Air side thermal performance of wavy fin heat exchangers produced by selective laser melting

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Abstract. Wavy fins are widely used for off-road vehicle coolers, due to their dust resistance. In this study, heat exchanger elements with wavy fins were examined in an experimental study. Due to independence of tooling and degrees of freedom in design, rapid prototyping technique selective laser melting was used to produce heat exchanger elements with high dimensional accuracy. Tests were conducted for air side Reynolds number $Re$ of $1400 \rightarrow 7400$ varying wavy amplitude and wave length at a constant water flow rate of $9.0m³/h$ inside the tubes. The effects of wavy amplitude and wave length on the air side thermal performance were studied. Experimental correlation equations for $Nu$ and $ψ$ were derived by regression analysis.

1. Introduction
Thermal management has become more important in recent years due to its potential to save energy, especially in vehicle applications. As crucial element of thermal management heat exchanger, i.e. radiators, charge air coolers, evaporators and condensers are designed to ensure the engine’s function and to provide air conditioning. Due to the dominant thermal resistance on the air side of heat exchangers, an improvement of heat exchangers can be realized by enhancing the air side convection heat transfer coefficient, derived by fin optimization. Different fin geometries e.g. plain fins, louvered fins, offset strip fins and wavy fins are used in compact heat exchangers. However, the latter are beneficial with respect to their resistance to dust adherence and their thermal hydraulic performance.

In wavy fins and at laminar flow regime, the swirl flow induced by wall corrugations gives the fluid a movement perpendicular to the main flow direction, thus mixing the flow and increasing the heat transfer [1][2]. Dong et al. [1][3][4] developed experimental correlations depicting thermal hydraulic performance of wavy fins. They investigated the effect of geometry parameters on thermal hydraulic performance of wavy fins in studies and showed that wavy length and wavy fin amplitude have the superior impact on the thermal hydraulic performance. Thus, the present study focusses on these two geometric parameters.

A limiting factor of the experimental studies with wavy fins is the dependence on tooling, as wavy fins are generally produced by rolling or stamping to produce a high number of respective fin geometry at low costs. Moreover, Joshi et. al. [5] reports on manufacturing irregularities i.e. edge roundness and burred edges have impact on thermal hydraulic measurements. For these reasons and to avoid high tooling costs and to take benefit from the high degree of freedom in creating desired geometries, the process of selective laser melting (SLM) was used to produce the heat sinks [6].
et al. [7] revealed the potential of SLM for heat sink production. They performed thermal hydraulic measurements with 4 different heat sink geometries produced by SLM due to their geometric complexity and showed their superior performance compared to conventional heat sinks. A side effect of SLM is a higher surface roughness. Strano et al. [8] conducted experiments measuring the surface roughness of a structure produced by SLM. The result was a model depicting surface roughness dependent on the build orientation in the SLM-process. Ventola et al. [9] also performed experiments stating the potential of SLM for usage in electronics cooling. Benefiting from the rough surfaces due to SLM, they found heat transfer can be enhanced by 35% compared to a smooth single finned surface.

In the present paper, a new method of fin development using SLM and experimental test bench results is introduced. This paper is organized as follows. In section 2 we depict test bench, heat sink geometries, test conditions and evaluation method. In section 3 results are presented and discussed. Finally, in section 4 main conclusions and outlook are provided.

2. Methodology

2.1 Test bench

![Test bench schematic](image)

**Figure 1.** Test specimen with flow arrangement and simplified illustration of the test bench.

Figure 1 depicts the test apparatus for the thermal hydraulic measurements in this study schematically. The test bench consists of a closed water side circuit, an open air side circuit, test specimen and measurement tools. The two mass flows are led together in a cross-flow as shown in figure 1, heating the air inside in the test specimen. The water side circuit with the hot water tank and the pump circulates water around the test specimen. Inlet and outlet temperature of the water side are measured by two pre-calibrated Pt100 resistance thermometers within an accuracy of 0.1K very close to the inlet and outlet. Moreover, the water channel was isolated. V-cone flow meter for mass flow determination and precision differential pressure transducers for pressure drop measurement were installed. The open air side circuit provides mass flow on the air side with a fan to charge the different fin geometries with various mass flow rates. Similar to the water side, inlet and outlet temperature measurements on the air...
side are detected by two pre-calibrated Pt100 resistance thermometers within an accuracy of 0.1K. V-cone flow meter and precision differential pressure transducers measured mass flow and pressure drop, respectively.

2.2 Heat sink geometries
Figure 2 shows the varying geometry parameters of a wavy fin. The heat exchanger elements were produced by SLM with a M1 Concept laser machine using stainless steel 316L. Table 1 gives the technical details of the test specimen. Nine different heat exchanger elements were tested with an air flow length of $L=150$mm, a fin height of $F_h=12$mm, test element width of $F_w=30$mm and a fin thickness of $t=0.3$mm. The dimensional accuracy of the test specimen tested by metallographic analysis is within an accuracy of ±0.1mm. Moreover, porosity was defined with metallographic examination of the test elements as <1% as seen in figure 3 and therefore the impact of porosity on the thermal hydraulic measurements can be neglected.

| No | Wavy length $WL$ (mm) | Wavy amplitude $2A$ (mm) | $2A/WL$ (-) |
|----|-----------------------|--------------------------|-------------|
| 1  | 12.5                  | 2                        | 0.16        |
| 2  | 7.5                   | 1                        | 0.13        |
| 3  | 7.5                   | 1.5                      | 0.20        |
| 4  | 12.5                  | 1                        | 0.08        |
| 5  | 7.5                   | 2                        | 0.27        |
| 6  | 10                    | 1                        | 0.10        |
| 7  | 10                    | 1.5                      | 0.15        |
| 8  | 12.5                  | 1.5                      | 0.12        |
| 9  | 10                    | 2                        | 0.20        |

![Figure 2. Geometric parameters of wavy fin and test specimen.](image)

2.3 Test conditions
Air mass flow rate $m_{air}$ was varied from 0.002 to 0.01kg/s, resulting a Re range from 1400-7400, as defined by equation (1):
Here, \( u \) denotes the average flow velocity, \( \nu \) represents the kinematic viscosity. The assumptions of air as an incompressible fluid was made and substance data of air was considered constant to logarithmic air temperature. The hydraulic diameter \( d_h \) can be defined as:

\[
d_h = \frac{4 \cdot A_{\text{free}} \cdot V}{A_0 \cdot A_{\text{front}}}
\]

The heat exchanger volume \( V \), frontal area \( A_{\text{front}} \), the total air side heat surface area \( A_0 \) and the air side minimum free flow area \( A_{\text{free}} \) are needed for this definition. On the water side, the inlet temperature is 90°C with a constant water flow rate of 9.0m³/, engendering a water side \( Re = 40000 \) and Nusselt Numbers \( Nu = 152 \) as defined by:

\[
Nu = \frac{\alpha \cdot d_h}{\lambda}
\]

\( \lambda \) is the thermal conductivity and \( \alpha \) the heat transfer coefficient. The difference of the water side inlet and out temperature \( t_{h2o,\text{in}} - t_{h2o,\text{out}} \) was smaller than 0.5K. Thus, a dominance of the water side heat transfer is guaranteed and an isothermal wall temperature at the specimen can be assumed. The data was accessed under stationary conditions.

2.4 Evaluation method

The heat transfer rate for the air side \( Q_{\text{air}} \) can be obtained by

\[
Q_{\text{air}} = m_{\text{air}} \cdot c_{p,\text{air}} \cdot (t_2 - t_1)
\]

Here, \( c_{p,\text{air}} \) is specific heat at constant pressure for air, \( t_2 \) and \( t_1 \) represent the outlet and inlet air temperature, respectively. By assuming the heat transfer rate for the air side \( Q_{\text{air}} \) being equal to the heat transfer rate for the water side, we define heat transfer effectiveness \( \phi \) for every measurement point as

\[
\phi = \frac{Q_{\text{air}}}{m_{\text{air}} \cdot c_{p,\text{air}} \cdot (t_1 - t_{h2o})}
\]

with the temperature of the water \( t_{h2o} \). The overall heat transfer coefficient \( kA \) for the test specimen is:

\[
kA = m_{\text{air}} \cdot c_{p,\text{air}} \cdot NTU
\]

Here, \( NTU \) is the number of transfer units. Heat transfer coefficient air side \( \alpha_{\text{air}} \) is calculated with equation (7). Note that the term representing the water side heat resistance can be neglected due to the dominance of the water side heat transfer resulting in a very high heat transfer coefficient \( d_{h20} \).

\[
\frac{1}{\alpha_{\text{air}} \cdot \eta_a \cdot A_0} = \frac{1}{kA} - \frac{s_t}{\lambda_t A_t} - \frac{1}{\alpha_{h2o} A_{h2o}}
\]

\( s_t, \lambda_t \) and \( A_t \) are thickness, heat conductivity and heat surface area of the tube, respectively. Surface effectiveness \( \eta_a \) and fin efficiency \( \eta_f \) of the wavy fins are defined as:
\[
\eta_a = 1 - \frac{A_f}{A_0} \left( 1 - \eta_f \right)
\]  
(8)

\[
\eta_f = \frac{\tanh(m \cdot l)}{m} = \left( \frac{2 \alpha_{air}}{\lambda_f \cdot l} \right)^{0.5} \quad m = 0.5 F_h
\]  
(9)

\(\lambda_f\) and \(A_f\) are the thermal heat conductivity and the surface area of the fin, respectively. With an iterative calculation, \(\alpha_{air}\) can be gained from equations (7)-(9). Finally, air side Nusselt number \(Nu\) can be calculated with equation (3). In order to evaluate pressure drop we calculate Darcy friction factor \(\psi\) which can be defined as:

\[
\psi = \frac{d_h \cdot \Delta p}{0.5 L \cdot \rho \cdot u^2}
\]  
(10)

Here, \(\Delta p\) and \(\rho\) denote the pressure drop and density, respectively. In order to evaluate the measurement results, a goodness factor \(JF\) is introduced:

\[
JF = \frac{Nu}{\psi}
\]  
(11)

As defined in equation (12), \(\delta_l(x)\) represents the thickness of the laminar boundary layer on a horizontal flow board in dependence of the the flow length \(x\), \(u\) and \(v\). \(\delta_t(x)\) denotes the thickness of the viscous sublayer in turbulent regime on a horizontal flow board [10]. An evaluation of \(\delta_l(x)\) and \(\delta_t(x)\) becomes necessary for identifying the impact of surface roughness on the measurements.

\[
\delta_l(x) = 5 \left( \frac{x \cdot v}{u} \right)^{0.5}
\]  
(12)

\[
\delta_t(x) = \frac{50x}{Re_x \left( 0.5 c_f \right)^{0.5}}
\]  
(13)

With Adams et al. [11] an equivalent sand grain roughness \(k_s\) can be calculated as function of average surface roughness \(R_a\) as defined by:

\[
k_s = 5.863 \cdot R_a
\]  
(13)

Taking into account instrument errors, geometry tolerances and property uncertainties, the estimated uncertainties for the \(Nu\) and \(\psi\) are 7.9\% and 5.4\%, respectively.

**3. Results**

Results are presented as Nusselt numbers \(Nu\) and Darcy friction factors \(\psi\) against \(Re\) to express the influence of wavy length \(WL\) and wavy amplitude \(2A\) on heat transfer and pressure drop performance with measurement points and correlations. Figure 4 shows the effect of \(WL\) on \(Nu\) and \(\psi\) at constant \(WL=12.5mm\) and a varying \(2A\). As expected, an increasing wavy amplitude \(2A\) leads to a higher \(Nu\) and \(\psi\) with a double wavy amplitude causing an increase in \(Nu\) and \(\psi\) up to 34\% and 39\%, respectively. Figure 5 depicts \(Nu\) and \(\psi\) as function of \(Re\) for a constant \(2A=2mm\) and a varying \(WL\). It can be seen that a reduction of \(WL\) from 12.5mm to 7.5mm leads to an increase in \(Nu\) and \(\psi\) up to 45\% and 77\%, respectively. To conclude, a lower wavy amplitude to wavy length ratio \(2A/WL\) shows
an increasing effect on $Nu$ and $\psi$. This can be explained by an enhanced mixing of the flow and vortex generation, thus creating higher pressure drop and heat transfer, as also found by [1][2][3].

Figure 4. Nusselt number $Nu$ (a) and Darcy friction factor $\psi$ (b) as function of Reynolds number $Re$ for a constant wavy length $WL$ of 12.5mm and varying wavy amplitude $2A$ from 1mm to 2mm.

Figure 5. Nusselt number $Nu$ (a) and Darcy friction factor $\psi$ (b) as function of Reynolds number $Re$ for a constant wavy amplitude $2A$ of 2mm and varying wavy length $WL$ from 7.5mm to 12.5mm.

As $2A/WL$ has been identified as the superior influence factor on thermal hydraulic performance [1], the goodness factor of the measurement results with various $2A/WL$ against Re is shown in figure 6. A trend is revealed that $JF$ is better for small $2A/WL$. If the test specimen with the same $2A/WL=0.2$ are compared, it is indicated that increasing $2A$ and $WL$ for a constant $2A/WL$ have a better $JF$. That is also verified by the experimental correlation of Dong et al. [1].

As mentioned above, SLM causes higher surface roughnesses. Strano et al. [9] conducted a study of the effect of build orientation in the process of SLM on the surface roughness. As a result, the maximum of surface roughness is found at $R_a=17.0\mu m$ for a structure consisting of stainless steel 316L, which is the same material used for building the heat sink geometries. Using equation (13), an equivalent sand grain roughness $k_s=99.7\mu m$ can be calculated. Comparing equivalent sand grain roughness $k_s=99.7\mu m$ to $\delta_l(x)$, it can be shown with equation (12) that after a flow length of 0.07mm at $Re=2210$, laminar boundary layer has reached the same height as maximum $k_s$ at maximum $Re$ in laminar regime. For maximum $Re=7400$ in turbulent regime the thickness of the viscous sublayer $\delta_l(x)$ at a flow length of 1.1mm has reached the same height as maximum $k_s$. This indicates that
surface roughness to have no effect on the measurements as beyond that point $\delta$ is higher than $k_s$ and 1.1mm compared to the total flow length $L$ of 150mm is neglectible small. The impact of surface roughness will be examined by further experiments as well as by computational fluid dynamics simulations.

![Figure 6. Goodness factor $JF$ as function of Reynolds number $Re$ for different wavy amplitude wavy length ratios $2A/WL$.](image)

For heat exchanger design quantitative calculations for the thermal hydraulic performance are meaningful, thus figure 7 draw comparision between the experimental data for $Nu$ and $\psi$ given and the predicted data given by equations (14-15), derived by multiple regression analysis. Equation 14 can describe 98.7% of $Nu$ of the total measurement data within ±20% with an average deviation of 7.7% and equation 15 predicts $\psi$ 52.1% of the total measurement data within ±20% with an average deviation of 25.4%. In comparison to the experimental uncertainties, the experimental correlations do not fit very well to the measurement results, especially for $\psi$. This may be due only varying two parameters, the small number of test specimen and the large differences in thermal hydraulic performance. An improved correlation shall be derived by further measurements, i.e. additional geometry parameter variations like fin height, fin pitch, total flow length.

$$Nu = 1.371 \times Re^{0.643} \times \left(\frac{4F_p}{2A}\right)^{0.435} \times \left(\frac{WL}{L}\right)^{-0.212} \times \left(\frac{2A}{WL}\right)^{1.108}$$  \hspace{1cm} (14)

$$\psi = 2.420 \times Re^{-0.223} \times \left(\frac{4F_p}{2A}\right)^{-0.272} \times \left(\frac{WL}{L}\right)^{-1.603} \times \left(\frac{2A}{WL}\right)^{1.149}$$  \hspace{1cm} (15)

4. Main conclusions and outlook
A new way of experimental fin development has been proposed in this paper. Test samples were produced by SLM and tested on a thermal hydraulic test bench. A detailed study quantified the impact of geometry parameters wavy length and amplitude on heat transfer and pressured drop. The advantages of SLM for experimental fin development are high dimensional accuracy and degrees of freedom in design, giving opportunity to conduct experimental tests with new fin geometries, not being producable by conventional production techniques. Moreover, as experimental tests are essential
for fin development, the process of fin development can be accelerated by rapid prototyping. The independence of conventional manufacturing processes as another asset gives the chance to conduct geometry parameter variations with state-of-the-art fin geometries, i.e. wavy fins, offset strip fins and louvered fins. Furthermore, simulations and further experiments will be conducted for a more detailed evaluation of the impact of surface roughness.

**Figure 7.** Comparison of experimental data and predicted data for Nusselt number $Nu$ (a) and Darcy friction factor $\psi$ (b).

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