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EXPERIMENTAL ANALYSIS OF PERFORMANCE PARAMETER OF PLATE FIN HEAT EXCHANGER

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***Abstract*— Compact heat exchangers, which have a high heat transfer surface area per unit volume of the exchanger, are among the most important elements of many cryogenic components. Compact heat exchangers are heat exchangers with a surface area density (β) more than 700 m2/m3 on one or more sides of a two-stream or multi-stream heat exchanger. Compact heat exchangers of the plate fin variety are frequently employed in the chemical, automotive, cryogenic, and space industries. Since plate fin heat exchangers are primarily employed in nitrogen liquefiers, they must be extremely successful since, if their efficiency falls below 87%, no liquid nitrogen is produced. Therefore, it becomes vital to evaluate these heat exchangers' performance before putting them into use.**

The heat exchanger test rig is used in the laboratory to test the readily accessible plate fin heat exchanger, which has rectangular offset strip geometry. The experiment is carried out under balanced conditions, meaning that the mass flow rates on both sides of the fluid stream are equal and that varied mass flow rates are used throughout. Finding the heat exchanger's efficiency for various. The literature contains a number of correlations that can be used to estimate the heat transfer and flow friction characteristics of a plate fin heat exchanger. The various performance parameters—such as efficiency, heat transfer coefficient, and pressure drop—that can be determined experimentally are then compared to the values derived from the various correlations. The effectiveness of heat exchangers, particularly cryogenic heat exchangers, is decreased by longitudinal heat conduction through walls; therefore, the effectiveness and overall heat transfer coefficient are determined by taking the effect of longitudinal heat conduction into account using Kroeger's equation.

***Index Terms*— Fin, Plate fins, LMTD, NTU, Effectiveness of fins.**

I. INTRODUCTION

Plate fin exchanger is a type of compact heat exchanger where the heat transfer surface area is enhanced by providing the extended metal surface interface between the two fluids and is called as the fins. Out of the various compact heat exchangers, plate-fin heat exchangers are unique due to their construction and performance. They are characterized by high effectiveness, compactness, low weight and moderate cost. As the name suggests, a plate fin heat exchanger (PFHE) 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. Separating plate acts as the primary heat transfer surface and the appendages known as fins act as the secondary heat transfer surfaces intimately connected to the primary surface. Fins not only form the extended heat transfer surfaces, but also work as strength supporting member against the internal pressure. The side bars prevent the fluid to spill over and mix with the second fluid. The fins and side bars are brazed with the parting sheet to ensure good thermal link and to provide the mechanical stability. Figure. 1.1 shows the exploded view of two layers of a plate fin heat exchanger. Such layers are arranged together in a monolithic block to form a heat exchanger.

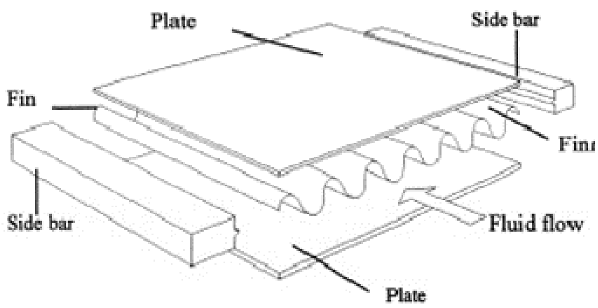

Fig. 1.1 Exploded view of a plate fin heat exchanger

II. LITERATURE REVIEW

Suzuki et al [1] in order to study the thermal performance of a staggered array of vertical flat plates at low Reynolds number has taken a different numerical approach by solving the elliptic differential equations governing the flow of momentum and energy. The validation of their numerical model has been done by carrying out experiments on a two dimensional system, followed by those on a practical offset strip fin heat exchanger. The experimental result was in good agreement with the performance study for the practical offset-strip-fin type heat exchanger in the range of Reynolds number of Re<800.

Tinaut et al [2] developed two correlations for heat transfer and flow friction coefficients for OSFs and plane parallel plates. The working fluid for OSF was engine oil and water was taken for analyzing the parallel plate channels. By using the correlations of Dittus and Boelter and some expressions of Kays and Crawford they obtained there correlations. For the validation of their results they compared there correlations with correlations of Weitng. Although there were some differences between the results but there correlations have been found acceptable upon comparing their results to the data obtained from other correlations.

Manglik and Bergles[3] carried an experimental research on OSFs. They investigated the effects of fin geometries as non dimensional forms on heat transfer and pressure drop, for their study they used 18 different OSFs. After their analysis they arrived upon two correlations, one for heat transfer and another one for pressure drop. The correlations were developed for all the three regions. They compared there results from the data obtained by other researchers in the deep laminar and fully turbulent regions. There correlations can be acceptable when comparing the results of the expressions to the experimental data obtained by Kays and London [16].

III. 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 sent here for its performance analysis and to establish the correlation for j and f factors, which is manufactured by APOLLO heat exchangers Mumbai, for BARC Mumbai. Figure 3.1 shows the test rig of 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.. This Project is basically an experimental set-up, which is build up for the thermal performance testing of the plate fin heat exchanger for studying its performance. The procured heat exchanger is an Aluminum Plate Fin Heat Exchanger and which was manufactured at Apollo Heat exchangers For BARC Mumbai. As per the information gathered from the BARC Mumbai this heat exchanger is designed for operating at high pressure and is to be used for low temperature applications. The properties such as effectiveness, NTU, overall heat transfer coefficient, Colburn factor j and skin friction co-efficient f etc are calculated in order to measure its performance.

Test Rig

Fig.3.1 Photograph of the Experimental Setup

IV. PERFORMANCE ANALYSIS

The main aim of present work is to calculate the performance parameters like, effectiveness, overall heat transfer coefficient of the plate fin heat exchanger. In order to find the performance of present heat exchanger a number of experiments were carried out at different mass flow rates and at different hot fluid inlet temperature under balanced flow.

Table 5.1 shows the experimentally observed data

Table 5.2 Performance of heat exchanger

Flow Rate (lit/min)	Mass flow rate (kg/s)	ε_h	ε_c	NTU	UA_0 W/K	Re_h	Re_c	ΔT_{Hotend}	$\Delta T_{Coldend}$
300	0.0057	89.902	83.749	13.64	80.95	298.67	199.72	8.73	5.87
400	0.0074	90.236	85.90	15.24	117.37	416.04	278.22	7.97	5.53
500	0.0103	91.134	86.920	15.00	160.89	542.44	362.75	7.44	5.05
550	0.0116	92.08	86.365	16.24	196.09	610.9	406.46	7.72	4.53
588	0.0127	92.001	86.247	15.70	207.64	668.83	445	7.64	4.45
650	0.0142	92.786	85.371	17.49	258.57	747.83	497.56	7.98	3.94
300	0.0057	88.108	83.064	11.76	69.74	280.24	186.48	4.3	3.02
400	0.0078	89.937	85.480	13.25	107.60	419.28	277.56	3.36	2.33
500	0.0101	88.48	86.814	9.79	102.89	491.54	399.77	3.48	3.04
600	0.011	90.004	86.597	11.28	135.41	702.63	465.13	3.46	2.58
650	0.014	90.572	85.114	12.00	174.87	789.66	525.48	3.49	2.21

In order to compare our experimental results with the values that are obtained from theoretical correlations, some graphs are plotted for which the experiment is conducted at different mass flow rates and at two different hot inlet temperatures of 66 and 96 . Some of the graphs are shown below:

5.2. Variation of Effectiveness with Mass Flow Rate

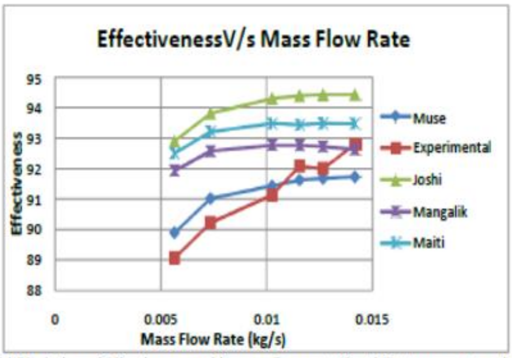

Fig. 5.1 Variation of effectiveness with mass flow rate (hot inlet temperature=96)

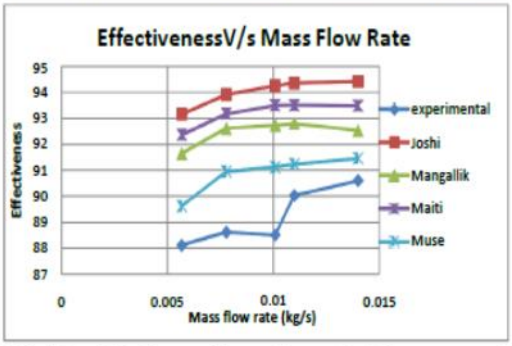

Fig. 5.2 Variation of effectiveness with mass flow rate (hot inlet temperature=66)

Figure 5.1 and 5.2 shows the variation of effectiveness obtained experimentally as well as with theoretical correlations and that obtained with simulation software Aspen with mass flow rate. It is seen that in both the cases effectiveness increases with mass flow rate. Experimental hot effectiveness first increases, then becomes almost constant for certain mass flow rates and then again increases. However from two figures it can be seen that the value of experimental effectiveness is more when hot inlet temperature is 96 as compared to effectiveness value when hot inlet temperature is 66 . So it can be concluded that with increase in hot inlet temperature effectiveness increases.

5.3. Variation of Overall thermal Conductance with Mass flow rate

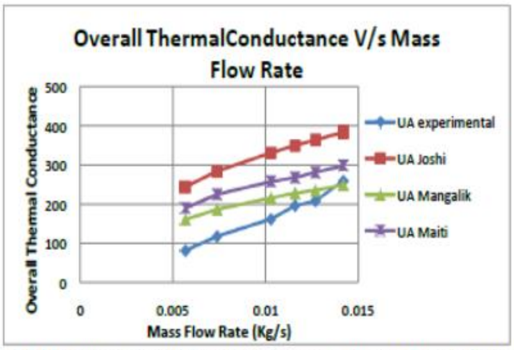

Fig. 5.3 Variation of overall thermal conductance with mass flow rate (hot inlet temperature of 96)

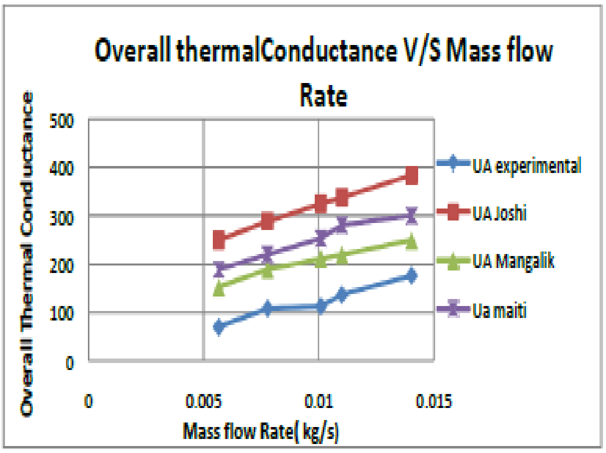

Fig. 5.4 Variation of overall thermal conductance with mass flow rate (hot inlet temperature of 66)

Figure 5.3 and 5.4 shows the variation of overall thermal conductance with mass flow rate for hot inlet temperature of 96 and 66 respectively. It can be seen that the theoretical as well as experimental overall heat transfer coefficient increases with increasing mass flow rate. It is due to the fact that with increasing mass flow rate the Reynolds number increases and as a result Colburn factor (j) also increases which is directly proportional to heat transfer coefficient, so overall thermal conductance increases.

5.4. Variation of Hot and Cold Effectiveness with Mass Flow Rate

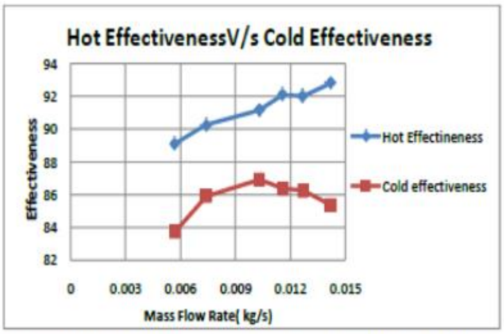

Fig. 5.5 Variation of Hot and Cold effectiveness with mass flow rate (hot inlet temperature of 96)

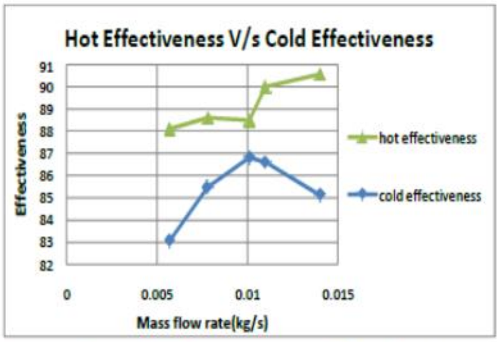

Fig. 5.6 Variation of Hot and Cold effectiveness with mass flow rat (hot inlet temperature of 66)

Figure 5.5 and 5.6 show how the experimental hot and cold effectiveness varies with the mass flow rate for hot inlet temperature of 96 and 66 respectively. It is seen that both hot and cold effectiveness increases with increasing mass flow rate and try to approach other and there is an optimum mass flow rate for each hot inlet temperature at which the gap between the two effectiveness is minimum and then again increases. Also after the optimum point the cold effectiveness again decreases, this is because a heat exchanger is designed for a particular mass flow rate and inlet temperatures at which it gives maximum effectiveness, after which its performance deteriorates. Also the imbalance increases because of heat loss to the environment as we are not able to provide the complete insulation. It can also be seen from the graphs that at lower hot inlet temperature the imbalance i.e. difference between the two effectiveness is less as compared to the imbalance at high temperatures.

V. CONCLUSIONS

The hot test is conducted to determine the thermal performance parameters of the available plate fin heat exchanger at different mass flow rates and two different hot inlet temperatures of 96 0C and 66 0C. An average effectiveness of 91% is obtained. It is found in both the cases that the effectiveness and overall thermal conductance increases with increasing mass flow rate It is also found that hot fluid effectiveness increases with flow rate of the fluid and agrees within 4% with the effectiveness value calculated by different correlations and that obtained by using the simulation software, Aspen. Also the pressure drop increases with increasing mass flow rate and experimental values are more as compared to theoretical results because the losses in pipes and manufacturing irregularities have not been taken in to account.

For a particular hot inlet temperature there is an optimum mass flow rate at which the difference between the hot and cold effectiveness of the heat exchanger is minimum and at this point the imbalance is also minimum. We found that the insulation which is provided in the heat exchanger has a significant effect on its performance. It is expected that the imbalance i.e. difference between the hot and cold end temperature can be brought to a minimum level if a perfect insulation like vacuum is provided.

VI. FUTURE SCOPE

Present tests are conducted at room temperatures and in future we can perform the experiment at low temperatures in order to check the performance of the present heat exchanger for Cryogenic applications. In cold testing air at about 100K will be used as the cold fluid. In cold test in place of heater a cold box will be used.

REFERENCES

[1] Suzuki, K., Hiral, 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.

[2] Tinaut F. V., Melgar A. and Rehman Ali A. A. 1992 Correlations for Heat Transfer and Flow Friction Characteristics of Compact Plate Type Heat Exchangers. International Journal of Heat and Mass Transfer. 35(7):1659:1665

[3] Manglik and Bergles A. E. 1995 Heat Transfer and Pressure drop Correlations for Rectangular Offset Strip Finn Compact Heat Exchangers. Experimental Fluid Science 10:171- 180.

[4] Hu S and Herold K. E. 1995a Prandtl Number Effect on Offset Strip Fin Heat Exchanger Performance: Predictive Model for Heat Transfer and Pressure Drop. International Journal of Heat and Mass Transfer 38(6) 1043-1051

[5] Hu S and Herold K. E. 1995b Prandtl number Effect on Offset Strip Fin Heat Exchanger Performance: Experimental Results. International Journal of Heat and Mass Transfer 38(6) 1053-1061.

[6] Zhang L. W., Balachandar S., Tafti D. K. and Najjar F. M. 1997. Heat Transfer Enhancement Mechanisms in Inline and Staggered Parallel Plate Fin Heat Exchanger. International Journal of Heat and Mass Transfer 40(10):2307-2325

[7] Dejong N. C., Zhang L. W., Jacobi A. M., Balchandar S. and Tafti D. K. 1998. A Complementary Experimental and Numerical Study of Flow and Heat Transfer in Offset Strip Fin Heat Exchangers. Journal of Heat Transfer 12:690:702

[8] Bhowmik H., Kwan- Soo Lee 2009. Analysis of Heat Transfer and Pressure Drop Characteristics in an Offset Strip Fin Heat Exchanger. International Journal of Heat and Mass Transfer 259-263

[9] Saidi A. and Sudden B. 2001. A Numerical Investigation of Heat Transfer Enhancement in Offset Strip Fin Heat Exchangers in Self Sustained Oscillatory Flow. International Journal of Numerical Methods for Heat and Fluid Flow. 11(7): 699-716

[10] Dong J., Chen J., Chen Z. and Zhou Y. 2007. Air Side Thermal hydraulic Performance of Offset Strip Fin Heat Exchangers Fin Alumunium Heat Exchangers. Applied Thermal Engineering 27:306-313

[11] Michna J. G., Jacobi A. M. and Burton L. R. 2005. Air Side Thermal- Hydraulic Performance of an Offset Strip Fin Array at Reynolds Number up to 12, 0000. Fifth International Conference on Enhanced Compact and Ultra Compact Heat Exchangers. Science, Engineering and Technology 8-14.