Experimental Study on Heat Transfer Characteristics of Triangular Curved-Surface Vortex-Generator Fins Postprint

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Abstract

To investigate the heat transfer characteristics of triangular vortex generator fins, experimental studies were conducted on 27 different tube-fin heat exchangers in an adiabatic wind tunnel. The fin pitches of the tube-fin heat exchangers were 1.7 mm, 2.0 mm, and 2.3 mm, respectively; the tube outer diameter considered was 8.9 mm. The experimental results present the variation of the JF factor, Nusselt number Nua, and friction factor fa with Reynolds number Rea. The results show that the heat transfer performance and friction factor of triangular curved vortex generator fins are both higher than those of plain fins, and the benefits outweigh the losses. The heat transfer performance of the two types of vortex generator fins is similar, but the JF factor of the triangular curved vortex generator I fin with a flow divider is higher than that of vortex generator II fin under the same conditions.

Full Text

Preamble

Experimental Study on the Heat Transfer Characteristics of Curve Triangular Vortex Generators

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Abstract

To investigate the heat transfer characteristics of curved triangular vortex generator fins, an experimental study was conducted on 27 different tube-fin heat exchangers in a thermally insulated wind tunnel. The fin spacing of the heat exchangers was varied at 1.7 mm, 2.0 mm, and 2.3 mm, while the tube outer diameter under consideration was 8.9 mm. The experimental results present the JF factor, Nusselt number (Nua), and friction factor (fa) as functions of Reynolds number (Rea). The results demonstrate that the heat transfer performance and friction coefficient of curved triangular vortex generator fins are both higher than those of plain fins, with the benefits outweighing the losses. The two types of vortex generator fins exhibit similar heat transfer performance, but the JF factor for fins with flow redistributors and curved triangular vortex generators (Type I) is higher than that for vortex generator Type II fins under the same conditions.

Keywords: tube-fin heat exchangers; curved triangular vortex generators; heat transfer characteristics

Introduction

Automotive radiators, internal cooling systems for diesel locomotive engines, and air conditioning systems represent important applications that demonstrate the significance of tube-fin heat exchangers. The flow structure formed by tubes and fins within the air flow channel is profoundly influenced by the tube arrangement configuration, which plays a critical role in determining the resulting flow patterns. While form drag can indeed be substantially reduced through the use of flat tubes, the welding technique employed to attach fins to tubes causes excessive energy consumption and environmental pollution, representing a significant reason for avoiding flat tubes. Nevertheless, an effective alternative solution involves using circular tubes, as in this case the fins and tubes are joined through an expansion process, making the manufacturing procedure more economical.

When air flows over circular tubes, two major defects emerge: the expansion of the wake region and subsequent flow pattern deterioration, which cause substantial pressure drop losses and degrade the heat transfer characteristics of the exchanger. Modifying the geometric shape of the air flow channel surface represents one method to address these two deficiencies.

To date, scholars have extensively investigated various fin types equipped with different vortex generators (VGs) to understand heat transfer enhancement through wake region reduction. Fiebig and Jacobi and Shah comprehensively reviewed early research on heat transfer performance and the application of VGs in compact heat exchangers. Leu, Wu, and Jang found that tube-fin heat exchangers with 45° angled block-type VGs installed behind tubes exhibited better thermal performance because the VGs generate longitudinal vortices that drive fluid into the wake region, thereby improving heat transfer. Joardar and Jacobi numerically investigated flow and heat transfer enhancement using common-flow-up array VGs in tube-fin heat exchangers, with results showing that three VG configurations increased the JF factor by 70% compared to the baseline plain fin case, though pressure loss increased correspondingly by 41%.

He, Han, and Tao et al. proposed using punched winglet-type VGs to enhance the air-side heat transfer performance of tube-fin heat exchangers, and their numerical results indicated that for punched VGs, the effectiveness of the primary vortex in improving heat transfer was not entirely dominant, while "corner vortices" had a significant impact on heat transfer performance. Yang, Bao, and Wang found that installing or punching novel longitudinal VGs on heat exchanger fin surfaces could narrow the wake region behind tubes. Hu, Su, and Wang et al. numerically analyzed the relationship between secondary flow intensity and heat transfer characteristics produced by triangular VGs with different configurations behind tubes. Wais et al. numerically investigated the influence of fin thickness and winglet orientation on mass and thermal efficiency in cross-flow heat exchangers, finding that the presence of winglets enhanced fluid and thermal exchange with the heat transfer surface by directing flow toward the wake region behind tubes, altering temperature and velocity distributions. Gong, Wang, and Lin numerically studied flow and heat transfer characteristics in circular tube-fin heat exchangers equipped with fin-punched curved triangular, trapezoidal, and rectangular VGs in the tube wake region, with results showing that curved VGs not only effectively reduced wake region area but also generated secondary flow to improve heat transfer capability on fin surfaces connected to the wake region. Lin, Liu, and Mei Lin et al. conducted a numerical study on interrupted annular groove fins with different groove positions, finding that interrupted annular grooves had dual functions of fluid guidance and separation vortex suppression, thereby reducing wake region area; however, at low Reynolds numbers, interrupted annular groove surfaces could not effectively improve heat transfer under the same pumping power criterion, while at higher Reynolds numbers, better heat transfer performance was achieved.

Wang, Chen, and Lin validated the air-side performance of tube-fin heat exchangers with semi-dimple VGs along fin surfaces. When airflow passed over the dimpled surface, flow separation could occur, generating recirculation flow and up-wash flow. Kanokjaruvijit and Martinez-Botas noted that considering economics, manufacturing, and pressure loss comprehensively, the hemispherical shape represented the optimal choice. Yang, Wun, and Chen et al. experimentally investigated the pressure drop and heat transfer characteristics of plain fins with different micro-dimple VG arrangements, finding that fins with micro-dimple VGs were more advantageous than conventional fin geometries.

If the mass flow rate through the tube front stagnation point is reduced, local pressure drop losses may also decrease. To some extent, the aforementioned micro-dimple VGs have minimal impact on flow distribution. Nevertheless, few reports have addressed other methods for reducing pressure drop losses in the region immediately upstream of the tube stagnation point. In light of this, researchers proposed installing flow redistributors (FRs) in front of the tube stagnation point to reduce mechanical energy losses around the tube. To test the influence of FRs combined with curved-edge triangular VGs on fin heat transfer and friction performance, researchers conducted performance tests on actual heat exchangers with excellent fins (punched curved-edge triangular VGs), also considering the effect of fin spacing.

1. Experimental Setup

Heat exchanger performance tests were conducted on a specific test platform, as shown in Figure 1 [Figure 1: see original paper]. The test platform consists of an open-loop low-speed wind tunnel, a water circulation system, and a silicon-controlled heater. The heat exchanger is installed in the experimental test section of the wind tunnel.

1.1 Air Loop and Water Loop

The air system primarily consists of an electric motor, induced draft fan, diffuser section, transition section, test section, convergence section, calibration section, and test section. All factors related to airflow through the wind tunnel and participation in the heat transfer process of the heat exchanger are set in the test section. Airflow velocity is controlled through the opening degree of the butterfly valve at the fan outlet, with the valve opening size determining the flow rate.

The water system mainly includes a water pump group, heating equipment, water flow rate regulation device, water temperature regulation device, water volume flow rate measurement device, water tank, and vent pipe. The water loop system contains two heating cabinets of different sizes with a total heating power of approximately 90 kW. After heating, hot water flows through pipelines to the heat exchanger under the action of the water pump. Circulating water is heated using a silicon-controlled heater, then cooled by air flow in the wind tunnel, and finally returns to the water tank through the water heater.

1.2 Temperature Measurement

Resistance thermometers, commonly used temperature measurement tools in medium-low temperature regions, are employed for air temperature measurement. Comprising a thermal resistance network and a double-armed bridge, they measure resistance. Air temperature can be calculated based on the linear relationship between temperature and metal resistance. However, hot water temperature is directly measured using glass rod thermometers with an accuracy of up to 0.05°C.

1.3 Pressure Measurement

Two static pressure holes are located on the measurement section walls, with two static pressure tubes connected to an inclined micromanometer. Therefore, the static pressure at the inlet and the air-side pressure drop across the heat exchanger can be calculated by reading the liquid column height of the inclined micromanometer. The water-side inlet and outlet pressures of the heat exchanger are measured using spring-type pressure gauges with an accuracy of up to 0.16 class. The air-side mass flow velocity is calculated through three local flow velocities measured by a pitot tube flowmeter located in the test section. The pitot tube flowmeter must be connected to a compensation-type micromanometer. According to the air-side volume flow rate of the tested heat exchanger, seven different test points are selected, with each test point measured three times to obtain average values.

Table 1 Geometric dimensions of fin-and-tube heat exchangers

Where: S1 = transverse distance between tubes; S2 = longitudinal distance between tubes; Do = tube outer diameter; = fin thickness; S3 = distance between tube and flow redistributor; = angle between bottom edge before starting point and hole center with stagnation point; Vl = vortex generator length; Vh = vortex generator height; a = flow redistributor length; b = flow redistributor width; c = flow redistributor height.

During the experiment, the water volume flow rate, inlet water temperature, air volume flow rate, and inlet air temperature are fixed, with only the air velocity adjusted within a certain range. Before data acquisition, heat balance between the air side and water side must be achieved, with the heat balance error between the two sides maintained at less than 5%.

This study tested two different curved-edge triangular VG types and compared them with fins punched only with FRs and plain fins. Detailed geometric parameters are listed in Table 1, with schematic diagrams shown in Figures 2 [Figure 2: see original paper], 3 [Figure 3: see original paper], and 4 [Figure 4: see original paper]. All tubes and fins are made of copper, and the test section is covered with a 10 cm thick cotton quilt to reduce heat loss to the surroundings.

Figure 3. Flow redistributors and curve triangular VGs (I) [Figure 3: see original paper]
Figure 4. Flow redistributors and curve triangular VGs (II) [Figure 4: see original paper]
Figure 1. Schematic diagram of experimental setup for real heat exchanger performance: 1 electromotor, 2 centrifugal ventilator, 3 soft tube, 4 diffuser section, 5 transition section, 6 inlet air temperature sensor, 7 heat exchanger, 8 outlet air temperature sensor, 9 test section, 10 converging section, 11 transition section, 12 rectification net, 13 calming section, 14-15 turning, 16 rectification net, 17 divergent duct, 18 joint body, 19 air dividing duct, 20 air outlet, 21 air inlet, 22-24 butterfly valve, 25 ventilator outlet, 26-30 supporting frame [Figure

1: see original paper]

Figure 2. Schematic view of circular tube bank fin heat exchanger with CTVGs and FRs [Figure 2: see original paper]

2. Data Reduction

Using the averaging method, the heat transfer rate in the calculations is determined from the heat transfer rates on the air side and water side as follows:

The overall heat transfer coefficient-area product KF is:

The log mean temperature difference is:

The overall heat transfer resistance is:

The water-side heat transfer coefficient Hw is obtained from:

The Dittus-Boelter correlation is used to calculate the water-side Nusselt number Nu_w (water-side Reynolds number Re_w):

The overall fin efficiency   equals the actual heat transfer rate of the fin and base divided by the heat transfer rate when the fin and base temperatures are identical. This can be expressed using fin efficiency ( ), fin surface area (F), and total surface area (F):

Here, F and F are the base and fin surface areas, respectively. The fin efficiency   is calculated using the approximate method described by Schmidt:

The air-side average Nusselt number is:

The air-side Reynolds number is:

Air density   is calculated from the ideal gas law using average pressure and average temperature:

The air-side friction factor is:

Here, De is the hydraulic diameter on the air side, calculated by:

A heat exchanger's performance is typically expressed through heat transfer and friction characteristics. While it is difficult to simultaneously reduce friction and increase heat transfer in a heat exchanger, this remains a designer' s objective. Therefore, the JF factor is widely used in research to compare different heat exchanger designs. The JF factor is a dimensionless number describing both heat transfer and friction characteristics of a heat exchanger, calculated as:

where Nu_plain and f_plain are the Nusselt number and friction factor for plain fins, respectively.

3. Data Uncertainty

To validate the experimental system, the Nusselt number (Nu) for plain fins was compared with the well-known correlation by Gray. Figure 5 [Figure 5: see original paper] describes the variation of Nu and f with Re . The maximum difference in Nu between the present data and Gray' s correlation is 8.65% at Re = 11757.7, while the minimum difference is 0.46% at Re = 6005.8. The maximum difference in f between the present data and the well-known correlation by Chang and Wang is 12.39% at Re = 3289.8, while the minimum difference is 2.03% at Re = 7423.4. The geometric similarity of fin thickness is not strictly limited in the experimental model, which affects fluid flow in the inlet region and consequently influences f and Nu . From effective reported results, the pressure drop in inlet and outlet regions is only a very small portion of the total pressure drop for this type of heat exchanger. In our experimental setup, a sufficient number of tube rows are used to further reduce these effects.

Figure 5. Validation of experimental results: (a) Nu ; and (b) f [Figure 5: see original paper]

4. Experimental Results and Discussion

4.1 Effect of Flow Redistributors and Curved Triangular VGs (I)

Figures 6 [Figure 6: see original paper], 7 [Figure 7: see original paper], and 8 [Figure 8: see original paper] compare the friction factor (f) and Nusselt number (Nu) for fins with CTVGs (I), fins with CTVGs (I) and FRs, fins punched with FRs only, and plain fins, for Reynolds numbers ranging from 300 to 5000 and fin spacings from 1.7 to 2.3 mm. Other parameters remain the same as listed in Table 1.

According to Figure 6(a), Nu increases with increasing Reynolds number, and the Nu distribution difference between fins with CTVGs (I) and fins with both CTVGs (I) and FRs is relatively small. Notably, under identical conditions, fins with CTVGs exhibit higher Nu than plain fins, with the Nu difference ranging from approximately 15.45% to 3.01% for Reynolds numbers between 300 and 5000. Similar analysis applies to Figures 7 and 8, which only vary in fin spacing. This demonstrates that the heat transfer performance of CTVGs is superior to that of plain fins for Reynolds numbers in the 300-5000 range. Fins punched with CTVGs (I) can generate longitudinal vortices around them that propagate along the main flow direction. Additionally, fins with both CTVGs and FRs show higher Nu than plain fins when the Reynolds number exceeds this range, with the Nu difference ranging from approximately 17.88% to 3.49% for Reynolds numbers between 300 and 5000.

As shown in Figure 6(b), the friction factor (f) is a function of Reynolds number, decreasing as the latter increases. For Reynolds numbers between 300 and 5000, the f distribution for fins with both CTVGs and FRs is close to that for fins with CTVGs only. It should be noted that when the Reynolds number exceeds this range, fins with both CTVGs and FRs exhibit higher f than plain fins. Similar analysis applies to Figures 7 and 8, which only vary in fin spacing.

Figure 6. Heat transfer performance of CTVGs (I) and FRs, CTVGs (I), FRs only, and plain fin: (a) Nu ; and (b) f under a fin spacing of 1.7 mm [Figure 6: see original paper]

Figure 7. Heat transfer performance of CTVGs (I) and FRs, CTVGs (I), FRs only, and plain fin: (a) Nu ; and (b) f under a fin spacing of 2 mm [Figure 7: see original paper]

Figure 8. Heat transfer performance of CTVGs (I) and FRs, CTVGs (I), FRs only, and plain fin: (a) Nu ; and (b) f under a fin spacing of 2.3 mm [Figure 8: see original paper]

4.2 Effect of Flow Redistributors and Curved Triangular CTVGs (II)

Figures 9 [Figure 9: see original paper], 10 [Figure 10: see original paper], and 11 [Figure 11: see original paper] compare the friction factor (f) and Nusselt number (Nu) for fins with CTVGs (II), fins with CTVGs (II) and FRs, fins punched with FRs only, and plain fins, for Reynolds numbers between 300 and 5000 and fin spacings ranging from 1.7 to 2.3 mm. Other parameters remain the same as in Table 1.

As shown in Figure 9, Nu increases with increasing Reynolds number, and the Nu distribution difference between fins with CTVGs (II) and fins with both CTVGs (II) and FRs is relatively small. It should be noted that for Reynolds numbers between 300 and 5000, fins with CTVGs exhibit higher Nu than both plain fins and fins with FRs only under the same conditions. Similar analysis applies to Figures 10(a) and 11(a), which only vary in fin spacing. This demonstrates that the heat transfer performance of CTVGs is superior to that of plain fins for Reynolds numbers in the 300-5000 range. Fins punched with CTVGs (II) can generate longitudinal vortices around them that propagate along the main flow direction. Additionally, fins with both CTVGs (II) and FRs show higher Nu than plain fins when the tested Reynolds number exceeds the 300-5000 range.

As shown in Figure 9(b), the friction factor (f) is a function of Reynolds number, decreasing as the latter increases. For Reynolds numbers between 300 and 5000, the f distribution for fins with both CTVGs (II) and FRs is close to that for fins with CTVGs (II) only. It should be noted that when the tested Reynolds number exceeds this range, fins with both CTVGs (II) and FRs exhibit significantly higher f than plain fins. Similar analysis applies to Figures 10(b) and 11(b), which only vary in fin spacing.

Figure 9. Heat transfer performance of CTVGs (II) and FRs, CTVGs (II), FRs only, and plain fin: (a) Nu ; and (b) f under a fin spacing of 1.7 mm [Figure 9: see original paper]

Figure 10. Heat transfer performance of CTVGs (II) and FRs, CTVGs (II), FRs only, and plain fin: (a) Nu ; and (b) f under a fin spacing of 2 mm [Figure 10: see original paper]

Figure 11. Heat transfer performance of CTVGs (II) and FRs, CTVGs (II), FRs only, and plain fin: (a) Nu ; and (b) f at a fin spacing of 2.3 mm [Figure 11: see original paper]

4.3 Effect of Fin Spacing

Fin spacing is a parameter that affects the heat transfer performance of heat exchangers equipped with fins containing CTVGs and FRs. The fin spacing was varied from 1.7 to 2.3 mm, with other parameters remaining the same as in Table 1.

Figure 12 [Figure 12: see original paper] describes the variation of Nu and f with increasing Reynolds number for plain fin heat exchangers. From Figure 12(a), we can see that Nu increases with Reynolds number, but when the fin spacing is Tp = 2 mm, its Nu is slightly smaller than at Tp = 2.3 mm. For Reynolds numbers between 500 and 5000, the Nu difference is approximately 12.48% to 2.06%. Figure 12(b) shows that regardless of whether the fin spacing is Tp = 1.7 mm, 2 mm, or 2.3 mm, f decreases with increasing Reynolds number. However, when the fin spacing is Tp = 1.7 mm, its f is smaller than at Tp = 2 mm or 2.3 mm. For Reynolds numbers between 500 and 5000, the f difference is approximately 28.82% to 25.83%.

Figure 12. Effect of fin spacing on heat transfer performance of plain fins: (a) Nu ; and (b) f [Figure 12: see original paper]

Figure 13 [Figure 13: see original paper] describes the variation of Nu and f with increasing Reynolds number for fins punched with FRs. From Figure 13(a), we can see that Nu increases with Reynolds number. For Reynolds numbers between 500 and 5000, the Nu difference is approximately 4.14% to 0.58%. Figure 13(b) shows that regardless of whether the fin spacing is Tp = 1.7 mm, 2 mm, or 2.3 mm, f decreases with increasing Reynolds number. However, when the fin spacing is Tp = 1.7 mm, its f is smaller than at Tp = 2 mm or 2.3 mm. For Reynolds numbers between 500 and 5000, the f difference is approximately 8.96% to 3.81%.

Figure 13. Effect of fin spacing on heat transfer performance of fins punched with FRs: (a) Nu ; and (b) f [Figure 13: see original paper]

Figure 14 [Figure 14: see original paper] describes the variation of Nu and f with increasing Reynolds number for fins punched with CTVGs (I). From Figure 14(a), we can see that Nu increases with Reynolds number, but when the fin spacing is Tp = 2.3 mm, its Nu is slightly smaller than at Tp = 2 mm. For Reynolds numbers between 500 and 5000, the Nu difference is approximately 10.27% to 4.09%. Figure 14(b) shows that regardless of whether the fin spacing is Tp = 1.7 mm, 2 mm, or 2.3 mm, f decreases with increasing Reynolds number.

However, at low Reynolds numbers, the f at Tp = 1.7 mm is smaller than at Tp = 2 mm or 2.3 mm. For Reynolds numbers between 500 and 5000, the f difference is approximately 11.63% to 10.68%.

Figure 14. Effect of fin spacing on heat transfer performance of CTVGs (I): (a) Nu ; and (b) f [Figure 14: see original paper]

Figure 15 [Figure 15: see original paper] describes the variation of Nu and f with increasing Reynolds number for fins punched with CTVGs (II). From Figure 15(a), we can see that Nu increases with Reynolds number, but when the fin spacing is Tp = 2 mm, its Nu is slightly smaller than at Tp = 2.3 mm. For Reynolds numbers between 500 and 5000, the Nu difference is approximately 13.89% to 5.49%. Figure 15(b) shows that regardless of whether the fin spacing is Tp = 1.7 mm, 2 mm, or 2.3 mm, f decreases with increasing Reynolds number. The f at Tp = 1.7 mm is smaller than at Tp = 2 mm or 2.3 mm. For Reynolds numbers between 500 and 5000, the f difference is approximately 15.85% to 12.45%.

Figure 15. Effect of fin spacing on heat transfer performance of CTVGs (II): (a) Nu ; and (b) f [Figure 15: see original paper]

Figure 16 [Figure 16: see original paper] describes the variation of Nu and f with increasing Reynolds number for fins punched with CTVGs (I) and FRs. From Figure 16(a), we can see that Nu increases with Reynolds number, but when the fin spacing is Tp = 2 mm, its Nu is slightly smaller than at Tp = 2.3 mm. For Reynolds numbers between 500 and 5000, the Nu difference is approximately 9.12% to 4.96%. Figure 16(b) shows that regardless of whether the fin spacing is Tp = 1.7 mm, 2 mm, or 2.3 mm, f decreases with increasing Reynolds number. The f at Tp = 1.7 mm is smaller than at Tp = 2 mm or 2.3 mm. For Reynolds numbers between 300 and 5000, the f difference is approximately 11.23% to 7.04%.

Figure 16. Effect of fin spacing on heat transfer performance of CTVGs (I) and FRs: (a) Nu ; and (b) f [Figure 16: see original paper]

Figure 17 [Figure 17: see original paper] describes the variation of Nu and f with increasing Reynolds number for fins punched with CTVGs (II) and FRs. From Figure 17(a), we can see that Nu increases with Reynolds number, but when the fin spacing is Tp = 2 mm, its Nu is slightly smaller than at Tp = 2.3 mm. For Reynolds numbers between 300 and 5000, the Nu difference is approximately 11.05% to 7.04%. Figure 17(b) shows that regardless of whether the fin spacing is Tp = 1.7 mm, 2 mm, or 2.3 mm, f decreases with increasing Reynolds number. The f at Tp = 1.7 mm is smaller than at Tp = 2 mm or 2.3 mm. For Reynolds numbers between 300 and 5000, the f difference is approximately 11.78% to 7.47%.

Figure 17. Effect of fin spacing on heat transfer performance of CTVGs (II) and FRs: (a) Nu ; and (b) f [Figure 17: see original paper]

4.4 JF Factor Comparison

Figures 18 [Figure 18: see original paper], 19 [Figure 19: see original paper], and 20 [Figure 20: see original paper] describe the variation of the JF factor with Reynolds number. For Reynolds numbers between 300 and 5000, the JF factor difference between fins with CTVGs and FRs and fins with FRs only is approximately 17.99% to 15.14%. Meanwhile, the JF factor difference between fins with CTVGs and FRs and fins with CTVGs only is approximately 0.05% to 3.05%.

As shown in Figure 18(a), the JF factor increases slightly with increasing Reynolds number. The JF factor for fins punched with FRs only is smaller than that for fins punched with both CTVGs (I) and FRs. Figure 18(a) compares the JF factors for fins punched with CTVGs (I), fins punched with both CTVGs (I) and FRs, and fins punched with FRs only at a fin spacing of 1.7 mm, as a function of Reynolds number. Under identical conditions, the JF factor for fins punched with both CTVGs (I) and FRs is smaller than that for fins punched with CTVGs (I) only. Additionally, at the same Reynolds number, the JF factor for fins punched with FRs only is always smaller than that for fins punched with CTVGs (I). This indicates that the comprehensive heat transfer performance of fins punched with both CTVGs (I) and FRs is inferior compared to fins punched with CTVGs (I) only when the fin spacing is 1.7 mm.

Figure 18(b) describes the JF factor for fins punched with FRs only and fins punched with CTVGs (II) only, which increases slightly with Reynolds number, while the JF factor for fins punched with both CTVGs (II) and FRs decreases significantly with increasing Reynolds number. The JF factor for fins punched with FRs only is smaller than that for fins punched with CTVGs (II) only. Furthermore, Figure 18(b) compares the JF factors for fins punched with CTVGs (II), fins punched with both CTVGs (II) and FRs, and fins punched with FRs only at a fin spacing of 1.7 mm, as a function of Reynolds number. Under identical conditions (except at low Reynolds numbers), the JF factor for fins punched with both CTVGs (II) and FRs is smaller than that for fins punched with CTVGs (II) only. When the Reynolds number exceeds 2000, the JF factor for fins punched with both CTVGs (II) and FRs is lower than that for both fins punched with CTVGs (II) only and fins punched with FRs only. Additionally, at the same Reynolds number, the JF factor for fins punched with FRs only is always smaller than that for fins punched with CTVGs (II). This further demonstrates that the comprehensive heat transfer performance of fins punched with both CTVGs (II) and FRs is inferior compared to fins punched with CTVGs (II) only when the fin spacing is 1.7 mm.

Figure 18. JF factor comparison between fins punched with CTVGs (I) and fins punched with CTVGs (II) at a fin spacing of 1.7 mm [Figure 18: see original paper]

As shown in Figure 19(a), the JF factor increases slightly with increasing

Reynolds number. The JF factor for fins punched with FRs only is smaller than that for fins punched with both CTVGs (I) and FRs. Figure 19(a) compares the JF factors for fins punched with CTVGs (I), fins punched with both CTVGs (I) and FRs, and fins punched with FRs only at a fin spacing of 2 mm, as a function of Reynolds number. Under identical conditions, the JF factor for fins punched with both CTVGs (I) and FRs is smaller than that for fins punched with CTVGs (I) only. Additionally, at the same Reynolds number, the JF factor for fins punched with FRs only is always smaller than that for fins punched with CTVGs (I). This indicates that the comprehensive heat transfer performance of fins punched with both CTVGs (I) and FRs is better compared to fins punched with CTVGs (I) only and fins punched with FRs only when the fin spacing is 2 mm.

Figure 19(b) describes the JF factor for fins punched with both CTVGs (II) and FRs and fins punched with CTVGs (II) only, which decreases with increasing Reynolds number, while it remains constant with increasing Reynolds number for fins punched with FRs only. The JF factor for fins punched with FRs only is smaller than that for fins punched with CTVGs (II) only. Furthermore, Figure 19(b) compares the JF factors for fins punched with CTVGs (II), fins punched with both CTVGs (II) and FRs, and fins punched with FRs only at a fin spacing of 2 mm, as a function of Reynolds number. Under identical conditions, the JF factor for fins punched with both CTVGs (II) and FRs is smaller than that for fins punched with CTVGs (II) only.

Figure 19. JF factor comparison between fins punched with CTVGs (I) and fins punched with CTVGs (II) at a fin spacing of 2 mm [Figure 19: see original paper]

As shown in Figure 20(a), the JF factor decreases significantly with increasing Reynolds number. The JF factor for fins punched with FRs only is smaller than that for fins punched with both CTVGs (I) and FRs. Figure 20(a) compares the JF factors for fins punched with CTVGs (I), fins punched with both CTVGs (I) and FRs, and fins punched with FRs only at a fin spacing of 2.3 mm, as a function of Reynolds number. Under identical conditions, the JF factor for fins punched with both CTVGs (I) and FRs is larger than that for fins punched with CTVGs (I) only. Additionally, at the same Reynolds number, the JF factor for fins punched with FRs only is always smaller than that for fins punched with CTVGs (I). This indicates that the comprehensive heat transfer performance of fins punched with both CTVGs (I) and FRs is better compared to fins punched with CTVGs (I) only.

Figure 20(b) describes the JF factor, which decreases significantly with increasing Reynolds number. The JF factor for fins punched with FRs only is smaller than that for fins punched with CTVGs (II) only. Furthermore, Figure 20(b) compares the JF factors for fins punched with CTVGs (II), fins punched with both CTVGs (II) and FRs, and fins punched with FRs only at a fin spacing of 2.3 mm, as a function of Reynolds number. Under identical conditions (except at low Reynolds numbers), the JF factor for fins punched with both CTVGs (II) and FRs is smaller than that for fins punched with CTVGs (II) only. At the same Reynolds number, the JF factor for fins punched with FRs only is always smaller than that for fins punched with CTVGs (II).

Figure 20. JF factor comparison between fins punched with CTVGs (I) and fins punched with CTVGs (II) at a fin spacing of 2.3 mm [Figure 20: see original paper]

Conclusions

To accurately evaluate the heat transfer characteristics of curved triangular VGs and flow redistributors, this study analyzed two CTVG designs and compared them with each other and against plain fins or fins punched with FRs.

1. Regardless of fin spacing, the heat transfer characteristics of fins punched with CTVGs and FRs or CTVGs only are higher than those of plain fins. Meanwhile, flow redistributors (FRs) do not improve the heat transfer performance of plain fins at the considered fin spacings.

2. Under identical conditions, the heat transfer characteristics of fins punched with both CTVGs and FRs are very similar to those of fins punched with CTVGs only. Furthermore, under identical conditions, the JF factor for fins punched with CTVGs (I) and FRs is higher than that for fins punched with CTVGs (II), with the JF factor difference ranging from 0.05% to 3.05% for Reynolds numbers between 700 and 4900. This indicates that fins punched with FRs and CTVGs (I) exhibit better heat transfer performance.

3. The study found that Nu increases with increasing Reynolds number for fins punched with CTVGs and FRs, CTVGs only, or FRs only. When the fin spacing is Tp = 1.7 mm, Nu is smaller than at Tp = 2 mm or Tp = 2.3 mm. When the fin spacing is Tp = 1.7 mm, the friction factor f is smaller than at Tp = 2 mm or Tp = 2.3 mm.

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Note: Figure translations are in progress. See original paper for figures.