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PERFORMANCE EVALUATION OF PLATE FIN CROSS FLOW HEAT EXCHANGER WITH DIFFERENT FIN SURFACES

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Abstract — *The demand for high performance heat exchange devices having small special dimensions is increasing due to their requirement in applications in aerospace and automobile vehicles, cooling of electronic equipment and artificial organs. The accurate prediction of the thermal performance of a compact heat exchanger in the design stage is highly desirable for most aerospace applications. When an extended surface is needed on only one fluid side or when the operating pressure needs to be contained on one fluid side, a plate-fin exchanger may be selected. PFCFHE is having different fin surfaces like, rectangular fins, triangular fins, offset strip and perforated fins having rectangular cross section for plate fin surfaces. The paper is concerned with the performance evaluation of such plate fin heat exchangers (PFCFHEs), having rectangular and triangular fins for plate fin surfaces with fin densities varying between 16 to 20 fins per inch. The thermal performance of each configuration is based on the ε-NTU method. The extensive experimental research data of Kays and London available in the form of graphs were translated into algebraic relations for developing computational models. From the present work, a generalized heat transfer correlations is proposed based on geometrical parameters for the effectiveness of the heat exchanger. The generalization is with respect to the different fin surface geometry and not with the different combination of hot and cold fluids. The correlation is applicable to air-to-gas or gas-to-air or combination of both configurations having both the fluids unmixed. Further, guidelines were proposed for an easy way of designing a PFCFHE by modes of geometrical based correlations, and the same was compared with traditional ε-NTU method.*

Keywords- *Plate-Fin Cross Flow Heat Exchanger, Rectangular Fin, Triangular Fin, Effectiveness, Fin Efficiency*

I. INTRODUCTION

The design theory of compact heat exchangers is based on either the ε-NTU or the log-mean temperature difference methods of analysis. The objective of the present work was to investigate the thermo fluid performance of Plate Fin Cross Flow Heat Exchanger (PFCFHE). For the analysis two different fin surfaces like rectangular and triangular were considered. The fin density for both the hot and cold sides varying from 16 to 20 fins per inch was decided from standard designs for PFCFHE, having rectangular cross section. In this work, the PFCFHE was used as air-to-gas or gas-to-air or combination of both configurations having both the fluids unmixed. For the design point of view constant thermo physical properties were assumed and hence constant heat transfer coefficient for the whole exchanger design. Calculation of effectiveness involves a number of parameters. A single correlation predicting effectiveness for all the combination is highly desirable for design purpose. Keeping this in view, an attempt was made to develop a correlation on the basis of geometrical parameters of all possible combinations of PFCFHE for plain rectangular and triangular fin surfaces. The extensive experimental research data of Kays and London in the form of graphs were also translated into algebraic relations for the purpose of developing computational models. This generalized correlation would satisfy for different combinations of fin surfaces like plain rectangular fins and triangular fins. Further, guidelines were proposed for an easy way of designing a CHE by modes of geometrical based correlations, and the same was compared with traditional ε-NTU method. A generalized computational program was developed in MATLAB 6.5.1.

II. MECHANICAL CONSTRUCTION

A. Fin surface geometry for a PFCFHE

The flow through the rectangular and triangular channel for PFCFHE is shown as in Fig.1. The sectional view for the rectangular PFCFHE is shown in Fig.2, based on which the geometrical parameters were calculated. The related calculative terms for both the PFCFHEs were given as per the Table 1 [2, 1]. The ε-NTU method is used for the thermal design and the design was developed for the rating of the plate fin heat exchanger. Finite element analysis of PFCFHE was also developed to treat the compact cross flow heat exchanger on a general basis.

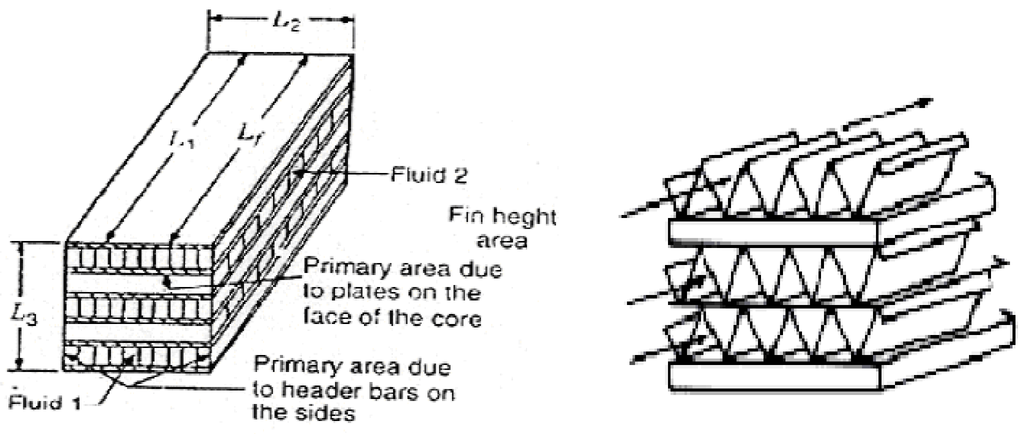

Fig. 1: Rectangular and triangular plate-fin cross flow heat exchangers

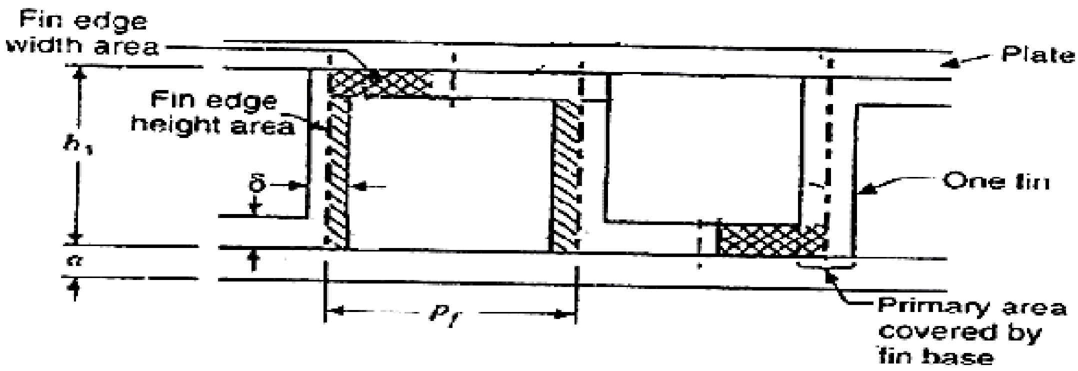

Fig. 2: Geometry for the rectangular plate fin cross flow heat exchanger

Table 1 Geometrical Parameters For Rectangular And Triangular Fin Surfaces Of PFCFHE

PARTICULAR	*RECTANGULAR FIN*	TRIANGULAR FIN
FREE FLOW AREA (A_{fr})	$b_1 L_2 N_p - [(b_1 - \delta) + P_f] * \delta N_{fin}$	$b_1L_2N_p - 2\left[\left(y-\sqrt{2\delta^2}\right)*\delta*N_{fin}+\delta^2*N_{fin}\right]$ $y=\sqrt{P_f^2+b_1^2}$
FIN AREA (A_f)	$2(b_1-\delta) L_1 N_{fin} + 2(b_1-\delta) \delta N_{fin} + (P_f-\delta) \delta N_{fin} + 2P_f \delta N_{fin}$	$2[(y-\delta)*L_1*N_{fin}]$
COLBURN FACTOR (j) & FRICTION FACTOR (f) FOR Re≤2000	$j = 0.8341(Re)$^$^{-0.7259}$ $f = j+0.035$	$j = 0.997(Re)^{-0.735} *(N_{fin})^{-0.0695}$ $f = j+0.0087$
COLBURN FACTOR (j) & FRICTION FACTOR (f) FOR Re>2000	$j = 0.011 (Re)^{-0.1426}$ $f = j+0.026$	$j = 0.046(Re)^{-0.043}$ $f = j+0.00583$
ml FOR FIN EFFICIENCY	$\frac{m \times b}{2}$ $m=\sqrt{\frac{2h}{K\delta}}$	$m*(y/2)$ $m=\sqrt{\frac{2h\left(1+\frac{\delta}{L1}\right)}{K\delta}}$

The method was illustrated through an example of single pass compact heat exchanger. The cases of constant properties were considered. The agreement of the final results of Finite element analysis was excellent with the experimental results given by Ravikumar and Seetharamu [3,4].

B. Empirical Correlation for the PFCFHE

Proposed correlation for the PFCFHE, effectiveness is given by

Effectiveness, $\varepsilon = m^*(AB)^n$

$$AB = \left(\frac{A_{fr}}{A}\right)_{hot}^{-0.086} \times \left(\frac{A_{fr}}{A}\right)_{cold}^{-0.1536} \times (VR)_{hot}^{0.165} \times (VR)_{cold}^{0.1836} \times (St)_{hot}^{0.1239} \times (St)_{cold}^{0.2372} \quad (1)$$

Where, m & n are constants based on the selection of the fin surface for hot and cold side of the heat exchanger.

Table 2 : Constants for different combinations of fin surfaces

Surface selected		m	n
Hot side	**Cold side**		
Rectangular or Triangular	Rectangular or Triangular	**0.8638**	**0.8690**

The suggested correlation is valid only for working fluids, air & normal gas with varying fin density between 16 to 20 FPI. By calculating the effectiveness for different conditions, it can be seen that, the correlations are valid for PFCFHE having hot side fluid flow length varying from 0.2-0.516 m, cold side fluid flow length varying from 0.138-0.438 m, & plate spacing for hot and cold fluid sides varying from 3-8mm and 2.54-8mm resp. with same inlet mass flow rates of both the sides.

III. HYDRAULIC DESIGN METHODOLOGY of PFCHE

The pressure drop for the PFCFHE is consists of; Frictional losses associated with fluid flow over the heat transfer surface, Momentum effect i.e. pressure drop or rise due to the fluid density changes in the core, Pressure drop associated with sudden contraction and expansion at the core inlet and outlet, and Gravity effect due to the change in elevation between the inlet and outlet of the exchangers. But the gravity effect is generally neglected for gases.

Pressure drop for the PFCFHE, which is given by Kays & London:

$$\frac{\Delta P}{P_1} = \frac{G^2 V_1}{2 g_c P_1}\left[\left(K_c + 1 - \sigma^2\right) + 2\left(\frac{\rho_i}{\rho_o} - 1\right) + \left(f \frac{L}{r_h} \frac{\rho_i}{\rho_m}\right) - \left(1 - \sigma^2 - Ke\right)\left(\frac{\rho_i}{\rho_o}\right)\right] \quad \textbf{(2)}$$

Values for K_c & K_e are directly taken from the standard available plots from Kays & London. This correlation is directly adopted for the present work which gives the best results for the chosen limits.

IV. FOOTNOTES

A. COMPARISON OF CORRELATION AND ε - NTU METHOD

Finally the Matlab results were compared with results obtained from ε - NTU approach for different flow conditions and for different exchanger design dimensions. Table 3 and 4 gives the physical properties and the initial flow field conditions for both fin surface configurations. Different flow field parameters were calculated as given per Table 5.

Table 3 Physical Parameters for PFCFHE

Quantity	Fluid 1 (Hot)	Fluid 2 (Cold)
Fluids	Air	Air
Surface Index	16 fins/inch	20 fins/inch
Plate Spacing (mm)	3.17	2.54
Plate Thickness (mm)	0.152	0.152
Number of Stacks	43	43

Table 4: Different initial Flow field conditions for PFCFHE

No	Flow Quantity (Kg/S)		Inlet temperature (°C)		Flow Lengths (m)		Inlet Pressures (KPa)	
	Hot	Cold	Hot	Cold	Hot	Cold	Hot	Cold
1	0.575	0.544	355.5	56.1	0.216	0.139	389.55	55.158
2	0.756	0.756	627	65.5	0.244	0.183	389.55	103.422
3	0.756	0.504	627	65.5	0.244	0.183	389.55	103.422
4	1.008	1.008	627	65.5	0.244	0.183	389.55	172.37
5	1.008	1.008	627	65.5	0.305	0.244	389.55	172.37

Table 5: Results for Performance Evaluation of Plate fin Heat Exchanger

No	Heat Transfer Coefficients (W/m^{2}K)		NTU (Exp.)	Effectiveness (Exp.) (%)	Effectiveness (Corr.) (%)	Pressure Drop (KPa)	
	Hot	Cold				Hot	Cold
1	393.63	278.76	1.965	62.98	62.45	7.40	4.96
2	420.15	408.48	2.583	66.87	65.90	10.10	9.72
3	417.81	291.11	3.211	79.60	76.58	9.58	5.75
4	526.37	504.33	2.420	65.78	65.00	17.32	10.1
5	419.15	426.82	3.302	70.60	68.30	12.50	8.27

B. PERAMETRIC ANALYSIS of PFCFHE

The effectiveness was found out by varying the different physical parameters like, heat transfer area; free flow area etc. Fin density was varying from 16-20 fins per inch for both the sides. The effectiveness obtained by experimental data and by the correlations had best agreement for all operating conditions with an approximate variation of ±5% for any combinations of the above fin surfaces.

From Fig 3, it was observed that as the number of fin increases for hot side (keeping cold side fins fixed), the ratio of free flow area to heat transfer area (A_{fr}/A_h) decreases. So the effectiveness was increased. Same trends were also observed with cold side. It was also noticed that rectangular fins have higher A_{fr}/A_h than that of triangular fins which indicate triangular fins have higher heat transfer area and hence higher effectiveness (Fig 4).

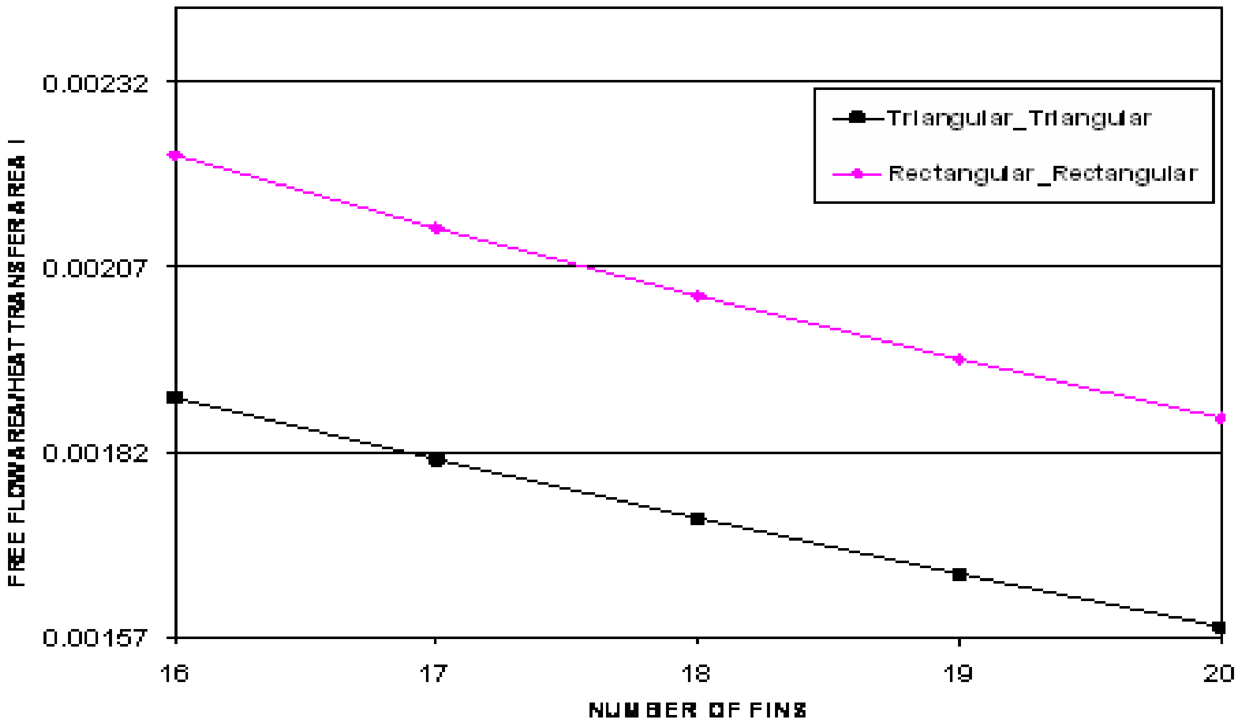

Fig. 3 Variation in Afr/Ah with Different Fin Configurations for Different Number Of Fins

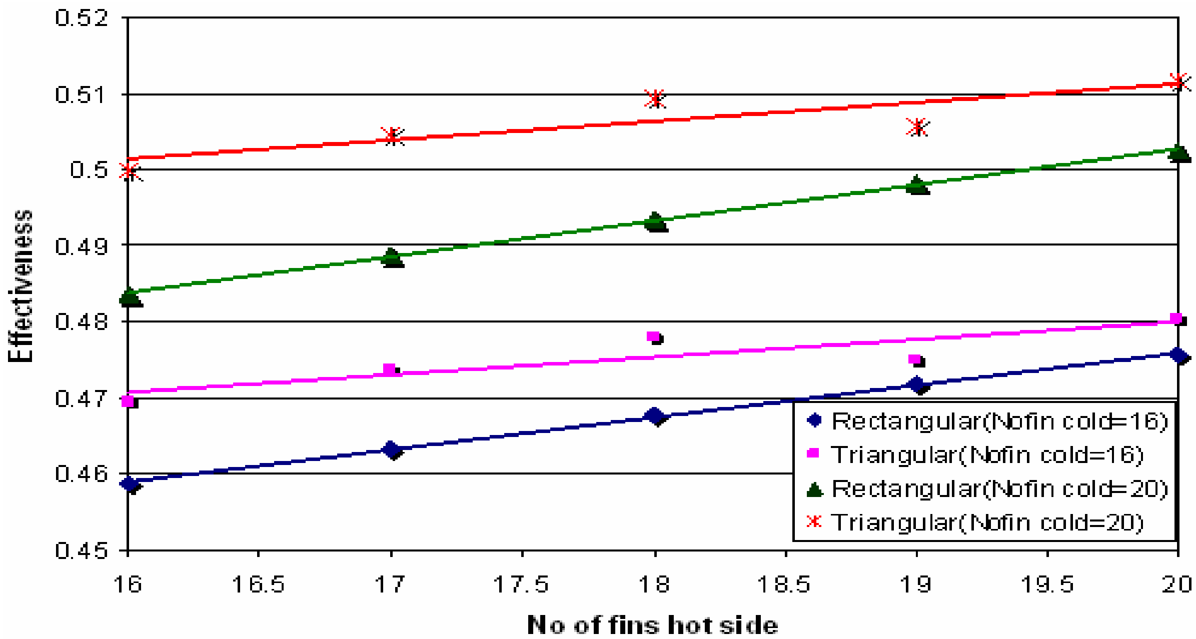

Fig. 4 Variation in Effectiveness for Different Number of Fins for Different Fin Configurations

According to the minimum and max range of area ratio, Fig 5 represents parametric study of effectiveness with respect to different hot side Stanton numbers, volume ratios (VRH) for offset strip fin having all the inlet conditions Fig 5 shows the combined effects of A_{fr}/A_h , Volume Ratio (VR) and Stanton Number (St No) on effectiveness. Range of area ratio was considered as 0.006 to 0.003, range of volume ratio was kept 0.36 to 0.8 and range of Stanton numbers was 0.004 to 0.023, while the same parameters for the cold side were kept constant as 0.00308, 0.6409 and 0.01247 respectively. Effectiveness was calculated from the correlation developed. For all the area ratios effectiveness increases with increase in Stanton number. This was due to increased heat transfer coefficient. It was noted that effect of changing VR was greater than that of $A_{fr}/A_{h.}$ However Triangular fin surfaces result in higher pressure drop compared to the rectangular fin surfaces (Fig 6). Therefore, more pumping power was required for triangular fin surface heat exchanger. Based on the developed correlation parametric analysis was carried out having fin density 20 FPI for both hot and cold sides. According to the

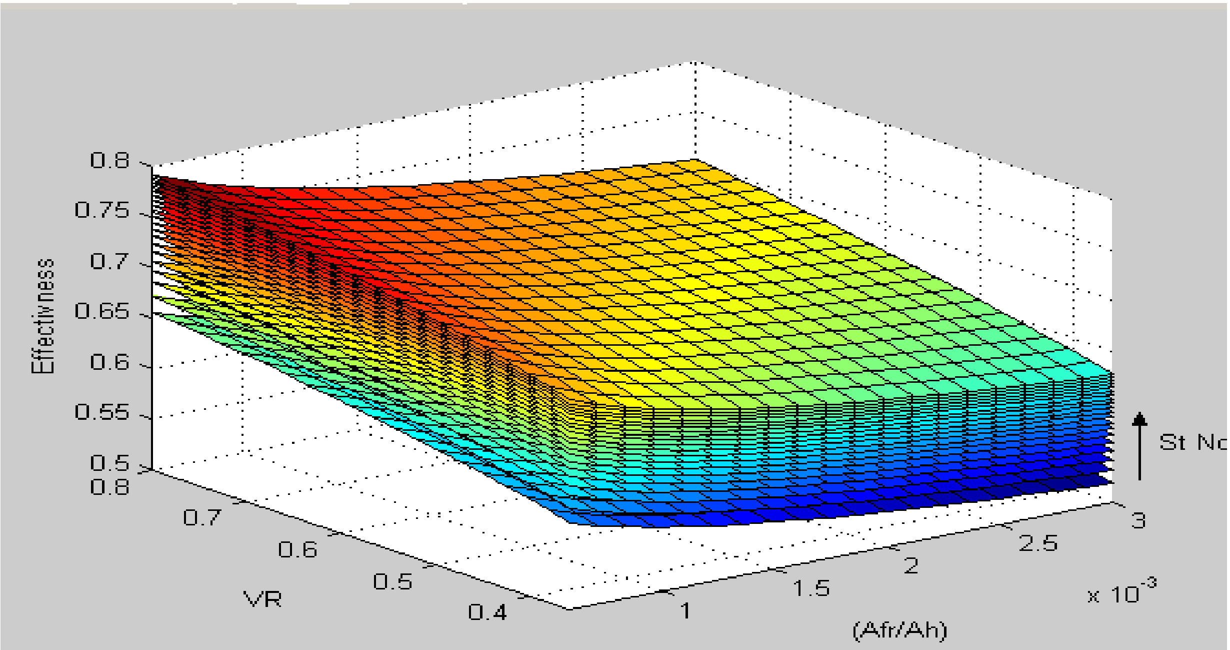

Fig. 5 Parametric Analysis for Effectiveness in Matlab

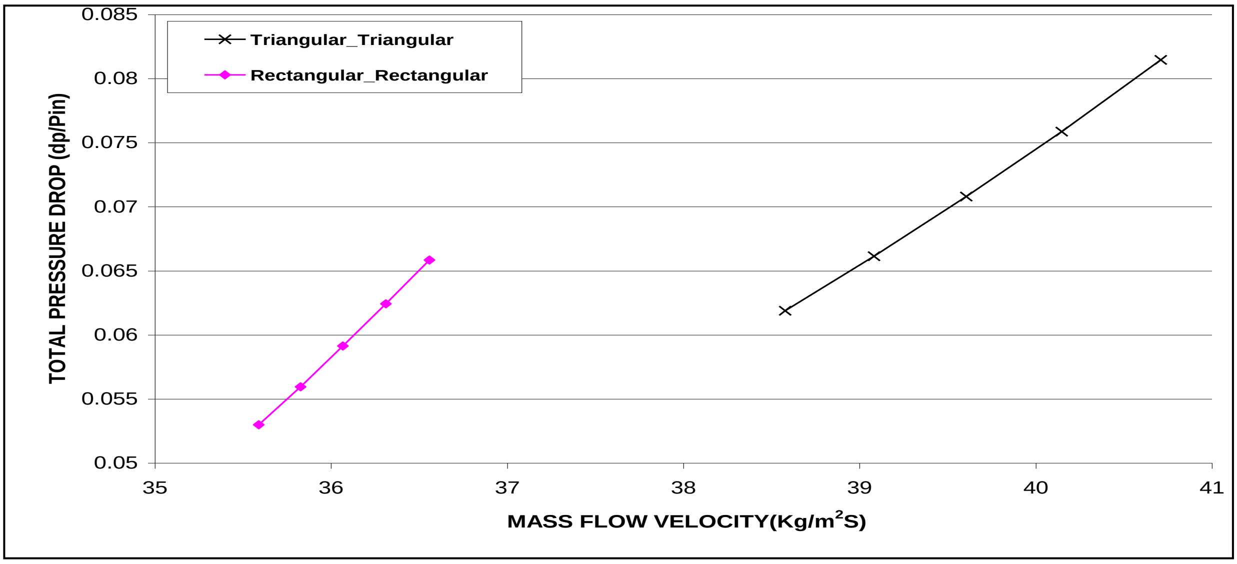

Fig.6 Variation in Total Pressure Drop for Triangular and Rectangular Fins

PFCFHE, designed as air-to-air and air-to-gas heat exchangers were analyzed thermally and hydraulically with different fin configurations such as, plain rectangular fin surface and triangular fin surfaces. the developed correlation gives the best approximation of effectiveness for the PFHE for different fin surfaces for different operating conditions with a maximum error of ±5% for any combinations of the plate fin surfaces. This correlation is valid only for working fluids, air and gas and having hot side fluid flow length varying from 0.2 - 0.516 m, cold side fluid flow length varying from 0.138 -0.438 m, and plate spacing for hot and cold fluid sides varying from 3 - 8 mm and 2.54 - 8 mm respectively.

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