Available online at www.sciencedirect.com

ScienceDirect

International Journal of Heat and Mass Transfer 51 (2008) 3683–3692

International Journal of HEAT and MASS TRANSFER

www.elsevier.com/locate/ijhmt

Numerical study on laminar convection heat transfer in a channel with longitudinal vortex generator. Part B: Parametric study of major influence factors

J.M. Wu [a,b], W.Q. Tao [a,*]

[a] State Key Laboratory of Multiphase Flow in Power Engineering, School of Energy and Power Engineering, Xi'an Jiaotong University, Xi'an City 710049, China

[b] School of Environment and Chemical Engineering, Xi'an Polytechnic University, Xi'an City 710048, China

Received 12 October 2006; received in revised form 17 March 2007
Available online 24 May 2008

Abstract

This paper presents the influences of main parameters of longitudinal vortex generator (LVG) on the heat transfer enhancement and flow resistance in a rectangular channel. The parameters include the location of LVG in the channel, geometric sizes and shape of LVG. Numerical results show that the overall Nusselt number of channel will decrease with the LVGs' location away from the inlet of the channel, and decrease too with the space between the LVG pair decreased. The location of LVG has no significant influence on the total pressure drop of channel. With the area of LVG increased, the average Nusselt number and the flow loss penalty of channel, especially when $\beta = 45^\circ$ will increase. With the area of LVG fixed, increasing the length of rectangular winglet pair vortex generator will bring about more heat transfer enhancement and less flow loss increase than that increasing the height of rectangular winglet pair vortex generator. With the same area of LVG, delta winglet pair is more effective than rectangular winglet pair on heat transfer enhancement of channel, and delta winglet pair-b is more effective than delta winglet pair-a. Delta winglet pair-a results in a higher pressure drop, the next is rectangular winglet pair and the last is delta winglet-b. The increase of heat transfer enhancement is always accompanied with the decrease of field synergy angle between the velocity and temperature gradient when the parameters of LVG are changed. This confirms again that the field synergy is the fundamental mechanism of heat transfer by longitudinal vortex. The laminar heat transfer of the channel with punched delta winglet pair is experimentally and numerically studied in the present paper. The numerical result for the average heat transfer coefficient of the channel agrees well with the experimental result, indicating the reliability of the present numerical predictions.
© 2007 Published by Elsevier Ltd.

Keywords: Vortex generator; Heat transfer enhancement; Flow loss; Field synergy principle

1. Introduction

In the companion paper [1] of the present article, the longitudinal vortex generator was introduced and the major related studies [2–14] were reviewed in detail. Thus, for the simplicity of presentation, the present article will not repeat such contents. From the review of [1], it is concluded that even though a large amount of researches, both numerical and experimental, have been conducted, following two aspects need further study for a deeper understanding of the LVG performance and the essence of heat transfer enhancement by LVG. First, in engineering practice the LVG is punched from the base sheet with a finite thickness, hence, there is a corresponding hole under the LVG. However, in most of the existing literatures, either the LVG thickness or the hole was not the taken into account. And it is also very limited for such parametric

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

0017-9310/$ - see front matter © 2007 Published by Elsevier Ltd.
doi:10.1016/j.ijheatmasstransfer.2007.03.031

Nomenclature

a	transverse space between the winglet pair defined in Fig. 1 (m)
b	thickness of vortex generator (m)
B	width of channel (m)
h	height of vortex generator (m)
H	height of channel (m)
f	fanning frictional factor
l	chord length of vortex generator (m)
Nu	Nusselt number
s	streamwise coordinate of LVG defined in Fig. 1 (m)

Greek symbol

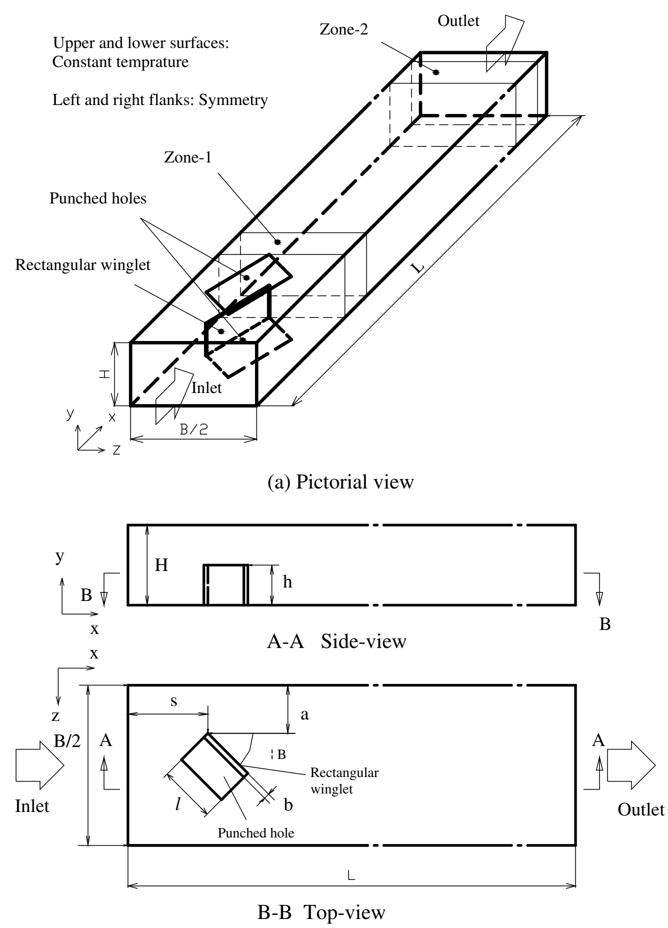
β	attack angle (°)

Subscripts

m	average
0	channel without vortex generator

study, which took both the LVG thickness and the hole into account. Second, the heat transfer enhancement by LVG was usually explained by the traditional enhancement mechanisms, including the re-development of a boundary layer, the mixing caused by swirling of the LVG and the LVG-induced disturbances. From our preliminary study shown in [1], the fundamental mechanism of heat transfer enhancement by the LVG is the improvement of the synergy between velocity and temperature gradient. The major purpose of this paper is to present our numerical results of a parametric study for the influences of the major parameters and reveal the fundamental mechanism of the LVG enhancement. To describe the performance of LVG, we take the comparison of the heat transfer and friction factor results between the duct with LVG and that without LVG at the same other conditions. It should be noted that there are a number of comparison criteria for the enhanced heat transfer surfaces as can be found in [16]. For such comprehensive comparison it needs another full paper, and is not the task of the present study. It may be useful to indicate that the field synergy principle is a new idea or concept of the heat transfer enhancement mechanism, it is not the comparison criterion for which a number of comparison criteria can be found in [15]. From the results presented later, it can be found that any heat transfer enhancement by the LVG is always accompanied by a better synergy between velocity and temperature gradient, thus un-doubtfully demonstrated that the fundamental reason for LVG enhancement is in the improvement of the field synergy.

For the readers' convenience, the computed channel is shown in Fig. 1. The governing equations and boundary conditions have been described in detail in [14], and for the simplicity of presentation they are not shown in this paper. The numerical results will be presented in detail in this paper. The mechanism of heat transfer by LVG will be discussed from the field synergy principle, and the domain average synergy angle will be used to show the degree of synergy between velocity and temperature gradient. A brief introduction to the field synergy principle has been provided in the companion paper [14], and for the simplicity of presentation, it will not be re-stated here. In the following, the numerical predicted effects of the LVG's location, size and shape will first be presented, followed by a comparison between experimental measurement and numerical prediction. Finally, some conclusions will be drawn.



(b) Plane view of the RWLVG in channel

Fig. 1. Rectangular channel with a pair of LVG.

2. Effects of the RWLVG's location on heat transfer enhancement, field synergy and flow loss

As we know, LV mainly influences the heat transfer in the downstream of the RWLVG. Thus, we can be sure that the location of the RWLVG in the channel will influence the global heat transfer for a given channel. In this section, the influences of the location of RWLVG pair on the heat transfer enhancement and the synergy angle of the channel is numerically computed. The location of the RWLVG is described by the parameters of a and s (see Fig. 1). For a convenient presentation the relative values of the ratios of s to H and a to H are adopted. In the following computations Reynolds number of channel is set to 1600.

When $a/H = 0.5$, the variations of average Nusselt number and average synergy angle in the whole field vs s/H are shown in Fig. 2. 1 It shows that when the RWLVG pair is moved away from the inlet of channel (increasing the value of s), the channel average synergy angle will increase and the heat transfer enhancement will decrease. Because with the RWLVG moving to the channel inlet, the influence range of the LV is getting larger in streamwise direction, the synergy between the velocity and the temperature fields is improved in larger region. As a result, the average synergy angle in entire field will decrease and the heat transfer will be enhanced.

When $s/H = 4.0$, the variations of average Nusselt number and average synergy angle in the whole field vs a/H are shown in Fig. 3. With decreasing the transverse space between the RWLVG pair (decreasing the value of a), average synergy angle increases and heat transfer enhancement decreases. Especially, when $a/H = 0.1$, the average Nusselt number for the case of $\beta = 45°$ will drop to the value for the case of $\beta = 30°$ at the same condition. The reason is that the LV pair generated by RWLVG pair in the channel rotates in the opposite directions. With decreasing the transverse space of a, the interaction between the LV pair in their inner side is getting stronger. As a result, each vortex will be weakened by the other, hence, heat transfer is deteriorated and the synergy between velocity and temperature becomes worse. Fig. 4 shows the comparison of the

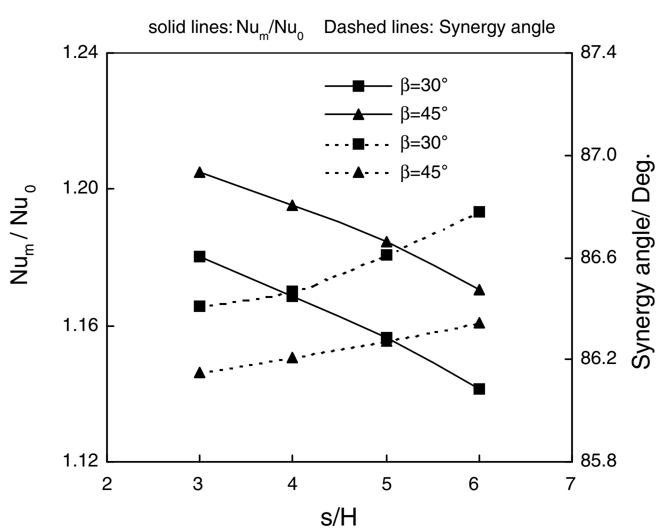


Fig. 2. The influence of streamwise location of RWLVG on the heat transfer enhancement and field synergy of channel ($Re = 1600$).

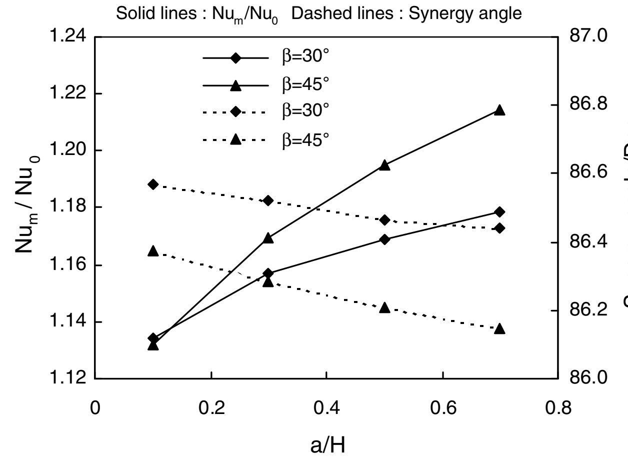
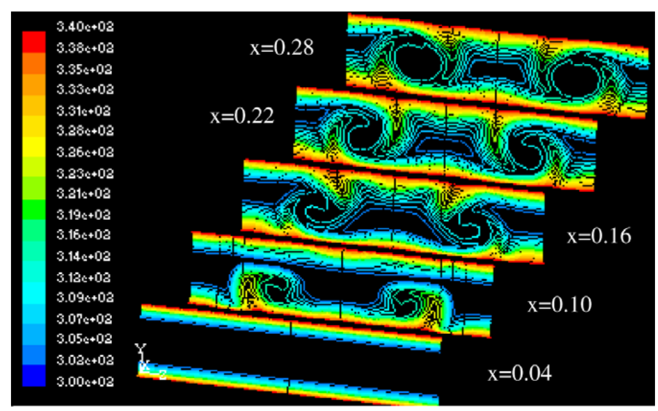
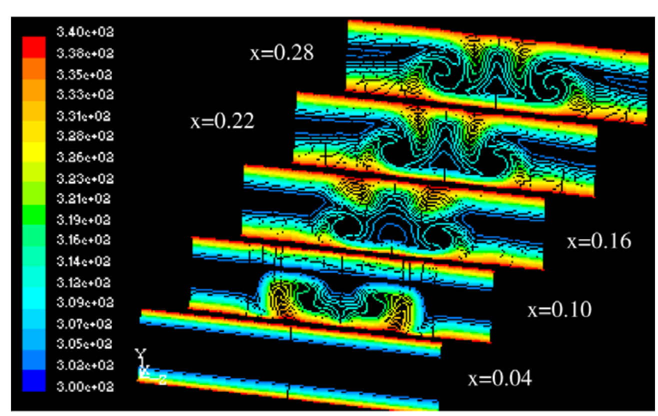


Fig. 3. Influence of the transverse space between the RWLVG pair on heat transfer enhancement and field synergy.



(a) $a/H = 0.5$



(b) $a/H = 0.1$

Fig. 4. Comparison of temperature profiles at different cross section ($\beta = 45°$).

temperature profiles at the different cross sections under the conditions of $\beta = 45°$ between the cases of $a/H = 0.5$ with $a/H = 0.1$. We can clearly observe that only the central region of the channel is disturbed by LVs for the case of $a/H = 0.1$, and temperature profiles at the both side regions of the channel are still layered. The influence range by LVs for the case of $a/H = 0.5$ is wider than that for the case of $a/H = 0.1$, and of course the convection heat transfer in the former case is enhanced more than in the later case. Therefore, to a given width of the channel, the transverse space between the RWLVG pair should not be too small so that the two LVs generated will not be interacted each other and the spanwise LV's influences may reach a wider distance.

As far as the pressure drop is concerned, numerical results revealed that for a given channel with the same LVG, the value of s almost has no effect on the total flow loss of the channel, while decreasing the value of a, will bring about a slight increase of total flow loss of channel because of the stronger interaction between the LV pair.

3. Effects of the geometric sizes of RWLVG on heat transfer enhancement, field synergy and flow loss

As an example, the rectangular winglet pair is still used as the longitudinal vortex generator. Due to that the rectangular winglet pair is punched out from the fin wall, the thickness of RWLVG is taken as constant. Therefore, under the condition of a given attack angle, geometric sizes of RWLVG include its chord length of l and height of h. The location of RWLVG is given as: $a/H = 0.5$ and $s/H = 4.0$, Reynolds number is still 1600. As we know, the size of h will decide the shielded part of the flow cross section in the height direction. Thus, a dimensionless ratio of h to H is employed to express the relative height of RWLVG. Similarly, the shielded part of the flow cross section in the spanwise direction is decided by the size of chord length l, therefore, another dimensionless ratio of l to $B/2$ is used to express the relative chord length of the RWLVG.

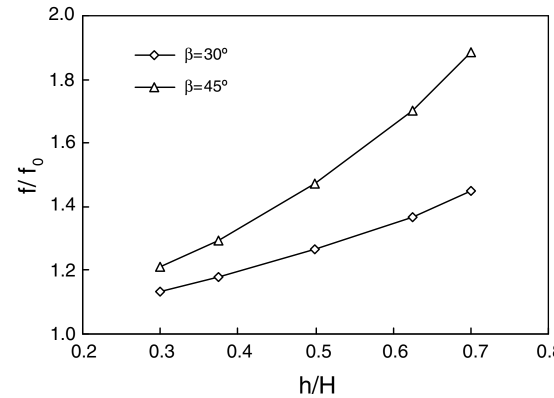


Fig. 6. Influence of the height of RWLVG on pressure loss.

When $l/0.5B = 0.5$, the influences of the height of the RWLVG on the heat transfer enhancement, average synergy angle and increase of pressure loss in the channel are shown in Figs. 5 and 6. Fig. 5 shows that with increasing the height of the RWLVG, the synergy angle decreases, and the heat transfer enhancement increases. As we know, the cross-sectional flow area of the channel decreases when h is increased, and the fluid velocity at the side edge of RWLVG increases at the same volume flow rate. Therefore, the strength of the generated LVs increases, thus improving the heat transfer performance. However, the associated penalty of pressure drop is also increased greatly with increasing h, especially, for the case of $\beta = 45°$. For example, when $h/H = 0.7$, the average friction factor of the channel with RWLVG in condition of $\beta = 45°$ will increase by 88.7% compared with the plain channel without LVG, which is about 28.3 % higher than that of $h/H = 0.5$ (see Fig. 6). Therefore, the suitable height of RWLVG pair in the channel should be $h/H = 0.5$.

When $h/H = 0.5$, the effects of the chord length of the RWLVG on the heat transfer enhancement, synergy angle

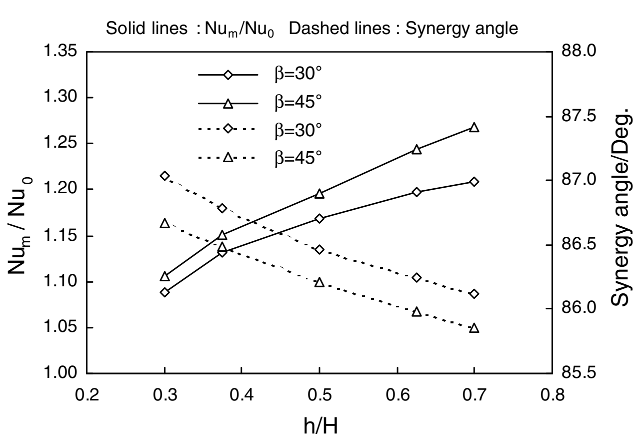


Fig. 5. Influence of the height of RWLVG on heat transfer enhancement and synergy angle.

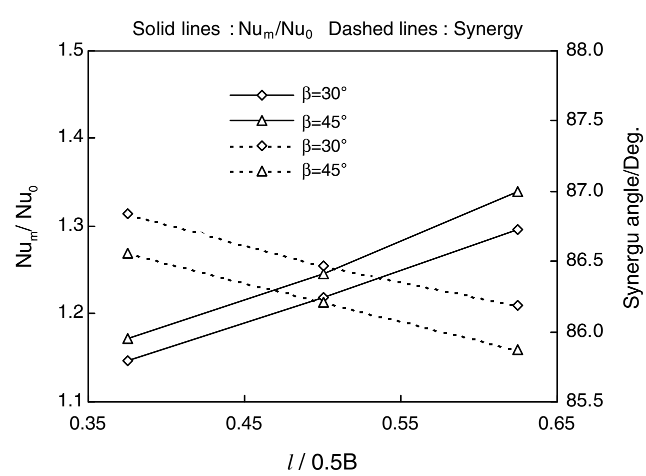


Fig. 7. Effect of length of LVG on heat transfer enhancement and field synergy.

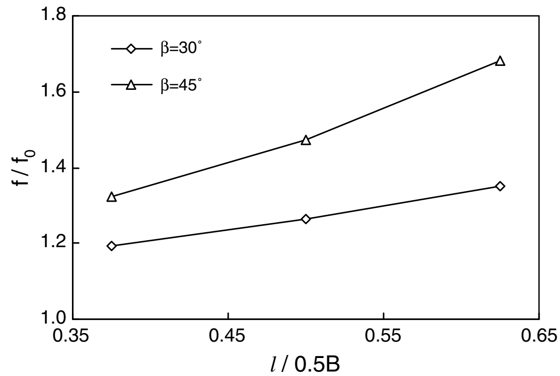


Fig. 8. Effect of length of LVG on pressure loss.

and penalty of pressure loss are shown in Figs. 7 and 8. The LVs are generated when fluid flows over the side-edge of the RWLVG because of inertia and boundary layer separation. The number of generated LVs will increase with increasing the chord length of l. This leads to two positive results. One is that the total strength of the LV will increase due to the superposition of the LVs, the other is that the transverse influence range of LVs is getting wider. Therefore, the field synergy between the velocity and temperature fields will be better , and the heat transfer enhancement is improved. Meanwhile, with the increasing the chord length, the area of the RWLVG also increases. This leads to an increase of the form drag of RWLVG and more pressure drop of the channel. However, the increase rate of pressure drop with increasing l for the case of $\beta = 30°$ is much lower than that for the case of $\beta = 45°$ (see Fig. 8). So, it is feasible to enhance the heat transfer by properly increasing the chord length of the RWLVG for the case of $\beta = 30°$.

From the above results, we can know that the increase of the area of RWLVG will bring about more heat transfer enhancement either by increasing the height or length of LVG. To investigate which one is more effective to heat transfer enhancement, the computed results for the cases with the same area but different height and length of RWLVG pair are compared in Table 1. The area of the RWLVG for cases 1 and 2 is the same (150 mm^2), the length of RWLVG for case 1 is larger than that for case 2, and of course the height of RWLVG for case 1 is smaller than that for case 2. The area of the RWLVG for cases 3 and 4 is the same (250 mm^2), the length of RWLVG for case 3 is larger than that for case 4, and the height of RWLVG for case 3 is smaller than that for case 4. The attack angle of RWLVG pair for every case can be 30° or 45°. From the data in Table 1, we can find that the RWLVG pair with a larger length and smaller height can cause the synergy angle of the channel to be smaller, heat transfer enhancement to be more and the pressure drop increase to be smaller. Therefore, when the area of RWLVG is given, properly increasing the length of RWLVG and decreasing its height is a good idea to get more heat transfer enhancement and to prevent flow loss from increasing too fast.

Table 1
Comparison between the influences of the length and the height of the RWLVG

Case	RWLVG sizes (mm): $l \times h$	Area (mm^2)	Attack angle (°)	Nu_m/Nu_0	Synergy angle (°)	f/f_0
1	20 × 7.5	150	30	1.132	86.78	1.180
			45	1.150	86.51	1.294
2	15 × 10		30	1.098	86.84	1.191
			45	1.122	86.56	1.322
3	20 × 12.5	250	30	1.198	86.24	1.368
			45	1.243	85.99	1.700
4	25 × 10		30	1.247	86.19	1.350
			45	1.289	85.87	1.681

4. Effects of the LVG's shape on heat transfer enhancement, field synergy and flow loss

Because the influence range in the transverse direction (called transverse influence range hereafter) of the LV generated by winglet pair type vortex generator is larger than that of LV generated by wing type vortex generator, so the winglet pair vortex generator is more acceptable in the practical application. The laminar flows and heat transfer in the channels with two kinds of delta winglet vortex generators (DWLVG) (called DWLVG-a and DWLVG-b) were numerically calculated, respectively, and the results are compared with those of the channel with RWLVG presented in the above sections. For the comparisons being meaningful, the following conditions are taken. The geometric sizes of the channels for the three cases are kept the same as the one in the above computations. The height of DWLVG-b is half of channel height, which is same as that of RWLVG, and the height of DWLVG-a is the same as the channel height. The locations of RWLVG and DWLVG-a in the channels are $s/H = 4.0$, and $a/H = 0.5$. The location DWLVG-b in the channel is $s/H = 4.0$, and $a/H = 0.4$. Taking $a/H = 0.4$ in the channel with DWLVG-b is to have enough area to punch the DWLVG out of the fin wall. The numerical predictions in Section 2 show that this will not bring much effect on the result. The areas of RWLVG, DWLVG-a and DWLVG-b are the same, 200 mm^2. The attack angle of the three LVGs is set to 30°.

The comparisons of heat transfer enhancement and field synergy between the three cases are shown in Fig. 9. It is found that for different Reynolds numbers the heat transfer is enhanced more by DWLVG than by RWLVG. The average synergy angles between the velocity and temperature gradient for the cases with DWLVG are smaller than that for the case with RWLVG. Moreover, the DWLVG-b is more advantageous for heat transfer enhancement than the DWLVG-a under higher Reynolds number condition. These can be explained as follows. The LVs will be generated at the side edge RWLVG due to the inertia and boundary layer separation of rectangular winglet, but for

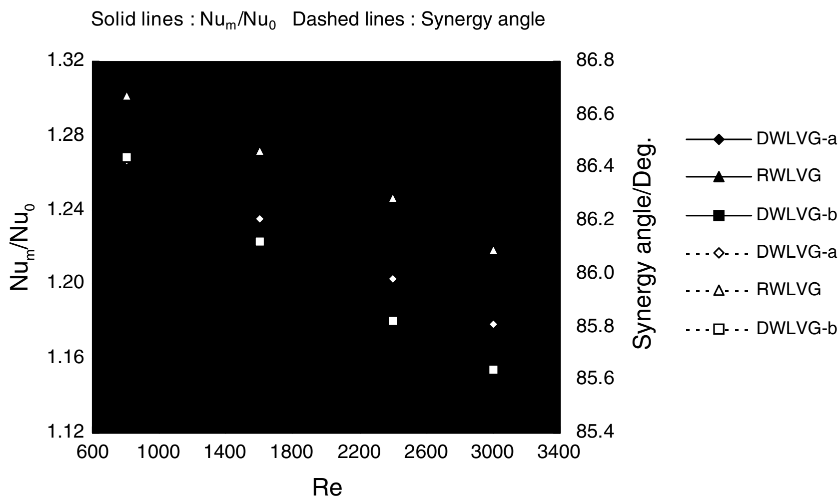


Fig. 9. Influence of LVG's shape on heat transfer enhancement and field synergy.

delta winglet LVG, however, the LVs are generated at the leading edge. Under the condition of same area, the longer the side edge (the long side of the rectangular winglet) or the leading edge(the bevel side of the triangle winglet) the more the LVs generated and the stronger the LVs' strength, and of course, the stronger the disturbance of LVs to the flow, therefore, the more the heat transfer enhancement. The length of the leading edge of DWLVG-a is longer than that of the side edge of RWLVG, and the length of the leading edge of DWLVG-b is longer than that of DWLVG-a. Thus, DWLVG-b is more advantageous than DWLVG-a, and DWLVG-a is superior to the RWLVG for enhancing heat transfer.

Fig. 10 shows the differences of the flow loss between the three channels with different kinds of LVGs. From Fig. 10, we can find that the pressure loss for the channel with DWLVG-a is the largest. This is because the height of DWLVG-a is equal to that of the channel so that the shielding of DWLVG-a to the cross sectional flow area in the vertical direction is severe. The flow loss for the channel with DWLVG-b is the smallest due to its small height. Although the height of DWLVG-b is the same as that of RWLVG, which is half of the channel height, its influence to the flow is mild, not so abrupt. Combining the effects of the LVG's shape to heat transfer enhancement and flow loss, DWLVG-b may be taken as a kind of LVG with better performance. Again the conclusion that properly increasing the length of LVG and decreasing the height of can obtain more heat transfer enhancement and prevent flow loss from increasing too fast is verified.

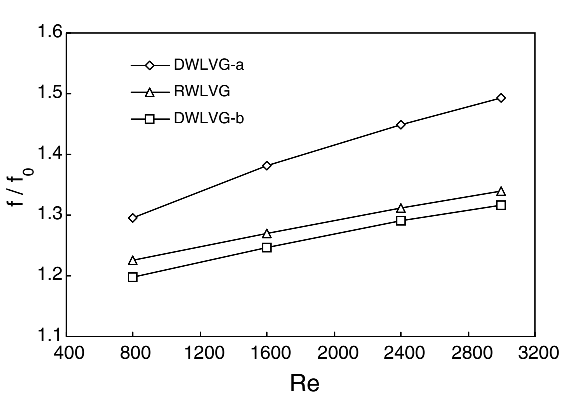


Fig. 10. Influence of LVG's shape on pressure loss.

5. Comparisons with experiment

5.1. Experimental apparatus

The test was conducted in a small tunnel shown in Fig. 11. The wall of the tunnel was made of organic glass with thickness of 10 mm. The cross section of the test section is 162 mm in width and 62 mm in height. A compound aluminum plate is mounted in the middle of the test section by two small bakelite supporting posts as shown in Fig. 12. So actually the test section is composed of two channels through which air flows over the two surfaces of the compound aluminum plate. The average convection heat transfer on the up and down surfaces of the compound aluminum plate was tested. The compound aluminum plate with a pair of punched delta winglets, as shown in Fig. 13, was made of two layers of aluminum plate with thickness of 1 mm and a thin electric heating film (0.15 mm) used to heat air through the aluminum plates. The sizes of the aluminum plates and the position of delta winglet are shown in Fig. 14. To avoid the heat transfer through the walls of the test section, the out surface of the walls are thermally isolated with the foam plates of thickness 100 mm.

The inlet temperature measuring mesh contains 8 thermocouples, and the outlet temperature measuring mesh contains 16 thermocouples. The surface temperature of the up and down aluminum plates was measured by 8 ther-

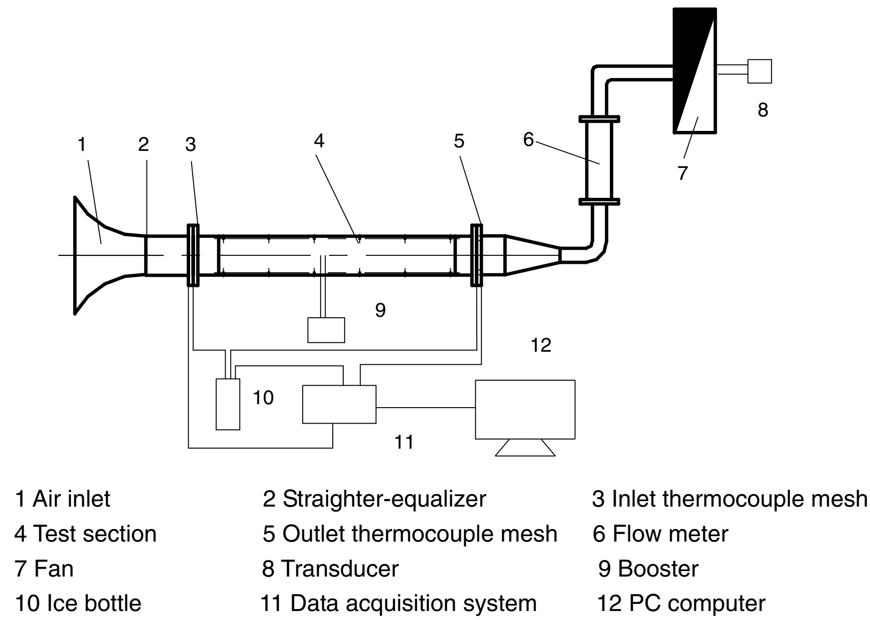


Fig. 11. Schematic diagram of test facility.

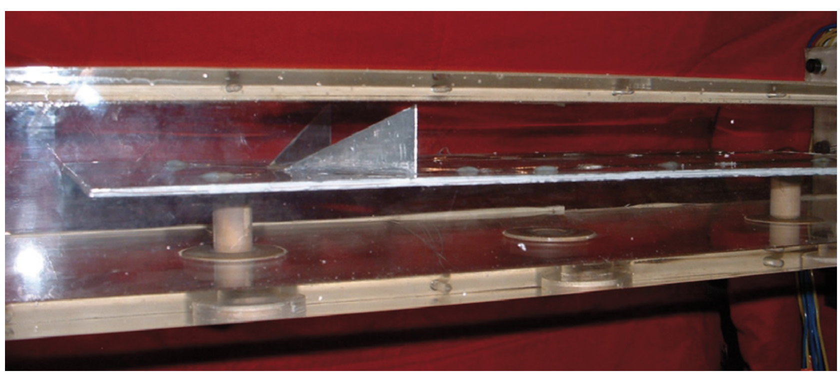


Fig. 12. Structure of the test section.

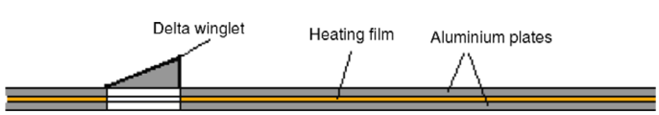


Fig. 13. Structure of the compound aluminum plate.

mocouples, respectively. The measurement uncertainty of these thermocouples was within ±0.2 °C. The air volumetric flow rate was metered by a rotor flow meter with a relative error of 2.5%. To keep the flow in the channels in laminar state, the air volumetric flow rate was controlled within 9–33 $m^3 h^{-1}$, which corresponds to the frontal velocities from 0.25 to 1 m/s. The power of the heating film is adjusted by a booster and controlled within 35 ± 1 W. The energy un-balance between the heat gained by air and heating power was within 5%. The uncertainty in the reported experimental value of the surface convection heat transfer coefficient was estimated by the method suggested by Moffat [16], and is less 5% in the present experimental conditions.

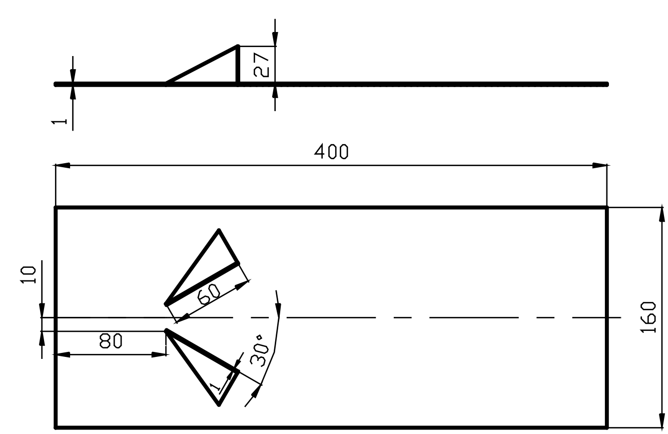


Fig. 14. Geometric parameters of aluminum plate (example of $\beta = 30°$).

5.2. Experimental results and comparison

The surface average convection heat transfer for the five compound aluminum plates, including the plain plate with-

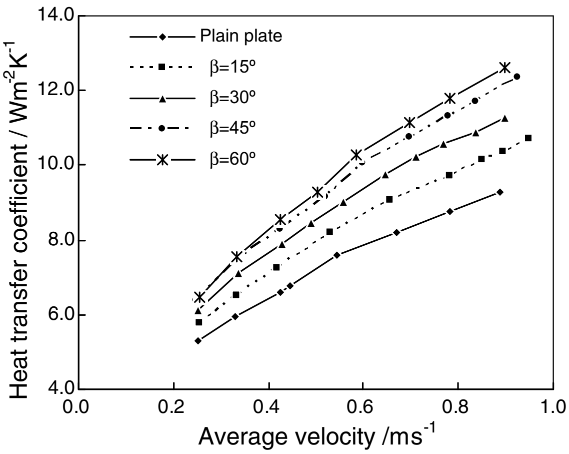


Fig. 15. Experimental results of heat transfer coefficient vs average velocity.

out LVG and the ones with punched delta winglet pair at four different attack angles, in a channel flow were tested. Fig. 15 shows the experimental results. In comparison with the plain plate, the heat transfer coefficient of the plate with delta winglet pair at $\beta = 15°$ is increased by 8–11%, those at $\beta = 30°$, 45°, 60° is increased by15–20%, 21–29%, 21–34%, respectively. The heat transfer enhancement is increased with the increase of the average velocity in the channel. We can find that the heat transfer coefficient for the case of $\beta = 45°$ is just slightly higher than that of $\beta = 60°$. The result is coincident with the one given by Fiebig [7] for delta winglet pair, even though the tested channel in our experiment has some differences with the one by Fiebig [7].

To validate the present models and methods used in the present numerical solutions, the heat transfer in the tested channel is numerically simulated. Due to the symmetry, only half of the tested channel is computed as shown in Fig. 16. As a bell-type inlet and a straighter–equalizer were adopted in the test tunnel (see Fig. 11), a uniform inlet boundary condition is employed. To use the developed boundary condition at outlet, the computed domain is extended 20 times the height of channel in downstream. The walls of tested section are taken as adiabatic. The surfaces of two bakelite supporting post are taken as adiabatic walls too. The heating film is treated as solid with inner heat source. Aluminum plates including delta winglet pair are treated as solid conjugated with air in the computation. Fig. 17 shows the comparisons between the numerical and experimental results for the five cases. Deviations between tested and simulated results of heat transfer for every case is less than 10%. The agreement between the numerical and experimental results proves that the models and methods used in the present study are feasible and the numerical results are reliable.

6. Conclusions

Extending our study of part A [14], the influences of main parameters of longitudinal vortex generator (LVG) on heat transfer enhancement and flow resistance are numerically computed and analyzed in the present paper. The major findings are as follows:

(1) The increase of heat transfer enhancement is always companied by the decrease of field synergy angle between the velocity and temperature gradient when the parameters of LVG are changed. This confirms again that the field synergy is the fundamental mechanism of heat transfer by longitudinal vortex.
(2) The heat transfer enhancement of channel will decrease with the increase of LVGs' location from the channel inlet, and with the decrease of the transverse space between the LVG pair .The location of LVG has no significant influence on the flow loss of channel.

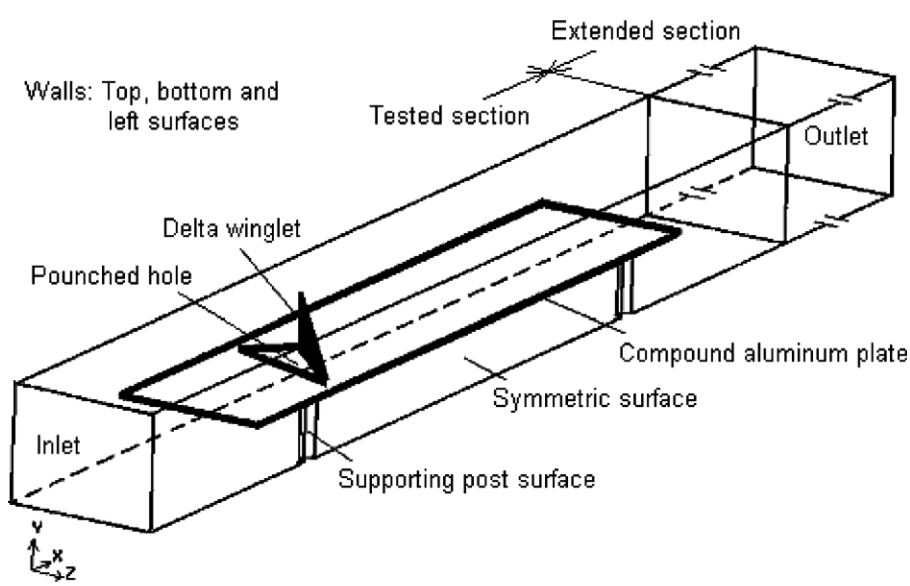


Fig. 16. The schematic diagram of computed domain.

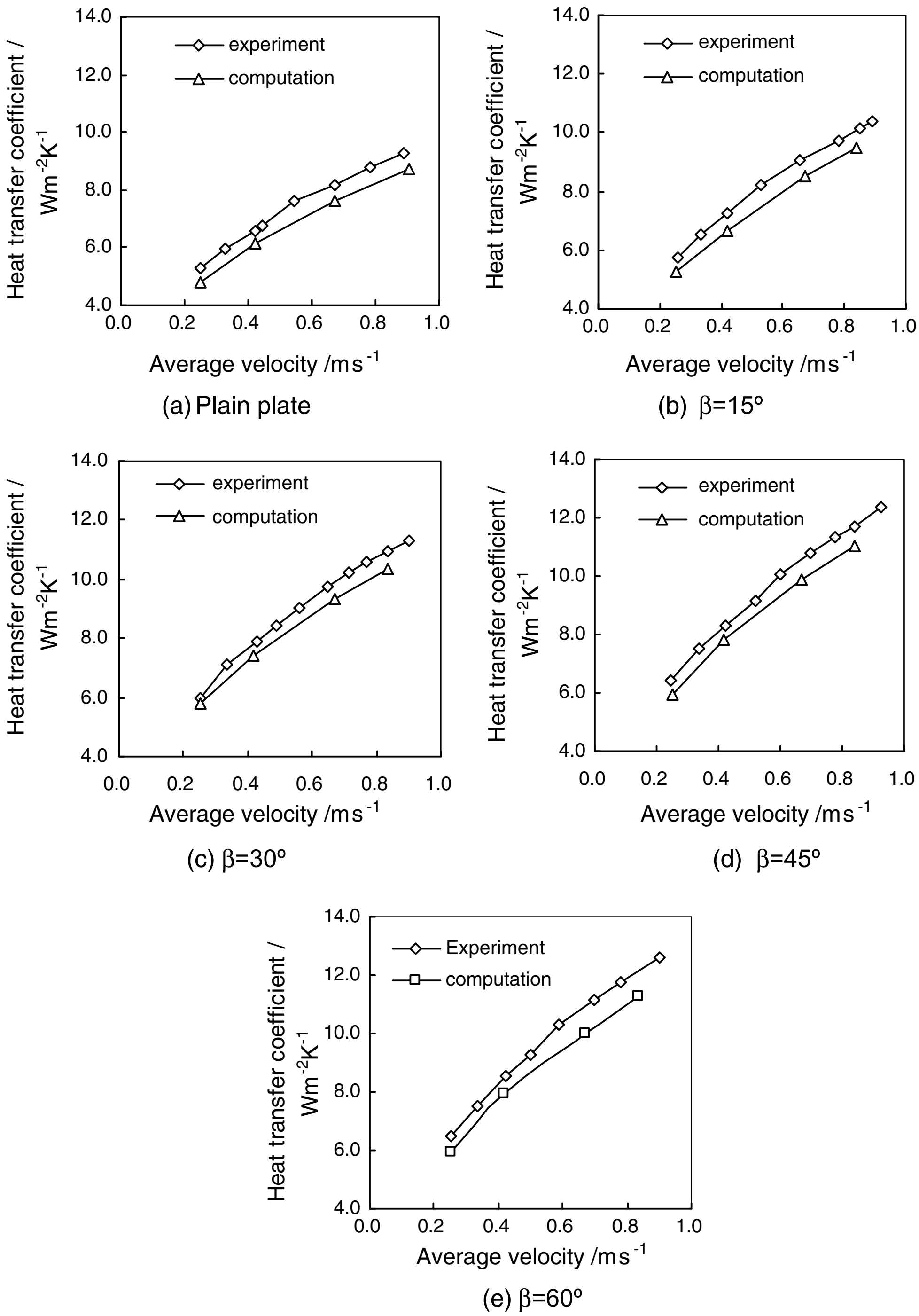


Fig. 17. Comparisons of the numerical and experimental results.

(3) With the increasing area of LVG, both the heat transfer enhancement of the channel and the pressure loss will increase. At a fixed area of LVG, properly increasing the length of RWLVG and decreasing the height of RWLVG is a good idea to get more heat transfer enhancement and to prevent from a significant flow loss increase.

(4) With the same area of LVG, DWLVG is more effective than RWLVG on the channel heat transfer enhancement, and DWLVG-b is more effective than DWLVG-a. DWLVG-b leads to less pressure loss in comparison with DWLVG-a and EWLVG. DWLVG-b may be regarded as the best one among LVGs compared.

(5) The surface average convection heat transfer for the five compound aluminum plates, including the plain plate without LVG and the ones with punched delta winglet pair at four different attack angles, in a channel flow were tested as well as simulated. Deviations of heat transfer between the numerical and experi-

mental results for every case is less than 10%. The agreement between the numerical and experimental results proves that the models and methods used in the present study are feasible and the numerical results are reliable.

Acknowledgements

Supports from the National Basic Research of China (973 Program) (2007CB 206902), and the National Natural Science Foundation of China (No. 50476046) are greatly appreciated.

References

[1] J.M. Wu, W.Q. Tao, Numerical study on laminar convection heat transfer in a rectangular channel with longitudinal vortex generator, Part A: verification of field synergy principle, Int. J. Heat Mass Transfer 51 (2008) 1179–1191.

[2] M. Fiebig, U. Brockmeier, N.K. Mitra, T. Guntermann, Structure of velocity and temperature fields in laminar channel flows with vortex generators, Numer. Heat Transfer, Part A 15 (1989) 281–302.

[3] G. Biswas, H. Chattopadhyay, Heat transfer in a channel with built-in wing type vortex generators, Int. J. Heat Mass Transfer 35 (1992) 803–914.

[4] M. Fiebig, P. Kallweit, N. Mitra, S. Tiggelbeck, Heat transfer enhancement and drag by longitudinal vortex generators in channel flow, Exp. Therm. Fluid Sci. 4 (1991) 103–114.

[5] S. Tiggelbeck, N.K. Mitra, M. Fiebig, Experimental investigations of heat transfer enhancement and flow losses in a channel with double rows of longitudinal vortex generators, Int. J. Heat Mass Transfer 36 (1993) 2327–2337.

[6] G. Biswas, P. Deb, S. Biswas, Generation of longitudinal streamwise vortices – a device for improving heat exchanger design, ASME J. Heat Transfer 116 (1994) 558–597.

[7] M. Fiebig, Embedded vortices in internal flow: heat transfer and pressure loss enhancement, Int. J. Heat Fluid Flow 16 (1995) 376–388.

[8] G. Biswas, K. Torii, D. Fuji, K. Nishino, Numerical and experimental determination of flow structure and heat transfer effects of longitudinal vortices in a channel flow, Int. J. Heat Mass Transfer 39 (1996) 3441–3451.

[9] Y. Chen, M. Fiebig, N.K. Mitra, Heat transfer enhancement of a finned oval tube with staggered punched longitudinal vortex generators, Int. J. Heat Mass Transfer 43 (1998) 417–435.

[10] C.C. Wang, J. Jo, Y.T. Lin, C.S. Wei, Flow visualization of annular and delta winglet vortex generators in fin-and-tube heat exchanger application, Int. J. Heat Mass Transfer 45 (2002) 3803–3815.

[11] J.S. Leu, Y.H. Wu, J.Y. Jang, Heat transfer and fluid flow analysis in plate-fin and tube heat exchangers with a pair of block shape vortex generators, Int. J. Heat Mass Transfer 47 (2004) 4327–4338.

[12] S.M. Pesteei, P.M.V. Subbarao, R.S. Agarwal, Experimental study of the effect of winglet location on heat transfer enhancement and pressure drop in fin-tube heat exchangers, Appl. Therm. Eng. 25 (2005) 1684–1696.

[13] J.E. O'Brien, M.S. Sohal, Heat transfer enhancement for finned-tube heat exchangers with winglets, ASME J Heat Transfer 127 (2005) 171–178.

[14] K.M. Kwak, K. Torri, K. Nishino, Simultaneous heat transfer enhancement and pressure loss reduction for finned-tube bundles with first or two transverse rows of built-in winglets, Exp. Therm. Fluid Sci. 29 (2005) 625–632.

[15] R.L. Webb, N.H. Kim, Principles of Enhanced Heat Transfer, second ed., Taylor & Francis, Boca Raton, 2005 (Chapter 3).

[16] R.J. Moffat, Describing the uncertainties in experimental results, Exp. Therm. Fluid Sci. 1 (1988) 3–17.