Experimental and Numerical Investigations of Serrated Finned-Tubes in Cross-Flow

by

R. Hofmann\textsuperscript{a} and K. Ponweiser

Institute for Thermodynamics and Energy Conversion
Vienna University of Technology
A-1060 Vienna, Getreidemarkt 9/E302, AUSTRIA
\textsuperscript{a}Tel.: +43-1-58801-30217, Fax: +43-1-58801-30299
Rene.Hofmann@tuwien.ac.at, K.Ponweiser@tuwien.ac.at

1 Abstract

A three dimensional steady state numerical analysis of fluid flow and heat transfer downstream a circular tube with a periodic array of fins is performed. Four different models of a single row finned-tube heat exchanger, with solid and segmented fins, are investigated. The fins are arranged circularly or helically on the bare tube in cross-flow. The thermal field has to be examined for convective transport phenomena. The effect of the parametrically varied Reynolds-number to the Nusselt-number is to be analyzed. Especially the difference between solid and serrated fins has to be investigated. For modeling turbulence, a renomalization group theory (RNG) based on a $k-\epsilon$ turbulence model with enhanced wall functions is applied to resolve near wall effects between the adjacent fins.

Additionally, experimental investigations at an industrial scale test facility, of the Institute for Thermodynamics and Energy Conversion is performed at different finned-tube bundle configurations. The tube bundles are arranged at the equal transverse pitch, and in case of up to eight consecutively arranged tubes, with the equal longitudinally pitch in staggered formation. Thus, a maximum total number of 88 tubes in different configurations are investigated. Also at the gas-side, a flow rectifier consisting of three fine-wire meshes in close arrangement, followed by an inflow channel, which serves to calm the fully developed turbulent flow, should provide the same inlet conditions as for the simulations. At the water-side, an even cooling flow distribution in the tubes is achieved by installing orifices after the inlet collector. All investigations are made under hot conditions, so that the effect
of the pressure recovery through the tube bundle has to be considered. The Reynolds-Number was varied in the range between 4500 and 35000.

2 Motivation

Whenever gas/water heat exchangers are used, the heat transfer coefficient on the gas-side is inherently lower than at the water-side. For water tube heat exchangers, e.g. finned-tubes are applied to enhance the gas-side heat transfer. Many possibilities for improving heat transfer at the air-side exist i.e. the heat-transferring surface can be enlarged by an arrangement of annular fins or other elements, finned tubes with segmented fins show a somewhat higher turbulence than those with smooth fins, since the boundary layer has to be established at each individual segment \[1\], and staggered arrangement of the tubes in the bundle also increases turbulence. A higher pressure drop is caused by resistance in the flow channel and turbulence. Optimizing a finned-tube heat exchanger also results in minimizing the pumping power.

Experimental investigations at solid finned-tubes have been studied extensively in literature. From the among correlations that exist in literature only just a few investigations were performed for segmented fins. Also some varying influences of in-line and staggered tube arrangement were investigated. As an example, Weierman and Taborek \[2\] found that in-line arrangement should only be used for special cases, because of the disadvantage of possible bypass flow between the tube bundles.

3 Test Facility

A test rig for heat transfer and pressure-drop measurements on finned-tube bundles in cross-flow is in operation at the laboratory of the Institute for Thermodynamics and Energy Conversion. This test facility allows measuring at Reynolds numbers in the range between 4500 and 35000 and flue gas mass-flows from 0.6 to 4.5 kg/s; the layout is shown in Figure $3.1$.

The finned-tube bundle is admitted with hot gas of up to 400$^\circ$C which is generated by combustion of natural gas. Air is sucked in using a Venturi nozzle and a smaller ISA 1932 inlet nozzle for low Re-numbers, which are used for mass flow measurement of the combustion air. The burner is designed as duct burner. Downstream of two 90$^\circ$ bends there is an mixer application, followed by a transition piece to a rectangular cross-section, 500$mm$ in width and 1000$mm$ in height, containing a flow rectifier, consisting of three fine-wire
meshes in close arrangement. In the finned-tube heat exchanger the tubes are arranged horizontally with a given transversal and longitudinal pitch. The tube bundle consists of max. 88 tubes, which are arranged in 8, 6, 4, 2 consecutive columns, or a single tube row consisting of 11 horizontal tubes each. The experimental investigation requires a number of measurements to be taken simultaneously in order to evaluate and determine the amount of transferred heat as well as gas-side pressure drop. All measurement systems viz. thermocouples, RTD’s, pressure transducers, and a humidity sensor, were pre-calibrated before application. The measured values are transmitted to the process computer using a data acquisition system from National Instruments and the LabView 7E program system.

Solid and segmented finned-tube geometries have been tested and numerical analyzed in order to characterize the influence on heat transfer and pressure drop and thus to optimize heat-exchanger geometry. The main advantages of the segmented U-shaped fin geometry (Figure 3.2) are larger contact area between fin and tube (heat conduction) and the possibility of closer fin spacing\footnote{a larger fin base provides enough space for laser welding manufacturing} which allows a larger total outside surface area at equal fin height of the finned-tube bundle. Thus, an equal or smaller installation size of the heat exchanger can be achieved.

The characteristic parameters of the analyzed segmented U-shaped finned-tube are presented in Table 1.

Figure 3.1: Test facility
The measurements were performed at the gas-side as well as at the water-side. To obtain exact heat transfer correlations, each calculated point is validated after measuring, according to a validation model, introduced by J. Tenner et al. [3]. This curve fitting method uses equations of mass-balances, of energy-balances, and measurement value equations. The basic concept of the validation is to use all measured values with their variances and co-variances to fulfill all side conditions. After applying this criterion, all measured values can be developed into correlations for the prediction of the Nusselt numbers as functions of the Reynolds number.

4 Measurement Results

According to the method of dynamic similarity an objective function of Nusselt is defined as follows

\[ Nu = f(Re, Pr). \] (4.1)

Following dimensional analysis, the power law for the heat transfer correlation can be achieved.

\[ Nu = CRe^m Pr^n \] (4.2)

\(C\) and \(m\) are functions of geometric parameters, whereas \(n\) depends on fluid properties. In our case all \(Nu\)-Correlations are calculated at constant \(Pr\).
Table 1: Specifications of finned tube

<table>
<thead>
<tr>
<th>Fin Geometry</th>
<th>U-serrated</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bare tube diameter</td>
<td>38.0 mm</td>
</tr>
<tr>
<td>Tube thickness</td>
<td>3.2 mm</td>
</tr>
<tr>
<td>Number of fins per m</td>
<td>295</td>
</tr>
<tr>
<td>Average fin height</td>
<td>20.0 mm</td>
</tr>
<tr>
<td>Average fin thickness</td>
<td>0.8 mm</td>
</tr>
<tr>
<td>Average tube length</td>
<td>495 mm</td>
</tr>
<tr>
<td>Average segment width</td>
<td>4.3 mm</td>
</tr>
<tr>
<td>Number of tubes in flow-direction</td>
<td>8, 6, 4, 2, 1</td>
</tr>
<tr>
<td>Number of tubes per row</td>
<td>11</td>
</tr>
<tr>
<td>Longitudinal tube pitch</td>
<td>79 mm</td>
</tr>
<tr>
<td>Transversal tube pitch</td>
<td>85 mm</td>
</tr>
<tr>
<td>Outside surface area for 8 tube rows</td>
<td>84.48 m²</td>
</tr>
<tr>
<td>Net free area of tube row</td>
<td>0.2292 m²</td>
</tr>
</tbody>
</table>

Figure 4.1 shows the dimensionless heat transfer coefficient for a representative number of measured points in the $Re$-range of 8, 6, 4, and 2 serrated finned-tube rows in staggered arrangement and for a single tube row. In Figure 4.2, the pressure drop coefficient for 8 tube rows in staggered arrangement is presented. The index of zeta indicates the position of the measurement.

5 Numerical Procedure

Steady state numerical investigations were performed to predict three dimensional flow and heat transfer of a single tube in cross-flow. Four different models with solid and segmented fins were investigated. Starting from an single bare tube in cross-flow, the computational domain was discretized step-wise since the spirally segmented finned-tube model was able to be meshed. Therefore, the computational domain was split up into several parts of simple descriptive geometries and meshed each separately. To evaluate the momentum and heat transfer, some boundary conditions have to be defined. The wall temperature of the bare tube was defined to be constant with an

\[2\text{ i.e. fins, tube, fluid between fin and tube, inlet region, and outlet region} \]
appropriate heat transfer coefficient of the adjacent fluid inside the tube. The upstream inlet condition was chosen as "velocity inlet" with an turbulent intensity of $Tu = 5\%$ and a given temperature of $T_{in} = 480\, K$, and the downstream outlet condition was chosen as "pressure outlet" to an ambient pressure of about 1 bar. To obtain the same behavior as in the test rig, the effect of the symmetry of 11 tubes are on top of each other was chosen in a way, that symmetry conditions were applied on the top and bottom of the computational domain. For discretizing the spiral fins, periodic conditions were applied on the right and left hand side of the domain, seen from the flow direction above the finned-tube bundle, see figure 3.2. To reproduce this periodic effect the minimum fin thickness in the model starts form 0.04 $mm$. For turbulence modeling, a k-$\epsilon$ based RNG turbulence model is applied. Since the condition of the dimensionless wall distance for standard wall functions between the adjacent fins could not be met ($y^+ > 30$, logarithmic law of the wall) \footnote{3}, enhanced wall functions were applied. Thus a high number of cells were needed and a large computational effort is involved for the simulation of these complex geometries, associated with these dense grids. Finally parallel computing was applied and a simulation was stopped if either steady or periodic behavior of the residuals undergo the termination criterion.

The grid generation was performed using ANSYS, Gambit 2.3.16 and for

\footnote{3: roughly speaking}
Figure 4.2: Pressure drop measurement results of 8 consecutive finned solid/serrated tube rows

the CFD-calculations ANSYS, Fluent 6.3.26 was applied.

6 Comparison and Discussion

First, an global/average approach of the comparison between the measurement and simulation will be presented. As seen in figure 6.1, a comparison between literature, simulation, and measurement for a single tube row shows a good agreement. The equations of Weierman [5] have a measurement uncertainty of about ±10% for serrated finned-tubes in a staggered equilateral layout.

The exponents for the Nusselt correlations vary from about 0.5 to 0.6. The heat transfer calculation at only one finned-tube row, according to VDI WA 6, is based on the calculation at a tube surrounded by the flow. The flooding length at the smooth tube is used in this case as the characteristic length. The measured results for a single tube row show a higher inaccuracy. But in the measured Re-range, approximately the same gradient of the exponent in the power law, see equation (4.2), can be observed within the four equations.

The global behavior of the pressure drop coefficient is presented in figure 6.2. Especially for low Re-numbers a good agreement between measurement and simulation can be found. As mentioned above, the index in the graph
Figure 6.1: Global heat transfer behavior of a single finned tube in cross-flow represents the measurement location.

The investigated finned geometries have different geometrical constants, i.e. fin height, fin pitch, fin thickness, and fin width. To determine the optimal performance, the local heat transfer coefficient around the finned surface has to be calculated. Figure 6.3 presents the temperature contours of the helical serrated finned-surface. In the figure the fluid flow is in positive X-direction. The finned-tube is admitted with dry air of $T_{in} = 480$ K and a velocity of $v_{in} = 9$ m/s. As seen, the lowest temperature lies in the wake of the tube. For the solid material the same steel as for the real test tubes was applied. Since the numerical calculations are still continuing, only a few first results can be presented in this article. As a next step, comparisons between the temperature distribution of solid and segmented fins will be performed.

These comparisons between measurement results at global performance and numerical investigations of local and global heat transfer behavior in a single finned-tube row (e.g. turbulences, horseshoe vortices), will provide further knowledge of fluid flow and local heat transfer distribution.

$\zeta_1$ is calculated from the pressure differences on the air-side at the tube bundle wall, $\zeta_2$ is calculated from pressure differences in the channel center.
Figure 6.2: Global pressure drop behavior of a single finned tube in cross-flow

Figure 6.3: Contour of the local Temperature distribution in [K] of the single finned-tube
References


