

**An Experimental Evaluation
of the Thermal-Hydraulic Impact
of Delta-Wing Vortex Generators
in Plain-Fin-and-Tube Heat Exchangers**

A. I. El Sherbini and A. M. Jacobi

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For additional information:

Air Conditioning and Refrigeration Center
University of Illinois
Mechanical & Industrial Engineering Dept.
1206 West Green Street
Urbana, IL 61801

(217) 333-3115

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AN EXPERIMENTAL EVALUATION OF THE THERMAL-HYDRAULIC IMPACT OF DELTA-WING VORTEX GENERATORS IN PLAIN-FIN- AND-TUBE HEAT EXCHANGERS

A.I. EISherbini
A.M. Jacobi

ABSTRACT

Longitudinal vortex generation is a technique for enhancing heat transfer. Recent studies have tried to simulate the conditions for vortex generators in heat exchangers. In this work, the effectiveness of delta-wing vortex generators for fin-and-tube heat exchangers is experimentally evaluated in full-scale testing. The heat transfer and pressure drop performance of a heat exchanger are tested before and after adding a single row of vortex generators. A heat transfer enhancement of 31% was achieved, with a pressure drop penalty smaller than 10%. The results demonstrate the high potential for this method of heat transfer augmentation in plain-fin-and-tube heat exchanger applications.

NOMENCLATURE

ϵ	Heat exchanger effectiveness
α	Wing angle of attack
Δ	Wing aspect ratio ($2b/c$)
A_T	Total surface area
A_F	Fin surface area
A_t	Outside surface area of tubes
b	Wing base length
C	Heat capacity
c	Wing chord length
f	friction factor
h	Convection heat transfer coefficient
j	Colburn factor
n	Number of tubes
Nu	Nusselt number
Pr	Prandtl number
q	Heat transfer rate
R_{as}	Air-side thermal resistance
R_c	Heat capacity ratio (C_{min}/C_{max})
R_T	Total thermal resistance
Re	Reynolds number
T	Temperature
U	Heat transfer conductance

Subscripts

1	The first partition of the exchanger
2	The second partition of the exchanger
a	Air-side
i	Inlet
m	Midpoint between partitions
o	Outlet
p	Single tube pass
r	Refrigerant-side

INTRODUCTION

Heat exchanger performance has been important in meeting energy demands, and lower manufacturing costs and heat exchanger volumes; this importance continues to motivate the study of enhancement techniques for heat exchanger applications in HVAC & R systems. In liquid or refrigerant-to-air fin-and-tube heat exchangers, the air-side thermal resistance is the largest single contributor to the overall thermal resistance. Therefore, many techniques for enhancing heat transfer focus on the air-side heat transfer surface. One method to reduce the air-side thermal resistance is the generation of longitudinal vortices using passive vortex generators such as those shown in Figure 1. The pressure difference between the upstream and downstream sides of a vortex generator causes flow separation and the formation of secondary vortices that are carried through the heat exchanger by the flow. The resulting longitudinal vortices modify the thermal boundary layer and improve heat transfer. As with other techniques, this heat transfer enhancement is accompanied by a pressure drop penalty.

Several investigators have studied the structure of longitudinal vortices for different generators, surface geometries, and flow regimes. Jacobi and Shah (1995) and Fiebig (1995, 1997, and 1998) have presented thorough reviews of the research on vortex generators for enhancing heat transfer. In order to understand the mechanisms involved with vortex generators and their impact, some researchers have examined flows over flat plates for both the laminar and turbulent regimes. For example, Gentry and Jacobi (1997) tested delta-wings on a flat plate for a Reynolds number range of 600 to 1000 based on plate length. They reported average heat transfer enhancements of 50 to 60%. Yanagihara and Torii (1993) investigated the effects of arrays of delta winglets on the local and average heat transfer of a laminar boundary layer. Their experiments studied co-rotating and counter-rotating longitudinal vortices. The arrays of counter-rotating vortices tended to merge in the common flow region, while the co-rotating vortices did not merge. The best heat transfer augmentation was achieved for the counter-rotating vortices with the smallest distances between generators and the steepest angles of attack.

Channel flows have received more attention, because of their importance in heat exchanger applications. The most common types of vortex generators used in channel flows are delta and rectangular wings, and winglet pairs. The wings and winglets can either be attached to the fins or formed by stamping them out of the fins, as shown in Figure 1. Fiebig (1998) concluded that for channel flows, punched winglets perform better than wings. However, attached wings outperform winglets. In all cases, the heat transfer and pressure drop increase with the angle of attack until reaching an attack angle corresponding to vortex breakdown. The aspect ratio and position of winglets have smaller effects than the angle of attack.

Fiebig *et al.* (1986) had earlier tested vortex generators in a channel flow using the method of unsteady liquid crystal thermography (LCT). At a Re of 1360, they achieved enhancements of 20 to 60%. In 1991, Fiebig and co-workers extended their experiments to include rectangular wings and winglets. They found delta wings and winglets to be superior to rectangular ones; with a delta-wing providing more than 50% enhancement in heat transfer and a corresponding 45% increase in drag coefficient. Gentry and Jacobi (1998) used another method, the naphthalene sublimation technique, to measure the performance of vortex generators. For delta-wings in a channel flow, they reported average enhancements of 20-50% with a pressure drop penalty of 50-110%, for a Re range of 400-2000. Numerical investigations support these favorable assessments of the thermal-hydraulic performance of vortex generators in channel flows. The computations of Biswas and Chattopadhyay (1992) predicted enhancements of up to 34% with friction factor increases of 79% at the channel exit for built-in delta-wings. The corresponding heat transfer and pressure drop results for stamped wings were lower. Brockmeier *et al.* (1993) compared their numerical results for delta-wings in a channel flow to experimental data for other basic and high performance surfaces. They found the performance of the vortex generators to be superior to those of plain-fin, offset strip, and louvered-fin geometries.

Other research focused on channels with tubes, to better simulate heat exchangers. In 1993, Fiebig and co-workers tested heat exchanger elements with 3 tube rows and a delta-winglet pair downstream of each tube. For inline tube arrangement they measured a 55-65% increase in heat transfer, with a pressure drop increase of 20-45%. The staggered arrangement resulted in less enhancement and pressure loss. Fiebig *et al.* (1994) extended their work to compare round to flat tubes. They reported a 100% heat transfer enhancement for flat tubes, with a similar increase in pressure drop. It is believed (Jacobi and Shah 1995) that the arrangement of the flat tubes and winglets caused the enhancement potential to be exaggerated in the study by Fiebig *et al.* (1994), because of the poor performance of the baseline geometry. Gentry *et al.* (1996) assessed the potential of vortex generators for evaporator heat exchangers. Based on flat plate experimental data, they selected friction and Colburn *j*-factor multipliers for the established plain-fin heat exchanger data. The modified *f* and *j*-factors were compared to the plain ones using known performance evaluation criteria. They concluded that vortex generators would improve heat exchanger performance, with smaller heat exchanger sizes for a fixed heat duty or more heat transfer for a fixed size. Russell *et al.* (1982)

used the transient melt line method to test vortex generators in full-scale flat tube heat exchangers. However, they did not test the same heat exchanger geometry without vortex generators. Instead, they compared their measurements to existing plain-tube correlations. They reported 50% improvement in heat transfer, with an increase in friction factor of 20% at Re of 1000, based on hydraulic diameter.

Numerical studies have also addressed vortex generators in channel flows with tubes. Biswas *et al.* (1994a) reported that a delta-winglet pair downstream of the tube results in up to 240% local enhancement of the heat transfer in the recirculation zone. For punched delta-winglets in such a flow configuration, the numerical predictions of Fiebig *et al.* (1995) showed a 31% enhancement at Re=300. Jahromi *et al.* (1999) reported 20-50% enhancements with a similar configuration for a Re range of 400-1200. The ratio of the increase in Nusselt number to the increase in friction factor ranged between 0.65 and 0.78.

Although research on vortex generators has been motivated by their potential application in heat exchangers, realistic heat exchanger conditions have not been adequately tested, due to experimental limitations, geometrical considerations, or other restrictions. The objective of this study is to experimentally evaluate the effectiveness, in terms of heat transfer and pressure drop, of using delta-wing vortex generators in full-scale fin-and-tube heat exchangers. For this purpose, a heat exchanger (a refrigerator evaporator) is tested in full-scale with delta-wings, and compared to an otherwise identical coil. The thermal performance and friction results are used to assess the effectiveness of the vortex generators for the heat exchanger.

EXPERIMENTAL TEST APPARATUS

The tests were conducted in the closed circuit wind tunnel described in detail by Davis and co-workers (1996). A schematic of the tunnel is shown in Figure 2. The temperature and humidity of air were controlled in the flow conditioning chamber using 4 pre-heaters, 4 cooling coils connected to a chiller, and 2 after heaters. The heaters, which supplied a maximum input of 3 kW, were controlled by variable transformers. A 2.24 kW blower delivered air from the thermal conditioning chamber to the flow conditioning section. In order to thoroughly mix the flow and obtain a uniform temperature distribution, a static mixer at the fan outlet and a centrifugal mixer were used, in addition to 2 small mixing fans. A uniform velocity distribution was achieved by using a 3.2-mm-cell honeycomb, followed by 4 screens, based on the recommendations of NASA (Scheiman 1981). The screens were 70.4% open, with at least 75 screen-mesh sizes between them. A 12-to-1 cubic contraction provided a smooth transition between the flow conditioning section and the test section. The top of the test section was fastened to the sides by screws to avoid air leakage, allowing a height of 50.8 mm. Interior contractions were used to smoothly reduce the tunnel span from a maximum of 610 mm to the testing width of the heat exchanger, excluding the tube bends. In order to measure the pressure drop across the heat exchanger, four pressure taps in two upstream and downstream locations were connected to an electronic manometer. Air inlet and outlet temperatures were measured by 7 thermopiles at each side, with spanwise intervals of 76 mm between them. Each thermopile consisted of five 0.25-mm-diameter, type-T thermocouples calibrated against NIST-certified ASTM thermometers. The resulting uncertainty in the average air temperature was less than $\pm 0.09^\circ\text{C}$. The measured temperature profiles were flat to within $\pm 0.1^\circ\text{C}$. Approach air velocity measurements, using hot-wire anemometer, showed the velocity profiles to be flat to within $\pm 3.4\%$.

A 152-mm-diameter return pipe closed the air circuit. The air flow rate was measured by an ASME Standard orifice plate with 76.2 mm bore diameter. Air passing through the pipe was conditioned before reaching the orifice plate, following the ASME recommendations (1989). This conditioning was achieved through turning vanes in the 90° elbow and a honeycomb downstream of the elbow. The pressure drop across the orifice plate was measured by a pressure transducer that was calibrated to an accuracy of $\pm 0.3\%$ of the full-scale.

The coolant-side flow was provided by a chiller used to cool a single-phase ethylene glycol aqueous solution with a concentration of 32.6% by volume. A gear pump with a 375 W motor circulated the mixture into the heat exchanger (and the coolers of the thermal conditioning section). Two platinum RTDs were used to measure coolant inlet and outlet temperatures, with an uncertainty of $\pm 0.05^\circ\text{C}$. Care was taken to thoroughly mix the flow before reaching the RTDs so that measurements represented the bulk fluid temperature. This mixing was achieved using 90° elbows and mixing cups. A Coriolis-effect mass flow meter of $\pm 0.15\%$ uncertainty was connected downstream from the tested heat exchanger to measure the coolant mass flow rate.

A plain-fin heat exchanger was tested in the wind tunnel to compare its performance to that with the delta-wing vortex generators attached. The fins were brazed to the tubes in order to eliminate the effects of thermal contact resistance on heat exchanger performance. The testing length and height of the heat exchanger were 451 mm and

50.8 mm, respectively. The fin length was 203 mm, with a fin density of 2 fins/cm. The exchanger had 8 rows and 2 columns of tubes, with a tube outside diameter of 9.53 mm. The hydraulic diameter was 6.94 mm.

The heat exchanger was then tested after mounting the delta-wing vortex generators. The wings were fixed at the leading edges of the fins, as in Figure 3. The wings were chosen to have an aspect ratio, Λ , of 1 and angle of attack, α , of 55° . Table 1 provides the dimensions for the two sizes of wings that were used in these experiments.

Table 1 Dimensions for the delta-wing vortex generators used.

Delta wing designation	VG1	VG2
Wing base, b (mm)	4.2	10.5
Wing chord, c (mm)	8.4	21.0
Chord to coil hydraulic diameter (c/d_h)	1.21	3.026
Aspect ratio, $\Lambda (=2b/c)$	1.0	1.0
Angle of attack, α ($^\circ$)	55	55

PROCEDURE AND DATA INTERPRETATION

Energy balances between air and coolant sides were used to check the performance of the test apparatus. For 90% of the data points, the energy transfer rates were within 5% of each other. An ε -NTU method was then used to analyze the heat exchanger performance. The heat exchanger can be divided into two halves, as shown in Figure 4, with the assumption that the dividing line is adiabatic. Each partition has 8 passes in cross-flow. The upper partition has an overall counter flow arrangement, while the lower one has an overall parallel flow arrangement. From an energy balance on each stream, the following expressions can be written:

$$q_1 = C_{a1}(T_{ai} - T_{ao1}) \quad (1)$$

$$q_1 = C_r(T_{rm} - T_{ri}) \quad (2)$$

$$q_2 = C_{a2}(T_{ai} - T_{ao2}) \quad (3)$$

$$q_2 = C_r(T_{ro} - T_{rm}) \quad (4)$$

where C_r is the refrigerant heat capacity, C_{a1} and C_{a2} are the air heat capacities for the upper and lower partitions, respectively. The two heat capacities are assumed equal, with their sum equal to C_a . Since $C_a < C_r$, the heat rates can be expressed in terms of effectiveness as

$$q_1 = \varepsilon_1 C_{a1}(T_{ai} - T_{ri}) \quad (5)$$

$$q_2 = \varepsilon_2 C_{a2}(T_{ai} - T_{rm}) \quad (6)$$

The effectiveness of all the individual passes in both partitions were considered equal, since the convective heat transfer coefficient was assumed uniform throughout the heat exchanger. The air flow was unmixed, due to the fins, whereas the coolant flow was mixed. The effectiveness of a single pass in such a configuration (Incropera and DeWitt 1996) is

$$\varepsilon_p = \{1 - \exp[-R_c(1 - \exp(-NTU_p))]\} / R_c \quad (7)$$

where R_c is the heat capacity ratio, $R_c = C_{a1}/C_r = C_{a2}/C_r$. The number of transfer units for a single pass is related to the thermal conductance by

$$NTU_p = UA_p / R_c \quad (8)$$

Following the method of Domingos (1969), outlined by Shah and Mueller (1985), the effectiveness of the overall counter-flow partition becomes

$$\varepsilon_1 = \frac{\left(\frac{1 - \varepsilon_p C_r}{1 - \varepsilon_p}\right)^n - 1}{\left(\frac{1 - \varepsilon_p C_r}{1 - \varepsilon_p}\right)^n - C_r} \quad (9)$$

where n is the number of passes in the partition. Likewise, the effectiveness of the overall parallel-flow partition is

$$\varepsilon_2 = \frac{1 - (1 - \varepsilon_p (1 + C_r))^n}{1 + C_r} \quad (10)$$

In equations (1) to (10), the inlet and outlet temperatures, mass flow rates, and specific heat of both the air and refrigerant are known. The ten equations can be solved simultaneously for the following 10 unknowns: q_1 , q_2 , T_{ao1} , T_{ao2} , T_{rm} , ε_1 , ε_2 , ε_p , NTU_p , and UA_p . The overall thermal conductance of the heat exchanger can then be found from

$$UA_T = 2nUA_p \quad (11)$$

The overall conductance, UA_T , is the reciprocal of the overall thermal resistance, $R_T = 1/UA_T$. Figure 5 shows a thermal resistance network for the heat exchanger. The air-side resistance, R_{as} , is a combination of the resistances due to air-side tube convection, fin convection, and fin-to-tube contact resistance. The total thermal resistance, R_T , is equivalent to the sum of air-side resistance, tube conduction resistance, and refrigerant convection resistance inside the tubes. A modified Wilson plot is used to obtain the air-side heat transfer resistance of the heat exchanger, R_{as} . For a given air flow rate, the total thermal resistance, R_T , is plotted against the reciprocal of the refrigerant Nusselt number ($1/Nu_r$) at various coolant flow rates. In Figure 6, Wilson plots of sample measurements are shown. Each line represents a fixed air-side Reynolds number. The air-side resistance, R_{as} , is found by extrapolating the line to intercept the ordinate, R_T . The infinite refrigerant Nusselt number ($1/Nu_r = 0$) indicates zero convection resistance at the tube-side; thus the intercept value of R_T gives R_{as} . The Gnielenski correlation was used to model the refrigerant Nusselt number, since it is believed to be most appropriate for the Reynolds number range under consideration (Bhatti and Shah 1987)

$$Nu_r = \frac{(f/8)(Re - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \quad (12)$$

where f is the friction factor obtained from the Filonenko correlation for the tube-side friction,

$$f = (0.790 \ln Re_r - 1.64)^{-2} \quad (13)$$

Since the tested heat exchanger had a zero fin-to-tube contact resistance, the air-side resistance in Figure 5 can be expressed as

$$R_{as} = \frac{1}{hA_t + \eta_f hA_f} \quad (14)$$

where A_t is the outside surface area of the tubes, A_f is the surface area of fins, and η_f is the fin efficiency. Equation 14 is used to get the convection coefficient, h . The fin efficiency is calculated based on the sector method proposed by Carrier and Anderson (1944). Care was taken in applying the method to accurately account for the shape of the fins.

The uncertainties in the measurements were propagated to the calculated quantities using the method of Kline and McClintock (1953). The resulting uncertainties in the air-side Reynolds number and thermal resistance were less than 5% and 7%, respectively. The j and f factor data had uncertainties less than 9% and 9.6%, respectively.

RESULTS AND DISCUSSION

The thermal performance of the heat exchanger after attaching the delta-wing vortex generators is compared to its original performance without the generators in Figure 7. The figure presents the heat transfer results in terms of air-side thermal resistance. The data show only a small scatter, within the experimental uncertainty of R_{as} of $\pm 7\%$. The smaller size of delta-wings, VG1, caused the air-side resistance, R_{as} , of the heat exchanger to drop by 13 to 16.6%. The delta-wings generated longitudinal vortices that were carried along the fins by the flow, causing an increase in flow mixing and, therefore, an enhancement in convection to the fins. When VG1 wings were replaced by VG2, R_{as} decreased more. The larger wings generated stronger vortices, for the same air flow rate. The drop in R_{as} due to VG2 was 20 to 21.5%. It should be noted that the wings were attached to the fins using a non-conducting tape. Hence, the wings did not increase the effective heat transfer area of the fins, and the decrease in the fin resistance can be entirely attributed to the generated vortices. The wing-to-fin area ratio was 0.23% for VG1 and 1.4% for VG2. Thus, the enhancement can be achieved with an insignificant addition in fin material. The delta-wings can also be stamped out of the fins, in which case there is no addition of material. The heat transfer results are presented in terms of Colburn j -factors in Figure 8. The use of VG1 wings caused the j -factor to increase by 14 to 18.3%. The enhancement magnitudes differ slightly from those of R_{as} , due to the nonlinear relation between j and R_{as} and the differences in the reference values of comparison. The increase in j due to VG2 delta-wings ranged from 24.7 to 31.3%, with a maximum uncertainty of $\pm 9\%$.

These results are very encouraging, considering that only a single row of vortex generators was used with 8 tube rows. Prior research has shown that the use of multiple rows of vortex generators can significantly increase the heat transfer enhancement (Tiggelbeck *et al.* 1993, Biswas *et al.* 1994b, and Chen *et al.* 1998). Because of such enhancements, a heat exchanger with vortex generators can be smaller in size, for the same heat duty. This size reduction can reduce cost and allow more compactness.

The pressure drop measurements are presented in Figure 8, which shows the friction factor data for the heat exchanger before and after mounting VG1 and VG2 delta-wings. The figure indicates that the vortex generators did not cause a significant change in the friction factor. The absence of a pressure drop penalty at the tested conditions is a very interesting result. It is common for heat transfer enhancement techniques to be accompanied by a pressure-drop penalty. In particular, previous investigations have shown that enhancements by way of vortex generation are associated with increases in pressure drop of the same order as the heat transfer enhancement. For the conditions and geometry of this test, the change in pressure drop was within the measurement uncertainty. The uncertainty in f reached a maximum of 9.6% in the Reynolds number range corresponding to heat transfer measurements. It is expected that the addition of multiple rows of generators to the heat exchanger would increase the pressure drop; however, with the increased heat transfer, the small change in pressure drop would be a key advantage over other enhancement methods. It is notable that studies of vortex generators in heat exchanger elements tend to report less pressure drop penalties than studies with channel flows. The relative contribution to pressure drop by tubes is much higher than by channels. Therefore, the increase in pressure drop due to delta-wings is more pronounced in the case of channel flows.

The overall performance of the delta-wing vortex generator is evaluated using the London area-goodness factor, j/f , and compared to the baseline, plain-fin, performance in Figure 9. For a Reynolds number range common to these tests, VG1 wings improved the j/f factor by 17 to 20.2%. The corresponding increase in the j/f ratio for VG2 was 28.9 to 33.5%. These values differ slightly from the increases in j , due to the small differences in f -values.

In this work, the fins were brazed to the tubes, so that no thermal contact resistance existed between them. The contact resistance in a heat exchanger is connected in series to the fin resistance, R_f , as shown earlier in Figure 5. If there is a contact resistance, the reduction in R_f due to vortex generators will have a smaller effect on R_{as} . Therefore, the heat transfer enhancement is anticipated to be lower when contact resistance is important. The impact on enhancement will depend on the ratio of the contact to fin resistances. For fins with collars, the contact area is relatively large and the contact-to-fin resistance ratio is small. In this case, the heat transfer enhancement due to the vortex generators will be only slightly lower than the case of no contact resistance. The collarless fins, such as those used in refrigerator evaporators, have higher contact-to-fin resistance ratios; however, recent work by ElSherbini *et al.* (2000) shows that a thin layer of frost eliminates the effect of contact resistance.

The use of vortex generators during frost growth on a surface was studied by Storey and Jacobi (1999), who found that vortices increased the maximum frost thickness by only 7.2%. The increased frost deposition increases the density of the frost layer and slightly improves its conduction resistance. Storey and Jacobi concluded that the net effect of delta-wings under frost conditions is to provide overall heat transfer enhancement. Therefore, the use of vortex generators in refrigeration applications appears particularly promising in view of the results now presented.

The use of delta-wings in heat exchangers may cause some practical concerns. The sharp edges of the wings require ergonomic consideration. Also, handling the heat exchangers may deflect the wings and change their angles of attack. In order to address such concerns, it is recommended to place the delta-wings downstream of the leading edges. There are other types of vortex generators that are suitable for heat exchanger applications, such as hemispherical bumps. Gentry and Jacobi (1998) studied hemispherical bumps in flat plate and channel flows and showed good potentials for enhancing heat transfer using this method. It is recommended that such vortex generators be tested on heat exchangers in full-scale.

The results of the current work have demonstrated a favorable impact of delta-wings on the performance of heat exchangers. The full-scale tests of single rows of vortex generators showed considerable heat transfer enhancements. The insignificant pressure-drop penalty further confirmed that impact. For best performance, the configuration of the wings as well as their locations and number of rows need to be optimized.

CONCLUSIONS

The thermal-hydraulic effectiveness of using delta-wing vortex generators to augment heat transfer in fin-and-tube heat exchangers was evaluated. The heat transfer and pressure drop performance of a plain-fin heat exchanger were tested in full-scale, and compared to the performance after attaching a single row of delta-wing vortex generators. The wings resulted in reductions in the air-side thermal resistance and increases in the Colburn j -factor. Heat transfer enhancements of up to 31.3% were achieved, with no significant increase in pressure drop. The results show a high potential for delta-wing vortex generators as effective devices for improving the performance of plain-fin heat exchangers. The heat transfer enhancements measured in this investigation are in accordance with previous studies for channels and heat exchanger models. However, this work tested vortex generators in a full-scale heat exchanger used as a refrigerator evaporator. The friction factor results, which were not affected by the vortex generators, showed lower pressure losses than anticipated. A simple performance evaluation using the j/f factor confirmed the high effectiveness of the vortex generators. The findings can be extended to other heat exchanger configurations and fin geometries, since the vortex generators act on improving the fin resistance. A fin-to-tube contact resistance would decrease the thermal enhancement of the vortex generators according to the value of the contact resistance. A low contact-to-fin resistance ratio results in a small decrease in enhancement. Frost would also cause the enhancement to drop slightly because the longitudinal vortices increase frost deposition. This increase causes a small increase in frost density and thickness. It is recommended to use multiple rows of delta-wings, and optimize their configuration, in order to achieve better performance. For practical considerations, the delta-wings are suggested to be located some distance downstream of the leading edges of fins. Other types of vortex generators, such as hemispherical bumps, are recommended for full-scale tests on heat exchangers.

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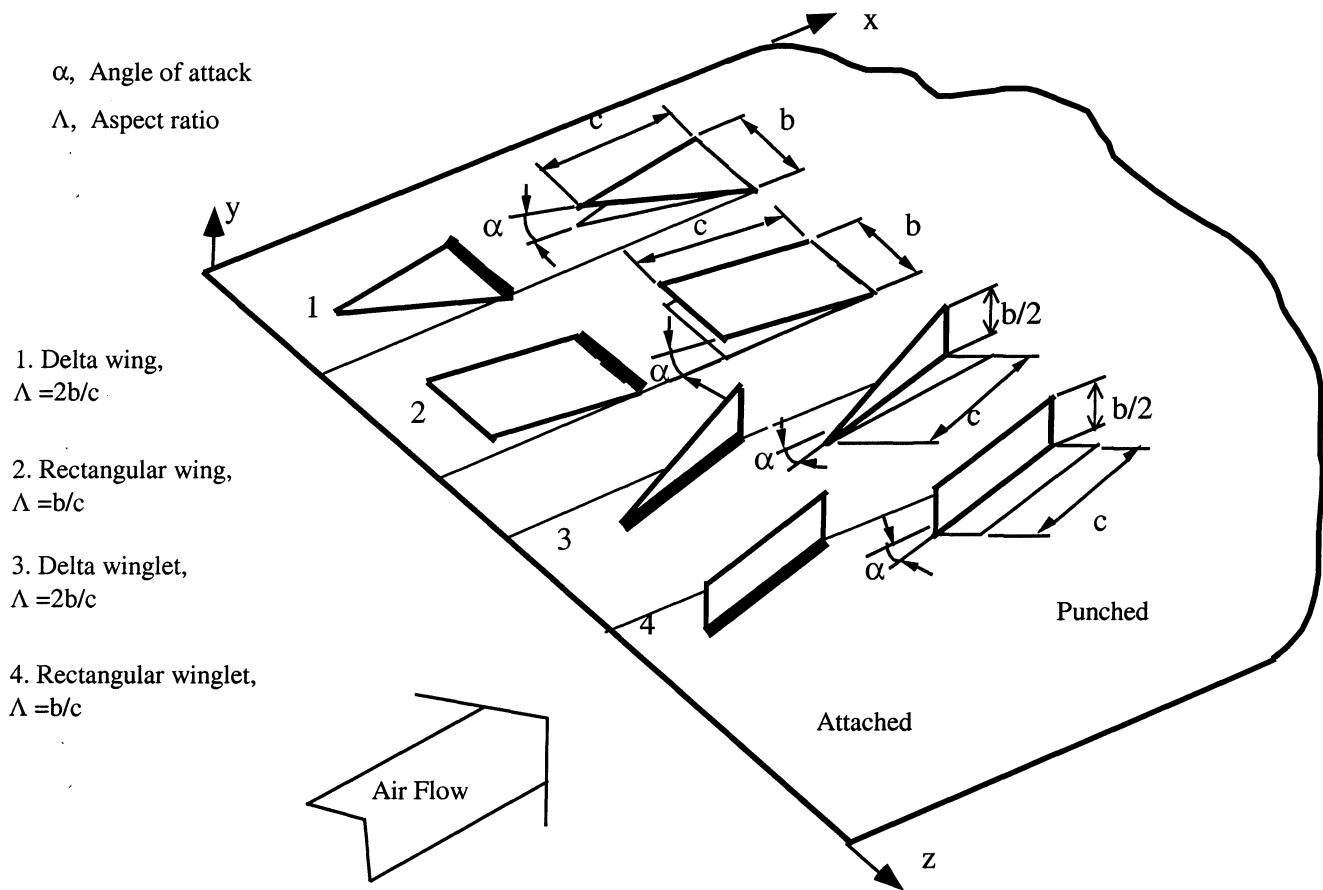
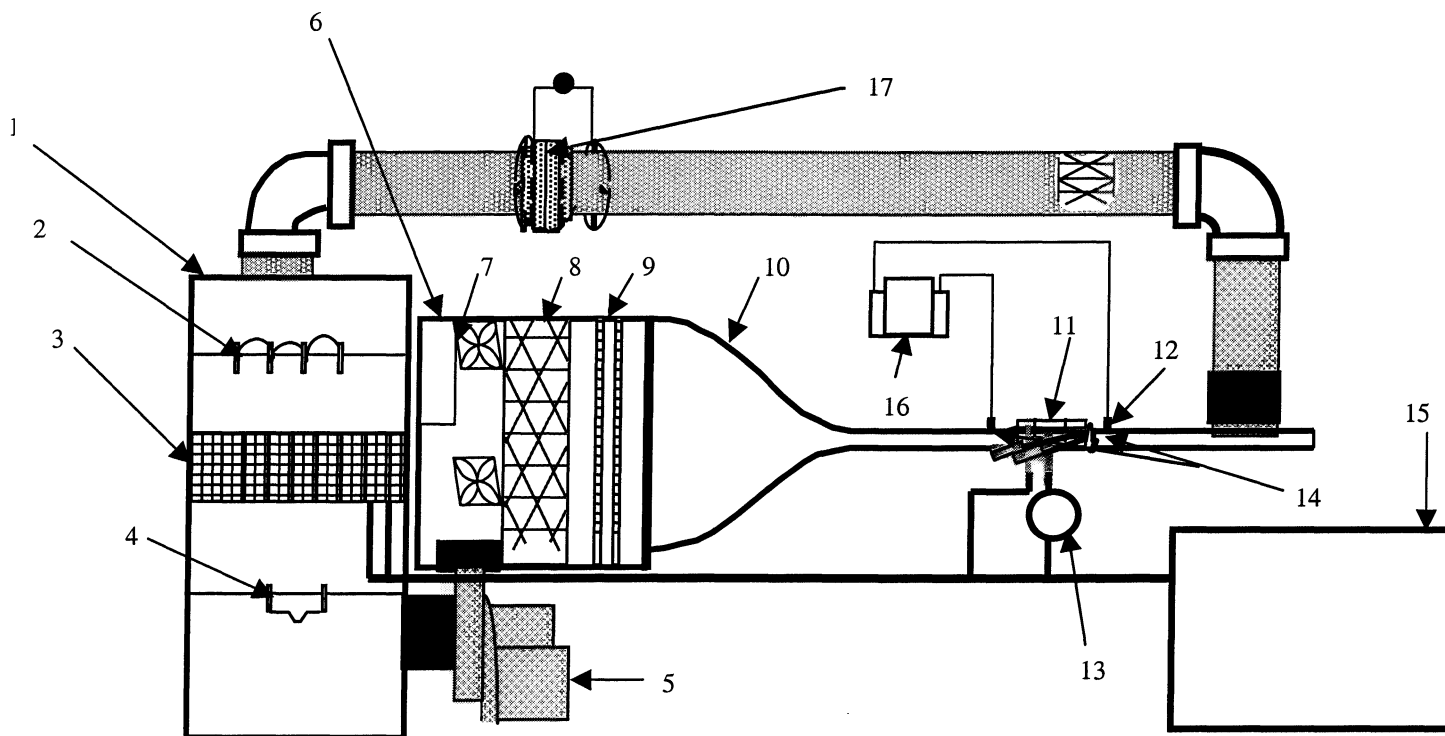


Figure 1. Punched and attached vortex generators. A wing intersects the fin at its base, while a winglet intersects at its chord. The geometrical definitions for each type are shown. (From Jacobi and Shah, 1995).



1	Thermal conditioning chamber	10	Contraction
2	Pre-heaters	11	Test section
3	Cooling coils	12	Pressure taps
4	After heaters	13	Coolant mass flow meter
5	Blower	14	Thermocouples
6	Flow conditioning chamber	15	Chiller
7	Mixers	16	Micro-manometer
8	Honeycomb	17	Air flow meter
9	Screens		

Figure 2. Schematic of the wind tunnel apparatus used for testing heat exchangers.

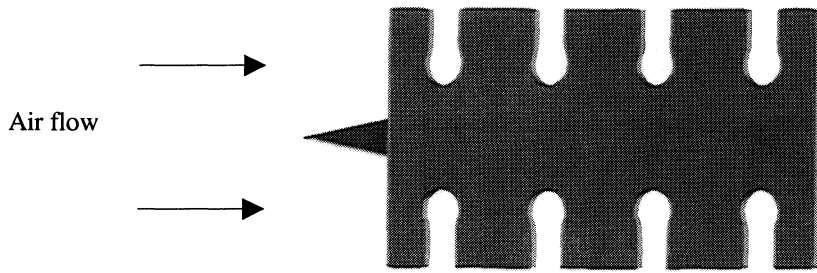


Figure 3. A delta-wing attached to the leading edge of a fin. The fins extended over 8 rows and 2 columns. The figure shows only part of the fin.

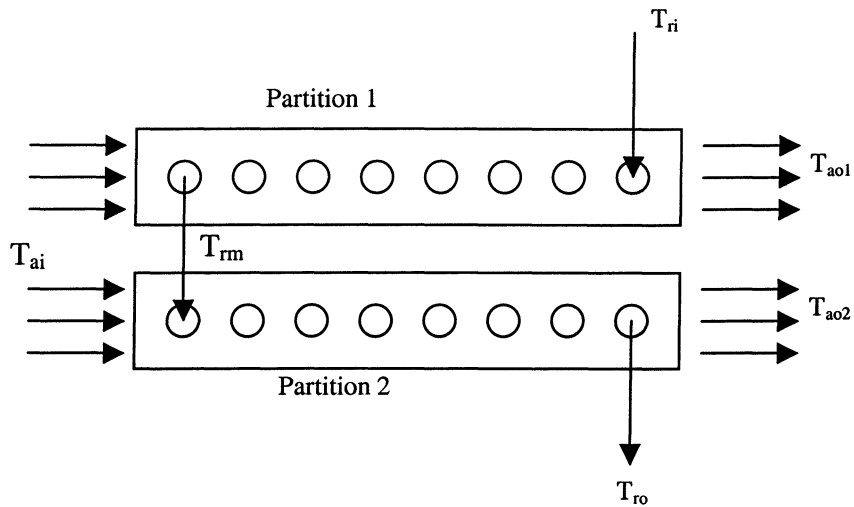


Figure 4. Heat exchanger configuration for data interpretation. Both partitions are in cross-flow. The overall arrangement is counter-flow for the upper partition, and parallel-flow for the lower one.

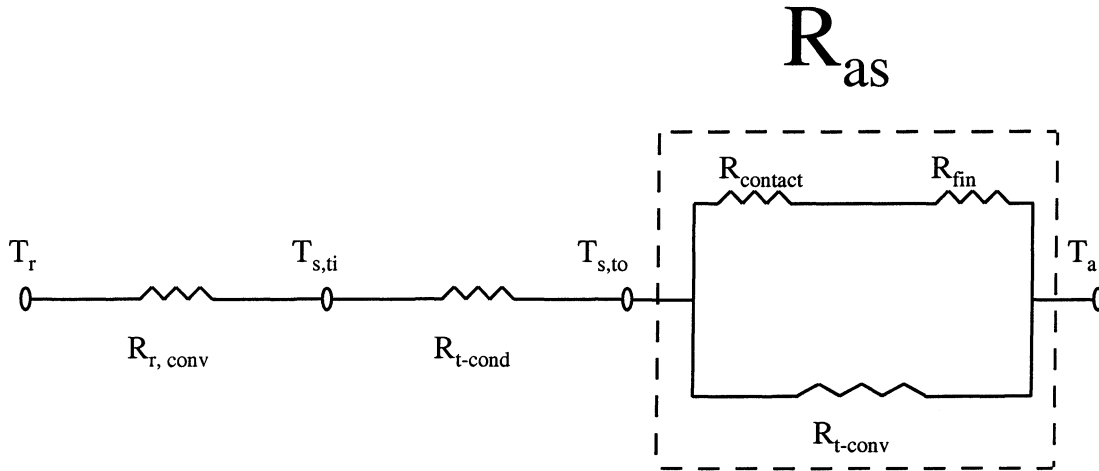


Figure 5. Thermal resistance network for the heat exchanger. The air-side resistance is the equivalent of the fin, contact, and outside tube convection resistances. The air-side resistance is in series with the resistances due to conduction through the tubes and convection inside them. $T_{s,ti}$ and $T_{s,to}$ are the inside and outside tube surface temperatures, respectively.

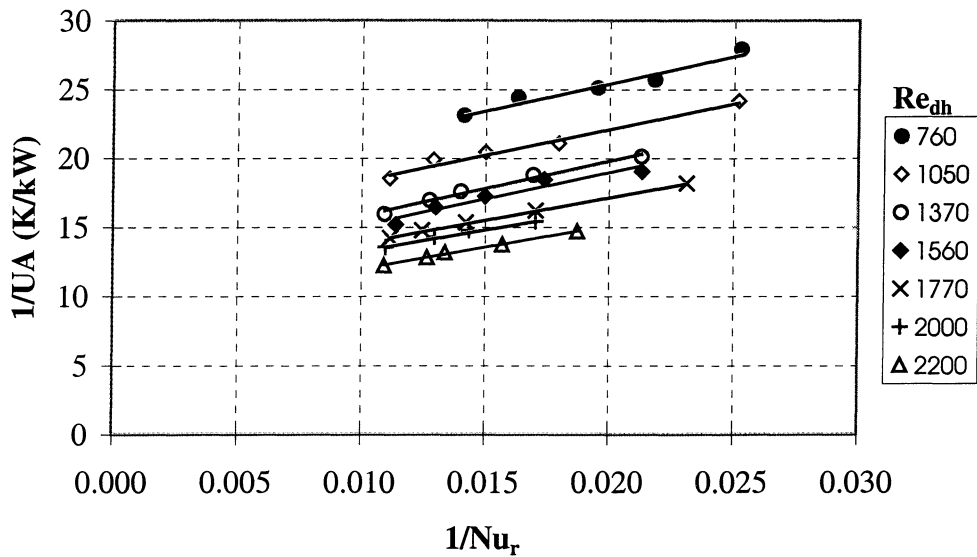


Figure 6. Modified Wilson plots for the tested heat exchanger at different air-side Reynolds numbers. For a given air flow rate, the air-side resistance, R_{as} , is the ordinate-intercept of the straight line.

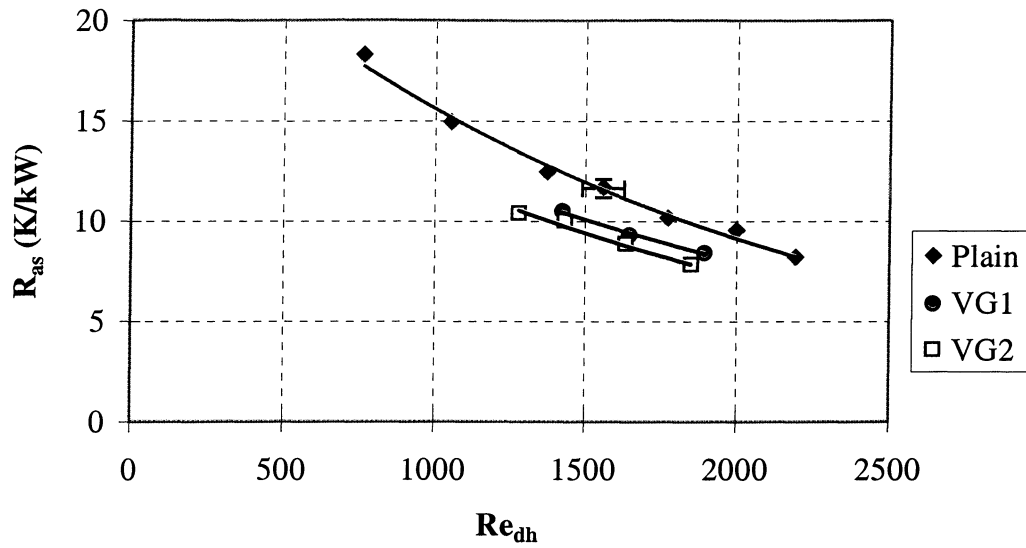


Figure 7. Air-side thermal resistance, R_{as} , for a plain coil, and the same coil with a single row of VG1 and VG2 delta-wings. VG1 decreased R_{as} by 13 to 16.6%, and VG2 decreased it by up to 21.5%.

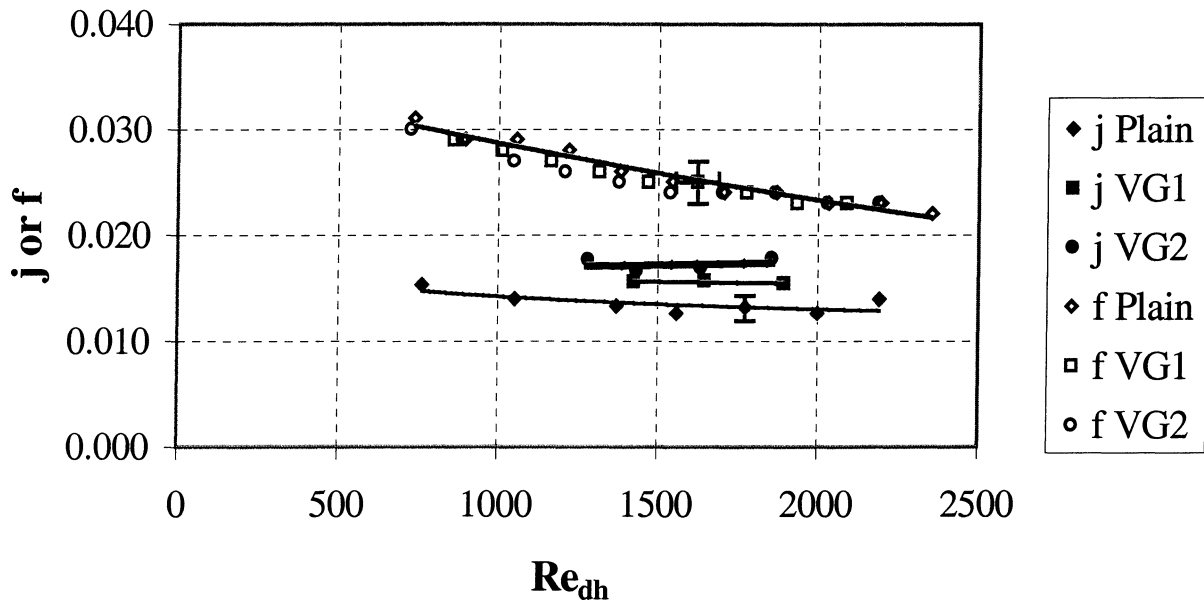


Figure 8. Colburn j and friction factor, f , results for a plain coil, and the same coil with VG1 and VG2. The j factor increased by up to 31.3% because of the vortex generators. No significant change in the friction factor was measured. The large enhancement in heat transfer was not accompanied by a pressure drop penalty.

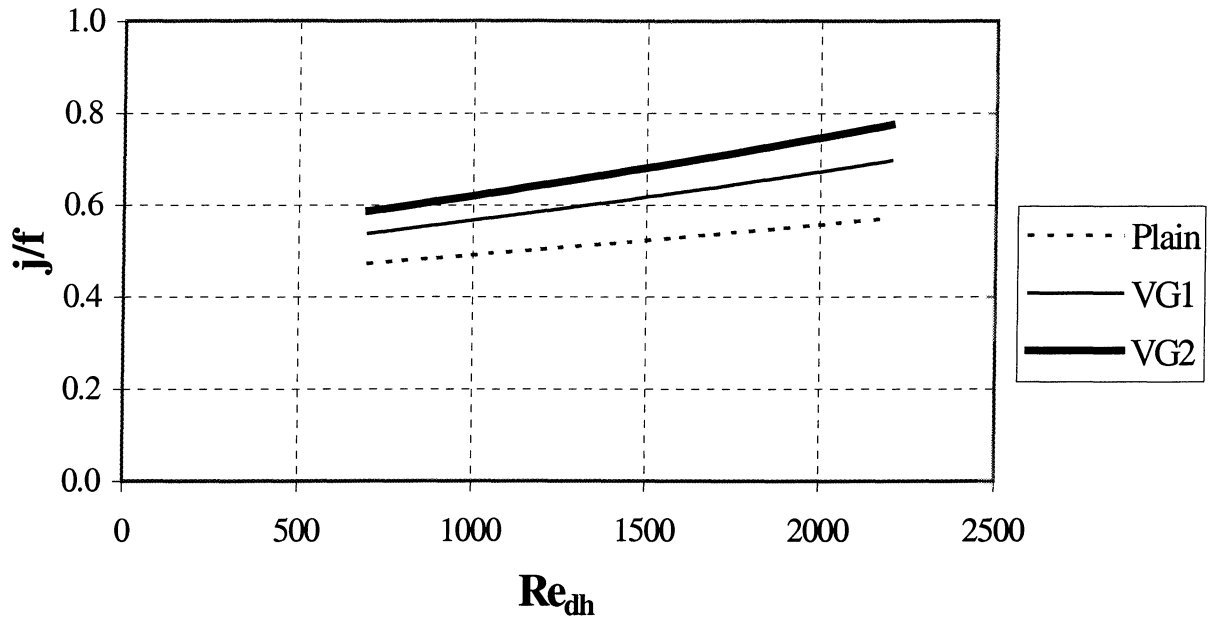


Figure 9. Area goodness factor, j/f , for the plain coil, and the same coil with VG1 and VG2 vortex generators. The increase in j/f values due to the delta-wing vortex generators demonstrates their advantageous overall performance on heat exchangers.