure drops within specified constraints. The air-to-gas or gas-to-air PFHE is used, having both the fluids unmixed. The surface configurations are selected as per standards given in Kays and London [1]. To analyze such cross flow compact heat exchangers, a numerical code is developed in MATLAB. In order to examine the effect of operating parameters on core geometry of PFHE, parametric analysis is also carried out. The key operating parameters like effectiveness, inlet temperature of fluids, mass flow rate of fluids, and inlet pressure of fluids are varied for the same.

II. Plate Fin Heat Exchanger

A plate fin heat exchanger is a type of compact exchanger that consists of a stack of alternate flat plates called parting sheets and corrugated fins brazed together as a block. Streams exchange heat by flowing along the passages made by the fins between the parting sheets. The fins serve as a secondary heat transfer surface and mechanical support for the internal pressure between layers. A solid bar called as header bar is used to prevent entry of one fluid into the channels for the other fluid. Fig. 1 gives the idea about construction of the PFHE. The most common area of application is in cryogenic processing such as liquefied natural gas production, hydrogen purification and helium separation and liquefaction.

The plate-fin exchangers are generally designed for moderate operating pressures (less than about 700kPa gauge (100psig)), although they are available commercially for

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operating pressures up to about 8300kPa gauge (1200psig) [5]. The temperature limitation for plate-fin exchangers depends on the method of bonding and the materials employed. Such exchangers have been made from metals which can withstand about 840^oC (1550^oF) and ceramic materials which can withstand about 1150^oC (2100^oF) with a peak temperature of 1370^oC (2500^oF). Plate fin exchangers have been built with a surface area density of up to 5900m^2/m^3 (1800ft^2/ft^3). There is total freedom in selecting the fin surface area on each fluid side, as required by the design, by varying the fin height and fin density. Although typical fin densities are 120 to 700fins/m (3 to 18fins/in.), applications exist for as many as 2100fins/m (53fins/in.). Common fin thickness ranges between 0.05mm and 0.25mm (0.002in. to 0.01in.). Fin heights may range from 2mm to 25mm (0.08in. to 1.0in.). A plate-fin exchanger with 600fins/m (15.2fins/in.) provides about 1300m^2 (400ft^2/ft^3) of heat transfer surface area per cubic meter of volume occupied by the fins [5].

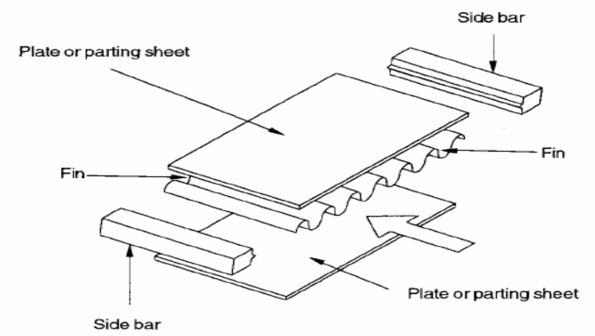

Fig. 1 Basic components of plate-fin heat exchanger [5]

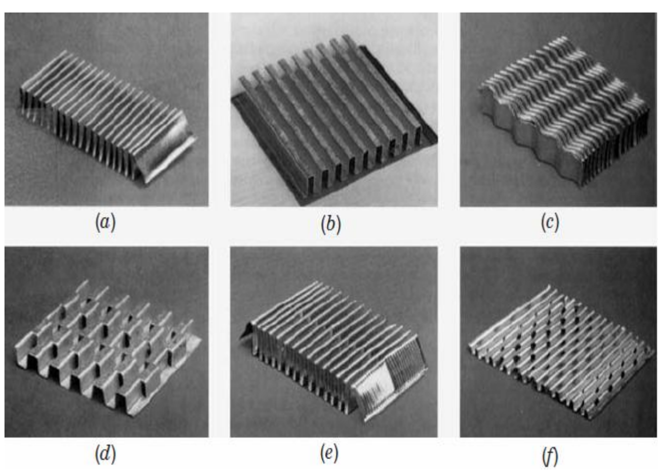

Fig. 2 Different corrugated fin configurations used in PFHE (a) plain triangular fin; (b) plain rectangle fin; (c) wavy fin; (d) offset strip fin; (e) multi-louver fin; (f) perforated fin.

A large number of fin geometries are available for plate-fin heat exchangers. Different corrugated fin configurations used in PFHE are shown in Fig. 2.

The unique characteristics of compact extended surface exchangers, compared to conventional shell-and-tube exchangers are as follows.

- Availability of numerous surfaces having different orders of magnitude of surface area density
- Flexibility in distributing surface area on the hot and cold sides as warranted by design considerations
- Substantial cost, weight, or volume savings.

III. Thermodynamic Design Methodology

For extended surface exchanger, the sizing problem can be solved by finding the physical size i.e. length, width, height, and surface areas on each side of exchanger. Input to the thermo dynamic and fluid dynamic procedures are the surface heat transfer and flow friction characteristics, geometrical properties, and thermo physical properties of fluids, in addition to the design specifications.

As a case study, the problem given by Shah and Sekulic [5] for heat recovery from the exhaust gas to preheat incoming air in a solid oxide fuel cell cogeneration system is studied. This exchanger is a gas-to-air single-pass cross flow heat exchanger operating at an effectiveness of 0.83. The operating conditions of the heat recovery system are shown in Table I. Also the geometric parameters are given in Table II. Both fins and plates (parting sheets) are made from Inconel 625 alloy (its thermal conductivity as 18W/mK). The plate thickness is taken to be 0.5mm.

TABLE I
Operating Conditions of Fluids

Operating Condition	Air side	Gas side
Flow rate, (kg/s)	2	1.66
Inlet Temperature, (^oC)	200	900
Inlet Pressure, (kN/m^2)	200	160
Pressure drop, (kN/m^2)	8.79	9.05

TABLE II
Geometric Parameters of PFHE

Geometric Parameters	Values
Plate Spacing (mm)	2.49
Hydraulic diameter (m)	0.00154
Fin thickness (mm)	0.102
Area density (m^2/m^3)	2254
Fin area/total area	0.785

The thermo-fluid analysis is carried out for a heat exchanger by considering offset strip fin surface. The design is based on the sizing problem to evaluate fluid flow length on both the sides and no-flow length of the heat exchanger. For a cross flow exchanger, determining the core dimensions on one fluid side does not fix the dimensions on the other fluid side. In such a case, the design problem is solved simultaneously on both fluid sides.

The heat duty for both the gas and air is

$$q = \left(m\, c_p\right)_1 \left(T_{1,i} - T_{1,o}\right) = \left(m\, c_p\right)_2 \left(T_{2,o} - T_{2,i}\right) \qquad (1)$$

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The *ntu* on each fluid side is given by the approximations as

$$ntu_h \approx ntu_c \approx 2\,NTU$$
$$ntu_g = 1.11\,NTU$$
$$ntu_{liq} = 10 \times C \times NTU \qquad (2)$$

The mass flow velocity (G) on each fluid side is evaluated using

$$G = \left[\frac{2g_c}{(1/\rho)_m \Pr^{2/3}} \frac{\eta_o \Delta p}{ntu} \left(\frac{j}{f} \right) \right]^{1/2} \qquad (3)$$

The heat transfer coefficient h is calculated as follows:

$$h = Nu\left(\frac{k}{D_h}\right) \quad \text{or} \quad h = \frac{jc_pG}{\Pr^{2/3}} \qquad (4)$$

Here the colburn factor 'j' is based on the fin configuration used which is given by Kays and London [1] for different fins.

The Overall heat transfer coefficient U on the fluid 1 side is given as [14]:

$$\frac{1}{U} = \frac{1}{(\eta_o h)_1} + \frac{1}{(\eta_o h_f)_1} + \frac{\delta_w A_1}{k_w A_w} + \frac{\alpha_1/\alpha_2}{(\eta_o h_f)_2} + \frac{\alpha_1/\alpha_2}{(\eta_o h)_2} \qquad (5)$$

Where, $\alpha_1/\alpha_2 = A_1/A_2$ and $\alpha = A/V$ (6)

For plate fin exchanger α's are related to β's by

$$\alpha_1 = \frac{b_1\beta_1}{b_1+b_2+2\delta_w}, \quad \alpha_2 = \frac{b_2\beta_2}{b_1+b_2+2\delta_w} \qquad (7)$$

The overall surface efficiency of the fin 'η_o' in (5) is defined as:

$$\eta_o = 1 - \frac{A_{fin}}{A}(1-\eta_{fin}) \qquad (8)$$

By assuming the negligible temperature difference between the plates that share the fins i.e. the fins can be considered to be insulated at the center. Therefore the fin efficiency can be calculated as for the case with an adiabatic tip:

$$\eta_{fin} = \frac{tanh(m(b/2))}{m(b/2)} \qquad (9)$$

Where

$$m = \sqrt{\frac{2\alpha}{K_f\delta}} \qquad (10)$$

NTU is calculated from effectiveness formula given as

$$\varepsilon = 1 - exp\left[\frac{NTU^{0.22}}{R}\left(exp\left(-R * NTU^{0.78}\right) - 1\right)\right] \qquad (11)$$

$$A = \frac{NTU \times C_{min}}{U} \qquad (12)$$

The fluid flow length on each fluid side is now can be calculated from the definition of hydraulic diameter of the surface as follows:

$$L = \frac{D_h A}{4A_o} \qquad (13)$$

In order to observe the variation in core dimensions, effectiveness for the same operating parameter is varied from 0.8 to 0.9 in step of 0.02. For the design point of view, the thermo-physical properties are assumed constant which lead to a constant heat transfer coefficient for the whole exchanger design.

IV. Results and Discussions

Fig. 3 shows the variation of various flow lengths with effectiveness. For the same inlet conditions for both the fluid sides, (i.e., hot and cold sides), Fig. 3 shows flow lengths increases with increase in effectiveness and no-flow lengths decreases with increase in effectiveness. This is because of as the effectiveness increases the heat transfer rate also increase. For the same geometry of fins, higher the temperature drop results higher heat transfer rate. This is only possible when the length of the fluid flow is longer. However, no-flow length is reduced for the same volume of the heat exchanger.

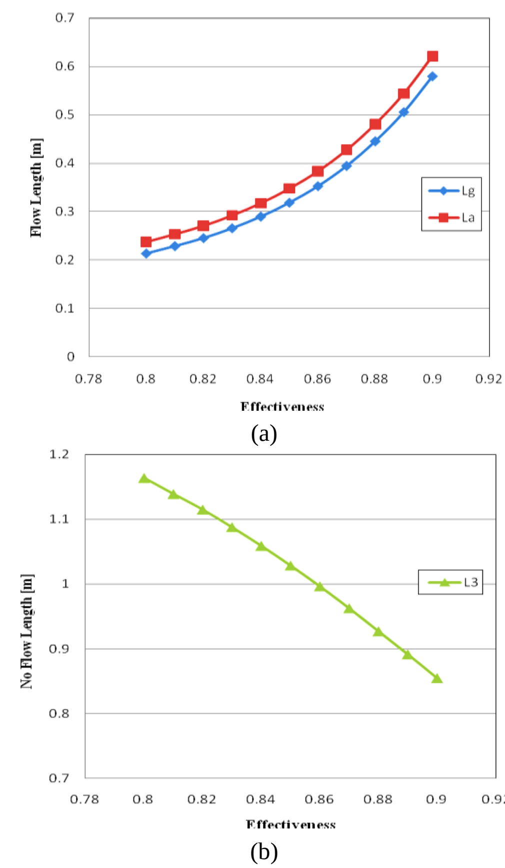

Fig. 3 Variation of effectiveness on core dimension

The effect of operating parameters on the size of the heat exchanger is established by the parametric analysis. The parametric analysis is carried out considering the offset strip fin surface having rectangular fins for both hot and cold sides. Other physical parameters are maintained constant. Ratios of mass flow rates, temperatures, pressures for air to gas are considered as variables.

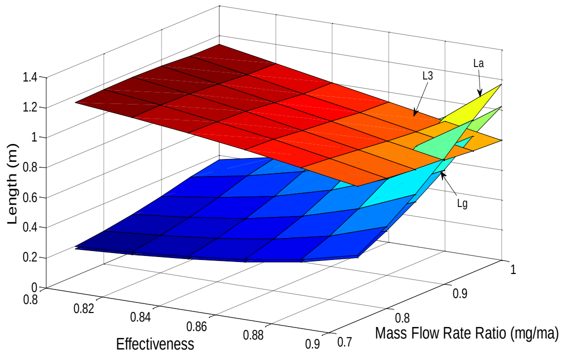

Fig. 4 Effect of mass flow rate ratio and effectiveness on core dimensions

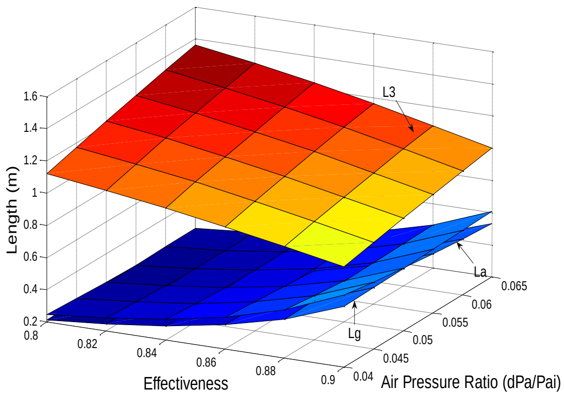

Fig. 5 Effect of air pressure ratio on core dimensions

Fig. 4 shows the effect on core dimensions for different gas to air mass flow rate ratio and effectiveness for offset strip fin having all the inlet conditions unchanged. Range of gas to air mass flow rate ratio is considered from 0.75 to 1 by varying gas side mass flow rate from 1.5 to 2kg/s at constant air side mass flow rate 2kg/s. Different surfaces in the figure indicate variation of different flow length of exchanger with different mass flow rate ratio and effectiveness. Top most surfaces corresponds no-flow length (L_3) which shows lower value at higher effectiveness and remains nearly unaffected by varying mass flow rate ratio. The other two surface indicating gas flow length (L_g) and air flow length (L_a) shows nearly equal profile. Both the lengths increase with increase in effectiveness and mass flow rate ratio. However, rate of increase in flow lengths with effectiveness is comparatively smaller than that with mass flow rate ratio. Hence it can be argued that mass flow rate is more responsive than effectiveness for the core size.

Fig. 5 shows the effect on core dimensions for air pressure ratio and effectiveness. The pressure ratio (dp/p_i) i.e. pressure drop to inlet pressure is varied from 0.04 to 0.065 in step of 0.005. From Fig. 5, it is observed that as pressure ratio of air increases no-flow length increases. It can be also seen that change in length of fluid flow is negligible with change in pressure ratio. Same profile for both fluid flow length and no-flow length is observed at different gas pressure ratio (Fig. 6). The pressure ratio (dp/p_i) for gas side i.e. pressure drop to inlet pressure is varied from 0.05 to 0.08 in step of 0.01. From both the figures (Fig. 5 and Fig. 6) higher no-flow length is examined at lower effectiveness and maximum pressure ratio. It is concluded from above figures that effectiveness has more pronounced effect on core size than pressure ratio of both the sides.

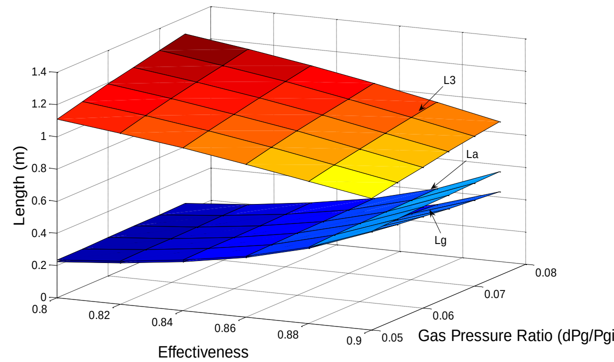

Fig. 6 Effect of gas pressure ratio on core dimensions

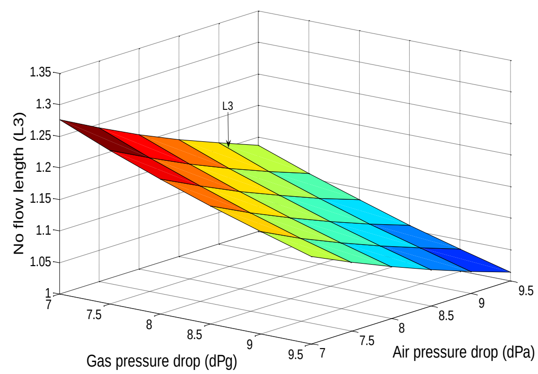

Fig. 7 Effect of gas and air pressure drops on no-flow length (L_3)

As no-flow length is more susceptible with air and gas pressure ratio than fluid flow length, the effect of pressure drops on no-flow length for both the fluid is studied independently (Fig. 7). From the figure, it is noted that rate of decrease in length with increase in gas side pressure drop is higher than that of air side pressure drop. This is due to fact that higher pressure drop induces higher mass flow velocity which increases the overall heat transfer coefficient resulting lower heat transfer area, for which the flow length should be high which leads to decrease in no-flow length for the same volume of core.

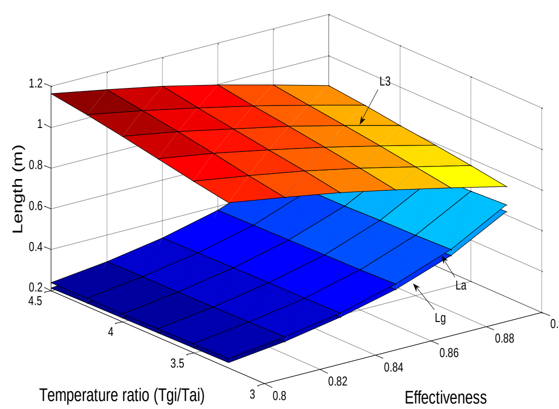

Fig. 8 Effect of temperature ratio and effectiveness on core dimensions

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Fig. 8 shows effect on core dimensions for effectiveness and inlet temperature ratio varying from 3.25 to 4.5 in step of 0.25 by increasing gas inlet temperature from 650°C to 900°C at constant air side inlet temperature i.e. 200°C. The effect of inlet temperature ratio is negligible on flow lengths and has considerable on no-flow length.

V. Conclusion

In the present study, the numerical investigation for effect of operating parameters on plate fin heat exchanger is carried out. From the parametric analysis of plate fin cross flow heat exchanger, it is concluded that the core dimensions are strongly dependent on the effectiveness of the heat exchanger. The core dimensions have nearly linear variations with the effectiveness, and the flow lengths always increased with the effectiveness. As the flow lengths and no-flow length are inversely proportional for same volume of core, no-flow length decreases with increasing effectiveness.

Also the flow lengths have negligible effect with the temperature and pressure of any fluid but have significant effect with the variation in mass flow rates. But in case of no-flow length, it shows varying effect with the slight change in operating parameters. With increasing inlet pressure or pressure drop of any fluid the flow length on that fluid side will increase while the other fluid flow length and no-flow length will decrease. It is also clear that the effect of change in mass flow rate ratio on the flow lengths is negligible at lower value of effectiveness but it shows very step growth at higher effectiveness. This parametric study provides guidelines for sizing the system and design optimization.

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