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Abstract
Systematic numerical simulations are carried out for a steady-state, laminar and constant-property air flow through a passage of a compact fin-tube heat exchanger specified with a constant wall temperature distribution and a uniform inlet velocity profile. A pair of delta winglet-type vortex generators (VGs) is punched out of the plain fin surface near the tube as a heat transfer enhancement device. A variety of VG configurations are investigated and their effects on the thermal-hydraulic (heat transfer) performance of the compact fin-tube heat exchanger are presented in terms of the Colburn factor $j$ and the friction factor $f$ for a specific Reynolds number ($Re$=400). In addition, the highest net enhancement in the thermal-hydraulic performance, defined as the ratio of the heat transfer enhancement to the pressure loss penalty, is sought. It is found that a moderate degree of the upstream shifting and the spanwise shifting of VG toward the tube contribute to the net enhancement. Nearly 9% of the net enhancement is achieved with an optimum configuration investigated presently.

Key words: Compact Fin-Tube Heat Exchanger, Winglet-Type Vortex Generator, Heat Transfer Enhancement, Pressure Loss Penalty, Net Enhancement, Numerical Simulation

1. Introduction

In gas-to-liquid heat exchangers, the gas-side convective heat transfer coefficient is usually 10 to 50 times smaller than that on the liquid-side, and thereby resulting in lower heat transfer rate per unit surface area. This is due to the lower thermal conductivity and density of the gas relative to most of liquids. Thus a large surface area becomes a typical characteristic of gas-to-liquid heat exchangers. A common approach to construct high density surfaces is to make use of extended surfaces or fins. Figure 1 depicts a schematic of a typical compact fin-tube heat exchanger, of which heat transfer surface area per unit volume exceeds $700m^2/m^3$ (1). For such compact fin-tube heat exchangers, in general, plain fins have been employed basically as secondary surfaces, and nowadays these are followed by enhanced fin configurations such as offset-strip fins, louvered fins, perforated fins and corrugated or wavy fins. These special fin configurations improve the heat transfer coefficient by altering the flow field (2), disrupting boundary layer and/or mixing the bulk flow through secondary motions (3). However, it should be pointed out that since most compact heat exchangers are designed and have been optimized for flow conditions characterized by relatively large Reynolds numbers (1000 and above, typically), the heat
transfer performance of all these traditional surfaces degrade considerably as Reynolds number decreases (4).

Fig. 1 Schematic of a typical fin-tube heat exchanger.

Heat transfer enhancement using vortex generators (VGs) is, on the other hand, a promising technique for compact fin-tube heat exchangers, especially for circular tube geometries. This enhancement method is based on boundary-layer thinning, flow swirling, and flow destabilization (5). Furthermore, appropriate placement of VGs on the fins reduces a poor heat transfer region in the tube wake (6). Considerable work has been done on vortex-induced heat transfer enhancement for circular finned tube heat exchangers.

Fiebig et al. (7) examined the potential of VGs on the simultaneous heat transfer enhancement and pressure-loss reduction of circular finned tube heat exchangers. A single-row circular tube of 50mm diameter and 20mm fin spacing formed an experimental test core. A pair of winglet-type VGs was employed in common-flow down configuration. The winglet had 10mm height, 45° angle of attack and an aspect ratio of 2. An optimum configuration proposed was to place a pair of winglets right behind the tube with their spanwise separation of two tube diameters. In the Reynolds number range of 2000 (based on channel height) to 5000, the optimum configuration yielded heat transfer enhancement up to 20% and pressure-loss decrease up to 10% compared to the plain finned tube geometry. This enhancement was attributed to the delayed boundary layer separation on the tube by longitudinal vortices generated, which introduced high momentum fluid into the region behind the cylinder.

Fiebig et al. (8) extended the study to three-row heat exchanger geometry. The average heat transfer enhancement was 55 to 65% for the inline tube arrangement, and 9% for the staggered arrangement. The corresponding pressure drop increase in the Reynolds number range of 600 to 2700 (based on channel height) was 20 to 45% and 3% for inline and staggered tube arrangements, respectively. In this study, VGs were in common-flow down configuration.

Torii et al. (9) tested three-row tube geometry with VGs. Winglet pairs with 15° angle of attack were installed beside the first tube row only. This geometry was unique in that the winglet pairs were in common-flow up configuration, thus forming a narrow flow passage between the tube and each VG. They reported that a significant decrease of pressure loss together with a moderate increase in heat transfer was achieved at a particular
configuration. They attributed this enhancement to a separation delay, reduction of form drag of the tube, and a resultant removal of poor heat transfer zone from the wake. But they did not show the reason why such a remarkable enhancement was gained at that particular configuration.

The present study is motivated by various findings from the previous studies. Their configurations seem to be very promising to enhance the heat transfer while keeping the pressure loss low, which ultimately leads to net enhancement in heat transfer performance. To do so, a number of geometric parameters of VG are to be investigated systematically and their effect on the heat transfer performance is to be clarified. In general, the geometric parameters include an angle of attack, an area ratio of VG to fin, an aspect ratio of VG, a tilt angle of VG to the fin surface, a tube arrangement (inline or staggered), the number of tube rows, and so on. In the present study, a reference configuration is first defined, which is similar to that proposed by Torii et al. and then 25 cases of different configurations are identified with three geometric variations introduced in Section 2.1. Numerical simulations have been carried out systematically for those configurations and their influence on the basic heat transfer and pressure drop characteristics has been examined exhaustively. Flow condition is fixed at a Reynolds number of 400 which is one of the most frequently encountered design values for compact fin-tube heat exchangers.

Nomenclature

\[
\begin{align*}
A & \quad \text{total heat transfer area, m}^2 \\
B & \quad \text{width of the fin, m} \\
Br & \quad \text{Brinkman number, } \mu u_m^2/k(T_{\text{w,m}}-T_m), \text{ dimensionless} \\
B(x) & \quad \text{local width of the fin exposed to the fluid along the flow passage, m} \\
c_p & \quad \text{specific heat of air at constant pressure, J/kg K} \\
D & \quad \text{diameter of the cylindrical tube, m} \\
D_h & \quad \text{hydraulic diameter of the flow passage, } 2H, \text{ m} \\
f & \quad \text{Fanning friction factor, } H/L((\Delta p/\rho u_{in}^2)), \text{ dimensionless} \\
H & \quad \text{spacing between the fins, m} \\
h & \quad \text{local heat transfer coefficient, W/(m}^2\cdot\text{K}) \\
h_m & \quad \text{overall (flow-length average) heat transfer coefficient, W/(m}^2\cdot\text{K}) \\
h_x & \quad \text{spanwise-average heat transfer coefficient, } q_x/(T_{\text{w,m}}-T_m), \text{ W/(m}^2\cdot\text{K}) \\
j & \quad \text{Colburn factor, } Nu_m/(RePr^{1/3}), \text{ dimensionless} \\
L & \quad \text{length of the flow passage, m} \\
l & \quad \text{chord length of the VG, m} \\
Nu & \quad \text{local Nusselt number, } hD_o/k, \text{ dimensionless} \\
Nu_m & \quad \text{flow-length average Nusselt number, } h_xD_o/k, \text{ dimensionless} \\
Nu_x & \quad \text{spanwise-average Nusselt number, } h_xD_o/k, \text{ dimensionless} \\
Pe & \quad \text{Péclet Number, } u_mD_o/\alpha, \text{ dimensionless} \\
Pr & \quad \text{Prandtl number, } \mu c_p/k, \text{ dimensionless} \\
\Delta p_t & \quad \text{total pressure difference, Pa} \\
Q & \quad \text{heat flow rate, W} \\
q_w & \quad \text{local wall heat flux, W/m}^2 \\
q_x & \quad \text{spanwise-average wall heat flux, } k[(\partial T/\partial n)_{w,m}], \text{ W/m}^2 \\
Re & \quad \text{Reynolds number, } \mu u_mD_o/\mu, \text{ dimensionless} \\
S_t & \quad \text{transverse pitch of the cylindrical tubes, m} \\
S_l & \quad \text{longitudinal pitch of the cylindrical tubes, m} \\
s & \quad \text{height of VG, m} \\
T_m & \quad \text{bulk-mean fluid temperature, K} \\
T_{w,m} & \quad \text{spanwise-average wall temperature, K}
\end{align*}
\]
\( \Delta T_{lm} \) log-mean temperature difference, K
\( t \) thickness of VG, m
\( u_{in} \) inlet velocity, m/s
\( u_m \) mean velocity, m/s
\( x_P, z_P \) local points indicating the position of VGs relative to the reference tube

Greek symbols
\( \alpha \) thermal diffusivity, \( k/\rho c_p \), m\(^2\)/s
\( \beta \) angle of attack of VG, deg
\( \delta \) thickness of the fin, m
\( k \) thermal conductivity, W/(mK)
\( \mu \) dynamic viscosity, Pa s
\( \rho \) density, kg/m\(^3\)

2. Numerical simulation

2.1 Statement of the problem

Figure 2 illustrates a schematic of the two-row staggered fin-tube geometry with delta winglet-type vortex generators installed at the first row of tubes. Two rectangular plates in parallel form a flow passage, which is \( H \) in height, \( B (=7.5H) \) in width and \( L (=26H) \) in length. Circular tubes of outer diameter \( D (=5.4H) \) are arranged in staggered form and their streamwise and spanwise pitches (\( S_l \) and \( S_t \)) are 13\( H \) and 15\( H \), respectively. Each VG has a chord length \( l (=5.4H) \) and height \( s (=0.9H) \). The geometric parameters varied and investigated here are the streamwise and the spanwise positions of VG, \( \Delta x \) and \( \Delta z \), and the angle of attack, \( \Delta \beta \), to the main flow direction as illustrated in Fig. 2(b). Note that the reference configuration, denoted here by ref., is similar to that proposed by Torii et al. \(^9 \) but is different in that some modifications are made in some aspects, i.e., (1) the number of tube rows being reduced from three to two, (2) both entrance length and exit length of the fin plate being shortened from \( S_l \) to \( S_l/2 \), and (3) the spanwise pitch of tubes being changed from 13.39\( H \) to 15\( H \). The present variations (\( \Delta x \), \( \Delta z \) and \( \Delta \beta \)) are defined as the differences from the reference and they are respectively referred to as \( x \)-shifting, \( z \)-shifting and \( \beta \)-adjusting in this paper. Table1 summarizes all configurations considered in the present study.

![Fig. 2 Schematic of two-row fin-tube geometry: (a) top view, (b) a local coordinate system identifying position of VG, (c) side view.](image-url)
2.2 Governing equations and numerical schemes

The three-dimensional continuity, Navier-Stokes and energy equations for laminar air \((Pr=0.691)\) flow characterized by steady state, constant-property conditions, with negligible axial conduction \((Pe>>1)\) and viscous dissipation \((Br<<1)\) are

\[
\frac{\partial u_i}{\partial x_i} = 0 \tag{1}
\]

\[
\frac{\partial u_j}{\partial x_j} = -\frac{1}{\rho} \left( \frac{\partial p}{\partial x_j} \right) + \nu \left( \frac{\partial^2 u_i}{\partial x_i x_j} \right) \tag{2}
\]

\[
\frac{\partial T}{\partial x_j} = \alpha \left( \frac{\partial^2 T}{\partial x_i x_j} \right) \tag{3}
\]

The governing equations are discretized and solved by the finite volume method using a commercially available CFD code \(^{(10)}\). Second order spatial discretization scheme MARS (Monotone Advection and Reconstruction Scheme) is chosen for the convective terms because of its least sensitivity to the unstructured mesh employed currently. The SIMPLE is used for the pressure field calculation. More details on the numerical scheme can be found in Song and Nishino \(^{(11)}\).

2.3 Computational domain and boundary conditions

Figure 3 depicts the computational domain and boundary conditions imposed. Only a quarter of the experimental test core is employed for the main solution domain. The solid parts such as the fin, the tubes and VGs are excluded in the course of grid generation.

As shown in Fig. 3, a sufficiently long extended domain is added to the downstream of the main solution domain to assure the solution stability and accuracy in the computation.
This special care is to prevent the outlet boundary from cutting strong recirculating flows behind the second tube row in the main domain. Hence, a conventional outlet boundary condition is imposed on the downstream cross-section of the extended domain. For other parts of the solution domain, the following simple and satisfactory boundary conditions are applied:

- **Inlet**: Predetermined and uniform velocity and constant fluid temperature ($T_i=293K$) are assumed.
- **Outlet**: Zero gradients of $u$, $v$, $w$, $p$ and $T$ are assumed over the outlet plane.
- **Side planes**: Symmetric plane boundary condition is imposed.
- **Free stream planes** (on the top and bottom surfaces of the extended domain and the holes stamped): Cyclic boundary conditions are applied.
- **Solid surfaces** (on the surfaces of the fins, the tubes and VG): No-slip and constant temperature ($T_w=323K$) boundary conditions are assumed at all solid walls.

2.4 Grid independence test

The dependency of the computed $j$- and $f$-factors on the grid density is investigated to improve the accuracy of the computational results. A grid independence test is conducted for the reference configuration at $Re=400$.

In order to capture steep gradients of local quantities associated with flow and heat transfer more accurately, the grids are sufficiently refined in the regions close to the tubes and VGs, as seen in the magnified insets of Fig. 4. Additional local mesh refinement is made in the normal $y$-direction of the computational domain due to the same reason. Then two parameters $n_x$ and $n_z$, which are the numbers of grids in the streamwise and spanwise directions, are adjusted. Figure 5 shows the results of the grid independence test. There is no appreciable difference in the computed results of the $j$- and $f$-factors with respect to the total number of grids, mainly, due to the precaution against the grid sensitive regions. The grid of 518,000 is chosen, where the numbers of basic grids in $x$-, $y$- and $z$-directions are $130\times30\times37$ for the main domain and $60\times30\times37$ for the extended one, and the number of locally refined grid is about 307,000. The same basic grid density guarantees the accuracy of the computational results for the other configurations.
3. Definition and evaluation of performance parameters

Throughout this paper, the Reynolds number \( Re \) is based on the inlet velocity \( u_{in} \) and the hydraulic diameter \( D_h = 2H \) of the flow passage, defined as

\[
Re = \frac{\rho u_{in} D_h}{\mu}
\]

Pressure loss data are represented in terms of the Fanning friction factor \( f \) specified with the total pressure difference \( \Delta p_t \) across the main solution domain as follows:

\[
f = \frac{H}{L} \left[ \frac{\Delta p_t}{\rho u_{in}^2} \right]
\]

Here the total pressure difference \( \Delta p_t \) is evaluated by integrating local total pressure over the cross-sections of the inlet and outlet of the main domain.

The spanwise-average Nusselt number \( Nu_x \) is defined in Eq. (6) in order to demonstrate the effect of VGs on local heat transfer enhancement (see, Figs. 9 and 11), as follows:

\[
Nu_x = \frac{h_x D_h}{k} = \frac{q_x D_h}{k(T_{w,m} - T_m)} = \frac{D_h[(\partial T / \partial n)_{w,m}]}{(T_{w,m} - T_m)}
\]

where \( h_x, q_x, T_{w,m} \) and \( T_m \) are the spanwise-average heat transfer coefficient, the spanwise-average heat flux, the spanwise-average wall temperature and the bulk-mean fluid
temperature, respectively. Definitions of $T_{w,m}$ and $T_m$ in Eq. (6) are expressed in Eqs. (7) and (8), respectively.

$$T_{w,m} = \frac{1}{B(x)} \int_{B(x)}^{\infty} T_w \, dz$$

where $B(x)$ is the local fin width exposed to the fluid along the flow passage.

$$T_m = \frac{1}{A c u_m} \int u T \, dA_c$$

where $A_c$ and $u_m$ are the flow cross-sectional area and the mean velocity at a streamwise position along the flow passage.

The overall (flow-length average) heat transfer coefficient $h_m$ is based on the heat flow rate $Q$ computed by the energy balance in the main solution domain and the logarithmic mean temperature difference (LMTD) $\Delta T_{lm}$ defined as

$$\Delta T_{lm} = \frac{(T_{w,l} - T_{m,l}) - (T_{w,0} - T_{m,0})}{\ln((T_{w,l} - T_{m,l}) / (T_{w,0} - T_{m,0}))}$$

where ($T_{w,l}$-$T_{m,l}$) and ($T_{w,0}$-$T_{m,0}$) are the temperature differences between the wall and the fluid at the inlet and outlet of the main solution domain, respectively.

4. Results and discussion

Validation of the present computation is provided by Song and Nishino (11) and found to be well established, and therefore described only briefly here. Steady-state computations have been performed for a hydrodynamically and thermally developing flow between parallel plates with constant wall temperature boundary condition (CWT), for which the theoretical solutions of heat transfer and pressure drop are available from the literature. As shown in Fig. 6, the computational results of heat transfer and flow-friction in terms of the Colburn factor $j$ and the Fanning friction factor $f$ as a function of the Reynolds number $Re$ are in excellent agreement with the corresponding theoretical solutions of Mercer et al. (12) and Liu (13), respectively.

It may be mentioned that no unsteadiness is confirmed in the flow and temperature fields of the unsteady solutions for the present flow condition ($Re=400$) and thereby justifying the present computation to be steady-state.

Figures 7 through 11 demonstrate some beneficial effects of placing VGs near the first tube row. The reference configuration defined in Section 2.1 is chosen for this purpose, and it is compared with the baseline one. Figure 7 shows the flow and temperature fields for the baseline and reference configurations at $y=H/2$, represented in terms of temperature.

![Fig. 6 Validation of CFD code: a comparison of computational $j$ and $f$ with theoretical values for flow through parallel plates (11).](image-url)
contours and streamlines, respectively. It is clearly seen that the presence of the VG near the first tube change the flow and temperature fields behind the tube wake and the region downstream of the VG. This change ultimately serves to give rise to heat transfer enhancement of those areas.

Figure 8 compares the local Nusselt number $Nu$ distributions for the baseline and reference configurations, where $Nu$ is defined by Eq. (6) after replacing $q_w$ and $(T_{w,m} - T_m)$ with the local wall heat flux $q_w$ and the logarithmic mean temperature difference $\Delta T_m$. It is clearly seen that the presence of VG augments heat transfer considerably. In particular, the augmentation is found in the area close to VG as well as in the area behind the first tube row. The reduction of that poor heat transfer area is clear evidence of the overall heat transfer enhancement resulting from the reference configuration.

![Fig. 7 Local fluid temperature distributions and streamlines for flow past the baseline (above) and reference (below) configurations at $y=H/2$.](image1)

![Fig. 8 Local Nusselt number distributions for the baseline (above) and reference (below) configurations.](image2)

Figure 9 shows the spanwise-average Nusselt number variations in the streamwise direction. For the baseline configuration, an abrupt $Nu_x$ drop is observed at the entrance. This is followed by a slight rise in front of the first tube, presumably due to the formation of horseshoe vortices. Then, $Nu_x$ decreases and reaches a minimum right behind the tube. The reason for that decrease is the presence of a large and long wake zone behind the tube. In the wake, the heat transfer slightly increases and meets a second peak at the stagnation point (or line), then again repeats the same variation that is observed in the downstream of the first tube row. For the reference configuration, heat transfer is further enhanced by longitudinal vortices generated by VG. The enhancement is pronounced near the VG and in
the first tube and their wakes, as is also evident from Fig. 8.

Fig. 9 Streamwise variation of the spanwise-average Nusselt numbers $N_u$ for the baseline and reference configurations.

Fig. 10 Streamwise variations of the dimensionless pressure drop ($\frac{dp}{\rho U_{in}^2}$) for the baseline and reference configurations.

In gas-to-liquid heat exchangers, the pressure loss expenditure is of the same importance as the heat transfer enhancement $^2$. Figure 10 indicates the streamwise variations of static pressure drop in dimensionless form. As compared to the baseline configuration, the reference one leads to higher pressure drop, presumably due to the form drag of VG.

In Fig. 11, heat transfer enhancement and pressure loss penalty are represented in terms of $\frac{N_u}{N_{u,0}}$ and $\frac{dp}{dp_{0,0}}$. A local heat transfer is augmented by 66% immediately behind VG and the local heat transfer enhancement $\frac{N_u}{N_{u,0}}$ is generally larger than the local pressure loss penalty $\frac{dp}{dp_{0,0}}$.

Fig. 11 Streamwise variations of the heat transfer enhancement $\frac{N_u}{N_{u,0}}$ and pressure loss penalty $\frac{dp}{dp_{0,0}}$ for the reference configuration.

Figures 12 through 14 address the computational results of heat transfer and pressure loss for all configurations considered here to account for geometric variations such as $z$-shifting, $x$-shifting, $\beta$-adjusting and their combinations.

First of all, it is found in Fig. 12 that $z$-shifting of VG toward the tube brings about larger heat transfer enhancement than pressure loss enhancement for all cases. It is clearly observed that the heat transfer increases to a maximum of 22.5% at the position corresponding to the configuration, $-0.8Hz$. On the contrary, the pressure loss increases continuously with steeper rise near the tube. As a result, on the basis of $z$-shifting, there is a maximum in terms of the net enhancement in thermal-hydraulic performance at the $z$-position corresponding to the configuration, $-0.4Hz$, as can be found in Fig. 14.

Figure 13(a) depicts local Nusselt number distributions on the top wall of the solution domain resulting from each $z$-shifting. Compared to the baseline configuration, the presence
of VG appears to make a considerable difference in the local Nusselt number distribution. It is also observed that z-shifting toward the tube reduces the poor heat transfer zone behind the first tube row.

The influence of x-shifting on the heat transfer and pressure loss characteristics is found in Figs. 12, 13(b) and 14, respectively. As compared to the reference configuration, upstream shifting of VG results in a slight heat transfer increase (-1Hx) but a further shifting causes deterioration of heat transfer with the same rate of pressure loss rise (-2Hx), and hence bringing about low net enhancement. Figure 13(b) illustrates the effect of x-shifting on the local Nusselt number distribution. It is conjectured that the heat transfer enhancement by the upstream shifting of VG is closely related to the boundary layer thickness. Based on x-shifting, a maximum heat transfer enhancement of 19.3% is obtained at the configuration of -1Hx, while the highest net heat transfer of 8.6% of all VG configurations investigated presently is gained with the configuration of +1Hx.

An angle of attack is a very important parameter associated with the heat transfer and pressure loss characteristics. Figure 12 indicates that the β-adjusting from 10° to 20° makes a monotonous increase in both heat transfer and pressure loss. The heat transfer enhancement is mainly due to enlargement of the effective width of VG to the main flow. This generates larger longitudinal vortices and thereby increases more the heat transfer. On the other hand, an excessive enlargement of the effective VG width will also bring about a larger form drag and hence increase pressure loss penalty.

The results of some combined parameters are found in Fig. 12. It is interesting to note that when the angle of attack is fixed at 20° and some x-shifting is made the heat transfer enhancement exceeds 25%, and the highest heat transfer enhancement of 26.5% is obtained for the -0.8Hz+5° configuration.

Net enhancement in the thermal-hydraulic performance is presented in Fig. 14 in terms of the performance parameter (j/j₀)/(f/f₀) for all VG configurations investigated in the present study. In contrast to the result provided in Fig. 12, the configuration of +1Hx shows the best thermal-hydraulic performance with value of 8.6%. In the meantime, the configuration of -1.2Hz shows the poorest performance with value of 3%. As a consequence, it can be concluded that effects of geometric parameters of VG bring about approximately 5% variation of the net enhancement.

![Heat transfer enhancement, j/j₀ and Pressure loss penalty, f/f₀](image.png)

**Fig. 12** Thermal-hydraulic performances for parametric VG configurations in terms of the heat transfer enhancement j/j₀ and the pressure loss penalty f/f₀ at Re=400.
Fig. 13 Local Nusselt number distributions for the baseline and some VG configurations on the top wall ($y=H$): each for (a) $z$-shifting, (b) $x$-shifting and (c) $\beta$-adjusting.

Fig. 14 Net enhancement in the thermal-hydraulic performance for parametric VG configurations.

5. Concluding remarks

Systematic numerical simulations have been carried out for flow through a passage of a compact fin-tube heat exchanger with built-in single pair of VGs. Various geometric parameters of VGs installed beside the first row of two-row staggered tube bundles at $Re=400$ are considered. It is expected that the computational results presented in this paper can be used not only to gain new insight into the heat transfer and flow-friction characteristics for compact fin-tube heat exchangers with built-in VGs, but also to design further advanced heat transfer surfaces.

The conclusions of the present study can be summarized as follows:
1. Spanwise shifting of VGs toward the tube-side shows a clear tendency that the heat transfer increases to a maximum at a certain position (-0.8Hz) while the pressure loss
Increases continuously with a steeper rise near the tube. Consequently, there is a maximum of the net enhancement in the thermal-hydraulic performance at a certain z-position of VGs (-0.4Hz, presently).

2. Upstream shifting of VGs brings about increase in both heat transfer and pressure loss up to a certain position (-1Hx). However further upstream shifting leads to deterioration of heat transfer together with the same rate of pressure loss rise (-2Hx), and thereby resulting in low net enhancement.

3. Increasing attack angle from 10° to 20° results in a monotonous increase in both heat transfer and pressure loss. It is found that the rate of net enhancement is almost constant.

4. The maximum heat transfer enhancement of 26.5% is achieved at the configuration with the spanwise shifting of -0.8Hz and with the attack angle adjusting of 20° (denoted by -0.8Hz+5° in the paper). However, its net enhancement is at most 3.9%. The highest net enhancement, 8.6%, is gained with the configuration of +1Hx.

References


(10) STAR-LT 2005 Methodology, (2005), CD-adapco JAPAN Co. LTD, Chapter 4.

