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Heat Transfer Analysis and Optimal Design of a Double-pipe Heat Exchanger with Swirl Fins

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Keywords：double-pipe heat exchanger, neural network, genetic algorithm, optimized design.

ABSTRACT

An investigation of thermohydraulic performance of a double-pipe heat exchanger with swirl fins were performed by using numerical simulation and single objective optimization method. In this study, the geometric parameters of swirl fins include the ratio of width of inner fin roots to circumference of inner flow passage, the number of fins, the height of fins and the swirl angle of fins were examined. Besides, fixed Reynolds number and constant inlet temperature were assumed for the simulation. The friction factor and Nusselt number were analyzed for getting the relations between fin geometry and thermohydraulic characteristics in the pipe. Moreover, the neural network method with the genetic algorithm was used to build performance model of heat exchanger for design optimization. The performance evaluation criteria (PEC) of the heat exchanger was introduced as the index for searching the optimal design parameters. In addition, the optimal design parameters were verified with numerical simulations and compared with other parameters. The results show the robustness of optimal solution. This study also demonstrates the accuracy of the single-objective optimization method and the practicality of numerical modeling with neural network training. The design methodology in this study can be used as a reference for the design of other types of heat exchanger.

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INTRODUCTION

Nowadays, heat exchangers are extensively used in various fields of the industry. The main purpose of heat exchangers is transferring heat between two or more fluids, i.e. energy conversion. The transport of thermal energy and efficiency of system are both important tasks for engineers. There are several types of heat exchangers according to their configurations, such as double-pipe heat exchanger (DPHE), shell-and-tube heat exchanger, plate heat exchanger, spiral heat exchangers. This study focused on DPHEs. An experiment study of the thermal performance of SCO2 in a DPHE was conducted by Ma et al. (2016). The effect of SCO2-side mass flux on total heat transfer coefficient was investigated. In recent years, the applications of DPHE on solar thermal energy storage of geothermal system is studied. Templeton et al. (2016) discovered the effect of solar thermal energy storage on a geothermal system by the proposed transient heat transfer model. In this study, the techno-economic performance of geothermal energy production was discussed through the examination of rate of heat extraction and injection.

The design and modification of the configuration of heat exchanger is an effective strategy for enhancing the thermal performance. Some of common methods for passive heat transfer enhancement are twisted pipes and corrugated pipes. The effect of twisted pipes or corrugated pipes in terms of aspect ratios and twist ratios on thermos-hydraulic performances of double pipe heat exchangers was investigated by (Lou et al. (2021), Bhadouriva et al. (2015), Luo et al. (2020), Dizaji et al. (2015), Wang et al. (2019)). The results shown that the mixing of hot and cold fluids in the pipe can be enhanced by the secondary flow generated by structural design. The use of twisted tape inserts in a tube is also an effective passive method to enhance the convective heat transfer in the tube (Naphon (2006), Thejaraju et al. (2020), Abed et al. (2018)). The working fluid flow swirly in pipe with twisted tape insert and generates the centrifugal force, which interrupts the development of boundary layer and makes the thermal energy transmitted across the layers of fluid, as a result, the efficiency of heat transfer was enhanced. However, both friction factor and Nusselt number of pipe with twisted tape insert were increased.

There are some researches focused on the design of pipe with baffles or fins in order to promote the performance of double pipe heat exchanger. Salem et al. (2017) experimentally investigated the hydrothermal performance of double pipe heat exchangers with/without single segmental perforated baffles (SSPBs) with different holes spacing, void, cut, pitch ratios and inclination angle. A great number of case studies (21 runs for the case without baffles and 231 runs with SSPBs) were performed on twelve heat exchangers. It noted that the change of baffle increases the throttling for the annulus flow and produces better impingement. The impingement becomes stronger, breaks the boundary layer and creates an obvious enhancement of heat transfer. However, a significant increase in the pressure drop is also accompanied with increasing of heat transfer. The correlations for the averaged Nu and averaged Fanning friction factor are addressed.

A numerical analysis was carried out to examine the performance of DPHE with rectangular, triangular and parabolic fins by Kumar et al. (2015). The object of research is to find out the best fin configuration under the same fin base width, height and number of fins. It showed that the pressure drop for a concave parabolic showed minimum value for all mass flow rates. Even though the thermal performance of concave parabolic fin is being slightly reduced. Syed et al. (2015) studied the convective heat transfer in an innovate design of DPHE with longitudinal fins of variable thickness of the tip. In the study, the configuration of DPHE was constructed with longitudinal fins been mounted at the outer surface of the inner pipe. The product of friction factor and the Reynolds number, the Nusselt number and the Colburn j-factor were introduced to assess the performance of DPHE. This study indicated that the ratio of tip to base angles is an important parameter for DPHE in cutting down the cost, weight and frictional loss, in improving the thermal performance. The choice of tip angle highly depends on the number of fins and their height. Ahmad et al. (2017) investigated the performance of exponential fins by numerical simulation. The work introduced exponential fin to evaluate the performance through conjugate heat transfer problem in the finned annulus of a double pipe. The effects of the number of fins, the ratio of the radii of inner and outer pipes, the fin thickness and the ratio of conductivity of the solid material used to manufacture the inner pipe and the fin on thermo-hydraulic performance of DPHEs were discussed. The results also showed that the Nusselt number decreases monotonically while increasing the number of fins for the shorter fins ($H^* = 0.2, 0.4$). While the Nusselt number increases at first and then reduces as the number of fins increased for $H^* = 0.8$. Their study also indicated that the performance of exponential fin is 0.02 - 15.09% higher than the triangular fin. The influence of different configurations and fin geometric parameters on fluid flow, heat transfer coefficient and pressure drop were studied (2017). The correlations for heat transfer and pressure drop were developed in their studies for the reference in designs of DPHEs. The results indicate that using helical baffles in the annulus side improve the heat transfer rate. However, it accompanies an increment of pressure drop. The high-pressure drop is caused due to the entrance region effect. In addition, the value of average thermal performance enhancement factor shows higher when helical baffles are used in laminar flow regime. Rao and Levy (2008) investigated a double pipe finned heat exchanger by using a semi empirical methodology. They compared three types (internally finned, externally finned, both internally and externally finned) of fin arrangements. The results show that the temperature variation of core flow of only inner fins or only outer fins cases is almost the same. The maximum heat is transferred in the case when there are both external and internal fins. M.A. Ali and S.N. Shehab (2023) examines heat convection in a double-pipe heat exchanger using a dimpled tube across various Reynolds numbers. The study explores how geometrical parameters (dimple arrangements, distribution angle, pitch ratio) affect water flow's hydrothermal characteristics. Results indicate a 50% higher Nusselt number for staggered dimpled tubes compared to inline ones, with optimum performance seen at a 60º distribution angle. Thermal performance factor (TPF) values range from 1.67 to 5.22 for inline arrangements and 4.91 to 8.633 for staggered arrangements. Key findings include increased Nusselt number with reduced pitch ratio and smaller dimple distribution angles. Heeraman et al. (2023) investigate heat transfer and friction factor in a double pipe heat exchanger using twisted tape (TT) inserts with dimples. The study analyzes the effects of dimple configuration on heat transfer and friction characteristics, considering various parameters such as dimple diameter (D), diameter-to-depth ratio (D/H), and Reynolds number (Re) ranging from 6000 to 14000. Results show that TT inserts with dimples significantly enhance heat transfer and friction factor compared to plain tubes and TT inserts without dimples, especially at higher Reynolds numbers. Dimple configuration influences flow dynamics and contact rates, with friction factor being more sensitive across the observed Reynolds number range.

MATHEMATICL MODEL AND NUMERICAL METHOD

In this study, the DPHE is based on two concentric pipes with different diameters. The structure of inner pipe is composed of multiple sections and multiple fins and angle changes as shown in Fig. 1. The cross-section of the fins is trapezoidal. Moreover, the entrance and exit section are connected with straight fins and setting as adiabatic sections. Finally, it composited the DPHE.

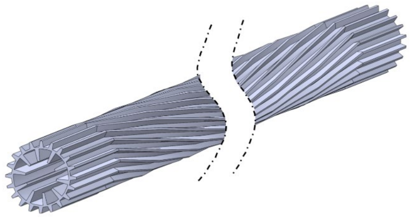
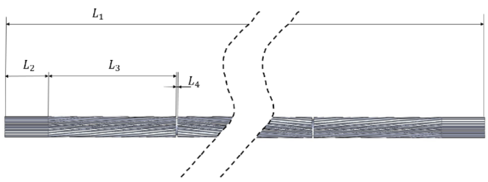


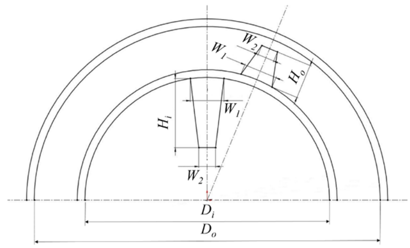
Fig. 1. Structure of inner fins.

The hot fluid passes through the inner pipe of the DPHE, which consists of 20 sections with fin variations. The total length of the double-pipe is L_1. It is composed of two adiabatic sections of length L_2 at the entrance and exit and 20 sections of length L_3 with swirl fins. The gap length between each section is L_4. The configuration of geometry is shown in Fig. 2(a).

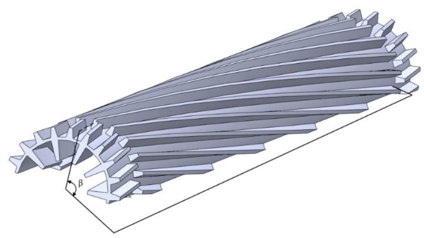
The inner pipe geometry is shown in Fig. 2(b), which is mainly composed of two circular pipes. The outer and inner pipe diameters are D_o and D_i, respectively. In addition, there are fins allocated inside and outside of the inner pipe to enhance the heat transfer efficiency. The height of the inner fins H_i is twice the height of the outer fins H_o. The shape of the fin is trapezoidal, and the width W_1 at the root is twice the width W_2 at the tip of the fin. The number of outer fins N_o is twice the number of inner fins N_i. The swirl angle of fin (β) is defined as shown in Fig. 2(c), where the swirl angle of the subsequent fins of each section is opposite in direction. The fins at the gap junction are staggered in rotational position.



(a) axial



(b) radial



(c) swirl angle

Fig. 2. Inscription of dimensions of pipe

Four parameters including the ratio of width of inner fin roots to circumference of inner flow passage (R_c), the number of fins (N_i), the height of fins (H_i) and the swirl angle of fins (β) are considered for their effect on exchanger performance.
The ratio of fins is defined as:

$$R_c = \frac{N_i W_1}{2\pi R_i} \tag{1}$$

The dimension of the DPHE is shown in Table 1. In this study, the water is used as working fluid.

Table 1. Dimensions of the DPHE

Parameters	**Size (mm)**
Total length (L_1)	3100
Adiabatic section (L_2)	50
The length of fins (L_3)	148.1
The gap between fins (L_4)	2
Diameter of outer pipe (D_o)	22.624
Diameter of inner (D_i)	16
Thickness of wall of pipe (t)	0.5

The material of pipe is stainless steel SUS316. The thermos-physical properties of the water and SUS316 are shown in Table 2. The insignificant variation (lower than 7%) of properties of water were shown under the range of temperature used in the study. In order to reduce the efforts of computation, the constant thermal properties were used in this study.

Table 2. Thermophysical properties of the water and SUS316

Properties	Water	SUS316
Density (kg/m^3)	998.2	7980
Specific heat (J/kgK)	4182	502
Thermal conductivity (W/mK)	0.6	16.3
Viscosity (kg/ms)	0.001	

In this study, the governing equations of mass, momentum, energy were employed. The k-ω turbulence model was used for complex flows under adverse pressure gradient and separation. The flow was assumed to be steady, incompressible. The buoyancy and radiation heat transfer effects were neglected. In addition, the thermophysical properties of the fluid were assumed to be constant. The uniform flow with constant temperature were adopted at the flow inlet boundary. The pressure outlet boundaries were set at outlets. The adiabatic thermal boundary condition was applied on the outer surface of outer pipe.

The present study is based on the SIMPLEC algorithm. The SIMPLEC algorithm reduces convergence difficulties associated with highly skewed meshes. Pressure and velocity correction schemes were implemented in the model algorithm to achieve a converged solution when both the pressure and velocity satisfy the momentum and continuity equations. The under-relaxation scheme was employed to avoid divergence in the iterative solutions.

In this study, the parameters introduced for parametric analysis and rating the thermohydraulic performance are listed as follows.

(1) Reynolds number

$$Re = \frac{\rho V D_h}{\mu} \quad (2)$$

where D_h is hydraulic diameter

(2) Overall heat transfer coefficient

$$U = \frac{Q_{avg}}{A\Delta LMTD} \quad (3)$$

where $Q_{avg} = (Q_h + Q_c)/2$ is the average heat transfer rate

(3) Nusselt number

$$Nu = \frac{hD_h}{k_f} \quad (4)$$

where k_f is thermal conductivity of fluid, and h is convective heat transfer coefficient. The calculation of Nusselt number is based on inner fluid.

(4) Fraction factor

$$f_c = \frac{\Delta p}{2\rho u^2}\left(\frac{D_h}{L_m}\right) \quad (5)$$

where $\Delta p = P_{in} - P_{out}$

The fraction factor is calculated based on the average of inner and outer fluid.

(5) Performance evaluation criterion

$$PEC = \frac{Nu/Nu_{ref}}{\left(F_c/F_{c,ref}\right)^{1/3}} \quad (6)$$

where Nu_{ref} is the reference value obtained for R_c = 0.25, N_i = 10, H_i = 4.5 mm, $\beta = 90^\circ$

Results and discussion

In this study, the DPHE was investigated numerically. The Reynolds number of both inlet boundaries at inner and outer pipes are 5000. The ratio of width of inner fin roots to circumference of inner flow passage (R_c), the number of fins (N_i), the height of fins (H_i) and the swirl angle of fins (β) were used for parametric analysis. Subsequently, the resulting performance corresponding to parameter data set were used to establish the heat exchanger performance model by a neural network. Then the genetic algorithm can be applied to provide optimal design for desired requirements.

Parametric analysis

Before parametric analysis, a validation study of pipe flow was performed to compare the result with Dittus-Boelter equation. The deviation between CFD solutions and Dittus-Boelter equation is about 6.2%. Moreover, the grid independent tests were also performed and showed that the current model provide satisfactory numerical accuracy.

The range of design parameters discussed in this study is listed in Table 3. Fig. 3 shows the streamlines in the double-pipe, where counter-flow type heat exchange was considered. The swirling flow is generated due to swirl fins on both sides of the inner pipe. The swirl directions difference in each section also causes the mixing effect and secondary flow.

Table 3. The range of design parameters for modelling and optimal design

Parameters	Range
the ratio of width of inner fin roots to circumference of inner flow passage (R_c)	0.025 ~ 0.4
number of fins (N_i)	8 ~ 12
height of fins (H_i)	2.5 ~ 5 mm
swirl angle of fins (β)	60° ~ 120°

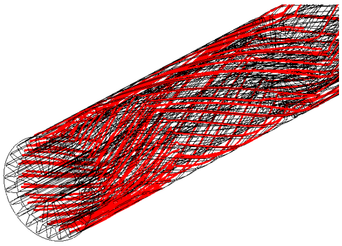
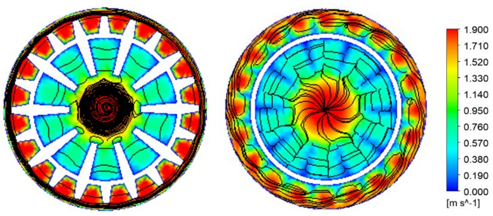


Fig. 3. Partial view of streamlines

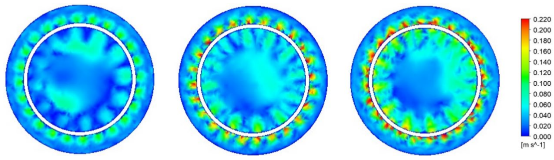
Fig. 4 depicts the distribution of velocity in cross section of the pipe. The cross section of outer flow channel is comparatively smaller to that of inner flow. Hence, the flow velocity of outer flow channel is higher for the same Reynolds number. The distribution of velocity also reveals the strong disturbance occur in the gap between two sections with fins. The disturbance disrupts the existing boundary layer and enhancing the heat transfer rate. It also accompanies the increment of pressure drop.



(a) (b)

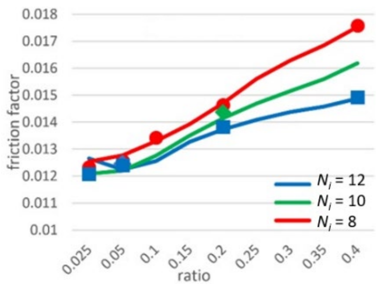
Fig. 4. Distribution of velocity and streamlines at a cross section of the pipe (a) section with swirl fin (b) the gap between two sections with fins

Fig. 5 shows the effects of the ratio of width of inner fin roots to circumference of inner flow passage (R_c) on secondary flow patterns in cross section of the pipe. The secondary flow becomes obvious with higher R_c. It causes the increment of pressure drop and friction factor. The effects of the number of fins (N_i) and the ratio of width of inner fin roots to circumference of inner flow passage (R_c) on friction factors are shown in Fig. 6. It depicts that the friction factor increases as R_c increased and the influence of N_i on friction factor is not significant when R_c is small. Fig. 7 depicts the effects of the number of fins (N_i) on secondary flow patterns. The results show that the intensity of secondary flow is higher when N_i is low. The secondary flow occurs significantly with a change of swirl angle. Fig. 8 shows the effects of the height of fins (H_i) on secondary flow. It depicts that the longer fins, the greater secondary flow strength. The effects of the swirl angle of fins (β) on flow field are illustrated in Fig. 9. It reveals that the larger the swirl angle, the greater the secondary flow intensity and the larger the friction factor. Fig. 10 shows the effects of β and the ratio of width of inner fin roots to circumference of inner flow passage (R_c) on friction factors. When the swirl angle of the fins increases, it makes the fluid flow more swirly. Consequently, the collision force at the junction of swirl change will be stronger and result in a significantly disturbance of the fluid in the pipe.

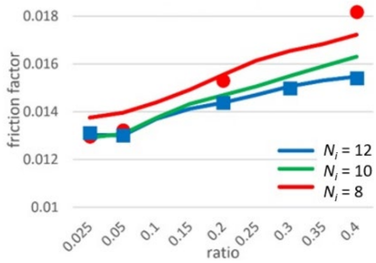


(a) $R_c = 0.05$ (b) $R_c = 0.2$ (c) $R_c = 0.4$

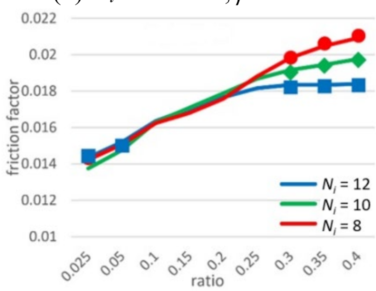
Fig. 5. Effects of the ratio of width of inner fin roots to circumference of inner flow passage (R_c) on secondary flow patterns at cross section of the pipe at z = 1550 mm



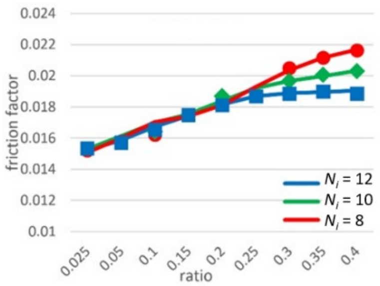
(a) $H_i = 2.5$ mm, $\beta = 60^{\circ}$



(b) $H_i = 2.5$ mm, $\beta = 120^{\circ}$

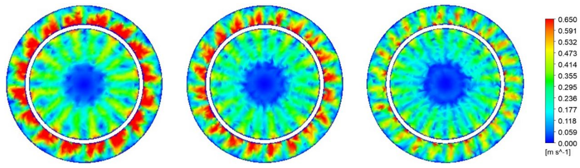


(c) $H_i = 5$ mm, $\beta = 60^{\circ}$



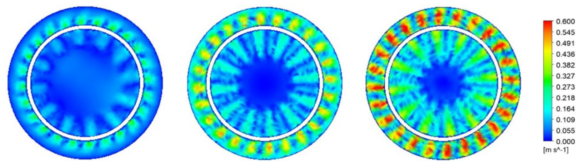
(d) H_i = 5 mm, β = 120°

Fig. 6. Effects of the number of fins (N_i) and the ratio of fins (R_c) on friction factors



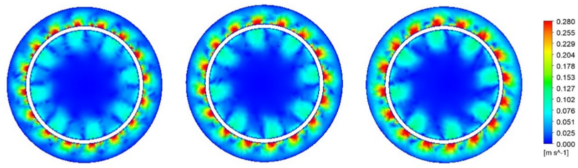
(a) N_i = 8 (b) N_i = 10 (c) N_i = 12

Fig. 7. Effects of the number of fins (N_i) on secondary flow patterns in cross section of the pipe



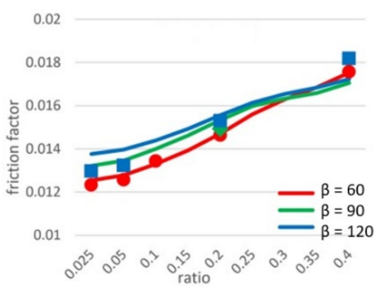
(a) H_i = 2.5 mm (b) H_i = 4 mm (c) H_i = 5 mm

Fig. 8. Effects of the height of fins (H_i) on secondary flow patterns in cross section of the pipe

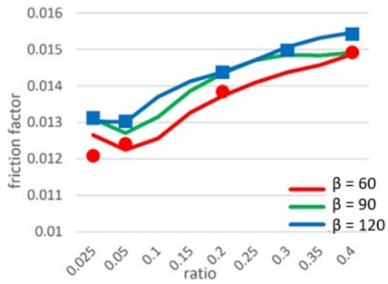


(a) β = 60° (b) β = 90° (c) β = 120°

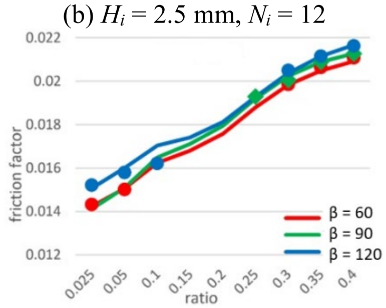
Fig. 9. Effects of the swirl angle of fins (β) on secondary flow patterns in cross section of the pipe



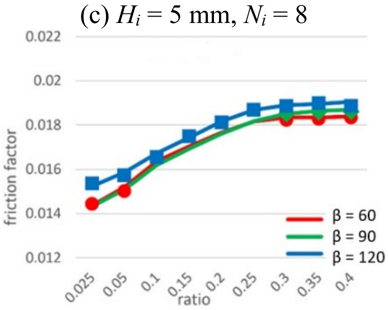
(a) H_i = 2.5 mm, N_i = 8



(b) H_i = 2.5 mm, N_i = 12



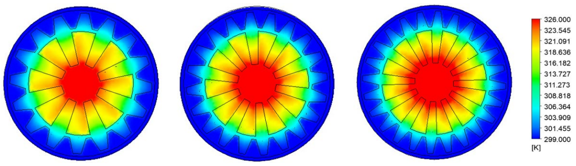
(c) H_i = 5 mm, N_i = 8



(d) H_i = 5 mm, N_i = 12

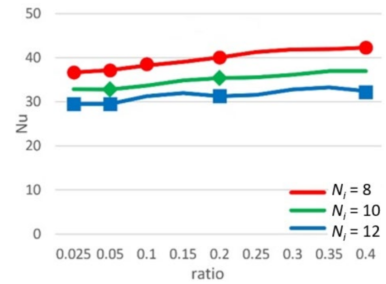
Fig. 10. Effects of the swirl angle of fins (β) and the ratio of fins (R_c) on friction factors

Fig. 11 illustrates the effects of the number of fins (N_i) on temperature distribution. It can be seen that when the number of fins is increased, the flow channel will be too narrow. Hence, the hot fluid in the center of the pipe is not easy to dissipate heat. Fig. 12 depicts the effects of the number of fins (N_i) and the ratio of width of inner fin roots to circumference of inner flow passage (R_c) on Nusselt number. It shows the trends of variation of R_c on Nusselt number under the same N_i. The magnitude of these three curves changes is small. When the R_c is increased by 16 times, the number of Nusselt number only increases by 6%, and this phenomenon is more obvious when the N_i is large. Therefore, when the R_c is increased, the increment of Nusselt number is relatively insignificant. When comparing the relationship between N_i and Nusselt number, it can be found that the N_i and Nusselt number are in inverse proportion. When the N_i increases by 1.5 times, the Nusselt number decreases by 24%, even that the heat transfer area also increases by 20% due to an increase of the number of fins. From the flow field analysis, it is known that a smaller number of fins can cause greater disturbance in the pipe. Therefore, for the heat transfer benefit, the flow field constructed by flow passage is important than an increase of heat transfer area.

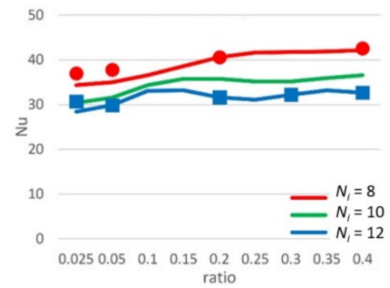


(a) $N_i = 8$ (b) $N_i = 10$ (c) $N_i = 12$

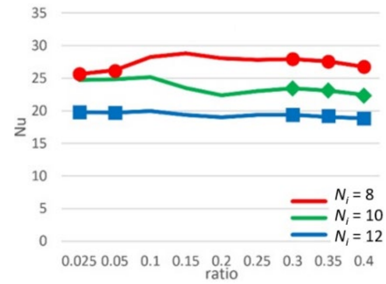
Fig. 11. Effects of the number of fins (N_i) on temperature distribution in cross section of the pipe



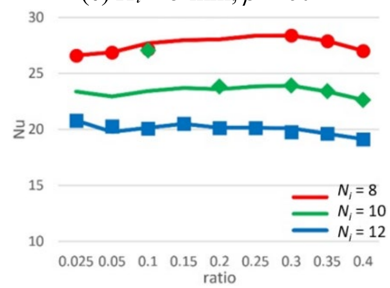
(a) $H_i = 2.5$ mm, $\beta = 60^{\circ}$



(b) $H_i = 2.5$ mm, $\beta = 120^{\circ}$



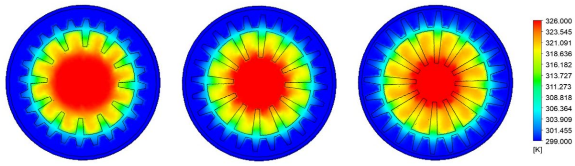
(c) $H_i = 5$ mm, $\beta = 60^{\circ}$



(d) $H_i = 5$ mm, $\beta = 120^{\circ}$

Fig. 12 Effects of the number of fins (N_i) and the ratio of fins (R_c) on Nusselt number

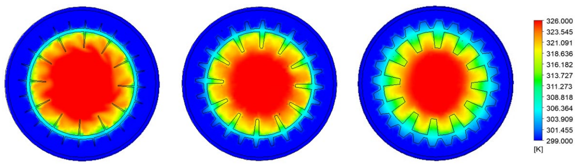
The effects of H_i on temperature distribution in cross section of the pipe are illustrated in Fig. 13. It shows that the central temperature in the pipe is higher with longer fins. Affected by the length of the fins, the center of the fluid passes quickly. It makes the difficulty in heat dissipation of hot fluid in the center of pipe. Hence, there is a relatively large space for heat transfer when the height of fins is short.



(a) $H_i = 2.5$ mm (b) $H_i = 4$ mm (c) $H_i = 5$ mm

Fig. 13. Effects of the height of fins (H_i) on temperature distribution in cross section of the pipe.

Fig. 14 shows the temperature distribution in cross section of the pipe with different R_c of fins. The width of fins becomes smaller when R_c is decreased. The influence of fins on the high temperature fluid in center region is insignificant and the heat dissipation is comparatively poor.



(a) $R_c = 0.05$ (b) $R_c = 0.2$ (c) $R_c = 0.4$

Fig. 14. Effects of the ratio of width of inner fin roots to circumference of inner flow passage (R_c) on temperature distribution in cross section of the pipe.

NEURAL NETWORKS MODELLING AND OPTIMAL DESIGN

In the recent decade, the neural networks (NN) were widely used for system modelling and optimization (Kant and Sangwan (2015), Sun et al. (2020)). The NN model consists of multiple layers of nodes, where input and output variables are represented as nodes of the input and output layer, respectively. Layers of nodes between the input and output layers are hidden layers. Each node connects to another has an associated weight and threshold. In this study, the performance model of the heat exchanger was constructed using the NN fitting app of MATLAB which uses a two-layer feed-forward network with sigmoid hidden neurons and linear output neurons. The network was trained by the Levenberg-Marquardt backpropagation algorithm.

To describe the performance model of the proposed DPHE, the parameters including the ratio of width of inner fin roots to circumference of inner flow passage (R_c), the number of fins (N_i), the height of fins (H_i) and the swirl angle of fins (β) are adopted as input parameters. The Nusselt number of Eq.(4) and Friction factor of eq.(5) are chosen as model outputs,

$$Nu = f(R_c, N_i, H_i, \beta) \quad (7)$$

$$f_c = g(R_c, N_i, H_i, \beta) \quad (8)$$

then they can be modelled by NN. To construct an NN model for the relationship between design parameters and performance indices, sufficient data sets are required. The procedure for constructing neural network model is shown in Fig. 15. In this study, 124 data sets were used to train the NN model of the first generation. Subsequently, another 10 data sets were used for model validation. If the resulting average error of predicted perform indices is greater than 5%, then these 10 data sets will be included to the training data sets and retrain the NN model to form the 2nd generation model. For each iteration, 10 validation data sets will be introduced to exam the model accuracy of the updated model until the average error is less than 5%. In general, validation data sets are selected evenly within the parameter space, however, a couple of data sets can be added around where there has high Nu and low f_c. The validation error of the NN prediction model to those obtained by ANSYS Fluent for each generation is shown in Fig. 16. A total of 164 data sets used for constructing the NN model is shown in Fig. 17. After construction the parameter-performance NN model, a searching algorithm such as genetic algorithms can be used for determining the optimal design parameters for the performance evaluation criterion (PEC) of the heat exchanger. The schematic diagram of optimal design procedure is shown in Fig. 18.

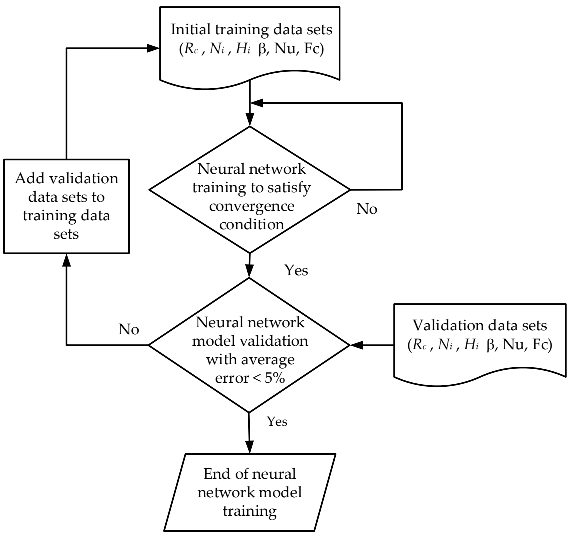


Fig. 15. Procedure for neural network model construction

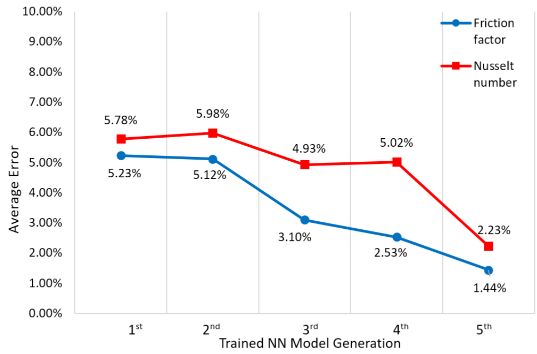


Fig. 16. Validation error of NN models

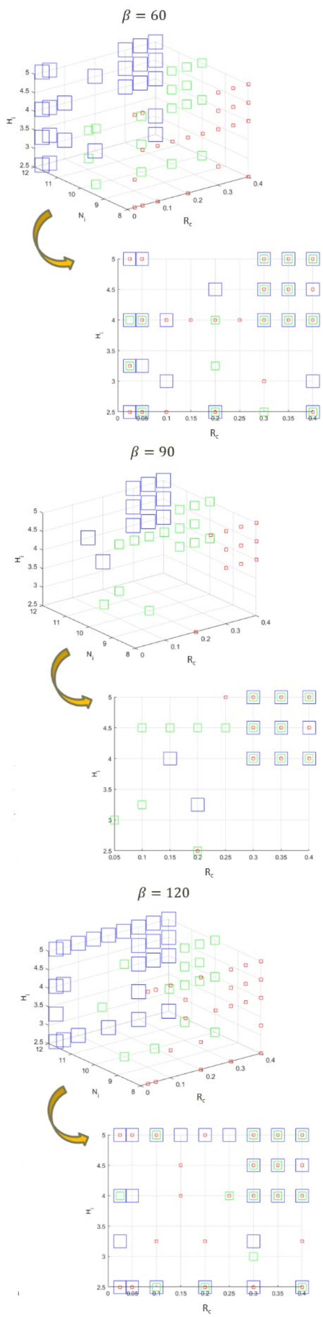


Fig. 17. The distribution of training design parameters

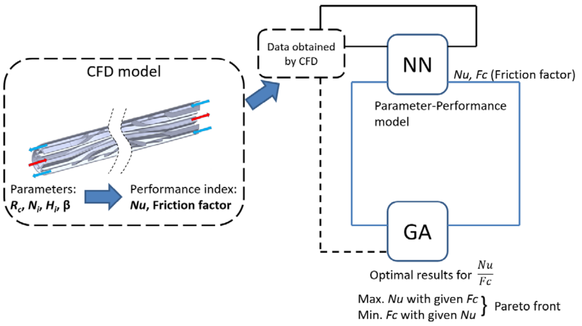


Fig. 18. Schematic diagram of optimal design using a genetic algorithm and neural network model

The genetic algorithm was proposed by John Holland in 1975. This algorithm is inspired by the process of natural selection (survival of the fittest). Genetic algorithms are commonly used to generate solutions for an optimization problem through biologically inspired operators such as mutation, crossover and selection. According to the range of design parameters specified in Table 3. The predicted PEC of optimal design parameters is 1.44 by the genetic algorithm in association with the NN performance model, which corresponding to the parameter sets of (R_c, N_i, H_i, β) = (0.229, 8, 2.5, 76.4). To validate the optimality of the obtained result, corresponding *PEC* of 6 parameter sets around the optimal set were investigated. As shown in Table 4, the model predicted results and CFD results illustrate the agreement among these parameter sets, which also confirms the superiority and reliability proposed approach.

Fig. 19 shows the scatter plot of *PEC* corresponding to training data points and results obtained by GA. This figure illustrates the magnitude of *PEC* from 0.613 to 1.44 by colors. The point with optimal *PEC* implies that it has high *Nu* value with low pressure drop. The black line represents Pareto front for the studied heat exchanger design, which indicates the maximum value of Nusselt numbers can be achieved under a specified friction factor. It provides a good suggestion for feasible solutions either with a constraint of pumping power or required heat exchange performance.

CONCLUSIONS

In this study, a numerical analysis was employed to simulate the fluid flow in the DPHE. The characteristics of thermal and flow field in the pipe with inner swirl fins was investigated via examining the influence of geometric parameters of fins including the ratio of width of inner fin roots to circumference of inner flow passage, the height of fins, the number of fins, and the swirl angle of fins. The velocity field in cross-section of pipe, Nusselt number, and friction factor were also discussed to explore the effects of variations of parameters on the fluid. Moreover, the performance model of DPHE was established using CFD data and neural networks method with validations. Then, a genetic algorithm was utilized for optimal design. The PEC of the heat exchanger was introduced as the index for searching the optimal parameter set. The main conclusions of the current investigation are listed in the following:

(1) In a pipe with swirl fins, the mixing of fluid occurs in the intersection of rotating segments and it causes perturbations. The friction factor increases with the increment of pressure drop caused by the disturbances. The disturbances interrupt the boundary layer and then enhance the heat transfer. Therefore, the efficiency of heat transfer at the intersections of rotating segments are better than others.

(2) An increasing of Rc, Hi and β results in larger pressure drop. In addition, the number of fins is inversely proportional to friction factor under same ratio of width of inner fin roots to circumference of inner flow passage.

(3) The Nu increases with the increase of swirl angle β. However, compare with other parameters, the influence of swirl angle β on Nu is not obvious. For the same Rc, the Nu decreases with an increasing of Hi or Ni. It reveals that the thermal performance not only affected by the heat transfer area but also influenced by flow conditions. In addition, the Nu is increased with an increasing of Rc under the conditions of short fins or fewer fins.

(4) This study showed that good performance predictions can be carried out by a well-trained neural network model. The optimal flow passage design corresponding to the PEC design was obtained with Rc = 0.2291, Ni = 8, Hi = 2.5mm, β = 76.4° for the proposed DPHE with swirl fins. The Pareto front also revealed that a 20% increase of Nu can be achieved at the cost of a 40% increase of friction factor. The design methodology in this study can be applied to the design of other types of heat exchangers.

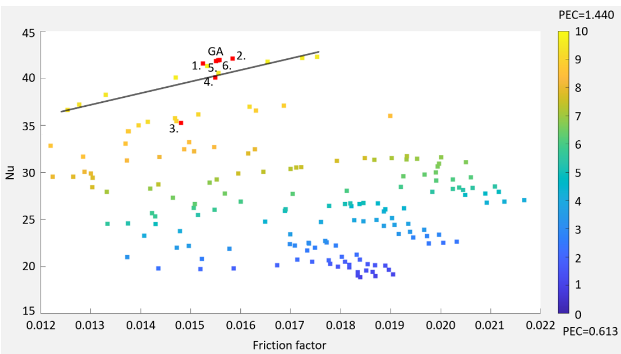


Fig. 19. The scatter plot of *PEC* values in (Nu, f_c) coordinate

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NOMENCLATURE

A area of inlet

D_h hydraulic diameter

D_o diameter of outer pipe

D_i diameter of inner pipe

f_c fraction factor

H_i height of the inner fins

H_o height of the outer fins

h convective heat transfer coefficient

k_f thermal conductivity of fluid

L_1 total length of the double pipe

L_2 length of adiabatic section

L_3 length of fin

L_4 gap length between each section

Nu Nusselt number

N_i number of inner fins

N_o number of outer fins

P_{in} pressure of inlet

P_{out} pressure of outlet

Δp pressure difference between inlet and outlet

PEC performance evaluation criteria

Q_{ave} average heat transfer rate

Q_h heat transfer rate of hot fluid

Q_c heat transfer rate of cold fluid

R_c the ratio of width of inner fin roots to circumference of inner flow passage

Re Reynolds number

W_1 width at the root of fins

W_2 width at the tip of fins

β swirl angle of fin

μ viscosity

ρ density

$\Delta LMTD$ logarithmic mean temperature difference

具旋轉鰭片之雙套管式熱交換器熱傳分析及最佳化設計

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摘 要

本文採用數值模擬和單目標最佳化方法對具旋轉鰭片之雙套管熱交換器的熱流性能進行了研究。研究討論的項目為旋轉鰭片的幾何參數，包括內鰭片根部寬度與內流道週長之比、鰭片數量、鰭片高度以及鰭片旋轉角。此外，模擬也假設固定雷諾數和固定入口溫度。透過分析摩擦係數和努塞爾數(Nu)，得出鰭片幾何形狀與管道熱流特性之間的關係。在最佳化研究方面，採用神經網路方法結合遺傳演算法建立熱交換器性能模型，並引入熱交換器性能評估標準(PEC)作為尋找最優設計參數的指標。除此之外，本研究也透過數值模擬驗證了最佳設計參數，並與其他參數進行比較。研究結果顯示了最佳解的強健性。這項研究也證明了單目標最佳化方法的準確性以及神經網路訓練數值建模的實用性。本研究的設計方法可為其他型式熱交換器的設計提供參考。