Experimental Investigation on Cooling Performance of Additively Manufactured Channels

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Industrial supervisor: Mats Kinell
Examiner: Roland Gårdhagen
Abstract

Industrial gas turbines are in use to generate work output and energize society. During energy production, gas turbines experience high temperature and pressure levels. Materials cannot resist these extreme conditions with the available technology today, and effective cooling systems are required to maintain high work outputs without any failure. Active cooling methodologies are required for gas turbine technology since the technology available in the present stage does not allow to reach the required cooling levels only with the passive techniques. Additive Manufacturing (AM) increased the design space and made it possible to design complex and more effective cooling features, such as internal cooling channels on the gas turbine blades. However, AM is still a new and developing technology, and surface roughness is available on the additively manufactured parts. Surface roughness affects the pressure loss and heat transfer behavior for the mentioned cooling channels on the blades of the turbine parts. The effect of surface roughness on the flow behavior needs to be examined experimentally to be able to use the cooling advantage of the AM technology, effectively.

In the cooling systems, higher heat transfer rates and lower pressure losses are preferred to ensure effective cooling performance. The study aims to identify the surface roughness effect on the cooling performance of the additively manufactured mini channels. Pressure, temperature, and mass flow measurements were recorded during the experiments to calculate the non-dimensional numbers such as Darcy friction factor and Nusselt number to identify the cooling performance of the channels, on the wide range of Reynolds numbers.

For the present thesis, experimental investigations were done on the additively manufactured STAL15 superalloy mini channels, with the quasi Steady-State Heat Transfer rig in the Fluid Dynamics Lab of Siemens Energy in Finspång, Sweden. With the output of the study, the database for the additively manufactured materials in Siemens Energy was enlarged. The results will help to find accurate mathematical correlations between the surface roughness properties, pressure losses, and heat transfer behaviors and model the effects on the computer simulations.
Acknowledgements

This thesis work was performed in the Fluid Dynamics Laboratory of Siemens Energy in Finspång, Sweden. I would like to thank Siemens Energy and Linköping University for this cutting-edge research opportunity.

I would like to thank Dr. Mats Kinell, my industrial supervisor, who guided and supported me tremendously during the entire project. I learned a lot from the discussion sessions and lectures on the whiteboard in the laboratory. You kept me in a fully motivated condition in even hard times with your positive attitude and encouraged me to go further and more detailed in the project. The time we spent together in the laboratory was pure fun and instructive. I feel super lucky to have you as not only a supervisor but also as a friend.

Dr. Karl-Johan Nogenmyr, who used the data of the experiments from the thesis work to model the surface roughness on the computer simulations and worked closely with me during the thesis project, I would like to thank you for your support and comments about the results of the experiments, and the experimental methodology. The meetings we had increased my knowledge and motivations.

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It would not be possible without the Engineering Materials Laboratory of Linköping University to get the surface roughness measurements for the test objects. I would like to Johan Moverare and Per-Olof Karlsson for their positive attitude and super-fast help on the cutting of the test objects, which is a critical part of the surface roughness measurement procedure. Also, I would like to thank my colleagues from Siemens Energy, Marcel Olma, and Sebastian Richter for their help on the surface measurements.

My parents, Songül and Kazım Firat, whatever I would write here would not be enough to describe your support in every aspect during my entire life. I would like to thank you for everything.

My brother, Egemen Firat, I am thankful for the time you spent with us in the world. I am proud of you and always will love you. Rest in peace.
## Nomenclature

### Abbreviations and Acronyms

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Meaning</th>
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<tbody>
<tr>
<td>LiU</td>
<td>Linköping University</td>
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<tr>
<td>UN</td>
<td>United Nations</td>
</tr>
<tr>
<td>SDGs</td>
<td>Sustainable Development Goals</td>
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<tr>
<td>AM</td>
<td>Additive Manufacturing</td>
</tr>
<tr>
<td>CNC</td>
<td>Computer Numerical Control</td>
</tr>
<tr>
<td>qSSHT</td>
<td>quasi Steady-State Heat Transfer</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl Number</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds Number</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt Number</td>
</tr>
<tr>
<td>Gr</td>
<td>Grashof Number</td>
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<tr>
<td>SLM</td>
<td>Selective Laser Melting</td>
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### Latin Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
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<tbody>
<tr>
<td>u, V</td>
<td>Velocity</td>
<td>m.s⁻¹</td>
</tr>
<tr>
<td>T</td>
<td>Temperature</td>
<td>Celsius</td>
</tr>
<tr>
<td>c</td>
<td>Specific heat</td>
<td>J.kg⁻¹.K⁻¹</td>
</tr>
<tr>
<td>L</td>
<td>Length</td>
<td>m</td>
</tr>
<tr>
<td>P</td>
<td>Pressure</td>
<td>Pa</td>
</tr>
<tr>
<td>H</td>
<td>Head</td>
<td>-</td>
</tr>
<tr>
<td>g</td>
<td>Acceleration of gravity</td>
<td>Pa</td>
</tr>
<tr>
<td>D</td>
<td>Diameter</td>
<td>m</td>
</tr>
<tr>
<td>f</td>
<td>Friction factor</td>
<td>-</td>
</tr>
<tr>
<td>K</td>
<td>Loss coefficient</td>
<td>-</td>
</tr>
<tr>
<td>A</td>
<td>Area</td>
<td>m²</td>
</tr>
<tr>
<td>k</td>
<td>Thermal conductivity</td>
<td>W.m⁻¹.K⁻¹</td>
</tr>
<tr>
<td>h</td>
<td>Heat transfer coefficient</td>
<td>W.m⁻².K⁻¹</td>
</tr>
<tr>
<td>M</td>
<td>Mach number</td>
<td>-</td>
</tr>
<tr>
<td>R</td>
<td>Universal gas constant</td>
<td>J.K⁻¹.mol⁻¹</td>
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### Greek Symbols

<table>
<thead>
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<th>Symbol</th>
<th>Description</th>
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<tr>
<td>δ</td>
<td>Thickness</td>
<td>m</td>
</tr>
<tr>
<td>μ</td>
<td>Dynamics viscosity</td>
<td>kg.m⁻¹.s⁻¹</td>
</tr>
<tr>
<td>ρ</td>
<td>Density</td>
<td>kg.m⁻³</td>
</tr>
<tr>
<td>ε</td>
<td>Absolute roughness</td>
<td>mmm</td>
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### Subscripts and superscripts

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Meaning</th>
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<tbody>
<tr>
<td>∞</td>
<td>sign of free stream value</td>
</tr>
<tr>
<td>t</td>
<td>sign of thermal value</td>
</tr>
<tr>
<td>S</td>
<td>sign of value at the surface</td>
</tr>
<tr>
<td>p</td>
<td>sign of constant pressure</td>
</tr>
<tr>
<td>w</td>
<td>sign of value at the wall</td>
</tr>
<tr>
<td>0</td>
<td>sign of value for the smooth surfaces</td>
</tr>
<tr>
<td>L</td>
<td>length</td>
</tr>
<tr>
<td>1</td>
<td>inlet or upstream</td>
</tr>
<tr>
<td>2</td>
<td>outlet or downstream</td>
</tr>
<tr>
<td>a, b, c</td>
<td>constant value</td>
</tr>
<tr>
<td>d</td>
<td>Darcy</td>
</tr>
<tr>
<td>n</td>
<td>Normal</td>
</tr>
<tr>
<td>cs</td>
<td>cross-section</td>
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1 Introduction

1.1 Background

In 2015, United Nations (UN) set 17 Sustainable Development Goals (SDGs) to be achieved by 2030. The goals have a clear focus to provide sustainable economic growth all over the world. SDGs aim for a better future for the next generations by ending poverty in many aspects. [1]

Sufficient energy production may be a key to achieve most of the goals proposed by the UN. Generating the needed energy for the entirety of humanity, inequalities that people are suffering in the present circumstances could solve in the future. Energizing society with sustainable solutions will not affect just humankind but also all living organisms in the universe.

Gas turbines are internal combustion engines that can transform the chemical energy of the fluid fuel to kinetic energy. A gas turbine consists of three main parts, namely compressor, combustion chamber, and turbine. Compressors compress the air, and the compressed air mixes with the fuel and burns in the combustion chamber, then the fluid turns the turbines to generate the work output.

In 1903, Jens William Aegidius Elling invented the first gas turbine that could generate higher energy than the needed energy input to run the components of the turbine. This gas turbine is considered the primary gas turbine with the potential of net power production. The first gas turbine was only able to produce 11 hp net power, which is approximately equal to 8 KW. [2] With the scientific advancements, now Siemens SGT6-9000HL can provide 593 MW power output with combined-cycle efficiencies greater than 63% [3], and gas turbine technology still has the potential to be more efficient and powerful.

Modern gas turbine components have high rotational velocities and temperatures to generate a massive amount of power output. The power output of the gas turbine can improve further by increasing the turbine inlet temperature. On the other hand, there is a limitation for higher temperatures for these highly stressed components. Insufficient cooling may cause creep problems on the turbine parts. To ensure higher maintenance intervals and reliable engines, the cooling performance of the machine is critical. Thus, among the developments in material science, efficient cooling is needed to achieve higher energy outputs without any potential component failure. [4] [5]

Today, passive and active cooling methodologies are in use to protect the turbine parts from high temperatures. In the passive cooling method, layers with low thermal conductivity materials exist between the metal surface and hot gas. However, this might not be enough to ensure the desired temperatures for the turbine parts, and active cooling is needed. External and internal active cooling with liquid and air
are the possible cooling solutions available in the present stage. However, the liquid cooling systems are not practical except the spray cooling for the thrust boosting in turbojet engines since cooling liquid makes the methodology bulky to use. Thus, the internal air cooling method is one of the best active cooling available for now.

With the introduction of Additive Manufacturing (AM), the possible design space expanded compared to traditional manufacturing methods. AM allows manufacturing complex-shaped features without design constraints faced with conventional techniques. Even it is still a growing technique with shortcomings, it has a huge impact in critical sectors such as aerospace and medical with success. AM offers complex-shaped products with less weight and a reduced number of parts. Furthermore, it reduces the waste material during production compared to a conventional machining method such as computer numerical control (CNC), where the desired component is produced by removing materials from a workpiece. But in AM, materials are added to create the product. [6] Cooling characteristics for the gas turbine components can be more effective with the AM techniques. Maximum allowable turbine inlet temperature can increase to produce higher power outputs since it can produce more complex cooling features with low conductivity materials.

1.2 Literature Review

In the earlier 20th century, Johann Nikuradse [7] conducted avant-garde research with artificially roughened pipes. The sand grains cemented to the walls of the flow pipes to achieve surface roughness, and the ratio of average sand grain diameter to the internal diameter of the flow pipe was defined as relative roughness. The test objects in the experimental study had six different relative roughness levels (from 0.000986 to 0.03333). From the results of the experiments, it is observed that, for low Reynolds numbers (i.e. Laminar regime), the surface roughness levels available on his test samples do not affect the resistance factor, and are the same with smooth pipes. However, for higher Reynolds numbers (i.e. Transition and turbulent regime), it is observed that an increase in Reynolds number increases the resistance factor.

With the experimental study done in 1983, Wu and Little [8] became the first scientists who discovered the higher friction factor levels for rough surfaces even in the laminar region, compared to the smooth ones. The study showed that, even for rough pipes, the slope of the straight line for friction force and Reynolds number is the same as smooth pipes. For rough pipes, tremendously earlier critical Reynolds numbers were experienced, such as Re = 350.

In 2013, Huang et al. [9] investigated the friction factor for pipes with larger relative roughness levels than Nikuradse (up to 0.41666). With the higher relative roughness, he confirmed the different flow behavior for rough pipes than smooth ones, in the laminar flow regime. Larger friction effects were present for the rough pipes, which could be predicted with the quadratic equation of the relative roughness. Only two of his test samples with the relative roughness of 0.008421 and 0.016183 were able to follow the same line with the smooth pipes, and starting from 0.032372 relative roughness, deviations were observed. Moreover, earlier transition Reynolds numbers
were validated for rough pipes, and even earlier were present with the increased relative roughness levels.

Studies presented until now were focused on the friction effect of the surface roughness, but not heat transfer. In 2003, a published review of Gian Luca Morini [10] for the experimental studies done on heat transfer in microchannels defined the subject as a scientific open question.

Recent experimental studies were performed in The Pennsylvania State University to investigate the heat transfer and pressure losses in additively manufactured mini channels by Stimpson et al. [11] [12]. Test coupons were printed with direct metal laser sintering (DMLS) which has a high potential for producing gas turbine components with additive manufacturing technology. Cobalt-chrome-molybdenum-based superalloy (CoCr) and Inconel 718 were the print materials of the test samples. Test channels were printed together, side by side, on one single part, which may lead to problems with the heat transfer between each channel. Additionally, the mass flow rate was measured for the single part, not one by one for each channel, and then calculated for each test channel by using the pressure ratios. Results of the study showed that significantly increased pressure drops (i.e. Darcy friction factor) and heat transfer rates (i.e. Nusselt numbers) are present for additively manufactured rough flow channels. Surface roughness measurements were performed, and it is suggested that relative arithmetic mean roughness (Ra/Dh) is the best parameter to relate to the relative sand grain roughness used by Nikuradse.

1.3 Aim

As mentioned earlier in the Background section (1.1), additively manufactured materials have a tremendous potential to produce gas turbines with higher work outputs by increasing the temperature limit on the parts of the turbines with better cooling performance achieved. However, AM technology is still developing, and it also has shortcomings. In the current availability of AM, it is not possible to produce the surfaces of the produced materials as smooth as conventional methods. The mystery of the surface roughness gives no opportunity to perform accurate computational simulations to investigate fluid flow features and heat transfer properties on additively manufactured materials. Modeling the effect of the surface roughness could help computer simulations to be more accurate and user-friendly. Experimental investigation of mentioned features and properties is needed in the current phase to understand the cooling performance of additively manufactured parts.

The present thesis aims to understand the effect of surface roughness on pressure drop and heat transfer performance with the help of data acquired from the experimental analysis. The mentioned phenomenons will be related to the non-dimensional numbers, namely, the Darcy friction factor and the Nusselt number. Understanding the friction effect will ensure understanding the cooling performance characteristics also. The quasi Steady-State Heat Transfer (qSSHT) rig of Siemens Energy was used to perform the experiments. Test objects with both circular and non-circular flow channels with different dimensions were examined for different flow regimes.
2 Theory

The study examines the cooling performance of the AM materials by using aerodynamics science and steady-state forced convection theoretical knowledge. This section will introduce aerodynamics flow characteristics and features, heat transfer modes, and parameters to ensure a more measurable understanding of the studied parameters and conditions in the present thesis. Theoretical background will provide supportive and critical information regarding the applied methodology and data interpretation of the study.

2.1 Aerodynamics

2.1.1 Boundary Layer

In 1904, the boundary layer phenomenon was defined firstly by Ludwig Prandtl. The boundary layer is a thin layer close to the interaction section of the fluid flow and the solid surface, where the viscous effects are dominant over inertia forces due to the friction between the surface and fluid flow, and viscous effects are negligible outside the boundary layer. Hence the majority of the friction effects and heat transfer occurs inside the mentioned thin layer of the flow, and the present thesis focuses on the investigation of the surface roughness on the friction and heat transfer, a comprehensive understanding of the boundary layer theory is a requirement.

Velocity and thermal boundary layers develop due to the mentioned viscous effects. [14]

**Velocity Boundary Layer**

While the fluid flows over or inside an object, shear stress occurs between the flow and object. As a consequence of the mentioned friction force parallel to flow direction, the fluid particles tend to stick to the surface. The fluid velocity on the surface will be equal to zero, and the specified phenomenon can be called a no-slip boundary condition. As a result of the friction force between each adjacent velocity streamline, the velocity of the next ones also decreases. With an increase in the distance between the surface and fluid flow, the flow velocity increases until it reaches the free stream velocity.

Thus, the velocity boundary layer is described as a thin layer with a thickness of \( \delta \) between the region where the flow velocity is zero and where the flow velocity is equal to 99 percent of the free stream velocity. [13]

The schematic illustration of the velocity boundary layer is shown in Figure [1].
Thermal Boundary Layer

Similar to the velocity boundary layer, the interaction between the fluid flow and the surface causes changes in the fluid properties. In the velocity boundary layer case, the phenomenon was introduced with the velocity changes. For the current case, due to the viscous effects, there is a difference between the free-stream fluid temperature and the wall surface temperature, which causes the development of the boundary layer.

When fluid flows with a uniform temperature of $T_\infty$ over or inside the object with a wall temperature of $T_s$, the fluid particles adjacent to the object wall will be in thermal equilibrium, which is also known as no-temperature jump condition. The heat is transferred with conduction mode between the surface and the first layer of the flow. Then, the mentioned particles will transfer energy to adjacent flow layers, and the process will result in the development of the thermal boundary layer.

The length of the boundary layer depends on the distance where the equilibrium between the free stream fluid property is ensured. With a more detailed explanation, the temperature difference between the fluid flow and the wall surface equals 99 percent of the temperature difference between the free stream flow and the wall surface. Thermal boundary thickness ($\delta_t$) is the distance from the wall where the stated balance is established. \[13\]

The schematic illustration of the thermal boundary layer is shown in Figure 2.
Prandtl Number

Prandtl number is a non-dimensional parameter, which defines the relative thickness of the velocity and thermal boundary layers. The quantity is a ratio of molecular diffusivity done by momentum to heat.

\[ Pr = \frac{\text{Molecular diffusivity of momentum}}{\text{Molecular diffusivity of heat}} = \frac{\partial}{\alpha} = \frac{\mu C_p}{k} \] (1)

From the description and Equation 1, the ratio smaller than unity means that the molecular diffusivity of the heat is higher than the momentum one, and the thermal boundary layer will be thicker than the velocity boundary layer, and the reverse case is also applicable. For the third condition, the unity ratio means that the molecular diffusivity of heat and momentum are equal. Thus, both the velocity and thermal boundary layer will have the same thicknesses. [13]

2.1.2 Flow Regimes

The flow regime of the fluid is classified according to the behavior of the flow streamlines. Mainly there are two fluid regimes named laminar and turbulent, and the region where flow change characteristics from laminar to turbulent regime define the transition region.

In the laminar region, the fluid motion shows highly ordered behavior with smooth streamlines with lower velocities. As the velocity of the fluid increases, the flow does not change its regime from laminar to turbulent suddenly. Between the two regions, the transition of the boundary layer occurs, where the fluid behavior is unpredictable. Inside the transition region, the fluid may act like laminar, turbulent, or neither laminar nor turbulent. After the transition region, the flow becomes turbulent. In contrast to laminar flow, the fluid behavior could be defined as chaotic in the turbulent regime since the motion of the fluid is highly disordered and there are velocity fluctuations.

The transition from the laminar to the turbulent boundary layer is tremendously complex and dependent on numerous factors. Some of the parameters affecting the transition point are Mach number, surface temperature, pressure gradient, and surface roughness. Experimental studies performed by Alber L. Braslow show that the transition Reynolds number decreases with the increase in Mach number until the Mach number of 4. However, after the mentioned Mach number, the transition Reynolds number increases significantly. Additionally, the heated wall surface results in a reduced Reynolds number for the transition. In contrast, a cooled wall surface increases the stability of the laminar boundary layer and delays the transition point. Favorable pressure gradients result in delayed transition points, and unfavorable ones trigger the transition. Surface roughness reduces the stabilization and disturbs the laminar flow, and earlier transition points are present with higher surface roughnesses. [15]
Reynolds Number

Reynolds number is the ratio of inertia forces to viscous forces and can be used to predict the behavior of the flow.

\[
Re = \frac{\text{Inertia forces}}{\text{Viscous forces}} = \frac{VL_c}{\nu} = \frac{\rho VL_c}{\mu} \tag{2}
\]

As shown in Equation 2, the inertial forces are represented with the density and velocity of the fluid, and inertial viscous forces are proportional to the viscosity of the fluid. The fluid particles stick together and alter an ordered behavior (i.e. laminar regime) with high enough viscous forces. On the other hand, an increase in velocity results in the dominance of inertial forces over viscous ones. At some point, the viscous forces become insufficient to stick the flow particles together, and fluid changes behavior. The numerical quantity where the flow changes its regime from laminar to turbulent could be defined as a critical Reynolds number.

2.1.3 Pressure Losses and Darcy Friction Factor

The conservation of the energy law suggested by Émilie du Châtelet, states that the energy inside the universe can not be destroyed but can change from one to another. During the fluid flow, the fluid transforms its mechanical energy to thermal energy. While the fluid flows over or inside the object, the fluid properties might not stay the same through the flow path. Head losses are the transformation of the available mechanical to unavailable heat energy. Equation 3 describes the head losses on the fluid flow.

\[
H_L = \frac{P_1}{\rho_w} - \frac{P_2}{\rho_w} + \frac{V_1^2 - V_2^2}{2g} \tag{3}
\]

When assumed that the cross-sectional area for the fluid flow is constant over the fluid domain, there will be no significant velocity variation, and it is possible to say that the head losses are mainly proportional to the pressure difference.

On the other hand, features such as sudden contraction, expansion may disturb the fluid flow (i.e. induced turbulence), which causes the difference in the fluid velocity. Thus, for this case, the total head loss should include both the pressure loss caused by the frictional effect on the domain and the induced turbulence. 

Surface Friction Loss

Friction force occurs between two bodies due to the relative motion of the bodies. The friction force causes losses since the friction force is in the opposite direction with the direction of movement and the net force is equal to applied force minus friction force. To overcome the mentioned friction and ensure the movement of the fluid, an external agent (i.e. pressure, temperature difference) is needed. The relationships given in this section are only applicable for the incompressible flows.
In the 17th century, experimental studies were done by Gotthilf Heinrich Ludwig Hagen and Jean Léonard Marie Poiseuille individually, and both scientists have discovered the same relationship for the pressure losses in the laminar region. Hagen-Poiseuille law is given in Equation 4 \[16\]

\[ H_L = \frac{32\mu LV}{\rho_w D^2} \quad (4) \]

In the 19th century, Henry Philibert Gaspard Darcy suggested the given formula in Equation 5 to relate the pressure losses in the turbulent region for smooth pipes.

\[ H_L = (a \frac{L}{D} + b \frac{L}{D^3}) V + (b \frac{L}{D} + c \frac{L}{D^3}) V^2 \quad (5) \]

The problem with the formula was it was not dimensionally homogenous (i.e. dimensions of the variables on each side of the equation are not the same). In 1845, Julius Ludwig Weisbach published a dimensionally homogenous formula, as shown in Equation 6 which relates the friction effects on the surface with the pressure drop.

\[ H_L = fL \frac{V^2}{D} \quad (6) \]

The Darcy-Weisbach formula given in Equation 7 is still in use to relate the pressure drop inside the pipe flow with the Darcy friction factor on both laminar and turbulent flow regimes for both smooth and rough pipes. As input, the pressure difference between the inlet and outlet is required. Thus, to use the formula, an experimental investigation is needed.

\[ f_d = 2 \frac{D}{L} \frac{\Delta P}{\rho_{mean} V_{mean}^2} \quad (7) \]

On the other hand, it is possible to predict the Darcy friction factor by using the theoretical relationships acquired from the previous experimental studies without knowing the pressure drop along the channel. \[17\]

For the laminar flows, by grouping the Reynolds number terms from the Hagen-Poiseuille law (see Equation 4) and other remaining terms from the Weisbach formula given in Equation 6, Equation 8 can be obtained. In the laminar region, the friction factor is dependent on Reynolds number, and surface roughness does not affect the friction factor. The formula is only applicable for the circular channels. For the non-circular ones, the constant on the numerator should change depending on the aspect ratio of the channel. \[13\] \[16\]

\[ f_d = \frac{64}{Re} \quad (8) \]
However, until the 20th century, an understanding of the turbulent flow regimes could not be achieved. Johann Nikuradse, a Ph.D. student of Prandtl, performed experiments with artificially roughened pipes, and with help of the results, suggested Equation 9 to predict the Darcy friction factor on the turbulent flow regimes.

\[
\frac{1}{\sqrt{f_d}} = -2\log_{10} \frac{2.51}{Re^{\frac{1}{2}}} \tag{9}
\]

Theodore von Kármán used data from the experimental work of Nikuradse and obtained the formula (see Equation 10), which does not contain the Reynolds number but predicts the Darcy friction factor by using the relative roughness of the channel. Equation 10 is applicable for the fully turbulent flows only.

\[
\frac{1}{\sqrt{f_d}} = -2\log_{10} \frac{\varepsilon}{3.7D} \tag{10}
\]

In 1939, Colebrook-White Equation given in Equation 11 was published, which combines the studies of Johann Nikuradse and Theodore von Kármán.

\[
\frac{1}{\sqrt{f_d}} = -2\log_{10} \left( \frac{\varepsilon}{3.7D} + \frac{2.51}{Re^{\frac{1}{2}}} \right) \tag{11}
\]

The Colebrook-White Equation predicts the Darcy Friction Factor for different Reynolds numbers and different surface roughness levels. 

In 1944, Lewis Ferry Moody generated the Moody Diagram shown in Figure 3 by using formulas for the laminar (see Equation 8) and turbulent (see Equation 11) flow regimes. The diagram represents the Darcy Friction Factor for channels with different relative roughness levels over various Reynolds numbers.

\[
Re = \frac{\rho Vd}{\mu}
\]

Figure 3: Moody Diagram.
**Induced Turbulence Loss**

Due to the experienced sudden cross-sectional changes on the flow path, the flow behavior changes and causes induced turbulence. The mentioned phenomenon can be in any part of the flow path. But for the pipe flow examined in the present thesis, the cross-sectional area along the flow path inside the channel is constant. However, there is a sudden contraction in the channel inlet, and there is a flow expansion after the test channel.

As described in the head losses, the energy is conserved, but it can change its form. Due to the friction forces on the pipe wall, the mechanical energy transforms to heat energy. With the effect of the induced turbulence, both energy transformation and head losses increase. Thus, together with the friction losses, the induced turbulence losses should be considered to model all head losses experienced during the fluid flow.

When assumed that the density of the fluid on the inlet and outlet of the pipe is the same, the head loss Equation given in [3] could be rearranged and written as in Equation [12]

\[
\frac{P_1 - P_2}{\rho_w} = H_L - \frac{V_1^2 - V_2^2}{2g} \quad (12)
\]

By using the relationship between the cross-sectional area and the fluid velocity given in Equation [13] [20], the head loss equation becomes as Equation [14]

\[
V_1 A_1 = V_2 A_2 \quad (13)
\]

\[
\frac{P_1 - P_2}{\rho_w} = H_L - \frac{V_1^2}{2g} \left[ \left( 1 - \frac{A_1}{A_2} \right)^2 \right] \quad (14)
\]

Borda-Carnot equation given in [15] predicts the head losses due to the induced turbulence.

\[
H_L = K \frac{V^2}{2g} \quad (15)
\]

\[
K = \left( 1 - \frac{A_1}{A_2} \right)^2
\]

The \( H_L \) term in Equation [14] could be replaced with the given definition of \( H_L \) in Equation [15] to obtain Equation [16] and [17] for the inlet and outlet losses respectively.

\[
\frac{P_1 - P_2}{\rho_w} = \frac{V_1^2 \rho_w}{2g} \left[ K_1 - 1 + \left( \frac{A_1}{A_2} \right)^2 \right] \quad (16)
\]

\[
\frac{P_1 - P_2}{\rho_w} = \frac{V_2^2 \rho_w}{2g} \left[ K_2 - \left( \frac{A_2}{A_1} \right)^2 + 1 \right] \quad (17)
\]
Equation 16 and 17 could be rewritten as Equation 19 and 20 by using the following relationship between the mass flow rate and velocity (see Equation 18).

\[ \dot{m} = \rho_w VA \]  

(18)

\[ V^2 = \frac{\dot{m}^2}{A^2 \rho_w^2} \]

\[ \frac{P_1 - P_2}{\rho_w} = \frac{\dot{m}^2}{2gA_1^2 \rho_w} \left[ K_1 - 1 + \left( \frac{A_1}{A_2} \right)^2 \right] \]  

(19)

\[ \frac{P_1 - P_2}{\rho_w} = \frac{\dot{m}^2}{2gA_2^2 \rho_w} \left[ K_2 - \left( \frac{A_2}{A_1} \right)^2 + 1 \right] \]  

(20)

And when assuming that the inlet and outlet area of the pipe is the same \( (A_1 = A_2) \), the induced turbulence loss equation becomes as in [21] [16]

\[ P_1 - P_2 = K \frac{\dot{m}^2}{2gA^2 \rho_w} \]  

(21)

The loss coefficient \( K \) in Equation 21 should be selected for inlet and outlet separately from the tables given in Idelchik’s research paper [21] according to Reynolds numbers and cross-sectional area ratios.

2.2 Heat Transfer

Heat transfer is due to the temperature difference, and the heat flows from the higher temperature point to the lower one.

Transfer of the heat occurs with three different modes, namely, conduction, convection, and radiation.

2.2.1 Modes

Conduction

As mentioned previously, the heat transfer appears due to the difference in the temperatures and in the conduction mode, the energy transfers from the higher energetic to lower energetic particles. Heat transfer with the conduction mode occurs with the atomic and molecular activities (i.e. Lattice vibration and electron motion for the solids and transitional, vibrational, rotational, and electrons motions for gases and liquids).
As a result of the experiments performed by Joseph Fourier, Fourier Law presents conduction heat transfer mode mathematically as shown in Equation \[22\] \[23\]

\[
Q = -kA_n \frac{T_2 - T_1}{L}
\]  

(22)

The given conduction mode law is for steady-state, one-dimensional, and constant thermal conductivity cases only. The outcomes of the experiments done by Fourier showed that the thermal conductivity \(k\) changes for different materials. Thus, transferred heat is proportional to the thermal conductivity of the material, the temperature difference, and the thickness of the material. For the conduction mode, a material medium is a must.

**Convection**

The convection heat transfer mechanism requires a medium like the conduction mode. But it also needs a motion of the molecules. The fluid motion continues until the equilibrium. Depends on the source of the fluid movement, the mechanism defines as natural, forced, or mixed convection. In the natural convection case, the motion occurs naturally due to the density differences. Differently, an external factor (i.e. fan, pressurized air) creates forced convection. Furthermore, mixed convection is the combination of natural and forced ones.

Newton’s law of cooling equation given in Equation \[23\] represents the rate of convection heat transfer on a body, mathematically. The constant \(h\) stands for convection heat transfer coefficient, and it depends on the material. \[13\]

\[
\dot{Q}_{\text{conv}} = hA_s(T_s - T_{\infty})
\]  

(23)

Additional theory about the convection heat transfer mechanism will be given in Section \[2.2.2\]

**Radiation**

An object which has a higher temperature than absolute zero (i.e. zero Kelvin) emits radiation. Due to the temperature, molecular activity rises, and it results in radiation heat transfer. Thus, with an increase in the temperature, heat transfer with radiation mode magnifies. Different from conduction and convection mode, radiation mode does not require any medium to transfer the heat.

The net rate of radiation heat transfer done between a surface and a much larger surface is represented by Equation \[24\] by the Stefan-Boltzmann law of radiation, mathematically.
\[ \dot{Q}_{\text{rad}} = \varepsilon \sigma (T_s^4 - T_{\text{surr}}^4) \]  

(24)

The black body is an ideal surface that emits radiation at the maximum possible rate. The emissivity \( \varepsilon \) of the black body is 1, and for the other bodies, emissivity values change between 0 and 1. [24]

### 2.2.2 Nusselt Number

As mentioned at the beginning of the Theory Section, thermal performance investigations use the steady-state forced convection theoretical background. The data acquired from the experiments are input to calculate the non-dimensional numbers to understand the cooling performance.

Nusselt number (Nu) is a non-dimensional heat transfer coefficient, and it is the ratio of convective to conductive heat transfer as given in Equation [25].

\[ Nu = \frac{hL}{k} \]  

(25)

From the formulation, the Nusselt number represents the increment in the heat transfer due to the convection mode. Nusselt number of 1 indicates that the heat transferred only with the conduction mechanism. But Nusselt number of 3 means that the motion of the fluid caused a three times increment on the heat transfer.

The given description of the Nusselt number shows the importance of the parameter on the investigation of the heat transfer performance of the materials.

As previously mentioned, in the forced convection case, the fluid movement is caused by an external agent. Thus, there will be significant fluid velocity, and the Nusselt number is a function of the Reynolds and Prandtl number.

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

(26)
Nusselt number Correlations for Internal Flow

For the fully developed laminar flow inside both circular and non-circular channels, Nusselt numbers are given in Table 5 for both constant heat flux and constant surface temperature boundary conditions.

Table 5: Nusselt numbers for fully developed laminar flow. [24]

<table>
<thead>
<tr>
<th>Shape</th>
<th>a/b</th>
<th>Constant heat flux</th>
<th>Constant wall temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D$</td>
<td></td>
<td>4.36</td>
<td>3.66</td>
</tr>
<tr>
<td></td>
<td>1</td>
<td>3.61</td>
<td>2.98</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>4.12</td>
<td>3.39</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>5.33</td>
<td>4.44</td>
</tr>
<tr>
<td>$\infty$</td>
<td>8</td>
<td>6.49</td>
<td>5.60</td>
</tr>
<tr>
<td>$\sqrt{3}a$</td>
<td>8</td>
<td>8.24</td>
<td>7.54</td>
</tr>
</tbody>
</table>

For the turbulent flow case, the boundary condition has no significant effect compared to the laminar region. Thus, the following correlations are applicable for both constant heat flux and constant surface temperature setups.
Dittus-Boelter correlation

\[ Nu_D = 0.023Re_D^{0.8}Pr^n \] (27)

The given correlation in Equation 27 is to predict the Nusselt number uses the mean temperature for the calculation of the properties. Thus, for the higher temperature variations, the accuracy of this correlation is low. The constant \( n \) is 0.4 for the heating fluid and 0.3 for the cooling one. The applicable Prandtl number range for the Dittus and Boelter correlation is from 0.6 to 100, and the correlation is valid for Reynolds numbers higher than 10000.

Sieder-Tate correlation

\[ Nu_D = 0.027Re_D^{0.8}Pr^{1/3} \left( \frac{\mu}{\mu_s} \right)^{0.14} \] (28)

The properties except for the \( \mu_s \) on Equation 28 to predict the Nusselt number by using the Sieder-Tate correlation are at the mean temperature. And \( \mu_s \) is at the surface temperature. Consequently, Sieder and Tate correlation is more reliable than Dittus-Boelter if the temperature difference between the fluid and surface is significant. The applicable Prandtl number range for the Dittus and Boelter correlation is from 0.7 to 16700, and similar to the Dittus-Boelter correlation, the correlation is valid for Reynolds numbers higher than 10000.

Gnielinski correlation

\[ Nu_D = \frac{(f/8)(Re_D - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \left[ 1 + \left( \frac{D}{L} \right)^{2/3} \right] \left( \frac{Pr}{Pr_s} \right)^{0.11} \] (29)

Gnielinski correlation offers more accurate than the previously mentioned ones. Rather than the \( Pr_s \), all properties given in Equation 29 are at mean temperature. But, \( Pr_s \) is at the temperature of the surface. Different than the previous two correlations, the friction factor is also taken into account. Moreover, the Prandtl number range is tremendously enhanced (0.5 to 2000). Gnielinski correlation is valid from the Reynolds numbers of 2300 to \( 10^6 \), and friction factor in the formula could be found on the Moody diagram, or calculated from the Colebrook-White Equation given in [11] [24] [25]
Entrance Effects

As previously mentioned, the flow behavior changes due to the sudden contraction and expansion in the inlet and outlet of the pipe, respectively. The phenomenon causes additional pressure drops and changes in the heat transfer performance. Pressure losses come from the induced turbulence losses already discussed in the Darcy friction section.

Now, the subsection will introduce the entrance effect on the heat transfer case. Mills conducted experimental studies to investigate the turbulent heat transfer in the entrance region of the circular conduit. The study shows that the Nusselt number has a higher value in the entrance than the other regions. If the flow properties are constant, a constant value of the Nusselt number is present after the entrance region. From the introduced entrance effects, it is possible to say that, for the channels with longer lengths, the entrance effects are less dominant when compared with the shorter ones.

Mills suggests three reasons which cause the entrance effects for the turbulent flows. In the end, three suggestions come to the same conclusion. There is different behavior in the boundary layer due to the circulations and separation in the entrance region, which causes unusual turbulence. The boundary layer is not fully developed, and there is a discontinuity in the heating condition. Thus, the shape and dimensions of the entrance can have a huge influence on flow behavior. Moreover, the Reynolds number of the flow affects the entrance effects. Mills investigated different inlet conditions such as bellmouth, sharp edge, T-piece, elbow, round bend, orifice plate type, and long/short calming sections in the Reynolds number range of 10 000 to 110 000. For the present thesis, the inlet condition is a sharp edge, 90 degrees.

Equation [30] models the entrance effect for the Nusselt number calculations. $\frac{N_u}{N_{u,\infty}}$ is the Gneilinski correlation Nusselt number and $N_u$ is the increased Nusselt number due to the complex flow features in the entrance region. [26]

$$\frac{N_u}{N_{u,\infty}} = 1 + \frac{8.7}{L_D + 5} \tag{30}$$
2.3 Additive Manufacturing

Additively manufacturing, with another name 3D printing, is a production method that differs from the traditional manufacturing methods such as machining, molding, forming, and joining, with the way it produces materials. In the AM method, to build the desired good, layers are added one by one. It does not require any prior preparation which for example, the molding method needs a mold to inject the materials inside it. Neither, it does not require material removal like CNC machining. Moreover, the complexity of material affects the production processes of the previously mentioned methods.

AM offers to manufacture complex geometries without bulky premanufacturing activities. Thus, it makes it easier to generate products that are not easy to produce with the conventional methods, and prototype realizations became faster thanks to the AM methods. However, there are still a variety of parameters to consider during additively manufacturing. Such parameters like heat power, traveling speed, and layer thickness need to be defined carefully. For instance, high layer thicknesses can reduce the manufacturing time and the cost, but it also affects the quality of the material and porosity of the surface.

Selective Laser Melting (SLM) methodology was used to manufacture the materials examined in the present thesis. In SLM, a layer with the material powder is added, and the laser melts the selected locations to manufacture the desired product. After each layer is done, the new powder is provided to generate the next layer, and the laser melts the powder to produce the new layer. The process repeats until to obtain the desired products. [27] [28]

2.4 Surface Roughness Parameters

The $Ra$ parameter is the arithmetic average of the profile heights in the surface evaluation profile. It represents the mean surface roughness. The average maximum height of the surface evaluation profile is represented with $Rz$, and $Rp$ stands for the distance of the maximum peak point to the mean line of the surface evaluation profile. $Rt$ parameter is the vertical distance between the maximum and minimum points obtained in the surface evaluation profile. [29]
3 Experimental Apparatus

In this section, the qSSHT rig of Siemens Energy will be introduced to ensure a clear understanding of the experimental setup used in the study. The needed data to calculate the parameters to understand the cooling performance of the additively manufactured materials were measured and gathered with the experimental setup. Thus, the experimental setup should be presented in detail to ensure the credibility of the outcome of the study, clearance, and repeatability.

3.1 Test Rig

The schematic, and Computer Aided Design of the test rig are presented in Figures 4, 5, respectively.

![Schematic representation of the test rig.](image)

**Figure 4: Schematic representation of the test rig.**
To measure the pressure losses and heat transfer performance, mass flow inside the test channels is needed. Fluid flow due to the external agent results with the forced convection, and the external agent for the fluid motion comes from the pressurized air system in the Fluid Dynamics Laboratory of Siemens Energy in Finspång.

The manual ball valve in the open position activates the mass flow inside the test rig. Before the fluid goes into the test rig, Coriolis mass flow meter is employed to measure the mass flow rates, which is critical for the calculations of the examined parameters (i.e. Reynolds number). Additionally, a regulating valve is available on the inlet and outlet of the test rig compartment to adjust the amount of mass flow going inside the test compartment. The data acquired by the Coriolis was transferred to the computer by a USB connection with Datascan 7220 Analog Input Measurement Processor.

The test object is inside a copper block, which has high conductivity and can help to ensure the constant temperature boundary condition. The temperature of copper block and test objects increases using two nozzle heaters (for the Nusselt number tests only). In the interactions of the copper block and test object, and PT-100 temperature sensors inside the copper block, MX-4 High-Performance Thermal Paste is applied to obtain perfect contact and increase the heat transfer rates between the materials. The required power to generate the heat comes from the RS PRO Bench Power Supply (RS-6500D).

To measure the temperature for the inlet and outlet fluid flow and the copper block, PT-100 temperature sensors are located in the corresponding positions. PT-100 sensors connect to the Agilent 34970A Data Acquisition System, and data is transferred to the computer by an Ethernet connection.

Pressure taps are located in the various positions on the inlet and outlet of the test compartment to measure the pressures. NetScanner 9116 Intelligent Pressure Scanner system connects to the computer with an Ethernet connection. Additionally, to measure the atmospheric pressure, the Rosemount 3051C Coplanar Pressure Transmitter device is used. Similar to the Coriolis connection, data achieved from the
Rosemount 3051C transfer to the computer with a USB connection from the Datascan 7220.

Until this point, the measurement devices to acquire the required data to investigate the thermal performance of the additively manufactured test object were introduced. As represented, in the end, all data measurements are transferred to the computer. To monitor, and take a time average of the measured data, the in-house software Siemens RigVIEW is used. However, to examine the stabilization of the data, before sampling them, monitoring the change in the properties with time is crucial. RigVIEWClient is the software used to observe the real-time graphs for the measurements (i.e., pressure, temperature, mass flow). For the hot experiments, a Python script is used to monitor the stabilization of the results.

Figure 6 illustrates the mentioned connections of the experimental apparatus together with the computer and software connections of the qSSHT rig of Siemens Energy and taken from the database of the company.

![Figure 6: Schematic representation of the experimental instrumentation.](image)

### 3.2 Test Objects

<table>
<thead>
<tr>
<th>ID</th>
<th>Design Diameter [mm]</th>
<th>Measured Diameter [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.75</td>
<td>0.79275</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>1.04800</td>
</tr>
<tr>
<td>3</td>
<td>1.5</td>
<td>1.55700</td>
</tr>
<tr>
<td>4</td>
<td>2</td>
<td>2.07500</td>
</tr>
</tbody>
</table>
Table 7: List of the non-circular test objects.

<table>
<thead>
<tr>
<th>ID</th>
<th>Design Height [mm]</th>
<th>Measured Height [mm]</th>
<th>Design Width [mm]</th>
<th>Measured Width [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.75</td>
<td>0.85778</td>
<td>0.75</td>
<td>0.85778</td>
</tr>
<tr>
<td>6</td>
<td>1</td>
<td>1.07690</td>
<td>1</td>
<td>1.07690</td>
</tr>
<tr>
<td>7</td>
<td>1.5</td>
<td>1.57007</td>
<td>1.5</td>
<td>1.57007</td>
</tr>
<tr>
<td>8</td>
<td>2</td>
<td>2.08742</td>
<td>2</td>
<td>2.08742</td>
</tr>
<tr>
<td>9</td>
<td>0.75</td>
<td>0.85891</td>
<td>1</td>
<td>1.06037</td>
</tr>
<tr>
<td>10</td>
<td>0.75</td>
<td>0.84958</td>
<td>1.5</td>
<td>1.58873</td>
</tr>
<tr>
<td>11</td>
<td>0.75</td>
<td>0.80810</td>
<td>2</td>
<td>2.08030</td>
</tr>
<tr>
<td>12</td>
<td>1</td>
<td>1.07649</td>
<td>1.5</td>
<td>1.58443</td>
</tr>
<tr>
<td>13</td>
<td>1</td>
<td>1.07799</td>
<td>2</td>
<td>2.16219</td>
</tr>
<tr>
<td>14</td>
<td>1.5</td>
<td>1.59339</td>
<td>2</td>
<td>2.08240</td>
</tr>
</tbody>
</table>

The test object lists were given in Tables 6 and 7 for the circular and non-circular test objects respectively. The material of the objects in the Tables is STAL15 superalloy. With the manufacturing capabilities available today, it is not possible to realize products with the exact dimensions. As shown in the tables, the design dimensions and measured ones differ for the channels. The performed study is tremendously sensitive to the dimensions of the test channels. Thus, microscopic pictures were taken to estimate the exact dimensional properties of the test objects. And for the processing of the data, the measured dimensions were used. Additionally, the length of the test objects is 90 [mm].

In addition to the given STAL15 test objects, circular Aluminum 2 [mm] test object was used to validate the experimental setup since it has a "smooth" surface, and the results obtained from this test sample should follow the theoretical lines.

Microscopic pictures for the "smooth" Aluminum test object and additively manufactured STAL15 circular and non-circular test object were given in Figure 7. The surface roughness effects can be seen in the microscopic pictures.

![Microscopic pictures of circular aluminum 2 [mm] (a) circular STAL15 2 [mm] (b) and non-circular STAL15 2 x 2 [mm] (c)](image)

Figure 7: Microscopic pictures of circular aluminum 2 [mm] (a) circular STAL15 2 [mm] (b) and non-circular STAL15 2 x 2 [mm] (c)

After the experiments were done for all test objects, the test objects were cut in the flow direction and perpendicular to the flow direction. Then, to get the surface roughness parameters were measured inside the flow channels with MarSurf PS 10 roughness measuring device. MarSurf PS 10 extracts a wide range of measurement parameters from the surface evaluation profile. However, for the present thesis, only Ra, Rz, Rt, and Rp values will be represented.
The surface evaluation profile is the 2-dimensional representation of the texture of the surface for the sampled position of the surface with the probe tip of the measurement device. Sample surface evaluation profile is provided in Figure 8 for the STAL15 2 [mm] circular test object.

![Surface evaluation profile for STAL15 2 [mm].](image)

From the evaluation profile, parameters such as Ra, Rz, Rt, and Rp were obtained (see 2.4).

Surface parameters for the STAL15 test objects are given in Tables 8 and 9.

<table>
<thead>
<tr>
<th>ID</th>
<th>Ra [mm]</th>
<th>Rz [mm]</th>
<th>Rp [mm]</th>
<th>Rt [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.01256</td>
<td>0.06430</td>
<td>0.03363</td>
<td>0.09288</td>
</tr>
<tr>
<td>2</td>
<td>0.01082</td>
<td>0.05806</td>
<td>0.03240</td>
<td>0.07965</td>
</tr>
<tr>
<td>3</td>
<td>0.01249</td>
<td>0.06486</td>
<td>0.03589</td>
<td>0.08480</td>
</tr>
<tr>
<td>4</td>
<td>0.01243</td>
<td>0.06313</td>
<td>0.03533</td>
<td>0.08109</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>ID</th>
<th>Ra [mm]</th>
<th>Rz [mm]</th>
<th>Rp [mm]</th>
<th>Rt [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.01167</td>
<td>0.05710</td>
<td>0.02861</td>
<td>0.08121</td>
</tr>
<tr>
<td>6</td>
<td>0.01043</td>
<td>0.05609</td>
<td>0.02979</td>
<td>0.07416</td>
</tr>
<tr>
<td>7</td>
<td>0.01123</td>
<td>0.06019</td>
<td>0.03558</td>
<td>0.07681</td>
</tr>
<tr>
<td>8</td>
<td>0.01121</td>
<td>0.05301</td>
<td>0.02918</td>
<td>0.06375</td>
</tr>
<tr>
<td>9</td>
<td>0.01287</td>
<td>0.06678</td>
<td>0.03693</td>
<td>0.09924</td>
</tr>
<tr>
<td>10</td>
<td>0.01166</td>
<td>0.06190</td>
<td>0.03491</td>
<td>0.08135</td>
</tr>
<tr>
<td>11</td>
<td>0.01164</td>
<td>0.06037</td>
<td>0.03429</td>
<td>0.08178</td>
</tr>
<tr>
<td>12</td>
<td>0.01281</td>
<td>0.06604</td>
<td>0.03664</td>
<td>0.08963</td>
</tr>
<tr>
<td>13</td>
<td>0.01198</td>
<td>0.06268</td>
<td>0.03474</td>
<td>0.06015</td>
</tr>
<tr>
<td>14</td>
<td>0.01123</td>
<td>0.05856</td>
<td>0.03258</td>
<td>0.07757</td>
</tr>
</tbody>
</table>

From the surface measurement, it could be seen that obtained parameters are comparable for all test objects since the materials and printing techniques for the objects are the same.
3.3 Test Procedure

The test rig has to be tested for the leakages before performing any experiments. Leakage free experimental setup is critical to achieving accurate results. In the presence of leakage, the measured mass flow rates and pressures will not be credible. Therefore, the leakage test is one of the most crucial parts of the experimental procedure of the study.

The first leakage test methodology is based on eye inspection. As a first step of the eye inspection test, the rig should be fully pressurized, and the soap-water mixture is sprayed on the surface of the test rig, especially the positions with a high risk of leakages such as pressure taps. Then, the rig is inspected visually for some time, and the occurrence of bubbles means that there is leakage in the observed locations. The eye inspection methodology could not be credible for the significantly small leakages or the positions which are hard to see during the test. Thus, another leakage test methodology with higher measurability is needed to ensure higher accuracy.

The second methodology to investigate the leakages in the test setup uses the measurements from the rig. For this methodology, similar to the previous one, the rig needs to be pressurized. When the maximum available pressure is obtained from the central pressurized air system, the main air supply valve to the rig should be closed. Then, pressure measurements and the mass flow rate measurement from the rig need to be followed from the RigView channel monitor screen. The system should maintain its pressure, and there should be no mass flow through to the experimental setup to be able to continue with the experiments.

In case of observation of any leak in the rig, fixing them is a must. The leakages in the pressure taps could be fixed with X60: Cold Curing Glue for Experimental Tests, and for the threads where plastic and metal interaction is present, Loctite Threadlocker Red 271 could be the solution. On the other hand, thread seal tape could be used for the threads with metal and metal interaction.

To achieve the data required in the study, mainly two different test methodologies were used. The pressure losses in the channels were obtained by performing the Darcy friction factor tests where the heaters are closed. Oppositely, for the heat transfer performance evaluations, where Nusselt number calculations were performed, heaters should be activated. The test with active heaters could be called a Nusselt number test since the test object will be cooled down with forced convection which is caused by the external agent. Darcy friction factor and Nusselt number test procedures are shown step by step in Tables 10 and 11.

The methodology used to calculate the examined parameters for the thermal performance investigation is sensitive to the Mach number. Thus, Mach numbers should be kept as small as possible during the experiments, and the flow should be incompressible (M<0.3).
Table 10: Darcy friction factor test procedure

<table>
<thead>
<tr>
<th>Step</th>
<th>Activity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Safety check (Hoses, connections)</td>
</tr>
<tr>
<td>2</td>
<td>Open RigView, Zero adjustment for Coriolis, and pressure measurement device</td>
</tr>
<tr>
<td>3</td>
<td>Leakage test</td>
</tr>
<tr>
<td>4</td>
<td>Start tests with the lowest mass flow rate, follow pressures and mass flow until stabilization</td>
</tr>
<tr>
<td>5</td>
<td>Log the data</td>
</tr>
<tr>
<td>6</td>
<td>Check the data using plots (Re, Darcy), also be sure that the Mach number is low (i.e. no compressible effects)</td>
</tr>
<tr>
<td>7</td>
<td>Change mass flow for the next point</td>
</tr>
</tbody>
</table>

Table 11: Nusselt number test procedure

<table>
<thead>
<tr>
<th>Step</th>
<th>Activity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Safety check (Hoses, connections)</td>
</tr>
<tr>
<td>2</td>
<td>Open RigView, Zero adjustment for Coriolis, Rezero for pressure measurement device</td>
</tr>
<tr>
<td>3</td>
<td>Leakage test</td>
</tr>
<tr>
<td>4</td>
<td>Increase the temperature of the test object and copper block</td>
</tr>
<tr>
<td>5</td>
<td>Open mass flow fully, start saving the data</td>
</tr>
<tr>
<td>6</td>
<td>Monitor stabilization of Nu/Nu0 (with .py code) also check Re to fill the plots better</td>
</tr>
<tr>
<td>7</td>
<td>Log the data</td>
</tr>
<tr>
<td>8</td>
<td>Change mass flow for the next point</td>
</tr>
</tbody>
</table>
4 Method

The data obtained from the experimental setup are needed to be post-processed to understand the thermal performance of the different test objects. Since the number of test objects and data points for each object is tremendously high, and for some cases, an iterative calculation is necessary, scripts were used to calculate the required parameters.

Three main parameters were used to represent the results from the study and ensure the comparison between different test objects, namely, Reynolds number, Darcy friction factor, and Nusselt number. The last two mentioned parameters will be plotted over the first parameter to show the change in the parameters for different flow regimes and Reynolds numbers.

4.1 Reynolds Number

Reynolds number could be calculated with the given Equation 2. However, in the experimental setup, there is no direct velocity measurement from the experimental apparatus. The Coriolis mass flow meter enables the measurement of the mass flow rates going through the test object. Thus, the Reynolds number was calculated with Equation 31.

\[ Re = \frac{\dot{m} D_h}{\mu A_{cs}} \]  

Where \( \dot{m} \) is mass flow rate, \( D_h \) is the hydraulic diameter of the channel, \( \mu \) is the viscosity of the flow and \( A_{cs} \) is the cross-sectional area of the flow channel. Now, the calculation of the required variables to obtain the Reynolds number will be introduced.

**Hydraulic Diameter**

For the circular channels, the value from the microscopic measurements could be used directly. However, for the non-circular ones, Equation 32 was used to calculate the hydraulic diameters. Nevertheless, the perimeter values were taken from the microscopic pictures.

\[ D_h = \frac{4A_{cs}}{P} \]  

27
Viscosity

Sutherland’s law given in Equation 4 was used to calculate the viscosity of air for the different temperatures.

\[ \mu = \mu_{\text{ref}} \left( \frac{T}{T_{\text{ref}}} \right)^{3/2} \frac{T_{\text{ref}} + S}{T + S} \]  

(33)

4.2 Darcy Friction Factor

Darcy friction factor was calculated by using the given relationship in Equation 7. However, as introduced in the Theory Chapter, pressure losses arise due to the surface friction losses and induced turbulence losses. Furthermore, induced pressure losses are caused by the sudden contraction and expansion in the inlet and outlet, respectively. The calculated Darcy friction factor should be independent of the losses caused by the experimental setup. In other words, the study aims to investigate only the pressure losses inside the channel, without the additional losses in the air inlet and outlet of the test channel. Thus, pressure losses caused by the cross-sectional area change in the inlet and outlet (i.e. induced turbulence losses) should be calculated separately and subtracted from the measured pressure difference from the experimental setup.

The introduced net pressure losses, which are "pure" pressure losses that come from only the surface friction, can be formulated as given in Equation 34.

\[ \Delta P_{\text{surface friction}} = \Delta P_{\text{measured from the rig}} - \Delta P_{\text{induced turbulence}} \]  

(34)

Induced turbulence pressure losses due to the entrance effects on the sudden contraction in the inlet and expansion on the outlet of the test objects were calculated by Equation 21 without the gravitational acceleration term and manipulated mass flow rate term with the velocity term by using the relationship given in Equation 18. The induced turbulence losses are present for both laminar and turbulent flow regimes. For the laminar regime, the flow inside the channel is laminar except for the mentioned regions where induced turbulence occurs (i.e. flow separation, circulations). In Equation 35, the final mathematical representation is shown for the calculation of the induced turbulence pressure losses.

\[ \Delta P_{\text{induced turbulence}} = K \frac{D_w V^2}{2} \]  

(35)

To calculate the induced turbulence pressure losses in the inlet and outlet, velocities and densities in the inlet and outlet were calculated separately. The selected values for the loss coefficient K from Idelchik’s research paper [21] can be seen in Table 12.
Table 12: Loss coefficient values for the inlet and outlet for different flow regimes.

<table>
<thead>
<tr>
<th></th>
<th>Inlet</th>
<th>Outlet</th>
</tr>
</thead>
<tbody>
<tr>
<td>Laminar Region</td>
<td>0.5</td>
<td>2</td>
</tr>
<tr>
<td>Turbulent Region</td>
<td>0.5</td>
<td>1</td>
</tr>
</tbody>
</table>

The final and detailed form of the Darcy-Weisbach formula to calculate the Darcy friction factor is shown in Equation 36:

\[
f = \frac{D}{L} \frac{\Delta P\text{ measured from the rig}}{0.5(\rho_{inlet}V_{inlet}^2 + \rho_{outlet}V_{outlet}^2)} - \left( K_{inlet} \frac{\rho_{inlet}V_{inlet}^2}{2} + K_{outlet} \frac{\rho_{outlet}V_{outlet}^2}{2} \right)
\]

(D6)

**Density**

Equation of state given in Equation 37 was used to calculate the density values in the inlet and outlet of the test channel by using the temperature and pressure values in the corresponding regions.

\[
\rho = \frac{P}{RT}
\]

(37)

**Velocity**

Equation 38 uses the mass flow, density, and the cross-sectional area of the channel to calculate the velocity. With the Equation, flow velocities in the inlet and outlet of the channel were obtained.

\[
V = \frac{\dot{m}}{\rho A}
\]

(38)

**Mach Number**

As stated before, the experiment is not independent of the Mach number, and the parameter needed to be less than 0.3 to ensure the incompressible flow. However, 0.3 is not a strict limit that defines the flow compressibility. Thus, it is ideal to ensure the Mach number is as small as possible during the experiments. Equation 39 was used to get the Mach number for different operating points and monitor the value, but the quantity was not used in any other calculation during the study.

\[
M = \frac{V}{Speed\ of\ sound}
\]

(39)
4.3 Nusselt Number

The wall temperature inside the channel is not constant because of the heating of the fluid flow and the low conductivity of the additively manufactured materials (i.e. 8.42 W/mK for STAL15). On the other hand, the outer wall temperature of the channel is constant (i.e. constant temperature boundary condition). The fluid enters inside the test channel with temperatures similar to room temperature, and inside the test channel, fluid temperature increases. The phenomenon means, in regions close to the inlet of the test channels, the average metal temperature is lower than the average temperature of the regions close to the channel outlet (see Figure 9).

An iterative solution is needed to calculate the Nusselt number since heat transfer coefficient is unknown for the test channel and different values of the parameter will be used until the correct value is reached. It makes the calculations more complex compared to the Darcy friction factor ones. Recorded data will be using from PT100 sensors and the Coriolis mass flow meter.

\[ q = \frac{T_{\text{gas}} - T_{\text{copper}}}{\frac{1}{P_i h} + \frac{\ln(r_o/r_i)}{2\pi k} + \frac{1}{P_o h_{\text{contact}}}} \]  \hspace{1cm} (40)

\[ T_w = T_{\text{gas}} - \frac{q}{P_i h} \]  \hspace{1cm} (41)

\[ T_{\text{gas},i+1} = T_{\text{gas},i} - \frac{q dx}{mc_p} \]  \hspace{1cm} (42)

The first methodology to calculate the Nusselt number is with the given formulas for the axisymmetric objects which comes from the 1D Fourier heat transfer law, and the Python code will be used for the iterations. The approach is valid for the circular test object since the temperature distribution for these objects is axisymmetric. On the other hand, for the non-circular test object, in-house computer software C3d turbine cooling program was used, and the methodology for the software will be introduced after the first one.

The test channel was divided into 100 cross-sections in the flow direction to calculate the local parameters. The local values of heat flux and wall temperatures for the discretized parts were calculated with Equation 40 and 41. The calculation starts from the first section, where is the inlet of the channel. For the inlet value of the gas temperature, the measured inlet air temperature from the experimental setup was
taken, and Equation 42 was used to calculate the gas temperature in the following section. After the first section calculations were done, the gas temperatures, heat flux values, and inner wall temperatures were calculated for the remaining ones. Additionally, for the heat transfer coefficient of the contact, the ratio of the conductivity of the thermal paste to the gap distance (see Equation 43) was used, where the thermal paste was applied.

\[
h_{\text{contact}} = \frac{\text{Thermal conductivity of the MX4}}{\text{Thermal paste gap}}
\]  

(43)

The difference between the calculated outlet gas temperature value and the measured one was checked. Then, different heat transfer coefficients were applied to get the difference between the resulting outlet temperature values and the value measured from the experiments lower than 0.1 K. Obtained heat transfer coefficient value from the iterations, which ensures the difference limit for the outlet gas temperature, was used in Equation 25 to calculate the Nusselt number.

For a better and visualized understanding of the Nusselt number calculation methodology, the flow chart is provided in Figure 10.

The second and more accurate method to calculate the Nusselt number is with C3d turbine cooling software since it considers the effect of various parameters and is optimized for this purpose. C3d performs iterations with the different heat transfer coefficients and calculates the outlet air temperature and mass flow. The program compares the calculated variables with the measurements from the experiments. Modifies the heat transfer coefficients until the difference between the quantities is within the desired limit.

To use the turbine cooling software C3d, the geometry of the test object needs to be created and meshed. Geometry was created and meshed in Siemens NX. Dimensions of the test channel were taken from the microscopic picture measurements. The software supports various mesh types, but in the present study, the mesh type used was hexahedral C3D8 (Eight-node brick element).

C3d software relates the mesh elements generated for the solid part of the test objects on the Siemens NX. Thus, the number of mesh elements in each branch of the C3d model should be the same. In the C3d model, nodes should be generated from the inlet to the outlet of the channel. An additional node needs to be placed after the outlet of the test channel to determine the outflow temperature. The extra branch should not cause any additional pressure drop. Between the nodes, branches were created. The diameters and cross-sectional areas for each branch were defined.

The material of the test object was defined. Temperature and pressure boundary conditions were set in the inlet and outlet nodes of the test object for the operating point with the highest mass flow rate. Additionally, the copper temperature for the
The first operating point was defined for the outer surface of the test object, and the value was assumed to be the same as the outer wall temperature of the test object. The heat transfer coefficient for the outer wall of the test object was set as 10000. The value means the heat transfer between the copper block and test object is assumed as tremendously high, and the temperature of the outer wall of the test object is the same as the copper block temperature. Visual illustration of the boundary conditions was given in Figure 11.

Then, the initial value for the heat transfer coefficient (Nusselt number) and fanning friction factor, which is one-fourth of the Darcy friction factor, were introduced to the model. The flow network was coupled to the mesh.

After the case file is ready, the conjugate solver was run to converge in the first operating point, and the file was saved. With the help of the Python script, the solver was run for all different operating points in the batch mode, and the calculated Nusselt numbers for the examined Reynolds number range were written to text files.

\( T_{\text{diff}} \) in the chart (Figure 10) stands for the difference between the calculated \( T_{\text{gas}} \) for the outlet of the test object, and measured one in the experiments from the outlet of the test objects with the PT100 temperature sensors.

![Flow chart for Nusselt number calculation methodology.](image)

**Figure 10:** Flow chart for Nusselt number calculation methodology.
Figure 11: Applied boundary conditions in C3d.
5 Results

In this section, Darcy friction factor and Nusselt number results obtained from the study will be given. The x-axis will be the Reynolds number for both tests, and in the y-axis, Darcy friction factor and Nusselt values will be given for the corresponding Reynolds number.

Darcy friction factor results will be plotted together with the theoretical lines for the laminar and turbulent regimes. For the laminar regime Equation 8, and for the turbulent regime, the Colebrook-White equation (see Equation 11) for the smooth surfaces ($\epsilon$ is equal to 0) will be used.

Except for the validation case and one complete result of all test objects, the Nusselt number results will be shown in the ratio form ($\text{Nu}/\text{Nu}_0$) to show the enhancement compared to the smooth surfaces.

5.1 Test Rig Validation

Darcy friction factor and Nusselt number test results for the Aluminum 2 [mm] test object are given in Figures 12 and 13, respectively.

![Figure 12: Rig validation with Darcy friction factor.](image)

In the Darcy friction factor results shown in Figure 12, the Aluminum 2 [mm] test object followed the theoretical line for the laminar region and the Colebrook-White equation for the smooth surfaces (i.e. $\epsilon = 0$) in the turbulent regime. However, in the transient region between the laminar and turbulent ones, Darcy friction factor results were not able to follow the theoretical lines of either the laminar or the turbulent regimes. Furthermore, close to the transition regions, values started to differ compared to the theoretical lines. Darcy friction factor results were stuck on
a constant value after Reynolds numbers of 35 000, and after Reynolds numbers of 55 000 friction factor values increased.

![Nu vs Re graph]

Figure 13: Rig validation with Nusselt number.

Nusselt number results given in Figure 13 shows that Nusselt numbers obtained for the Aluminum 2 [mm] test object differ from the Gnielinski line for the smooth surfaces. However, except for the points on the transition region, the results match the Gnielinski enhanced line, in which the entrance effects on the heat transfer (see Equation 30) were applied.

### 5.2 Darcy Friction Factor

![Darcy friction factor vs Re graph]

Figure 14: Darcy friction factor results.

The results obtained from the Darcy friction factor tests for all test samples are given in Figure 14. The lowest Reynolds number obtained in the experiments was 300, and
the highest was 80 000. In general, for the channels with smaller hydraulic diameters obtained maximum Reynolds numbers were lower than the ones obtained with the bigger hydraulic diameters. Only the comparison of the circular and non-circular objects with similar hydraulic diameters violates the trend. Additionally, the Darcy friction factors were higher for the smaller channels, and generally, the transitions from the laminar to the turbulent regime were started in the earlier Reynolds numbers.

For the laminar region, all channels followed a similar trend with the theoretical line. However, friction factor values were higher than the smooth surfaces. The friction factor values started from higher values and decreased until the start of the transition. With the start point of the flow phase change from laminar to turbulent, Darcy friction factor values were increased, and the increase continued even in the turbulent regime until Reynolds number of 6 000. After a particular Reynolds number in the turbulent flow regime, the Darcy friction factor tends to stick on a constant value. Moreover, there were drops in the Darcy friction factor for the last data points obtained in the highest possible Reynolds numbers.

To ensure a better comparison, the results are grouped in separate plots and illustrated from Figures 15 to 20.

![Figure 15: Darcy friction factor results for circular channels.](image)

Darcy friction factor comparison for the circular objects with different diameters is given in Figure 15. For the circular test objects, the lowest friction factor was obtained with the STAL15 2 [mm] channel, which has the largest hydraulic diameter in the comparison case, and the highest Darcy friction factor was experienced with the lowest hydraulic diameter, which is the STAL15 0.75 [mm] test object. Friction results obtained for the 1.5 [mm] channel is slightly higher than the one calculated for the 2 [mm] test object. However, with the STAL15 1 [mm] test sample, drastically higher friction values were obtained, compared to the STAL15 2 [mm] and 1.5 [mm] samples.
The earliest transition was started for the STAL15 0.75 [mm] object, from the Reynolds number of 1600, and for the channels with higher hydraulic diameters, transition points were obtained at higher Reynolds numbers. Until the Reynolds numbers of 3000, data obtained for the STAL15 2 [mm] channel follow the laminar theoretical line.

To compare the effect of the channel shape on the Darcy friction factor, square-shaped objects, which have similar hydraulic diameters with the circular objects given in Figure 15, were added to the plot and are shown in Figure 16. The comparison shows that, in the laminar region, the obtained Darcy friction factor levels were tremendously similar for the objects with similar hydraulic diameters. However, slightly higher Darcy friction factor levels were obtained for the channels with square shapes, in the turbulent regime.
In Figure 17, Darcy friction factor results for the non-circular test samples with at least one side of the channel having 2 [mm] diameter are plotted together.

The minimum friction factor for the compared objects was obtained for the STAL15 2 x 2 [mm], and the highest one is for 0.75 x 2 [mm]. Additionally, with the reduction of the dimension of the other side than the 2 [mm] one of the test objects, the Darcy friction factor increases.

Transitions were started from the similar Reynolds numbers for STAL15 2 x 2 [mm] and 1.5 x 2 [mm] test objects, also, the object with the higher channel height showed a later transition than the one with lower. The similar trend was experienced for STAL15 1 x 2 [mm] and 0.75 x 2 [mm] objects. However, an opposite trend in the Reynolds number for the transition start points was available in this case.

![Figure 18: Darcy friction factor results for 1.5 [mm] channels.](image)

Non-circular objects where at least one side length of the channel are 1.5 [mm] are given in Figure 18. For both 1.5 x 1.5 [mm] and 1.5 x 2 [mm] test objects, almost the same Darcy friction factors, and transition points were obtained, even with different hydraulic diameters. However, the highest Reynolds number was achieved with the STAL15 1.5 x 2 [mm], which has the highest hydraulic diameter in this comparison case. The highest friction factor values were obtained for the object with the lowest hydraulic diameter in the comparison case, namely, STAL15 0.75 x 1.5 [mm], and the second-highest Darcy friction factors were shown with 1 x 1.5 [mm]. Moreover, from the comparison of the latest mentioned test objects, STAL15 0.75 x 1.5 [mm] was followed the laminar line longer than the STAL15 1 x 1.5 [mm] sample. For the 0.75 x 1.5 [mm] object, the deviation between the smooth surfaces in the laminar region was higher than the other compared ones.
Figure 19: Darcy friction factor results for 1 [mm] channels.

Figure 19 illustrates the comparison case for the rectangular test objects with at least one side of the channel is 1 [mm]. The lowest Darcy friction factor was reached with the STAL15 1 x 2 [mm] object, which has the highest hydraulic diameter of the case, and with the increase in the hydraulic diameter, there was a reduction in the obtained friction factor values. STAL15 1 x 1.5 [mm] test object was experienced earlier transition than 1 x 1 [mm] and 0.75 x 1 [mm] ones. Except for the 0.75 x 1 [mm] test sample, similar friction factor values were achieved in the laminar regime for all other channels.

Figure 20: Darcy friction factor results for 0.75 [mm] channels.

Darcy friction factor results for the 0.75 [mm] sided objects are given in Figure 20 show that for the lower hydraulic diameters, higher Darcy values were experienced. Additionally, higher maximum Reynolds number values and transition start points were obtained for the objects with higher hydraulic diameters. For the compared test samples, high deviations from the laminar theoretical line were seen.
Darcy friction factor values obtained in the turbulent regime are given in Tables 13 and 14 for the circular and non-circular objects respectively. Hydraulic diameters of the test channels and surface roughness parameters are also shown in the tables to illustrate the effect of the parameters on Darcy friction factor values.

Table 13: Darcy friction factor values for circular test channels, together with hydraulic diameters and surface roughness parameters.

<table>
<thead>
<tr>
<th>ID</th>
<th>( \text{fd [-]} )</th>
<th>( \text{Dh [mm]} )</th>
<th>( \text{Ra/Dh [-]} )</th>
<th>( \text{Rz/Dh [-]} )</th>
<th>( \text{Rp/Dh [-]} )</th>
<th>( \text{Rt/Dh [-]} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.26293</td>
<td>0.792</td>
<td>0.01586</td>
<td>0.08119</td>
<td>0.04246</td>
<td>0.11728</td>
</tr>
<tr>
<td>2</td>
<td>0.17363</td>
<td>1.048</td>
<td>0.01032</td>
<td>0.05540</td>
<td>0.03092</td>
<td>0.07600</td>
</tr>
<tr>
<td>3</td>
<td>0.10340</td>
<td>1.557</td>
<td>0.00802</td>
<td>0.04165</td>
<td>0.02305</td>
<td>0.05446</td>
</tr>
<tr>
<td>4</td>
<td>0.08991</td>
<td>2.075</td>
<td>0.00599</td>
<td>0.03042</td>
<td>0.01703</td>
<td>0.03908</td>
</tr>
</tbody>
</table>

For the circular objects with an increase in the hydraulic diameter (Dh), a decrease in the Darcy friction factor values (\( \text{fd} \)), and a decrease in the relative roughness parameters (Ra/Dh, Rz/Dh, Rp/Dh, Rt/Dh) were observed.

Table 14: Darcy friction factor values for non-circular test channels, together with hydraulic diameters and surface roughness parameters.

<table>
<thead>
<tr>
<th>ID</th>
<th>( \text{fd [-]} )</th>
<th>( \text{Dh [mm]} )</th>
<th>( \text{Ra/Dh [-]} )</th>
<th>( \text{Rz/Dh [-]} )</th>
<th>( \text{Rp/Dh [-]} )</th>
<th>( \text{Rt/Dh [-]} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.28657</td>
<td>0.0857</td>
<td>0.01362</td>
<td>0.06663</td>
<td>0.03339</td>
<td>0.09476</td>
</tr>
<tr>
<td>6</td>
<td>0.17788</td>
<td>1.0760</td>
<td>0.00969</td>
<td>0.05213</td>
<td>0.02768</td>
<td>0.06892</td>
</tr>
<tr>
<td>7</td>
<td>0.11440</td>
<td>1.5700</td>
<td>0.00715</td>
<td>0.03833</td>
<td>0.02266</td>
<td>0.04892</td>
</tr>
<tr>
<td>8</td>
<td>0.09480</td>
<td>2.0870</td>
<td>0.00537</td>
<td>0.02540</td>
<td>0.01398</td>
<td>0.03054</td>
</tr>
<tr>
<td>9</td>
<td>0.22528</td>
<td>0.9490</td>
<td>0.01356</td>
<td>0.07037</td>
<td>0.03891</td>
<td>0.10457</td>
</tr>
<tr>
<td>10</td>
<td>0.20677</td>
<td>1.1070</td>
<td>0.01053</td>
<td>0.05592</td>
<td>0.03153</td>
<td>0.07349</td>
</tr>
<tr>
<td>11</td>
<td>0.14679</td>
<td>1.1640</td>
<td>0.01000</td>
<td>0.05186</td>
<td>0.02945</td>
<td>0.07026</td>
</tr>
<tr>
<td>12</td>
<td>0.15241</td>
<td>1.2810</td>
<td>0.01000</td>
<td>0.05155</td>
<td>0.02860</td>
<td>0.06997</td>
</tr>
<tr>
<td>13</td>
<td>0.12750</td>
<td>1.4250</td>
<td>0.00840</td>
<td>0.04399</td>
<td>0.02438</td>
<td>0.04221</td>
</tr>
<tr>
<td>14</td>
<td>0.11088</td>
<td>1.8050</td>
<td>0.00622</td>
<td>0.03244</td>
<td>0.01805</td>
<td>0.04298</td>
</tr>
</tbody>
</table>

The mentioned trend for the circular test objects is also applicable for most of the non-circular test objects. However, there are a few cases that violate the trend. For example, test object 12 has a higher hydraulic diameter than test object 11, but also higher Darcy value was obtained for it. Moreover, relative roughness parameters were obtained almost identical for the mentioned test objects.
5.3 Nusselt Number

The Nusselt number test results for the test objects are given in Figure 21, together with the Gnielinski line for the smooth surfaces.

![Figure 21: Nusselt number results.](image)

The lowest Reynolds number achieved in the experiment was 560, and the highest was 81 000. The calculated Nusselt number results were in the interval of 1.18 to 285. As a general trend, higher Nusselt numbers were obtained for the higher Reynolds numbers. However, dramatic drops in the Nusselt numbers were seen for the last points which have the highest Reynolds numbers for the test samples.

In the turbulent regime, except for the STAL15 0.75 [mm] circular test channel, and 0.75 x 0.75 [mm] non-circular one, the achieved Nusselt numbers were higher than the Nusselt numbers calculated with the Gnielinski correlation. Lower values were seen for the Nusselt number from the experiments for the mentioned exceptions.

Behavior changes were seen in the transient region for all of the test objects. In the transition region, the Nusselt number results were reduced more than the reduction trend seen in the turbulent regime with the decrease in the Reynolds number. Additionally, values were below the theoretical line.

To represent the increase in the heat transfer compared to the smooth surfaces, due to the surface roughness, the calculated Nusselt numbers were divided to the Nusselt number calculated with the Gnielinski correlation (for the smooth surfaces) for the same test object. The ratio represents the enhancement factor in the heat transfer and the achievements on the heat transfer performance with the additively manufactured materials.
For all test objects, Nusselt enhancement results are given in Figure 21. Aluminum 2 [mm] test sample results were added to the plot to compare the smooth surfaces with the additively manufactured porous ones.

The highest enhancement factor for the Aluminum 2 [mm] object was obtained as 1.28 at the highest Reynolds number. The factor means the obtained Nusselt number is 28 percent higher than the Nusselt number calculated with the Gnielinkski correlation.

Below the 10 000 Reynolds number, higher or similar enhancement factors were achieved in the Aluminum test sample compared to the additively manufactured objects with small diameters. STAL15 0.75 [mm], 0.75 x 0.75 [mm], 1 [mm], 1 x 1 [mm] and 0.75 x 1 [mm] test samples could be example for the mentioned trend. However, after 10 000 Reynolds numbers, higher enhancements in the heat transfer were seen with the rough surfaces.

Nusselt enhancement factors were started from the smaller values for the lower Reynolds numbers, and ratios were increased with the increments of Reynolds numbers. After the Reynolds number of 25 000, the enhancement factor tends to stick in a constant value, except for the objects that were not possible to achieve a higher Reynolds number than the mentioned value. Moreover, for the objects which were able to stick in the constant enhancement factors, dramatic drops on the enhancements were present for the end of the maximum possible Reynolds numbers.

Similar to the Darcy friction factor experiments, higher Reynolds numbers were present for the higher hydraulic diameters. In general, higher heat transfer enhancements were achieved for the objects with higher hydraulic diameters. Moreover, compared to the circular ones, lower Reynolds numbers were obtained for the square objects with similar hydraulic diameters.
The same methodology used in the Darcy friction factor plots will be used to compare the objects with the important comparison cases. Grouped results of the Nusselt enhancement factor for the test samples are given in Figures 23 to 28.

Figure 23: Nusselt enhancement results for circular channels.

Nusselt enhancement results for the circular test objects are given in Figure 23. The enhancement factors were stuck on a constant value of approximately 1.9 for the STAL15 2 [mm] and 1.5 [mm] test objects and these objects have the highest ones in the comparison case. Nusselt enhancement factor values for the STAL15 0.75 and 1 [mm] test objects were increased with the increase of Reynolds numbers. However, no constant values were obtained for the mentioned 0.75 and 1 [mm] test samples. Higher values of Nusselt enhancement factors were present for the STAL15 1.5 and 2 [mm] objects, except for the region where a constantly increasing trend was experienced in the Reynolds number interval of 18 000 and 26 000 for the STAL15 1 [mm] sample.

Figure 24: Nusselt enhancement results for circular and square channels.
In Figure 24, square objects with similar hydraulic diameters to the circular ones were added to the previous comparison case. In the results, two different trends were experienced. For the STAL15 0.75 [mm] and 1 [mm] comparisons, higher enhancements were obtained for the square test channels. However, for the 1.5 and 2 [mm] cases, a lower heat transfer enhancement ratio were present for the square channels.

Nusselt number enhancement results for the non-circular STAL15 test objects with at least one side of the channel is 2 [mm] are given in Figure 25. The results show that higher enhancements were achieved with the 1.5 x 2 [mm] test sample. The enhancement factors of the test object have become constant at approximately 1.92. On the other hand, the lowest values were seen for the 0.75 x 2 [mm] and 2 x 2 [mm] samples, with approximate values of 1.75. Between these mentioned maximum and minimum ones, 1 x 2 [mm] test object were placed with 1.82 enhancement factors in the constant region. Additionally, below the Reynolds number of 8 000, similar enhancements factors were present for all compared test objects in the case.
The enhancement results are shown for 1.5 [mm] sided non-circular test objects in Figure 26. The results show that, between the Reynolds numbers of 3 000 and 8 000, all test objects except the STAL15 0.75 x 1.5 [mm] object, were shown the identical Nusselt enhancements behavior. On the other hand, lower values were obtained for the mentioned exception in the given interval. Constant increment factors were not obtained for the exceptional object, namely STAL15 0.75 x 1.5 [mm]. However STAL15 1 x 1.5 [mm] and 1.5 x 2 [mm] test samples reached the higher stable Nusselt increment factors compared to the square, STAL15 1.5 x 1.5 [mm] test sample.

![Figure 27: Nusselt enhancement results for 1 [mm] channels.](image)

For the 1 [mm] sided objects, the enhancement factor results of the Nusselt number are given in Figure 27. From Reynolds number 3 000 to 13 000, the test objects were in the order such as the object with a lower hydraulic diameter gives less increment, and vice versa. However, with an increase in the Reynolds numbers STAL15 1 x 1.5 [mm] and 1 x 2 [mm] started to stick on the constant values of increment factors. The obtained constant value for the STAL15 1 x 1.5 [mm] test sample was than the one achieved with STAL15 1 x 2 [mm]. The mentioned constant value trend was not seen for the STAL15 0.75 x 1 [mm] and 1 x 1 [mm] objects, the values for these objects were shown constant increasing trend until the maximum possible Reynolds numbers were achieved for them.
Figure 28: Nusselt enhancement results for 0.75 [mm] channels.

The Nusselt enhancement results for the 0.75 [mm] sided test objects are given in Figure 28. The constant values were only achieved with the STAL15 0.75 x 2 [mm] test object, and the objects with smaller diameters were not able to show the mentioned behavior in the comparison case. Higher Nusselt ratios were seen for the STAL15 0.75 x 1.5 [mm] object compared to the channel with 0.75 x 2 [mm]. Additionally, except for the last second highest points of the STAL15 0.75 x 0.75, the lowest enhancements were seen for this object.

Table 15: Nusselt number enhancement ratio values for circular test channels, together with hydraulic diameters and surface roughness parameters.

<table>
<thead>
<tr>
<th>ID</th>
<th>Nu/Nu0 [-]</th>
<th>Dh [mm]</th>
<th>Ra/Dh [-]</th>
<th>Rz/Dh [-]</th>
<th>Rp/Dh [-]</th>
<th>Rt/Dh [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>1.94088</td>
<td>1.557</td>
<td>0.00802</td>
<td>0.04165</td>
<td>0.02305</td>
<td>0.05446</td>
</tr>
<tr>
<td>4</td>
<td>1.90369</td>
<td>2.075</td>
<td>0.00599</td>
<td>0.03042</td>
<td>0.01703</td>
<td>0.03908</td>
</tr>
</tbody>
</table>

Table 16: Nusselt number enhancement ratio values for non-circular test channels, together with hydraulic diameters and surface roughness parameters.

<table>
<thead>
<tr>
<th>ID</th>
<th>Nu/Nu0 [-]</th>
<th>Dh [mm]</th>
<th>Ra/Dh [-]</th>
<th>Rz/Dh [-]</th>
<th>Rp/Dh [-]</th>
<th>Rt/Dh [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>7</td>
<td>1.80841</td>
<td>1.5700</td>
<td>0.00715</td>
<td>0.03833</td>
<td>0.02266</td>
<td>0.04892</td>
</tr>
<tr>
<td>8</td>
<td>1.74188</td>
<td>2.0870</td>
<td>0.00573</td>
<td>0.02540</td>
<td>0.01398</td>
<td>0.03054</td>
</tr>
<tr>
<td>11</td>
<td>1.74480</td>
<td>1.1640</td>
<td>0.01000</td>
<td>0.05186</td>
<td>0.02945</td>
<td>0.07026</td>
</tr>
<tr>
<td>12</td>
<td>1.82512</td>
<td>1.2810</td>
<td>0.01000</td>
<td>0.05155</td>
<td>0.02860</td>
<td>0.06997</td>
</tr>
<tr>
<td>13</td>
<td>1.87433</td>
<td>1.4250</td>
<td>0.00840</td>
<td>0.04399</td>
<td>0.02438</td>
<td>0.04221</td>
</tr>
</tbody>
</table>
5.4 Surface Roughness Parameters and Darcy Friction Factor Trend Comparison with Other Studies

In this section of the study, obtained Darcy friction factor values will be plotted over the surface roughness measurement data, namely Ra and Rz parameters. The data are contrasted with the previous similar studies performed by Siemens Energy in Sweden by Anna Nyhlen [31], The Pennsylvania State University [11] [12], and Nikuradse [7] will be added to the plots.

The results shown in this section could be considered as a validation of the results of the work and will show the importance of the results gathered with the project, and how it extends the existing knowledge of the effect of the surface roughness on the pressure losses.

In the comparison cases given in Figures 29 and 30, each data point represents one particular test sample. For both plots, the data from the Fluid Dynamics Laboratory of Siemens Energy STAL15 circular and non-circular test objects (which are the interest of the present thesis), Inconel 939, CM247LC test objects were taken.

The manufacturing materials of the test coupons of Stimpson et al. from The Pennsylvania State University were cobalt-chrome-molybdenum-based superalloy (CoCr) and Inconel 718. In this comparison case, Ra values are used as surface roughness parameters.

![Darcy friction factor vs Ra/Dh](image)

Figure 29: Darcy friction factor vs Ra/Dh.
Figure 29 shows the increasing trend is exponential with the increase in the Ra parameter, and the results obtained with the thesis work follow the general trend from the previous similar studies. However, there are some data points observed with different behavior than the trend.

In Figure 30, the data from the Fluid Dynamics Lab, including the data gathered for the present thesis, compared with the results obtained by Johann Nikuradse, with the sand grain roughened pipes, artificially. Different than the previous comparison case, Rz values are taken as the surface roughness parameters, since the definition used by Nikuradse fits with the Rz parameter.

![Darcy friction factor vs Rz/Dh](image)

**Figure 30: Darcy friction factor vs Rz/Dh.**

Except for 1 test sample, the other 5 test samples prepared by Nikuradse have lower roughness levels than the available roughness in the thesis work. Thus, it is hard to make a proper comparison, but still, it could be assumed that the data would follow the exponential trend with higher roughness levels.
6 Discussion

6.1 Test Rig Validation

The test rig, test procedure, and data processing methodology were validated with the smooth circular test object, namely Aluminum 2 [mm]. The obtained data for the smooth channel follows the theoretical calculations for both the Darcy friction factor and Nusselt number tests. The constant Darcy friction factor values were seen at the end of the Reynolds numbers interval in Figure 12 could be explained with the choked flow in this stage of the experiment. More precisely, in the experimental setup, the maximum upstream pressure is limited with the pressurized system, and on the downstream, pressure could be reduced by opening the pressure valve on the outlet. But in the mentioned stage, even with the reduction in the downstream pressure, mass flow can not be increased further. As introduced in the theory section, sudden contraction in the inlet region of the test channel changes the boundary layer behavior. Due to the separation and circulations in the mentioned part, the heating condition has a discontinuity. The mentioned effect results in higher Nusselt number values than the ones calculated from the Gnielinski correlation for smooth surfaces, and when the influence of the entrance effects on the Nusselt number values was taken into account, the validation test was successful (see Figure 13).

6.2 Darcy Friction Factor

In Figure 14, it is seen that for all AM test channels, there was an offset between the laminar theoretical line and obtained Darcy friction factor levels. Similar trends were seen in the study performed by Pennsylvania State University with different test materials.[11] [12] Theoretically, in the laminar regime, surface roughness should not have any influence on the Darcy friction factor (see Equation 8). The contrast between the theoretical values and calculated ones could be explained by the wrong estimation of the hydraulic diameters of the test objects. The hydraulic diameter has a tremendous influence on the results of the performed study. Microscopic pictures of the test objects were taken from the inlet and outlet of the test channels to identify the hydraulic diameters of each test object, and obtained values were used to calculate both Reynolds number and Darcy friction factor. However, there is no information about the surface features inside the channel for the entire flow path, and there could be unexpected features that reduce the available flow area for the fluid flow or disturb the flow and affect the flow field.

After the end of the laminar regime, boundary layer transition starts for each test object. At the start of the transition, Darcy friction factor values increase, and after some point, the values stick on a constant value. As a consequence of the reduced cross-sectional area, the flow velocity in the outlet of the test channel is higher than the velocity before the inlet of the channel. Thus, the boundary layer transition starts
from the end of the test channels, and flow firstly becomes turbulent at the end of the test object, while it still remains laminar on the previous parts of the objects. The explained transition behavior means that the flow inside the flow path is not always fully turbulent at the beginning of the turbulent regime, and with an increase in the mass flow rates, the start point of the turbulent flow inside the test channel moves from the outlet to inlet. Until the fully turbulent flow is reached, the Darcy friction factor values do not become stable in the turbulent regime. Additionally, dramatic drops in the Darcy friction factor at the end of the turbulent regime could be explained by the compressible effects. In the mentioned part, both outlet and inlet valves become almost fully open, Mach numbers increases, and flow becomes compressible. Mach numbers were calculated and monitored for all operation points to avoid compressible effects. However, compressibility effects were present even for the Mach numbers lower than the mentioned 0.3 value.

In Figure 14 for the objects STAL15 1.5 x 1.5 [mm] and 1.5 x 2 [mm] which have different hydraulic diameters, almost the same Dary friction factor values were available. When the results were considered with the surface roughness measurements given in Table 4, both test objects had similar relative roughness levels. Thus, even with different hydraulic diameters, it is possible to achieve similar Darcy friction factor values.

It was found that, for the same test materials, comparable surface roughness parameters are present. However, since the hydraulic diameters are not the same, relative roughness parameters, which were calculated by dividing the roughness values to measured hydraulic diameters of each test sample, differ. For the circular test objects, there is a clear relationship between the Darcy friction factor and relative roughness parameters. The trend is mostly applicable for the non-circular test objects also. But for the non-circular ones, irregularities were seen. Different flow fields inside the test objects due to the channel shapes may be the reason for the observed violations.

### 6.3 Nusselt Number

Normally, the Nusselt number values are higher for the rough surfaces. However, in the obtained results it was observed that, for the test objects with small diameters, lower values than the smooth surfaces for the Nusselt number were present in the low Reynolds numbers. The behavior could be due to the significantly low mass flow rates available inside the test channels, because of the small hydraulic diameter and high relative roughness levels. Low mass flows mean that forced convection was lower for the mentioned case.

With an increase in the Reynolds numbers, Nusselt enhancement factor values were increased. Afterward, the values were stuck on constant values for most of the test objects, except for the ones with smaller hydraulic diameters. The behavior could be explained by the choking effect inside the test channels. Additionally, for the objects which were able to achieve the constant Nusselt numbers, dramatic drops were present in the last points of the highest Reynolds numbers, which could also be
because of the choking effects.

From Table it could be possible to say increased relative roughness results with the increased Nusselt enhancement for the circular test objects. However, for the non-circular test object data given in Table, two different trends were present between the relative roughness parameter and Nusselt enhancements. It might be the possible reason that Nusselt number enhancement is not related to only one of the parameters shown in the Tables, and it has a more complex relationship, such as a combination of the given parameters. Thus, further data analysis and experimental studies needed to investigate the relationship and give more clear discussion about the behavior.

### 6.4 Surface Roughness Parameters and Darcy Friction Factor Trend Comparison with Other Studies

Comparison of the results from different sources given in Figures 29 and 30 was important to validate the found general trend. While most data points follow the trend, there were some points with different behavior. Measurement errors, such as wrong determination of the hydraulic diameters or surface roughness parameters, or problems during the experiments could be the reason for the exceptional data points. Overall, the comparisons show the relationship between the Darcy friction factor and relative surface parameters accurately and could be used to obtain the mathematical equations for further modeling purposes of the surface roughness effects on the pressure drops.
7 Conclusions

Experimental studies were done with the additively manufactured STAL15 test objects with varying diameters and channel shapes to investigate the cooling performance of the materials. Due to the inherent surface roughness present in the AM parts, higher pressure drops and heat transfer were observed compared to the smooth surfaces. For the cooling performance, higher heat transfer rates (Nusselt numbers) and lower pressure losses are preferred, since higher heat transfer would ensure lower temperatures on the components, but higher pressure losses would cause losses on the acquired energy from the gas turbine components, or require additional energy input to maintain the pressure levels stable on the system.

Outcomes of the study could be concluded as followings;

(i) For the higher relative roughness, higher Darcy friction factor values were present. The trend was always applicable for the circular test objects, for the non-circular ones, because of the different flow fields for the non-circular object, there were some violations, but still, it was applicable for most of the cases.

(ii) Darcy friction factor values were increased exponentially with an increase in the surface roughness parameters.

(iii) Higher Darcy values were observed for the non-circular test objects, compared to the circular ones with similar hydraulic diameters.

(iv) From the Nusselt enhancement factor point of view, similar to the Darcy case, higher enhancement factors were present for the higher relative roughness levels, for the circular test channels. On the other hand, the outcome was violated by the non-circular test objects.

(v) Lower heat transfer enhancements were achieved with the square test objects, compared to the circular ones with similar hydraulic diameters.
8 Perspectives

To ensure higher roughness modeling accuracy and more information about the cooling performance, further experimental investigation with more test material would be needed. Obvious trends were achieved with the surface roughness parameters and Darcy friction factor. However, it was not possible to conclude such a trend for the Nusselt number. Further data analysis or experimental investigations might help to find out the mathematical relationships between the surface roughness parameters and Nusselt numbers.

Dimensions of the flow channel were estimated from the microscopic pictures taken from the inlet and outlet of each test sample. The methodology is open for errors and hydraulic diameter has a tremendous effect on the obtained values. A more accurate methodology to detect the hydraulic diameters for the entire flow path could help to reduce the possible errors in the study.

Since the non-circular test samples had sharp corners, another investigation could be the effect of the round corners on the STAL15 test objects. The print direction could affect the surface roughness and flow behavior, and in the present study test objects were printed perpendicular to the ground. Thus, it could be interesting to perform experiments with the objects printed with angles.

For the Nusselt number tests, it was not able to achieve the constant value in the high Reynolds numbers for the test samples with smaller diameters due to the choking effects. The pressurized system in the test setup was able to provide 8 bars, to overcome the choking and be able to get the constant data on the mentioned region, higher pressures for the pressurized system could be considered.

Flow visualization inside the test channels would help to understand the effect of the channel shape and surface features on the flow field.
References


A Appendix

The given Python codes in this chapter were developed by the scientists of Siemens Energy in Finspång, Sweden. The codes were used on the previous studies and may be used for future studies. The author of the present thesis adjusted the codes for the current thesis work.

```python
# import math, sys
import matplotlib.pyplot as plt
import numpy as np

# geometrical data
D = [Value] # hydraulic diameter [m]
A = [Value] # cross section flow area [m^2]
Pi = [Value] # inner perimeter [m]
L = [Value] # length of the test object [m]

OutputFile="output_file_name.txt"

# Tin (inlet air temperature), Tout (outlet air temperature) = C [from PT100 sensors]
# mdot (mass flow rate) = g/s (from Coriolis)
# Pin (inlet pressure), Pout (outlet pressure), InOut1 (differential pressure), InOut5 (differential pressure) = kPa (from psi 9116 pressure scanner)
# Patm (atmospheric pressure) = Pa (from Rosemount coplanar pressure transmitter)

solveCases = [
    {"Tin": [Value], "Tout": [Value], "mdot": [Value], "Pin": [Value], "Pout": [Value], "Patm": [Value], "InOut1": [Value], "InOut5": [Value]} ,
    {"Tin": [Value], "Tout": [Value], "mdot": [Value], "Pin": [Value], "Pout": [Value], "Patm": [Value], "InOut1": [Value], "InOut5": [Value]} ,
    {"Tin": [Value], "Tout": [Value], "mdot": [Value], "Pin": [Value], "Pout": [Value], "Patm": [Value], "InOut1": [Value], "InOut5": [Value]} ,
    {"Tin": [Value], "Tout": [Value], "mdot": [Value], "Pin": [Value], "Pout": [Value], "Patm": [Value], "InOut1": [Value], "InOut5": [Value]} ,
]

plot = 0

def mu(T): #Sutherland's Law
    mu0 = 1.716e-5
    T0=273.11
    S=110.56
    return mu0*math.pow(T/T0, 1.5) * (T0+S)/(T+S)

def rho(T,p): # T in Kelvin
    R=8314/28.97
    return p/(R*T)
```
for ii, caseinfo in enumerate(solveCases):
    Tinlet = caseinfo["Tin"]+273.15
    Toutlet = caseinfo["Tout"]+273.15
    mdot = caseinfo["mdot"]/1000
    Pinlet = caseinfo["Pin"]*1000 + caseinfo["Patm"]
    Poutlet = caseinfo["Pout"]*1000 + caseinfo["Patm"]
    IO1 = caseinfo["InOut1"]*1000
    IO5 = caseinfo["InOut5"]*1000
    print("Mass flow: {} g/s".format(mdot*1000))
    print("Inlet temperature: {}°C".format(Tinlet-273.15))
    print("Outlet temperature: {}°C".format(Toutlet-273.15))
    print("Inlet pressure: {} Pa".format(Pinlet))
    print("Outlet pressure: {} Pa".format(Poutlet))
    print("dp_read: {} Pa".format(Pinlet-Poutlet))
    print("dp1_read: {} Pa".format(IO1))
    print("dp5_read: {} Pa".format(IO5))
    
    Re = mdot * D / (mu(Tinlet) * A)
    ReWrite.append(Re)
    
    if Re <2500:
        inloss = 0.5
        outloss = 2
    else:
        inloss = 0.5
        outloss = 1
    
    Ubulk_inlet = mdot/(rho(Tinlet, Pinlet) * A)
    Ubulk_outlet = mdot/(rho(Toutlet, Poutlet) * A)
    velheadIn = Ubulk_inlet**2*rho(Tinlet, Pinlet)/2
    velheadOut = Ubulk_outlet**2*rho(Toutlet, Poutlet)/2
    dp = Pinlet-Poutlet - outloss*velheadOut - inloss*velheadIn
    dp1 = IO1 - outloss*velheadOut - inloss*velheadIn
    dp5 = IO5 - outloss*velheadOut - inloss*velheadIn
    Mach = Ubulk_outlet / 343
    MachWrite.append(Mach)
    
    print("dp: {} Pa".format(dp))
    print("dp1: {} Pa".format(dp1))
    print("dp5: {} Pa".format(dp5))
    print("Darcy friction factor: {}".format(dp*D/L / (0.5*(velheadIn+velheadOut))))
    print("Darcy friction factor dp1: {}".format(dp1*D/L / (0.5*(velheadIn+velheadOut))))
print("Darcy friction factor dp5: {}".format(dp5*D/L / (0.5*(velheadIn+velheadOut))))

dpWrite.append(dp*D/L / (0.5*(velheadIn+velheadOut)))
dp1Write.append(dp1*D/L / (0.5*(velheadIn+velheadOut)))
dp5Write.append(dp5*D/L / (0.5*(velheadIn+velheadOut)))

with open(OutputFile,"w") as outFile:
    outFile.write("Reynolds
")
    for Re in ReWrite:
        outFile.write("{}
".format(Re))
    outFile.write("Darcy
")
    for dpn in dpWrite:
        outFile.write("{}
".format(dpn))
    outFile.write("Darcy dp1
")
    for dp1n in dp1Write:
        outFile.write("{}
".format(dp1n))
    outFile.write("Darcy dp5
")
    for dp5n in dp5Write:
        outFile.write("{}
".format(dp5n))
    outFile.write("Mach
")
    for Mach in MachWrite:
        outFile.write("{}
".format(Mach))

#

Code Listing 1: Darcy friction factor evaluation Python code.
```python
# geometrical data
D = [Value] # hydraulic diameter [m]
A = [Value] # cross section flow area [m^2]
Pi = [Value] # inner perimeter [m]
L = [Value] # length of the test object [m]

OutputFile="output_file_name.txt"

# Tin (inlet air temperature), Tout (outlet air temperature), Tcu (copper temperature) = C from PT100 sensors
# mdot (mass flow rate) = g/s (from Coriolis)
# Pin (inlet pressure), Pout (outlet pressure) = Pa (from psi 9116 pressure scanner)

hcontact = 1e4 #8.5/0.02e-3 #1e5 #term pasta gap 0.02 [mm] MX-4: 8.5 [W/m K]
kmet = [Value] # thermal conductivity of the test object [W/mK]
Do = [Value] # qSSHT rig test object outer diameter [m]
solveCases = [
    
    "Tin": [Value],
    "Tout": [Value],
    "Tcu": [Value],
    "mdot": [Value],
    "Pin": [Value],
    "Pout": [Value],
]

Cp = 1005. # [J/kgK]
plot = 0
def mu(T): #Sutherland's Law
    mu0 = 1.716e-5
    T0=273.11
    S=110.56
    return mu0*math.pow(T/T0, 1.5) * (T0+S)/(T+S)

def rho(T, p): # T in Kelvin
    R=8314/28.97
    return p/(R*T)

def k(T): # T in Kelvin
    Cv = 720
    R=8314/28.97
    return mu(T)*Cv*(1.32 + 1.77*R/Cv)

def getheatflux(Ti, To, ri, ro, hi, ho, Pi, Po, kmet):
    return (Ti - To) / (1/(Pi*hi) + math.log(ro/ri)/(math.pi*2*kmet) + 1/(Po*ho)) # heat flux per meter pipe [W/m]
```
```python
def entranchEnhance(xrD):
    return 1+8.7/(xrD+5)

def gnielinski(Re, Pr):
    f = 0.3164/np.power(Re, 0.25)
    return f/8*(Re−1000)*Pr/(1+12.7*np.sqrt(f/8)*(np.power(Pr, 2/3.)−1))

def getOutletTemperature(Tcopper, hcontact, Tinlet, mdot, h):
    Po = math.pi*Do
    xvec = np.linspace(0, L, 100)
    Tgas = Tinlet
    dx = xvec[1]−xvec[0]
    Tvec = []
    TmetIn = []
    TmetOut = []

    for x in xvec:
        q = getheatflux(Tgas, Tcopper, D/2, Do/2, h, hcontact, Pi, Po, kmet)
        a = −q / (2*math.pi*kmet)
        b = Tgas − q/(Pi*h) # inner metal temperature
        Tgas = q/dx/(mdot*Cp)
        Tvec.append(Tgas−273.15)
        TmetIn.append(b−273.15)
        TmetOut.append(a*np.log(Do/D) + b −273.15)

    if plot:
        plt.plot(xvec, Tvec)
        plt.plot(xvec, TmetIn)
        plt.plot(xvec, TmetOut)
        plt.plot(xvec, np.ones(len(xvec))*(Tcopper−273.15), '−−')
        plt.xlabel("Length [m]")
        plt.ylabel("Gas temperature [C]")
        plt.show()
    return Tgas, np.average(TmetIn)+273.15 # outlet temperature

h = 1600
ReWrite=[]
NuWrite=[]
GnWrite=[]

for caseinfo in solveCases:
    Tinlet = caseinfo["Tin"]+273.15
    Toutlet = caseinfo["Tout"]+273.15
    Tcopper = caseinfo["Tcu"]+273.15
    mdot = caseinfo["mdot*"]/1000
    Pinlet = caseinfo["Pin*]
    Poutlet = caseinfo["Pout*"

    print("Mass flow: {0:.6f} kg/s".format(mdot))
    print("Inlet temperature: {0:.1f} K".format(Tinlet))
    print("Outlet temperature: {0:.1f} K".format(Toutlet))
```

print("Copper temperature: {0:.1f} K".format(Tcopper))
print("Inlet pressure: {0:.4f} Pa".format(Pinlet))
print("Outlet pressure: {0:.4f} Pa".format(Poutlet))

Tdiff = 1
print("Iterating to find htc...")
while np.abs(Tdiff) > .10:
    Toutest, Tmet = getOutletTemperature(Tcopper, hcontact, Tinlet, mdot, h)
    Tdiff = Toutlet - Toutest
    if Tdiff > 0:
        h *= 1.001
    else:
        h *= 0.999
getOutletTemperature(Tcopper, hcontact, Tinlet, mdot, h)

Re = mdot * D / (mu(Tinlet) * A)
print("Found HTC: {0:.2f} W/m2K".format(h))
print("HTC (LMTD): {}".format(mdot*Cp*(Toutlet-Tinlet)*np.log((Tmet-Tinlet)/(Tmet-Toutlet))*(L*Pi*(Toutlet-Tinlet))))
print("Nusselt number: {}".format(D*h/k(Tinlet)))
print("Re: {}".format(Re))
print("Nusselt Gneilinski: {}".format(gneilinski(Re, 0.71)))
ReWrite.append(Re)
NuWrite.append(D*h/k(Tinlet))
GnWrite.append(gneilinski(Re, 0.71))

### Friction factor

if Re <2500:
    inloss = 0.5
    outloss = 2
else:
    inloss = 0.5
    outloss = 1

Ubulk_inlet = mdot/(rho(Tinlet, Pinlet) * A)
Ubulk_outlet = mdot/(rho(Toutlet, Poutlet) * A)
velheadIn = Ubulk_inlet**2*rho(Tinlet, Pinlet)/2
velheadOut = Ubulk_outlet**2*rho(Toutlet, Poutlet)/2
dp = Pinlet-Poutlet - outloss*velheadOut - inloss*velheadIn
print("Darcy friction factor: {}".format(dp*D/L / (0.5*(velheadIn+velheadOut))))

with open(OutputFile, "w") as outfile:
    outfile.write("Reynolds
")
    for Re in ReWrite:
        outfile.write("{}
".format(Re))
    outfile.write("Nusselt
")
    for dpn in NuWrite:
        outfile.write("{}
".format(dpn))
    outfile.write("Gneilinski
")
    for dp1n in GnWrite:
        outfile.write("{}
".format(dp1n))

Code Listing 2: Nusselt number evaluation Python code.
import math, sys
import matplotlib.pyplot as plt
import numpy as np

# geometrical data
D = [Value] # hydraulic diameter [m]
A = [Value] # cross section flow area [m^2]
Pi = [Value] # inner perimeter [m]
L = [Value] # length of the test object [m]

measfile = "input_file_name.DIF"

skip_lines = 3 # no of lines to skip in indata file after header
hcontact = 1e4 #8.5/0.02e−3 #1e5 #term pasta gap 0.02 [mm] MX−4: 8.5 [W/m K]
kmet = [Value] # thermal conductivity of the test object [W/mK]
Do = [Value] # QSSH rig test object outer diameter [m]

Cp = 1005. # [J/kgK]
plot = 0
def mu(T): #Sutherland's Law
    mu0 = 1.716e−5
    T0=273.11
    S=110.56
    return mu0*math.pow(T/T0 , 1.5 ) *(T0+S)/(T+S)

def rho (T, p): # T in Kelvin
    R=8314/28.97
    return p/(R*mu(T))

def k (T) : # T in Kelvin
    Cv = 720
    R=8314/28.97
    return mu(T)*Cv*(1.32 + 1.77*R/Cv)

def getheatflux(Ti , To , ri , ro , hi , ho , Pi , Po , kmet ) :
    return ( Ti−To ) / (1/(Pi*hi ) + math.log(ri/ro )/(math.pi*2*kmet)
    + 1/(Po*ho ) )# W/m heat flux per meter pipe

def entranchEnhance(xrD) : # the accumulated increase of Nu
    return 1+8.7/(xrD+5)

def gnielinski(Re, Pr):
    f = 0.3164/np.power(Re, 0.25)
    return f/8*(Re−1000)*Pr/(1+12.7*np.sqrt(f/8)*(np.power(Pr, 2/3.))−1)

def getOutletTemperature(Tcopper, hcontact, Tinlet, mdot, h):
    Po = math.pi*Do
    xvec = np.linspace(0, L, 100)
    Tgas = Tinlet #
dx = xvec[1] - xvec[0]
Tvec = []
TmetIn = []
TmetOut = []

for x in xvec:
    q = getheatflux(Tgas, Tcopper, D/2, Do/2, h, hcontact, Pi, Po, kmet)
    a = -q / (2*math.pi*kmet)
    b = Tgas - q/(Pi*h)  # inner metal temperature
    Tgas -= q*dx/(mdot*Cp)
    Tvec.append(Tgas-273.15)
    TmetIn.append(b-273.15)
    TmetOut.append(a*np.log(Do/D) + b -273.15)

if plot:
    plt.plot(xvec,Tvec)
    plt.plot(xvec,TmetIn)
    plt.plot(xvec,TmetOut)
    plt.plot(xvec,np.ones(len(xvec))*(Tcopper-273.15),'-r')
    plt.xlabel("Length [m]")
    plt.ylabel("Gas temperature [C]")
    plt.show()

return Tgas, np.average(TmetIn)+273.15  # outlet temperature

measdata = {}
labels = []
timepoint = 0
with open(measfile) as f:
    for id, line in enumerate(f):
        if id == 0:
            continue
        if id == 1:
            for stuff in line.split("t"):  
                measdata[stuff.strip()] = {}  
                measdata[stuff.strip()]['data'] = []  
                labels.append(stuff.strip())
            continue
        if id == 2:
            for id, stuff in enumerate(line.split("t")):
                measdata[labels[id]]['unit'] = stuff
            continue
        if id == 3:
            for id, stuff in enumerate(line.split("t")):
                measdata[labels[id]]['info'] = stuff
            continue
        if id < skiplines:
            continue
        for id, stuff in enumerate(line.split("t")):
            try:
                getfloat = float(stuff)
            except:
                getfloat = stuff
            measdata[labels[id]]['data'].append(getfloat)
timepoint += 1

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if False:
    for entry in measdata:
        try:
            plt.plot(measdata[entry]['data'])
            plt.title(measdata[entry]['info'])
            plt.show()
        except:
            pass

h = 800
h_hist = []
Nu = []
NuNu0 = []
Re = []
h_LMTD = []
if True:
    for i in range(timepoint):
        Tcopper = 0.5*(measdata['PT−100 3']['data'][i]+measdata['PT−100 4']['data'][i]) + 273.15
        Tinlet = measdata['PT−100 1']['data'][i] + 273.15
        Toutlet = measdata['PT−100 2']['data'][i] + 273.15
        mdot = measdata['Coriolis 10']['data'][i]/1000.
        Tdiff = 1
        while np.abs(Tdiff) > .10:
            Toutest, Tmet = getOutletTemperature(Tcopper, hcontact, Tinlet, mdot, h)
            Tdiff = Toutlet - Toutest
            if Tdiff > 0:
                h *= 1.001
            else:
                h *= 0.999
                getOutletTemperature(Tcopper, hcontact, Tinlet, mdot, h)
        h_hist.append(h)
        h_LMTD.append(mdot*Cp*(Toutlet−Tinlet)*np.log((Tcopper−Tinlet)/(Tcopper−Toutlet))/(L*Pi*(Toutlet−Tinlet))))
        Nu.append(D*h/k(Tinlet))
        Re.append(mdot * D / (mu(Tinlet) * A ))
        NuNu0.append(D*h/k(Tinlet)/gnielinski(mdot * D / (mu(Tinlet) * A ), 0.71))
    plot = 1

plt.figure(1)
plt.plot(Nu)
plt.xlabel("t [s]")
plt.ylabel("Nu")
plt.grid()
plt.figure(2)
plt.plot(Re)
plt.xlabel("t [s]")
plt.ylabel("Re")
plt.grid()
Remin = \max(3000, \ \text{min}(Re))
Remax = \text{max}(Re)
Rerange = \text{np.linspace}(Remin, Remax, 100)
plt.figure(3)
plt.plot(Re, Nu, '. ')
plt.plot(Rerange, [gnielinski(Re_i, 0.71) for Re_i in Rerange])
plt.xlabel("Re")
plt.ylabel("Nu")

plt.figure(4)
plt.plot(Nu / Nu0)
plt.xlabel("t\ [s]\)"
plt.ylabel("Nu/Nu0")
plt.grid()
plt.show()

### Friction factor
if False:
    f_darcy1 = []
f_darcy2 = []
Re = []

for i in range(timepoint):
    P0 = measdata["Patm"["data"]][i] # atmospheric pressure in Pa
    Tinlet = measdata["PT−100 1"["data"]][i] + 273.15
    Toutlet = measdata["PT−100 2"["data"]][i] + 273.15
    mdot = measdata["Coriolis 10"["data"]][i]/1000.
    P4in = measdata["P16"["data"]][i]*1000 + P0 # meas pressure in point 4
    P4out = measdata["P14"["data"]][i]*1000 + P0 # meas pressure in point 4
    Re_i = mdot * D / (mu(Tinlet) * A)
    Re.append(Re_i)
    dp_5 = measdata["In5−Ut5"["data"]][i]
    if Re_i < 2500:
        inloss = 0.5
        outloss = 2
    else:
        inloss = 0.5
        outloss = 1
    Ubulk_inlet = mdot/(rho(Tinlet, P4in) * A)
    Ubulk_outlet = mdot/(rho(Toutlet, P4out) * A)
    velheadIn = Ubulk_inlet**2*rho(Tinlet, P4in)/2
    velheadOut = Ubulk_outlet**2*rho(Toutlet, P4out)/2
    dp_channel = P4in – P4out + outloss*velheadOut – inloss*velheadIn
    dp1.append(dp_channel)
dp2 . append ( dp _ channel _ alt )

f _ darcy1 . append ( dp _ channel + D / TOlength / ( 0.5 * ( velheadIn +
velheadOut )))

f _ darcy2 . append ( dp _ channel _ alt + D / TOlength / ( 0.5 * ( velheadIn +
velheadOut )))

fig , ax = plt . subplots ()

ax . loglog ( Re , f _ darcy1 , '. ' )
ax . loglog ( Re , f _ darcy2 , '. ' )
ax . loglog ( np . linspace ( 100 , 5000 , 200 ) / np . linspace ( 100 , 5000 , 200 ) )
ax . loglog ( np . linspace ( 2300 , 50000 , 200 ) , [ 0.3164 / np . power ( Re i ,
0.25 ) for Re i in np . linspace ( 2300 , 50000 , 200 ) ] )

plt . show ()
plt . plot ( Re , dp1 , '. ' )
plt . plot ( Re , dp2 , '. ' )
plt . show ()

Code Listing 3: Nusselt number stabilization evaluation Python code.