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Numerical Studies on the Inherent Interrelationship between Field Synergy Principle and Entransy Extreme Principle for Enhancing Convective Heat Transfer

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Abstract

In 1998 Guo et al. integrated the boundary-layer energy equation along the thermal boundary layer thickness, and noted that at outside boundary the temperature gradient is zero and the convection term is actually the inner production of vector velocity and temperature gradient, they obtained:

$$\rho c_p \int_0^{\delta_t} (\vec{U} \bullet gradT)dy = -\lambda(\frac{\partial T}{\partial y})_{y=0} = q_w$$

For a fixed flow rate and temperature difference, the smaller the intersection angle (θ) between velocity and temperature gradient the larger the heat transfer rate. This idea is called field synergy principle (FSP). Tao et al. extended this idea to elliptic fluid flows, and showed that FSP can unified all mechanisms for enhancing single phase heat transfer.

In 2007 Guo and his co-workers proposed another new concept: entransy to describe the potential of a body to transfer thermal energy and the entransy dissipation extreme principle (EDEP). For a body with specific heat c_v,density ρ, volume V and temperature T its entransy is defined as:

$$E = [\frac{1}{2}(\rho c_v T)V]\cdot T$$

where the first part can be regarded as the amount of internal energy owned by the body and the second part is the temperature to which this amount of energy is attached. Obviously the first part is conserved during any transport process while the second part can be upgraded and degraded depending on specific conditions. It is indicated that for any heat transfer process the entransy of the system is always dissipated, which can be regarded as the indication of the irreversibility of the transport process. It is also demonstrated in [4] that for a heat transfer process with given boundary temperature condition the best one has the maximum entransy dissipation, while for that with given boundary heat flux condition the best one has minimum entransy dissipation. The combination of the two cases is called the entransy extremum principle

The purpose of this paper is to reveal the inherent interrelationship between the ideas of field synergy principle and the entransy extremum principle. From the physical intuitive considerations it is naturally to expect that for given heat transfer conditions a better synergy should lead to a larger entransy dissipation (for given temperature condition) or less entransy dissipation (for given flux condition). Numerical simulations are conducted for six examples of convective heat transfer, including laminar flow and turbulent flow, inner flow and outer flow, simple geometries and complicated geometries. All the numerical results demonstrate the correctness of the above physically intuitive expectation.

Key words: field synergy principle, entransy extremum principle, enhancement of convective heat transfer

1. Introduction to FSP

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 . Among the three modes of heat transfer the focus of the present work is concentrated on the convective heat transfer . Generally speaking, at preliminary stage (i.e., before 1960s), most studies focused on revealing the fundamental mechanism 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”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. Since then, heat transfer enhancement has become one of the hottest research subjects in the field of heat transfer. After 1990s, the technology of heat transfer enhancement has evolved from the so-called second-generation technology to the third-generation technology [1-4] and significant achievements have been achieved. In 2002, the fourth-generation concept of heat transfer enhancement technology was proposed in [5].

During the last few decades, great achievements on convective heat transfer enhancement have been obtained and various kinds of technologies have been adopted for single-phase convective heat transfer enhancement, i.e., (1) mixing the main flow and/or the flow in the wall region by using rough surface, insert, vortex generators, etc., (2) reducing the boundary layer thickness by using interrupted fins or jet impingement, etc., (3) creating velocity gradient at wall, etc. Many such techniques are presented in [6,7]. However, the essence of the convective heat transfer enhancement was still unclear. Although some explanations can account for the mechanism of the heat transfer enhancement in some special cases, they was no unified principle or theory to explain the physical mechanism for the enhancement of single-phase convective heat transfer process till the end of the last century.

In 1998, Guo and his co-workers [8-11] firstly proposed the concept of enhancing single-phase convective heat transfer for the parabolic fluid flow situation by transforming the convective term of the energy equation into the form of dot production of velocity vector and the temperature gradient, and integrating the energy equation over the thermal boundary layer. Consider a 2-D boundary-layer steady-state flow over a cold flat plate at zero incident angle as indicated in Fig.1 the energy equation is 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 integration of of Eq. (1) over the thermal boundary layer yields:

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

That is :

$$\rho c_p\int_0^{\delta_t}(\vec{U}\bullet gradT)dy=-\lambda\left(\frac{\partial T}{\partial y}\right)_{y=0}=q_w \tag{3}$$

The product of the velocity vector and the temperature gradient can be given by

$$\overline{U}\cdot gradT=\left|\overline{U}\right|\cdot\left|gradT\right|\cos\beta \tag{4}$$

with β denoting the intersetion angle between the velocity vector and the temperature gradient.

It can be seen that the convective heat transfer performance can be effectively improved by reducing the intersection angle between the velocity vector and the temperature gradient. According to the Webster's New Word Dictionary [12] "synergy" means combined or cooperative action or force. Hence this idea is so-called field synergy principle, and the intersection angle synergy angle. Later, Tao et al. [13,14] extended the FSP to the case of elliptic flow and test its applicability via many numerical examples. In [15] Guo et al. further describe the meanings of synergy. It is pointed that the synergy between the velocity vector and the temperature gradient means that: (a) The intersection angle between the velocity and the temperature gradient should be as small as possible; (b) The local values of the three scalar fields should all be simultaneously large; (c) The velocity and temperature profiles at each cross section should be as uniform as possible. This is the complete understanding of the terminology "synergy"

From then on, extensive works have been done to apply it for the development of heat transfer enhancement technology.

Intrinsically, the strength of the convective heat transfer relies on the synergy between the velocity and temperature fields. The question is how to characterize the synergy degree between two fields. The most useful application of the FSP is to reveal for the entire flow field where the synergy is bad and hence it is there enhancement technique should be adopted. In this regard, the local synergy angle between velocity and the temperature gradient is the most suiable one.

The local synergy angle between the velocity vector and the temperature gradient is defined as

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

With the local field synergy angle, many studies were conducted to obtain a general index to describe the field synergy degree in the entire flow system. Zhou [16] proposed five different mean synergy angles. Those are defined, respectively, by modulus average, vector average, volumetric average, field average and simple arithmetic mean local field synergy angles. It is found that except the simple arithmetic mean local field synergy angle, the rest are in accordance with each other qualitatively. For the case of air flowing across a certain finned tube, the variations of the mean synergy angles of different definitions with fluid velocity are plotted in Fig. 1. Clearly, there are no great qualitative differences between the variation trends of the different mean filed synergy angles. As it is the variation trend of field synergy angle that is used to guide practical problems, it is safe to adopt any one of them to qualitatively explain the reason/ mechanism of the heat transfer enhancement. Usually, the mean synergy angle based on the volumetric average or the modulus average are employed, which can be written as

Volumetric average $$\beta_{\mathrm{m}} = \frac{\sum \beta_{\mathrm{i}}\,\mathrm{d}V_{\mathrm{i}}}{\sum \mathrm{d}V_{\mathrm{i}}} \tag{6a}$$

Modulus average $$\beta_m = \arccos\frac{\sum |\vec{u}| \bullet |\mathrm{grad}t| \bullet \cos\beta_i \bullet dV}{\sum |\vec{u}| \bullet |\mathrm{grad}t| \bullet dV} \tag{6b}$$

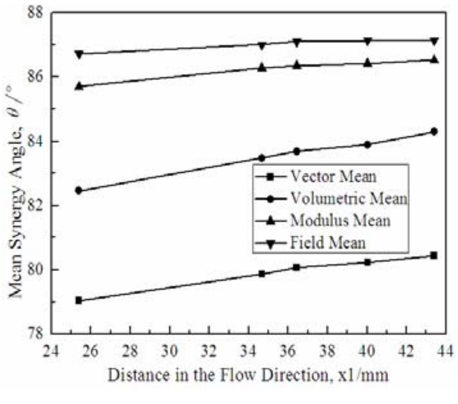
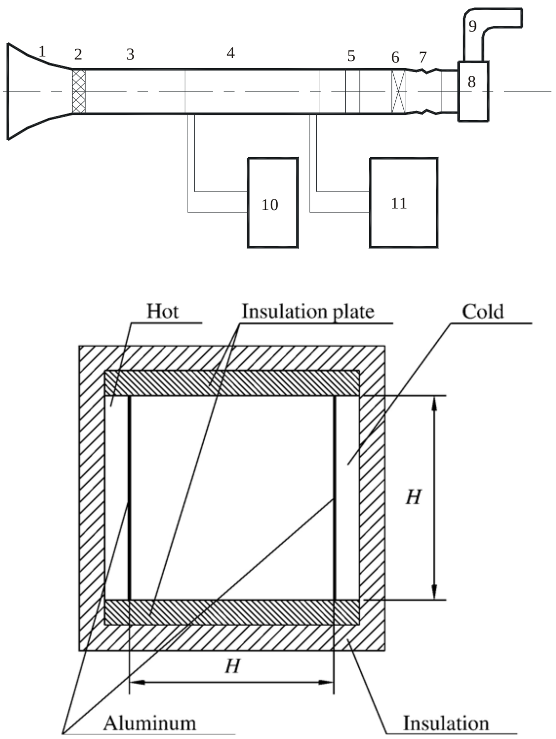


Fig. 1 Variations of the mean synergy angle with different definition

It should be noted that thedefinition of Eq.(6b) is most agreeable to the complete understanding of the concept of synergy.

A lot of papers have been published to validate or to apply the FSP to develop the enhancement technques for convective heat transfer[17-37],among whom reference[20] is especially worth mentioning. A special experimental system was designed, fabricated, and installed to demonstrate that when velocity vector is perpendicular to the fluid temperature gradient, flow velocity has noe effect on the heat transfer. The experimental apparatus is an open flow system with air as working fluid as sketched in Fig. 2 (a). Fig. 2 (b) also illustrates the cross-sectional view of the heat exchanger (i.e., heat transfer test section). The square duct consists of two horizontal PVC walls of small thermal conductivity and two vertical aluminum walls which are bounded by two narrow vertical channels through which hot and cold water goes, respectively. Due to the high thermal conductivity of aluminum, the two vertical walls can be regarded as practically isothermal by strictly controlling the water temperature differences between inlet and outlet of the channel, while the two horizontal walls of the square duct are well insulated and can be considered as adiabatic. From the above description, it can be seen that the axial fluid flow in the square duct is just normal to the imposed temperature difference between the two vertical aluminum walls of the duct. Although the natural convection due to the imposed temperature difference may cause a weak secondary flow field in the thin thermal boundary layer along the solid wall, in the major part of the duct, the main stream flow is much stronger than the secondary flow in the cross section and thus the combined fluid velocity and the temperature gradient is almost everywhere perpendicular to each other. Under such a circumstance, the heat transfer between the hot wall and the cold wall should be independent on the velocity of the main flow stream.



(a) Test rig
1. Inlet; 2. Filter; 3. Developing section; 4. Test section; 5. Flow meter; 6. Valve; 7. Flexible duct; 8. Centrifugal fan; 9. Release vent; 10. Hot water thermostat reservoir; 11. Cold water thermostat reservoir

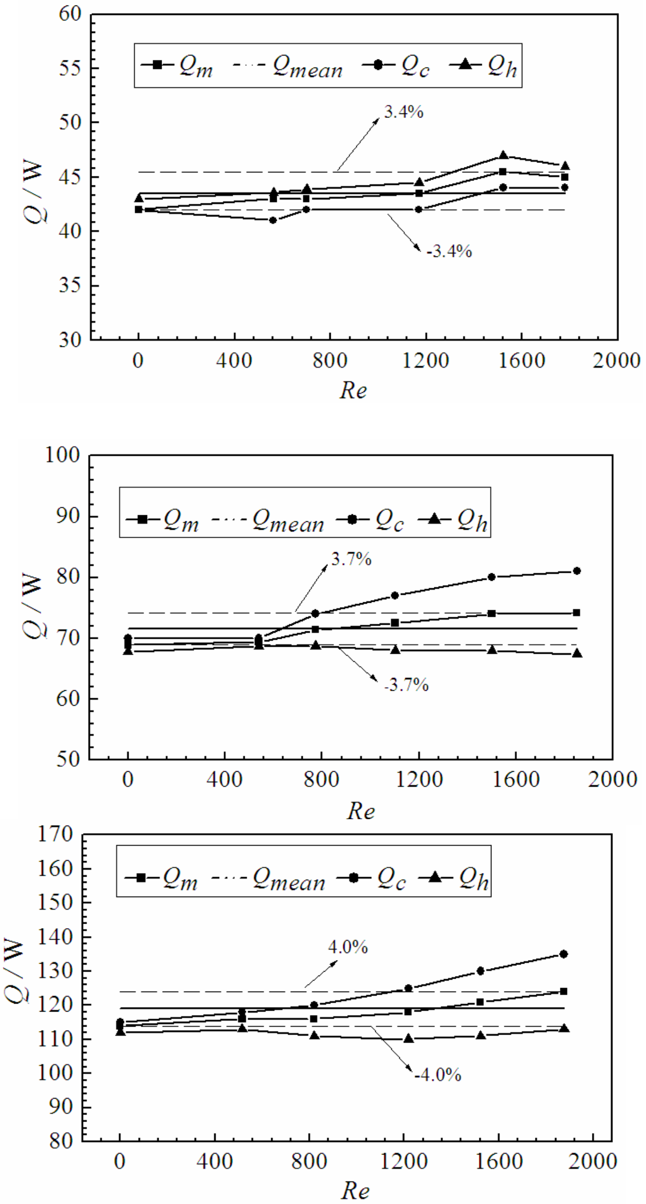
(b) Cross-sectional view of the test duct

Fig. 2 Schematic diagram of the test set up

A series of tests were conducted to measure the heat flux between the hot and cold walls based on the above-described test rig with controlled temperature difference of 10，20 and 30℃. Fig. 3 presents the measured heat flux between the hot and cold walls for different *Re*. Clearly, as indicated in Fig. 3, the magnitude of the heat flux is only influenced by the temperature difference, completely independent on the axial mean fluid velocity. For all the tests, the energy balances between the hot wall and cold wall are fairly good with the maximal deviation of about 4% which is mainly due to the testing error [20].

2. Introduction to EDEP

2.1 Definition of entransy In 2007 Guo and his co-workers proposed another new concept: entransy to describe the potential of a body to transfer thermal energy and the entransy dissipation extremum principle (EDEP) for the optimization of heat transfer process[38]. Its physical meaning of entransy is the ability of a body to transfer its thermal energy (heat) to the environment.



(a) $\Delta T = 10\text{K}$

(b) $\Delta T = 20\text{K}$

Figure 4 Variations of heat flux with Reynolds number at different temperature differences

The basic definition of the entransy of a body is:

$$E = \frac{1}{2}\left[(\rho C_V T)\times V\right]\times T \qquad (7)$$

In the above equation, the term in the square brackets is the thermal energy stored in body(relative to the reference temperature of zero degree), and the temperature standing outsie the brackets is the temperature that this energy is attached. It is this temperature shows the ability of transferring heat, hence it can be regarded as the potential of this amount of heat.

To understand why the coefficient of 1/2 is needed in Eq.(7), the reversible heating process of an object with temperature, T, and specific heat at constant volume, c_v can be used to show the necessity. For a reversible process, the temperature difference between the object and the heat source and the heat added are infinitesimal. Continuous heating of the object implies an infinite number of heat sources that heat the object successively. The temperature of these heat sources increases infinitesimally with each source giving an infinitesimal amount of heat to the object. The temperature represents the potential of the heat. Hence the ‘‘potential energy” of the thermal energy increases in parallel with the increasing thermal energy (thermal charge) when heat is added. When an infinitesimal amount of heat is added to an object, the increment in ‘‘potential energy” of the thermal energy can be written as the product of the thermal charge and the thermal potential (temperature) differential

$$dE = QdT \tag{8}$$

where $Q = Mc_V T$, with M being the mass of the body. If absolute zero is taken as the zero temperature potential, then the ‘‘potential energy” of the thermal energy in the object at temperature T is

$$E = \int_0^T QdT \tag{9}$$

The word “potential energy” is quoted because its unit is J K, not Joules. For a constant specific heat

$$E = \int_0^T QdT = \int_0^T Mc_V TdT = \frac{1}{2} Mc_V T^2 \tag{10}$$

2.2 Entransy dissipation and its extremum principle In the heat transfer process, the amount of the energy is conserved, while the ability of transferring heat is reduced because of the thermal resistance. That is to say, there is entransy dissipation in the heat transfer process. For any heat transfer process between two substances the one with a higher temperature loses entransy, while the other at lower temperature gains the entransy. However, as a whole the lost part is always larger then the gained part, leading to the dissipation of entransy. The entransy dissipation reflects the loss of heat transfer ability caused by the irreversible property in the heat transfer process. Thus entransy is the indicator of the irreversibility of a heat transfer process.

The optimization of a heat transfer process should make the peocess entrabsy dissipation extremum. According to Guo et al.[38], the entransy dissipation extremum principle (EDEP) can be described as follows: For a given temperature condition the best heat transfer process of a fluid-solid system has the maximum dissipation of its entransy; While for a given heat flux boundary condition, the best heat transfer process of the fluid-solid system has the minimum entransy dissipation.

Since the proposal of the entransy concept in [38], a number of paper have been published showing various applications in different fields, and References [39-42] may be consulted in this regard.

3. Inherent Interrelationship between FSP and EDEP and Numerical Validation

3.1 Intuitive consideration From FSP, the best heat transfer process has the largest convective heat transfer rate at the same flow rate and the same temperature difference between fluid and wall. Suppose the wall keep a uniform temperature (T), then from EDEP, the best heat transfer process has the maximum entransy dissipation. If the wall keeps uniform heat flux (q) then the best heat transfer process should have the minimum entransy dissipation. That is, the better the synergy, the larger (or the smaller) the entransy dissipation for given temperature (or given heat flux) condition,respectively. This means that synergy between velocity and fluid temperature gradient should have inherent consistency with the dissipation of entransy. The above intuitive consideration for FSP and EDEP is very meaningful and understandable!However, so far there is no diret demonstration in the open literature for the inherent relationship between FSP and EDEP. In this section, several numerical example will be provided to show such inherent consistency between FSP and EDEP.

3.2 Entransy balance equation for convecive heat transfer To proceed, the entransy balance formulation for convective heat transfer will be provided. For the convective heat transfer when fluid is cooled, as shown in Fig. 4a, we have

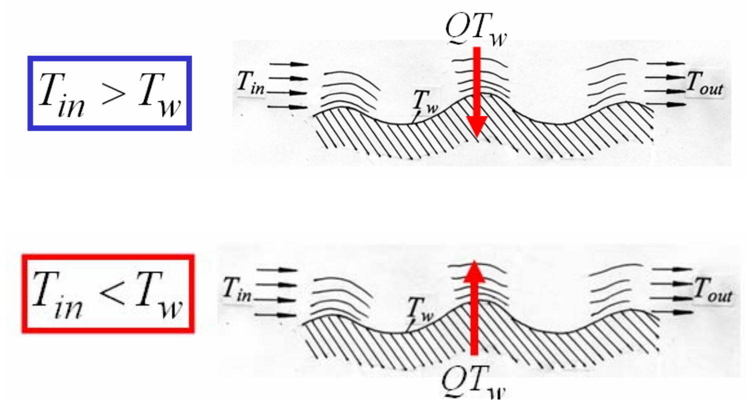
$$\frac{1}{2}C_v q_m T_{in}^2 = \frac{1}{2}C_v q_m T_{out}^2 + \rho q_m (T_{in} - T_{out}) T_w + \Delta E \tag{11}$$

The term at the left hand side is the entransy flow-in carried by the fluid, while the first term at the right hand side is the entransy flow-out carried by the fluid, the second term at the right hand side is the entransy flows to the wall, and the last term is the entransy dissipated during this heat transfer process.

If fluid is heated (Fig. 4b), thenentransy balance equation is:

$$\frac{1}{2}C_v q_m T_{out}^2 = \frac{1}{2}C_v q_m T_{in}^2 + \rho q_m (T_{out} - T_{in}) T_w - \Delta E \tag{12}$$

The meanings of the four terms can be clearly understood from the comparison with Eq.(11).



(a) Fluid cooled

(b) Fluid heated

Fig. 4 Entransy balance of convective heat transfer

3.3 Numerical validation In the following six numerical examples will be provided to validate the consistency between FSP and EDEP. All the physical problems simulated take the following assumptions:

1. Flow and heat transfer are in steady state;
2. Fluid thermo-physical properties are constant (except Example 5);
3. For Examples 1-4 wall temeprature is given,while for example 5 heat flux is given ;
4. Fluids are incompressible and the energy dissipation due to the shear stress is neglected.

FVM is used to discretize the governing equations. SIMPLE-family algorithms are adopted to deal with the linage between velocity and pressure. Numerical solutions were conduted by using the software FLUENT 6.26. After the converged solutions are obtained, the omain averaged synergy angle is determined by a UDF incorporated into FLUENT.

Example 1 Turbulent gas flow cooled by an H-type finned tube (RNG k-epsilon model)

As shown in Fig. 5a gas flow and heat transfer over H-type finned tube surface is studied at the periodically fully developed flow region. The H-type fin is rectangle in shape with a slot in its center part, making its appearance lookes like an English captal letter H. There are four rectangular votex generators in the fin. The grid system ggenerated by GAMBIT is presented in Fig5b, with total 580,000 grids. Comparison was made for the H-type fin without vortex generators with total 340,000 grids. From the numerical results of temperature and velocity fileds, the domain averaged synergy angle, β ,the domain averaged Nusselt number, Nu and the entransy dissipation of the process ΔE were determined. The variations of the three paramters with Reynolds number are shown in Figure 6. It can be seen that for this given wall temperature case, for the surface with votex generators its Nusselt number is higher than that without votex generators, its synergy angle is lower than that without votex generators, and ist entansy dissipation is higher than that without votex generators. That is the FSP is fully consistent with EDEP.

Example 2 Laminar air flow cooled/heated by continuously finned tubes with/without vortex generators

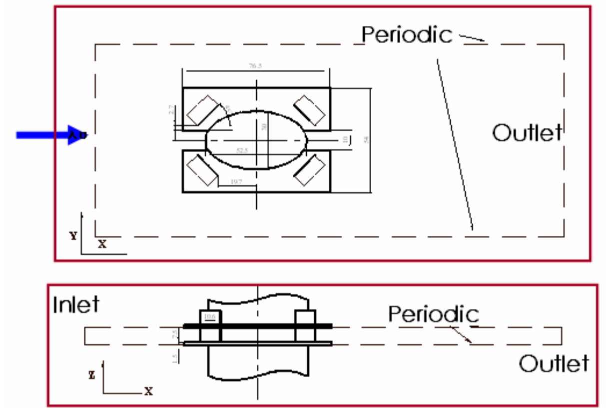
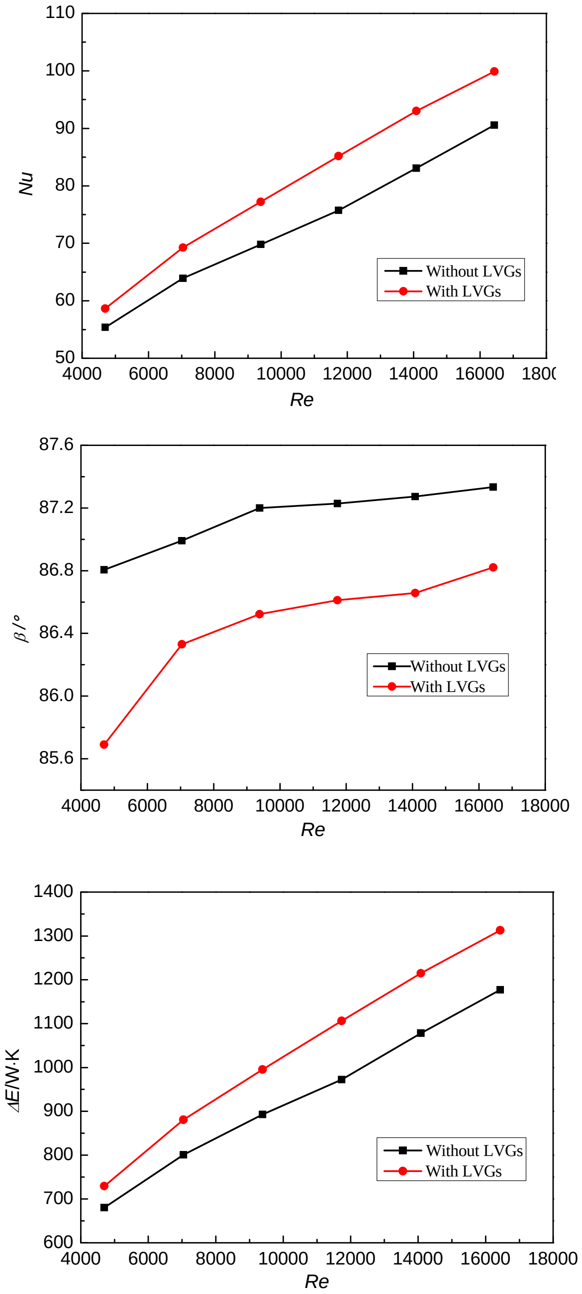


Fig. 5 Example 1 –heat transfer of H-type finned tube surface



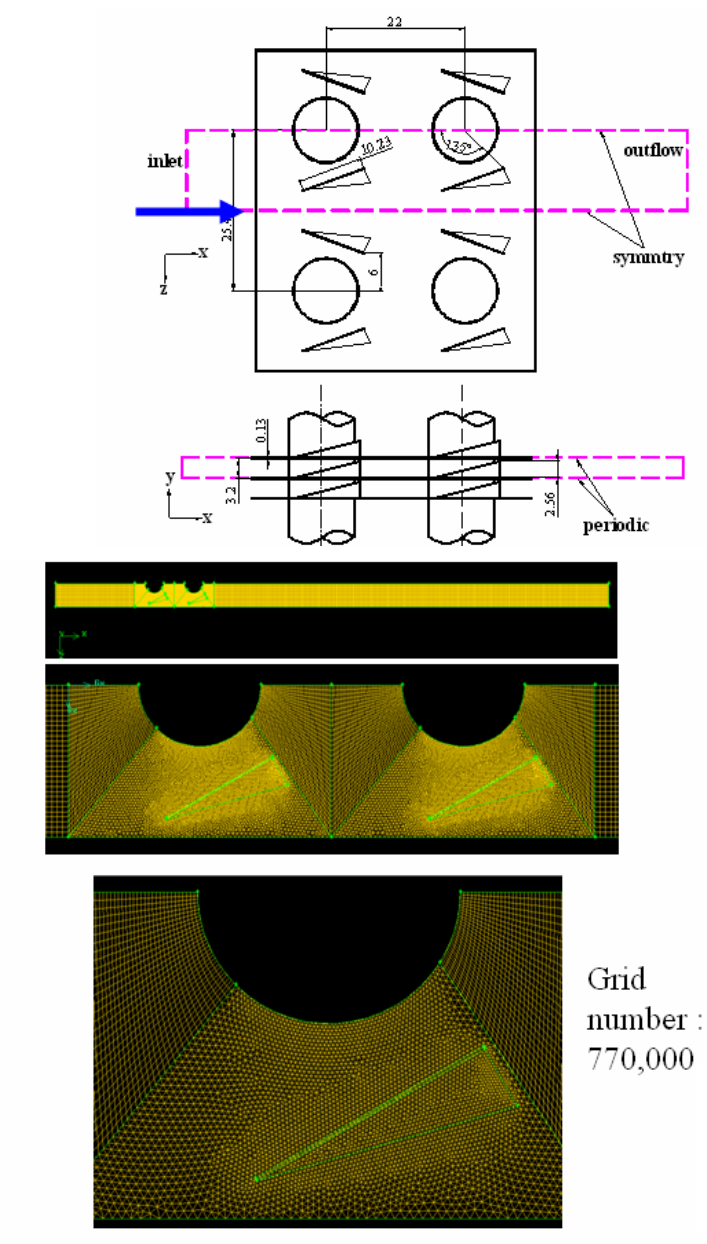
(a) Nu vs. Re

(b) β vs. Re

(c) Entransy dissipation vs Re

Figure 6 Variation of Nu, β and ΔE with Re of Example 1

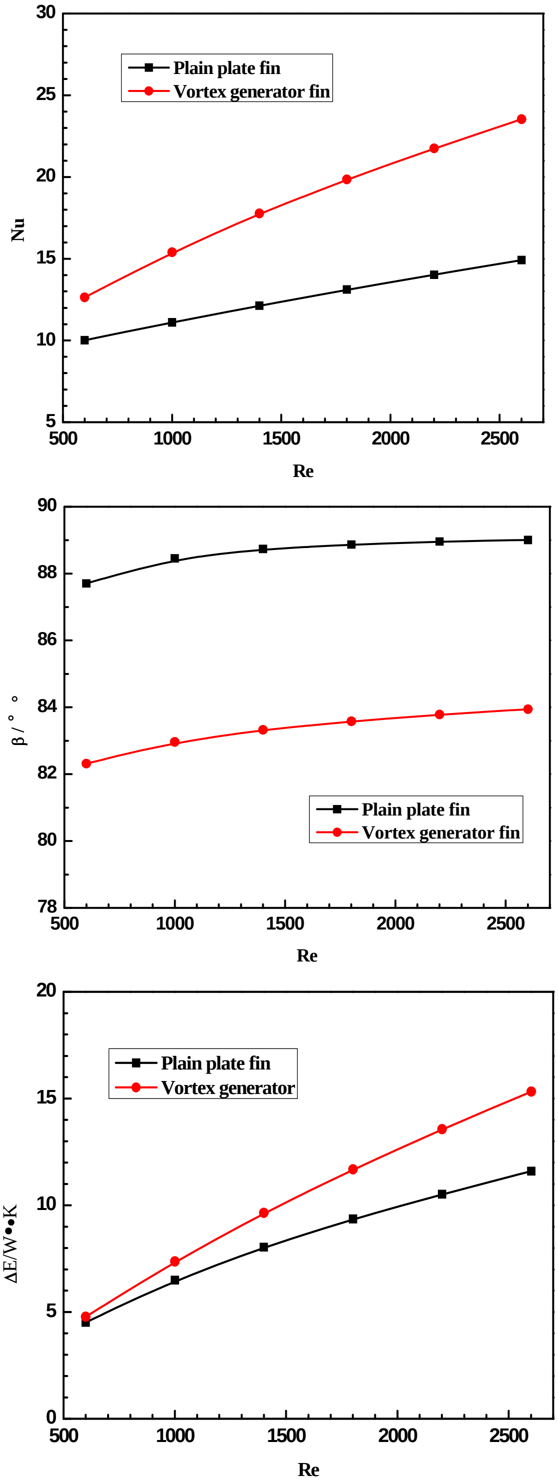
In Figure 7a, the computational domain for two-row tubes with continuous fin is presented, where dashed lines are the computational boudaries. Here the inlet boundary was set 1.5 times of the streamwise fin length ahead of the fin and the outflow boundary was set 5 times of the streamwise fin length behind the fin region. In such a was for the inflow boundary uniform inlet velocity and temperature may ne assumed and for the out flow boundary the one-way coordinate assumption [43] may be used. The totle gird numbers were 770000. Figgure 7b is the top view of the grid system. The same simulations were also performed for two-row tubes with continuous fin without votex generators. The variations of the above-mentioned three parameters with Reynolds number are shown in Figure 8. It is interesting to note that the simulations were conducted for both fluid heated case and fluid cooled case, and both cases obtained the same results.



(a) Fin geometry

(b) Grid system

Figure 7 Continuously finned two tubes with vortex generators



(a) Nu vs. Re

(b) β vs. Re

(c) Entransy dissipation vs Re

Figure 8 Variation of Nu, β and ΔE with Re of Example 2

Example 3 **Laminar air flow cooled by five-row finned tube**

In Figure 9 two kinds of six-row fin-and-tube surfaces are presented. The final grid system of the surface with vortex generators was 1936288,which is not shown for the simplicity. The comparisons of the three parameters for the surfaces with and without vortex generators are provided in Figure 10.

Example 4 Fully developed turbulent flow in tubes of constant temperature with/ without dimples by k-epsilon model

The geometries of two tubes simulated are shown in Figure 11(a), and the grid systems are presented in Figure 11(b). The numerical results for the three parameters are provided in Figure 12.

The comparisons presented in Figures 10 and 12 once again demonstrate the inherent consistency between the FSP and EDEP.

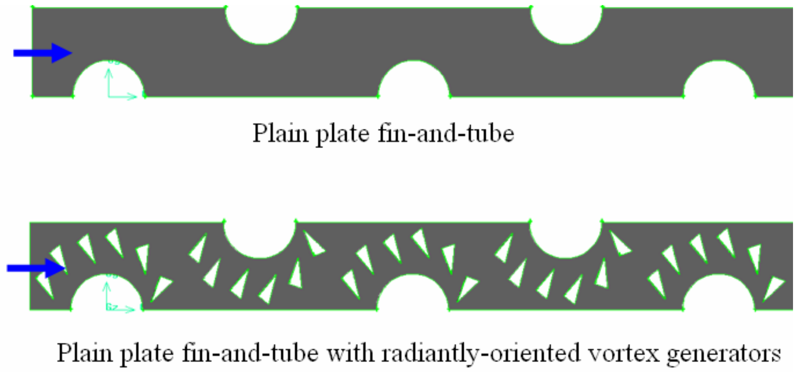
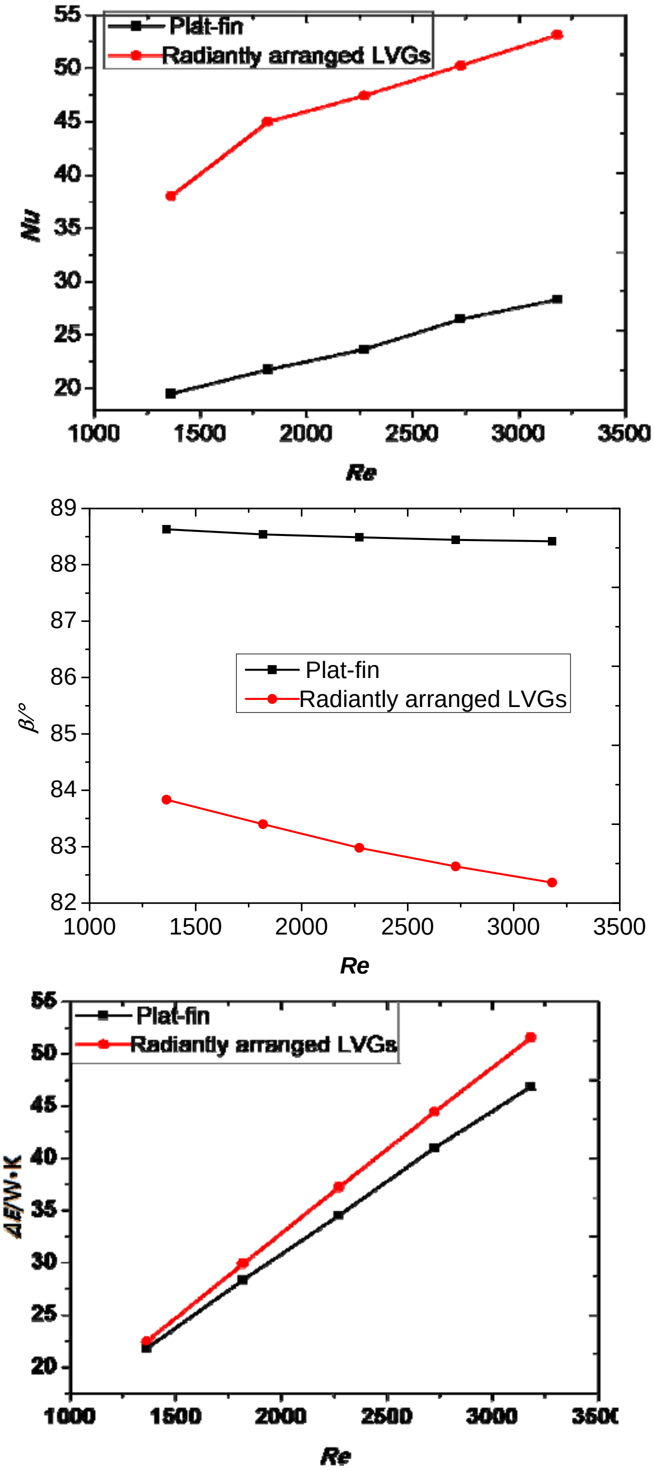


Figure 9 Two kinds of six-row fin-and-tube surfaces

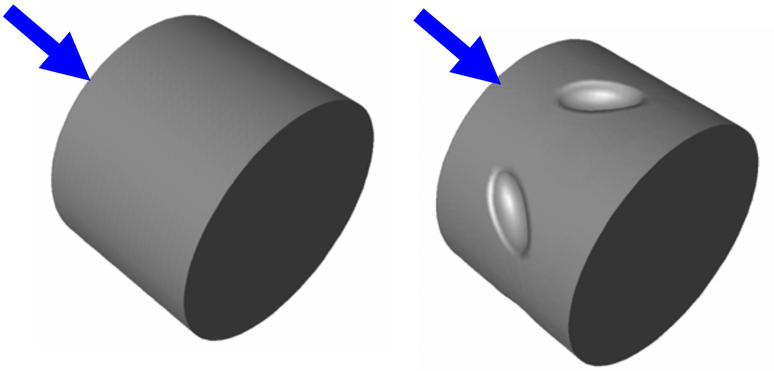


(a) Nu vs. Re

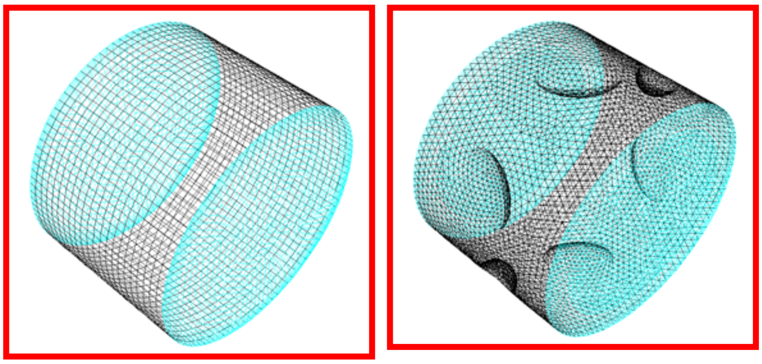
(b) β vs. Re

(c) Entransy dissipation vs. Re

Figure 10 Variation of Nu, β and ΔE with Re of Example 3

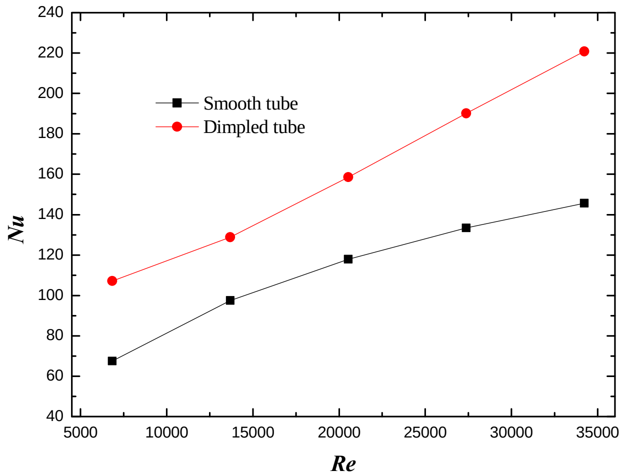


(a) Geometri

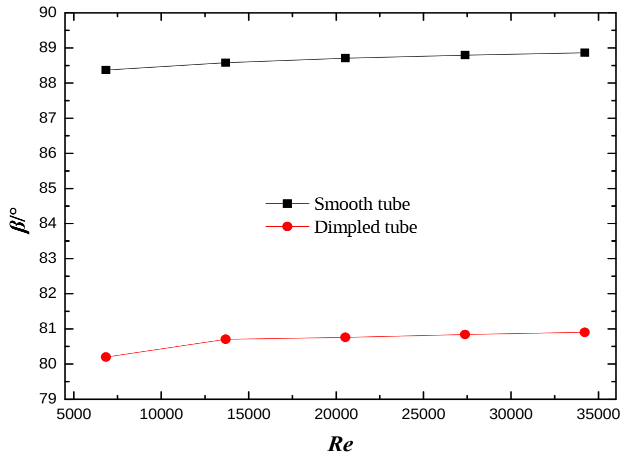


(b) Grid systems

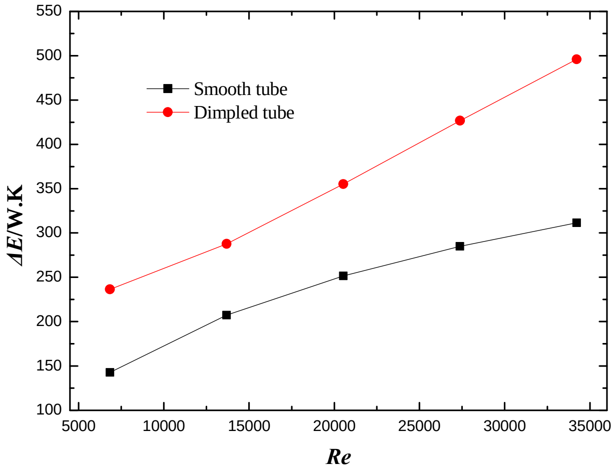
Figure 11 Fully developed convective heat transfer in tubes with/without dimples



(a) Nu vs. Re



(b) β vs. Re

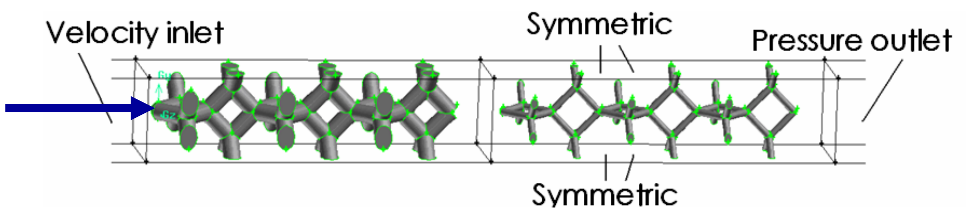


(a) Entransy dissipation vs. Re

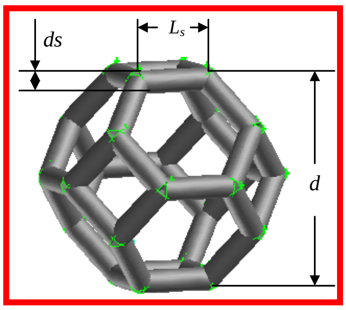
Figure 12 Variation of Nu, β and ΔE with Re of Example 4

Example 5 Turbulent heat transfer of heated air flowing through composite porous structure with uniform heat flux condition

Air flow and heat transfer through two porous sturctures (Figure 13(a)) are simulated with uniform heat flux boundary condition at the porous material surfaces. A unit structure of the porous material is shown in Figure 13(b), which has 14 surfaces. Two diameters of the rod in the unit, ds, are used: one is larger and the other is smaller, being called as dense and sparse, respectively. Numerical results of the comparisons of the three parameters are presented in Fig. 14. It can be observed that the combination of s-s has the highest Nusselt number, the smallest averaged synergy angle and the smallest entransy dissipation. Theses results are in good accordance with the FSP and EDEP. The inherent consistensy for the given boundary heat flux condition is also demonstrated.

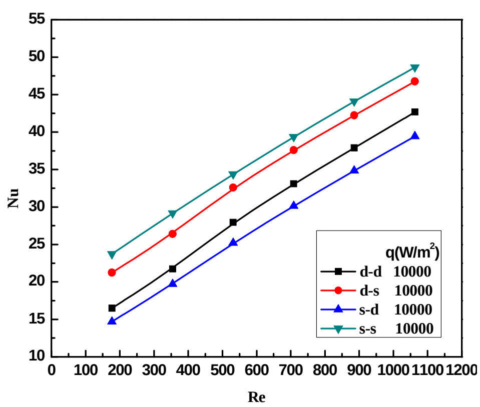


(a) Geometry of composite porous structure

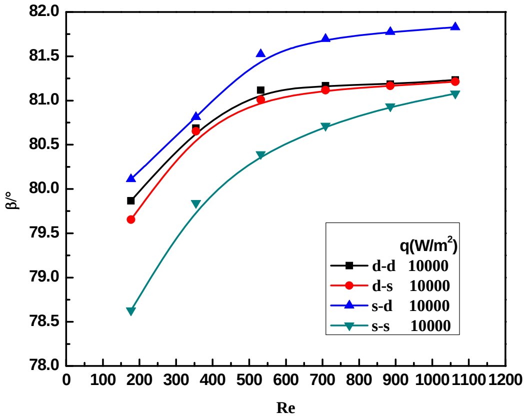


(b) Unit of porous structure

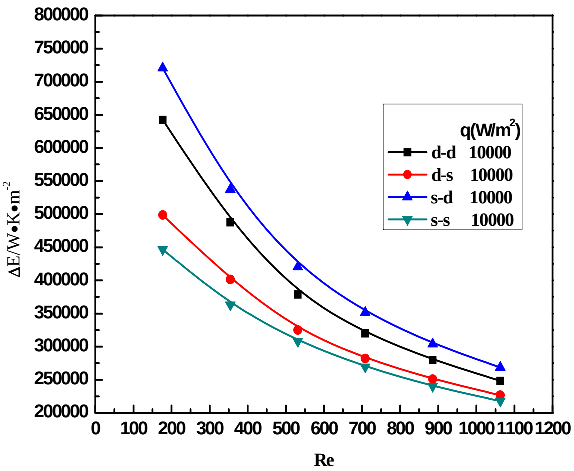
Figure 13 Laminar air flow through composite porous structure with constant heat flux



(a) Nu vs. Re



(b) β vs. Re



(c) Entransy dissipation with Re

Figure 14 Variation of Nu, β and ΔE with Re of Example 5 ($q = 100000\mathrm{W/m^2}$)

4. A Unique Formulation of EDEP

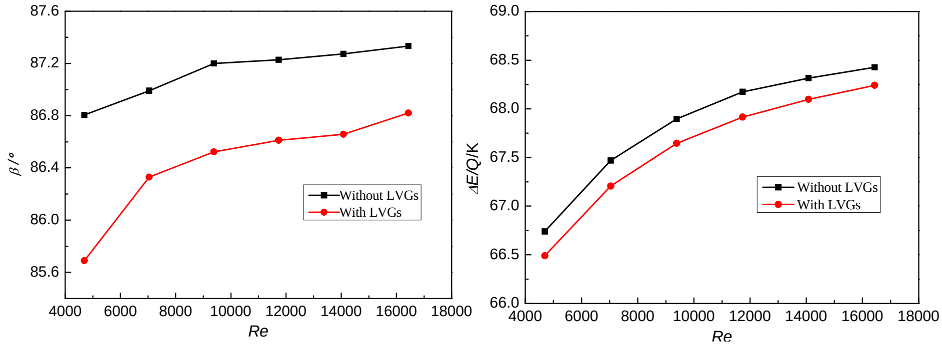
As presented above, the present formulation of entransy extremum principle is, in some extent, not very convenient to the users, because its description is dependent on the boundary conditions: for given wall temperature boundary (T-boundary) the entransy dissipation is the maximum while for given wall heat flux boundary (q-boundary) the entransy dissipation is the maximum minimum. It is our expectation that since the dissipation of the entransy is taken as the indicator of irreversibility, the optimized situation should have the minimum entransy dissipation based on some un-discovered unit. From the performance evaluation [44] of heat transfer enhancement technologies we may get some hint: in the performance evaluation of heat transfer enhancement techniques, we compare the heat transfer based on the same pumping power, why we do not compare the entransy dissipation in the same way: i.e., we should compare the entransy dissipation based on the same amount of heat transferred, that is an optimized heat transfer process should have the minimum entransy dissipation per unit energy transferred! For the given heat flux case, above statement is can be easily derived from exisiting formulation of EDEP; The key issue is to verify this idea for the isothermal boundary condition for which acording to the present formulation of EDEP the best situation has the maximum entransy dissipation. Before we present numerical demestration of this unique formulation,we first discuss this concept from point of view of dimensional analusis.

The physical meaning of the entransy dissipation per unit energy transferred can be well understood from its dimension:

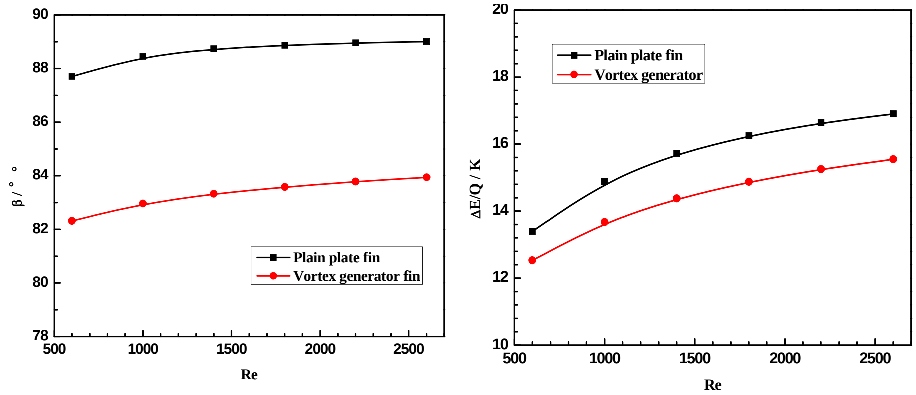
$$\left[\frac{\text{energy}\times\text{temperature}}{\text{energy}}\right]=\left[\text{temperature}\right] \tag{13}$$

That is the physical meaning of the entransy dissipation per unit energy transferred is the temperature, or more appropriately, the temperature difference. Since temperature difference is the most essential driving force of heat transfer, the best heat transfer at any condition should have the minimum temperature difference, hence minimum entransy dissipation per unit energy transferred.

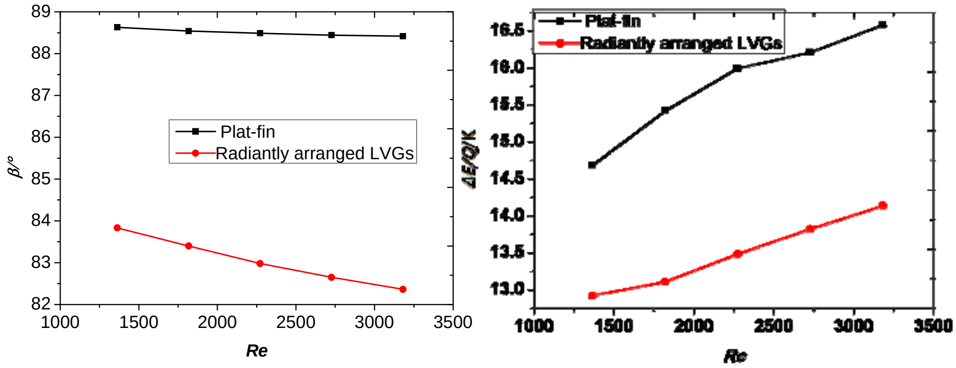
In the following the above-mentioned 5 exmaples will be reconsidered from view pint of entransy dissipation per unit energy transferred. Numerical results are presented in Figs. 15, where the variations of the averagd synergy angle with Re are shown in left figures and and the variation of entransy dissipation per unit energy transferred with Re are shown in the right figure. From the figures the unique formation of EDEP are verified obviously.



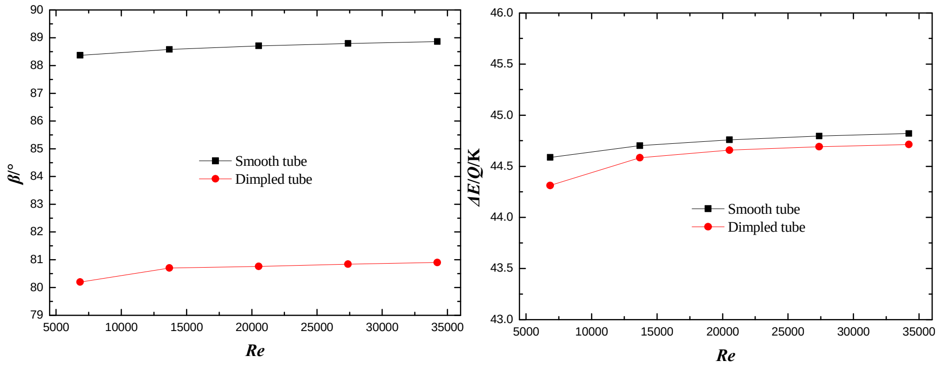
(a) β and $\Delta E / Q$ vs. Re (Example 1)



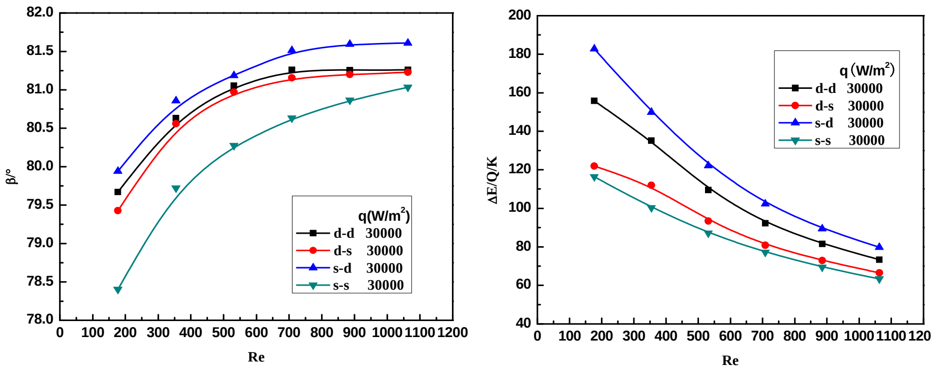
(b) β and $\Delta E / Q$ vs. Re (Example 2)



(c) β and $\Delta E / Q$ vs. Re (Example 3)



(d) β and $\Delta E / Q$ vs. Re (Example 4)



(a) β and $\Delta E / Q$ vs. Re (Example 2)

Figure15 Comparisons of variation trends of β and $\Delta E / Q$ with Re for five examples

In the study of heat transfer enhancement, one of our major goals is to develop high-efficiency and low-resistance heat transfer element. Low-resistance is related to pressure field, hence, it is our expectation that for the enhancment technique possessing high heat efficiency and low resistannce, the three fields, i.e. velocity, temperature gradient and pressure gradient should have, at least in some sense, a better synergy between them. And with this unique formulation of EDEP, we may expect that the high-efficiency and low-resistance heat transfer element should has the minimum of entransy dissipation and the minimum of pumping power consumption based on unit heat transferred. Further more research work is needed in order to verify the expectation and our group is going on this way.

5. Conclusion

From above presentation following conclusions may be made:

1. The new physical quantity ENTRANSY of a substance may be regarded being composed of two parts: one is proportional to its internal thermal energy which is always conserved, the other is the temperature at which this energy is attached. The temperature part may be degraded or upgraded during any heat transfer process.
2. The entransy of a substance represents its ability to transfer its internal thermal energy to the environment. During any heat transfer process the entransy gained is always less then the entransy lost. The difference is the entransy dissipation.
3. The field synergy principle and the entransy extremum principle are inherently consistent in that a better synergy corresponds to a less entransy dissipation for unit energy transferred.
4. The entransy extremum principle may be reformulated by following unique expression: an optimized heat transfer process with any boundary condition dissipates the minimum entransy for unit energy transferred. The dimension of the entransy dissipation per unit energy transferred is temperature (unit of K), hence entransy dissipation per unit energy may be regarded as an equivalent temperature difference of the heat transfer
5. The best heat transfer enhancing technique at given condition may be obtained by satisfying following object function: the minimum of entransy dissipation and the minimum pumping power consumption per transferring unit energy.

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References

[1] Bergles A.E. Heat transfer enhancement - The maturing of second-generation heat transfer technology, Heat Transfer Engineering.1997.Vol. 18. No. 1.Pp. 47-55.

[2] Bergles A.E. Heat transfer enhancement - The encouragement and accommodation of high heat fluxes, ASME. Journal of Heat Transfer. 1997.Vol.119.No.1. Pp. 8-19.

[3] Rohsenow W. M., Hartnett J.P., Chi Y. I. Handbook of Heat Transfer, third ed., McGraw Hill, New York, 1998.

[4] Bergles A.E. Enhanced heat transfer: Endless frontier, or mature and routine? Journal of Enhanced Heat Transfer.1999.Vol. 6.No.2-4.Pp. 79-88.

[5] Bergles A.E. ExHFT for fourth generation heat transfer technology, Experimental Thermal and Fluid Science. 2002. Vol. 26.No.2-4.Pp.335-344.
[6] Bergles A.E. Application of Heat Transfer Augmentation. Hemisphere Pub Co., London, 1981.
[7] Webb R.L. Principles of Enhanced Heat Transferm, John Wiley & Sons, New York, 1994.
[8] Guo Z.Y., Li D.Y., Wang B.X. A novel concept for convective heat transfer enhancement, International Journal of Heat and Mass Transfer. 1998. Vol.41.No.14.Pp. 2221-2225.
[9] Wang S., Li Z.X., Guo Z.Y. Novel concept and device of heat transfer augmentation, in: Proceedings of the Eleventh Inthernational Conference on Heat Transfer, Kyongju, Korea, 1998, Pp. 405-408.
[10] Guo Z.Y., Wang S. Novel concept and approaches of heat transfer enhancement, in: Proceedings of Symposium on Energy Engineering in the 21st Century, Begell House, New York, 2000, Pp. 119-126.
[11] Guo Z.Y. Mechanism and control of convective heat transfer - Coordination of velocity and heat flow fields, Chinese Science Bulletin.2001. Vol. 46.No.7. Pp. 596-599.
[12] Webster's Third New International Dictionary ,1961, New York
[13] Tao W.Q., Guo Z.Y., Wang B.X. Field synergy principle for enhancing convective heat transfer-its extension and numerical verifications, International Journal of Heat and Mass Transfer. 2002.Vol. 45.No.18.Pp. 3849-3856.
[14] Tao W.Q., He Y.L., Wang Q.W., Qu Z.G., Song F.Q. A unified analysis on enhancing single phase convective heat transfer with field synergy principle, International Journal of Heat and Mass Transfer. 2002.Vol.45. No.24. Pp. 4871-4879.
[15] Guo Z.Y., Tao W.Q., Shah R.K. The field synergy (coordination) principle and its applications in enhancing single phase convective heat transfer, International Journal of Heat and Mass Transfer. 2005. Vol. 48. No.9. Pp. 1797-1807.
[16] Zhou J.J. Mechanisms of heat transfer enhancement and energy saving of the compact heat exchanger surfaces and study on their optimization, PhD thesis, Xi'an Jiaotong University, Xi'an, 2006.
[17] Shi M.H., Wang H., Hao Y.L. Experimental study of forced convective heat transfer in porous meduim with centrifugal force, Journal of Engineering Thermophysics.2002. Vol. 23.No.4. Pp.473-476.
[18] Wang H., Shi M.H. Numerical simulation of drying process in centrifugal fluidized bed, Journal of Chemical Engineering. 2002. Vol. 53. No.10. Pp. 1040-1045.
[19] Song F.Q., Qu Z.G., He Y.L., Tao W.Q. Study of air heat transfer passing through finned tube banks at low velocity, Journal of Xi'an Jiaotong University. 2002. Vol.36. No.9. Pp. 899-902.
[20] Ma L.D., Li Z.Y., Tao W.Q. Experimental verification of the field synergy principle, International Communications in Heat and Mass Transfer. 2007. Vol.34. No.3. Pp. 269-276.
[21] Zeng M., Tao W.Q. Numerical verification of the field synergy principle for turbulent flow, Journal of Enhanced Heat Transfer. 2004. Vol. 11.No.4. Pp.453-459.
[22] Zeng M. Experimental and numerical studies of fluid flow and heat transfer characteristics in wavy tubes, PhD thesis, Xi'an Jiaotong University, Xi'an, 2004.
[23] Li Z.Y. Study on the turbulent fluid flow and heat transfer in rotating ducts, PhD thesis, Xi'an Jiaotong University, Xi'an, 2001.
[24] Chen Q., Meng J.A. Field synergy analysis and optimization of the convective mass transfer in photocatalytic oxidation reactors, International Journal of Heat and Mass Transfer. 2008. Vol.51. No.11-12. Pp. 2863-2870.
[25] Chen Q., Ren J.X., Guo Z.Y. Field synergy analysis and optimization of decontamination ventilation designs, International Journal of Heat and Mass Transfer. 2008. Vol.51. No.3-4. Pp.873-881.

[26] He Y.L., Wu M., Tao W.Q., Chen Z.Q. Improvement of the thermal performance of pulse tube refrigerator by using a general principle for enhancing energy transport and conversion processes, Applied Thermal Engineering. 2004. Vol.24. No. 1. Pp. 79-93.
[27] Wu L.B., Li Z., Song Y.Z. Field synergy principle of heat and mass transfer, Chinese Science Bulletin. 2009. Vol. 54.No. 24. Pp. 4604-4609.
[28] Chen Q., Wang M., Guo Z.Y. Field Synergy principle for energy conservation analysis and application, Advances in Mechanical Engineering. 2010. 9 pp.
[29] Song W.M., Meng J.A., Li Z.X. Optimization of flue gas convective heat transfer with condensation in a rectangular channel, Chinese Science Bulletin. 2011.Vol.56.No.3.Pp. 263-268.
[30] Chen Q., Ren J.X., Guo Z.Y. Fluid flow field synergy principle and its application to drag reduction, Chinese Science Bulletin. 2008. Vol. 53. No.11. Pp.1768-1772.
[31] Lei Y.G. Theoretical and experimeantal investigation on heat transfer augmentation of internal and external flow with high efficiency, PhD thesis, Xi'an Jiaotong University, Xi'an, 2009.
[32] He Y.L., Lei Y.G., Tian L.T., Chu P., Liu Z.B. An analysis of three-field synergy on heat transfer sugmentation with low penalty of pressure drop, Journal of Engineering Thermophysics. 2009. Vol. 30.No.11.Pp. 1904-1906.
[33] Liu W., Liu Z.C., Ming T.Z., Guo Z.Y. Physical quantity synergy in laminar flow field and its application in heat transfer enhancement, International Journal of Heat and Mass Transfer. 2009. Vol.52 . No.19-20. Pp. 4669-4672.
[34] Chen Y.P. g, Lee T.S., Low H.T. Numerical simulation of conjugate heat transfer in electronic cooling and analysis based on field synergy principle, Applied Thermal Engineering. 2008. Vol.28. No.14-15. Pp. 1826-1833.
[35] Cheng Y.P., Qu Z.G., Tao W.Q., He Y.L. Numerical design of efficient slotted fin surface based on the field synergy principle, Numerical Heat Transfer Part a-Applications. 2004. Vol. 45.No.6. Pp. 517-538.
[36] Qu Z.G., Tao W.Q., He Y.L. Three-dimensional numerical simulation on laminar heat transfer and fluid flow characteristics of strip fin surface with x-arrangement of strips, ASME. Journal of Heat Transfer. 2004, Vol.126. No.4. Pp. 697-707.
[37] Tao W.Q., He Y.L., Qu Z.G., Cheng Y.P. Applications of the field synergy principle in developing new type heat transfer enhanced surfaces, Journal of Enhanced Heat Transfer. 20004. Vol.11. No.4. Pp. 435-451.
[38] Guo Z. Y., Zhu H. Y., and Liang X. G. Entransy—a physical quantity describing heat transfer ability, Int. J. Heat Mass Transfer. 2007. Vol. 50, No. 13–14, Pp. 2545-2556.
[39] Guo Z.Y., Cheng X., and Xia Z.Zh,, Least dissipation principle of heat transport potential capacity and its application in heat conduction optimization, Chinese. Sci. Bull.@003. Vol. 48, No. 4. Pp. 406-410.
[40] Meng J. A., Liang X. G., and Li Z. X. Field synergy optimization and enhanced heat transfer by multi-longitudinal vortexes flow in tube, Int. J. Heat Mass Transfer. 2005. Vol.48. No. 16. Pp. 3331-3337.
[41] Meng J.A., Liang X.G., Li Z.X., and Guo Z.Y. Numerical study on low Reynolds number convection in alternate elliptical axis tube, J. Enhanced Heat Transfer. 2009. Vol. 11, no.
[42] Wu J. and Liang X.G. Application of entransy dissipation extremum principle in radiative heat transfer optimization, Sci. China Ser. E-Technol. Sci. 2008. Vol.51. No. 8. Pp. 1306-1314.
[43] Tao W.Q. Numerical heat transfer, 3ed edition. Xi'an Jiaotong University Press, Xi'an, 2001
[44] Fan J.F., Ding W.K., Zhang J.F., He Y.L., Tao W.Q.International Journal of Heat and Mass Transfer. 2009，Vol.52. No.1-2.Pp.33-44