Numerical studies on the effect of baffle on the heat transfer and flow in cross-corrugated triangular ducts

C H Liang¹*, C N Feng¹, T Y Lei¹ and Z X Li²
¹School of Mechanical and Electrical Engineering, Guilin University of Electronic Technology, Guilin, China
²Department of Environmental Engineering, Shanxi University, Shanxi, China
E-mail: lianghang@guet.edu.cn

Abstract. The cross-corrugated triangular ducts are suitable for plate heat exchanger because of its high heat transfer capability and strong mechanical strength. In this numerical study, a SST k-ω model is used, 6 different cases for installing different baffles in the cross-corrugated triangular ducts are involved. The frictional resistance and heat transfer coefficient in the 1000-6000 range of Reynolds number are studied, the influence of baffle on flow-pattern and temperature distribution are analyzed. Combined with the comprehensive heat transfer index, a good improvement plan was proposed. The results show that baffle can effectively increase the heat transfer area, change the mainstream direction, enhance the fluid disturbance, and enhance the heat transfer. In addition, case 6 is the best enhanced heat transfer scheme. Compared with the case without the baffle, the Nu of case 6 is 1.5-1.6 times that of the original duct, and the heat transfer efficiency can be increased by 13% under the same fan power. It is a feasible choice to set up baffles in the middle and lower part of the upstream channel of the cross-corrugated triangular plate flow channel.

1. Introduction
Heat exchanger is a type of device that transfers part of heat from hot fluid to cold fluid. It is widely used in HVAC, chemical engineering, electric power engineering, petroleum refining and other engineering fields. Structurally, the heat exchangers can be divided into movable tube sheet heat exchanger, fixed tube-plate heat exchanger, plate heat exchanger and so on. Specifically, the plate heat exchanger has been widely applied and developed due to its compact, economical, simple and outstanding performance.

Parallel plate duct structure is a common and simplest plate heat exchanger structure. However, the heat transfer efficiency is low. Many researches has been done trying to solve this problem, Scott K [1] and Zhang[2] developed a cross corrugated sinusoidal/triangular duct structure for plate heat
exchangers to enhance heat and mass transfer. The schematic is shown in Figure 1. Many corrugated sheets are stacked together to form a flow channel in the plate heat exchanger, and adjacent plates have a certain angle to forming flow channel and separating the fluid. The structure makes the cross-corrugated triangular plate have higher mechanical strength. And due to the contact expansion characteristic in the cross-corrugated triangular duct, the internal flow has been turbulent at the low Reynolds number[3], thus greatly increased the convection heat transfer coefficient. The air flow through the channel periodically, producing a large number of vortices to achieve high efficiency[2], cross-corrugated triangular heat exchanger is an ideal heat exchanger. Some experiments and theoretical studies have been carried out to optimize the performance of the heat transfer of cross-corrugated duct[4]. Liu and Niu[5] investigated the influence of duct sizes and apex angles on the performances of cross-corrugated total heat exchangers. They found that when the apex angle was 90°, the j/f (Colburn factor to friction factor) factor is enhanced by 4.1-7.0 times.

![Figure 1. Schematic of cross-corrugated triangular duct.](image)

Many researchers have installed baffles or barriers in shell and plate heat exchangers to improve heat transfer performance[6-8]. The shape of baffles is usually designed to be similar to that of the flow channels. In order to enhance the heat transfer characteristics of the backward-facing step flow in a channel. Promvonge[9] studied the effect of multiple 60° V-baffle fitted on a channel in Reynolds number of 5000–25000 range, the results show that the V-baffle provides the drastic increase in Nusselt number, friction factor, and the thermal enhancement factor values over the smooth wall channel. They found that maximum thermal enhancement factor of about 1.87 at lower Reynolds number, e/H (blockage ratio) = 0.10, and PR (pitch ratio) = 1. Li and Gao[10] recently proposed a structure that half baffles and full baffles are set in the traditional cross-corrugated triangular duct to improve heat transfer, and predicted the results of heat transfer enhancement by simulation. The results show that both two kinds of baffles can effectively improve the Nusselt number. Compared with the structure of the full baffle, the friction factor of half baffle channel is within the acceptable range, but the comprehensive heat transfer enhancement effect is not deeply analyzed in their research. In general, there is still the possibility of improvement on the optimization of thermal design of the cross-corrugated ducts by inserting baffle in the channels.
The objective of this paper is to improve the heat transfer performance of cross-corrugated triangular ducts by inserting different baffles at the channel. This study numerically studied 6 different schemes for installing baffle schemes on the cross-corrugated triangular duct, there are 7 cases including non-baffle channels. The temperature distribution and flow field of cross-corrugated triangular duct with 10 cycles were simulated, the apex angle of the model was 60°, the Reynolds number range from 1000 to 6000 which is practical in nature cross-corrugated plate heat exchangers. Figure 2 shows the schematic of a cross-corrugated triangular duct with baffles (case2). In order to further optimize the heat transfer of cross-corrugated triangular duct, we compare the heat transfer, air pressure drop and comprehensive heat transfer coefficient of 6 different cases. In addition, the influence of different baffle on the main stream is analyzed.

![Figure 2. Schematic of a cross-corrugated triangular duct with baffles(case2).](image)

2. Mathematical models

2.1. Governing equations

Usually, the transition flow of cross-corrugated triangular duct is modeled by turbulence model. A SST k-ω model is employed in this investigation as the model has been previously validated by Zhang[11].

The numerical model for this study is based on some assumptions like steady three-dimensional turbulent, incompressible flow, no radiation heat transfer and constant fluid properties. Based on the assumptions above, the duct flow is governed by the continuity, the Navier-Stokes equations and the energy equation.

\[
\frac{\partial (\rho \bar{u}_i)}{\partial x_i} = 0
\]

\[
\frac{\partial}{\partial x_j} (\rho \bar{u}_i \bar{u}_j) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i} \left( \mu \frac{\partial \bar{u}_i}{\partial x_j} - \rho \bar{u}_i \bar{u}_j \right)
\]
\[
\frac{\partial}{\partial x_j} \left( \rho c_p \mu T \right) = \frac{\partial}{\partial x_j} \left( \frac{\mu c_p}{Pr} \frac{\partial T}{\partial x_j} - \rho c_p \mu T \right)
\]  
(3)

where \( \rho \) is the density of the fluid, \( c_p \) is the specific heat of the fluid, \( P \) is the pressure and \( \mu \) is the dynamic viscosity.

The Nusselt number is defined as

\[
Nu = \frac{hD_h}{\lambda}
\]
(5)

The channel-average friction factor is calculated by

\[
f = \frac{(P_i - P_o)D_h}{\frac{1}{2} \rho u_m^2 L}
\]
(6)

where \( L \) is the length of the channel, (m); \( P_i \) and \( P_o \) are pressure at inlet and outlet, respectively, (Pa).

The PEC (performance evaluation criteria) number is defined as

\[
PEC = \frac{Nu / Nu_0}{(f / f_0)^{\frac{1}{5}}}
\]
(7)

where \( Nu_0 \) and \( f_0 \) are the Nusselt number and friction factor of the cross-corrugated triangular ducts without the baffle (case 1), respectively.

2.2. geometry

The geometry of the cross-corrugated triangular duct is shown in Figure 2. The apex angle of the cross-corrugated triangular ducts is 60° and hydraulic diameter \( H \) was 12.99 mm. Totally there are 10 cycles in z direction, the baffle is located in the middle of the Z direction of each cycle. As shown in Figure 3, the triangular upper channel is divided into 4 equal parts at height, namely a1-a4. By setting one or more baffles in these four locations, six different baffle cases were designed. Case 1 means non-baffle flow channel that used for comparison. Table 1 shows the baffle position.

2.3. Solution method

Seven computational mesh has been constructed to resolve the geometry in this study. The details of the mesh structure of case 2 is shown in Figure 4. Each grid-independence test has also been done. For simplification, a flow channel with half-baffle (case 2) is selected to illustrate grid independence check procedure: Three different grid densities of 945410, 1216902 and 1724721 nodes are used in the calculation. Nusselt number and friction factor difference are checked for these grid systems. The difference in the results between 1216902 nodes and 1724721 nodes is less than 1%. To save computation time, grid system with 1216902 nodes is chosen. The solution criterion for continuum and momentum equations is \( 10^{-4} \), and for energy equation is \( 10^{-7} \).
Table 1. Baffle position of seven cases.

| Case  | Baffle position                      |
|-------|-------------------------------------|
| Case1 | no baffle                           |
| Case2 | a1+a2                               |
| Case3 | a1+a2+a3+a4                         |
| Case4 | a2                                   |
| Case5 | a3                                   |
| Case6 | a4                                   |
| Case7 | a2+a4                                |

Figure 3. Schematic of different cases.  

Figure 4. Part of mesh structure (case2).

The continuum and energy equations are discretized onto the meshes mentioned above. Boundary conditions to be defined on all boundaries of the computational domain. Flow channel inlet is defined as mass-flow-inlet boundary, a constant mass flow rate is imposed. At the top and bottom walls of the flow channel, the no-slip wall and uniform temperature condition is applied. The flow channel exit conditions are defined as outflow.

The governing equations are solved by using finite-volume methods along with a pressure-correction algorithm using Fluent 16.0. The N-S equations are solved by SIMPLE scheme and the diffuse term is solved by the second-order central difference scheme.

3. Discussions

3.1. Model validation

The comparisons of calculated Nu and f with data from literature[2] is shown in Figure 5. The maximum deviations for fully developed Nusselt number and friction factor is 9.13% and 5.31%, respectively. Generally, it can be clearly seen the good agreement between the present results and relevant literature.
Figure 5. Comparisons of calculated Nu and f with data from literature[2]: (a) Fully developed friction factor; (b) Fully developed Nusselt number.

3.2. Flow distribution and friction factors

The Figure 6 shows the velocity vectors in y-z plane at x=0.5. In non-baffle channel (case1), the parallel flow is the predominant flow pattern in the corrugation troughs on the upper wall. In the corrugation troughs of the lower wall, a small portion of the fluid forms a clockwise vortex. For case2/4/5, in the troughs of the lower wall, the flow pattern and vortices form are nearly the same as that of case1. In the corrugation troughs of the upper wall, case2 creating a counterclockwise vortex behind the baffle. The case4 and 5 form two relatively low speed vortexes behind the baffle. For case3/6/7, the impact and separate of the fluid on the troughs of the lower wall. In case7, the two baffles divide the fluid into three strands, and the parallel flow almost vanishes, the fluid exchange between the near-wall area and main stream region is more abundant, which enhance the heat transfer effect.

Figure 7 shows the friction factor of channel. The results show that the friction factor in case3, different with other cases, increases with the Re number. This may be due to the dominant pressure drag in the frictional resistance. The friction factor of the case3 is the highest, which is 60 to 150 times that of the case1. It is found that friction coefficient increases when a baffle is set in a cross-corrugated triangular duct, when the baffle area of the channel is large and meanwhile located in the middle of the upstream wall, the friction factor will be larger.
3.3. Temperature distribution and Nusselt numbers

On the Figure 6 shows the temperature distribution in y-z plane at x=0.5. In case 1, the isothermal line near the upper wall and the lower wall is very steep. Observation case 3/6/7, a higher temperature gradient is found on the lower wall than the non-baffle channel (case 1). In the case 3/5/6, it is found that a higher temperature gradient on the upper wall than the case 1.

Figure 8 shows the Nusselt numbers of channel with 7 cases. The Nu in all cases increases with the Re. Case 3 achieves the most efficient heat transfer effect and the Nu is 2.7 to 3.7 times of the
non-baffle channel (case 1). In case 7, the Nu is 1.7 to 2 times of case 1. The Nu of Case 5/6 is slightly lower than case 7. It can be deduced that strengthening turbulence in the lower channel is more conducive to enhancing heat and momentum transfer.

3.4. Enhancement heat transfer comprehensive index

Figure 9 shows the PEC of channel with case 2-7. From the figure, we can get the PEC number greater than 1 when the whole case of case 6 and the low Reynolds number of case 7. For case 3, although it achieves the most efficient heat transfer effect, at high Reynolds numbers, PEC is only 0.5. Although the PEC of case 4/5 is slightly less than 1, it is also acceptable for certain heat exchange requirements. The results show that, the heat transfer efficiency of case 6 can be increased by 13% under the same fan power and low Reynolds number.

4. Conclusion and implications

The main work of this study is to numerically study the heat transfer and flow of cross-corrugated triangular ducts with six different baffles and analyze. The results can be concluded as follows:

(1) The setting of the baffle in the cross-corrugated triangular duct enhances the disturbance of the fluid, changes the main direction and flow pattern, thus enhancing the heat transfer, but the friction coefficient is inevitably increased.

(2) In this study, the comprehensive enhanced heat transfer effect of case 6 is better than others. The Nu of case 6 is 1.5-1.6 times that of the non-baffle channel, and the heat transfer efficiency can be increased by 13% under the same fan power and low Reynolds number.

(3) For cross-corrugated triangular ducts, combined with velocity vectors and Nusselt numbers, it is found that the enhancing turbulence in the troughs of the lower wall is more conducive to enhancing heat transfer, and setting up baffles at the bottom of upper wall is a good choice.

Acknowledgments: This project was jointly supported by The National Natural Science Foundation of China, No. 51566002, and by Innovation Project of Guang Xi Graduate Education, No. 2018YJCX08.

References

[1] Scott K and Lobato J 2003 *Ind. Eng. Chem. Fundam*. 42(22) 5697-701
[2] Zhang L Z 2005 *Numer. Heat Transfer, Part A* 48(4) 387-405
[3] Zhang L Z 2016 *ENERGY* 101 390-401.
[4] Liang C H and Tong X M 2017 *Appl. Sci.* 7(6) 554
[5] Liu X P and Niu J L 2015 *Int. J. Heat Mass Transfer* 84 542-9
[6] Kwankaomeng S and Promvonge P 2010 *Int. Commun. Heat Mass Transfer* 37 857-66
[7] Ahmed H E 2016 *Appl. Therm. Eng.* 102 1422-32
[8] Tsay Y L, Chang T S and Cheng J C 2005 *Acta Mech* 174 63-76
[9] Promvonge P 2010 *Int. Commun. Heat Mass Transfer* 37 835-40
[10] Li Z X and Gao Y Y 2017 *Int. J. Heat Mass Transfer* 108 658-70
[11] Zhang L and Che D F 2011 *Numer. Heat Transfer, Part A* 60 410–440.