CONVECTIVE HEAT TRANSFER ENHANCEMENT: MECHANISM, TECHNIQUES AND PERFORMANCE EVALUATION

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Abstract In this keynote lecture the existing mechanisms for enhancing single phase convective heat transfer are reviewed and the fundamental mechanism, that is, reducing the intersection angle between fluid velocity and temperature gradient, is presented in detail. This basic idea is called field synergy principle (FSP). A great number examples are provided to demonstrate the validity of the FSP. Some typical convective heat transfer phenomena are analyzed and found that their characteristics can be well understood by the FSP. An effective way for improving convective heat transfer performance of an existing heat transfer structure is to reveal the locations with bad synergy (i.e., large local synergy angle) and improve it by changing local structure of the surface. Examples of new enhanced surfaces are provided which are developed under the guidance of the FSP. Then the thermo-hydraulic performance comparisons of the enhanced structures compared with the reference one are discussed under three constraints: identical pumping power, identical pressure drop and identical flow rate. All the three constraints can be unified in a picture with log (f_e / f_o) and log(Nu_e / Nu_o) as abscissa and ordinate, respectively. An enhanced technique can be expressed in this plot and under what constraint it enhances heat transfer can be clearly identified.

1. Introduction

1.1 Background

Heat transfer process is absolutely among the most intimate physical processes related to the human society. Although the basic principles of heat transfer theory have been built up at least for more than half- century, its development is still one of the hottest topics in the field of the applied thermal science and engineering. Among the three modes of heat transfer convective heat transfer is the most active and most widely applicable one. Especially in the waste heat recovery engineering, the waste heat (or thermal energy) is usually stored in gases or water with their temperatures appreciably higher than the environmental temperature. An efficient recovery of this part of thermal energy is an important way to increase the energy utilization efficiency. An approximate estimation of the industrial waste heat in China is made by the authors' group. Based on the national statistics of the year 2010 [1] energy consumption of industry takes 71.1% of the total national energy consumption. It is estimated that the energy utilization efficiency in China is about 33% [2] , and the rest 67 % is released as industrial waste energy most part of which still can be reused. A survey of the industrial waste heat shows that a major part of the industrial waste heat is carried away by hot gases. In order to efficiently reuse waste heat enhanced convective heat transfer techniques are of great importance. The authors' group has been working on this subject for a number of years. The focus of the present lecture is thus concentrated on the enhancement of single phase convective heat transfer, from the mechanism, techniques to performance evaluation plot..

Generally speaking, before 1970s, most studies of convective heat transfer focused on revealing the mechanisms of convective heat transfer and establishing correlations between Nusselt number and Reynolds number, and there was almost no such a term as "heat transfer enhancement/

augmentation/intensification" in the open literature and textbooks. Later, the energy crisis in 1970s broke this situation. The dilemma greatly shocked the global economy and forced people to reduce the excessive energy consumptions and efficiently utilize the available energy sources, i.e., seeking methods to enhance heat transfer in a certain process with minimal energy consumption. It is estimated that heat transfer processes are involved in the usage of about 80% of all kinds of energy in the world [3]. In the transfer of heat, or thermal energy, people pay penalty of temperature difference and fluid pumping power, and both of them are closely related to energy consumption. Hence, convective heat transfer enhancement then became one of the hottest research subjects in the field of heat transfer. To the authors' knowledge the terminology of enhancement of heat transfer was first published in journal in 1979 [4]. After 1990s, the technology of heat transfer enhancement has evolved from the so-called second- generation technology to the third-generation technology [5,6] and significant achievements have been obtained. In 2002, the fourth-generation concept of heat transfer enhancement technology was proposed in [7]. In the recent decade the importance of the waste heat recovery is more deeply recognized by researchers all over the world which further strengthens the study on enhancement of convective heat transfer.

1.2 Introduction to field synergy principle

During the last few decades, great achievements on convective heat transfer enhancement have been obtained and various kinds of technologies have been adopted Generally speaking such techniques can be classified into four aspects i.e., (1) mixing the main flow and/or the flow in the wall region by using rough surface, insert, vortex generators, etc.[8], (2) reducing the thermal boundary layer thickness by using interrupted fins or jet impingement, etc.[9], (3) increasing velocity gradient at wall, and (4) adopting different kinds of fins [8,9] . However, up to the end of last century, the essence of the single phase convective heat transfer enhancement was still unclear. Although some explanation can account for the mechanism of some heat transfer enhancement techniques, but it may fail for other enhancing devices. For example, the mechanism of "decreasing thermal boundary layer thickness" can well explain why off-set fin can enhance heat transfer compared with the plain ducts , but it can not account for why wavy channel can enhance heat transfer compared with the plain duct. In a word, there was no unified principle or theory which could explain the physical mechanism for all kinds of enhancement techniques of single-phase convective heat transfer process till the end of the last century.

In 1998, Guo and his co-workers [10-12] firstly proposed a new understanding of enhancing single-phase convective heat transfer for the parabolic fluid flow situation. A brief introduction is presented as follows.

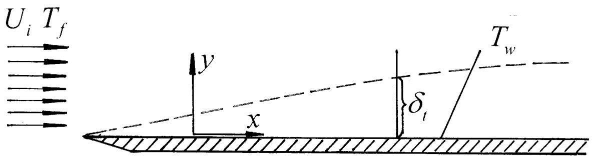


Fig. 1 Thermal boundary layer over a plate

Consider a 2-D boundary-layer steady-state flow over a flat plate at zero incident angle as indicated in Fig.1, the energy equation can be expressed as follows:

$$\rho c_{\mathrm{p}}\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) \tag{1}$$

The integral of Eq. (1) over the thermal boundary layer shown in Fig.1 is

$$\int_0^{\delta_{\mathrm{t}}}\rho c_{\mathrm{p}}\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)\mathrm{d}y=-k\left.\frac{\partial T}{\partial y}\right|_{\mathrm{w}} \quad q_{\mathrm{w}} \tag{2}$$

where the condition of $\left.\frac{\partial T}{\partial y}\right|_{\delta_t} = 0$ is introduced.

It can be seen from Eq. (2) that higher convection term leads to higher heat flux on the plate wall. Increasing the integral value of convection term over the thermal boundary layer can thus lead to the increase in the heat flux at the wall, i.e., enhancing the convective heat transfer. Mathematically the convective term $\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)$ is the dot production of two vectors，velocity and temperature gradient：

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\vec{U}\bullet\nabla T=\left|\vec{U}\right|\left|\nabla T\right|\cos\vartheta \quad (3)$$

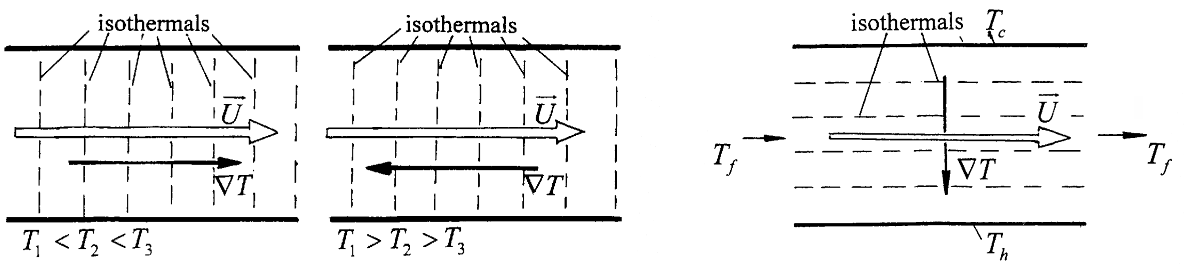
where θ is the local intersection angle between velocity and temperature gradient. For a case with given oncoming flow rate and temperature difference, the larger the absolute value of $\cos\theta$, the larger the heat flux. For the simplicity of presentation in the following discussion we will take the heating case as an example, and say that the smaller the intersection angle the larger the heat transfer flux.

If we take the two vectors as two forces, then according to the Webster Dictionary [13]: when several actions or forces are cooperative or combined, such situation can be called “synergy”. Hence the above new concept about enhancement of convective heat transfer is called field synergy (coordination) principle.

The extension of the field synergy principle from parabolic flow to elliptic flow was completed in [14] which is not presented here for simplicity.

In 2005 Guo and his co-workers reclaim the concept of FSP in [15] as follows. The synergy (coordination) between the velocity vector and the temperature gradient means that: (1) the intersection angle between these two vectors should be small; (2) the magnitudes of local velocity and temperature gradient should be simultaneously large ; (3) for the case of inner flow, the cross sectional velocity and temperature profiles should be as smooth as possible. These statements can be regarded as a general description of the field synergy principle

From the above discussion two extreme or asymptotic situations can be deduced [16]. These two extreme situations are shown in Fig. 2.



Fluid heated Fluid cooled

(a) Perfect synergy (b)The worst synergy

Fig. 2 Two extreme situations of convective heat transfer

Figure 2(a) presents the perfect synergy where the intersection angle either equals zero (fluid is heated) or equals 180 degrees (fluid is cooled). In that situation the heat transfer coefficient (or the Nusselt number) is linearly proportional to the magnitude of the fluid velocity (or the Reynolds number). This case will be called deduction 1 of FSP. The worst synergy situation is shown in Fig. 2(b), where the fluid velocity vector just perpendicular to the temperature gradient. For this case no matter how large the velocity is it does not have any effect on the convective heat transfer. This will be called deduction 2 of FSP..

1.3 Indicators of synergy

Two indicators have been proposed: field synergy number and field synergy angle. In [15] the field synergy number is introduced. By re-formulizing Eq.(2) into a non-dimensional form with the convective term expressed in vector form, finally we can obtain:

$$Fc = \frac{Nu_x}{Re_x Pr} = \int_0^1 (\bar{U} \cdot \nabla \bar{T}) \mathrm{d}\bar{y} \qquad (4)$$

where Re_x, Nu_x and Pr are, respectively, local Reynolds number, local Nusselt number and Prandtl number , $\bar{U}$, $\overline{\nabla T}$ and $\bar{y}$ are the non-dimensional velocity vector , non-dimensional temperature gradient and dimensionless distance normalized by the oncoming flow velocity , reference temperature difference and the thickness of the boundary layer, respectively.

It can be seen from Eq.(4) that the maximum upper limit of Fc equals 1,when the velocity vector is in perfect coordination with the temperature gradient, and both dimensionless velocity and temperature gradient are all the way equal to 1 in the boundary layer. In that perfect case we will have:

$$Nu_x = Re_x Pr \qquad (5)$$

Actually, Fc for all the existing convective heat transfer processes encountered in engineering are far below 1 by about two orders, showing a large room for the heat transfer enhancement study. The field synergy number is a criterion to judge how far a specific convective heat transfer process differs from its ideal performance evaluated the process as a whole.

In many engineering practices it is required to improve some existing heat transfer surfaces or structures such that the heat transfer enhancement can be made with mild or less pressure drop penalty. In order to develop enhancing technique more effectively, the enhancing element should be positioned in the place where the local synergy between velocity and temperature gradient is worse or the worst. In this regard, the local synergy angle is the only choice to show the goodness of synergy, which is defined by:

$$\theta = \cos^{-1}\left(\frac{U \cdot \nabla T}{|U||\nabla T|}\right) \qquad (5)$$

The local synergy angle can be obtained by numerical simulation. If some commercial software, such ad FLUENT, is used, with a user defined subroutine this angle can be obtained from predicted velocity and temperature data with ease [17,18] . When a specific convective case is analyzed from FSP point of view the local synergy angle can identify where the local synergy is bad hence enhancement technique is needed.

With the local synergy angle at hand researchers have tried to get domain averaged synergy angle appropriately. Five definitions for the domain averaged synergy angle has been tested [19] and it is found that except the simple arithmetic mean method all other four method can get qualitatively the same variation trend with some other parameters, among whom the domain integration mean is recommended：

$$\theta_m = \arccos \frac{\sum |\vec{U}| \bullet |gradt| \cos\theta_i dV}{\sum |\vec{U}| \bullet |gradt| dV} \qquad (6)$$

It can be seen that the cosine of this average synergy angle θ_m is the mean value of the domain integration for the local cosine value.

It is worth mentioning that when the averaged synergy angle is used, what we are interested in is its variation trend and relative magnitude, rather than its absolute value.

1.4 Techniques for enhancing single phase convective heat transfer

All techniques discussed in this paper are of passive type. For the single phase flow and heat transfer passive techniques can be divided into four aspects, simply speaking, they are: 1) mixing fluids;2) reducing boundary-layer thickness;3) increasing wall velocity gradient; and 4) adopting fins. It has been shown in [16] that all the four aspects can be unified by the FSP. It now becomes very clear that the FSP can combine all the existing mechanisms into the fundamental enhancement mechanism: improving the synergy between velocity and temperature gradient. Since the proposal of the FSP in 1998 by Guo and his co-workers and later enhanced by several researchers, this new concept has been accepted by more and more researchers in the international heat transfer community, and several hundreds papers have been published. In 2011 after made a comprehensive review of the new enhancement techniques Bergles implored the international heat transfer community to pay more attention to the field synergy principle [20]. He wrote: 'In addition to keeping an eye out for new literature, it is recommended that the practitioner of enhanced heat transfer consider two more fundamental and philosophical works that appeared recently. Guo [33] (Ref.[12] of the present paper-authors) advanced the Field Coordination Principle, which states that the coordination between the fluid velocity and the temperature gradient determines the convective heat transfer enhancement.” . In this lecture apart from above introduction we will further present application examples of enhancing techniques designed under the guidance of the FSP.

1.5 Performance evaluation methods for enhancing techniques

It is very crucial on how to quantitatively evaluate the thermo-hydraulic performance improvement for a given enhancement technique. Performance evaluation is normally made by comparing the performance of the enhanced surface with a corresponding referenced structure. In order to make such performance evaluation clear and simple, following two issues should be claimed.

First, what are the comparison conditions. Usually enhanced surface and the referenced one have different structure, including different heat transfer surface area, different characteristic length, different cross section for fluid flow, etc. In order to make a meaningful comparison, following practices, or assumptions are often adopted [22]:

(1) The thermophysical properties of fluid are constant;
(2) Heat transfer area used for calculating the convective heat transfer coefficient of enhanced surface is the same as that of the reference one;
(3) The cross sectional area used for calculating the average velocity of enhanced surface is the same as that of the reference one;
(4) The reference dimension used for calculating the dimensionless characteristic number of enhanced surface is the same as that of the reference one.

The second important issue is what are the comparison contents? In most engineering practices for developing enhancing techniques, among the four objectives indicated in [21]three objectives are closely related to heat transfer rate per identical flow rate, or per identical pressure drop, or per identical pumping power. Thus, it is rationale to take the three constraints, i.e., comparing the enhanced surface with the reference one for identical flow rate, identical pressure drop and identical pumping power as the comparison contents, or evaluation objectives [22-29].

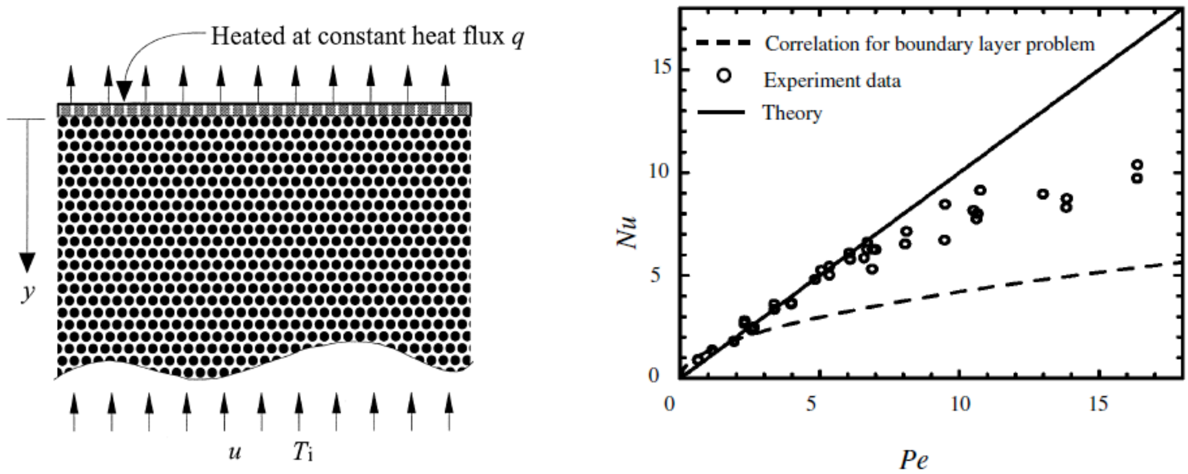
To visually present the performance evaluation results for enhancing techniques different kinds of plots have been suggested in literatures. However, among the existing comparison plots none can be used to indicate whether the heat (thermal energy) transmitted by the enhanced surface per unit pumping power can be increased compared with the reference one, or whether the heat transmitted per unit pressure drop can be increased. In 2009 Fan and her co-workers [30] proposed a log (Nu_e / Nu_o) vs. log (f_e / f_o) plot for a visual presentation of comparison result. In this new plot different enhanced techniques for the same reference system can easily and clearly be compared for their energy-saving performance. In the following discussion its applications will be illustrated.

2. Verifications of FSP

2.1 Verification of FSP deduction 1

The FSP deduction 1 says: when fluid flow velocity is parallel to fluid temperature gradient the heat transfer rate will be linearly proportional to flow velocity, or $Nu : \mathrm{Re}$. At the time when the FSP was just proposed, two research papers were published independently, and they were not specially designed to verify the FSP, but coincidently their results demonstrate deduction 1.One of the two experimental studies is now introduced as follows.

Zhao and Song [31] proposed an enhanced heat transfer model to meet the demand for dissipating high heat flux as shown in Fig. 3(a). A permeable plate is imbedded in a semi-infinite porous medium. A fluid at infinity of temperature T_i flows upwards through the porous medium and is heated by the downward facing permeable plate with constant heat flux. When the oncoming velocity is not large the flow and heat transfer in the porous medium region may be assumed one dimensional.It can be shown that for such a case we have: $Nu = \mathrm{Re}\,\mathrm{Pr} = Pe$ In order to verify above analysis a test facility was designed .Their test results are presented in Fig. 3(b). It can be clearly observed that when Pe is less than 6-7, test data agree with analytical results exactly and it is a perfect synergy case where flow velocity direction coincides with fluid temperature gradient!



(a) Ideal model (b) Test results

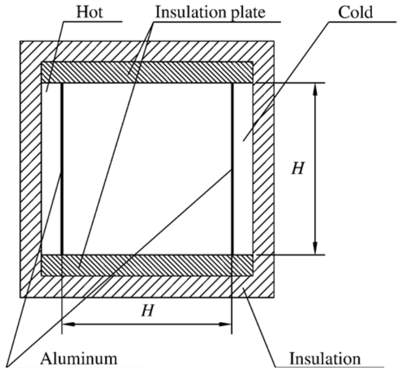
Fig. 3 Test results of forced convection in porous medium

Another experimental verification can be found in [32].

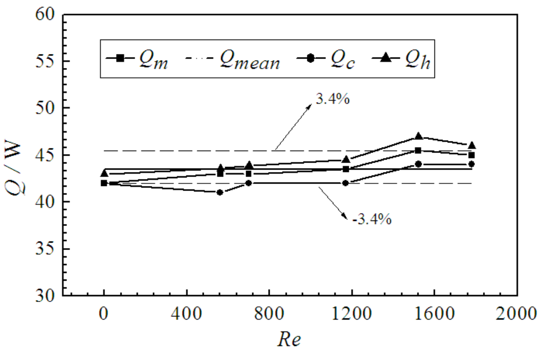
2.2 Verification of FSP deduction 2

The FSP deduction 2 says: when fluid flow velocity is normal to fluid temperature gradient no matter how large the velocity is it does not make any contribution to the convective heat transfer. This was done by the present authors' group [33]. The major resultss are presented in Fig. 4. Figure

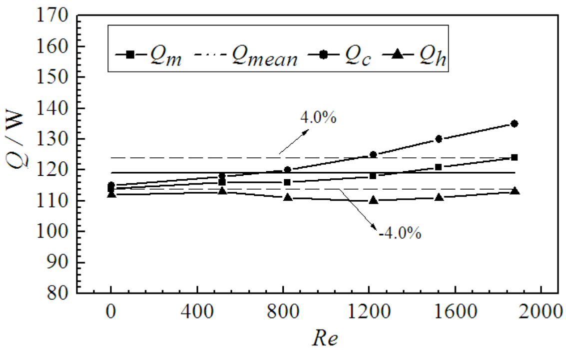
4(a) is the cross section of the test wind tunnel. The top and bottom are adiabatic and the right and left wall are kept at constant but different temperature, forming a situation where fluid temperature gradient is perpendicular to the main flow direction. The results show that the heat transfer rate is only depends on the temperature difference of the two walls, and the flow rate does not have its effect.



(a) Cross-sectional view of the test duct



(b) $\Delta T = 10^{\circ}$C



(c) $\Delta T = 30^{\circ}$ C

Fig. 4 Experimental results of the special test

2.3 Verification of FSP for turbulent heat transfer

Numerical verification of FSP for turbulent heat transfer are presented in [34,35], and will not be presented in this paper. It is the present authors' understanding that as long as the governing equation for energy is expressed by the diffusion-convection type within which the convection part includes the dot production of temperature gradient and velocity, no matter whether the flow is parabolic or elliptic, transient or steady, laminar or turbulent, the field synergy principle is always applicable.

3. Contributions of FSP to the development of convective heat transfer theory

3.1 FSP revealing the condition for velocity to play a role in convective heat transfer

Convective heat transfer has long been understood as heat transfer between a moving fluid and a solid wall, and the faster the velocity ,the greater the convective heat transfer . Before the proposal of the FSP, no one concerned the velocity direction relative to the fluid temperature gradient. The field synergy principle indicates that for fluid velocity to have some role to play it must be not perpendicular to fluid temperature gradient. This gives a very important guidance to enhance convective heat transfer with less penalty in fluid pressure drop or pumping power.

3.2 FSP revealing the upper limit of exponent m in correlation of $Nu \sim \mathrm{Re}^m$

It has been demonstrated above that the upper limit of m in the correlation of $Nu \sim \mathrm{Re}^m$ is equal to 1 when velocity direction fully coincides with temperature gradient, either at the same direction (fluid heated) or in the opposite direction (fluid cooled).

The first and second contributions of the FSP are very simple, yet are very important, and they provide us very useful tool to judge whether some newly proposed correlations or inventions are correct (feasible) or not. In the authors' opinion the first and second contributions of FSP may be regarded as a kind of milestone in the development of convective heat transfer theory.

3.3 FSP explaining fundamental reasons of characteristics for some basic and enhanced heat transfer cases

3.3.1 Laminar fully developed heat transfer in tube: $Nu_q > Nu_T$

In almost all heat transfer books it is presented that for the laminar fully developed heat transfer in circular tube the Nusselt number of constant heat flux boundary condition is $Nu_q = 4.36$, while that of constant wall temperature $Nu_T = 3.66$, about 16 % less than Nu_q. It turns out that the local synergy between velocity and temperature gradient near the wall of the constant heat flux case is better than that of constant wall temperature case [36].

3.3.2 Very high heat transfer coefficient at stagnation point of impinging jet

Impinging jet is a well known enhancing technique for cooling or heating and widely used in engineering. The local heat transfer coefficient at the stagnation point is especially high, because at this point velocity direction just coincides with fluid temperature gradient: the same if fluid is heated or the opposite if fluid is cooled.

3.3.3 Role of fins

It has been shown by the present authors that the roles of fin are not only increasing the heat transfer surface but also greatly improving the synergy between fluid velocity and the temperature gradient [37].

3.3.4 Heat transfer characteristics of flow across tube banks

He and her co-workers [38] numerically simulated heat transfer and pressure drop characteristics of plate fin-and-tube banks with tube number from 2 to 4 for laminar flow situation . Among the effects of five parameters studied they found that the spanwise pitch effect is the strongest. At fixed oncoming flow velocity the variations of average Nusselt number and synergy angle with S_1(spanwise tube pitch) and S_2 (longitudinal tube pitch) for two-row case are presented in Fig. 5. It can be observed that the variation trend of Nusselt number with S_1 and S_2 are fully consistent from view point of the FSP.

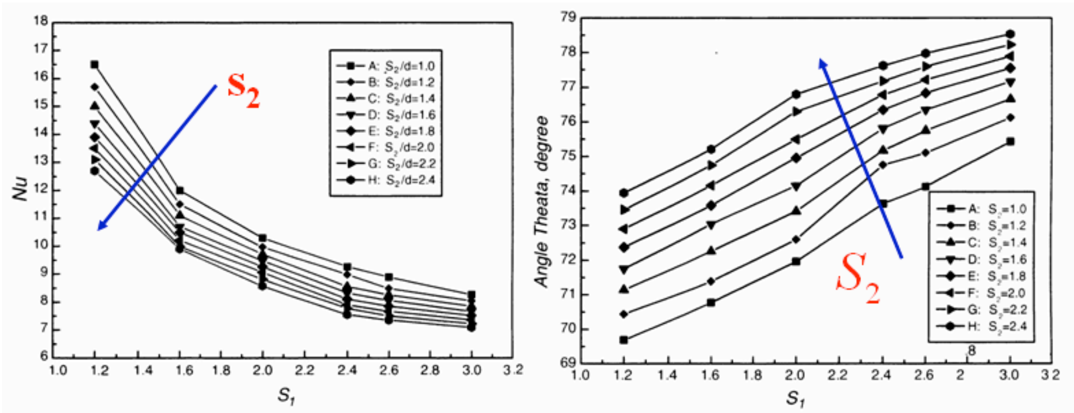
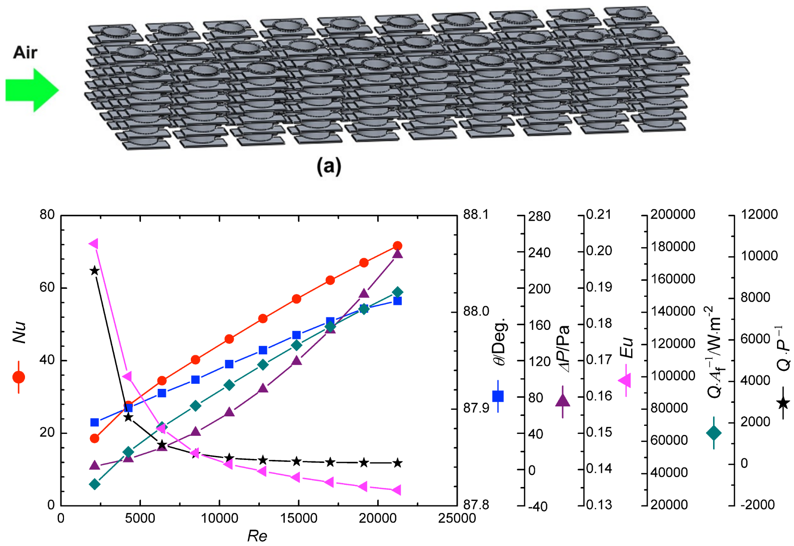


Fig. 5 Heat transfer characteristics across two-row fined tube bank-- Variation of Nuseelt number and synergy angle with S_1 and S_2

3.3.5 Heat transfer characteristics of flow across tube bank with H-type fins

In recent years, H-type finned tubes (Figs. 8(a)) are widely used in boiler economizer and heat exchanger of waste heat recovery. H-type finned tube is derived basically from the rectangle-type finned tube. It has excellent anti-wear and anti-fouling performance because of its unique groove structure in fin surface, i.e., some heat transfer areas of the fin in the separation zone are removed to reduce the negative effect on heat transfer. This is of great importance for the application in the heat exchangers of waste heat recovery. In [39] for a tube bank of ten rows a systematic numerical study has been conducted for the effects of geometric parameters on heat transfer and pressure drop characteristics of H-type finned tube. The effects of eight parameters are examined: number of tube rows, fin thickness, slit width, fin height, fin pitch, spanwise tube pitch, longitudinal tube pitch and Reynolds number on heat transfer and fluid flow characteristics.

Fig.8(b) shows the variations of Nusselt number, intersection angle, pressure drop, Euler number, heat transfer rate per unit frontal area and heat transfer rate per pumping power with *Re*. The domain average intersection angle θ increases with the increase of *Re*, which implies the deterioration of the synergy between the temperature gradient and velocity with *Re*. Even though the variation of the synergy angle is only about 0.3 degree, for an angle as large as 88 degree, a minor change in the angle will result in an appreciable difference in cosine. It is the deterioration of synergy with the increase in Reynolds number that make the increase in Nusselt number not linearly proportional to the increase in Reynolds number.



(d) Effects of *Re* number on Nusselt number, intersection angle, pressure drop, Euler number, heat transfer rate per unit frontal area(QA_f^{-1}) and heat transfer rate per pumping power QP^{-1})

Fig. 6 Heat transfer characteristics across H-type finned tube bank

3.3.6 Heat transfer characteristics of flow across vortex generators

The longitudinal vortex generator (LVG) has been a well-known enhancing technique since 1990s [40-42]. The heat transfer enhancement by the LVGs is usually explained as that the generated longitudinal vortices disturb, swirl and mix the fluid flow, break the boundary layer developing and make it thinner. Wu and Tao [17,18] conducted both numerical and experimental studies to reveal the fundamental mechanism of heat transfer enhancement by LVGs. Their results show that the increase of heat transfer enhancement is always companied by the decrease of field synergy angle between the velocity and temperature gradient when any parameter of LVG is changed. This confirms that the improvement in field synergy is the fundamental mechanism of heat transfer enhancement by longitudinal vortex.

3.3.7 The role of nano-particles in heat transfer enhancement

Enhancement of heat transfer with nanofluids is usually attributed to enhancement of effective thermal conductivity due to presence of nano-particles. In addition random motion and dispersion effect of ultrafine nano-particles are also considered in favor of enhancing heat transfer of nanofluids. Actually all these ingredients lead to the improvement of synergy between velocity and fluid temperature gradient, as demonstrated by Bhattacharya et al. [45]. They conducted 3D numerical study of conjugate heat transfer in rectangular microchannel heat sink with Al_2O_3/H_2O nanofluid, and found that at a Reynolds number of 250, volume averaged value of $\cos(\theta)$ increases by 22.4% (from 0.00567 to 0.00694) as nanoparticle concentration changes from 0 vol% (pure water) to 2 vol%. Thus it can be inferred that use of nanofluid leads to a better synergy between velocity and temperature gradient vector.

3.4 FSP guiding the developments of enhancing techniques with high efficiency

3.4.1 Design of slotted fin surface with ‘front sparse and rear dense’ rule.

As indicated above, plate fin-and-tube heat exchangers are widely used in various engineering fields, and many types have been developed, among which the slotted fin surface is an efficient one. Slots are uniformly distributed in the mainflow direction. However，a more appropriate arrangement of slots along flow direction should be from sparse to dense as shown by Pattern 2 in Fig.7. Cheng et al.[46] made three designs of the slots arrangement along flow direction for a plate fin of a three-row tube-and-fin heat exchanger with different degrees of front sparse and rear dense as shown in Fig. 8. The simulated results for laminar heat transfer show that the first design, i.e., Slit 1 , has the best synergy and highest Nusselt number for both the identical pressure drop and identical pumping power constraints. For the identical pressure drop case, the results are presented in Fig. 9 as an example. This new type of slotted fin was adopted by a heat exchanger factory in China, and two heat exchanger with 12 tube-rows of plain plate fin and the new type slotted fins were manufactured. Compared with the heat exchanger with plain plate fin surface, in the entire range of air velocity the increases in the overall heat transfer coefficient of the slotted fin heat exchanger is lager than the increase in air pressure drop. At the air velocity of 5-6m/s, the enhancement of overall heat transfer coefficient is about 26%, while the pressure drop penalty is only about 22%.

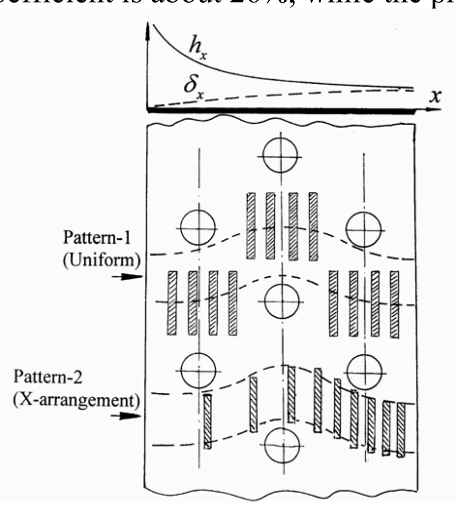
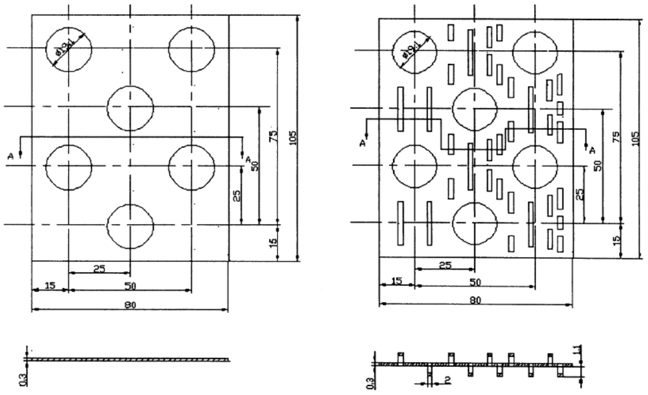
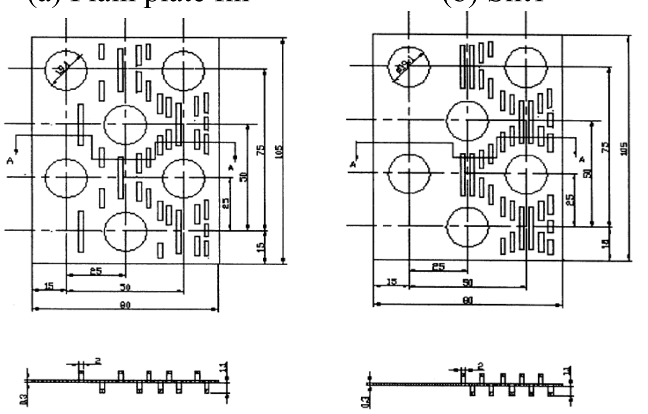


Fig. 7 Arrangements of slots

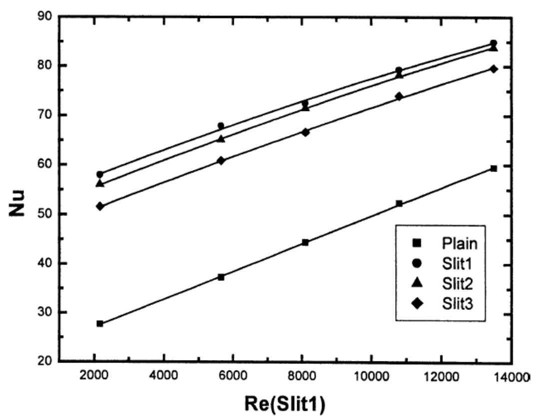


(a) Plain plate fin (b) Slit1

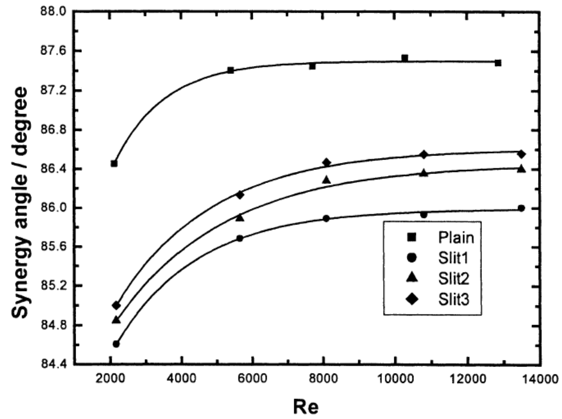


(c) Slit 2 (d) Slit 3

Fig. 8 Three designs of slots for plate fin surface



(a) Nu vs. Re



(b) Synergy angle vs. Re

Fig. 9 Numerical results of Nusselt number and synergy angles of the three designs of slots for plate fin surface

3.4.2 Design of an alternating elliptical axis tube (AEAT)

See [47] for details.

3.4.3 Design of plain fin with radiantly arranged winglets around each tube

See [48] for details.

3.4.4 Improvement of bipolar channel for PEMFC

See [49-52] for details.

4. Performance evaluation of enhanced structures

Any enhancement technique will introduce additional fluid pressure drop, therefore thermo-hydraulic performance comparison is necessary to determine whether an enhanced technique is worthy of application. Plot is a convenient way for a visual presentation of comparison results. However, none of the existing comparison plots can simultaneously indicate whether the technique can enhance heat transfer under the three constraints. In this section a plot method proposed by Fan et al. [43] which can simultaneously express the performance of an enhanced technique evaluated from the three constraints will be presented.

4.1 A unified log-log plot for performance evaluation

The new performance evaluation plot takes the two ratios, $(f_e / f_0)_{\mathrm{Re}}$ and $(Nu_e / Nu_0)_{\mathrm{Re}}$ (or $(j_e / j_0)_{\mathrm{Re}}$, as the two coordinates and the thermo-hydraulic performance of the enhanced surface can be expressed by a point on the plot.

The new performance evaluation plot (NPEP) for heat transfer enhancement techniques oriented for energy-saving is presented in Figure 10.

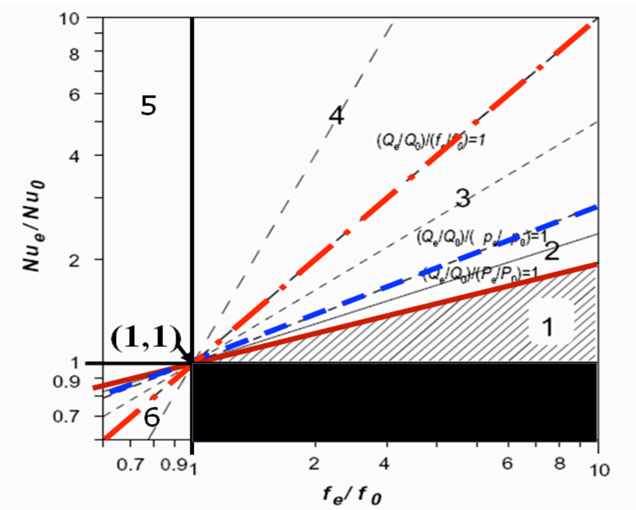
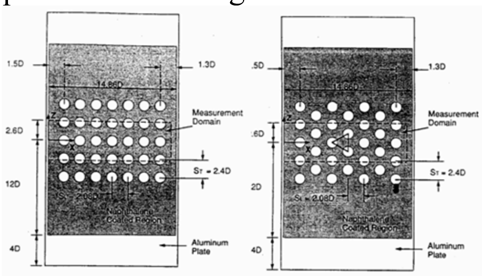


Fig. 10 Composition of the new performance evaluation plot

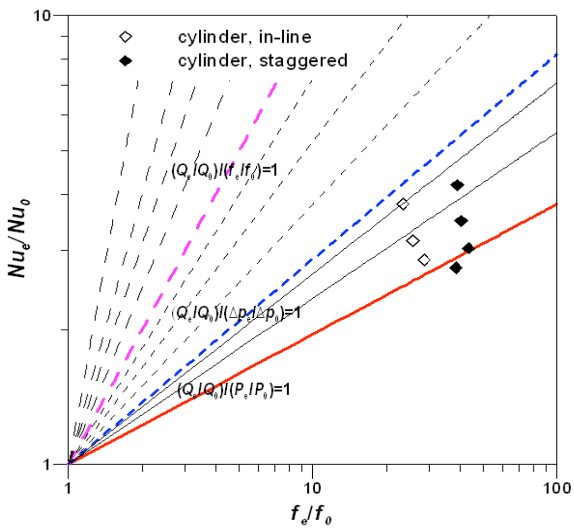
The abscissa of the figure presents the friction coefficient ratio between the enhanced and referenced surfaces under the same Reynolds numbers, and the ordinate presents the Nusselt number (or j-factor, or Stanton number) ratio between the enhanced and referenced surfaces under the same Reynolds numbers. The composition of this NPEP can be described as follows. The plot divides the whole plane into four quadrants with (1,1) as its origin. In quadrant I both ratios are larger than one, implying that heat transfer is enhanced and pressure drop is increased, which is the most frequently encountered situation in heat transfer enhancement study; In quadrant II heat transfer is enhanced while pressure drop is decreased, a very rarely happen situation in practical engineering; In quadrant III both heat transfer and pressure drop are decreased which sometimes happens in engineering practice as the shell side heat transfer and pressure drop of helical baffle compared with those of segmental baffles; finally in quadrant Ⅳ heat transfer is deteriorated while pressure drop is increased, and structures characterized by any point in this quadrant should not be adopted. Any point in the log-log picture is called working point. Our focus is put on quadrant I. It can be divided into four regions indicated by the four Arabic numbers in boldface in Figure 10. Any working point in Region 1 is characterized by enhanced heat transfer without energy-saving, that is the heat transfer enhancement ratio is less than the increase of pumping power consumption. Region 2 is featured by enhanced heat transfer with the same pump power consumption, i.e., where at identical pumping power the enhanced surface presents higher heat transfer rate than referenced one. In Region 3 the enhanced surface presents higher heat transfer rate than the referenced one under the identical pressure drop constraint. Finally Region 4 is the most favorable one where the augmentation of heat transfer rate is larger than the increase of friction coefficient under the same flow rate. The boundary line of Regions 1 and 2 is the dividing line of heat transfer enhancement or deterioration compared with the referenced ones based on the identical pumping power consumption. The boundary line of Regions 2 and 3 is the dividing line of heat transfer enhancement and deterioration compared with the referenced ones based on the identical pressure drop. The boundary line of Regions 3 and 4 is the dividing line judging the level of the ratio of increase of heat transfer rate and friction coefficient compared with the referenced ones under the same fluid flow rate.

4.2. Application examples

Short cylinders are widely used to enhance heat transfer in a rectangle channel with larger aspect ratio (ratio of channel width over channel height).. For the seven configurations reported in [54], their performances are compared in Figure 11. It can be seen that compared with smooth duct such enhanced techniques can enhancing heat constraint.



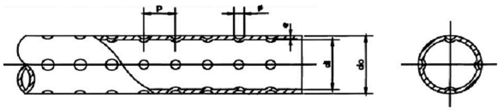
(a) Short cylinder enhancement



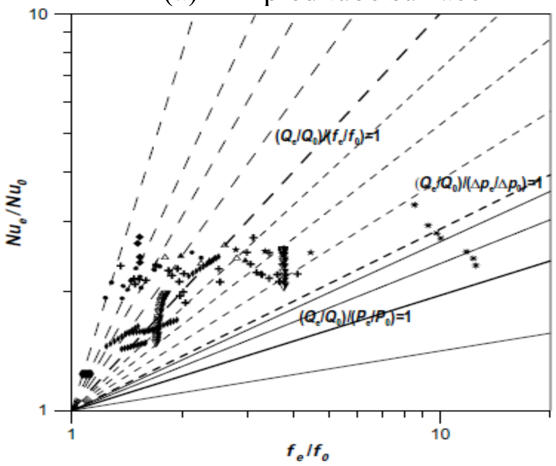
(b) Comparison in NPEP

Figure 11 Examples of enhancing technique under identical pumping power

In the past two decades, a new type of enhanced surface, called dimpledsurfaces, were widely studied [58-63]. Dimples are arrays of indentations arranged along flow direction. These are usually spherical in shape. Figures. 12(a) shows dimples for a tube. Because dimples do not protrude into the flow too much, hence the introduced form drag is not significant. The vortex structure in the dimples enhances local and downstream heat transfer with a minor pressure drop penalty. Figure 12(b) presents the performance comparison of different dimpled surface of turbulent flow inside cooling channel .From the figure it can be seen that working points of these dimpled surfaces mainly located in Regions 3 and 4, and only five working points in Region 2, implying that most of the dimpled surfaces possess high effectiveness .



(a) Dimpled tube surface



(b) Dimpled surface performance comparison

Figure 12 Examples of enhancing technique under identical flow rate

A comprehensive comparison study of techniques for enhancing air side heat transfer was conducted in [63] oriented for the application of solar receiver. Twenty seven enhanced structures were compared based on the correlations of heat transfer and friction factors. Then the NPEP was used for their thermo-hydraulic performance comparison. From their comparison, it can be concluded that under the identical flow rate constraint, arc shaped ribs has the highest heat transfer enhancement ratio, followed by U-shaped turbulators and inclined ribs with gap. Under the identical pumping power constraint wire screen mesh packing with porosity 0.937 performs best. While for the multipass flat channel the enhancement of heat transfer rate is much less than the increase of pumping power consumption, thus from energy-saving view point, it performs the worst.

5.Conclusions

In this lecture a comprehensive review on the field synergy principle of enhancing single phase convective heat transfer has been conducted , a great number of enhancing techniques have been analyzed from the view point of the FSP, the new performance evaluation plot has been introduced and the thermo-hydraulic performance of a number of enhancing techniques has been evaluated and compared by using this NPEP. Following conclusions can be obtained.

1. The basic mechanism of enhancing single phase convective heat transfer, whether laminar or turbulent, transient or stead, elliptic or parabolic, is to make a better synergy between velocity and fluid temperature gradient, i.e., to reduce the synergy angle between velocity and

temperature gradient. All the existing explanations for enhancing single phase convective heat transfer can be unified by the FSP.

2. The best situation of synergy is that the local velocity and temperature gradient coincide everywhere in the flow field. If fluid velocity and the temperature gradient are in the same direction, fluid is heated along the flow direction, otherwise it is cooled. For this case the heat transfer rate increases linearly with fluid velocity and the exponent m in the correlation of $Nu \propto \mathrm{Re}^m$ reaches its maximum value of one.
3. The worst situation of synergy occurs when fluid velocity is perpendicular to the fluid temperature gradient. For this case no matter how large is the fluid velocity it does not make any contribution to convective heat transfer. When fluid velocity is used to enhance convective heat transfer, not only its magnitude but also its direction should be taken into account.
4. For fluid convective heat transfer over an existing surface structure revealing the locations where the synergy is bad and improving it is an efficient way for improving the existing or developing a new enhancing techniques. Since the synergy angle can not be measured directly ,numerical simulation plays an important role in this regard.
5. To judge the synergy of entire fluid domain, the average synergy angle may be used. Numerical practices show that except the simple arithmetic mean the variation trends of the average synergy angle of different methods are the same and can be used to make qualitative analysis.
6. By using FSP the characters of a number of typical convective heat transfer cases can be well-explained and understood. It is found that the roles of fins are not only extending the cheat transfer surface but also improving the synergy between velocity and temperature gradient.
7. For thermo-hydraulic performance comparison of enhancing techniques for single phase heat transfer, the three comparison constraints, i.e., the identical pumping power, identical pressure drop and identical fluid flow rate, can be expressed in a unified plot with log $(f_e / f_o)$and log (Nu_e / Nu_o) as abscissa and ordinate, respectively. Thermo-hydraulic performance comparison of an enhancing technique with respect to the reference structure can be conducted by identify the location of its working point in the picture with the two ratios of f_e / f_o and Nu_e / Nu_o being obtained either by experimental measurement of by numerical simulation.
8. In this picture the first quadrant can be divided into four regions. When a working point located in the first region under the identical pumping power constraint heat transfer is deteriorated; Working point in the second and third regions is characterized by enhanced heat transfer under the constraint of identical pumping power and identical pressure drop, respectively. Working point in the fourth region represents a technique which can have a larger Nu_e / Nu_o ratio than the ratio of f_e / f_o under the same flow rate, which is quite difficult to realize.
9. This NPEP is most suitable for comparisons of a number of techniques with respect to the same reference structure. Under the same ratio of f_e / f_o the higher the ratio of Nu_e / Nu_o the better the technique; For the same ratio of Nu_e / Nu_o, the less the ratio of f_e / f_o the better the technique. In addition the position of the working point can clearly indicates under what condition the technique can enhance heat transfer. The technique with working point in the fourth region is the best, that in the first region is the worst, techniques with working points in the third region and second region, respectively, are in between.

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