Available online at www.sciencedirect.com
ScienceDirect
International Communications in Heat and Mass Transfer 34 (2007) 269–276
International Communications in HEAT and MASS TRANSFER
www.elsevier.com/locate/ichmt

Experimental verification of the field synergy principle☆

Liang-Dong Ma, Zeng-Yao Li, Wen-Quan Tao *

State Key Laboratory of Multiphase Flow in Power Engineering, School of Energy and Power Engineering, Xi' an Jiaotong University, Xi' an, PR. China

Available online 15 December 2006

Abstract

In this paper, the basic idea of the field synergy principle (FSP) is briefly reviewed and is validated experimentally by incompressible flow through a square duct with an imposed temperature difference between vertical walls and perfectly insulated on the horizontal walls. This creates a situation where the steamwise flow velocity is normal to the cross section temperature gradient. The experimental results show the independency of crosswise heat transfer rate on the steamwise flow velocity. Detailed discussion is provided to account for some minor deviation from the expected results of FSP.

Keywords: Field synergy principle; Experimental study; Enhanced heat transfer

1. Introduction

The enhancement of convective heat transfer is an everlasting subject for both the researchers of heat transfer community of academia and the technicians in industry. Numerous investigations, both experimental and numerical, have been conducted and great achievements have been obtained. The passive techniques for single phase heat transfer [1] can be grouped usually into three types of enhanced methods: Decreasing the thermal boundary layer thickness [2], increasing the interruption in the fluids [3] and increasing the velocity gradient near a heat transfer wall [4].

However, up to the end of last century, there was no unified theory which can reveal the essence of single phase convective heat transfer enhancement common to all enhancement methods. In 1998, Guo and his co-workers proposed a novel concept for enhancing convective heat transfer of laminar parabolic flow [5] which is now called field synergy principle (FSP). According to this principle, the reduction of the intersection angle between fluid velocity and temperature gradient is the fundamental mechanism for enhancing convective heat transfer. According to FSP it can be seen that the most perfect case is the one where the intersection angle is zero in the entire flow domain (first deduction) and the worst case is that the flow velocity is everywhere normal to the local temperature gradient for which the fluid flow doesn't make any contribution to the convective heat transfer (second deduction).

Since the proposal of FSP by Guo et al. for the parabolic flow, the concept of the FSP was soon extended to the laminar elliptic convective heat transfer [6]. Numerical verification was conducted in [7] to demonstrate that this

☆ Communicated by P. Cheng and W.-Q. Tao.

* Corresponding author.
E-mail address: wqtao@mail.xjtu.edu.cn (W.-Q. Tao).

0735-1933/$ - see front matter © 2006 Elsevier Ltd. All rights reserved.
doi:10.1016/j.icheatmasstransfer.2006.11.008

Nomenclature

c_p	specific heat capacity, J kg^{-1} K^{-1}
H	width of square duct, m
L	length of square duct, m
m	mass flow rate, kg s^{-1}
Q	total heat transfer rate, W
Re	Reynolds number
T	temperature, K
u, v, w	velocity component in x, y and z direction, m s^{-1}
x, y, z	Cartesian coordinates, m

Greek symbols

θ	intersection angle between velocity and temperature gradient, degree
v	kinetic viscosity, m^2 s^{-1}
ρ	air density, kg m^{-3}

Subscripts

c	cold fluid
h	hot fluid
m	mean
wh	temperature difference of hot water
wc	temperature difference of cold water

principle can unify all the three mechanisms indicated above. A comprehensive review on the FSP can be found in [8]. The experimental results in [9,10] give a strong support to the first deduction of FSP. However, to the authors' knowledge, the second deduction from the FSP has never been experimentally verified. This situation stimulates the present authors to conduct the present experimental study.

The main purpose of the present paper is to provide experimental example for the validity of the second deduction. To be specific, we try to present an experimental demonstration that when the main flow direction is normal to the direction of heat flux in fluid, the magnitude of the flow velocity does not affect the heat transfer rate at all. Although such demonstration does not provide any information on how to enhance heat transfer, conceptually it is very important in two aspects: First the FSP is verified from the other side, and in conjunction with those verifications provided in the literatures reviewed below, the verification of FSP is thus completed; Second, it is the first time in the heat transfer literature showing that the fluid flow normal to heat flux does not contribute the heat transfer rate at all.

In the following the FSP is briefly reviewed and some mathematical expressions are provided.

2. Brief review on field synergy principle

For the reader's convenience and for the further discussion, the major idea of FSP is briefly reviewed here. Integrating the steady-state energy equation of single phase convective heat transfer over the domain Ω, we can obtain following expression of the convection term

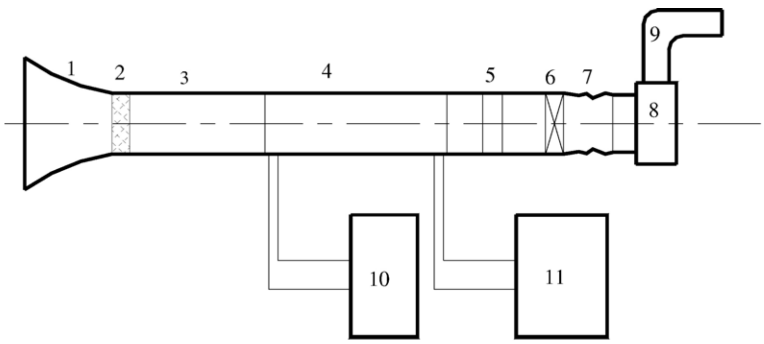
$$\iint_{\Omega} \rho c_{\mathrm{p}}(\vec{u} \cdot \nabla T)\mathrm{d}x\mathrm{d}y = \iint_{\Omega} \rho c_p|\vec{u}| \cdot |\nabla T|\cos\theta \mathrm{d}x\mathrm{d}y = \mathrm{FM} \tag{1}$$

where θ is the intersection angle between fluid velocity and temperature gradient. From Eq. (1), it is clear that under the same other conditions, decreasing the intersection angle θ will make FM larger, i.e. enhancing the convective heat transfer rate. Thus, it can be concluded that, in order to enhance convective heat transfer, we must reduce the intersection angle θ, i.e. improve the synergy between velocity and temperature gradient. This is the major idea of the field synergy principle. As can be seen from Eq.(1), if the velocity direction is everywhere normal to the local temperature gradient, the heat transferred by the fluid motion will be zero. In other words, in that case the fluid motion does not make any contribution to heat transfer (as stated above, this is the second deduction of FSP). Since the proposal of FSP, we have been considering to experimentally demonstrate the second deduction.

The design of such an experiment is of a great challenge. The major difficulty is to create such a flow field which is everywhere normal to fluid temperature gradient. Having stimulated from some numerical simulations, we decided to establish an axial flow within a square duct whose two lateral walls are maintained at constant but different temperatures, while the other two walls are adiabatic. In such case there will be global temperature gradient from the hot wall to the cold one, whose direction is normal to the axial flow. If the idea of the FSP is valid, then the heat transfer rate from the hot wall to the cold wall of the duct will only be dependent on the temperature difference of the two walls, but nothing to do with the axial flow velocity. In the implementation of the conceptual design, we met the second difficulty. In order to create a fluid temperature gradient which is always normal to the axial fluid flow, the two lateral walls have to be isothermal, otherwise an axial temperature gradient in fluid will exist, which violates the normal condition of the two vectors. This requirement discarded the possibility of using electrical heating for the hot wall. Thus for both hot wall and cold wall fluid heating or cooling have to be used. Then the second difficulty comes. In order to measure the heat transfer from the hot wall to the cold wall more accurately, we need a higher temperature difference between fluid inlet and outlet of each wall. However, from the isothermal requirement, this difference should be as small as possible. Finally we make some compromise between the measurement accuracy and the normal condition: the temperature difference of both the heating water and cooling water are allowed only about 1 °C, and this one degree difference is taken place in an axial direction as long as 2 m. In the execution of test one more difficulty (third difficulty) occurred, which will be described in the later presentation. In the following the specially designed test facility is presented.

3. Experimental apparatus

As mentioned above a special experimental system was designed, fabricated, and installed to demonstrate the second deduction of FSP. As shown in Fig. 1, the experimental apparatus is an open flow system with air as working fluid. It consists of three major components, (1) inlet and flow measuring section, (2) heat transfer test



1. Inlet
2. Filter
3. Developing section
4. Test section
5. Flow meter
6. Valve
7. Flexible duct
8. Centrifugal fan system
9. Discharge
10. Hot water thermostat reservoir
11. Cold water thermostat reservoir

Fig. 1. Schematic diagram of the experimental set up.

section and (3) fan assembly. The air is supplied by a centrifugal fan installed at the end of the set up. The test section is a square duct shown in Fig. 2(a). Fig. 2(b) shows a cross-sectional view of the heat exchanger. The square duct consists of two vertical aluminum walls and two horizontal PVC walls of small thermal conductivity. The dimension of the two vertical walls is 2000(mm) × 140(mm) × 1(mm). As it can be seen from Fig. 2(b), the two aluminum walls are bounded by two narrow vertical channels through which hot and cold water goes through respectively. The water temperature differences between inlet and outlet for each channel were mainly controlled within 1.0 °C, with the maximum temperature variation being less than 1.5 °C. Thus due to the high thermal conductivity of aluminum from which the walls are made, the two vertical walls can be regarded as practically isothermal [11]. The two horizontal walls of the square duct are well insulated and can be considered as adiabatic. The whole test section was well insulated.

The developing section (1000 mm long and 140 mm width) was designed to provided good streamlines flow and to avoid any flow disturbances to the upstream in the test section during experimental measurement.

Calibrated copper-constantan thermocouples were used to measure the air temperature at the inlet and exit of the test section. Nine thermocouples distributed uniformly in the cross section of the inlet were used to determine the average temperature of air of the cross section. The same measurement method was employed for the exit air temperature.

Hot water and cold water from two constant temperature reservoirs were pumped to the vertical channels of the heat exchanger, respectively. The hot water was heated by a controlled electrical resistance. And the cold water was cooled by a refrigeration system. The water mass flow rate was measured by volume method. The temperature difference of water between the inlet and exit of the two channels was measured by a thermopile consisted of five thermocouples.

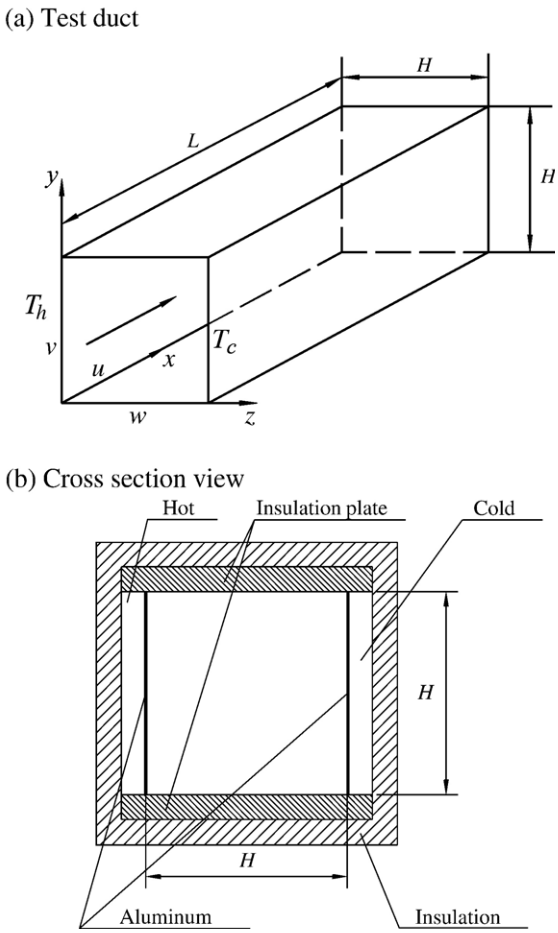


Fig. 2. Geometry of the square duct. (a) Test duct; (b) Cross section view.

From the above description of the test rig, 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. But the natural convection caused by the two side wall temperature difference leads to a weak secondary flow field in a thin thermal boundary layer along the solid wall. However in the major part of the duct the main stream flow velocity is of several orders of magnitude larger than the secondary flow in the cross section, the intersection angle between the combined fluid velocity and temperature gradient is almost everywhere perpendicular to each other, and hence according to the FSP, the mainstream velocity makes almost no contribution to the heat transfer from the hot wall to the cold wall and it is the natural convection in the cross section that contributes the convective heat transfer which is a relatively small amount and should not be affected by the axial flow velocity. In the preliminary test the third difficulty as mentioned above occurred. In the measurement of the heat transfer rate, we required that this amount of heat calculated from the heating water and from the cooling water should the same, or their deviation should be within about 5%. However, the required deviation was hardly satisfied, and often the heat transfer from the hot wall was larger than that of the cold wall. After examining many possible factors, we finally realized that apart from the natural heat transfer in the enclosure, there are heat transfers at the outside surface of the two vertical walls by both natural convection and radiation. When the absolute value of the temperature difference of the outside surface of the hot wall and cold wall are not equal, this additional heat transfer rate of the two vertical surfaces are not equal. The hot water within the channel of the hot wall actually released heat to two sinks: one to the environment, and the other to the moving air in the duct. On the other hand the cold water in the channel of cold wall absorbed heat from both environment and from the moving air in the duct. In order that the combined natural convection of the hot wall and the cold wall have the same amount of heat transfer, following conditions are required; (1) the mean temperature of the duct flow should be the same as the environment; (2) and the temperature of the hot wall and that of the cold wall should be apart from the environment temperature the same value but in opposite direction. Only when the test procedure is in such condition, the heat balance between the hot wall and cold wall is quite satisfactorily.

Steady-state was regarded being established when all temperature readings did not deviate over a 10–15 minutes period [12]. And all the test data, including thermocouples and thermopiles reading, flowmeter reading of air and the mass flow rate of water, were taken at steady state condition for each experimental run. Usually, a series of runs with different Reynolds number for a constant Rayleigh-number were taken by adjusting the flow rate successively.

4. Data reduction procedure

The main objective of data reduction was to obtain the average heat transfer rate of the two sides of heat exchanger. In each cross section the heat transferred from the hot surface to the cold surface by the natural convection. And it was expected that this heat transfer rate was basically determined by the temperature difference between the hot and cold surfaces, rather than by the main stream flow velocity. Thus the major parameters are the heat transfer rate and the air flow rate in axial direction. For the convenience of presentation a Reynolds number characterized by the streamwise velocity and the duct height was also calculated.

The total heat transfer rate at the side of hot wall is determined by

$$Q_{\mathrm{h}} = m_{\mathrm{h}} c_{\mathrm{p}} \Delta T_{\mathrm{wh}} \tag{2}$$

where m_{h} is the mass flow rate of hot water, c_{p} is the specific heat, and ΔT_{wh} is the temperature difference of hot water between the channel inlet and outlet. Similarly, the total heat transfer rate at the side of cold wall is given by

$$Q_{\mathrm{c}} = m_{\mathrm{c}} c_{\mathrm{p}} \Delta T_{\mathrm{wc}} \tag{3}$$

The mean heat transfer rate is determined by

$$Q_{\mathrm{m}} = 0.5(Q_{\mathrm{h}} + Q_{\mathrm{c}}) \tag{4}$$

The Reynolds number is defined as

$$Re = \frac{u_{\mathrm{m}} H}{v} \tag{5}$$

where u_{m} is the axial volumetric mean velocity, v is the kinetic viscosity, and H is the width of the square duct. The reference temperature of air was taken as the average of the temperatures of hot and cold walls.

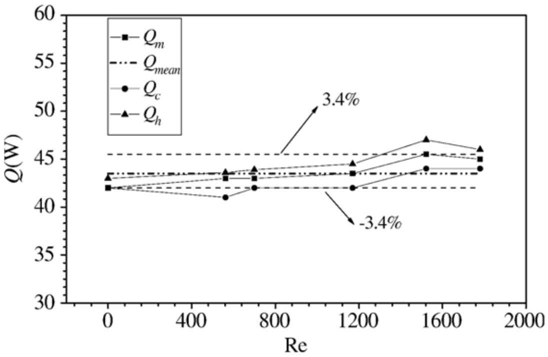


Fig. 3. Variation of the total heat transfer rate with Re (Δt=10 °C).

The uncertainty of the thermocouple used in the apparatus was 0.2 °C. And the experimental uncertainty of the air and water mass flow measurement was 2%. An uncertainty analysis of the reduced data was conducted along the line proposed in [13,14]. The uncertainty in Reynolds number is estimated about 3.3~7.7%, and that in heat transfer rate about 3.4~8.6%.

5. Results and discussion

Three levels of the wall temperature difference (10, 20 and 30 °C) were used in this study.

Figs. 3–5 give the variation of total heat transfer rate at three temperature difference levels, respectively, where Q_{mean} is the arithmetical mean value of Q_{m} for the variation range of flow rate. From the figures, following features may be noted. First, the total heat transfer rates at the three levels of the wall temperature differences are mainly independent on the streamwise flow rate. It is to be noted that the low end of the Reynolds number is actually zero. This implies that within a wide variation range of the streamwise velocity the heat transfer across the main stream is basically not affected. Second, at the three wall temperature difference levels the heat transfer rates are different, with the higher temperature difference corresponds to a higher heat transfer rate. Third, the energy balances between the hot wall and cold wall are generally good, with the deviation being increased with the increase in wall temperature difference. In all runs the energy balance may be regarded being within 4.0%. Fourth, in the two heat transfer rates from hot and cold walls, the heat transfer rate from the cold wall, Q_{c}, increases a bit more appreciably with the streamwise flow velocity while that from the hot wall is remained unchanged within the measurement uncertainty. This phenomenon will be analyzed in the following section.

As shown in Fig. 1, our test system is an open one. As mentioned above, the air temperature at the duct inlet was equal to the environment temperature. In the experiments the air inlet temperature was also equal to the average temperature between hot and cold walls. Thus to create the wall temperature difference, the hot wall temperature has to be higher than the environment while the cold wall temperature should be lower the ambient temperature. This caused some problem. When the cold wall temperature was lower than the dew point of the inlet air, some condensation occurred at the cold wall, which would increase the heat transferred to the cold water. With the increase in the streamwise flow velocity, the condensation film adhering at the cold wall could be more easily removed from the wall, thus enhancing the condensation heat transfer. The above discussion can well explain why $Q_{\mathrm{c}}>Q_{\mathrm{h}}$ in

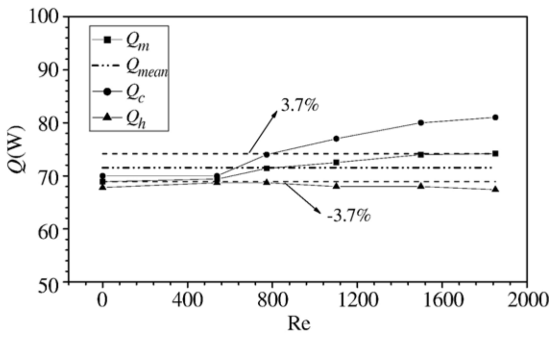


Fig. 4. Variation of the total heat transfer rate with Re (Δt=20 °C).

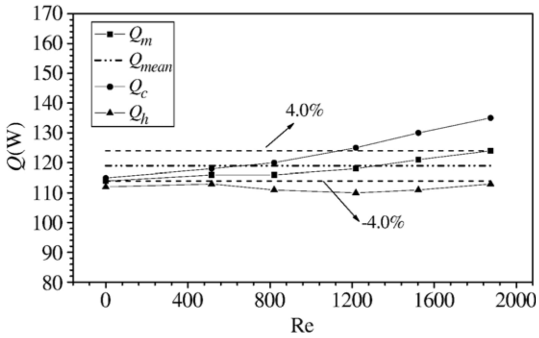


Fig. 5. Variation of the total heat transfer rate with Re (Δt=30 °C).

Figs. 4 and 5, and why Q_c increases with Re. As mentioned above, for the open flow system we can not eliminate this weakness by simply increasing the cold wall temperature, otherwise the heat balance will be severely violated. Fortunately this weakness does not affect the major purpose of this experimental study: to demonstrate experimentally that when the main flow velocity is normal to the heat flux direction, the flow velocity does not make any contribution to the heat transfer rate.

6. Conclusion

With a specially designed test apparatus, the heat transfer is investigated experimentally in a square duct with an imposed horizontal temperature difference across the main flow direction and perfectly insulated top and bottom walls. Test results show that the heat transfer rate between the vertical hot and cold walls only depends on the temperature difference across the section, and it is basically not affected by the streamwise flow velocity in the square duct. This result verifies the second deduction of the FSP, which means that if a flow is normal to a temperature gradient the flow doesn't make any contribution to the heat transfer occurring in the temperature gradient direction.

Discussion is provided to account for some minor deviation from the expected results of FSP. Detailed description of the difficulties met in such experimental study was presented and the minor deviation comes from these measurement difficulties. It is also pointed that this weakness does not affect the major purpose of this experimental study

Acknowledgements

This work is supported by the National Key Fundamental Project of R & D of China (Grant No. 2007CB206902), National Natural Science Foundation of China (Grant No.50425620), and the Research Foundation of Doctorial Project of Ministry of Education of China (RFDP 20030698015).

References

[1] R.L. Webb, Principles of Enhanced Heat Transfer, John Wiley & Sons Inc., New York, 1994, p. 89.
[2] Y.A. Cengel, Heat Transfer, A Practical Approach, WCB McGraw-Hill, Boston, 1998, p. 378.
[3] L.B. Wang, G.D. Jiang, W.Q. Tao, H. Ozoe, Numerical simulation on heat transfer and fluid flow characteristics of arrays with non-uniform plate length positioned obliquely to the flow direction, ASME Journal of Heat Transfer 120 (1998) 991–998.
[4] B. Yu, J.H. Nie, W.Q. Tao, Experimental study on the pressure drop and heat transfer characteristics of tube with internal wave-like longitudinal fins, Heat and Mass Transfer 35 (1999) 65–73.
[5] Z.Y. Guo, D.Y. Li, B.X. Wang, A novel concept for convective heat transfer enhancement, International Journal of Heat and Mass Transfer 41 (2002) 2221–2225.
[6] W.Q. Tao, Z.Y. Guo, B.X. Wang, Field synergy principle for enhancing convective heat transfer—its extension and numerical verification, International Journal of Heat and Mass Transfer 45 (2002) 3849–3856.
[7] W.Q. Tao, Y.L. He, Q.W. Wang, Z.G. Qu, F.Q. Song, A unified analysis on enhancing convective heat transfer with field synergy principle, International Journal of Heat and Mass Transfer 45 (2002) 4871–4879.
[8] Z.Y. Guo, W.Q. Tao, R.K. Shah, The field synergy (coordination) principle and its applications in enhancing single phase convective heat transfer, International Journal of Heat and Mass Transfer 48 (2005) 1797–1807.

[9] H. Wang, M.H. Shi, Experimental investigation of forced convection heat transfer in centrifugal fluidized bed, Journal of Chemical Industry and Engineering (China) 54 (5) (2003) 596–600.
[10] T.S. Zhao, Y.J. Song, Forced convection in a porous medium heated by permeable wall perpendicular to flow direction: analyses and measurements, International Journal of Heat and Mass Transfer 44 (2001) 1031–1037.
[11] S.L. Braga, F.E.M. Saboya, Turbulent heat transfer and pressure drop in an internally finned equilateral triangular duct, Experimental Thermal and Fluid Science 12 (1996) 57–64.
[12] Mafizul Huq, A.M. Aziz-ul Huq, Experimental measurements of heat transfer in an internally finned tube, International Communications in Heat and Mass Transfer 25 (5) (1998) 619–630.
[13] S.J. Kline, F.A. McClintock, Describing uncertainties in single sample experiments, Mechanical Engineering 75 (1953) 3–8.
[14] S.J. Kline, The purpose of uncertainty analysis, ASME Journal of Fluids Engineering 117 (1985) 153–160.