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Multi-Objective Optimization Design for Cooling Unit of Automotive Exhaust-Based Thermoelectric Generators

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In order to improve the performance of cooling units for automotive thermoelectric generators, a study is carried out to optimize the cold side and the fin distributions arranged on its inner faces. Based on the experimental measurements and numerical simulations, a response surface model of different internal structures is built to analyze the heat transfer and pressure drop characteristics of fluid flow in the cooling unit. For the fin distributions, five independent variables including height, length, thickness, space and distance from walls are considered. An experimental study design incorporating the central composite design method is used to assess the influence of fin distributions on the temperature field and the pressure drop in the cooling units. The archive-based micro genetic algorithm (AMGA) is used for multi-objective optimization to analyze the sensitivity of the design variables and to build a database from which to construct the surrogate model. Finally, improvement measures are proposed for optimization of the cooling system and guidelines are provided for future research.

Key words: ATEG, cooling unit, multi-objective optimization, DOE

INTRODUCTION

With the continuous sharp growth in world energy consumption, energy recovery techniques have long been in significant demand. Considering the huge amount of thermal energy in automobile exhaust, waste heat recovery has become a focus of research in vehicle energy savings. Therefore, a new technology, namely an automotive thermoelectric generator (ATEG), has been developed and has found use in an expanding array of applications. ATEG was developed on the basis of thermoelectric modules (TEMs) and offers reliable thermoelectric energy conversion with advantages such as zero emissions and low noise levels.

Usually, an entire thermoelectric generator (TEG) system consists of a heat exchanger, TEMs, a cooling unit and clamping devices. The hot sides of TEMs are in contact with the heat exchanger so as to retrieve the heat in the exhaust, and the cold sides are cooled by the cooling unit so that a temperature difference over the upper and lower surfaces of the TEMs is created. According to the principle of the Seebeck effect in Eq. 1,

$$\Delta V = (S_{\mathrm{A}} - S_{\mathrm{B}})(T_{\mathrm{h}} - T_{\mathrm{c}}), \quad (1)$$

where ΔV is the open-circuit voltage of TEMs, S_{A} and S_{B} are the Seebeck coefficients which are decided by the thermoelectric materials, and T_{h} and T_{c} are the temperatures of the hot and cold sides, respectively. From this we can infer that the generating capacity of TEMs depends mainly on the temperature difference between the heat exchanger and the thermoelectric cooler (TEC). Actually, the result is still not satisfying using current cooling methods to transfer heat rapidly and maintain temperature stability at the cold sides. Therefore, in addition to optimization of the heat exchanger and thermoelectric materials, the design and optimiza-

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tion of the TEC is an effective way to promote overall TEG performance, and should be discussed.

Hsiao et al.[1] applied TEGs on an automobile to recover waste heat, and found that a TEM presents better performance on the exhaust pipe than on the radiator, based on the simulation models and experiments. Liu et al.[2] designed a new system called a "four-TEG" system and assembled it into a prototype vehicle, and a maximum power of 944 W was obtained, which completely meets the automotive application. Gou et al.[3] proposed many suggestions to obtain better performance of TEGs, including enhanced cold-side heat transfer capacity. Thacher et al.[4] tested an ATEG in a light truck to measure the performance and determine which factors are important for optimizing ATEG design, and the results showed that insulating the exhaust and lowering the coolant temperature had a significant positive effect on the power. Kim et al.[5] fabricated a TEG using the engine water coolant of passenger vehicles, and the results revealed that the conventional radiator could be replaced by the proposed TEG without additional devices or redesign of the engine water cooling system of the existing radiator. He et al.[6] compared four different cooling methods for a TEG system, and showed that with the water cooling method, the co-flow and counter-flow methods did not need to be distinguished because of the small difference between them, but that they should be distinguished for the air cooling method. Du et al.[7] developed a TEG model coupled with exhaust and cooling channels and investigated the comprehensive design optimization of the cooling channel, and it was revealed that the net output power is generally higher with liquid cooling than air cooling. Deng et al.[8] constructed a new engine cooling system in which a TEG cooling unit was inserted; the results showed that coolant temperature exceeds its boiling point at high power and low vehicle speed, and may lead to a 15% decrease of the TEMs' generating capacity. Rezania et al.[9,10] applied a parallel micro-channel heat sink to a TEG, and the experimental result indicated that there was a unique flow rate that gave the maximum net power in the system at each temperature difference, and the temperature difference between the hot and cold sides of the TEG increased slowly with the flow rate at a constant imposed heat flux. Su et al.[11] built an entire TEG system which was validated by bench tests, and the performance of three different types of TECs was analyzed to determine the most efficient one, with results showing that the stripe-shaped TEC offered the best performance.

However, the study for TEC optimization is still not enough. Therefore, in order to promote the overall efficiency of heat transfer and increase the temperature difference between the hot and cold sides, multi-field coupled analysis was introduced to obtain evaluation indexes of a cooling device including the convective heat transfer, friction loss and compensation coefficients, and the change laws between these indexes and fluid flow or temperature were also analyzed to obtain the optimal optimization result.

DESIGN OF EXPERIMENT

Design of experiment (DOE) is one branch of modern mathematical statistics, which recognizes the key experimental factors and analyzes the relationships between input and output parameters so that the optimal parameter combination could be achieved with less time required for testing. In this paper, local sensitivity and global sensitivity are combined for the analysis of weight of impact factors, so that the impact factors are screened. The experimental design method used in the paper is central composite design (CCD), and there are three types (see Fig. 1). The first is circumscribed CCD (a), which may exceed the limiting interval easily; the second is inscribed CCD (c), which is likely to form an uneven sample space; and the last is face-centred CCD (b), which is compatible with quadratic response surface models.

WEIGHT ANALYSIS AND OPTIMIZATION DESIGN METHOD

The study of the relationship between evaluation indexes and impact factors in TEC is an indispensable part of the process of model parameterization, and weight analysis of impact factors can provide guidance for model modification as well as for the layout of fins in the TEC.

Optimization of the TEC is required to improve heat transfer efficiency while reducing the influence of friction loss, and therefore it is an optimization problem with multiple factors and constraints. In this paper, weight analysis is performed on the basis of a response surface method, and the later optimization is based on screening. According to the number of impact factors, a corresponding number of experimental points are generated, and based on these experiments, the approximation relationships between impact factors and performance parameters are fitted and projected on the response surfaces, which vary with the impact factors.

DESIGN PARAMETERS

The enhancement of heat transfer for a stripe-shaped TEC is achieved by arranging fins in the interior of the cavity to enlarge the heat convection area and increase the heat transfer coefficient, such that heat transfer is enhanced without a sharp increase in loss of friction.

Optimization Model

Based on the passive heat transfer enhancement theory, there are two ways to arrange the fins. One is with the fins perpendicular to the flow direction and arranged transversely inside the cavity (see

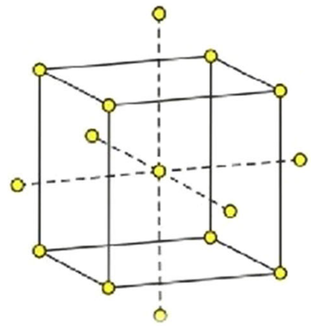

Fig. 1. Design samples spaces of circumscribed CCD (a) and face-centred CCD (b) and inscribed CCD (c).

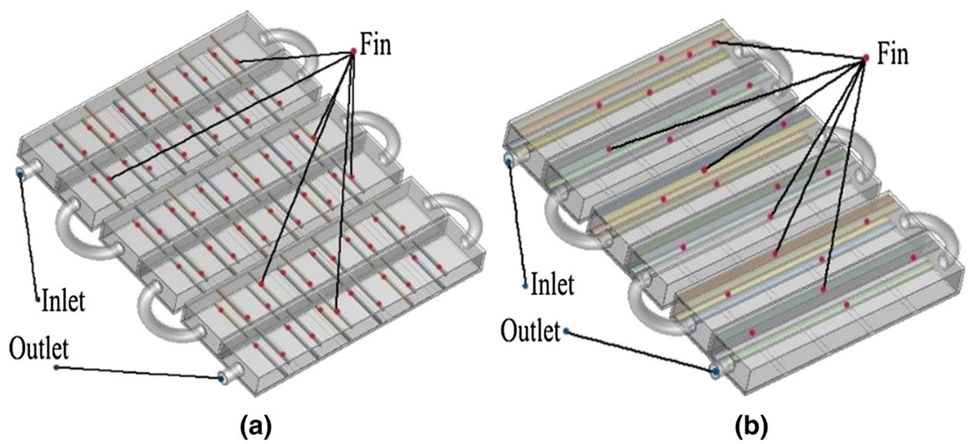

Fig. 2. TEC with fins arranged transversely (a) and longitudinally (b).

Fig. 2a). Therefore, the hydraulic diameter would compress and expand periodically with the variation of the flow cross-section area, and the near-wall fluctuation velocity would be increased, so that enhancement of heat transfer is achieved. The second is with the fins parallel to the flow direction and arranged longitudinally inside the cavity (see Fig. 2b), so that the walls are transformed into a plate–fin heat transfer surface to block the development of a boundary layer.

Performance Parameters and Impact Factors

In this study, evaluation indexes, which include convective heat transfer coefficient, friction loss coefficient and temperature difference between inlet and outlet, are also defined as performance parameters. With the cooling liquid assumed as an incompressible ideal fluid, and fluid density as a constant, then the friction loss coefficient would be linearly related to the static pressure drop. As a consequence, static pressure drop would be used as an output parameter instead of friction loss coefficient, and all performance parameters which are defined as output are:

$$\mathrm{DROP} = ave(\mathrm{Pressure})@\mathrm{inlet} - \mathrm{ave}(\mathrm{Pressure})@\mathrm{oulet}, \tag{2}$$

$$\mathrm{DT} = \mathrm{ave}(\mathrm{Temperature})@\mathrm{inlet} - \mathrm{ave}(\mathrm{Temperature})@\mathrm{oulet}, \tag{3}$$

$$\mathrm{CONVERF} = \mathrm{ave}(\text{Convective Heat Transfer Coefficient})\ @\text{Fluid Solid Interface Side}, \tag{4}$$

where DROP is the static pressure drop, CONVERF is the convective heat transfer coefficient, DT is the temperature difference between the inlet and out, and Fluid Solid Interface Side is the interface between the coolant and water tank.

There are two factors influencing the temperature difference between the inlet and outlet, which are the inlet flow mass (INVMASS) and the inlet temperature (INT), and the inlet flow mass also has an effect on the convective heat transfer coefficient. According to the difference between the two fin layouts, the other impact factors are also determined, which are fin thickness (ds_RX), fin distance away from walls (ds_L), fin spacing (ds_D) and fin

Table I. Limiting intervals of fin geometry

Impact factors	Fins arranged transversely Limiting intervals (mm)	Fins arranged longitudinally Limiting intervals (mm)
ds_RX	[1, 3]	[1, 3]
ds_L	[40, 50]	[4, 6]
ds_D	[40, 50]	[5, 7]
ds_H	[1, 5]	[1, 5]

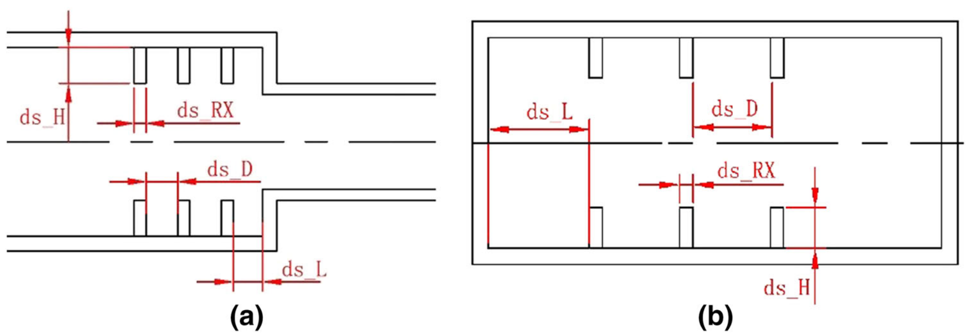

Fig. 3. Layout of water tank for fins arranged transversely (a) and longitudinally (b).

height (ds_H); their limiting intervals are shown in Table I.

When fins are arranged inside the cavity transversely, the layout of one single sub water tank is shown in Fig. 3a. In order to enhance heat transfer while fluid resistance is rarely increased, the fluid equivalent diameter in the place of compression and expansion should meet as:

$$\frac{De_1}{De_1} \geq 0.9, \quad (5)$$

where De_1 is the fluid equivalent diameter in the case of compression, and De_2 is the fluid equivalent diameter in the case of expansion.

When fins are arranged inside the cavity longitudinally, the layout of one single sub water tank is shown in Fig. 3b, and the geometry of the fins should meet as:

$$\frac{\delta}{De_2} \leq 0.05 \sim 0.1, \quad (6)$$

where δ is fin thickness, and De_2 is the equivalent diameter of the single channel.

Face-centred CCD is employed to design the experiment. Five levels for each impact factor are picked in the limiting interval, and all factors and the values of their levels are put into the design platform in turn, so 45 groups of experiments are generated. As shown in Fig. 4, the connection chart of impact factors and performance parameters is plotted. Left in the chart are the experimental schemes composed by each level of impact factors; different levels of different impact factors are interlaced and the whole sample space is covered. The test is carried out as the schemes and all test results are recorded.

Where AREA is the convective heat transfer area and OUT is the outlet temperature.

OPTIMIZATION CALCULATION

Change Law of Convective Heat Transfer Coefficient

An increasing convective heat transfer coefficient is one of the most effective measures to enhance heat transfer. According to the fin layouts, the relationships between different impact factors and convective heat transfer coefficient vary. The impact diagram of factor weight plotted in Fig. 5 revealed that apart from inlet flow mass (IVMASS) and inlet temperature (INT), as for fins arranged transversely, the first three impact factors are fin height (ds_H), fin distance away from walls (ds_L) and fin spacing (ds_D), and as for fins arranged longitudinally, the first three impact factors are ds_RX, ds_H, and ds_D.

When fins are transversely arranged, the enhancement of heat transfer is mainly reliant upon the change of fluid into a pulsating flow and intensification of the radial flow, so that the overall turbulence intensity would be improved. When fins are longitudinally arranged, the overall thermal resistance is decreased by blocking the development of the boundary layer, and then the heat transfer would be enhanced. According to the correlation curves between a convective heat transfer coefficient and each single variable, the first three factors which influence the convective heat transfer coefficient most are plotted in Fig. 6.

It is indicated that with the increase of fin thickness, the convective heat transfer coefficient would increase when fins are arranged transversely and longitudinally, but a greater increase occurs with the former.

As for fins arranged transversely, because of the collision between fluid and fins, a vortex zone is formed and the flow becomes turbulent. Meanwhile, when the fluid flows through the top of the fins, affected by the near-wall friction, fluid velocity at the bottom is smaller than that in the main flow region. Thus, the circulation between the layers is strengthened. However, when fin thickness continues to increase, the loss of fluid momentum would be too big to afford the fluid to hit the next fin at a high velocity, so turbulence intensity is going to be decreased. Therefore, the convective heat transfer coefficient would stop increasing and would finally tend to be stable.

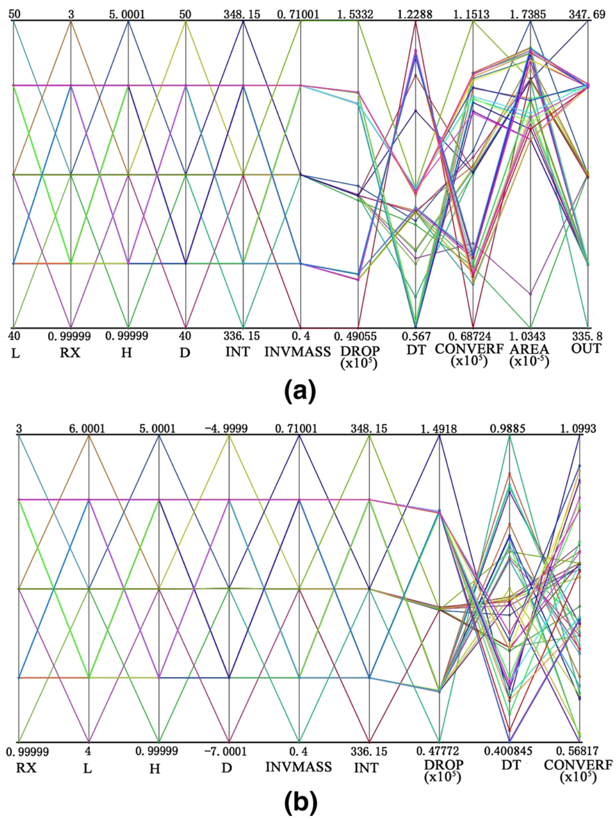

Fig. 4. Sample space filling diagram for fins arranged transversely (a) and longitudinally (b).

As for fins arranged longitudinally, the enlargement of fin surface area is equivalent to the blocking of the development of a boundary layer. If fin thickness is small, the friction coefficient of the fin surface would decrease as the Reynolds number increases, and the turbulence intensity is strengthened mainly through the increase of surface friction. As a result, with the increase of fin thickness, the turbulence intensity would be decreased. However, as long as fin thickness is large enough, the relationship between surface friction and the Reynolds number would not be obvious, and the turbulence intensity is mainly determined by the Reynolds number. Therefore, the convective heat transfer coefficient would increase continuously with the increase of thickness. But once the fin thickness increases to an extent value, it would be hard for fluid to divide the boundary layer when it flows through fins, and the convective heat transfer coefficient would no longer be increased.

Combined with Fig. 5, Fig. 6 shows that the fin spacing makes a negative contribution to the convective heat transfer coefficient, which means there would be a minimum convective heat transfer coefficient value under a certain value of fin spacing. When fins are arranged longitudinally, fluid in the central part between two adjacent fins is turbulence. As fin spacing increases in a certain range, the turbulence area expands constantly, and leads to an increasing convective heat transfer coefficient. When fins are arranged transversely, the variation of fin spacing is equal to the change of fluid pulsation frequency. For turbulence, there is a corner frequency for the heat transfer enhancement of

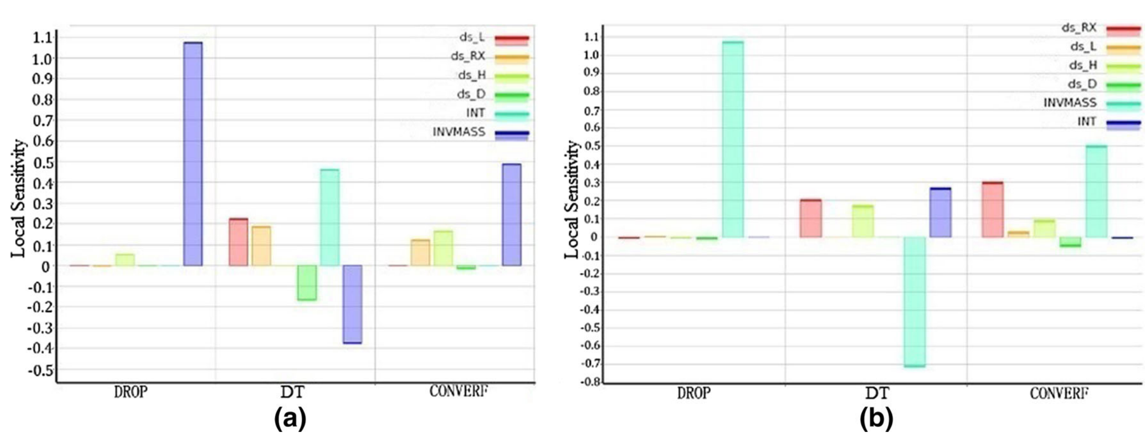

Fig. 5. Impact diagram of factor weight for fins arranged transversely (a) and longitudinally (b).

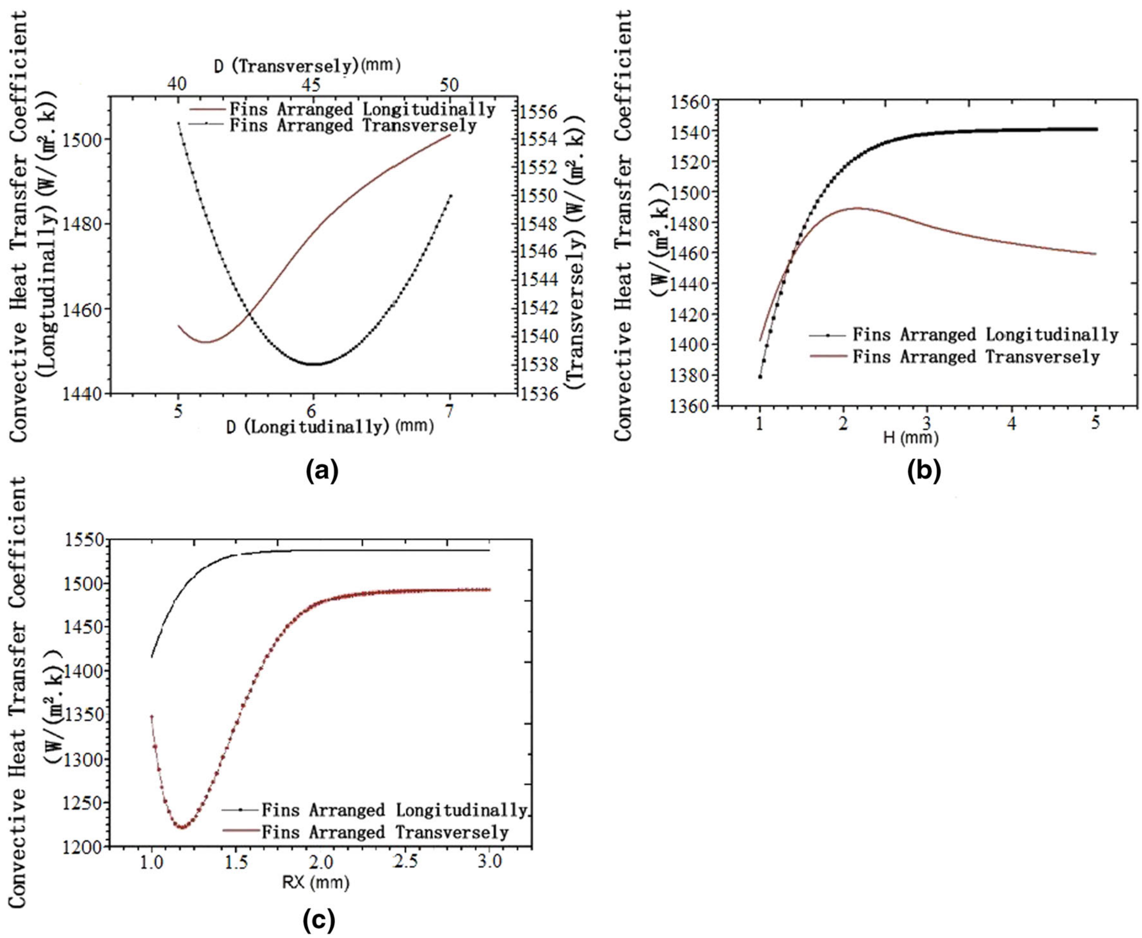

Fig. 6. Change law between convective heat transfer coefficient and fin spacing (a), fin height (b) and fin thickness (c).

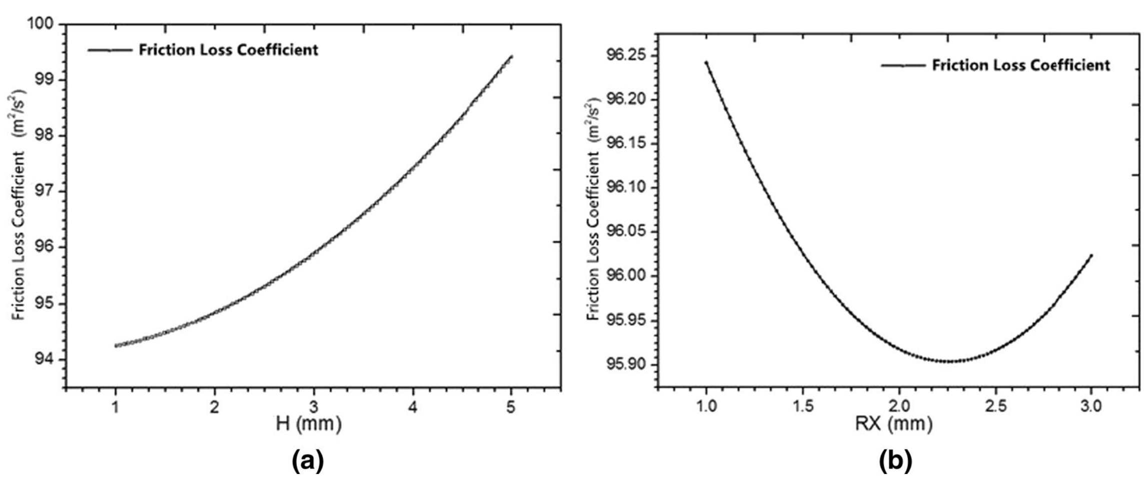

Fig. 7. Change law between friction loss coefficient and fin height (a) and fin thickness (b).

pulsating flow, which means the effect of enhancement is worst under this frequency (see Fig. 6a).

As shown in Fig. 6b, with the increase of fin height, the convective heat transfer coefficient increases rapidly first, then the rate of increase keeps on dropping until the convective heat transfer coefficient finally achieved the asymptotic value. When fin height is increased, the top of the fins would be closer to the main flow region, and the collision between fluid and fins would be intensified, so that the overall turbulence intensity is increased. However, if the fin height is too high, the thermal resistance along the direction of height would be increased, especially when fins are arranged longitudinally, the increase would be extraordinarily obvious, which may even make the heat difficult to transfer from the top to the root of the fins, so that the convective heat transfer coefficient would be decreased.

Change Law of Friction Loss Coefficient

According to Fig. 5, for fins arranged transversely, the influence of fin height on the friction loss coefficient is most obvious, but when fins are arranged longitudinally, apart from inlet flow and inlet temperature, the impact of the other factors on the friction loss coefficient is relatively small.

As shown in Fig. 7, when fins are arranged transversely, with the increase of fin height, the friction loss coefficient is increased due to the increase of fluid pressure caused by the decrease of cross-sectional area. In addition, there is an inflection point in the correlation curve between friction loss coefficient and fin thickness, and the friction loss coefficient would be at its minimum under this inflection point.

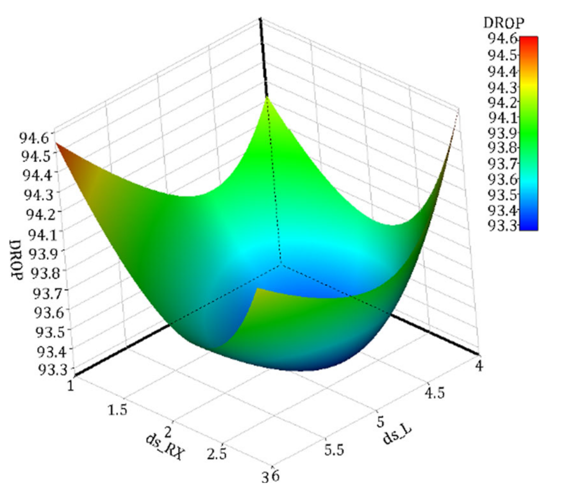

Fig. 8. Relationship between friction loss coefficient and fin thickness and fin distance away from walls.

As for fins arranged longitudinally, it is known that fin thickness, fin spacing and fin distance away from walls are all negatively contribute to the friction loss coefficient; thus, there would be a minimum value of friction loss coefficient (see Fig. 8).

Change Law of Temperature Difference between the Inlet and Outlet

When fluid flows in a TEC, it dissipates heat to the environment and the overall temperature falls as a consequence. Assuming the fluid is ideal, it is reasonable to take the temperature difference between the inlet and outlet as the valuation index of the heat dissipated by the TEC. As shown in Fig. 5, for fins arranged transversely, impact factors which effect the temperature difference obviously are fin distance away from walls, fin thickness and fin spacing, and for fins arranged longitudinally, impact factors which effect the temperature difference obviously are fin thickness and fin height.

When fins are longitudinally arranged, the increase of fin height is equivalent to the enlargement of the heat transfer area, although the thermal resistance along the direction of fin height would also be increased. Therefore, the temperature difference would increase at first and then decrease when the fin height continues increasing (see Fig. 9a). On the other hand, with the increase of fin thickness, the temperature difference would increase to the maximum value, and then begin to fall slightly (see Fig. 9b). Figure 10 shows that there is a maximum temperature difference value under the comprehensive effect of fin thickness and fin height.

When fins are transversely arranged, turbulence intensity and the contact area between fins and the main flow region are all affected by fin thickness, and there is an inflection point in the correlation

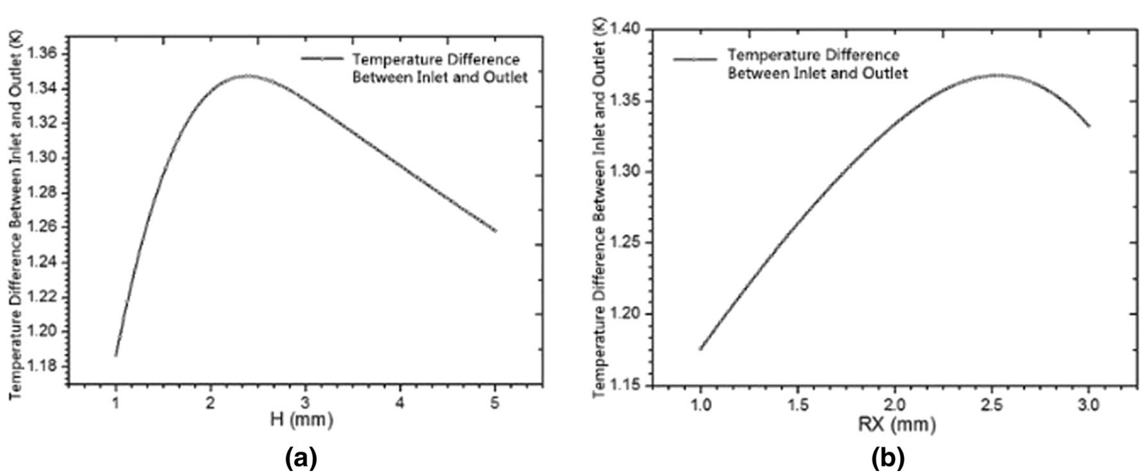

Fig. 9. Change law between temperature difference and fin height (a) and fin thickness (b).

curve between temperature difference and fin thickness (see Fig. 11a). Besides, the location of fins in the main flow region could also be adjusted by varying the fin distance away from the walls to make the synergistic effect between the flow field and temperature optimal. As shown in Figs. 11b and c, the maximum value of temperature difference all existed with the variation of fin spacing or fin distance away from walls. Combined with Fig. 12, it is known that there is a maximum tem-

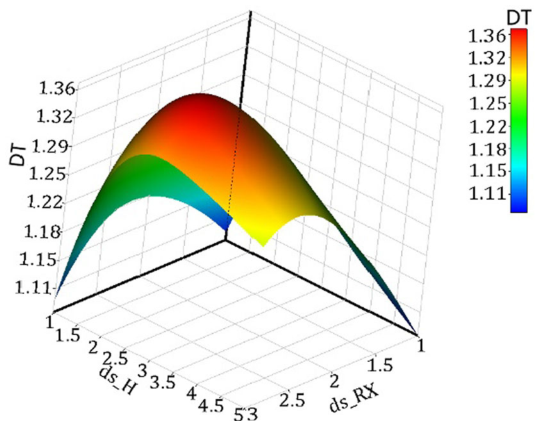

Fig. 10. Relationship between temperature difference and fin thickness and fin height.

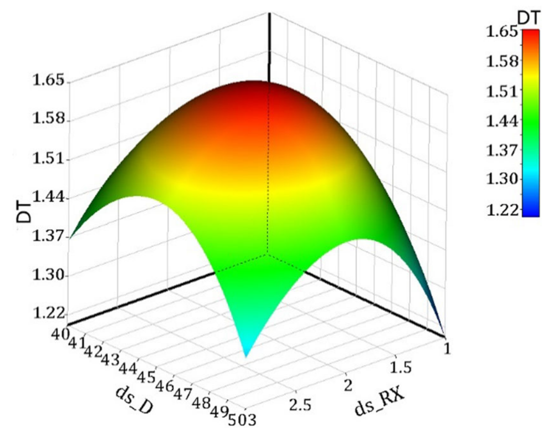

Fig. 12. Relationship between temperature difference and fin thickness and fin spacing.

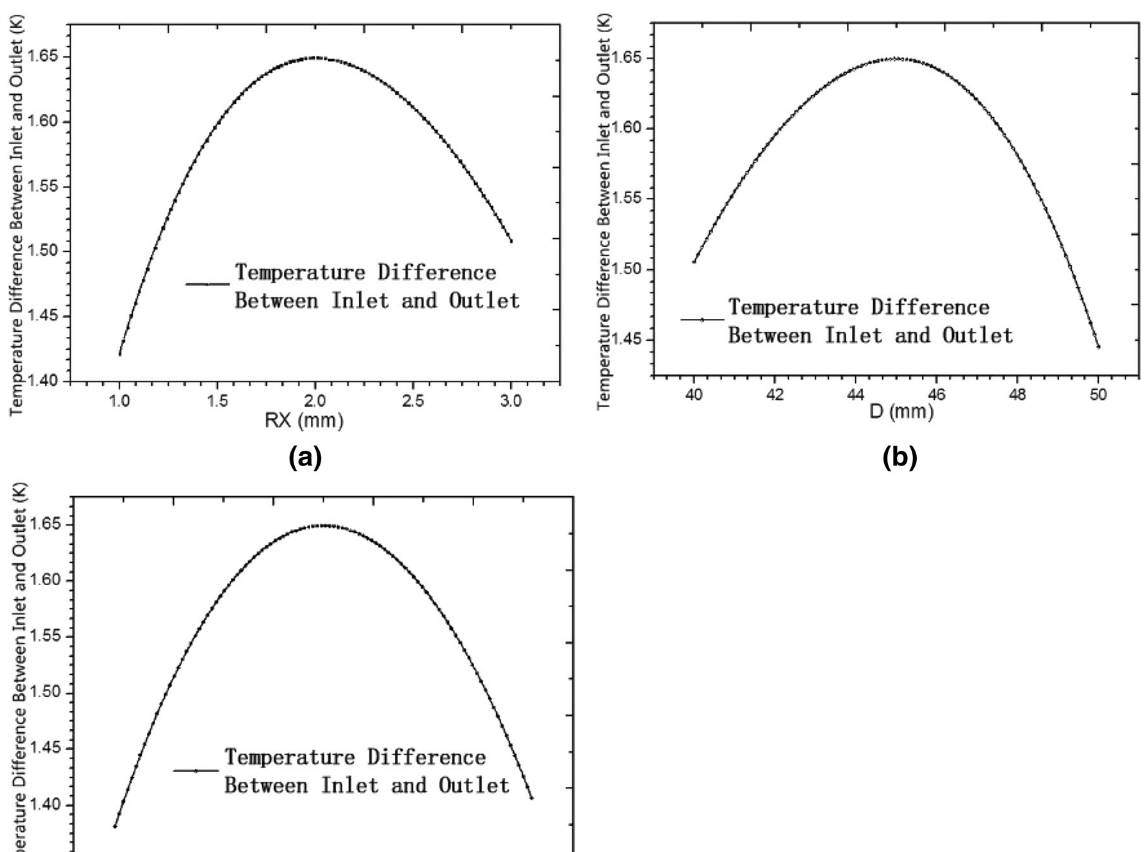

Fig. 11. Change law between temperature difference and fin thickness (a) and fin spacing (b) and fin distance away from walls (c).

Table II. First three candidates of optimization results for fins arranged longitudinally

	ds_RX	ds_L	ds_H	ds_D	INVMASS	INT
Candidate A	2.8	5.2	3.6	5.2	0.41	346.33
Candidate B	2.3	5.5	3.5	6.0	0.41	347.31
Candidate C	2.0	4.7	2.8	6.7	0.40	342.74

Table III. First three candidates of optimization results for fins arranged transversely

	ds_RX	ds_L	ds_H	ds_D	INVMASS	INT
Candidate A	2.3	45.2	2.8	45.3	0.41	347.29
Candidate B	1.9	42.9	3.4	42.9	0.41	347.84
Candidate C	2.6	49.8	3.2	48.4	0.40	348.10

Table IV. Performance parameters of optimization candidates

	DROP	DT	CONVERF	ϖ
Fins arranged transversely				
Candidate A	52.10	2.59	1301.4	20.086
Candidate B	53.05	2.56	1309.1	20.684
Candidate C	51.33	2.34	1301.4	21.914
Fins arranged longitudinally				
Candidate A	51.69	1.89	1239.5	27.35
Candidate B	50.41	1.86	1253.6	27.14
Candidate C	49.72	1.77	1257.9	28.16

perature difference on the value-taking surface composed by fin thickness and fin spacing.

RESULTS ANALYSIS

Design of a TEC is a comprehensive problem with multiple parameters and multiple constraints. The primary objective is to meet the demand of cooling performance, but the corresponding fluid resistance should also be considered; therefore, a blind pursuit of cooling performance is not advisable.

Based on the relationships between impact factors and performance parameters established in the previous paragraphs, the goal-driven optimization method is adopted to obtain the optimal design. The optimization method used is screening, and 1000 groups of experimental samples are screened out. In addition, the sample spaces are sorted by the evaluation index grade. Temperature difference is regarded as the optimization goal of the highest grade in this paper,; the second place is pressure drop. The first three candidates of the optimization results are shown in Tables II and III.

Table IV indicates that for fins arranged transversely, the temperature differences in the three schemes are all bigger than if the fins were arranged longitudinally. Besides, with the same increase of fin height, the enlargement of convective heat transfer area for fins arranged longitudinally is more, while the convective heat transfer coefficient is smaller when compared with fins arranged transversely.

However, in all the schemes, friction loss is larger when fins are arranged transversely. Therefore, a new performance parameter-compensation coefficient is defined, as follows, to analyze the performance of two TECs furthermore.

$$\varpi = \frac{\Delta h}{\Delta T} \times 100\%, \tag{7}$$

where ϖ is the compensation coefficient, Δh is the pressure drop, and ΔT is the temperature difference.

Actually, the compensation coefficient is the friction loss coefficient value per the temperature difference, and the results show that for fins arranged transversely, the compensation coefficient is smaller. Meanwhile, considering the principle of temperature difference maximization, the optimal optimization result turns out to be candidate A of fins arranged transversely.

CONCLUSION

Based on the theory of heat transfer enhancement, cooling efficiency is improved through changing the layout of fins in the cavity transversely or longitudinally, enlarging the convective heat transfer area and increasing the convective heat transfer coefficient of the walls. The study focuses on the influence of the geometric parameters and arrangement of fins on cooling performance. According to the response surface method, fin thickness, fin spacing, fin height and fin distance away from walls are regarded as impact factors. The convective heat transfer coefficient, friction loss coefficient, and temperature difference between the inlet and outlet are regarded as performance parameters, and the relationships between these performance parameters and each impact factor are analyzed. Furthermore, the optimization method (screening) is adopted to screen out the optimization scheme candidates, and the optimization result is determined by comparing the compensation coefficients.

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