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To Determine the Thermal Parameters of Plate Fins Heat Exchanger

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ABSTRACT:- Plate heat exchangers are most critical component of many cryogenic components because the heat transfer surface area per unit volume of the exchanger is very high. The surface area density β is greater than 700m2/m3 of multi stream heat exchanger. Plate fins heat exchanger is widely used in automobile cryogenic space applications and chemical industries. It is mostly used for nitrogen liquefiers; the liquid nitrogen is not produced for they need to be highly efficient. The heat exchanger effectiveness is less than 86% so that it is necessary the heat exchanger effectiveness test before putting them in operation. The geometry of the available plate heat exchanger has rectangular off strip and heat exchanger test ring is used for its testing in the laboratory. The experiment is conducted under balanced condition i.e. the mass flow rate for both sides of fluid stream is same, and the experiment is carried out at different mass flow rates. The effectiveness of heat exchanger is found out for different mass flow rates. Various correlations are available in the literature for estimation of heat transfer and flow friction characteristics of the plate fin heat exchanger, so the various performance parameters like effectiveness, heat transfer coefficient and pressure drop obtained through experiments is compared with the values obtained from the different correlations. The longitudinal heat conduction through walls decreases the heat exchanger effectiveness, especially of cryogenic heat exchangers, so the effectiveness and overall heat transfer coefficient is found out by considering the effect of longitudinal heat conduction using the Kroeger's equation.

KEYWORDS:- heat exchanger, mass flow rate, heat transfer.

INTRODUCTION

A heat exchanger is a device built for efficient heat transfer from one medium to another. It is to transfer heat from a hot fluid to cold fluid across an impermeable wall. It is work on the principle is to facilitate an efficient heat flow from hot fluid to cold fluid. The heat is flow is a due to temperature difference between the two fluids, the area where heat is transferred, and the conductive/convective properties of the fluid and the flow state. This relation is based on the Newton's law of cooling, which is given in Equation.

$$Q = H*A* T$$

Where
h = heat transfer coefficient [W/m^2K]
A = heat transfer area [m^2]
T = temperature difference

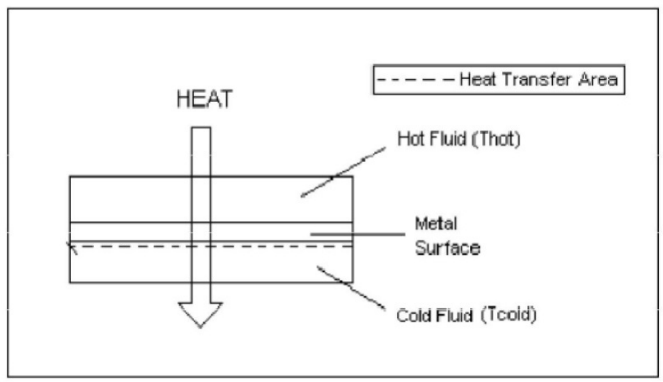


Fig.1. Basic heat transfer mechanism

They are widely used in space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, petroleum refineries, natural gas processing, cryogenics applications and sewage treatment. Heat exchangers are one of the essential components in engineering systems and the design and construction of heat exchangers is often vital for the proper functioning of such systems.

LITERATURE SURVEY

Patankar and Prakash [1] presented a two dimensional analysis for the flow and heat transfer in an interrupted plate passage which is an idealization of the OSFs heat exchanger. The main aim of the study is investigating the effect of plate thickness in a non-dimensional form t/H on heat transfer and pressure drop in OSF channels because the impingement region resulting from thick plate on the leading edge and recirculating region behind the trailing edge are absent if the plate thickness is neglected. Their calculation method was based on the periodically fully developed flow through one periodic module since the flow in OSF channels attains a periodic fully developed behaviour after a short entrance region, which may extend to about 5 (at the most 10) ranks of plates (Sparrow, et al. 1977). Steady and laminar flow was assumed by them between Reynolds numbers 100 to 2000. They found the flow to be mainly laminar in this range although in some cases just before the Reynolds no. 2000 there was a transition from laminar to turbulence. Especially for the higher values of t/H. They used the constant heat flow boundary condition with each row of fins at fixed temperature. They made there analysis for different fin thickness ratios t/H= 0, 0.1, 0.2, 0.3 for the same fin length L/H = 1, and they fixed the Prandtl number of fluid = 0.7. For proper validation they compared there numerical results with the experimental results of [London and Shah] for offset strip fin heat exchangers. The result indicate reasonable agreement for the f factors, but the predicted j factor are twice as large as the experimental data. They concluded that the thick [plate situation leads to significantly higher pressure drop while the heat transfer does not sufficiently improve despite the increased surface area and increased mean velocity.

Joshi and Webb [2] developed an analytical model to predict the heat transfer coefficient and the friction factor of the offset strip fin surface geometry. To study the transition from laminar to turbulent flow they conducted the flow visualization experiments and an equation based on the conditions in wake was developed.

Suzuki et al [3] 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 [4] 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 [17]. 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[5] 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.

Hu and Herold [6] presented two papers to show the effect of Prandtl no. on heat transfer and pressure drop in OSF array. Experimental study was carried out in the first paper to study the effect for which they used the seven OSFs having different geometries and three working fluids with different Prandtl number. At the same time the effect of changing the Prandtl number of fluid with temperature was also investigated. The study was carried out in the range of Reynolds number varying from 10 to 2000 in both the papers. The results of the two studies showed that the Prandtl number has a significant effect on heat transfer in OSF channel. Although there is no effect on the pressure drop.

Zhang et al [7] investigated the mechanisms for heat transfer enhancement in parallel plate fin heat exchangers including the inline and staggered arrays of OSFs. They have also taken into account the effect of fin thickness and the time dependent flow behaviour due to the vortex shedding by solving the unsteady momentum and energy equation. The effect of vortices which are generated at the leading edge of the fins and travel downstream along the fin surface was also studied. From there study they found that only the surface interruptions increase the heat transfer because they cause the boundary layers to start periodically on fin surfaces and reduce the thermal resistance to transfer heat between the fin surfaces and fluid. However after a critical Reynolds number the flow becomes unsteady and in this regime the vortices play a major role to increase the heat transfer by bringing the fresh fluids continuously from the main stream towards fin surface.

Dejong et al [8] carried out an experimental and numerical study for understanding the flow and heat transfer in OSFs. In the study the pressure drop, local Nusselt number, average heat transfer and skin friction coefficient on fin surface, instantaneous flow structures and local time averaged velocity profiles in OSF channel were investigated. They compared there results with the experimental results obtained by Dejong and Jacobi [1997] and unsteady numerical simulation of Zhang et al [1997]. There results indicate that the boundary layer development, flow separation and reattachment, wake formation and vortex shedding play an important role in the OSF geometry.

H. Bhowmik and Kwan-Soo Lee [9] studied the heat transfer and pressure drop characteristics of an offset strip fin heat exchanger. For their study they used a steady state three dimensional numerical model. They have taken water as the heat transfer medium, and the Reynolds number (Re)in the range of 10 to 3500. Variations in the Fanning friction factor f and the Colburn heat transfer j relative to Reynolds number were observed. General correlations for the f and j factors were derived by them which could be used to analyze fluid flow and heat transfer Characteristics of offset strip fins in the laminar, transition, and turbulent regions of the flow.

Saidi and Sudden [10] carried out a numerical analysis of the instantaneous flow and heat transfer for OSF geometries in self-sustained time-dependent oscillatory flow. The effect of vortices over the fin surfaces on heat transfer was studied at intermediate Reynolds numbers where the flow remains laminar, but unsteadiness and vortex shedding tends to dominate. They compared there numerical results with previous numerical and experimental data done by Dejong, et al. (1998).

The plate fins heat exchanger which is counter flow heat exchanger with off strip fins geometry is used on this test. The heat exchanger which manufactured by APOLLO Mumbai was sent here to determine its thermal performance parameter and to establish the correlation for j and f factor. the core dimension and thermal data is given on table 3.1 and 3.2 this experimental set- up which is made up to determine thermal performance testing of the plate fins heat exchanger and studying of its performance . this heat exchanger is kind of Aluminum plate fins which manufactured by APOLLO Mumbai. This heat exchanger was designed for operating at high pressure and to be used low temperature applications. The various property such as over all heat transfer coefficient effectiveness NTU colburn factor j and skin friction coefficient factor f etc are calculated and measure its performance.

CORE DATA

Table 3.1(a) Dimensions of procured plate fin heat exchanger

	INTERNAL (HOT SIDE)	EXTERNAL (COLD SIDE)
FIN	OSF	OSF
NO. OF PASSAGE	5	6
NO. OF PASS	1	1
FLOW RATE	COUNTER FLOW	COUNTER FLOW

CORE SIZE

Table 3.1(b) Dimensions of procured plate fin heat exchanger

FLOW LENGTH/EFFECTIVE FLOW LENGTH	1000mm/900mm
TOTAL HEIGHT	105mm
TOTAL WIDTH/EFFECTIVE WIDTH	86mm/74mm

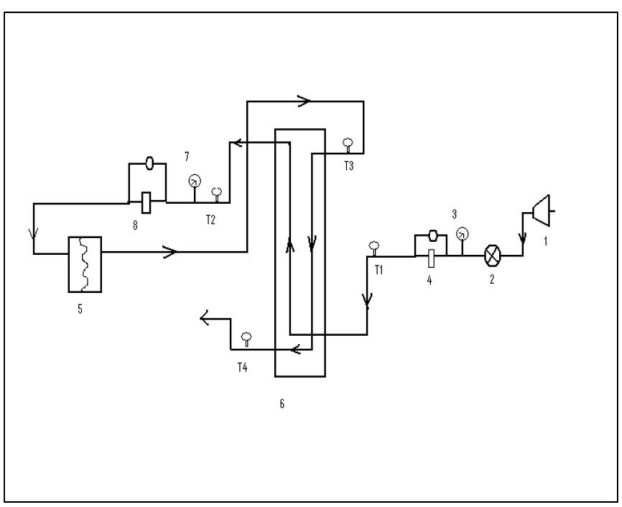
THERMAL DATA

Table 3.2 Procured design data of plate fin heat exchanger

HEAT LOAD	5.5 KW	
	HOT SIDE	COLD SIDE
FLUID	HELIUM (HP)	HELIUM (LP)
FLOW RATE	5 g/s	4.8 g/s
INLET TEMP.	36.65 ^{0}C	-189.45 ^{0}C
OUTLET TEMP.	-177.25 ^{0}C	24.39 ^{0}C
PRESSURE DROP	0.003 kg/cm^2	0.02 kg/cm^2
OPERATING PRESSURE	7.35 kg/cm^2	7.05 kg/cm^2

Test Rig

1: Compressor
2: Control Valve
3,7: Pressure Taps
4,8: U- Tube manometer
5: Heater 6:Test section
T_1, T_2, T_3, T_4 are RTD's



Estimation of average temperature

Since the fluid outlet temperatures are not known for the rating problem, the average fluids mean temperatures have to be predicted first. The fluid properties at the predicted mean temperatures of 342.02 K and 338.71 K for hot and cold fluid are obtained from property package, Gaspak.

The properties of hot gas at the mean temperature,

Specific heat, C_p □ 1.04086 KJ/kgK
Viscosity, □ □ 0.0000198 $Pa-s$
Prandtl number, Pr □ 0.7169882 Density, ρ □ 1.14249kg/m^3
The properties of cold gas at the mean temperature are, Specific heat, C_p □ 1040.67 KJ/kgK

Viscosity, □ □ 0.0000192 $Pa-s$
Prandtl number, Pr □0.7167 Density, ρ □ 1.321kg/m^3

Overall heat transfer coefficient and NTU

The overall heat transfer coefficient is given by

$$\frac{1}{U_O A_0} = \frac{1}{\eta_{oh} h_h A_h} + \frac{a}{K_w A_w} + \frac{1}{\eta_{oc} h_c A_c}$$

where

A_w= lateral conduction area =W×L×(2N_p +2)= 0.074× 0.866 (2×5+2) = 0.786

$$\frac{1}{U_O A_0} = \frac{1}{0.763\times142.76\times4.354} + \frac{0.0008}{165\times0.786} + \frac{1}{0.87593\times179.9486\times5.2152}$$

$(U_O A_O)_h$ □ 300 W/K

Overall heat transfer coefficient

$$\text{Uoh} = \frac{U_O A_{O)} h}{Aoh} = \frac{300}{4.354} = 68.96$$

$$U_{oc} = \frac{U_O A_O}{A_{oc}} = \frac{300}{5.452} = 55.025$$

Number of transfer units

$$N_{tu} = \frac{U_O A_O}{C_{min}} = \frac{300}{9.8876} = 30.34$$

Effectiveness of heat exchanger without considering the effect of longitudinal conduction

$$E = \frac{1-e^{-N_{tu}(1-C_r)}}{1-C_r e^{-N_{tu}(1-C_r)}} = \frac{1-e^{-30.34(1-0.9999)}}{1-0.9999e^{-30.34(1-0.9999)}} =$$

= 0.96804

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.

CALCULATIONS

The temperatures values which are obtained experimentally are firs of all corrected using the calibration chart, and also the pressure values are converted in units of Pa or bar, and then

used for further calculations.
T_1 = 39.82°c T_2 = 88.83 °c T_3 = 96.66°c T_4 = 43.48°c P_1 = 1.21 bar, P_2 = 1.18 bar,

Flow rate 550 ltr/min
1 Mass flow rte = Volume flow rate $\frac{p_4}{RT_4}$

$$= \frac{550\times10^{-3}}{60} \times \frac{1.01325\times10^5}{287\times316.98} = 0.01\text{kg/s}$$

2 Heat capacity of hot and cold fluids,

$C_c = m_c \times C_{pc}$ =0.010087KW/K
For hot fluid
$C_h = m_h \times C_{ph}$KW/K
3 Capacity rate ratio= $C_r = \frac{C_{min}}{C_{max}}$ = 0.9994

4 Effectiveness ε_h= 92.08
ε_c = 86.365

5 Number of transfer units, NTU = 15.009
After considering the effect of longitudinal heat conduction. Same steps as described in Chapter 4 are followed, but here the NTU value is assumed in such a way that the effectiveness obtained from Kroger's equation matches with the experimental value of effectiveness.

6. Overall Heat transfer conductance,UA_0
UA_0 = NTU ×C_{min}
= 15.009×0.010087 = 160.898 W/K

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. °c. Some of the graphs are shown below:

Variation of Effectiveness with Mass Flow Rate

Effectiveness V/s Mass Flow Rate

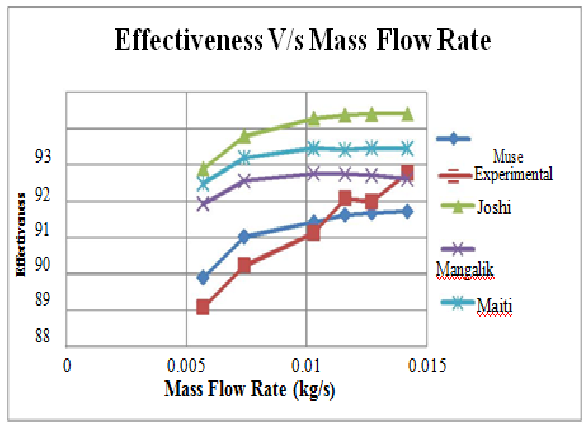


Fig. 5.1 Variation of effectiveness with mass flow rate (hot inlet temperature=96

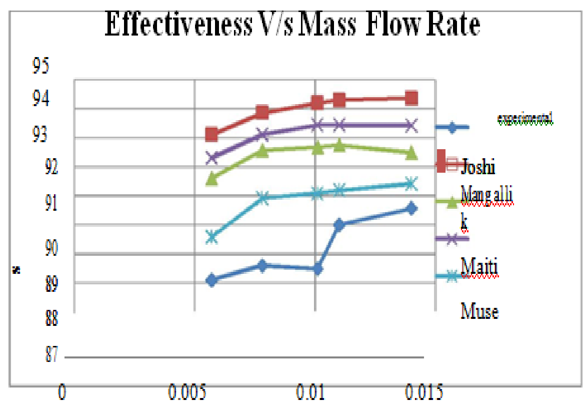


Fig. 5.2 Variation of effectiveness with mass flow

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°c as compared to effectiveness value when hot inlet temperature is 66°c. So it can be concluded that with increase in hot inlet temperature effectiveness increases.

Variation of Overall thermal Conductance with Mass flow rate

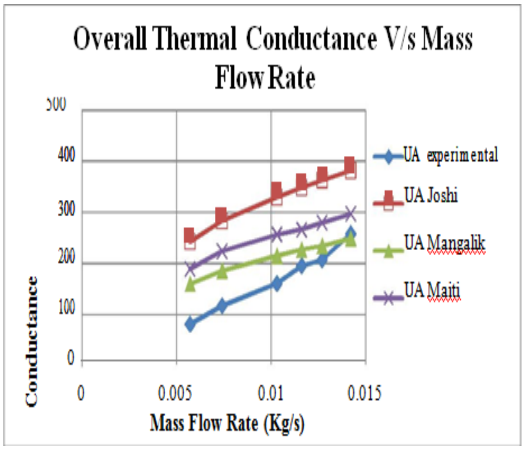


Fig. 5.3 Variation of overall thermal conductance with mass flow rate (hot inlet temperature of 96°c)

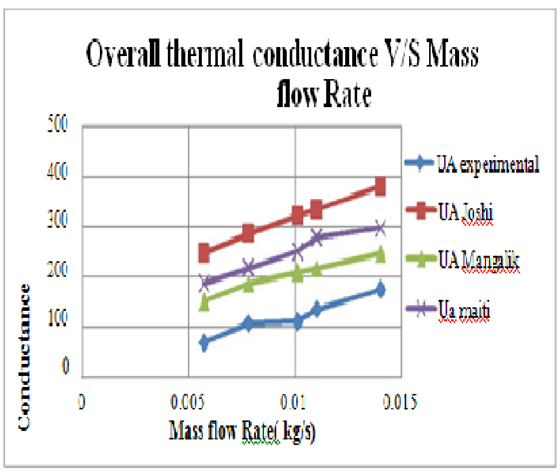


Fig. 5.4 Variation of overall thermal conductance with mass flow rate (hot inlet temperature of 66°c

Figure 5.3 and 5.4 shows the variation of overall thermal conductance with mass flow rate for hot inlet temperature of 96°c and 66°c 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.

Variation of Pressure Drop with Mass Flow Rate

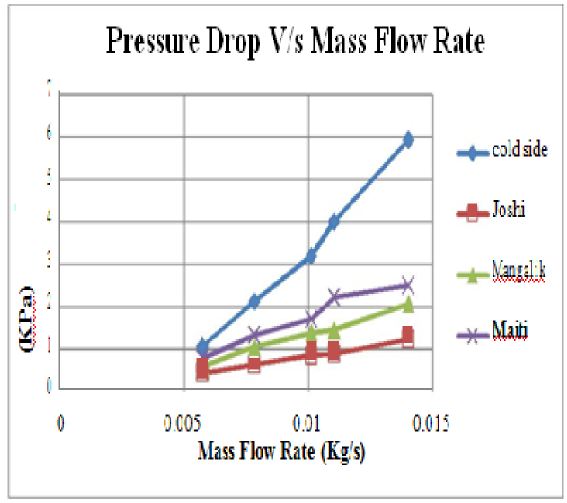


Fig. 5.5 Variation of pressure drop with mass flow rate

Figure 5.5 shows how the pressure drop in the heat exchanger varies with varying mass flow rate and also the comparison between experimental and theoretical pressure drop. It can be seen that the pressure drop increases with mass flow rate for each case. However the experimental pressure drop is much more as compared to the theoretical pressure drop because in theoretical calculations we have not taken in to account the pressure drop taking place in piping's and also the manufacturing irregularities and header losses.

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 and 66_. 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.

Scope for Future Work

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.

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