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Review

A comprehensive review and comparison on heatline concept and field synergy principle

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ABSTRACT

A comprehensive review and comparison on heatline concept and field synergy principle have been made based on more than two hundreds of related publications. The major conclusions are as follows. Both heatline concept and field synergy principle are important contributions to the developments of single-phase convective heat transfer theories. The role and function of heat line concept is to visualize the heat transfer path while that of field synergy principle is to reveal the fundamental mechanism of heat transfer enhancement and to guide the development of enhanced structures. None of them can be used to deduce the other, nor none of them can be derived from the other. Hence, there is no problem of mutual remake between them at all. If heatlines are constructed by solving a Poisson equation additional computational work should be done; However, either the synergy number or the synergy angle both can be obtained by using numerical results without additional computational work. Further research needs for both heatline concept and field synergy principle are also provided.

Contents

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Nomenclature

Latin symbols

c_p	specific heat
E_x,E_y,E_z	component of energy vector
Fc	field synergy number
FSP	field synergy principle
grad	gradient
H	heatline function
i,j	indices in x,y direction
J_x,J_y	total (conduction and convection) heat flux in x and y direction
k	thermal conductivity
Nu	Nusselt number
Pr	Prandtle number
Re	Reynolds number
T	temperature
$\overrightarrow{U}$	velocity vector
u, v	velocity component in x,y direction
V	volume
x,y	coordinates

Greek symbols

∇	gradient
δ_t	thickness of thermal boundary layer
θ	synergy angle
ρ	fluid density
ψ	stream function

Subscript

m	average
i	local

1. Introduction

In 2015 Bejan published a paper concerning the concept of heatline and the field synergy principle (FSP) [1]. In the abstract of his paper, following sentences were written: "Both concepts, heatlines and synergy, are about visualizing the physics of convection, which is the combination (superposition) of heat conduction lines and enthalpy flow lines over a material in motion. Heatlines and synergy are reviewed here comparatively. This comparison reveals that synergy is a remake of heatlines, and that synergy has no physical connection with heat transfer enhancement." Basak et al. [2] repeated these conclusions and further ask such a question to FSP as "the designer does not know how to access the field synergy angle for better heat transfer … Finally, it is concluded that, synergy is a repetition of the heatlines". Recently Bejan [3] further insisted "that synergy is a 1998 remake of the existing (1983) concept and method of heatfunction and heatlines. It is not original."

Here four important questions have been raised:

(1) Are heatlines concept and synergy principle both about visualizing the physics of convection heat transfer?
(2) Is synergy a remake of heatline?
(3) Does synergy have no physical connection with heat transfer enhancement?
(4) How to access the field synergy angle for improving heat transfer?

To the authors understanding, both heatline concept and field synergy principle are important developments in the theory of convective heat transfer. The above four questions are worth paying our attentions and further discussing. Even though a quite comprehensive review on the heatline concept and its applications has already been made in [2], the authors' focus is mainly on the transport process in enclosure; even though extension of the heatline concept for visualizing heat transfer route to 3D cases has been presented in literatures but this is not mentioned in [2]. Ref. [2] repeated the comments made in [1] to the FSP without citing Guo's responses [4]. which, to the authors knowledge, are not scientific, hence not fair. Furthermore, Ref. [3] further insisted the author's opinion. It is thus strongly required that a more general and comprehensive review and comparison for the heatline concept and the FSP should be conducted in order to promote the healthy atmosphere of free discussion between authors with different opinions in the international heat transfer community and to promote further development of convective heat transfer theory. We thus made a comprehensive review of the publications for the both subjects since they were proposed, and more than two hundred related technical journal papers were carefully read for the heatlines concept and the FSP. This comprehensive review paper is the outcome of our comparative study on the two subjects.

Because of large number of published papers related to the heatline concept it is impossible to introduce each paper individually. After careful examination of all papers known to the present authors we found that we can classify them according to the problem categories. We will first show what kind of different category problems have been studied and give some representative useful results related to heatline concept. As for the field synergy principle after a brief introduction focus will be put on its roles on developing the convective heat transfer theory and examples for guiding development of heat transfer enhancement techniques. For each subject discussion on further research needs will be presented.

The composition of the paper is presented above and will not be restated here for simplicity. Apart from detailed comparison between the heatline concept and the FSP, an another purpose of this paper is to discuss with Bejan and his co-workers through the large number of examples cited in this paper that we can come to the same understanding on heatlines and FSP. We sincerely welcome any practical comments to this paper. We are sure that only through such practical and dispassionate discussion any academic dispute could be solved.

2. Brief introduction to heatline concept

In 1983 Kimura and Bejan published a technical note in ASME Journal of Heat Transfer titled by "The heatline visualization of convective heat" [5]. In this paper the heat line concept was first proposed and defined. By mimicking the following definition of stream function in fluid mechanics of Cartesian coordinate:

$$u = \partial\psi/\partial y;\ v = -\partial\psi/\partial x \tag{1}$$

which automatically satisfies two-dimensional mass conservation requirement, they defined heat line function, denoted by H, as follows:

$$J_x = \frac{\partial H}{\partial y} = \rho u c_P T - k\frac{\partial T}{\partial x}; \tag{2a}$$

$$J_y = \frac{\partial H}{\partial x} = \rho v c_P T - k\frac{\partial T}{\partial y} \tag{2b}$$

where J_x,J_y are the total (conduction and convection) heat flux in x and y direction, respectively. Obviously,

$$\frac{\partial J_x}{\partial x} + \frac{\partial J_y}{\partial y} = \frac{\partial^2 H}{\partial x \partial y} - \frac{\partial^2 H}{\partial y \partial x} = 0 \tag{3}$$

This implies that by definition of Eq. (2) the energy conservation for 2-D steady situation is automatically satisfied. In order to get the value of H at different location within the computational domain, following Poisson equation should be solved:

$$\frac{\partial^2 H}{\partial x^2} + \frac{\partial^2 H}{\partial y^2} = \rho c_P\left[\frac{\partial}{\partial y}(uT) - \frac{\partial}{\partial x}(vT)\right] \tag{4}$$

For any 2D practical convective heat transfer problem once numerical solutions are obtained, the right side source term of Eq. (4) can be computed, hence, the Poisson equation for H can be solved numerically with ease.

The contours of H represent the route, along which heat energy transfers within 2-dimensional body. Those contours are different from isotherms, which can show heat transfer direction only for conduction problems. Therefore H function can vividly show the convective heat transfer direction, i.e., the visualization of convective heat transfer process. That is why so many published papers about heatline adopt the word "visualization", i.e. visualization of convective heat transfer, as adopted by Bejan himself in [5].

In 1987 Trevisa and Bejan extended the heatline concept to mass transfer and proposed the concept of massline [6]. The basic concept is the same as heatline and will not be further discussed in this paper.

3. Applications of heatline concept

Since the proposal of the heatline concept, it has been widely adopted for visualizing convective heat transfer for different problems. A search of literature by Web of Science obtains paper publication information shown in Fig. 1. In order to make an objective and impartial evaluation on the role and contribution of heatline concept to heat transfer the present authors read almost all the journal published papers about heatline. We found that in the past three decades this concept has been successfully extended to following aspects of convective heat transfer, for example:

(1) 2D cylindrical coordinates [7–9] and polar coordinates [10,11];
(2) unsteady 2D problems [8,12];
(3) turbulent heat transfer [13];
(4) convective heat transfer in porous media [14];
(5) heat transfer in anisotropic media [15];
(6) reacting flow and heat transfer [16,17].

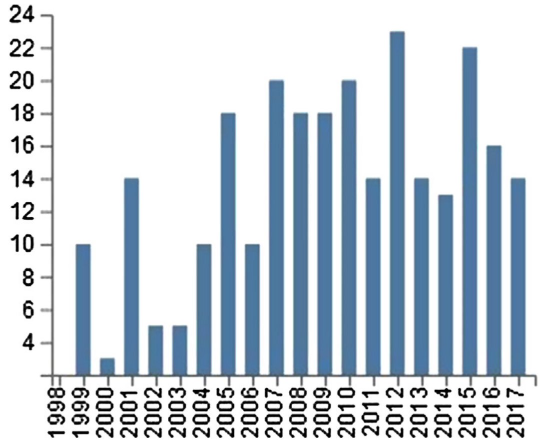


Fig. 1. Paper publication information related to heat line concept.

In the following typical categories of convective heat transfer problems studied by the heatline concept will be presented and some useful results will be provided.

3.1. Two-dimensional natural convection in different kinds of cavities

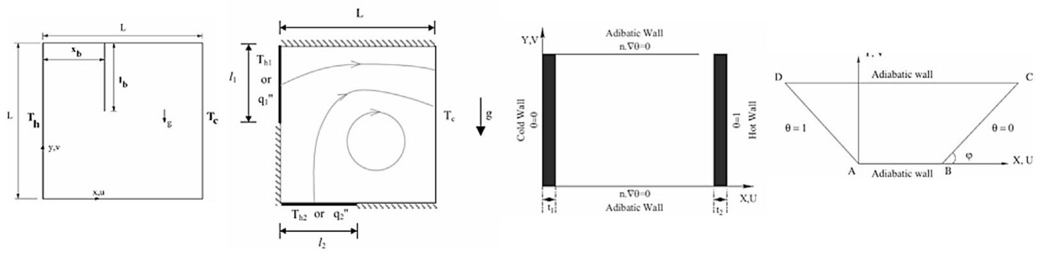
This category of heat transfer problems has been studied by using heatline concept very thoroughly. Fig. 2 presents the geometries, fluids that have been studied and the related Refs. [18–72]. As much as 34 different types are included, and each group may have different cases. In Fig. 3 the results of streamlines, isotherms and heat lines are presented for natural convection in a 2D square enclosure with two thick vertical walls [30]. Apart from the streamlines and isotherms, the heat lines vividly show how the heat is transferred from the right hot wall to the cold right wall through the air. And it also can be clearly observed that with the increase in *Ra* number the vortex region increases where fluid flow constitutes a closed cycle but no any contribution is made for the transfer of heat from left to right wall.

3.2. Natural convection in 2D enclosure with porous medium

A large number of different cases of enclosures (cavities) with porous medium have been studied by using the heatline concept to visualize heat transfer. In Fig. 4 they are grouped by sixteen types [73–91]. In each type there are some minor different either in given condition or in partial geometries. In Fig. 5 the isotherms (Fig. 5(a)), streamlines (Fig. 5(b)) and heatlines (Fig. 5(c)) in entrapped porous triangular cavities are presented [81]. How the heat transfers from the lateral walls to the top and bottom walls are clearly indicated in Fig. 5(c).

3.3. Natural convection in 2D enclosure with open inlet and outlet

In Fig. 6(a) different types of natural convection in 2D enclosure with open inlet and outlet, which have been investigated by heatline concept are collected [92–95]. The heatlines of Problem type 3 are provided in Fig. 6(b), where it can be clearly seen how the heat transfer routes change with Grashof number.

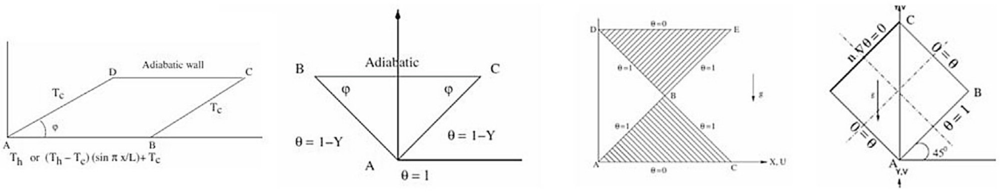


(1) Square cavity with dividing plate [18,19]

(2) Square cavity with discrete wall heat sources [20-26]

(3) Conjugated square cavity [27-31]

(4) Trapezoidal cavities [32-38]

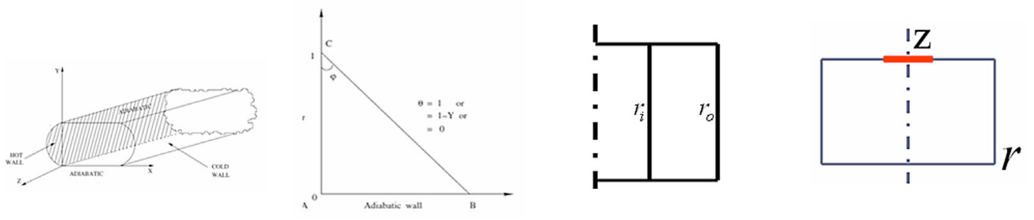


(5) Rhombic cavity [39,40]

(6) Equilateral triangle cavity [41,42]

(7) Two entrapped triangle cavities [43-45]

(8) Tilted square cavity [46-48]

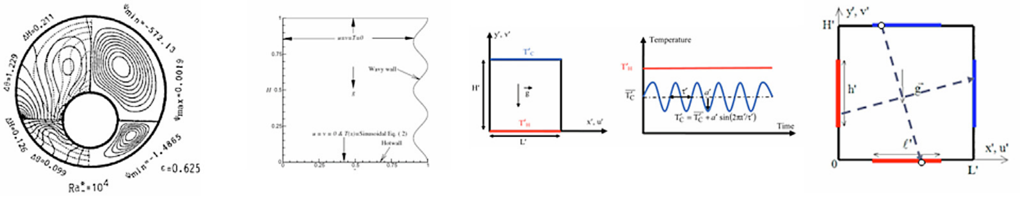


(9) Enclosure with convex/concave side wall [49]

(10) Triangular enclosure [50,51]

(11) Vertical cylindrical annulus [7,9]

(12) Vertical cylindrical enclosure [8]



(13) Horizontal annulus [10]

(14) Complicated enclosure [52]

(15) Enclosure periodically cooled from above [53]

(16) Enclosure with surface radiation [54,55]

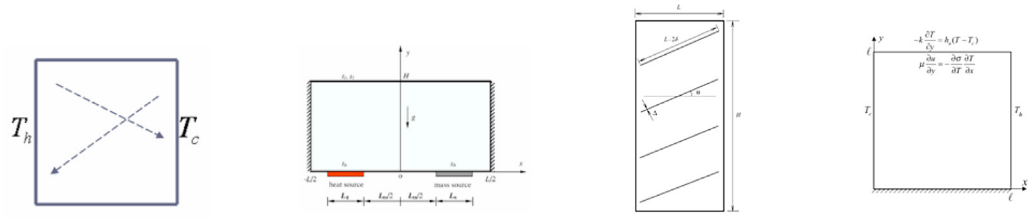
Fig. 2. Natural convection in 2D enclosure with different geometries, boundary condition and fluids.

3.4. Inverse problems of 2D natural convection

The heatline concept was also adopted to study the inverse problems of 2D enclosures, and Fig. 7(a) presents four such kind of problems studied in literature [96–99]. Fig. 7(b) shows the heatlines for the problem type 4. For this case the enclosure right wall is heated and left wall is adiabatic, while the top and bottom walls are cooled. Because of the effect of natural convection the heat transferred to the bottom is basically not directly from right wall to bottom. Rather it goes first round the inner cylinder then to the bottom wall.

3.5. 2D forced and mixed convection

In Fig. 8(a) ten types of forced or mixed convection problems are provided [100–111], among whom the heatline for problem type 7 is shown in Fig. 8(b). It can be observed that above the metal

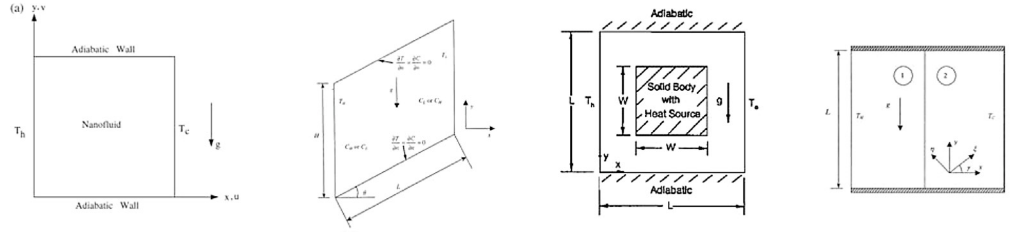


(17) With participating medium [56]

(18) With discrete thermo-solutal sources [57]

(19) A vertical stack of parallelogrammic partial enclosures [58]

(20) Effect of surface tension [59]

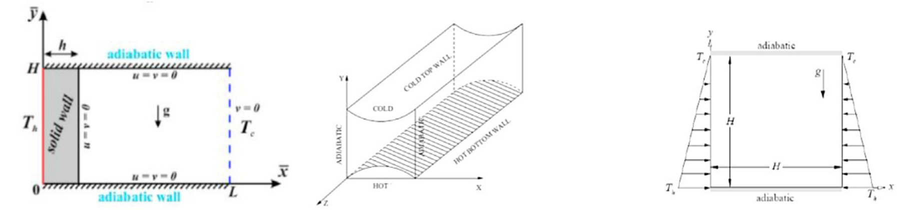


(21) Nanofluids in enclosure [60]

(22) Double diffusive enclosure [61]

(23) With internal isolated heat source [62]

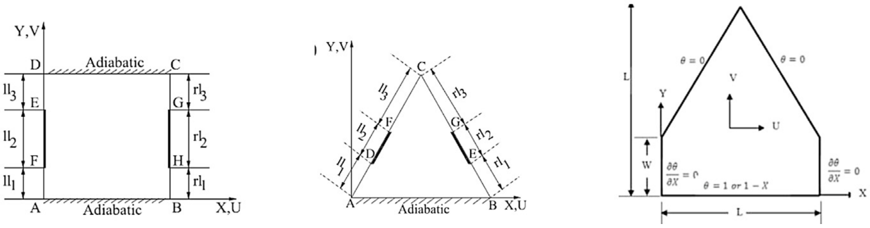
(24) With anisotropic media [63]



(25) Nanofluids in enclosure with a thick wall [64]

(26) Enclosure with convex/concave top walls [65]

(27) Nanofluids in enclosure with sidewall linear heating [66]



（28）Square cavity with discrete heating sources [67]

(29) Triangular cavity with discrete heating sources [67]

(30) Natural Convection inside prismatic enclosure [68]

Fig. 2 (*continued*)

foam there is a heat rout circulation, which does not contribute to heat transfer at all.

3.6. 2D boundary layer problems

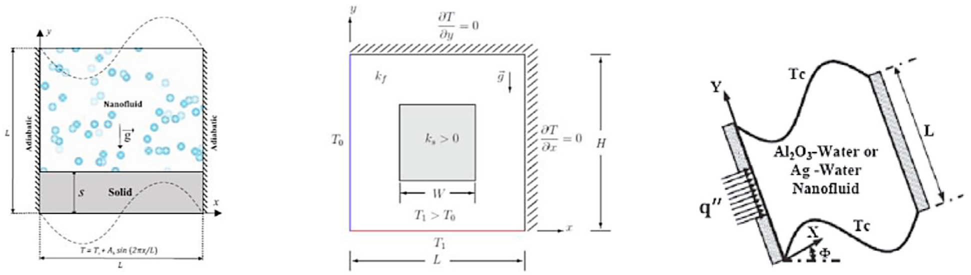
Boundary-layer type convective heat transfer problems were also investigated by heatline concept and Fig. 9(a) illustrated four types of problems investigated [112–114]. How the heat from surrounding hot air transfers to the cold wall for the problem type 3 is very vividly represented by the heatlines in Fig. 9 (b).

3.7. Heat transfer with reaction of electro-magnetic force

In Fig. 10 eight special situations are presented, which are characterized with the existence of either reaction or electro-magnetic force [115–123]. Details in the heatlines can be found in the cited references.

3.8. 2D unsteady problems

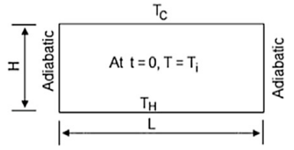
In Refs. [8,12,124] heatline concept was used to visualize 2D unsteady heat transfer for buoyancy-driven flow in a cylindrical enclosure, natural convection in a cylindrical enclosure



(31) Cavity with nanofluid and sinusoidal temperature variations on horizontal walls [69]

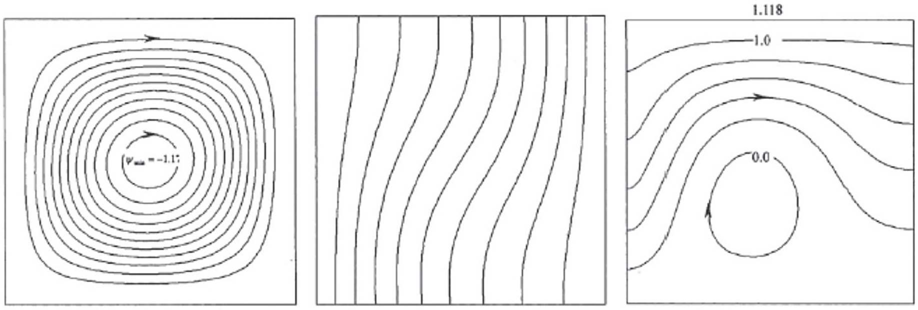
(32) Conjugate heat transfer in enclosure heated and cooled at the adjacent [70]

(33) Wavy cavities with nanofluids and subjected to a discrete heating [71]



(34) Solidification of a eutectic solution in a rectangular cavity [72]

Fig. 2 (*continued*)

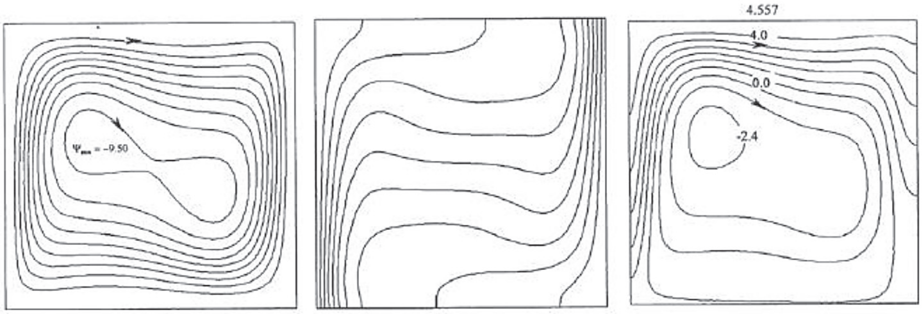


Stream lines

Isotherms

Heatlines

(a) Ra=1000, Pr =0.71



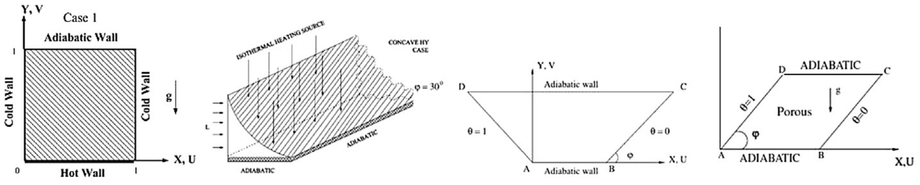
Stream lines

Isotherms

Heatlines

(b) Ra=10000, Pr =0.71

Fig. 3. Heat transfer route visualization of type 3 of Fig. 2.

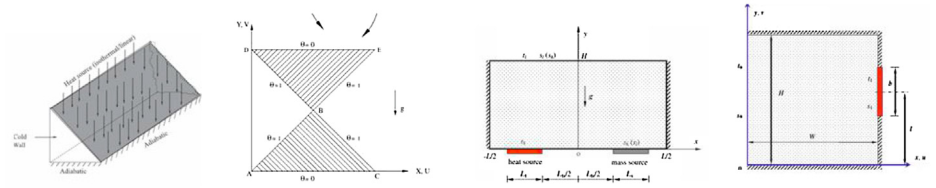


(1) Square enclosure [73-76]

(2) Right angled triangular enclosures with a concave /convex hypotenuse [77]

(3) Trapzoidal enclosures [33,78]

(4) Rhombic enclosures [79]

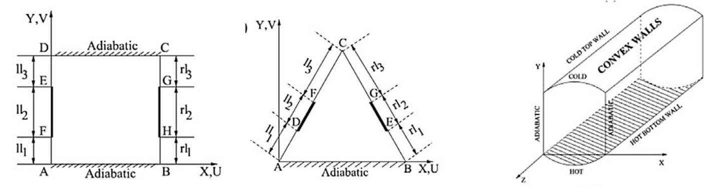


(5) Right angled triangular enclosures [50,51,80]

(6) Entrapped triangular cavities [81]

(7) With localized heating and salting from below [82]

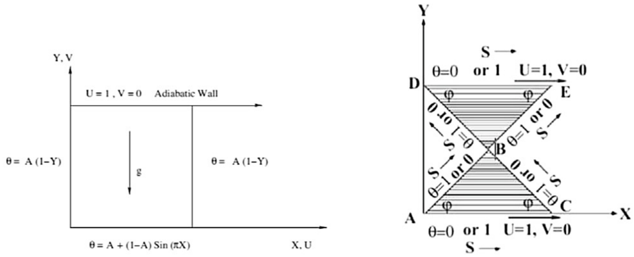
(8) With thermal and solute source [83,84]



(9) Square enclosure with discrete heating elements in the wall [85]

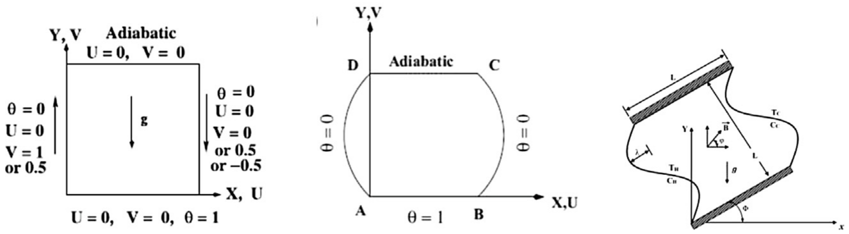
(10) Triangle enclosure with discrete heating elements in the wall [85]

(11) Enclosure with concave and convex top and bottom walls [86]



(12) Mixed convection within porous square cavities [87]

(13) Entrapped porous triangular cavities with moving walls [88]



(14) Porous square enclosure with various moving walls [89]

(15) Role of Forchheimer term porous enclosures of four natural convection in enclosures of four shapes [90]

(16) Double-diffusive MHD natural convection within tilted sinusoidal corrugated porous enclosure [91]

Fig. 4. Natural convection in 2D enclosure with porous medium.

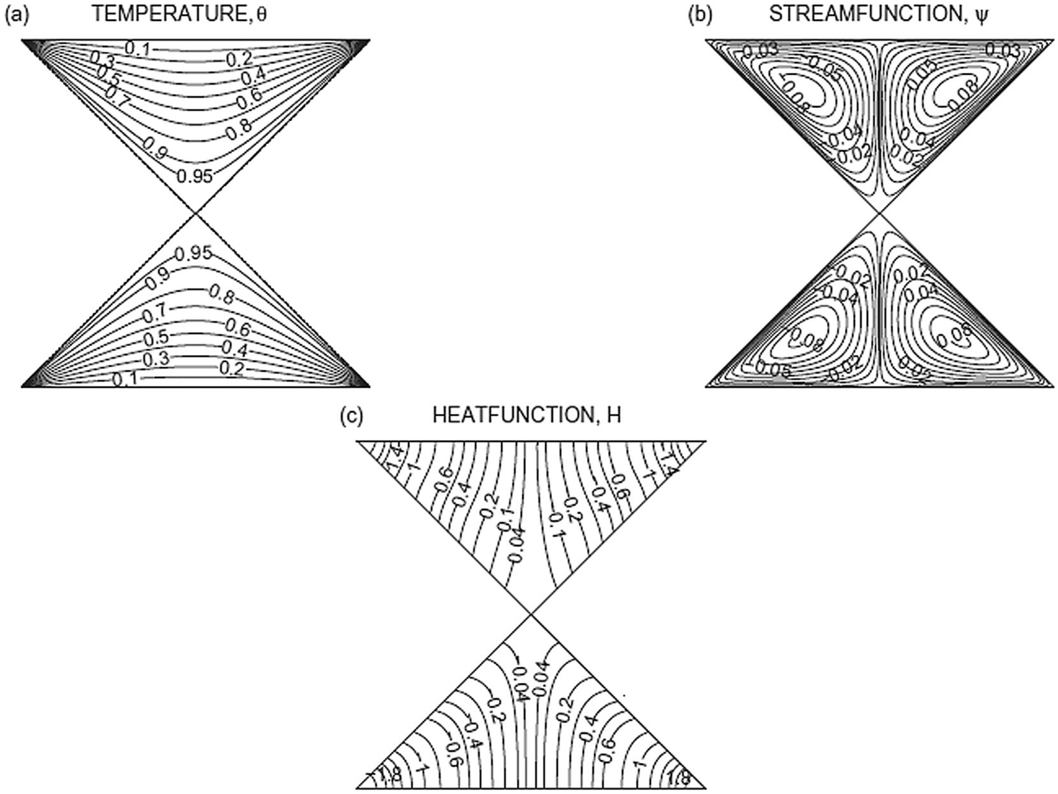
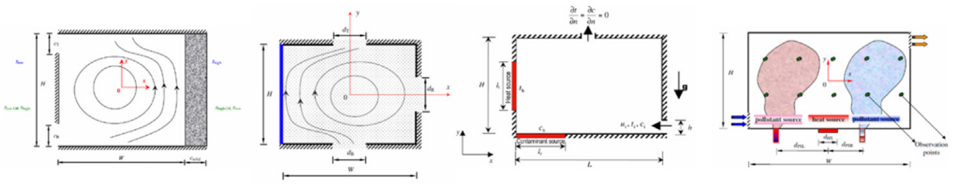


Fig. 5. Heat transfer route visualization of type 6 of Fig. 4.



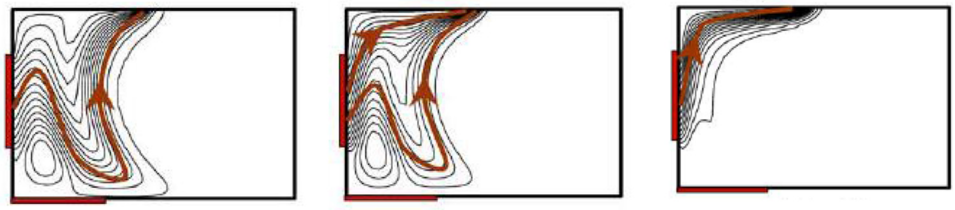
(1) Vented enclosure [92]

(2) Enclosure with multiple outlets [93]

(3) Ventilated enclosure [94]

(4) Ventilated enclosure with in/outlets at opposed walls [95]

(a) Four types of problems



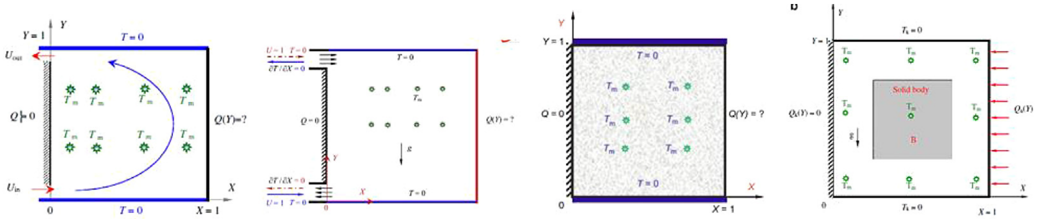
(a) Gr=1.8×10^4

(b) Gr=2.5×10^4

(c) Gr=10^5

(b) Heatlines of problem type 3

Fig. 6. Natural convection in 2D enclosure with open inlet and outlet.



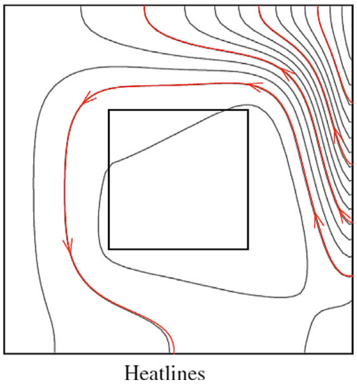
(1) Laminar mixed convections [96]

(2) Turbulent slot-vented enclosure [97]

(3) Square enclosure estimating source profile [98]

(4) Square enclosure with internal body [99]

(a) Four types of problems



(b) Heatlines of problem type 4 for Ra=10^5

Fig. 7. Inverse problems of 2D convection in enclosure.

nonuniformly heated at the top wall and heat transfer in a partially heated open cavity. The graphic presentation of the numerical results of the unsteady heat lines are not so clear, hence it is not copied here. Interested readers may consult above reference.

4. Comments on the heatline concepts

4.1. Role and function of the heat line concept

From above presentation it can be concluded that the heatline concept can clearly show the thermal energy (heat) transfer routs in fluid, which are definitely different from the direction of gradient of fluid isotherms. This should be regarded as an important contribution to the convective heat transfer. As indicated very objectively by Costa in [125] after 23 years of the publication of [5], “The method was invented two decades ago. It has evolved, and is now mature and ready to be used as a systematic analysis tool. The streamline, heatline, and massline methods can be treated through common procedures, which can be easily implemented in CFD packages”.

However, from all the literatures we examined, no paper claimed that heatline concept could be used to guide the study on enhancing convective heat transfer, nor any such example was found. To be objective, the present authors have carefully read the very comprehensive review paper on the heatline [2], where about 40 cases were analyzed by heatlines, and for any case some conclusion words were presented to show the role of the heatline concept. We excerpted several typical words as follows:“Overall, the heatlines presented a clear picture of the heat flow during boundary layer flow involving hot and cold vertical walls”; “Overall, the heatlines are useful to visualize the convective heat flow over a vertical cylinder.” We could not excerpted more because of the limitation of copyright. From the present review and many other review papers on the heatline concept such as [2,125], it can be definitely concluded that the role and function of heatline is to visualize the heat (or energy) transfer route.

4.2. Do we need to solve the partial differential equation of heatline function

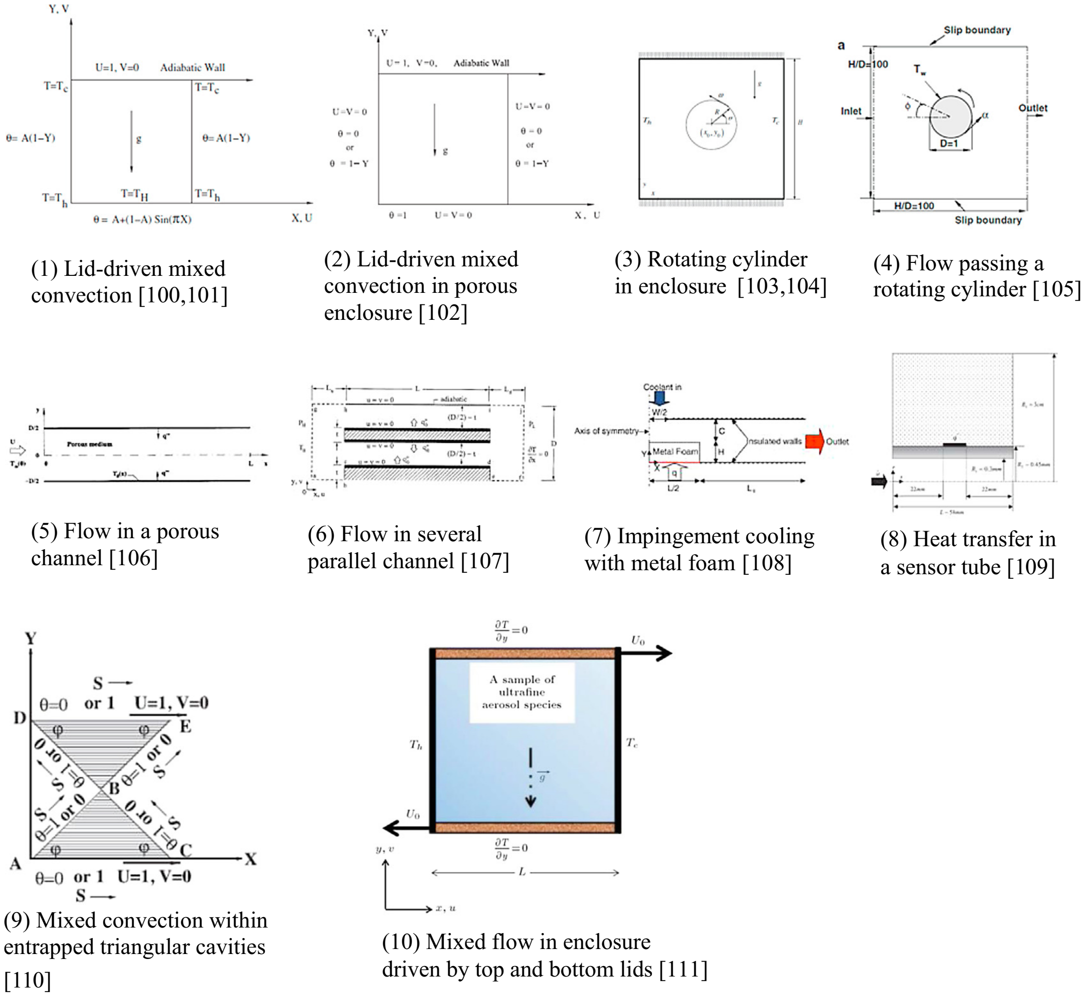
In the pioneer paper of the heatline concept [5] and many other papers, say [2], the heatline function is introduced and a partial differential equation is derived, such as Eq. (4). As already pointed out in 2009 and 2010 [126,127] by Hooman et al., the introduction of heatline function and partial differential equation is not necessary. This introduction has two disadvantages. First it will waste computational time. Because to draw the heatline numerical simulations of fluid flow and heat transfer are necessary, and once numerical solutions of velocity and temperature are available, one can compute the energy vector with ease:

$$E_x = \rho u c_p T - k\frac{\partial T}{\partial x}; E_y = \rho v c_p T - k\frac{\partial T}{\partial y}; \vec{E} = E_x\vec{i} + E_y\vec{j} \quad (5)$$

And the tangent of energy vector is the route of energy transfer, i.e., the heatline.

The most important drawback of introducing heatline function is that heatline function is a remake of stream function, which is only limited to two-dimensional case, thus it is not possible to obtain the counterpart of Eq. (4) in three-dimensional case. This restricts the usage of heatline concept only for the two-dimensional situations. However, it is really can be used for 3D situations as described below.

It is interested to note that from numerical point of view E_x, E_y and E_z are the total heat flux (diffusion plus convection) in x, y and



(1) Lid-driven mixed convection [100,101]

(2) Lid-driven mixed convection in porous enclosure [102]

(3) Rotating cylinder in enclosure [103,104]

(4) Flow passing a rotating cylinder [105]

(5) Flow in a porous channel [106]

(6) Flow in several parallel channel [107]

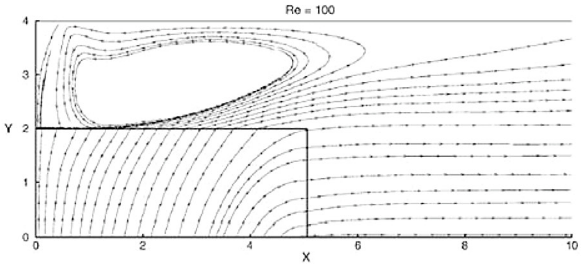
(7) Impingement cooling with metal foam [108]

(8) Heat transfer in a sensor tube [109]

(9) Mixed convection within entrapped triangular cavities [110]

(10) Mixed flow in enclosure driven by top and bottom lids [111]

(a) Ten types of 2D forced and mixed convection



(b) Heatlines for problem type 7

Fig. 8. 2D forced and mixed convection.

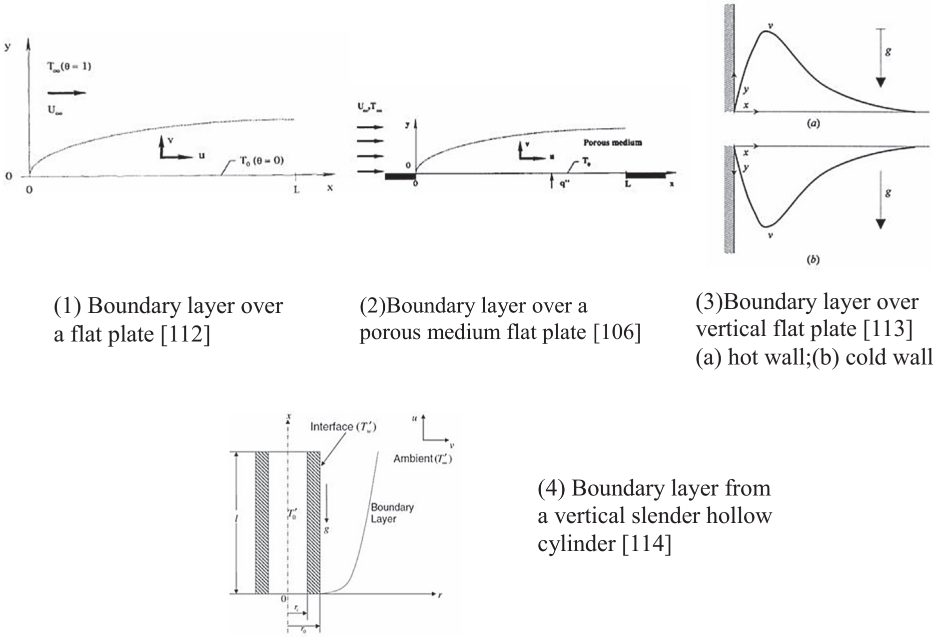
z directions, respectively [128,129]. For any problem when its numerical solutions of velocity and temperatures are obtained, the local values of the total heat flux in different direction (energy vector components) can be determined easily.

4.3. Further research needs

The concept of heatline is still useful for 3D case if we regard the heatline is the tangent of the 3D energy vector. In order to show the heat transfer route in a 3-D problem, the three components (E_x, E_y, and E_z) of the energy flux density vector are calculated from the converged results of velocity and temperature.

$$\vec{E} = \left(\rho u c_p T - k\frac{\partial T}{\partial x}\right)\vec{i} + \left(\rho v c_p T - k\frac{\partial T}{\partial y}\right)\vec{j} + \left(\rho w c_p T - k\frac{\partial T}{\partial z}\right)\vec{k} \tag{6}$$

These three components are used to trace an imaginary path of the heat flow. Some example is given in [130].



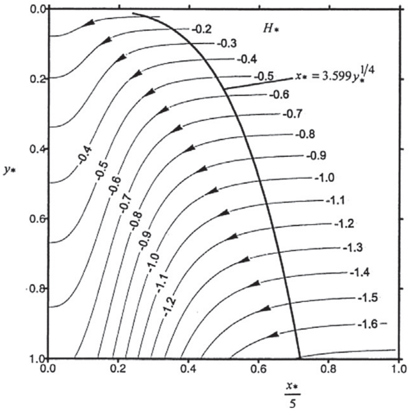
(1) Boundary layer over a flat plate [112]

(2)Boundary layer over a porous medium flat plate [106]

(3)Boundary layer over vertical flat plate [113] (a) hot wall;(b) cold wall

(4) Boundary layer from a vertical slender hollow cylinder [114]

(a) Four types of boundary layer heat transfer problems



(b) Heatlines of cold wall of problem type 3

Fig. 9. 2D boundary layer problems.

For 3D unsteady situation the application of the heatline concepts is much more complicated, some preliminary studies have been done [131], and much more researches are needed in this aspect.

5. Brief introduction to field synergy principle

The enhancement of convective heat transfer is an everlasting subject in heat transfer study. Numerous investigations, both experimental and numerical, have been conducted, and great achievements have been obtained. However, up to the end of the last century, there was no unified theory, even for the single phase convective heat transfer. In the literatures, there exist three mechanisms for single phase convective heat transfer enhancement: (1) Decreasing the thermal boundary layer thickness, such as the offset fin: (2) Increasing the interruption in the fluids: inserted devices and corrugated tubes are based on this mechanism; (3) Increasing the velocity gradient near a heat transfer wall. The so-called centre-blocked longitudinal finned tube is of this type. Each mechanism can explain some techniques but often fails for the others [132]. Up to the end of the last century, no unified theory exists even for the single phase convective heat transfer.

In 1998, Guo and his co-workers integrated the boundary energy equation along the thermal boundary layer [133] in the Cartesian coordinates (Fig. 11),

$$\int_0^{\delta_t} \rho c_p \left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} \right) dy = -\int_0^{\delta_t} \lambda \frac{\partial T}{\partial y} dy \tag{7a}$$

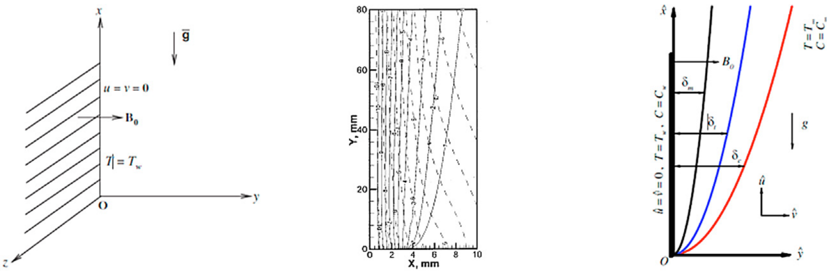
Noting that at outer boundary:

$$\left.\frac{\partial T}{\partial y}\right)_{y=\delta_t} = 0 \tag{7b}$$

and $$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \overleftarrow{U} \cdot gradT \tag{8}$$

the integration leads to

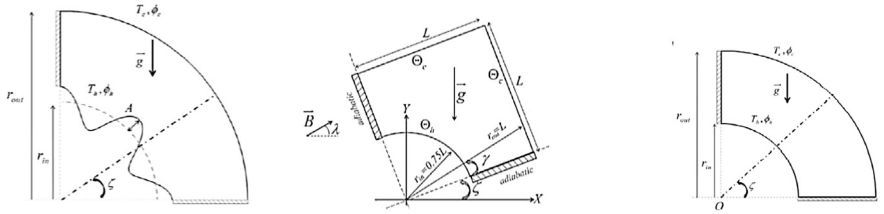
$$\int_0^{\delta_t} \rho c_p (\overleftarrow{U} \cdot gradT) = -\lambda \left.\frac{\partial T}{\partial y}\right)_{y=0} = q_w \tag{9}$$



(1) Hall-current effects on magnetohydrodynamic natural convection flow [115]

(2) Heat transfer with chemical reaction [116,117]

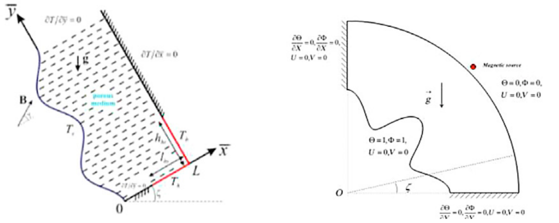
(3) Conjugate natural convection in presence of strong cross magnetic field [118]



(4)Enclosure with a hot sinusoidal wall and Brownian motion [119]

(5) Nanofluid heat transfer with magnetic field effect [120]

(6) MHD free convection of nanofluid with Brownian and thermophoresis effects [121]



(7) MHD natural convection in inclined wavy open porous cavity with nanofluid [122]

（8）Magnetic field effect on nanofluid motion with Brownian and thermophoresis effects [123]

Fig. 10. Convection with reaction of magnetic force.

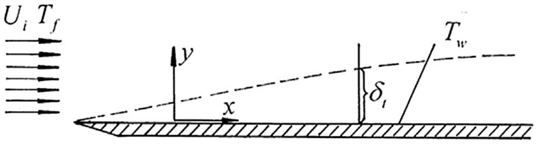


Fig. 11. Thermal boundary layer.

Mathematically,

$$\overleftarrow{U}\cdot gradT = \left|\overleftarrow{U}\right||gradT|\cos\theta \tag{10}$$

We can conclude that for a fixed flow rate and temperature difference, the smaller the intersection angle between fluid velocity and its temperature gradient, the larger the heat transfer rate. Webster Dictionary says [134]: when several actions or forces are cooperative or combined, such situation can be called "synergy". This idea is then called "field synergy (coordination) principle" (FSP), and the intersection angle the "synergy angle" (included angle).

Since most convective heat transfer processes are of elliptic type, extending the FSP to elliptic situations is of great importance. This had been implemented in [135].

Later the field synergy principle was further summarized [136] and the essence is that the better the synergy of velocity field and temperature gradient field, the higher the convective heat transfer rate under the same other conditions. The synergy of the two vector fields implies that the synergy angle between the velocity and the temperature gradient should be as small as possible.

Come here a question naturally arises: how to judge the goodness of field synergy for a specific convective heat transfer? In this regard, two indicators have been proposed: field synergy number and field synergy angle. When we consider the whole heat transfer process, the field synergy number should be adopted. It can be derived as follows [136,137]: Reformulating the boundary layer energy equation into a non-dimensional form with the convective term expressed in vector form, we obtain:

$$Re_x Pr \int_0^1 (\overline{U}\cdot\overline{\nabla T})\mathrm{d}\overline{y} = Nu_x \tag{11}$$

Following field synergy number can be defined

$$Fc = \frac{Nu_x}{Re_x Pr} = \int_0^1 (\bar{U} \cdot \nabla \bar{T}) \mathrm{d}\bar{y} \tag{12}$$

It can be seen 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: $Nu_x = Re_x Pr$, or more generally:

$$Nu_x \approx Re_x Pr \tag{13}$$

It has been shown in [137] that for all the existing single phase convective heat transfer their synergy number are much less than 1, showing a large room for their enhancement.

It should be noted that the field synergy number Fc and the Stanton number St have identical formulas, but their physical meaning and applications are quite different. St comes from the analogy between heat and momentum transfer. The group $St = C_f/2Pr^{2/3}$ was dealt by the chemical engineer Colburn, called Reynolds-Colburn analogy. It can be used directly to infer heat transfer data from measurements of the shear stress, or vise verse. It can be also extended to turbulent flow, which is much harder to predict analytically. Fc comes from the integration of dot production of the velocity vector and the temperature gradient vector over the whole domain, which stands for the dimensionless heat source strength (i.e., the dimensionless convection term) over the entire domain and physically is the indication of the degree of synergy between the velocity and temperature gradient fields, that is, the larger the Fc, the better the synergy between the velocity and temperature gradient fields, then the better the heat transfer performance.

In the application of FSP, it is often desirable to reveal for an existing heat transfer configuration where the field synergy is worse, hence there improvement is needed. In this regard, the local synergy angle is the unanimous one. To determine the domain averaged synergy angle, several definitions have been tried. It has been shown in [138] that for the same situation different definitions of the domain average synergy angles may have different values, but their variation trends are the same. This gives us quite wide flexibility to adopt a definition for the domain averaged synergy angle: we are interested in its variation trend and relative magnitude, rather than its absolute value.

Usually following domain integration mean definition is recommended:

$$\theta_m = \arccos \frac{\sum \left|\vec{U}\right| \cdot |gradT| \cos\theta_i dV}{\sum \left|\vec{U}\right| \cdot |gradT| dV} \tag{14}$$

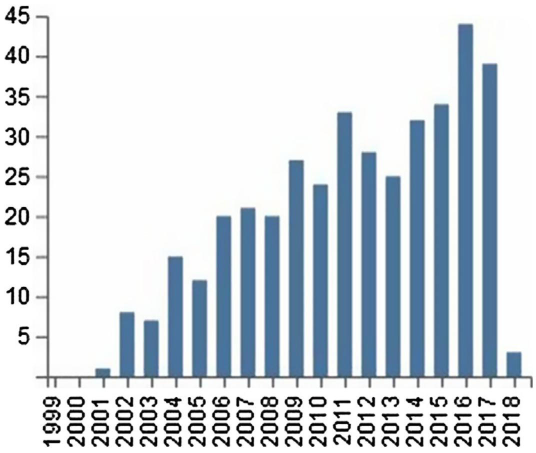
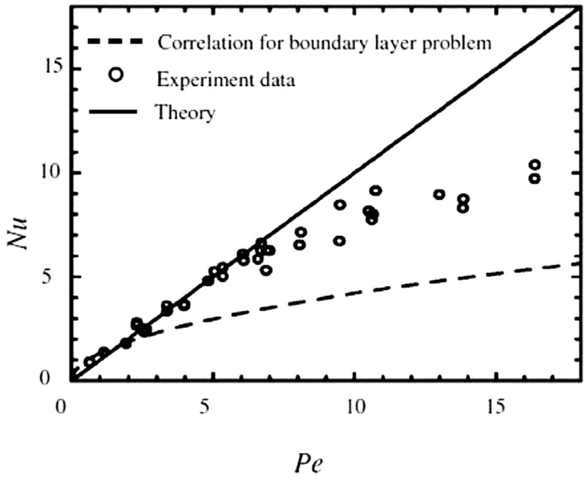


Fig. 12. Paper publication information related to field synergy principle.

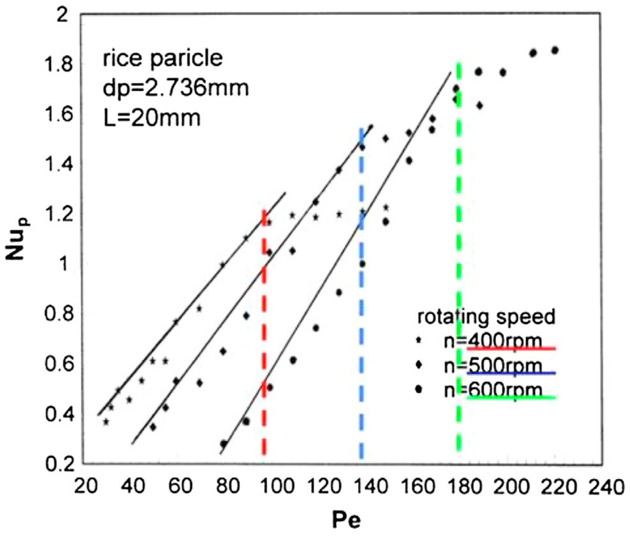
Since the proposal of FSP, it has been widely tested, verified and applied to variety of convective heat transfer problems, and great achievements have been obtained. The statistics of paper publications are shown in Fig. 12. In the following validations and applications of FSP will be selectively presented.

6. Validation of field synergy principle

According to FSP, there are two extremely situations for single phase convective heat transfer: when the fluid temperature gradient is fully parallel to fluid velocity the heat transfer rate is the maximum, and fluid may be heated or cooled depending on the direction of the fluid temperature gradient being the same or the opposite with velocity direction; The other extreme situation is that the fluid temperature gradient is normal to the velocity, then no matter how large is the velocity it does not make any contribution to heat transfer. Such brand new concept was never read in heat transfer textbook, and variety of doubts were raised in heat transfer community when the field synergy principle was just proposed. Thus at the initial stage of the proposal of FSP a number of numerical and experimental validations were conducted to validate the principle. Here only some experimental works will be mentioned.



(a) Test results for forced convection in porous medium [139]



(b) Test results in centrifugal fluidized bed [140]

Fig. 13. Two examples of the most favorable heat transfer situation.

For the 1st situation two papers published at the beginning of this century, which were independently conducted, provides results that when fluid velocity is parallel to the temperature gradient Nusselt number is proportional to Reynolds number as shown in Fig. 13 [139,140]. In the two figures it can be observed that when Reynolds number is less than a certain value *Nu* is proportional to *Re*.

For the second situation, no existing results are available in the literature, and special test facility was built in [141] to create such a situation that temperature gradient is normal to axial velocity direction. Test results definitely show that for a fixed temperature gradient the axial fluid velocity has no any effects (within the measurement uncertainty range) on the heat transfer. Partial test results are presented in Fig. 14. Test results show that the heat transfer rate depends on the temperature difference between hot and cold walls, but does not depend on flow rate (within the measurement uncertainty). The reasons why heat transfer rates at low and high flow rate deviate a bit and this deviation increases with the temperature difference are explained in details in [141].

7. Applications of field synergy principle

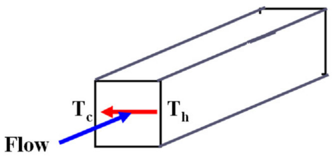
7.1. Examples of heat transfer processes well-explained by FSP

In the heat transfer textbooks and references there are a large number of cases for which why the heat transfer is enhanced or deteriorated is not well explained. With the FSP such situation can be explained well. Parts of them are summarized by the present authors in [142]. For the simplicity of presentation only three examples are described in details and other cases are only mentioned briefly as follows.

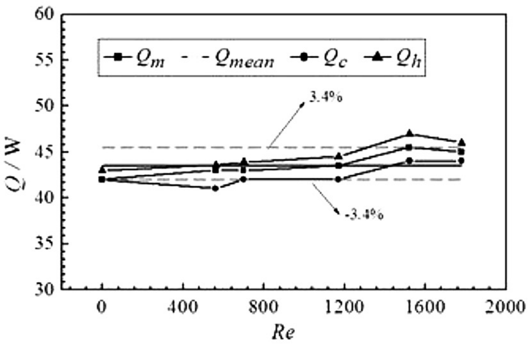
The first example is the very high local heat transfer coefficient at the stagnation point of jet impingement [143]. It is because that at the stagnation point the jet velocity direction is exactly parallel to the fluid temperature gradient: the fluid is heated if the direction is the same, and cooled if the direction is the opposite. When conjugated heat transfer occurs in the impinged plate the synergy in the vicinity of the stagnation point will be soon degraded because of the transverse transmission of heat conduction within the plate [144].

The second example is why longitudinal vortex can effectively enhance heat transfer. In literatures different explanations were proposed, and usually it is attributed to the following effects of the generated vortices: disturbing, swirling and mixing the fluid flow, breaking the fluid boundary layer and making it thinner. The experimental and numerical studies in [145,146] show that the increase of heat transfer enhancement is always inherently accompanied by the decrease in the field synergy angle between fluid velocity and temperature gradient and vise versa, confirming that the improvement of field synergy degree is the fundamental mechanism of heat transfer enhancement by vortex generator. Recent publications [147–153] related to the vortex generator all confirm the results presented in [145,146].

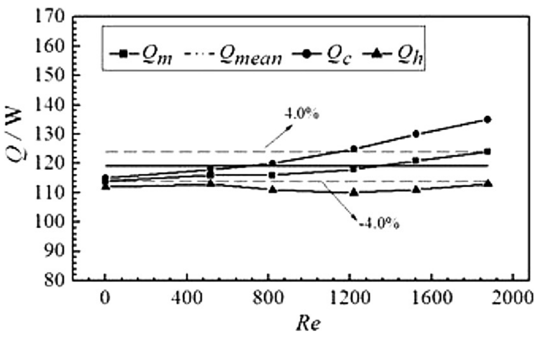
The third example related to the solar energy application. In [154] it is revealed that the solar air heater shown in Fig. 15(a), called flat-plate-collector (FPC), is very ineffective, while that



(a) Schematic diagram of test rig

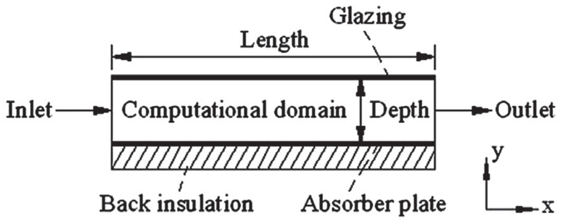


(a) $T_h - T_c = 10$ C

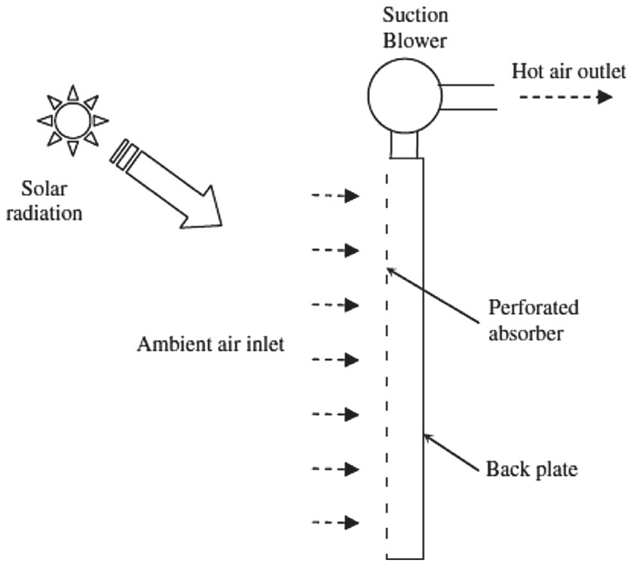


(b) $T_h - T_c = 30$ C

Fig. 14. Examples of the most unfavorable situation.



(a) Schematic diagram of FPC



(b) Schematic diagram of UTC

Fig. 15. Two types of solar air heater.

presented in Fig. 15(b), called unglazed transpired collector (UTC), is much better. The essential reason is that in FPC the air temperature gradient is almost normal to the flow air velocity, with an angle as large as 88°, while in the UTC the average synergy angle between air temperature gradient and velocity is much smaller, only about 43°. A much better synergy greatly improves the heat transfer significantly.

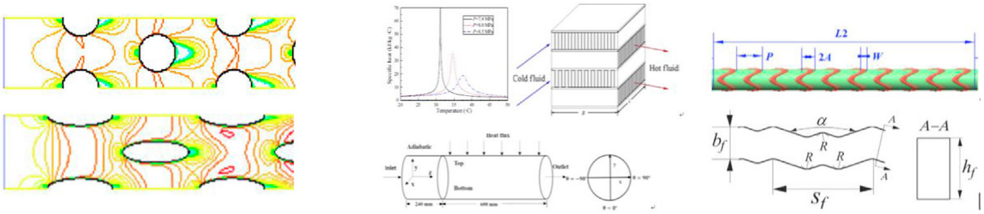
Apart from the above three examples, all single phase convective heat transfer enhanced measures and techniques known to the authors, can be well explained by FSP. A number of typical examples were collected in [142], and for the simplicity of presentation they will not be restated here. In Fig. 16, some examples [155–186] published since the publication of the review paper of [143] are grouped in eighteen cases for readers' reference.

7.2. Heat/mass transfer enhancing techniques developed by FSP

The most important application of FSP is to guide the design of enhanced surfaces or structures. Four examples are briefly described as follows.

7.2.1. Slotted fin surface with slits positioned 'front sparse and rear dense'

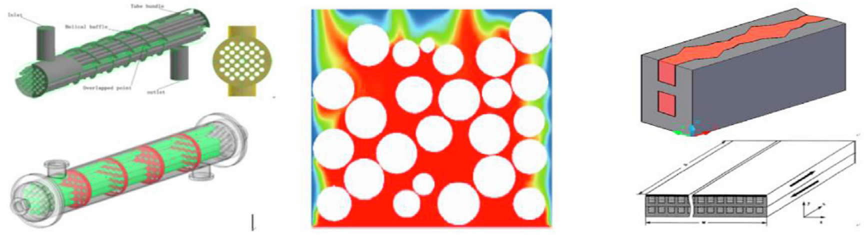
The common practice of the slotted fin surfaces is that the slits are uniformly distributed along the streamwise direction. (Pattern 1 of Fig. 17(a)). Such design seems plausible but actually not scientific. Because at the fin inlet the synergy between velocity and fluid temperature gradient is quite good, but gradually becomes worse along the flow direction [142]. Cheng et al. [187] compared three



(1) Circular tube vs. elliptical tube [155,156]

(2) SP CO2 heat transfer [157,158]

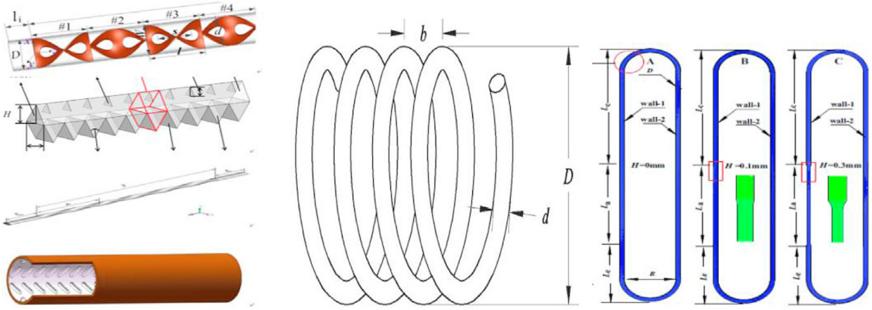
(3) Different conjugate and wavy tubes [159-164]



(4) Effect of baffles [165]

(5) Fluid-particle he at transfer in packed beds [166]

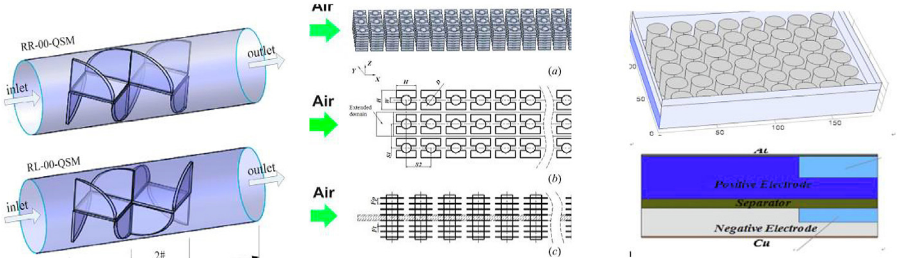
(6)Microchannel and mixer [167-170]



(7) Special enhanced techniques For internal flow [171-173]

(8) Turbulent heat transfer in helical coil [174]

(9) Oscillating heat pipe Heat transfer in pulsed heat pipe [175]

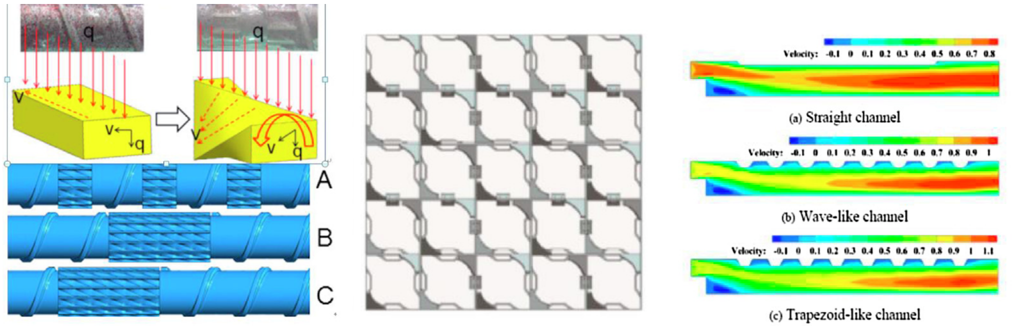


(10) Effect of static mixer [176]

(11) Effect of fin [177,178]

(12) Cooling technique of lithium iron battery [179,180]

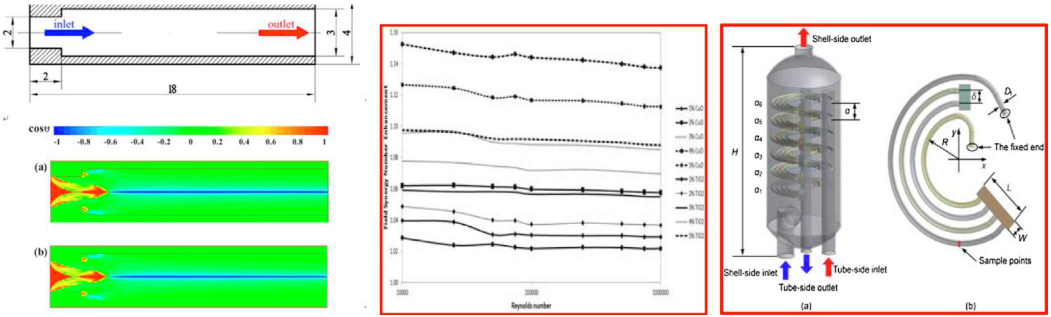
Fig. 16. Miscellaneous enhanced examples well-explained by FSP.



(13) Enhanced heat transfer in homogenizing section of polymer plasticization [181]

(14) Enhanced heat transfer of fuel assembly by mixer vane spacer [182]

(15) Enhanced heat and mass transfer in polysilicon chemical vapor deposition reactor [183]



(16) Micro-cylindrical combustor [184]

(17) Enhancing heat transfer of nanofluid [185]

(18) Enhancing heat transfer by flow-induced vibration [186]

Fig. 16 (*continued*)

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 (Fig. 18(a)–(d)). The final selection of Slit 1 has the best synergy and highest Nusselt number, and its enhancing effect has also been validated by tests of two heat exchanger with plain plate fin and the new type slotted fins. At the air velocity of 5–6 m/s, the enhancement of overall heat transfer coefficient of the new heat exchanger is about 26%, while the air-side pressure drop penalty is about 22% [142].

7.2.2. Alternating elliptical axis tube (AEAT)

For conventional convective heat transfer in a straight duct, the main flow velocity almost parallel to the fluid temperature isotherms, leading to a bad synergy between velocity and fluid temperature gradient. Meng proposed a so-called alternating elliptical axis tube (AEAT) in [188] in which second flow is induced at cross section to improve synergy.

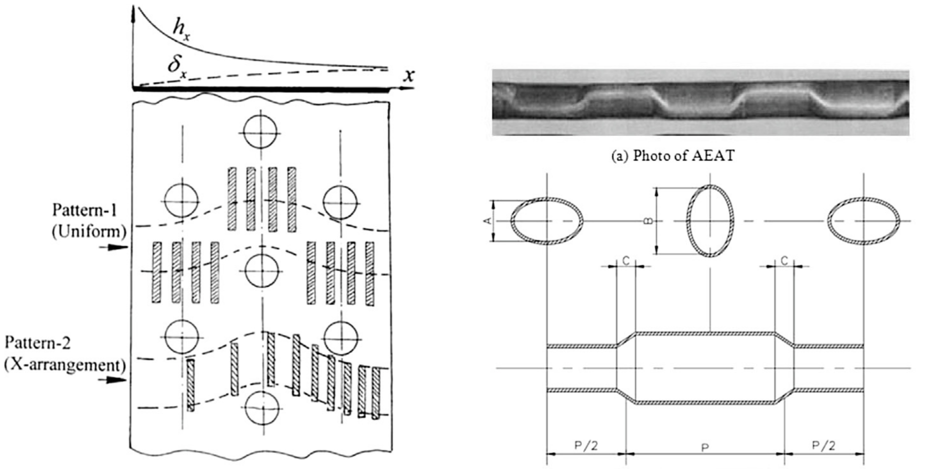
Fig. 17(b) presents a photo and schematic views of the AEAT. Apart from numerical studies experiments were conducted in [189] to validate numerical results. Fig. 19 presents some numerical results for the tube at the middle cross section of P-segment (Fig. 17(b)). It can be seen that for the elliptic tube the local velocities are more or less parallel to the local isotherms (Fig. 19(a)), implying that the local temperature gradient is almost normal to the local velocity, very bad synergy between velocity and temperature gradient. While for the AEAT there are more local regions where the isotherms are almost perpendicular to the local velocity (Fig. 19(b)) because of multiple vortex structure, which is a very perfect situation for enhancing heat transfer. Fig. 19(c) shows that compared with the straight elliptic tube the AEAT has a smaller average intersection angle by from 0.25° to 0.45°, which may lead to a difference in cosine about 30–40%.

7.2.3. Tube-and-fin surface enhanced with radiantly arranged LVGs

In Example 3 of Fig. 17(c), around each tube there are twelve small longitudinal vortex generators (LVGs) [190]. This structure has quite good heat transfer performance verified by experimental study [191]: the original tube bank with wavy fin and 6 tubes can be replaced by 5 tubes of radially arranged LVGs. The comparisons of Nusselt number and synergy angles are presented in Fig. 20 between four surfaces: the radially-arranged LVGs, a wavy fin, a plain plate fin and a plain plate fin with common-flow-down LVGs. A significant improvement in synergy of the surface with radial arrange of LVGs can be clearly identified.

7.2.4. Channel with blocks to enhance mass transfer in PEMFC

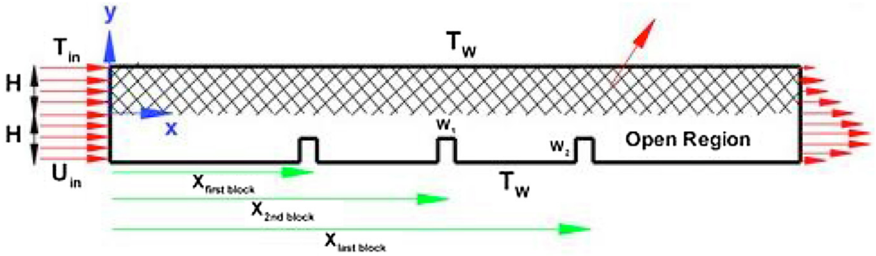
In the anode of a bipolar plate of PEMFC reactant gas (hydrogen) is supplied into the channel. When it is going forward part of it diffuses through the gas diffusion layer to the catalyst layer where the electrochemical reaction is taken, liberating electrons for the external circuit. An appropriate design of the flow field is of great significance for PEMFC performance. According to the FSP in order to enhance this mass transport process the gas mass concentration isotherms should not be parallel to the channel bottom (top). However, in the conventional designs both the main flow velocity and the mass concentration isotherms are almost parallel to the channel bottom (top) surface, a very unfavorable synergy for the convective mass transfer. Recently the FSP was used to modify the channel geometry for a better synergy between velocity and mass concentration gradient [192–195] by introducing some partial blocks (indents) in the flow channel or using wavy channel such that the local synergy can be enhanced (Fig. 21(a) [194]). Some numerical results for flow and transport are illustrated in Fig. 21 (b). In [193] the effect of channel shape on the synergy angle are computed (Fig. 21(c)), showing that great improvement in the



(a) Slotted fin surface [187]

(b) Alterative elliptic axis tube (AEAT) [188]

(c) Radiantly arranged LVGs enhanced fin [190,191]



(d) Channel with blocks to enhance mass transfer in PEMFC

Fig. 17. Four examples of enhanced techniques designed by FSP.

synergy angle can be obtained compared with the straight duct. The experimental study in [196] for fuel cell with channels of partial blocks (indents) demonstrated an enhancement of fuel cell performance by 15%.

8. Comments on field synergy principle

8.1. Role and function of FSP in convective heat transfer

Come here it is appropriate to summarize the major contributions of FSP to the convective heat transfer theory. FSP can be applied to answer following four questions, which have never been answered (or correctly and completely answered) in previous literatures.

1) Whether fluid flow alway make a contribution to convective heat transfer?

The fluid velocity must not be perpendicular to the fluid temperature gradient!

2) What is the upper limit of the exponent n in the correlation of $Nu \approx Re^n$ for single phase convective heat transfer?

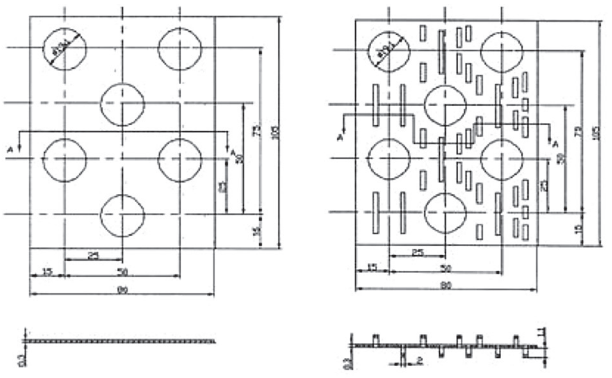
For the most favorable situation n can reach its upper limit of 1.

3) What is the fundamental mechanism for enhancing single phase convective heat transfer?

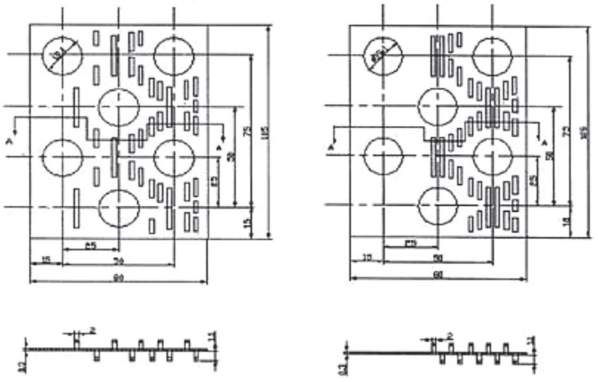
It is the improvement of synergy between fluid velocity field and temperature gradient field. So far all the numerical simulation results for the enhanced techniques known to the present authors, either laminar flow or turbulent flow, parabolic flow or elliptic flow, natural convection or forced convection, steady state or unsteady state, show that any enhancement of heat transfer is inherently accompanied by the improvement of synergy, and vise versa.

4) How to improve an existing heat transfer surface (structure) more effectively?

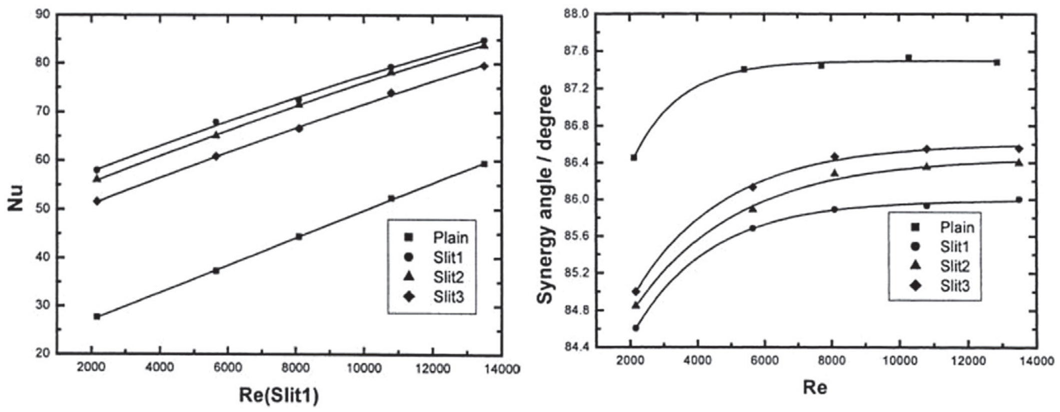
Simulating the heat transfer performance of the surface by numerical methods, revealing the local region with the worst synergy and then improving the local synergy by different techniques. Such numerical try and error simulations maybe should be conducted several times, however, it is much easier and less expensive compared with experimental method.



(a) Plain plate fin (b) Slit1



(c) Slit 2 (d) Slit 3



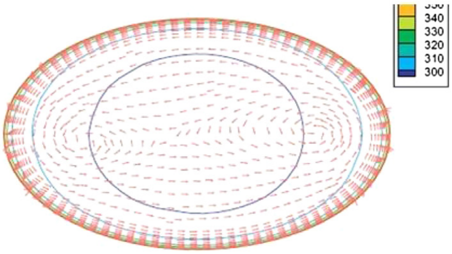
(e) Nu vs. Re of three designs (f) Synergy angle vs. Re of three designs

Fig. 18. Design of slotted fin surface with "front sparse and real dense" principle.

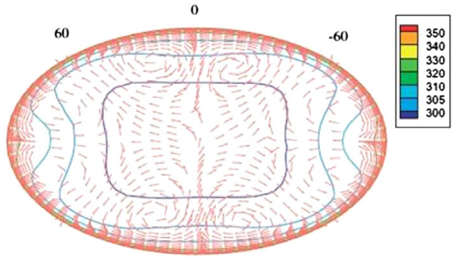
8.2. Further research needs

Since the proposal of FSP in 1998, twenty years have passed. A huge number of numerical and experimental studies have verified this basic principle both qualitatively and quantitatively. In addition the international heat transfer community has given FSP very positive evaluation. For example, 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 [197]. 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 [34] (Ref. [198] 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." (The another new one is related to turbulent heat transfer put forward by Kasagi—authors). In the preface of Advances of Heat Transfer, Vol. 46 (2014) within which the present authors' review paper [142] was included, the editors Sparrow, Cho, Abraham and Gorman wrote: "A contribution by Wen-Quan Tao and Ya-Ling He proposes a means of enhancing single-phase convective heat transfer. That method, termed field synergy principle, reduces the intersection angle between the fluid velocity and temperature gradients. Clear guidelines are proposed to accomplish the optimization of practical problems:".

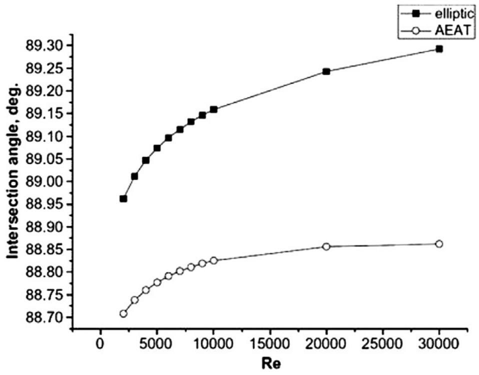
However, as the developments of many other theories, FSP also needs further development so that it can play more important and valuable roles. To the authors' knowledge, further researches include following aspects.



(a) Straight elliptic tube

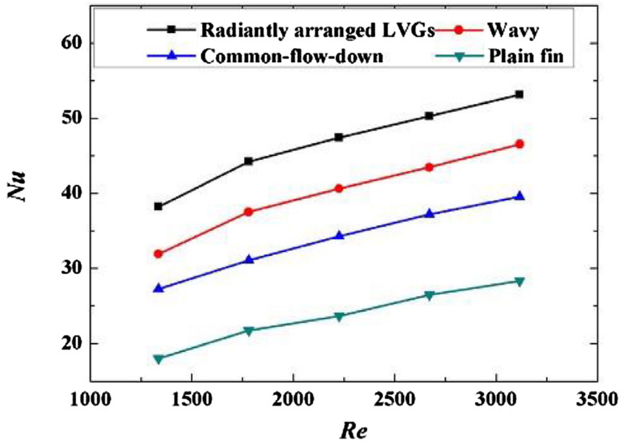


(b) AEAT

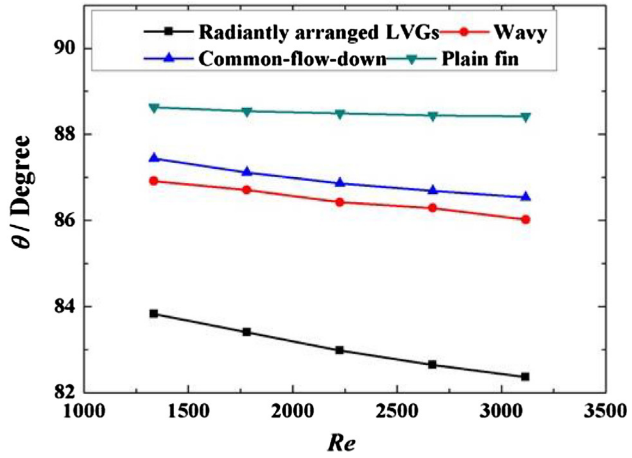


(c) Synergy angle vs. Reynolds number

Fig. 19. Numerical results of AEAT.



(a) *Nu* vs. *Re*



(b) Synergy angle vs. Reynolds number

Fig. 20. Comparison between four surface structures.

(1) Developing method for pre-designing surface structure which can make a better synergy for given flow and heat transfer conditions. To the authors' knowledge this is actually an inverse problem of convective heat transfer, i.e., by given flow rate, wall temperature or heat flux, the heat transfer surface structure is to be found which can make a desired synergy between velocity and fluid temperature gradient. Although proposals have been put forward by many authors [199–201], the problem is far being solved.

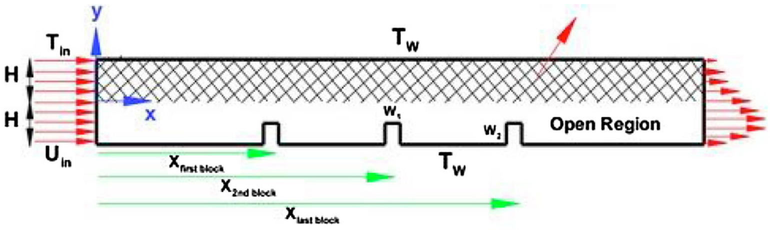
(2) Demonstrating whether FSP can be applied to the elliptical heat transfer of compressible flow. From existing publication we are sure that FSP can be applied for any type of incompressible flow, including laminar and turbulent, steady and unsteady, Newtonian and non-Newtonian, boundary layer or recirculating flow, etc.; For compressible flow [202] pointed out that if the temperature gradient is replaced by stagnation enthalpy gradient, FSP can be applied to the boundary layer compressible flow and heat transfer. Whether this statement is also valid for the elliptical compressible flow further research is needed.

(3) Further verifying appropriate indicators for FSP. It is widely accepted that both the field synergy number and the domain-averaged synergy angle can serve as the indicator for FSP, and the internal consistency between these two indicators have been numerically demonstrated in [203]. It is also well-known that a local synergy angle should be used in order to find the position on surface where local heat transfer is worse and enhancement measure should be adopted. Recently Zhu and Zhao [204] pointed out that the practice of evaluating the average synergy angle in the entire flow domain turns out to be imprecise. They proposed that (1) For laminar flow, the synergy angle should be evaluated within the thermal boundary layer because the synergy angle outside the thermal boundary is valueless, which may weaken the precision and sensitivity of the analytical result; (2) For turbulent flow, the FSP analysis must be further refined to the viscous sublayer. The feasibility of these statements needs further studies.

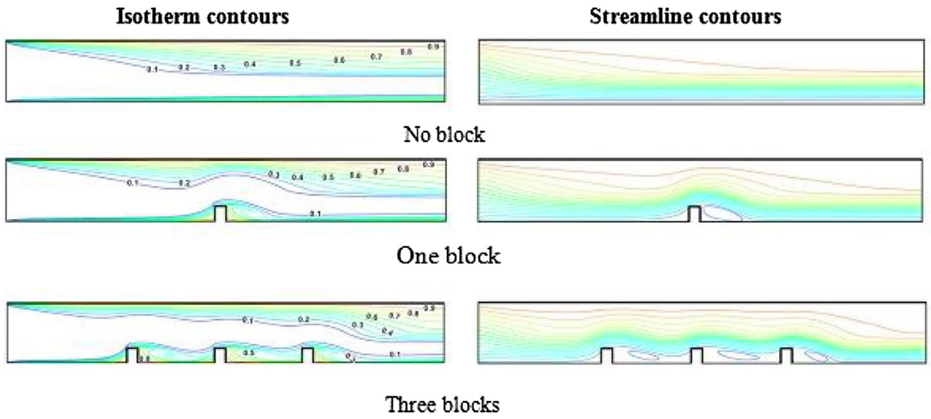
9. Conclusions

From the above comprehensive discussion and comparison following conclusions may be obtained:

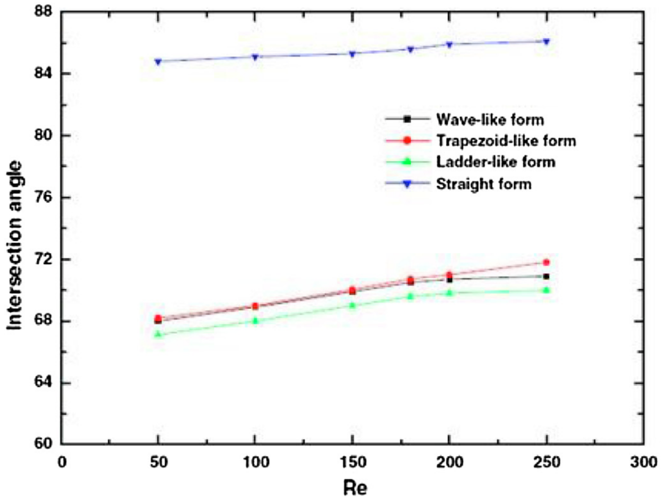
1) The major function of heat line concept is to visualize the heat transfer path which is a picture that cannot be directly given by temperature isotherms; The field synergy principle reveals that in order to enhance convective heat transfer the synergy between velocity field and the fluid temperature gradient field should be improved;



(a) Flow channel with blocks [195]



(b) Simulated results of isotherms and streamlines [195]



(c) Reduction of synergy angle by complicated channel [193]

Fig. 21. Enhancement of mass transfer in flow channel by indents (blocks).

2) To the authors knowledge no paper about heatline published so far has adopted the heatline concept for improving heat transfer; On the other hand, the filed synergy principle does not possess the function for visualization of heat transfer path. To the authors knowledge no paper about field synergy principle published so far has claimed that FSP can visualize the heat transfer path.
3) According to the original proposal of heatlines concept it is necessary to solve a Poisson equation in order to get heatlines, which definitely increasing the computational load.; For the field synergy principle either the synergy number or the synergy angle both can be obtained by using numerical results and no additional solution of differential equation is needed.
4) Both the heatline concept and the field synergy principle are two independently developed important contributions in the development process of heat transfer theory. None of them can be used to deduce the other, nor none of them can be derived from the other. Hence, there is no problem of mutual remake between them at all.
5) Both the heatline concept and the field synergy principle need further development, such that the former can be applied to 3D situations and the later can play more important and valuable role in developing heat transfer enhancement techniques.

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