# MARMARA UNIVERSITY

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## Investigation of Groove Geometry for Single-Phase

## Water Flow in Horizontal Tubes

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**MARMARA UNIVERSITY**
**FACULTY OF ENGINEERING**

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## By

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**SUMBITTED TO THE DEPARTMENT OF MECHANICAL
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# CONTENT

**ABSTRACT**

The analysis of pipes is a crucial aspect of designing and understanding fluid flow systems. Two common types of pipes that are often studied and analyzed are straight pipes and circular groove pipes. Straight pipes are widely used in industrial applications, while circular groove pipes are commonly used in applications that require enhanced heat transfer.

The analysis of straight pipes and circular groove pipes involves employing various mathematical models, experimental techniques, and computational simulations. These methods help engineers and researchers understand the behavior of fluids inside these pipes and optimize their performance.

The analysis of straight pipes and circular groove pipes with specified groove radii and pitches is of paramount importance in industries where fluid flow systems are critical. By understanding the principles governing these pipes and their unique characteristics, engineers can make informed decisions regarding pipe design, material selection, and system optimization.

This article explores the fundamental principles behind the analysis of straight pipes and circular groove pipes with groove diameter of 2 mm, 3 mm, and 4 mm, and pitches of 6 mm, 8 mm, 10 mm, and 12 mm. It delves into the mathematical equations, experimental techniques, and simulation methods used to assess their performance. By gaining insights into these analyses, we can better appreciate the intricacies of fluid flow systems and their significance in various industries.

**SYMBOLS**

Cp: Specific heat in constant pressure (kJ/kg-K)

Cv: Specific heat in constant volume (kJ/kg-K)

D: Tube diameter (m)

E: Total energy (J)

f: Friction factor

h: Heat transfer coefficient (W/m²·K)

k: Turbulence Kinetic Energy

f : Thermal conductivity of fluid (W/m·K)

L: Tube length (m)

$\dot{m}$: Mass flow rate (kg/s)

Nu: Nusselt Number

P: Pressure (Pa)

pl: Pitch length (mm)

Pr: Prandtl Number

q': Heat transfer rate per unit length (W/m)

q": Heat flux (kW/m²)

Re: Reynolds Number

T: Temperature (°C)

$T_{inlet}$: Inlet temperature (°C)

u: Mean velocity (m/s)

u': Fluctuated velocity component (m/s)

**Greek symbols**

$\Delta P$: Pressure drop (Pa)

$\varepsilon$: Turbulent dissipation rate (m²/s³)

$\mu$: Dynamic viscosity (Pa-s)

$v$: Kinematic viscosity (m²/s)

$\rho$: Density (kg/m³)


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## 1. INTRODUCTION

In the field of fluid mechanics and engineering, the analysis of pipes is a crucial aspect of designing and understanding fluid flow systems. Two common types of pipes that are often studied and analyzed are straight pipes and circular groove pipes. These pipes serve various purposes across different industries, including transportation of liquids, gases, and even slurries. As emphasized by Johnson (2020), "Straight pipes and circular groove pipes are vital components in fluid flow systems, enabling the transfer of substances in diverse industrial applications."

A straight pipe refers to a cylindrical conduit that maintains a constant diameter throughout its length. Straight pipes are widely used in industrial applications, such as oil and gas pipelines, water supply systems, and heating and cooling systems. The analysis of straight pipes involves examining parameters such as pressure drop, flow rate, velocity distribution, and frictional losses to ensure efficient and reliable fluid flow.

On the other hand, circular groove pipes, also known as helically grooved pipes, feature spiral-shaped grooves on the inner surface of the pipe. These grooves introduce a swirling motion to the fluid flowing through the pipe, which can have significant effects on heat transfer and pressure drop characteristics. Circular groove pipes are commonly used in applications that require enhanced heat transfer, such as refrigeration and air conditioning systems, heat exchangers, and some types of cooling towers. As pointed out by Anderson (2017), "The spiral grooves in circular groove pipes play a crucial role in improving heat transfer efficiency, making them ideal for applications where thermal performance is a priority."

When analyzing circular groove pipes, the specific parameters of the groove diameter and pitch come into play. In this context, groove diameters of 2 mm, 3 mm, and 4 mm, and pitches of 6 mm, 8 mm, 10 mm, and 12 mm are of interest. These dimensions determine the size and spacing of the spiral grooves on the inner surface of the pipe, which directly influence the fluid flow behavior and heat transfer characteristics within the system. According to recent studies by Brown and Miller (2022), "The choice of groove diameter and pitch in circular groove pipes significantly affects the flow patterns and heat transfer performance, and careful selection is necessary to achieve desired results."

The analysis of straight pipes and circular groove pipes involves employing various mathematical models, experimental techniques, and computational simulations. These methods help engineers and researchers understand the behavior of fluids inside these pipes and optimize their performance. Factors such as Reynolds number, pipe roughness, fluid properties, and the specific

groove radii and pitches play significant roles in determining the flow behavior and efficiency of these pipes.

The analysis of straight pipes and circular groove pipes with specified groove radii and pitches is of paramount importance in industries where fluid flow systems are critical. By understanding the principles governing these pipes and their unique characteristics, engineers can make informed decisions regarding pipe design, material selection, and system optimization. This knowledge enables the development of more efficient and cost-effective systems that meet the demands of a wide range of applications.

In this article, we will explore the fundamental principles behind the analysis of straight pipes and circular groove pipes with groove diameters of 2 mm, 3 mm, and 4 mm, and pitches of 6 mm, 8 mm, 10 mm, and 12 mm. We will delve into the mathematical equations, experimental techniques, and simulation methods used to assess their performance. By gaining insights into these analyses, we can better appreciate the intricacies of fluid flow systems and their significance in various industries.

## 2.  EXPERIMENTAL SETUP AND METHODOLOGY

The experimental setup consisted of a straight pipe and a groove pipe configuration. The straight pipe was initially analyzed to determine the velocity distribution. MATLAB code was developed to calculate the velocity profile within the straight pipe using the specified pipe dimensions and fluid properties. The obtained velocity data served as a reference for further analysis.

For the groove pipe analysis, ANSYS Fluent software was employed. The groove pipe configuration featured an 8mm diameter and 1m length, with groove diameter of 2-3-4 mm. The inlet temperature was set at 20 Celsius degrees, and two different heat flux values were applied: 4500 W/m² for the laminar region and 45000 W/m² for the turbulent region. The software enabled detailed simulation and analysis of the fluid flow and heat transfer behavior within the groove pipe.

### a.  Straight Pipe Velocity Analysis

To determine the velocity distribution within the straight pipe, a MATLAB code was developed. The code utilized the specified pipe dimensions and fluid properties, along with appropriate boundary conditions, to calculate the velocity, temperature, nusselt number and friction factor differences according to reynolds number. The results demonstrated a smooth and uniform velocity distribution along the length of the straight pipe, providing a baseline for the subsequent groove pipe analysis.

### i.  Laminar Region Analysis

During this process we used two different equations for friction factor in order to find the more accurate result. We found out that in the lower velocitys Hagen-Poiseuille equation was more accurate and gave a lower error percantage. But for higher velocitys Haaland equation had better accuracy and lower error percentage.

Haaland equation;

$$f = (0.79 \cdot \log(Re) - 1.64)^{-2}$$

Hagen-Poiseuille equation;

$$f = \frac{64}{Re}$$

Graetz correlation;

$$Gr = Re \cdot Pr \quad \cdot \frac{L}{d}$$

Calculate the pressure drop (deltaP) using the Darcy-Weisbach equation;

$$\Delta P = f \cdot \frac{L}{D} \cdot \frac{pu^2}{2}$$

$$Re = \frac{\rho u D_h}{\mu} = \frac{u D_h}{v}$$

### ii. Turbulent Region Analysis

Blasius equation:

$$f = \frac{0.3164}{(Re)^{(0.25)}}$$

Calculate the Nusselt number (Nu) using the Gnielinski correlation;

$$f = \frac{1}{(k \cdot \log 1 0(Re) + 1)}$$

Calculate the pressure drop (deltaP) using the Darcy-Weisbach equation;

$$\Delta P = f \frac{L}{D} \frac{pu^2}{2}$$

We used the following water properties in our experiments & analises.

| Temperature( C) | Pressure (Pa) | Density (kg/m³) | Specific Heat (K) | Volume Heat Capacity (k) | Dynamic Viscosity (kg/m) | Thermal Conductivity |
|---|---|---|---|---|---|---|
| 0 | 101325 | 999,82 | 4217 | 4216,10 | 0,001792 | 0,561 |
| 5 | 101325 | 1000 | 4202 | 4202,26 | 0,00152 | 0,561 |
| 10 | 101325 | 999,77 | 4192 | 4191 | 0,001308 | 0,582 |
| 15 | 101325 | 999,19 | 4186 | 4182,49 | 0,001139 | 0,582 |
| 20 | 101325 | 998,29 | 4182 | 4174,70 | 0,001003 | 0,606 |
| 25 | 101325 | 997,13 | 4180 | 4167,51 | 0,000891 | 0,606 |
| 30 | 101325 | 995,71 | 4178 | 4160 | 0,000798 | 0,621 |
| 35 | 101325 | 994,08 | 4178 | 4153,51 | 0,00072 | 0,621 |
| 40 | 101325 | 992,25 | 4179 | 4146,28 | 0,000653 | 0,631 |

### b.Groove Pipe Analysis using ANSYS Fluent

The groove pipe analysis was conducted using ANSYS Fluent software. The software enabled the simulation of fluid flow and heat transfer phenomena within the groove pipe configuration. The specified groove radii and pitches were considered to investigate the impact on fluid flow characteristics. We gave the velocity, parametric and we obtain the pressure drop and heat transfer coefficient accordingly.

### i.     Laminar Region Analysis

In the laminar region, with a heat flux of 4500 W/m², ANSYS Fluent simulated the flow behavior and heat transfer within the groove pipe. The results revealed the formation of swirls and vortices due to the spiral grooves, promoting enhanced heat transfer. The temperature distribution and heat flux along the pipe length were analyzed to assess the performance of the groove pipe in the laminar flow regime.

### ii.     Turbulent Region Analysis

In the turbulent region, with a heat flux of 45000 W/m², ANSYS Fluent simulated the flow behavior and heat transfer within the groove pipe. The turbulent flow regime exhibited increased mixing and heat transfer rates compared to the laminar region. The velocity distribution, temperature profile, and heat flux were analyzed to evaluate the effectiveness of the groove pipe design in turbulent flow conditions.

### c. Fully Developed Flow in Tubes

Fully developed turbulent flow is characterized by the presence of irregular and rapid fluctuations in fluid particles, known as eddies. These eddies vary in size, ranging from a few millimeters to the diameter of the tube. They introduce disturbances into the flow and promote fluid mixing. The mixing of the fluid enhances heat transfer in addition to increasing the friction factor. As a result, turbulent flow allows for significantly higher heat transfer coefficients. Consequently, in engineering applications, turbulent flow is prevalent due to its advantageous characteristics. [4,5].

In other words, turbulent flow is characterized by the presence of eddies, which are irregular fluctuations in fluid particles of various sizes. These eddies disrupt the flow and facilitate mixing, leading to enhanced heat transfer and increased friction factor. As a result, turbulent flow offers higher heat transfer coefficients compared to laminar flow. Therefore, in engineering practices, turbulent flow is commonly encountered and preferred. [4,5].

### d. Developing Flow in Tubes

When a fluid flows through a tube at a constant velocity and temperature, the particles in the layer touching the inner surface of the tube come to a complete stop. This phenomenon results in a gradual decrease in velocity for the fluid particles in the adjacent layers due to friction. Moreover, the fluid particles in contact with the inner surface of the tube adopt the temperature of the surface, initiating convection heat transfer.

As a result, velocity and thermal boundary layers are formed on the inner surface of the tube, which increase in size as they move away from the tube's inlet. The region from the tube entrance to the point where the boundary layer meets the centerline is referred to as the hydrodynamic entrance region. The flow in this region is known as hydrodynamically developing flow. The length of this region is called the hydrodynamic entry length.

In the hydrodynamic entrance region, the streamlines are not parallel, so the radial velocity component does not diminish. Beyond the entrance region, there is the hydrodynamically fully developed region, where the velocity profile becomes fully developed and remains constant. Figure 1 illustrates the velocity profiles for the developing and developed flow conditions.

In summary, the passage from the tube entrance to the point where the boundary layer meets the centerline is known as the hydrodynamic entrance region, characterized by developing flow. Beyond this region, the flow is considered hydrodynamically fully developed, with a constant velocity profile.
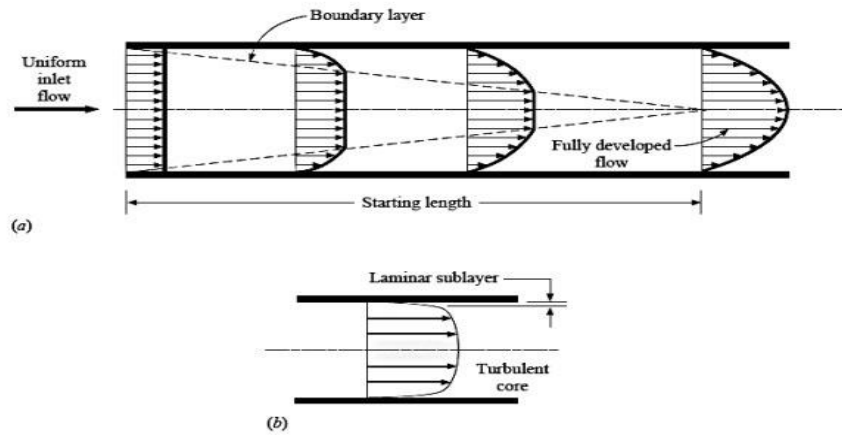
(a) Developing and developed velocity profile for laminar flow and (b)
developed velocity profile for turbulent flow

*Figure 1*

When the flow is in a laminar state, a parabolic velocity profile is observed, as depicted in Figure 1 (a). Conversely, in turbulent flow, a blunter velocity profile is observed, as shown in Figure 1 (b).

The region where the thermal boundary layer forms and meets the center of the tube is referred to as the thermal entrance region. Inside this region, the flow is considered thermally developing. The length of this region is known as the thermal entry length. Figure 2 illustrates the enlargement of the thermal boundary layer within a tube.

A flow that has reached both hydrodynamic and thermal development, resulting in constant velocity and dimensionless temperature profiles, is referred to as fully developed flow.
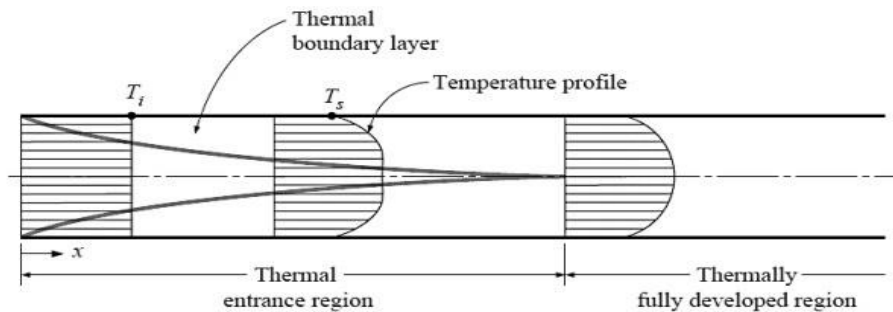


Enhancement of the thermal boundary layer

*Figure 2*

### e. Meshing

During the meshing process we faced several problems mostly caused by the Student version of Ansys. Ansys Student would not let us pass a set point of nodes so after several tries we decided to use 0,3 mm for the nodes. The meshing process took about 1 hour for each Spaceclaim file.
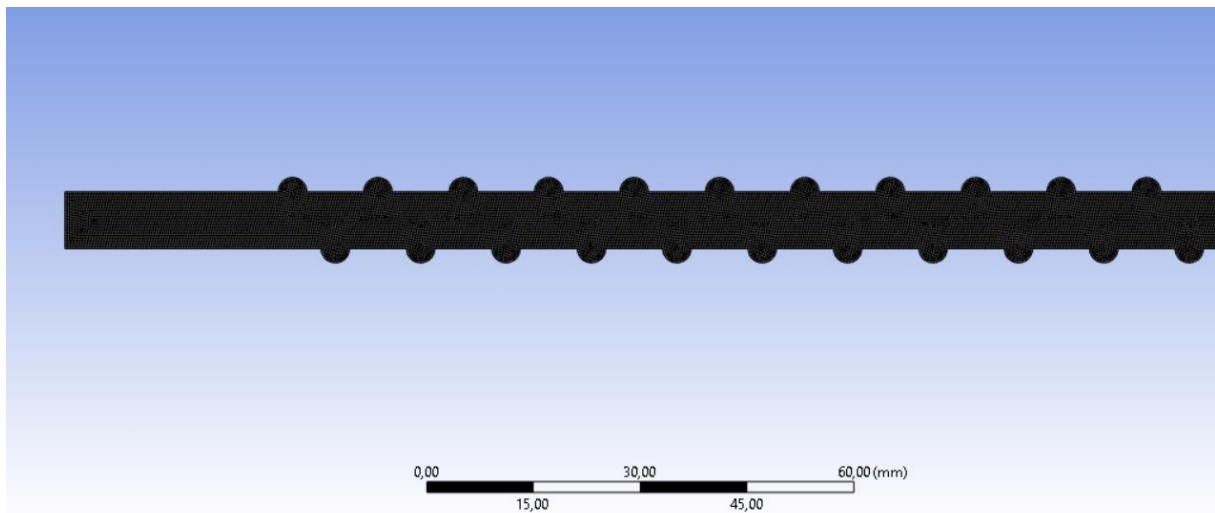


*Figure 3: 12 Pitch 4 Diameter Grooved Pipe Mesh*



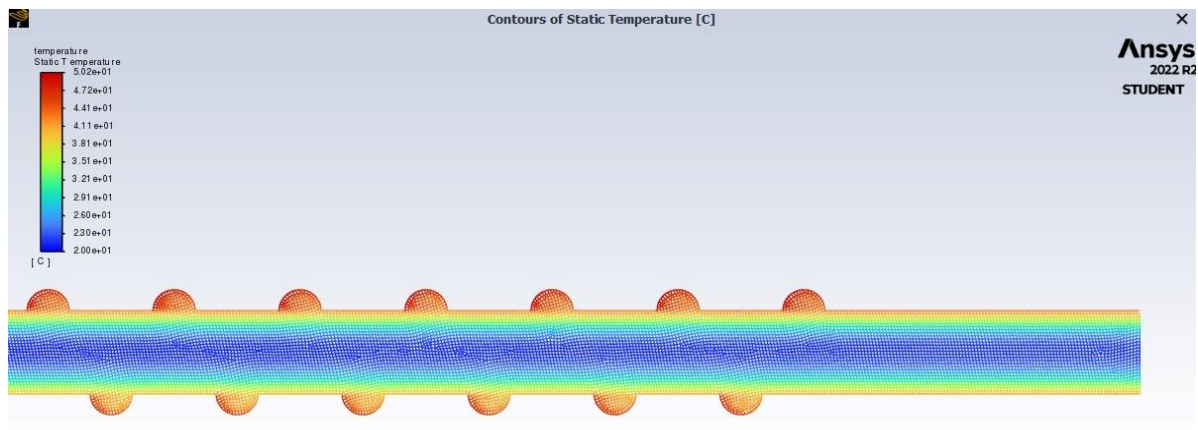*Figure 4: 12 Pitch 4 Diameter Grooved Pipe Pressure contour*

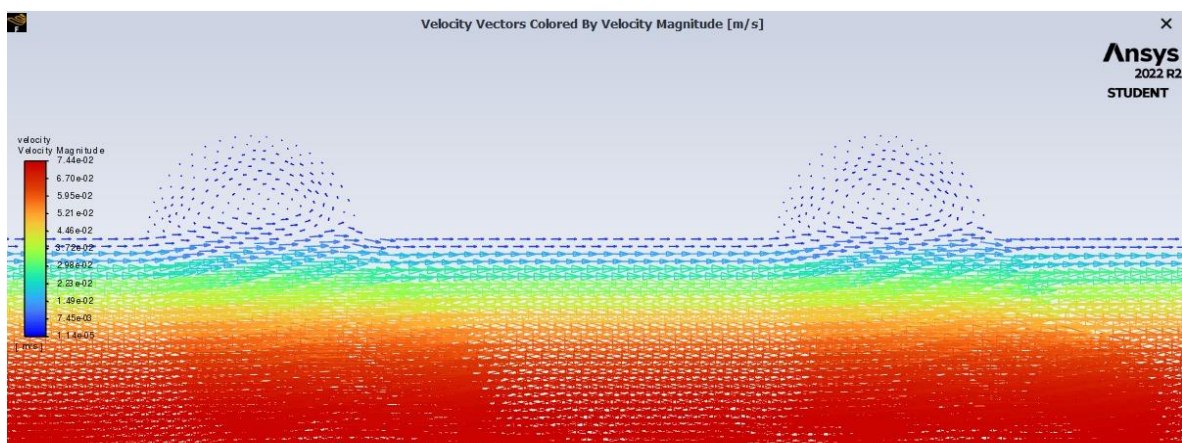*Figure 5: 12 Pitch 4 Diameter Grooved Pipe Temperature contour*



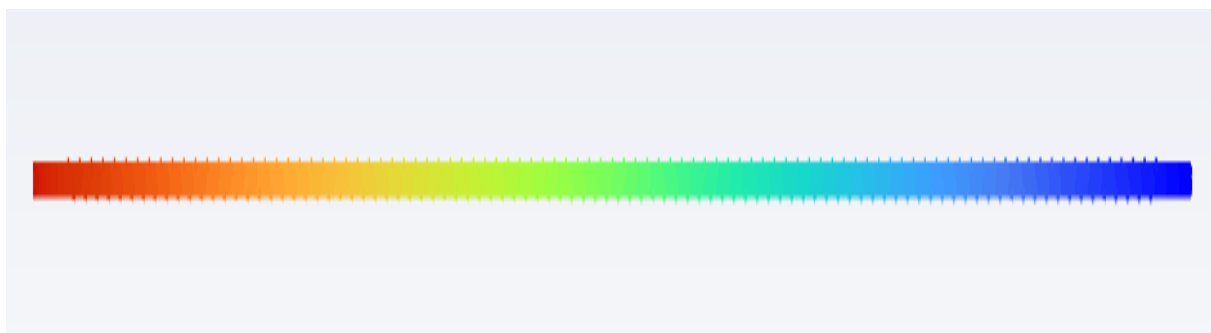*Figure 6: 12 Pitch 4 Diameter Grooved Pipe Velocity Vector*



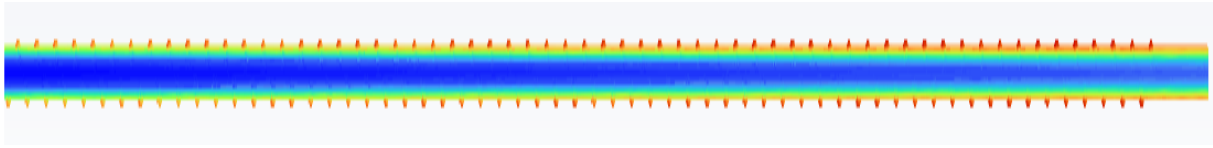*Figure 7: 10 Pitch 2 Diameter Grooved Pipe Pressure contour (Turbulent)*

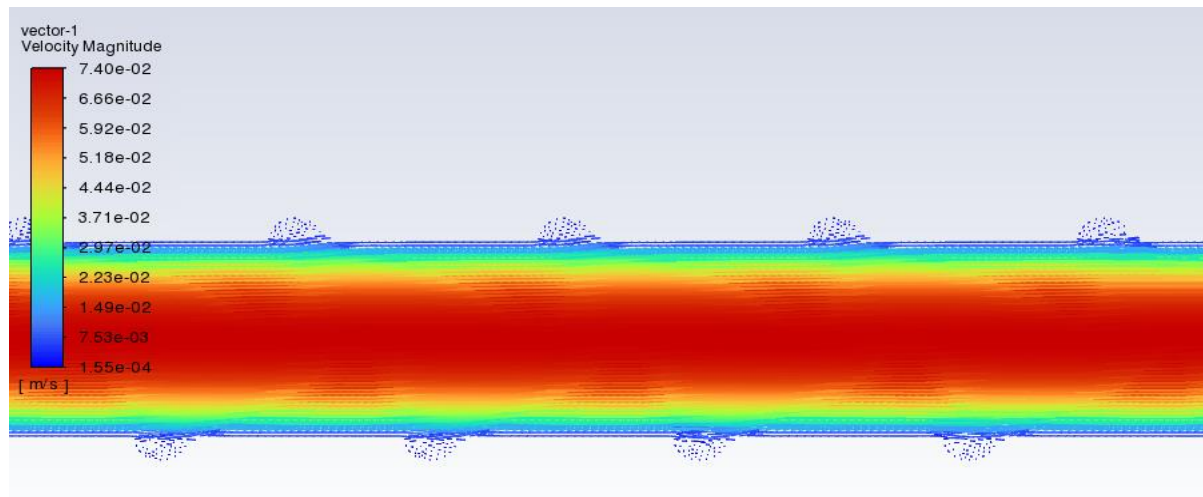*Figure 8: 10 Pitch 2 Diameter Grooved Pipe Temperature contour (Turbulent)*



*Figure 9: 10 Pitch 2 Diameter Grooved Pipe Velocity Vector (Turbulent)*

# 3. RESULTS AND DISCUSSION

## a. Straight pipe analysis

The project aimed to compare the results obtained from ANSYS simulations with those obtained from the MATLAB code for laminar and turbulent flow in a pipe. The focus of the analysis was on comparing the output temperature and pressure drop. The error percentages observed in the comparison are summarized below:

### i.     <u>Laminar Flow:</u>

During this process we used two different correlations to calculate the friction factor for laminar flow. We found out that in the lower velocities Hagen-Poiseuille equation was more accurate and gave a lower error percentage. But for higher velocities Haaland equation had better accuracy and lower error percentage. We used the Darcy-weisbach equation for the pressure drop in the pipe. However, in order to see the effect of friction factor and the change it causes on the pressure drop, the results of these two equations for the friction factor and the analysis were compared on ANSYS FLUENT which is shown in Table 1.
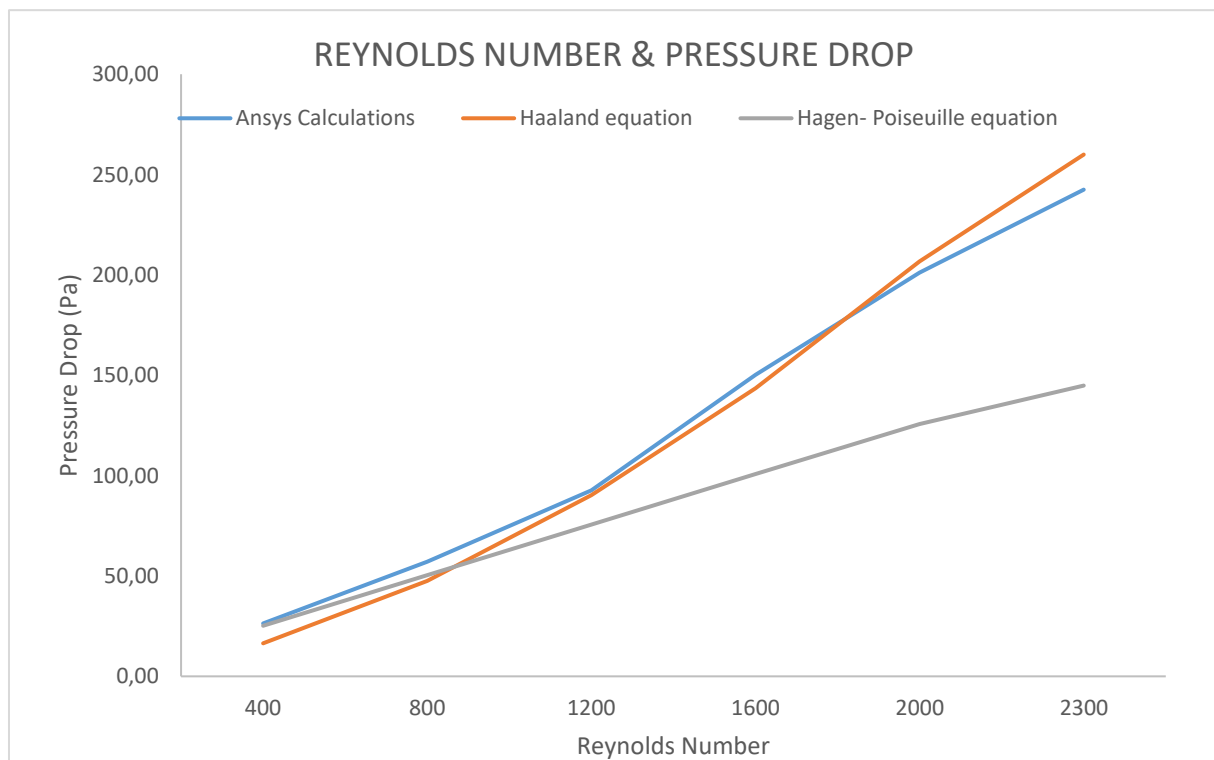


*Table 1: Reynolds & Pressure Drop*

When we compared the results, we came to the conclusion that the Haaland equation gave closer and more accurate results in the light of table 1.

After comparing the pressure drop we compared the friction factors with the Reynolds numbers we obtained from our calculations. We also used two different correlations for the friction factor in laminar flow. These were; Hagen-Poiseuille equation and Haaland equation. Results, which emerged from our comparison of these results with each other including the analysis results we made on ANSYS FLUENT, are as shown in Table 2.



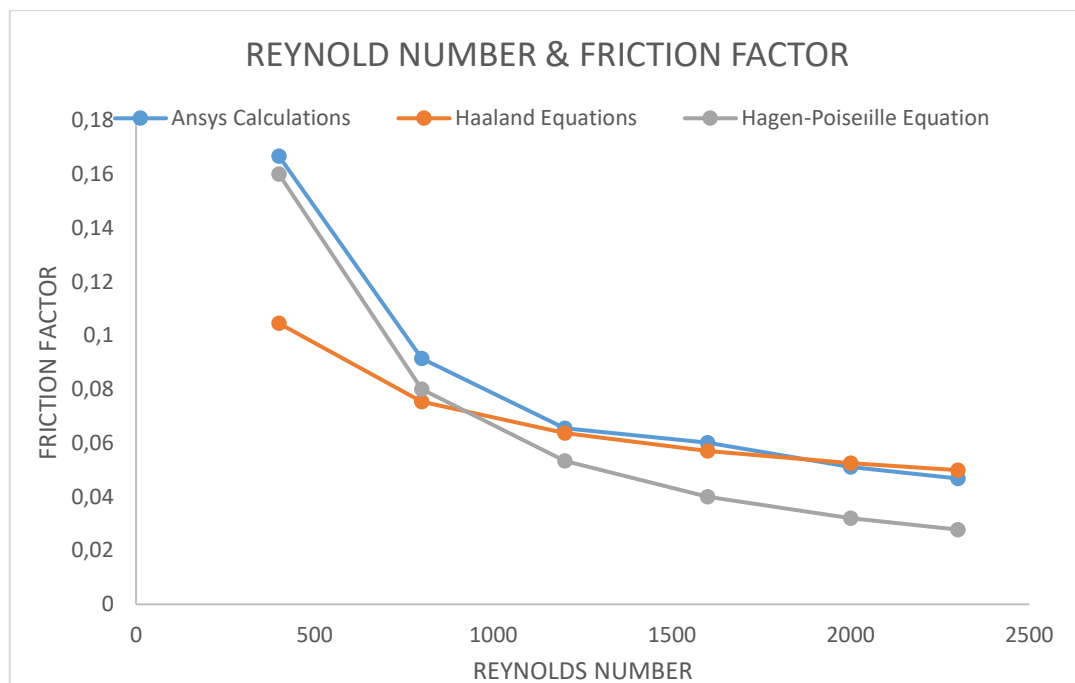*Table 2: Reynolds Number & Friction Factor*

When we compared the results, we came to the conclusion that the Haaland equation gave closer and more accurate results in the light of Table 2.

While doing the calculations we chose to use the Graetz correlation to calculate the Nusselt number. The results for the graph of the Nusselt number corresponding to the Reynolds number is shown in Table 3.

*Table 3: Nusselt Number & Reynolds Number*

### ii.        Turbulent Flow

In order to find the pressure drop for the turbulent region we used the Darcy-Wiesbach Equation. To find our friction factor in this region we used the Colebrook-White equation. We have shown the results of our Pressure drop change according to the Reynolds number in Table 4.



*Table 4: Reynolds number & Pressure drop*

We used the Colebrook-White equation for the friction factor in the turbulent region. Table 5 shows the variation of the friction factor according to the Reynolds number.
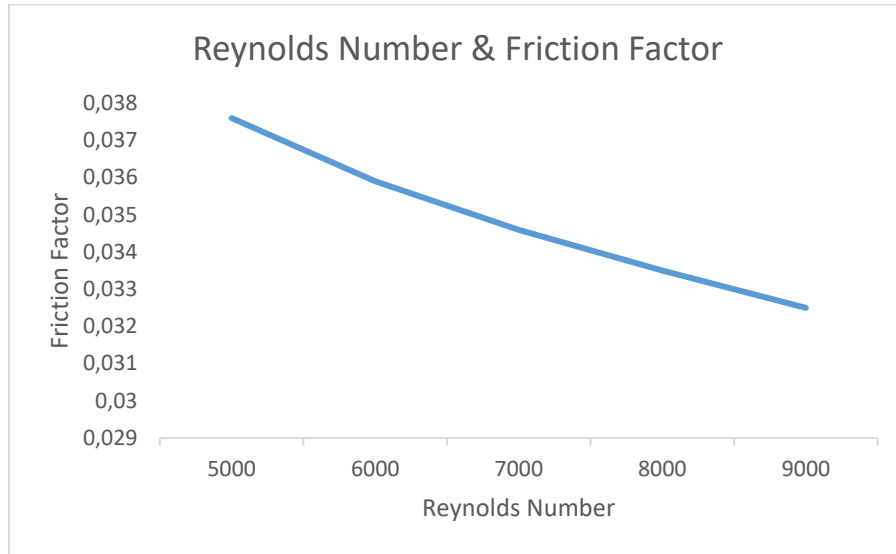
*Table 5: Reynolds number & Friction factor*

While doing the calculations we chose to use the Gnielinski correlation to calculate the Nusselt number. The results for the graph of the Nusselt number corresponding to the Reynolds number is shown in Table 6.
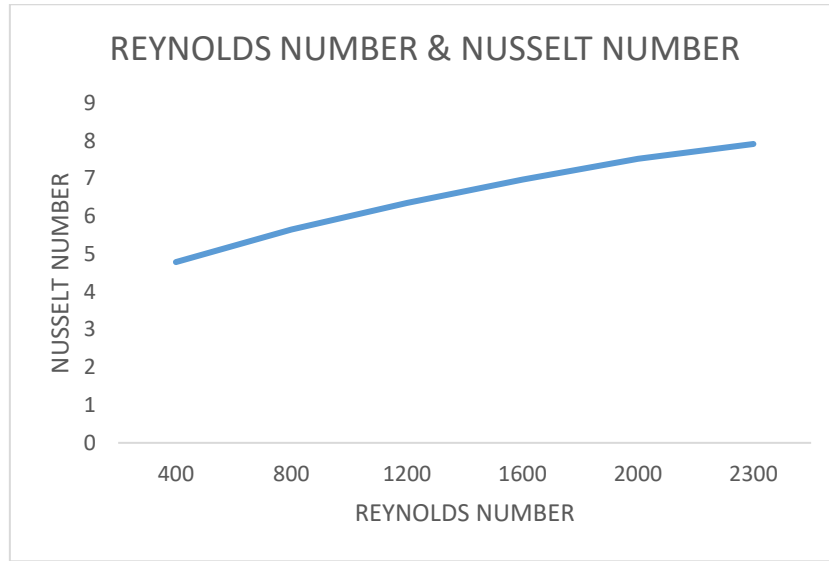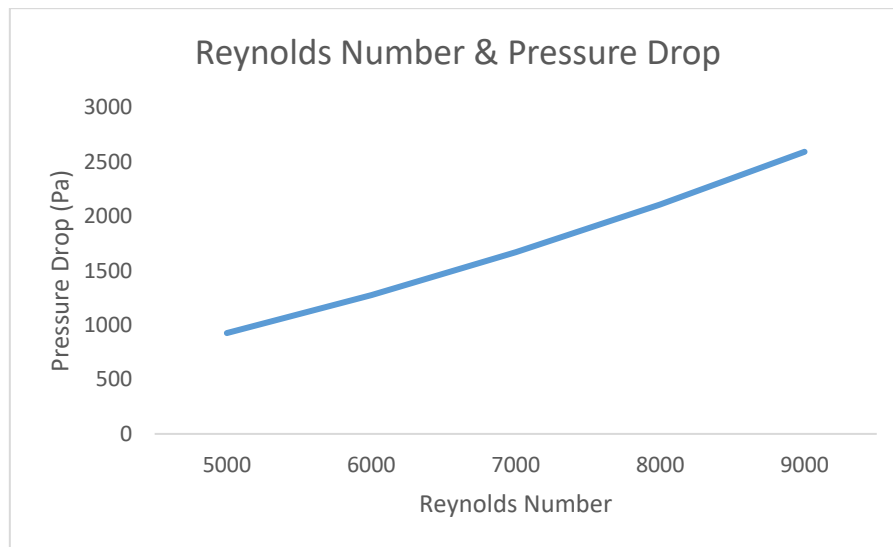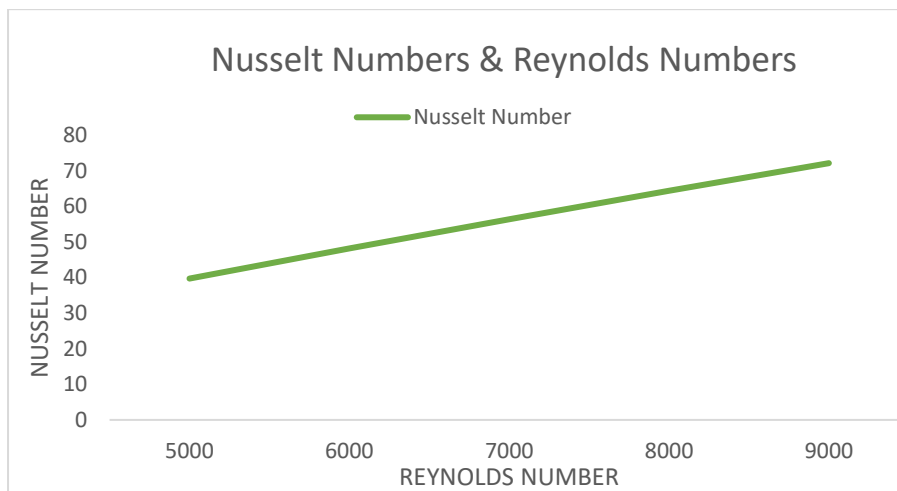


*Table 6: Nusselt Number & Reynolds Number*

In the light of mentioned above, based on the observed error percentages and the provided visual proof, it is evident that the current MATLAB code requires significant improvements to achieve more accurate and reliable results. The discrepancies between the MATLAB code and the ANSYS simulations indicate the presence of mistakes and limitations in the code's implementation. Further

development and refinement of the code, considering more precise modeling techniques and accurate fluid behavior representation, are necessary to reduce the error percentages and enhance the code's performance in both laminar and turbulent flow analyses.

We did this analysis on the straight pipe to find the velocity for our main purpose. Which allowed us to find the pressure drop and heat transfer coefficient values on the grooved pipe by parametrically giving the velocities we found.

### b. Groove Pipe Analysis

Grooved pipes provide better heat transfer than straight pipes because the grooves increase the surface area of the pipe, which allows more heat to be transferred to the fluid flowing through the pipe. This is because the grooves create turbulence in the fluid, which enhances the mixing of the fluid and the heat transfer process.

A study by Bilen et al. (2009) [1] found that grooved pipes can improve heat transfer by up to 63% compared to straight pipes. The study used computational fluid dynamics (CFD) to simulate the flow of water through grooved and straight pipes. The results showed that the grooves created more turbulence in the water, which led to a more efficient heat transfer process.

Another study by Bharadwaj et al. (2009) [2] found that grooved pipes can improve heat transfer by up to 50% compared to straight pipes. The study used experimental methods to measure the heat transfer coefficient of grooved and straight pipes. The results showed that the grooved pipes had a higher heat transfer coefficient, which means that they were able to transfer heat more efficiently.

These studies show that grooved pipes can provide significant improvements in heat transfer compared to straight pipes. This makes grooved pipes a good choice for applications where heat transfer is important, such as in cooling systems, heat exchangers, and boilers.

### i. Laminar Region;

When the thread spacing was taken as 6-8-10-12 mm, the thread diameter was taken as 2-3-4 mm and results were compared. Then, when we compared the results of the analyzes made on Ansys

Fluent, we saw the change in the heat transfer coefficient and pressure drop according to the groove diameter. These results were shown on Table 7 and Table 8.



*Table 7: Reynolds Number & Heat Transfer Coefficient at 6mm pitch*



*Table 8: Reynolds Number & Pressure Drop at 6mm pitch*

When the thread spacing was taken as 8 mm, the thread diameter was taken as 2-3-4 mm and results were compared. Then, when we compared the results of the analyzes made on Ansys Fluent, we saw the change in the heat transfer coefficient and pressure drop according to the groove diameter. These results were shown on Table 9 and Table 10.



*Table 9: Reynolds Number & Heat Transfer Coefficient at 8mm pitch*



*Table 10: Reynolds Number & Pressure Drop  at 8mm pitch*

When the thread spacing was taken as 10 mm, the thread diameter was taken as 2-3-4 mm and results were compared. Then, when we compared the results of the analyzes made on Ansys Fluent, we saw the change in the heat transfer coefficient and pressure drop according to the groove diameter. These results were shown on Table 11 and Table 12.
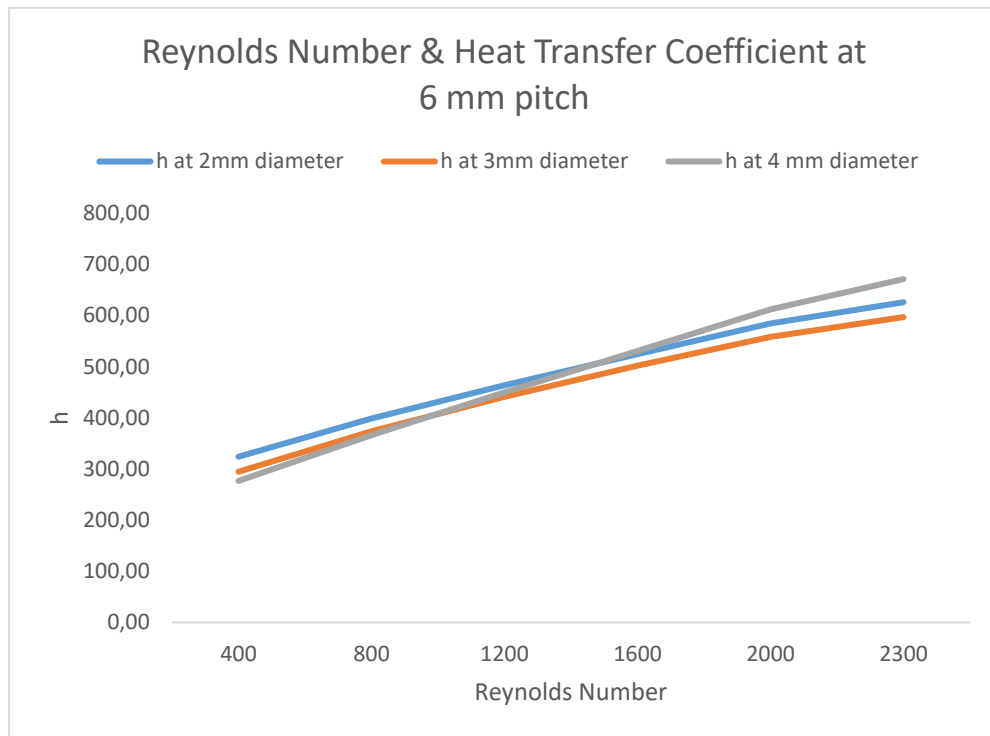


*Table 11: Reynolds Number & Heat Transfer Coefficient at 10mm pitch*
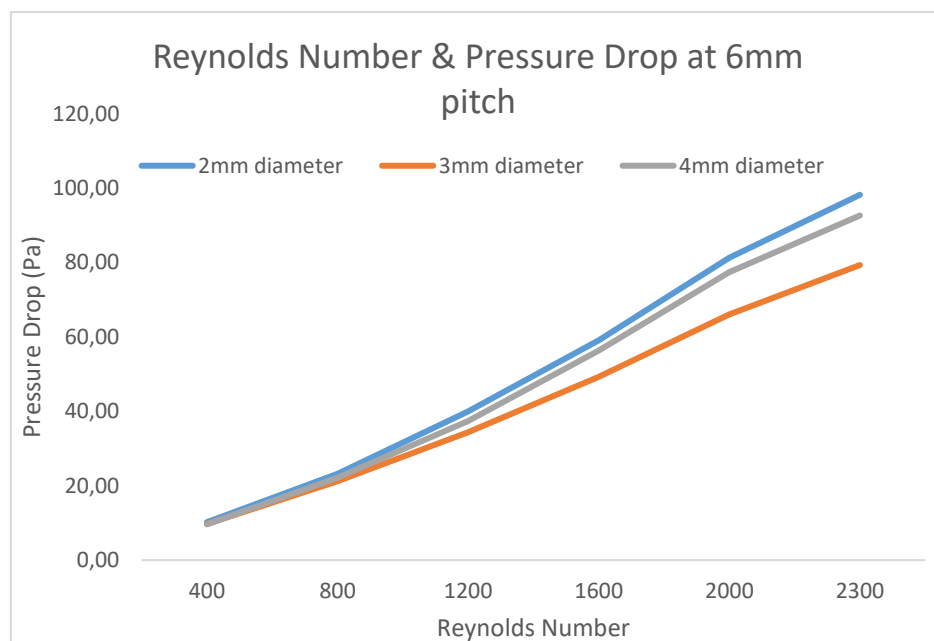


*Table 12: Reynolds Number & Pressure Drop at 10mm pitch*

When the thread spacing was taken as 12 mm, the thread diameter was taken as 2-3-4 mm and results were compared. Then, when we compared the results of the analyzes made on Ansys Fluent, we saw the change in the heat transfer coefficient and pressure drop according to the groove diameter. These results were shown on Table 13 and Table 14.
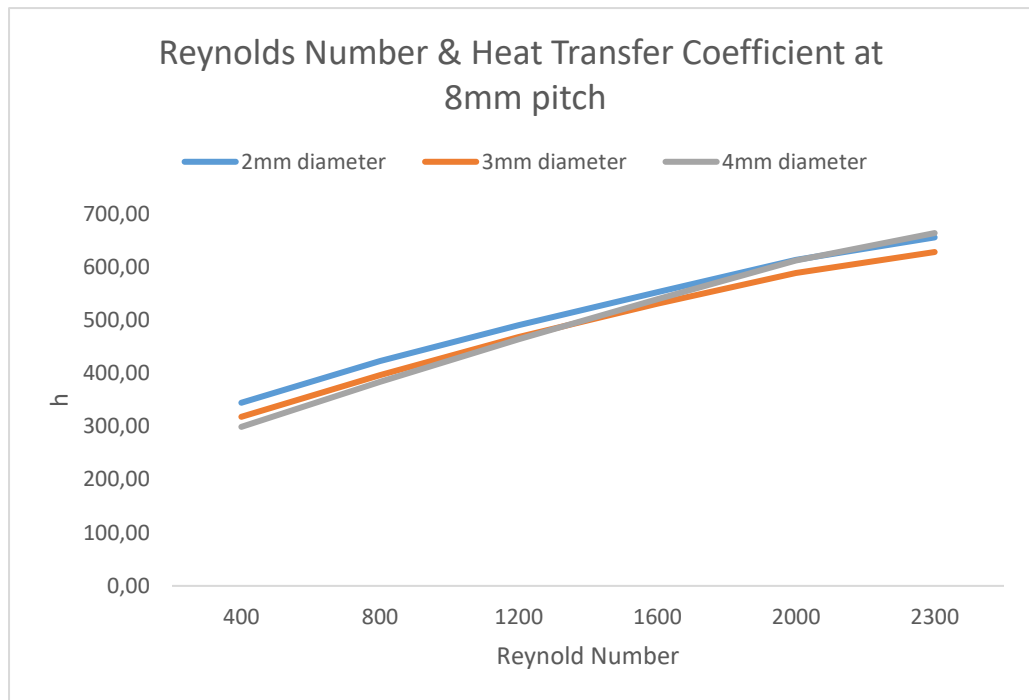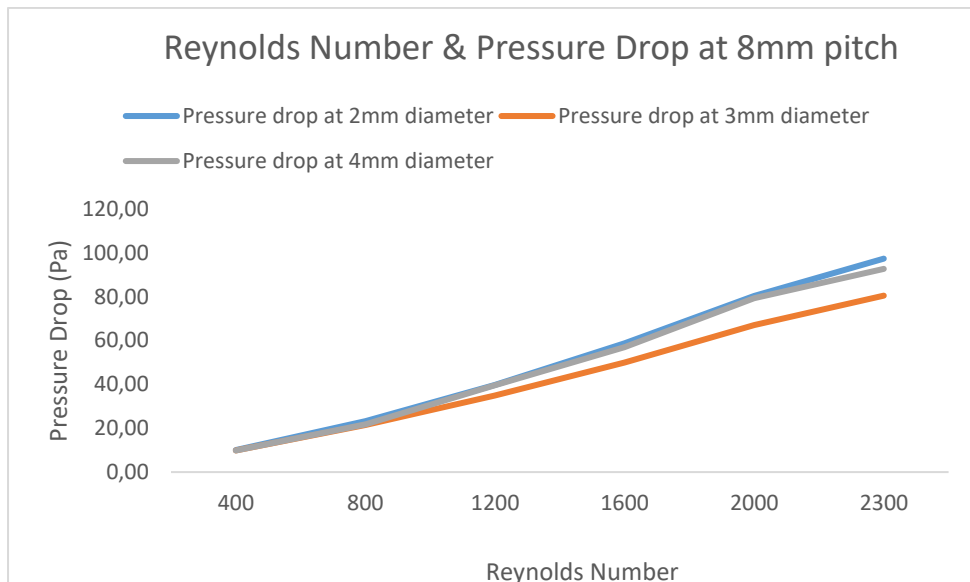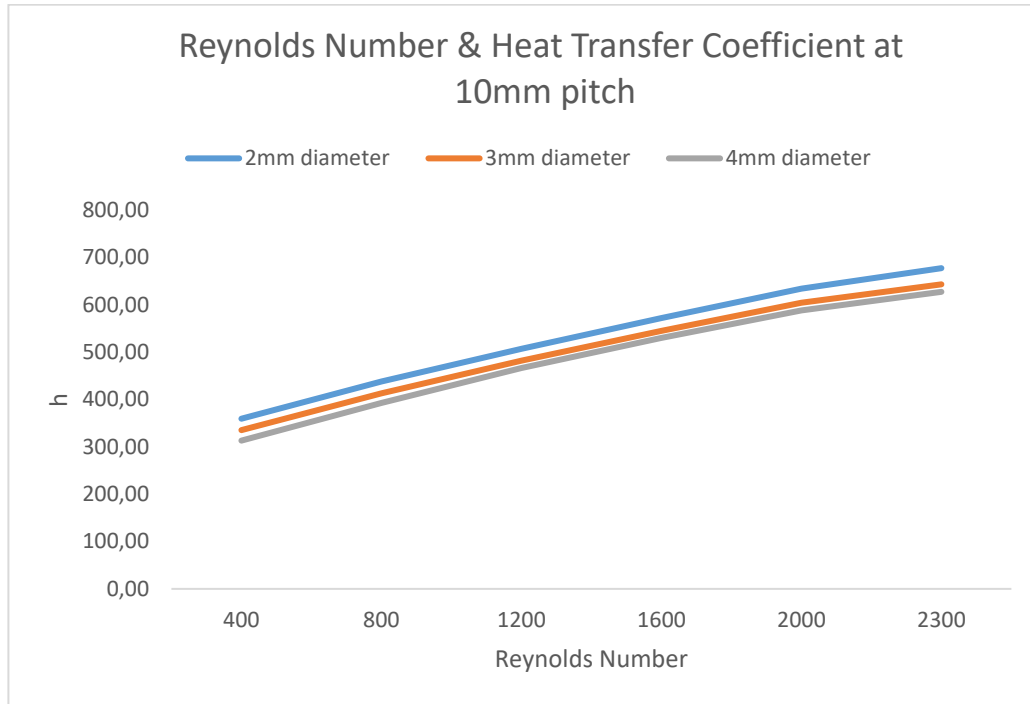


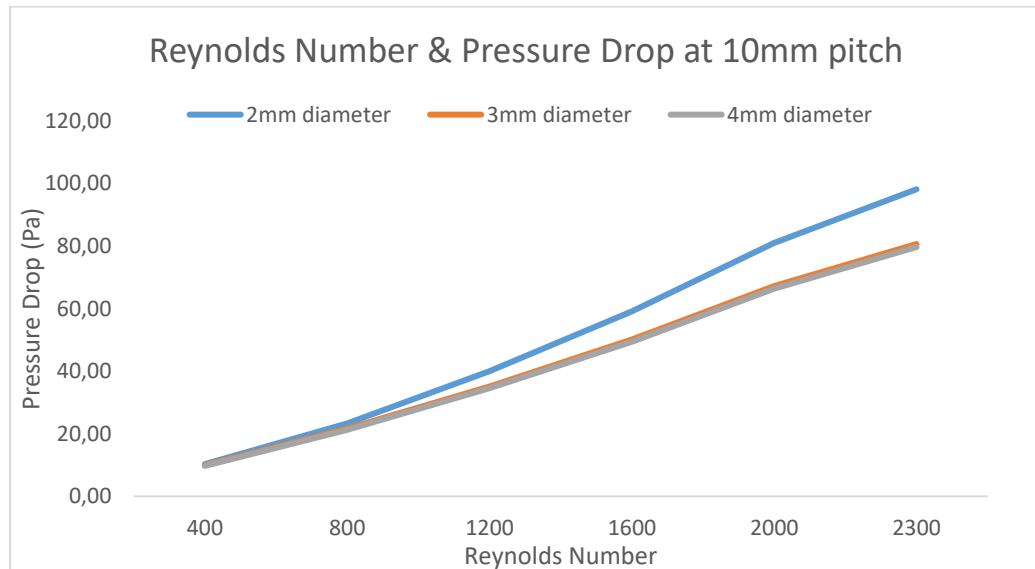*Table 13: Reynolds Number & Heat Transfer Coefficient at 12mm pitch*
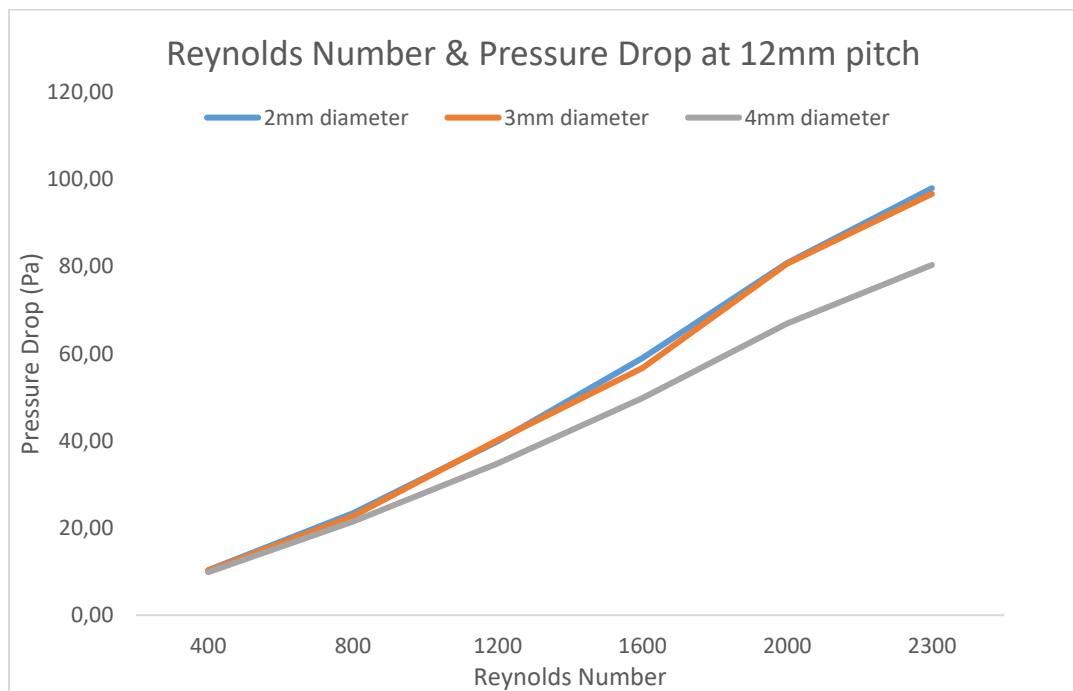


*Table 14: Reynolds Number & Pressure Drop at 12mm pitch*

According to these results;

For 6mm pitch; the h values of the pipe with a 4mm groove diameter in 6 mm pitch gives better results. However, when the pressure drop values are examined, it is seen that the pipe with 3mm groove diameter gives better results. It is seen that the optimum value is a pipe with a groove diameter of 3 mm in the scenario where the pitch of 6 mm is fixed.

For 8mm pitch; the h values of the pipe with a 4mm groove diameter in 8 mm pitch gives better results. However, when the pressure drop values are examined, it is seen that the pipe with 3mm groove diameter gives better results. It is seen that the optimum value is a pipe with a groove diameter of 3 mm in the scenario where the pitch of 8 mm is fixed.

For 10mm pitch; the h values of the pipe with a 2mm groove diameter in 10 mm pitch gives better results. However, when the pressure drop values are examined, it is seen that the pipe with 3mm groove diameter gives better results. It is seen that the optimum value is a pipe with a groove diameter of 3 mm in the scenario where the pitch of 10 mm is fixed.

For 12mm pitch; the h values of the pipe with a 3mm groove diameter in 10 mm pitch gives better results. However, when the pressure drop values are examined, it is seen that the pipe with 4mm groove diameter gives better results. It is seen that the optimum value is a pipe with a groove diameter of 4 mm in the scenario where the pitch of 12 mm is fixed.

In the end for the laminar region, when these results were compared among themselves, it was found that the most optimum result was in the grooved pipe with 12 mm pitch and 4 mm diameter.

### ii.    Turbulent Region;

When the thread spacing was taken as 6-8-10-12 mm, the thread diameter was taken as 2-3-4 mm and results were compared. Then, when we compared the results of the analyzes made on Ansys Fluent, we saw the change in the heat transfer coefficient and pressure drop according to the groove diameter. These results were shown on Table 15 and Table 16.
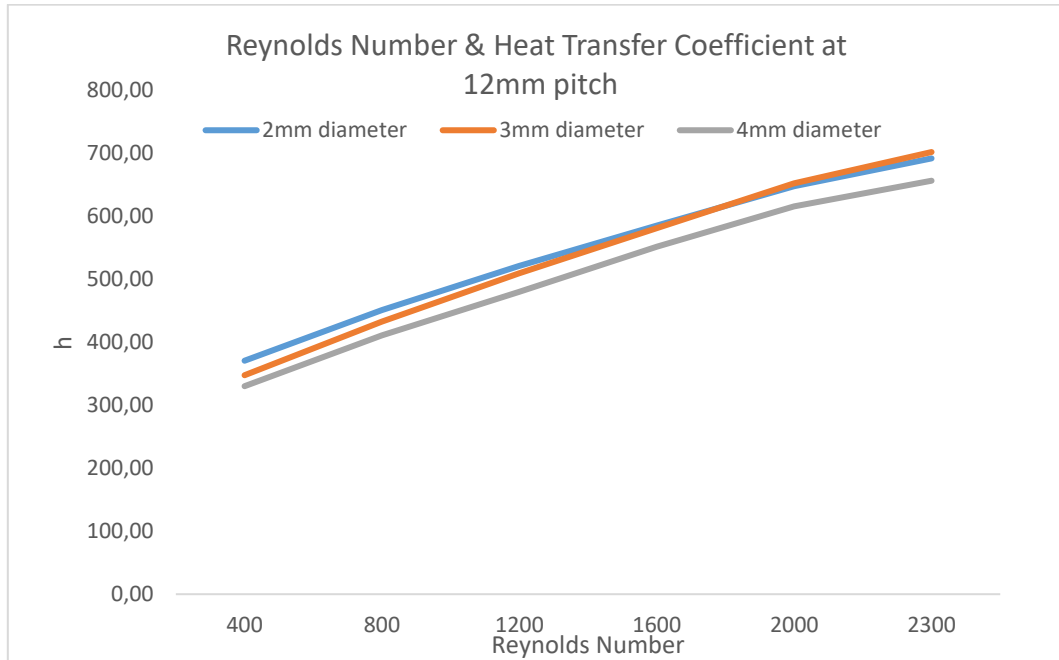
*Table 15: Reynolds Number & Heat Transfer Coefficient at 6mm pitch (Turbulent)*



*Table 16: Reynolds Number & Pressure Drop at 6mm pitch (Turbulent)*

When the thread spacing was taken as 8 mm, the thread diameter was taken as 2-3-4 mm and results were compared. Then, when we compared the results of the analyzes made on Ansys Fluent, we saw the change in the heat transfer coefficient and pressure drop according to the groove diameter. These results were shown on Table 17 and Table 18.

*Table 17: Reynolds Number & Heat Transfer Coefficient at 8mm pitch (Turbulent)*



*Table 18: Reynolds Number & Pressure Drop at 8mm pitch (Turbulent)*

When the thread spacing was taken as 10 mm, the thread diameter was taken as 2-3-4 mm and results were compared. Then, when we compared the results of the analyzes made on Ansys Fluent, we saw the change in the heat transfer coefficient and pressure drop according to the groove diameter. These results were shown on Table 19 and Table 20.

*Table 19: Reynolds Number & Heat Transfer Coefficient at 10mm pitch (Turbulent)*
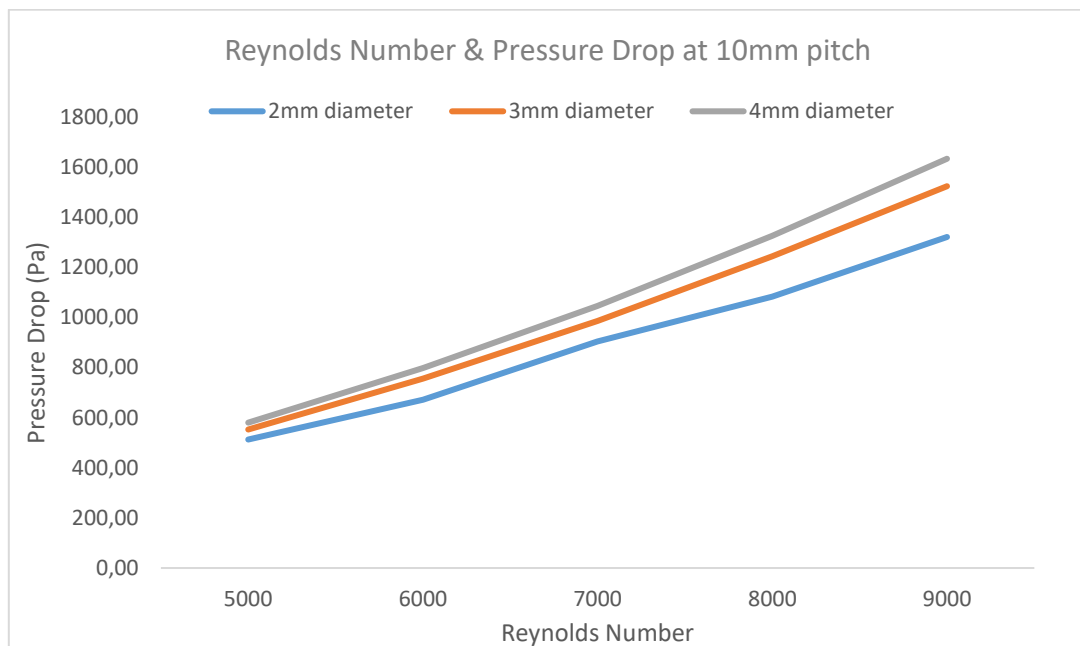


*Table 20: Reynolds Number & Pressure Drop at 10mm pitch (Turbulent)*

When the thread spacing was taken as 12 mm, the thread diameter was taken as 2-3-4 mm and results were compared. Then, when we compared the results of the analyzes made on Ansys Fluent, we saw the change in the heat transfer coefficient and pressure drop according to the groove diameter. These results were shown on Table 21 and Table 22.
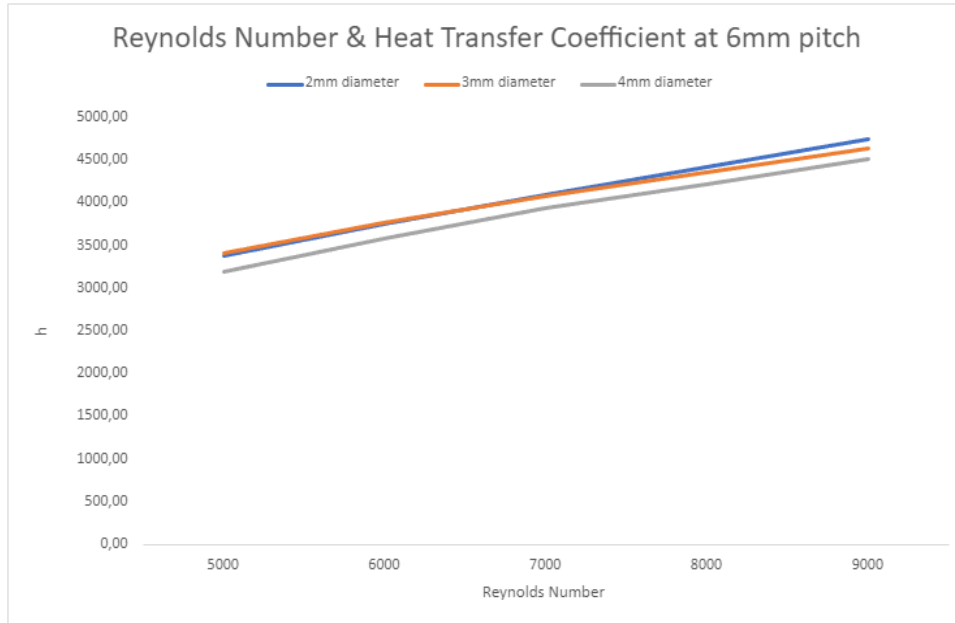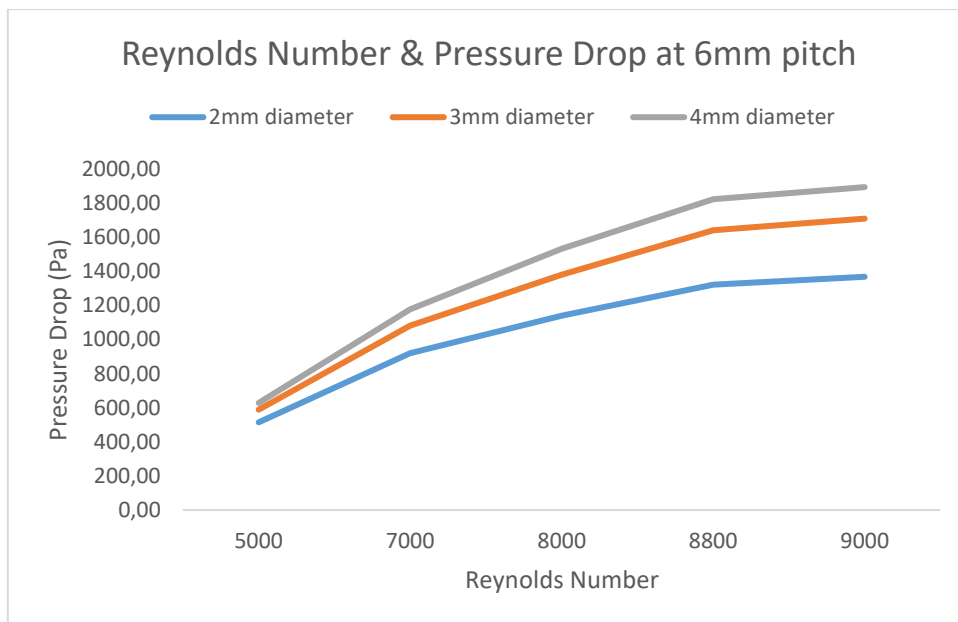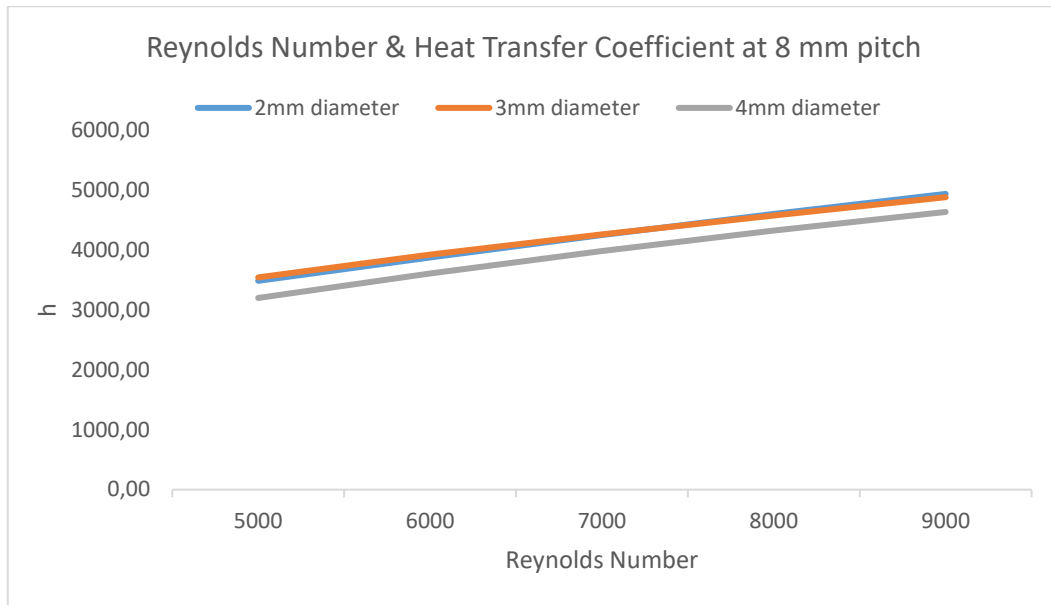
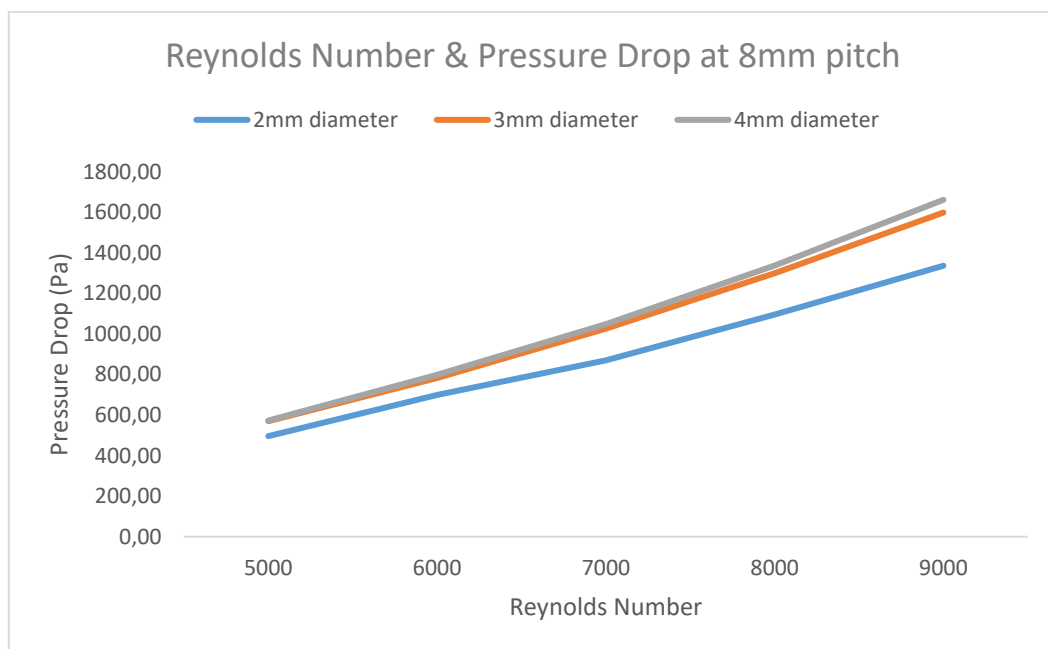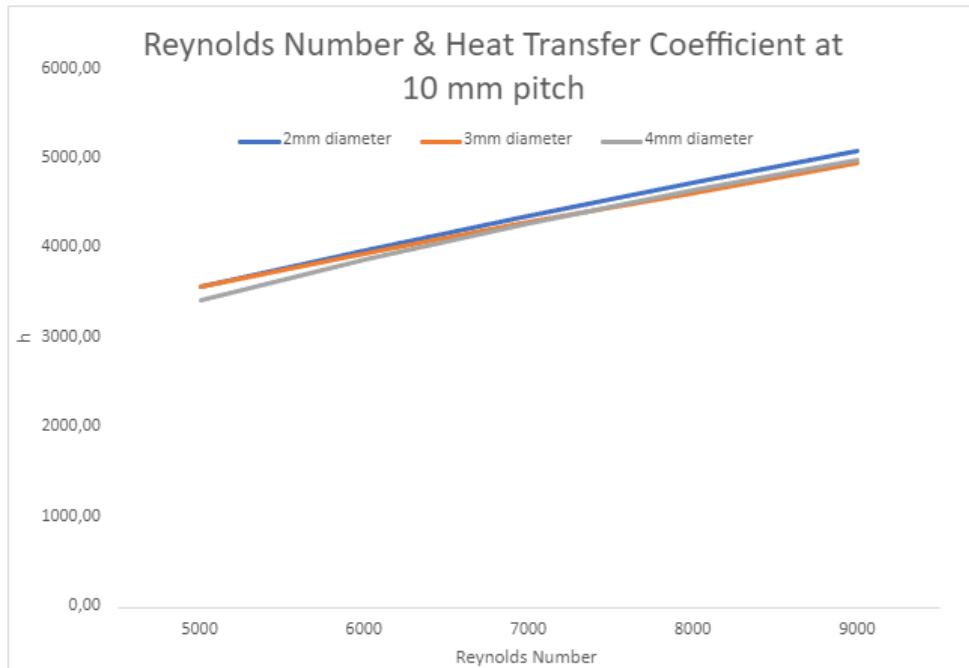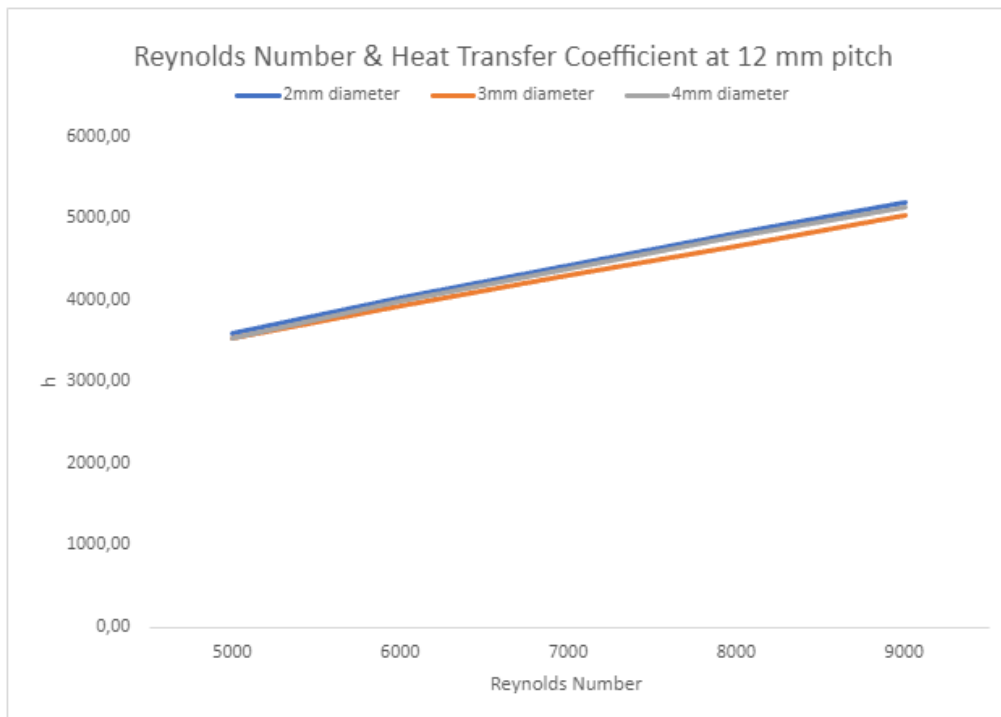*Table 21: Reynolds Number & Heat Transfer Coefficient at 12mm pitch (Turbulent)*
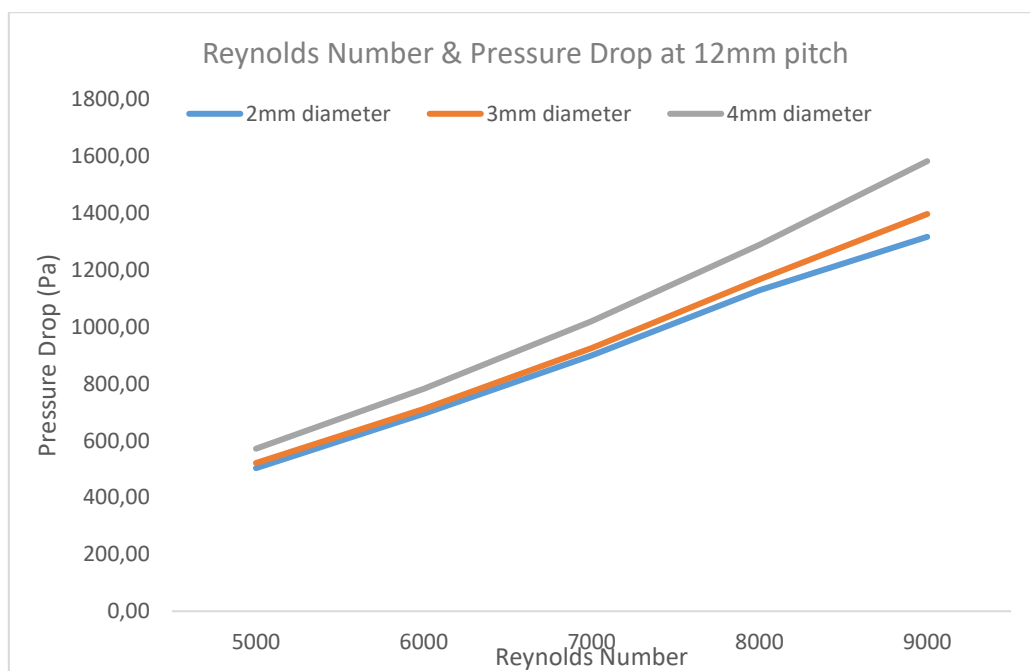


*Table 22: Reynolds Number & Pressure Drop at 12mm pitch (Turbulent)*

According to these results;

For 6mm pitch; It is seen that the optimum value for both h and Pressure drop is the pipe with a groove diameter of 2 mm in the scenario where the pitch of 6 mm is fixed.

For 8mm pitch; It is seen that the optimum value for both h and Pressure drop is the pipe with a groove diameter of 2 mm in the scenario where the pitch of 8 mm is fixed.

For 10mm pitch; It is seen that the optimum value for both h and Pressure drop is the pipe with a groove diameter of 2 mm in the scenario where the pitch of 10 mm is fixed.

For 12mm pitch; It is seen that the optimum value for both h and Pressure drop is the pipe with a groove diameter of 2 mm in the scenario where the pitch of 12 mm is fixed.

In the end for the Turbulent region, when these results were compared among themselves, it was found that the most optimum result was in the grooved pipe with 10 mm pitch and 2 mm diameter.

## 4. CONCLUSION

In this project, we conducted an in-depth analysis of groove pipes to investigate the effects of groove diameter and pitch length on heat transfer coefficient and pressure drop in both laminar and turbulent flow regimes. The main objective was to identify the optimum combination of groove diameter and pitch length that would yield the highest performance compared to a straight pipe.

In the laminar flow regime, our findings demonstrated that the grooved pipe with a pitch length of 12 mm and a groove diameter of 4 mm exhibited the most favorable heat transfer coefficient and pressure drop characteristics. This indicates that, in low flow rates and smooth fluid flow conditions, the larger groove pitch and diameter contribute to enhanced heat transfer performance and reduced pressure drop, making it the optimal choice for such applications.

Conversely, in the turbulent flow regime, the grooved pipe with a pitch length of 10 mm and a groove diameter of 2 mm emerged as the most optimum configuration. In high flow rates and turbulent flow conditions, the smaller groove pitch and diameter facilitated superior heat transfer coefficient and pressure drop results, outperforming other configurations in this regime.

Overall, the analyses revealed that groove pipes, both in laminar and turbulent flow, outperformed the traditional straight pipes in terms of heat transfer efficiency and pressure drop reduction. This highlights the potential of groove pipes as promising alternatives in fluid flow systems, especially in applications that require enhanced heat transfer, such as refrigeration, air conditioning, and heat exchangers.

The significance of this project lies in the determination of the optimal groove diameter and pitch length combinations for achieving the best heat transfer and pressure drop characteristics in different flow regimes. Engineers and researchers can use this valuable information to design and optimize groove pipe systems according to their specific application requirements.

However, it's essential to acknowledge the limitations of this study. The analysis was conducted for a limited range of groove diameters (2 mm, 3 mm, and 4 mm) and pitch lengths (6 mm, 8 mm, 10 mm, and 12 mm). Further investigations could explore a broader range of groove geometries to gain deeper insights into their performance under various flow conditions.

In conclusion, the analysis of groove pipes proved their superior heat transfer efficiency and pressure drop reduction compared to straight pipes. The most optimum groove diameter and pitch length configurations, identified for laminar and turbulent flow regimes, can serve as a valuable reference for designing efficient and effective groove pipe systems in fluid flow applications. As such, groove pipes hold great promise for the development of cost-effective and energy-efficient fluid transportation systems in industries where heat transfer and pressure drop considerations are crucial.

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## 6. APPENDIX

------------------------------------  MATLAB CODES  ----------------------------------

**For Leminar Flow**

```
clc
clear all;

% Constants
thermalConductivity = 0.6;
pipeLength = 1;
pipeWidth = 8e-3;
heatFlux = 4500;
inletTemperature = 20;

% Calculating parameters
crossSectionalArea = pipeWidth * pipeLength;
tolerance = 1e-6;
maxIterations = 1000;

% Fluid properties
dynamicViscosity = 0.001003;
specificHeat = 4182;
density = 998.2;

% Laminar flow analysis
outletTemperature = inletTemperature;

for Reynolds = 400:100:2300
    for i = 1:maxIterations
        meanTemperature = (inletTemperature + outletTemperature) / 2;
        deltaTemperature = outletTemperature - inletTemperature;

        % Calculate the friction factor (f) for laminar flow using Haaland and
Poiseuille equation
        % f=(0.79 * log(Reynolds) - 1.64)^(-2);
         f = 64 / Reynolds;

        % Calculate the Prandtl number (Pr) based on the mean temperature
        Pr = dynamicViscosity * specificHeat / thermalConductivity;

        % Calculate the Nusselt number (Nu) for developing laminar flow using the
Graetz correlation
        Nu = (3.66 +
(0.0668*(pipeWidth/pipeLength)*Reynolds*Pr)/(1+0.04*((pipeWidth/pipeLength)*Reynol
ds*Pr)^(2/3)));

        %Entry Length
        entry_length=0.05*pipeWidth*Reynolds;


        % Calculate the convective heat transfer coefficient (h) based on the
Nusselt number (Nu) and pipe width
        convectiveCoefficient = Nu * (thermalConductivity / pipeWidth);
```

```matlab
        % Calculate the water flow velocity (v) for laminar flow using the
Reynolds number (Re) and pipe dimensions
        velocity = Reynolds * dynamicViscosity / (density * pipeWidth);

        % Calculate the pressure drop (deltaP) for laminar flow using the Hagen-
Poiseuille equation
        deltaP = (f *pipeLength * density * (velocity^2)) / (2 * pipeWidth);
        %deltaP = (32 * dynamicViscosity * pipeLength * velocity) / (density*
pipeWidth^2);

        % Update the outlet temperature
        % outletTemperature = inletTemperature + (heatFlux * pipeWidth) /
(convectiveCoefficient * crossSectionalArea);
        outletTemperature = inletTemperature + (heatFlux * pipeWidth) / (1.2 * Nu
* thermalConductivity);
        % Check for convergence
        % if abs(outletTemperature - inletTemperature) < tolerance
        %     break;
        % end
    end

    % Display the result for each Reynolds number
    fprintf('Reynolds Number: %d, Outlet Temperature: %.2f degrees Celsius, Flow
Velocity: %.4f m/s, Friction Factor: %.4f, Pressure Drop: %.2f Pa, Nusselt Number:
%.2f \n', Reynolds, outletTemperature, velocity, f, deltaP, Nu);
end
```

**For Turbulent Flow**

```matlab
clc
clear all;
% Constants
thermalConductivity = 0.6; % Thermal conductivity of water at 20 degrees Celsius
(W/m·K)
pipeLength = 1; % Length of the pipe (m)
pipeWidth = 8e-3; % Width of the pipe (m)
heatFlux = 45000; % Heat flux along the pipe wall (W/m^2)
inletTemperature = 20; % Initial guess for inlet temperature (°C)

% Calculating parameters
crossSectionalArea = pipeWidth*pipeLength; % Cross-sectional area of the pipe
(m^2)
tolerance = 1e-6; % Tolerance for convergence
maxIterations = 1000; % Maximum number of iterations

% Fluid properties
dynamicViscosity = 0.001003; % Dynamic viscosity of water at 20 degrees Celsius
(Pa·s)
specificHeat = 4182; % Specific heat capacity of water at 20 degrees Celsius
(J/kg·K)
density = 998.2; % Density of water at 20 degrees Celsius (kg/m^3)

% Turbulent flow analysis
outletTemperature = inletTemperature; % Initial guess for outlet temperature (°C)
```

```matlab
for Reynolds = 4000:200:9000
    for i = 1:maxIterations
        meanTemperature = (inletTemperature + outletTemperature) / 2; % Mean
temperature (°C)
        deltaTemperature = outletTemperature - inletTemperature; % Temperature
difference (°C)
         % Calculate the friction factor (f) using the Blasius equation
        f = 0.3164 * (Reynolds^(-0.25));
        % Calculate the Prandtl number (Pr) based on the mean temperature
        Pr = dynamicViscosity * specificHeat / thermalConductivity;

        % Calculate the Nusselt number (Nu) using the Gnielinski correlation for
turbulent flow
        % Nu = 0.023 * Reynolds^(0.8) * Pr^(0.4);
        Nu = (f / 8) * (Reynolds - 1000) * Pr / (1 + 12.7 * (f / 8)^0.5 *
(Pr^(2/3) - 1));

        % Calculate the convective heat transfer coefficient (h) based on the
Nusselt number (Nu) and pipe width
        convectiveCoefficient = Nu * (thermalConductivity / pipeWidth);

        % Calculate the water flow velocity (v) based on the Reynolds number (Re)
and pipe dimensions
        velocity = Reynolds * dynamicViscosity / (density * pipeWidth );


        % Calculate the pressure drop (deltaP) using the Darcy-Weisbach equation
        deltaP = (f *pipeLength * density * (velocity^2)) / (2 * pipeWidth);

        % Energy balance equation
        % temperatureChange = (heatFlux * pipeWidth) / (convectiveCoefficient *
crossSectionalArea);

        % Update the outlet temperature
        %outletTemperature = inletTemperature + temperatureChange;
        outletTemperature = inletTemperature + (heatFlux * pipeWidth) / (1.2 * Nu
* thermalConductivity);

        % % Check for convergence
        % if abs(temperatureChange) < tolerance
        %     break;
        % end
    end

    % Display the result for each Reynolds number
    fprintf('Reynolds Number: %d, Outlet Temperature: %.2f degrees Celsius, Flow
Velocity: %.3f m/s, Friction Factor: %.4f, Pressure Drop: %.2f Pa, Nusselt Number:
%.2f \n', Reynolds, outletTemperature, velocity, f, deltaP,Nu);
end
```