Experimental Investigation and CFD Analysis Performance of Fin and Tube Heat Exchanger with Different Types of Fins

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Abstract- **Air-side heat transfer and friction characteristics of five kinds of fin-and-tube heat exchangers, with the number of tube rows (N = 12) and the diameter of tubes (Do = 18 mm), have been experimentally inves-tigated. The test samples consist of five types of fin configurations: crimped spiral fin, plain fin, slit fin, fin with delta-wing longitudinal vortex generators (VGs) and mixed fin with front 6-row vortex-generator fin and rear 6-row slit fin. The heat transfer and friction factor correlations for different types of heat exchangers were obtained with the Reynolds numbers ranging from 4000 to 10000. It was found that crimped spiral fin provides higher heat transfer and pressure drop than the other four fins; fin tube heat exchangers are mostly employed to gas to liquid heat exchangers; it is an effective way to reduce the air side thermal resistance which often accounts for about 90% of the overall thermal resistance. Fin and Tube heat exchangers are widely used in various engineering fields such as heating, ventilation and air conditioning, refrigeration, automobiles and air intercoolers. Thus the study of behavior of fin and tube heat exchangers becomes quite necessary. . The heat transfer and friction factor correlations for different types of heat exchangers were obtained with the Reynolds numbers ranging from 4000 to 10000. It was found that crimped spiral fin provides higher heat transfer and pressure drop than the other four fins. The air-side performance of heat exchangers with the above five fins has been evaluated under three sets of criteria and it was shown that the heat exchanger with mixed fin (front vortex-generator fin and rear slit fin) has better performance than that with fin with delta-wing vortex generators, and the slit fin offers best heat transfer performance at high Reynolds numbers. Based on the correlations of numerical data. . The heat transfer performances for optimized vortex-generator fin and slit fin at have been com-pared with numerical method.**

Keywords: Plain Fin, Mix Fin, Slit Fin

I. INTRODUCTION

Fin-and-tube heat exchangers are extensively employed in chemical engineering, refrigeration, and HVAC (heating, ventilation and air conditioning) applications such as compressor intercoolers, air-coolers and fan coils. The dominant thermal resistance is usu-ally on the air side in practical applications, and therefore the use of finned surfaces on the air side is very common to effectively improve the overall thermal performance of heat exchangers. These finned surfaces include crimped spiral fin, plain fin, slit fin and fin with delta-wing longitudinal vortex generator and so on. A Heat Exchanger is a device to transfer heat from hot fluid to cold fluid across an impermeable wall. Fundamental of Heat Exchanger principle is to facilitate an efficient heat flow from hot fluid to cold fluid. This heat flow is a function of the temperature, difference between the two fluids, the area where heat is transferred, and the conductive/convective properties of fluid and the flow state this relation was formulated by Newton and called as Newton's law of cooling, which is given in equation (1.1)

$$Q = h*A*\Delta t$$

Where h:-heat transfer co-efficient [w/m^2k], where fluids conductive/convective properties and the flow state comes in the picture,

A:-heat transfer area [m^2] , and

Δt:-temperature difference[k].

Fins are used to increase the effective surface area of heat exchanger tubing. Finned tubes are used when the heat transfer coefficient on the outside of the tubes is appreciably lower than that on the inside. Various types of fins can be used to increase the effective heat transfer such as plain fin, slit fin, spiral fin, fin with delta wing vortex generator fin. The aim was to find the best suitable fin for a considered application. Various researchers have done remarkable work in the field and have analyzed ,various possibilities in different working condition.

II. EXPERIMENTAL SYSTEM AND PROCEDURE

The experiments are conducted in an open wind tunnel. The system consists of two loops: air loop and steam loop. The air loop is provided to blow air across the finned bundles of test core, and the steam loop is designed to supply slightly superheated stream through the tubes of test core. The extended (finned) surfaces are prepared for the test core, which are placed in the test section. The steam–air system is employed for accomplishment of steam-to-air heat exchange, as schematically shown in Fig. 1.

Air is induced to the wind tunnel by a centrifugal blower. The inlet air temperature and the temperature difference between in-let and outlet through the test core are measured

by two sets of multi-point T-type copper-constantan thermocouple grids. Each set contains 16 calibrated thermocouples within the accuracy of 0.1 LC, and the junctions of thermocouples are connected in series to give a single reading. Steam is generated in an electrically heated boiler, of which the power can be adjusted by six trans-formers. After flowing through the over-heater, the steam is superheated to provide the desired 1–3 LC of superheat. The superheated steam temperature is directly read with an ethanol thermometer, which is inserted in the hole that designed in the steam inlet header.
The wall static pressures before and after the test
cores are measured by a U-tube water column manometer, and the super-heated steam pressure is read by a U-tube mercury column manometer. The air velocity is measured by a Pitot-tube meter, which is located in the flow metering duct far downstream of the test core. The Pitot-tube meter is connected to the inclined draft Five kinds of fin-and-tube heat exchangers are tested in this study. The slit fin pattern in this paper was popularly investi-gated, and the geometry of slit fin is based on the available re-ported results by Wang et al. [5–7]. The geometry of VGs is based on the available reported results by Fiebig [21]. These re-sults [21]

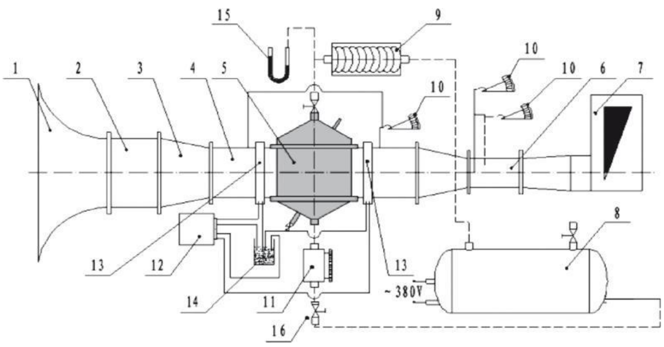


Fig.1 Schematic of Experimental Facility

Problem Defination

Various researchers have thoroughly studied the flow pattern over various types of heat exchangers; whereas these studies were mainly focused on small tube diameter and small tube rows, which are usually used in condensers or evaporators in refrigeration engineering. However in some application such as intercooler of multistage compressor, the number of tube rows and outside diameter of tubes are large therefore, it is essential to investigate the heat transfer and friction characteristics of fin and tube heat exchangers with large tube diameter and large number of tube rows.

In the present study air side heat transfer and friction performance of large number of tube rows(N=12) and large tube diameter(D_0 =18) are numerically studied and compared

Objective of the Study

The main objective of this study is to compare the effect of different types of fin in finned tube heat exchanger i.e. to evaluate their performance parameters in a cross flow fin and tube heat exchangers.

III. PLAIN FIN,SLIT FIN,FIN WITH VORTEXGENRATOR

3.1 PLAIN FIN

Provided a j factor independent of fin spacing but a friction correlation *f* increasing as fin spacing decreased. Yan and sheen concluded that both heat transfer and pressure drop increase as fin spacing decreases, from fin spacing 2.0 to 1.4 mm. disagreement in the literature as to effect of fin spacing may be due to geometrical variation and experimental uncertainties; however most investigators concluded that f is higher for a smaller fin spacing and j is relatively independent of fin spacing.[1][2][3][4]

Modified and extended by Wang and clang (1988) to include a broader set of data, and that correlation and *f* factor correlation by Wang et al.(1996) are usefulbecause of the wide range of parameters covered. In order to enhance generality abumadiet.al (1988) may be useful. Rich (1973) and Wang.et.l (1996) reported that heat transfer and friction factors do not depend strongly on fin spacing. Gray and Webb (1986)

3.2 SLIT FIN

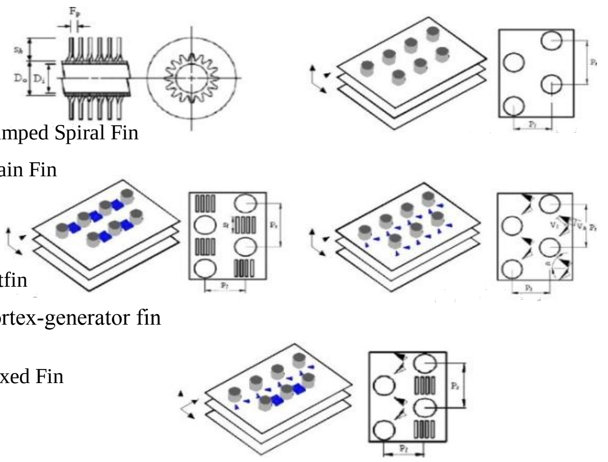
Fin spacing may have stronger effect on slit fin patterns. Wang et.al (1999b) confirmed both heat transfer coefficient and pressure drop decreased as fin spacing increased. As with other geometries there is a relatively small effect of number of tube rows on friction factor and decreased j factor for an increasing number of tube rows.[12]

3.3 FIN WITH VORTEX GENERATOR

In recent years vortex generators such as fins,ribs, wings,etc. have been successfully used for heat transfer enhancement of the modern thermal systems. Vortex generators form secondary flow by swirl and destabilize the flow. They generate the longitudinal vortices and create rotating and secondary flow in the main flow which can raise turbulent intensity, mix the warm and cold fluid near and in the center of channel and increase the heat transfer in the heat exchangers. Different types of vortex generators such as rectangular and triangular are available.

In 1994 firebug et al experimentally compared the effect of vortex generator on the heat transfer and flow losses in heat exchangers for the Reynolds number between 600-3000 for the staggered fin-tube arrangement, their results showed that the heat exchanger element with round tubes and vortex generator increases heat transfer only 10%but about 100% in flat tube. They also showed that pressure drop in flat tube bank with vortex generator is nearly half than that of round tube.

The effect of two pairs of vortex generator in a flat tube bank was experimentally studied by wang.et.al. They showed that heat transfer enhancement in flat tube bank with and without vortex generator was 47.5 % in constant mass flow. 41.4% in constant power consumption; and 37.5 for constant pressure drop.[6][8][10]



Crimped Spiral Fin

Plain Fin

Slitfin

Vortex-generator fin

Mixed Fin

IV. DATA REDUCTION

The main purpose of data reduction is to determine the air side heat transfer and friction characteristics, Nusselt number (Nu) or Colburn factor (j) and friction factor (f) of the heat exchangers from the experimental data, which are recorded at steady-state conditions during each test run, and to find out corresponding power-law correlations of Nu vs. Re, and f vs. Re for each case. The average value of the inlet and outlet temperatures of air side is used to evaluate the thermal properties of air, and the thermal properties of steam are determined by the steam pressure.

The saturated steam gauge pressure in the tubes is around 200 mm Hg. The steam-side heat transfer rate Q_{steam} is given as

Q steam ¼ msteamrsteam ð1Þ

where m_{steam} is the vapor mass flowrate, and r_{steam} is the latent heat of steam at the corresponding pressure. The air-side heat transfer rate Q $_{air}$ is given as

Q air ¼ maircp;airÞtair ð2Þ

where m_{air} is the air mass flowrate.

The total heat transfer rate is defined as the average of the air side and the steam-side heat transfer rates

Q $_{ave}$ ¼ ðQ $_{steam}$ þ Q $_{air}$Þ=2: ð3Þ

The total heat transfer coefficient, UA product, is calculated from the following relationship:

UA ¼ Q $_{ave}$=Dt$_m$ ð4Þ

Where Dt$_m$ is the logarithmic-mean temperature difference, defined by

t ðt$_s$_ t$_{in}$Þ _ ðt$_s$_ t$_{out}$Þ 5

D m ¼ ln ts _tin ð Þ

ts −tout

Where t_{in} is the inlet temperature of air, t_{out} is the outlet tempera-ture, and t_s is the saturated temperature of steam at the correspond-ing pressure.

V. METHODOLOGY

Present work is investigated numerically using commercial CFD software ANSYS Fluent. The work consists of

- Selecting the computation domain and associated governing equations along with boundary conditions.
- Creating the required 3D computational domain in ANSYS Workbench Design Modeler.
- Meshing the model using ANSYS Workbench.
- Select the solver formulation.
- Chose the basic equation to solved: laminar or turbulent (or in viscid), heat transfer models.
- Specify the material properties.
- Specify the boundary conditions.
- Adjust the solution control parameter.
- Initialize the flow field.
- Calculate a solution.
- Examine the results.
- Save the result

VI. MATHEMATICAL MODELING AND NUMERICAL IMPLEMENTATION

This chapter deals with the mathematical and numerical aspects of the present problem. Under mathematical modelling, assumptions made in the formulation of the problem, applicable governing equation and associated boundary conditions needs to be applied, have been discussed. Numerical implementation discusses and lists all models, values of constants required to run the simulation along with all pre-processing activities of solid modelling and meshing.

6.1Mathematical Modeling

Mathematical modelling is the actual representation of any system in mathematical form. For this simple approximations and idealizations are employed. Different laws such as physical laws and conservation laws such as mass, momentum and energy conservation laws are employed. In this present chapter for solving the present problem the assumptions made are given. The basic governing equations required are given. Boundary conditions and computational domain are discussed. For solving the problem numerically solid model, meshing and required solver set up is discusse

7.1 Numerical simulation for vortex-generator fin

7.2 Physical model

In our previous study [30], we indicated that heat transfer and friction performance for the fin with vortex generators is indepen-

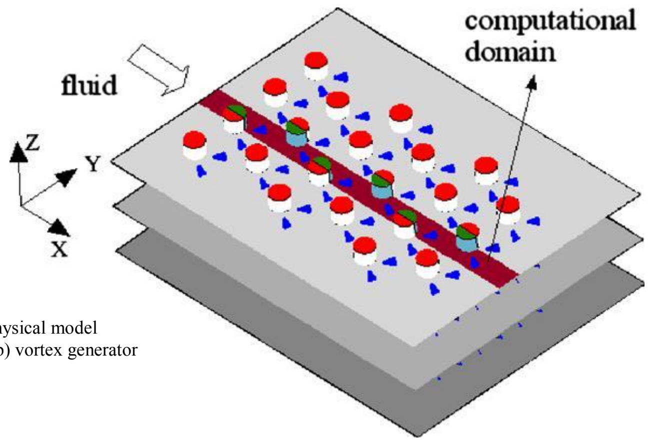
dent of the number of tube rows when N P 6. The]

schematic dia-gram of the fin with vortex generators is shown in Fig. 8 with six rows of tubes in the flow direction. Table 1 lists all the geometric dimensions of the heat exchanger. The air flow direction is x-direction, fin span-wise direction is y-direction and fin thickness direction is z-direction. The actual length of the computational domain is 7.5 times of the length of vortex-generator fin. That is, the do-main is extended 1.5 times of the length of vortex-generator fin for the entrance section to ensure the inlet uniformity, and at the exit, the domain is extended 5 times of the length of vortex-generator fin in order to make sure that the exit flow boundary has no flow recirculation.

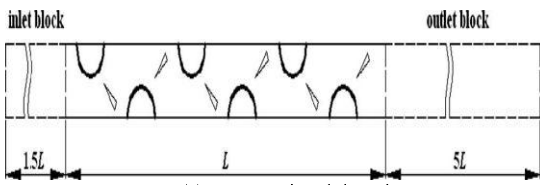
Due to the geometrical symmetry of the flow domain, only one-half of the heat exchanger element has been computed. The flow is assumed to be steady, turbulent and no viscous dissipation. The fluid is considered incompressible ideal gas with constant physical properties. The dimensionless equations for continuity, momentum and energy may be expressed in tensor rotation as

r _ ðqV/Þ ¼ r _ ðC/grad/Þ þ S/:

In the above equation, the dependent variable, /; stands for the velocity components, temperature, k and e, $C_/$ and $S_/$ represent the appropriate diffusion coefficients and the source terms, respectively **(a)**



physical model
(b) vortex generator



(c) computational domain
Schematic diagram of vortex-generator fin.

7.3 SOLID MODELING

Models are constructed to study the air side performance of fin and tube heat exchangers

TABLE 1. Observations Pressure drop

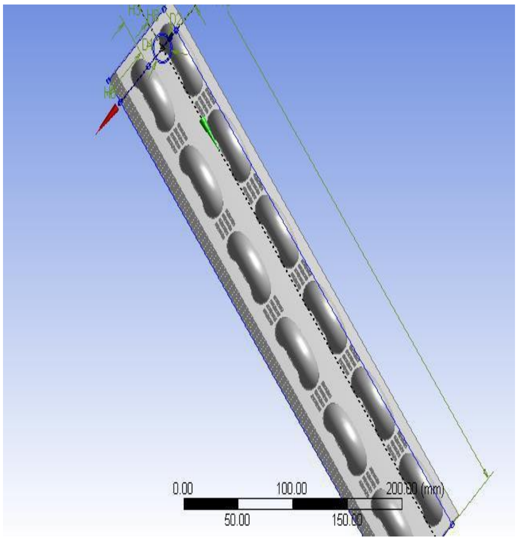
Reynolds number	4000	5205	6506	7808	9109
Plain fin	41.56	69.84	103.95	145.72	194
slit fin	41.61	62.05	94.02	137.37	174.6
Vortex generator fin	33.42	57.42	86	121	163.04

Table 2.Temperature Outlet

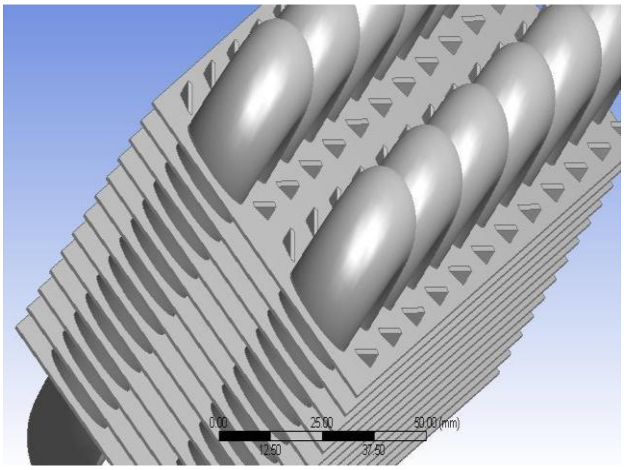
Reynolds number	4000	5205	6506	7808	9109
Plain fin	613	615	599.31	593	590.26
slit fin	302	450	443.93	478.74	436.38
Vortex Generator fin	728.33	713.41	699.69	688.7	682.72

Table 3. Frication Factor

Reynolds number	4000	5205	6506	7808	9109
Plain fin	0.351	0.331	0.316	0.307	0.306
slit fin	0.351	0.294	0.285	0.289	0.270
Vortex generator fin	0.282	0.272	0.261	0.255	0.252



Geometric dimensions of fin tube heat exchanger:-Internal diameter (D_i):- 16 mm
Outer diameter (D_o):- 18 mm Fin thickness:- 0.5 mm
Fin collar outside diameter (D_c):- 19 mm
Fin pitch: - 4.1 mm
Transverse tube pitch:
Longitudinal tube pitch: - 34
Dimension of slit:-
Slit length:- 12 mm
Slit width:- 2.2 mm
Slit spacing:- 2



Enlarged view of finned tube heatex changer with vortex generator

VII. CONCLUSIONS

In the present study, the air-side heat transfer of five kinds of fin-and-tube heat exchangers have been experimentally investi-gated with the Reynolds number ranging from 4000 to 10000, and the optimization of heat exchanger with VGs is also addressed. The main conclusions are summarized as follows:

(1)Before optimization, at high Reynolds numbers, the heat exchanger with slit fin offers best heat transfer performance.

(2) The larger attack angle, higher length and smaller height of vortex generators will lead to better overall performance of
Heat exchangers with VGs.

(3) As Reynolds number increases; the pressure drop increases & fluid outlet temperature decreases. That concludes that with
increase in Reynolds number the disturbances are increased and the heat transfer reduces

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