DIPLOMA THESIS

Department Mineral Resources and
Petroleum Engineering, Chair of
Petroleum Production and Processing
& Department of Economics, Mining
University of Leoben.

By:

Nan JIA EISENBERGER

Supervision of:

Ao. Univ.-Prof. Dipl.-Ing. Dr. techn. Wilhelm Brandstätter

Leoben, July, 2010

Optimization's potentials for oil guide plate of guide bearing in a hydro-generator

Andritz Hydro GmbH

EIDESSTATTLICHE ERKLÄRUNG

Ich erkläre an Eides statt, dass ich diese Arbeit selbständig verfasst, andere als die angegebenen Quellen und Hilfsmittel nicht benutzt, und mich auch sonst keiner unerlaubten Hilfsmittel bedient habe.

AFFIDAVIT

I declare in lieu of oath that I did this Master's Degree in hand by myself using only literature cited at the end of this volume.

_____
Nan JIA EISENBERGER

Danksagungen

Abstract

In this diploma thesis, which was carried out for the company Andritz Hydro GmbH, the optimization potentials with regard to friction losses for a guide plate conveying concept of guide bearings in hydro-generators were investigated. With the help of numerical flow simulations the steady, isothermal and incompressible flow in the gap between the rotating runner and the guide plate in the oil container for the hydro-generator 'Glendoe' was calculated, which is a part of a hydroelectric plant located in Scotland. These calculations were performed with the free, open source CFD software package OpenFOAM ('Open Field Operation and Manipulation').

By variation of geometric parameters, such as the gap width between the rotor and the guide plate, an attempt to maximize the flow rate through the gap while reducing the correlated friction losses was made. With the help of so-called dimensionless numbers (e.g. friction loss coefficient), the influence of certain parameters are presented. The results of these numerical flow calculations were compared with analytical solutions found in the literature for simplified geometries (enclosed rotating discs with different gap widths between the disc and casing). Subsequently, attempts to find a general factor for the key factors have been made, by which it is possible to quickly predict the most important parameters, such as friction losses, flow rate and pressure, for the guide plate conveying concept.

Kurzfassung

Im Rahmen dieser Diplomarbeit, welche für die Firma Andritz-Hydro GmbH durchgeführt wurde, wurden Verbesserungspotentiale hinsichtlich der Reibungsverluste für ein Scheibenförderungskonzept in Führungslagern von Hydro-Generatoren untersucht. Mit Hilfe von numerischen Strömungssimulationen wurde die stationäre, isotherme und inkompressible Strömung zwischen einem rotierenden Mantel- bzw. Scheibenabschnitt und einer gegenüberliegenden stationären Wand für den Hydro-Generator 'Glendoe' berechnet, welcher ein Teil eines Wasserkraftwerks in Schottland ist. Diese Berechnungen wurden mit dem frei zugänglichen CFD-Softwarepaket OpenFOAM („Open Field Operation and Manipulation') durchgeführt.

Durch Variationen von geometrischen Parametern, wie zum Beispiel der Spaltweite zwischen Spurkopfring und Förderscheibe, wurde versucht den Volumenstrom durch den Spalt zu maximieren bei gleichzeitiger Reduktion der Reibungsverluste. Mit Hilfe von so genannten dimensionslosen Kennzahlen (z.B.: Reibbeiwert) kann der Einfluss gewisser Parameter dargestellt werden. Die Ergebnisse der numerischen Strömungsberechnungen wurden mit analytischen Lösungen aus der Fachliteratur für vereinfachte Geometrien (geschlossene rotierende Scheiben mit verschiedenen Spaltweiten zwischen Scheibe und Gehäuse) verglichen. In weiterer Folge wurde versucht einen Gesamteinflussfaktor für die wichtigsten Einflussfaktoren zu ermitteln, mit dessen Hilfe es möglich ist, eine schnelle Vorhersage der wichtigsten Kenngrößen, wie zum Beispiel Reibungsverluste, Volumenstrom und Drücke, für das Scheibenförderungskonzept zu treffen.

Contents

List of figures

List of tables

1 Introduction

1.1 Background

Andritz-Hydro GmbH is a global supplier of electro-mechanical systems and services for Hydro Power plants. The company is a leader in the world market for hydraulic power generation. Intensive research and development work, including virtual tools as Computer Aided Engineering (CAE) and Computational Fluid Dynamics (CFD), form the solid basis of their design capabilities. Building and testing prototypes are processes which are both expensive and time-intensive, and therefore CFD is an attractive way to support the development process of new components and to optimize these components concerning their efficiency.

In this work CFD is used to investigate the lubricating flow for a guide bearing concept of a generator. Figure 1-1 shows a schematic illustration of the main constructive components and flow directions of lubricating oil [1]. The rotor side spaces in such a system represent one of the most important sources to the overall hydraulic losses.

Figure 1- 1 : Schematic illustration of the main constructive components and flow directions of lubricating oil [1].

A circulating oil flow is constituted in the oil container due to the rotating runner. The geometric parameters of the guide plate, as for example the distance to the rotor or its length, influences the friction loss which is engendered between rotor and the opposite stationary walls

as the highly viscous oil flows through the spacing between them. A parameterised, simplified CFD model of the guide bearing concept was created to identify the key parameters influencing the friction losses in the system. In addition to minimizing the friction losses even further parameters (e.g., pressure drop or discharge flow rate) play a significant role in order to guarantee the lubrication of the guide bearing.

The results of this diploma work have been casted into the empirical formulas for the main variables which describe the fluid flow in the guide bearing lubrication system, similar to that of Linnecken [2], Geis [3] or Dubbel [4] for a free rotating disc or a enclosed rotating disc (see Figure 1-2).

Figure 1- 2 : a) A rotating disc in stationary fluid [5] b) Enclosed rotating disc [6]

1.2 Previous Research

In the year 2006 Andritz-Hydro made a study [1] to compare the analytical approaches found in the literature for calculating the disc friction losses in a rotor-stator system. Most of these empirical formulas, as for example by Linnecken [2] or Schlichting [5], are based on the interpretation of experimental data and measurements. A comparison between these analytical models and CFD simulations with ANSYS CFX [7] has been also made to find an appropriate model for estimating the friction losses of the guide plate in the guide bearing concept.

In 2007 ICE Strömungsforschung GmbH [8] developed a parameterised simplified model of this rotor-stator system on the basis of the OpenFOAM CFD toolbox [9] for Andritz-Hydro to find the key parameters which influence the friction losses or oil flow rate in the gap between the rotor and stator disc. In this thesis this model has been used as framework to make further simulations to develop simple analytical formulas, which can describe the fluid flow in the system.

1.3 Objective of this thesis

There were multiple goals for this work:

Evaluation of OpenFOAM ('Open Source Field Operation and Manipulation') [9] with respect to its suitability for flows as they occur in rotor-stator systems. For this purpose a commonly used case has been investigated, i.e. the 'Enclosed rotating disc'. These preliminary studies were used in addition to find an appropriate turbulence model for further calculations in this work.

A parameterised, simplified CFD model of the guide bearing lubrication system was created to identify the main parameters (geometric and boundary conditions) influencing the friction losses, flow rate and pressure drop in the system. Therefore a systematic numerical study with OpenFOAM was performed in which more than $1,000$ simulations were analyzed.

From the results of these simulations qualitative and quantitative design rules showing the influence of parameter variations on pressure drop, friction torque and flow rates through the rotor side spaces can be given. Simple correlations for the main variables are derived to gain a better understanding of the flow mechanisms associated with the lubrication of guide bearings.

Comparison of these results with correlations can be found in the literature (i.e., Linnecken, Geis, Dubbel, etc.) for rotor-stator systems.

Own empirical formulas are carried out with the help of Matlab to predict and evaluate the most important flow variables of the parameterised model 'Glendoe', i.e. friction losses at rotor and stator walls, flow rate and the pressure differences between inlet and outlet of the rotor-stator distance.

1.4 Thesis Outline

This thesis is organised in the following way:

In this chapter a short introduction to the initial situation from oil guide plate of guide bearing in a hydro-generator is given.

Chapter 2 presents the mathematical models which are used for the CFD simulations in this thesis. It starts with a short review on the Navier-Stokes equation for incompressible fluids. Further on a description of turbulent fluid flows and turbulence modelling is made.

Chapter 3 gives an overview about the open-source CFD software OpenFOAM [9], [10]. The theoretical background as well as solvers, the selected turbulence models, the mesh generation process and the post-processing are described.

In Chapter 4 the results from the numerical simulations of the validation case 'Enclosed rotating disc' are presented. These results are compared with analytical correlations found in the literature (e.g., Linnecken, Geis, Dubbel, etc.).

A general overview of the guide bearing lubrication concept is presented in chapter 5.

Chapter 6 explains the parameterised CFD model for the example geometry 'Glendoe'. It gives a description of the almost completely automated solution process with OpenFOAM, from pre- to post-processing, which is controlled by a script written in Python [11].

In Chapter 7 the results for the parameter variation study and the appropriate correlations for the example geometry 'Glendoe' are presented.

The empirical formulas for the parameterised model 'Glendoe' are shown in Chapter 8.

In Chapter 9 the conclusions on this thesis are given and some recommendations for possible future work are suggested.

2 Mathematical Model

2.1 Governing equations of continuum mechanics

The Navier-Stokes equations are partial differential equations, which are named after the two 19th century scientists Claude-Louis Navier and George Gabriel Stokes. These equations describe the motion of fluid substances, which can flow such as water, oil, air, etc. [12], [13]. The solutions of Navier-Stokes equations can be found with the help of CFD simulations.

2.1.1 Navier – Stokes equations

The Navier-Stokes equations based on Newton second law [13]:

- For solid mass:

$$F = m \cdot a \tag{2.1}$$

Here, the m represents the mass and a is acceleration.

- For a continuum:

$$\rho \ (\underbrace{\frac{\partial \vec{u}}{\partial t} + \vec{u} \nabla \vec{u}}_{(I)}) = \nabla \sigma \cdot f \tag{2.2}$$

Where ρ describes the fluid density (i.e., mass per volume), the term (I) at the left side is the acceleration, ∇ defines the del operator, $\vec{u}$ is the velocity vector, σ is the stress tensor (i.e., force per area) and f represents the body force vector (i.e., force per volume).

In this project, the governing equations are based on continuity and momentum equations which read:

- **Continuity equation**: The general form of continuity equation is replenished by the mass conservation equation [12], [13], [14].

$$\frac{\partial \rho}{\partial t} + \nabla \ \rho \ \vec{u}) = 0 \tag{2.3}$$

Where ρ defines the fluid density, $\vec{u}$ is the velocity vector, ∇ describes the del operator and the convective term $\rho \ \vec{u}$ represents the net flow of mass across boundaries.

- **Momentum equation**: applied with Newton's second Law of Motion to a fluid element [12], [13], [14].

$$\frac{\partial \rho \vec{}}{\partial} + \nabla (\rho\ \vec{u} \cdot \vec{u}) = -\nabla + \nabla \sigma - f \tag{2.4}$$

Where, f is the body force vector (i.e., the gravity and centrifugal accelerations), p defines the pressure and σ represents a surface stress tensor.

2.1.2 Incompressible flow of Newtonian fluids

The governing equations for a steady-state, single phase flow are [15]:

- **Continuity equation:**

$$\nabla (\rho\ \vec{u}) = 0 \tag{2.5}$$

- **Momentum equation:**

$$\rho \underbrace{\overbrace{(\vec{u} \cdot \nabla \vec{})}^{(I)}}_{(II)} = \underbrace{\overbrace{-\nabla}^{(III)}}_{(IV)} + \underbrace{\nabla^2 \cdot \vec{u}}_{(V)} + \underbrace{f}_{(VI)} \tag{2.6}$$

The completely term (I) on the left side of the equation 2.6 represents the inertia (per volume). The second term (II) on the left side is the convective acceleration. The first term (III) on the right side represents the divergence of stress, where the term (IV) is the pressure gradient and the term (V) describes the viscosity of the fluid. The final term of the right side (VI) is the other body forces (i.e., the gravity and centrifugal accelerations).

Due to the constant density and temperature the equations 2.5 and 2.6 can be simplified [12]:

- **Continuity equation:**

$$\nabla \vec{u} = 0 \tag{2.7}$$

- **Momentum equation**

$$\vec{u} \cdot \nabla \vec{} - v\ \nabla^2 \vec{u} = -\nabla + f \tag{2.8}$$

Where the ν represents the kinematic viscosity, $\vec{u}$ describes the velocity vector, p is the pressure and f describes the body force vector (e.g., gravity acceleration or centrifugal force).

2.2 Turbulence Models

2.2.1 What is Turbulence ?

The turbulent flow is a type of fluid (e.g., gas or liquid) flow in which the fluid at a point moves in irregular directions [16], [17]. The most kinds of fluid flows are turbulent flows, Table 2-1 shows the characteristics of fluid turbulence observed in nature:

Irregularity	Flow too complicated to be fully described with detail and economically. Deterministic approaches are impossible (to date).
Three Dimensionally	Turbulence is always rotational and flow fluctuations have three-dimensional components even if the mean flow is one- or two-dimensional. Turbulence flows always exhibit high levels of fluctuating vorticity.
Diffusivity	Rapid mixing and increased rates of momentum, heat, mass transfer, etc.
Dissipation	The kinetic energy of turbulence is dissipated to heat under the influence of viscosity since viscous shear stresses perform mechanical deformation work that increases the internal energy of the fluid. The energy source to produce turbulence must come from the mean flow by interaction of shear stresses and velocity gradients.

Table 2-1: Characteristics of fluid turbulence observed in nature [17], [18], [19].

The turbulent flow occurs always at high Reynolds number (i.e. high flow velocity) and it is rotational. In laminar flow, the fluid motion is very orderly that it moves in straight lines parallel to the walls. It stays stable and changes not with time [20]. The laminar flow is appeared at low Reynolds number (i.e. low flow velocity), transitional flow is an intermediate flow condition between the laminar and turbulent flow. The initial condition of the transition to turbulent can be explained by considering the stability of laminar flow to small disturbances. The equation 2.9 represents a general equation for the calculation of Reynolds number.

- General equation for Reynolds number [20]:

$$\text{Re} = \rho \cdot u \cdot \frac{l}{\mu} \qquad (2.9)$$

Here, ρ is the fluid density, u represents mean velocity, l is the characteristic length and μ describes the dynamic viscosity.

The Reynolds number gives a measure of the ratio of inertial forces (i.e., associated with convective effects) and viscous forces [20]. For a given value of the Reynolds number, the critical Reynolds number defines the boundary between laminar and turbulent flow. If Re > Re$_{krit}$, the turbulent flow is generated and if Re < Re$_{krit}$, the flow is laminar or the transitional flow between laminar and turbulent (i.e. boundary layer). Reynolds number can only be compared with the same or similar geometrics.

Calculation of Reynolds number for the validation case 'Enclosed Rotating Disc':

- Reynolds number for the validation case 'Enclosed rotating disc' [1]:

$$\text{Re} = U \cdot \frac{R}{\nu} = \omega \; R^2 = \frac{(\frac{2 \cdot \pi \cdot N}{60})}{\nu} \cdot R^2 \tag{2.10}$$

Where U is the circumferential velocity, R represents the radius, ω is the angular velocity, N is the rotation speed and ν describes the kinematic viscosity.

As can be seen in Table 2-2, there are a several literatures such as Geis [3], Dubbel [4], Sigloch [6], etc., which have investigated the delineation of laminar, transitional and turbulent flow with Reynolds number for the validation case 'Enclosed Rotating Disc'.

Sigloch	Re < $3 \cdot 10^5$	Couette flow
	Re > $3 \cdot 10^5$	Turbulent flow
Dubbel	Re < $3 \cdot 10^4$	Laminar flow
	Re = $3 \cdot 10^4$ - $6 \cdot 10^5$	Boundary layer
	Re > $6 \cdot 10^5$	Turbulent flow
Schlichting	Re < $3 \cdot 10^5$	Laminar flow
	Re > $3 \cdot 10^5$	Turbulent flow
Geis	Re < $3 \cdot 10^4$	Laminar flow, combined boundary layer
	10^4 < Re < 10^5	Laminar flow, discrete boundary layer
	10^5 < Re < $2 \cdot 10^6$	Turbulent flow, combined boundary layer
	Re > $2 \cdot 10^6$	Turbulent flow, discrete boundary layer

Table 2-2: The Reynolds numbers of Sigloch, Dubbel, Schlichting and Geis for the validation case 'Enclosed rotating disk'.

2.2.2 Turbulence modelling

There are numerous turbulence models available for application with CFD simulations. The turbulence models k - epsilon, RNG k - epsilon and SST k - omega are used for this work.

For the different turbulence models, the Figure 2-1 shows the classification of them. In Table 2-3, the strengths and weaknesses about the different turbulence models are listed.

Figure 2- 1 : Classification of turbulence models in OpenFOAM [18].

Model Names	Advantages	Disadvantages
Spalart-Allmaras	. A small amount of calculations . The better results for complexity boundary problems	. The results with Spalart-Allmaras model are not extensively tested. . Lack of sub-models . Consideration without combustion and buoyancy equations

Standard $k - \cdot$	. More applications, good for moderately complex behaviour
	. Economical
	. More accumulated performance of data.
	. Results are considerable accuracy
		. Mediocre results for complex flow with:
		  - severe pressure gradients,
		  - strong streamline curvature,
		  - swirling flows.

RNG $k - \cdot$	. Advantages are similar as above (i.e. Standard $k - \cdot$).
	. Moreover, better solutions for the following problems:
	  - jet impingement,
	  - separated flows,
	  - secondary flows,
	  - swirling flows.
		. This model is subjected to limitations.
		. Due to isotropic eddy viscosity assumption.

SST $k - \omega$	. Advantages are similar as above (i.e. Standard k – ε). . More accurate result for a wide class of flows than $k - \omega$ model. . Incorporates a damped cross-diffusion . Better for the near-wall and far-Field zones.	. Be used in the Low-Reynolds number turbulence model. . Too sensitive to the inlet free-stream turbulence properties . Produce too large turbulent levels in stagnation regions and regions with strong acceleration . This tendency is much less pronounced than with a normal k-ε model through.
RSM	. Consideration with more detail of physical mechanism. . Consideration with anisotropy Effects of turbulent flow.	. Requires more cpu effort (2-3 times) . Tightly coupled momentum and turbulence equations.

Table 2-3: The advantages and disadvantages for different turbulence models [20].

Reynolds averaged Navier-Stokes equations are basic equations for eddy viscosity and Reynolds stress turbulence models. The **Spalart-Allmaras** model is a rather new one-equation mixing length model that it is defined by an algebraic formula, which is designed for external aerodynamic boundary layer economical computations [20], [21]. But it is weakly for this model to accurately describe flows involving separation and recirculation. The case 'Enclosed Rotating Disc' presents a rotor with opposing stationary walls in a closed system, wherefore recirculation would be generated. It can be said, that the turbulence model Spalart-Allmaras is irrelevant for this project. The **Reynolds Stress Model (RSM)** is based also on the Reynolds averaged Navier-Stokes equation that accounts for anisotropic Reynolds's stresses in the flow (e.g., for heat transport) [22].

The two-equations turbulence models **Standard k-epsilon**, **RNG (i.e. renormalization group) k-epsilon** and **SST k-omega** are selected for the preliminary simulations for the validation case, which are defined by turbulent kinetic energy k and the dissipation rate of the turbulent kinetic energy ε in the k-epsilon models or the specific dissipation rate ω in the k-omega model [21]. The k-epsilon model has its limitations, but it is the simplest and widely used in OpenFOAM. The RNG k-epsilon model is designed for moderate rotation, swirling flows and complex shear flows and has a higher computational overhead. The SST k-omega model can improve the k-epsilon model at the near-wall region [21]. The detailed introduction about these two k-epsilon and the SST k-omega turbulence models would be presented in the following chapters.

2.2.3 K – epsilon turbulence models

The Standard k-epsilon model is a two-equation turbulence model, which one is defined by turbulent kinetic energy k (i.e., to express the turbulence velocity, per unit mass) and the other one is defined by the dissipation rate of the turbulent kinetic energy ε (i.e., to express the

turbulence length scale). The instantaneous kinetic energy $k(t)$ of a turbulent flow is the sum of mean kinetic energy K and turbulent kinetic energy k [21].

$$k(t) = K +$$
$$K = \frac{1}{2}(U^2 + {}^{r2} + V^2) \tag{2.11}$$
$$k = \frac{1}{2}(\overline{u'^2} + \overline{{}'^2} + \overline{v'^2})$$

Here, in equation 2.11, the U, V and W describe the mean velocity components. u', v' and w' are the fluctuating velocity components. The k defines the kinetic energy, which is produced by the flow fluctuations [20].

The k - epsilon model contracts that the averaged flow can be influenced by the turbulent viscosity μ and Reynolds stresses τ_j acting on it [20]. If the turbulent kinetic energy k and the dissipation rate of turbulent kinetic energy ε are given than the turbulent viscosity can be expressed as:

$$\mu_t \propto l \propto {}^{\frac{1}{2}} \frac{k^{\frac{3}{2}}}{\varepsilon} = \frac{k^2}{\varepsilon} \tag{2.12}$$

$$\tau_{j} = - \qquad = \left(\begin{array}{cc} & \\ & \end{array} \right) \tag{2.13}$$

The turbulent viscosity (μ) is isotropic, in other words that the ratio between Reynolds stress and mean rate of deformation is the same in all directions. In OpenFOAM the turbulent kinematic viscosity is used with $v_t = \frac{u_t}{\rho}$. The Reynold's stresses (τ_j) in equation 2.13 are considered proportional to the dissipation rate reduced by the eddy turbulent kinetic energy. The δ_{j} is a Kronecker delta, which assures that the normal Reynolds stress are each appropriately accounted with it, $\delta_{j} = 0$ when $i \neq {}^{'}$, $\delta_{j} = 1$ when $i = {}^{'}$.

The length scale l for the large vortices is used in the k - epsilon model to define the length scale ε for small vortices, for which a transport equation is used in the model [23]. The dissipation rate of turbulent kinetic energy ε is presented in equation 2.14 and in equation 2.15:

$$\varepsilon = \frac{k^{\frac{2}{3}}}{l} \tag{2.14}$$

$$\varepsilon = \frac{\mu}{\rho} \cdot \frac{\partial \overline{'}_i}{\partial_k} \cdot \frac{\partial \overline{'}_j}{\partial_k} \tag{2.15}$$

2.2.3.1 Standard k – epsilon turbulence model

- **Turbulent kinetic energy k**

The transport form of the turbulent kinetic energy k is as following [21], [23]:

$$\frac{\partial\rho}{\partial} + \frac{\partial}{\partial_j}(\rho\,'_jk) = \frac{\partial}{x_i}[\rho\,'_i\cdot\frac{\overline{u'_iu'_j}}{2} + \,')] - \rho\,\overline{'_iu'_j}\cdot\frac{\partial\,'_i}{\partial_j} - \mu\,\frac{\partial\overline{'}_i}{\partial\,'_j}\cdot\frac{\partial\overline{'}_i}{\partial\,'_j} \tag{2.16}$$

(I)   (II)   (III)   (IV)   (V)

The transient term (I) at the left side describes the accumulation of k, the second term (II) at the left side is the transport of k by convection, the first term at the right side (III) presents the transport of k by pressure, the term (IV) is the turbulence production of k due from the mean flow and the last term at the right side defines the rate of viscous dissipation of k.

The terms (III) and (V) at the right side of equation 2.16 are replaced using scalar diffusion transport terms for (III) and the time-averaged term for (V) to result in the following :

$$\frac{\partial}{\partial} + \frac{\partial}{\partial} \quad - \frac{\partial}{\partial}[\quad] \quad \frac{\partial}{\partial} - \cdot \tag{2.17}$$

The σ is a turbulent Schmidt number of the k equation, which stays constant. The i or j = 1 corresponds to the x-direction, i or j = 2 the y-direction and i or j = 3 the z-direction.

- **Dissipations rate of the turbulent kinetic energy ε**

The transport equation for the dissipation rate of turbulent kinetic energy is as following [21], [23]:

$$\frac{\partial}{\partial} + \frac{\partial}{\partial} \quad - \frac{\partial}{\partial}[\quad] \quad (\quad \frac{\partial}{\partial} \quad) \tag{2.18}$$

(I)   (II)   (III)   (IV)   (V)

It is similar to the transport equation for k, the term (I) at the left side represents the rate of change of ε, the term (II) is the rate of destruction of ε, the first term at the right side describes the diffusive transport of ε, the term (IV) is the rate of production of ε and the last term (V) at the right side defines the transport of ε by convection.

- **The fitting equations about k and ε for this project**

For the steady-state flow of an incompressible fluid at constant temperature the transient terms in equations 2.17 and 2.18 can be neglected. The simplified equations of the turbulent kinetic energy k and the dissipation rate of turbulent kinetic energy ε are following:

- The turbulent kinetic energy k for this project:

$$\frac{\partial}{\partial x_i}(\rho U_i k) = \frac{\partial}{\partial x_i}[\frac{\mu_t}{\sigma_k}\frac{\partial k}{\partial x_i}] - \rho \overline{u'_i u'_j}\frac{\partial U_i}{\partial x_j} - \rho \cdot \varepsilon \tag{2.19}$$

$$\frac{\partial}{\partial x_i}(\rho U_i \varepsilon) = \frac{\partial}{\partial x_j}[\frac{\mu_t}{\sigma_\varepsilon}\frac{\partial}{\partial x_i}] + C_1 \frac{\varepsilon}{k}(-\overline{u'_i u'_j}\cdot\frac{\partial U_i}{\partial x_j}) - C_2 \rho \frac{\varepsilon^2}{k} \tag{2.20}$$

The σ, C_1 and C_2 are constants of the ε equation, which are presented in Table 2-4.

C_u	C_1	C_2	σ	σ
0.09	1.44	1.92	1.0	1.3

Table 2-4: Constants for the equations of k – epsilon model.

- **The equation of turbulent viscosity μ_t [23]**

The eddy viscosity in the mixing length model can be defined by:

$$\mu_t = C\rho \vartheta l = \rho C_\mu \frac{k^2}{\varepsilon} \tag{2.21}$$

Where, C_μ is a dimensionless constant.

2.2.3.2 RNG k – epsilon turbulence model

The RNG k – epsilon model was derived using a statistical technique (called also renormalization group theory) [20]. This model is similar in form to the standard k – epsilon model, but it has more strengths than the standard k – epsilon model (see the table 2.2-3). The RNG k – epsilon model has an additional term in its ε - equation, which can account for the different scales of motion through changes to the production term [23], [24].

- **Turbulent kinetic energy** k

$$\underbrace{\frac{\partial \, k}{\partial t}}_{(I)} + \underbrace{\frac{\partial}{\partial x_i}(\rho U_i k)}_{(II)} = \underbrace{\frac{\partial}{\partial x_i}[(\mu + \frac{u_t}{\sigma_k}) \cdot \frac{\partial k}{\partial x_i}]}_{(III)} - \underbrace{\rho \overline{u'_i u'_j} \frac{\partial U_i}{\partial x_j}}_{(IV)} - \underbrace{\rho \cdot \varepsilon}_{(V)} \qquad (2.22)$$

The transient term (I) at the left side describes accumulation of k, second term (II) at the left side is the transport of k by convection, the first term at the right side (III) presents the generation, the term (IV) is the diffusion and the last term at the right side defines the rate of viscous dissipation of k.

- **Dissipations rate of the turbulent kinetic energy** ε

$$\underbrace{\frac{\partial \, '}{\partial}}_{(I)} + \underbrace{\frac{\partial}{\partial}(\rho \, U_i k)}_{(II)} = \underbrace{\frac{\partial}{\partial}[(\mu + \frac{u}{\sigma}) \cdot \frac{\partial}{\partial}]}_{(III)} - \underbrace{\gamma_{1\varepsilon}^{*} \cdot \frac{\varepsilon}{k} \cdot (\rho \overline{u'_i u'_j} \frac{\partial U_i}{\partial_j})}_{(IV)} - \underbrace{\gamma_{2\varepsilon} \rho \frac{?}{k}}_{(V)} \qquad (2.23)$$

The terms (I), (II) and (III) are the same as the equation 2.23, the term (IV) resents also the diffusion and the last term at the right side defines the destruction.

- **The fitting equations about** k **and** ε **for this work:**

$$\frac{\partial}{\partial x_i}(\rho U_i k) = \frac{\partial}{\partial x_i}[(\mu + \frac{u_t}{\sigma_k}) \cdot \frac{\partial k}{\partial x_i}] - \rho \overline{u'_i u'_j} \frac{\partial U_i}{\partial x_j} - \rho \cdot \varepsilon \qquad (2.24)$$

$$\frac{\partial}{\partial}(\rho \, U_i k) = \frac{\partial}{\partial}[(\mu + \frac{u}{\sigma_\varepsilon}) \cdot \frac{\partial}{\partial}] - \gamma_{1\varepsilon}^{*} \cdot \frac{\varepsilon}{k} \cdot (\rho \overline{u'_i u'_j} \frac{\partial U_i}{\partial_j}) - \gamma_{2\varepsilon} \cdot \rho \frac{?}{k} \qquad (2.25)$$

There are several constants, which are derived explicitly in the RNG procedure. The constants of RNG k – epsilon model are presented in Table 2-5:

C_u	C_1	C_2	$\sigma_{\ }$	$\sigma_{\ }$
0.0845	1.42	1.68	0.7194	0.7194

Table 2-5: Constants for the equations of RNG k – epsilon model.

2.2.3.3 SST k – omega turbulence model

The turbulence model SST (Shear-Stress Transport) k-omega is a type of hybrid model, which combines two models in order to better calculate the flow in the near-wall region [22], [23]. This model is designed in response to the problem of the k – epsilon model's unsatisfactory near-wall performance for boundary layers with adverse pressure gradient. The SST k – omega

model is similar to the k – epsilon model, but the SST k – omega model uses a turbulence frequency variable omega (ω), which takes the place of ε as the second variable [22], [23].

- **Turbulent kinetic energy k for SST k – omega model**

$$\underbrace{\frac{\partial(\rho k)}{\partial}}_{(I)} + \underbrace{\frac{\partial}{\partial}(\rho U_i k)}_{(II)} = \underbrace{\frac{\partial}{\partial}[(\mu + \frac{u_t}{\sigma_k}) \cdot \frac{\partial}{\partial}]}_{(III)} - \underbrace{(2\mu_t \frac{\partial^T_i}{\partial_j} \cdot \frac{\partial^T_i}{\partial} - \frac{2}{3}\rho k \frac{\partial^T_i}{\partial_j} \delta_{ij})}_{(IV)} - \underbrace{\beta^* \rho k \omega}_{(V)} \quad (2.26)$$

The transient term (I) at the left side is a transient term that it describes accumulation of k, second term (II) at the left side is the transport of k by convection, the first term at the right side (III) presents the turbulent diffusion transport of k, the term (IV) defines the rate of production of k and the last term at the right side is the rate of dissipation of k. σ and β are equation constants (see table 2.2-7).

- **Turbulent frequency ω for SST k – omega model**

$$\underbrace{\frac{\partial \rho \omega}{\partial}}_{(I)} + \underbrace{\frac{\partial}{\partial}(\rho^{\,}_i \omega)}_{(II)} = \underbrace{\frac{\partial}{\partial}[(\mu + \frac{\mu}{\sigma_{\omega,1}}) \cdot \frac{\partial \omega}{\partial}]}_{(III)} + \underbrace{\gamma_2 (2\rho \frac{\partial^T_i}{\partial_j} \cdot \frac{\partial^T_i}{\partial_j} - \frac{2}{3}\rho \omega \frac{\partial^T_i}{\partial_j} \delta_j)}_{(IV)} - \underbrace{\beta_2 \rho \omega^2}_{(V)} + \underbrace{2\frac{\rho}{\sigma_{\omega,2}} \frac{\partial}{\partial_k} \frac{\partial \omega}{\partial_k}}_{(VI)}$$

$$(2.27)$$

Here, the terms (I) to (V) of the general equation 2.27 are the usual terns for accumulation, convection, diffusion, production and dissipation of ω. The last term at the right side presents the cross-diffusion, which plays a role in the transition of the modelling from ε to ω.

- **The fitting equations about k and ω for this work**:

$$\frac{\partial}{\partial}(\rho U_i k) = \frac{\partial}{\partial}[(\mu + \frac{u_t}{\sigma_k}) \cdot \frac{\partial}{\partial}] - (2\mu_t \frac{\partial^T_i}{\partial_j} \cdot \frac{\partial^T_i}{\partial} - \frac{2}{3}\rho k \frac{\partial^T_i}{\partial_j} \delta_{ij}) - \beta^* \rho k \omega \quad (2.28)$$

$$\frac{\partial}{\partial}(\rho^{\,}_i \omega) = \frac{\partial}{\partial}[(\mu + \frac{\mu}{\sigma_{\omega,1}}) \cdot \frac{\partial \omega}{\partial}] + \gamma_2 (2\rho \frac{\partial^T_i}{\partial_j} \cdot \frac{\partial^T_i}{\partial_j} - \frac{2}{3}\rho \omega \frac{\partial^T_i}{\partial_j} \delta_j) - \beta_2 \rho \omega^2 + 2\frac{\rho}{\sigma_{\omega,2}} \frac{\partial}{\partial_k} \frac{\partial \omega}{\partial_k} \quad (2.29)$$

In the equations, $\sigma_{\,}$, $\gamma_{\,}$, β and $\sigma_{\,}$ describe the equation constants, which are:

β	$\beta_{\,}$	$\sigma_{\,}$	$\sigma_{\,}$	$\sigma_{\,}$	$\gamma_{\,}$
0.09	0.083	1.0	2.0	1.17	0.44

Table 2-6: Constants for the equations of SST k – omega model.

2.2.4 Summary of turbulence models

The two k – epsilon models, Standard k – epsilon, RNG k – epsilon and SST k – omega are used for the validation case 'Enclosed Rotating Disc' in this thesis to preview which turbulence model is better for the situation of this work. After the consideration about the applicability, availability, the computationally economic and the accuracy of the different turbulence models, the RNG k – epsilon model was applied to carry out the further simulations with OpenFOAM in this thesis.

The RNG k – epsilon model provides better results for separated, secondary and swirling flows and decides not only the problems for the motion with high Reynolds number such as the Standard k – epsilon model, moreover it can account for low Reynolds number effects and provide an appropriate treatment of the near-wall region [22], [23]. The k – epsilon model is only suitably for the free-stream turbulent region far away from the wall. The results with the SST k – omega model depend strongly on the mesh resolution near the wall and it is not computational economically [24].

2.2.5 Near–wall treatment for turbulent flows

Turbulent flows are affected by the presence of walls. Very close to the wall viscous damping and kinematic blocking reduces the velocity fluctuations. In the outer part of the near-wall region, the turbulence is strongly affected by the production of kinetic energy due to large velocity gradients. The near-wall modelling has therefore a high impact on the accuracy of the numerical solutions [7].

The following Figure 2-2 and Figure 2-3 show the turbulent boundary layer and the subdivisions of the near-wall region (plotted in semi-log coordinates). In the innermost layer, the viscous sublayer, the flow is laminar and viscosity plays the dominant role in momentum transfer ($y^+ = 0$ - 5). In the buffer layer ($y^+ = 5$ - 60) both effects, viscosity and turbulence, are important. The turbulent motion of the fluid appeared in the fully-turbulent or log-layer region ($y^+ > 60$). Here the turbulence plays the application role [25].

Figure 2- 2 : Turbulent boundary layers [20].

Figure 2- 3 : Subdivisions of the Near-wall region [20].

The k – epsilon models are designed for turbulent core flows, somewhat far away from walls. Therefore considerations have to been made for the treatment of the near-wall region.

Wall-functions can be used to bridge the viscosity-affected region between the wall and the fully-turbulent region [7]. The wall-function uses an empirical law to circumvent the inability of the k – epsilon model to predict a logarithmic velocity profile near a wall [23].

The wall-function depends on the Reynolds number, which is based on the distance to the wall using the friction velocity [7].

- The friction velocity is:

$$u_\tau = \frac{\tau_w}{\rho})^{\frac{1}{2}} \tag{2.30}$$

- The velocity at $y^+ = \mathfrak{N}u_\tau y / \mu$ using the friction velocity:

$$u^+ = \frac{U}{u_\tau} \tag{2.31}$$

Where the u_τ is the friction velocity, u^+ describes the dimensionless velocity.

The law of the wall for mean velocity yields

$$u^+ = \frac{1}{\kappa} \ln(y^+) + \mathfrak{I} \tag{2.32}$$

Where, u^+ describes the dimensionless velocity and y^+ defines the dimensionless wall distance.

κ is the Karman's constant, which is 0.4187. C presents also a constant, is about 5.1.

The logarithmic law for mean velocity is known to be valid for $30 < y^+ < 300$. For meshes with $y^+ < 11.63$ the laminar stress-strain relationship is used [7].

The wall-function can substantially save the computational resources in most flows with high Reynolds number. The reason for this is that the solution variables change most rapidly in the viscosity-affected near-wall region and it is not necessary to be resolved. The wall-function is economical, robust and accurate that it is a popular practical option to solve the problems within the near-wall region [9].

3   OpenFOAM Overview

OpenFOAM (OpenField Operation and Manipulation) is free software under the GNU General Public License (GPL), which can be downloaded from the website http://www.opencfd.co.uk. The development of CFD tools was started at Imperial College in London and they have the GPL license since 2004 with the announcement of OpenFOAM 1.0 [26], [27]. The OpenFOAM 1.6 version is the latest version, which released in July 2009. And it was evaluated in the course of this thesis.

OpenFOAM is primarily a C++ toolbox for the customization and extension of numerical solvers for continuum mechanics problems, including computational fluid dynamics (CFD). It is used for a wide range of problems involving complex fluid flows to solid dynamics. The applications can be divided into two different categories: solvers and utilities, of which the former perform the actual calculations and the latter, provide a range of functionalities for pre– and post-processing such as ParaView (see Figure 3-1) [26].

Figure 3- 1 : The structure of OpenFOAM [27].

3.1   Structure of OpenFOAM cases

The basic file structures of the cases in OpenFOAM are similar, which have only the slight differences stemming from the particular choice of solver. The basic file structure is based on the main directories **0**, **constant** and **system** and the subdirectories with specific tasks. Table 3-1 shows the structure of the case, which is specific for this thesis:

Table 3-1: The file structure of OpenFOAM case for this work.

Due to setting up a simulation with OpenFOAM various steps must be undertaken, such as:

- Generating the geometry and mesh - Setting initial conditions. (Subsection 3.1.1)

- Select the turbulence and fluid properties. (Subsection 3.1.2)

- Select the numerical schemes and algorithms for the solution of the system equations. (Subsection 3.1.3)

- Fix the general simulations settings. (Subsection 3.1.4)

3.1.1 Pre – processing – Geometry and Mesh generation

The mesh is created by using the dictionary file *blockMeshDict* [9]. This dictionary contains the following informations [23]:

- A header part about the version of the software, the validation case, the directories containing useful files and a scaling factor to convert the vertex coordinates.

- A second part which contains the vertex coordinates, the block definitions and a list of the patches in the following form:

  ➤ Vertices are defined by their three coordinates:

➢ The blocks are defined in this form: the vertices that they contain, and the characteristics (e.g., Number of cells in each direction, cell expansion ratio) [9] of the mesh:

```
blocks
(
    hex (0 1 2 3 20 21 22 23) (52 5 1) simpleGrading (1 1 1)
    hex (1 4 5 2 21 24 25 22) (4 5 1) simpleGrading (1 1 1)
    hex (4 6 7 5 24 26 27 25) set1 (8 5 1) simpleGrading (1 1 1)
```

```
    hex (10 12 16 14 30 32 36 34) (4 143 1) simpleGrading (1 1 1)
    hex (12 13 17 16 32 33 37 36) set2 (8 143 1) simpleGrading (1 1 1)
    hex (13 18 19 17 33 38 39 37) (114 143 1) simpleGrading (1 1 1)
);
```

With the e.g. **set1** and **set2** cell sets can be selected.

➢ The **patches** are defined by a name and the vertices used to create them. Each patch in the list is a compound entry containing the patch type and a list of block faces that make up the patch [9]:

```
patches
(
    wall spurkopf_1
    (
        (13 33 38 18)
    )
    wall spurkopf_2
    (
        (13 33 27 7)
    )
```

```
patch oelspiegel
    (
        (19 39 38 18)
        (9 29 28 8)
    )
    empty frontAndBack
    (
        (0 3 2 1)
        (1 2 5 4)
        (4 5 7 6)
        (6 7 9 8)
        (11 10 2 3)
        (5 12 13 7)
        (11 15 14 10)
        (10 14 16 12)
        (12 16 17 13)
        (13 17 19 18)
        (20 21 22 23)
        (21 24 25 22)
        (24 26 27 25)
        (26 28 29 27)
        (23 22 30 31)
        (25 27 33 32)
        (31 30 34 35)
        (30 32 36 34)
        (32 33 37 36)
        (33 38 39 37)
    )
);
```

Utility for mesh generation in pre – processing ➔ blockMesh

BlockMesh <root> -case <casename>

The utility **blockMesh** generates the mesh and writes out the mesh data to points and faces, cells and boundary files from the input dictionary *blockMeshDict* located in the **constant/polyMesh** directory for the case. The *boundary* dictionary contains the base types (e.g., patch, wall, empty) and has the following form:

```
11
(
    spurkopf_1
    {
        type        wall;
        nFaces       114;
        startFace    58262;
    }
    spurkopf_2
    {
        type        wall;
        nFaces       51;
        startFace    58376;
    }
```

```
    oeltopf
    {
        type        wall;
        nFaces       377;
        startFace    58651;
    }
    axis
    {
        type        wall;
        nFaces       178;
        startFace    59028;
    }
    oelspiegel
    {
        type        patch;
        nFaces       148;
        startFace    59206;
    }
    frontAndBack
    {
        type        empty;
        nFaces       58808;
        startFace    59354;
    }
)
```

After generating the mesh the utility **checkMesh** can be used to check the validity of the mesh:

Utility to check the generated mesh in pre – processing ➔ **_checkMesh_**

checkMesh <root> -case <casename>

OpenFOAM is strictly a 3D-code and therefore a special utility has to be used to create an axisymmetric mesh [9].

Utility to create an axi – symmetric grid in pre – processing ➔ **_makeAxialMesh_**

makeAxialMesh <root> -case <casename>

The application of the utility *makeAxialMesh* needs two inputs beyond the uniquitous root and case inputs [9], [29]:

- Name of the boundary that is the symmetry axis

- Name of the boundary that has to be split into two wedge-boundaries (i.e., front and back)

It generates a wedge of 5° angle and 1 cell thick running along the plane of symmetry straddling one of the coordinate planes. The type of the axis-patch is set to *empty*, the type of the newly created patches to *wedge* type. Figure 3-2 shows an axisymmetric geometry using the wedge patch type as an example.

Figure 3-2 : Axisymmetric geometry using the wedge path type [29].

The initial volume fields and derived boundary types are contained in the time directory **0**. The directory contains several files for the different volume fields, e.g. pressure p, velocity U, turbulent kinetic energy k, dissipation rate of turbulent kinetic energy ε, and so on. The figure below shows the boundary field for the pressure p as an example.

There are 3 principal entries in field data files:

- *dimensions* focus on the dimensions of the field, here kinematic pressure with dimension $\mathrm{m}^2/\mathrm{s}^2$.

- The *internalField* data which can be uniform, described by a single value; or nonuniform, where all the values of the field must be specified.

- The *boundaryField* data that includes boundary conditions and data for all the boundary patches.

  ➢ In this project, the boundary condition for pressure p has set as **zeroGradient.** It means that the normal gradient of pressure is zero at walls.

  ➢ *wedge* is used for front and back in axisymmetric cases.

3.1.2 Fluid properties

There is a **transportProperties** dictionary for the physical properties for the case [9]:

```
transportModel  Newtonian;

nu        nu [0 2 -1 0 0 0 0] 3.52e-05;
```

For our project, the transportModel is selected a linear viscous fluid model (i.e., Newtonian). The nu presents the kinematic viscosity. If the fluid flow is turbulent such as in this work, there are also **turbulenceProperties** and **RASProperties** dictionaries in which the turbulence model (e.g., RNG k – epsilon model) can be chosen and even the coefficients for every single model can be edited.

3.1.3 Schemes and solution algorithms

The **fvSolution** dictionary in the **system** directory controls solvers, tolerances and algorithms for the systems of equations solved to obtain every variable [9]. Also under-relaxation factors for the variables are defined in this file. A part of **fvSolution** dictionary is following:

```
solvers
{
  P
  {
          solver            GAMG;
          tolerance         1e-06;
          relTol            0.1;
          smoother          GaussSeidel;
          nPreSweeps              0;
          nPostSweeps             2;
          cacheAgglomeration    true;
          nCellsInCoarsestLevel   10;
          agglomerator
          faceAreaPair;
          mergeLevels             1;
  }
  U
  {
        solver                PBiCG;
        preconditioner        DILU;
        tolerance             1e-05;
        relTol                0.1;
  }
```

The gradient solver Generalised Geometric-Algebraic Multi-grid (GAMG) is used for the pressure equation. For the velocity field (U) and the turbulence equations (k and epsilon) the Preconditioned (bi-) conjugate gradient solver PBiCG was used.

The **fvSchemes** dictionary determines the numerical schemes for terms appearing in the constituent equations, e.g. interpolation schemes, time derivatives, gradient, divergence, Laplacian terms, etc [9]. The figure below shows a detail of the dictionary:

```
gradSchemes
{
    default                 Gauss   linear;
    grad(U)                 Gauss   linear;
}
divSchemes
{
    default                         none;
    div(phi,U)              Gauss   upwind;
laplacianSchemes
{
    default                         none;
    laplacian(nuEff,U) Gauss linear corrected;
    laplacian((1|A(U)),p)   Gauss       linear
corrected;
}
interpolationSchemes
{
    default                         linear;
    interpolate(U)                  linear;
}
```

3.1.4 Simulation control

In the **system** directory, the **controlDict** dictionary controls essential settings for the simulation as the used solver (**application**), the starting time (**startTime**), the end time (**endTime**) and the time step (**deltaT**) [9]. Furthermore, the timing of writing output, its format and compression are determined. A part of the **controlDict** dictionary would look like:

```
application     simpleFoam;

startFrom       latestTime;

startTime       latestTime;

stopAt          endTime;

endTime         50000;

deltaT          1;

writeControl    timeStep;

writeInterval   2500;
```

When all of the settings are made, the case can be started from the command line using one of the solvers [9], such as simpleFoam. To run this solver on the described case the following command is used.

simpleFoam <root> -case <casename>

3.2 OpenFOAM Solver – simpleFoam

OpenFOAM is supplied with numerous pre-configured solvers, utilities and libraries [9]. For this thesis the steady-state solver **simpleFoam** for incompressible, turbulent flow was chosen. The solver is based on the SIMPLE algorithm described below.

The SIMPLE (Semi-Implicit Method for Pressure-Linked Equations) algorithm is a guess-and-correct technique to determine the values for pressure in a staggered grid. It is based on the Navier-Stokes equation with an iterative procedure, which is summed as in Figure 3-3 [21], [23], [30]:

Figure 3- 3 : Description of the steps in the algorithm [21], [23], [30].

- **START**

  Set the boundary conditions, such as estimation of a starting guess for the pressure field $p*$.

- **STEP 1**

  Solve the discretised momentum equation to compute the velocity components $u*,v*$ in the intermediate velocity field. This step results in finding values for $u*,v*$ are based on $p*$.

- **STEP 2**

  The pressure correction p' is defined as the difference between the correct pressure field p and the guessed pressure field $p*$, and for the velocity is also the same theory:

  $$p = p*+p'$$
  $$u = u*+u' \tag{3.1}$$

  The calculation of pressure correction p' is done with the discretised continuity equation by finding the mass imbalance b' and between total mass flow inflow and the total mass flow outflow of the guessed velocities.

- **STEP 3**

  Correct the pressure and velocity components using the pressure correction, where the pressure correction p' is added to the initial guessed pressure $p*$, to get the new pressure field p :

  $$p = p*+\alpha_r p' \tag{3.2}$$

  If $\alpha_r = 1$ → the guessed pressure $p*$ is corrected by p'.

  If $\alpha_r = 0$ → apply no correction at all, which is also undesirable.

  If $0 < \alpha_r < 1$ → add guessed pressure field $p*$ a fraction of the correction pressure field p' that is large enough to move the iterative improvement process forward, but small enough to ensure stable computation.

  In OpenFOAM, the value of α_r (for pressure) is equal 0.3 and α_r (for velocity) is equal 0.7 [21], [23], [30].

- **STEP 4**

    The other discretised transport equation is solved with the PBiCG (Preconditioned (bi-) conjugate gradient) method to obtain the values for the remaining scalar variables ϕ.

- **CONVERGENCE**

    The outputs are tested for convergence after the step 4. It means that the mass imbalance is very close to zero and if it is not within the value required for convergence, the program loops back to the beginning. The further calculation is used with the newly calculated pressure, velocity and other scalar values as the next starting guess. The calculation process continues until convergence (i.e., iteration) occurs.

3.3 Post – processing with Utilities and ParaView

The utilities of OpenFOAM are divided into supporting pre – processing and post – processing tasks. The utilities for pre – processing are previously described in chapter 3.3.1. The post – processing utilities and ParaView [9] are used to analyse the solutions.

3.3.1 Post – processing utilities

The utilities are used as:

$$<utility> <root> \text{ -}case <casename> \text{ }[\text{-}optionalParameters]$$

Where <utility> defines the name of the utility (e.g., calcMassFlow), <root> is the path to the root directory and <casename> presents the path of the actual case, relative to the directory.

Table 3-2 presents the types, distributions and applications of the different utilities, which are used in this project [9]:

Types of utilities	Distribution	Name	Application
Standard utilities	. Post – processing data converters.	- foamtoVTK	Legacy VTK file format writer
	. Miscellaneous post – processing	-ptot	The total pressure is calculated for each time.
		-wallGradU	Calculates and writes the gradient of velocity at the wall.

		-wallShearStress	Calculates and writes the wall shear stress, for the specified times.
	. Post – processing at the walls	-yPlusRAS	Calculates and reports the values of y^+ for all wall patches, for the specified times.
Own utilities	. Post – processing for our calculations	-calcMassFlow	Calculates and reports the mass fluxes at boundaries and faceSets.
		-calcPressureDifference	Calculates and writes the static or total pressure differences at boundaries.
		-torqueIncompressible	Calculates and reports the friction losses at walls for an incompressible fluid.

Table 3-2: Distribution and application of used utilities in this project.

3.3.2 ParaView

This chapter presents the design and features of a visualization tool, called ParaView [31]. It is an open-source, multi-platform data analysis and visualization application for 2D or 3D data sets. ParaView supports distributed computation models to process large data sets and provides a graphical user interface for the creation and dynamic execution of visualization tasks. The data exploration can be interactively in 3D or programmatically using ParaView's batch processing capabilities.

Figure 3-4 shows a part of the sample ParaView session, which offers the **Menu Bar**, **Tool Bar**, **Display Area** is at the right side, Left Panel at the left side (e.g. Pipeline Browser, Properties, Display and Information Planes) [31]. ParaView can be controlled from the **Left Plane**.

Figure 3- 4 : The session of ParaView [31].

The top of **Menu Bar** provides menu buttons for loading and saving data, creating sources and filters, viewing other windows, displaying help and the other standard functionally. The new filters are selected from the *Source* or *Filter* menu on the **Menu Bar**. In this thesis, the filter *glyphs* in the vector field was mostly used, which can be scaled by scalars, vector component or vector magnitude and can be oriented using a vector field [31]. It can be also selected quickly from **Tool Bar**, which includes the common filters. The *Filters* and **Tool Bar** provide the visualization algorithms for operating on datasets.

The top of the **Left Panel** presents the **Pipeline Browser** that provides a list of instantiated sources and filters. Below the selection window shows the Properties, Display and Information Planes [31]:

Pipeline Browser displays the visualization pipeline in the current ParaView session and lists the modules opened in ParaView.

- **Properties Panel** contains the input selections for the case, such as **Times** (i.e., at the top of right side), **Region** and **Field Status** [31]:

  ➢ With **Time** selected the time data for each time.

  ➢ With **Region Status** selected the internalMesh region and/or individual patches.

  ➢ With **Field Status** selected the fields read into the case module.

- **Display Panel** controls the visual representation of the selected module, e.g. colours and the geometry manipulation tools, etc.

- **Information Panel** gives case statistics such as mesh geometry and size [31].

The **Display Area** is at the right side in Figure 3-4 with white colour, where shows the 3D representation.

More information about ParaView can be found in ParaView tutorial and ParaView user's guide for the OpenFOAM version 1.6.

4 The validation case 'Enclosed Rotating Disc'

Rotor-stator flows are found commonly in turbomachinery. The study of these flows is important to further improve their efficiency. The geometry of those rotor-stator systems is often very complex in real world and most of these flows are turbulent. Many studies can be found in the literature [32], [33] which examined the turbulent behaviour of the flow between a rotating disc and an opposing stationary casing. Another important issue are the disc friction losses generated in such rotor-stator devices. Most of the published work [32], [33] is concerned with a simplified rotor-stator system, a disc rotating in a closed casing, and has been presented in terms of a non-dimensional torque coefficient. In this thesis such a simplified configuration has been investigated numerically to validate the suitability of the CFD code OpenFOAM for these fluid flows and to find an appropriate turbulence model for further simulations.

Subsection 4.1 presents the theoretical background of the validation case. In Subsection 4.2 the basic equations for the calculation of friction losses, the given operating and geometrical data and the existing empirical formulas are described. In Subsection 4.2.1 and 4.2.2, the numerical solutions for the two - and three - dimensional simulations with different turbulence models are presented and compared with empirical formulas and with results of a commercial CFD-software (FLUENT 12.0).

4.1 Theoretical background

In turbomachines it is often the case that rotating discs are enclosed in a casing, with the disc surrounded by radial and axial boundaries. Figure 4-1 shows the fluid flow related to rotating discs enclosed in the middle of a casing. Here, b is the thickness of the disc, s describes the axial spacing and t is the radial clearance (i.e., the spacing between the outside radius of disc and the outside radius of casing).

The flow structure is strongly influenced by the spacing s between the rotating disc and casing during rotation. When the spacing s is sufficiently large (i.e., multiple of the boundary layer thickness at the disc edge), two separated boundary layers can be found on the disc and casing. Due to the centrifugal forces the fluid in the boundary layer at the disc wall passes radially outward and in the boundary layer at the casing wall back in the direction of disc axis. The thicker fluid layer between the rotating disc and casing has no significant radial velocity and rotates in circumferential direction with some angular velocity as laminar core flow [1], [6]. A laminar shear flow with linear velocity distribution (Couette-flow) is generated, when the spacing s is small.

Figure 4- 1 : Enclosed rotating disc [1], [6].

After Daily and Nece [33] four different flow regimes can be identified. Regime I is characterised by merged laminar boundary layers with no 'core' between the rotating disc and the stationary wall, Regime II having separated boundary layer with a 'core' rotating with constant angular velocity, Regime III with merged turbulent boundary layers and Regime IV with separated turbulent boundary layers (see Figure 4-2). The flow is defined by the disc Reynolds number $\mathrm{Re} = R^2\omega\,\nu$ and the gap ratio $G = s/R$, where R is the radius, ω is the angular velocity, ν is the kinematic viscosity and s is the axial distance between the rotating disc and the casing [32].

Figure 4- 2 : Profiles of the radial velocity component in the gap between a rotating and stationary disc [32].

4.2 Overview of the validation case

Due to the rotationally symmetric geometry of the validation case it is possible to make the 2D simulations with the help of an axisymmetric model and the 3D simulations with a segment of 10° degrees with periodic boundaries.

Geometry:

The geometry of the validation case with large gap s is shown in Figure 4-3 with the parameters b = 200 mm, s = 50 mm, t = 15 mm and R = 900 mm.

Figure 4- 3 : Basic geometry for the validation case in OpenFOAM.

Here R is the disc radius, s is the axial clearance, b presents the length of casing and t describes the radial clearance.

The given operating and geometrical data, parameters and assumptions which are utilized as basic conditions for the CFD simulations are given in the table below:

Operating and Geometric data:

Operating data			Geometrical data		
Revolution speed	N [RPM]	100 to 2,000	**Outside radius of disc**	R [m]	0.9
Oil grade	ISO-VG Nr.	46	**Width of disc**	b [m]	0.2
Oil temperature	T [°C]	37	**Axial clearance**	s [m]	0.009 and 0.05
Density	ρ [kg/m³]	852.33	**Tip clearance**	t [m]	0.015
Dynamic viscosity	μ [mPa·s]	45.2	**Equivalent surface roughness**	k_s [m]	0

Table 4-1: The operating and geometric data for the validation case.

Parameters and assumptions:

The 2D and 3D CFD simulations with OpenFOAM 1.6 are conducted for the following parameters and assumptions:

➢ The axial spacings **s** between rotating disc and casing wall are equal 9 mm and 50 mm.

➢ Seven different revolution speeds (N = 100, 200, 300, 400, 500, 1000 and 2000 RPM).

➢ Smooth disc and casing walls (Equivalent surface roughness, i.e. $k_s \approx 0$).

➢ Isothermal, incompressible, single phase flow.

➢ Steady-state condition.

Basic data and equations for this case

The important goal for this validation case is to investigate the friction power losses at Disc Side A and Disc Side B (see Figure 4-3). The calculations are made with following equations:

$$P_{v,A(B)} = \omega \ T_{v,A(B)} \tag{4.1}$$

Where $\omega = \cdot \pi \cdot N / 60$, which presents the angular velocity with dimension $[1/s]$. $P_{v,A(B)}$ are the friction power losses for Disc Side A or Disc Side B with dimension $[W]$ and $T_{v,A(B)}$ describes the friction torque with dimension $[Nm]$, which is:

$$T_{v,A(B)} = \,_{v,A(B)} \cdot c_{A(B),T} \tag{4.2}$$

$c_{v,A(B)}$ are the coefficients of friction and dimensionless. $c_{A(B),T}$ are the friction coefficients for Disc Side A and Disc Side B, which are given:

$$c_{A,T} = \frac{\pi}{2} \cdot \rho \ U^2 \cdot R^2 \cdot b = \frac{\pi}{2} \cdot \rho \ \omega \cdot R^4 \cdot b \tag{4.3}$$

$$c_{B,T} = \frac{\pi}{2} \cdot \rho \ U^2 \cdot R^3 = \frac{\pi}{2} \cdot \rho \ \omega \cdot R^5 \tag{4.4}$$

where ρ is the density in $[kg/m^3]$, $U = \omega \ R$, which is the circumferential velocity in $[m/s]$, R is the radius in $[m]$ and b in equation 4.3 describes the casing width in $[m]$.

Inserting the equations 4.1, 4.3 and 4.4 into equation 4.2 gives the coefficients of friction c_A and c_B:

$$c_A = \frac{2 \cdot P_{v,A}}{\pi \cdot \rho \ \omega \cdot R^4 \cdot b} \tag{4.5}$$

$$c_B = \frac{2 \cdot P_{v,B}}{\pi \cdot \rho \ \omega \cdot R^5} \tag{4.6}$$

The friction loss depends strongly on the Reynolds number, which are defined for Disc Side A and Disc Side B as:

$$\text{Re}_A = \frac{'J \cdot t}{\nu} = \frac{\vartheta \cdot R \cdot t}{\nu} \tag{4.7}$$

$$\text{Re}_B = \frac{'J \cdot s}{\nu} = \frac{\vartheta \cdot R \cdot s}{\nu} \tag{4.8}$$

Where the ν is kinematic viscosity in Re_A (i.e., $kg/(s \cdot m)$) is the Reynolds number for Disc Side A and Re_B is the Reynolds number for Disc Side B.

At first, two-dimensional numerical simulations with an axisymmetric model for one revolution speed (400 U/min) and different turbulence models are made to evaluate a suitable turbulence model for this work (see the Chapter 4.2.1.1). Afterward the results of OpenFOAM are compared with the empirical formulas from Dubbel, Geis, GleitLRB, Schlichting and Sigloch for the rotor-stator system, which are shown in Table 4-2:

Source	Limitation	Coefficient of Friction	Approximation equations	
			Laminar	Turbulent
Dubbel	$Re < \cdot 10^5$	$c_s = 2/Re_s$	$c_s \approx \dfrac{0.6016}{\sqrt{Re}}$	
	$3\cdot 10^4 < Re < \cdot 10^5$	$c_s = \dfrac{3.78/2\pi}{\sqrt{Re}}$		
	$Re > \cdot 10^5$	$c_s = \dfrac{0.0714/2\pi}{\sqrt[5]{Re}}$		$c_s \approx \dfrac{0.0114}{\sqrt[5]{Re}}$
Geis	$Re < 10^4$	$c_s = /Re$	Laminar, merged boundary layer	
	$10^4 < Re < 10^5$	$c_s = \dfrac{0.925\cdot(s/R)^{0.1}\cdot 2\pi}{\sqrt{Re}}$	Lanimar, separated boundary layer	
	$10^5 < Re < \cdot 10^6$	$c_s = \dfrac{0.02\cdot 2/\pi}{\sqrt[6]{s/R}\cdot\sqrt[4]{Re}}$	Turbulent, merged boundary layer	
	$Re > \cdot 10^6$	$c_s = \dfrac{0.0255\cdot(s/R)^{0.1}\cdot 2/\pi}{\sqrt[5]{Re}}$	Turbulent, separated boundary layer	
GleitLRB	Small gap laminar	$c_s = \dfrac{1}{Re_s}(2+\dfrac{s^2}{R^2})$		
	Large gap laminar	$c_s = \dfrac{0.67\cdot\sqrt{R}/(2\sqrt{2}\cdot\pi)}{\sqrt{Re}}$	$c_s = \dfrac{0.3\cdot\sqrt{R}}{\sqrt{Re}}$	
Linnecken	All flat Re-Numbers	$c_s = \dfrac{1}{Re_s} + \dfrac{0.08\cdot(1+s/R)}{\pi\cdot\sqrt[5]{Re}\cdot(2+s/R)}$	$\left(\dfrac{s}{R}\right)_{opt} = s\cdot$ $(\dfrac{1+ .16\cdot Re^{0.4}}{0.0255Re^{0.8}-})$	

Schlichting	Small gap $Re < 0^5$	$c_s = \ / Re_s$	$c_s \approx \dfrac{0.42}{\sqrt{Re}}$
	Large gap $Re < \cdot 10^5$	$c_s = \dfrac{2.67/2\pi}{\sqrt{Re}}$	
	$Re > \cdot 10^5$	$c_s = \dfrac{0.0622/2\pi}{\sqrt[5]{Re}}$	
Sigloch	$s < \delta_{max}$ Couette-Flow	$c_s = \ / Re_s$	$c_s \approx \dfrac{0.4416}{\sqrt{Re}}$
	$s \gg \delta_{max}$ Couette-$Re < \cdot 10^5$ Flow	$c_s = \dfrac{0.64}{\sqrt{Re}}\left(1 - 0.31\cdot e^{-12\frac{s}{R}}\right)$	
	$Re > \cdot 10^5$ Turbulent Flat $k_s \approx)$ $(\delta_{max} = \dfrac{0.526}{\sqrt[5]{Re}}\cdot R)$	$c_s = \dfrac{0.023}{\sqrt[5]{Re}}\left(1 - 0.5\cdot e^{-12\frac{s}{R}}\right)$	$c_s \approx \dfrac{0.0115}{\sqrt[5]{Re}}$

Table 4-2: Analytical correlations for the friction coefficient for the validation case 'Enclosed rotating disc' [1].

4.2.1 Two–Dimensional axisymmetric simulations

Subsections 4.2.1 and 4.2.2 present some preliminary results and observations from the simulations carried out in this project. In subsection 4.2.1 the results of the 2D numerical simulations at a constant rotational speed of 400 RPM and three different turbulence models, Standard $k – epsilon$, RNG $k – epsilon$ and SST $k – omega$ are presented. For these simulations OpenFOAM 1.6 and FLUENT 12.0 were used. Based on the results of these calculations, a suitable turbulence model for the further investigations was found. Afterward simulations with seven different rotational velocities and the selected turbulence model have been performed. The solutions of them are shown in subsection 4.2.3.

4.2.1.1 2D calculation of the validation case at 400 RPM

The purpose of this chapter is to compare the 2D OpenFOAM results with the results of two different commercial CFD solvers (FLUENT and CFX) and with empirical equations based on experiments for the validation case.

OpenFOAM setup:

An axisymmetric mesh was generated with the *blockMesh* utility using the wedge type patches for the axisymmetric planes. Figure 4-4 shows the first mesh made for this case (i.e., $b = 200$ mm, $R = 900$ mm, $s = 50$mm and $t = 15$ mm) with 5, 445 cells.

Figure 4- 4 : Details of the mesh for the validation case with s = 50 mm.

simpleFoam was used as solver for the simulations of the steady-state, incompressible turbulent fluid flow. Three different turbulence models (described in Chapter 2) are used to evaluate an appropriate model for the resulting turbulent flow at a disc Reynolds number of approximately 6.4E+5 (i.e., 400 RPM).

Friction losses for Disc Side A and Disc Side B – Comparison of Results

With the given data (see Table 4-1) and mesh (see Figure 4-4) simulations are calculated to 40, 000 time steps with an axisymmetric two-dimensional model and three different turbulence models in OpenFOAM and FLUENT. The results for the friction losses are shown in Table 4-3.

Used empirical formulas and CFD softwares	Mesh Size	Average y⁺ - values [-] Disc A / Disc B	Friction Loss [kW] Disc Side A	Friction Loss [kW] Disc Side B
Geis				48.70
Linnecken			47.70	54.00
CFX 5.4 – *3D, SST-k-omega* (by Andritz Hydro)	84,000	-	52.70	51.30
FLUENT 12.0 - *2D axisymmetric, Standard k – epsilon*	5,445	21.3 / 18.1	57.83	49.57

OpenFOAM 1.6 – *2D axisymmetric Standard k – epsilon*	**5,445**	**6.0 / 13.4**	**43.28**	**45.53**
FLUENT 12.0 – *2D axisymmetric, SST k – omega*	**5,445**	**23.1 / 19.3**	**66.17**	**52.84**
OpenFOAM 1.6 – *2D axisymmetric, SST k – omega*	**5,445**	**5.1 / 13.6**	**43.60**	**45.60**
FLUENT 12.0 – *2D axisymmetric, RNG k – epsilon*	**5,445**	**21.2 / 18.0**	**50.70**	**49.08**
OpenFOAM 1.6 – *2D axisymmetric RNG k – epsilon*	**5,445**	**5.1 / 10.1**	**43.00**	**42.80**

Table 4-3: Comparison of CFD codes with analytical results - Calculated friction losses for the validation case (Mesh Size: 5,445 Cells).

The first three ranks in Table 4-3 (i.e., the solutions of Geis, Linnecken equations and CFX [34]) had been by Andritz-Hydro and all of turbulence models in OpenFOAM lead to large deviations from them. Except for the FLUENT results with the turbulence model *RNG k – epsilon* the other two FLUENT results with *Standard k – epsilon* and *SST k – omega* models show also smaller deviations compared to the given results (see Table 4-3). The reasons for this can be:

- The different y^+ values (see the y^+ values in Table 4-3) lead to diverse friction losses at Disc Side A and B and the reason for this is that the two CFD-solvers OpenFOAM and FLUENT calculate with different wall-functions at the boundary layer with the same turbulence model.

➢ **OpenFOAM**

- All 3 turbulence models are implemented as High Re-Number models.

- All 3 turbulence models use Standard Wall-Function, which is accurate in the near-wall region (i.e., at the first cell of boundary layer) if the y^+ values are greater than 30 (see Figure 2-3).

  - OpenFOAM uses a logarithmic wall-function (i.e., Standard Wall-Function) always, as far as the author is aware of, even it is not valid when the grid is very fine, whereas FLUENT uses some kind of blending (Linear Wall-Function) if the grid is fine [39].

➢ **FLUENT**

- Only the *k – epsilon* and *RNG k – epsilon* model use the Standard Wall-Function:

  - For example, *RNG k – epsilon* model calculates with Standard Wall – Function in OpenFOAM and FLUENT, but FLUENT has the additional options for this model [7]:

- ■ Transitional Flow

- ■ Swirl Dominated Flow

- • In order to capture the laminar and transitional boundary layers correctly, the mesh of *SST k – omega* model must have a y^+ value of approximately one. If the y^+ is too large (i.e., $y^+ > 5$), then the transition onset location moves upstream with increasing y^+ value [7].

- - The original mesh is not suitable for each turbulence model and the mesh quality on the boundary layer influences the y^+ values.

The different friction losses at the casing and disc walls lead to diserve flow characteristics, which are shown in the following figures:

The characteristics of fluid flow

All computations have been performed for the same revolution speed of 400 RPM. The results of flow velocities between the casing and disc walls are calculated with the three turbulence models *Standard k – epsilon*, *RNG k – epsilon* and *SST k – omega* in OpenFOAM 1.6 and FLUENT 12.0, which are shown in x-, y- and z- directions (i.e., from Figure 4-5 to Figure 4-10).

Figure 4- 5 : The velocity in the x direction with turbulence models *Standard k – epsilon*, *RNG k – epsilon* and *SST k – omega* in OpenFOAM.

Fluent with different turbulence models

Figure 4- 6 : The velocity in the x direction with turbulence models *Standard k – epsilon*, *RNG k – epsilon* and *SST k – omega* in FLUENT.

For the axial velocity the two *k – epsilon* turbulence models (i.e., *Standard k – epsilon* and *RNG k – epsilon*) in OpenFOAM show the same flow structure in the radial clearance with 4 Taylor vortices, but the *SST k – omega* model shows 6 Taylor vortices. FLUENT results in Figure 4-6 show everywhere 5 Taylor vortices in the radial clearance with different turbulence models.

Almost the same results can be seen in Figure 4-7 and Figure 4-8 in the y direction. It means that 4 Taylor vortices in the radial clearance with *k – epsilon* models are shown in OpenFOAM results and 6 Taylor vortices can be seen with the *SST k – omega* model. FLUENT results present everywhere 5 Taylor vortices in the radial clearance.

Paraview for OpenFOAM 1.6 with different turbulence models

Figure 4- 7 : The velocity in the y direction with turbulence models *Standard k – epsilon*, *RNG k – epsilon* and *SST k – omega* in OpenFOAM.

Fluent with different turbulence models

Figure 4- 8 : The velocity in the y direction with turbulence models *Standard k − epsilon*, *RNG k − epsilon* and *SST k − omega* in FLUENT.

In the z direction, the *k − epsilon* turbulence models show 2 Taylor vortices in the radial clearance and the *SST k − omega* model shows 3 Taylor vortices in OpenFOAM (see Figure 4-9). In FLUENT, all turbulence models show 2 Taylor vortices in the radial clearance.

Paraview for OpenFOAM 1.6 with different turbulence models

Figure 4- 9 : The velocity in the z direction with turbulence models *Standard k − epsilon*, *RNG k − epsilon* and *SST k − omega* in OpenFOAM.

Fluent with different turbulence models

Figure 4- 1 0 : The velocity in the z direction with turbulence models *Standard k − epsilon*, *RNG k − epsilon* and *SST k − omega* in FLUENT.

All three turbulence models in FLUENT show the same number of Taylor vortices in the radial gap, but the characteristics of fluid flow in OpenFOAM are different between the two *k − epsilon* models and the *SST k − omega* model. The reason for it has been discussed on page 40.

<u>In order to find a suitable mesh for each turbulence model several investigations are made:</u>

From Chapter 3 can be concluded that the *RNG k − epsilon* turbulence model is usually considered a reasonable selection regarding rotating or swirling flows. For weakly to moderately strained flows, the RNG model tends to give results comparable to the *Standard k − epsilon* model. In rapidly strained flows, the *RNG k − epsilon* model yields a lower turbulent viscosity than the *Standard k − epsilon* model. Thus, the RNG model is more responsive to the effects of rapid strain and streamlines curvature than the *Standard k − epsilon* model, which explains the superior performance of the RNG model for certain classes of flows including rotational flow [40].

So to speak, the turbulence models *RNG k − epsilon* and *SST k − omega* are theoretically suitable for this case, because they can better response to the problem at the boundary layer in the near-wall region. But, with the first mesh in Figure 4-4 the *SST k − omega* model can not accurately predict the fluid flow in the near-wall region, therefore more test simulations with *SST k − omega* model and different grids are performed in one of the used CFD program, i.e. FLUENT.

The refined mesh used with the *SST k − omega* model is shown in Figure 4-11.

Figure 4- 1 1 : Finer mesh for *SST k − omega* model.

Used empirical formulas and CFD softwares	Friction loss at Disc Side A [KW]	Friction loss at Disc Side B [KW]
Gies		48.70
Linnecken	47.70	54.00
CFD Calculation with **CFX**	52.70	51.30
FLUENT 12.0 − *2D axisymmetric, SST k − omega* (first mesh)	66.17	52.84
FLUENT 12.0 − *2D axisymmetric, SST k − omega* (refined mesh)	51.55	46.28

Table 4-4: Friction losses at Disc Side A and B with *SST k − omega* model in FLUENT 12.0.

After the FLUENT simulations with the refined mesh the friction losses at Disc Side A and Disc Side B are shown in Table 4-4, in which can be seen that it has similar friction loss at Disc Side A in comparison to the CFX results and the friction loss at Disc Side B shows that it is less than Geis and the CFX results by about 10%. Though the refined mesh yields better results than the first mesh and it has proven the above-mentioned assumption, but the computing time of the simulation is significantly increased and it is not economically. With OpenFOAM the refined mesh can not help us to get better results as with FLUENT, because it uses the logarithmic wall-function and FLUENT uses an enhanced wall treatment method if the grid is fine [39].

Therefore the simulations with *SST k – omega* model in OpenFOAM are omitted and the turbulence model *RNG k – epsilon* will be further used within this work.

Due to the lower y^+ values (see Table 4-3) at the disc and casing walls different coarsed meshes have been tested with the turbulence model *RNG k - epsilon*, which are based on the first mesh (i.e., 5, 445 cells). Table 4-5 shows the results with coarsed meshes, in which can be seen that the new mesh with 1, 441 cells fits better to the given results from the first three ranks and it will be used for further simulations.

Used empirical formulas and CFD softwares	Friction loss at Disc Side A [KW]	Friction loss at Disc Side B [KW]
Geis		48.7
Linnecken	47.7	54.0
CFD Calculation with **CFX**	52.7	51.3
OpenFOAM 1.6 – *2D axisymmetric RNG-k – epsilon* **(coarse mesh with 5, 445 cells)**	43.0	42.8
OpenFOAM 1.6 – *2D axisymmetric RNG-k – epsilon* **(coarse mesh with 2, 145 cells)**	51.5	44.1
OpenFOAM 1.6 – *2D axisymmetric RNG-k – epsilon* **(coarse mesh with 1, 441 cells)**	52.9	46.1
OpenFOAM 1.6 – *2D axisymmetric RNG-k – epsilon* **(coarse mesh with 915 cells)**	52.9	45.5

Table 4-5: Friction losses at Disc Side A and B with *RNG k – epsilon* model in OpenFOAM 1.6 and FLUENT 12.0.

To reaffirm if the calculated results with OpenFOAM are representative, the simulations were carried out again with the same new mesh (i.e., 1, 441 cells) in FLUENT and the results comparisons are shown in Table 4-6:

Used empirical formulas and CFD softwares	Friction loss at Disc Side A [KW]	Friction loss at Disc Side B [KW]
Geis		48.7
Linnecken	47.7	54.0
CFD Calculation with **CFX**	52.7	51.3
FLUENT 12.0 - *2D axisymmetric RNG-k-epsilon*	53.6	48.1
OpenFOAM 1.6 – *2D axisymmetric RNG-k – epsilon*	52.9	46.1

FLUENT 12.0 – *3D (10° Segment) - RNG k – epsilon*	**52.5**	**49.3**
OpenFOAM 1.6 – *3D (10° Segment) - RNG k – epsilon*	**51.4**	**47.8**

Table 4-6 : Friction losses at Disc Side A and B with 2D and 3D geometries and *RNG k – epsilon* model in OpenFOAM 1.6 and FLUENT 12.0.

Table 4-6 shows that the 2D calculations with OpenFOAM lead to similar friction loss at Disc Side A, but lower friction loss at Disc Side B by about 10% on average compared to CFX, Linnecken and Geis. About 4% deviation is existing between the OpenFOAM and FLUENT results at Disc Side B. 3D-simulations are made to verify the results of the two-dimensional calculations in OpenFOAM and FLUENT (see Table 4-6), which show reduced friction losses by about 1 [KW]. So to speak, the results with the 2D axisymmetric geometry and turbulence model *RNG k – epsilon* inspire confidence.

4.2.1.2 Comparison of 2-D OpenFOAM simulations with empirical formulas Linnecken, Geis, Dubbel, etc.

For two different gap widths, i.e., s = 9 and 50 mm, the numerical calculations were performed with OpenFOAM using the *RNG k – epsilon* model. Then, the results of these will be compared with the known empirical formulas from the literature. The used geometries and meshes are shown in Figure 4-12:

Figure 4- 1 2 : Basic meshes with 1411 cells for s = 50 and 9 mm, t = 15 mm.

Figure 4-12 shows the meshes for the validation case with different gap widths, both having 1, 411 cells. The figure at the left hand side is for the axial clearance of 50 mm and tip clearance of 15 mm, the figure at the right hand side is for the axial clearance of 9 mm and a tip clearance of 15 mm.

A lot of empirical equations to predict the disc friction losses were suggested by many researchers, for example by Linnecken, Geis, Dubbel and so on. In order to investigate the quality of the introduced approximate equations by the different authors and to reveal the effects of the two-dimensional axisymmetric geometry on the friction losses in the rotor-stator system, several comparisons were made. Figure 4-13 and Figure 4-14 show the friction losses from the OpenFOAM simulations for Disc Side A and B compared to the empirical equations and FLUENT:

Figure 4- 1 3 : Comparison of simulation results in OpenFOAM for Disc Side A and B with using different equations and FLUENT. (Tip clearance t = 15 mm, Radius R = 900 mm and Gap clearance s = 50 mm).

Figure 4- 1 4 : Comparison of simulation results in OpenFOAM for Disc Side A and B with using different equations and FLUENT. (Tip clearance t = 15 mm, Radius R = 900 mm and Gap clearance s = 9 mm).

From the curves c = f (Re), i.e., friction coefficient over the Reynolds number, can be seen that:

- **Disc Side A**

    The left hand sides of Figure 4-13 and Figure 4-14 show the plots of the dimensionless friction coefficients over Reynolds numbers for Disc Side A with gap ratio t/R = 0.0167. Both curves of the OpenFOAM results in black colour show the same trend.

It means that the friction coefficients are continuously decreasing with increasing Reynolds numbers after 300 RPM (Re = 4.8E+5). From Figure 4-13 can be seen that the OpenFOAM results lay between the CFX and FLUENT results after a Reynolds number of 4.8E+5. Although the OpenFOAM results in Figure 4-14 show the same trend as in Figure 4-13, they have a relatively large deviation in comparison to FLUENT, CFX or the analytical solution of Linnecken. The reason for this is:

➢ The mesh quality in the axial clearance can affect the friction coefficient in the radial gap. Both simulations with s = 9 and 50 mm are calculated with the same mesh which have 8 cells in the axial gap (see Figure 4-16) and it is too fine for s = 9 mm using wall functions, therefore a higher friction coefficient can be found in Figure 4-14.

The y^+ values for the 2D OpenFOAM calculations with s = 50 mm at low Reynolds number as an example are presented in Table 4-7, which shows that the y^+ values for the first cell layer near the wall are less than 30 (i.e., they are in the so-called laminar region ($y^+ < 11$) or buffer layer ($11 < y^+ < 30$)).

2D OpenFOAM caculations with s = 50 mm		
Revolution per Minute	Y⁺ values for Disc Side A	Y⁺ values for Disc Side B
100	3.60	2.46
200	4.77	3.72
300	24.30	20.11
400	31.22	26.50
500	37.94	35.78
1, 000	69.65	74.13
2, 000	128.41	140.41

Table 4-7 : The y^+ values for 2D OpenFOAM simulations with s = 50 mm.

- **Disc Side B**

The right hand sides of Figure 4-13 and Figure 4-14 show the results comparison for Disc Side B with large gap ratio (i.e., s/R = 0.056) and small gap ratio (i.e., s/R = 0.01). All of the solutions are calculated with the same radius R and tip clearance t (i.e., R = 900 mm and t = 15 mm).

For the large gap ratio the friction coefficients of OpenFOAM lay between Dubbel, FLUENT and Geis calculations after 300 RPM. The small gap ratio shows that the results of OpenFOAM and FLUENT are similar after 300 RPM. The reason for the friction coefficients at low Reynolds numbers is the same as for Disc Side A due to the low y^+ values.

Summing up the above two points can get the conclusions:

- The friction coefficient should be decreasing with increasing Reynolds number.

- The critical Reynolds number is estimated as about $5 \cdot 10^5$ for the validation case. If the $Re > 5 \cdot 10^5$, the fluid flow at the near-wall region is turbulent (i.e., $y^+ > 30$). The turbulence model with Standard Wall-Function shows certain deviations if the y^+ values lay between $11 < y^+ < 30$.

- By increasing the gap ratio s/R the friction coefficient c would be decreased.

- The friction coefficients at the axial and radial clearances depend on the mesh quality in the axial clearance. In order to investigate the reliability of the 2D OpenFOAM simulation, i.e., why the friction coefficient at Disc Side A shows great deviations to the other results, the CFD simulations are carried out with a three-dimensional model (a 10° segment with periodic boundary conditions) again and presented in the next subsection 4.2.2.

4.2.2 Three–Dimensional simulations with a 10° segment

To verify the solutions from the last subsection, the validation case is carried out again with a 3D model. The new geometries of the validation case are constructed for an axial clearance of 9 mm with 33, 500 cells and for 50 mm with 36, 630 cells, which are shown in the following figure:

Figure 4- 1 5 : New geometries for 3D simulations.

4.2.2.1 Comparison of 3-D OpenFOAM simulations with FLUENT and empirical formulas, i.e. Linnecken, Geis, Dubbel, etc.

The three-dimensional simulations are based on the above geometries and Figure 4-16 and Figure 4-17 show the comparisons of the OpenFOAM results for Disc Side A and B with given empirical equations and FLUENT:

Figure 4- 1 6 : Comparison of simulation results in OpenFOAM for Disc Side A and B with using different equations and FLUENT. (Tip clearance t = 15 mm, Radius R = 900 mm and Gap clearance s = 50 mm).

Figure 4- 1 7 : Comparison of simulation results in OpenFOAM for Disc Side A and B with using different equations and FLUENT. (Tip clearance t = 15 mm, Radius R = 900 mm and Gap clearance s = 9 mm).

- **<u>Disc Side A</u>**

    3D OpenFOAM simulations show approximately the same friction coefficients at Disc Side A for s = 50 mm (i.e., the large gap ratio s/R = 0.056) as 2D OpenFOAM simulations after 400 RPM, i.e. Re = 6.4E+5 (see Figure 4-16). Table 4-8 shows the

reason, why the deviations between the 2D and 3D OpenFOAM calculations for revolution speeds lower than 400 RPM are existing:

2D OpenFOAM caculations with s = 50 mm			3D OpenFOAM caculations with s = 50 mm		
Revolution per Minute	Y+ values for Disc Side A	Y+ values for Disc Side B	Revolution per Minute	Y+ values for Disc Side A	Y+ values for Disc Side B
100	3.60	2.46	100	3.58	2.39
200	4.77	3.72	200	4.10	5.85
300	24.30	20.11	300	18.10	7.44
400	31.22	26.50	400	25.80	12.95
500	37.94	35.78	500	31.29	17.56
1, 000	69.65	74.13	1, 000	57.20	39.26
2, 000	128.41	140.41	2, 000	105.01	77.16

Table 4-8 : The y^+ values for 3D OpenFOAM simulations with s = 50 mm.

For the small gap ratio s/R = 0.01 (see Figure 4-17), the 2D and 3D OpenFOAM results show significant deviations for Disc Side A. The reason for this is:

➢ The mesh quality in the axial clearance can affect the friction coefficients in the radial gap. Due to a different grid size distribution in the axial clearance s = 9 mm, i.e. the 2D geometry has 8 cells and the 3D geometry has 5 cells in the axial clearance, the friction losses at Disc Side A of the two-dimensional calculations are higher.

- **Disc Side B**

For the large gap ratio (s = 50 mm) the calculated friction coefficients of Disc Side B with the 3D model show approximately the same results as the FLUENT (2D) and CFX (3D) calculations after 400 RPM. Small deviations to the two-dimensional OpenFOAM results after 200 RPM (see the y^+ values in Table 4-8) can be observed.

The right hand sides of Figure 4-17 shows the deviations between the 2D and 3D results due to the different cell-numbers in the axial clearances.

Summing up the above points leads to the conclusions:

- The 3D simulations with s = 9 and 50 mm show similar friction losses at Disc Side B as the 2D axisymmetric model.

- Approximately the same friction losses at Disc Side A after 400 RPM can be seen in Figure 4-16, because the *RNG k – epsilon* model performs correct in the near-wall region, i.e., $y^+ > 30$ (see Table 4-8).

- The simulations for 2D and 3D are calculated with different mesh resolutions of 5 and 8 cells in the axial clearance s = 9 mm, which lead to the deviations in the results at Disc Side A (see Figure 4-17).

- After several comparisons it was concluded that the OpenFOAM simulations (2D and 3D) show similar results. Therefore for further simulations a 2D axisymmetric geometry with *RNG k – epsilon* turbulence model was used for the application case.

4.2.2.2 Fluid profile presentation

Keeping in mind the different geometries for the 2D and 3D simulations (i.e., 2D simulations use an axisymmetric model of 5° with wedge boundary conditions, 3D simulations use a 10°-segment with periodic boundary conditions), similar fluid flow characteristics in the radial clearance for the large gap ratio at 400 RPM, can be described (see Figure 4-18 to Figure 4-20).

Figure 4- 1 8 : Comparison of Fluid flow profile between 2D and 3D OpenFOAM simulations in the x direction.

For the axial direction the both OpenFOAM results, 2D and 3D show 4 Taylor vortices in the radial clearance and similar velocities.

Figure 4- 1 9 : Comparison of Fluid flow profile between 2D and 3D OpenFOAM
simulations in the y direction.

From the right hand side of Figure 4-17 can be seen that the friction coefficients for the 3D
simulations are larger than 2D results for Disc Side B due to higher radial velocities for the small
gap ratio (see Figure 4-19). In the radial clearance the 2D and 3D simulations show the same
number of Taylor vortices which leads to similar friction coefficients for Disc Side A (see Figure
4-18).

Figure 4- 2 0 : Comparison of Fluid flow profile between 2D and 3D OpenFOAM
simulations in the z direction.

The velocities in z-direction of 2D and 3D simulations 400 RPM show the same flow behaviour
(see Figure 4-20).

Figure 4- 2 1 : Comparison of the turbulent kinetic energy 'k' between 2D and 3D
OpenFOAM simulations.

Figure 4- 2 2 : Comparison of the dissipations rate 'epsilon' of the turbulent kinetic energy
between 2D and 3D simulations.

Figure 4- 2 3 : Comparison of the kinematic turbulence viscosity 'nut' between 2D and 3D simulations.

Figure 4-21 to Figure 4-23 show Contour-plots of the turbulent kinetic energy 'k', the dissipations rate 'epsilon' and the kinematic turbulent viscosity 'nut' for the 2D and 3D OpenFOAM simulations at 400 RPM.

5 Overview of the application project 'Oil Guide Plate in a Guide Bearing'

The 'Oil Guide Plate in a Guide Bearing' is the application project for this work, in which a stationary disc is situated in a closed distance **s** from a rotating disc and a circulating oil flow is constituted in the oil container due to the rotating runner. The geometric parameters of the guide plate, as for example the distance to the rotor or its length influences the friction losses which is engendered between rotor and the opposite stationary walls as the highly viscous oil flows through the spacing between them. A parameterised, simplified CFD model of the guide bearing concept was created to identify the key parameters influencing the friction losses in the system. In addition to minimizing the friction losses even further parameters (e.g. pressure drop or discharge flow rate) play a significant role in order to guarantee the lubrication of the guide bearing.

The problem under study is a steady, incompressible, isothermal and single phase flow. The numerical simulations are carried out with an idealized axisymmetrical geometry of the generator 'Glendoe' (see Figure 5-1 and Figure 5-2). For the turbulent, swirling flow inside the oil container the *RNG k – ε* turbulence model was used. In the following subsections a short description of the geometry, fluid properties and the main flow characteristics are presented. Further on a comparison between the validation and the application cases is made to highlight the differences concerning the fluid flow in both cases.

<u>The idealized geometry:</u>

Figure 5-1 presents the simplified geometry model of 'Glendoe', with which the OpenFOAM simulations are carried out. The geometry and the mesh are made with the pre - processing utility *blockMesh* which is part of the free CFD software OpenFOAM. The geometry is divided into 10 blocks shown with different colours in Figure 5-1. The establishing circulating oil flow inside the oil container due to the rotating runner is shown with white arrows.

Figure 5- 1 : The simplified geometry for the lubrication system of the generator 'Glendoe'.

The differences between the idealized and realized geometry models are:

-   In reality, the oil guide plate is not about the whole circumference. It is constructed of 12 disconnected segments.

-   The lower guide bearing pads are not considered in the idealized geometry, which are situated beside the rotor wall, named Rotor_1, at the outside radius. (see Figure 5-2).

-   The oil level is modelled as a frictionless wall for the numerical simulations due the fact that only a single phase flow is considered.

The following figure and table show the dimensions of the geometry and fluid properties for the CFD model 'Glendoe':

Figure 5- 2 : Dimensions and wall boundaries for the CFD model

Parameters	Descriptions of parameters	Dimension
D	Width of stator.	0.01 [m]
H_K	Spacing between the bottom of container and rotor.	0.123 [m]
h_{oil}	Oil level.	0.35 [m]
R_{ka}	Outside radius of rotor.	0.55 [m]
R_{ki}	Inside radius of rotor.	0.448 [m]
R_{sa}	Outside radius of stator.	0.55 [m]
R_{si}	Inside radius of stator.	0.4475 [m]
R_{Ta}	Outside radius of oil container.	0.98 [m]
R_{Ti}	Inside radius of oil container.	0.441 [m]
s	Spacing between rotor and stator.	0.01 [m]
η	Dynamic viscosity at 46°C – Oil ISO-VG46	29.81 [mPa.s]
ρ	Density – Oil ISO-VG46	846.3 [kg/m^3]

Table 5-1: Geometric data and fluid properties for the CFD model.

Standard condition:

- Incompressible, isothermal fluid flow (ρ = constant, T = constant)

- Steady-state, single phase flow

- The surface roughness at rotor and stator walls is neglected.

- Ten revolution speeds: N = 150, 300, 375, 450, 500, 600, 750, 900, 1050 and 2000 RPM.

The first simulations showed that a 'quasi-stationary' state is reached after 40, 000 iterations (time steps). Figure 5-3 presents the plot of mass flow rate over time, in which the mass flow rate changes only slightly at the monitor point after 40, 000 iterations. After the consideration about the 'quasi-stationary' state and high CPU-times to finish all simulations in the framework of this thesis, '40, 000' is registered as 'last Timestep' in the *controlDict* dictionary for all further simulations (see Figure 6-8).

Figure 5- 3 : Determination of time steps for the simulation.

The analysis of the simulations is focused on the spacing between rotor and stator i.e. the red area in Figure 5-4. Therefore two iso-surfaces at the inlet and outlet of the axial clearance are defined with the help of three cellsets to calculate the mass flow rates and pressure (static and total) differences at these surfaces after the simulations reached their convergence criterium. The inlet area is equal the sum of cellset 1 and cellset 3, and the outlet area is the sum of cellset 2 and cellset 3.

Figure 5- 4 : Cellset definition for the creation of the control surfaces.

5.1 Comparison of fluid flows about the validation case and the application case

The validation case in name 'Enclosed Rotating Disc' (in the following referred to as 'Validation case') and the application case in name 'Oil Guide Plate of Guide Bearing' (furthermore named 'Application case') are simulated with the same standard conditions (i.e., incompressible, isothermal, steady-state, single phase flow and neglected surface roughness on walls) and the *RNG* $k - \varepsilon$ turbulence model with Standard Wall-Function in OpenFOAM. The basic geometries of these two cases are different, because the validation case has a narrow radial clearance of 15 mm (i.e., the spacing between Disc Side A and casing wall). But the application case has a wide radial clearance between outside radius of rotor and oil container of 430 mm, in which the Taylor vortices cannot be formed as in the validation case. Therefore the two cases are only comparable with respect to the flow field within the axial spacing (see Subsection 5.1.1).

In the following subsections the fluid flow for the two cases and associated velocity profiles in axial, radial and azimuth directions are compared for similar Reynolds numbers (Re = 4.5E+5 at 300 RPM for the validation case and Re = 4.8E+5 at 500 RPM for the application case). The x-axis was chosen as the axis of rotation. In the validation case two separated boundary layers can be found in the axial clearance on the disc and casing wall. Due to the centrifugal forces the fluid in the boundary layer at the disc wall passes radially outward and in the boundary layer at the casing wall in the direction to the disc axis back. A small flow of the fluid remains in the radial clearance and forms Taylor vortices at a specific revolution speed.

In the application case a large flow of fluid comes from the oil container into the axial clearance (i.e., rotor-stator distance) due rotation. In the rotor-stator region, a radial outflow on the rotating disc and some backflow can be observed along the stationary disc at certain revolution speeds. A small part of them retains in the spacing between rotor and inside radius of the oil container.

5.1.1 The fluid flows and velocity profiles in the x direction for the two cases

Here the fluid flow in the axial clearances for both cases is analysed. In order to see clearly the fluid flow in the axial clearances, an enlarged vector plot is shown in Figure 5-5. The left side of this figure shows the velocity vectors for the validation case in the upper, middle and lower part of the axial clearance. The two separated boundary layers on the disc and casing walls can be seen clearly. The right side shows an enlarged view of the fluid flow for the application case which explains how the fluid flow passes through the rotor-stator clearance. The oil flows radially outward at the rotor side with high velocities and backflow is provided along the stator side, which enters the axial gap at the outside radius of the guide plate.

Validation case

Upper Part

Middle Part

Bottom Part

Disc Side A

Disc Side B

Comparative region

Application case

Stator_2

Rotor_2

Comparative Region

Enlarged view of fluid flow in comparative
regions for the both cases

Figure 5- 5 : Enlarged view of fluid flows in the axial clearance for the validation (left) and the application (right) cases.

In order to investigate accurately the flow patterns in the axial clearances iso-surfaces at nine different radii are defined (see Figure 5-6), which correspond to the total height of the axial clearances for these two cases (i.e., from 0.11 m to 0.83 m for the validation case and from 0.45775 m to 0.53975 m for the application case).

Figure 5- 6 : Definition of nine iso-surfaces at different radial heights in the axial clearance
for the two cases.

The velocity profiles at these iso-surfaces in the axial, radial and tangential directions are presented in the following subsections.

Velocity profiles in the axial direction for both cases:

Figure 5-7 shows the axial velocity profiles for the validation case, in which the highest axial velocities can be found in the middle of the axial clearance. From the plot can be seen that the axial velocities are decreasing with increasing radius inside the axial gap. This corresponds to the radially outward flow at the disc side and the radially inward flow at the casing wall.

Figure 5- 7 : Axial velocity profiles for the validation case.

Figure 5- 8 : Axial velocity profiles for the application case.

Figure 5-8 shows the axial velocity profiles for the application case, which presents similar velocity profiles at higher radii (from 0.509 to 0.53975 m) inside the axial clearance as the validation case. The axial velocity is negative near the stator wall at lower radii (from 0.45775 to 0.4885 m). This indicates the existence of a local re-circulating region because the fluid makes a sharp turn of 90 degrees before it enters the axial gap between the rotor and stator (see Figure 5-11). The maximum axial velocities can be found almost in the middle of the axial clearance except for the iso-surface at a radius of 0.45575 m.

5.1.2 The fluid flows and velocity profiles in the y direction for the two cases

In Figure 5-9 and Figure 5-10 the distributions of radial velocities for both cases are presented. In general the radial velocities are much greater than the axial ones and the trend of the radial velocity profiles are different to the axial velocity profiles. From the radial velocity profiles of the validation case (Figure 5-9) can be seen that all profiles of the radial component cross one point, which is situated at about 4.5 mm of the gap width in axial direction. This corresponds to the unique feature of the flow in a rotor–stator system - the asymmetry of the distributions of the physical variables. The positive radial velocity presents the outward flow and the negative velocity shows the backflow along the casing wall. The magnitudes of the radial velocities are almost of the same size at the left and right side of the crossing point. The radial velocities at the disc size are marginally greater.

67

>

Figure 5- 9 : Radial velocity profiles for the validation case.

Figure 5- 1 0 : Radial velocity profiles for the application case.   .

The radial velocity profiles for the application case are essential different to the validation case, which is shown in Figure 5-10. For the application case the radial velocities are relatively higher than in the validation case due to the higher circumferential velocities.

No distinctive turning point can be seen for the application case considering the radial velocities. The highest radial velocities are shown for the iso-surface after the inlet corner at r = 0.45775 m. Here the fluid is accelerated because of the sharp turn of 90 degrees and enlargement of the flow area from 7 to 10 mm. This is in opposite to the trend presented for the validation case. The lowest radial velocities also occur at this iso-surface because a local re-circulating zone is established at the stator side. A detailed view of this flow pattern can be seen in the scaled (Uy from – 0.2 to 0.2 m/s) vector plot shown in Figure 5-11.

Figure 5- 1 1 : Scaled vector plot of velocities in y-direction for the application case.

Figure 5-11 also shows the re-entering fluid along the stator wall from the outside radius of the axial clearance. The radial velocities of this backflow are decreasing with decreasing radius inside the axial gap till to the re-circulating zone at the inlet corner. It can be also proved in Figure 5-10 that the highest negative radial velocities are occurring at the in- and outside of the axial clearance. The velocity curves for the other iso-surfaces are a little bit disordered, which depends on how strong the backflow and recirculation zone are developed along the axial clearance.

5.1.3 The fluid flows and velocity profiles in the z direction for the two cases

The tangential velocity profiles for the validation and application cases are shown in Figure 5-12 and Figure 5-13. The similarities between these two cases are:

- The tangential velocities are similar due to the approximately the same Reynolds number.

- The tangential velocities are enlarged with increased radial heights.

- The application and validation cases in Figure 5-12 and Figure 5-13 show a clearly increasing trend from stator wall to rotor wall due to the narrow rotor-stator distance (i.e., s = 9 mm for validation case and s = 10 mm for application case).

Figure 5- 1 2 : Tangential velocity profile for the validation case.

Figure 5- 1 3 : Tangential velocity profile for the application case.

6 Parameterised model – 'Glendoe'

The generator model 'Glendoe' is selected by Andritz-Hydro for further detailed investigations with the parameterised model. This work based on numerical flow simulations was performed to predict mass influx, pressure differences and energetically losses in the rotor-stator system. The influence of the possible geometrical parameters was systematically investigated in the following subsections, in which described how the primary parameters are defined and how the results can be automatically worked out in OpenFOAM.

Figure 6-1 shows the basic geometry of 'Glendoe', which is already given as an example in Chapter 5:

Figure 6- 1 : Geometric data for model 'Glendoe'.

6.1 Parameter description

This subsection presents the six primary geometric parameters which are:

- Rotor-stator distance **s**

- The spacing **t** between inside radius of rotor and oil container

- The spacing **t'** between the inside radius of stator and oil container

- Extension of guide plate length at the outside radius - Parameter **a**

- Modification of guide plate length at the inside and outside radius - Parameter **at'**

- The setting angle α of the stator.

The next subsections show an overview on all geometry parameters. In order to investigate influencing parameters a systematic numerical study was performed in which more than 1, 000 simulations were analysed.

6.1.1 s – The rotor–stator distance

The spacing **s** between rotor and stator is one of the important parameter, which can affect directly the flow rate and relational friction losses. In order to prove it the parameter **s** is varied from 5 to 15 mm and the other parameters stay constant, which can be seen in Figure 6-2. The original rotor-stator distance is 10 mm.

Figure 6- 2 : Description of parameter **s**.

6.1.2 t – The spacing between inside radii of rotor and oil container

The parameter **t** presents the spacing between inside radius of rotor and oil container, which is also possibly to influence the results. Because the friction loss of 'Rotor_3' is affected strongly by the width of the spacing between the inside radius of rotor and oil container, the inlet volume of the fluid is automatically increased, when the parameter **t** is enlarged. Figure 6-3 presents the varied parameter **t** from 3.5 to 28 mm and the parameter **s** is 10 mm. Simultaneously, the simulations would be also carried out with the parameter **s** with 5 and 15 mm and the parameter **t** is varied from 3.5 to 28 mm.

Figure 6- 3 : Description of parameter **t**.

6.1.3 t' – The difference between the inside radii of stator and oil container

The consideration about the parameter **t'** is made because the flow rate would be possibly increased with an enlarged inlet gap to the axial spacing. The stator of all variants would be shortened by 7 mm. The geometry in Figure 6-4 shows an example with the variant '**s** = 10 mm, **t** = 7 mm and **t'** = 7 mm', in which the parameters **s** and **t** stay constant and stator is reduced by 7mm (i.e., **t'**=7 mm) in radial direction.

Figure 6- 4 : The stator is shortened by 7 mm in the radial direction for an example '**s** = 10 mm, **t** = 7 mm'.

6.1.4 a – The difference between the outside radii of rotor and stator

Here, the stator would be extended by 7 mm in the radial direction for each variation and geometry of it is based on Figure 6-3. Figure 6-5 shows an example in which the parameters **s** and **t** are the same one (i.e., **s** = 10 mm, **t** = 7 mm and **a** = 7 mm), but the whole stator is extended by 7 mm in the radial direction which is defined as parameter **a**. The reason to insert this parameter is that the backflow is possible circumvented by the enlarged stator in the radial direction and therefore increasing the mass flow rate through spacing **s**.

Figure 6- 5 : The stator is extended by 7 mm in the radial direction for an example '**s** = 10 mm, **t** = 7 mm'.

6.1.5 at' – The distance between inside and outside radii of Rotor and Stator

The parameter **at'** is a combination of the parameters **a** and **t'**, which are described above. The whole stator is shifted radially outward by a maximum distance of 7 mm for each variant. The considerations for this approach are to maximize the incoming mass flow rate at the inside radius and to prevent the backflow at the outside radius. The variant with '**s** = 10 mm, **t** = 7 mm, **at'** = 7 mm' is shown in Figure 6-6 as an example, which presents the used geometry for this variation.

Figure 6- 6 : The stator is shifted by 7 mm in the radial direction for an example '**s** = 10 mm, **t** = 7 mm'.

6.1.6 α – The setting angle

The simulations are calculated with different setting angles **α** to investigate if the angle influences the volume flow rate through the rotor–stator distance. Figure 6-7 shows the geometric examples for maximum and minimum setting angle **α**, which are 3.5° and $- 3.5^{\circ}$ because the smallest spacing between rotor and stator is restricted to 5 mm. Parameter **t** is defined as 7 mm.

Figure 6- 7 : Description of parameter **Alpha.**

6.2 Automatically parametrised grid generation

The geometrical data, length intervals, physical properties and solver inputs are defined in an input data file in name '*Variation data*' (see **Appendixes A**) before the numerical calculation starts, in which the different parameters can be simultaneously updated. An example input data file is shown in Figure 6-8:

```
Rki: 0.448
Rka: 0.550
Rsi: 0.4475
delta_Rsi: 0.0
Rsa: 0.550
delta_Rsa: 0.0
Rti: 0.441
delta_Rti: 0.0
Rta: 0.980
s_min: 0.007
s_max: 0.021
Hk: 0.123
Hk_min: 0.11
Hk_max: 0.13
Hoel: 0.350
S: 0.01
S_min: 0.005
S_max: 0.02
D: 0.01
Alpha: 0.0
Alpha_min: -3.5
Alpha_max: 3.5
mu: 0.02981
Dichte: 846.3
Drehzahl: 15.708,31.416,39.270,47.124,52.360,62.832,78.540,94.248,109.956,125.664
Foerderhoehe: 0.1
Iter: 40000
writeInterval: 2500
```

Figure 6- 8 : Shortened *Variation data* for parameter input in OpenFOAM.

The simulations would be automatically broken if the defined parameters exceeded the provided intervals. The simulation is calculated to 40, 000's iteration with different revolution speeds from 15.708 to 125.664 rad/s (i.e., from 150 to 2, 000 RPM) and the write interval is 2, 500.

The whole numerical simulation with OpenFOAM is controlled with a Python script in name '*versatileScheibe.py*' (see **Appendixes B)** with the objective to monitor the entire workflow from pre-processing (e.g., grid generation) to post-processing (e.g., automatically output of results).

A cut-out of the script is shown in Figure 6-9, in which several queries for the geometric parameters are made.

```python
            .
            .
            .
    def calculation(self,values,keep=False):
        Spaltweite_Oeltopf = (values["Rki"])-((values["Rti"])+(values["delta_Rti"]))
        Spaltweite_Scheibe = (values["S"])-(((values["Rsa"])-(values["Rsi"]))/2.0) *
abs((math.sin((values["Alpha"])/180.*math.pi)))
        Rsi_Neu = (values["Rsi"])+(values["delta_Rsi"])
        Rsa_Neu = (values["Rsa"])+(values["delta_Rsa"])
        Nmin = 9.549296*(values["Drehzahl"])
        fac = self.eval("Dichte*Drehzahl*72",values)
        if (values["Alpha"])<(values["Alpha_min"]) \
            or (values["Alpha"])>(values["Alpha_max"]) \
            or (values["S"])<(values["S_min"]) \
            or (values["S"])>(values["S_max"]) \
            or Spaltweite_Scheibe<0.005 \
            or Rsi_Neu<0.99*(values["Rsi"]) \
            or Rsi_Neu>1.01*(values["Rsi"]) \
            or Rsi_Neu-((values["Rti"])+(values["delta_Rti"]))<0.0065 \
            or Rsa_Neu<0.99*(values["Rsa"]) \
            or Rsa_Neu>1.01*(values["Rsa"]) \
            or Spaltweite_Oeltopf<(values["s_min"]) \
            or Spaltweite_Oeltopf>(values["s_max"]) \
            or (values["Hk"])>(values["Hoel"]) \
            or (values["Hk"])<(values["Hk_min"]) \
            or (values["Hk"])>(values["Hk_max"]) \
            or (values["Rki"])<(values["Rti"])+(values["delta_Rti"]) \
            or (values["Rka"])>(values["Rta"]):
            print "Ungueltige Geometrieangaben"
            .
            .
            .
```

Figure 6- 9 : A shortened Python script.

OpenFOAM is based on templates (*blockMesh* templates as example can be found in the next subsection), with which the calculations can be simple parameterised and automated. With the Python script single runs, variations (geometric or physical) or optimization studies with OpenFOAM can be carried out. The simulation would be automatically broken, when the parameter values are violating the pre-assigned values.

6.2.1 Different BlockMesh templates for the pre-processing in OpenFOAM

According the predetermined parameter values in the input file '*Variation data*' a suitable mesh is automatically selected from three different templates. Figure 6-10 shows a shortened *blockMesh* dictionary to explain how the mesh is defined with specified parameters and the relationship for the different parameters. The complete *blockMesh* dictionaries for the three meshes are presented in **Appendixes C, D, and E.**

```
                                        .
                                        .
                                        .
vertices
(
        (0 $Rti+delta_Rti$ -0.01)
        ($Hk-S-D$ $Rti+delta_Rti$ -0.01)
        ($Hk-S-D$ $Rki$ -0.01)
        (0 $Rki$ -0.01)
        ($Hk-S$ $Rti+delta_Rti$ -0.01)
        ($Hk-S$ $Rki$ -0.01)
        ($Hk$ $Rti+delta_Rti$ -0.01)
        ($Hk$ $Rki$ -0.01)
        ($Hoel$ $Rti+delta_Rti$ -0.01)
        ($Hoel$ $Rki$ -0.01)
        ($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsi-My)$$My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_ Alpha*(Rsi-My)+delta_Rsi$ -0.01)
        (0 $Rsi$ -0.01)
        ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsi-My)$$My-Sin_Alpha*(Hk-S-Mx)+Cos_Alpha*
(Rsi-My)+delta_Rsi$ -0.01)
        ($Hk$ $Rsi$ -0.01)
        ($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsa-My)$$My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_Alpha*(Rsa-My)+delta_Rsa$ -0.01)
        (0 $Rsa$ -0.01)
        ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsa-My)$$My-Sin_Alpha*(Hk-S-Mx)+Cos_Alpha*
(Rsa-My)+delta_Rsa$ -0.01)
        ($Hk$ $Rka$ -0.01)
        ($Hk-S-D$ $Rta$ -0.01)
                                        .
                                        .
                                        .
```

Figure 6- 1 0 : A shortened mesh example.

Figure 6-10 shows as an example the definition of the block vertices for one mesh template. The inserted parameter values from the input file would be directly accepted here and then a suitable mesh is automatically built.

6.2.2 Automatically calculations and analysis for post-processing in OpenFOAM

After mesh generation the simulation process can be started. This is also controlled by the Python script. The number of time steps, the output writing interval and the residual tolerances can be specified by the user.

After the numerical calculations were terminated some post-processing utilities are started by the script to evaluate the relevant values for the rotor-stator system. Figure 6-11 presents the result data for an example of 'Glendoe' with 500 RPM, in which the flow rate, static pressure difference, total pressure difference and friction losses of rotor and stator walls (e.g., Rotor_2, Rotor_3, Rotor_total, Stator_2 and the Stator_total) are summarized.

```
Zusammenfassung_Foerderscheibe.S_0.01.Alpha_0.L_0.1025.Hk_0.123.s_0.007.N_500.001
Volumenstrom in Liter pro Minute - Scheibenaustritt: 1951.1
Statischer Druck-Scheibenaustritt [Pa]: 106074
Statischer Druck-Scheibeneintritt [Pa]: 98918
Totaldruck-Scheibenaustritt [Pa]: 143740
Totaldruck-Scheibeneintritt [Pa]: 107144
Spurkopf-Verluste parallel zum Spalt [Watt]:  -8217
Scheiben-Verluste parallel zum Spalt [Watt]:   1343
Spurkopf-Verluste parallel zum Oeltopf [Watt]:  -5158
Spurkopf-Verluste gesamt [Watt]: -29683
Scheiben-Verluste gesamt [Watt]:   2601
```

Figure 6- 1 1 : Evaluated results for 'Glendoe' with revolution speed 500 RPM.

7 Results presentation of parameterised model 'Glendoe'

This chapter presents some results and observations about the parameterised model 'Glendoe' from simulations carried out in this project. All the simulations have been carried out with the *RNG k – epsilon* turbulence model. The 2-D axisymmetric grids with different geometry parameters for 'Glendoe' and their standard conditions have been presented Chapter 5. There are six primary geometry parameters (**s, t, t', a, at'** and **α**) in order to investigate if the mass flow rate, static and total pressure differences and friction losses associated with them yield better results than the original geometry (i.e., **s** = 10 mm, **t** = 7 mm). A systematic numerical study was performed, which included more than 1,000 simulations.

7.1 The setting angle – Parameter α

Before the other variations are made, the influence of the parameter **α** is at first investigated with the original geometry (**s** = 10 mm, **t** = 7 mm and **α** = 0°). 35 optimization steps have been performed with different setting angles in the range from $-3.5°$ to $3.5°$.

To get a maximum volume flow rate and simultaneously minimum friction losses in the rotor-stator system was the application goal for this work. Therefore the volume flow rate, friction losses at Rotor_2, Rotor_3 and Stator_2 are depicted in the following figures with histograms for the different varied setting angles, with respect to the results of the original case in percentage. In Figure 7-1 to Figure 7-4 positive setting angles are marked with green colour, which presented an enlarged inlet surface area. The red marked columns present the negative setting angles.

Figure 7- 1 : Comparison of volume flow rate for varied setting angles with 0º.

Figure 7- 2 : Comparison of friction loss at Rotor_2 for varied setting angles with 0º.

Figure 7- 3 : Comparison of friction loss at Rotor_3 for varied setting angles with 0º.

Figure 7- 4 : Comparison of friction loss at Stator_2 for varied setting angles with 0º.

Negative setting angles

For the volume flow rate the varied negative setting angles from – 0.596º to – 2.214º showed an approximately result like the original geometry ($\alpha = 0°$). The friction losses at Rotor_2, Rotor_3 and Stator_2 are reduced slightly, about 1%, 0.1% and 2.5%. Above an angle of - 3º the volume flow rates are decreased about 2%, compared to $\alpha = 0°$. The friction losses at Rotor_2 and Rotor_3 are increased by about 6% and 0.22%, at Stator_2 the friction loss is decreased by about 2.5%. It means that the mass flow rate is reduced with decreased inlet surface area. Due to the fluid flows with a higher radial velocity in the rotor-stator distance **s** the volume flow rate is not reduced too much, but the friction loss at Rotor_2 is increased with the higher inflow

velocity by the variants with negative setting angles. The backflow between the rotor and stator is also reduced, therewith decreasing the friction loss at stator_2.

Positive setting angles

For setting angles of 1° and 1.75° the volume flow rates are decreased by about 1%, the friction losses at Rotor_2 and Stator_2 are increased by 1% and 3%, the friction loss at Rotor_3 is decreased by only about 0.2% and hence has negligible impact on the results. For the larger positive setting angles from 2.625° to 3.5° the volume flow rates are increased significantly by about 21%, but the friction losses at Rotor_2 and Stator_2 are also increased by about 35% and 28%. At Rotor_3 the friction loss stays at about 0.16% in comparison to the original case. It means that the inlet surface is enlarged and gets more inflow, but also still higher friction losses at Rotor_2 and Stator_2.

A brief summary is that the variations of setting angles can not influence the friction loss at Rotor_3 and the mass flow rate is increased although with an enlarged inlet surface area, but the friction losses at Rotor_2 and Stator_2 are also strongly increased. When the mass flow rate is increased by about 22%, the friction losses at Rotor_2 and Stator_2 are also increased by about 35% and 17% on average. So to speak, the effects of these variations are cancelled. Therefore the further simulations have been dispensed with this parameter.

7.2 Rotor–stator distance – Parameter s

The rotor-stator distance **s** describes the spacing between the rotor and stator. To determine if parameter **s** has a greatly influence, **s** is varied from 5 to 15 mm with constant **t**, which is defined from 3.5 to 28 mm (see Chapter 6.1.1). The following subsections show the trend-lines of volume flow rate, total pressure difference and friction losses at rotor and stator walls while the parameter **s** changed. All the results are compared with the original case, i.e. **s** = 10 mm and **t** = 7 mm. The yellow columns in the figures indicate the original case for each revolution speed. The plots of the different variables (e.g., volume flow rate, static and total pressure differences and friction losses at rotor and stator walls) for 5, 10 and 15 mm are presented with blue, black and red colours.

7.2.1 t = 3. 5 mm and s = 5, 10, 15 mm

The left hand side of Figure 7-5 shows the volume flow rate vs. Reynolds numbers for varying **s** and constant **t** = 3.5 mm, which lead to smaller volume flow rates due to the narrow radial clearance. All of the three variants have a crank in the trend curves and intersected with each other, the reason for it is that the axial velocities are strongly increased at certain revolution velocities (e.g., the axial velocities in the spacing between inside radius of stator and the oil container are increased by 59.8%, 52.3% and 69.3% for **s** = 5 mm with 900 RPM, for **s** = 10

mm with 750 RPM and for **s** = 15 mm with 600 RPM) which can be seen in the left side of Figure 7-5 and Table 7-1.

Figure 7- 5 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **s** (**t** = 3.5 mm = constant).

s=5mm t=3,5mm			s=10mm t=3,5mm			s=15mm t=3,5mm		
revolution speed	Maximal axial velocity	Comparison the current with previous revolution speed in percentage	revolution speed	Maximal axial velocity	Comparison the current with previous revolution speed in percentage	revolution speed	Maximal axial velocity	Comparison the current with previous revolution speed in percentage
[U/min]	[m/s]	(+)	[U/min]	[m/s]	(+)	[U/min]	[m/s]	(+)
150	1,11		150	1,18		150	1,05	
300	2,00	80,8%	300	1,80	52,0%	300	1,47	39,8%
375	2,36	17,6%	375	2,00	11,4%	375	1,59	8,1%
450	2,66	12,8%	450	2,16	7,9%	450	1,65	4,2%
500	2,84	6,7%	500	2,22	2,5%	500	1,63	-1,3%
600	3,14	10,9%	600	2,83	27,7%	600	2,77	69,3%
750	3,53	12,3%	750	4,31	52,3%	750	3,35	21,1%
900	5,64	59,8%	900	4,99	15,7%	900	4,02	20,1%
1050	6,89	22,3%	1050	6,19	24,1%	1050	4,96	23,4%

Table 7-1: The maximal axial velocities for varied **s** and constant **t** = 3.5mm.

Figure 7- 6 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **s** (**t** = 3.5 mm = constant).

Figure 7-6 shows the friction losses at the rotor walls (see Figure 7-7). The plots for varying parameter **s** at Rotor_2 present lower friction losses than the original case for Reynolds numbers between 1.35E+5 and 6.70E+5 (i.e., 150 to 750 RPM). The curves of friction losses at Rotor_3 for **s** = 5, 10 and 15 mm are almost identicall due to the same radial clearance. Up to a Reynolds number of 5.40E+5 (i.e., 600 RPM) the friction losses at Rotor_3 of all variations are distinctly higher than in the original case (see Table 7-2), because the fluid flows with higher axial velocity than in the original case due to the decreased radial clearance. After certain revolution speeds the re-circulation zone in the original case (**t** = 7 mm) is larger than in the variants with '**t** = 3.5 mm'. Therefore, after 750 RPM the variants present lower friction losses at Rotor_3 than the original case. The detailed comparison to the original case can be found in Table 7-2.

Figure 7- 7 : Simplified rotor-stator configuration.

With decreased rotor-stator distance, i.e. **s** = 5 mm, the friction losses at Stator_2 are higher because the fluid passes through the axial clearance (i.e., the distance between rotor and stator) with an increasing velocity and there is no backflow by this variation (see Figure 7-8).

Figure 7- 8 : Comparison of friction losses at Stator_2 vs. Reynolds numbers for varied **s** (**t** = 3.5 mm = constant).

A brief summary for these variants

By decreasing the spacing between inside radius of oil container and rotor to half of the original case the volume flow rates of all variants are also decreased by 40% due to the reduced inlet volume between inside radius of stator and oil container. But the friction losses at rotor and stator walls show only minor changes between the variants. With these variants it is not possible to get better results than the original case, therefore **t** = 3.5 mm would be not considered in further simulations.

s = 5 mm, t = 3, 5 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
450	4,05E+05	-42,66%	38,18%	-19,34%	71,26%	55,46%
750	6,74E+05	-45,78%	12,73%	-37,29%	0,76%	-2,73%
900	8,09E+05	-21,82%	62,21%	4,19%	-9,81%	59,86%
s = 10 mm, t = 3, 5 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
450	4,05E+05	-52,75%	-9,77%	-32,96%	71,31%	-3,44%
750	6,74E+05	-33,64%	21,74%	1,35%	0,69%	15,32%
900	8,09E+05	-31,86%	23,44%	2,25%	-10,28%	11,93%
s = 15 mm, t = 3, 5 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
450	4,05E+05	-64,23%	-26,90%	-31,22%	71,44%	-14,78%
750	6,74E+05	-48,02%	0,10%	2,44%	0,80%	-6,47%
900	8,09E+05	-44,95%	2,30%	4,17%	-10,18%	-9,35%

Table 7-2 : Comparison of results between original case for **s** = 5, 10 and 15 mm and the variations for **t** = 3.5 mm.

7.2.2 t = 7 mm and s = 5, 10, 15 mm

Here, the parameter **s** is varied and parameter **t** stays constant (i.e. the spacing between inside radius of rotor and oil container is 7 mm). The results are presented from Figure 7-9 to Figure 7-11. The variant with '**s** = 5 mm' shows higher radial velocities in the axial clearance (i.e., the distance between rotor and stator) and therefore higher volume flow rates than the original case, because in the original case the backflow in established along the stator wall in the axial clearance. The volume flow rate is decreased about 10% still up to Re = 4.5E+5 (i.e., 500 RPM) and increased 2% at higher Reynolds numbers by **s** = 5 mm, compared to the original case (see Table 7-3). The reason for it is that more fluid flows though the larger rotor-stator distances, therefore the original case presented higher volume flow rates at lower revolution speed. At certain revolution speeds the volume flow rate of **s** = 5 mm is increased directly with enlarged rotation speeds and is also higher than the original case due to the fact that backflow is formed in the larger axial clearance, i.e. **s** = 10 mm. The variant with '**s** = 15 mm' shows a significantly reduced volume flow rates than the original case due to stronger backflow in the enlarged axial clearance.

Figure 7- 9 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **s** (**t** = 7 mm = constant).

The total pressure difference between inlet and outlet of the axial gap is shown in Figure 7-9, in which the curves show the same trend as volume flow rate. Due to the smaller static pressure difference and the lower flow velocity in the axial clearance the variant with '**s** = 15 mm' has a lower total pressure difference than the original case and the variant with '**s** = 5 mm'.

Figure 7- 1 0 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **s** (**t** = 7 mm = constant).

The variant **s** = 5 mm shows a slightly increased friction loss at Rotor_2 caused by the higher inflow velocity to the axial clearance. At Rotor_3 the variants with '**s** = 5 and 15 mm' show the same friction losses as 10 mm due to the constant **t**.

The friction losses at Stator_2 show the expected trend, which is increased with decreased axial gap width **s**.

Figure 7- 1 1 : Comparison of friction losses at Stator_2 vs. Reynolds numbers for varied **s** (**t** = 7 mm = constant).

A brief summary for these variants

For the axial gap variant **s** = 5 mm the volume flow rate is decreased about 10% still to 450 RPM and then increased only about 2% at higher Reynolds numbers. Relative to volume flow rate the friction losses at Rotor_2 and Stator_2 are increased about 15% and 80% at lower Reynolds number and about 3% and 50% at higher Reynolds numbers. The detailed percentages for all variants are showed in Table 7-3.

s = 5 mm, t = 7 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
300	2,70E+05	-10,42%	59,40%	19,97%	0,36%	89,92%
750	6,74E+05	2,37%	29,14%	3,29%	0,45%	55,63%
1050	9,44E+05	0,54%	33,20%	3,10%	0,48%	55,89%
s = 15 mm, t = 7 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
300	2,70E+05	-17,20%	-15,99%	-0,17%	0,00%	-12,61%
750	6,74E+05	-24,22%	-11,51%	1,02%	0,06%	-12,02%
1050	9,44E+05	-21,39%	-12,77%	0,89%	-0,04%	-12,20%

Table 7-3 : Comparison of results between original case for **s** = 5, 15 mm and the variation for **t** = 7 mm.

The variant with '**s** = 15 mm' shows a reduced volume flow rate by about 20%, and at the same time the total pressure difference is decreased about 20%. The friction losses at Rotor_2 and Rotor_3 are similar to the original case, the friction loss at Stator_2 is decreased about 12% (see Table 7-3).

7.2.3 t = 14 mm and s = 5, 10, 15 mm

The parameter **t** is constant (i.e., **t** = 14 mm) and the rotor-stator distance **s** is varied from 5 to 15 mm. Figure 7-12 shows that the volume flow rates of the variant with 's = 10 mm and t = 14 mm' are higher than in the original case (see Table 7-4). At low revolution speeds (e.g., for 150 and 300 RPM) the variant with **s** = 15 mm has about 10% higher volume flow rates than **s** = 10 mm, because the backflow is not strong at low revolution speeds. This is enlarged with increasing Reynolds numbers, the volume flow rates are decreased and they are less than in the original case. Due to the high radial velocity in the axial clearance the volume flow rates of variant with 's = 5 mm' show only small divergence to the original case 's = 10 mm, t = 7 mm' at low Reynolds numbers and presented about 15% higher mass flows above Re = 5.40E+5 (see Table 7-4).

Figure 7-1 2 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **s** (**t** = 14 mm = constant).

The variant with 's = 5 mm' presents the highest total pressure difference caused by higher radial velocities in the axial gap and the increased static pressure differences. The axial velocities are reduced with enlarged radial clearance between the inside radius of oil guide plate and the oil container, therefore the variant with 't = 14 mm' has decreased total pressure difference than the original case. Due to the larger axial gap the variant with 's = 15 mm' has the reduced static pressure difference and lower radial velocity between inlet and outlet of axial clearance, therefore it shows a lower total pressure difference as the variant with 's = 10 mm, t = 14 mm'.

The results for the friction losses at Rotor_2, Rotor_3 and Stator_2 are shown in Figure 7-13 and Figure 7-14. Because of the same distance between the inside radius of rotor and oil container (i.e., **t** = 14 mm) the friction losses at Rotor_3 for all variants are similar and they are less small than the original case due to the reduced axial velocity in the narrow radial clearance **t**.

Figure 7- 1 3 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers
for varied **s** (**t** = 14 mm = constant).

Figure 7- 1 4 : Comparison of friction losses at Stator_2 vs. Reynolds numbers for varied **s**
(**t** = 14 mm = constant).

The variant with '**s** = 10 mm, **t** = 14 mm' has the same friction loss at Stator_2 as in the original case, because of the same rotor-stator distance. They are higher with **s** = 5 mm cause of the smaller axial clearance.

A brief summary for this variant

The variant with '**s** = 10 mm, **t**= 14 mm' presents higher volume flow rates about 30% and approximately the same or lower friction losses at rotor and stator walls as the original case (see Table 7-4), which yield to better results than the original case.

s = 5 mm, t = 14 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	-23,16%	37,53%	12,13%	-15,19%	125%
450	4,05E+05	3,36%	18,80%	-0,21%	-0,07%	74%
750	6,74E+05	15,10%	10,55%	-0,07%	-11,60%	62%
s = 10 mm, t = 14 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	30,80%	-22,22%	-0,27%	-15,75%	-4,76%
450	4,05E+05	30,30%	-22,57%	-2,84%	-1,25%	-0,46%
750	6,74E+05	28,03%	-22,23%	-2,76%	-12,31%	-2,93%
s = 15 mm, t = 14 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	42,98%	-39,63%	-0,81%	-16,02%	-32,14%
450	4,05E+05	0,75%	-31,51%	-2,95%	-1,40%	-9,78%
750	6,74E+05	-5,15%	-24,08%	-2,71%	-12,03%	-11,32%

Table 7-4 : Comparison of results between original case for **s** = 5, 10, 15 mm and the variations for **t** = 14 mm.

At low Reynolds numbers, from 1.35E+5 to 3.37E+5, the variant '**s** = 15 mm, **t** = 14 mm' shows also increased volume flow rates by about 35% and reduced friction losses, compared to the original case. The volume flow rate is decreased by about 2% on average, the total pressure difference is also decreased by about 30% on average and the friction losses at the rotor and stator walls are also reduced.

For the variant with '**s** = 5 mm, **t** = 14 mm' the volume flow rate is increased after Re = 4.05E+5 about 15% on average compared to the original case, the friction losses at Rotor_2 is reduced petty and at Stator_2 is increased about 70% (see Table 7-4).

7.2.4 t = 21 mm and s = 5, 10, 15 mm

Except for 150 RPM with '**s** = 5 mm' all of the curves presented increased volume flow rates (see Figure 7-15) and their detailed percentages are presented in Table 7-5. The variant with '**s** = 10 mm, **t**= 21 mm' has the largest flow rates, the flow rates by variant '**s** = 5 mm, **t**= 21 mm' are smaller than for the variant '**s** = 15 mm, **t** = 21 mm' still up to 500 RPM, because the backflow in the larger axial clearance is not so strong at lower revolution speeds. But it is quickly increased with increased revolution speeds, therefore the more fluid passes through the 5 mm than 15 mm after 600 RPM.

Figure 7- 1 5 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **s** (**t** = 21 mm = constant).

The plots of the total pressure differences for the variant with '**t** = 21 mm' in Figure 7-15 showed the same trend as the above presented variant with **t** = 14 mm, which can be seen in Figure 7-12.

Figure 7- 1 6 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **s** (**t** = 21 mm = constant).

The variant with '**s** = 5 mm' has the highest friction losses at Rotor_2 and Stator_2 caused by the small gap (see Figure 7-7, Figure 7-15 and 7-16). The friction losses at Rotor_2 for **s** = 10 and 15 mm showed a small difference due to the similar inflow velocities. The friction losses at Rotor_3 for all variations are approximately the same or lower than it in the original case.

Figure 7- 1 7 : Comparison of friction losses at Stator_2 vs. Reynolds numbers for varied **s** (**t** = 21 mm = constant).

A brief summary for these variants

The variant with '**s** = 10 mm, **t** = 21 mm' shows the best result, due to the strongly increased volume flow rates by about 50% and only small differences in comparison to the original case for the friction losses at rotor and stator walls.

s = 5 mm, t = 21 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	-19,78%	21,92%	9,97%	-16,57%	138,10%
450	4,05E+05	11,15%	4,00%	-3,21%	2,42%	83,51%
750	6,74E+05	24,22%	-4,25%	-3,04%	-14,80%	73,09%
s = 10 mm, t = 21 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	42,58%	-31,92%	-1,62%	-17,13%	2,38%
450	4,05E+05	48,34%	-34,93%	-5,58%	0,97%	5,21%
750	6,74E+05	48,29%	-35,47%	-5,72%	-15,76%	2,66%
s = 15 mm, t = 21 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	66,38%	-49,79%	-2,16%	-17,68%	-28,57%
450	4,05E+05	21,55%	-42,40%	-5,89%	0,94%	-6,99%
750	6,74E+05	13,12%	-32,89%	-5,44%	-14,93%	-10,87%

Table 7-5 : Comparison of results between original case for **s** = 5, 10, 15 mm and the variations for **t** = 21 mm.

Almost the same results shows the variant with '**s** = 15 mm, **t** = 21 mm'. Here, the volume flow is increased about 20% after 450 RPM but still lower than the variant with '**s** = 10 mm', the friction loss at Rotor_2 is similar to the original case and the friction loss at Rotor_3 is decreased only about 10% on average. Stator_2 has also reduced friction loss compared to the original case.

The variant with 's = 5 mm, t = 21 mm' has about 18% higher flow rates and the total pressure difference is just like in the original case after 300 RPM, but the friction loss at Stator_2 is almost doubled.

7.2.5 t = 28 mm and s = 5, 10, 15 mm

The variant with 's = 10 mm, t = 28 mm' shows higher volume flow rates after 300 RPM in the left hand side of Figure 7-18. At the Reynolds number of 1.35E+5 the variant with 's = 15 mm' has higher flow rate than the variant with 's = 10 mm, t = 28 mm', because of reduced backflow at low revolution speeds. With increasing revolution speeds the variant **s** = 15 mm shows an increased flow rate by about 40%. After 300 RPM the volume flow rate of variant **s** = 5 mm is increased about 30% (see Table 7-6).

The total pressure differences of all variants show a similar trend like **t** = 14 and 21 mm, but here the variant with '**s** = 5 mm' presented lower total pressure difference than the original case due to the reduced inflow velocity with increased parameter **t**.

Figure 7- 1 8 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **s** (**t** = 28 mm = constant).

Due to the high radial velocity in the axial clearance between rotor-stator distance the variant with '**s** = 5 mm' shows the largest friction losses at Rotor_2 and Stator_2. All variants give reduced friction losses at rotor walls, at Stator_2 the variant with '**s** = 15 mm' showed decreased friction loss, the variant with '**s** = 10 mm' has increased friction loss and '**s** = 5 mm' has almost doubled the friction loss at Stator_2 compared to the original case (see Figure 7-19 and Figure 7-20).

Figure 7- 1 9 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers
for varied **s** (**t** = 28 mm = constant).

Figure 7- 2 0 : Comparison of friction losses at Stator_2 vs. Reynolds numbers for varied **s**
(**t** = 28 mm = constant).

A brief summary for these variants

The variant with '**s** = 10 mm, **t** = 28 mm' shows the increased volume flow rate after 150 RPM, which is about 70% higher than for the original case. But the total pressure differences are strongly reduced by about 45% on average. The friction losses at rotor walls are similar to the original case and the friction loss at Stator_2 is increased only about 10%, which can be found in following table:

s = 5 mm, t = 28 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	-16,17%	6,23%	7,55%	-19,34%	152,38%
450	4,05E+05	19,52%	-9,69%	-6,21%	5,92%	94,08%
750	6,74E+05	32,93%	-18,41%	-5,73%	-14,05%	88,83%
s = 10 mm, t = 28 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	52,18%	-39,71%	-3,23%	-19,89%	11,90%
450	4,05E+05	66,48%	-44,88%	-8,23%	4,07%	11,55%
750	6,74E+05	70,08%	-48,34%	-9,16%	-15,57%	13,63%
s = 15 mm, t = 28 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	83,81%	-55,21%	-3,77%	-20,44%	-21,43%
450	4,05E+05	44,90%	-51,72%	-8,66%	4,05%	-2,19%
750	6,74E+05	34,76%	-43,74%	-9,17%	-14,79%	-4,51%

Table 7-6 : Comparison of results between original case for **s** = 5, 10, 15 mm and the variations for **t** = 28 mm.

Almost the same results gave the variant with '**s** = 15 mm, **t** = 28' which presents an increased mass flow rate about 40% after 375 RPM. It is less than the variant with '**s** = 10 mm, **t** = 28 mm', because of the stronger backflow in the larger axial clearance **s**. The friction loss at Stotor_2 is also minimal reduced about 2% on average and the total pressure difference is here strongly reduced about 50%, compared to the original case (see Table 7-6).

The variant with '**s** = 5 mm, **t** = 28 mm' has about 16% reduced volume flow at low revolution speeds compared to the original case, which is increased by about 20 % on average with enlarged Reynolds numbers. The total pressure difference and the friction loss at Rotor_2 are similar as the original case. The friction loss at Rotor_3 is decreased by about 10%, but Stator_2 shows high friction losses. With this variant can not yield better results than with the original case.

A brief summary for varied parameter s

The parameter **s** has a significantly influence. The variants with '**s** = 10 mm' and constant parameter **t** showed the best results. The variants with '**s** = 5 mm' lead to higher radial velocities in the distance between rotor and stator but it restrains the inflow volume due to the narrow axial gap, compared to the original case. The backflow is increased with enlarged axial clearance while the radial gap **t** stays constant, therefore the variant with '**s** = 15 mm' shows lower volume flow rates than the variant with '**s** = 10 mm'.

7.3 The spacing between inside radius of rotor and oil container – Parameter *t*

The geometry parameter **t** represents the spacing between inside radius of rotor and oil container, i.e., the radial clearance in the rotor-stator system.

Enlarging parameter **t** leads to a larger inlet volume that means an increased volume flow rate. Several simulations have been made with varied parameter **t** to investigate how much the volume flow rate is increased and how the friction losses at rotor and stator walls are changed with varied radial clearance. The results are presented in following subsections and are compared with the original case '**s** = 10 mm, **t** = 7 mm'.

The diagrams with the numerical results show the volume flow rate, total pressure difference and friction losses at rotor and stator walls versus 10 Reynolds numbers, in which the yellow columns are describing the original case and the line plots are indicating the results from varying parameter **t**.

7.3.1 s = 5 mm, t = 7, 14, 21 and 28 mm

The simulations are started with constant rotor-stator distance **s** and different radial clearances **t**. The left side of Figure 7-21 shows the volume flow rates as a function of Reynolds number for different parameter **t** (i.e., **t** is varied from 7 mm to 28 mm). The flow rates are increased with enlarged radial clearances due to the enlarged inflow volume.

Figure 7- 2 1 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t** (**s** = 5 mm = constant).

The total pressure difference depends on the static pressure difference and the radial velocity between inlet and outlet of the axial gap. In Figure 7-21 it can be seen that the total pressure

difference is increased with decreased radial clearance **t** due to the faster inflow velocity in the axial gap.

Figure 7- 2 2 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t** (**s** = 5 mm = constant).

All variants show the similar behaviour for the friction loss at 'Rotor_2' and they are nearly equal the friction loss in the original case '**s** = 10 mm, **t** = 7 mm' (see Figure 7-22), in which the variant with '**t** = 7 mm' shows the highest friction loss at Rotor_2 due to the higher inflow velocity.

Due to the same width of the radial clearance as the original case the variant with '**s** = 5 mm, **t** = 7 mm' presented similar friction loss at Rotor_3 can be observed and the other variants have decreased or approximately the same friction loss at this rotor wall like the original case (see Table 7-7).

Figure 7- 2 3 : Comparison of friction losses at Stator_2 vs. Reynolds numbers for varied **t** (**s** = 5 mm = constant).

All of the variants have higher friction losses at Stator_2 than the original case, because there is no backflow in the narrow rotor-stator distance and higher radial velocities in the gap. The radial

velocity is increased in the constant narrow axial clearance (e.g., **s** = 5 mm), when the radial clearance **t** is enlarged. Therefore the variant with 't = 28 mm' shows the highest friction losses at Stator_2.

<u>A brief summary for this variant</u>

For the variant with '**s** = 5mm, **t** = 7 mm' the volume flow rate is reduced about 20% at low Reynolds numbers because of the reduced inflow volume and higher total pressure differences due to increased radial velocities (see Table 7-7). Due to the same radial gap width between inside radius of rotor and oil container as in the original case this variant showed similar friction loss at Rotor_3. The friction loss at Rotor_2 is increased by about 20% at low Reynolds numbers and about 5% at high Reynolds numbers. Distinctly increased friction loss can be found at Stator_2 (see Table 7-7). So to speak, with this variant we can not obtain better results than with the original case.

After 300 RPM the variants with **t** = 14, 21 and 28 mm have increased volume flow rates about 10%, 20% and 30% on average. As an example the variant with '**t** = 14 mm' shows that the volume flow rate is increased by about 10% after 300 RPM and therefore results in larger total pressure differences. Up to 375 RPM the friction loss at Rotor_2 is increased by about 15% and then has similar friction losses like the original case. Friction loss at Stator_2 is strongly increased because of higher radial velocities. The variants with '**t** = 21 and 28 mm' have higher volume flow rates, higher friction losses at Stator_2 than the original case and approximately the same friction losses at rotor walls like **t** = 14 mm. These three variants perform better for the Reynolds numbers larger than Re = 3.37E+5 (i.e., after 375 RPM).

s = 5 mm, t = 7 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	-29,14%	54,11%	13,75%	0,28%	115,48%
375	3,37E+05	-2,98%	56,01%	21,60%	1,19%	81,68%
750	6,74E+05	2,37%	29,14%	3,29%	0,45%	55,63%
s = 5 mm, t = 14 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	-23,16%	37,53%	12,13%	-15,19%	125,00%
375	3,37E+05	9,81%	32,93%	19,26%	-8,84%	94,18%
750	6,74E+05	15,10%	10,55%	-0,07%	-11,60%	62,01%
s = 5 mm, t = 21 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	-19,78%	21,92%	9,97%	-16,57%	138,10%
375	3,37E+05	24,74%	42,37%	33,63%	-7,01%	113,87%
750	6,74E+05	24,22%	-4,25%	-3,04%	-14,80%	73,09%
s = 5 mm, t = 28 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	-16,17%	6,23%	7,55%	-19,34%	152,38%
375	3,37E+05	33,14%	24,85%	29,90%	-6,38%	125,51%
750	6,74E+05	32,93%	-18,41%	-5,73%	-14,05%	88,83%

Table 7-7 : Comparison the results with original case in certain percentage for **s** = 5 mm and **t** = 7, 14, 21 and 28 mm.

7.3.2 s = 10 mm, t = 7, 14, 21 and 28 mm

Figure 7-24 to Figure 7-26 show the volume flow rate, total pressure difference and friction losses at rotor and stator walls over Reynolds numbers for the axial clearance **s** = constant = 10 mm and the radial clearances are varied from 7 to 28 mm, which presented a similar trend like for **s** = 5 mm. The volume flow rates of the variant with '**s** = 10 mm' are substantially higher than the variant with '**s** = 5 mm'. The total pressure differences are significantly reduced with enlarged radial clearances due to the decreased fluid velocities. All of variants have similar friction losses at Rotor_2 as the original case and show reduced friction losses at Rotor_3.

Figure 7- 2 4 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t** (**s** = 10 mm = constant).

Figure 7- 2 5 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t** (**s** = 10 mm = constant).

Due to the larger distance between the inside radius of stator and the oil container the system has more inflow volume and stronger backflow occurs for the variant ' **s** = 10 mm, **t** = 28 mm', which yields an increased friction loss at Stator_2.

Figure 7- 2 6 : Comparison of friction loss at Stator_2 vs. Reynolds numbers for varied **t** (**s** = 10 mm = constant).

<u>A brief summary for these variants</u>

The axial clearance with 10 mm presented generally higher volume flow rates than the original case, i.e. the volume flow rates are increased with enlarged distance between inside radius of rotor and oil container **t**. For example the volume flow rate is increased about 30% by **t** = 14 mm, 45% by **t** = 21 mm and 70% by **t** = 28 mm, compared to the original case (see Table 7-8). The friction losses at Rotor_3 for all three variants are reduced in similar way. Due to the stronger backflow the variant with 28 mm shows the highest friction loss at Stator_2, but it is only about 15% higher than in the original case (see Table 7-8).

The variants with '**s** = 10 mm, **t** = 14 and 21 mm' perform better, because the friction losses are similar or smaller than in the original case and mass flow rates are increased between 30% and 45%. Although the variant with '**t** = 28 mm' presents the highest volume flow rate, it can not be proposed as the best variant. Since the total pressure difference is decreased about 50% compared to the original case.

s = 10 mm, t = 14 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
450	4,05E+05	30,30%	-22,57%	-2,84%	-1,25%	-0,46%
750	6,74E+05	28,03%	-22,23%	-2,76%	-12,31%	-2,93%
1050	9,44E+05	28,10%	-22,37%	-3,00%	-11,27%	-0,25%
s = 10 mm, t = 21 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
450	4,05E+05	48,34%	-34,93%	-5,58%	0,97%	5,21%
750	6,74E+05	48,29%	-35,47%	-5,72%	-15,76%	2,66%
1050	9,44E+05	47,67%	-35,84%	-5,90%	-13,83%	7,43%
s = 10 mm, t = 28 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
450	4,05E+05	66,48%	-44,88%	-8,23%	4,07%	11,55%
750	6,74E+05	70,08%	-48,34%	-9,16%	-15,57%	13,63%
1050	9,44E+05	66,40%	-46,09%	-8,45%	-12,36%	16,15%

Table 7-8 : Comparison the results with original case in certain percentage for **s** = 10 mm and **t** = 14, 21 and 28 mm.

7.3.3 s = 15 mm, t = 7, 14, 21 and 28 mm

Here, the rotor-stator distance is enlarged to 15 mm and the radial clearance **t** is varied from 7 to 28 mm. The results are presented in the following figures. The volume flow rates and total pressure differences of these variants show the similar trends like the variants with **s** = 5 and 10 mm (see Figure 7-27). Due to the enlarged distance between rotor and oil container the variants **t** = 21 and 28 mm have higher volume flow rates over all Reynolds numbers. Up to Re = 3.37E+5 (i.e., N = 375 RPM) the variant with **t** = 14 mm has a larger volume flow rate and after this Reynolds number the volume flow rates are minimally reduced, compared to the original case. The reason for this is that more backflow occurs along the stator wall side with increased rotor-stator distance and it would become stronger with greater revolution speeds. The variant with '**s** = 7 mm' proved that the volume flow rate is reduced because of the decreased inflow volume and backflow.

Figure 7- 2 7 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t** (**s** = 15 mm = constant).

Due to the backflow and higher velocity differences between inlet and outlet for these variants (i.e., **s** = 15 mm) the total pressure differences are clearly increased with reduced radial clearance compared to the original case (see right hand side of Figure 7-27).

The variant with '**s** = 15 mm, **t** = 7 mm' shows increased friction losses at rotor walls due to the higher axial velocities and the friction losses of the other variants are slightly reduced or approximately the same as the original case (see Figure 7-28 and Figure 7-29).

Figure 7-2 8 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t** (**s** = 15 mm = constant).

Figure 7-2 9 : Comparison of friction loss at Stator_2 vs. Reynolds numbers for varied **t** (**s** = 15 mm = constant).

With enlarged inflow volume the friction loss at Stator_2 is increased, thereby the backflow is also stronger.

A brief summary for these variants

The variant with '**s** = 15 mm, **t** = 7 mm' shows a reduced volume flow rate, about 10% than the original case at low Reynolds numbers. After Re = 4.05E+5 the volume flow rate is decreased about 23% (see Table 7-9). The friction losses at rotor walls are similar as the original case and the friction loss at Stator_2 is reduced by about 10%. This variant shows not better results than the original case.

Up to a Reynolds number of 3.37E+5 the variant with '**t** = 14 mm' shows an increased volume flow rate by about 20% compared to the original case. After this Reynolds number the volume flow rates are reduced minimally (see Table 7-9). The friction losses at rotor walls are similar as the original case, the total pressure difference is decreased by about 20% and the friction loss at

Stator_2 is also reduced by about 30% up to Re = 3.37E+5. At the higher Reynolds numbers the friction loss is reduced only by about 10%. This variant shows better results at low Reynolds numbers, i.e. smaller than the Re = 4.05E+5.

The variant with **t** = 21 and 28 mm has clearly increased volume flow rates by about 50% and 70% at low Reynolds number (i.e., Re = 1.35E+5 to 3.37E+5). Above Re = 4.05E+5 these two variants show a strongly reduced total pressure difference and similar friction losses at rotor and stator walls (see Table 7-9). The mass flow rates are increased by 20% to 40% on average.

s = 15 mm, t = 7 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	-4,87%	-41,35%	-0,27%	0,00%	-21,43%
450	4,05E+05	-24,49%	-16,58%	0,40%	-0,39%	-7,38%
750	6,74E+05	-24,22%	-11,51%	1,02%	0,06%	-12,02%
s = 15 mm, t = 14 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	42,98%	-39,63%	-0,81%	-16,02%	-32,14%
450	4,05E+05	0,75%	-31,51%	-2,95%	-1,40%	-9,78%
750	6,74E+05	-5,15%	-24,08%	-2,71%	-12,03%	-11,32%
s = 15 mm, t = 21 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	66,38%	-49,79%	-2,16%	-17,68%	-28,57%
450	4,05E+05	21,55%	-42,40%	-5,89%	0,94%	-6,99%
750	6,74E+05	13,12%	-32,89%	-5,44%	-14,93%	-10,87%
s = 15 mm, t = 28 mm						
revolution speed	Re	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
[U/min]	[-]	[%]	[%]	[%]	[%]	[%]
150	1,35E+05	83,81%	-55,21%	-3,77%	-20,44%	-21,43%
450	4,05E+05	44,90%	-51,72%	-8,66%	4,05%	-2,19%
750	6,74E+05	34,76%	-43,74%	-9,17%	-14,79%	-4,51%

Table 7-9 : Comparison the results with original case in certain percentage for **s** = 15 mm and **t** = 14, 21 and 28 mm.

The general increased volume flow rates of all variants leads to the conclusion that the flow rate through the axial gap is influenced by parameter **t** and it is increased with enlarged radial clearance.

A brief summary for varied parameter t

The above analyses showed that the volume flow rate depends on the width of the radial clearance. The variant with the largest radial clearance shows the highest volume flow rate.

7.4 The results comparisons of parameters t', a and at' for s = 5, 10 and 15 mm with the original case 's = 10 mm and t = 7mm'

The above subsection showed the results for constant axial clearance **s** and varied radial clearance **t**, from 7 mm to 28 mm. In order to investigate how much difference can be seen when only the inflow volume is varied and the axial and radial clearances stay constant, further simulations have been carried out with the new geometry parameter **t'**, i.e., the spacing between inside radius of stator and oil container. In order to investigate if the backflow can be retarded in the rotor-stator distance by extension of the guide plate in the radial direction, the parameter **a** has been considered in the variation. Finally, simulations with a combination of these two new parameters **t'** and **a**, which are indicated as parameter **at'**, are made.

Here, '**t'** = 7 mm' means that the distance between inside radius of stator and oil container **t'** is enlarged by 7 mm, i.e. the spacing between inside radius of stator and oil container is 14 mm now and the radial clearance **t** stays always 7 mm (see Chapter 6). The variant with '**a** = 7 mm' means that the outside radius of the stator is greater than the outside radius of rotor by 7 mm. The last variant with parameter **at'** means that the whole stator is shifted 7 mm outside, which is a combination of parameters **t'** and **a**.

With these three new parameters and constant parameters **s** and **t** the simulations have been carried out and the results will be compared with the original case '**s** = 10 mm, **t** = 7 mm', which are shown in following subsections. The yellow columns in the figures present the results of the original case and the other plots describe the results with different parameters **t, t', a** and **at'**. Here, the parameter **t** will be considered again to compare the influences of all the parameters.

7.4.1 The variations with constant axial clearance s = 5 mm

For the variants with '**s** = 5 mm' the simulations are automatically stopped after 600 RPM, because the mesh is distorted too much through the variation with parameter **at'** and the solution gets unstable with the parameterised model at higher revolution speeds.

From the last subsection it can be inferred that parameter **t** gives us more substantial impact while axial clearance **s** stays constant. With increased radial clearance **t** and constant axial clearance **s** the volume flow rate is enlarged. The variants with '**t** = 7 mm' do not lead to better results than the original case. The following figures show the results of the variants with '**s** = 5 mm, **t** = 14 and 21 mm' as examples and are compared with the original case.

Combination of all parameters for the variants with 's = 5 mm, t = 14 mm'

Figure 7- 3 0 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 5 mm = constant, **t** = 14 mm).

Figure 7- 3 1 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 5 mm = constant, **t** = 14 mm).

Figure 7- 3 2 : Comparison of friction loss at Stator_2 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 5 mm = constant, **t** = 14 mm)

Combination of all parameters for the variants with 's = 5 mm, t = 21 mm'

Figure 7- 3 3 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 5 mm = constant, **t** = 21 mm).

Figure 7- 3 4 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 5 mm = constant, **t** = 21 mm).

Figure 7- 3 5 : Comparison of friction loss at Stator_2 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 5 mm = constant, **t** = 21 mm).

The results of the selected examples have proven that the volume flow rates and friction losses at Stator_2 are increased with enlarged radial clearances and they are higher than in the original case. The friction losses at Rotor_2 gave a similar trend and are approximately the same as in the original case. Friction losses at Rotor_3 are reduced in both examples, except for 500 RPM. The variant with '**at'** = 7 mm' has a greatly reduced total pressure difference due to the decreased static pressures.

The variant of '**s** = 5 mm' with shortened stator (i.e., parameter **t'**) and enlarged radial clearance (i.e., parameter **t**) show better results compared to the original case.

7.4.2 The variations with constant axial clearance s = 10 mm

The variants with '**s** = 10 mm, **t** = varied = 14, 21 and 28 mm' show generally similar or better results than the original case '**s** = 10 mm, **t** = 7 mm', which can be seen in the following subsections.

For the flow variables, i.e. the volume flow rate, total pressure difference and friction losses at rotor and stator walls, the variants with shortened stator **t'** = 7 mm and enlarged radial clearance **t**, i.e., '**s** = 10 mm, **t** = 7 mm, **t'** = 7mm' and '**s** = 10 mm, **t** = 14 mm', '**s** = 10 mm, **t** = 14 mm, **t'** = 7mm' and '**s** = 10 mm, **t** = 21 mm', '**s** = 10 mm, **t** = 21 mm, **t'** = 7mm' and '**s** = 10 mm, **t** = 28 mm', show the best results. The reason for this is that the volume flow rates are increased with enlarged inlet volume, i.e. the distance between the inside radius of stator and oil container. Therefore the parameter **t'**, **at'** and enlarged **t** show the highest volume flow rates for each variant (see Figure 7-37, Figure 7-44, Figure 7-47 and Figure 7-50). But the variants with '**at'** = 7 mm' have a significantly reduced total pressure differences by about 40% on average. The variants with '**a** = 7 mm' have similar volume flow rates as in the original case but higher friction losses at Stator_2 than original case and the other variants.

Combination of all parameters for the variants with 's = 10 mm, t = 7 mm'

In the following figures the reason for flattening of the volume flow rates between 300 and 375 RPM are seen in Figure 7-36:

- According to the simulation plan the rotational velocities should be increased by 150 RPM increments from 150 to 1, 200 RPM. The 375 RPM is the original revolution speed for the generator 'Glendoe', which is only 75 RPM higher than 300 RPM. Therefore the trend for the volume flow rate at this point is not clear (see Figure 7-37). But with increased radial clearance and constant axial clearance significantly higher volume flow rates can be recognized due to the increased inflow volume (see Figure 7-37, Figure 7-44 and Figure 7-47).

- The backflow at 375 RPM passes through along the axial clearance (see Figure 7-36). At 300 RPM the backflow is formed only at the end of the axial clearance and the re-

circulation zone can be found at the beginning of the axial clearance, which is marked with white colour.

Figure 7- 3 6 : Fluid flow characteristics of axial clearance at 300 and 375 RPM.

Figure 7- 3 7 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 10 mm = constant, **t** = 7 mm).

Figure 7- 3 8 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 10 mm = constant, **t** = 7 mm).

Except for the variant with 't = 14 mm' the friction losses at Rotor_3 are similar to the original case due to the same circumferential velocities and are decreased with enlarged radial clearance. The friction losses at Rotor_2 show approximately the same results as the original case.

At Stator_2 the friction loss is increased with enlarged length of stator, therefore the variant with 'a = 7 mm' has the highest friction loss.

Figure 7- 3 9 : Comparison of friction loss at Stator_2 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 10 mm = constant, **t** = 7 mm).

Combination of all parameters for the variants with 's = 10, t = 14 mm'

Comparison of the variant with '**t** = 7 mm' in Figure 7-44 shows that the volume flow rates are increased with enlarged radial clearance **t**, compared to the original case.

All of the variants show significantly increased volume flow rates from 900 to 1050 RPM (see Figure 7-37, Figure 7-44, Figure 7-47, Figure 7-50, Figure 7-54, Figure 7-57, Figure 7-60 and Figure 7-63) and are not influenced by the different parameter. In order to investigate this behaviour shown in the plots the region between the stator_4 wall and the bottom of the oil container has been examined closer.

- Figure 7-40 shows three radial iso-surfaces between the bottom of oil container and Stator_4 at 0.538, 0.491 and 0.457 m, where the flow velocities will be depicted in the problem region for 900 and 1050 RPM.

Figure 7- 4 0 : Three iso-surfaces at different radial heights between the bottom of oil container and Stator_4.

Figure 7- 4 3 : Radial velocity profile in the region between bottom of oil container and Stator_4 for the three radial iso-surfaces at 1050 RPM.

- At 1050 RPM, the fluid passes in general with higher radial velocities through the distance between the bottom of oil container and Stator_4 than at 900 RPM. The radial velocities at Stator_4 by 1050 RPM are about 30% higher than by a revolution speed of 900 RPM, which is the reason for the rapid change between 900 and 1050 RPM in the plots for the volume flow rate for all variants.

Figure 7- 4 4 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 10 mm = constant, **t** = 14 mm).

The total pressure differences of all variants are decreased compared to the original case due to the reduced fluid velocities. Figure 7-45, Figure 7-48 and Figure 7-51 show similar trends for friction losses at Rotor_3. They are smaller than in the original case. The friction losses at Rotor_3 of all variants are not be influenced by the width of radial clearance, when the radial clearance **t** $\geq$ 14 mm.

The friction losses at Rotor_2 show approximately the same results as the original case and at Stator_2 the variant with extended length of the stator, i.e. **a** = 7 mm, shows the highest friction loss.

Figure 7- 4 5 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 10 mm = constant, **t** = 14 mm).

Figure 7- 4 6 : Comparison of friction loss at Stator_2 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 10 mm = constant, **t** = 21 mm).

Combination of all parameters for the variants with 's = 10 mm, t = 21 mm'

Figure 7- 4 7 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 10 mm = constant, **t** = 21 mm).

Figure 7- 4 8 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 10 mm = constant, **t** = 21 mm).

Figure 7- 4 9 : Comparison of friction loss at Stator_2 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 10 mm = constant, **t** = 21 mm).

Combination of all parameters for the variants with 's = 10 mm, t = 28 mm'

Figure 7- 5 0 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 10 mm = constant, **t** = 28 mm).

Figure 7- 5 1 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 10 mm = constant, **t** = 28 mm).

Figure 7- 5 2 : Comparison of friction loss at Stator_2 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 10 mm = constant, **t** = 28 mm).

For the variants with 't = 21 mm' the total pressure differences are reduced by about 42% on average and the volume flow rates are increased by about 60%. The variants with 't = 28 mm' show also highly decreased total pressure differences by about 50% on average and volume flow rates are increased by about 70% (see Table 7-10).

The variants with 't = 7 mm' have also increased volume flow rates by about 18%, the variants with 't = 14 mm' show higher volume flow rates by about 45% on average, about 30% reduced total pressure differences and decreased or similar friction losses at rotor and stator walls, compared to the original case (see Table 7-10). About 60% increased volume flow rate are reached by variants with 't = 21 mm' and the highest volume flow rate has the variant with 't = 28 mm', but these two variants have both significantly reduced total pressure differences by about 42% and 51% and the friction losses are less decreased or similar as the original case.

7.4.3 The variations with constant axial clearance s = 15 mm

Similar results as the variants 's = 10 mm' show the variants with 's = 15 mm', but volume flow rates are lower than the variants with 's = 10 mm' (see Figure 7-54, Figure 7-57, Figure 7-60 and Figure 7-63) due to the establishing backflow. The total pressure differences of these variants with 'at' = 7 mm' are strongly reduced compared to the other variants. The variants with 'a = 7 mm' have the largest friction losses at Stator_2, because of the extended length of the stator.

Combination of all parameters for the variants with 's = 15 mm, t = 7 mm'

Figure 7-54 shows that the volume flow rates of all variants are similar or smaller than in the original case after 375 RPM due to the strong backflow. Here, the plots show also a rapid increase by 375 RPM just like the variants with 's = 10 mm' and the reason for this was discussed in Subsection 7.4.2.

As examples the fluid flow characteristics showing the backflow in the axial gap for the variants with 's = 10 mm, t = 7 mm' and 's = 15 mm, t = 7 mm' are presented in Figure 7-53. It can be seen that the variant with 's = 10 mm' has higher radial velocity at beginning of the axial clearance and has not so much backflow than the variant with 's = 15 mm'. Therefore it shows higher volume flow rate than the variant with 's = 15 mm'.

Figure 7- 5 3 : The backflow comparison between 's = 10 mm'at the left side and 's = 15 mm' at right side with scaled radial velocities.

Figure 7- 5 4 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 15 mm = constant, **t** = 7 mm).

The total pressure differences are also reduced compared to the original case and the variant with '**at'** = 7 mm' shows the lowest total pressure difference due to the reduced static pressure differences.

The friction losses at Rotor_2 are similar as in the original case. Due to the enlarged radial clearance the friction losses at Rotor_3 with variant '**t** = 14 mm' are lower than with '**t** = 7 mm'. The friction losses at Stator_2 are decreased with shortened length of stator and increased with extended stator length. Therefore the variant with '**t'** = 7mm' shows the lowest friction loss at Stator_2 and with '**a** = 7 mm' the highest.

Figure 7- 5 5 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 15 mm = constant, **t** = 7 mm).

Figure 7- 5 6 : Comparison of friction loss at Stator_2 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 15 mm = constant, **t** = 7 mm).

The variants with '**t** = 7 mm' can not yield to better results than the original case, because the volume flow rates are not significantly increased. Total pressure differences are decreased, the friction losses at rotor walls are almost the same one like in the original case and the maximum difference of the friction losses at Stator_2 between the variations and original case are about 20 %.

Combination of all parameters for the variants with 's = 15 mm, t = 14 mm'

Figure 7-57 shows the volume flow rates for variants with '**t** = 14 mm' which are higher than for the variants with '**t** = 7 mm' and they are increased with enlarged radial clearance **t** while axial clearance **s** stays constant (see Figure 7-57, Figure 7-60 and Figure 7-63). Compared to the original case, the total pressure differences are reduced by about 30% on average and the worst total pressure difference results with the variant '**at'** = 7 mm' due to the reduced static pressure difference.

Figure 7- 5 7 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 15 mm = constant, **t** = 14 mm).

Figure 7- 5 8 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 15 mm = constant, **t** = 14 mm).

Figure 7- 5 9 : Comparison of friction loss at Stator_2 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 15 mm = constant, **t** = 14 mm).

Figure 7-58 shows similar friction losses at Rotor_3, which proved once again the theory of variants with '**s** = 10 mm':

-   From certain width of radial clearance **t** (i.e., t ≥ 14 mm) the friction loss at Rotor_3 will not be significantly changed (see Figure 7-57, Figure 7-60, Figure 7-63).

With variants '**t** = 14 mm' the volume flow rates are not really increased, total pressure differences are reduced by about 30% on average and the friction losses at rotor and stator walls are approximately the same or reduced by a small amounts. Compared to the variants with '**s** = 10 mm' they are not seen to be optimal.

Combination of all parameters for the variants with 's = 15 mm, t = 21 mm'

The volume flow rates are again increased with enlarged radial clearance and they are higher than in the original case. The flat trend-line from 300 to 375 RPM was explained in the last subsection. In Figure 7-60 it can be seen that the total pressure differences are strongly reduced.

Figure 7- 6 0 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 15 mm = constant, **t** = 21 mm).

Figure 7- 6 1 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 15 mm = constant, **t** = 21 mm).

Figure 7- 6 2 : Comparison of friction loss at Stator_2 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 15 mm = constant, **t** = 21 mm).

Compared to the variant with '**s** = 10 mm, **t** = 21 mm, **t'** = 7 mm, **a** = 7 mm and **at'** = 7 mm' the variant with '**s** = 15 mm, **t** = 21 mm, **t'** = 7 mm, **a** = 7 mm and **at'** = 7 mm' shows the lowest volume flow rate increase except for 150 and 300 RPM due to the missing or less backflow at lower revolution speeds. The variants show strongly reduced total pressure differences and similar or less decreased friction losses at rotor and stator walls. So to speak, the results of the variants with '**s** = 15 mm' are not the better than the variants with '**s** =10 mm'.

Combination of all parameters for the variants with 's = 15 mm, t = 28 mm'

The plots of the variants with '**s** = 15 mm' show the same trends and the same flow behaviour as the last variants and they have higher volume flow rates than the original case but smaller than the variants with '**s** = 10 mm, **t** = 28 mm, **t'** = 7 mm, **a** = 7 mm and **at'** = 7 mm'. Similar total pressure differences are seen for both variants, which are much smaller than in the original case.

Figure 7- 6 3 : Comparison of volume flow rate and total pressure difference vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 15 mm = constant, **t** = 28 mm).

Figure 7- 6 4 : Comparison of friction losses at Rotor_2 and Rotor_3 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 15 mm = constant, **t** = 28 mm).

Figure 7- 6 5 : Comparison of friction loss at Stator_2 vs. Reynolds numbers for varied **t'**, **a** and **at'** (**s** = 15 mm = constant, **t** = 28 mm).

The friction losses at rotor walls stay similar for these two variants (i.e., **s** = 10 and 15 mm) and the friction losses at Stator_2 are decreased by only about 15% on average.

This variant shows not better results than the variant with '**s** = 10 mm, **t** = 28 mm, **t'** = 7 mm, **a** = 7 mm and **at'** = 7 mm'. The width of axial clearance **s** = 10 mm shows the highest volume flow rates and similar friction losses at rotor and stator walls.

7.5 Fluid flow comparsions between selected variants with the original case

From the simulation results in Subsection 7.4 can be inferred that the variants 's = 5 and 15 mm' yield no better results than the variants 's = 10 mm', therefore the variants with 's = 10 mm' are selected and discussed here:

- Table 7-10 shows the variations of all varied parameters with 's = 10 mm', it can be seen that higher volume flow rates compared to the original case and they are increased with enlarged radial clearance **t**. The variants with the largest radial clearance (i.e., **t** = 28 mm) shows the highest volume flow rates.

Variants	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
	[%]	[%]	[%]	[%]	[%]
t = 7 mm	18,16%	-24,92%	0,05%	-0,21%	1,93%
t = 14 mm	45,75%	-31,73%	1,08%	-8,64%	8,85%
t = 21 mm	60,19%	-42,24%	0,38%	-10,46%	9,09%
t = 28 mm	69,69%	-51,03%	-2,63%	-9,96%	-5,02%

Table 7-10 : Comparisons between the variants with 's = 10 mm' and the original case in percentage.

- Although the variants with '**t** = 21 and 28 mm' show the highest volume flow rates, they also lead to very strongly reduced total pressure differences. It means that the total pressure difference is decreased with enlarged radial clearance due to the decreased static pressure difference and dynamic pressure.

- The variants with '**t** = 14 mm' are selected as examples to compare with the original case the fluid flow characteristics, because they show higher volume flow rates of about 46% than the original case, decreased total pressure differences by about 32% and similar friction losses at rotor and stator walls (see Figure 7-44 to Figure 7-46):

  ➤ **s** = 10 mm, **t** = 14 mm

  ➤ **s** = 10 mm, **t** = 14 mm, **t'** = 7 mm

  ➤ **s** = 10 mm, **t** = 14 mm, **a** = 7 mm

  ➤ **s** = 10 mm, **t** = 14 mm, **at'** = 7 mm

  ➤ **s** = 10 mm, **t** = 21 mm

Table 7-11 shows the comparison of the five variants with the original case in percentage. The best results show the variants with 's = 10 mm, **t** = 21 mm', 's = 10 mm, **t** = 14 mm, **at'** = 7 mm' and 's = 10 mm, **t** = 14 mm, **t'** = 7 mm'. The variant with 's = 10 mm, **t** = 14 mm, **a** = 7

mm' shows similar results as the variant with 's = 10 mm, t = 14 mm, t' = 7 mm', i.e., the volume flow rate is decreased about 1%, total pressure difference is reduced by about 3% and friction loss at Stator_2 is increased by about 13%.

s = 10 mm, t = 14 mm					
Variants	**Volume flow rate**	**Total pressure difference**	**Friction loss at Rotor_2**	**Friction loss at Rotor_3**	**Friction loss at Stator_2**
	[%]	**[%]**	**[%]**	**[%]**	**[%]**
s = 10 mm, t = 14 mm	33,99%	-20,65%	1,43%	-8,45%	-1,22%
s = 10 mm, t = 14 mm, t' = 7 mm	39,86%	-31,17%	1,10%	-8,89%	-5,78%
s = 10 mm, t = 14 mm, a = 7 mm	38,83%	-34,28%	0,86%	-8,42%	8,28%
s = 10 mm, t = 14 mm, at' = 7 mm	46,45%	-44,64%	0,93%	-8,81%	2,87%
s = 10 mm, t = 21 mm	51,72%	-30,76%	0,76%	-10,42%	5,89%

Table 7-11 : Comparisons between the variants with 't = 14 mm' for constant s = 10 mm and original case in percentage.

Therefore the variants with 's = 10 mm, t = 21 mm', 's = 10 mm, t = 14 mm, at' = 7 mm' and 's = 10 mm, t = 14 mm, t' = 7 mm' are selected as examples for flow characteristics comparisons with the original case.

7.5.1 Fluid flow characterising for selected variations

In this subsection the fluid flow of the original case and the selected variants at certain revolution speeds, i.e. 375 and 600 RPM, are discussed.

Fluid flow characteristics in axial direction by 375 RPM

Figure 7-66 to Figure 7-69 show the fluid velocities for the original case and selected variants at 375 RPM in the axial direction. It can be seen that for the original case the highest axial velocities are occurring due to the narrow radial distance between the inside radius of stator and oil container. An enlarged spacing between the inside radius of stator and oil container leads to an increased inlet volume and therefore to reduced axial velocities in this spacing.

s = 10 mm, t = 7 mm at 375 RPM

Figure 7- 6 6 : Flow velocities in axial direction – Original case 's = 10 mm, t = 7 mm' at 375 RPM.

s = 10 mm, t = 14 mm, t' = 7 mm at 375 RPM

Figure 7- 6 7 : Flow velocities in axial direction – Variation 's = 10 mm, t = 14 mm, t' = 7 mm' at 375 RPM.

s = 10 mm, t = 14 mm, at' = 7 mm at 375 RPM

Figure 7- 6 8 : Flow velocities in axial direction – Variation 's = 10 mm, **t** = 14 mm, **at'** = 7 mm' at 375 RPM.

s = 10 mm, t = 21 mm at 375 RPM

Figure 7- 6 9 : Flow velocities in axial direction – Variation 's = 10 mm, **t** = 21 mm' at 375 RPM.

Fluid flow characteristics in radial direction by 375 RPM

Figure 7-70 to Figure 7-73 show the fluid velocities for the original case and selected variants at 375 RPM in the radial direction. For this revolution speed, backflow occurs for the original case in the axial clearance. Although the other selected variants have the same width of the axial clearance, they do not show such a strong backflow along the Stator_2 wall due to the enlarged radial clearance.

Figure 7- 7 0 : Flow velocities in radial direction – Original case 's = 10 mm, t = 7 mm' at 375 RPM.

Figure 7- 7 1 : Flow velocities in radial direction – Variation 's = 10 mm, t = 14 mm, t' = 7 mm' at 375 RPM.

Figure 7- 7 2 : Flow velocities in radial direction – Variation '**s** = 10 mm, **t** = 14 mm, **t'** = 14 mm, **at'** = 7 mm' at 375 RPM.

Figure 7- 7 3 : Flow velocities in radial direction – Variation '**s** = 10 mm, **t** = 21 mm' at 375 RPM.

For 375 RPM, the comparisons of selected variants with the original case in percentage are presented in Table 7-12. All of the selected variants have higher volume flow rates by about 60%, because the original case has strong backflow at this revolution speed and the inlet volume of fluid in the axial clearance between the inside radius of oil container and stator for the selected variants is increased. While the axial clearance **s** stays constant, the volume flow rate is also increased with enlarged radial clearance **t**, therefore the variant with 's = 10 mm, t = 21 mm' shows the largest volume flow rate, by about 72% in comparison to the original case. The reason for it can be seen in Figure 7-73. The inlet volume is increased and a re-circulation zone in the radial clearance **t** is formed. The backflow of the re-circulation zone in the radial clearance mixes with the inflow from the oil container at the beginning of the rotor-stator distance and passes through the axial clearance. This re-circulation zone at the beginning of the axial gap is decreased in comparison to the variants with 't = 14 mm', therefore there is more inlet volume in the axial clearance. This is the reason why the variant with 's = 10 mm, t = 21 mm' shows lower friction loss at Rotor_2 than the other two variants. All of the variants show similar reduced friction losses at Rotor_3 as the original case. Due to the shortened length of stator the variant with 's = 10 mm, t = 14 mm, t' = 7 mm' shows an increased friction loss at Stator_2. The highest friction loss at Stator_2 shows the variant with 's = 10 mm, t = 14 mm, at' = 7 mm'. The reason for it is that a stronger backflow is formed at the outlet of the axial clearance than for the other variants (see Figure 7-72).

The selected variants compared with original case at 375 RPM					
Variants	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
	[%]	[%]	[%]	[%]	[%]
s = 10 mm, t = 14 mm, t' = 7 mm	63,65%	-10,70%	33,92%	-10,66%	6,34%
s = 10 mm, t = 14 mm, at' = 7 mm	62,36%	-15,18%	33,67%	-9,96%	22,26%
s = 10 mm, t = 21 mm	72,47%	-10,32%	30,72%	-8,13%	17,47%

Table 7-12: The comparisons of selected variants and original case at 375 RPM in percentage.

Fluid flow characteristics in axial direction by 600 RPM

For a revolution speed of 600 RPM the fluid flow velocities show similar results as at 375 RPM in the axial direction. They are depicted from Figure 7-74 to Figure 7-77. With enlarged spacing between inside radius of oil container and stator the selected variants show lower axial velocities at beginning of the axial clearance but they are higher at the outside radius of it, compared to the original case.

s = 10 mm, t = 7 mm at 600 RPM

Figure 7- 7 4 : Flow velocities in axial direction – Original case 's = 10 mm, t = 7 mm' at 600 RPM.

s = 10 mm, t = 14 mm, t' = 7 mm at 600 RPM

Figure 7- 7 5 : Flow velocities in axial direction – Variation 's = 10 mm, t = 14 mm, t' = 7 mm' at 600 RPM.

s = 10 mm, t = 14 mm, at' = 7 mm at 600 RPM

Figure 7- 7 6 : Flow velocities in axial direction – Variation '**s** = 10 mm, **t** = 14 mm, **at'** = 7 mm' at 600 RPM.

s = 10 mm, t = 21 mm at 600 RPM

Figure 7- 7 7 : Flow velocities in axial direction – Variation '**s** = 10 mm, **t** = 21 mm' at 600 RPM.

Fluid flow characteristics in radial direction by 600 RPM

Except for the original case in Figure 7-78 the backflow regions in the selected variants are not recognizable (see Figure 7-79 to Figure 7-81), but the radial velocities in the axial clearance are increased at this revolution speed.

Figure 7- 7 8 : Flow velocities in radial direction – Original case 's = 10 mm, t = 7 mm' at 600 RPM.

Figure 7- 7 9 : Flow velocities in radial direction – Variation 's = 10 mm, t = 14 mm, t' = 7 mm' at 600 RPM.

Figure 7- 8 0 : Flow velocities in radial direction – Variation '**s** = 10 mm, **t** = 14 mm, **at'** = 7 mm' at 600 RPM.

Figure 7- 8 1 : Flow velocities in radial direction – Variation '**s** = 10 mm, **t** = 21 mm' at 600 RPM.

Stronger backflow is established with increasing revolution speeds for the original case. This is the reason why the volume flow rates of all selected variants are higher than in the original case. The variant with 's = 10 mm, t = 21 mm' shows the highest increased volume flow rate by about 50%, because of the largest radial clearance. The backflow at the beginning of the radial clearance (i.e., from the re-circulation zone in the spacing between inside radius of rotor and oil container) mixes with the inflow from the oil container and passes through the axial gap. As a consequence the volume flow rate is increased (see Figure 7-81). The variant with 's = 10 mm, t = 14 mm, t' = 7 mm' has the same inlet volume as the variant with 's = 10 mm, t = 21 mm' due to the enlarged distance between the inside radius of oil container and stator, but smaller radial clearance. Therefore it has only on about 6% higher volume flow rate than the variant with 's = 10 mm, t = 21 mm'. All of the selected variants have reduced friction losses at rotor and stator walls, because the fluid velocities of them are lower than the original case at this revolution speed. Due to the same width of the radial clearance the variant with 's = 10 mm, t = 14 mm, t' = 7mm' and 's = 10 mm, t = 14 mm, at' = 7 mm' show similar friction losses at Rotor_3. With increased parameter t the friction loss at Rotor_3 is reduced, because of the decreased fluid velocity between the inside radius of rotor and oil container. The white marked region in Figure 7-81 shows that the oil in the container has lower radial velocities than the other variants at beginning of the Rotor_2. Therefore the variant with 's = 10 mm, t = 21 mm' has the lowest friction loss at Rotor_2 in comparison to the other selected variants. With enlarged radial clearance the axial velocity is reduced and the associated friction loss at Rotor_3 is lower. It can be seen in Table 7-13 that the friction losses at Rotor_3 of all variants are lower than the original case with 's = 10 mm, t= 7 mm' and the variants with 's = 10 mm, t = 14 mm' show higher friction losses at Rotor_3 than the variant with 's = 10 mm, t = 21 mm'. Due to the shortened length of stator the variant with 's = 10 mm, t = 14 mm, t' = 7 mm' shows the lowest friction loss at Stator_2.

The selected variants compared with original case at 600 RPM					
Variants	Volume flow rate	Total pressure difference	Friction loss at Rotor_2	Friction loss at Rotor_3	Friction loss at Stator_2
	[%]	[%]	[%]	[%]	[%]
s = 10 mm, t = 14 mm, t' = 7 mm	36,85%	-32,07%	-3,42%	-12,98%	-10,46%
s = 10 mm, t = 14 mm, at' = 7 mm	43,74%	-45,40%	-3,73%	-12,96%	-2,57%
s = 10 mm, t = 21 mm	49,38%	-34,31%	-6,12%	-16,04%	-0,63%

Table 7-13 : The comparisons of selected variants and original case at 600 RPM in percentage.

8 Empirical formulas for the model 'Glendoe'

One of the objectives of this diploma thesis work was to develop improved empirical correlations for the most important flow variables of the parameterised model 'Glendoe' with OpenFOAM. At first, the results of OpenFOAM will be compared with the existing empirical formulas, i.e. Dubbel, Geis, GleitLRB, Linnecken and Schlichting in this chapter. Then four considered functions for the volume flow rate, static and total pressure differences and the friction losses at rotor and stator walls are presented. With the help of Matlab suitable empirical formulas have been identified after comparison of the results given by OpenFOAM. Matlab is a software package http://www.mathworks.com/products/matlab/ for numerical calculations (e.g., matrix manipulation) and for the visualization of data (e.g., plotting of functions and data) in the technical-scientific area [36], [37].

8.1 Compared OpenFOAM results of 'Glendoe' with existing empirical formulas

Several empirical equations to predict the friction losses in a rotor-stator system were suggested by a number of researchers. Most of the published works on disc friction are concerned with plane thin discs rotating in enclosed cylindrical casings (e.g., for the validation case 'Enclosed rotating disc') and it differs to the present case studies, because:

- 'Glendoe' has the thin spacing between rotor and stator at one side, but the other side shows a wide distance between outside radius of rotor and oil container.

- The circumferential velocities of the validation case are obviously higher than within the application case 'Glendoe'.

In order to investigate how much deviation in the calculation of the friction coefficients between existing empirical formulas and OpenFOAM simulations exist, several comparisons are made, which are shown in Figure 8-1 and Table 8-2. The used equations to calculate the friction coefficient are presented in Table 4-2 and Table 8-1:

Description	Dimension	Equation
Friction loss	[Watt]	$P_{V,Rotor\_2} = \omega \; T_{V,Rotor\_2}$
Friction torque	[Nm]	$T_{V,Rotor\_2} = \xi_s \cdot c_{S,T}$
Friction coefficient	[-]	c_s

Friction factor	[Nm]	$c_{S,T} = \dfrac{\pi}{2} \cdot \rho \ U^2 \cdot R^3 = \dfrac{\pi}{2} \cdot \rho \ \omega \ \cdot R^5$

Table 8-1 : Used derivations for friction coefficient c_s.

Figure 8- 1 : Results comparisons between existing empirical formulas and OpenFOAM simulation.

Figure 8-1 shows the plot of friction coefficient over Reynolds number for the CFD-model 'Glendoe', in which can be seen that the results of the OpenFOAM simulations lie near to the empirical equations of 'Dubbel' and 'Geis' after 375 RPM. Table 8-1 shows a detailed results comparison in percentage between the empirical formulas (i.e., Linnecken, Geis and Dubbel) and OpenFOAM. The equation of 'Dubbel' gives similar friction coefficients, which have about 10% deviation in comparison to the CFD results after 375 RPM.

Drehzahl		Re	C_s values				Deviation to the existing empirical formulars in percentage		
[rad/sec]	[U/min]	[-]	OF	Linnecken	Geis	Dubbel	Linnecken	Geis	Dubbel
15,71	150	1,35E+05	1,43E-03	1,62E-03	1,30E-03	1,64E-03	-11,55%	10,45%	-12,62%
31,42	300	2,70E+05	8,31E-04	1,26E-03	1,09E-03	1,16E-03	-33,93%	-23,75%	-28,26%
39,27	375	3,37E+05	6,87E-04	1,17E-03	1,03E-03	1,04E-03	-41,33%	-33,35%	-33,72%
47,12	450	4,05E+05	8,86E-04	1,11E-03	9,84E-04	9,45E-04	-20,00%	-10,01%	-6,31%
52,36	500	4,50E+05	8,56E-04	1,07E-03	9,59E-04	8,97E-04	-20,28%	-10,75%	-4,60%
62,83	600	5,40E+05	8,07E-04	1,02E-03	9,16E-04	8,19E-04	-20,79%	-11,89%	-1,42%
78,54	750	6,74E+05	7,56E-04	9,59E-04	8,67E-04	7,33E-04	-21,12%	-12,73%	3,20%
94,25	900	8,09E+05	7,17E-04	9,14E-04	8,28E-04	6,69E-04	-21,49%	-13,36%	7,24%
109,96	1050	9,44E+05	7,26E-04	8,78E-04	7,97E-04	6,19E-04	-17,37%	-8,90%	17,19%
125,66	1200	1,08E+06	7,06E-04	8,49E-04	7,70E-04	5,79E-04	-16,86%	-8,34%	21,96%

Table 8-2 : The results comparisons between the existing empirical formulas and OpenFOAM simulation in percentage.

At low Reynolds numbers, i.e. from 150 to 375 RPM, the friction coefficients are significantly decreased with increasing Reynolds number and then show a suddenly increased friction coefficient at 450 RPM. The reason for this is that the used turbulence model (RNG k – epsilon)

is not applicable to low Reynolds numbers, because it is a high Reynolds number model using Standard Wall-Functions in the near-wall region (see Figure 2-3) and the first row of computational mesh cells is located in the laminar sublayer. Table 8-3, for information purposes contains the y^+ values, which are below 11 from 150 to 375 RPM. But the Standard Wall-Function approach is valid only, when $y^+ > 30$. At 450 RPM the average y^+ values are suddenly increased to 22.39, which means by about 250% in comparison to 375 RPM. Therefore the friction coefficients in Figure 8-1 show a jump between these two revolution speeds.

s = 10 mm, t = 7 mm			
[U/min]	Rotor_1	Rotor_2	Rotor_3
150	5,89	4,09	4,33
300	17,63	6,28	6,69
375	21,48	6,35	7,82
450	24,84	22,39	8,88
500	32,00	28,49	21,54
600	39,06	34,46	26,26
750	46,04	40,24	30,89
900	53,37	47,22	35,48
1050	74,38	65,04	48,82
1200	97,12	84,22	63,21

Table 8-3 : The average y^+ values for the rotor walls in OpenFOAM simulation.

8.2 Presentation of MATLAB results

The analysis of OpenFOAM results shown in the last chapter revealed that the parameters **s** and **t** have the greatest influence. There are seven geometries variations to investigate due to the different widths of the axial and radial clearances:

- Three variations with constant **s** and varied **t**:

    ➢ **s** = constant = 5 mm, **t** = varied = 7 mm, 14 mm, 21 mm, 28 mm

    ➢ **s** = constant = 10 mm, **t** = varied = 7 mm, 14 mm, 21 mm, 28 mm

    ➢ **s** = constant = 15 mm, **t** = varied = 7 mm, 14 mm, 21 mm, 28 mm

- Four variations with constant **t** and varied **s**:

    ➢ **t** = constant = 7 mm, **s** = varied = 5 mm, 10 mm, 15 mm

    ➢ **t** = constant = 14 mm, **s** = varied = 5 mm, 10 mm, 15 mm

    ➢ **t** = constant = 21 mm, **s** = varied = 5 mm, 10 mm, 15 mm

    ➢ **t** = constant = 28 mm, **s** = varied = 5 mm, 10 mm, 15 mm

Four empirical formulas have been defined and with the help of Matlab the constants of the equations are calculated to fit the CFD results. The functions describe the variables of fluid flow for example the mass flow rate y, as a function of $y = f(x_1, x_2)$ or $y = f(x_1, x_3)$. Reynolds number

is defined as x_1, x_2 is **t** – the radial clearance and x_3 indicates as **s** – the axial clearance. The variables are scaled with their respective maximum values, since then the datafit can be performed easier and more accurate. Furthermore the effect of different magnitudes can be thus minimized.

The functions in Table 8-4 are based on the parameters **Re** and **t** (i.e., the radial clearance is varied from 7 mm to 28 mm) while parameter **s** stays constant.

Functions for s=constant, t= varied	Own empirical formulas
Function 1	$y_1 = c + c_2 + c_3 \cdot \dfrac{t}{t_{max}}) \cdot e^{c_4 \cdot \frac{Re}{Re_{max}}}$
Function 2	$y_2 = c_1 + c_2 \cdot \dfrac{t}{t_{max}} + c_3 \cdot \dfrac{Re}{Re_{max}} + c_4 \cdot \dfrac{t}{t_{max}} \cdot \dfrac{Re}{Re_{max}} + c_5 \cdot (\dfrac{Re}{Re_{max}})^2 + c_6 \cdot \dfrac{t}{t_{max}} \cdot (\dfrac{Re}{Re_{max}})^2$
Function 3	$y_3 = c_1 + c_2 \cdot \dfrac{t}{t_{max}}) \cdot (c_3 \cdot \dfrac{Re}{Re_{max}} + c_4 \cdot (\dfrac{Re}{Re_{max}})^2)$
Function 4	$y_4 = c_1 + c_2 \cdot \dfrac{t}{t_{max}}) \cdot (\dfrac{Re}{Re_{max}})^{c_3}$

Table 8-4 : The used empirical formulas with parameter **t** in Matlab.

Table 8-5 shows the used functions which are based on the parameters **Re** and **s** (i.e., the axial clearance is varied from 5 mm to 15 mm) while parameter **t** stays constant.

Functions for s=varied, t= constant	Own empirical formulas
Function 1	$y_1 = c + c_2 + c_3 \cdot \dfrac{s}{s_{max}}) \cdot e^{c_4 \cdot \frac{Re}{Re_{max}}}$
Function 2	$y_2 = c_1 + c_2 \cdot \dfrac{s}{s_{max}} + c_3 \cdot \dfrac{Re}{Re_{max}} + c_4 \cdot \dfrac{s}{s_{max}} \cdot \dfrac{Re}{Re_{max}} + c_5 \cdot (\dfrac{Re}{Re_{max}})^2 + c_6 \cdot \dfrac{s}{s_{max}} \cdot (\dfrac{Re}{Re_{max}})^2$
Function 3	$y_3 = c_1 + c_2 \cdot \dfrac{s}{s_{max}}) \cdot (c_3 \cdot \dfrac{Re}{Re_{max}} + c_4 \cdot (\dfrac{Re}{Re_{max}})^2)$
Function 4	$y_4 = c_1 + c_2 \cdot \dfrac{s}{s_{max}}) \cdot (\dfrac{Re}{Re_{max}})^{c_3}$

Table 8-5 : The used empirical formulas with parameter **s** in Matlab.

8.3 Example with 's = constant = 10 mm, t = varied = 21 mm'

In Subsections 8.3 and 8.4 a comparison of two selected examples with variants 's = constant = 10 mm, t = varied = 21 mm' and 's = varied = 10 mm, t = constant = 21 mm' are presented for each flow variable (i.e., volume flow rate, pressure differences and the friction losses). The construction of these two subsections is:

1. The constants in the Matlab functions for the variants with 's = constant, t = varied' and 's = varied, t = constant' are shown at first in these subsections.

2. Then, the results comparisons between Matlab functions and OpenFOAM can be seen in the figures to investigate which functions are suitable to OpenFOAM results.

3. To comprehend the exact deviations between the results of Matlab functions and OpenFOAM a table with the essential percentages is shown after the figures.

4. Then, a rating table for all variants in 's = constant, t = varied' and 's = varied, t = constant' is made.

5. The comparative charts between the fitting functions and OpenFOAM results are shown at end.

The Matlab calculations are carried out with the functions in Table 8-4 for the variants with constant **s** and varied **t**. There are a few constants belonging to the defined functions, which are presented in Table 8-6 for the variant with '**s** = 10 mm, **t** = varied':

s=constant=10mm t=varied=21mm		c_1	c_2	c_3	c_4	c_5	c_6
Function 1	Mass flow rate	-0.272734	0.298929	0.172857	0.993552		
	Static pressure difference	-0.460498	0.335086	0.052157	1.310011		
	Total pressure difference	-0.180597	0.209580	-0.086287	1.808065		
	Friction loss at Rotor_2	-0.094920	0,057634	-0.005310	2.974593		
	Friction loss at Rotor_3	-0.097958	0.059852	-0.008012	2.922295		
	Friction loss at Stator_2	-0.138767	0.074645	0.012963	2.551199		
Function 2	Mass flow rate	0.060106	0.042766	0.299351	0.449012	0.115464	0.032061
	Static pressure difference	-0,054713	-0.013034	0.243003	0.043017	0.539175	0.242106

	Total pressure difference	-0.056712	0.038669	0.299167	-0.156520	0.888781	-0.499565
	Friction loss at Rotor_2	0.086503	-0.003963	-0.704075	0.068666	1.625493	-0.175281
	Friction loss at Rotor_3	0.076975	-0.000245	-0.662104	0.061223	1.577617	-0.216783
	Friction loss at Stator_2	0.034936	0.023700	-0.363882	-0.174175	1.114091	0.342079
Function 3	Mass flow rate	0,594298	0.669030	0.898226	-0.123173		
	Static pressure difference	0.725568	0.265849	0.352500	0.985508		
	Total pressure difference	0.892681	-0.482253	0.126572	1.146504		
	Friction loss at Rotor_2	0.981753	-0.106838	-0.374896	1.382503		
	Friction loss at Rotor_3	0.983755	-0.156627	-0.375777	1.368788		
	Friction loss at Stator_2	0.821775	0.192056	-0.302326	1.252588		
Function 4	Mass flow rate	0.459239	0.516985	0.881627			
	Static pressure difference	0.734834	0.269145	1.906672			
	Total pressure difference	1.128371	-0.609644	1.831932			
	Friction loss at Rotor_2	0.821775	-0.111562	2.838470			
	Friction loss at Rotor_3	1.011579	-0.160269	2.831898			
	Friction loss at Stator_2	0.798165	0.186879	2.621574			

Table 8-6 : The constants of the empirical formulas for 's = constant, t = varied'.

Mass flow rate

Figure 8-2, Figure 8-4, Figure 8-8, Figure 8-10, Figure 8-12 and Figure 8-14 show the results comparisons between empirical formulas and the OpenFOAM results for this variant, i.e. mass flow rate, static pressure difference, total pressure difference, friction loss at Rotor_2, friction loss at Rotor_3 and friction loss at Stator_2 over Reynolds numbers. The red curves represent the OpenFOAM results and the results of Function 1 to Function 4 are shown in grey, green, blue and yellow colours respectively.

Figure 8- 2 : Mass flow rate vs. Reynolds numbers - Matlab functions compared to the OpenFOAM results.

The plot for the mass flow rate in Figure 8-2 shows that Function 2 and Function 4 fit next to the OpenFOAM results and Function 3 has relative great deviation. In Table 8-7 it is evident that Function 2 and Function 4 present approximately the same mass flows rates and have the smallest deviation to the CFD results.

OpenFOAM Reults Mass flow rate [kg/s]	Re	Results presentation of each functions with Matlab.				Matlab results of each functions compared to OpenFOAM results in			
		fun1	fun2	fun3	fun4	fun1	fun2	fun3	fun4
16,6	1,35E+05	19,1	18,0	1,4	14,4	15,00%	8,43%	-91,54%	-13,48%
27,5	2,70E+05	24,9	27,0	5,6	26,5	-9,54%	-2,07%	-79,57%	-3,69%
31,6	3,37E+05	28,1	31,6	8,8	32,2	-11,29%	-0,19%	-72,28%	1,96%
37,7	4,05E+05	31,5	36,3	12,7	37,9	-16,54%	-3,55%	-66,40%	0,62%
41,0	4,50E+05	33,8	39,6	15,6	41,6	-17,54%	-3,50%	-61,89%	1,46%
46,9	5,40E+05	38,9	46,2	22,5	48,9	-17,15%	-1,54%	-52,01%	4,19%
54,4	6,74E+05	47,2	56,4	35,1	59,4	-13,21%	3,73%	-35,49%	9,30%
61,7	8,09E+05	56,7	67,1	50,5	69,8	-8,13%	8,82%	-18,09%	13,15%
79,8	9,44E+05	67,4	78,3	68,8	80,0	-15,53%	-1,86%	-13,79%	0,21%
90,0	1,08E+06	79,7	90,0	90,0	90,0	-11,52%	0,00%	0,00%	0,00%

Table 8-7 : The comparisons of mass flow rates between the empirical formulas with Matlab and OpenFOAM variations.

Mass flow rate	s = 5 mm, t = varied				s = 10 mm, t = varied				s = 15 mm, t = varied			
	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm
fun1				✓			✓					✓
fun2	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
fun3												
fun4	✓	✓	✓	✓	✓	✓	✓	✓				

Table 8-8 : Assessment of own empirical formulas about volume flow rate for all variants with 's = constant, t = varied'.

From Table 8-8 can be seen that for the variant with 's = 15 mm, t = varied' only Function 2 is suitable to fit the OpenFOAM results for the mass flow rate. Therefore it was decided that Function 2 will be the best fit for this variable.

Figure 8-3 presents a 3-D graph for the mass flow rate comparison between OpenFOAM (i.e., data) and Function 2 in Matlab (i.e., fit). It shows that Function 2 can fit the OpenFOAM results

very accurately. The fit and data curves lay closely side by side and the minimal inaccuracy between them is displayed in the relative error plot at the right hand side of Figure 8-3. The Matlab solvers have error tolerances in the form of a scalar relative error tolerance and a vector of relative error tolerances [38]. The relative error is the magnitude of the actual error and it can be obtained by taking the difference between the simulated value (i.e., OpenFOAM) and reference value (i.e., Matlab functions), e.g., relative error = (fit-data)/data [38]. The solutions of the default relative error tolerance are usually interpreted graphically in Matlab [38].

The maximal and minimal relative errors with Function 2 are about 3% and 6% for the mass flow rate.

Figure 8- 3 : Comparison of mass flow rate between OpenFOAM (data) and Function 2 with Matlab (fit).

Static pressure difference

For the static pressure difference Function 2, Function 3 and Function 4 (see Figure 8-4) show good results with low deviation after 375 RPM in comparison to OpenFOAM. The detailed percentages are presented in Table 8-9.

Figure 8- 4 : Static pressure difference vs. Reynolds numbers - Matlab functions compared to the OpenFOAM results.

OpenFOAM Reults Static pressure difference [Pa]	Re	Results presentation of each functions with Matlab.				Matlab results of each functions compared to OpenFOAM results in			
		fun1	fun2	fun3	fun4	fun1	fun2	fun3	fun4
424	1,35E+05	-896	-918	710	862	-311,33%	-316,54%	67,49%	103,36%
2009	2,70E+05	2668	2409	2841	3233	32,82%	19,91%	41,39%	60,92%
3765	3,37E+05	4666	4468	4425	4933	23,93%	18,68%	17,54%	31,03%
6075	4,05E+05	6867	6835	6391	7004	13,03%	12,51%	5,21%	15,29%
7880	4,50E+05	8426	8555	7890	8562	6,93%	8,56%	0,13%	8,66%
12208	5,40E+05	11812	12360	11362	12122	-3,24%	1,25%	-6,93%	-0,71%
19890	6,74E+05	17591	18931	17701	18497	-11,56%	-4,82%	-11,01%	-7,00%
29414	8,09E+05	24444	26645	25502	26199	-16,90%	-9,41%	-13,30%	-10,93%
34991	9,44E+05	32517	35459	34723	35162	-7,07%	1,34%	-0,77%	0,49%
45449	1,08E+06	42103	45449	45449	45449	-7,36%	0,00%	0,00%	0,00%

Table 8-9: The comparisons of static pressure differences between the empirical formulas with Matlab and OpenFOAM variations.

Static pressure difference	s = 5 mm, t = varied				s = 10 mm, t = varied				s = 15 mm, t = varied			
	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm
fun1												
fun2	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
fun3	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
fun4	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓

Table 8-10 : Assessment of own empirical formulas about static pressure difference for all variants with 's = constant, t = varied'.

Table 8-10 shows that Function 2 to Function 4 are suitable not only for this example with 's = 10 mm, t = varied', moreover for all variations with 's = constant and t = varied'. From the following figures (i.e. from Figure 8-5 to Figure 8-7) it can be seen how exact the Matlab functions fit the OpenFOAM results.

Figure 8- 5 : Comparison of static pressure difference between OpenFOAM (data) and Function 2 with Matlab (fit).

Figure 8- 6 : Comparison of static pressure difference between OpenFOAM (data) and Function 3 with Matlab (fit).

Figure 8- 7 : Comparison of static pressure difference between OpenFOAM (data) and
Function 4 with Matlab (fit).

Function 3 and Function 4 show a similar trend and the minimal and maximal relative errors are about 5% and 6%. Although the Function 2 has a greater deviation at low Reynolds numbers, the relative errors of it are less than the other two functions.

Total pressure difference

Similar curve trends are given for the total pressure differences vs. Reynolds number, which can be seen in Figure 8-8. The Function 2 and Function 4 have approximately the same total pressure differences and they lie next to the OpenFOAM results. Function 3 and Function 1 gave higher deviations than Function 2 and Function 4.

Figure 8- 8 : Total pressure difference vs. Reynolds numbers - Matlab functions compared to the OpenFOAM results.

The detailed analysis is proven in Table 8-11 that Function 2 and Function 4 gave the best fit to the CFD simulations. There is a minimal difference in the results between these two functions.

OpenFOAM Reults		Results presentation of each functions with Matlab.				Matlab results of each functions compared to OpenFOAM results in			
Total pressure difference [Pa]	Re	fun1	fun2	fun3	fun4	fun1	fun2	fun3	fun4
2534	1,35E+05	119	542	1858	2636	-95,29%	-78,61%	-26,66%	4,02%
8707	2,70E+05	5597	8877	7434	9384	-35,72%	1,95%	-14,62%	7,78%
13893	3,37E+05	8811	14075	11581	14085	-36,58%	1,31%	-16,64%	1,38%
19186	4,05E+05	12463	20071	16726	19724	-35,04%	4,61%	-12,82%	2,80%
23725	4,50E+05	15119	24438	20650	23923	-36,27%	3,01%	-12,96%	0,84%
34526	5,40E+05	21071	34126	29736	33410	-38,97%	-1,16%	-13,87%	-3,23%
52900	6,74E+05	31772	50904	46325	50145	-39,94%	-3,77%	-12,43%	-5,21%
76229	8,09E+05	45277	70658	66740	70061	-40,60%	-7,31%	-12,45%	-8,09%
92871	9,44E+05	62205	93271	90873	92952	-33,02%	0,43%	-2,15%	0,09%
118943	1,08E+06	83603	118943	118943	118943	-29,71%	0,00%	0,00%	0,00%

Table 8-11 : The comparisons of total pressure differences between the empirical formulas with Matlab and OpenFOAM variations.

For the total pressure difference between inlet and outlet of the axial clearance only one general empirical formula is possible due to the fact that Function 2 is not suitable for the variants with 's= 15 mm, t = varied', which is shown in Table 8-12.

Total pressure difference	s = 5 mm, t = varied				s = 10 mm, t = varied				s = 15 mm, t = varied			
	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm
fun1	✓				✓				✓			
fun2	✓	✓	✓	✓	✓	✓	✓	✓				
fun3												
fun4	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓

Table 8-12 : Assessment of own empirical formulas about total pressure difference for all variants with 's = constant, t = varied'.

Calculated total pressure differences with Function 4 are in comparison with the OpenFOAM results (see Figure 8-9).

Figure 8- 9 : Comparison of total pressure difference between OpenFOAM (data) and
Function 4 with Matlab (fit).

Friction loss at Rotor 2

Figure 8-10 indicates that the Function 1, Function 2 and Function 4 are the best choices for
this variant. But the detailed percentages of Function 2 in Table 8-13 shows that the friction loss
at Rotor_2 presented higher deviations at low Reynolds numbers than Function 1 and Function
4.

Figure 8- 1 0 : The friction loss at Rotor_2 vs. Reynolds numbers - Matlab functions
compared to the OpenFOAM results.

OpenFOAM Reults / Friction loss at Rotor_2 [watt]	Re	Results presentation of each functions with Matlab.				Matlab results of each functions compared to OpenFOAM results in			
		fun1	fun2	fun3	fun4	fun1	fun2	fun3	fun4
365	1,35E+05	-1510	2415	1379	241	-513,72%	561,63%	277,92%	-33,90%
2042	2,70E+05	1584	1314	5518	1726	-22,44%	-35,67%	170,21%	-15,49%
3638	3,37E+05	3603	2422	8596	3237	-0,96%	-33,44%	136,28%	-11,01%
5855	4,05E+05	6071	4668	12415	5455	3,69%	-20,27%	112,04%	-6,84%
7739	4,50E+05	7978	6777	15327	7356	3,09%	-12,43%	98,05%	-4,95%
12574	5,40E+05	12580	12479	22070	12343	0,05%	-0,76%	75,52%	-1,84%
23110	6,74E+05	21936	24639	34383	23156	-5,08%	6,61%	48,78%	0,20%
38125	8,09E+05	35589	41328	49536	38879	-6,65%	8,40%	29,93%	1,98%
60735	9,44E+05	55392	62474	67448	60250	-8,80%	2,86%	11,05%	-0,80%
88282	1,08E+06	84368	88282	88282	88282	-4,43%	0,00%	0,00%	0,00%

Table 8-13: The comparisons of friction losses at Rotor_2 between the empirical formulas with Matlab and OpenFOAM variations.

Till now, the Function 1 and Function 4 gave the best results for the example ‘**s** = 10 mm, **t** = 21 mm’. However, Function 1 can not fit the OpenFOAM results for all variations with ‘**s** = constant, **t** = varied’, because it has a relatively high deviation at the largest radial clearance **t** = 28 mm. Therefore the general preferred empirical formula for the calculation of the friction loss at Rotor_2 is Function 4.

Friction loss at Rotor_2	s = 5 mm, t = varied				s = 10 mm, t = varied				s = 15 mm, t = varied			
	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm
fun1	✓	✓	✓		✓	✓	✓		✓	✓	✓	
fun2												
fun3												
fun4	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓

Table 8-14 : Assessment of own empirical formulas about friction loss at Rotor_2 for all variants with ‘**s** = constant, **t** = varied’.

The comparison of the friction loss at Rotor_2 between Function 4 with Matlab and OpenFOAM is shown in Figure 8-11, which indicates that the maximal deviation between fit and data is about 0.5%. The minimal deviation is about 1.5%, which is only about 10% lower than OpenFOAM results.

Figure 8- 1 1 : Comparison of friction loss at Rotor_2 between OpenFOAM (data) and Function 2 with Matlab (fit).

Friction loss at Rotor_3

The friction loss comparisons at Rotor_3 between Matlab functions and OpenFOAM are shown in the following diagram (i.e., Figure 8-12), in which can be seen that the Function 2 and Function 4 fit the OpenFOAM results best. The detailed deviations in percentage are presented in Table 8-15 and they show that the Function 4 is the best choice. Function 2 yields to greater deviations than Function 4 (see Figure 8-15 and Figure 8-16).

Figure 8- 1 2 : The friction loss at Rotor_3 vs. Reynolds numbers - Matlab functions compared to the OpenFOAM results.

OpenFOAM Reults		Results presentation of each functions with Matlab.				Matlab results of each functions compared to OpenFOAM results in			
Friction loss at Rotor_3 [watt]	Re	fun1	fun2	fun3	fun4	fun1	fun2	fun3	fun4
300	1,35E+05	-1352	1657	1037	184	-550,52%	452,48%	245,50%	-38,74%
1466	2,70E+05	918	847	4146	1309	-37,40%	-42,20%	182,81%	-10,74%
2464	3,37E+05	2392	1689	6459	2451	-2,93%	-31,45%	162,14%	-0,51%
4095	4,05E+05	4188	3387	9329	4125	2,26%	-17,29%	127,81%	0,74%
5453	4,50E+05	5571	4978	11517	5560	2,17%	-8,71%	111,20%	1,95%
9389	5,40E+05	8899	9277	16584	9317	-5,21%	-1,19%	76,63%	-0,77%
17457	6,74E+05	15629	18436	25836	17454	-10,47%	5,61%	48,00%	-0,02%
29027	8,09E+05	25385	31001	37222	29270	-12,55%	6,80%	28,23%	0,84%
45504	9,44E+05	39443	46917	50682	45313	-13,32%	3,10%	11,38%	-0,42%
66337	1,08E+06	59879	66337	66337	66337	-9,73%	0,00%	0,00%	0,00%

Table 8-15 : The comparisons of friction loss at Rotor_3 between the empirical formulas with Matlab and OpenFOAM variations.

Friction loss at Rotor_3	s = 5 mm, t = varied				s = 10 mm, t = varied				s = 15 mm, t = varied			
	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm	t = 7 mm	t = 14 mm	t = 21 mm	t = 28 mm
fun1	✓				✓				✓			
fun2	✓	✓			✓	✓			✓	✓		
fun3												
fun4	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓

Table 8-16 : Assessment of own empirical formulas about friction loss at Rotor_3 for all variants with 's = constant, t = varied'.

A 3-D-plot of the friction losses at Rotor_3 is shown in Figure 8-13. At the right hand side of this figure the relative error between the fit and data files can be seen.

Figure 8- 1 3 : Comparison of friction loss at Rotor_3 between OpenFOAM (data) and Function 4 with Matlab (fit).

Therefore, there are two general empirical formulas for the friction loss at Stator_2, i.e. Function 2 and Function 4 (see Figure 8-18).

The fit and data comparison in Matlab shows also that Function 2 has a greater deviation than Function 4 at low Reynolds numbers. The minimal error of Function 2 is higher than Function 4 but the maximum error is lower (see Figure 8-15 and Figure 8-16).

Figure 8- 1 5 : Comparison of friction loss at Stator_2 between OpenFOAM (data) and Function 2 with Matlab (fit).

Figure 8- 1 6 : Comparison of friction loss at Stator_2 between OpenFOAM (data) and Function 4 with Matlab (fit).

8.4 Example with 's = varied = 15 mm, t = constant = 14 mm'

The example 't = 14 mm, **s** = varied = 15 mm' belongs to the variations with 't = constant, **s** = varied'. The results comparisons about mass flow rate, static pressure difference, total pressure difference and the friction losses at rotor and stator walls are made between Matlab functions and OpenFOAM results in excel diagrams, at first. After the optical comparisons, detailed analysis of the relative error in percentages is made to show how much deviations are between the Matlab functions and the OpenFOAM results. Based on above analysis suitable empirical functions are evaluated for this example. The same process will be made with all variants for the variations 't = constant, **s** = varied', from which the general empirical formulas are determined. Finally, 3-D surface plots for the numerical and analytical results are presented.

The Matlab functions for constant **t** and varied **s** are presented in Table 8-19. The constants for these functions are shown in the following table:

t=constant=14mm s=varied=15mm		c_1	c_2	c_3	c_4	c_5	c_6
Function 1	Mass flow rate	-0.603950	0.757298	-0.043660	0.721845		
	Static pressure difference	-0.415845	0.380902	-0.077522	1.382304		
	Total pressure difference	-0.240177	0.254410	-0.081970	1.647364		
	Friction loss at Rotor_2	-0.097473	0.056804	-0.001720	2.974431		
	Friction loss at Rotor_3	-0.110122	0.062484	-0.000524	2.888160		
	Friction loss at Stator_2	-0.07506	0.073128	-0.03869	2.833923		
Function 2	Mass flow rate	-0.088040	0.298923	1.272035	-1.063502	-0.191471	0.604310
	Static pressure difference	-0.092923	0.036483	0.374082	-0.128805	0.884951	-0.328142
	Total pressure difference	-0.013293	-0.027320	0.170910	0.027508	0.964525	-0.511851
	Friction loss at Rotor_2	0.091033	-0.007123	-0.709110	0.045169	1.604315	-0.074866
	Friction loss at Rotor_3	0.083125	-0.002895	-0.675478	0.015965	1.577300	-0.023569
	Friction loss at Stator_2	0.079574	-0.06011	-0.65822	0.443659	1.746961	-1.07671
Function 3	Mass flow rate	-1.096612	-0.165033	0.994832	-0.118828		
	Static pressure difference	0.997082	-0.356934	0.036359	1.148894		
	Total pressure difference	0.903603	-0.410366	0.113233	1.136008		
	Friction loss at Rotor_2	0.976669	-0.034890	-0.369409	1.359907		
	Friction loss at Rotor_3	0.973663	-0.009754	-0.369432	1.363711		
	Friction loss at Stator_2	1.004172	-0.59274	-0.35045	1.497176		
Function 4	Volume flow rate	0.954548	-0.141914	0.891195			
	Static pressure difference	1.171627	-0.419370	1.907363			
	Total pressure difference	1.121624	-0.509224	1.847119			
	Friction loss at Rotor_2	1.004661	-0.036058	2.841722			
	Friction loss at Rotor_3	1.002254	-0.010530	2.813074			
	Friction loss at Stator_2	1.178033	-0.6957	2.604357			

Table 8-19 : The constants of the empirical formulas for '**t** = constant, **s** = varied'.

Mass flow rate

Figure 8-17 shows that the Function 2 is well suited to fit the numerical results for all mass flow rates.

Figure 8- 1 7 : Mass flow rate vs. Reynolds numbers - Matlab functions compared to the OpenFOAM results.

The detailed percentage deviations between the results of the analytical functions and CFD results are shown in Table 8-20, in which can be seen that the Function 1 and Function 3 have the larger deviations to OpenFOAM and Function 2 shows the best results.

OpenFOAM Reults Mass flow rate [kg/s]	Re	Results presentation of each functions with Matlab.				Matlab results of each functions compared to OpenFOAM results in			
		fun1	fun2	fun3	fun4	fun1	fun2	fun3	fun4
16,7	1,35E+05	10,6	17,6	0,9	9,4	-36,26%	5,28%	-94,38%	-43,58%
21,7	2,70E+05	15,1	20,8	3,8	17,5	-30,50%	-3,85%	-82,68%	-19,46%
22,7	3,37E+05	17,4	22,8	5,8	21,3	-23,15%	0,68%	-74,19%	-6,13%
25,6	4,05E+05	19,9	25,1	8,4	25,1	-22,20%	-2,12%	-67,00%	-2,09%
27,1	4,50E+05	21,6	26,7	10,4	27,5	-20,08%	-1,48%	-61,47%	1,72%
30,4	5,40E+05	25,2	30,2	15,0	32,4	-17,09%	-0,73%	-50,63%	6,47%
34,8	6,74E+05	31,0	36,2	23,4	39,5	-10,90%	4,16%	-32,72%	13,48%
40,1	8,09E+05	37,3	43,2	33,7	46,4	-6,89%	7,78%	-15,94%	15,81%
53,8	9,44E+05	44,3	51,1	45,9	53,3	-17,63%	-4,88%	-14,65%	-0,91%
60,1	1,08E+06	51,9	60,1	60,1	60,1	-13,51%	0,00%	0,00%	0,00%

Table 8-20: Assessment of own empirical formulas about mass flow rate for all variants with 's = 15 mm, t = 21 mm'.

In order to investigate if the Function 2 and Function 4 are suitable for all variants, the following analysis is made:

Volume flow rate	t = 7 mm, s = varied			t = 14 mm, s = varied			t = 21 mm, s = varied			t = 28 mm, s = varied		
	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm
fun1												
fun2	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
fun3												
fun4		✓			✓			✓			✓	

Table 8-21: Assessment of own empirical formulas about mass flow rate for all variants with 's = varied, t = constant'.

From Table 8-21 can be seen that the Function 4 shows the best result in comparison to OpenFOAM only for the variant with '**s** = 10 mm' and Function 2 is suitable for all variants. Figure 8-18 shows the fit and data graph in Matlab between the Function 4 and OpenFOAM.

Figure 8- 1 8 : Comparison of mass flow rate between OpenFOAM (data) and Function 4 with Matlab (fit).

<u>Static pressure difference</u>

The trend curves (see Figure 8-19) of Function 2 and Function 4 are similar to the example with '**s** = 10 mm, **t** = 21 mm' for the static pressure differences. Table 8-22 shows the percentage deviation between the results of OpenFOAM and Matlab functions for the static pressure difference.

Figure 8- 1 9 : Static pressure difference vs. Reynolds numbers - Matlab functions compared to the OpenFOAM results.

OpenFOAM Reults		Results presentation of each functions with Matlab.				Matlab results of each functions compared to OpenFOAM results in			
Static pressure difference [Pa]	Re	fun1	fun2	fun3	fun4	fun1	fun2	fun3	fun4
359	1,35E+05	-1874	-777	530	643	-622,09%	-316,48%	47,67%	79,04%
1389	2,70E+05	433	1805	2121	2411	-68,80%	29,98%	52,66%	73,58%
2802	3,37E+05	1735	3381	3303	3680	-38,07%	20,67%	17,90%	31,33%
4502	4,05E+05	3176	5180	4771	5225	-29,45%	15,06%	5,98%	16,06%
5834	4,50E+05	4201	6481	5890	6388	-27,99%	11,08%	0,96%	9,49%
9166	5,40E+05	6436	9346	8482	9045	-29,78%	1,96%	-7,46%	-1,33%
15487	6,74E+05	10280	14264	13214	13804	-33,62%	-7,90%	-14,68%	-10,87%
22458	8,09E+05	14880	20008	19037	19554	-33,74%	-10,91%	-15,23%	-12,93%
25989	9,44E+05	20348	26544	25921	26246	-21,71%	2,13%	-0,26%	0,99%
33928	1,08E+06	26900	33928	33928	33928	-20,72%	0,00%	0,00%	0,00%

Table 8-22: Assessment of own empirical formulas about static pressure difference for all variants with 's = 15 mm, t = 21 mm'.

From Table 8-23 it can be concluded that Function 2, Function 3 and Function 4 can be used as empirical formulas to determine the static pressure differences.

Static pressure difference	t = 7 mm, s = varied			t = 14 mm, s = varied			t = 21 mm, s = varied			t = 28 mm, s = varied		
	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm
fun1							✓			✓		
fun2	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
fun3	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
fun4	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓

Table 8-23: Assessment of own empirical formulas about static pressure difference for all variants with 's = varied, t = constant'.

The following figures are the fit and data graphs between Matlab functions (i.e., from Function 2 to Function 4) and OpenFOAM results. It can be seen that Function 2 and 4 show higher maximal errors than the CFD data by about 8%. Function 3 shows the largest minimal relative error by about 5%.

Figure 8- 2 0 : Comparison of static pressure difference between OpenFOAM (data) and
Function 2 with Matlab (fit).

Figure 8- 2 1 : Comparison of static pressure difference between OpenFOAM (data) and
Function 3 with Matlab (fit).

Figure 8-2 2 : Comparison of static pressure difference between OpenFOAM (data) and Function 4 with Matlab (fit).

Total pressure difference

Function 2 and Function 4 for the total pressure differences lie in proximity of the OpenFOAM results, which can be also seen in Table 8-24. The Function 3 shows greater deviations than Function 2 and 4 and Function 1 has lower total pressure differences between the inlet and outlet of the axial gap by about 50% on average than the CFD results.

Figure 8-2 3 : Total pressure difference vs. Reynolds numbers - Matlab functions compared to the OpenFOAM results.

OpenFOAM Reults		Results presentation of each functions with Matlab.				Matlab results of each functions compared to OpenFOAM results in			
Total pressure difference [Pa]	Re	fun1	fun2	fun3	fun4	fun1	fun2	fun3	fun4
2183	1,35E+05	-4579	-2315	2528	3474	-309,77%	-206,07%	15,79%	59,12%
9110	2,70E+05	3258	9880	10111	12498	-64,24%	8,45%	10,99%	37,19%
13292	3,37E+05	7789	17324	15751	18821	-41,40%	30,34%	18,50%	41,60%
24596	4,05E+05	12887	25824	22749	26429	-47,61%	4,99%	-7,51%	7,45%
30723	4,50E+05	16563	31971	28085	32108	-46,09%	4,06%	-8,59%	4,51%
45301	5,40E+05	24718	45516	40443	44964	-45,44%	0,48%	-10,72%	-0,74%
72541	6,74E+05	39134	68770	63005	67714	-46,05%	-5,20%	-13,15%	-6,65%
104133	8,09E+05	56967	95932	90772	94871	-45,29%	-7,88%	-12,83%	-8,89%
126272	9,44E+05	78877	126842	123595	126164	-37,53%	0,45%	-2,12%	-0,09%
161772	1,08E+06	106017	161772	161772	161772	-34,46%	0,00%	0,00%	0,00%

Table 8-24: Assessment of own empirical formulas about total pressure difference for all variants with '**s** = 15 mm, **t** = 21 mm'.

For all variants in this variation Function 2 and Function 4 are feasible, which are defined as general empirical formulas for total pressure difference.

Total pressure difference	t = 7 mm, s = varied			t = 14 mm, s = varied			t = 21 mm, s = varied			t = 28 mm, s = varied		
	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm
fun1												
fun2	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
fun3												
fun4	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓

Table 8-25: Assessment of own empirical formulas about total pressure difference for all variants with '**s** = varied, **t** = constant'.

Figure 8-24 and Figure 8-25 show the comparisons of total pressure differences between Matlab functions (i.e., Function 2 and Function 4) and the CFD results.

Figure 8- 2 4 : Comparison of total pressure difference between OpenFOAM (data) and Function 2 with Matlab (fit).

Figure 8- 2 5 : Comparison of total pressure difference between OpenFOAM (data) and Function 4 with Matlab (fit).

Friction loss at Rotor_2

With Function 3 calculated friction losses at Rotor_2 present the highest deviation to the CFD results. For this example, Function 1 and Function 4 lie next to the OpenFOAM results. The differences between Matlab functions and OpenFOAM are shown in Table 8-26.

Figure 8- 2 6 : Friction loss at Rotor_2 vs. Reynolds numbers - Matlab functions compared to the OpenFOAM results.

OpenFOAM Reults		Results presentation of each functions with Matlab.				Matlab results of each functions compared to OpenFOAM results in			
Friction loss at Rotor_2 [Watt]	Re	fun1	fun2	fun3	fun4	fun1	fun2	fun3	fun4
368	1,35E+05	-1610	2394	1431	249	-537,60%	550,53%	288,89%	-32,44%
1712	2,70E+05	1685	1304	5724	1782	-1,58%	-23,84%	234,37%	4,10%
3715	3,37E+05	3836	2475	8918	3346	3,25%	-33,38%	140,06%	-9,93%
6018	4,05E+05	6464	4825	12880	5641	7,42%	-19,83%	114,03%	-6,26%
7987	4,50E+05	8495	7023	15901	7610	6,36%	-12,07%	99,09%	-4,72%
13022	5,40E+05	13396	12956	22898	12777	2,87%	-0,50%	75,84%	-1,88%
23849	6,74E+05	23361	25588	35672	23987	-2,05%	7,29%	49,58%	0,58%
39665	8,09E+05	37902	42907	51393	40299	-4,44%	8,17%	29,57%	1,60%
63021	9,44E+05	58992	64837	69977	62482	-6,39%	2,88%	11,04%	-0,86%
91592	1,08E+06	89851	91592	91592	91592	-1,90%	0,00%	0,00%	0,00%

Table 8-26: Assessment of own empirical formulas about friction loss at Rotor_2 for all variants with 's = 15 mm, t = 21 mm'.

The same comparisons are made for all variants with 's = varied, t = constant' and the results are shown in Table 8-26, which yield that Function 2 is only suitable for certain variants, but Function 4 presents the best results for all variants.

Friction loss at Rotor_2	t = 7 mm, s = varied			t = 14 mm, s = varied			t = 21 mm, s = varied			t = 28 mm, s = varied		
	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm
fun1	✓	✓		✓	✓							
fun2												
fun3												
fun4	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓

Table 8-27: Assessment of own empirical formulas about friction loss at Rotor_2 for all variants with 's = varied, t = constant'.

Function 4 is determined as the general empirical formula for the friction loss at Rotor_2 and its difference to OpenFOAM results are presented in Figure 8-27.

Figure 8- 2 7 : Comparison of friction loss at Rotor_2 between OpenFOAM (data) and Function 4 with Matlab (fit).

Friction loss at Rotor_3

The friction loss comparisons at Rotor_3 between Matlab functions and OpenFOAM is shown in the following diagram (see Figure 8-28), in which can be seen that Function 1, Function 2 and Function 4 can fit the OpenFOAM results better. The detailed deviations in kind of percentage in Table 8-28 show that Function 4 has the smallest differences to the OpenFOAM results.

Figure 8- 2 8 : Friction loss at Rotor_3 vs. Reynolds numbers - Matlab functions compared to the OpenFOAM results.

OpenFOAM Reults Friction loss at Rotor_3 [Watt]	Re	Results presentation of each functions with Matlab.				Matlab results of each functions compared to OpenFOAM results in			
		fun1	fun2	fun3	fun4	fun1	fun2	fun3	fun4
304	1,35E+05	-1448	1546	1066	197	-576,47%	408,45%	250,80%	-35,32%
1441	2,70E+05	1190	873	4266	1382	-17,44%	-39,44%	196,03%	-4,10%
2403	3,37E+05	2898	1801	6645	2578	20,60%	-25,03%	176,55%	7,28%
3999	4,05E+05	4975	3601	9598	4323	24,40%	-9,96%	140,01%	8,11%
5901	4,50E+05	6572	5266	11849	5815	11,37%	-10,77%	100,80%	-1,46%
9795	5,40E+05	10406	9729	17063	9712	6,24%	-0,67%	74,20%	-0,85%
18230	6,74E+05	18129	19176	26582	18118	-0,55%	5,19%	45,81%	-0,62%
30372	8,09E+05	29280	32081	38297	30279	-3,60%	5,63%	26,09%	-0,31%
46918	9,44E+05	45278	48387	52145	46740	-3,49%	3,13%	11,14%	-0,38%
68252	1,08E+06	68436	68252	68252	68252	0,27%	0,00%	0,00%	0,00%

Table 8-28: Assessment of own empirical formulas about friction loss at Rotor_3 for all variants with 's = 15 mm, t = 21 mm'.

The same trend can be seen for the other variants with 's = varied, t = constant', therefore Function 4 is the empirical formula of friction loss at Rotor_3 for them (see Table 8-28 and Table 8-29).

Friction loss at Rotor_3	t = 7 mm, s = varied			t = 14 mm, s = varied			t = 21 mm, s = varied			t = 28 mm, s = varied		
	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm
fun1												
fun2												
fun3												
fun4	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓

Table 8-29: Assessment of own empirical formulas about friction loss at Rotor_3 for all variants with 's = varied, t = constant'.

Figure 8-29 shows the differences of friction loss at Rotor_3 between Function 4 (fit) and OpenFOAM results (data). The relative errors of them are also presented at the right side of this figure.

Figure 8- 2 9 : Comparison of friction loss at Rotor_3 between OpenFOAM (data) and Function 4 with Matlab (fit).

_Friction loss at Stator_2_

The plots for the friction losses at Stator_2 show that Function 2 and 4 present better results than Function 1 and 3. They gave only minimal differences to OpenFOAM. This can also be seen in Table 8-30, which shows the relative errors for all Matlab functions in comparison to the OpenFOAM results.

Figure 8-3 0 : Friction loss at Stator_2 vs. Reynolds numbers - Matlab functions compared to the OpenFOAM results.

OpenFOAM Reults		Results presentation of each functions with Matlab.				Matlab results of each functions compared to OpenFOAM results in			
Friction loss at Stator_2 [Watt]	Re	fun1	fun2	fun3	fun4	fun1	fun2	fun3	fun4
57	1,35E+05	-334	84	201	57	-686,52%	48,05%	252,77%	0,39%
278	2,70E+05	-66	209	804	348	-123,68%	-24,85%	189,32%	25,18%
570	3,37E+05	107	481	1253	620	-81,19%	-15,55%	119,83%	8,74%
939	4,05E+05	317	901	1810	1000	-66,26%	-4,08%	92,73%	6,54%
1240	4,50E+05	478	1257	2234	1316	-61,48%	1,40%	80,18%	6,15%
1992	5,40E+05	862	2160	3217	2116	-56,72%	8,43%	61,51%	6,23%
3931	6,74E+05	1632	3971	5012	3769	-58,47%	1,01%	27,50%	-4,11%
6620	8,09E+05	2737	6360	7221	6064	-58,66%	-3,93%	9,08%	-8,40%
8864	9,44E+05	4311	9317	9832	9064	-51,37%	5,11%	10,92%	2,26%
12869	1,08E+06	6574	12869	12869	12869	-48,92%	0,00%	0,00%	0,00%

Table 8-30: Assessment of own empirical formulas about friction loss at Stator_2 for all variants with '**s** = 15 mm, **t** = 21 mm'.

After the assessment for an empirical formula about the friction loss at Stator_2, Function 2 and Function 4 yield better results than the other two functions (see Table 8-30 and Table 8-31). Therefore they will be preferred as general empirical formulas for all variants.

Friction loss at Stator_2	t = 7 mm, s = varied			t = 14 mm, s = varied			t = 21 mm, s = varied			t = 28 mm, s = varied		
	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm	s = 5 mm	s = 10 mm	s = 15 mm
fun1												
fun2	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓
fun3												
fun4	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓	✓

Table 8-31: Assessment of own empirical formulas about friction loss at Stator_2 for all variants with '**s** = varied, **t** = constant'.

Here, the comparison between Matlab functions (i.e., Function 2 and Function 4) and the CFD results for the friction loss at Stator_2 is made and shown in Figure 8-31 and Figure 8-32.

Friction loss at Stator_2 between OpenFOAM (data) and Function 2 (fit)

Figure 8- 3 1 : Comparison of friction loss at Stator_2 between OpenFOAM (data) and
Function 2 with Matlab (fit).

Friction loss at Stator_2 between OpenFOAM (data) and Function 4 (fit)

Figure 8- 3 2 : Comparison of friction loss at Stator_2 between OpenFOAM (data) and
Function 4 with Matlab (fit).

8.5 A brief summary about empirical formulas

In the last subsections the results comparison between Matlab functions and OpenFOAM simulations has been made for mass flow rate, static and total pressure difference, friction losses at Rotor_2, at Rotor_3 and at Stator_2. The previously presented fitting functions are available separately for the parameters **s** and **t**. They are not dependent of each other.

To investigate if a common empirical formula is possible for all variations (i.e., for '**s** = constant, **t** = varied' and '**s** = varied, **t** = constant') a comparison has been made in this subsection (see Table 8-32), in which can be seen that one general empirical formula (i.e., Function 2) exists for the mass flow rate. For the static pressure differences Function 2, Function 3 and Function 4 would fit well the OpenFOAM results; Function 4 gives the best results for the total pressure differences and the friction losses at rotor walls. For the friction loss at Stator_2 Function 2 and Function 4 represent the best fits for the CFD results of parametrised model 'Glendoe'.

	s = constant, t = varied				t = constant, s = varied			
	Function 1	Function 2	Function 3	Function 4	Function 1	Function 2	Function 3	Function 4
Mass flow rate		✓				✓		
Static pressure difference		✓	✓	✓		✓	✓	✓
Total pressure difference				✓		✓		✓
Friction loss at Rotor_2				✓				✓
Friction loss at Rotor_3				✓				✓
Friction loss at Stator_2		✓		✓		✓		✓

Table 8-32 : The suitable empirical formulas for the variations with '**s** = constant, **t** = varied' and '**s** = varied, **t** = constant'.

8.5.1 Comparison of the variants with 's = constant = 10 mm, t = varied = 14 mm' and 's = varied = 10 mm, t = constant = 14 mm'

Above presented functions are based separately on the parameter **Re**, **s** and **t**:

- $y = f(x_1, x_2)$

- $y = f(x_1, x_3)$

Here, $x_1 = $ **Re**, $x_2 = $ **t** and $x_3 = $ **s**.

To investigate the deviations between the two variations '**s** = constant, **t** = varied' and '**s** = varied, **t** = constant' several comparisons between the fitting Matlab functions and OpenFOAM results are made. The variants with '**s** = constant = 10 mm, **t** = varied = 14 mm' and '**s** =

varied = 10 mm, **t** = constant = 14 mm' are selected as an example and presented in following figures.

Figure 8-３３: The comparison of the results between the fitting Matlab function (Function 2) and OpenFOAM for volume flow rate.

Figure 8-33 shows the comparison for the volume flow rate between Function 2 and OpenFOAM. The volume flow rates of the variants with '**s** = constant = 10 mm, **t** = varied = 14 mm' and '**s** = varied = 10 mm, **t** = constant = 14 mm' show minimal deviations between them and present also small differences to the OpenFOAM results after 500 RPM.

Figure 8-３４: The comparison of the results between the fitting Matlab functions (Function 2 and Function 4) and OpenFOAM for static pressure difference.

Function 2 and Function 4 show approximately the same trend for the static pressure differences between these two variations (see Figure 8-34). But Function 2 can fit the results of OpenFOAM better at low Reynolds numbers than Function 4.

Figure 8- 3 5 : The comparison of the results between the fitting Matlab function (Function 4) and OpenFOAM for total pressure difference.

For the total pressure difference the deviations of the two variations with Function 4 are minimal (see Figure 8-35).

Figure 8- 3 6 : The comparison of the results between the fitting Matlab function (Function 4) and OpenFOAM for the friction loss at Rotor_2.

Function 4 of the two variations shows approximately the same friction losses for rotor and stator walls as the OpenFOAM results (see Figure 8-36, Figure 8-37 and Figure 8-38).

Figure 8- 3 7 : The comparison of the results between the fitting Matlab function (Function 4) and OpenFOAM for the friction loss at Rotor_3.

Figure 8- 3 8 : The comparison of the results between the fitting Matlab function (Function 4) and OpenFOAM for the friction loss at Stator_2.

From the comparisons for this example it can been seen that the fitting Matlab function with 's = constant = 10 mm, t = varied = 14 mm' shows approximately the same results as it with 's = varied = 10 mm, t = constant = 14 mm' for each flow variable (i.e., volume flow rate, static and total pressure difference and the friction losses at rotor and stator walls). Thereafter comparisons between all variants in 's = constant, t = varied' and 's = varied, t = constant' have been made as this example. So to speak, a common empirical formula, which includes the parameter **s** and **t,** is not necessary.

8.5.2 Decision of the empirical formulas for the different flow variables

In summary Function 2 shall be used to estimate volume flow rate and static pressure difference as a function of Reynolds number, whilst for all other physical parameters (i.e., total pressure difference and the friction losses at rotor and stator walls) Function 4 should be used (see Table 8-33):

Flow variables	Fitting function	Matlab Functions
Volume flow rate	2	$y_2 = c_1 + c_2 \cdot \dfrac{t}{t_{max}} + c_3 \cdot \dfrac{Re}{Re_{max}} + c_4 \cdot \dfrac{t}{t_{max}} \cdot \dfrac{Re}{Re_{max}} + c_5 \cdot (\dfrac{Re}{Re_{max}})^2 + c_6 \cdot \dfrac{t}{t_{max}} \cdot (\dfrac{Re}{Re_{max}})^2$
Static pressure difference	2	$y_2 = c_1 + c_2 \cdot \dfrac{s}{s_{max}} + c_3 \cdot \dfrac{Re}{Re_{max}} + c_4 \cdot \dfrac{s}{s_{max}} \cdot \dfrac{Re}{Re_{max}} + c_5 \cdot (\dfrac{Re}{Re_{max}})^2 + c_6 \cdot \dfrac{s}{s_{max}} \cdot (\dfrac{Re}{Re_{max}})^2$
Total pressure difference	4	
Friction loss at Rotor_2	4	$y_4 = (c_1 + c_2 \cdot \dfrac{t}{t_{max}}) \cdot (\dfrac{Re}{Re_{max}})^{c_3}$
Friction loss at Rotor_3	4	
Friction loss at Stator_2	4	$y_4 = (c_1 + c_2 \cdot \dfrac{s}{s_{max}}) \cdot (\dfrac{Re}{Re_{max}})^{c_3}$

Table 8-33: The fitting Matlab functions for the different flow variables.

The coefficients of the individual fit functions (y_2, y_4) have been computated in Matlab and a quick and efficient pre-design of guidance bearing 'Glendoe' is now possible and a common empirical formula including both parameters **s** and **t** is not necessary.

9 Concluding Remarks and Future Work

9.1 Conclusions

This subsection presents the conclusions from the findings of this work and points out commonalities. The results depend strongly on the geometry parameters **s**, **t** and the Reynolds number:

- The volume flow rate and the friction losses at rotor and stator walls are increased with enlarged revolution speed.

- 10 mm is the optimal axial clearance (i.e., the rotor-stator distance).

- When the rotor-stator distance stays constant the volume flow rate is increased with enlarged radial clearance **t** and enlarged inflow volume, which depends on the spacing between inside radius of oil container and stator. The inflow volume is increased with shortened and shifted stator in the direction to the outside radius of the oil container, i.e. parameter **t'** and **at'**.

- Total pressure difference is reduced with increased width of radial clearance **t** and constantly rotor-stator distance.

- At lower Reynolds numbers the turbulence model RNG k – epsilon is not well suited due to the fact that the y^+ values in the first cell layer near the walls are smaller than 30 (i.e., the results stay in buffer layer when $11<y^+<30$). Therefore different results between OpenFOAM and FLUENT have been observed, because OpenFOAM uses standard logarithmic wall-functions and FLUENT uses some kind of blending function in the buffer layer.

- It was possible to obtain general empirical formulas with the help of Matlab for mass flow, static pressure difference, total pressure difference and the friction losses at rotor and stator walls as a function of Reynolds number.

9.2 Future work

Starting from the results achieved in this thesis, several directions for future research are thinkable:

- Consideration for the friction loss at Rotor_1: add the guide bearing between the outside radius of oil container and rotor in the geometry to observe if a strong reduced total pressure difference plays a role here and how the friction loss at Rotor_1 is changed.

- Make more macro-meshes for the parameterised model. Because there are only three different macro-meshes used in this work, with extension of the stator region the meshes are distorted at the outside radius of the axial clearance. This can have a negative impact on the accuracy of the results.

- Different turbulence models should be used for the differing ranges of Reynolds number. Simulations with the same turbulence model for all revolution speeds are prone to errors.

- Measurement data would help to verify the accuracy of the simulations for the Glendoe bearing system.

- Two-phase flow (i.e., oil-air) simulations can help us to determine the oil surface height in the oil container.

- Above presented empirical formulas are only suitable for the generators, which have the approximately same geometric dimensions as 'Glendoe'.

Nomenclature

Latin Letters

-	a	Acceleration	[m/s^2]
-	b	Thickness of the Disc	[m]
-	b'	Mass Imbalance	
-	c_v	Friction Coefficient	[-]
-	C_μ	Dimensionless Constant	[-]
-	D	Width of Stator	[m]
-	f	Body Force Vector	
-	F	Force	[N]
-	G	Gap Ratio	[-]
-	h_{oil}	Height of Container	[m]
-	H_K	Spacing between the Bottom of Container and Rotor	[m]
-	k	Turbulent Kinetic Energy	[m²/s²]
-	k_s	Equivalent Roughness	[m]
-	K	Mean Kinetic Energy	[m²/s²]
-	l	Characteristic Length	[m]
-	m	Mass	[kg]
-	N	Revolution Speed	[1/min]
-	p	Pressure	[Pa]
-	p_v	Friction Power Loss	[Watt]
-	p'	Pressure Correction	
-	$p*$	Initial guessed Pressure	[Pa]

-	R	Radius	[m]
-	R_{ka}	Outside Radius of Rotor	[m]
-	R_{ki}	Inside Radius of Rotor	[m]
-	R_{sa}	Outside Radius of Stator	[m]
-	R_{si}	Inside Radius of Stator	[m]
-	R_{Ta}	Outside Radius of Oil Container	[m]
-	R_{Ti}	Inside Radius of Oil Container.	[m]
-	s	Axial Spacing	[m]
-	s	Spacing between Rotor and Stator	[m]
-	t	Radial clearance	[m]
-	t	Time	[s]
-	T	Temperature	[^{o}C]
-	T_v	Friction Torque	[Nm]
-	u	Velocity	[m/s]
-	$\vec{u}$	Velocity Vector	[m/s]
-	u_τ	Friction Velocity	[m/s]
-	u^+	Dimensionless Velocity	[-]
-	u'	Velocity Correction	
-	u^*	Initial guessed Velocity	[m/s]
-	U	Circumferential Velocity	[m/s]
-	u, v, w	Cartesian Velocities	[m/s]
-	x, y, z	Cartesian Coordinates	
-	y^+	Dimensionless Wall Distance	[-]

Greek Letters

- ε Dissipation Rate of Turbulent Kinetic Energy $[m^2/s^3]$

- β Equation constant

- δ_j Kronecker Delta

- μ Dynamic Viscosity $[Pa \cdot s]$

- μ_t Turbulent Viscosity $[Pa \cdot s]$

- ν Kinematic Viscosity $[m^2/s]$

- ρ Density $[kg/m^3]$

- σ Stress Tensor

- σ_k Turbulent Schmidt Number of the k Equation

- σ_ε Turbulent Schmidt Number of the ε Equation

- σ Equation constant

- κ Karman's constant=0.41

- τ_j Reynold's Stress

- ω Angular Velocity $[rad/s]$

Symbol

- ∇ Del operator

- ISO-VG Nr.   Oil Grade

Dimensionless numbers

- c   Friction Coefficient

- Re   Reynolds number

- Re_{krit} Critical Reynolds number

Abbreviations

- 2D   Two Dimensional

- 3D   Three Dimensional

- ANSYS   ANalysis SYStem

- CAE   Computer Aided Engineering

- CFD   Computational Fluid Dynamics

- CPU   Central Processing Unit

- GAMG   Geometric-Algebraic Multi-grid

- GmbH   Gesellschaft mit beschränkter Haftung

- GPL   General Public License

- LES   Large Eddy Simulation

- MATLAB   Abbreviation of 'Matrix-Laboratorium'

- OpenFOAM   OpenField Operation and Manipulation

- PBiCG   Preconditioned (bi-) conjugate gradient

- RANS   Reynold Averaged Navier-Stokes

- RNG   Renormalization Group

- RPM   Revolution per Minute

- RSM   Reynolds Stress Model

- SST   Shear Stress Transport

- VTK   Visualization Toolkit

Appendices

Appendix A

Parameterfile - Eingabe von Geometriewerten und stoffspezifischen Daten
#
Alle Längeneinheiten wie Radien und Höhen in Metern angeben,
Winkel in Grad, dyn. Viskosität in [Pa.s], Dichte in [kg/m³] und
die Drehzahl in [rad/sec].
#
Kurzbeschreibung der Variablennamen:
Rki: Innenradius-Spurkopf, Rka: Aussenradius-Spurkopf
Rsi: Innenradius-Scheibe, Rsa: Aussenradius-Scheibe, delta_Rsi bzw. delta_Rsa: Änderung Scheibenlänge
Rti: Innenradius-Öltopf, Rta: Aussenradius-Öltopf, delta_Rti: Änderung Öltopfinnenradius
s_min und s_max: Minimale bzw. maximale Spaltweite Öltopf-Spurkopf
Hk: Spurkopfhöhe, Hkmin und Hkmax: minimale und maximale Spurkopfhöhe
Hoel: Ölspiegelhöhe
S: Spaltweite Spurkopf-Scheibe, D: Scheibendicke
Alpha: Anstellwinkel Scheibe, Alpha_min bzw. Alpha_max: minimaler bzw. maximaler Anstellstellwinkel
mu und Dichte: Dynamische Viskosität und Dichte des verwendeten Öls bei T=konst.
Drehzahl: Spurkopfdrehzahl in rad/sec.
Förderhöhe: für die Berechnung des Grenzwertes des Förderdrucks
#
Iter: Anzahl der zu rechnenden Zeitschritte
writeInterval: Wie oft soll ausgeschrieben werden?
pRes: Solver-Toleranz für den Druck
uRes: Solver-Toleranz für die Geschwindigkeiten
kRes und epsRes: Solver-Toleranzen für die turb. kinetische Energie und die Dissipationsrate
rRes und nuRes: Solver-Toleranzen für Reynolds-Spannungen und die turbulente kinematische Viskosität
#
Starten von Einzellauf, Variation und Optimierung:
Einzellauf: Parameter: Wert
Variation: Parameter: Wert,Wert,Wert
Optimierung: Parameter: ? Wert oder Wert ? oder Wert ? Wert - Achtung: kein Beistrich zwischen Wert und ?
#

Rki: 0.448
Rka: 0.550
Rsi: 0.4475
delta_Rsi: 0.0
Rsa: 0.550
delta_Rsa: 0.0
Rti: 0.441
delta_Rti: 0.0
Rta: 0.980
s_min: 0.007
s_max: 0.021

Hk: 0.123
Hk_min: 0.11
Hk_max: 0.13
Hoel: 0.350
S: 0.01
S_min: 0.005
S_max: 0.02
D: 0.01
Alpha: 0.0
Alpha_min: -3.5
Alpha_max: 3.5
mu: 0.02981
Dichte: 846.3
Drehzahl: 15.708,20.944,31.416,47.124,52.360,57.596,62.832,68.068,73.304,78.540,94.248,109.956,
157.080,209.440
Foerderhoehe: 0.1
Iter: 40000
writeInterval: 2500
pRes: 1e-08
uRes: 1e-06
kRes: 1e-06
epsRes: 1e-06
rRes: 1e-06
nuRes: 1e-06

Appendix B

```python
# versatileScheibe.py - Pythonscript zur Parametrisierung und Optimierung
# des CFD-Modells "Scheibenfoerderung-Oelstroemung" mit OpenFOAM
# Import der PyOptimize- und PyFoam-Libraries

import os,sys
import math

from os import path,system

from PyOptimize.VersatileExternCommandFunction import
VersatileExternCommandFunction
from PyFoam.Execution.AnalyzedRunner import AnalyzedRunner
from PyFoam.Execution.ConvergenceRunner import ConvergenceRunner
from PyFoam.Execution.ParallelExecution import LAMMachine
from PyFoam.Execution.UtilityRunner import UtilityRunner
from PyFoam.LogAnalysis.BoundingLogAnalyzer import BoundingLogAnalyzer
from PyFoam.RunDictionary.SolutionDirectory import SolutionDirectory
from PyFoam.RunDictionary.ParameterFile import ParameterFile
from PyFoam.RunDictionary.ParsedParameterFile import FoamFileParser,ParsedParameterFile

# Name des OpenFOAM-Templates

template="Scheibe.template"

# Definition der Klasse

class FoamScheibeFunction(VersatileExternCommandFunction):
    def __init__(self,paraFile):
        VersatileExternCommandFunction.__init__(self,paraFile)

# Definition von Konstanten und Expressions

    def additionalConstants(self):
        return { "pi":3.1415926 }

    def expressions(self):
        return { "Mx" : "Hk-S-(D/2)"
               ,"My" : "(Rsi+delta_Rsi)+((Rsa+delta_Rsa)-(Rsi+delta_Rsi))/2"
               ,"Cos_Alpha" : "cos(Alpha/180.*pi)"
               ,"Sin_Alpha" : "sin(Alpha/180.*pi)" }

# Parameterkontrolle - zBsp.: die Spaltweite darf nicht kleiner als 5 mm werden

    def calculation(self,values,keep=False):
        Spaltweite_Oeltopf = (values["Rki"])-((values["Rti"])+(values["delta_Rti"]))
        Spaltweite_Scheibe = (values["S"])-(((values["Rsa"])-(values["Rsi"]))/2.0) *
abs((math.sin((values["Alpha"])/180.*math.pi)))
        Rsi_Neu = (values["Rsi"])+(values["delta_Rsi"])
        Rsa_Neu = (values["Rsa"])+(values["delta_Rsa"])
        Nmin = 9.549296*(values["Drehzahl"])
        fac = self.eval("Dichte*Drehzahl*72",values)
```

```python
    if (values["Alpha"])<(values["Alpha_min"]) \
        or (values["Alpha"])>(values["Alpha_max"]) \
        or (values["S"])<(values["S_min"]) \
        or (values["S"])>(values["S_max"]) \
        or Spaltweite_Scheibe<0.005 \
        or Rsi_Neu<0.99*(values["Rsi"]) \
        or Rsi_Neu>1.01*(values["Rsi"]) \
        or Rsi_Neu-((values["Rti"])+(values["delta_Rti"]))<0.0065 \
        or Rsa_Neu<0.99*(values["Rsa"]) \
        or Rsa_Neu>1.01*(values["Rsa"]) \
        or Spaltweite_Oeltopf<(values["s_min"]) \
        or Spaltweite_Oeltopf>(values["s_max"]) \
        or (values["Hk"])>(values["Hoel"]) \
        or (values["Hk"])<(values["Hk_min"]) \
        or (values["Hk"])>(values["Hk_max"]) \
        or (values["Rki"])<(values["Rti"])+(values["delta_Rti"]) \
        or (values["Rka"])>(values["Rta"]):
        print "Ungueltige Geometrieangaben"
        return 1.0e3

# Name des SolutionDirectories

    solDir="Foerderscheibe.S_%g.Alpha_%g.L_%g.Hk_%g.s_%g.N_%g" % (values["S"],
values["Alpha"],Rsa_Neu-Rsi_Neu,values["Hk"],Spaltweite_Oeltopf,Nmin)
    case=SolutionDirectory(template,paraviewLink=False,archive=None).cloneCase(solDir)

# Gittererstellung mit blockMesh und andere Templates herrichten

    blockDict=path.join(case.polyMeshDir(),"blockMeshDict")
    transport=path.join(case.constantDir(),"transportProperties")
    rotation=path.join(case.systemDir(),"rotationDict")
    setVelocity=path.join(case.initialDir(),"U")
    calcPressure=path.join(case.systemDir(),"calcPressureDifferenceDict")
    calcMass=path.join(case.systemDir(),"calcMassFlowDict")
    control=path.join(case.systemDir(),"controlDict")
    fvsol=path.join(case.systemDir(),"fvSolution")

    self.doCalcOnFile(control+".template",control,values)
    self.doCalcOnFile(fvsol+".template",fvsol,values)

    if (values["Rsi"])-(values["Rki"])>0.007:
        extension="2"
    elif (values["Rsi"])-(values["Rki"])<-0.007:
        extension="3"
    else:
        extension="1"
    self.doCalcOnFile(blockDict+".template."+extension,blockDict,values)
    block=UtilityRunner(argv=["blockMesh","-
case",case.name],silent=True,logname="blockMesh")
    block.start()
    self.doCalcOnFile(transport+".template",transport,values)
    self.doCalcOnFile(rotation+".template",rotation,values)
    self.doCalcOnFile(calcMass+".template",calcMass,values)
```

```
        self.doCalcOnFile(calcPressure+".template",calcPressure,values)
        self.doCalcOnFile(setVelocity+".template",setVelocity,values)
# Erstellen der Stroemungskontrollflaechen - Scheibenaustritt bzw. Scheibeneintritt

        faceset=UtilityRunner(argv=["cellSetBoundary","-case",case.name,"set1","set3","Eintritt"],
silent=True,logname="cellSetBoundary")
        faceset.start()
        faceset=UtilityRunner(argv=["cellSetBoundary","-case",case.name,"set2","set3","Austritt"],
silent=True,logname="cellSetBoundary")
        faceset.start()


# Erzeugen eines 5-Grad-Segments
        axial=UtilityRunner(argv=["makeAxialMesh","-
case",case.name],silent=True,logname="makeAxialMesh")
        axial.start()


# Kopieren, Umbenennen und Loeschen
        deltaT=ParameterFile(case.controlDict()).readParameter("deltaT")
        os.system("cp -r "+path.join(case.polyMeshDir(),"sets")+"
"+path.join(case.name,deltaT,"polyMesh"))
        os.system("rm -rf "+case.polyMeshDir())
        os.system("mv "+path.join(case.name,deltaT,"polyMesh")+" "+case.polyMeshDir())
        os.system("rm -r "+path.join(case.name,deltaT))


# Starten des Solvers

        run=ConvergenceRunner(BoundingLogAnalyzer(),argv=["simpleFoam","-
case",case.name],silent=True,logname="Solver")
        run.start()


# Auswertungs-Utilities: Massenstrom, Verlustmoment und Druck usw.

        pUtil=UtilityRunner(argv=["calcPressureDifference","-case",case.name,"-
latestTime"],silent=True,logname="Pressure")
        pUtil.add("Paus","Average pressure on Austritt = .*\] (.+)")
        pUtil.add("Pein","Average pressure on Eintritt = .*\] (.+)")
        pUtil.start()

        mUtil=UtilityRunner(argv=["calcMassFlow","-case",case.name,"-
latestTime"],silent=True,logname="MassFlow")
        mUtil.add("massenstrom","Flux at Austritt .* = .*\] (.+) Relat")
        mUtil.start()

        tUtil=UtilityRunner(argv=["torqueLossIncompressible","-case",case.name,"-factor",fac,"-
latestTime"],silent=True,logname="Torque")
        tUtil.add("spurkopf_a","Patch: spurkopf_1 .* = (%f%) .+ = (%f%) .+ = (%f%)")
        tUtil.add("scheibe_a","Patch: scheibe_1 .* = (%f%) .+ = (%f%) .+ = (%f%)")
        tUtil.add("spurkopf_b","Patch: spurkopf_2 .* = (%f%) .+ = (%f%) .+ = (%f%)")
        tUtil.add("scheibe_b","Patch: scheibe_2 .* = (%f%) .+ = (%f%) .+ = (%f%)")
        tUtil.add("spurkopf_c","Patch: spurkopf_3 .* = (%f%) .+ = (%f%) .+ = (%f%)")
        tUtil.add("scheibe_c","Patch: scheibe_3 .* = (%f%) .+ = (%f%) .+ = (%f%)")
        tUtil.add("scheibe_d","Patch: scheibe_4 .* = (%f%) .+ = (%f%) .+ = (%f%)")
        tUtil.start()
```

```python
        ptotUtil=UtilityRunner(argv=["ptot","-case",case.name,"-latestTime"],silent=True)
        ptotUtil.start()

        test=ParsedParameterFile(calcPressure)
        test.content["pressureField"]="ptot"
        test.writeFile()

        totpUtil=UtilityRunner(argv=["calcPressureDifference","-case",case.name,"-latestTime"],
silent=True,logname="totalPressure")
        totpUtil.add("Ptotalaus","Average pressure on Austritt = .*\] (.+)")
        totpUtil.add("Ptotalein","Average pressure on Eintritt = .*\] (.+)")
        totpUtil.start()

        gradVelUtil=UtilityRunner(argv=["wallGradU","-case",case.name,"-
latestTime"],silent=True)
        gradVelUtil.start()

        wallShearUtil=UtilityRunner(argv=["wallShearStress","-case",case.name,"-
latestTime"],silent=True)
        wallShearUtil.start()

        vtkUtil=UtilityRunner(argv=["foamToVTK","-case",case.name,"-
latestTime"],silent=True,logname="foamToVTK")
        vtkUtil.start()

        massenstrom=float(mUtil.get("massenstrom")[0])
        volumenstrom=(massenstrom/(values["Dichte"]))*60000
        scheibenaustrittsdruck=float(pUtil.get("Paus")[0])
        totaldruckaustritt=float(totpUtil.get("Ptotalaus")[0])
        scheibeneintrittsdruck=float(pUtil.get("Pein")[0])
        totaldruckeintritt=float(totpUtil.get("Ptotalein")[0])
        spurkopf1=float(tUtil.get("spurkopf_a")[0])
        spurkopf2=float(tUtil.get("spurkopf_a")[1])
        spurkopf3=float(tUtil.get("spurkopf_a")[2])
        scheibe1=float(tUtil.get("scheibe_a")[0])
        scheibe2=float(tUtil.get("scheibe_a")[1])
        scheibe3=float(tUtil.get("scheibe_a")[2])
        spurkopf4=float(tUtil.get("spurkopf_b")[0])
        spurkopf5=float(tUtil.get("spurkopf_b")[1])
        spurkopf6=float(tUtil.get("spurkopf_b")[2])
        scheibe4=float(tUtil.get("scheibe_b")[0])
        scheibe5=float(tUtil.get("scheibe_b")[1])
        scheibe6=float(tUtil.get("scheibe_b")[2])
        spurkopf7=float(tUtil.get("spurkopf_c")[0])
        spurkopf8=float(tUtil.get("spurkopf_c")[1])
        spurkopf9=float(tUtil.get("spurkopf_c")[2])
        scheibe7=float(tUtil.get("scheibe_c")[0])
        scheibe8=float(tUtil.get("scheibe_c")[1])
        scheibe9=float(tUtil.get("scheibe_c")[2])
        scheibe10=float(tUtil.get("scheibe_d")[0])
        scheibe11=float(tUtil.get("scheibe_d")[1])
        scheibe12=float(tUtil.get("scheibe_d")[2])
        f=open(path.join(solDir,"Zusammenfassung_"+solDir),"w")
```

```python
        f.write("Zusammenfassung_"+solDir+":\n")
        f.write("Volumenstrom in Liter pro Minute - Scheibenaustritt: %5.1f\n" % volumenstrom)
        f.write("Statischer Druck-Scheibenaustritt [Pa]: %6.0f\n" % scheibenaustrittsdruck)
        f.write("Statischer Druck-Scheibeneintritt [Pa]: %6.0f\n" % scheibeneintrittsdruck)
        f.write("Totaldruck-Scheibenaustritt [Pa]: %6.0f\n" % totaldruckaustritt)
        f.write("Totaldruck-Scheibeneintritt [Pa]: %6.0f\n" % totaldruckeintritt)
        f.write("Spurkopf-Verluste parallel zum Spalt [Watt]: %6.0f\n" %
(spurkopf4+spurkopf5+spurkopf6))
        f.write("Scheiben-Verluste parallel zum Spalt [Watt]: %6.0f\n" %
(scheibe4+scheibe5+scheibe6))
        f.write("Spurkopf-Verluste parallel zum Oeltopf [Watt]: %6.0f\n" %
(spurkopf7+spurkopf8+spurkopf9))
        f.write("Spurkopf-Verluste gesamt [Watt]: %6.0f\n" %
(spurkopf1+spurkopf2+spurkopf3+spurkopf4+spurkopf5+spurkopf6+spurkopf7+spurkopf8+
spurkopf9))
        f.write("Scheiben-Verluste gesamt [Watt]: %6.0f\n" %
(scheibe1+scheibe2+scheibe3+scheibe4+scheibe5+scheibe6+scheibe7+scheibe8+scheibe9+sch
eibe10+scheibe11+scheibe12))
        f.close()

        if druckdifferenz<(values["Dichte"])*(values["Foerderhoehe"])*9.81:
        if (scheibenaustrittsdruck-101325.0)<(values["Dichte"])*(values["Foerderhoehe"])*9.81:
        print "Druck am Scheibenaustritt zu gering"
        return 1.0e3

        tEnd=ParameterFile(case.controlDict()).readParameter("endTime")
        tLast=case.getLast()
        if(float(tLast)<float(tEnd)):
            massenstrom=1.0e3
        else:
            massenstrom=-(float(mUtil.get("massenstrom")[0]))
        return massenstrom

# Importieren der Optimierungsalgorithmen

from PyOptimize.SimplexAlgorithm import SimplexAlgorithm
from PyOptimize.PowellAlgorithm import PowellBrentAlgorithm

fun=FoamScheibeFunction(sys.argv[1])

if fun.isVariation():
    result=fun.doVariation(keep=True)
    print result
elif fun.isOptimization():
#    alg=SimplexAlgorithm(fun,verbose=True)
    alg=PowellBrentAlgorithm(fun,verbose=True)
    alg.run()
    alg.report()
    final=alg.solution()
    print "Running with optimal solution"
#    fun.func(final,keep=True)
else:
    sol=fun.func([],keep=True)
```

Appendix C

```
/*--------------------------------------------------------------------------*\
| =========                 |                                              |
| \\      /  F ield         | OpenFOAM: The Open Source CFD Toolbox |
|  \\    /   O peration     | Version:  1.6                          |
|   \\  /    A nd           | Web:    http://www.openfoam.org       |
|    \\/     M anipulation  |                                        |
\*--------------------------------------------------------------------------*/

// FoamX Case Dictionary.

FoamFile
{
    version     2.0;
    format      ascii;
    class       dictionary;
    object      blockMeshDict;
}

// * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * //

arguments       "/home/.../foerderScheibe" off;

convertToMeters 1;

vertices
(
    (0 $Rti+delta_Rti$ -0.01)
    ($Hk-S-D$ $Rti+delta_Rti$ -0.01)
($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsi-My)$ $My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_ Alpha*(Rsi-My)+delta_Rsi$ -0.01)
    (0 $Rsi$ -0.01)
    ($Hk-S$ $Rti+delta_Rti$ -0.01)
    ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsi-My)$ $My-Sin_Alpha*(Hk-S-
Mx)+Cos_Alpha* (Rsi-My)+delta_Rsi$ -0.01)
    ($Hk$ $Rti+delta_Rti$ -0.01)
    ($Hk$ $Rki$ -0.01)
    ($Hoel$ $Rti+delta_Rti$ -0.01)
    ($Hoel$ $Rki$ -0.01)
    ($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsa-My)$ $My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_Alpha*(Rsa-My)+delta_Rsa$ -0.01)
    (0 $Rsa$ -0.01)
    ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsa-My)$ $My-Sin_Alpha*(Hk-S-Mx)+
Cos_Alpha*(Rsa-My)+delta_Rsa$ -0.01)
    ($Hk$ $Rka$ -0.01)
    ($Hk-S-D$ $Rta$ -0.01)
    (0 $Rta$ -0.01)
    ($Hk-S$ $Rta$ -0.01)
    ($Hk$ $Rta$ -0.01)
    ($Hoel$ $Rka$ -0.01)
    ($Hoel$ $Rta$ -0.01)
    (0 $Rti+delta_Rti$ 0.01)
```

```
   ($Hk-S-D$ $Rti+delta_Rti$ 0.01)
   ($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsi-My)$ $My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_Alpha*(Rsi-My)+delta_Rsi$ 0.01)
   (0 $Rsi$ 0.01)
   ($Hk-S$ $Rti+delta_Rti$ 0.01)
   ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsi-My)$ $My-Sin_Alpha*(Hk-S-
Mx)+Cos_Alpha* (Rsi-My)+delta_Rsi$ 0.01)
   ($Hk$ $Rti+delta_Rti$ 0.01)
   ($Hk$ $Rki$ 0.01)
   ($Hoel$ $Rti+delta_Rti$ 0.01)
   ($Hoel$ $Rki$ 0.01)
   ($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsa-My)$ $My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_Alpha*(Rsa-My)+delta_Rsa$ 0.01)
   (0 $Rsa$ 0.01)
   ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsa-My)$ $My-Sin_Alpha*(Hk-S-Mx)+
Cos_Alpha* (Rsa-My)+delta_Rsa$ 0.01)
   ($Hk$ $Rka$ 0.01)
   ($Hk-S-D$ $Rta$ 0.01)
   (0 $Rta$ 0.01)
   ($Hk-S$ $Rta$ 0.01)
   ($Hk$ $Rta$ 0.01)
   ($Hoel$ $Rka$ 0.01)
   ($Hoel$ $Rta$ 0.01)
);

blocks
(
   hex (0 1 2 3 20 21 22 23) ($int(round(((Hk-S-D)/2)*1000))$ $int(round(((Rki-
(Rti+delta_Rti))/2)* 1000))$ 1) simpleGrading (1 1 1)
   hex (1 4 5 2 21 24 25 22) ($int(round((D/2)*1000))$ $int(round(((Rki-
(Rti+delta_Rti))/2)*1000))$ 1) simpleGrading (1 1 1)
   hex (4 6 7 5 24 26 27 25) set1 ($int(round((S/2)*1000))+1$ $int(round(((Rki-
(Rti+delta_Rti))/2)*1000))$ 1) simpleGrading (1 1 1)
   hex (6 8 9 7 26 28 29 27) ($int(round(((Hoel-Hk)/2)*1000))$ $int(round(((Rki-
(Rti+delta_Rti))/2)* 1000))$ 1) simpleGrading (1 1 1)
   hex (3 2 10 11 23 22 30 31) ($int(round(((Hk-S-D)/2)*1000))$ $int(round(((Rsa-
Rsi)/2)*1000))$ 1)  simpleGrading (1 1 1)
   hex (5 7 13 12 25 27 33 32) set3 ($int(round((S/2)*1000))+1$ $int(round(((Rka-
Rki)/2)*1000))$ 1) simpleGrading (1 1 1)
   hex (11 10 14 15 31 30 34 35) ($int(round(((Hk-S-D)/2)*1000))$ $int(round(((Rta-Rka)/3)*
1000))$ 1) simpleGrading (1 2 1)
   hex (10 12 16 14 30 32 36 34) ($int(round((D/2)*1000))$ $int(round(((Rta-Rka)/3)*1000))$ 1)
simpleGrading (1 2 1)
   hex (12 13 17 16 32 33 37 36) set2 ($int(round((S/2)*1000))+1$ $int(round(((Rta-
Rka)/3)*1000))$ 1) simpleGrading (1 2 1)
hex (13 18 19 17 33 38 39 37) ($int(round(((Hoel-Hk)/2)*1000))$ $int(round(((Rta-Rka)/3)*
1000)) $ 1) simpleGrading (1 2 1)
);

edges
(
);
```

```
patches
(
    wall spurkopf_1
    (
        (13 33 38 18)
    )
    wall spurkopf_2
    (
        (7 27 33 13)
    )
    wall spurkopf_3
    (
        (9 29 27 7)
    )
    wall scheibe_1
    (
        (10 30 32 12)
    )
    wall scheibe_2
    (
        (12 32 25 5)
    )
    wall scheibe_3
    (
        (5 25 22 2)
    )
    wall scheibe_4
    (
        (2 22 30 10)
    )
    wall oeltopf
    (
        (0 3 23 20)
        (3 11 31 23)
        (11 15 35 31)
        (15 14 34 35)
        (14 16 36 34)
        (16 17 37 36)
        (17 19 39 37)
    )
    wall achse
    (
        (8 6 26 28)
        (6 4 24 26)
        (4 1 21 24)
        (1 0 20 21)
    )
    patch oelspiegel
    (
        (19 18 38 39)
        (9 8 28 29)
    )
    empty frontAndBack
```

```
    (
        (3 0 1 2)
        (2 1 4 5)
        (5 4 6 7)
        (7 6 8 9)
        (11 3 2 10)
        (12 5 7 13)
        (15 11 10 14)
        (14 10 12 16)
        (16 12 13 17)
        (17 13 18 19)
        (23 22 21 20)
        (22 25 24 21)
        (25 27 26 24)
        (27 29 28 26)
        (31 30 22 23)
        (32 33 27 25)
        (35 34 30 31)
        (34 36 32 30)
        (36 37 33 32)
        (37 39 38 33)
    )
);

mergePatchPairs
(
);


// ************************************************************************* //
```

Appendix D

```
/*--------------------------------------------------------------------------*\
| =========                 |                                                |
| \\      /  F ield         | OpenFOAM: The Open Source CFD Toolbox          |
| \\    /   O peration      | Version:  1.6                                  |
|  \\  /    A nd            | Web:     http://www.openfoam.org               |
|   \\/     M anipulation   |                                                |
\*--------------------------------------------------------------------------*/

// FoamX Case Dictionary.

FoamFile
{
    version     2.0;
    format      ascii;
    class       dictionary;
    object      blockMeshDict;
}

// * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * //

arguments       "/home/.../foerderScheibe" off;

convertToMeters 1;

vertices
(
    (0 $Rti+delta_Rti$ -0.01)
    ($Hk-S-D$ $Rti+delta_Rti$ -0.01)
    ($Hk-S-D$ $Rki$ -0.01)
    (0 $Rki$ -0.01)
    ($Hk-S$ $Rti+delta_Rti$ -0.01)
    ($Hk-S$ $Rki$ -0.01)
    ($Hk$ $Rti+delta_Rti$ -0.01)
    ($Hk$ $Rki$ -0.01)
    ($Hoel$ $Rti+delta_Rti$ -0.01)
    ($Hoel$ $Rki$ -0.01)
    ($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsi-My)$ $My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_Alpha*(Rsi-My)+delta_Rsi$ -0.01)
    (0 $Rsi$ -0.01)
    ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsi-My)$ $My-Sin_Alpha*(Hk-S-Mx)+
Cos_Alpha* (Rsi-My)+delta_Rsi$ -0.01)
    ($Hk$ $Rsi$ -0.01)
    ($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsa-My)$ $My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_Alpha*(Rsa-My)+delta_Rsa$ -0.01)
    (0 $Rsa$ -0.01)
    ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsa-My)$ $My-Sin_Alpha*(Hk-S-Mx)+
Cos_Alpha* (Rsa-My)+delta_Rsa$ -0.01)
    ($Hk$ $Rka$ -0.01)
    ($Hk-S-D$ $Rta$ -0.01)
    (0 $Rta$ -0.01)
    ($Hk-S$ $Rta$ -0.01)
```

(Hk Rta -0.01)
($Hoel$ Rka -0.01)
($Hoel$ Rta -0.01)
(0 $Rti+delta_Rti$ 0.01)
($Hk-S-D$ $Rti+delta_Rti$ 0.01)
($Hk-S-D$ Rki 0.01)
(0 Rki 0.01)
($Hk-S$ $Rti+delta_Rti$ 0.01)
($Hk-S$ Rki 0.01)
(Hk $Rti+delta_Rti$ 0.01)
(Hk Rki 0.01)
($Hoel$ $Rti+delta_Rti$ 0.01)
($Hoel$ Rki 0.01)
($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsi-My)$ $My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_Alpha*(Rsi-My)+delta_Rsi$ 0.01)
(0 Rsi 0.01)
($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsi-My)$ $My-Sin_Alpha*(Hk-S-Mx)+
Cos_Alpha* (Rsi-My)+delta_Rsi$ 0.01)
(Hk Rsi 0.01)
($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsa-My)$ $My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_Alpha*(Rsa-My)+delta_Rsa$ 0.01)
(0 Rsa 0.01)
($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsa-My)$ $My-Sin_Alpha*(Hk-S-
Mx)+Cos_Alpha* (Rsa-My)+delta_Rsa$ 0.01)
(Hk Rka 0.01)
($Hk-S-D$ Rta 0.01)
(0 Rta 0.01)
($Hk-S$ Rta 0.01)
(Hk Rta 0.01)
($Hoel$ Rka 0.01)
($Hoel$ Rta 0.01)
);

blocks
(
    hex (0 1 2 3 24 25 26 27) ($int(round(((Hk-S-D)/2)*1000))$ $int(round(((Rki-
(Rti+delta_Rti))/2)* 1000))$ 1) simpleGrading (1 1 1)
    hex (1 4 5 2 25 28 29 26) ($int(round((D/2)*1000))$ $int(round(((Rki-
(Rti+delta_Rti))/2)*1000))$ 1) simpleGrading (1 1 1)
    hex (4 6 7 5 28 30 31 29) ($int(round((S/2)*1000))+1$ $int(round(((Rki-(Rti+delta_Rti))/2)*
1000))$ 1) simpleGrading (1 1 1)
    hex (6 8 9 7 30 32 33 31) ($int(round(((Hoel-Hk)/2)*1000))$ $int(round(((Rki-
(Rti+delta_Rti))/2)* 1000))$ 1) simpleGrading (1 1 1)
    hex (3 2 10 11 27 26 34 35) ($int(round(((Hk-S-D)/2)*1000))$ $int(round(((Rsi-
Rki)/2)*1000))$ 1) simpleGrading (1 1 1)
    hex (2 5 12 10 26 29 36 34) ($int(round((D/2)*1000))$ $int(round(((Rsi-Rki)/2)*1000))$ 1)
simpleGrading (1 1 1)
    hex (5 7 13 12 29 31 37 36) set1 ($int(round((S/2)*1000))+1$ $int(round(((Rsi-
Rki)/2)*1000))$ 1) simpleGrading (1 1 1)
    hex (11 10 14 15 35 34 38 39) ($int(round(((Hk-S-D)/2)*1000))$ $int(round(((Rsa-
Rsi)/2)*1000))$ 1) simpleGrading (1 1 1)
    hex (12 13 17 16 36 37 41 40) set3 ($int(round((S/2)*1000))+1$ $int(round(((Rsa-
Rsi)/2)*1000))$ 1) simpleGrading (1 1 1)

```
   hex (15 14 18 19 39 38 42 43) ($int(round(((Hk-S-D)/2)*1000))$ $int(round(((Rta-Rka)/3)*
1000))$ 1) simpleGrading (1 2 1)
   hex (14 16 20 18 38 40 44 42) ($int(round((D/2)*1000))$ $int(round(((Rta-Rka)/3)*1000))$ 1)
simpleGrading (1 2 1)
   hex (16 17 21 20 40 41 45 44) set2 ($int(round((S/2)*1000))+1$ $int(round(((Rta-
Rka)/3)*1000))$ 1) simpleGrading (1 2 1)
   hex (17 22 23 21 41 46 47 45) ($int(round(((Hoel-Hk)/2)*1000))$ $int(round(((Rta-Rka)/3)*
1000))$ 1) simpleGrading (1 2 1)
);

edges
(
);

patches
(
   wall spurkopf_1
   (
      (17 41 46 22)
   )
   wall spurkopf_2
   (
      (13 37 41 17)
      (7 31 37 13)
   )
   wall spurkopf_3
   (
      (9 33 31 7)
   )
   wall scheibe_1
   (
      (14 38 40 16)
   )
   wall scheibe_2
   (
      (16 40 36 12)
   )
   wall scheibe_3
   (
      (12 36 34 10)
   )
   wall scheibe_4
   (
      (14 10 34 38)
   )
   wall oeltopf
   (
      (0 3 27 24)
      (3 11 35 27)
      (11 15 39 35)
      (15 19 43 39)
      (19 18 42 43)
      (18 20 44 42)
```

```
      (20 21 45 44)
      (21 23 47 45)
  )
  wall achse
  (
      (8 6 30 32)
      (6 4 28 30)
      (4 1 25 28)
      (1 0 24 25)
  )
  wall oelspiegel
  (
      (23 22 46 47)
      (9 8 32 33)
  )
  empty frontAndBack
  (
      (3 0 1 2)
      (2 1 4 5)
      (5 4 6 7)
      (7 6 8 9)
      (11 3 2 10)
      (10 2 5 12)
      (12 5 7 13)
      (15 11 10 14)
      (16 12 13 17)
      (19 15 14 18)
      (18 14 16 20)
      (20 16 17 21)
      (21 17 22 23)
      (27 26 25 24)
      (26 29 28 25)
      (29 31 30 28)
      (31 33 32 30)
      (35 34 26 27)
      (34 36 29 26)
      (36 37 31 29)
      (39 38 34 35)
      (40 41 37 36)
      (43 42 38 39)
      (42 44 40 38)
      (44 45 41 40)
      (45 47 46 41)
  )
);

mergePatchPairs
(
);


// ************************************************************************* //
```

Appendix E

```
/*--------------------------------------------------------------------------*\
| =========                 |                                  |
| \\      /  F ield         | OpenFOAM: The Open Source CFD Toolbox       |
| \\    /   O peration      | Version:  1.6                    |
|  \\  /    A nd            | Web:     http://www.openfoam.org          |
|   \\/     M anipulation   |                                  |
\*--------------------------------------------------------------------------*/

// FoamX Case Dictionary.

FoamFile
{
    version     2.0;
    format      ascii;
    class       dictionary;
    object      blockMeshDict;
}

// * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * * //

arguments       "/home/.../foerderScheibe" off;

convertToMeters 1;

vertices
(
    (0 $Rti+delta_Rti$ -0.01)
    ($Hk-S-D$ $Rti+delta_Rti$ -0.01)
    ($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsi-My)$ $My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_Alpha*(Rsi-My)+delta_Rsi$ -0.01)
    (0 $Rsi$ -0.01)
    ($Hk-S$ $Rti+delta_Rti$ -0.01)
    ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsi-My)$ $My-Sin_Alpha*(Hk-S-
Mx)+Cos_Alpha* (Rsi-My)+delta_Rsi$ -0.01)
    ($Hk$ $Rti+delta_Rti$ -0.01)
    ($Hk$ $Rsi$ -0.01)
    ($Hoel$ $Rti+delta_Rti$ -0.01)
    ($Hoel$ $Rsi$ -0.01)
    ($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsa-My)$ $My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_Alpha*(Rsa-My)+delta_Rsi$ -0.01)
    (0 $Rsa$ -0.01)
    ($Hk$ $Rki$ -0.01)
    ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rki-My)$ $My-Sin_Alpha*(Hk-S-
Mx)+Cos_Alpha* (Rki-My)$ -0.01)
    ($Hoel$ $Rki$ -0.01)
    ($Hk$ $Rka$ -0.01)
    ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsa-My)$ $My-Sin_Alpha*(Hk-S-
Mx)+Cos_Alpha* (Rsa-My)+delta_Rsa$ -0.01)
    ($Hk-S-D$ $Rta$ -0.01)
    (0 $Rta$ -0.01)
```

```
        ($Hk-S$ $Rta$ -0.01)
        ($Hk$ $Rta$ -0.01)
        ($Hoel$ $Rka$ -0.01)
        ($Hoel$ $Rta$ -0.01)
        (0 $Rti+delta_Rti$ 0.01)
        ($Hk-S-D$ $Rti+delta_Rti$ 0.01)
        ($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsi-My)$ $My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_Alpha*(Rsi-My)+delta_Rsi$ 0.01)
        (0 $Rsi$ 0.01)
        ($Hk-S$ $Rti+delta_Rti$ 0.01)
        ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsi-My)$ $My-Sin_Alpha*(Hk-S-
Mx)+Cos_Alpha* (Rsi-My)+delta_Rsi$ 0.01)
        ($Hk$ $Rti+delta_Rti$ 0.01)
        ($Hk$ $Rsi$ 0.01)
        ($Hoel$ $Rti+delta_Rti$ 0.01)
        ($Hoel$ $Rsi$ 0.01)
        ($Mx+Cos_Alpha*(Hk-S-D-Mx)+Sin_Alpha*(Rsa-My)$ $My-Sin_Alpha*(Hk-S-D-Mx)+
Cos_Alpha*(Rsa-My)+delta_Rsi$ 0.01)
        (0 $Rsa$ 0.01)
        ($Hk$ $Rki$ 0.01)
        ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rki-My)$ $My-Sin_Alpha*(Hk-S-
Mx)+Cos_Alpha* (Rki-My)$ 0.01)
        ($Hoel$ $Rki$ 0.01)
        ($Hk$ $Rka$ 0.01)
        ($Mx+Cos_Alpha*(Hk-S-Mx)+Sin_Alpha*(Rsa-My)$ $My-Sin_Alpha*(Hk-S-
Mx)+Cos_Alpha* (Rsa-My)+delta_Rsa$ 0.01)
        ($Hk-S-D$ $Rta$ 0.01)
        (0 $Rta$ 0.01)
        ($Hk-S$ $Rta$ 0.01)
        ($Hk$ $Rta$ 0.01)
        ($Hoel$ $Rka$ 0.01)
        ($Hoel$ $Rta$ 0.01)
);

blocks
(
    hex (0 1 2 3 23 24 25 26) ($int(round((((Hk-S-D)/2)*1000))$ $int(round(((Rsi-
(Rti+delta_Rti))/2)* 1000))$ 1) simpleGrading (1 1 1)
    hex (1 4 5 2 24 27 28 25) ($int(round((D/2)*1000))$ $int(round(((Rsi-
(Rti+delta_Rti))/2)*1000))$ 1) simpleGrading (1 1 1)
    hex (4 6 7 5 27 29 30 28) ($int(round((S/2)*1000))+1$ $int(round(((Rsi-(Rti+delta_Rti))/2)*
1000))$ 1) simpleGrading (1 1 1)
    hex (6 8 9 7 29 31 32 30) ($int(round(((Hoel-Hk)/2)*1000))$ $int(round(((Rsi-
(Rti+delta_Rti))/2)* 1000))$ 1) simpleGrading (1 1 1)
    hex (3 2 10 11 26 25 33 34) ($int(round((((Hk-S-D)/2)*1000))$ $int(round(((Rsa-
Rsi)/2)*1000))$ 1) simpleGrading (1 1 1)
    hex (5 7 12 13 28 30 35 36) set1 ($int(round((S/2)*1000))+1$ $int(round(((Rki-
Rsi)/2)*1000))$ 1) simpleGrading (1 1 1)
    hex (7 9 14 12 30 32 37 35) ($int(round(((Hoel-Hk)/2)*1000))$ $int(round(((Rki-
Rsi)/2)*1000))$ 1) simpleGrading (1 1 1)
    hex (13 12 15 16 36 35 38 39) set3 ($int(round((S/2)*1000))+1$ $int(round(((Rka-
Rki)/2)*1000))$ 1) simpleGrading (1 1 1)
```

```
    hex (11 10 17 18 34 33 40 41) ($int(round(((Hk-S-D)/2)*1000))$ $int(round(((Rta-Rka)/3)*
1000))$ 1) simpleGrading (1 2 1)
    hex (10 16 19 17 33 39 42 40) ($int(round((D/2)*1000))$ $int(round(((Rta-Rka)/3)*1000))$ 1)
simpleGrading (1 2 1)
    hex (16 15 20 19 39 38 43 42) set2 ($int(round((S/2)*1000))+1$ $int(round(((Rta-
Rka)/3)*1000))$ 1) simpleGrading (1 2 1)
    hex (15 21 22 20 38 44 45 43) ($int(round(((Hoel-Hk)/2)*1000))$ $int(round(((Rta-Rka)/3)*
1000))$ 1) simpleGrading (1 2 1)
);

edges
(
);

patches
(
    wall spurkopf_1
    (
        (15 38 44 21)
    )
    wall spurkopf_2
    (
        (12 35 38 15)
    )
    wall spurkopf_3
    (
        (14 37 35 12)
    )
    wall scheibe_1
    (
        (10 33 39 16)
    )
    wall scheibe_2
    (
        (16 39 36 13)
        (13 36 28 5)
    )
    wall scheibe_3
    (
        (5 28 25 2)
    )
    wall scheibe_4
    (
        (10 2 25 33)
    )
    wall oeltopf
    (
        (0 3 26 23)
        (3 11 34 26)
        (11 18 41 34)
        (18 17 40 41)
        (17 19 42 40)
        (19 20 43 42)
```

```
      (20 22 45 43)
    )
    wall achse
    (
      (8 6 29 31)
      (6 4 27 29)
      (4 1 24 27)
      (1 0 23 24)
    )
    wall oelspiegel
    (
      (22 21 44 45)
      (14 9 32 37)
      (9 8 31 32)
    )
    empty frontAndBack
    (
      (3 0 1 2)
      (2 1 4 5)
      (5 4 6 7)
      (7 6 8 9)
      (11 3 2 10)
      (13 5 7 12)
      (12 7 9 14)
      (16 13 12 15)
      (18 11 10 17)
      (17 10 16 19)
      (19 16 15 20)
      (20 15 21 22)
      (26 25 24 23)
      (25 28 27 24)
      (28 30 29 27)
      (30 32 31 29)
      (34 33 25 26)
      (36 35 30 28)
      (35 37 32 30)
      (39 38 35 36)
      (41 40 33 34)
      (40 42 39 33)
      (42 43 38 39)
      (43 45 44 38)
    )
);

mergePatchPairs
(
);


// ************************************************************************* //
```

Appendix F

```
function data = fitNaNData()

defPath = 'C:/Dokumente und Einstellungen/jleix/Projects/Nan';

[data,path,fDataName] = readNaNData(defPath);
if isempty(data)
    return
end

str = {data.name};
[datNr,ok] = listdlg('PromptString','Select a data set:',...
            'SelectionMode','single',...
            'ListString',str);
if (~ok)
    return;
end

data = data(datNr);
[y,x] = meshgrid(data.y,data.x);
xdata = [y(:) x(:)];
ydata = data.data(:);

h = figure(datNr);
set(h,'Position',[10 10 800 400],'name',data.name);
%surface(x,y,data.data);grid on; view(3);
c0 = [1 1 1 1];
tolx = 1e-20;
toly = 1e-20;
maxFunEval = 1e7;
maxIter = 1e5;

fhandle = @fitFun4;
funcName = func2str(fhandle);

while 1

    ans = inputdlg({'Start values'},'Input lsq parameter',1,{['[' num2str(c0) ']']});
    c0 = str2num(ans{1});

    ans = inputdlg({'Tolerance x', 'Tolerance y','Max function eval','Max iterations'},...
        'Input lsq parameter',1,{num2str(tolx), num2str(toly) , num2str(maxFunEval),
num2str(maxIter)});
    tolx = str2num(ans{1});
    toly = str2num(ans{2});
    maxFunEval = str2num(ans{3});
    maxIter = str2num(ans{4});

    mmx = max(xdata);
    for j=1:size(xdata,2)
        xdataScal(:,j) = xdata(:,j)/mmx(j);
```

```
      end
      mmy = max(ydata);
      ydataScal = ydata/mmy;


      options = optimset('LargeScale','on','tolx',tolx,'tolfun',toly,...
          'maxfunevals',maxFunEval,'Jacobian','on',...
          'MaxIter',maxIter,'DerivativeCheck','on');
      [coef,resnorm,residual,exitflag,output,lambda,jacobian] = ...
          lsqcurvefit(fhandle,c0,xdataScal,ydataScal,[],[],options);
      z = feval(fhandle,coef,xdataScal);
      z = reshape(z,size(x))*mmy;

      subplot(1,2,1); cla; hold on
      surface(x,y,data.data,'FaceColor','green');
      surface(x,y,z,'FaceColor','red');
      legend('data','fit',2)
      grid on; view(3)

      subplot(1,2,2); cla
      surface(x,y,data.data-z,'FaceColor','green');
      legend('error',2)
      grid on; view(3)

      stat([],fDataName,funcName,data.name,c0,coef,data.data,z);

      repeat = questdlg('Repeat?');
      if strcmp(repeat,'Yes')
          c0 = coef;
      else
          break;
      end

end

saveFile = questdlg('Save result to file?');
if strcmp(saveFile,'Yes')
    fName = checkfile(path,'*','w');
    stat(fName,fDataName,funcName,data.name,c0,coef,data.data,z);
    saveas(h,fName,'fig');
end


function stat(fName,dataFileName,funcName,dataName, c0,coef,data,approx)

fid = 1;
if ~isempty(fName)
    fid = fopen([fName '.dat'],'w');
end

fprintf(fid,'\n Result %s',dataName);
fprintf(fid,'\n\t Data file: %s',dataFileName);
fprintf(fid,'\n\t Fit function: %s',funcName);
```

```
fprintf(fid,'\n\t Coeficient: ');
fprintf(fid,'\n\t\t Start: '); fprintf(fid,'%f ',c0);
fprintf(fid,'\n\t\t Final: '); fprintf(fid,'%f ',coef);
error = data(:)-approx(:); mdata = max(data(:)); lsq = sqrt(sum(error.^2))/length(error);
fprintf(fid,'\n\t Data statistic: ');
fprintf(fid,'\n\t\t Data (min mean max): %e %e %e', min(data(:)), mean(data(:)), max(data(:)));
fprintf(fid,'\n\t\t Fit (min mean max) : %e %e %e', min(approx(:)), mean(approx(:)),
max(approx(:)'));
fprintf(fid,'\n\t Error statistic: ');
fprintf(fid,'\n\t\t Absolute (min mean max): %e %e %e', min(error), mean(error), max(error));
fprintf(fid,'\n\t\t Relative (min mean max): %e %e %e', min(error)/mdata, mean(error)/mdata,
max(error)/mdata);
fprintf(fid,'\n\t\t Square (abs rel mean)  : %e %e %e', lsq, lsq/mdata, lsq/length(error));
fprintf(fid,'\n\n');

if fid~=1
    fclose(fid);
end


function [data,defPath,fName] = readNaNData(defPath)


[fName,defPath] = checkfile(defPath,'*','r');
if isempty(fName)
    data = [];
    return
end

fid = fopen(fName,'r');
cmpText = 'Drehzahl ;;Re';

text = fgetl(fid);
count = 1;
dataLines = [];
names = {};
while (text~=-1)
    if strncmp(text,cmpText,length(cmpText))
        c = strread(text,'%s','delimiter',';');
        names{end+1} = c{4};
        dataLines(end+1) = count;
    end
    count = count+1;
    text = fgetl(fid);
end
fclose(fid);

fid = fopen(fName,'r');
dataLines(end+1) = count;

data = [];
for i=1:length(names)
    temp = struct('name','','x',[],'y',[],'data',[]);
```

```
      fgetl(fid); % read first line
      temp.name = names{i};
      col = fgetl(fid); % read 1.parameter line
      col = strread(col,'%s','delimiter',';');
      ind = find(~strcmp(col,''))';
      start = strfind(col{ind(1)},'=');
      for j=ind
          col{j} = str2num(col{j}(start+1:end));
      end
      temp.y = [col{ind}];
      fgetl(fid); % read dimensions

      jmax = dataLines(i+1)-dataLines(i)-3;
      for j=1:jmax
          text = fgetl(fid);
          value = strread(text,'%f','delimiter',';');
          temp.x(end+1) = value(ind(1)-1);
          temp.data(end+1,:) = value(ind);
      end

      data(end+1).name = temp.name;
      data(end).x = temp.x;
      data(end).y = temp.y;
      data(end).data = temp.data;
  end
  fclose(fid);
```

References (Bibliography)

[1] M. Lang, B. Streibl, Zur Überprüfung der analytischen Rechenmodelle für Scheiben- und Mantelreibungsverlust, Technischer Bericht von Andritz-Hydro, 2006

[2] Linnecken H. Der Radreibungsverlust, insbesondere bei Turbomaschinen, AEG-Mitteilungen, 1-2 1957

[3] T. Geis, Strömung und reibungsinduzierte Leistungs- und Wirkungsadverluste in komplexen Rotor-Stator Zwischenräumen, Karlsruhe, Deutschland, 2002

[4] K. H. Grote, H. Dubbel, J. Feldhusen, Taschenbuch fuer den Maschinenbau, *22. Auflage,* Springer-Verlag Berlin Heidelberg, 2005

[5] Schlichting H., Grenzschicht-Theorie, Verlag Braun, Karlsruhe, 5.Auflage, 1965

[6] Sigloch H. , Technische Fluidmechanik, VDI Verlag, 3.Auflage, 1996

[7] FLUENT Inc., *User's guide for FLUENT 6.3*, http://www.fluent.com, 05. January. 2010

[8] http://www.ice-sf.at/ 05. January. 2010

[9] OpenFOAM Inc., *User's guide for OpenFOAM 1.6*, http://www.openfoam.com/docs/user/, 06. January. 2010

[10] http://openfoamwiki.net/index.php/Main_Page, 05. January. 2010

[11] http://www.python.org/, 10. January. 2010

[12] R. Temam, Navier-Stokes equations and nonlinear functional analysis, second Edition, SIAM Philadelphia, 1995

[13] http://en.wikipedia.org/wiki/Navier%E2%80%93Stokes_equations, 05. January. 2010

[14] http://www.cfd-online.com/Wiki/Navier-Stokes_equations, 06. January. 2010

[15] Panton, R.L.: *Incompressible flow*: John Wiley and Sons, 1984.

[16] Papanastasion, T. C, Georgiou, G. C., and Alexadrou, A. N.: *Viscous fluid flow*: CRC Press, 1999.

[17] H. Tennekes, J. L. Lumely, A First Course in Turbulence, The MIT Press, Cambridge, MA, 1981

[18] B.E. Launder and D.B. Spalding. The numerical computation of turbulent flows. Comp. Methods Appl. Mech. Eng., 3:269-289, 1974.

[19] M. Abbot, D. Basco, Computational Fluid Mechanics, Longman Scientific and Technical Pubs., London, UK, 1989

[20] http://www.bakker.org/dartmouth06/engs150/, 17. January. 2010

[21] H. K. Versteeg, W. Malalasekera, *An introduction to Computational Fluid Dynamics*, The finite volume method, England, 1995.

[22] S. B. Pope, *Turbulent flows*, Cambridge University Press, Cornell University, 2000

[23] A. M. Hansen, CFD simulation of a fin-and tube heat exchanger, Group for chemical fluid flow processes, Aalborg University, Esbjerg, 2008.

[24] http://www.cfd-online.com/Wiki/RNG_k-epsilon_model, 09. January.2010

[25] http://www.cfdnode.com/archives/159, 18. January. 2010

[26] http://en.wikipedia.org/wiki/OpenFOAM, 20. January. 2010

[27] http://www.ltt-rostock.de/mediawiki/images/1/1e/OpenFOAM.pdf, 23. January. 2010

[28] http://www.opencfd.co.uk/openfoam/doc/boundaries.html#x24-1350005.2, 25. January. 2010

[29] http://openfoamwiki.net/index.php/Contrib_MakeAxialMesh, 05. February . 2010

[30] http://openfoamwiki.net/index.php/The_SIMPLE_algorithm_in_OpenFOAM, 06. February. 2010

[31] http://www.itk.org/Wiki/ParaView, 27. January. 2010

[32] M. Lygren, Simulation of turbulent flow between a rotating and a stationary disc, Master Thesis, Department of Applied Mechanics, Thermodynamic and fluid Dynamics, Norwegian University of Science and Technology, 2001

[33] J. W. Daily and R. E. Nece, Chamber dimension effects on induced flow and frictional resistance of enclosed rotating discs, 217-232, ASME J. Basic Engng 82, 1960

[34] http://www.ansys.com/products/fluid-dynamics/cfx/ 13.October.2010

[35] J. P. Vanto, Rotating Fluids in Engineering and Science, Mineola, N.Y., USA, 1993

[36] http://www.esi.ac.at/~susanne/MatlabSkriptum.pdf, 10. 05. January. 2010

[37] http://de.wikipedia.org/wiki/Matlab, 10. May. 2010

[38] L.F.Shampine, I. Gladwell, S. Thompson, Solving ODEs with MATLAB, 2003

[39] A. Karvinen, H. Ahlstedt, Simulation of Three-Dimensional Diffuser, Department of Energy and Process Engineering, Tampere University of Technology, 2009

[40] Z. X. Yuan, N. Saniei, X. T. Yan, Turbulent heat transfer on the stationary disk in a rotor-stator system, Department of Mechanical and Industrial Engineering, Southern Illinois University at Edwardsville, Edwardsville, IL 62026, USA, 2002