CFD Analysis on Flow Through Plate Fin Heat Exchangers with Perforations

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Abstract—This paper aims at optimizing the performance of plate fin heat exchangers with the help of perforations using FLUENT software. The widespread use of the heat exchangers design has ensured that there are numerous dimensional variations and shown that changes in dimensional parameters affect the performance. It is then important to understand how the geometry of compact heat exchanger can affect its performance. Therefore an investigation into the parametric effect on the global performance on types of plate fin heat exchangers (plain fin, circular and elliptical perforated fin, strip offset fin with and without perforations) are modeled and simulated at same boundary conditions (low Reynolds number). From the results, the heat transfer behaviour, Nusselt number, j and f factors are analyzed and compare all the parameters for different types of plate fin heat exchangers.

Index Terms—Fluent, heat transfer, modeling, perforated fin, plate fin heat exchanger (PFHE), strip offset fin.

I. INTRODUCTION

Plate fin heat exchangers are widely used in automobile, aerospace, cryogenic and chemical industries. They are characterized by high effectiveness, compactness (high surface area density), low weight and moderate cost. Although these exchangers have been extensively used around the world for several decades, the technologies related to their design and manufacture remain confined to a few companies in developed countries. Recently efforts are being made in India towards the development of small plate fin heat exchangers for cryogenic and aerospace applications. Its focus, however, is on the basic heat transfer and flow friction phenomena applicable to all plate fin heat exchangers. In order to best match the heat transfer and pressure drop in PFHE, many experimental and theoretical studies have been performed to understand the heat transfer flow behavior in various fin geometries at similar boundary conditions. It is well known that heat transfer in thermal entry region is more compared to thermal end region. The results based on the simplified model with short computational fin length cannot properly reflect the overall heat transfer coefficient of a practical PFHE. Moreover, in order to best match the heat transfer coefficient and pressure loss in the PFHE fins, the local distributions of j and f factors along the fins, especially at flow-interrupted locations, need to be well understood.

Kays and London [1] carried out early experimental investigations on various fin geometries. For this paper, experimental data is taken from reference [2].

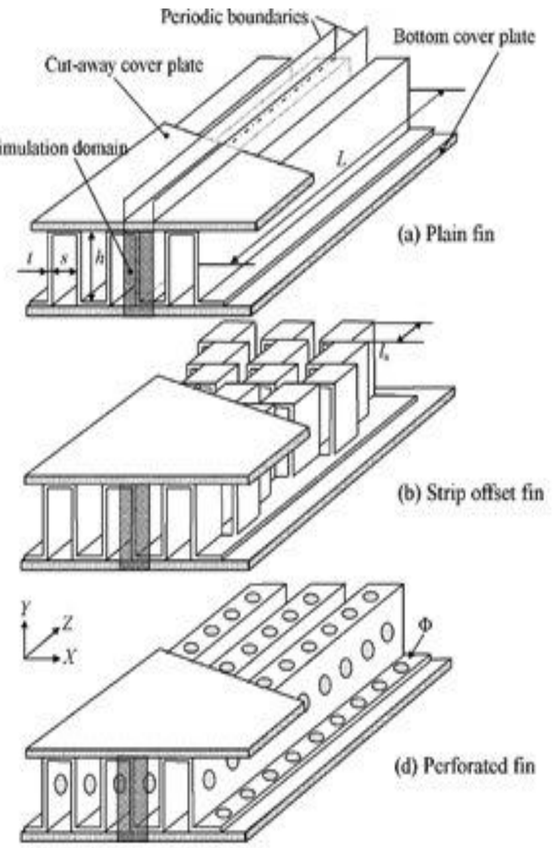

Fig. 1 Perspective view of PFHE

The purpose of this paper is to study the pressure drop and heat transfer in the PFHE fins, the fin models are created in proper three dimensional calculation domains and finely meshed. The flow and heat transfer process in the fins are obtained and represented with the local pattern and distribution of the j, f factors and Nusselt number. The behaviour of heat transfer enhancement in the flow-interrupted fins are carefully illustrated and discussed.

II. MODEL APPROACH

CFD MODELS: The perspective view of three-dimensional structures of the basic PFHE fins, plain fin, and strip offset fin, and perforated fin, is shown in Fig.1.1. In order to study the heat transfer and pressure drop behaviors and make a comparison among those fins, some dimensions are equal such as the fin thickness, fin height, fin spacing distance, and total fin array length, and they are listed in Table 1 in detail.

To accurately simulate the characteristics of heat transfer and pressure drop and at the same time simplify the modeling and computation processes, the following considerations are made in the present work.

A proper fin length of 0.306 m is selected. The length is not very long so as to simplify the modeling and computation. For example, the studied strip offset fin has 50 periodic offset fins and the thermal entry region is about 3-9 periodic offset fins long with Re 285. Therefore, the flow and heat transfer behaviors in both thermal entry and fully developed regions can be investigated.

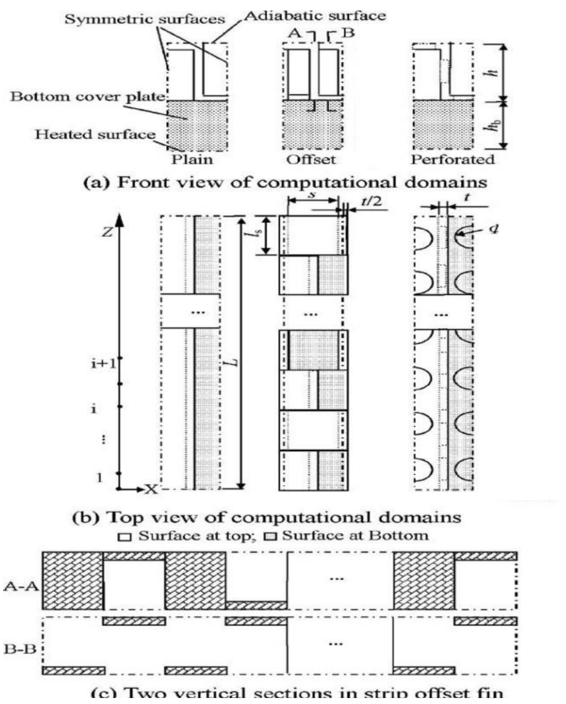

Fig. 2 CFD models of the plate fins

Symmetry boundaries, as shown in Fig. 1, are used to reduce the extent of each computational model to a symmetric subsection of the overall fin.

Not only the fin thickness in the X-direction but also the thickness of top and bottom fins in the Y-direction is considered in creating fin models.

The fins models are created in three-dimensional domains in ANSYS workbench, as schematically shown in Fig. 2, the fins are finely meshed. In order to obtain faster computation speed, only the locations with significant flow changes such as velocity boundaries and flow-interrupted places are with concentrated grid density. Each fin model is meshed up to grid convergence.

SOLUTION ALGORITHM: The working fluid is water, thermodynamics properties of which are assumed constant in the present studies: density = 983 kg/m^3, viscosity = $4.7{*}10{-}4$ Pa s, Thermal conductivity k = 0.655 W/m K, and specific heat Cp =4180J/kg K. The Reynolds number in the present work is 285, while the Prandtl number is fixed at 3. Both fins and bottom cover plates are made of aluminum with a thermal conductivity of 206 W/m K. The height of the bottom cover plate is 2.5 mm.

On inlets of the fins, the velocity inlet condition with an inlet temperature of 60^0C is applied and listed in Table 2. The pressure outlet condition is adopted on the fin outlets. The bottom surface of the cover plate is modeled in the constant heat flux boundary condition with a constant heat flux of 30,000 W/m^2. No slip wall condition is used and the symmetry boundary condition is applied on the two side surfaces of the fins.

Table 1 Fin Geometries

Fin type	Fin thickness t (mm)	Fin height h (mm)	Fin spacing distance s (mm)	Fin array length (mm)	Specifications
Plain fin	0.152	2.26	1.52	306	-
Perforated fin (circular)	0.152	2.26	1.52	306	Dia = 0.8 mm
Strip offset fin	.152	2.26	1.52	306	Pitch = 7.6 mm
Perforated fin (elliptical)	0.152	2.26	1.52	306	minor dia = 0.8mm

Major hole diameter for elliptical perforations is varies in 4 aspect ratios.

Elliptical perforations of fixed vertical minor axis diameter =0.8mm

Elliptical perforations, minor diameter to major diameter aspect ratio = (1:1.25), (1:1.5), (1:1.75), (1:2)

Table 2 Inlet Fluid Velocities for Fins

Reynolds number	Plain fin	offset fin	Perforated fin
285	0.075	0.076	0.0735

The commercial code FLUENT Version 15.0 is employed in the present numerical simulation. A model with steady-state three-dimensional incompressible laminar flow is employed for the solution of the problem. The maximum residual tolerance of the conservation equations is kept at less than $1e^{-5}$.

To analyze the heat transfer and flow behaviors in the fins, Colburn factor, j, and Fanning friction factor, f, are computed from the CFD simulation results by the following procedure.

Colburn factor, j, used for the evaluation on heat transfer performance, is defined as

Equation (1) ------------ $$J = \frac{Nu}{RePr^{1/3}}$$

The pressure drop can be expressed in terms of the friction factor, f, defined as

Equation (2) -------------f = $\dfrac{P_L D_h}{2u^2 L}$

Where,

P_L is the pressures drop of the fin array in the flow direction.

III MODEL VALIDATION

Plain fin modeled and simulated at boundary conditions mentioned above, and obtained computed results are validated with experimental results as in reference number [2], similarly Circular perforated fin is modeled and simulated; Elliptical perforated fin in different aspect ratios are modeled and simulated; Strip offset fin with and without perforations also modeled and simulated in workbench.

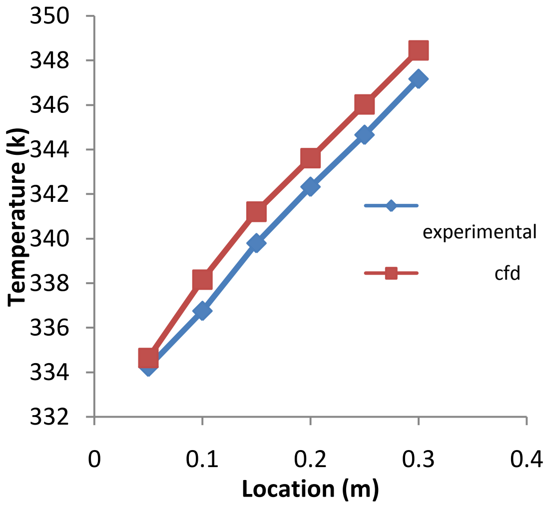

Fig. 3 Temperature distributions along the flow direction of plain fin

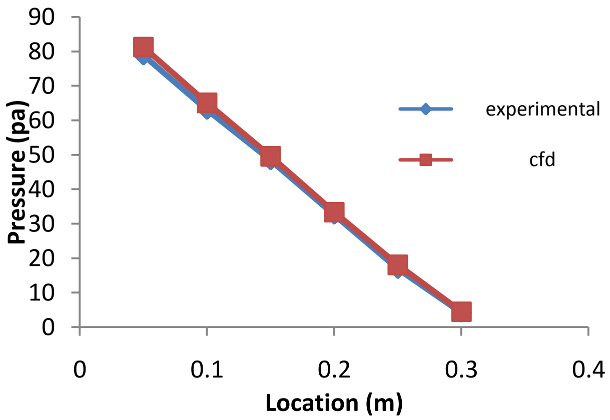

Fig. 4 Pressure distributions along the flow direction of plain fin

In validation fig: 3, 4; some error will be exempted, between the computed results and experimental results. Taken it as an experiment setup error.

IV. RESULTS AND DISCUSSIONS

Global average j and f factors. The global average j and f factors of the fins are calculated by Equation 1a, 1b; the plain fin has the lowest j in tested Reynolds number due to the heat transfer in other three flow-interrupted fins enhanced by thermal boundary layer interruption with offset strip fins, holes. Results show that j factors of the strip offset is high, and the strip offset fin with elliptical perforations has the highest j factor at Re 285 and it has the greatest pressure loss especially at large values of Re.

Local heat transfer and pressure behaviors. Further insights into the heat transfer behaviors in the fins are carried out by examining variations of the local stream wise average j factors along the flow direction, as presented. In all cases, heat transfer is enhanced at both the fin entrance and exit due to the thermal entry effect and end effect. As Re increases, the thermal entry effect enhances but the end effect becomes weak.

The fluid is blocked by leading surfaces and separated by tailing surfaces, resulting in j factor increases at these locations. As we know from results in previous two-dimensional models, the flow in the strip offset fin is not periodic in two strip fin lengths but four strip fin lengths, obviously because the fin thickness in the Y-direction is considered in the present simulation and the results are compared in four cases as shown below.

The above results shows the, validity of the plain fin simulation model is verified by comparing the computed results with the corresponding experimental data Ref [2], and by using the similar boundary conditions ,all the other types of plate fin heat exchangers are modeled and simulated. Finally all the results for plate fin heat exchangers are compared as cases below.

Case 1: Plain fin vs. circular perforated

Case 2: Circular perforated fin vs. Elliptical perforated fin

Case 3: Plain fin vs. Strip offset fin

Case 4: Strip offset fin vs. strip offset fin with perforations

a) Comparison of Pressure distribution along the flow direction for different fins

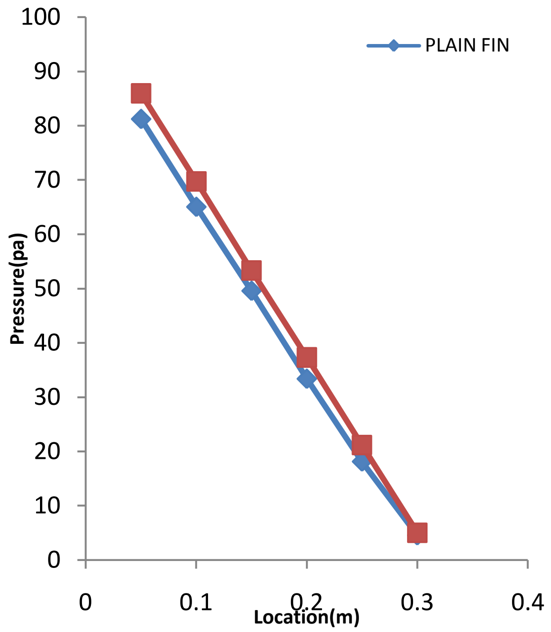

Fig. 5 Comparison of pressure distribution for plain fin and circular perforated fin

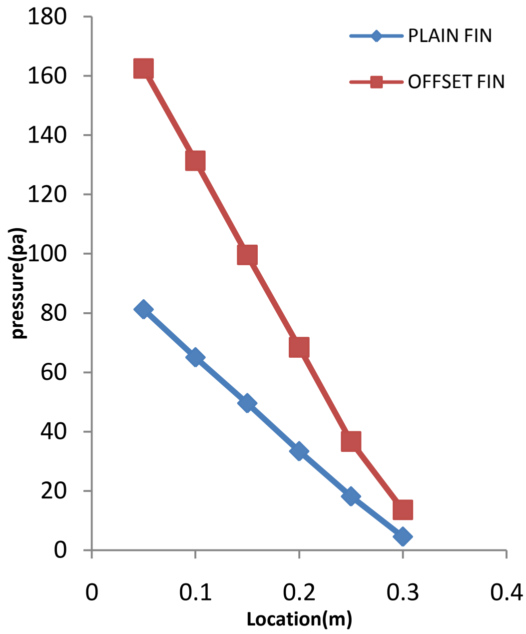

Fig. 7 Comparison of pressure distribution for plain fin and offset fin

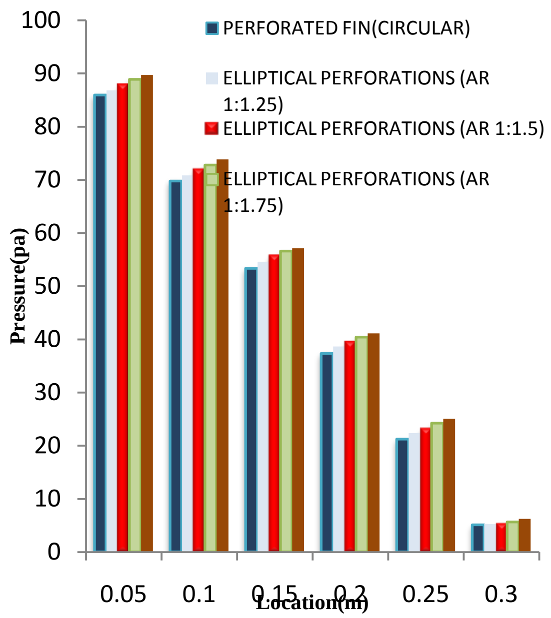

Fig. 6 Comparison of pressure distribution for circular and elliptical perforated fins

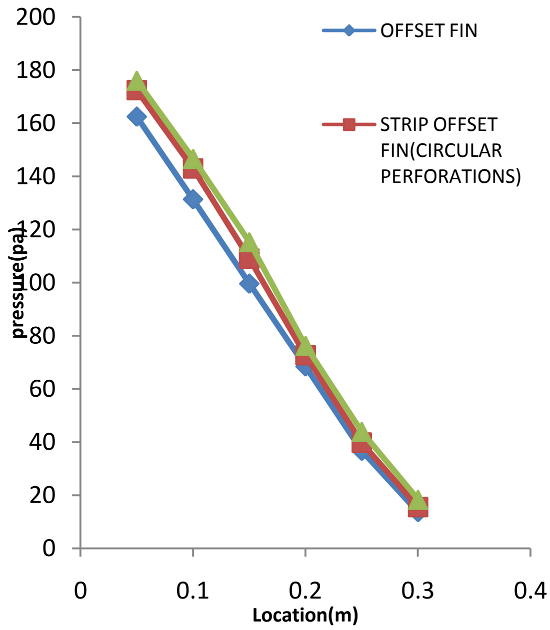

Fig. 8 Comparison of pressure distribution for offset fin and offset fin with perforations

b) Comparison of Temperature distribution along the flow direction for different fins

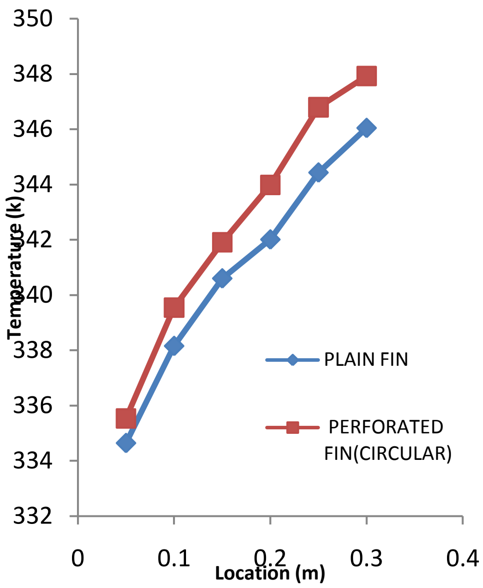

Fig. 9 Comparison of temperature distribution for plain fin and circular perforated fin

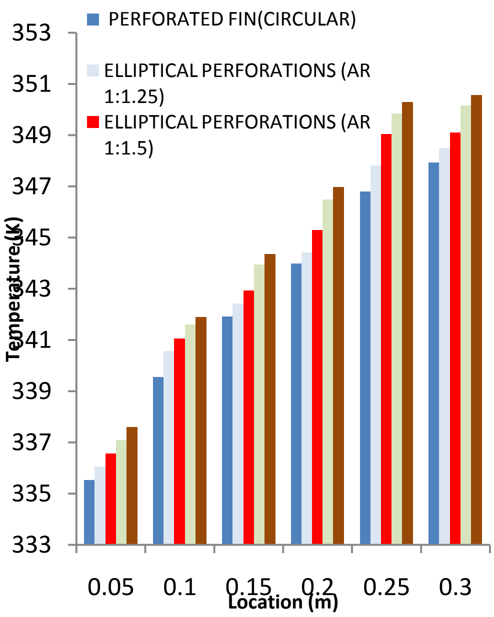

Fig. 10 Comparison of temperature distribution for circular and elliptical perforated fins

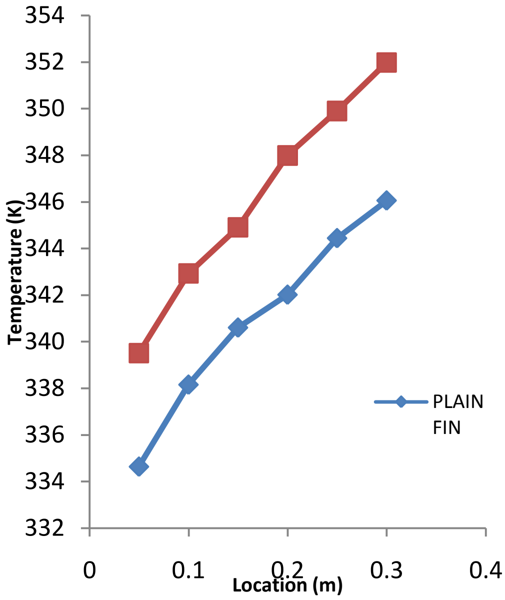

Fig. 11 Comparison of temperature distribution for plain fin and offset fin

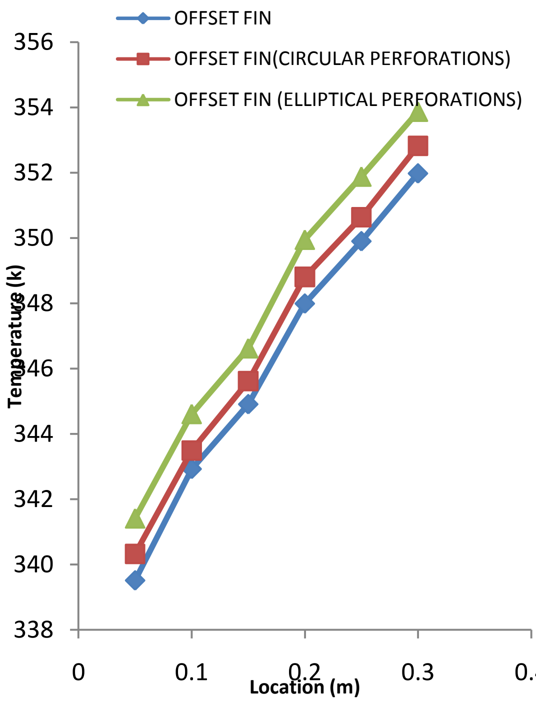

Fig. 12 Comparison of temperature distribution for offset fin and offset fin with perforations

c) **Comparison of Local Nusselt number along the flow direction for different fins**

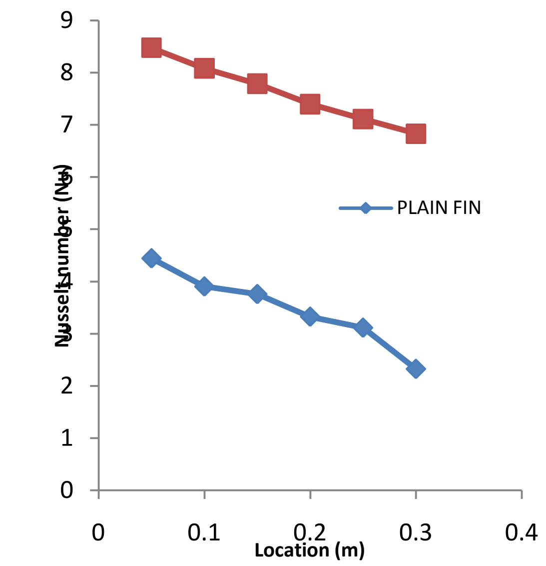

Fig. 13 Comparison of Nusselt number distribution for plain fin and circular perforated fin

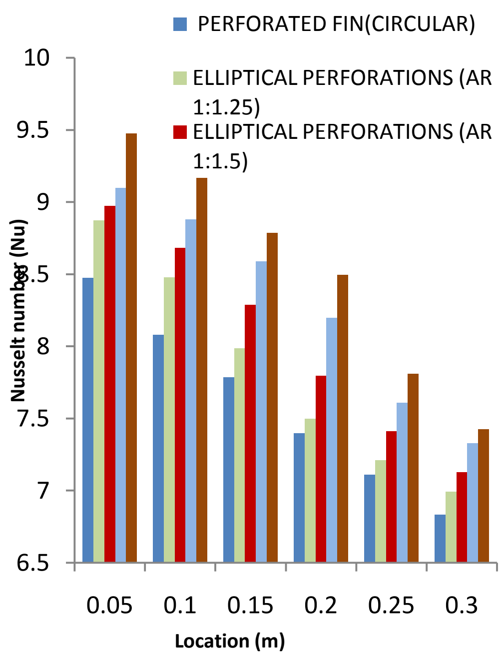

Fig. 14 Comparison of Nusselt number distribution for circular and elliptical perforated fins

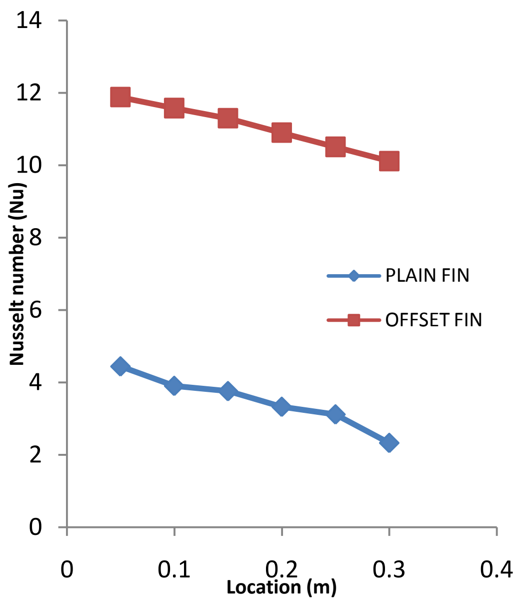

Fig. 15 Comparison of Nusselt number distribution for plain fin and offset fin

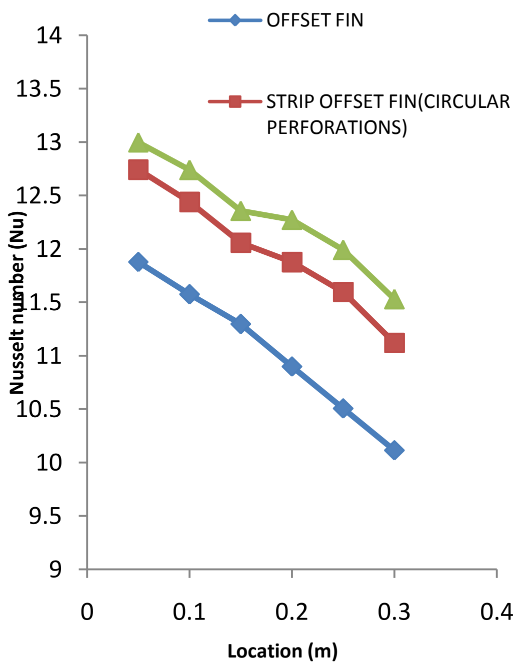

Fig. 16 Comparison of Nusselt number distribution for offset fin and offset fin with perforations

d) Comparison of colburn j factor along the flow direction for different fins

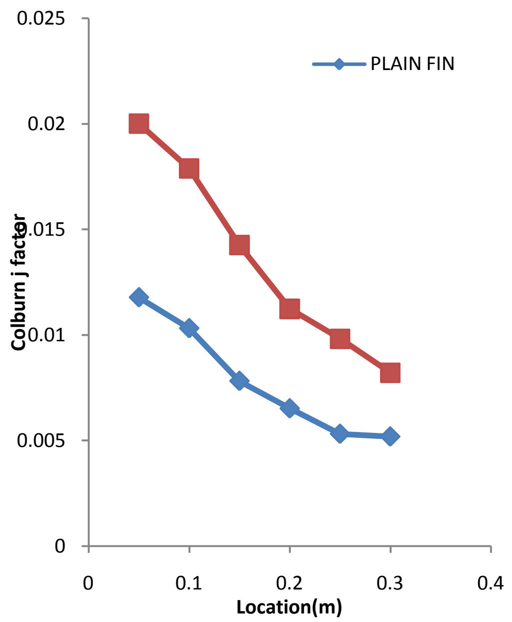

Fig.17 Comparison of colburn factor distribution for plain fin and circular perforated fin

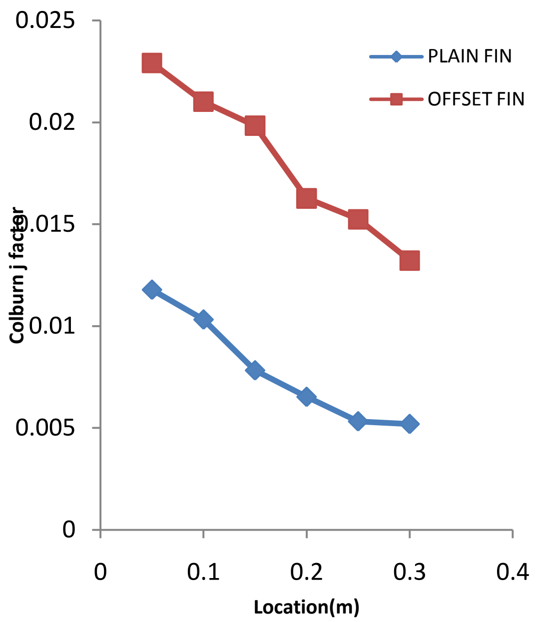

Fig. 19 Comparison of colburn factor distribution for plain fin and offset fin

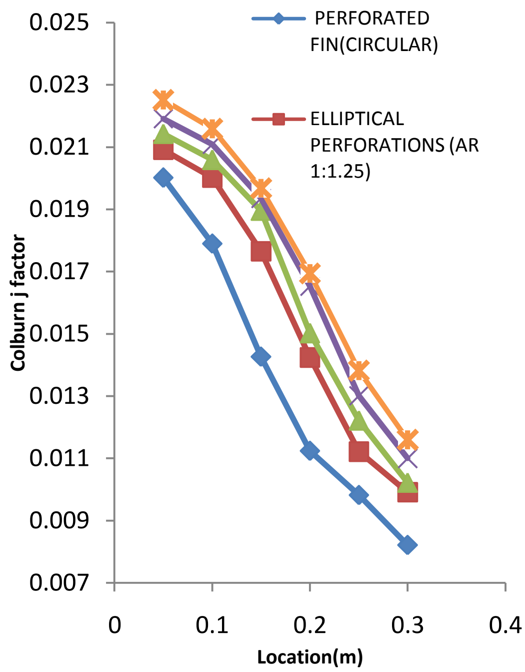

Fig. 18 Comparison of colburn factor distribution for circular and elliptical perforated fins

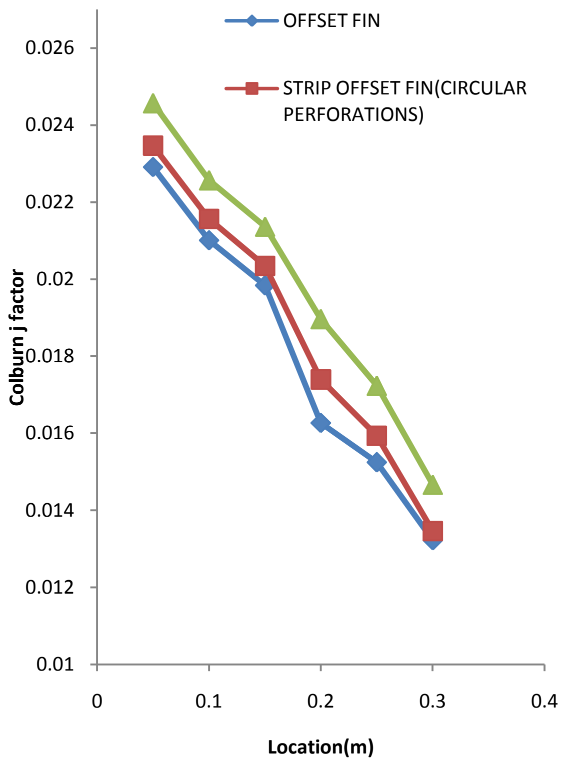

Fig. 20 Comparison of colburn factor distribution for offset fin and offset fin with perforations

e) Comparison of friction factor along the flow direction for different fins

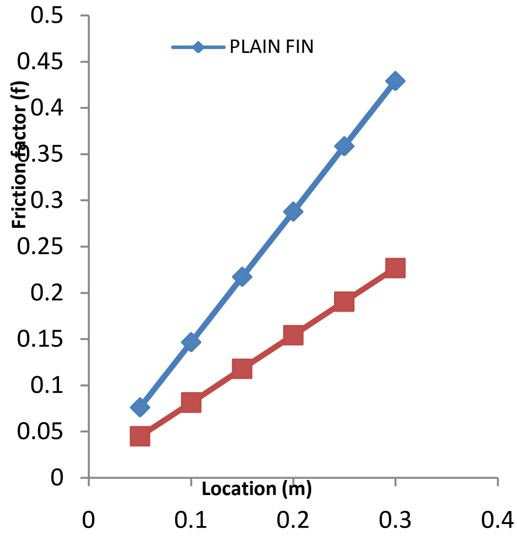

Fig. 21 Comparison of friction factor distribution for plain fin and circular perforated fin

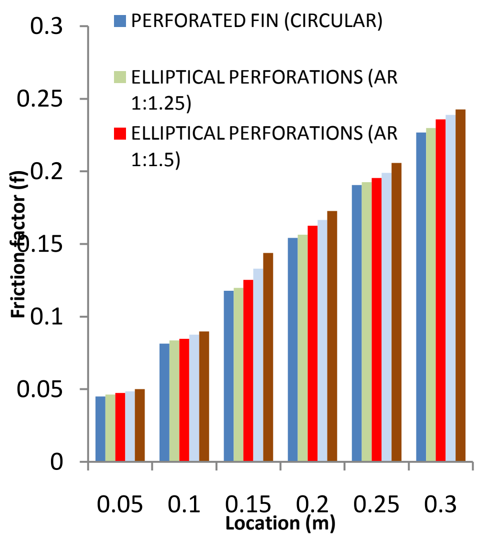

Fig. 22 Comparison of friction factor distribution for circular and elliptical perforated fins

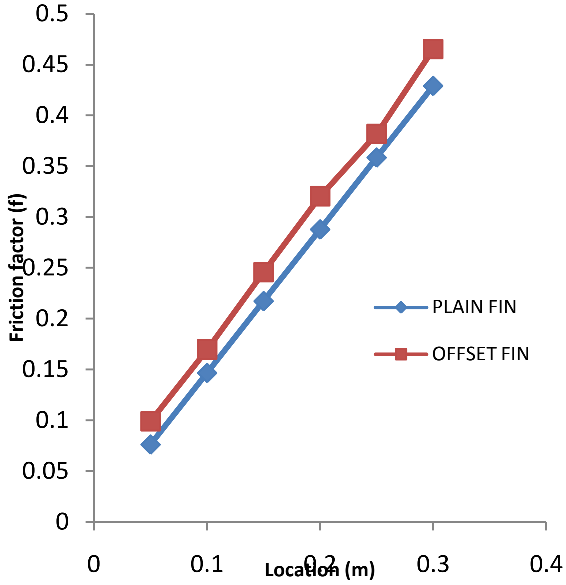

Fig. 23 Comparison of friction factor distribution for plain fin and offset fin

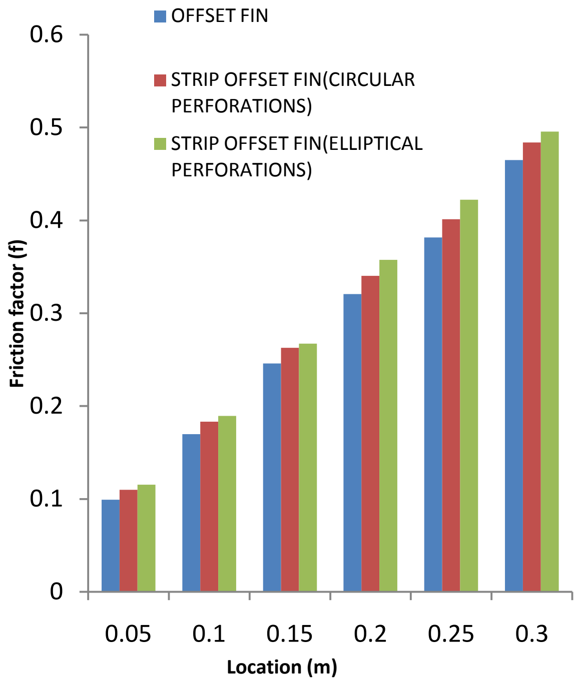

Fig. 24 Comparison of friction factor distribution for offset fin and offset fin with perforations

IV. CONCLUSIONS

Heat transfer and pressure drop in complete three-dimensional geometries of the plain fin, strip offset fin, circular and elliptical perforated fins, are carefully investigated, in which the fin thickness, spacing, length, thermal entry effect, are taken into account in this work. CFD simulations are carried out for the basic fins of PFHE at Reynolds number 285. The validity of the simulation models is verified by comparing the computed results of the plain fin with the corresponding experimental data Ref [2]. Good agreement has been obtained between the computed results and the experimental results.

Furthermore, simulation for calculation of the local Nusselt numbers, j factor, and f factor is presented, based on which, the heat transfer and pressure drop characteristics in the fins are obtained and analyzed in detail. Influences of the offset fins in the strip fin, holes in the perforated fin, on the pressure drop and heat transfer are investigated and the results are compared for different PFHE.

Hence It is observed that, strip offset fin with elliptical perforations have more heat transfer and friction factor comparatively.

V. NOMENCLATURE

D_h = hydraulic diameter

f = fanning friction factor

j = colburn factor

L = fin array length

Nu = Nusselt number (=h D_h / k)

Pr = Prandtl number

Re = Reynolds number

s = fin spacing

T = temperature (k)

t = fin thickness

v = fluid velocity

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