Convective heat transfer with nanofluids

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Erklärung

Declaration


I hereby assure to have produced this Master thesis without help of others and only with the quoted sources. All information from external sources is marked as such. This thesis has not been presented in this or a similar form to an examining authority before.

(Ort,Datum)                                           (Unterschrift)
Abstract

The present Master thesis is concerned with forced convection heat transfer in laminar and turbulent flow with nanofluids. Nanofluids are defined as a colloidal suspension of particles in a base fluid, where the particles have a characteristic length of less than 100 nm. Experiments were conducted to determine the qualification of nanofluids for laminar and turbulent flow forced convection heat transfer. The experiments were conducted in two different devices: Firstly, a stainless steel pipe with an inner diameter of 3.7 mm, heated directly by a DC current in the pipe wall, and secondly, a tubular heat exchanger, which the fluid was cooled down in. The tested nanofluids were not only assessed considering Nu/Re, as it has been found to be common in a short literature review, but also by taking into account the pressure drop in different ways. A way of considering pressure drop in non-dimensional quantities was introduced that had not been seen in literature. In some cases, an opposite assessment for the fluid could be found from comparing Nu/Re of base fluid and nanofluid and comparing $h/\Delta p$. Difficulties during validation of the test rig had called for system improvement; an extensive error investigation was conducted on the test rig and the calculation. The error investigation resulted in changes concerning the calculation and the test rig.
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Nomenclature

Latin symbols

\( C_m \)  Mass-wise concentration of nanoparticles is nanofluid -

\( C_v \)  Local volumetric concentration of nanoparticles is nanofluid -

\( c \)  Specific heat capacity \( \text{J/(kg K)} \)

\( d \)  Diameter (of test section pipe) \( \text{m} \)

\( f \)  Friction factor of (nano)fluid in test section -

\( Gz \)  Graetz number -

\( h \)  Average heat transfer coefficient in test section \( \text{W/(m}^2\text{ K)} \)

\( h_x \)  Local heat transfer coefficient \( \text{W/(m}^2\text{ K)} \)

\( k \)  Heat conduction coefficient \( \text{W/(m K)} \)

\( L \)  Length of heated test section \( \text{m} \)

\( l_j \)  Length of particular section \( j \) of test section \( \text{m} \)

\( LMTD \)  Logarithmic mean temperature difference of tubular heat exchanger \( \text{K} \)

\( m \)  Mass flow through test rig \( \text{kg/s} \)

\( Nu \)  Average Nusselt number in test section -

\( Nu_x \)  Local Nusselt number -

\( p \)  Pressure \( \text{Pa} \)

\( P \)  Electric power \( \text{W} \)

\( Pr \)  Prandtl number of fluid in test section -

\( \dot{q} \)  Heat flux density from test section pipe to water \( \text{W/m}^2 \)

\( \dot{Q} \)  Heat flux from test section pipe to water \( \text{W} \)

\( Re \)  Reynolds number -

\( r_k \)  Ratio between thermal conductivity of nanofluid and base fluid -

\( r_{\mu} \)  Ratio between dynamic viscosity of nanofluid and base fluid -

\( t \)  Temperature \( \text{°C} \)

\( t_{in} \)  Fluid temperature at inlet to test section \( \text{°C} \)

\( t_{out} \)  Fluid temperature at outlet of test section \( \text{°C} \)

\( UA \)  UA number for tubular heat exchanger \( \text{W/K} \)

\( v \)  Fluid velocity in test section \( \text{m/s} \)

\( V \)  Volume flow rate \( \text{m}^3/\text{s} \)

\( x \)  Distance from start of heating in test section to place \( j \) \( \text{m} \)
Dimensionless distance from start of heating in test section

**Greek symbols**

- $\Delta$: Difference
- $\gamma$: Factor for Gnielinski's 1995 correlation for Nusselt numbers in the transition region $2300 < \text{Re} < 10^4$
- $\mu$: Dynamic viscosity, kg/(m s)
- $\rho$: Density, kg/m$^3$
- $\xi_1$: Friction factor in Gnielinski's correlation for the local Nusselt number and the 1995 correlation for the average turbulent Nusselt number
- $\xi_2$: Friction factor in Gnielinski's 1975 correlation for the average turbulent Nusselt number

**Subscripts**

- $\text{avg}$: Average
- $\text{Bl}$: Blasius
- $\text{bf}$: Base fluid
- $\text{cf}$: Cooling fluid
- $\text{des}$: Desired
- $\text{el}$: Electric
- $\text{f}$: Fluid
- $\text{Gn}$: Gnielinski
- $\text{HE}$: Heat Exchanger
- $\text{HP}$: Hagen-Poiseuille
- $\text{ht}$: Heat transfer
- $i$: Inside
- $\text{in}$: Inlet
- $j$: Local value in place j
- $\text{lam}$: Laminar
- $\text{nf}$: Nanofluid
- $\text{out}$: Outlet
- $p$: Particle
- $\text{Re}$: Reynolds
tube  Test section tube from stainless steel

turb  Turbulent

w  Wall (of test section pipe)

Δp  Pressure drop

Abbreviations

DC  Direct current
HD  Hydrodynamic
TC  Thermocouple
THE  Tubular heat exchanger
1 Introduction

Heat transfer coefficients are limited for different cooling technologies, depending on the fluid used in the process, as Figure 1.1 shows. Recent developments in electronic applications tend smaller devices with higher computing power, which leads to higher power densities and thus call for more efficient cooling.

![Order of Magnitude for Heat Transfer Coefficients Depending on Cooling Technology](image)

**Figure 1.1:** Heat transfer coefficients achievable with different cooling technologies and fluids [1]

If a heat source demands higher heat transfer coefficients to enable cooling at required conditions, either a different cooling technology can be chosen, if such is available, or using a different fluid with better heat transfer properties can be considered to increase the heat transfer coefficient with a given cooling technology. Changing the fluid offers two possible advantages: Firstly, existing cooling facilities and equipment can still be used with another fluid, which saves investments. Secondly, the known technology with the highest heat transfer coefficient can theoretically be enhanced, offering a higher overall achievable maximum heat transfer coefficient.

If a fluid shall be replaced, the new fluid has to meet the same conditions as the before used. Nanofluids offer a good chance to fulfil this demand, because here the possibility is given to mix nanoparticles into an existing fluid to enhance the fluid’s heat transfer coefficient while the fluid properties in respect to applicable temperature range, pressure range and corrosion behaviour remain.

In the present work, focus lies on the investigation of heat transfer coefficients achievable with nanofluids. Heat transfer was experimentally investigated in single-phase forced convection heat transfer in laminar and turbulent flow. Determining the heat transfer coefficients with nanofluids is motivated mainly by two reasons: On one hand, the heat transfer coefficient is the practically most relevant quantity for heat transfer equipment design; on the other hand, some researchers found higher enhancements in heat transfer coefficients than enhancements in thermal conductivity (e.g. [2]). The latter might be caused by mechanisms that are beyond the known mechanisms relevant for known suspensions of particles in fluid in macro scale. The present work, the Nusselt numbers achieved with nanofluids are compared among others to established heat transfer correlations which are valid for known fluids at present time.
1.1 Nanofluid – definition

The term “Nanofluid” was first used by Choi in 1995 [3]. He defined them as fluids containing particles of sizes below 100 nm. Nanofluids can be classified in different aspects: Base fluid, particle material, particle size, particle concentration, dispersant and pH-value of the nanofluid. In some cases, dispersants are used to stabilise the particles in the nanofluid and prevent the particles from sedimenting.

Usually classic heat transfer fluids as water, oil and ethylene glycol, are used as base fluids. Different materials have been used to manufacture the nanoparticles; they can generally be grouped into metallic (i.e. copper in [4]), metal-oxide (i.e. CuO and Al2O3 in [5]), chalcogenides (sulphides, selenides and tellurides, mentioned in [6]) and other particles, such as carbon nanotubes (i.e. in [7]). Sizes for single particles usually differ in literature between 20 nm and 100 nm (for example 20 nm in [8] and below 100 nm in [9]).

1.2 Nanofluid assessment

The suitability of nanofluids as heat transfer fluids is assessed by consideration of different quantities in the literature. Generally, there are two groups of research papers: The first group, forming the clear majority, looks at the heat transfer behaviour exclusively, the second group also considers viscosity and the additional pressure drop caused by the nanofluid respectively.

1.2.1 Fluid assessment considering only heat transfer behaviour

The most widely spread method of comparing a nanofluid to its base fluid is to consider the achieved Nusselt numbers at the same Reynolds number and therefore the same flow regime. This way of comparison takes into account the heat transfer behaviour and the thermal conductivity of the fluid via the Nusselt number and the density and dynamic viscosity of the nanofluid via the Reynolds number. Thereby, the viscosity of the nanofluid is included in this approach, but only in the Reynolds number. Still, it is not possible to determine from this way of comparison, if a possible enhancement in Nusselt numbers is followed by the penalty of a higher pumping power for the fluid. Therefore, this approach will be listed in this sub-chapter.

Xuan and Li [9] and Li and Xuan [10] report a remarkable increase of the Nusselt number by up to 60% with growing particle concentration and flow velocity in their experiment, where they consider turbulent and laminar flow through a straight brass tube with an inner diameter of 10 mm. They used a water-based nanofluid with Cu particles (diameter less than 100 nm) with 0.3, 0.5, 0.8, 1, 1.5 and 2 vol% concentration. The particles were covered with fatty acid to prevent aggregation. They only mentioned the penalty caused by the nanoparticles by stating that the pressure drop was almost like with pure water, they did not quantify their statement. They neglected the penalty in pumping power.

Yang et al. [11] reported a heat transfer coefficient between 15% and 22% higher for a nanofluid of graphite particles in different oils. The tests were conducted at laminar flow and temperatures between 50°C and 70°C. The nanofluid was assessed by considering the heat transfer coefficient at different Reynolds numbers, particle concentrations, temperatures, nanoparticle sources and base fluids. The pressure drop was not mentioned.

Wen and Ding [12] tested a nanofluid consisting of deionized water, SDBS dispersant (sodium dodecylbenzene sulfonate) and Al2O3-particles with a concentration of 0.6, 1.0 and 1.6 vol-%. The particle size was between 27 nm and 56 nm. The experiments were conducted at laminar flow. Between 41% and 47% higher local heat transfer coefficient was found with the nanofluid than with the base fluid. Only Nusselt vs. Reynolds numbers were considered at different particle concentrations.
Li et al. [4] found 6% – 39% enhancement of Nusselt numbers for nanofluid in their experiments with deionized water and copper nanoparticles in different sizes and concentrations. Only the heat transfer properties were considered for performance assessment.

Xuan and Li [13] tested water-based nanofluid with copper particles of 26 nm diameter in a flat tube. They found an increase of 39% in Nusselt numbers for the nanofluid with 2 vol-% particles in turbulent flow of Reynolds numbers higher than Re = 10000. No information about pressure loss is given.

Heris et al. [5] tested water-CuO- and water-Al2O3-nanofluids at particle concentrations between 0.3 and 3.0 vol-%. For comparison to the base fluid they considered Nusselt numbers vs. Peclet numbers, which leads to the extinction of the viscosity from the comparison. This way of comparison offers even less insight to the pressure drop behavior of the nanofluid than the before seen Nusselt-over-Reynolds approach.

In Heris et al. [14], also water-Al2O3-nanofluids are tested; again the fluid assessment considers Nusselt number and heat transfer coefficient vs. Peclet numbers. Pressure drop stays disregarded.

Lai et al. [8] tested nanofluids consisting of distilled water and Al2O3 particles with a size of 20 nm. 8% enhancement of Nusselt number was measured; Nusselt numbers were directly compared to each other, disregarding the Reynolds number.

Ding et al. [15] measured heat transfer with multi-walled carbon nanotubes in aqueous solution with a concentration of 0.1 – 1.0 vol-% and 0.5 wt-% of gum Arabic as dispersant. They found an enhancement in heat transfer of 350% in Re = 800. Pressure drop was not mentioned.

Jung et al. [16] also experimented with water-Al2O3-nanofluids, they measured more than 32% increment for the heat transfer coefficient for a concentration of 1.8 vol-%. Pressure drop in the rectangular micro channel was not regarded.

Duangthongsuk and Wongwises [17] tested water-TiO2-nanofluid with a particle concentration of 0.2 vol-% with 4000 < Re < 13000. They used different models to predict the thermophysical values of the nanofluid, but assessed the nanofluid’s qualification as heat transfer fluid only on the basis of Nu/Re.

1.2.2 Fluid assessment considering heat transfer and pressure drop

Williams et al. [18] tested water-based nanofluids with ZrO2 and Al2O3-particles. The flow profile was wide with Reynolds numbers between Re = 9000 and Re = 63000. At temperatures between 21°C and 76°C and particle concentrations of 0.9 – 3.6 vol% (Al2O3) and 0.2 – 0.9 vol% (ZrO2), heat transfer and pressure loss behavior could be described with traditional equations if the effective nanofluid properties were used in calculating the dimensionless numbers.

Duangthongsuk and Wongwises [19] experimented with water-TiO2-nanofluid containing 0.2 vol-% of particles, which had a diameter of 21 nm in a horizontal double-tube counter flow heat exchanger. Heat flux boundary conditions were varied as well as flow rate of nanofluid and heating water in turbulent flow. The results showed 6-11% higher heat transfer coefficient with nanofluid than with pure water and a little penalty in pressure drop (not quantified).

1.2.3 Result

A clear tendency can be seen in the reviewed literature: Pressure loss is mostly not regarded for nanofluid assessment. For practical purposes, pressure loss is an important issue however to receive information about the cost of heat transfer in form of pumping power with a given heat transfer fluid. The nanofluids tested during this thesis were assessed under consideration of the pressure loss.
2 Experimental setup description

2.1 Components

A closed-loop system was used to conduct forced convection heat transfer experiments. It was assembled from components which are listed in Table 2.1:

Table 2.1: Components used in experimental setup

<table>
<thead>
<tr>
<th>Component</th>
<th>Manufacturer</th>
<th>Model</th>
<th>Details</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pump</td>
<td>Ismatec</td>
<td>MCP-Z</td>
<td>60 – 6000 rpm</td>
</tr>
<tr>
<td>Pump head</td>
<td>Micropump</td>
<td>170-000</td>
<td>40 – 3840 ml/min gear pump head</td>
</tr>
<tr>
<td>Mass flow meter</td>
<td>Micromotion</td>
<td>FlowMeter 2700</td>
<td>Coriolis Mass Flow Meter</td>
</tr>
<tr>
<td>DC power source</td>
<td>Elektro-Automatik GmbH</td>
<td>PSI 9080-100</td>
<td>80 V/100 A/3000 W max.</td>
</tr>
<tr>
<td>Voltage meter</td>
<td>FLUKE</td>
<td>45</td>
<td></td>
</tr>
<tr>
<td>Thermocouple glue</td>
<td>omega.com</td>
<td>Omegabond 101</td>
<td>Thermal conductivity: 1 W/(mK)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Electrical resistivity: 10^{13} Ωm</td>
</tr>
<tr>
<td>Thermocouple cable</td>
<td>Omega Engineering Inc.</td>
<td>TT-T-30-SLE(ROHS)</td>
<td>Copper and Constantan wire</td>
</tr>
<tr>
<td>Sheathed thermocouples</td>
<td>Omega Engineering Inc.</td>
<td>T-Type</td>
<td>Stainless Steel, Inconel or SuperOMEGAACLAD®XL Sheaths</td>
</tr>
<tr>
<td>Absolute pressure transducers</td>
<td>ClimaCheck</td>
<td>PA-22S</td>
<td>Range: 0 – 10 bar</td>
</tr>
<tr>
<td>Differential pressure transducer</td>
<td>GE Druck</td>
<td>PTX5060-TA-A3-CC-H0-PA</td>
<td>Range: 0 – 1.5 bar</td>
</tr>
<tr>
<td>Pipe connections, valves</td>
<td>Swagelok</td>
<td>various equipment</td>
<td></td>
</tr>
<tr>
<td>Data acquisition</td>
<td>Agilent</td>
<td>2x 34970A Data Acquisition / Switch units</td>
<td>used with 2x 34907A Multifunction Module DIO/Totalize/DAC and 2x 34901A 20 Channel Multiplexer, Vee Pro 7 Software</td>
</tr>
<tr>
<td>Thermostat</td>
<td>Lauda</td>
<td>Ecoline Staredition Re204</td>
<td></td>
</tr>
<tr>
<td>Insulation</td>
<td>Armaflex</td>
<td>XG</td>
<td>Used as inner layer on test section and on all other pipes in the rig</td>
</tr>
<tr>
<td>Cover insulation</td>
<td>Isover</td>
<td>7600</td>
<td>Used as outer layer on test section</td>
</tr>
</tbody>
</table>
2.1.1 Heating

The test section is heated directly by applying a direct current to the pipe wall. The current is transferred to the test section through copper clamps. The voltage fall between these copper clamps is measured; the connections to the voltage meter are screwed to the copper clamps. The current is measured using the internal current meter of the DC power source.

2.1.2 Pressure drop

Pressure drop over the test section is measured with a differential pressure transducer. The pipes connected to the differential pressure transducer are connected to the test section to measure static pressure drop over the test section and intermediate pipe connections. On top of that, absolute pressure is measured at both differential pressure transducers’ taps before and after the test section.

2.1.3 Temperature measurement

The thermocouples used in the setup are listed in Table 2.2:

<table>
<thead>
<tr>
<th>Number of thermocouples with 1st/2nd/3rd system generation</th>
<th>Position in the test-rig</th>
<th>Kind of placement</th>
</tr>
</thead>
<tbody>
<tr>
<td>16/16/19</td>
<td>Test section, heated part</td>
<td>Outside wall of pipe</td>
</tr>
<tr>
<td>2/2/3</td>
<td>Flow development region</td>
<td>Outside wall of pipe</td>
</tr>
<tr>
<td>0/0/1</td>
<td>Test section, after heated part</td>
<td>Outside wall of pipe</td>
</tr>
<tr>
<td>0/2/2</td>
<td>Copper clamp (connection between DC source and test section pipe), one thermocouple per clamp</td>
<td>On copper clamp body</td>
</tr>
<tr>
<td>1/2/2</td>
<td>Inlet of test section</td>
<td>In fluid</td>
</tr>
<tr>
<td>2/3/3</td>
<td>Outlet of test section</td>
<td>In fluid</td>
</tr>
<tr>
<td>1</td>
<td>Inlet of tubular heat exchanger, nanofluid side</td>
<td>In fluid</td>
</tr>
<tr>
<td>1</td>
<td>Outlet of tubular heat exchanger, nanofluid side</td>
<td>In fluid</td>
</tr>
<tr>
<td>1</td>
<td>Inlet of tubular heat exchanger, cooling water side</td>
<td>In fluid</td>
</tr>
<tr>
<td>1</td>
<td>Outlet of tubular heat exchanger, cooling water side</td>
<td>In fluid</td>
</tr>
<tr>
<td>1</td>
<td>Thermocouple connection box</td>
<td>In box</td>
</tr>
<tr>
<td>1/1/0</td>
<td>40cm above tubular heat exchanger (ambient temp.)</td>
<td>In ambient air</td>
</tr>
</tbody>
</table>

The thermocouples on the pipe wall were glued to the pipe wall. This way of fixing suited three purposes: Firstly, the thermocouples were mechanically fixed on the wall; secondly, the thermocouples were electrically insulated from the electrically heated pipe wall by the glue and thirdly, the thermocouples were thermally connected to the test section pipe. The glue is well thermally conductive (k = 1 W/mK), in combination with a good insulation to the ambient for the test section it was assumed that the thermocouples' measurement error due to losses to the ambient could be neglected. This assumption is confirmed by calculation in chapter 2.3.3.4.
2.1.4 Insulation

The test section is insulated in radial direction with two layers of insulation: The inner layer consists of Armaflex insulation foam, the outer layer consists of mineral wool insulation. In axial direction, the test section is connected to plastic pipes to provide thermal and electrical insulation.

The rest of the pipes in the test stand are insulated with one layer of Armaflex insulation foam in different thicknesses, depending on the available space around the pipes.
2.2 First test rig generation

In this chapter the test rig is described as it existed at the start of this thesis.

The fluid flow is driven by a gear pump. From there, the fluid flows to a hydrodynamic development region and from there into the test section. The test section consists of a stainless steel pipe with an outer diameter of 4mm and an inner diameter of 3.7 mm. From there, the fluid is conducted into a Coriolis mass flow meter, then to a tubular heat exchanger and back to the pump. The tubular heat exchanger is cooled by water from a second closed loop, where it is cooled by a thermostat. The thermostat temperature can be adjusted. If the cooling power of the thermostat is insufficient, an additional heat exchanger located in the fluid loop of the thermostat can assist. The additional plate heat exchanger is cooled with tap water in an open loop.

At each end of the test section, plastic pipes were connected to the test section pipe to insulate the test section axially from the rest of the setup. On the inlet side, the flow development region was connected to the insulating plastic pipe; upstream of the flow development region, a second plastic pipe was connected. Left to this second plastic pipe, a union cross was connected, where the absolute pressure transducer and a pipe leading to the differential pressure transducer was connected.
2.2.1 Electric connection

A closer look at the test rig revealed hot electrical connection cables at high heating power. The electric connection to the test section pipe was made with a copper sheet at each end of the test section, bent around the pipe. The ends were held together by a screw which also held the cable connecting the DC power source. After removing the insulation, several pictures could be taken with an infra-red camera, as is shown in Figure 2.2. The first picture shows the cable to the DC power source from the right, leading to a screw joint, where the voltage meter was connected to the inlet (darker, in the background). The two parts of the screw joint were bridged with a wire, which was thinner than the others, showing a high temperature in the first picture. In the lower part of the first picture, the clamp surrounding the test section pipe is visible with a screw, which tightens this clamp and connects the wire. It can be seen that these parts are much warmer than the test section, especially the thin connection wire, which showed a temperature of more than 150°C in the IR-camera spot measurement, as can be seen in the middle picture of Figure 2.2, while the test section temperature is around 25°C.

Several consequences for the second rig generation were drawn from this insight. Firstly, all the cables were chosen articulately larger (35 mm² instead of 2.5 mm² cross-section) to avoid an electrical heating in the cable. Secondly, the clamps connecting wire and pipe were enhanced to avoid a high electrical resistance and thus a possible additional heat source in the electrical connections. Thirdly, the connection between the clamps and cables were chosen accordingly to grant a low electrical resistance.

All changes that have been made to the system are described in chapter 2.3.1.

Figure 2.2: IR pictures of the old clamp
2.2.2 Recorded data

From the data recorded in the experiments, different characteristic values are calculated to assess the heat transfer of the fluid used in each experiment.

The data logged in each measurement is summed up in Table 2.3.

Table 2.3: Logged experimental data

<table>
<thead>
<tr>
<th>Name</th>
<th>Measured quantity</th>
<th>Location of sensor</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>time</td>
<td>Time</td>
<td></td>
<td>s</td>
</tr>
<tr>
<td>t_ref</td>
<td>Temperature in logger box</td>
<td>in air</td>
<td>°C</td>
</tr>
<tr>
<td>t_1 – t_15</td>
<td>Temperature in position 1-15 of test section</td>
<td>on outside pipe wall</td>
<td>°C</td>
</tr>
<tr>
<td>t_17</td>
<td>Temperature in position 16 of test section</td>
<td>on outside pipe wall</td>
<td>°C</td>
</tr>
<tr>
<td>t_18</td>
<td>Ambient temperature behind data logger</td>
<td>in air</td>
<td>°C</td>
</tr>
<tr>
<td>t_19 + t_20</td>
<td>Temperatures of flow development region upstream of test section</td>
<td>on outside pipe wall</td>
<td>°C</td>
</tr>
<tr>
<td>t_21</td>
<td>Fluid temperature at inlet 1</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_22</td>
<td>Fluid temperature at test section outlet</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_23</td>
<td>Fluid temperature after union tee after test section outlet</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_24</td>
<td>Fluid temperature of nanofluid at outlet of tubular heat exchanger</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_25</td>
<td>Fluid temperature of cooling water inlet to tubular heat exchanger</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_26</td>
<td>in fluid</td>
<td>°C</td>
<td></td>
</tr>
<tr>
<td>t_27</td>
<td>in fluid</td>
<td>°C</td>
<td></td>
</tr>
<tr>
<td>t_28</td>
<td>Fluid temperature at outlet, variable position</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_29</td>
<td>Temperature of electrical connection clamp at inlet</td>
<td>on clamp</td>
<td>°C</td>
</tr>
<tr>
<td>t_30</td>
<td>Temperature of electrical connection clamp at outlet</td>
<td>on clamp</td>
<td>°C</td>
</tr>
<tr>
<td>t_31</td>
<td>Fluid temperature at inlet 2</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>Pin</td>
<td>Pressure at test section inlet</td>
<td></td>
<td>bar</td>
</tr>
<tr>
<td>PDiff</td>
<td>Pressure drop over test section</td>
<td></td>
<td>bar</td>
</tr>
<tr>
<td>Pout</td>
<td>Pressure at test section outlet</td>
<td></td>
<td>bar</td>
</tr>
<tr>
<td>Flow</td>
<td>Mass flow</td>
<td></td>
<td>kg/h</td>
</tr>
<tr>
<td>Density</td>
<td>Density of fluid</td>
<td></td>
<td>kg/m³</td>
</tr>
</tbody>
</table>
2.3 Second test rig generation

![Diagram of second test rig generation](image)

Figure 2.3: Scheme of second test rig generation

2.3.1 Changes compared to the first generation

The existing test section pipe was equipped with new electrical connection clamps, each a set of two copper blocks. Figure 2.4 shows one new electrical connection clamp after mounting it on the test section pipe. In preparation, the two copper blocks had been drilled while the two halves were held together; a hole with an inner diameter of 4.1 mm was made. The clamps were fixed to the test section by adding solder into the gap between the 4.1 mm hole in the clamp and the pipe, then applying the clamps, and bolting them together. Then the prepared clamps were heated up, so the solder could melt to form an even and uniform connection between pipe and clamp. The bolt holding together the two clamp parts was tightened again when the solder was still hot and liquid.

![New electrical connection clamp after mounting](image)

Figure 2.4: New electrical connection clamp after mounting
After soldering the clamps together, the cables (35 mm² cross-section area) were equipped with connection shoes by soldering the cables to the shoes. The copper shoes had a galvanised coating, which was removed on all surfaces used to establish electric contact.

After soldering the shoes to the cables, the shoes were fixed to the clamps using the same bolt, which was used before to press both halves of the copper clamp together; both surfaces on the shoe and on the clamps were cleaned before the connection to remove dirt and copper oxide. The clamps were equipped with thermocouples to enable measuring the temperature of the clamps.

When the test section was disconnected from the system to apply the new clamps, the pipe showed a layer of white solid material on the inner wall, probably nanoparticles. The pipe was cleaned on the inside using a small brush attached to a long wire, which again was attached to a battery drill and pushed into the pipe while rotating. This procedure was done twice, after that no visible layer was left.

The clamps were equipped with thermocouples to enable measuring the temperature of the clamps.

It should be noted that the pipe was bent at the inlet during mounting. It was straightened out as much as possible, but the possibility remains that the cross-section was not circular any more at the bending spot, which might result in locally higher velocities around that area or even induce eddies. There is no crack in the pipe, as pressure tests showed.

![Figure 2.5: IR picture of new clamp running 35 Amperes in Re=8000, inlet temperature not controlled](image)

**Figure 2.5** shows the Infrared picture of the new inlet clamp at a current of I = 35 A, taken from above the cooling bath. It can be seen that the clamp is now cooler (28.9°C) than the heated part of the pipe (31.3°C, value from a different infrared picture), whereas the old inlet clamp had heated up to 118°C, as can be seen in chapter 2.2.1.

During mounting the rig, it was discovered that two pipe connections located between the pressure sensors had not been drilled up to 3.7 mm inside diameter. This was corrected, later tests showed pressure loss values, which were closer to the expected values than before.
2.3.2 System behaviour after modification

The changed setup was tested with distilled water in laminar and turbulent flow, each with 25°C and 40°C inlet temperature. The water was not changed during the two days of testing. The evaluation was done considering the average and the local Nusselt numbers from steady-state data, which was recorded over three minutes. The correlations in this chapter are calculated with locally determined Reynolds and Nusselt numbers, a change compared to the calculation as it had been made when the system behaviour of the seconds rig generation was evaluated. The influence of the mentioned change in calculation can be seen in chapter 2.4.2.2.

2.3.2.1 25°C inlet temperature tests

The laminar results in 25°C inlet temperature show an enhancement for the average data, as can be seen in Figure 2.6:

![Figure 2.6: Average Nusselt numbers, distilled water, laminar, 1st and 2nd generation test rigs, 25°C inlet](image)

The data recorded with the changed setup (orange squares) is very close to the VDI Heat Atlas (see [20]) prediction (less than 3% deviation, except for Re = 2000) and does not show the bend that the older measurements showed between Reynolds numbers Re = 1200 and Re = 1500. The situation is different looking at the local Nusselt numbers, as can be seen from the comparison of Figure 2.7 and Figure 2.8. Figure 2.7 shows that the local Nusselt numbers recorded with the new clamps are closer to the prediction in average, whereas the older data in Figure 2.8 are slightly below the prediction. Anyway, the new results do not form a line as smooth as those recorded with the old setup. Especially single points drop out of line, especially the last and next-to-last value of each row and in some cases the value calculated from the temperature of thermocouple 6 or 8. A control of the thermocouple plugs on the backside of the data acquisition box showed that some thermocouples were not perfectly connected, but thermocouples 15 and 16 (last and next-to-last) were well connected. Even thermocouples 6 and 8 did not seem disconnected or wrongly connected. Another explanation for the dropping-out values may be that the thermocouple connection to the pipe may have been damaged during the process of applying the new clamps. The pipe, very unstable due to the thin wall, was bent slightly during handling...
due to the weight of the insulation; it cannot be excluded that the glue, which the thermocouples are fixed to the pipe with, may have cracked in some places.

Figure 2.7: Setup with new clamp: Local Nu vs. x*, distilled water, laminar flow, 25°C inlet

Figure 2.8: Old setup: Local Nu vs. x*, distilled water, laminar, 25°C inlet
The turbulent tests with 25°C inlet temperature showed no mentionable difference (less than 2%) in average Nusselt numbers for the system with new clamps (orange squares) to earlier tests, as can be seen in Figure 2.9.

The local Nu numbers paint an analogical picture in turbulent as in laminar flow, as Figure 2.10 and Figure 2.11 show. Like in laminar flow, the recorded data points form a line less smooth than it was recorded with the setup before the changes. Again, thermocouples 15 and 16 (two measuring point with highest x*-value in a measurement series) show a deviation from the overall trend.

One big difference in the new setup is that the local Nusselt number at the inlet is now higher rather than lower than the correlation, as is was in in the old setup. At thermocouple 1 (point at smallest x*-value), the local Nusselt number now exceeds the correlation by up to 19%, whereas in the old system, the local Nusselt number was partly below (in Re = 8076 and Re = 6066) and partly above the correlation (in Re = 3508 and especially in Re = 2515). The correlations themselves are different for the old and the new system, because the positions of the thermocouples in relation to the assumed start of heating were changed by introducing the new clamps. This fact makes the biggest difference in case of the first thermocouple.
Figure 2.10: 2nd generation setup, turbulent, distilled water, 25°C inlet

Figure 2.11: 1st generation setup, turbulent, distilled water, 25°C inlet
2.3.2.2 **40°C inlet temperature tests**

At 40 °C inlet temperature the situation is considerably different from the 25°C inlet temperature tests. In laminar flow, the average Nusselt numbers of the system with the new clamps (orange squares) are further away from the prediction than those of the old system, as Figure 2.12 shows.

![Figure 2.12: Comparison 1st and 2nd rig generation, laminar, distilled water, 40°C inlet](image)

The average Nusselt numbers of the modified system are generally higher than in the correlation (between 13% and 27% higher), which means that the recorded temperatures on the outside wall of the pipe must have been lower than the correlation would have predicted. The reasons for that might be that the modified system was equipped with significantly bigger copper connection cables than the old system. The new cables do not only have the effect of better electrical conduction but unfortunately also the effect of better thermal conduction to the ambient resulting in a heat loss to the ambient, which might be interpreted in the measurements as heat transferred to the test fluid. This motivated full insulation of the connection cables in the third test rig generation.

At 40°C inlet temperature, the local Nusselt numbers achieved with both setups seem fairly similar, as can be seen in Figure 2.13 and Figure 2.14. The most obvious change is a much higher value at the position of the first thermocouple in the second generation system (up to 3.15 times higher local Nusselt number than the correlation for Re = 483), now maybe induced by a heat flux out of the system through the electric connection, which helped cooling down the system. In case of the three higher Re numbers, the same effect as in 25°C inlet temperature is visible: Thermocouples 15 and 16 show certain deviation from the overall trend. Also again visible here is a deviation in thermocouple 6 to higher Nu number, although this can only be seen in the test at Reynolds Re = 1969 (purple triangles) and Re = 1464 (yellow circles) experiments. One possible explanation for this is a failure in measurements caused by loose contact between thermocouple and pipe or loose contact in the plug, which is plugged into the data acquisition system.
The situation in the turbulent tests in 40°C inlet temperature is not very different from the situation of turbulent tests with 25°C inlet temperature. Here too, the average Nusselt numbers are almost the same in the new tests (orange squares) as in the old tests (less than 2% difference), as can be seen in Figure 2.15. Possibly the influence is not as big as in laminar flow, because the heat flux through the connections is limited and has a smaller share in the overall heat flux in the system.
Looking at the local Nu numbers in 40°C inlet temperature and turbulent flow in Figure 2.16 and Figure 2.17, again the same tendency can be seen as in laminar flow. Again at the position of the first thermocouple (on the very left), a considerably higher Nu value is recorded, in all cases higher than the prediction, up to 40% higher for the modified system. This again might be due to extra heat flux out of the system through the electrical connections. Also similar to what was seen before, thermocouples 15 and especially 16 show quite a drop in Nusselt number, though here only in higher flow rates. Other than that, it can be seen here again, that thermocouple 6 is out of line in the new setup in Re = 2466 and Re = 1969 (though not in the other two). Here, thermocouple 6 is the only thermocouple, which indicates a less smooth behaviour than in the old setup. The picture was quite different in 25°C inlet temperature, where other thermocouples showed out-of-line values too.
Figure 2.16: New configuration, local Nu numbers, turbulent, 40°C inlet, distilled Water

Figure 2.17: Old setup, Local Nu numbers, turbulent, distilled water, 40°C inlet
2.3.3 Investigation of error sources

Difficulties with the second generation test rig were almost the same as with the first generation, namely the slope of the local Nusselt numbers in turbulent flow, more articulate with higher Reynolds numbers. Several possible reasons for this slope were investigated:

- Temperature dependency of electrical resistance of stainless steel pipe
- Heat transfer to the ambient by fin effect of electric connection cables
- Errors in flow temperature measurement
- Axial conduction, heat loss to ambient
- Temperature gain of test fluid due to shear stress in the flow

2.3.3.1 Temperature dependency of electric resistivity

Temperature dependency follows in a first approximation a linear law such as

\[ R(T) = R(T_0) \left( 1 + \alpha_{T_0} (T - T_0) \right), \]

as found in [21].

With a temperature coefficient of \( \alpha_{T_0} = 0.001 \) \( \text{K}^{-1} \) and a resistivity at 20°C of \( R(T_0) = 7 \times 10^{-7} \) \( \Omega \text{m} \) (see [22]), it can be concluded that the electrical resistance at 60°C, which is the highest temperature occurring in the system, can only deviate by 4% from the value at 20°C. It shall be added, that the highest occurring temperature difference between inlet and outlet of the test section is about 15K, so the deviation in electrical resistance can only be 1.5%. This effect was neglected in the search for the reasons for the slope in local Nusselt numbers because the influence is too low to be the reason for the difference in slope.

Figure 2.18 shows an example calculation with data from a turbulent experiment with Reynolds number \( \text{Re} = 4006 \).

![Figure 2.18: Influence of temperature dependent resistivity on local Nusselt numbers](image)

In Figure 2.18, the red squares represent the Nusselt numbers calculated with different heat dissipation in each pipe section, depending on the pipe temperature in each section, which has an influence on the pipe’s electrical resistance. The blue diamonds represent the local Nusselt numbers, calculated with the assumption, that the heat flux density to the water in the test section is homogeneously distributed along the test section wall. The overall heat added to the water is here calculated with mass flow, heat capacity and temperature difference between test section inlet and outlet.
The differences between the Nusselt numbers calculated incorporating the resistance’s temperature dependency to the Nusselt numbers calculated without incorporating this dependency are about 5% at the inlet and about 4% at the outlet, finding generally lower local Nusselt numbers from the calculation minding the temperature dependency. The fact that the differences between both calculations are so similar at inlet and outlet, leads to the assumption that this offset can mainly be interpreted as heat losses to the ambient. The remaining deviation can be assumed to be caused by considering the temperature dependency.

2.3.3.2 Heat transfer to ambient through electric cables

A conflict of interests exists between good electrical conductivity and good thermal insulation that had not been foreseen when the new clamps were designed. The clamps and/or electrical connection cables might now be too big, so that they provide more than enough electrical conductivity, but too much thermal conductivity. This possible effect was investigated mostly looking at the 40°C inlet experiments, because possible effects were assumed to be more pronounced with a higher temperature difference between test equipment and the ambient.

Figure 2.19 shows temperatures of pipe wall, fluid, clamps and the room at a laminar experiment conducted at 25°C inlet temperature. It can be seen that the inlet clamp’s temperature is very close to the fluid temperature (the measurement showed only 0.01K deviation). The outlet clamp, however, is a little warmer than the fluid (1.5K); its temperature is between the fluid and the wall temperature. This indicates a heat flux from pipe wall to the clamp and from there to the ambient. Here the ambient is 5.6K cooler than the clamp. It also indicates a heat flux from the pipe wall to the clamp, which may be an explanation for the two last wall temperatures being lower than the third last wall temperature.

Figure 2.19: Clamp temperatures in laminar experiment, 25°C inlet

Distilled water, Re=1478; 25°C inlet

It was observed during the experiments at 40°C inlet temperature that the electric cables were quite warm. Also the thermocouples located at the clamps showed considerable lower temperatures than those at the pipe wall. In case of the $Re = 1472$ experiment at 40°C inlet temperature shown in Figure 2.20, the thermocouple at the inlet clamp showed 1.8K less than inlet fluid temperature, and the outlet clamp thermocouple showed 2.6K less than outlet fluid temperature. Due to the good thermal connection it can be assumed, that there is a considerable heat loss to the ambient through the electrical connections in the current setup.
In turbulent experiments, the situation seems slightly different, as can be seen from Figure 2.22 and Figure 2.21.

In 25°C inlet temperature (see Figure 2.21), no heat is exchanged between fluid and inlet clamp, the whole equipment has about room temperature at the inlet clamp. The first thermocouple on the heated wall is 1 cm away from the inlet clamp. At the outlet clamp we can see, that the thermal connection to the fluid is much better than to the ambient, but anyway there is a heat flux to the ambient, apparently the clamp is cooled by the ambient, which is 10.4K cooler than the outlet clamp.
Figure 2.22: Clamp temperatures in turbulent experiment, 40°C inlet

The same effect can be seen in 40°C inlet temp (see Figure 2.22), just more articulate, because there’s a temperature difference to the ambient already at the inlet clamp (14.9K) and an even higher temperature difference at the outlet clamp (25.1K), which there leads to a seriously lower clamp temperature, 3.9K colder than the fluid. The distance between the thermocouple and the fluid is about 3.5 mm, consisting of 0.15 mm stainless steel, 3 mm of copper and (estimated) 0.3 mm of glue. Calculating the radial resistance of the stainless steel pipe and assuming, that the measured temperature is the same on the surface of the copper clamp surface, this means, that a heat flux of 44 W would flow to the ambient through the outlet clamp. Given a total electric power of 760 W, this is a loss of almost 6% through this clamp. Comparing the electrical power and the fluid’s energy gain through the pipe, this indicates only a power loss of 0.6% to ambient (which seems very small, considering the relatively high temperature difference to the ambient), so the heat that is lost through the outlet clamp could be generated somewhere else than in the pipe wall.

On the other hand, the clamp temperatures are lower than the fluid temperatures in the turbulent tests, where the electrical current is high. It can be assumed that the clamps and connections are so big that only negligible heat is generated in the clamps or other parts of the electrical connection.

It was concluded from this investigation that the electrical connection cables must be insulated for further experiments. Still this cannot be the solution to the problem of the “wrong” slope of the local Nusselt numbers in turbulent flow experiments, because even if there is heat flux to the ambient through the cables, it cannot origin from the middle of the test section pipe, because axial conduction has too small an influence, as was seen in chapter 2.3.3.4.

2.3.3.3 Errors in flow temperature measurement

It was observed that the positive slope of local Nusselt numbers in turbulent flow experiments vanishes if not the measured outlet temperature is used for calculation, but a temperature 0.5K to 1K lower than the measured value. This observation immediately suggested looking after the measurement of the fluid temperature at the outlet.

For this purpose, an experiment was conducted on the second rig generation at 25°C inlet temperature at an average Reynolds number of $Re = 4000$. A sheathed thermocouple with an outer diameter of 1 mm was inserted into the test section from the end, where a union tee fitting grants access to the test section.
pipe. The thermocouple was stuck through a PTFE plug, which then was fixed to the union tee instead of a third pipe. The thermocouple was prepared before mounting by wrapping it around a thin rod to form a coil. The size of the coil was fit to the inside diameter of the test section to guarantee contact to the wall to provide a specified position for the thermocouple tip, which was bent just so much that it was located in the middle of the pipe. This preparation was tested in a short piece of tube of the same dimension as the test section. It proved to be able to reliably grant the thermocouple tip to be located in the middle of the pipe.

The temperature of this thermocouple was measured at different positions in the pipe, each for three minutes. The data acquired this way were normalized by adding a correction value that was calculated before by comparing the inlet and outlet temperatures to the average value of all measurements.

It can be seen in Figure 2.23 that the fluid temperature at the thermocouple tip was still rising after the inside edge of the clamp, the edge pointing to the middle of the test section. It was assumed earlier that heating ends at the inside edge because from there the resistance is much less in the copper block than in the test section wall, so no heat would be dissipated after the inside edge of the clamps. The fluid temperature however rose until it reached a steady value, 47 mm downstream of the supposed end of heating, the inside edge of the copper clamp.

A possible explanation for this effect may be that the mixing process in turbulent flow is limited, so fluid temperature in the middle of the pipe reaches the average value only 47 mm after the supposed end of heating. This would mean a measurement or homogenisation problem is present. This can be further investigated with a static mixer built in after the test section, which would allow the assumption of a homogeneous temperature profile over the whole cross-section of the pipe. This test could not be conducted during this thesis, because the static mixer could not be installed in the system due to missing parts.
Another explanation for the temperature rise after the assumed end of heating would be that heating takes place even downstream of the inside edge of the outlet clamp. This was investigated on the third generation test rig (see chapter 2.4.2.1), but it could be not clarified what had happened in the first and second test rig generation.

Assuming a longer heated length would influence the calculation, because the local fluid temperature is calculated as a linear interpolation between inlet and outlet temperature. If the point, where the outlet temperature is reached is assumed further away from the inlet, the slope of the fluid temperature will turn out flatter at the same inlet and outlet temperature. This will result in a higher local temperature difference between fluid and pipe wall, leading to a lower calculated local Nusselt number. This effect will be increasing as distance from the inlet gets bigger.

The influence on the local Nusselt numbers shall be explained looking at tests in turbulent flow. In Figure 2.24, local Nusselt numbers are shown as they are calculated using the length between the inside edges of the electrical connection clamps as heated length. All local Nusselt numbers show a steeper slope than the correlations, the values deviate up to 21% from the correlations.

In Figure 2.25, the same experiments are shown as in Figure 2.24, but this time the heated length is assumed to be the sum of the length between the electrical connection clamps plus 47 mm on the outlet side, as it could be measured before (see Figure 2.23). It can be seen that in this case the slope of the experimentally determined local Nusselt numbers fits the slope of the correlations very well (less than 6% deviation).

![Local Nu, turbulent, 40°C inlet, 1st rig generation](image)

Figure 2.24: Local Nusselt numbers assuming the end fluid warming at the electric clamp
Further investigations are necessary to determine if it is legitimate to assume the heated length of the test section longer than just the length between the electric connection clamps. A longer heated length is not assumed for the evaluation of the tests conducted on the first generation test rig, because it cannot be said with certainty how the exact configuration was at the outlet concerning thermocouples and axial electrical insulation.

### 2.3.3.4 Axial conduction, heat loss to ambient

Calculation investigations showed that axial conduction cannot be the reason for the phenomena described above, also losses to the ambient are negligible, especially in tests done at 25°C inlet temperature, which is very close to the ambient temperature (between 24°C and 29°C).

*Table 2.4* shows results for three different ways to calculate the temperature on the inside of the pipe wall. An older calculation, prepared before the start of this thesis, did not take into account the heat source in the wall, but instead assumed conduction through the pipe wall. The differences are small compared to the average temperature difference between inside pipe wall and bulk fluid of approximately 5K; still, the formula, which considers the wall as the heat source,

\[
t_{w,i,j} = t_{w,o,j} + \frac{\dot{Q}}{4\pi L k_{pipe}} \frac{\varphi (1 - \ln(\varphi - 1)) - 1}{\varphi - 1}
\]

(2.2)

with \( \varphi = \frac{d\bar{c}}{d_1} \),

will be used in future calculations (column 2).
2.3.3.5 **Temperature gain of test fluid due to shear stress in the flow**

The local Nusselt numbers in turbulent flow showed a rising slope (see i.e. Figure 2.17). This means that the temperature difference between fluid and wall is smaller toward the end of the test section pipe than the correlation predicts, given that the heat flux density is equally distributed over the pipe wall. A smaller temperature difference than given by the correlation can be caused by a lower wall temperature or a higher fluid temperature. A higher fluid temperature than expected can be caused by viscous heating in the fluid, this has been investigated here.

The percentage of heating caused by viscous friction in the fluid can be calculated with

\[
\kappa = \frac{\Delta t_{\text{visc}}}{\Delta t_q} = \frac{2 \text{Br} \text{Po} A}{d_i^2} = \frac{\dot{m} \Delta p}{Q \rho},
\]

as seen in [23]. The Brinkman number is given in terms of the dynamic viscosity, mass flow, heat flux, pipe length, pipe cross-section and density as

---

**Table 2.4: Calculated inside wall temperatures using different formulas and assumptions**

<table>
<thead>
<tr>
<th>1: Old calculation, °C</th>
<th>2: Heat source in wall, °C</th>
<th>3: Difference 2-1, K</th>
<th>4: Wall source, loss to ambient, wall conduction, °C</th>
<th>5: Difference 4-2, K</th>
</tr>
</thead>
<tbody>
<tr>
<td>44,68</td>
<td>44,79</td>
<td>0,11</td>
<td>44,79</td>
<td>-0,003</td>
</tr>
<tr>
<td>45,71</td>
<td>45,82</td>
<td>0,11</td>
<td>45,82</td>
<td>-0,003</td>
</tr>
<tr>
<td>45,13</td>
<td>45,25</td>
<td>0,11</td>
<td>45,24</td>
<td>-0,003</td>
</tr>
<tr>
<td>45,53</td>
<td>45,64</td>
<td>0,11</td>
<td>45,64</td>
<td>-0,003</td>
</tr>
<tr>
<td>45,61</td>
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<td>0,11</td>
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</tr>
<tr>
<td>45,79</td>
<td>45,91</td>
<td>0,12</td>
<td>45,90</td>
<td>-0,003</td>
</tr>
<tr>
<td>45,92</td>
<td>46,03</td>
<td>0,12</td>
<td>46,03</td>
<td>-0,003</td>
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<tr>
<td>46,29</td>
<td>46,40</td>
<td>0,12</td>
<td>46,40</td>
<td>-0,003</td>
</tr>
<tr>
<td>46,31</td>
<td>46,42</td>
<td>0,12</td>
<td>46,42</td>
<td>-0,003</td>
</tr>
<tr>
<td>46,54</td>
<td>46,66</td>
<td>0,12</td>
<td>46,66</td>
<td>-0,003</td>
</tr>
<tr>
<td>46,66</td>
<td>46,78</td>
<td>0,12</td>
<td>46,78</td>
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</tr>
<tr>
<td>46,89</td>
<td>47,01</td>
<td>0,12</td>
<td>47,00</td>
<td>-0,003</td>
</tr>
<tr>
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<td>47,60</td>
<td>47,72</td>
<td>0,12</td>
<td>47,72</td>
<td>-0,003</td>
</tr>
</tbody>
</table>
\[ \text{Br} = \frac{\mu \dot{m}^2}{\frac{Q}{L} A^2 \rho^2}. \] (2.4)

The Poiseuille number is given as

\[ \text{Po} = \frac{f}{4} \text{Re} = \frac{\Delta p}{2 \Delta \rho} \frac{d_l^2}{A \mu}. \] (2.5)

It could be seen that the share of viscous heating was less than 1% in all experiments. The effect of viscous heating was thus neglected in all calculations.
2.4 Third test rig generation

2.4.1 Changes compared to the second generation

A new test section was built for the third test rig generation. The previous test section had been damaged during handling; it was possible that the cross-section of the pipe was not circular any more in all places. The opportunity was used to implement a few other changes as well:

2.4.1.1 Thermocouples

Thermocouples fixing

Thermocouples were attached to the test section pipe in a three-step-procedure:

- First, the soldered thermocouple tips were bent around a piece of pipe with an outer diameter of 4 mm, the same as the test section. Then these “hooks” were coated with a layer of glue on the inside of the “hook” to guarantee electrical insulation against the test section pipe.
- After the glue had dried, the prepared thermocouples were fixed to the test section pipe by wrapping them around the test section two times and fixing them with a thin layer of glue. When this glue had dried, the position on the pipe of each thermocouple was measured.
- A final protective layer of glue was added by wrapping them around the test section pipe to provide good mechanical stability.

Wrapping the thermocouples around the test section pipe twice was a measure of providing a good mechanical connection to the pipe as well as a measure to avoid thermal conduction from the ambient to the thermocouple tip. It also brought forward a better precision in determining where the point of temperature measurement is located along the pipe. The thermocouples are hand-soldered, which results in a soldered tip with a length of 5 – 10 mm. Along the tip, it is not possible to know where the contact between the two thermocouple wires is made and thus where the temperature is actually measured. By wrapping the thermocouple around the pipe, the place of temperature measurement has the same distance to the beginning of heating, no matter where exactly the connection between the copper and the constantan wire had been made in the thermocouple tip.
Thermocouples positions

In the newly created test section, thermocouples were not any more placed equally-distanced on the test section, but the distance between two thermocouples decreased close to the inlet. The positions are given in Table 2.5. The overall length of the heated part of the test section (between the inside edges of the electrical connection clamps) is 1468 mm.

<table>
<thead>
<tr>
<th>TC number</th>
<th>Distance from the edge of the electrical connection clamp facing the heated part, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5</td>
</tr>
<tr>
<td>2</td>
<td>18</td>
</tr>
<tr>
<td>3</td>
<td>41</td>
</tr>
<tr>
<td>4</td>
<td>66</td>
</tr>
<tr>
<td>5</td>
<td>111</td>
</tr>
<tr>
<td>6</td>
<td>214</td>
</tr>
<tr>
<td>7</td>
<td>320</td>
</tr>
<tr>
<td>8</td>
<td>423</td>
</tr>
<tr>
<td>9</td>
<td>528</td>
</tr>
<tr>
<td>10</td>
<td>632</td>
</tr>
<tr>
<td>11</td>
<td>735</td>
</tr>
<tr>
<td>12</td>
<td>840</td>
</tr>
<tr>
<td>13</td>
<td>945</td>
</tr>
<tr>
<td>14</td>
<td>1047</td>
</tr>
<tr>
<td>15</td>
<td>1155</td>
</tr>
<tr>
<td>16</td>
<td>1257</td>
</tr>
<tr>
<td>17</td>
<td>1359</td>
</tr>
<tr>
<td>18</td>
<td>1463</td>
</tr>
</tbody>
</table>

One thermocouple is positioned after the electrical connection clamp at the outlet of the test section; the distance is 5 mm from the outside edge of the clamp. Three thermocouples are positioned upstream of the inlet clamp (on the hydrodynamic development section), the distances to the outside edge of the clamp are 5, 100 and 194 mm. The clamps themselves are 15mm wide each.

2.4.1.2 Flow profile development section

The flow profile development section consisted of a separate piece of pipe in the former rig generations. Pipe connections had to be used to connect it to the test section. It was feared that the connections might disturb the flow after it had passed the flow development section, so one piece of pipe was used to create the new test section and flow development section to be able to avoid connections. The advantage of a flow profile development section made from a separate piece of pipe was that it could be thermally and electrically insulated from the test section in axial direction. After showing that axial conduction is very
small (see chapter 2.3.3.4), it was decided to accept this small disadvantage to the benefit of not disturbing the flow between flow profile development section and test section.

### 2.4.1.3 Minor changes

In the first and second test rig generation, the mass flow meter was located between the test section and the tubular heat exchanger. This brought about that the mass flow meter had to cool down or heat up to the test section outlet temperature to allow the test rig to reach steady-state whenever heating power and/or pump speed was changed. This resulted in long waiting times between tests, especially in low flow rates, because the mass flow meter as a heavy stainless steel apparatus had to be heated or cooled by a relatively small mass flow of test fluid. The mass flow meter was moved to between tubular heat exchanger and pump, because there fluid temperature is kept at the same level during a set of experiments with the same inlet temperature. This change had a second positive effect on the test rig: The water column in the charge funnel, which is connected to the mass flow meter, is now located directly upstream of the pump. The fluid pressure at the pump inlet is now as high as possible with the used components, which results in the possibility of running the pump at higher speed because of less danger of cavitation.

A union tee with a valve to the ambient was added between pump and test section to enable drying the system with air. It would not have been possible to dry the system properly with air otherwise because the only other available air inlet to the system would have been positioned on the other side of the pump. The pump cannot be operated dryly, but it has a high flow resistance, even against passing through gas.

The axial insulation is accomplished not with plastic tubes any more, but with pipe adaptors made from PTFE, replacing stainless steel adapters. This allows a more compact setup and moves the point of electrical and thermal insulation as close to the test section as possible.

The clamps, which connect the test section pipe to the DC power source, could be made from one piece of copper and then be soldered to the pipe for the third test rig generation. This had not been possible when the first generation’s pipe was modified, because then pipe connectors had already been squeezed onto the pipe, which had a much larger outer diameter than the pipe, so it was impossible to slide on clamps with only 0.1 mm bigger hole diameter than the pipe’s outside diameter.

The cables connecting the test section clamps to the DC power source were insulated with Armaflex foam insulation to limit the heat flux to the ambient through the cables.

A static helical mixer was ordered to be added to the outlet of the test section to ensure measuring the bulk fluid temperature, especially in laminar flow. It could not be installed before the end of this thesis due to missing connections, which had a too long delivery time.

### 2.4.2 Investigation of error sources

Following possible errors were investigated while building the new test section:

- Insufficient insulation of the test section pipe in axial direction (fluid thermocouple conduction)
- Influence of using local material properties in calculation

#### 2.4.2.1 Insufficient electrical insulation of the test section pipe

Electrical measurements have shown that the test section was not well insulated from the rest of the setup. A current meter showed an electric current of 23.2 A in the test section and 4.5 A in the rest of the test rig pipes when the DC power source displayed a current output of 27.8 A. A “squeaking” sound could be heard during these measurements, the source was clearly the insulating plastic tubes.
The thermocouples for fluid temperature measurement at inlet and outlet had provided electrical connection by leaning against the pipe wall on both sides of the plastic tubes used for axial insulations. The exact position of these thermocouples was not controlled in previous experiments, so it could not be determined which experiments were influenced by this error. By the time this error was discovered, the new test section for the third rig generation was already installed, so the influence of this error can only be investigated with the new test section.

The thermocouple tips were re-positioned to the middle of the insulating plastic tubes to avoid the possibility of an electrical connection through the thermocouples. The current measurements were repeated after this change, no current was measurable in the rest of the test rig’s pipes. The “squeaking” sound had stopped; it was assumed that it was caused by the electrical connection from thermocouple sheathing to the pipe wall. The thermocouple showed a blackened spot when it was revised for re-positioning, this too was assumed to be caused by the electrical connection, possibly a connection only in one point with a high current density.

All thermocouple cables have been measured for electric current, all thermocouples showed a current between 0.05 A to 0.14A, with an instrument’s offset of 0 to 0.1A. It was concluded that none of the thermocouples were providing an electrical connection to a point outside of the test section.

The electric current-measurements were conducted with a FLUKE 39 Power Meter and a FLUKE 80i-110s AC/DC current probe.

Additional tests were made to determine voltage fall along the test section. The voltage was measured with a Tillquist TQ-703 Multimeter, the measuring tips were dipped into saltwater to provide good electrical connection to the measured probe. It was seen that the voltage fell proportionally to the length, so it could be concluded that heat was generated homogenously along the test section pipe.

After re-positioning the fluid thermocouple tips to the middle of the insulating plastic tubes, the electric resistance was measured between both ends of each insulating plastic tube. The instrument displayed infinitely high resistance; it was concluded that neither the pipe nor the fluid or any solid layers on the tube wall provide an electrical connection to the rest of the test rig.

2.4.2.2 Influence of using local fluid properties in calculation

The difference between using local fluid properties in the calculation versus using fluid properties calculated with the average fluid temperature was investigated while building the new test section. This was a purely mathematical investigation and was independent from the work on new test section, but both was done simultaneously.

The influence of local fluid properties had been investigated before the start of this thesis, but had then been found negligible. It was considered again at this point out of missing alternatives for the source of errors seen in the evaluation of experimental data.

In calculation and comparison of the experimentally determined Nusselt numbers and established correlations for Nusselt numbers, fluid properties (density, viscosity, specific heat capacity and thermal conductivity) of the base fluid were calculated using temperature dependent functions. In earlier versions of the calculation, fluid properties had been calculated using the average fluid temperature.

The effect on the local Nusselt numbers in turbulent flow shall be discussed here. In contrast to the local Nusselt correlations for laminar flow, the correlations for turbulent flow strongly depend on the Reynolds and Prandtl numbers. It therefore makes a considerable difference, if Reynolds and Prandtl numbers are calculated from fluid properties that themselves are calculated using the local temperature or the average fluid temperature in the test section. The effects on the correlations for local Nusselt numbers can be seen
in the following figures, experimental data from the first rig generation are shown to compare to the correlations:

**Figure 2.27**: Experimental data at 25°C inlet and correlations using average fluid properties

**Figure 2.28**: Experimental data at 25°C inlet and correlations using local fluid properties

The biggest difference between Figure 2.27 and Figure 2.28 is that in Figure 2.28, where local fluid properties are used in the calculation, the slope of the local Nusselt numbers from the correlation is rising, much like the slope of the measured Nusselt numbers. The different slopes of the local Nusselt numbers...
from correlation or determined in experiments, as can be seen in Figure 2.27, had been the biggest objection against the quality of the experimental setup in earlier investigations.

The reason for the increasing slope of the correlation for local Nusselt numbers is that the fluid in the test section is heated up flowing through the test section. The temperature difference between inlet and outlet differs from experiment to experiment, but does not exceed 15K. Still the effect on the fluid’s properties, especially on viscosity, is so big that Reynolds and Prandtl numbers at the inlet can be considerably different from those at the outlet of the test section. Consequently, the correlation must be calculated with different Reynolds and Prandtl numbers in each part of the test section, resulting in a rising slope of the correlation curve.

In Figure 2.28, the experimentally found values with higher Reynolds numbers are not only compared to Gnielinski’s correlation from 1975, valid for $2300 < Re < 10^6$, but also to Gnielinski’s later correlation from 1995, valid actually only for $Re > 10^4$. The latter was chosen to show that towards the end of the test section, Reynolds numbers of the flow increase and lean towards the 1995 prediction, which generally gives between 13% and 20% higher values than the correlation from 1975.

The effect made by the different calculation becomes even more articulate looking at the experiments conducted with an inlet temperature of 40°C:

![Local Nu, turbulent, 40°C inlet, 1st rig generation](image)

Figure 2.29: Experimental data at 40°C inlet and correlations using average fluid properties
Figure 2.30: Experimental data at 40°C inlet and correlations using local fluid properties

Apparently, in 40°C inlet temperature another effect takes place, as can be seen in Figure 2.30. The two tests with the highest Reynolds numbers (Re = 11811 in grey-green diamonds and Re = 7936 in orange squares) exceed even Gnielinski’s correlation from 1995, which did not take place in the test with highest Reynolds number in 25°C inlet (compare Figure 2.28). In both inlet temperatures however, the Nusselt values at the test section inlet are lower than both correlations. It is reckoned that here conduction to the ambient was responsible for these deviations from the correlations.
2.4.3 Recorded data

From the data recorded in the experiments, different characteristic values are calculated to assess the heat transfer of the fluid used in each experiment.

The data logged in each measurement is summed up in Table 2.6:

<table>
<thead>
<tr>
<th>Name</th>
<th>Measured quantity</th>
<th>Location of sensor</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>time</td>
<td>Time</td>
<td></td>
<td>s</td>
</tr>
<tr>
<td>t_ref</td>
<td>Temperature in logger box</td>
<td>in air</td>
<td>°C</td>
</tr>
<tr>
<td>t_1 - t_15</td>
<td>Temperature in position 1-15 of test section</td>
<td>on outside pipe wall</td>
<td>°C</td>
</tr>
<tr>
<td>t_17 - t_19</td>
<td>Temperature in position 16-18 of test section</td>
<td>on outside pipe wall</td>
<td>°C</td>
</tr>
<tr>
<td>t_20 - t_22</td>
<td>Temperatures of flow development region</td>
<td>on outside pipe wall</td>
<td>°C</td>
</tr>
<tr>
<td>t_24</td>
<td>Temperature of test section pipe after outlet clamp</td>
<td>on outside pipe wall</td>
<td>°C</td>
</tr>
<tr>
<td>t_25</td>
<td>Temperature of electrical connection clamp at inlet</td>
<td>on clamp</td>
<td>°C</td>
</tr>
<tr>
<td>t_26</td>
<td>Temperature of electrical connection clamp at outlet</td>
<td>on clamp</td>
<td>°C</td>
</tr>
<tr>
<td>t_27 + t_28</td>
<td>Inlet temperature, two thermocouples</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_29</td>
<td>Outlet temperature in axial insulation part</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_30</td>
<td>Outlet temperature in straight pipe after axial insulation</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_31</td>
<td>Outlet temperature in straight pipe after union tee</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_32</td>
<td>Fluid temperature of nanofluid at inlet of tubular heat exchanger</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_33</td>
<td>Fluid temperature of nanofluid at outlet of tubular heat exchanger</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_35</td>
<td>Fluid temperature of cooling water inlet to tubular heat exchanger</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>t_36</td>
<td>Fluid temperature of cooling water outlet to tubular heat exchanger</td>
<td>in fluid</td>
<td>°C</td>
</tr>
<tr>
<td>Pin</td>
<td>Pressure at test section inlet</td>
<td></td>
<td>bar</td>
</tr>
<tr>
<td>PDiff</td>
<td>Pressure drop over test section</td>
<td></td>
<td>bar</td>
</tr>
<tr>
<td>Pout</td>
<td>Pressure at test section outlet</td>
<td></td>
<td>bar</td>
</tr>
<tr>
<td>Flow</td>
<td>Mass flow</td>
<td></td>
<td>kg/h</td>
</tr>
<tr>
<td>Density</td>
<td>Density of fluid</td>
<td></td>
<td>kg/m³</td>
</tr>
</tbody>
</table>

Channels 16, 23 and 34 are not used for measurements because they showed high deviation from the bath temperature in calibration (23 and 35) or no value at all (16).
2.4.4 System behaviour of third generation test rig

In the last days of this thesis, tests in 25°C and 40°C inlet temperature and both laminar and turbulent flow were conducted with distilled water with the third generation test rig. The laminar tests showed a heat loss of up to 26%, only three of 15 tests had a heat loss of less than 10%. Unfortunately, these tests could not be repeated due to a lack of time. The reason for these high losses, which had not been seen in any other tests conducted until this point, could not be investigated. Because of this problem, the laminar tests are not shown here.

The turbulent tests did not show this high heat loss, they could be evaluated with respect to the systems heat transfer behaviour in comparison with established correlations. One aim for the turbulent tests had been to conduct tests in turbulent flow with Reynolds numbers higher than 10000. For these Reynolds numbers, fully turbulent flow can be assumed, while the development of turbulence between Reynolds numbers Re = 2300 and Re = 10000 depends on many different factors and thus is difficult to determine (compare [24]). This aim could not be reached for the tests with 25°C inlet temperature, because the cooling power provided by the thermostat and the additional plate heat exchanger was not sufficient to remove as much heat as necessary for tests with a Reynolds number higher than Re = 8000 in 25°C inlet temperature. In 40°C however, tests could be conducted up to a Reynolds number of Re = 13000.

2.4.4.1 25°C inlet temperature

![Average Nu vs. Re, distillated water, turbulent flow, all test rig generations, 25°C inlet](image)

Figure 2.31 shows good agreement for the average Nusselt numbers determined with the third generation test rig as well to earlier tests as to Gnielinski’s correlation from 1975. The deviation from the older tests is mostly less than 7%; only in Re = 2500 a bigger deviation can be seen, apparently the flow in the new setup (purple circles) reaches turbulent flow already at lower Reynolds numbers than in the older setups (orange squares and blue diamonds). The deviation from the Gnielinski’s 1975 correlation is slightly larger with the third rig generation than with the older configurations. The new rig’s deviation is up to 3.4%, the older rigs’ deviations were less than 2% for Re > 3000.
In Figure 2.32, local Nusselt numbers of the third test rig are displayed. It can be seen that a thermocouple in the middle of the test section was badly connected. This was discovered between the test at $Re = 4115$ and $Re = 6162$ and was repaired, so the two tests with higher Reynolds numbers didn’t show this irregularity. Unfortunately there was not enough time to repeat the tests with the malfunctioning thermocouple.

Figure 2.33: 2nd generation test rig, turbulent, distilled water, 25°C inlet
It can be seen from comparison of Figure 2.32 and Figure 2.33 that both test rig generations show a good agreement with Gnielinski’s 1975 correlation; deviation is less than 6% at most spots on the test section (except for the inlet region, the values from the wrongly connected thermocouples in the third test rig and the last two thermocouples in each rig). The third test rig generation is equipped with more thermocouples close to the test section inlet, it can be seen that the correlation is met well by those thermocouples in higher Reynolds numbers (less than 5% deviation), but is exceeded in the tests with lower Reynolds numbers by up to 37% in case of Re = 2574. This may be caused by locally laminar flow at the inlet, the test with an average Reynolds number of Re = 2574 shows a local Reynolds number at the inlet of only Re_{x1} = 2375.

It had already been visible in the second rig generation that local Nusselt numbers from the next-to-last thermocouple’s temperature showed higher value than the trend. In the third rig generation, this phenomenon seems to have become worse, now the next-to-last and the last thermocouple’s temperature lead to local Nusselt numbers that are strongly out of trend. In the three tests with higher Reynolds numbers in Figure 2.32, the next-to-last local Nusselt number is up to 12% higher than the correlation, while the other values deviate less than 6%. This should be further investigated or the values from the two last thermocouples should be disregarded in the evaluation of heat transfer with different fluids.

2.4.4.2 40°C inlet temperature

![Graph: Average Nu, turbulent, 40°C inlet](image)

Figure 2.34: Average Nu vs. Re, distilled water, turbulent flow, all test rig generations, 40°C inlet

As Figure 2.34 shows, the third test rig generation shows similar behaviour, judging from average Nusselt numbers, than the older versions of the test rig. Deviations between Nusselt numbers achieved with the different test rigs are mostly less than 5%. One test, in Re = 2500 on the newest test rig, shows a three times as high Nusselt number than the correlation, the reasons for that are not known. An error in data processing may be suspected, but could not be further investigated until the end of this thesis. It can be seen that in 40°C inlet temperature, the Nusselt numbers exceed Gnielinski’s equation from 1975 clearly (by up to 8%), while leaning towards Gnielinski’s correlation from 1995, which could not be seen in 25°C inlet temperature (compare Figure 2.31). For Reynolds numbers of Re = 10000 and higher, where
Gnielinski’s 1995 correlation is valid, the deviation of all tests conducted at such high Reynolds numbers is less than 3%.

Figure 2.35: 3rd generation test rig, turbulent, distilled water, 40°C inlet

Figure 2.36: 2nd generation test rig, turbulent, distilled water, 40°C inlet

Local Nusselt numbers of the third generation test rig are shown in Figure 2.35 and can be compared to those determined with the second generation test rig in Figure 2.36. It can be seen that the slope of the local Nusselt numbers agrees better with the correlations’ slopes in the third test rig generation. On the
other hand, especially the tests with higher Reynolds numbers do not form a smooth line in Figure 2.35, while the values formed a fairly smooth line in Figure 2.36, apart from single values.

It can be seen in Figure 2.35 that the exceptionally high average Nusselt number achieved in the Re = 2539 test seen in Figure 2.34 is caused by local Nusselt numbers which are also generally much higher than the correlation, especially toward the test section outlet. No explanation is available for this; it has to be further investigated.

The phenomenon of higher deviation in local Nusselt numbers for the two last measuring points with high x*-values is more articulate in 40°C inlet than in 25°C (compare Figure 2.32). Especially the next-to-last measuring point is articulately out of trend, showing a deviation of up to 15% from Gnielinski’s correlation from 1995, while the rest of the values deviates less than 4% from the correlation, except for the inlet area, where deviations go up to 9%. Further investigation is necessary here.

The deviation of local Nusselt numbers from Gnielinski’s 1995 correlation for tests with an average Reynolds number of Re = 10000 and higher is mostly less than 4%. Only at the inlet, deviation get as high as 9% for the first value and the next-to-last local Nusselt numbers deviate by up to 15%, as it was mentioned above.
3 Calculation

All signals that are recorded during experiments were recorded over three minutes. The recording was only started after making sure that the system had reached steady state. Steady state was assumed when the signal reaching steady state last showed less than 0.05K variation during three minutes. In first and second test rig generation, this was the temperature after the mass flow meter, because the mass flow meter had the biggest mass of all devices in the system and its temperature had to assimilate to the test section outlet temperature each time the flow or heating conditions were changed.

Quality control measures were taken after the experiment to ensure the system had been in steady-state indeed during the experiment. One measure was to calculate the standard deviation from the recorded signals. The system was considered steady-state if the maximum standard deviation of all recorded temperatures was less than 0.2K. This may have been a stronger deviation than the deviation seen at the mass flow meter before the experiment start, because the wall temperatures on the test section fluctuated more strongly in the transition region between Re = 2000 and Re = 3000 than the temperature of the mass flow meter. In most cases, when either laminar or turbulent flow was stable, the maximum standard deviation was less than 0.05K. The recorded data was analysed, this chapter is dedicated to explain detailed formulas.

3.1 Material properties

The material property data are based on reference data for the base fluid. The thermal conductivity for water is calculated using a formula from Pátek et al. [25]; specific heat capacity, density and dynamic viscosity for water are calculated from polynomials that had been calculated from values taken from the NIST database [26]. For Antifrogen N (AFN), fluid properties were obtained from the supplier Clariant [27].

With these values for the base fluid, thermal conductivity and dynamic viscosity of the nanofluid are calculated from base fluid values and constant factors:

\[
k_{nf,j} = k_{bf,j} \cdot r_k
\]

\[
\mu_{nf,j} = \mu_{bf,j} \cdot r_k
\]

These constant factors were determined experimentally by Seyed Aliakbar Mirmohammadi and Mohammadreza Behi in KTH laboratories as part of their Master thesis work [28], still on-going by the end of this thesis. The factors were determined at 20°C; by the end of this thesis data for higher temperatures were not available. It was assumed that the ratios are the same at all temperatures occurring in the conducted experiments.

Specific heat capacity and density of the nanofluid are calculated from literature values for the base fluid and the particles and the concentration. \(C_v\) stands for the volume concentration and \(C_m\) for the mass concentration of particles in the nanofluid.

\[
\rho_{nf,j} = (1 - C_{v,j})\rho_{bf,j} + C_{v,j} \rho_p
\]

\[
\rho_f = (1 - C_m)\rho_{bf} + C_m \rho_p
\]

With the mass concentration given by the nanofluid manufacturer, volumetric concentration in equation (3.3) can be calculated with:
\[ C_v = \frac{C_m}{C_m + (1 - C_m) \frac{\rho_p}{\rho_{bf}}} \quad (3.5) \]

All four fluid properties are temperature dependent, but the dependency of the specific heat capacity is so small (less than 0.02% difference at a temperature change of 15K, which is the highest temperature difference occurring during experiments) that it was neglected. The specific heat is calculated using the mean fluid temperature, calculated from the fluid temperature at the inlet \(t_{in}\) and the fluid temperature at the outlet \(t_{out}\):

\[ t_{f,ave} = \frac{t_{out} + t_{in}}{2} \quad (3.6) \]

The other three fluid properties were calculated locally, using the fluid temperature at each spot in the pipe, calculated from the inlet temperature \(t_{in}\) and the heating to the considered point with the distance \(x_j\) to the start of heating:

\[ t_{f,j} = t_{in} + \frac{\dot{q} \pi d_i x_j}{m c_{nf}} \quad (3.7) \]

Especially the dynamic viscosity is strongly temperature dependent.

Fluid properties used in the average calculation were nevertheless generally calculated with the average fluid temperature.

For means of comparison, the thermal conductivity is calculated with Maxwell’s formula for conductivity of small spheres in another continuous medium, as found in \([29]\):

\[ k_{nf,Maxwell} = k_f + \frac{k_p + 2 k_f + 2 C_v (k_p - k_f)}{k_p + 2 k_f - C_v (k_p - k_f)} \quad (3.8) \]

### 3.2 Test design calculation

Before each experiment, volume flow rate and electrical heating power were calculated, taking into account the desired Reynolds number, the inlet temperature and the average temperature difference between pipe wall and fluid. The desired average temperature difference between the inner pipe wall and the bulk fluid was chosen to be 5K. This value must not be chosen too small; otherwise share of the thermocouples’ measurement error in the measured signal will be too high. The value of 5K was suggested from experience in the research group of Prof. Björn Palm and Ehsan Bitaraf Haghighi.

The detailed calculation is as follows:

\[ P_{el,des} = \Delta t_{h,t,avg,des} h_{avg,des} (\pi d_i L) \quad (3.9) \]

Here the average heat transfer coefficient is achieved from the average Nusselt number:

\[ h_{avg,des} = \frac{N_{u,des} k_{nf}}{d_i} \quad (3.10) \]

The average Nusselt number for laminar flow can be assumed as the empirical value for the asymptote of the mean Nusselt number for low values of \((Re Pr d_i/l)\), as seen in \([20]\):
The average Nusselt number for turbulent flow can, as seen in [30], be estimated by

\[ \text{Nu}_{\text{des,turb}} = 0.116 \left( \text{Re}_{\text{des}}^{2/3} - 125 \right) \text{Pr}_{\text{nf}}^{1/3} \left( 1 + \left( \frac{d_i}{L} \right)^{2/3} \right) \left( \frac{\mu_w}{\mu_f} \right)^{0.14} \] (3.12)

with \( \text{Pr}_{\text{nf}} = \frac{c_{\text{nf}} \mu_{\text{nf}}}{k_{\text{nf}}} \). (3.13)

The thermal conductivity is calculated as shown in chapter 3.1.

The quotient of viscosity at the wall and in bulk fluid in (3.12),

\[ \frac{\mu_w}{\mu_f}, \] (3.14)

was assumed to be = 1 and was thereby neglected in the test design calculation.

The volume flow rate is calculated as

\[ \dot{V}_{\text{des}} = \text{Re}_{\text{des}} \mu_{\text{nf}} \frac{\pi}{4} d_i/\rho_{\text{nf}} \] (3.15)

Density and dynamic viscosity are calculated as shown in chapter 3.1. The average fluid temperature for this process is achieved with

\[ t_{f,\text{avg,des}} = t_{\text{in,des}} + \frac{1}{2} (t_{\text{out,des}} - t_{\text{in,des}}) \] (3.16)

Here the inlet temperature is given for each test, usually 25°C or 40°C. The outlet temperature is calculated with

\[ t_{\text{out,des}} = t_{\text{in,des}} + \frac{P_{\text{el}}}{\rho_{\text{nf}} \dot{V}_{\text{des}} c_{\text{nf}}} \] (3.17)

### 3.3 Heat transfer in the test section

#### 3.3.1 Local calculation

##### 3.3.1.1 Laminar flow

The local Nusselt numbers \( \text{Nu}_{x,j} \) are calculated from the local heat transfer coefficients \( h_{x,j} \).

\[ \text{Nu}_{x,j} = \frac{h_{x,j} d_i}{k_{\text{nf},j}} \] (3.18)

The local heat transfer coefficients are calculated for each of the 16 parts of the test section with:

\[ h_{x,j} = \dot{q} (t_{w,i,j} - t_{f,j}) \] (3.19)

Here the heat flux density is calculated assuming a homogenous dissipation of the electrical power in the pipe wall of the test section:
\[ q = \frac{(t_{\text{out}} - t_{\text{in}}) \dot{m} c_{nf}}{\pi d_i L} \]  

In (3.19), the local fluid temperature is calculated using equation (3.7). The wall temperature \( t_{w,i,j} \) on the inside wall of the test section pipe is calculated from the measured outside wall temperature \( t_{w,o,j} \) using Fourier’s heat conduction equation with the assumptions that the heat flux to the ambient equals zero and the pipe wall acts as an inner heat source. The equation for \( t_{w,i,j} \) is:

\[ t_{w,i,j} = t_{w,o,j} + \frac{\dot{Q}}{4 \pi L k_{\text{pipe}}} \left( \frac{\varphi}{\varphi - 1} \right) \left( 1 - \ln(\varphi - 1) - 1 \right) \]  

with

\[ \varphi = \frac{d_o^2}{d_i^2}. \]

The local Nusselt numbers are usually shown in a graph vs. a dimensionless distance from the start of heating, \( x^* \):

\[ x_j^* = \frac{1}{Gz} = \frac{x_j}{d_i \Re_j \Pr_j} = \frac{\pi}{4} \frac{k_{nf,j} x_j}{\dot{m} c_{nf}} \]

The calculated local Nusselt numbers from the experiments using distilled water in the system are compared to the correlation for hydrodynamically developed laminar flow mentioned in VDI Heat Atlas [20] to evaluate the system’s behaviour:

\[ \text{Nu}_{x,\text{VDI},j} = \left\{ \begin{array}{ll} 4.354^3 + 1 + \left[ 1.302 \sqrt{Gz_j - 1} \right]^3 \frac{1}{2} \left( \frac{\Pr_j}{\Pr_{w,j}} \right)^{0.11} & \text{if } 0.00005 \leq x^* \leq 0.0015 \\ 3.302 x^{* - 1/3} - 1; & \text{if } x^* < 0.00005 \\ 1.302 x^{* - 1/3} - 0.5; & \text{if } 0.00005 \leq x^* < 0.0015 \\ 4.364 + 8.68 (10^3 x^*)^{-0.506} \cdot e^{-41 x^*}; & \text{if } x^* \geq 0.0015 \end{array} \right. \]  

Equation (3.24) is valid for all possible Gz numbers. In graphical comparison, the correlation is shown assuming the ratio of Prandtl numbers in (3.24) to be = 1, otherwise too many correlation lines very close next to each other would be created in the figures.

Comparison to the Shah correlation for thermally developing but hydraulically developed laminar flow in a round pipe with uniform heat flux as found in [31] was relinquished, because the difference between the predicted Nusselt numbers was always smaller than 0.03%:

\[ \text{Nu}_{x,\text{Shah},j} = \left\{ \begin{array}{ll} 3.302 x^{* - 1/3} - 1; & \text{if } x^* < 0.00005 \\ 1.302 x^{* - 1/3} - 0.5; & \text{if } 0.00005 \leq x^* < 0.0015 \\ 4.364 + 8.68 (10^3 x^*)^{-0.506} \cdot e^{-41 x^*}; & \text{if } x^* \geq 0.0015 \end{array} \right. \]  

### 3.3.1.2 Turbulent Flow

In turbulent flow, the calculation is analogue to the calculation for laminar flow, but the measured local Nusselt numbers are compared to Gnielinski’s correlations for turbulent flow. Two correlations are consulted for comparison, Gnielinski’s correlation from 1995 (see [24]) for fully turbulent flow (Re > 10^9)
and the derivation from Gnielinski’s correlation for average Nusselt numbers from 1975 (see [32]), valid for \(2300 < \text{Re} < 10^6\):

\[
\text{Nu}_{x,\text{avg},2300<\text{Re}<10^6,j} = \frac{\xi_{2,j} (\text{Re}_j - 1000) \text{Pr}_j}{1 + 12.7 \sqrt{\xi_{2,j} \left( \frac{\text{Pr}_j}{\text{Pr}_{\text{w},j}} \right)^{2/3}} \left( \frac{d_j}{x_j} \right)^{2/3}} \left( 1 + \frac{1}{3} \left( \frac{d_j}{x_j} \right)^{2/3} \right) \left( \frac{\text{Pr}_j}{\text{Pr}_{\text{w},j}} \right)^{0.11},
\]

\[
\xi_{2,j} = \frac{1}{(1.82 \log(\text{Re}_j) - 1.64)^2},
\]

\[
(3.30)
\]

\[
(3.31)
\]

\[
(3.32)
\]

\[
(3.33)
\]

### 3.3.2 Average calculation

#### 3.3.2.1 Laminar flow

**Test section:**

The average Nusselt number is calculated from the local Nusselt numbers, each weighed by the length of the section in which the particular thermocouple, which leads to the local Nusselt number, is located:

\[
\text{Nu} = \frac{\sum_{j=1}^{16} \text{Nu}_{x,j} l_j}{L}
\]

To compare the experimentally found Nusselt numbers to correlations from the literature, the Nusselt number is plotted in a diagram as a function of the Reynolds number, as used in chapter 0.

\[
\text{Re} = \frac{v d_i \rho_{nf}}{\mu_{nf}}
\]

In (3.31), the Reynolds number is found with the mean fluid velocity \(v\) of the nanofluid in the test section, which is found as follows:

\[
v = \frac{\dot{m}}{\rho_{nf} d_i^2/4 \pi}
\]

The average Nusselt numbers found with distilled water in the system are then compared to two correlations. The first correlation is mentioned in VDI Heat Atlas [20] for hydrodynamically developed laminar flow to evaluate the system’s behaviour:

\[
\text{Nu}_{\text{VDI}} = \left( 4.364^3 + 0.6^3 + \left( 1.953 \sqrt{\text{Re} \text{Pr} d_i/L} - 0.6 \right)^3 \right)^{1/3} \left( \frac{\text{Pr}}{\text{Pr}_{\text{w}}} \right)^{0.11},
\]

\[
(3.34)
\]
In graphical comparison of different tests with water, the correlation is shown only for one set of data, otherwise too many correlation lines very close next to each other would be created in the figures. The difference between the correlations for the different tests with water was always smaller than 1%.

Secondly, Shah’s correlation\cite{31} for Hagen-Poiseuille flow in a round tube with uniform heat flux:

\[
\text{Nu}_{\text{Shah}} = \begin{cases} 
1.953 \sqrt[3]{x^*}, & x^* \leq 0.03 \\
4.364 + 0.0722/x^*, & x^* > 0.03
\end{cases}
\] (3.35)

Besides comparing the Nusselt vs. Reynolds values of nanofluids and base fluids, other criteria are also consulted. One of the quantities is the average heat transfer coefficient \(h\):

\[
h = \frac{\text{Nu} \ k_{nf}}{d_i}
\] (3.36)

### 3.3.2.2 Turbulent flow

The average calculation for turbulent flow follows the laminar calculation, but a few changes had to be made:

Gnielinski proposed different correlations to describe the average Nusselt numbers for turbulent flow. Gnielinski’s correlation from 1995, valid for \(Re > 10^4\) and found in [24] and, originally, in [33] is used for comparison:

\[
\text{Nu}_{\text{Gnielinski,Re}>10^4} = \frac{\xi_1 \ Re \ Pr}{1 + 12.7 \sqrt{\xi_1/8} \left( 1 + \left( \frac{d_i}{L} \right)^{2/3} \right) \left( \frac{Pr}{Pr_w} \right)^{0.11}}
\] (3.37)

with \(\xi_1\) as mentioned in (3.27) and the Prandtl numbers ratio as mentioned in (3.34).

For \(Re > 10^4\), equation (3.37) has less average deviation from experimentally found Nusselt numbers in the literature than Gnielinski’s equation from 1975 [32] had. Since the Reynolds numbers in the turbulent experiments carried out during this work are mostly in the area of \(2300 < Re < 10^4\) and equation (3.37) is not valid there, the experimentally found Nusselt numbers are also compared to Gnielinski’s equation from 1975:

\[
\text{Nu}_{\text{Gnielinski,2300<Re<10^4}} = \frac{\xi_2 \ (Re - 1000) \ Pr}{1 + 12.7 \sqrt{\xi_2/8} \left( 1 + \left( \frac{d_i}{L} \right)^{2/3} \right) \left( \frac{Pr}{Pr_w} \right)^{0.11}}
\] (3.38)

with \(\xi_2\) as mentioned in (3.29) and the Prandtl numbers ratio as mentioned in (3.34).

Experiments mentioned in the literature [33] showed that Nusselt numbers decrease strongly in the Reynolds region \(2300 < Re < 10^4\). Equation (3.38) predicts the Nusselt numbers well for \(2300 < Re < 10^4\) (see [33]). For \(Re < 10^4\) it always gives smaller values than equation (3.37).
Finally, Gnielinski proposed an equation to offer a continuous correlation without a point of discontinuity at Re=2300 in 1995. It links the equations for laminar flow to equation (3.37). This equation is also compared to the experimentally found Nusselt numbers. The equation is:

\[
\text{Nu}_{\text{Gn,2300}<\text{Re}<10^4} = \left((1 - \gamma) \text{Nu}_{\text{VDI,Re}=2300} + \gamma \text{Nu}_{\text{Gn,Re}>10^4,\text{Re}=10^4}\right) \left(\frac{\text{Pr}}{\text{Pr}_w}\right)^{0.11} \tag{3.39}
\]

with \(
\gamma = \frac{\text{Re} - 2300}{10^4 - 2300}.
\) \tag{3.40}

In equation (3.39), the VDI Heat Atlas correlation is calculated with Re = 2300 and Gnielinski’s correlation is calculated with Re = 10000, the Prandtl numbers ratio as mentioned in (3.34).

In graphical comparison of different tests with water, the correlations are shown only for one set of data, otherwise too many correlation lines very close next to each other would be created in the figures. The difference between the correlations for the different tests with water was always smaller than 1%.

### 3.4 Heat transfer in tubular heat exchanger

The tubular heat exchanger on the back side of the test rig can be used as a second unit to measure heat transfer. Different boundary conditions apply here, because the tubular heat exchanger works as a counter-flow heat exchanger, so neither heat flux density nor temperature are constant at the walls transferring the heat.

The NTU value (Number of Transfer Units) is calculated for all experiments to compare the heat exchange in the tubular heat exchanger with different fluids. NTU is defined as:

\[
\text{NTU} = \frac{UA}{m c_{nf}} = \frac{Q}{\text{LMTD} m c_{nf}} = \frac{t_{\text{HE,nf,in}} - t_{\text{HE,nf,out}}}{\text{LMTD}} \tag{3.41}
\]

With LMTD defined as

\[
\text{LMTD} = \frac{\left(t_{\text{HE,nf,in}} - t_{\text{HE,cf,out}}\right) - \left(t_{\text{HE,nf,out}} - t_{\text{HE,cf,in}}\right)}{\ln \left(\frac{t_{\text{HE,nf,in}} - t_{\text{HE,cf,out}}}{t_{\text{HE,nf,out}} - t_{\text{HE,cf,in}}}\right)} \tag{3.42}
\]
3.5 Pressure drop calculation

3.5.1 Laminar flow

Nanofluids and base fluids are also compared regarding the pressure drop $\Delta p$ caused by the flow through the test section and, derived from that, the friction factor $f$:

$$ f = \frac{\Delta p}{l_{d \eta}} \frac{d_i}{\rho_f} \frac{\rho_f}{\nu^2} \frac{2}{\nu^2} $$

(3.43)

The experimentally found friction factor $f$ is compared to the theoretical value, as calculated with this form of the Hagen-Poiseuille equation [34]:

$$ f_{hp} = \frac{64}{\nu} $$

(3.44)

3.5.2 Turbulent flow

The experimentally determined friction factor (see equation (3.43) is compared to Blasius' correlation (as can be found in [34]) in turbulent flow, valid for $3000 < \nu < 10^5$:

$$ f_{bl} = \frac{0.3164}{\sqrt{\nu}} $$

(3.45)
4 System calibration and correction

4.1 Pump testing / Flow control
The gear-pump used in the test rig displays the running speed in rounds per minute (rpm) and shows a pre-calibrated value for the volume flow that rises linearly with the running speed. The displayed value is used to adjust the system to the desired volume flow rate, depending on the desired Reynolds number of each experiment. The Reynolds number, which is actually achieved, is calculated from the mass-flow measured during the experiment.

The tests were conducted to determine how precise the pre-calibrated flow rate value is at present state. The result presented in Figure 4.1 showed less than 1% difference between the displayed and the measured value, for a volume flow of more than 650 ml/min the mistake is even smaller with a deviation of less than 0.5%. It was concluded that no changes had to be made to the pump. The flow rate used for the experiment evaluation is given by the mass flow meter, not by the pump.

4.2 Pressure measurement correction

4.2.1 Absolute pressure transducers
The system was equipped with two absolute pressure transducers (ClimaCheck PA-22S, range: 0…10bar), located close to the inlet of the test section and at the outlet of the test section, respectively. To be able to use them to measure the pressure drop along the straight tube and connections between them it was necessary to correct the differential offset. Nine tests at different pressure levels were conducted, each for one minute with no heat applied by the electric heating system and no flow in the system so the pressure difference between the two sensors should have been zero. The system was filled with Nitrogen while conducting the tests.

The average from each one-minute-test was calculated, Figure 4.2 shows all nine measurements and a curve that was fit to the measurements to enable a continuous error correction. The function describing the curve was used to correct the pressure drop whenever it was measured using the two absolute pressure transducers.
4.2.2 Differential pressure transducer

The system was equipped with one differential pressure transducer (GE Druck PTX5060-TA-A3-CC-H0-PA, range 0 – 1.5 bar), which is connected to the test section in the same places as the absolute pressure transducers. Thereby the same pressure drop is measured by both systems. For details see chapter 0.

The offset of the differential pressure transducer was recorded for one minute each at two pressure levels, the lowest and the highest pressure to be expected during experiments. The system was filled with Nitrogen; there was no flow in the system. A function was calculated to subtract the offset from the measured value before integrating the pressure difference into the following calculation. The offset can be seen in Figure 4.3.

\[ y = -0.0004x^2 + 0.0042x - 0.0026 \]

\[ y = 0.0009778817x - 0.0000811520 \]
4.3 Thermocouple correction

All thermocouples used in the test rig had been tested before they were built in on the test rig. Testing at 20°C, 23°C, 40°C, 50°C and 60°C showed that none of the thermocouples deviated by more than 0.1K from the temperature of the bath they were tested in. The bath temperature was measured with a T100 resistance thermometer.

Before the experiments with nanofluids, the 16 thermocouples on the outside wall of the test section were calibrated. For this process, the system was filled with Stockholm tap water. Measurements were conducted when the temperature in the water tank that supplies cooling/heating water to the tubular heat exchanger was set to 20°C, 30°C, 40°C, 50°C and 60°C, respectively. This tubular heat exchanger is used to cool the nanofluid when the experiments are conducted to determine the heat transfer coefficient. The pump ran at 500 rpm which resulted into a volume flow of about 360 ml/min (about 0.5 m/s average flow speed in the test section), which again resulted into a Reynolds number of between Re = 1800 at 20°C and Re = 4000 at 60°C in the test section. At 60°C bath temperature the flow cooled off by 0.3°C flowing through the test section (ambient temperature was 28.8°C). This was regarded so little that ideal insulation was assumed along the test section. This means that no heat flux through the glue connection between pipe and thermocouples could have caused a measurement error, but that only offset errors at different temperature levels had to be corrected. Omegabond 101 was used to fix the thermocouples on the pipe, a two-component epoxy adhesive with a high electrical resistance and high thermal conductivity (about 1 W/mK, compare [35]). But even if the heat flux to the ambient is not neglected, it still can be concluded that the temperature drop from the test section wall through the glue to the thermocouples is much less than the pressure drop from the thermocouples through the insulation to the ambient.

The temperatures at each of the 16 thermocouples were recorded for one minute after the system had reached steady-state. All pipes in the test rig were insulated, so it could be concluded, that the temperature was the same everywhere. Nevertheless, the wall thermocouples showed different data each, so for each thermocouple a mean value from the data collected during the recording time was calculated for each thermocouple at each of the five different temperatures. In a next step, a mean temperature was calculated at each of the five temperature levels. These reference temperatures were considered the correct temperatures in each temperature level in the following. Now the deviation from the reference temperature was calculated for every thermocouple in each temperature level and based on these data a deviation curve was fitted for each of the 16 thermocouples. Now a continuous measuring error correction is available for wall temperatures in the test section between approx. 21°C and 57°C, which were the mean wall temperatures during the tests. This correction was applied to all measured values in all experiments that are part of this thesis. The results of this procedure can be seen in Figure 4.4.
For each thermocouple, the individual deviation from the average temperature of all thermocouples is calculated at each of the six temperature levels. The red squares in Figure 4.4 show the average deviation of each thermocouple from each of the six mean temperatures at the six temperature levels. This information only says that in average the six corrected temperatures are after the correction the same as the six mean temperatures. The information provided by the red crosses adds the information, that in the process of calculating the average of the corrected values’ deviations from the mean temperatures at each of the six temperature levels, the standard deviation is close to zero. This again means that for all the thermocouples all the corrected values are close to the average temperatures at each of the six temperature levels.

Additional test were carried out with air and vacuum in the system. The measured temperatures showed the same tendency in differences of the measurement values, although the differences were so much higher that it seemed plausible to conclude that the temperature in the test section was not as homogenous as when water flowed through the system.
5 Experiments

The aim of the experiments was to measure the heat transfer behaviour of different nanofluids in different flow conditions. The pressure drop through the test section was measured to determine whether a potentially better heat transfer behaviour of a nanofluid was going along with a penalty of increased pumping power.

Heat transfer was measured in two devices simultaneously: The main device was the test section, designed to provide a constant heat flux boundary condition at the pipe wall. The minor device for heat transfer measurement was the tubular heat exchanger used for cooling down the nanofluid after it had been heated up in the test section. Results from the test section could be compared to established correlations for heat transfer, while results from the tubular heat exchanger were compared to each other employing a non-dimensional number, NTU (Number of Transfer Units).

After some first experiments on the first generation test rig, it was tried to find reasons for the limited agreement with Nusselt number correlations. Changes were made to the test rig, but problems with validation still persisted, so it was decided to make further changes. It proved later that not only issues in the experimental setup were responsible for the difficulties regarding validation, but also issues in the calculation. This was only discovered when the third test rig generation was already under construction, so despite better system behaviour regarding validation, no tests with nanofluids were conducted with the second test rig generation. On the third test rig generation, only validation tests could be conducted until the end of this thesis.

Table 5.1 shows the fluids tested on the first generation rig, which is described in chapter 2.2:

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Base fluid</th>
<th>Particle</th>
<th>Concentration, wt-%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Distilled Water</td>
<td>Distilled Water</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>KTH-CeO2-MS06</td>
<td>Distilled Water</td>
<td>CeO₂</td>
<td>9</td>
</tr>
<tr>
<td>KTH_AFN-BF001</td>
<td>Water-Antifrogen N, 50/50 vol-%</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>CeO2_KTH_MS018AB</td>
<td>Water-Antifrogen N, 50/50 vol-%</td>
<td>CeO₂</td>
<td>9</td>
</tr>
</tbody>
</table>

Each fluid was tested in laminar flow with Reynolds numbers of Re = 500 and higher and in turbulent flow with Reynolds numbers up to Re = 8000 or, in one case, Re = 12000. Tests were conducted at inlet temperatures to the test section of 25°C, 40°C and, partly, 55°C. After testing a base fluid, tests with a nanofluid based on the before tested base fluid were conducted at the same set of values for pump speed and electrical heating.

Due to the long search for the reasons of the unexpected system behaviour compared to correlations, experiments could only be conducted on the first generation test rig. The results of those experiments will be shown in this chapter.
5.1 Quantities used for nanofluid assessment

Several quantities are evaluated to assess the qualification of a nanofluid as a heat transfer medium.

a) \( \text{Nu vs. Re} \)

Firstly, as it is widespread in literature about nanofluids, Nusselt numbers vs. Reynolds numbers are compared to review heat transfer conditions using different fluids. This is most interesting to look at if pressure drop does not need to be considered, i.e. in applications, where removing heat can be critical and therefore a higher penalty in pumping power can be tolerated to be able to cool the system at all. It also can give an idea of the connection between flow regime and heat transfer with the liquids in question.

b) \( h \text{ vs. } \Delta \rho \)

Less popular but still known in literature, the heat transfer coefficient vs. pressure drop is considered to determine whether a different behaviour in heat transfer goes along with a change in pressure drop and thus pumping power. This comparison is important for practical applications; it can help to determine how big the investment in pressure drop (and hereby pumping power) must be to achieve a certain heat transfer coefficient with a heat transfer fluid.

c) \( h \text{ vs. volume flow} \)

For applications where pressure drop is not important, the development of the heat transfer coefficient at different volume flows can be considered.

d) \( \text{Nu/}f \text{ vs. Re} \)

Not seen in literature by the author, the dimensionless quotient of Nusselt number and friction factor is considered at different Reynolds numbers. The aim behind this quotient was to compare heat transfer coefficient and pressure drop of different fluids at different Reynolds numbers to get information about the fluid’s behaviour at different flow regimes. A dimensionless number was searched containing heat transfer coefficient and pressure drop. Here, Nusselt number and friction factor were chosen.

With Reynolds number being defined as

\[
\text{Re} = \frac{v_d \rho}{\mu},
\]

(5.1)

volume flow and Reynolds number will not be the same in the same pipe, if density and/ or dynamic viscosity of two compared fluids are different. This is the reason why comparison of fluids at the same Reynolds number can give a different impression of the heat transfer performance than looking at \( h \text{ vs. } \Delta \rho \) of a fluid: The conditions leading to the same Reynolds number may be different for both fluids, resulting in different heat transfer and pressure drop behaviour. This effect can be seen exemplarily in the comparison of Figure 5.1 and Figure 5.2. \( \text{Nu/}f \) is rather of academic than of practical interest.

The comparison of \( \text{Nu/}f \) does not contain more information than \( \text{Nu vs. Re} \), if the friction factor \( f \) follows the correlation or equation for \( f \). The reason for this is that the correlations or equations for \( f \) only depend on the Reynolds number, which leads to the same friction factor for any fluid at the same Reynolds number. Therefore, when \( \text{Nu/}f \) is plotted vs. Reynolds number, only a constant factor will distinguish the results from the comparison of \( \text{Nu vs. Re} \) and the pressure drop behaviour of the compared fluids will not be considered. The comparison will nevertheless be shown in the following to show by comparison to the \( \text{Nu vs. Re} \) diagrams if the friction factor of the nanofluid follows the correlation.

In both comparisons containing Nusselt and Reynolds numbers, a nanofluid with a higher Prandtl number will show a higher Nusselt number, if the fluid shows a heat transfer behaviour that is close to what correlations describe. It is therefore recommended to consult dimensional quantities for practical nanofluid assessment, especially \( h, \Delta \rho \) and volume flow.
Figure 5.1: Nanofluid assessment using Nusselt number and friction factor

In Figure 5.1, Figure 5.2 and Figure 5.3, distilled water as base fluid is compared to the nanofluid KTH-CeO2-MS06, where CeO$_2$-nanoparticles were added to distilled water.

In Figure 5.1, Nu/$f$ is plotted vs. Re, so the dimensionless quotient of heat transfer coefficient and pressure drop is compared at different flow regimes. The nanofluid shows a higher quotient of Nu/$f$ at the same flow regime.

In Figure 5.2, the ratio of heat transfer coefficient and the pressure drop is plotted vs. the volume flow rate. Instead of qualitatively showing the same result as Nu/$f$ vs. Re, in this way of comparison the base fluid shows a higher, or, in one point, the same performance. It can be assumed that this comparison is practically more relevant, because in most heat transfer applications the heat transfer coefficient, which can be achieved with a certain pumping power is more important than the knowledge of which flow regimes leads to which ratio of heat transfer and pressure drop. The jump of values around 400 ml/min is caused by the change from laminar to turbulent flow.
Figure 5.2: Nanofluid assessment with dimensioned quantities

Figure 5.3, where $h$ is plotted vs. $\Delta p$, shows the same result as Figure 5.2, namely that the base fluid performs better than the nanofluid. It shows the effect of change from laminar to turbulent flow regime less drastically than Figure 5.2, but the tendency is still visible. This comparison is wider spread in literature than plotting $h/\Delta p$ vs. volume flow rate and qualitatively the statement of both plots is the same. For these reasons, $h$ vs. $\Delta p$ was selected for means of comparing base fluids and nanofluids.

Figure 5.3: Nanofluid assessment using heat transfer coefficient and pressure drop
e) **NTU vs. Re**

Lastly, the heat transfer performance in the tubular heat exchanger is reviewed by looking at the NTU (Number of transfer units, dimensionless number indicating heat exchanger performance) in different Reynolds numbers. NTU can be reduced to a ratio of temperature differences, where the temperature difference of the nanofluid is divided by the logarithmic mean temperature difference in the counter flow heat exchanger (see chapter 3.4). The Reynolds number in the tubular heat exchanger is always lower in the same test than in the test section, because the inner diameter of the inner pipe of the heat exchanger is not 3.7 mm, as in the test section, but 6 mm.
5.2 First generation test rig validation

The system’s heat transfer behaviour with distilled water was compared to established correlations (given in chapter 3.3) to validate the system’s applicability. The comparison was made for local and average Nusselt numbers both in laminar and turbulent flow and both for 25°C and 40°C inlet temperature. It shall be added here that the setup at this point did not allow to reach Reynolds numbers higher than \( \text{Re} = 8000 \). This was not ideal for validation, because in the transition area between Reynolds number \( \text{Re} = 2300 \) and \( \text{Re} = 10000 \), the actual grade of turbulence depends strongly on smoothness of the pipe and form of inlet, which was difficult to control on the test rig.

5.2.1 Laminar validation

5.2.1.1 25°C inlet temperature

For laminar flow, experimentally found average Nusselt numbers were compared to Shah’s correlation [31] and the correlation mentioned in VDI Heat Atlas [20], as shown in Figure 5.4.

![Figure 5.4: Validation with average Nusselt numbers in laminar and 25°C inlet temperature](image)

It can be seen that a bend exists at approximately \( \text{Re} = 1500 \). For higher Reynolds numbers, the experimental data is close to the correlations, for lower Reynolds numbers, it is considerably lower (13% lower at \( \text{Re} = 1200 \)). The reason for this effect is not clear, but it is assumed that it is related to the kind of electrical connection to the test section pipe, because just this effect had vanished later in the second test rig generation, where shape of the electrical connection clamps had been changed.

The data correlation is much closer to the correlation for thermally developing and hydrodynamically developed flow than to the correlation for hydrodynamically and thermally developing flow. This was not sure before, because adaptors and pipe connections were built in between the flow development area and the test section in the first generation of the test rig.
The local Nusselt numbers, shown in a double-logarithmic diagram in Figure 5.5, are generally below the Shah correlation, although the deviation is mostly less than 10%. It’s becoming clear which part of the test section is responsible for the characteristic bend in the average Nusselt numbers: The entrance region of the tests with Re = 1500 and higher Reynolds numbers shows a higher than predicted Nusselt number than the tests with lower Reynolds number, while in the rear part of the tube all tests’ Nusselt numbers are close together. The difference appears to be in the front part of the test section. The reasons for this effect are unknown.

Figure 5.5: Validation with local Nusselt numbers in laminar and 25°C inlet temperature
5.2.1.2 40°C inlet temperature

Figure 5.6: Validation with average Nusselt numbers in laminar and 40°C inlet temperature

Figure 5.6 displays that average Nusselt numbers were mostly within 10% deviation from the VDI Heat Atlas correlation for the 40°C experiments. The same bend as in Figure 5.4 is recognisable, but the general level of values is higher in the 40°C experiments. In 40°C inlet temperature too, the experimental values are much closer to the correlation for thermally developing and hydrodynamically developed flow than to the correlation for both thermally and hydrodynamically developing flow.

Figure 5.7: Validation with local Nusselt numbers in laminar and 40°C inlet temperature
Figure 5.7 shows local Nusselt numbers for 40°C inlet tests. Like in 25°C inlet temperature, values close to the inlet show a higher deviation from the correlation than rear values do. Also similar to the result from the 25°C inlet tests is that the two tests with lower Reynolds numbers show lower Nusselt numbers than the correlation close to the inlet, while the two tests with higher Reynolds numbers exceed the correlation’s values, in 40°C inlet even stronger than in 25°C inlet.

It can be concluded that validation of the first generation test rig is limited for laminar flow. The bend in the average Nusselt numbers indicate a qualitative error in the rig, comparison of local Nusselt numbers with the correlation reveals partly strong deviation, especially in the region close to the inlet.

5.2.2 Turbulent validation

5.2.2.1 25°C inlet temperature

Figure 5.8 displays that average Nusselt numbers in turbulent and 25°C inlet temperature show a good agreement with Gnielinski’s correlation (valid for $2300 < Re < 10^6$) from 1975 [32] for Reynolds numbers of $Re = 3000$ and higher. This correlation is said to predict the experiments found in the literature well in the range of $2300 < Re < 10^6$ (see [33]). At $Re = 2300$, the experiments meet Gnielinski’s correlation for $2300 < Re < 10^4$ from 1995 well. It is assumed that the flow is in transition between laminar and turbulent between $Re = 2300$ and $Re = 3000$, stable turbulent flow is reached at about $Re = 3000$.

Local Nusselt numbers are shown in Figure 5.9. It can clearly be seen that local Nusselt numbers for $Re = 2545$ don’t follow Gnielinski’s correlation for local Nusselt numbers from 1975 (valid for $2300 < Re < 10^4$), but for tests with $Re = 3000$ and higher, very good to good agreement can be seen (deviation is maximally 11%).

The front part of the pipe seems to get a worse local heat transfer behaviour with rising Reynolds numbers, which can’t be explained easily. In lower Reynolds numbers, this behaviour does not occur, so it...
might have to do with higher power in the system and a negative influence of the electrical connection clamp design.

![Figure 5.9: Validation with local Nusselt numbers in laminar and 25°C inlet temperature](image)

### 5.2.2.2 40°C inlet temperature

![Figure 5.10: Validation with average Nusselt numbers in turbulent and 40°C inlet temperature](image)
Figure 5.10 shows the system’s behaviour in 40°C inlet temperature tests. These tests could be conducted up to Reynolds number $Re = 12000$. It can be seen that with increasing Reynolds numbers, the average Nusselt numbers tend to lean towards the values predicted by Gnielinski’s correlation for $10^4 < Re < 10^6$ from 1995. Below $Re = 3000$, it can be seen, much like in the 25°C inlet test, that the Nusselt numbers lean towards Gnielinski’s prediction for $2300 < Re < 10^4$, which approaches the laminar predictions at $Re = 2300$ continuously, so without a jump.

Figure 5.11: Validation with local Nusselt numbers in laminar and 40°C inlet temperature

In Figure 5.11, the local Nusselt numbers for the above mentioned tests are displayed. In contrast to the 25°C inlet tests in Figure 5.9, the present values don’t follow the correlation, not even in lower Reynolds numbers. While the experimental values fit Gnielinski’s correlation from 1975 well in the test section inlet, the trend with rising $X^*$-values is to exceed this correlation significantly (up to 21% in $Re = 7936$).

Due to the partly high deviations from the correlations, the results from this test rig should not be compared to data from other test rigs, but rather only to data also recorded on this test rig.
5.3 Experiments on the first generation test stand

5.3.1 KTH-CeO$_2$-MS06

Tests have been conducted with the nanofluid KTH-CeO2-MS06 at 25°C and 40°C test section inlet temperature in laminar and turbulent flow. See Table 5.2 for the nanofluid’s properties, supplied by [36]:

<table>
<thead>
<tr>
<th>Nanofluid ID</th>
<th>KTH-CeO2-MS06</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base fluid</td>
<td>Water</td>
</tr>
<tr>
<td>Particles</td>
<td>CeO$_2$</td>
</tr>
<tr>
<td>Particle concentration</td>
<td>9 wt-% / 1.34 vol-%</td>
</tr>
<tr>
<td>Particle density</td>
<td>7250 kg/m$^3$</td>
</tr>
<tr>
<td>Particle specific heat capacity</td>
<td>765 J/kg K</td>
</tr>
<tr>
<td>Primary particle size</td>
<td>10 nm</td>
</tr>
<tr>
<td>Average aggregate size</td>
<td>50 nm</td>
</tr>
<tr>
<td>pH</td>
<td>6</td>
</tr>
<tr>
<td>Viscosity ratio $\frac{\mu_{nf}}{\mu_{bf}}$</td>
<td>1.220 at 20°C</td>
</tr>
<tr>
<td>Thermal conductivity ratio $\frac{k_{nf}}{k_{bf}}$</td>
<td>1.024 at 20°C</td>
</tr>
<tr>
<td>Density ratio $\frac{\rho_{nf}}{\rho_{bf}}$</td>
<td>1.090 at 25°C</td>
</tr>
<tr>
<td>Heat capacity ratio $\frac{c_{nf}}{c_{bf}}$</td>
<td>0.928 at 25°C</td>
</tr>
<tr>
<td>Prandtl number ratio $\frac{Pr_{nf}}{Pr_{bf}}$</td>
<td>1.106 at 25°C</td>
</tr>
</tbody>
</table>
5.3.1.1 Laminar experiments

25°C inlet temperature:

Experiments with KTH-CeO2-MS06 show no enhancement in heat transfer in Re=500, but slightly higher Nusselt numbers with increasing Reynolds numbers, as can be seen in Figure 5.12. The largest enhancement of Nusselt number is 5% at Re = 1600 (determined by interpolation of the water value). The experiments with nanofluid show the same characteristic bend around Re = 1500 as the experiments with water. Two tests with the nanofluid show almost no deviation (less than 1%) from the VDI Heat Atlas correlation, which is calculated using Maxwell’s equation for thermal conductivity. All other tests show a significantly smaller Nusselt number, up to 12% lower than the correlation.

![Figure 5.12](image-url) Average Nu, laminar, 25°C inlet, 1st rig generation

Looking at the local Nusselt numbers in Figure 5.13 can give detailed information about the local sources of the deviations from the VDI Heat Atlas correlation. It can be seen that in lower Reynolds numbers, all values except for the last five are lower than the correlation’s values, the local Reynolds number is up to 29% lower than calculated with the correlation (at the inlet). The tests with higher Reynolds numbers also show lower values for most positions on the test section, but the higher the Reynolds number is, the higher the local Nusselt numbers are (close to the inlet). In those tests, which showed average Nusselt numbers close to the correlation’s value, the local Nusselt numbers are higher than the correlation in the front half of the test section and lower in the rear half. This adds up to a good agreement for the average Nusselt number.
Figure 5.13: Local Nusselt numbers of water-CeO$_2$-nanofluid in 25°C inlet, laminar flow

Figure 5.14: Heat transfer coefficient and pressure drop, laminar, 25°C inlet, water-CeO$_2$-nanofluid

Figure 5.14 shows that the same or lower heat transfer coefficient can be achieved with nanofluid when the same pressure drop occurs as with use of water. In case of the value pair at 0.025 bar pressure drop, water shows about 8% better heat transfer. The water value is the first of the three values above the bend (see Figure 5.12). The nanofluid value is below the bend, it shows a lower heat transfer coefficient.
Figure 5.15 shows that the tested nanofluid has a slightly better heat transfer performance (up to 3%) when same volume flow rates are compared and pressure drop is disregarded. The same value pair as in Figure 5.14 is exceptional, here water show a heat transfer coefficient, which is 8% higher than the nanofluid's.

The ratio of \( \frac{Nu}{f} \) is illustrated at different Reynolds numbers in Figure 5.16. It shows that with rising Reynolds number, the Nusselt number of the nanofluid increases stronger than the friction factor than in case of water. In \( Re = 1765 \) (highest Reynolds number for nanofluid), the nanofluid’s coefficient of \( \frac{Nu}{f} \) is 15% higher than the interpolated value for water at this Reynolds number.
In Figure 5.17, the Number of Transfer Units is considered to assess the nanofluid's heat transfer behaviour in the tubular heat exchanger. It can be seen that water gives a better performance in the tubular heat exchanger at all tested Reynolds numbers.

**40°C inlet temperature:**

**Figure 5.18: Average Nusselt numbers in laminar flow, 25°C inlet, water-CeO\textsubscript{2}-nanofluid**

Figure 5.18 shows that in Re = 1300 and Re = 1800, the Nusselt numbers which could be achieved with the nanofluid were higher than those in tests with water and higher than those given by the VDI Heat Atlas correlation (where Maxwell’s equation has been used to calculate thermal conductivity). The two
tests with higher Reynolds numbers even exceed the correlation’s values by up to 22%, yet similar behaviour had been seen for water (increase up to 15%, as can be seen in Figure 5.18 and chapter 5.2.1.

In Figure 5.19, it can be seen how the heat transfer behaviour of the test section deviates from the VDI Heat Atlas correlation. Local Nusselt numbers for the experiments with Re = 429 and Re = 864 are on or slightly under the line representing the correlation (mostly less deviation than 10%), which results in average Nusselt numbers being close to the correlation (see Figure 5.18). The average values for the two tests with higher Reynolds numbers are more than 10% higher than the value given by the correlation. In Figure 5.19 it can be seen that this deviation is mostly caused by higher local heat transfer in the first half of the test section, up to 108% for the first value in Re = 1788.

Figure 5.19: Local Nusselt numbers of water-CeO\textsubscript{2}-nanofluid in 40°C inlet laminar flow
Figure 5.20: Heat transfer coefficient and pressure drop, laminar, 40°C inlet, water-CeO$_2$-nanofluid

The rate of heat transfer coefficient to pressure drop using the water-CeO$_2$-nanofluid is similar or, in case of the third test, slightly higher (12% increment against the interpolated value for water) than the rate which can be achieved with water in the system, as can be seen in Figure 5.20.

Figure 5.21: Heat transfer coefficient vs. volume flow, laminar, 40°C inlet, water-CeO$_2$-nanofluid

When pressure drop is not considered, the nanofluid shows a higher heat transfer coefficient than water in all tests, as Figure 5.21 shows. Nanofluid reaches an up to 9% higher heat transfer coefficient than water in a volume flow of 160 ml/min.
Figure 5.22: Nusselt per friction factor of water-CeO$_2$-nanofluid, 40°C inlet, laminar

Figure 5.22 points out that the nanofluid shows a better Nu/f ratio in Re = 1300 (24% increment), in all other tests the rate is equal or smaller than the rate achieved with water.

Figure 5.23: Heat transfer behaviour of water-CeO$_2$-nanofluid in tubular heat exchanger

The nanofluid showed the same or up to 14% less NTU in the tubular heat exchanger compared to the interpolated values of the base fluid, as can be seen in Figure 5.23.

For both base fluid and nanofluid one data point less than in the other diagrams is available. For the tests with the lowest Reynolds number, the LMTD could not be calculated, because the thermocouples had measured a lower temperature at the nanofluid outlet of the tubular heat exchanger than for the cooling.
fluid inlet. This cannot be correct, so an error must have been present, which was corrected in the first test rig generation. In the third test rig generation, all thermocouples were calibrated and this error should not occur any more. This has not been tested until the end of the thesis.

5.3.1.2 Turbulent experiments

25°C inlet temperature:

![Figure 5.24: Average Nusselt numbers in laminar flow, 25°C inlet, water-CeO$_2$-nanofluid](image)

**Figure 5.24** shows generally slightly higher Nusselt numbers for water-CeO$_2$-nanofluid, up to 6% higher in $Re = 7300$. The nanofluid experiments also slightly exceed the values predicted with Gnielinski’s correlation from 1975 (by up to 5%), calculated with thermal conductivity given by Maxwell’s equation.
Figure 5.25 shows the local reasons for the average deviations. The Nusselt numbers from experimental data mostly show a good agreement with Gnielinski’s correlation from 1975 (valid for $2300 < Re < 10^6$) in the inlet area, but local Nusselt number increasingly exceed the correlation’s values towards the outlet, up to 11%. While surpassing Gnielinski’s equation from 1975, the local Nusselt numbers lean towards Gnielinski’s equation from 1995, which is only valid for $10^4 < Re < 10^6$, when the flow is fully turbulent. A possible reason is that the flow regime approaches fully turbulent flow towards the end of the test section.

Figure 5.26: Heat transfer coefficient and pressure drop, laminar, 25°C inlet, water-CeO$_2$-nanofluid
Figure 5.26 shows a generally better ratio of heat transfer coefficient to pressure drop through the test section for the base fluid, water.

Figure 5.27: Heat transfer coefficient vs. volume flow, turbulent, 25°C inlet, water-CeO$_2$-nanofluid

In Figure 5.27, water shows a better ratio of heat transfer vs. volume flow (11% higher) at one Reynolds number, Re = 2430. Here it can be assumed that due to differences in viscosity, the nanofluid is still in laminar flow while the base fluid already entered the transition region to turbulent flow. In all other cases the difference between water and nanofluid is less than 1%.

Figure 5.28: Nusselt per friction factor of water-CeO$_2$-nanofluid, 25°C inlet, turbulent

Figure 5.28: Nusselt per friction factor of water-CeO$_2$-nanofluid, 25°C inlet, turbulent
Comparing $\text{Nu}/f$ vs. Reynolds numbers, the nanofluid seems to perform better in all experiments. The nanofluid exceeds the base fluid’s $\text{Nu}/f$ ratio up to 19% at $\text{Re} = 2660$. It can be seen that the results are similar to what is shown in Figure 5.24, scaled by a constant factor.

For most tests, Figure 5.29 shows better heat transfer behaviour in the tubular heat exchanger for water. The rise of both values in the middle of the diagram is caused by change from laminar to turbulent flow in the heat exchanger. The transition to turbulent flow seems to happen at lower Reynolds numbers for the nanofluid, so there is a limited range of Reynolds numbers where the nanofluid shows higher NTUs than the base fluid, an increment up to 10% in $\text{Re} = 2000$ (comparison to interpolated water value).
40°C inlet temperature:

![Average Nu, turbulent, 40°C inlet](image)

**Figure 5.30: Average Nusselt numbers in turbulent flow, 40°C inlet, water-CeO$_2$-nanofluid**

In 40°C inlet temperature, the nanofluid shows Nusselt numbers up to 14% higher than those of water (in case of Re = 10570 test, determined by linear interpolation of water values), as can be seen in Figure 5.30. This is a larger enhancement in Nusselt number than it was measured in 25°C inlet temperature, so a temperature dependence of the heat transfer behaviour of the nanofluid can be suspected. The nanofluid also exceeds Gnielinski’s prediction from 1995 (calculated with thermal conductivity from Maxwell’s equation) for $10^4 < \text{Re} < 10^6$, while water in all values reached values between Gnielinski’s equations from 1995 and 1975. This might point to another effect than just classical mixing of particles and fluid involved in the heat transfer of this nanofluid.
Figure 5.31: Local Nusselt numbers of water-CeO$_2$-nanofluid in 40°C inlet, turbulent flow

Figure 5.31 shows the local Nusselt numbers of the nanofluid. The same tendency as in 25°C inlet temperature can be seen: The slope of the Nusselt numbers from experiments is steeper than the correlation's slope. Other than in the 25°C inlet temperature tests, the Nusselt numbers from the experiments exceed even Gnielinski's correlation from 1995 (by up to 19% for the third-last value in Re = 10570). An explanation is not available, but the same had been seen in tests with water (compare chapter 5.2.2.2, so this indicates a peculiarity of the test rig rather than an unusual behaviour of the nanofluid.

Figure 5.32: Heat transfer coefficient and pressure drop, turbulent, 40°C inlet, water-CeO$_2$-nanofluid
Figure 5.32 states a slightly better ratio of heat transfer coefficient to pressure drop for water, although the differences are not bigger than 6%.

Figure 5.33 also shows no big differences in heat transfer coefficient between base fluid and nanofluid, even when pressure drop is not considered here. Only in one volume flow rate, 1220 ml/min, the nanofluid has a heat transfer coefficient which is 1% higher than the base fluid’s.
Figure 5.34 shows more difference between base fluid and nanofluid than Figure 5.32 and Figure 5.33. The Nu/f ratio is generally higher for the nanofluid, up to 14% in case of the nanofluid test at Re = 10570 (compared to the interpolated value for water).

Figure 5.35: Heat transfer behaviour of water-CeO$_2$-nanofluid in tubular heat exchanger

In the tubular heat exchanger, higher NTU values are reached using water than in nanofluid use. The curve shows a bend in the transition area between laminar and turbulent flow like it was seen in 25°C inlet temperature (compare Figure 5.29), yet here the transition from laminar to turbulent flow seems to happen at the same Reynolds number range for both fluids.
5.3.2 CeO$_2$-KTH_MS018AB

With CeO$_2$-KTH_MS018AB, a nanofluid based on a 50/50 vol-% mixture of water and Clariant Antifrogen N (base fluid ID: KTH_AFN-BF001), tests have been conducted at 25°C inlet temperature for laminar flow and 40°C and 55°C inlet temperatures for turbulent flow. No turbulent tests were conducted at 25°C inlet temperature, because it was feared to overload the pump with pumping fluid with such a high viscosity at high speeds. Laminar tests at 40°C inlet temperature could not be conducted due to administrative problems.

The nanofluid properties, as supplied by [36], are listed in Table 5.3:

<table>
<thead>
<tr>
<th>Table 5.3: CeO$_2$-KTH_MS018AB properties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nanofluid ID</td>
</tr>
<tr>
<td>Base fluid</td>
</tr>
<tr>
<td></td>
</tr>
<tr>
<td>Particles</td>
</tr>
<tr>
<td>Particle concentration</td>
</tr>
<tr>
<td>Particle density</td>
</tr>
<tr>
<td>Particle specific heat capacity</td>
</tr>
<tr>
<td>Primary particle size</td>
</tr>
<tr>
<td>Average aggregate size</td>
</tr>
<tr>
<td>Dynamic viscosity ratio $\mu_{nf}$/$\mu_{bf}$</td>
</tr>
<tr>
<td>Thermal conductivity ratio $k_{nf}$/$k_{bf}$</td>
</tr>
<tr>
<td>Density ratio $\rho_{nf}$/$\rho_{bf}$</td>
</tr>
<tr>
<td>Heat capacity ratio $c_{nf}$/$c_{bf}$</td>
</tr>
<tr>
<td>Prandtl number ratio $Pr_{nf}$/$Pr_{bf}$</td>
</tr>
</tbody>
</table>

Comparability of tests at the same Reynolds number is limited, because tests were conducted at the same flow rates for the base fluid and the nanofluid. Due to the high difference in dynamic viscosity between base fluid and nanofluid (see Table 5.3), each pair of tests resulted in Reynolds numbers which were about 1.5 times higher for the base fluid than for the nanofluid.
### 5.3.2.1 Laminar experiments

**Figure 5.36: Average Nusselt numbers in laminar flow, 25°C inlet, AFN-CeO\(_2\)-nanofluid**

Figure 5.36 shows same or higher Nusselt numbers for nanofluid than for the corresponding base fluid, up to 32% enhancement in Nusselt number is reached in \(Re = 1600\). Both nanofluid and base fluid show the same bend as it was seen with water (compare chapter 5.2.1.1). The Nusselt numbers determined from base fluid experiments are up to 15% lower than the VDI Heat Atlas correlation’s values, in case of the nanofluid the deviation is up to 17%.

**Figure 5.37: Local Nusselt numbers of AFN-water base fluid in 25°C inlet, laminar flow**

```latex
\begin{figure}
\centering
\includegraphics[width=\textwidth]{figure5_36.png}
\caption{Average Nusselt numbers in laminar, 25°C inlet, 1st rig generation}
\end{figure}

\begin{figure}
\centering
\includegraphics[width=\textwidth]{figure5_37.png}
\caption{Local Nusselt numbers of AFN-water base fluid in 25°C inlet, laminar flow}
\end{figure}
```
Figure 5.37 shows the local Nusselt numbers of the base fluid. It can be seen that the test in \(Re = 2266\) already shows transition to turbulent flow in the rear half of the test section. The average value’s deviation, as can be seen in Figure 5.36, origins in the higher heat transfer in the rear part of the test section and a general deviation to higher values in the rest of the test section. The test with the average Reynolds number of \(Re = 1850\) shows higher Nusselt numbers in the front part and lower Nusselt numbers in the rear part of the test section, which evens out to a good agreement for the average Nusselt number, as Figure 5.36 shows. The two tests with lower Reynolds numbers show less than 10% deviation from the correlation in most parts of the test section, but in the inlet region, the local Nusselt numbers are up to 25% less than the correlation’s values.

Figure 5.37: Local Nusselt numbers of AFN-CeO₂-nanofluid in 25°C inlet, laminar flow

In the nanofluid test with the highest Reynolds number (\(Re = 1622\)), the local Nusselt numbers exceed the correlation by up to 107% close to the inlet, as Figure 5.38 shows. In the rear parts of the test section, local Numbers are up to 17% below the correlation’s values. The two tests with \(Re = 727\) and \(Re = 1080\) show a different behaviour: All local Nusselt numbers are lower than calculated with the correlation, mostly not more than 10%, but in the inlet area deviations increase up to 34%. The test at \(Re = 409\) again shows a different behaviour, here values are higher than mentioned by the correlation in the whole test section except for the measuring point closest to the inlet. The first value is 15% lower than the correlation; the rest is up to 27% higher.
In Figure 5.39, it can be seen that, with one exception, the base fluid has an up to 38% higher heat transfer coefficient-to-pressure loss ratio than the nanofluid, because the base fluid turns to turbulent flow at lower pressure drop due to a significantly lower viscosity.

Figure 5.40 qualitatively shows the same result as Figure 5.39, although pressure drop is not considered here. The base fluid exceeds the nanofluid's heat transfer coefficient in most cases, up to 38% in a volume flow rate of 1280 ml/min, again the base fluid shows turbulent flow already at lower volume flow rates, which can also be seen from comparison with Figure 5.36.
Figure 5.41 shows higher Nusselt-to-friction factor ratio for the nanofluid for Reynolds numbers of \( Re = 1350 \) and \( Re = 1622 \). The increase against the base fluid is up to 49% at \( Re = 1622 \). Again the same reason applies for this effect as discussed in Figure 5.40.

As Figure 5.42 shows, the performance of the heat exchanger is mostly higher using the base fluid, only in \( Re = 590 \) and \( Re = 730 \), the heat exchanger performs up to 8% better using nanofluid.
The nanofluid shows considerably better heat transfer performance only in two aspects: Comparing Nusselt vs. Reynolds numbers, the nanofluid shows higher values in a small range, and looking at the Nusselt-to-friction factor ratio, the nanofluid shows a much better value, but also only in a small range. Generally, the nanofluid shows advantages when similar flow regimes are compared (via same Reynolds numbers, see Figure 5.36 and Figure 5.41), but to achieve the same Reynolds numbers as the base fluid, higher volume flow rates and thereby pressure drop is necessary than with the base fluid, as can be seen in Figure 5.39 and Figure 5.40.

5.3.2.2 Turbulent experiments

40°C inlet temperature:

Figure 5.43: Average Nusselt numbers in laminar flow, 40°C inlet, AFN-CeO$_2$-nanofluid

Figure 5.43 shows that Reynolds numbers are in a different range for the base fluid than for the nanofluid, mostly due to the high differences in dynamic viscosity between the fluids. The nanofluid shows a Nusselt number which is 87% higher than the base fluid’s in Re = 2440, but only 8% higher in Re = 2801 (compared to the interpolated values of the base fluid). One reason can be that it is not correct to assume a constant viscosity ratio of nanofluid and base fluid at all occurring temperatures.
Figure 5.44: Local Nusselt numbers of AFN-water base fluid in 40°C inlet, turbulent flow

The test with the lowest average Reynolds number $\text{Re} = 2378$ in Figure 5.44 shows laminar behaviour, at least for lower $x^*$-values. The other three tests with higher average Reynolds numbers show a positive offset against the correlation except for the lowest $x^*$-value, exceeding the correlation by 10-20% with stronger deviation at higher $x^*$-values.

Figure 5.45: Local Nusselt numbers of AFN-CeO$_2$-nanofluid in 40°C inlet, turbulent flow

Figure 5.45, where the local Nusselt numbers of the test with the nanofluid are plotted, shows laminar or partly laminar flow for the tests at $\text{Re} = 1619$ and $\text{Re} = 1820$. The characteristic rising slope of the
turbulent tests is achieved from $Re = 2022$ on, which supports the assumption made in the discussion of Figure 5.43, that the nanofluid develops a turbulent flow profile at lower Reynolds numbers than the base fluid. The local Nusselt numbers determined from experiments show a stronger slope than the correlation, a feature which has also been seen with the base fluid in Figure 5.44. For low $x^*$-values, the local Nusselt numbers determined from experiments are close to the correlation, for higher $x^*$-values, they exceed the correlation by up to 39%.

**Figure 5.46: Heat transfer coefficient and pressure drop, turbulent, 40°C inlet, AFN-CeO$_2$-nanofluid**

Heat transfer coefficients are plotted vs. pressure drop in Figure 5.46. The nanofluid shows between 27% and 54% lower heat transfer coefficient at a corresponding pressure drop.

**Figure 5.47: Heat transfer coefficient vs. volume flow, turbulent, 40°C inlet, AFN-CeO$_2$-nanofluid**
Figure 5.47 displays that the pump could not supply the same volume flow for two corresponding tests of base fluid and nanofluid at the same pump speed. Clearly the high difference in viscosity was the reason for this high deviation. The volume flow with base fluid is 5-6% higher than with nanofluid at the same pump speed. Much like in Figure 5.46, the nanofluid shows a between 21% and 40% lower heat transfer coefficient than the base fluid (values for the base fluid interpolated for comparison).

Figure 5.48: $\text{Nu}/f$ vs. Reynolds numbers of AFN-CeO$_2$-nanofluid, 40°C inlet, turbulent

The big differences in dynamic viscosity once again complicate the comparison of nanofluid and base fluid at the same Reynolds number. Only two pairs of values can be compared, and these two comparisons reveal a very heterogeneous situation: $\text{Nu}/f$ is 58% higher for the nanofluid at $\text{Re} = 2440$, but only 8% higher in $\text{Re} = 2800$. Like in Figure 5.43, it is assumed that this situation emerges from a lower Reynolds number range for the transition from laminar flow to turbulent flow for the nanofluid.
The Reynolds number ranges for nanofluid and base fluid are even further apart from each other in the tubular heat exchanger with its wider inner diameter. Here, only one nanofluid NTU-value can be compared to the values for the base fluid. The NTU value at $Re = 1500$ is 2% lower for the nanofluid (value for the base fluid interpolated for comparison).
55°C inlet temperature:

Only three tests each were made with nanofluid and base fluid at 55°C inlet temperature to get a quick overview about the fluids’ behaviour at a higher temperature, thus results are limited for this series of experiments.

![Turbulent, 55°C inlet, 1st rig generation](image)

**Figure 5.50: Average Nusselt numbers in laminar flow, 55°C inlet, AFN-CeO$_2$-nanofluid**

In Figure 5.50, the nanofluid shows a 18% higher Nusselt number in Re = 2500, but a 1% lower Nusselt number in Re = 3330 (value for the base fluid interpolated for comparison). The Nusselt numbers for the base fluid exceed the correlation’s values by between 12% at Re = 2500 and 33% at Re = 5000. The nanofluid’s Nusselt numbers determined from the experiments differ between agreeing with the correlation at Re = 1670 and exceeding the correlation by 20% at Re = 3330.

**Figure 5.51** shows local Nusselt numbers for the base fluid. At Re = 2483, where the average Nusselt number is relatively close to the correlation (compare Figure 5.50 and discussion), the local Nusselt number is three times as high than the correlation at the test section inlet. Here, laminar flow in the inlet may be suspected, which is indicated by the strongly decreasing local Nusselt numbers at the inlet. From $x^* = 0.003$ on, turbulent flow seems reached and a good agreement with the correlation can be seen. With rising $x^*$, the slope of the Nusselt numbers determined with tests is higher than the correlation’s slope, the correlation is exceeded by up to 50%. The tests with higher average Reynolds numbers show an offset from the correlation of approximately 25%. On top of that, the steeper slope leads to deviations up to 46% at higher $x^*$-values.
Figure 5.51: Local Nusselt numbers of AFN-water base fluid in 55°C inlet, turbulent flow

Figure 5.52: Local Nusselt numbers of AFN-CeO\textsubscript{2}-nanofluid in 55°C inlet, turbulent flow

Figure 5.52 shows local Nusselt numbers for the nanofluid. The correlation does not apply for the test with the lowest average Reynolds number Re = 1671. For the other two tests, a similar behaviour as for the base fluid in higher Reynolds numbers can be seen. At an average Reynolds number of Re = 2510, the nanofluid shows clear attributes of turbulent flow in all parts of the pipe, but the base fluid showed signs of laminar flow in Re = 2483, as can be seen in Figure 5.51. The test with Reynolds number Re = 2510 using nanofluid shows only a slightly steeper slope than the correlation, but an offset of about 30\% at lower x*-values and 40\% at higher x*-values. With increasing average Reynolds number, the deviation in
slope also gets stronger. While the second value of $Re = 3330$ agrees with the correlation, deviation grows with rising $x^*$ up to 36% in $x^* = 0.006$.

Figure 5.53: Heat transfer coefficient and pressure drop, turbulent, 55°C inlet, AFN-CeO$_2$-nanofluid

Figure 5.53 shows between 35% and 57% lower heat transfer coefficient for nanofluid at approximately the same pressure drop than for the base fluid.

Figure 5.54: Heat transfer coefficient vs. volume flow, turbulent, 55°C inlet, AFN-CeO$_2$-nanofluid

Figure 5.54 shows the same situation as Figure 5.53, now for similar flow rates. Here too, the nanofluid shows much lower heat transfer coefficients than the base fluid.
Figure 5.55: Nu/f vs. Reynolds numbers of AFN-CeO$_2$-nanofluid, 55°C inlet, turbulent

Figure 5.55 shows the same ratio of Nusselt number vs. friction factor for the two tests that can be compared.

Figure 5.56: Heat transfer behaviour of AFN-CeO$_2$-nanofluid in tubular heat exchanger

Figure 5.56 shows a 23% smaller NTU for the nanofluid in Re = 1350 in the tubular heat exchanger. In Re = 1800, the nanofluid transfers heat slightly better with a 9% higher NTU than the base fluid (value for the base fluid interpolated for comparison).
6 Conclusion and Outlook

The present Master thesis concentrates on error investigation in the forced convection heat transfer test rig rather than on convective heat transfer tests with nanofluids. This process of error investigation was successful, as far as it could be seen in the final tests with water, conducted on the third generation test rig. The most important enhancements in the test rig was the introduction of new clamps used for the electrical connection of the test section pipe. In the calculation, the most effective change was to calculate Reynolds and Prandtl numbers for the correlations locally at each spot in the test section where the pipe wall temperature is measured instead of calculating the correlations for local Nusselt numbers with average fluid properties.

In the tested nanofluids, no clear advantage of a nanofluid could be recognised. CeO$_2$-water-nanofluid showed the same or in some situations slightly better heat transfer behaviour as the base fluid, water. Only the heat transfer in the tubular heat exchanger was remarkably better with the nanofluid, the NTU was up to 47% higher, a general trend of higher NTU could be recognised. It is not obvious, why an enhancement in heat transfer was measured only in the tubular heat exchanger but not in the test section. In case of the CeO$_2$-AFN nanofluid, the situation looks heterogeneous. In some aspects, the nanofluid seems better than the base fluid, but in other aspects the base fluid performs better.

In future work, the following measures should be taken:

- Single peculiarities seen in the third test rig generation must be investigated:
  - The local Nusselt number determined from the temperature measured by the next-to-last thermocouple is out of trend in turbulent flow.
  - The reason for the unusually high heat loss in laminar flow must be found
- A static mixer should be integrated in the system after the outlet in order to be able to measure the average fluid temperature at the outlet.
- The cooling power in the system should be increased, the currently installed components do not offer sufficient cooling power for turbulent tests at Reynolds numbers of Re = 10000 and more at 25°C inlet temperature.
Bibliography


