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Computational study of Staggered and Double Cross Flow Heat Exchanger

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ABSTRACT

The preliminary findings of a comparative study of heat transfer rate and pressure drop between conventional staggered flow and double cross flow heat exchanger is reported. Excepting for the tube arrangements, the shell and tube dimensions, materials and inlet conditions are retained the same for the two configurations. While in the conventional arrangement, adjacent rows of tubes are normal only to the fluid flow in the shell, in the double cross-flow arrangement, they are normal to both fluid flow direction in the shell as well as to each other. Shell dimensions are 100 cm × 20 cm × 20 cm and tube outside and inside diameters are 1 cm and 0.8 cm. The shell and tube materials are steel and copper. Water and air were considered as tube and shell side fluids respectively, with an overall arrangement of parallel flow. The tube flow Reynolds number was fixed at 2200 and the shell flow Reynolds number was varied from 20 to 120 in the laminar regime and 360 to 600 in the turbulent zone. The study reveals that the proposed configuration gives a maximum increase of about 27 per cent in the heat transfer rate per unit pressure drop over the conventional one.

Keywords: Staggered; Cross-flow; Heat exchanger; Laminar flow; Turbulent flow

1. INTRODUCTION

Heat exchangers are amongst the ubiquitously used equipment, with their utility ranging from condenser in domestic refrigerators, automobile radiators to robust industrial heat exchangers like pre-heaters, condensers and miniaturised ones typically those used for cryocoolers. It would not be an exaggeration to quote heat exchangers as indispensable for human survival as human body itself is an extremely complex heat exchanger. On an otherwise simplistic term, heat exchanger facilitates transfer of heat between two process streams. Heat exchangers are called by different names like condensers, boilers, pre-heaters, cooling towers, regenerators, etc., depending on the purpose they are used for. Performance and efficiency of heat exchangers are measured through the amount of heat transfer using least area of heat transfer and pressure drop[1]. Another way to judge the heat exchanger performance is to estimate the overall heat transfer coefficient. The capital and running cost for a given amount of heat transfer depends on the area required and pressure drop respectively. One however, needs to remember that the running cost would be dynamic. These equipment come in various forms such as shell and tube heat exchangers (STHX), plate-fin heat exchangers, compact heat exchangers, fin and tube heat exchangers, etc., with each form to best cater a particular requirement. There are lot of literatures and theories on designing heat exchangers. Great wealth of information on design of heat exchangers is available in literature, for instance[2]. The STHX is relatively simple to manufacture and is capable of heat transfer between fluid streams of same or different phase for a large temperature differences over a wide range of pressure. This makes them popular enough to be still widely used in various industrial applications[3-4].

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2. LITERATURE SURVEY

Shell and tube heat exchangers have been studied in breadth and depth to such an extent that it is a topic of discussion in books on heat transfer[1,3,4]. In many of the studies, performance enhancement has been a matter of prime consideration. Petinrin and Dare[5] conducted numerical studies on thermal and hydraulic performance of shell and tube heat exchangers. Li[6], *et al.* have reported numerical studies on thermal performance of two different heat exchangers for thermo-electric generators. Venkateshan and Eswaramurthi[7] presented a review of performance of heat exchangers. Kanojia[8], *et al*. have written on review of performance enhancement of HX using inserts. Some more review of literature can be found in Acharya[9].

An improvement in performance can be obtained by increasing in the heat transfer coefficient of shell side fluid and / or the tube side fluid. Both active and passive methods have been reported in Dewan[10], *et al.* Standard passive methods include inclusion of baffles and inserts for increasing the shell side and the tube side heat transfer coefficients. Both these result in an increased pressure drop. A conventional method of increasing the heat transfer coefficient of the shell side is by having cross-flow arrangements of tubes, with adjacent rows placed in a staggered manner. This helps in better heat transfer rates due to mixing of the shell side fluid. Considering x-axis oriented along the flow direction in the shell, in a conventional staggered arrangement the mixing due to directional changes happens only in one of the other two mutually orthogonal directions, i.e. along a direction perpendicular to the tubes.

Kumar and Krishnan[11] presented a novel technique of attempting to mix the shell fluid in two directions. This was achieved by placing the adjacent sets of tubes normal to each other and the shell fluid flow direction as well. This arrangement was named as double cross flow. By this arrangement, the shell fluid undergoes a directional change upon facing one set of tube, and since the alternate set of tube are placed normal to each other, it undergoes directional change perpendicular to the earlier one. The author(s) numerically compared the heat transfer and pressure drop characteristics of double cross flow (DCF) and staggered flow (SF) shell and tube heat exchangers (HX) for laminar flow. The author(s) have reported the DCF arrangement to exhibit superior heat transfer and pressure drop characteristics. The present article is an extension of their studies to larger regime of flow including turbulent flow, on a scaled up model of the heat exchanger. This study was carried for different scale according to the experimental step-up for future work. Shell side fluid flow in this project was laminar, transition and turbulent. Air was considered as shell fluid and water as the tube fluid, with air and water as hot and cold fluids respectively at the inlets.

3. METHODOLOGY

3.1 Geometry

The 3D computational models of SFHX and DCFHX are shown in Figs. 1 and 2, respectively. The list of legends used in Figs. 2 and 3 are as given in Fig. 3.

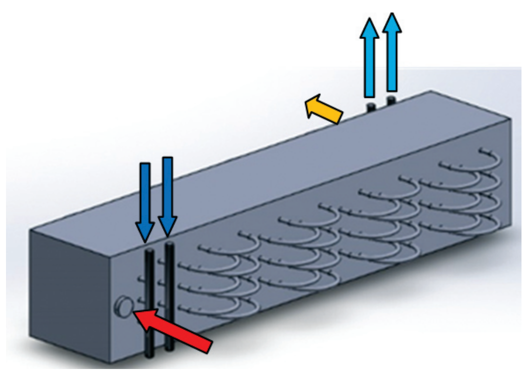

Figure 1. Staggered flow heat exchanger.

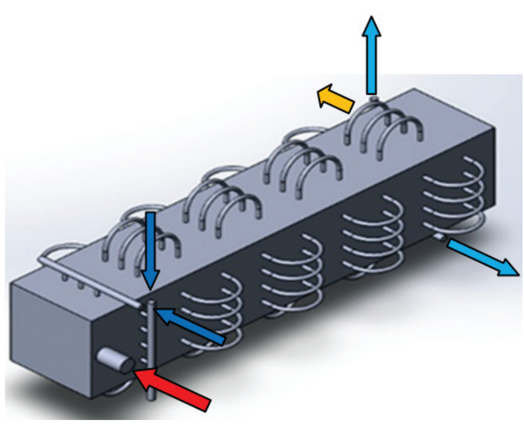

Figure 2. Double-cross flow heat exchanger.

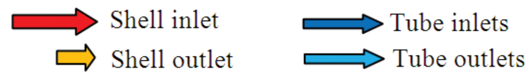

Figure 3. Legends for inlets and outlets.

The geometrical parameters for both the heat exchangers are as shown below in Table 1. To simplify numerical simulation, the following assumptions were made:

(a) The shell and tube side fluid are assumed to have constant thermal properties.
(b) The fluid flow and heat transfer processes are in steady state.
(c) The heat exchanger is well insulated, so heat loss to the surrounding is neglected.
(d) No slip condition at the walls of shell and tubes.
(e) Both shell and tube side fluid are assumed to be incompressible

Table 1. Geometric specifications of the heat exchangers

Dimensions (mm)		Material
Length of shell	1000	Mild Steel
Width of shell	200	
Height of shell	200	
Outer diameter of tube	10	Copper
Inner diameter of tube	8	
Longitudinal pitch, S_L	50	NA
Transverse pitch, S_T	40	

3.2 Governing Equation

Continuity Eqn -

$$\nabla \cdot \vec{u} = 0 \tag{1}$$

Momentum Eqn

$$\rho_f \frac{D\vec{u}}{Dt} = \nabla P + \mu \nabla^2 \vec{u} \tag{2}$$

Energy Eqn:

Convection: Solid – Fluid Interface

$$\rho_f c_{pf} \frac{DT}{Dt} = k_s \nabla^2 T \tag{3}$$

Conduction: Solid region

$$\rho_s c_{ps} \frac{\partial T}{\partial t} = k_s \nabla^2 T \tag{4}$$

3.3 Boundary and Initial Conditions

In order to obtain a well-posed system of equations, reasonable boundary and initial conditions for the computational domain have to be implemented. From Reynolds number find velocity and using that velocity find mass flow rate. The boundary conditions are shown in Table 2.

Couple energy boundary condition was chosen for the tube wall since the property fields in fluids both outside and inside the tube had to be solved for simultaneously.

3.4 Mesh Generation

The three-dimensional model was generated in Solidworks. The domain was extracted and discretised in ANSA using triangular mesh elements which are accurate and involve less computation effort. Finer mesh was adopted near

Table 2. Boundary conditions for computational analysis

Boundary type	Boundary condition
Shell inlet	Mass flow inlet, Air (Temperature -413 K)
Shell outlet	Pressure outlet (Gauge pressure – 0 Pa)
Tube-3 inlet	Mass flow inlet, Water (Temperature -300 K)
Tube-3 outlet	Pressure outlet (Gauge pressure – 0 Pa)
Tube-4 inlet	Mass flow inlet, Water (Temperature -300 K)
Tube-4 outlet	Pressure outlet (Gauge pressure – 0 Pa)
Shell wall	Wall (Adiabatic)
Tube wall (exposed to atmosphere)	Wall (Adiabatic)
Tube wall (exposed to air and water)	Wall (Coupled energy)

the wall surface to capture the boundary layer phenomena. The surface mesh was imported to Ansys Tgrid, where meshing of the entire model is carried out by tetrahedral element for higher accuracy. The entire geometry is divided into three fluid domains viz., Fluid shell, Tube3_fluid, and Tube4_fluid. The heat exchanger is discretised into solid and fluid domains in order to have better control over the number of nodes. The fluid mesh is made finer for simulating conjugate heat transfer phenomenon. The first cell height in the fluid domain from the tube surface is maintained at 100 microns to capture the velocity and thermal boundary layers. Once the meshes are checked for free of errors and minimum required quality it is exported to ANSYS Fluent pre-processor.

3.5 Grid Independence Test

Solution obtained by any numerical technique has to be invariably tested for its independence to mesh size, and hence was followed for in this study too. Mixed type of cells viz., tetra and tetrahedral having triangular faces at boundaries. Structured cells were preferentially used by dividing the geometry into several parts using the option of automatic methods available in ANSYS meshing client. The structuring of the mesh is generally done to reduce the numerical diffusion, especially near wall regions. The results of grid independence test for SFHX is as shown in Table 3. Initially, a relatively coarser mesh of 1.87 million cells was chosen, which was increased up to about 4.96 million cells. Based on relative change in heat transfer rate 2.23 million cells were considered for further analysis. A similar exercise was carried out for DCFHX, for which 2.41 million cells were chosen after grid independence test. The details are as shown in Table 4.

The geometry of SFHX and DCFHX after meshing is as shown in Figs. 4 and 5, respectively.

Table 3. Results of grid independence test for SFHX

No. of Cells (in lakhs)	Heat transfer rate (W)	Relative change (per cent)
18.7	210.31	-
22.3	231.56	10.1
35.3	232.81	0.54
49.6	233.14	0.14

Table 4. Results of grid independence test for DCFHX

No. of Cells (in lakhs)	Heat transfer rate (W)	Relative change (per cent)
18.4	220.42	-
24.1	258.94	17.47
34.2	259.71	0.29
50.3	260.04	0.13

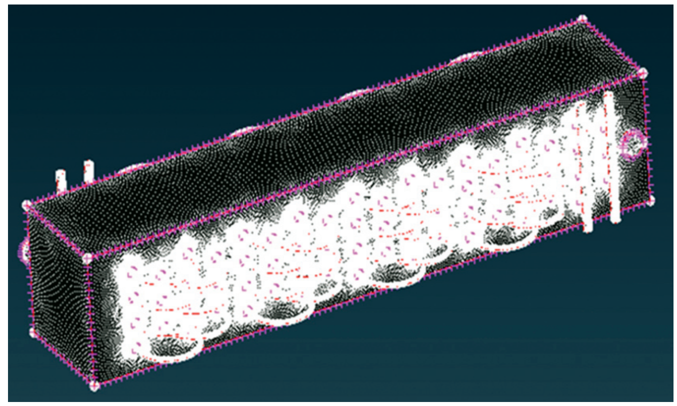

Figure 4. Meshed model of SFHX.

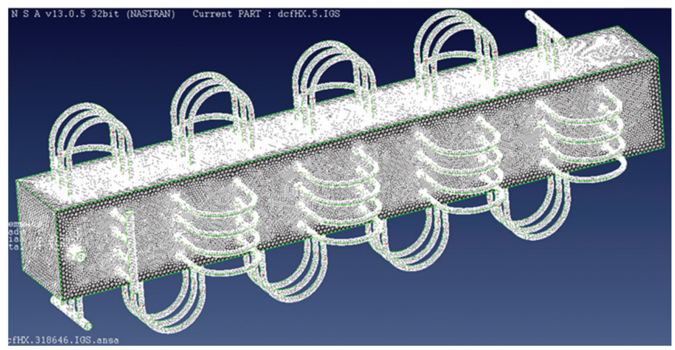

Figure 5. Meshed model of DCFHX.

4. RESULTS AND DISCUSSION

4.1 Variation of Pressure Drop with Reynolds Number

A comparison of variation of pressure drop with Reynolds number for the two heat exchangers is as shown in Fig. 6. It could be seen that variation of pressure drop shows a similar trend for both the heat exchangers, with the DCFHX having marginally lesser drop than SFHX. While this is a positive sign, a comparison of heat transfer bears more significance for heat exchangers. However it was decided to compare ratio of heat transfer to pressure drop, which is a better indicator of energy transferred to energy spent to do so. The variation of the ratio of heat transfer to pressure drop with respect to Reynolds number, for the two heat exchangers are as shown in Figs. 7 and 8.

4.2 Effectiveness

Figure 9 shows the variation of effectiveness with Reynolds number for the two heat exchangers. The effectiveness of DCFHX can be seen to be consistently higher than SFHX.

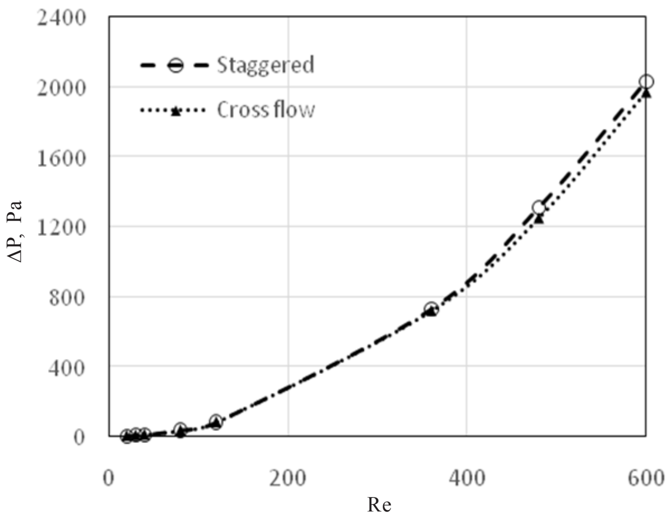

Figure 6. Variation of pressure drop.

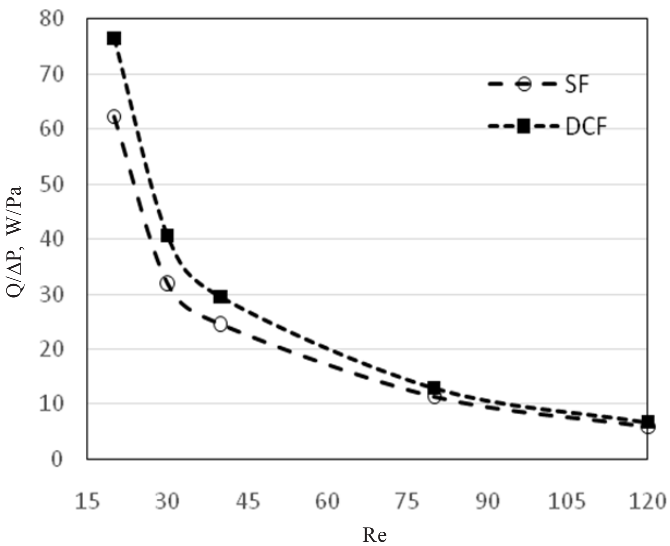

Figure 7. Variation of heat transfer rate per unit pressure drop for laminar flow.

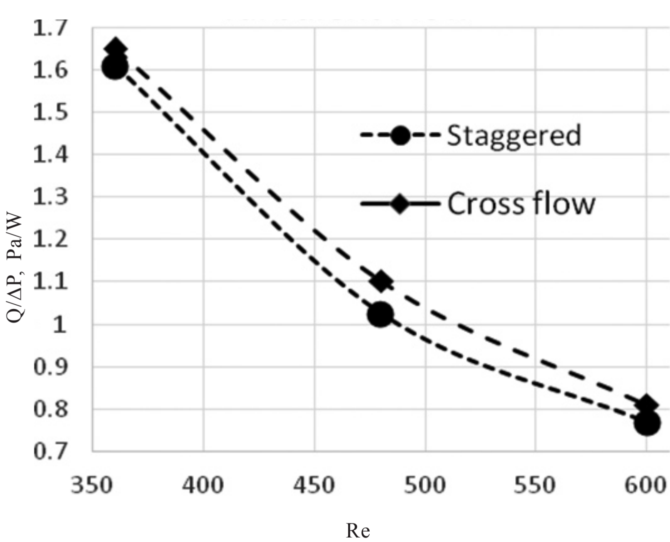

Figure 8. Variation of heat transfer rate per unit pressure drop for turbulent flow.

Figures 7-9 show that the DCFHX has an edge over the conventional SFHX for both laminar and turbulent flows. One of the reasons for better performance of the DCFHX could be attributed to better mixing. This is because of an additional change in direction that the shell fluid undergoes. The second possible reason is a lesser pressure drop. This could be because of a smoother directional changes, albeit an additional directional change. Both these together contribute for improvement shown by DCFHX which ranges from about 3 per cent to 27 per cent, with the lower one corresponding to turbulent and higher extreme lying in the laminar regime. Additionally, a third possibility of effect of vortex was too considered, which is discussed in the following section.

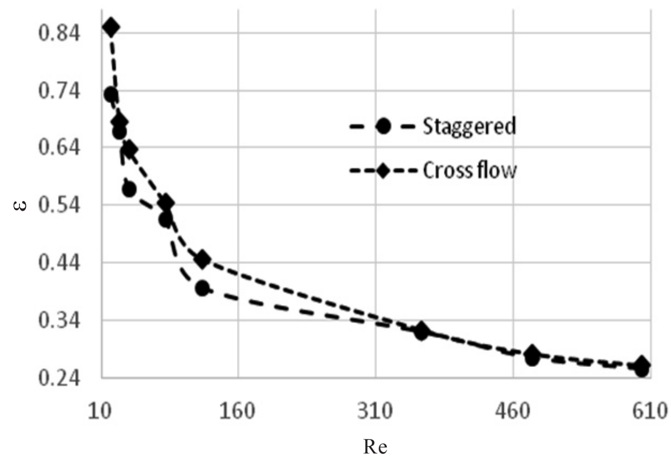

Figure 9. Variation of effectiveness with Reynolds number.

4.3 Velocity Contour of Staggered Arrangement for Unsteady 2D Analysis

A transient 2D analysis was conducted for the staggered arrangement. The flow pattern for two different Reynolds numbers are as shown in Figs. 10 and 11. Figure 10 indicates the formation of wakes behind the tubes for a Reynolds number of 40. Vortex formation was not noticed at this Reynolds number. However, for a Reynolds number of 225, the formation and shedding of vortex could be seen from Figure 11. It could be recalled that double cross flow arrangement fared better over

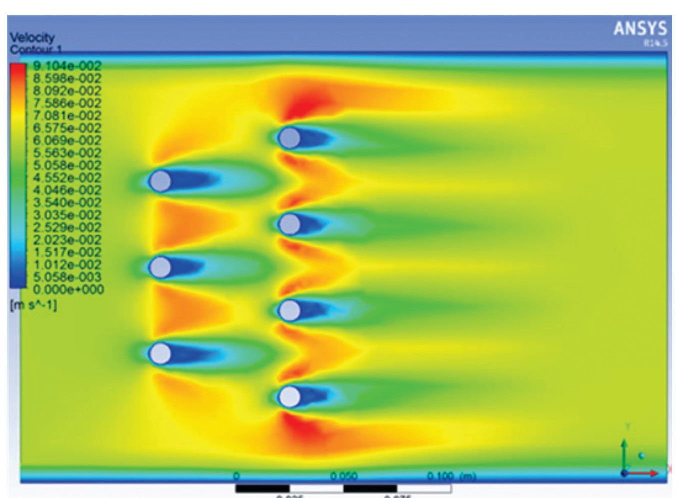

Figure 10. Velocity contours for Re = 40.

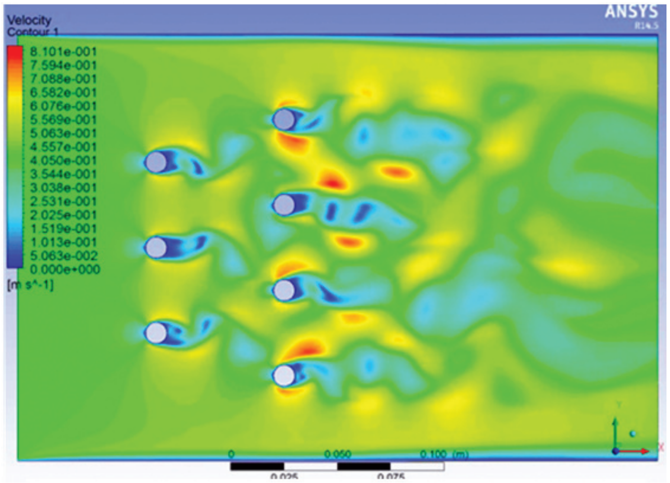

Figure 11. Velocity contours for Re = 225.

the staggered arrangement in both laminar and turbulent flow regimes for the same contact area between the shell fluid and the tube surfaces. The reason could possibly be explained as follows:

(a) At low Reynolds numbers, wherein no noticeable vortices are formed, the double cross flow arrangement helps in better heat transfer rate by reducing the amount of shell side fluid which does not participate in heat transfer

(b) Lesser directional changes leading to reduced pressure drop for the range of Reynolds number considered for study and

(c) At higher Reynolds numbers supporting formation of vortices, breaking up of vortices could be better achieved by the double cross arrangement than the staggered flow arrangement.

5. CONCLUSIONS

Computational studies on staggered and double cross flow heat exchanger for both laminar and turbulent flow have been presented. The heat transfer rate per unit pressure drop is higher in double cross flow heat exchanger by a minimum of 2.5 per cent to 27.1 per cent than the staggered type heat exchanger. The increase is more prominent in the laminar than the turbulent regime of vortex shedding. The results indicate DCF to perform better than staggered heat exchanger in terms of heat transfer rate per pressure drop.

Future work could include experimental validation and a more detailed study on vortex formation and shedding, and its effects on performance of the heat exchanger.

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CONTRIBUTORS

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In the current study, his contribution is conceptualisation of the idea, planning the research, guidance, preparation and submission of original and revised manuscripts.

Mr P. Gowtham obtained his ME in Heat Power Engineering from Coimbatore Institute of Technology, Coimbatore, in 2016. He works as an Assistant Professor at Government College of Engineering, Srirangam. His area of research: Heat transfer.

In the current study, he has conducted the CFD study discussed in this paper, prepared the draft and assisted in revising the manuscript.