Heat-Transfer Enhancement by Artificially Generated Streamwise Vorticity

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Abstract. Vortex-induced heat transfer enhancement exploits longitudinal and transverse pressure-driven vortices through the deliberate artificial generation of large-scale vortical flow structures. Thermal-hydraulic performance, Nusselt number and friction factor are experimentally investigated in a HEV (high-efficiency vortex) mixer, which is a tubular heat exchanger and static mixer equipped with trapezoidal vortex generators. Pressure gradients are generated on the trapezoidal tab initiating a streamwise swirling motion in the form of two longitudinal counter-rotating vortex pairs (CVP). Due to the Kelvin–Helmholtz instability, the shear layer generated at the tab edges, which is a production site of turbulence kinetic energy (TKE), becomes unstable further downstream from the tabs and gives rise to periodic hairpin vortices. The aim of the study is to quantify the effects of hydrodynamics on the heat- and mass-transfer phenomena accompanying such flows for comparison with the results of numerical studies and validate the high efficiency of the intensification process implementing such vortex generators. The experimental results reflect the enhancement expected from the numerical studies and confirm the high status of the HEV heat exchanger and static mixer.

1. Introduction

Transverse and longitudinal vortices enhance the exchange of fluid particles between different flow regions with relatively small increases in pressure loss [1]. The ability of these vortices to increase velocity fluctuations and flow momentum redistribution leads to better heat and mass transfer and intensifies convective phenomena and turbulent mixing with no need for external mechanical forces [2,3]. It was shown that most heat- and mass-transfer enhancement is provided essentially by streamwise vorticity, while the transverse stationary vorticity slightly enhances heat transfer in the region near the vortex generator, generally in the near-wake of the vortex generator [1,4].

Vortices can be created by flow separation behind vortex generators or turbulence promoters such as the transient structures produced by Kelvin-Helmholtz instabilities and the counter- or co-rotating vortices caused by pressure gradients upstream and downstream of the flow perturbators investigated in the present study [2, 3,5].
These structures can be artificially generated in open or internal flows such as those in multifunctional heat exchangers/reactors (MHE/R’s) widely used in industry. The high-efficiency vortex (HEV) is a MHE/R designed to exploit the vortex system described above. It is a circular tube equipped with rows of trapezoidal vortex generators. The tabs are fixed at a 30° inclination angle with respect to the wall [6], as shown in figure 1a.

Numerical results demonstrate that the convective heat transfer characterized by Nusselt number is enhanced by 500% in the HEV static mixer over that in empty-tube exchangers. Computations of global mixing efficiency based on turbulence kinetic energy dissipation rate show the ability of the HEV static mixer to enhance mixing by 40 times over that in a plain turbulent tube flow reactor [7]. A moderate increase in friction factor and pressure losses accompanies this valuable enhancement.

This paper is organized as follows: first, the flow structure is discussed. The experimental setup and measurement techniques are then introduced. Results and discussion are then presented, and finally some conclusions are drawn.

Figure 1. (a) HEV geometry and (b) flow structure produced behind a trapezoidal vortex generator

2. Flow hydrodynamics
Flow past trapezoidal vortex generators has been extensively studied due to its ability to enhance turbulent mixing, mass transfer and phase dispersion by generating complex coherent structures [8]. Mainly, two types of flow structures are observed downstream from a trapezoidal vortex generator: a counter-rotating vortex pair formed by the pressure difference across the vortex generator, and a periodic sequence of horseshoe-like structures shed from the trailing edges of the vortex generator [5], as shown in figure 1b.

Numerical studies examining the mean streamwise velocity distribution [3] show that the axial velocity above the tab is greater than that in the wake region; the low-pressure region thus created above the tab initiates a swirling motion that produces the two large symmetric counter-rotating vortices. This large motion entrains fluid particle transfer between the low-momentum region in the tab wake and the high-momentum fluid in the core region.

In addition to the counter-rotating vortex pairs, shear layers are formed on the front and rear edges of the generator, creating an energy transport that enhances heat transfer locally and globally. High-velocity fluctuations caused by these shear layers, which can be characterized by the turbulent kinetic energy, result in flow fluctuations and can form a self-sustained oscillatory flow. Due to the Kelvin-Helmholtz instability, these shear layers give rise to the periodic “hairpin vortices” that propagate on the vortex pair [9], as shown in figure 1b.

The following section presents an experimental study based on heat transfer measurements to examine and quantify the improvement in turbulent flow with aligned vorticity generators compared to turbulent straight pipe flow, and analyze the effects of the flow structure described on the thermal behavior of this heat exchanger.
3. Experimental setup and procedures

3.1. Flow loop and test section

The open flow loop used for the experiments consists of a reservoir from which water is circulated by a variable-speed gear pump. The reservoir temperature is held constant by a thermostat. The volume flow rate is measured by three parallel flowmeters with an accuracy of 2% over the measured range. Upstream of the flowmeters, a high-resolution ball valve is used further to adjust the flow rate after setting the required pump speed to cover the Reynolds numbers, which ranged between 2000 and 15000 (corresponding to flow rates between 130 L/h and 960 L/h). Before entering the test section, the fluid passes through a 2-meter-long straight tube whose length is 100 times its inner diameter, sufficient to ensure a fully developed velocity profile at the inlet of the test section and thus eliminate entrance effects. The circuit is equipped with a safety valve and a pulsation damper to limit any pressure fluctuations produced by the pump and to ensure continuity and stability of flow.

The HEV test section in figure 1a is composed of a stainless steel tube in the walls of which seven arrays of aligned trapezoidal vortex generators are inserted. Each array is constituted of four tabs inclined at 30° relative to the wall in the flow direction. The tube is 145 mm long, 22.7 mm in diameter and has wall thickness 1.15 mm. The distance between two successive tab rows is 22 mm.

At the test section inlet, the working fluid (water) properties taken at 25°C are as shown in Table 1. The flow Reynolds number is defined by:

\[
Re = \frac{U_m D}{\nu}
\]

All thermophysical properties of water are determined at the fluid bulk temperature \( T_b \).

3.2. Heating and data acquisition

The HEV heat exchanger is housed inside a stainless steel heating cylinder of 55 mm inner diameter. The 10 mm annular gap between the heat exchanger and the heating cylinder is filled with heat-conducting grease of 0.7 W/mK thermal conductivity. The outer cylinder is heated by the Joule effect by a continuous heating wire wound around it, as shown in figure 2. This indirect heating technique is used to provide a uniform wall heat flux to the HEV heat exchanger. The two ends of the heating wire are connected to a transformer; the electrical output needed to generate a 200 watt uniform heat flux in the wire is controlled by adjusting the voltage (50 V) and the current (4 A) with precision 1%.

The temperature gradient across the annular gap is used to calculate the constant heat flux provided on the HEV outer wall. For this purpose temperature is measured at 15 different longitudinal positions by two sets of type-K thermocouple beads with an accuracy of ±1.5°C, one set welded on the HEV heat exchanger outer wall and the other on the inner wall of the heating tube. Calibrated temperature sensors are used to measure the inlet and outlet water temperatures. The whole test section is insulated with a polystyrene box fitted with a sensitive heat fluxmeter to measure convective losses. The heat flux at the test wall is set by the transformer and the system is then allowed to reach thermal steady state for each value of flow Reynolds number. Inlet, outlet and longitudinal temperatures are measured continuously via an ‘Agilent’ data acquisition chain and numerically handled using ‘BenchLink Data Logger’ software.

Pressure drop is measured by two differential manometers covering all head losses with a precision depending on the measured value and varying between 0.1 and 0.25%. The friction factor is then calculated in terms of head losses. Pressure drop measurements are carried out under isothermal

<table>
<thead>
<tr>
<th>( \rho ) ( \text{kg m}^{-3} )</th>
<th>( C_p ) ( \text{J kg}^{-1}\text{K}^{-1} )</th>
<th>( \nu ) ( \text{m}^2\text{s}^{-1} )</th>
<th>( \kappa ) ( \text{W m}^{-1}\text{K}^{-1} )</th>
<th>( Pr )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>998</td>
<td>4186</td>
<td>( 10^{-6} )</td>
<td>0.60</td>
</tr>
</tbody>
</table>
conditions. Experimental uncertainties are evaluated for the derived quantities using the relative errors of measurement of their elementary terms as given by:

\[ \frac{\Delta z}{z} = \frac{1}{z} \sum_i \frac{\partial z}{\partial x_i} \Delta x_i \]  

where \( z \) is the derived quantity depending on \( n \) variables, \( x_1, x_2, ..., x_n \), while \( \Delta z \) and \( \Delta x \) represent the absolute error in the calculation of \( z \) and the measurement of \( x \) respectively.

4. Results and discussion

4.1. Convective heat-transfer coefficient

Several mechanisms interact and concomitantly produce the heat-transfer enhancement in the HEV heat exchanger. The shear layer detached from the vortex generator is the site of high production of turbulent kinetic energy whose value increases in the longitudinal direction up to the fourth tabs array [3,10]. Before the breakdown into turbulence, the Kelvin-Helmholtz instability in this layer generates hairpin vortices that produce a more uniform temperature distribution in the flow cross section that intensifies the heat transfer. The vortex generator acts as a thermal fin and injects additional heat into the main flow. The temperature evolution is related to the development of the thermal boundary layer, which is in turn controlled by the tab succession: each tab row disrupts the wall boundary layer; the boundary layer is then regenerated on the heat exchanger wall before being disrupted again by the next tab row downstream. This successive disruption of the wall boundary layer contributes to heat-transfer enhancement. However, a disadvantage of the HEV heat exchanger is the hot spots generated behind the tabs due to flow recirculation.

The temperatures \( T_e \) and \( T_w \) measured at the different longitudinal positions are used to calculate the local heat flux densities by:

\[ \varphi = \frac{\lambda}{e} (T_e - T_w) \]  

The longitudinally averaged flux density is then used as the heat source over 15 incremental longitudinal segments making up the total heat exchanger wall. The inlet and exit temperatures are used to calculate a mean temperature \( T_m \) in each segment. The local convective heat transfer coefficients \( h \) are calculated using temperatures \( T_w \) and \( T_m \):

\[ h = \frac{\varphi}{(T_w - T_m)} \]  

The local Nusselt numbers are then calculated by:

\[ Nu = \frac{hD}{\kappa} \]
The longitudinal evolution of the local Nusselt number calculated for different Reynolds numbers is shown in figure 3. The experimental data uncertainties are determined and the maximum uncertainty associated with the Nusselt number is estimated at 3.6%. The longitudinal increase of the Nusselt number is related to the increase in heat transfer coefficient, \( h \), produced by the flow hydrodynamics.

![Figure 3](image3.png)  
**Figure 3.** Local Nusselt number for different Reynolds numbers

Each row of trapezoidal tabs regenerates the vortex pairs continuously, sustaining the radial transport phenomena that carry hot fluid from the wall region to the core flow and thus homogenizing the cross-sectional temperature. The increase in the Reynolds number also promotes the convective heat transfer. The turbulent structures thus created in the flow replace the parallel laminar streamlines, enhance fluid mixing and intensify the effect of the embedded vortical structures to produce higher Nusselt numbers.

The pair of longitudinal vortices generates longitudinal vorticity and redistributes heat in the tube cross section. The secondary flow in the symmetry plane of the tab transfers heat from near the hot wall towards the cold region in the core flow, increasing the radial heat transfer. These results confirm the numerical results of Habchi *et al.* [10] for the same heat transfer conditions and flow rates.

### 4.2. Global thermal performance

The global convective heat transfer coefficient \( h_{\text{global}} \) and the global Nusselt number \( Nu_g \), which represents the ratio of convective to conductive heat transfer, define the thermal performance of a heat exchanger. The global Nusselt numbers are calculated using the longitudinally averaged heat flux density over the heat exchanger length \( \phi \), the heat exchanger average wall temperature \( T_w \), and the average fluid temperature \( T_m \). Measured values of the global convective heat transfer coefficient \( h_{\text{global}} \) and the global Nusselt number \( Nu_g \) are shown as a function of Reynolds number in table 2. Figure 4 shows the variation of global Nusselt number with Reynolds number. These results confirm those of previous numerical studies [3,10] showing a large increase in the Nusselt number with Reynolds number: for all experiments in the current study, Nusselt number increases with Reynolds number. This increase is attributed, among other parameters, particularly to the enhancement of turbulence intensity. The growth of Nusselt number in the longitudinal direction reflects also the renewal of the thermal boundary layer after each row of vortex generators.

Gnielinski [11] gives a correlation for Nusselt number in turbulent tube flow \( Nu_0 \) to Prandtl numbers ranging from 0.5 to 200 and Reynolds numbers from 2300 to 5×10^6:

![Figure 4](image4.png)  
**Figure 4.** Global Nusselt number, \( Nu_g \)
Table 2. Experimental values of $h_{global}$ and $Nu_g$

<table>
<thead>
<tr>
<th>$Re$</th>
<th>2160</th>
<th>4530</th>
<th>6390</th>
<th>8670</th>
<th>10460</th>
<th>12860</th>
<th>14920</th>
</tr>
</thead>
<tbody>
<tr>
<td>$h_{global}$ [Wm$^{-2}$k$^{-1}$]</td>
<td>1192</td>
<td>1781</td>
<td>2076</td>
<td>2300</td>
<td>2549</td>
<td>2682</td>
<td>3023</td>
</tr>
<tr>
<td>$Nu_g$</td>
<td>45</td>
<td>67</td>
<td>79</td>
<td>87</td>
<td>96</td>
<td>101</td>
<td>114</td>
</tr>
</tbody>
</table>

\[
Nu_0 = \frac{(f_0/8)(Re - 1000)Pr}{1 + 12.7(f_0/8)^{1/2}(Pr^{2/3} - 1)}
\]  

(6)

where $f_0$ is the friction factor in a smooth-pipe turbulent flow given by:

\[
f_0 = 0.079Re^{-0.25}
\]  

(7)

The numerical studies of Mohand Kaci et al. [3] predict a 500% increase in average convective heat transfer in the HEV flow over a turbulent tube flow for a matching range of Reynolds numbers. Figure 4 compares the global experimental Nusselt number $Nu_g$ with those calculated using the Gnielinski correlation in a turbulent tube flow, $Nu_0$. The results confirm the numerical predictions of [3]: in the low-Reynolds-number zone, where the flow is transitional in the plain tube and heat transfer is weak, the HEV heat exchanger shows relatively high Nusselt numbers. When a fully turbulent regime is attained, the enhancement is less important. To characterize this relative enhancement, we define the rate of relative heat-transfer intensification $\chi$ (Eq. 8) that reflects the increase in Nusselt number in the HEV heat exchanger compared to a turbulent tube flow:

\[
\chi = \frac{Nu_g - Nu_0}{Nu_0}
\]  

(8)

Figure 5 shows the index of relative intensification with a 5% error margin. From this figure, it can be concluded that the vortical enhancement of heat transfer by HEV geometry increases the Nusselt number by between 800% to 200% over that of a turbulent tube flow for the Reynolds numbers ranging between 2000 and 15000, respectively. The HEV produces considerable enhancement in the low-Reynolds-number zone with a relative enhancement that decreases for higher Reynolds numbers as the turbulent structures automatically start to appear in the plain tube, intensifying the convective transfer processes and reducing the sharp difference between Nusselt numbers in laminar regimes and the HEV flow.

Figure 5. Index of relative intensification of convective transfer, $\chi$

Figure 6. Variation of friction factor ratio, HEV/plain tube
4.3. Friction and Colburn factors

Based on isothermal differential pressure measurements between the entrance and exit from the heat exchanger, the friction factor is calculated using the Darcy-Weisbach equation (Eq. 9) and plotted against the Reynolds number. Over the length of the test section, the maximum pressure drop 850 Pa was attained in the highest-Reynolds-number run ($Re = 15000$):

$$f = \frac{\Delta P}{\frac{L}{D} \left( \frac{\mu U}{2} \right)}$$  \hspace{1cm} (9)

Figure 6 compares the ratio of the HEV flow friction factor to that in a turbulent tube flow (denoted $f_0$). The tab rows generate additional losses by friction due to flow interruption and thus increased inertial forces in the boundary layer. The HEV friction factors are 15 to 19 times above those in a tube flow. The uncertainty of the friction factor calculations varies between 0.1 and 0.25% depending on the measured values of head losses. These values prove the HEV to be among the most efficient exchangers compared to other heat exchangers/mixers producing similar heat-transfer enhancement with similar diameters, as shown in figure 7. Only the empty-pipe flow and the helical tube exhibit inferior friction factors. However, the heat transfer in these geometries remains modest. In the other exchangers, the gain in heat transfer is accompanied by elevated pumping costs, with friction factors reaching up to ten times those in the HEV flow, as for example in the Sulzer SMX. The intermittent and relatively slight protuberance of the trapezoidal tabs into the core flow compared to the voluminous inserts in the Kenics, V-nozzle and Sulzer geometries allows free fluid passage in most of the cross section and is reflected in the moderate friction factors.

Another criterion for judging the energy efficiency of a heat exchanger is the Colburn factor $j$:

$$j = \frac{Nu}{Re Pr^{1/3}}$$  \hspace{1cm} (10)

The Colburn factor quantifies the ratio of thermal power transferred to mechanical power consumed; a higher Colburn factor indicates more heat transfer. Numerical studies [3] show a Colburn factor of 0.01 for HEV flow, an improvement of 500% over the turbulent plain-tube flow ($j = 0.002$). The comparison of the Colburn factor for HEV flow with that for other geometries (figure 8) shows the HEV’s capacity to enhance heat transfer with moderate energy consumption. The high friction factors of the Sulzer SMX are reflected in a low Colburn factor. The weak convective transfer in straight and helical empty pipes keeps their Colburn factors relatively low even if their energy costs are less. The Helical Kenics with its complex fabrication and maintenance processes is the only geometry with an efficiency superior to that of the HEV.

Figure 7. Evolution of friction factors of different exchangers (adapted from Thakur et al. [12])

Figure 8. Colburn factor for common commercial heat exchangers (Mohand Kaci et al. [3])
5. Conclusions and perspectives
An experimental thermal study is carried out to investigate heat-transfer enhancement in a tube fitted with trapezoidal tabs acting as vortex generators and turbulence promoters. In the HEV multifunctional heat exchanger geometry studied here, seven aligned arrays of four tabs each create a complex vortex system that enhances radial heat and mass transfer by intensifying convective exchange phenomena. A global analysis of the results based on the available literature describing the predicted flow structure is presented. The values of Nusselt number and friction factor in the tube equipped with vorticity generators are higher than those in a plain tube. The protuberance of the solid tabs in the flow produces additional losses compared to a plain tube, but simultaneously contributes to heat transfer by the fin effect and also by extra mixing of cold and hot fluid particles. The overall effect of losses is moderate compared to those produced in similar devices. The experimental study validates the numerical results of [3,10] and confirms the capacity of the HEV heat exchanger to improve convective heat transfer while keeping pressure losses moderately low.

Equipment to test the alternated geometry, a variant of the current test section in which the tab arrays are no longer aligned but instead are alternated so that each row of trapezoidal tabs is rotated by 45° with respect to its predecessor, is under construction. Recent numerical results [8] have shown the interest of such modified geometry in terms of mass and heat transfer, but experimental validation is needed.

Nomenclature

- \( C_p \) specific heat (J kg\(^{-1}\)K\(^{-1}\))
- \( D \) heat exchanger diameter (m)
- \( e \) grease layer thickness (m)
- \( f \) friction factor
- \( h \) heat-transfer coefficient (Wm\(^{-2}\)K\(^{-1}\))
- \( j \) Colburn factor
- \( k \) turbulent kinetic energy (m\(^2\)s\(^{-2}\))
- \( L \) heat exchanger length (m)
- \( Nu \) Nusselt number
- \( Pr \) Prandtl number
- \( Re \) Reynolds number
- \( T \) temperature (K)
- \( U_m \) mean axial velocity (m s\(^{-1}\))
- \( \epsilon \) turbulent energy dissipation rate (m\(^3\)s\(^{-3}\))
- \( \kappa \) thermal conductivity of water (Wm\(^{-1}\)K\(^{-1}\))
- \( \lambda \) thermal conductivity of grease (Wm\(^{-1}\)K\(^{-1}\))
- \( \nu \) kinematic viscosity (m\(^2\)s\(^{-1}\))
- \( \rho \) fluid density (kg m\(^{-3}\))
- \( \phi \) heat flux density (Wm\(^{-2}\))

Subscripts
- \( e \) heating tube inner wall
- \( m \) tube section mean
- \( w \) HEV wall

References