1D Turbine Design Tool Validation and Loss Model Comparison: Performance Prediction of a 1-stage Turbine at Different Pressure Ratios

Jonas Persson
1D Turbine Design Tool Validation and Loss Model Comparison: Performance Prediction of a 1-stage Turbine at Different Pressure Ratios

Jonas Persson

Approved
2015-11-27

Examiner
Björn Laumert

Supervisor
Johan Dahlqvist

Commissioner

Contact person
ROYAL INSTITUTE OF TECHNOLOGY

School of Industrial Engineering and Management
Department of Energy Technology

Master of Science

1D Turbine Design Tool Validation and Loss Model Comparison:
Performance Prediction of a 1-stage Turbine at Different Pressure Ratios

by Jonas Persson

This project is performed in cooperation with GKN Aerospace and Siemens Industrial Turbomachinery, and is part of the Swedish TURBOPOWER Research Program.
Abstract

This work concerns the validation of two 1D Turbine Design Tools, AXIAL by Concepts NREC and TML by GKN Aerospace, and is purely computational. By using the KTH Test Turbine as a reference frame, these two software programs were set up to simulate its performance, and the results consequently validated against existing experimental data from the turbine. The main objective of this work is to investigate the performance prediction abilities of the 1D Design Tools for a variety of turbine parameters such as efficiency, mass flow, power output and degree of reaction, and study the accuracy of these predictions under given boundary conditions, namely turbine stage inlet pressure, temperature and pressure ratio. The main focus of the simulation was to evaluate the impact of the choice of loss model in the 1D Software Tools for estimation of losses. Thus, in order to gain a better understanding of differences and similarities among the scope of available loss models, as well as deviation models, a literature study was performed. Additionally, in order to extend the knowledge of the detailed performance prediction characteristics of these software tools in regard to the loss model implemented, the individual loss coefficients (profile, secondary, trailing edge, tip clearance and incidence) were studied and analysed. The impact of chosen pressure ratio on the 1D simulation accuracy was also investigated.

The software tool used and the loss model selected were both found to be of great significance to the accuracy of the simulated performance. The pressure ratio (PR) used for simulation also proved to be of great significance, with simulations performed at an elevated PR providing considerably more accurate results than at the design PR, suggesting that the majority of loss models are more accurate when estimating with higher PR.

The KTH Test Turbine stage validated in this work featured a number of special geometrical features of inconvenient nature for 1D simulations. In order to account for this, a number of correction coefficients were developed and implemented and their individual effect on the simulated performance studied. Another special feature of the turbine stage studied was the lean angle of the stator, which impact on the 1D simulations was also investigated. Additionally, a number of different user selectable parameters in AXIAL and their impact on the simulations were investigated. The geometry correction coefficients and stator lean angle were found to be of negligible impact to the overall estimated performance, while the user selectable parameters in AXIAL proved to be of relatively big influence on the simulated results.

Lastly, using the TML software tool, the concept of stator-rotor disc cavity flow known as 'purge flow' was simulated and validated against reference data. Purge flow serves to inhibit the inflow of hot air from the main annulus to the inner hub and simultaneously cool the rotor blades. The TML software was found to overestimate the losses associated with the use of purge flow, although providing relatively coherent trends for parameters such as efficiency, mass flow and power, suggesting that a correction coefficient applied to the overall losses from purge flow could potentially provide better overall accuracy in the simulations.
Acknowledgements

I would like to thank my supervisor Johan Dahlqvist, PhD Candidate at KTH, for his guidance and support throughout the entirety of this work, as well as Dr. Jens Fridh at KTH for his input and advice.

I would also like to thank Lars Hedlund at Siemens Industrial Turbomachinery for his valuable feedback regarding my work and his support concerning the KTH Test Turbine, as well as Staffan Brodin, Pieter Groth and Thomas Johansson at GKN Aerospace for their feedback and assistance concerning the TML software tool used extensively in this work.

I appreciate having been given the opportunity to conduct this project as part of the Swedish TURBOPOWER Research Program and to take part in the yearly conference as well as project meetings throughout the course of this work.

Last but not least, I would like to thank my parents for all the support they have always given me and for encouraging me to pursue my university studies.
# Contents

Abstract .......................................................... ii

Acknowledgements ................................................. iii

Contents ........................................................... iv

Abbreviations ......................................................... vii

Nomenclature ........................................................ viii

## 1 Introduction

1.1 Background ....................................................... 1
   1.1.1 Objectives .................................................. 2
   1.1.2 Limitations ................................................ 3
   1.1.3 Methodology ............................................... 4

1.2 Fundamental Turbomachinery Theory .......................... 4
   1.2.1 Blade and Turbine Stage Parameters ...................... 4
   1.2.2 Performance Parameters .................................. 6

1.3 Turbine Flow and Pressure Loss Components ............... 10
   1.3.1 Profile Loss .............................................. 10
   1.3.2 Secondary Loss .......................................... 11
   1.3.3 Tip Clearance Loss ...................................... 14
   1.3.4 Trailing Edge Loss ...................................... 16

1.4 KTH Test Turbine Facility .................................... 18
   1.4.1 Turbine Characteristics .................................. 18
   1.4.2 Air System ............................................... 19
   1.4.3 Control and Measuring Systems ......................... 20

## 2 Literature Review

2.1 Loss Models .................................................... 21
   2.1.1 Ainley/Mathieson ......................................... 21
   2.1.2 Dunham/Came ............................................. 27
   2.1.3 Kacker/Okapuu .......................................... 29
   2.1.4 Moustapha/Kacker ....................................... 35
   2.1.5 Benner/Sjolander/Moustapha .............................. 40
   2.1.6 Craig/Cox ................................................. 41

2.2 Deviation Models ............................................... 45
# Conclusion

## Bibliography

## List of Figures

## List of Tables

## A Blade Data

A.1 KTH Test Turbine Stage 4b ........................................ 152
A.2 KTH Test Turbine Legacy Stage .................................. 153
A.3 Stage by Ewen et al. (1973) .................................... 154

## B Experimental Data

B.1 KTH Test Turbine Stage 4b ........................................ 155
  B.1.1 Performance Simulation Data ............................... 155
  B.1.2 Purge Flow Leakage Simulation Data .................... 157
B.2 KTH Test Turbine Legacy Stage ................................ 158
B.3 Stage by Ewen et al. (1973) .................................... 159

## C 1D Software Loss Models

C.1 AXIAL ................................................................. 160
C.2 TML ................................................................. 161
Abbreviations

AM    Ainley & Mathieson
AR    Aspect Ratio
BSM  Benner & Sjolander & Moustapha
CC    Craig & Cox
DH    Dunham & Came
KO    Kacker & Okapuu
LE    Leading Edge
MK    Moustapha & Kacker
PR    Pressure Ratio
RMS   Root Mean Square
TE    Trailing Edge
VR    Velocity Ratio
Nomenclature

\( A \)  Flow area normal to flow direction \([\text{m}^2]\)

\( A_n \)  Annulus area \([\text{m}^2]\)

\( A_t \)  Blade passage throat area \([\text{m}^2]\)

\( C \)  True chord \([\text{m}]\)

\( C_x \)  Axial chord \([\text{m}]\)

\( c \)  Absolute velocity \([\text{m/s}]\)

\( c_x \)  Axial velocity component \([\text{m/s}]\)

\( c_\theta \)  Absolute circumferential velocity component \([\text{m/s}]\)

\( C_L \)  Lift coefficient [-]

\( d \)  Leading edge diameter \([\text{m}]\)

\( e \)  Blade back curvature \([\text{m}]\)

\( h \)  Blade height \([\text{m}]\)

\( h \)  Static enthalpy \([\text{J}]\)

\( h_0 \)  Total enthalpy \([\text{J}]\)

\( i \)  Incidence angle (gas flow angle relative to blade inlet angle) \([\degree]\)

\( k \)  Radial tip clearance \([\text{m}]\)

\( o \)  Throat opening \([\text{m}]\)

\( M \)  Mach number [-]

\( M \)  Torque \([\text{Nm}]\)

\( \dot{m} \)  Mass flow \([\text{kg/s}]\)

\( N \)  Rotational speed \([\text{RPM}]\)

\( \dot{W} \)  Power \([\text{W}]\)

\( P \)  Static pressure \([\text{Pa}]\)

\( P_0 \)  Total pressure \([\text{Pa}]\)

\( R \)  Degree of reaction [-]
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>( Re )</td>
<td>Reynolds number</td>
<td>[-]</td>
</tr>
<tr>
<td>( s )</td>
<td>Blade pitch</td>
<td>[m]</td>
</tr>
<tr>
<td>( t )</td>
<td>Blade trailing edge thickness</td>
<td>[m]</td>
</tr>
<tr>
<td>( t_{MAX} )</td>
<td>Blade maximum thickness</td>
<td>[m]</td>
</tr>
<tr>
<td>( T )</td>
<td>Static gas temperature</td>
<td>( ^\circ K )</td>
</tr>
<tr>
<td>( T_0 )</td>
<td>Total gas temperature</td>
<td>( ^\circ K )</td>
</tr>
<tr>
<td>( U )</td>
<td>Rotor blade speed</td>
<td>[m/s]</td>
</tr>
<tr>
<td>( v )</td>
<td>Gas velocity</td>
<td>[m/s]</td>
</tr>
<tr>
<td>( w )</td>
<td>Absolute relative velocity</td>
<td>[m/s]</td>
</tr>
<tr>
<td>( w_\theta )</td>
<td>Relative circumferential velocity component</td>
<td>[m/s]</td>
</tr>
<tr>
<td>( Y )</td>
<td>Pressure loss coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>( Z )</td>
<td>Ainley and Mathieson’s loading parameter</td>
<td>[-]</td>
</tr>
<tr>
<td>( \alpha )</td>
<td>Absolute flow angle, relative to meridional</td>
<td>( ^\circ )</td>
</tr>
<tr>
<td>( \alpha_m )</td>
<td>Vector-mean flow angle</td>
<td>( ^\circ )</td>
</tr>
<tr>
<td>( \beta )</td>
<td>Blade angle</td>
<td>( ^\circ )</td>
</tr>
<tr>
<td>( \beta )</td>
<td>Relative flow angle, relative to rotor</td>
<td>( ^\circ )</td>
</tr>
<tr>
<td>( \gamma )</td>
<td>Ratio of specific heats</td>
<td>[-]</td>
</tr>
<tr>
<td>( \gamma/\zeta )</td>
<td>Stagger angle</td>
<td>( ^\circ )</td>
</tr>
<tr>
<td>( \Delta )</td>
<td>Change</td>
<td>( ^\circ )</td>
</tr>
<tr>
<td>( \delta )</td>
<td>Deviation angle</td>
<td>( ^\circ )</td>
</tr>
<tr>
<td>( \delta )</td>
<td>Boundary layer overall thickness</td>
<td>[m]</td>
</tr>
<tr>
<td>( \delta^* )</td>
<td>Boundary layer displacement thickness</td>
<td>[m]</td>
</tr>
<tr>
<td>( \delta_e )</td>
<td>Boundary layer energy thickness</td>
<td>[m]</td>
</tr>
<tr>
<td>( \eta )</td>
<td>Turbine efficiency</td>
<td>[-]</td>
</tr>
<tr>
<td>( \mu )</td>
<td>Dynamic viscosity</td>
<td>[kg/ms]</td>
</tr>
<tr>
<td>( \rho )</td>
<td>Density</td>
<td>[kg/m(^3)]</td>
</tr>
<tr>
<td>( v )</td>
<td>Kinematic viscosity</td>
<td>[m(^2)/s]</td>
</tr>
<tr>
<td>( \nu )</td>
<td>Specific volume</td>
<td>[m(^3)/kg]</td>
</tr>
<tr>
<td>( \phi )</td>
<td>Flow coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>( \phi^2 )</td>
<td>Kinetic energy loss coefficient ( \left( \frac{\text{actual gas exit velocity}}{\text{ideal gas exit velocity}} \right)^2 )</td>
<td>[-]</td>
</tr>
<tr>
<td>( \psi )</td>
<td>Loading coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>( \omega )</td>
<td>Angular velocity</td>
<td>[rad/s]</td>
</tr>
</tbody>
</table>
Nomenclature

Subscripts

1 Stator inlet
2 Stator outlet/Rotor inlet
3 Rotor outlet
C Casing
H Hub
k Tip clearance
m Mean
m Meridional
mid Midspan
p Profile
s Secondary
ss Static-to-static
ts Total-to-static
tt Total-to-total
T Tip
Chapter 1

Introduction

1.1 Background

During the preliminary design phase of a turbine stage it is desirable to be able to predict the approximate performance of a design. This is most conveniently achieved using 1D mean line calculations on the stage and its blades. 1D mean line calculations estimate the performance at the mean-radius of the stage flow path, using blade and annulus geometrical parameters. The main advantage to this type of calculation procedure as compared to the more modern CFD simulation approach is its ease of implementation and low time consumption, making it still very widely used in industry to this day in competition from more advanced techniques.

The most important part of the 1D calculations is the estimation of stage aerodynamic losses, and for this purpose exists a number of different loss model systems developed and refined throughout the last century. These pressure loss systems, coupled with models for predicting the outlet flow angle from a blade row (deviation), are implemented in a number of commercial and non-commercial turbomachinery 1D mean line design software tools. The purpose of this project is to validate the performance prediction characteristics of two 1D design software suites using the KTH Test Turbine as a reference frame, while analysing the differences in predicted performance based on the loss model and deviation model implemented.

Studies of the significance of loss model selection and/or design tool for 1D mean line design and performance estimation have been carried out by several authors (see for example Dahlquist (2008), Guédez (2011), Jouybari Javaniyan et al. (2010), Klonowicz et al. (2014), Lozza (1982), Wei (2000)), investigating the impact of varying design parameters and turbines of differing
configurations. The studies by Wei (2000), Guédez (2011) and Mikaillian (2012) have all been performed using stages in the KTH Test Turbine as the reference frame. The study by Wei (2000) used a custom built 1D software tool for the simulation of turbine stage performance while the work by Guédez (2011) relied on the 1D tool LUAX-T developed by Lund University and the report by Mikaillian (2012) served to evaluate the performance prediction differences of two different in-house design software at Siemens Industrial Turbomachinery (SIT).

This present study, however, is aimed at validating two particular software tools: the commercial turbine and compressor design software AXIAL by Concepts NREC, and the non-commercial in-house turbine design software TML by GKN Aerospace. These programs do not require any mathematical input in the form of loss correlations, instead all correlations are already programmed into the software and the user can select which loss correlations should be applied. The capability of AXIAL to accurately predict the performance of a number of different turbine types and stage configurations has already been proven in a validation study by Dubitsky et al. (2003), making it interesting to also validate its accuracy in simulating the low aspect-ratio stage of the KTH Test Turbine.

1.1.1 Objectives

The main objective of this work is to validate the performance prediction accuracy of the two 1D software tools AXIAL by Concepts NREC and TML by GKN Aerospace at two different pressure ratios, while investigating the significance of the choice of loss model implemented in the programs. Concurrently, a number of other aspects related to the special geometry of the turbine stage simulated are to be analysed, and the set-up procedure and versatility of the 1D software tools investigated. A set of sub-goals defining the objective of this work has been formulated:

- Validate the results of the performance simulations performed with the 1D design software tools with experimental data.
- Investigate the impact of the pressure ratio on performance simulation and examine its dependence on loss model.
- Evaluate the impact of the choice of loss model for performance prediction, as well as the choice of deviation model (when applicable).
Analyse and compare the estimated partial losses of each loss model system in terms of their loss coefficients, in order to identify where the loss systems differ and determine the extent of the impact of these differences on performance estimation.

Simulate and analyse the Rotor/Stator loss distribution.

Analyse the effect of geometry-correction coefficients applied to the 1D software tools to account for unconventional stage features.

Investigate the user-selectable parameters of the software tools, and determine which are of major impact to the overall stage performance prediction.

Analyse the effect of the blade lean of the stator when performing 1D simulations.

Investigate the abilities of the TML software tool to simulate purge flow and examine its dependence on loss model.

1.1.2 Limitations

The present work is restricted by the following limitations:

This work is purely a validation study and thus no design recommendations are made.

The only loss models and deviation models analysed are those available in the evaluated 1D software tools as standard. Thus, no additional loss model-systems available in the open literature are taken into consideration in this work.

The present study only concerns loss models available in the open literature, and no proprietary systems are investigated.

No attempt is made to optimise the loss models by use of scaling factors, except for when the use of correction coefficients is deemed justified. In such cases, thorough justifications for the use of such coefficients are made. Hence, the aim of this report is not to optimise the performance of the 1D software and their loss models, but rather to examine their accuracy as is.

The flow medium analysed, air, is assumed to behave as an ideal gas.

Mechanical losses in the drive train are compensated for in the experimental performance data of the turbine stage used for validation of the 1D simulation results. Therefore, mechanical losses are not taken into account in the simulation, only aerodynamic losses from the expansion process are being evaluated.
1.1.3 Methodology

Firstly, a literature study is carried out aimed at improving the understanding of the annulus flow and flow components in turbines as well as the many loss models relevant to the 1D design software tools to be utilised. Secondly, the procedure for defining the computational domain of the turbine stages to be analysed in the 1D design software tools is described, together with discussion regarding general considerations of the setup of the individual tools. In this section, methods for handling special geometrical features of the turbine stage simulated in the 1D software are also discussed. Thereafter, the process of defining the loss models and other relevant aspects related to this selection is described. Lastly, the results of the performance simulations obtained from the two software and the different loss models are discussed. In this discussion, the results are also compared to those of other similar studies in order to try and establish common trends and discrepancies related to specific aspects of the turbine stages, boundary conditions and loss models utilised.

1.2 Fundamental Turbomachinery Theory

1.2.1 Blade and Turbine Stage Parameters

The basic geometrical parameters of a conventional turbine blade are seen in Fig. 1.2-1. The distance between blade leading edge and trailing edge is known as the chord length \(c\), and the angle between chord line and meridional is called the stagger angle \(\zeta\). Blade spacing is controlled by two main parameters, the pitch \(s\) and throat opening \(o\). The deviation angle \(\delta\) is equal to the blade outlet angle \(\beta_2\) minus the flow outlet angle \(\alpha_2\), and is an essential parameter in turbine performance evaluation. The degree to which the outgoing flow angle is estimated to deviate from the actual metal blade angle is governed by a deviation model. There exists a number of different deviation models, resulting in varying degrees of flow deviation when applied.

The incidence angle is an important parameter for turbine 'off-design' performance calculations. The incidence angle \(i\) is equal to the inlet flow angle \(\alpha_1\) minus the blade inlet angle \(\beta_1\), as seen in Fig. 1.2-2. At design conditions, the design incidence angle is determined as the inlet flow angle relative to blade inlet angle that gives rise to the highest possible efficiency, this angle will be the nominal incidence of \(0^\circ\). At so called 'off-design' conditions the angle of
incidence is different from the inlet angle of optimal efficiency, which can greatly affect cascade through-flow features and impact turbine performance. The complete incidence angle geometry is seen in Fig. 1.2-2.

The simplified geometry of a conventional turbine stage is illustrated by Fig. 1.2-3. A turbine stage normally consists of a stator followed by a rotor. The stator is a stationary blade row that turns the incoming expanding flow in the desired direction before it impacts with the rotor. The rotor then rotates, absorbing the kinetic energy of the expanding flow which can be utilised e.g.
in the form of shaft power or harnessed as electrical energy via a generator depending on the application. The flow through a turbine stage is shown in Fig. 1.2-4.

The sign convention for inlet and outlet blade and flow angles used in this report is shown in Fig. 1.2-5.

![Figure 1.2-5: Sign convention for blade and flow angles of stator and rotor blades.](image)

### 1.2.2 Performance Parameters

In this section, a number of parameters for turbomachinery performance evaluation will be presented. The geometrical properties of cascade flow and the thermodynamic properties of a typical expansion process in a turbine stage are shown in Figs. 1.2-6 and 1.2-7 respectively. All the performance parameters to be presented make reference to the data in these diagrams.

**Expansion Pressure Ratio**

The expansion pressure ratio (PR) is the ratio of inlet to outlet pressure of the turbine. The PR parameter is most often encountered in the total-to-total and total-to-static definitions. The present work utilises the total-to-static definition:

\[
PR_{ts} = \frac{P_{01}}{P_3}
\]  

(1.2-1)
Stage Efficiency

The total-to-total efficiency of a turbine stage is defined as (Dixon & Hall, 2010, p. 105):

$$\eta_{tt} = \frac{\text{actual work output}}{\text{ideal work output when operating to same back pressure}} = \frac{h_{01} - h_{03}}{h_{01} - h_{03ss}} \quad (1.2-2)$$

In cases where the exit velocity of a stage is not recovered, the total-to-static efficiency is normally preferred. This parameter is defined as:

$$\eta_{ts} = \frac{h_{01} - h_{03}}{h_{01} - h_{3ss}} \quad (1.2-3)$$

The static-to-static efficiency is not a normally used parameter. It is defined as (Based on Wei, 2000, p. 12):

$$\eta_{ss} = \frac{h_1 - h_3}{h_1 - h_{3ss}} \quad (1.2-4)$$
Isentropic Velocity Ratio

The isentropic velocity ratio (VR) is used for evaluation of turbomachinery performance at different points of operation. The total-to-static velocity ratio is defined as (Dahlqvist & Fridh, 2015, p. 1):

$$\nu_{ts} = \frac{U}{\sqrt{2\Delta h_{ts,\text{is}}}}$$  \hspace{1cm} (1.2-5)

Stage Power Output

The total power output of the turbine stage is (Vogt, 2007):

$$\dot{W} = \dot{m}(h_{01} - h_{03}) \ [W]$$  \hspace{1cm} (1.2-6)

Stage Torque Output

The total torque output of the stage is (Based on Vogt, 2010):

$$M = \frac{\dot{W}}{\omega} \ [Nm]$$  \hspace{1cm} (1.2-7)

Stage Degree of Reaction

The stage reaction is the ratio of static enthalpy drop in the rotor to static enthalpy drop across the stage:

$$R = \frac{h_2 - h_3}{h_1 - h_3}$$  \hspace{1cm} (1.2-8)

Assuming the flow through a turbine as being nearly isentropic, the stage reaction may also be written as:

$$R \approx \frac{p_2 - p_3}{p_1 - p_3}$$  \hspace{1cm} (1.2-9)

The stage reaction is an indicator of the blade geometries of the stage. Fifty-percent reaction usually means that stator and rotor blades are of similar geometry and camber angle, while a zero reaction turbine indicates that rotor and stator blades are of high and low camber respectively, which results in minimal pressure change through the rotor (Dixon & Hall, 2010, p. 101).
Pressure Loss Coefficient

The stagnation pressure loss coefficient is an important parameter for evaluating pressure losses in turbines, and the fundamental parameter for turbomachinery performance predictions in many loss systems. With reference to Fig. 1.2-7, for the stator row, this parameter is defined as (Ainley & Mathieson, 1951):

\[ Y = \frac{P_{01} - P_{02}}{P_{02} - P_2}; \]  \hspace{1cm} (1.2-10)

and for the rotor row the equivalent parameter would be:

\[ Y = \frac{P_{02} - P_{03}}{P_{03} - P_3}; \]  \hspace{1cm} (1.2-11)

Design Flow Coefficient

The design flow coefficient is the ratio of meridional flow velocity to blade speed:

\[ \phi = \frac{c_m}{U}; \]  \hspace{1cm} (1.2-12)

Low values of \( \phi \) implies highly staggered blades while high values imply low stagger (Dixon & Hall, 2010, p. 100-101).

Stage Loading Coefficient

The stage loading is equal to the ratio of stage stagnation enthalpy change to blade speed:

\[ \psi = \frac{\Delta h_0}{U^2}; \]  \hspace{1cm} (1.2-13)

For a purely axial turbine \( \Delta h_0 = U \Delta c_\theta \), and thus the stage loading coefficient can be written:

\[ \psi = \frac{\Delta c_\theta}{U}; \]  \hspace{1cm} (1.2-14)

High values of \( \psi \) is indicative of large flow turning, reducing the number of stages necessary for the required work output, however, high stage loading is limited by the impact on efficiency (Dixon & Hall, 2010, p. 101).
Turbine Constant

The turbine constant is a non-standard indicator of mass flow, used by certain companies in the industry (Mikaillian, 2012):

\[
C_T = \frac{\dot{m}}{\sqrt{\frac{p_1^2 - p_2^2}{p_1 v_1}}} [m^2], \quad (1.2-15)
\]

where \(v_1\) is the specific volume before stator inlet.

1.3 Turbine Flow and Pressure Loss Components

Due to the complexity of the flow effects that cause losses in blade cascades and turbine stages, it is necessary to evaluate the loss mechanisms individually in order to better understand the true contribution of different regions of the blade and blade passage to the overall loss. During the last century, a number of authors have attempted to establish exactly what causes these losses, and based on their conclusions they have tried to establish correlations which could predict them accurately. This chapter provides an overview of the different loss categories present in literature, while Chapter 2 examines the loss models of a select number of authors in higher detail.

1.3.1 Profile Loss

Flow through a turbine blade row causes boundary layers to form on the pressure and suction sides of the blades. Fig. 1.3-1 shows the typical turbulent boundary layer profile common in turbomachines (Stewart et al., 1960, p. 588-589). The displacement thickness varies with the flow features, however, it is usually greater on the suction side than the pressure side. These types of flow effects give rise to losses in the blade rows and turbine stages, known as profile losses.

The profile loss is generated in the central region of the blade from leading edge to trailing edge, away from inner and outer endwalls, and is usually assumed to be two-dimensional (Denton, 1993, p. 622). The boundary layer is either laminar or turbulent depending on the Reynolds number. In a certain range of Reynolds number the existence of either laminar or turbulent boundary layers is possible, often with a significantly greater loss generation when in the turbulent state (Denton, 1993, p. 626).
1.3.2 Secondary Loss

Secondary losses arise at the endwalls of the blade, caused partly by secondary flows generated by the passage of annulus boundary layers through the blade row (Denton, 1993, p. 622). The drivers of secondary flows are transverse static pressure gradients and mass forces acting on the flow medium in the blade passage. The endwall boundary layers are also altered in shape by the very same secondary flows that they give rise to (Lampart, 2009, p. 321-322). The losses attributable to endwall secondary flows are substantial, and may account for almost 30-50 percent of the total loss in a blade row. Fig. 1.3-2 shows the traditional secondary flow vortex system based on the assumption of inviscid flow (Sharma & Butler, 1987, p. 230).
However, later research has shown that the secondary flow field is more complex than previously thought. Rather than endwall boundary layers being restricted to passage vortexes along the endwalls of the cascade, the boundary layers in fact separate at the leading edge, causing a horseshoe vortex to form, as seen in Fig. 1.3-3. The horseshoe vortex consists of two legs, called the suction- and pressure-side legs, (Sharma & Butler, 1987, p. 231). The tightness of the vortices in the figures has been exaggerated for illustration purposes and is not consistent with reality. For example, the real number of rotations of the passage vortex on its way through the cascade has been observed to be about one and one half (Wang et al., 1997, p. 3).

Although the majority of the inlet boundary layer flow is consumed by this vortex-formation and subsequently formed passage vortex, the fluid particles in the inlet boundary layer closest to the endwall are subject to a convective process causing them to instead move along the suction side surface and exit the blade row on top of the main passage vortex (Sharma & Butler, 1987, p. 231), meanwhile rotating in the opposite direction to that of the passage vortex (Wang et al., 1997, p. 5).

On the pressure side of the blade, the vortex is subjected to the pressure gradient of the blade-passage and convected towards the suction side of the adjacent blade where it connects with the suction leg close to the point of lowest pressure (Sharma & Butler, 1987, p. 231), about 1/4 of the surface distance form the leading edge (Wang et al., 1997, p. 5). At this position a
new boundary layer is formed, from which the pressure side leg absorbs low-momentum fluid particles along its flow path, thereby forming the passage vortex as it grows in size (Sharma & Butler, 1987).

![Cascade endwall flow structure](image)

**Figure 1.3-3**: Cascade endwall flow structure (Reproduced from Sharma & Butler, 1987, Fig. 2).

**Loss Model Comparison**

The loss models presented in Chapter 2 have progressively evolved to reflect the developments in the understanding of secondary flows in cascades. The original secondary loss coefficient correlation by Ainley & Mathieson (1951) is based on a number of distance and area ratios of the blade as well as blade row flow angles. The subsequent models by Dunham & Came (1970) and Kacker & Okapuu (1982) identify the aspect ratio \( h/c \) as being a factor of great importance for the losses generated at the endwall, and modify earlier correlations to reflect this dependence. The subsequent work by Moustapha & Kacker (1990) proposes the leading edge geometry as being of major important for secondary losses and their correlation thus came to depend on the ratio of leading edge diameter to blade chord \( d/c \). The more recent work by Benner *et al.* (1995) makes no modifications to the secondary loss correlation by Moustapha & Kacker (1990). The Craig & Cox (1970) system uses a secondary loss factor dependent on inlet/outlet fluid angle, pitch and blade backbone length, with correction factors for aspect ratio and Reynolds number influence.
1.3.3 Tip Clearance Loss

Tip clearance losses are due to flow leakage over the tips of rotor blades. Since the tip leakage occurs very close to the casing, there may be an interaction between this type of loss and endwall loss. The loss mechanisms involved in tip clearance loss are heavily dependent on whether the blades are shrouded or unshrouded (Denton, 1993, p. 622-623). In Fig. 1.3-4 the tip leakage over an unshrouded rotor is illustrated, and in Fig. 1.3-5 the same process for a shrouded blade is shown.

For shrouded blades, stagnation usually arises in the area between endwall and leading edge, causing a horseshoe vortex to form. However, for unshrouded blades, unless the tip gap is very small, stagnation cannot take place and thus no horseshoe vortex will form. The flow at tip endwall is divided into two streams above the leading edge, a main stream through the tip gap and a cross-flow stream across the blade-channel passage, both exposed to convective forces steering them towards the suction side of the adjacent blades. Due to an adverse pressure gradient on the suction surface, the tip leakage flow separates from the endwall causing a tip
leakage vortex. This vortex in turn blocks the cross-flow stream, making it separate as well, forming a passage vortex (Lampart, 2009, p. 327-328). The developed tip leakage flow patterns are illustrated in Fig. 1.3-4.

Shrouded blade tip leakage features differ significantly from those of unshrouded blades. Fig. 1.3-5 shows the tip leakage process of a shrouded blade with a single tip seal. As the leakage flow passes through the tip seal clearance space it contracts to a jet. This jet has a smaller area than the seal clearance itself, determined by a contraction coefficient with a typical value of about 0.6. After forming at the blade seal, the jet is mixed out in an irreversible process taking place in the clearance space, causing a substantial decrease in meridional velocity, however, the swirl velocity of the leakage flow remains largely unaffected by this process. Lastly, after passing the seal clearance area, the leakage flow once again must unite with the main flow of the blade passage. Due to differences in meridional and swirl velocities between the two flows, further mixing losses will arise as this process takes place. The angle of re-injection of the seal leakage flow should be as small as possible in order to reduce losses (Denton, 1993, p. 637-638).

**Loss Model Comparison**

The tip clearance loss is included as a stand-alone component in most loss systems. Ainley & Mathieson (1951) and onwards use a linear tip clearance loss variation, dependent on the tip clearance, blade span, pitch and chord and a lift coefficient. Dunham & Came (1970) maintain the general structure of previous correlations but suggest that the tip clearance losses
are proportional to the tip clearance raised to the power of 0.78, and introduce an expression for calculating the equivalent tip clearance of a shrouded blade with multiple tip seals that better predicts the losses for this type of configuration. Kacker & Okapuu (1982) derive a new expression for unshrouded blades but retain the correlation by Dunham & Cane (1970) for losses in shrouded blades. Later loss systems by Moustapha & Kacker (1990) and Benner et al. (1995) only examine the profile and secondary loss mechanisms and do not propose a specific tip clearance loss correlation. The system by Craig & Cox (1970) uses a stand-alone component for tip clearance loss as well, dependent on the so called "lap" of the runner blade. Their correlation was developed for shrouded blades with correction factors for unshrouded configurations.

1.3.4 Trailing Edge Loss

Trailing edge loss arises due to mixing occurring as the adjacent flows of the two blade surfaces exit the blade row, however, at supersonic velocities shock losses also contribute. As seen in Fig. 1.3-6, for a low-speed turbine, the flow velocity of the boundary layer flow exiting the blade row is greatly reduced in the regions close to the surface, especially on the suction side where the boundary layer is thicker. Of the total two-dimensional losses of a blade, the mixing loss arising behind the trailing edge can constitute about a third for subsonic flow, and half for supersonic flow (Denton, 1993, p. 635).

For supersonic flow, the trailing edge loss mechanisms are more complex. As seen in 1.3-7, a triangular region of relatively low flow velocity forms immediately behind the trailing edge.
This region has close to uniform pressure conditions (base pressure), and is surrounded by shear layers, where flow velocity changes rapidly from low to high between the base region and free stream. The forming of the shear layers causes a change in flow direction, which generates expansion waves affecting the main flow. This expansion is sometimes followed by a weak shock, making the flow assume the base pressure, and other times the flow expands directly to the base pressure without any shock. Finally, the flows from pressure and suction side meet, and they do so in an area behind the base region termed the confluence region. As the two united flows turn to a common flow direction, shock waves are formed, of which one shock wave will propagate towards the suction surface of the adjacent blade, greatly affecting the pressure distribution of the main flow. The flow in the area downstream of the confluence region can be either subsonic or supersonic with a turbulent and unsteady blade wake often being present (Denton & Xu, 1990, p. 277).

Loss Model Comparison

Among the pressure based loss systems presented in Chapter 2, the method of handling the trailing edge loss differs greatly. Ainley & Mathieson (1951) use a correction factor applied to the total loss to account for the effect of trailing edge thickness when it differs from that of a default blade (2% of blade pitch). The review of the Ainley/Mathieson system conducted by Dunham & Cane (1970) makes no explicit changes to the trailing edge loss prediction. Just over a decade later, Kacker & Okapuu (1982) introduce a separate trailing edge loss component based on measurements of boundary layers at trailing edge, rather than using a correction factor for the trailing edge loss applied to the total loss as did the previous systems. The more mathematically complex system by Moustapha & Kacker (1990) appears to include the trailing edge loss as part of the profile loss coefficient. The method by Benner et al. (1995) only modifies the profile loss correlation of previous systems and makes no changes to the handling of the trailing edge loss component. The system by Craig & Cox (1970) uses a correction coefficient for trailing edge thickness applied to the basic profile loss parameters to account for its effect on efficiency, as well as an additional trailing edge loss increment term added on top of this as part of the same profile loss.
1.4 KTH Test Turbine Facility

The KTH Test Turbine facility has been in operation since 1989. The test turbine can be run with one to three stages and is driven by compressed air supplied by a screw compressor (Sodergard et al., 1989, p. 1). The stage configuration 4b currently utilised in the Test Turbine is seen in Fig. 1.4-1.

1.4.1 Turbine Characteristics

The test turbine consists of three housings, with the end housings being supported on a frame connected to the central housing by flanges. The first of the three housings, referred to as the inlet housing, is connected to an air system safety valve and inside this housing is installed a honeycomb with small holes meant to create a uniform inlet airflow. Once having passed through this grid, the airflow is accelerated in a narrowing channel, known as a bell mouth, before entering the guide blades. Thanks to interchangeable inner and outer turbine walls, blade cascades of varying disc and blade length can be tested (Sodergard et al., 1989, p. 27-28). The central housing does not support any of the internals, but rather functions as a cover for the test object and features fitted traversing pressure probes.

Figure 1.4-1: KTH Test Turbine 1-stage configuration 4b.
Lastly, the end housing contains the air outlet installations and houses the turbine shaft and bearings. A plate with closely spaced holes is fitted at the end of the annular channel, which purpose is to restrict the airflow in order to cause a pressure drop, preventing downstream flow disturbances from influencing the upstream flow of the turbine. A torque measuring device is connected to the outer end of the turbine shaft via a coupling (Sodergard et al., 1989, p. 30).

<table>
<thead>
<tr>
<th>Number of turbine stages</th>
<th>1 - 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max. outer diam. of a bladed turbine disk</td>
<td>600 mm</td>
</tr>
<tr>
<td>Inner diam. of a disk</td>
<td>280 mm</td>
</tr>
<tr>
<td>Max. speed of rotation</td>
<td>11,500 rpm</td>
</tr>
<tr>
<td>Braking power at 9000 rpm</td>
<td>750 kW</td>
</tr>
<tr>
<td>Max. braking torque at 7000 rpm</td>
<td>1017 kNm</td>
</tr>
<tr>
<td>Water flow to the brake, max.</td>
<td>3 kg/s</td>
</tr>
</tbody>
</table>

Table 1.4-1: KTH test turbine data (Dahlqvist & Fridh, 2015, Sodergard et al., 1989).

1.4.2 Air System

At the external intake tube for outside air, the air system is fitted with air filters. The compressor itself consumes 1100 kW. The air cooler should be supplied with inlet water of 20°C or less, in order to cool an airflow of 4.7 kg/s and 180°C to 30°C according to design specifications. After the cooler and before the turbine inlet a condensate separator is installed.

<table>
<thead>
<tr>
<th>Maximum working pressure</th>
<th>4 bar = 400 kPa</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air flow, volume at atmospheric pressure</td>
<td>3.95 m³/s</td>
</tr>
<tr>
<td>Air, massflow</td>
<td>4.7 kg/s</td>
</tr>
<tr>
<td>Number of compressor stages</td>
<td>2</td>
</tr>
<tr>
<td>Power input to compressor shaft</td>
<td>968 kW</td>
</tr>
<tr>
<td>Power input at no load</td>
<td>366 kW</td>
</tr>
<tr>
<td>Air outlet temperature, full power</td>
<td>180°C</td>
</tr>
<tr>
<td>Cooling water consumption</td>
<td>2.3 kg/s</td>
</tr>
<tr>
<td>Sound pressure level at 1 m distance</td>
<td>85 dB(A)</td>
</tr>
<tr>
<td>Air cooler, cooling capacity</td>
<td>180°C - 30°C</td>
</tr>
<tr>
<td>Air cooler, water flow (20°C-40°C)</td>
<td>8.5 kg/s</td>
</tr>
<tr>
<td>Main air tube, inner diameter</td>
<td>300 mm</td>
</tr>
</tbody>
</table>

Table 1.4-2: KTH test turbine air supply system data (Sodergard et al., 1989).
1.4.3 Control and Measuring Systems

The desired loading point of study is set via a manual control unit rack in the control room and accompanying control-software. In essence, the load is set by defining the shaft speed and the opening of the main compressor bypass valves. In the turbine are mounted a number of sensors such as barometers, thermocouples and humidity sensors. These sensors are connected to a datascanner system, from which raw data is collected and converted using a Labview code, TURBAQ, and an Excel program, TURBAN (Mikaillian, 2012).
Chapter 2

Literature Review

2.1 Loss Models

Loss models are mathematical systems for estimating the aerodynamic performance of blade rows in turbomachines. In this chapter, the loss models implemented in the different turbomachinery simulation 1D software tools used in the present work are presented in summary.

2.1.1 Ainley/Mathieson

Ainley & Mathieson (1951) present a method for estimating the performance of an axial-flow turbine. In their method, the evaluation of the turbine losses are simplified by the common practice of considering the flow path at one common diameter throughout each stage of the turbine, termed the ‘reference diameter’. Furthermore, the authors assume that the pressure-loss coefficient in each blade row is uninfluenced by the gas Mach-number, and they also assume that the blade row gas outlet angle is uninfluenced by the gas incidence angle (Ainley & Mathieson, 1951, pp. 2-3).

Profile Loss

Ainley & Mathieson (1951, p. 3) define the pressure-loss coefficient \( Y \) as:

\[
Y = \frac{P_{01} - P_{02}}{P_{02} - P_2}.
\]  

(2.1-1)
The profile loss coefficients for nozzle-type blades \((\beta_1 = 0)\) and impulse-type blades \((\beta_1 = -\alpha_2)\) at zero incidence can be deduced from Fig. 2.1-1:

\[
Y_{p(i=0)} = \left\{ Y_{p(\beta_1=0)} + \left( \frac{\beta_1}{\alpha_2} \right)^2 \left[ Y_{p(\beta_1=-\alpha_2)} - Y_{p(\beta_1=0)} \right] \right\} \left( \frac{t/c}{0.2} \right)^{-\beta_1/\alpha_2}
\]  

\text{(2.1-2)}

\text{Figure 2.1-1: Profile loss coefficients for conventional section blades at zero incidence.} \quad (t/c = 20 \text{ per cent}; R_e = 2 \times 10^5; M < 0.6.) \text{(Reproduced from Ainley & Mathieson, 1951, Fig. 4).}
where $\beta_1$ is the chosen blade angle at inlet to the row, $\alpha_2$ is the chosen gas flow angle at outlet from the row, $t$ is the maximum blade thickness and $c$ is the blade chord. The values of $Y_p(\beta_1=0)$ and $Y_p(\beta_1=-\alpha_2)$ are determined from Fig. 2.1-1. When working with impulse type blades equation 2.1-2 is only valid in the range $0.15 < t/c < 0.25$, and values that fall outside of these limits should use either 0.15 or 0.25 for lesser and greater $t/c$ respectively.

**Figure 2.1-2:** Positive stalling incidences of cascades of turbine blades. ($R_e = 2 \times 10^5; M < 0.6$)(Reproduced from Ainley & Mathieson, 1951, Fig. 7).
Up till now, the profile loss has only been calculated for zero incidence angle $i$. In order to calculate the profile loss at incidence angles other than zero the stalling incidence $i_s$ of the blade must first be determined. The stalling incidence is defined as the angle at which the profile loss is twice that of zero incidence angle and can be deduced from Fig. 2.1-2 for the desired $\beta_1$ and $\alpha_2$. Consequently, knowing $Y_{p(i=0)}$ and having determined the associated stalling incidence $i_s$ of the specific blade, the profile loss $Y_p$ at the desired incidence angle $i$ can be obtained using Fig. 2.1-3 which plots the variation of profile loss with incidence based on the assumption that $Y_p/Y_{p(i=0)}$ is a function of $i/i_s$.

![Figure 2.1-3: Variation of profile loss with incidence for typical turbine blading (Reproduced from Ainley & Mathieson, 1951, Fig. 6).](image)

**Secondary and Tip Clearance Loss**

Secondary losses are expressed as a coefficient of the drag they cause on the blades:

$$C_{D_s} = \lambda C_L^2/(s/c)$$  \hspace{1cm} (2.1-3)

where $C_L$ is the lift coefficient based on vector mean velocity, $s$ is the blade pitch, $c$ is the blade chord and $\lambda$ is an empirical factor of secondary loss, primarily dependent on the degree of acceleration that the gas is subject to while flowing through the blade row. In the same manner,
the tip clearance losses are also expressed as a coefficient of the drag that they cause on the blades:

\[ C_{Dk} = B(k/h)C_L^2/(s/c) \]  \hspace{1cm} (2.1-4)

where \( B \) is a constant, \( k \) the radial tip clearance and \( h \) the annulus height.

In order to make the secondary-drag coefficient and the tip clearance-drag coefficient more convenient to implement, Ainley & Mathieson (1951, p. 5) derive an expression for the sum of these drag coefficients converted to loss coefficients:

\[ Y_s + Y_k = [\lambda + B(k/h)][C_L/(s/c)]^2[\cos^2 \alpha_2 / \cos^3 \alpha_m] \]  \hspace{1cm} (2.1-5)

where the constant \( B \) is equal to 0.5 for a row with radial tip clearance or 0.25 for a row with shroud seal, and where

\[ C_L/(s/c) = 2(\tan \alpha_1 - \tan \alpha_2) \cos \alpha_m, \]

and the mean flow angle \( \alpha_m \) is calculated as

\[ \alpha_m = \tan^{-1}[(\tan \alpha_1 + \tan \alpha_2)/2], \]

and

\[ \lambda = f(A_2/A_1)^2/(1 + I.D./O.D.), \]

\[ A_1 = A_{n0} \cos \beta_1, \]

\[ A_2 = A_{n2} \cos \alpha_2, \]

and where \( I.D. \) and \( O.D. \) are the constant inner- and outer-annulus diameters respectively. Hence, the parameter \( \lambda \) is a function of \((A_2/A_1)^2/(1 + I.D./O.D.)\), this is plotted in Fig. 2.1-4.

**Effect of Reynolds Number**

The loss coefficients developed by Ainley & Mathieson (1951, p. 10) are based on turbine data of an average Reynolds number of \( 2 \times 10^5 \). When making loss calculations based on these correlations the results obtained will therefore be treated as if they were of a mean Reynolds number of \( 2 \times 10^5 \). Thus, in case the mean Reynolds number differs significantly from this value a correction should be made to the original results. This correction is made by assuming that
the turbine overall efficiency is consistent with the following relationship:

\[(1 - \eta) \propto R_e^{-1/5}\] (2.1-6)

where \(\eta\) is the isentropic total head efficiency.

**Trailing Edge Correction**

All loss coefficient calculations made correspond to a blade trailing-edge thickness of 2% of blade pitch. When the ratio of these two parameters differs from this value a correction factor should be applied to the total-loss coefficient. This correction factor can be deduced from Fig. 2.1-5.

**Total Loss Coefficient**

The total loss coefficient is defined as the sum of the profile-, secondary- and tip clearance-losses:

\[Y_t = Y_p + Y_s + Y_k.\] (2.1-7)
2.1.2 Dunham/Came

Dunham & Came (1970) present a review and evaluation of the method developed by Ainley & Mathieson (1951) and propose a set of improvements to their work.

Profile Loss

The authors propose a correction factor for the profile loss $Y_p$ defined by Ainley & Mathieson (1951) in order to increase the precision of the results for Mach numbers higher than unity. The new profile loss coefficient is written:

$$Y_{p,corrected} = Y_p \times [1 + 60(M_n - 1)^2]$$  \hspace{1cm} (2.1-8)

where $M_n$ is the Mach number, and where the correction factor itself is only applied when $M_n > 1$. 

Figure 2.1-5: Effect of trailing-edge thickness on blade loss coefficients (Reproduced from Ainley & Mathieson, 1951, Fig. 9).
Secondary Loss

Dunham & Came (1970, p. 253) suggest a modification to the secondary loss formula of equation 2.1-5. In essence, the authors replace the empirical function $\lambda$ used by Ainley & Mathieson (1951) by a different empirical correlation, creating the following expression for the secondary loss:

$$Y_s = 0.0334 \left( \frac{c}{h} \right) \left( \frac{\cos \alpha_2}{\cos \beta_1} \right) Z.$$  

(2.1-9)

where $Z$ is the loading parameter introduced by Ainley & Mathieson (1951):

$$Z = \left( \frac{C_L}{s/c} \right)^2 \left( \frac{\cos^2 \alpha_2}{\cos^3 \alpha_m} \right).$$  

(2.1-10)

The numerical constant 0.0334 of equation 2.1-9 is meant to serve as an estimation for the effect of wall boundary layer dependence on blade shape, which in turn is dependent on the blade aspect ratio ($\frac{c}{h}$), while also compensating for the use of reference diameter blade loading values for the estimation of effects arising at the blade ends. The coefficient $\left( \frac{\cos \alpha_2}{\cos \beta_1} \right)$ corrects the loading parameter $Z$ introduced by Ainley & Mathieson (1951).

Tip Clearance Loss

The authors (Dunham & Came, 1970, p. 253-254) argue that the tip clearance loss, rather than varying linearly as the model by Ainley & Mathieson (1951) suggest, should vary as the tip clearance to the power of 0.78:

$$Y_k \propto k^{0.78}$$  

(2.1-11)

Consequently, the authors replace the linear tip clearance parameter $\left( \frac{k}{h} \right)$ of equation 2.1-5 by the exponential parameter $B \frac{c}{h} \left( \frac{k}{c} \right)^{0.78}$. Hence, the final expression of the tip clearance loss is:

$$Y_k = B \frac{c}{h} \left( \frac{k}{c} \right)^{0.78} Z$$  

(2.1-12)

where $B$ is the same numerical constant used by Ainley & Mathieson (1951), however with slightly altered values of 0.47 for plain tip clearance and 0.37 for row with shroud seal. If the rotor has a shroud seal the equivalent tip clearance $k$ should be calculated using the following equation:

$$k = \frac{\text{geometrical } k}{(\text{number of seals})^{0.42}}.$$  

(2.1-13)
Effect of Reynolds Number

Dunham & Came (1970, p. 253) present a modification to the Reynolds number correction of equation 2.1-6 defined by Ainley & Mathieson (1951). Equation 2.1-6 is presented by Dunham & Came (1970) in the following form:

\[
(1 - \eta_{corrected}) = (1 - \eta) \left( \frac{Re_{\text{mean}}}{2 \times 10^5} \right)^{-0.2}.
\]  

(2.1-14)

In their revision, the authors modify the original correlation in order to apply it directly to the profile and secondary losses, utilising the appropriate Reynolds number for the blade row in question instead of the mean Reynolds number of the medium flowing through the turbine:

\[
(Y_p + Y_s)_{corrected} = (Y_p + Y_s) \left( \frac{Re}{2 \times 10^5} \right)^{-0.2}.
\]  

(2.1-15)

Verification of Loss System Improvements

In their testing of 25 turbines, Dunham & Came (1970) found the original method by Ainley & Mathieson (1951) to produce estimated efficiencies within ±3% of reference data for typical aircraft engine turbines, while smaller and unusually designed turbines fell significantly outside of this range. By instead utilising their own revised method, the efficiency of the vast majority of turbines was estimated to within ±2% of the test efficiency. A comparison of the results is shown in Fig. 2.1-6.

2.1.3 Kacker/Okapuu

Kacker & Okapuu (1982) present a major revision to the loss prediction system developed by Ainley & Mathieson (1951) and Dunham & Came (1970), onwards referred to as the AMDC loss system. The main modification to the AMDC system introduced by Kacker & Okapuu (1982) is the addition of a trailing edge loss coefficient (in place of the trailing edge loss factor previously utilised), and higher precision Mach number-correction in the calculation of the profile- and secondary losses. In terms of the profile loss, shock-loss components have been added to the calculation of the profile loss coefficient, and a more elaborate Reynolds number-correction procedure has been applied.
Profile Loss

The expression for profile loss presented by the authors is based on the profile loss coefficient by Ainley & Mathieson (1951) and Dunham & Came (1970), as displayed in equation 2.1-2. Due to improvements in turbine blade design since the time of the work by Ainley & Mathieson (1951), the authors suggest that the original profile loss correlation should be multiplied by a factor of $\frac{2}{3}$ to achieve more accurate results when working with modern blades (Kacker & Okapuu, 1982, p. 113). Equation 2.1-16 shows this new profile loss coefficient with the original coefficient denoted by $Y_{p, AMDC}$.

$$Y_p = 0.914 \left( \frac{2}{3} Y_{p, AMDC} K_p + Y_{SHOCK} \right)$$  \hspace{1cm} (2.1-16)

where $K_p$ is a coefficient accounting for the influence of Mach number at rotor inlet ($M_1$) and outlet ($M_2$):

$$K_p = 1 - \left( \frac{M_1}{M_2} \right)^2 (1 - [1 - 1.25(M_2 - 0.2)])\).$$  \hspace{1cm} (2.1-17)

The coefficient $K_p$ of equation 2.1-17 compensates for the fact that the velocity change of the working fluid increases with higher subsonic Mach number due to compressibility effects, in turn suppressing local separations and thinning the boundary layers (Kacker & Okapuu, 1982,
Chapter 2. Literature Review

p. 113-114). The $Y_{SHOCK}$ component of equation 2.1-16 represents the subsonic shock loss:

$$Y_{SHOCK} = \frac{\Delta P}{q_1} \left( \frac{p_1}{p_2} \right) \left( \frac{1 - \left(1 + \frac{\gamma - 1}{2} M_1^2 \right)^{\frac{-\gamma}{\gamma - 1}}}{1 - \left(1 + \frac{\gamma - 1}{2} M_2^2 \right)^{\frac{-\gamma}{\gamma - 1}}} \right). \quad (2.1-18)$$

where $\Delta P$ is the total pressure difference, $q_1$ is the dynamic pressure at inlet, and $p_1$ and $p_2$ are the static pressure at rotor inlet and outlet respectively.

![Figure 2.1-7: Inlet Mach number ratio for nonfree-vortex turbine blades (Reproduced from Kacker & Okapuu, 1982, Fig. 6).](image)

Shock losses develop at the blade leading edge and vary along the span of the blade, with the most pronounced losses appearing at the inner endwall. This is due to the local Mach number always being higher at the endwall, a radial variation necessary for flow equilibrium in the rotor (see Fig. 2.1-7). Equation 2.1-18 takes this into account by the shock loss span wise-variation parameter:

$$\frac{\Delta P}{q_1}_{SHOCK} = \left( \frac{R_H}{R_T} \right) \left( \frac{\Delta P}{q_1} \right)_{HUB} \quad (2.1-19)$$

where $R_H$ and $R_T$ are the rotor hub and tip radius respectively. The variation of shock loss with Mach number at the inner endwall is illustrated in Fig. 2.1-8.

Secondary Loss

The secondary loss calculation remains largely unchanged from the model by Dunham & Came (1970) (see eq. 2.1-9). However, the dependence on blade aspect ratio has changed, and instead
of the function $f_{(AR)} = \left( \frac{h}{c} \right)$ the following correlation is used to better predict the losses at lower aspect ratios:

$$
\begin{align*}
    f_{(AR)} &= 1 - 0.25 \sqrt{\frac{2-h/c}{h/c}} & \text{for } h/c \leq 2 \\
    f_{(AR)} &= \frac{1}{h/c} & \text{for } h/c > 2
\end{align*}
$$

The final expression for the secondary loss is:

$$
Y_s = 1.2 Y_{s,AMDC} K_s
$$

where $1.2$ is a numerical constant that compensates for the separation of secondary and trailing edge loss of the original model by Ainley & Mathieson (1951), in which the total loss was multiplied by a trailing edge loss correction factor whereas in the new system the two are separate coefficients. $K_s$ is the subsonic Mach number correction factor:

$$
K_s = 1 - \left( \frac{1}{h/b_x} \right)^2 (1 - K_p)
$$

where $h$ is the blade height, $b_x$ is the axial chord, and $K_p$ is a Mach number correction coefficient defined in equation 2.1-17. The Mach number correction coefficient is necessary to account for
the increase in losses close to the end wall, brought on by higher local acceleration in this area due to effects of compressibility (Kacker & Okapuu, 1982, p. 115).

**Tip Clearance Loss**

The tip clearance loss for unshrouded rotor blades is formulated:

\[
\frac{\Delta \eta}{\eta_0 \cos \alpha_2} \times \frac{R_{TIP}}{R_{MEAN}} = 0.93
\]

where \( \eta_0 \) is the total-to-total turbine efficiency at zero tip clearance, and \( \Delta \eta \) is the stage efficiency sought. However, the authors do not derive an expression for the tip clearance loss coefficient.

For tip leakage loss of shrouded blades, the AMDC system correlation (see equation 2.1-12) is utilised without modification (Kacker & Okapuu, 1982).

**Trailing Edge Loss**

Trailing edge losses are expressed in the form of an energy coefficient, \( \Delta \phi^2_{TET} \), which can be deduced from Fig. 2.1-9. The relationship shown in Fig. 2.1-9 is based on empirical measurements of profile boundary layers at blade trailing edge for entry nozzles and impulse blades. Intermittent values are interpolated in the same manner as the profile loss-coefficient in equation 2.1-2. The trailing edge kinetic energy coefficient is converted to the pressure loss coefficient according to:

\[
Y'_{TET} = \left[ 1 - \frac{2}{1 - \Delta \phi^2_{TET} - 1} \right]^{\frac{1}{\gamma - 1}} - 1
\]

\[
\gamma - 1
\]

\[
1 - \left( 1 + \frac{1 - \Delta \phi^2_{TET}}{2 M_2^2} \right)^{\frac{1}{\gamma - 1}} - 1
\]

\[
(2.1-24)
\]

**Effect of Reynolds Number**

The original Reynolds number correction coefficient of equation 2.1-15 is replaced by a coefficient that varies depending on the particular Reynolds number interval:

\[
f(Re) = \begin{cases} 
\left( \frac{Rec}{2 \times 10^5} \right)^{-0.4} & \text{for } Rec \leq 2 \times 10^5 \\
1.0 & \text{for } 2 \times 10^5 < Rec < 10^6 \\
\left( \frac{Rec}{10^6} \right)^{-0.2} & \text{for } Rec > 10^6 
\end{cases}
\]

\[
(2.1-25)
\]
Figure 2.1-9: Trailing edge loss [energy] coefficient correlated against the ratio of trailing edge thickness to throat opening (Reproduced from Kacker & Okapuu, 1982, Fig. 14).

As opposed to the AMDC loss system, the Reynolds number correction coefficient is only applied to the profile loss coefficient and not the secondary loss coefficient, since it is deemed unlikely by the authors that this factor would have any effect on any other losses but the profile loss (Kacker & Okapuu, 1982, p. 114).

Total Loss Coefficient

The above presented loss components can be described by a total loss coefficient, defined as:

\[ Y_T = Y_P f_{Re} + Y_S + Y_{TET} + Y_{TC}. \]  

(2.1-26)

In comparison, the original AMDC total loss coefficient has the form:

\[ Y_T = [(Y_P + Y_S) \text{REFAC} + Y_{TC}] Y_{TET}. \]  

(2.1-27)

Thus, the biggest difference between the two systems is the change from the trailing edge loss correction factor of the old loss system \( Y_{TET} \) to the inclusion of an independent trailing edge loss coefficient in the new loss system \( Y'_{TET} \). The second major change concerns the Reynolds number correction factor, which in the new loss system only applies to the profile loss coefficient.
and is denoted by $f_{Re}$, while in the old AMDC-system it is applied to both the profile and secondary loss coefficients and referred to here as REFAC (Kacker & Okapuu, 1982, p. 112).

**Verification of Loss System Improvements**

To verify their loss model, (Kacker & Okapuu, 1982, p. 117-118) collected empirical performance data for 33 turbines of different types. Comparing these data with the results of their own performance prediction model, most turbines fell within $\pm 1.5\%$ of the real efficiency. When using the old AMDC loss system the results were significantly worse however, mainly due to advances in blade profile design since the development of this performance prediction model. The system by Kacker & Okapuu (1982) used a factor of $\frac{2}{3}$ (see equation 2.1-16) applied to the profile loss of the AMDC system to correct for this phenomenon. Fig. 2.1-10 shows a comparison of experimental and predicted efficiency for the AMDC and KO loss systems.

![Figure 2.1-10: Comparison of predicted efficiency with experimental efficiency of 33 turbines (Reproduced from Kacker & Okapuu, 1982, Fig. 19 and 20).](image)

**2.1.4 Moustapha/Kacker**

The system by Moustapha & Kacker (1990) is a development of the loss models by Ainley & Mathieson (1951), Dunham & Came (1970), Kacker & Okapuu (1982) and Mukhtarov &
Krichakin (1969). The authors observe that the Ainley/Mathieson model's ability to predict turbine performance at off-design conditions is poor, as can be seen from Fig. 2.1-11, and revise the loss model by investigating the influence of the incidence angle on the accuracy of the off-design performance prediction.

**Figure 2.1-11:** Comparison of Ainley and Mathieson's correlation with turbine rig data (Reproduced from Moustapha et al., cited in Moustapha & Kacker, 1990, Fig. 5(b)).

**Profile Loss**

Based on previous studies by other authors, Moustapha & Kacker (1990, p. 272) conclude that the leading edge geometry is important to take into account when estimating the effect of incidence on the profile losses of turbine blades, as can be seen in Fig. 2.1-12. The Ainley/Mathieson model does not take blade leading edge shape into account when calculating profile losses, and thus their correlation is probably valid only for certain blade types of similar leading edge geometry (Moustapha & Kacker, 1990, p. 272). Their new profile loss correlation in the
form of a kinetic energy loss coefficient ($\phi^2$) is:

$$\Delta \phi^2_p = 0.778 \times 10^{-5} \chi' + 0.56 \times 10^{-7} \chi'^2 + 0.4 \times 10^{-10} \chi'^3 + 2.054 \times 10^{-19} \chi'^6$$

$$\Delta \phi^2_p = -5.1734 \times 10^{-6} \chi' + 7.6902 \times 10^{-9} \chi'^2$$

$$0 > \chi' > -800$$

where

$$\chi' = \left( \frac{d}{s} \right)^{-1.6} \left( \frac{\cos \beta_1}{\cos \beta_2} \right)^{-2} \left[ \alpha_1 - \alpha_{1\text{ (des)}} \right]$$

(2.1-29)

and $d$ is the leading edge diameter of the blade, and $\alpha_{1\text{ (des)}}$ is the design inlet gas angle.

Figure 2.1-12: Effect of incidence on profile losses (Reproduced from Moustapha & Kacker, 1990, Fig. 6).

Secondary Loss

In Fig. 2.1-13 the authors plot the turbine efficiency prediction dependence on incidence angle based on measurement data from multiple sources. The authors note that the Ainley/Mathieson model assumes that secondary losses are constant at incidence to stalling incidence ratios greater than unity ($i/i_s > 1$), causing underestimations for low aspect ratio cascades when operated at large positive incidence (Moustapha & Kacker, 1990, p. 273). The revised secondary loss
correlation takes the effect of the leading edge geometry into consideration, and is otherwise based mainly on loss correlations developed by Mukhtarov & Krichakin (1969) (Moustapha & Kacker, 1990). The new secondary loss correlation is:

\[
\left( \frac{Y_{des}}{Y_{des, s}} \right) = \begin{cases} 
\exp(0.9 \chi'' + 13 \chi''^2 + 400 \chi''^4) & 0.3 > \chi'' > 0 \\
\exp(0.9 \chi'') & 0 > \chi'' > -0.4 
\end{cases} 
\]

(2.1-30)

where

\[
\chi'' = \frac{\alpha_1 - \beta_1}{180 - (\beta_1 + \beta_2)} \left( \frac{\cos \beta_1}{\cos \beta_2} \right)^{1.5} \left( \frac{d}{c} \right)^{-0.3}
\]

(2.1-31)

and \( Y_{des} \) is the loss coefficient at design inlet gas angle.

\[ \begin{array}{c}
\text{[Y/Y(des)], measured} \\
\text{[\alpha_1 - \beta_1], degrees} \\
\end{array} \]

Figure 2.1-13: Effect of incidence on secondary losses (Reproduced from Moustapha & Kacker, 1990, Fig. 7).

**Verification of Loss System Improvements**

The improvements to profile and secondary loss prediction with the Moustapha/Kacker model compared to the Ainley/Mathieson model can be seen in Fig. 2.1-14 and Fig. 2.1-15 respectively. It is noted by the authors that there are a number of other factors besides from leading edge
geometry and blade channel acceleration which are also likely to have an effect on the incidence loss. These factors include the Mach number, Reynolds number, turbulence level, leading edge wedge angle, exit flow angle and axial loading distribution, however, due to insufficient data available concerning these factors, they were all excluded from the loss correlations (Moustapha & Kacker, 1990, p. 274).

**Figure 2.1-14**: Evaluation of Ainley and Mathieson profile incidence loss correlations (left) and improved profile incidence loss correlation (right) (Reproduced from Moustapha & Kacker, 1990, Fig. 8 and 11).

**Figure 2.1-15**: Evaluation of Ainley and Mathieson secondary incidence loss correlation (left) and improved secondary incidence loss correlation (right) (Reproduced from Moustapha & Kacker, 1990, Fig. 12 and 15).
2.1.5 Benner/Sjolander/Moustapha

Benner et al. (1995) have revised the profile loss correlation of previous work by Ainley & Mathieson (1951), Dunham & Came (1970), Kacker & Okapuu (1982) and Moustapha & Kacker (1990). By conducting cascade measurements on two turbine blades of different leading-edge geometries and varying incidence angle the authors could observe great limitations in the prediction accuracy of previous models in regard to the profile losses. Their new profile loss model incorporates the wedge angle into the correlation for profile losses instead of simply being a function of the leading edge diameter as was the case with the earlier model. The reasoning of the authors for incorporating the wedge angle is their observation that the it gives a good indication of the magnitude of discontinuity in curvature at the points where the circle of the leading edge joins the main profile. This discontinuity, they believe, influences the off-design behaviour of the blade (Benner et al., 1995, p. 1).

Profile Loss Coefficient

The incidence parameter $\chi$ used to predict profile loss is a further development of the one presented by Moustapha & Kacker (1990) (see equation 2.1-29), however, the new correlation introduces the dependence on wedge angle and lessens the influence of the leading edge diameter (Benner et al., 1995, p. 8):

$$\chi' = \left(\frac{d}{s}\right)^{-0.05} W e^{-0.2} \left(\frac{\cos \beta_1}{\cos \beta_2}\right)^{-1.4} [\alpha_1 - \alpha_{1(des)}].$$

(2.1-32)

The new profile loss correlation has been obtained by fitting polynomials to the data, resulting in the following expression:

$$\Delta \phi^2_p = 3.711 \times 10^{-7} \chi^8 - 5.318 \times 10^{-6} \chi^7 + 1.106 \times 10^{-5} \chi^6$$
$$+ 9.017 \times 10^{-5} \chi^5 - 1.542 \times 10^{-4} \chi^4 - 2.506 \times 10^{-4} \chi^3$$
$$+ 1.327 \times 10^{-3} \chi^2 - 6.149 \times 10^{-5} \chi$$

for $\chi \geq 0$

(2.1-33)

$$\Delta \phi^2_p = 1.358 \times 10^{-4} \chi^2 - 8.720 \times 10^{-4} \chi$$

for $\chi < 0$
The kinetic-energy coefficient ($\phi^2$) can be converted to a more commonly used pressure loss coefficient using the conversion formula:

$$Y = \frac{\left[1 - \frac{\gamma - 1}{2} M_2^2 \left(\frac{1}{\phi^2} - 1\right)\right]^{-\frac{1}{\gamma - 1}} - 1}{1 - \left(1 + \frac{\gamma - 1}{2} M_2^2\right)^{-\frac{1}{\gamma - 1}}}.$$  

(2.1-34)

Verification of Loss System Improvements

The improvements to profile loss prediction accuracy over the MK model is illustrated in Fig. 2.1-16.

![Figure 2.1-16: Evaluation of the profile loss correlation of Moustapha & Kacker (1990) (left) and the new correlation by Benner et al. (1995) (right) at off-design incidence (Reproduced from Benner et al., 1995, Figs. 10, 12).](image)

2.1.6 Craig/Cox

Craig & Cox (1970) present a method for performance estimation of axial steam and gas turbines. The authors have analysed over 100 cascade tests in order to form their correlation, which is based on velocity coefficients dependent on: Reynolds number, aspect ratio, blade angles and passage geometry, pitch to backbone length ratio, Mach number and incidence (Craig & Cox, 1970, p. 411).
Effect of Reynolds Number

In this method, the Reynolds number is of big importance for the performance prediction. Utilising a Reynolds number based on blade opening, the authors have formulated a correlation of profile loss dependence on Re in which surface roughness plays an important role (Craig & Cox, 1970, p. 411). The correlation is shown in Fig. 2.1-17 for a range of standard finish surface roughness.

![Figure 2.1-17: Profile loss ratio against Reynolds number effect (Reproduced from Craig & Cox, 1970, Fig. 3).](image)

Profile Loss

The basic profile loss for subsonic flow is a function of the lift parameter $F_L$ times blade pitch divided by blade backbone length ($F_L(s/b)$) and the blade passage contraction ratio. The contraction ratio is defined as the difference between throat width and blade inlet width. The lift parameter is read from Fig. 2.1-18, and the profile loss from Fig. 2.1-19. The profile loss also needs to be corrected for factors such as Reynolds number effect and incidence, and additional loss contributions from e.g. blade back radius and Mach number added. These factors are also obtained from diagrams in the report by Craig & Cox (1970, p. 411-416).
Chapter 2. Literature Review

The full profile loss correlation is formulated:

\[ X_p = x_{pb}N_{pr}N_{pi}N_{pt} + (\Delta x_p)_t + (\Delta x_p)_{s/e} + (\Delta x_p)_m \]  \hspace{0.5cm} (2.1-35)

where \( N_{pr} \) is the correction factor for Reynolds number effect, \( N_{pi} \) is the correction factor for incidence, \( N_{pt} \) is the trailing edge thickness correction factor and the three last terms \((\Delta x_p)_t\), \((\Delta x_p)_{s/e}\), \((\Delta x_p)_m\) represent the profile loss increment from trailing edge thickness, blade back radius and Mach number respectively.

Secondary Loss

The authors have assumed that the secondary loss is approximately inversely proportional to the aspect ratio when formulating their secondary loss correlation (Craig & Cox, 1970, p. 416). The secondary loss correlation is formulated:

\[ X_s = (N_s)_{r}(N_s)_{h/b}(x_s)_b \]  \hspace{0.5cm} (2.1-36)

where \((x_s)_b\) is the basic secondary loss factor and \((N_s)_{r}\) and \((N_s)_{h/b}\) are the correction factors for Reynolds number effect and aspect ratio effect respectively. The aspect ratio factor is shown in Fig. 2.1-20 and the basic loss factor is shown in Fig. 2.1-21.
Annulus and Cavity Loss

In order to account for losses from diffusion between adjacent stages and losses from wall cavities between stator and rotor an overall annulus loss factor $X_a$ has been developed (Craig & Cox, 1970, p. 417). The overall annulus loss factor is the sum of the annulus loss factor $X_{a1}$, dependent on the overall area ratio from inlet to outlet, and the cavity loss factors $X_{a2}$ and $X_{a3}$, dependent on the geometric dimensions of the cavity and the stage.

Tip Clearance Loss

The tip clearance loss correlation has been developed for shrouded blades, with correction coefficients being utilised for unshrouded blades. The efficiency loss from leakage flow is dependent on the lap, as defined in Fig. 2.1-22. Positive lap has been observed to reduce the effects of leakage, and an optimum lap exists for any given clearance. The reason for the beneficial effects of positive lap on leakage flow is the decrease in static pressure that it produces after the lap itself. The correlation for leakage loss only applies to cases where a flow disturbance precedes the seals, and in cases where the inlet flow to the runner blades would be uniform, the seal area...
must be multiplied by 1.5 in order to increase the effective discharge coefficient. The efficiency loss for unshrouded blades is approximately the same as for shrouded blades in conditions of close to constant axial velocity across the blade row and relative velocities that are far less than sonic. However, at the rear of low pressure cylinders, where unshrouded blades are commonly found in steam turbines, these conditions do not apply and a factor of 1.5 should be applied to the loss for an equal shrouded blade (Craig & Cox, 1970, pp. 418-419).

Figure 2.1-22: Shrouded efficiency loss (Reproduced from Craig & Cox, 1970, Fig. 21).

2.2 Deviation Models

Ainley/Mathieson

Ainley & Mathieson (1951, pp. 3-4) present two correlations for gas outlet angle, dependent on the Mach number at outlet. For Mach numbers $0 < M_2 < 0.5$ the correlation is:

$$
\alpha_2 = f(\cos^{-1}o/s) - 4(s/c)
$$

(2.2-1)
where $e$ is the blade back curvature, $s$ is the blade pitch, $o$ is the throat and the value of $f(\cos^{-1}o/s)$ is read from Fig. 2.2-1.

For a Mach number of unity at the outlet, the outlet gas angle is given as:

$$\alpha_2 = -\cos^{-1}(A_t/A_{n2})$$  \hspace{1cm} (2.2-2)

where $A_t$ is the passage throat area and $A_{n2}$ is the annulus area downstream of the blade row.

Islam & Sjolander (1999, p. 1) state that the outlet angle correlation at unity given by Ainley & Mathieson (1951) is equal to the gauge angle and may be written as:

$$\alpha_2 = \cos^{-1}(o/s).$$  \hspace{1cm} (2.2-3)

![Figure 2.2-1: Relationship between gas outlet angles and $\cos^{-1}(o/s)$ for straight-backed blades operating at low Mach numbers (Reproduced from Ainley & Mathieson, 1951, Fig. 5).](image)

Islam/Sjolander

The deviation model by Islam & Sjolander (1999, p. 5) was developed under the assumption that the deviation is determined by the blade loading towards the trailing edge. After forming a
functional correlation based on the relevant blade row parameters affecting the blade loading, the unknown coefficients were established using a database of deviation values and an optimisation algorithm. The resulting deviation correlation is (Islam & Sjolander, 1999, p. 5):

$$\delta = \frac{(AVDR)^3}{\xi^{1.45}} \left( \frac{s}{c} \right)^{1.1} (\alpha_1 + \beta_2)^{2.25} \xi^{1.45} \left( \frac{t_m}{c} \right)^{0.3} (22 + 0.22\beta_1^{1.64})$$  \hspace{1cm} (2.2-4)

where

- $\delta$ = flow deviation (in degrees)
- $\alpha_1$ = inlet flow angle (in degrees)
- $\beta_2$ = outlet metal angle (in degrees)
- $s/c$ = blade space-to-chord ratio
- $t_m/c$ = blade maximum thickness-to-chord ratio
- $\xi$ = stagger angle (in degrees).

**McDonald**

The model by McDonald is available in the TML software by GKN Aerospace, however, no source for this correlation could be found.

**Sheglyaev**

The model by Sheglyaev is available in the AXIAL software by Concepts NREC, however, no source for this correlation could be found.
Chapter 3

Turbine Simulation Setup

3.1 KTH Test Turbine Stage 4b

In this section, the geometrical representation of the KTH Test Turbine stage 4b in the 1D meanline software tools will be presented. The stage can be favourably used in 1D calculations due to its conventional blade and annulus design, the stage does however have a few complex geometrical features which will also be described in this chapter. The details of the KTH Test Turbine Facility were previously presented in Chapter 1.4.

3.1.1 Stage Geometry

For the stage representation in AXIAL by Concepts NREC, the geometry constituting the computational domain is set up by defining the dimensions of the stator and rotor blades axially and radially at hub and tip, as well as the midspan axial spacing of the blade rows at LE and TE and the axial cavity width between stator and rotor discs. In TML by GKN Aerospace, the geometry of each axial plane of the passage is defined independently, rather than as part of the stator and rotor geometry as in AXIAL. Thus, for each axial plane of the passage the type of plane is defined (being either a duct plane, LE plane or TE plane). Additionally, the axial positions of the planes from inlet, and the hub and tip radii are defined. Although the input-interfaces differ, the resulting 2D geometrical model is the same for both of the two software tools as seen in Fig. 3.1-1.
3.1.2 Stator and Rotor Geometries

The stator and rotor of the stage are of conventional design and feature un-twisted spanwise straight contours. The stator is slightly convergent and the rotor slightly divergent at the outer endwall, and the stator features a slight lean angle towards the pressure side surface of the blade. The constant geometry at hub, midspan and tip makes this type of blades well suited for 1D calculations. The majority of the geometrical blade data used in the simulations were acquired from measurements of Fig. 3.1-2 which shows a radial profile view of the stator and rotor. A summary of the main geometrical features of the blades is displayed in Table 3.1-1, with additional data available in Appendix A.1. Due to reasons of confidentiality the detailed blade specifications cannot be listed in this report.
Chapter 3. Turbine Experimental Setup

Figure 3.1-2: KTH Test Turbine stator and rotor geometries.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Stator</th>
<th>Rotor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hub radius [mm]</td>
<td>177.5</td>
<td>177.5</td>
</tr>
<tr>
<td>Span at LE ($h_{LE}$) [mm]</td>
<td>34.0</td>
<td>27.5</td>
</tr>
<tr>
<td>Span at TE ($h_{TE}$) [mm]</td>
<td>26.5</td>
<td>34.1</td>
</tr>
<tr>
<td>Nominal flow turning [$^\circ$]</td>
<td>108.1</td>
<td>128.7</td>
</tr>
<tr>
<td>Blade LE lean angle [$^\circ$]</td>
<td>12.2</td>
<td>0.0</td>
</tr>
<tr>
<td>Type of tip [-]</td>
<td>-</td>
<td>shrouded</td>
</tr>
<tr>
<td>Number of tip seals [-]</td>
<td>-</td>
<td>4</td>
</tr>
<tr>
<td>Tip seal clearance [mm]</td>
<td>-</td>
<td>0.5</td>
</tr>
<tr>
<td>Number of blades</td>
<td>42</td>
<td>58</td>
</tr>
<tr>
<td>Chord ($C$) [mm]</td>
<td>34.7</td>
<td>26.6</td>
</tr>
<tr>
<td>Aspect ratio ($h_{TE}/C$) [mm]</td>
<td>0.76</td>
<td>1.28</td>
</tr>
</tbody>
</table>

Table 3.1-1: Parameters of stator and rotor in the KTH Test Turbine stage 4b.

3.1.3 Stator Lean Angle

The stator blades of the turbine stage are slightly inclined towards their pressure side surfaces. The concept of blades tilting tangentially away from the radial stacking line is commonly referred to simply as ‘lean’. In low aspect ratio turbines, the use of stator lean can reduce secondary losses and improve inlet flow (Harrison, 1992). Blade lean is commonly used to alter the radial pressure gradient of the flow over an airfoil, and to increase blade reaction at the hub of low pressure ratio blades (Bennet, 2002). Blade lean increases the pressure at the hub endwall,
which counteracts the radial pressure gradient seen in unleaned blades and thus suppresses radial secondary velocities that cause accumulation of low-momentum fluid at the hub, which in turn could lead to boundary layer separation (Harrison, 1992). Stator lean is also used as a compliment to stator sweep to minimise the noise level of fan stages caused by rotor-stator wake interaction effects (Envia & Nallasamy, 1998).

Two types of lean exist referred to as simple and compound lean respectively (Fig. 3.1-3). Simple lean refers to a blade that is tilted tangentially from the radial stacking line, while compound lean is when the blade is tilted or bent at both ends. The stator of the KTH Test Turbine stage 4b used in the experiment and simulation utilises simple lean. An illustration of the stator blade row lean angle definition of the AXIAL software is seen in Fig. 3.1-4.

In addition to the type of lean, there also exists varying definitions for the angle itself. Due to the geometry of an airfoil, the relative lean angle will be different for every point on the surface. Hence, it is most common to measure the lean at either LE or TE locations, relative to the vertical centreline of hub. Although it is most common to measure the lean angle relative to the intersection of blade and hub, the practice of measuring at midspan and tip as well as using the mean of hub and tip relative lean angles is not unheard of. As can be seen from the Fig. 3.1-4, AXIAL uses the LE frame of reference for the lean angle parameter. It is important to be watchful of the lean angle definition since different points of reference can give greatly varying results. This fact is illustrated by Fig. 3.1-5, which shows the difference in calculated lean angle.
between the LE and TE approaches. These lean angles were calculated knowing the distance tangentially from the stator row vertical centreline to the LE and TE along the perimeter of the hub. As can be seen from the figure, the lean angle at TE is 4.1°, and the equivalent angle at LE is 12.2°, the latter of which should thus be used as parameter-input in the performance simulation in AXIAL.

Since the AXIAL software supports the use of lean angle geometry-input this parameter will be specified and used in the general performance simulation of stage 4b performed with this software. However, in order to evaluate the actual impact of the stator lean itself on the performance simulation in the AXIAL software, an additional comparison will be made between the simulated efficiency of the stage 4b both with and without lean angle. The change in radial pressure distribution resulting from stator lean will be analysed by comparing the simulated static pressure at hub, midspan and tip locations for the two cases. The stator lean angle in AXIAL is defined using the Lean_Angle_Tng parameter. Unfortunately, as will be seen in the results in Chapter 4, a small error was made when defining the lean angle in AXIAL, and the value used for the simulation was mistakenly set to 12.8° rather than the supposed 12.2°. However, in Chapter 4.4.5 this small discrepancy in effective lean angle is shown to have a negligible impact on the simulated performance.
Chapter 3. Turbine Experimental Setup

### 3.1.4 Stator Endwall Contouring

The stator of the turbine stage does not have a straight outer endwall, instead it features a contoured shape as seen in Fig. 3.1-6. While the shrouded tip of the rotor is straight and thus easy to describe geometrically in 1D-meanline software, the exact geometry of the contoured stator endwall is more complicated to account for in the 1D software. This is because the loss models which form the basis for the calculations of 1D software normally have not been developed to account for this type of geometry.

### Throat Area Correction

As is seen in Fig. 3.1-6, the outer endwall at casing is of a particular contoured shape. The 1D-meanline software AXIAL by Concepts NREC and TML by GKN Aerospace only take into account the blade dimensions at LE and TE, with stations at hub, mean, and tip. Fig. 3.1-7 illustrates the geometry input in AXIAL by Concepts NREC, and Fig. 3.1-8 illustrates the...
effect these geometry limitations have when applied to the evaluated geometry of the stator in the 1D software.

As can be clearly seen from Fig. 3.1-8, the real area of the airfoil and flow passage is greater than that being evaluated by the software. The discrepancy in flow passage area can be corrected for by using a coefficient for the calculated throat passage area, an area which is greater in reality than the resulting throat area of the model. The effect of the contoured endwall on flow features along airfoil and endwalls can be corrected for by applying coefficients to the loss components which are affected by this special geometrical feature.
To compensate for the throat area discrepancy a correction coefficient was calculated. By creating a schematic of two adjacent stator blades and locating the approximate location of the throat it was possible to estimate the difference in real throat area and the throat area of the model. The throat at midspan was determined to be located approximately 6.4 mm axially downstream of the TE, as seen in Fig. 3.1-9. As also can be seen from this figure, due to the high outlet blade angle of the stator (74.6°) the throat plane is relatively close to being parallel to axial. Thanks to this, it was reasonable to make the simplification to consider the throat plane as if it were completely parallel to axial. This eliminated the need for complex 3D-calculations since the necessary measurements could be obtained from the standard profile view drawings of the stage. In the profile view of the stator row in Fig. 3.1-10, the height of the blade passage at the point of the estimated throat location is visible. Thus, the height of the throat at its highest point is 1.7 mm greater than that of the blade passage model in the 1D software.

In Fig. 3.1-11 an attempt is made to illustrate the actual throat area plane between the TE of two stator blades. The resulting coefficient for throat area correction, referred to as
Coef\text{\_Area\_Throat\_Scale} in the AXIAL software, was calculated as:

\[ C_{\text{Area\_Throat}} = \frac{\text{Throat area of model}}{\text{Real throat area}} = \frac{(26.5 + \frac{1.0695 + 1.0889}{2} \times \frac{2.0 + 1.7}{2}) \times o(m)}{(26.5 + \frac{1.0695 + 1.0799}{2} \times \frac{2.0}{2}) \times o(m)} \approx 1.033 \]

![Diagram of stator throat plane](image)

**Figure 3.1-11:** Illustration of stator throat plane, simplified as if it were completely axial, with the area unaccounted for due to the contoured endwall geometry marked in yellow.

### Secondary Loss Correction

While the difference in airfoil area between the model and the real blade could potentially have an impact on profile loss estimation, studies suggest that the biggest impact of a contoured endwall is related to its effect on secondary flows. Secondary flow fields are the cause of high losses in turbine stages of low blade height, and this is due to the large portion that these flows occupy in the flow channel because of its limited height, causing them to interact at and around midspan (Ewen et al., 1973). By utilising a contoured vane endwall, the pitchwise pressure gradient in the high turning section of the blade row is reduced, which in turn minimizes the
potential for secondary flows (Ewen et al., 1973). Langston (2001) summarizes research results concerning the effect of endwall contouring, which indicate that a contoured outer endwall can increase the overall efficiency of a turbine, with the biggest reduction in losses occurring at the non-contoured endwall.

Due to the inability of 1D-meanline numerical models to take endwall contouring into account, an attempt was made in the present work to reflect its effects on the real flow through the blade row by modifying the secondary loss coefficient of the stator of the KTH stage 4b. The study by Ewen et al. (1973) found that the vane row loss reduction from the introduction of a contoured endwall was 6%, while a literature survey by Langston (2001) found studies that estimated the loss to 17% or similar. However, as a best approximation of the contoured endwall effect, the secondary loss coefficient $Coef_{Loss}$ in AXIAL by Concepts NREC was reduced to 80%, a value reportedly used by Siemens (SIT) for these kinds of simulations (L Hedlund 2015, personal communication, 17 April).

The method of reducing the secondary loss coefficient of the stator to account for non-standard geometrical features is not new. In a study by Dubitsky et al. (2003), the software AXIAL by Concepts NREC was utilised to predict the performance of a number of turbine configurations. In one of the cases, an exhaust recovery low power turbine featuring a convergent-divergent bladed nozzle was modelled. Since a passage of this special type of nozzle has more or less zero flow turning, this was accounted for by reducing the secondary flow loss by 80% in the program.

### 3.1.5 Annulus Contraction Between Stator and Rotor

In the axial gap region between stator and rotor (Fig. 3.1-12) the geometry of the passage is not possible to account for in an accurate manner in the 1D-meanline software programs. The cause for this is the contraction of the flow channel slightly downstream of the stator TE, at which point the casing radius is marginally smaller (203.2 mm) than the stator TE radius (204 mm). The 1D-meanline programs can only describe the geometry of the stator-rotor axial gap at the hub and tip locations of stator TE and rotor LE, and consequently assume a linear annulus contour between these points.

As seen in Fig. 3.1-13, between the hub and tip radius stations (red dots) of stator TE and rotor LE the flow channel is assumed to extend linearly between these points (blue lines). Hence, the contoured midsection of the casing outer endwall (green line) extending into the modelled
flow channel is not taken into account by the numerical calculations by default. This introduces a discrepancy between experiment and 1D-meanline stage geometries and could thus impact the performance estimation accuracy of the software. The potential effect of the midsection contraction on flow features and ultimately the estimated efficiency of the stage will therefore be unknown unless a correction factor is introduced.

The AXIAL software can be adapted to allow for this type of area correction using the parameter referred to as $Coef_{Area\_Scale\_out}$. Although the parameter is originally intended for area correction at the point of the TE, it should still be able to account for the area contraction at midsection sufficiently well since it is only marginally downstream of this point. Figure 3.1-14 shows the geometry at the TE of the stator in the upstream direction, with contours of the hub at TE and at the contracted part of the midsection visible. Thus, the yellow area of the figure is the area that needed to be corrected for, and the coefficient was calculated as:

$$C_{AreaScaleTE} = \frac{\text{Passage area at contraction}}{\text{Passage area at TE}} = \frac{(26.5 - \frac{(1.0695+1.0653 \times 0.8)}{2}) \times s(m)}{26.5 \times s(m)} \approx 0.968$$
3.1.6 Purge Flow Through Stator-Rotor Cavity

In the cavity between stator and rotor it is possible to apply an airflow, known as purge. Without the existence of purge flow in the stator-rotor cavity, ingress of hot air from the main annulus would occur, increasing the risk of material damage to the inner annulus structure. The purpose of applying a purge flow stream is to create a temperature boundary between the hot main annulus and the inner annulus structure, as well as providing a means of cooling of the rotor blades. In the quest for increased efficiency, turbine inlet temperatures are ever increasing, and ensuring proper cooling of the stage components using secondary flows, such as purge, is thus of great importance (Dahlqvist & Fridh, 2015). The location of the stator-rotor cavity and incoming purge flow is illustrated in Fig. 3.1-15.

The estimation of purge flow losses in TML is based on the mixing model by Hartsel (1972), although it is unknown in which way this model has been adapted for simulation of purge flow.
for use in the TML software. The correlation by Hartsel is formulated as follows:

\[
\frac{\Delta P_t}{P_{tg}} = -\frac{\gamma}{2} M_g^2 \xi \left[ 1 + \frac{T_t}{T_{tg}} - 2 \frac{V_c}{V_g} \cos \phi_c \right]
\]  

(3.1-1)

where \(\Delta P_t\) is the total pressure change from blade inlet to outlet due to viscous effects, trailing edge blockage and coolant primary flow mixing, \(P_{tg}\) is the total pressure at local mainstream condition, \(\gamma\) is the ratio of specific heats, \(M_g\) is the mach number at local mainstream conditions, \(\xi\) is the ratio of coolant-to-mainstream mass flowrates, \(T_t\) and \(T_{tg}\) is the total temperature of coolant and temperature at local mainstream condition respectively, \(V_c\) and \(V_g\) is the velocity of coolant and velocity at local mainstream condition respectively and \(\cos \phi_c\) is the angle of injection of coolant from mainstream direction as depicted in Fig. 3.1-16.

More recently, Dahlqvist & Fridh (2015) have proposed a correlation for predicting purge flow performance based on the work by Denton (1993) and using entropy as a quantifier of losses. Dahlqvist & Fridh (2015) summarised the entropy increase from the mixing of a main flow with a smaller flow as

\[
\Delta s_{visc} = c_p(\kappa - 1) M_{2g}^2 \left( 1 - \frac{c_{2p} \cos \gamma}{c_{2g}} \right)
\]  

(3.1-2)

where \(c_p\) is the specific heat at constant pressure, \(\kappa\) is the ratio of specific heats, \(M_{2g}\) is the Mach
number of the main flow at vane outlet, \( c_{2g} \) and \( c_{2p} \) are the absolute velocity at vane outlet for the main flow and purge respectively and \( \gamma \) is the flow angle difference of the two streams. The authors relate the entropy increase of the flow mixing to a change in efficiency by assuming the process taking place at constant pressure and temperature, resulting in the expression

\[
\Delta \eta_{\text{entropy}} = \frac{\Delta s_{\text{visc}} T_{g2}}{\Delta h_{is,g}}
\]

(3.1-3)

where \( T_{g2} \) is the static temperature at vane exit and \( h_{is,g} \) is the isentropic enthalpy drop of the main flow. The results presented in the work by Dahlqvist & Fridh (2015) will be compared to the results of the purge flow simulations performed in this work in Chapter 4.

### 3.1.7 Experimental Reference Data

#### General Performance Simulation

The experimental results obtained from the KTH Test Turbine were not collected as part of this work, rather, they were acquired separately by the Heat and Power Division at KTH at an earlier occasion. Two sets of experimental data were utilised for the KTH Test Turbine simulations performed, representing the static-to-static design pressure ratio of 1.23 and the elevated pressure ratio of 2.06. These datasets were used for validation of the 1D simulations performed with the AXIAL and TML software in the present work, and can be found in Appendix B.1.1. The design point of the turbine is at a speed of 4450 for the design pressure ratio, corresponding to a VR of 0.48.
The flow inlet parameters of the KTH Test Turbine experiments are also presented in Appendix B.1.1. An average value of these flow inlet parameters were used as input for the 1D simulations. However, since the AXIAL and TML software do not use static-to-static pressure ratio as input, a conversion to total-to-static was needed. This was performed by dividing the mean static inlet pressure with the mean static-to-static pressure ratio to obtain the static pressure at stage outlet, and then dividing the mean total inlet pressure with this static outlet pressure:

$$PR_{ts} = \frac{P_{01m}}{PR_{ssm}}$$

The three final aero inlet parameters together with the operating range used for simulation are displayed in Table 3.1-2.

<table>
<thead>
<tr>
<th>$PR_{ts}$</th>
<th>$P_{01}$ [kPA]</th>
<th>$T_{01}$ [K]</th>
<th>Operat. Range [RPM]</th>
<th>Corresponding VR [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.236</td>
<td>124.14</td>
<td>302.40</td>
<td>500,1000,.....,10000</td>
<td>0.054,0.108,.....,1.08</td>
</tr>
<tr>
<td>2.077</td>
<td>209.97</td>
<td>334.63</td>
<td>943,1886,.....,18860</td>
<td>0.054,0.108,.....,1.08</td>
</tr>
</tbody>
</table>

**Purge Flow Simulation**

The simulation of purge flow impact on performance simulation relies on two sets of experimental data for validation representing the design pressure ratio and the elevated pressure ratio respectively, data collected by the Heat and Power Division at KTH. The complete set of experimental data obtained from the KTH Test Turbine used as a reference for validation of the simulated results of purge flow are available in Appendix B.1.2. The average flow inlet parameters of the KTH Test Turbine measurements used for purge flow simulation are presented in Table 3.1-3. The purge flow leakage fractions of the total flow, which were investigated in the experiment and simulated in the 1D software, are listed in Table 3.1-4.

<table>
<thead>
<tr>
<th>$PR_{ts}$</th>
<th>$P_{01}$ [kPA]</th>
<th>$T_{01}$ [K]</th>
<th>Operat. Range [RPM]</th>
<th>Corresponding VR [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.236</td>
<td>128.67</td>
<td>302.76</td>
<td>4380</td>
<td>0.47</td>
</tr>
<tr>
<td>2.085</td>
<td>212.81</td>
<td>326.38</td>
<td>10530</td>
<td>0.60</td>
</tr>
</tbody>
</table>
### Table 3.1-4: 1D purge flow mass fractions (MFR).

<table>
<thead>
<tr>
<th>$PR_{ts}$</th>
<th>Purge Flow MFR [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.236</td>
<td>0.00 0.98 2.03 4.93</td>
</tr>
<tr>
<td>2.085</td>
<td>0.00 1.11 2.10 2.71 3.52 4.22</td>
</tr>
</tbody>
</table>

### 3.2 KTH Test Turbine Legacy Stage

The legacy stage of the KTH Test Turbine is based on the stage described by (Wei, 2000). This stage is very similar in design to stage 4b which was previously described. As in the case of the aforementioned stage, the legacy stage features conventional blade designs and a simple flow channel. The majority of the geometry for this stage was deduced from Fig. 3.2-1. The geometrical 2D representation of the KTH Test Turbine legacy stage in the 1D meanline software tools is favoured by the straight endwalls at hub and casing.

#### 3.2.1 Stage Geometry

The blade passage is similar to that of stage 4b described earlier, however, the legacy stage does not feature any convergent or divergent sections. The exact geometry of the annulus upstream of the stator and downstream of the rotor is not known from the work by Wei (2000), and is assumed to be equal to that of stage 4b. The parameters of the stage are shown in Table 3.2-1.

### Table 3.2-1: KTH Test Turbine legacy stage 1 geometry.

<table>
<thead>
<tr>
<th>Feature</th>
<th>Inlet</th>
<th>Stator LE</th>
<th>Stator TE</th>
<th>Rotor LE</th>
<th>Rotor TE</th>
<th>Outlet</th>
</tr>
</thead>
<tbody>
<tr>
<td>Axial Position [mm]</td>
<td>0</td>
<td>11.3</td>
<td>36.1</td>
<td>48.1</td>
<td>72.8</td>
<td>74.8</td>
</tr>
<tr>
<td>Radial Pos. at Tip [mm]</td>
<td>202</td>
<td>202</td>
<td>202</td>
<td>202.5</td>
<td>202.5</td>
<td>202.5</td>
</tr>
<tr>
<td>Radial Pos. at Hub [mm]</td>
<td>178</td>
<td>178</td>
<td>178</td>
<td>177.5</td>
<td>177.5</td>
<td>177.5</td>
</tr>
</tbody>
</table>

#### 3.2.2 Stator and Rotor Geometries

The stator and rotor of the stage are of conventional design without twisting, and were assumed to have spanwise straight contours and straight and horizontal tip sections. The majority of the blade data were measured from Fig. 3.2-1, which shows a profile view of the stator and rotor. Table 3.2-2 displays the main geometrical features of the stage, with the complete set of data available in Appendix A.2.
Table 3.2-2: Parameters of stator and rotor in the KTH Test Turbine legacy stage.
3.2.3 Experimental Reference Data

The experimental reference data of the stage was obtained from the work by Wei (2000) by deducing the efficiency of stage 1 from a graph showing the efficiency over the velocity ratio range (see Fig. 4.1.1 of Wei (2000, p. 67)). Due to the lack of other available reference data in the report, only the efficiency was used for validation of the 1D simulations. The extracted experimental data is available in Appendix B2. Regarding the flow aero inlet parameters, only the pressure ratio used was stated in the report, and the inlet pressure and total temperature were not available. However, since the pressure ratio for the measurements taken were said to be between 1.18-1.23 it is reasonable to assume that the inlet total pressure and temperature were similar to those of the previously described reference data for stage 4b, which utilised a pressure ratio of 1.23. Thus, the performance of the legacy stage was simulated using the same inlet parameters as for stage 4b, as listed in Table 3.2-3.

Table 3.2-3: 1D simulation inlet aero parameters of the KTH Test Turbine legacy stage.

<table>
<thead>
<tr>
<th>$PR_{ts}$</th>
<th>Tot. Inl. Pre. [kPa] $P_{01}$</th>
<th>Tot. Inl. Temp. [K] $T_{01}$</th>
<th>Operat. Range [RPM]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.236</td>
<td>124.14</td>
<td>302.4</td>
<td>500,1000,...,10000</td>
</tr>
</tbody>
</table>

3.3 Turbine Stage by Ewen et al. (1973)

The stage by Ewen et al. (1973) is characterized by its very low aspect ratio. A complete set of data for the geometry of the stage was not listed directly in the work by Ewen et al. (1973) that it is based on, and therefore the majority of the parameters listed here have been acquired manually by taking measurements from available figures of the stage in the report. A schematic of the turbine rig is shown in Fig. 3.3-1.

3.3.1 Stage Geometry

The parameters of the blade passage of the turbine have been deduced from Fig. 3.3-1 and are listed listed in Table 3.3-1. As seen from the figure, the stage features straight endwalls throughout.
3.3.2 Stator and Rotor Geometries

The majority of the dimensions of the stator and rotor were measured from Fig. 3.3-2. A summary of the blade parameters is listed in Table 3.3-2, with the complete specifications available in Appendix A.3.

3.3.3 Experimental Reference Data

Reference data for the stage was obtained from Ewen et al. (1973, p. 328, Fig. 3) by extracting datapoints from a figure showing the stage efficiency at 4 pressure ratios over a range of velocity ratios. The efficiency data was the only data available, and thus mass flow or output power validations etc. could not be made for the 1D simulation of the performance of this turbine. The extracted experimental data is available in Appendix B.3. The flow inlet parameters specified
Figure 3.3-2: Stage by Ewen, Huber and Mitchell (1973) stator and rotor geometries (Reproduced from Ewen et al., 1973, Fig. 12).

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Stator</th>
<th>Rotor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hub radius at LE [mm]</td>
<td>110</td>
<td>110</td>
</tr>
<tr>
<td>Span at LE ( h_{LE} ) [mm]</td>
<td>16.5</td>
<td>16.5</td>
</tr>
<tr>
<td>Span at TE ( h_{TE} ) [mm]</td>
<td>16.5</td>
<td>16.5</td>
</tr>
<tr>
<td>Nominal flow turning (^\circ)</td>
<td>74.8</td>
<td>106</td>
</tr>
<tr>
<td>Blade lean angle (^\circ)</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>Radius of LE ( r_{LE} ) [mm]</td>
<td>2.28</td>
<td>0.99</td>
</tr>
<tr>
<td>Thickness of TE ( t_{te} ) [mm]</td>
<td>1.02</td>
<td>0.635</td>
</tr>
<tr>
<td>Type of tip [-]</td>
<td>-</td>
<td>unshrouded</td>
</tr>
<tr>
<td>Number of tip seals [-]</td>
<td>-</td>
<td>1</td>
</tr>
<tr>
<td>Tip seal clearance [mm]</td>
<td>-</td>
<td>0.254</td>
</tr>
<tr>
<td>Number of blades</td>
<td>20</td>
<td>44</td>
</tr>
<tr>
<td>Chord ( C ) [mm]</td>
<td>51.3</td>
<td>22.0</td>
</tr>
<tr>
<td>Axial chord ( C_x ) [mm]</td>
<td>30.5</td>
<td>16.5</td>
</tr>
<tr>
<td>Stagger angle (^\circ)</td>
<td>53.5</td>
<td>-41.4</td>
</tr>
<tr>
<td>Aspect ratio ( h_{TE}/C ) [mm]</td>
<td>0.32</td>
<td>0.75</td>
</tr>
<tr>
<td>Pitch at midspan LE (RMS radius) ( s ) [mm]</td>
<td>37.2</td>
<td>16.9</td>
</tr>
<tr>
<td>Maximum blade thickness ( t_{mx} ) [mm]</td>
<td>7.5</td>
<td>3.8</td>
</tr>
<tr>
<td>Stator-rotor spacing [mm]</td>
<td>15</td>
<td></td>
</tr>
</tbody>
</table>

Table 3.3-2: Parameters of stator and rotor in the turbine detailed by Ewen et al. (1973).
in the report by Ewen et al. (1973) were used for the 1D simulations, and the data is listed in Table 3.3-3.

<table>
<thead>
<tr>
<th>$PR_{\text{tot}}$</th>
<th>Tot. Inl. Pre. [kPa]</th>
<th>$P_{\text{01}}$</th>
<th>Tot. Inl. Temp. [K]</th>
<th>$T_{\text{01}}$</th>
<th>Operat. Range [RPM]</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.02</td>
<td>345</td>
<td>444</td>
<td></td>
<td></td>
<td>7350, 8350, ..., 26350</td>
</tr>
</tbody>
</table>

### Table 3.3-3: 1D simulation inlet aero parameters of the turbine described in Ewen et al. (1973).

#### 3.4 Setup of AXIAL by Concepts NREC

**General Simulation Setup**

AXIAL is a commercial software tool used for 1D design and evaluation of turbines and compressors. The program features a graphical user interface with illustrations of turbine configuration and velocity triangles of the stage. The setup of AXIAL was performed by first entering the geometrical properties of the stage in question (see for example Table 3.1-1 for the KTH Test Turbine stage 4b), along with the 3 input parameters governing the incoming flow: total inlet pressure, temperature and total-to-static pressure ratio (in this case being those of Table 3.1-2).

The simulation itself was carried out using an official Excel plugin application, which supplies the software in real-time with the desired input parameters and outputs the simulated results directly into the same spreadsheet upon running the application. Thus, the main parameter defined in this Excel plugin is the operating range, being a number of set speeds (RPM). The program then simulated the performance of the stage for parameters such as efficiency, mass flow, power and degree of reaction for the entire operating range, and the results were plotted over the velocity ratio range (VR) as defined in Chapter 1.2.2.

As previously detailed in Chapter 3.1, three special geometrical features of the KTH Test Turbine stage 4b were corrected for in AXIAL using coefficients. These three being the area at stator throat, the area at stator exit and the stator endwall contouring effect on secondary losses. This was done by setting the coefficients $\text{Coef\_Area\_Scale\_out}$, $\text{Coef\_AThroat\_Scale}$ and $\text{Coef\_sLoss}$ to 0.968, 1.033 and 0.8 respectively. The stator lean angle was also defined in the AXIAL software using the $\text{Lean\_Angle\_Tng}$ parameter.

AXIAL has a large number of loss models available to the user, which can be applied independently to stator and rotor. In this work, only the loss systems based on classical loss models
in literature were evaluated, and none of the proprietary loss models of Concepts NREC were used. However, the AXIAL software does provide an alternative deviation model by Sheglyaev in addition to the default deviation by Ainley/Mathieson (see Chapter 2.2) which was also investigated. Also, the software features two proprietary modified secondary loss (abbreviated MSL) models for use with the AMDC family of loss models. According to the AXIAL software, one of these models, referred to in this work as MK-MSL1, has a proprietary limit on the secondary losses, and the other, MK-MSL2, is based on the same limit but also incorporates a secondary loss factor of 0.375 apparently proposed by Denton in 2008. These two secondary loss models were evaluated using the standard MK loss model as the foundation, only changing the model for secondary loss.

The partial loss coefficients evaluated in the program were profile, secondary, trailing edge, leakage and incidence loss. Losses of shock, wetness, partial admission, shroud, and disk friction were omitted. The loss models evaluated along with corresponding deviation models are displayed in Table 3.4-1.

**Table 3.4-1:** AXIAL loss models, deviation models and their abbreviations as used in this report. AM represents the deviation model by Ainley and Mathieson described in Chapter 2.2.

<table>
<thead>
<tr>
<th>Loss Model</th>
<th>Deviation Model</th>
<th>Abbrev.</th>
<th>AXIAL Input</th>
</tr>
</thead>
<tbody>
<tr>
<td>Craig/Cox</td>
<td>AM</td>
<td>CC</td>
<td>CRAIG+COX</td>
</tr>
<tr>
<td>Benner/Sjolander/Moustapha</td>
<td>AM</td>
<td>BSM</td>
<td>AMDC+KO+MK+BSM</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>AM</td>
<td>MK</td>
<td>AMDC+KO+MK</td>
</tr>
<tr>
<td>Ainley/Mathieson, Dunham/-Came</td>
<td>AM</td>
<td>AMDC</td>
<td>AMDC_OLD</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>Sheglyaev</td>
<td>MK-S</td>
<td>AMDC+KO+MK/DEV_SHEGL</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>AM</td>
<td>MK-MSL1</td>
<td>AMDC+KO+MK/S_AMDC_LIM</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>AM</td>
<td>MK-MSL2</td>
<td>AMDC+KO+MK/S_AMDC_LIM_0375</td>
</tr>
</tbody>
</table>

Besides from the loss models themselves, the Reynolds number limit at which pressure and secondary loss become independent also had to be specified in AXIAL, a parameter denoted by $Re_{FixPLoss}$ in the software. The default value in AXIAL for this parameter is 1e30, indicating virtually no limit for loss independence. This default value was used for the simulations, however, the effect of instead utilising the nominal value for each specific loss model as defined in the original systems (AMDC-family=200 000 and CC=450 000) was investigated.
The roughness of the blades also had to be defined in AXIAL using the parameters \textit{RoughnessType} and \textit{Roughness}. The roughness tolerance according to stage design specification is a standard RMS roughness of 1.6 \( \mu \text{m} \), thus the real value could in theory be somewhat lower. However, the simulation of the impact of roughness on performance estimation performed in this work used the maximum allowable 1.6 \( \mu \text{m} \), in order to avoid underestimation of losses.

The throat width of the blade row was automatically calculated by the AXIAL software based on the geometry specified, this same throat width was consequently used also as input for the TML software, which explicitly required this input parameter. Thus, this procedure is a way of ensuring that both software use the same throat width for their simulations. This is of big importance since the throat width is one of the major parameters controlling the mass flow of the stage.

### 3.5 Setup of TML by GKN Aerospace

**General Simulation Setup**

TML is a non-commercial command-line software tool for 1D design and evaluation of turbines. Input to the program is made via simple text files, with one defining the geometry of the stage and its blades and another the input aero parameters and loss models used for performance simulation, as well as the desired speeds making up the operating range at which the program should perform the simulation. In order to facilitate and speed up the command-line-based execution of TML simulations when working with multiple loss models, a script in the form of a batch file was written. Running this program then automatically initiated the consecutive execution of multiple TML input-files via the command-line, and appended the resulting output data to a common results-file for easy import into Excel for further data processing.

The geometry of the annulus is modelled by defining the hub and tip stations of the seven axial planes previously described in Chapter 3.1.1. The aero input parameters are the same as are used with the AXIAL software and are defined in Table 3.1-2. The TML software simulated the performance of the stage for the same set of parameters as in AXIAL, being efficiency, mass flow, power and degree of reaction for the entire operating range, and the results were plotted over the velocity ratio range as defined in Chapter 1.2.2.
The TML software does not support the use of the same correction coefficients that the AXIAL software does. Therefore, the correction coefficients of KTH Test Turbine stage 4b detailed in Chapter 3.1 and applied in the simulation in AXIAL were not used in TML. Thus, no correction for stator throat area, stator outlet area, and the endwall contouring effect on secondary losses, was made in TML. Neither is it possible to define a lean angle of the blades using the TML software which is the case in the AXIAL software, and thus the effect of the stator lean angle could not be simulated in TML. Hence, the simulation in TML was set up using solely the basic blade data of the stages, without applying any coefficients to correct for non-standard geometry.

The input for loss model selection in TML comprises two variables known as MLOSS and MANGLE. MLOSS represents the actual loss model being used and MANGLE the deviation model. In TML only one loss model at a time can be used for the entire stage, and no partial loss models can be set to govern certain loss components as can be done in AXIAL. TML does not have any option to set the Reynolds number limit for loss independence as is possible in AXIAL, neither does TML support the input of blade roughness.

**Table 3.5-1:** TML loss models and deviation models used for simulation, where AM represents the deviation model by Ainley and Mathieson, and \( \cos^{-1}(At/An) \) and \( \cos^{-1}(o/s) \) are simplifications of the same model, all described in Chapter 2.2.

<table>
<thead>
<tr>
<th>Loss Model</th>
<th>Deviation Model</th>
<th>Abbrev.</th>
<th>TML Input MLOSS/MANGLE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kacker/Okapuu</td>
<td>AM</td>
<td>KO-A</td>
<td>2/2</td>
</tr>
<tr>
<td>Kacker/Okapuu</td>
<td>McDonald</td>
<td>KO-M</td>
<td>2/3</td>
</tr>
<tr>
<td>Kacker/Okapuu</td>
<td>Islam</td>
<td>KO-I</td>
<td>2/5</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>AM</td>
<td>MK-A</td>
<td>3/1</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>McDonald</td>
<td>MK-M</td>
<td>3/3</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>Islam</td>
<td>MK-I</td>
<td>3/5</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>( \cos^{-1}(At/An) )</td>
<td>MK-C1</td>
<td>3/6</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>( \cos^{-1}(o/s) )</td>
<td>MK-C2</td>
<td>3/2</td>
</tr>
<tr>
<td>Moustapha/Kacker with Volvo Tip Clearance Correction</td>
<td>( \cos^{-1}(o/s) )</td>
<td>MK-C2V</td>
<td>3/7</td>
</tr>
</tbody>
</table>

The TML software does not feature an input for the number of tip seals of shrouded blades, therefore it was necessary to first convert the multi-seal tip clearance to the equivalent radial clearance of a single tip (see Dunham & Came (1970)) before defining this parameter in TML. In the case of the KTH Test Turbine Stage 4b with its 4 tip seals of 0.5 mm clearance, the equivalent radial clearance is calculated as:

\[
k' = \frac{k}{\text{(number of seals)}^{0.42}} = \frac{0.5}{4^{0.42}} = 0.279 \text{ mm}
\]
Purge Flow Simulation

The setup of the purge flow simulation in TML is performed by defining the leakage mass fraction of the main flow. Additionally, it is necessary to define the total inlet temperature and pressure ratio of the incoming purge flow. The incoming purge flow was assumed to have the same total temperature as the main flow between stator and rotor where it enters the turbine. Since the total temperature is maintained constant through the stator, the incoming purge flow will have the same temperature as at the inlet to the turbine stage. The static pressure ratio of the incoming purge flow to the main flow was set to unity, since the secondary flow should not be affected by any pressure gradient in a real scenario. The loss models used for purge flow simulation are seen in Table 3.5-2.

<table>
<thead>
<tr>
<th>Loss Model</th>
<th>Deviation Model</th>
<th>Abbrev.</th>
<th>TML Input</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kacker/Okapuu</td>
<td>cos⁻¹(At/An)</td>
<td>KO-C1</td>
<td>2/6</td>
</tr>
<tr>
<td>Kacker/Okapuu</td>
<td>cos⁻¹(o/s)</td>
<td>KO-C2</td>
<td>2/2</td>
</tr>
<tr>
<td>Kacker/Okapuu with Volvo Tip Clearance Correction</td>
<td>cos⁻¹(o/s)</td>
<td>KO-C2V</td>
<td>2/7</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>cos⁻¹(At/An)</td>
<td>MK-C1</td>
<td>3/6</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>cos⁻¹(o/s)</td>
<td>MK-C2</td>
<td>3/2</td>
</tr>
<tr>
<td>Moustapha/Kacker with Volvo Tip Clearance Correction</td>
<td>cos⁻¹(o/s)</td>
<td>MK-C2V</td>
<td>3/7</td>
</tr>
</tbody>
</table>
Chapter 4

Results

In this chapter, the results of the 1D performance prediction experiments detailed in Chapter 3 will be presented. The simulation results relating to AXIAL loss models will be displayed with circle markers (○) in the diagrams and the TML models with diamond markers (♢).

The definition of the complete set of performance parameters presented can be found in Chapter 1.2.2. The majority of the results will be presented as a function of the isentropic velocity ratio, which is a non-dimensional constant relating the turbine rotor speed to the total-to-static isentropic enthalpy drop of the stage (Dahlqvist & Fridh, 2015, p. 1):

\[ \nu_{ts} = \frac{U}{\sqrt{2\Delta h_{ts,e}}} \]

4.1 General Performance Simulation

In this section the results of the 1D simulation of the KTH Test Turbine stage 4b are presented separately for the design pressure ratio and the elevated pressure ratio. Additionally, a comprehensive comparison between the two pressure ratios simulated is made. The setup of the AXIAL and TML software tools used to obtain the results presented in this chapter were detailed in Chapter 3.4 and 3.5 respectively.

In the setup of the AXIAL software, coefficients were applied to correct for the effects of the stator lean angle on throat area as well as secondary losses, as described in Chapter 3.1.4. The discrepancy between the real stator outlet area and that of the 1D model, due to the annulus contraction between stator and rotor, was corrected for using a coefficient as detailed in Chapter
3.1.5 which was applied in AXIAL. The RMS roughness and Reynolds number-limit used for
the simulation in AXIAL where specified according to the description in Chapter 3.4.

The TML software setup did not support any of the geometry correction coefficients, roughness,
or Reynolds number-limit parameters, which were applied in the AXIAL software as described
in the foregoing paragraph (see Chapter 3.5). Thus, the results from TML are based on the
default loss models without any loss corrections.

4.1.1 Design Pressure Ratio

The estimated performance parameters of the stage at design PR using the AXIAL and TML
software compared to experimental data are displayed in Fig. 4.1-1 to 4.1-8. Starting out with
the efficiency parameter, the results show that all seven loss models in AXIAL underestimated
the total to static efficiency of the stage (Fig. 4.1-1). Out of the standard models, the CC
model came closest to the experimental data at approximately 4% underestimation, while the
BSM, MK, MK-S and MK-MSL1 models show a greater and almost identical degree of under-
estimation. The MK-OLD model provided the worst results, at -12% error. The MK-MSL2
model with the modified secondary loss coefficient gave the best results in terms of estimated
efficiency.

The results of the TML simulation show efficiency underestimations of between 5-8% for the
entire set of loss models with MK-C1 giving the best results and KO-M the worst (Fig. 4.1-
2). The models based on KO display a curios trend at VR around 0.60, where they dip down
uncharacteristically and inexplicably for an efficiency curve of this type.

In terms of the mass flow estimation, all AXIAL loss models overestimated this parameter to a
similar degree, while the TML models gave varying results (Fig. 4.1-3). The TML loss models
reliant on McDonald and Islam deviation (KO-M, KO-I, MK-M, MK-I) provided overestima-
tions, while models using the AM deviation (KO-A, MK-A) gave underestimations, and the
remaining three models, reliant on simplified AM deviation models (MK-C1, MK-C2 and MK-
C2V), provided the best compliance with exp. data. One of the reasons for the difference in
estimated mass flow between the loss models of the two software could be that they differ in
their calculated throat areas. The throat area of the stator and rotor is the main geometrical
parameter controlling the mass flow of a stage, and it is calculated automatically in both AX-
IAL and TML based on the blade geometry data input. However, a look at the throat areas
calculated by AXIAL and TML reveals that they are almost identical, with the AXIAL software stator and rotor throats being less than 1/10% (8310 vs. 8307 mm$^2$) and 2/3% (14496 vs. 14415 mm$^2$) greater than the TML throats respectively. Thus, the less than 1% greater throat areas that the AXIAL loss models are subject to as compared to the loss models in TML cannot explain the over 5% higher mass flow of the majority of the AXIAL models as compared to several of the TML models. Instead, the difference between the mass flow estimates among the individual loss models as well as the two software is mainly related to differences in estimated axial flow velocity.

Although all AXIAL models overestimated the mass flow, the vast majority (except for CC and MK-MSL2) still underestimated the aero power (Fig. 4.1-4). In TML all models underestimated the power output at design point, with MK-I, MK-C1, KO-M and KO-I giving the most accurate results being within 4% of the experimental data (Fig. 4.1-5). However, since the stage efficiency, mass flow and power parameters are intrinsically linked, the fact that the power predictions of some models might line up well with the exp. data is most probably more due to coincidence than evidence of superiority. To understand the true accuracy of the prediction of energy transferred in the turbine throughflow process, it is necessary to review the total enthalpy drop, also known as stage work, as seen in Fig 4.1-6. This parameter is greatly underestimated by all loss models in both AXIAL and TML, and consequently, the loss models mentioned at the beginning of the paragraph which were shown to underestimate the power output in spite of overpredicted mass flows thus did so because of the great underestimation of the enthalpy drop of the stage.

On another note, it should be pointed out that the predicted power of the KO-based models in TML exhibit the previously mentioned anomaly at higher VR observed in the efficiency diagram, concerning a sudden dip in predicted values resulting in a peculiar graph shape. Of course, this was to be expected since the two parameters both relate to the stage enthalpy drop, however, this once again brings in to question the trustworthiness and validity of the results of these specific models.

The estimations of reaction degree based on enthalpy and pressure both show similar trends of underestimation, although all models are of reasonable accuracy at higher VR (Fig. 4.1-7 and 4.1-8). At the lower range of the operating interval however, all models diverge drastically towards negative. Clearly, this is not a reasonable estimation, rather it seems as though none of the loss models are able to correctly estimate these parameters at low turbine speeds and
corresponding VR. Since the reaction degree prediction is dependent on enthalpy and pressure estimations at several different axial points of the stage, which in themselves might be subject to a relatively high degree of uncertainty, it should not be surprising that this parameter could potentially be among the most difficult to compute accurately.

**Figure 4.1-1**: Normalised total-to-static efficiency at design PR (AXIAL).

**Figure 4.1-2**: Normalised total-to-static efficiency at design PR (TML).
Figure 4.1-3: Normalised mass flow at design PR. Normalised to the point of maximum experimental efficiency (4800 RPM, VR 0.51).

Figure 4.1-4: Aero power at design PR (AXIAL).

Figure 4.1-5: Aero power at design PR (TML).
Chapter 4. Results

Figure 4.1-6: Normalised stage work output ($\Delta h_{tt}$) at design PR.

Figure 4.1-7: Enthalpy based degree of reaction at design PR.

Figure 4.1-8: Pressure based degree of reaction at design PR.
4.1.2 Elevated Pressure Ratio

The estimated performance parameters of the KTH stage 4b at elevated PR are shown in Figs. 4.1-9 to 4.1-20. The AXIAL software estimation of total to static efficiency showed that, out of the main loss models, the CC model provided the best and most accurate results with a discrepancy of ≈-2% compared to exp. data at design speed (Fig. 4.1-9). The BSM, MK, MK-S and MK-MSL1 gave close to identical results with all graphs displayed practically on top of each other in the diagram, although these estimated efficiencies were slightly less accurate than that of the CC model. The AMDC model provided the most inaccurate results, with an underestimation of ≈7%. The MK-MSL2 model with its Denton modified secondary loss coefficient was the only model to overestimate the efficiency (≈ +1%), and this model also provided a very good curve shape when compared to exp. data, a better shape than that of the second closest model CC.

The efficiency estimation of TML at the elevated PR provided relatively good results for the majority of the models (Fig. 4.1-10 and 4.1-11). Firstly, it is evident that the KO-based models exhibit the same peculiar efficiency-curve shape as was the case with the estimations at design PR. The same three models (KO-A, KO-M, KO-I) were also the most inaccurate of the nine models at the peak efficiency VR, while the rest of the models, all based on MK, predicted the efficiency to within 2% of the exp. data at this operating point. The MK-A, MK-C1 and MK-C2 models provided the best accuracy, an underestimation of ≈1%, all of which are based on the original AM deviation model.

As previously discussed in Chapter 4.1.1 concerning the simulations at design PR, the difference in mass flow estimate between the two software and the different loss models is related to the discrepancy in predicted axial flow velocities of the stage. However, in contrast to the results at design PR, the estimated mass flow at elevated PR also shows signs of a developed choking condition, as seen in Fig. 4.1-12. All loss models in AXIAL overestimated the mass flow, and also predicted a constant mass flow for VR less than 0.50, indicating that the software predicted a choked stage due to supersonic flow. Contrarily, the entire range of loss models utilized in TML underpredicted the mass flow at low VR, with the majority of the models showing a constant mass flow indicating choking in the VR range of 0.30 – 0.50. However, four of the models, MK-A, MK-C1, MK-M and KO-M, show a significant increase in mass flow at VR 0.40 to 0.50, providing a mass flow prediction very close to exp. data at and above a VR of around 0.50. This sudden increase in estimated mass flow is probably due to an interruption
and consequent cessation of the predicted choking of the stage. This is supported by the results of Fig. 4.1-13, which shows the estimated Mach number of the different loss models in TML at the point of maximum flow velocity of the stage at the stator exit.

By correlating the results of Mach number in Fig. 4.1-13 and mass flow in Fig. 4.1-12, it is seen that as the Mach number of the four aforementioned models (MK-A, MK-C1, MK-M, KO-M) decreases below unity, the mass flow simultaneously shoots up at the corresponding VR. This is made even clearer in Fig. 4.1-14, where the Mach number distribution of the four models has been overlaid on top of the mass flow curves. Thus, this link between the decrease of Mach number into subsonic territory and the dramatic rise in mass flow is evidence that the four loss models in question were limited by estimated choking, and once no longer under this constraint they provided the most accurate mass flow predictions out of the range of models studied.

The aero power estimations in AXIAL showed good accordance with exp. data particularly for the BSM and MK based models (Fig. 4.1-15). The CC and MK-MSL2 model overestimated the aero power, while the AMDC model was the only one of the AXIAL models to underestimate this parameter. The results of the TML simulation of aero power correspond better with the estimated mass flows than was the case with the models in AXIAL (Fig. 4.1-16 and 4.1-17). The models that underestimated the mass flow also underestimated the power to a similar degree, and the four models restricted by choking (MK-A, MK-C1, MK-M, KO-M), which displayed a sudden increase in mass flow at the mid VR-range due to the cessation of said choking, also exhibit the same trend for the estimated power. The estimated power of these four models show a better fit to exp. data at VR above 0.50 than the rest of the models, just as was the case with the estimated mass flow. However, among these four models, the accuracy of the KO-based model (KO-M) with exp. data is substantially lower than for the three MK-based models in terms of the estimated aero power.

Given that many of the models had been shown to overestimate the mass flow it would have been reasonable to expect that the estimated aero power of the same models would have been greater than the exp. data as well. However, the fact that the estimated aero power is lower than expected can be explained by the fact that the estimated efficiency is lower than exp. data as well, and thus the estimated work of the stage must be less than in reality, resulting in underestimated aero power in spite of the overestimations of mass flow. The estimated stage work is compared to exp. data in Fig. 4.1-18. In this figure, the most accurate models, provided by the TML software, are shown to underestimate the enthalpy drop by roughly 2%. Although
this is a relatively large underestimation, it is considerably better than was the case at design PR, where the same models were off by closer to 6%. Thus, this discrepancy, together with the higher accuracy in estimated mass flow, explains why the TML models in questions provided substantially better efficiency and power predictions at the elevated PR.

The reaction degree predictions at elevated PR shown in Fig. 4.1-19 and 4.1-20 mirror the accuracy and trend seen at design PR for the majority of loss models of both AXIAL and TML. This parameter is plagued by underestimations throughout the VR range and at the high loading points, corresponding to low VR values, the estimated degree of reaction is especially heavily underestimated. As was mentioned in the previous chapter concerning the results of the reaction degree simulations at design PR this parameter might be one of the most difficult to accurately predict since it is dependent on enthalpy/pressure estimations at several axial locations from stator inlet to rotor outlet, which in themselves carry a great deal of uncertainty.

![Figure 4.1-9: Normalised total-to-static efficiency at elevated PR, enlarged view (AXIAL).](image-url)
Figure 4.1-10: Normalised total-to-static efficiency at elevated PR (TML).

Figure 4.1-11: Normalised total-to-static efficiency at elevated PR, enlarged view (TML).

Figure 4.1-12: Normalised mass flow at elevated PR. Normalised to the point of maximum experimental efficiency (9800 RPM, VR 0.55).
Chapter 4. Results

**Figure 4.1-13:** Mach number at stator exit, the point of the highest absolute velocity of the stage, for the loss models in TML at elevated PR.

**Figure 4.1-14:** Illustration of relation between mass flow and Mach number, with the jump in mass flow coinciding with the abs. velocity decreasing below Mach 1.

**Figure 4.1-15:** Aero power at elevated PR (AXIAL).
Figure 4.1-16: Aero power at elevated PR (TML).

Figure 4.1-17: Aero power at elevated PR, enlarged view (TML).

Figure 4.1-18: Normalised stage work output ($\Delta h_{tt}$) at elevated PR.
4.1.3 Effect of Pressure Ratio on Performance Estimation

Besides from directly comparing the estimated performance parameters to exp. data for the specific PR, another way of validating the loss models of the 1D software is to evaluate how well the difference between the estimated results at design PR and elevated PR concur with the real difference of the exp. data for the two cases. This section examines the relative difference of the estimated results at design PR and elevated PR, in an attempt to establish whether the behaviour of the models with varying PR is realistic.
Overall Performance Parameters

The change in the overall performance parameters with an increase in PR from design to elevated condition is shown in Figs. 4.1-21 to 4.1-25. Generally, the majority of loss models concur well with the exp. data for most parameters, with the exception of the estimated efficiency. According to the exp. data, there is practically no difference between the two pressure ratios in terms of efficiency, however, the vast majority of loss models predicted an increase in the estimated efficiency for the elevated PR over design PR of around 4-6% at the design VR of 0.48. Hence, the fact that the change in mass flow from increased PR agrees relatively well with reference data in spite of the efficiency parameter not concurring does suggest that the general geometry of the stage is correctly modelled, while the loss models themselves are not correctly predicting the effect of this geometry on aerodynamic losses when working at low PR. Consequently, for the exp. data, the change in estimated power from elevation of PR is entirely due to the estimated increase in mass flow and not due to increased efficiency, whereas the 1D simulations incorrectly suggest that both of these factor would contribute to the increased power output.

Interestingly, the CC model estimated a significantly smaller increase in efficiency from PR elevation than the AMDC-based models of both AXIAL and TML, suggesting that special loss scheme of this model might be better configured for low PR, and thereby does not overestimating the impact of PR on airfoil performance to the same extent.

The diagrams of mass flow, aero power and reaction degree show a general trend of underestimation for the TML models using McDonald and Islam deviation models (KO-M, KO-I, MK-M, MK-I), especially in the low VR range. The MK-A model in TML is the only one to overestimate the parameter change associated with the elevation of PR for all of the parameters. The MK-A model along with MK-C1, MK-M and MK-M, reflect the rise in estimated mass flow and power at and around the design speed seen in the results for the elevated PR in Chapter 4.1.2, a phenomenon associated with the cessation of estimated choking of the stage, which is the main reason for the peculiar curve shapes of these loss models in Figs. 4.1-22 and 4.1-23.
Chapter 4. Results

Figure 4.1-21: Difference in estimated total to static efficiency between elevated and design PR.

Figure 4.1-22: Difference in estimated mass flow between elevated and design PR.

Figure 4.1-23: Difference in estimated aero power between elevated and design PR.
Loss Coefficients

This section examines the impact of the increase in PR on the loss distribution in stator and rotor using three select loss models. For this purpose, the CC and MK models in AXIAL and the MK-A model in TML have been chosen. The Figs. 4.1-26 to 4.1-28 visualise the percentage difference between the design PR and elevated PR for the profile, secondary, trailing edge, tip clearance and incidence loss coefficients. The incidence loss coefficient is only available in the AXIAL software and can be seen as a corrective coefficient, that varies between negative and positive in value as the VR increases and the incidence to the rotor goes from positive to negative. Normally, the effects of incidence are part of the profile loss correlation (AMDC-family) or as a
corrective coefficient to the profile loss (CC model). Due to the negative-to-positive variance of the incidence loss coefficient in AXIAL it cannot be handled the same way in the diagrams as the standard loss coefficients by using percentage differences, and thus the results for this loss component are presented as the real coefficient values in two separate graphs for the design and elevated PR relating to a separate y-axis on the right hand side.

The diagrams of the effects of increased PR show a great difference in loss coefficient trends for the CC model compared to the MK and MK-A model. The CC model estimates that profile losses are increased at the elevated PR for the rotor, whereas the MK and MK-A model both predict decreases in profile loss for this component. The losses in the stator are also estimated to rise, whereas the MK-based models show varying results, with the MK model (AXIAL) predicting a decrease and the MK-A model (TML) a very slight increase. Apart from this, the major trends seen in most graphs are a decrease in secondary loss, an increase in trailing edge loss and practically unaffected tip clearance loss.

In theory, the MK (AXIAL) and MK-A (TML) model should provide very similar results, since they are both based on the MK loss model and the AM deviation model, however, the results show that they respond differently to an increase in PR in some areas and most notably in terms of secondary loss. The MK model (AXIAL) predicts the secondary loss to be the loss coefficient that decreases the most with the increased PR both for stator and rotor (Fig. 4.1-27), while the MK-A model (TML) shows no decrease in secondary loss for the stator (Fig. 4.1-28a) and in terms of the rotor the decrease is substantially greater at low VR and shows a relatively small decrease in loss coefficient at high VR (Fig. 4.1-28b).
Figure 4.1-27: Difference in estimated MK stator (a) and rotor (b) loss coefficients between elevated and design PR (AXIAL).

Figure 4.1-28: Difference in estimated MK-A stator (a) and rotor (b) loss coefficients between elevated and design PR (TML).

A reasonable explanation for the difference in loss change with elevated PR between the MK model of AXIAL and the MK-A model of TML could be that the AXIAL software applies special proprietary correlations for the impact of e.g. roughness and Reynolds number limit on loss generation as opposed to the loss model-standard correlations of TML. Since roughness and Reynolds number limit both have different degrees of impact on performance depending on PR, as will be shown in Chapter 4.4.3 and 4.4.4, these proprietary coefficients could possibly interfere with the "normal" loss model behaviour, which could be part of the explanation for the discrepancy between the MK model in AXIAL and the MK-A model in TML.
4.1.4 Summary of Performance Estimation

In summary, it is evident that the performance estimation of stage 4b provided significantly more accurate results as compared to exp. data when conducted at the elevated PR rather than the design PR. This is illustrated in Fig. 4.1-29 which compares the accuracy of efficiency and mass flow estimates at the design point of design PR (≈4450 RPM, VR 0.48) (a), and the max. efficiency point of elevated PR (≈9500 RPM, VR 0.54-0.55) (b). Generally, it seems as though the models in AXIAL (displayed with round circles) provide more accurate efficiency predictions at the elevated PR, while the accuracy of the mass flow predictions do not improve noticeably. The loss models in TML (marked with diamonds), also provide much better estimations of efficiency at elevated compared to design PR, with all MK-based models now being within 2% of the exp. data. The mass flow estimations in TML improved for some of the models, whereas others got slightly worse. As a whole, the selection of TML models provided a much more cohesive set of mass flow predictions at elevated PR, as compared to the very dispersed results seen at design PR.

4.2 Loss Models and Loss Coefficients at Design PR

In this chapter, the loss models of AXIAL and TML are investigated in detail by analysing the loss coefficient distribution at design PR. The four main loss coefficients, profile, secondary, trailing edge and tip clearance loss, along with the total loss (sum of the coefficients) are presented for the entire VR range. Additionally, the models used in AXIAL also feature an
incidence loss coefficient calculated by the software, which is presented in the graphs of this chapter. The incidence loss of AXIAL appears to be a type of corrective coefficient, that varies between negative and positive in value as the VR increases and the incidence to the rotor goes from positive to negative. Normally the effects of incidence are part of the profile loss correlation (AMDC-family) or as a corrective coefficient to the profile loss (CC), and it is thus unknown exactly how the AXIAL software determines and calculates this variable.

4.2.1 Stator Loss Distribution

Selected Loss Models

Figs. 4.2-1 and 4.2-2 display the distribution of the stator loss coefficients for a couple of selected loss models in AXIAL and TML respectively. All of the four models examined (CC, MK, MK-A, KO-A) show a similar distribution of losses, with the secondary loss coefficient being the highest, followed by profile and trailing edge loss in descending order. The loss estimation of AXIAL and TML can best be compared by studying the MK model (AXIAL) and MK-A model (TML), since they in theory rely on the same loss model and deviation model. The total loss coefficient at design VR of MK is \(0.18\) and of MK-A \(0.15\), and both the secondary and profile loss coefficients are about 0.02 units greater for the MK model, while the trailing edge loss is very similar for the two models. Thus, the fact that the AXIAL models underestimated the efficiency to a greater degree than the equivalent TML models, as presented in Chapter 4.1.1, can partially be explained by this discrepancy in stator loss coefficient.
Chapter 4. Results

Loss Coefficient Comparison

In this section, the loss coefficients of the different loss models are compared. This enables the identification of loss models that differ significantly from the rest in terms of a particular loss coefficient. Figs. 4.2-3 to 4.2-6 show comparisons of profile, secondary, trailing edge and incidence loss coefficients for the stator.

The profile loss coefficient diagram shows three major groups of loss models at near identical values, the TML models using the MK-system are stacked on top of each other at $Y \approx 0.6$, the TML models using the KO-system are stacked above one another at $Y \approx 0.9$ and the majority of the AXIAL models are at $Y \approx 0.8$. The exceptions are the AMDC and CC models in AXIAL, which show much greater and lower values respectively compared to the majority of the models. This is to be expected for the AMDC model, since it was developed using now antiquated airfoils of lower aerodynamic performance than the blades of today.

The secondary loss coefficient diagram again shows a clear grouping of loss coefficients, with the entire range of TML models predicting more or less equal values and all AXIAL models providing very similar results between one another as well, the only exception being the MK-MSL2 model. The MK-MSL2 model gives a very low secondary loss coefficient due to its modified secondary loss correlation by Denton. This low value explains why this loss model predicts much higher levels of performance than the rest of the models (as seen in Chapter 4.1.1), however, compared to the rest of the models it appears as though this modified secondary loss coefficient is unreasonably low.
The trailing edge loss coefficients show no notable anomalies, instead all the AXIAL models give close to equal results, as do the TML models. On the whole, the trailing edge loss coefficients are very low for all loss models, and are thus of limited impact on the overall loss of the stator.

The incidence loss coefficients of the AXIAL loss models all show negative values indicating a favourable incidence, contributing to a decrease of the total loss coefficient. Since the incoming flow is axial and the stator inlet blade angle is \(-33.5^\circ\) this results in a relatively high negative incidence angle. Moderate negative incidence is known to have a positive effect on efficiency, although in this case it would appear as though the relatively high negative incidence in this case \((-33.5^\circ)\) is also evaluated by AXIAL as being of positive efficiency impact. The fact that the CC and BSM model differ from the rest of the models in terms of the incidence loss is not
surprising. The CC model relies on a different set of loss correlations than the AMDC-family of models and is therefore unlikely to give exactly the same results as the rest of the models in terms of the specific loss coefficients. The BSM model on the other hand is a modification of the MK model in regard to the incidence effect on profile losses (the addition of wedge angle dependence, see Chapter 2.1.5) and thus should be expected to exhibit a difference particularly in terms of the incidence loss parameter when compared to the other AMDC-family models. The close to zero value of the MK-MSL2 model in terms of incidence cannot be easily explained, since this model in theory should only be separated from the rest of the AMDC-family models in terms of the secondary loss coefficient predictions. However, a reasonable explanation for this could be that the incidence loss correlation in AXIAL is directly dependent on the specific flow field features of the blade passage, which would be affected by the difference in loss distribution brought on by a modification of the secondary loss component.
4.2.2 Rotor Loss Distribution

Selected Loss Models

The Figures 4.2-7 to 4.2-11 display the distribution of the rotor loss coefficients for a pair of the loss models in AXIAL and TML. All of the models examined except for KO-A show a similar distribution of losses, with the secondary loss coefficient being the highest, followed by profile, tip clearance and trailing edge loss in descending order. The KO-A model in TML shows a very strange behaviour in terms of its profile loss coefficient (Fig. 4.2-10) where the loss seems to increase at higher VR, rather than remain at a more or less constant level as do the other models. The reason for this behaviour is unknown, but it does explain the strange dip in efficiency and power occurring as the rotor speed increases past the design point, as seen in the main diagrams for the performance simulation of the stage in Chapter 4.1.1. The reason for this behaviour is unknown, although the KO model has previously been observed to predict considerably higher profile loss than the MK model at higher VR (equal to the region of higher negative incidence), as demonstrated in the work by Wei (2000). However, the profile loss of the KO model presented by Wei (2000, p. 71) at high VR is still far from the extreme level seen in the KO model in TML, and no sudden anomalous efficiency dips could be observed in his performance simulations with the model. Further into the report, in Chapter 4.2.2, it is shown that this peculiar distribution of profile loss in the rotor is a common feature of all the KO-based TML models.
Chapter 4. Results

Figure 4.2-7: Craig & Cox (CC) rotor loss coefficient distribution over the complete VR range at design PR (AXIAL).

The loss estimation of AXIAL and TML can best be compared by studying the MK model (AXIAL) and MK-A model (TML), since they in theory rely on the same loss model and deviation model. Both of these models display very similar curve shapes, and the only significant difference between them is in the region of low VR, where the secondary loss of the TML model is significantly higher than the equivalent AXIAL model. The total loss coefficient at design point (VR=0.48) of MK is \( \approx 0.35 \) and of MK-A \( \approx 0.31 \), and thus this is part of the explanation (together with the discrepancy in stator losses of the two models mentioned in Chapter 4.2.1) for why the AXIAL models underestimated the efficiency to a greater degree than the equivalent TML models (as seen in Chapter 4.1.1).

Figure 4.2-8: Moustapha & Kacker (MK) rotor loss coefficient distribution over the complete VR range at design PR (AXIAL).

The behaviour of the incidence loss coefficient of the AXIAL models is also clearly illustrated in
the diagrams of CC and MK (Figs. 4.2-7 and 4.2-8). As seen in the two diagrams, the incidence loss coefficient starts out at a positive value at low blade speed and VR and then transitions into a negative value slightly before the design VR of 0.48 (4450 RPM) and remains negative throughout the rest of the VR interval. This can be partially explained by the incidence angle to the rotor, which is initially positive but decreases as the speed of the rotor increases and crosses into negative in the vicinity of the design speed, which is the region of maximum efficiency. Since moderately negative angles of incidence are associated with decreased loss it makes sense that the coefficient in this region would be negative, and thus contribute to a decrease in the total loss coefficient. However, as the speed increases past the design point and the VR approaches unity, the negative incidence would become greater and greater and most probably of detrimental impact to the efficiency, although this is contrary to what can be deduced from the results in the diagrams.

Figure 4.2-11 is intended to illustrate the difference in loss coefficients when comparing the BSM model to the MK model. In theory, the BSM model is based on the MK system but features a modification of the profile loss coefficient and the incidence effect correlation (see Chapter 2.1.5), thus only these two coefficients would be expected to differ. As can be seen from the figure, the profile losses as well as the incidence losses do indeed differ between the two models, with the BSM profile coefficient being smaller and the incidence coefficient being greater at low VR and closer to positive at high VR (indicating a relatively more detrimental impact on efficiency due to incidence effects) compared to the MK model. However, also the secondary loss differs between the two systems, with the losses being slightly higher for the BSM system.
The reason for the secondary loss increase with BSM over MK is not easily explained and could indicate that the AXIAL software does not rely completely on the exact loss correlations of the loss models implemented but rather also implements proprietary correlations as part of the main loss system. However, it is important to acknowledge that the aforementioned loss coefficient differences between the two loss model systems are only present at off-design conditions above and below VR ≈ 0.48. This could indicate that the reason for the discrepancy between the two models may be due to a proprietary off-design loss-compensation scheme in AXIAL, dependent on the specific loss model used.

**Figure 4.2-10:** KO-A rotor loss coefficient distribution over the complete VR range at design PR (TML).

**Figure 4.2-11:** Comparison of BSM and MK loss models, displaying relative difference of loss coefficient distribution over the complete VR range (AXIAL).
Loss Coefficient Comparison

In this section, the loss coefficients of the different loss models are compared, this enables the identification of loss models that differ significantly from the rest in terms of a particular loss coefficient. Figures 4.2-12 to 4.2-16 show the comparisons of profile, secondary, tip clearance, trailing edge and incidence loss coefficients for the rotor.

The profile loss coefficient diagram shows three major groups of loss models at near identical values, the TML models using the MK system are stacked on top of each other at \( \approx 0.1 \), the TML models using the KO-system with their peculiar graph-shape are stacked close to one another, and the majority of the AXIAL models are at \( \approx 0.16 \). The exception is the AMDC model in AXIAL which shows higher values than those of the rest of the AXIAL models, just as was the case for the stator. As was mentioned in Chapter 4.2.1 concerning the stator, this behaviour was to be expected for the AMDC model since the blades were of lower aerodynamic performance at the time of the development of the AMDC model, and with the advent of the KO model a profile loss correction coefficient of \( 2/3 \) was introduced to lower the estimated profile losses of modern blades (see Chapter 2.1.3).

![Figure 4.2-12: Comparison of rotor profile loss coefficient distribution over the complete VR range at design PR.](image)

The secondary loss coefficient diagram again shows a clear grouping of loss coefficients just as was the case for the stator, with the entire range of TML models predicting more or less equal values at and above the design VR of 0.48, although the KO-based models differ greatly from the rest at low VR. All AXIAL models provided very similar results as well with the only exception being the MK-MSL2 model, which was also the case for the stator. The MK-MSL2 model gives a very low secondary loss coefficient due to its modified secondary loss correlation.
by Denton. This low value together with the equivalent low value of the stator secondary loss coefficient explain why this loss model predicts much higher levels of performance than the rest of the models as seen in Chapter 4.1.1, however, compared to the rest of the models it appears as though this modified secondary loss coefficient might be unreasonably low.

![Figure 4.2-13: Comparison of rotor secondary loss coefficient distribution over the complete VR range at design PR.](image)

The tip clearance loss coefficients are clearly divided into three groups. Firstly, the entire array of AXIAL models give practically identical results which seem to be independent of the VR, as illustrated by the flat curve shape. The same VR independence is seen among the MK-based models in TML which all show identical loss at a level slightly lower than the AXIAL models. Lastly, the third group of models is composed of the KO-based TML models which once again exhibit a behaviour very different from that of the majority of models. When using these models, the tip clearance loss is shown to decrease as the rotor speed increases.

The trailing edge loss coefficient distribution for the rotor echo the results of the stator and show no notable differences between the models, rather all the AXIAL models give close to equal results and so too do the TML models. On the whole, the trailing edge loss coefficients are very low for all loss models and are thus of limited impact on the overall loss of the rotor.

The incidence loss coefficients of the AXIAL loss models all show values going from positive to negative with increasing VR. This means that the incidence loss coefficient contributes to increased total loss at low VR and decreased total loss at high VR. This can be explained by the fact that the incidence to the rotor goes from positive to negative with increased rotor speed and VR and negative incidence is known to have a positive effect on efficiency. As was previously pointed out in Chapter 4.2.1 concerning the stator loss distribution, the fact that the CC and
BSM model differ from the rest of the models in terms of the incidence loss is not surprising. The CC model is not based on the same standard correlations as the AMDC-family of models, and is therefore not likely to coincide exactly with the estimated loss coefficients of the other systems. The BSM model which is a modification of the MK model in regard to the incidence effect on profile losses (the addition of wedge angle dependence, see Chapter 2.1.5) should therefore be expected to show a difference in regard to the estimated incidence loss parameter compared to the other AMDC-family models. Lastly, the reason for the deviation of the MK-MSL2 model from the other MK-models in terms of incidence loss is not obvious, since this model in theory should only be separated from the rest of the AMDC-family models in terms of the secondary loss coefficient predictions.
4.2.3 Effect of Incidence on Turbine Losses

As mentioned in the literature study in Chapter 2, the effect of blade incidence angle plays a major role in the performance of the blade row. Most important in a 1-stage axial turbine stage is the incidence to the rotor which varies depending on the rotor blade speed, whereas the incidence to the stator naturally is always constant. Figure 4.2-17 illustrates the efficiency dependence on rotor incidence angle for a number of loss models.

In figure Fig. 4.2-18, the loss dependence on rotor incidence for the CC and MK loss models is shown distributed on the loss coefficients. For positive rotor incidence angles the profile and incidence loss components are significantly higher than at negative values of incidence for
Figure 4.2-18: Visualisation of CC and MK rotor loss coefficient distribution at design PR as a function of rotor incidence angle (AXIAL).

the MK loss model, showcasing the detrimental effects of positive rotor incidence on turbine performance. For the MK model, the aerodynamic losses are observed to be largely independent of the rotor incidence angle at high negative incidence, while the secondary loss is the only loss coefficient (besides from incidence) affected in the case of the CC model. The relatively moderate increase of the total loss coefficient of the MK model as the incidence approaches the highly negative region was also seen in the work by Wei (2000, p. 69). As the author explains it, the fact that the MK total loss increases more "smoothly" than other loss models could potentially be because the MK off-design profile loss correlation was developed using performance data of more modern airfoils than the earlier AMDC-models it was originally based on (Wei, 2000, p. 74).

4.3 Total Loss Distribution

This section summarises and compares the total loss coefficients of the loss models in AXIAL and TML at design PR and elevated PR at the operating point of maximum efficiency (VR≈0.54).

4.3.1 Total Loss Coefficients at Design and Elevated PR

The total loss coefficients of the full set of loss models investigated are shown in Fig. 4.3-1 at both design PR (left column) and elevated PR (right column). In terms of the simulation in AXIAL, the MK-based models give very similar results concerning the loss coefficients. This correlates well with the predicted efficiencies which were found to be practically identical as
previously seen in Chapter 4.1, with the exception of course being the MK-MSL2 model with its Denton-modified secondary loss which results in drastically lower losses in both stator and rotor. Lastly, the CC model loss coefficients are the ones that stand out the most and are not only lower than the majority of AXIAL models both at the design and elevated PR but the relative difference between the two PR is also much smaller for this model than for the others. Since the CC model is not directly related to the AMDC-family of models it is not surprising that the loss distribution scheme itself would differ between this model and the rest, however it is somewhat more surprising that this is the only model that does not show a significant decrease of losses as the PR is increased. Though this behaviour helps explain why the relative difference of the estimated efficiency of the CC model was very small between design and elevated PR conditions as seen in Chapter 4.1.

The nine TML models investigated show a higher degree of coherence among one another than did the models in AXIAL. A difference between the KO-based and the MK-based models is however evident, with the estimated loss coefficients being higher for the former. The stator losses for the three KO-based models are nearly identical to one another, as are the stator losses for the three MK-based models. All of these loss models rely on different deviation models, and since the estimated losses of the models only differ for the rotor blade row it would appear as though the choice of deviation model only has a visible impact on the losses of this component. The reason for this could be the differences in inlet flow conditions between stator LE (constant

---

**Figure 4.3-1:** Total loss coefficients of the loss models in the AXIAL and TML software at the operating point of maximum efficiency, distributed on stator and rotor fractions. Design PR: $PR_{ts}$ 1.24, ≈5000 RPM, VR 0.54; Elevated PR: $PR_{ts}$ 2.08, ≈9400 RPM, VR 0.54.
high negative incidence and relatively low flow speed) and rotor LE (incidence close to zero and relatively higher flow speed) at the investigated operating point of maximum efficiency.

This loss comparison offers a good opportunity to compare the AXIAL and TML software directly by examining the results of the MK model in AXIAL and the MK-A model in TML, which in theory should be based on both the same loss model and deviation model. Hence, in an ideal scenario, and given that the geometry of the turbine stage were equivalently modelled in both software, the losses of these two models should be very similar to one another. However, as can be seen from Fig. 4.3-1, the MK model in AXIAL predicts higher losses both in the stator and rotor than does the MK-A model in TML. Another way of looking at these results is to display the percentage share of losses in stator and rotor at design and elevated PR, as illustrated in Fig. 4.3-2. From this graphic it is clear that although the absolute value of the loss coefficients differs notably between the two models, the actual share of stator and rotor losses at both PR is considerably more similar.

![Figure 4.3-2: Stator/rotor loss distribution (%) at design PR compared to elevated PR for the individual loss models in the AXIAL and TML software at the operating point of maximum efficiency, distributed on stator and rotor fractions. Design PR: $PR_{ts}$ 1.24, $\approx5000$ RPM, VR 0.54; Elevated PR: $PR_{ts}$ 2.08, $\approx9400$ RPM, VR 0.54.]

The effect of elevating the PR on stator and rotor loss contributions is illustrated in higher detail in Fig. 4.3-3, which shows big differences in the behaviour of the models in AXIAL compared to the models in TML. With increased PR, the loss models in AXIAL decrease their estimated loss coefficients of both stator and rotor, whereas the TML models only show a noticeable decrease in the losses of the rotor. It is not clear why an increase in the PR would lead to a decrease in
losses solely in the rotor (and not the stator as well) as in the case of the TML models, although it could possibly be related to the previously discussed difference between stator and rotor inlet flow conditions.

![Graph showing differences in total loss coefficients between elevated PR and design PR for individual loss models in AXIAL and TML software at the operating point of maximum efficiency, distributed on stator and rotor fractions. Design PR: \(\text{PR}_{ts} = 1.24\), \(\approx 5000\) RPM, \(\text{VR} = 0.54\); Elevated PR: \(\text{PR}_{ts} = 2.08\), \(\approx 9400\) RPM, \(\text{VR} = 0.54\).

**Figure 4.3-3:** Difference in total loss coefficients between elevated PR and design PR for the individual loss models in the AXIAL and TML software at the operating point of maximum efficiency, distributed on stator and rotor fractions. Design PR: \(\text{PR}_{ts} = 1.24\), \(\approx 5000\) RPM, \(\text{VR} = 0.54\); Elevated PR: \(\text{PR}_{ts} = 2.08\), \(\approx 9400\) RPM, \(\text{VR} = 0.54\).

It should be pointed out that the real efficiency of the turbine stage is virtually identical at design and elevated PR, as discussed in Chapter 4.1.3. Hence, in an ideal scenario the absolute value of the estimated loss coefficients should remain constant, and would only be allowed to change in the stator/rotor distribution. The fact that the majority of models show a net decrease in losses when comparing the elevated PR to the design PR could suggest that the losses at the lower PR condition are overestimated. Consequently, since the major decrease in losses from increasing the PR appear to occur in the rotor (Fig. 4.3-3), the losses in this component could possibly be the major source of loss-overestimation at design PR. This is especially true for the TML models, which show virtually no difference in stator losses but great difference in terms of the losses of the rotor when comparing the two PR. Thus, a way of adapting the software tools for better performance at the low PR condition would be to investigate the effect of downscaling the rotor losses.
4.3.2 Turbine Stage Flow Angles and Velocities

This study is not aimed at investigating the detailed flowfield of the turbine stage modelled, however, for reference purposes and for the benefit of future work on the subject a very brief review of these parameters will here be presented.

Table 4.3-1 provides an overview of the main flow angles at stator inlet, rotor inlet and rotor outlet for the standard MK models in AXIAL and TML. These flow angles are given at both design and elevated PR conditions. It is important to note that these two loss models, MK in AXIAL and MK-A in TML, in theory should provide very similar results since they are both supposedly based on the standard MK loss model together with the standard AM deviation model. Additionally, Fig. 4.3-4 and 4.3-5 illustrate the turbine stage velocity triangles of the MK-A model in TML at the operating point of maximum efficiency (VR 0.54) for the design PR and elevated PR respectively.

Table 4.3-1: Turbine stage velocity triangle data at design and elevated PR.

<table>
<thead>
<tr>
<th>Software &amp; Loss Model</th>
<th>Parameter</th>
<th>Design PR</th>
<th>Elevated PR</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Station</td>
<td>Inlet</td>
</tr>
<tr>
<td>AXIAL: MK</td>
<td>$\alpha$ [°]</td>
<td>0.0</td>
<td>74.0</td>
</tr>
<tr>
<td></td>
<td>$\beta$ [°]</td>
<td>-73.9</td>
<td>50.4</td>
</tr>
<tr>
<td>TML: MK-A</td>
<td>$\alpha$ [°]</td>
<td>0.0</td>
<td>76.7</td>
</tr>
<tr>
<td></td>
<td>$\beta$ [°]</td>
<td>-75.3</td>
<td>57.9</td>
</tr>
<tr>
<td></td>
<td>$c$ [m/s]</td>
<td>26.6</td>
<td>166.8</td>
</tr>
<tr>
<td></td>
<td>$c_x$ [m/s]</td>
<td>26.6</td>
<td>38.3</td>
</tr>
<tr>
<td></td>
<td>$c_y$ [m/s]</td>
<td>0.0</td>
<td>162.3</td>
</tr>
<tr>
<td></td>
<td>$w$ [m/s]</td>
<td>104.7</td>
<td>72.0</td>
</tr>
<tr>
<td></td>
<td>$w_y$ [m/s]</td>
<td>-101.3</td>
<td>61.0</td>
</tr>
<tr>
<td></td>
<td>$U$ [m/s]</td>
<td>101.3</td>
<td>5000</td>
</tr>
</tbody>
</table>

When comparing the stage flow angles in AXIAL and TML at design PR, the model in AXIAL estimated the flow angles as being circa 1 to 7 degrees smaller than for the equivalent model in TML. Most significantly, the absolute rotor outlet angle was 6.5 degrees smaller in AXIAL ($26.4^\circ$) compared to TML ($32.9^\circ$). In terms of the difference in flow angles at design and elevated PR, the only major difference between the two PR concerns the rotor outlet where there is a discrepancy of over $15^\circ$ for both the MK model in AXIAL and the MK-A model in TML. Hence, the rotor outlet angle is significantly greater at the design PR compared to the elevated PR.
Intuitively, a lesser absolute outlet angle at rotor outlet would be an indicator of higher efficiency due to the minimal kinetic energy loss in the direction of rotation of the rotor of this operating point, however, of course one also has to take into account the aerodynamic losses of the loss models. Hence, the higher predicted outlet angle of the MK-A model in TML in fact resulted in lower aerodynamic losses as calculated by the correlations of the loss model in the software, resulting in higher estimated efficiency than for the MK model in AXIAL. The same reasoning could be used to explain the difference in loss coefficient between design and elevated PR. In short, the relationship between the flow angles and the overall efficiency is complex and cannot be satisfactorily explained without a deeper understanding and analysis of the flow components than what is encompassed in the scope of the present work.
4.4 Geometry Adjustments and Parameter Selection in AXIAL

In this chapter, the effects of the geometry adjustment coefficients applied to the KTH Test Turbine stage 4b-model in AXIAL are investigated. Also, the impact of some of the most important user-definable parameters in AXIAL on the performance prediction of the stage is analysed. This is performed by determining the individual impact of the coefficients and parameters on predicted turbine efficiency and mass flow, along with other parameters when appropriate.

These same geometry correction coefficients and parameter choices have been used throughout all the simulations of the stage performance in AXIAL presented in Chapter 4 up until this point. The complete description of the development and implementation of these parameters can be found in Chapter 3. As stated in the aforementioned chapter, the TML software did not support geometry correction-coefficients and also did not feature the non-standard parameters present in AXIAL, and thus the simulations in TML were run at default setting without any type of modifications.

4.4.1 Effect of Stator Geometry Correction Coefficients

This section relates to the stator throat area coefficient (1.033) and stage midsection area coefficient (0.968) described in Chapter 3.1.4 and 3.1.5 respectively. The measure of introducing correction coefficients to account for the stator endwall contouring effect on stator throat area and the contraction of the annulus at midstage do not seem to have had a great impact on estimated efficiency or mass flow. Figure 4.4-1 illustrates the impact of the two area correction coefficients on the overall efficiency of the stage using two different loss models, CC and BSM, and Fig. 4.4-2 shows the combined effect of the two coefficients when applied simultaneously as is the case in the main simulation. From Fig. 4.4-1 it appears as though the two area correction coefficients more or less cancel each other out in terms of their combined impact on efficiency, with the throat area coefficient contributing to an efficiency decrease at low VR and an increase at high VR, whereas the stator outlet area coefficient gives the opposite result of equal magnitude. The final impact on efficiency, as seen in Fig. 4.4-2, is in the order of magnitude of 1/100 percent and thus of very limited importance to the overall efficiency of the stage.
Chapter 4. Results

The effect on mass flow mirrors that on efficiency in the sense that the two area coefficients predict equal change in mass flow in mutually opposite directions, as seen in Fig. 4.4-3. Predictably, the stator throat area correction coefficient which signifies an area increase results in an increased mass flow (just above 2%), and the stator outlet area coefficient which signifies an area decrease results in a mass flow decrease (just above 2%). The overall effect on mass flow when the two coefficients are combined is thus very limited, as seen in Fig. 4.4-4, and plays no major role in the estimated mass flow of the stage.

**Figure 4.4-1:** Individual effect of stator correction coefficients for throat and outlet area on estimated efficiency, using CC and BSM loss models.

**Figure 4.4-2:** Combined effect of stator correction coefficients for throat and outlet area on estimated efficiency, using CC and BSM loss models.
4.4.2 Effect of Stator Secondary Loss Correction

The decision to use a correction factor of 0.80 for the secondary loss coefficient of the stator, as described in Chapter 3.1.4, in order to account for the endwall contouring effect is shown to have a significant impact on most turbine performance parameters. As seen in Fig. 4.4-5 and 4.4-6, at the design speed of VR 0.48 ($\approx 4450$ RPM), the S-loss factor contributes to an increase in overall efficiency of 1.0-1.2% depending on loss model, an increase in mass flow of around 0.5%, an increase in aero power of 1.7-2.0%, and a decrease of the degree of reaction by 0.07 to 0.08 units.

The reason for the increase in efficiency is not surprising since a reduction of losses in any part
Chapter 4. Results

Figure 4.4-5: Effect of stator secondary loss coefficient factor (0.8) on estimated total-to-static efficiency (a) and mass flow (b) using CC, BSM and MK loss models.

Figure 4.4-6: Effect of stator secondary loss coefficient factor (0.8) on estimated aero power (a) and degree of reaction (b) using CC, BSM and MK loss models.

of the turbine will naturally affect its performance positively. Perhaps more surprising is the extent of the impact, an entire percentage-point, although this does seem reasonable when one considers that the secondary loss of the stator is the major source of loss in this blade row as seen in Chapter 4.2.1. The increase in mass flow caused by the use of the stator secondary loss coefficient of 0.80 also seems logical, since the reduction of secondary flows at the inlet of the stage should lead to less flow obstruction in the blade passage and thus higher flow rates.

4.4.3 Effect of Roughness

The simulation of the KTH Test Turbine stage 4b in AXIAL used a standard RMS roughness of 1.6 μm for the stator and rotor blades as discussed in Chapter 3.4. This section evaluates
the influence of roughness on predicted turbine performance in AXIAL, both in terms of overall performance and the impact on individual loss coefficients. Figure 4.4-7 shows the efficiency decrement from using a standard roughness of 1.6 µm as compared to zero roughness. The roughness parameter is shown to be responsible for an efficiency decrease of around -0.5% to -2.0% depending on loss model and rotor speed.

![Graph showing the effect of standard (1.6µm) vs. zero roughness on estimated efficiency, using CC, BSM and MK loss models.](image)

**Figure 4.4-7:** Effect of standard (1.6µm) vs. zero roughness on estimated efficiency, using CC, BSM and MK loss models.

Figure 4.4-8 and 4.4-9 show the influence of roughness on efficiency and mass flow for a standard range of roughness values (0.0 to 5.0 µm), and for two different loss models, CC and BSM, at the design and elevated PR. It seems as though the effect of roughness is more detrimental to the BSM model than the CC model in terms of the estimated efficiency. The simulated effect of roughness on mass flow is somewhat peculiar, although in the majority of cases the mass flow decreases with increasing roughness as would be expected, for the case of CC at design PR the trend is strangely the opposite.

Figure 4.4-10 illustrates the effect of roughness on the loss coefficients of the BSM loss model at design and elevated PR. As would be expected, increased roughness primarily effects the profile loss and only has a small effect on the estimated secondary loss. With elevated PR the profile losses caused by roughness are seen to increase considerably, equating to a total loss increase associated with standard roughness of about 50% more in the elevated PR case as compared to the design PR case.
Figure 4.4-8: Effect of roughness on estimated efficiency at design and elevated PR, using CC and BSM loss models.

Figure 4.4-9: Effect of roughness on mass flow at design and elevated PR, using CC and BSM loss models.

Figure 4.4-10: Effect of roughness on the rotor loss coefficients of the BSM loss model at design and elevated PR.
4.4.4 Effect of Reynolds Number Limit

This section evaluates the significance of the Re limit for profile and secondary loss correction of the AXIAL software. The nominal Re limit of the AMDC-family of loss models is 200 000 (see Chapter 2.1.3) and of the Craig & Cox model 450 000 (see Chapter 2.1.6). However, as earlier described in Chapter 3.4, the AXIAL simulations were carried out with the AXIAL recommended (and default) Re limit setting of 1e30 for all loss models, which signifies an unlimited Re limit for profile and secondary loss correction.

![Figure 4.4-11: Effect of default vs. nominal Re limit on estimated efficiency, using CC, BSM and MK loss models.](image)

Figure 4.4-11 illustrates the difference in efficiency when using the default 1e30 limit compared to the nominal values of 200 000 for the AMDC-family loss models and 450 000 for the Craig & Cox model. As can be clearly seen from the figure, the choice to use the default Re limit of 1e30 resulted in an efficiency increase of about 1.4% for the BSM model over using the nominal Re limit of 200 000. For the CC model, the effect of using 1e30 as opposed to the default value (450 000) for the Re limit did not have nearly as great an effect as for the BSM model. The reason for this can be better understood from Fig. 4.4-12, which illustrates that at the point of the AMDC Re limit of 200 000 the effect on the estimated efficiency is increasing rapidly when compared to higher Re (since the mean Re of the stage is above this limit), whereas at the point of the CC limit of Re 450 000 the efficiency impact curve has flattened (due to the set Re limit approaching the real Re) and thus there is no substantial difference between this value and higher values. This is especially true for the design PR which is the condition depicted in Fig. 4.4-11.
Figure 4.4-12: Effect of Re limit on estimated efficiency at design and elevated PR, using CC and BSM loss models.

Figure 4.4-13 illustrates the way the Re number limit affects the mass flow of the stage. It appears as though there is a big difference between the CC and BSM models at low PR, whereas the effects on the mass flow due to Re limit is nearly identical for the two models at elevated PR. In all cases, higher Re number limits increase the mass flow of the stage. This is probably caused by the estimated losses being lower for the higher Re limits, which implying less flow disturbances in the blade passages blocking the mass flow.

Figure 4.4-13: Effect of Re limit on mass flow, using CC and BSM loss models.

The impact on the loss coefficient distribution from the choice of Re limit is seen in Fig. 4.4-14 to 4.4-15 for the BSM and CC loss models at design and elevated PR. As would be expected, an increase in the defined Re limit results in decreased profile and secondary losses for both loss models. When simulating using the elevated PR the secondary losses are further decreased.
Figure 4.4-14: Effect of Re limit on the rotor loss coefficients of the BSM loss model at design and elevated PR.

Figure 4.4-15: Effect of Re limit on the rotor loss coefficients of the CC loss model at design and elevated PR.

compared to design PR, while the increase in PR does not seem to affect the profile losses significantly.

4.4.5 Effect of Stator Lean Angle

The stator of the KTH Test Turbine stage 4b has a positive LE lean angle of 12.2°, as detailed in Chapter 3.1.3. In the present chapter, the impact of the lean angle on stator and overall stage performance is analysed. However, as was also mentioned in the conclusion of Chapter 3.1.3, a small error was made when defining the lean angle in AXIAL and the angle specified was unfortunately a fraction too high at 12.8° as opposed to the real value of 12.2°. Thus, some experiments in this chapter have been conducted using the erroneous value of 12.8°, however,
as will also be shown in this chapter, this small discrepancy is of negligible importance to the actual impact of the stator lean angle on the predicted stage performance.

As seen in Fig. 4.4-16, the use of a lean angle of 12.8° has a very limited effect on the overall efficiency of the stage of around 0.05% increase at the design speed. The two loss models investigated are shown to differ at higher VR, where the CC model actually estimates a decrease in the overall efficiency attributable to the use of lean angle.

In Fig. 4.4-17 the simulation of the lean angle influence on overall efficiency is shown for a number of different lean angles in ascending order. The results show something unexpected, as the lean angle reaches 15° the trend of increasing efficiency contribution is abruptly interrupted and the results become unreasonable. It does thus appear as though the models used to simulate
the effect of blade lean angle in AXIAL cannot handle angles bigger than 15°, at least not for this specific turbine model. Alternatively, it could also be that the effect of the stator lean on the prediction of some other parameters of the stage (e.g. pressures, velocities, flow angles etc.) causes the numerical prediction of any of the latter to go out of range or fail to correctly compute, causing the sudden efficiency drop at 15°.

![Graph showing the effect of stator lean angle on stator loss distribution.](image)

**Figure 4.4-18**: Effect of stator lean angle on stator loss distribution, as compared to zero lean angle, for the BSM loss models at design PR.

The impact of the stator lean angle on the loss coefficient distribution of the BSM model for stator and rotor is illustrated in Fig. 4.4-18 and Fig. 4.4-19 respectively. The results show that the total loss coefficient of the stator in fact increases slightly with the increase of the lean angle, due to the secondary losses rising more than the combined fall of the profile, trailing edge, and incidence loss. The impact of the lean angle on the rotor losses appears to be of greater magnitude than on the stator losses, although still at a very small scale. In the rotor, both the profile, secondary and incidence loss fall as the stator lean angle increases, while the tip clearance and trailing edge loss remain constant. This results in a reduction of the total rotor loss coefficient of close to 1% at its peak at a lean angle of ≈15°. Beyond this point the loss coefficients of both stator and rotor diverge dramatically, and the continuous decrease/increase of the loss coefficients is interrupted. The probable cause for this was discussed in the previous section.

Figure 4.4-20 illustrates the effect that the lean angle of the stator has on the spanwise pressure distribution of the blade. The simulation shows that a positive lean angle results in an increase in static pressure at hub and a decrease in static pressure at tip, thus resulting in a more even spanwise pressure distribution than at zero lean. For a negative lean angle, the effect
Figure 4.4-19: Effect of stator lean angle on rotor loss distribution, as compared to zero lean angle, for the BSM loss model at design PR.

is the opposite resulting in a greater difference in the spanwise distribution of static pressure. The effects seen for a positive lean angle are beneficial since a more even spanwise pressure distribution reduces other non-desirable flow effects, such as secondary losses, which are driven by pressure gradients, as discussed in Chapter 3.1.3.

Figure 4.4-20: Effect of stator lean angle on stator spanwise static pressure distribution for the BSM loss model at design PR.

Lastly, the simulated effect of stator lean angle on rotor incidence angle along the span of the blade is shown in Fig. 4.4-21. As the lean angle of the stator increases, the incidence to the rotor at midspan stays largely unchanged, while it decreases at the hub and increases at the tip. The change of the incidence at the tip is shown to be significantly greater than at the hub for lean angles above $\approx 5^\circ$. 
4.5 Purge Flow

This chapter concerns the cavity purge flow simulations performed with the TML software tool as described in Chapter 3.5.

4.5.1 Loss Model Comparison

The purge flow impact on the performance of the KTH Test Turbine stage 4b has been simulated using six different loss models at design and elevated PR and validated against exp. data in a series of diagrams representing the main performance parameters. The results of the purge flow simulations at the two different PR are compared in Fig. 4.5-1 to 4.5-6, with the design PR results on the left (a) and the elevated PR results on the right (b). The purge flow simulation at design PR (1.24) was conducted at 4380 RPM (corresponding to a VR of 0.47), and the simulation at elevated PR (2.09) was conducted at 10530 RPM (corresponding to a VR of 0.60). Some clear trends can be seen from the diagrams, firstly, all the loss models used for the simulation give very similar results, especially at design PR. In terms of the agreement between simulation and exp. data the results of the design and elevated PR case are very similar, with an underestimation of the decrease in total to static efficiency and an overestimation of the decrease in total to total efficiency as well as an overestimation of the decrease in inlet mass flow and aero power. The one exception to the trend being the change in degree of reaction with increased purge rate, a parameter which is overestimated at design PR but underestimated by the majority of loss models at elevated PR.
The concurrence between simulated data and exp. data seems to be considerably better at the design PR than the elevated PR, especially for the mass flow and aero power parameters. The underestimation of the total to static efficiency loss caused by purge flow by the entire set of loss models at both design and elevated PR is not in line with the rest of the parameters whose losses are all overestimated. The total to total efficiency on the other hand complies much better with the exp. data, and the slight overestimation of the decrease of this parameter due to the applied purge flow is in line with the rest of the simulated parameters which also show overestimated losses. Thus, the simulated results indicate that when conducting purge flow simulations with TML it is advisable to use the total to total efficiency estimate, rather than the total to static variant, as the indicator of simulated performance.

Figure 4.5-1: Effect of purge flow on estimated total-to-static efficiency at design PR (4380 RPM, VR 0.48) and elevated PR (10350 RPM, VR 0.60).

Figure 4.5-2: Effect of purge flow on estimated total-to-total efficiency at design PR (4380 RPM, VR 0.47) and elevated PR (10350 RPM, VR 0.60).
Figure 4.5-3: Effect of purge flow on estimated inlet mass flow at design PR (4380 RPM, VR 0.47) and elevated PR (10350 RPM, VR 0.60).

Figure 4.5-4: Effect of purge flow on estimated aero power at design PR (4380 RPM, VR 0.47) and elevated PR (10350 RPM, VR 0.60).

Figure 4.5-5: Effect of purge flow on estimated enthalpy based degree of reaction at design PR (4380 RPM, VR 0.47) and elevated PR (10350 RPM, VR 0.60).
4.5.2 Operating Range Comparison

In this section, the effects of purge flow on stage performance across the entire operating range of the turbine is examined using the MK-C2V loss model in TML. Due to the lack of exp. data for purge flow across the operating range the results of this simulation are not validated, rather, this chapter serves merely to present the simulated results and further the understanding of how the TML software performs when running this type of simulation. A 'reference line' in the diagrams indicates the operating point at which the experimental data used for validation in the previous chapter was collected.

The results of the simulation are shown in Figs. 4.5-7 to 4.5-12, representing the total-to-static
efficiency, inlet mass flow, aero power and degree of reaction at design PR (a) and elevated PR (b). The design PR simulation shows results for zero, 0.98%, 2.03% and 4.93% purge flow rates, and at elevated PR the evaluated purge rates are at zero, 1.11%, 2.10%, 2.71%, 3.52% and 4.22%. At design PR, it is shown that an increase in purge flow decreases the efficiency of the stage, particularly in the design speed region. The inlet mass flow is also decreased with the increase of purge flow and seems to be largely independent of the rotor speed since the relative decrement is very similar across the entire VR range. Finally, it is seen that increased purge flow also increases the degree of reaction across the entire VR range, although the increase is greatest at lower VR. Overall, the trends at the elevated PR are seen to be the same as at the design PR but there are however some major differences, especially concerning the impact of purge flow on inlet mass flow. Firstly, it should be noted that the TML simulation of the overall performance of KTH stage 4b in general gives more accurate results (better agreement with test data) at elevated PR, as first noted in Chapter 4.1.4. Thus, this explains for example why the normalized total to static efficiency of the purge flow simulation is much higher than for the design PR case.

As just mentioned, the purge flow simulation of inlet mass flow at elevated PR yields some discrepant results as compared to the results at design PR (Fig. 4.5-8). In the zero purge flow scenario, the mass flow is almost perfectly constant with no abnormalities visible but when the purge flow fraction rises above zero a sudden drop in mass flow starts to appear in the 0.45-0.55 VR region, a drop which gets bigger as the purge flow fraction increases. This could potentially be related to choking which the stage is subject to when run at the elevated PR as discussed in Chapter 4.1.2. However, if choking in fact were the cause of this behaviour the mass flow would
Figure 4.5-9: Mach number at stator exit, the point of the highest absolute velocity of the stage, for the different purge flow rates at elevated PR.

Figure 4.5-10: Comparison of normalised aero power for varying purge flow rates at design and elevated PR using the MK-C2V loss model.

be expected to increase rather than decrease as the choking itself is released. Also, the Mach number of the flow at stator exit (Fig. 4.5-9) does not indicate that the flow becomes subsonic at the point at which the mass flow suddenly decreases either, further reducing the likelihood that stage choking is the reason for this behaviour. Thus, choking does not appear to be a good explanation for this phenomenon, and it would be necessary to study more parameters of the stage in detail in order to find a credible reason for this behaviour, as well as ensuring that this is not a computational error of the software. The sudden drop in mass flow in the mid VR range with increased purge rates consequently affects the estimated aero power in this region, as seen in Fig. 4.5-10b.

The effect of purge flow on the degree of reaction at elevated PR is similar to that of design PR. In the VR region of 0.45-0.55 however, the relative difference between the different purge
Figure 4.5-11: Comparison of enthalpy based degree of reaction for varying purge flow rates at design and elevated PR using the MK-C2V loss model.

Figure 4.5-12: Comparison of pressure based degree of reaction for varying purge flow rates at design and elevated PR using the MK-C2V loss model.

Flow fractions suddenly decreases as the higher purge flows decelerate in their rate of reaction degree increase, putting them more in line with the zero leakage condition.

4.6 Additional Turbines

This chapter describes the performance simulation of the KTH Test Turbine legacy stage by Wei (2000) and the stage by Ewen et al. (1973).
4.6.1 KTH Test Turbine Legacy Stage

The estimated total to total efficiency of the KTH Test Turbine legacy stage is displayed in Fig. 4.6-1 to 4.6-4. The simulation of the legacy stage was conducted using the same boundary conditions as the simulation of the stage 4b at design PR previously presented, i.e. using an equal total inlet temp. and pressure as well as PR. The results of the legacy stage simulation are however considerably more accurate than the simulation of stage 4b for these conditions, with most models falling within ±4% of exp. data at the design speed line rather than ±10% which was the case for stage 4b. The results of the AXIAL simulation show that the CC model comes closest to the experimental efficiency, while the AMDC and MK-MSL2 models give the most inaccurate results. The most accurate curve shape out of the set of loss models belongs to BSM.

The results of the TML simulation show the same positive trend, however with even better accuracy. All of the models simulated with this software are within 1% of the exp. data at the design speed with a generally good curve fit in the VR region of highest interest between 0.4 and 0.7. Outside of this interval however, the TML models based on the KO loss model display a somewhat strange curve shape, just as was the case for the simulation of KTH stage 4b. However, it should be pointed out that the anomalies themselves are not identical between the two stages. The previously observed dip in efficiency occurring right after the design speed line in the diagrams of stage 4b is not as clearly visible for the legacy stage (Fig. 4.6-3), where instead the rate of efficiency increase in the low VR range appears more inconsistent with exp. data.

![Figure 4.6-1: Total to total efficiency of the KTH Legacy Stage (AXIAL).](image)
Figure 4.6-2: Total to total efficiency of the KTH Legacy Stage, enlarged view (AXIAL).

Figure 4.6-3: Total to total efficiency of the KTH Legacy Stage (TML).

Figure 4.6-4: Total to total efficiency of the KTH Legacy Stage, enlarged view (TML).
4.6.2 Stage by Ewen et al. (1973)

The simulated efficiency of the Ewen-stage is seen in Figs. 4.6-5 and 4.6-6. As is evident from the figures, neither the loss models in AXIAL nor TML could satisfactorily predict the performance of this stage. Although the accuracy of the efficiency estimation at design point is relatively good for the majority of loss models in the two software, the results at off design VR are subpar and fail to reflect the real rate of efficiency increase as seen from the very bad agreement with the exp. data graph.

The velocity ratio definition in the work by Ewen et al. (1973) was slightly different from the main definition used for the simulations of the other stages in the present work. Rather than the total to static isentropic enthalpy drop being the main parameter, this alternate definition instead utilised the stage work, equivalent to the total to total enthalpy drop:

\[ \nu_{ts} = \frac{U}{\sqrt{2\Delta h_{tt}}} \]

![Figure 4.6-5: Total to total efficiency of the stage by Ewen et al. (1973) (AXIAL).](image-url)
Figure 4.6-6: Total to total efficiency of the stage by Ewen et al. (1973) (TML).
Chapter 5

Discussion

General Performance Simulation of KTH Stage 4b

The results of the simulation showed a few clear trends regarding the validity of certain loss models. Starting out with the simulation in AXIAL, the efficiency estimation of the CC loss model came the closest to the exp. data of the stage, which mirrors the trends seen in Wei (2000) and Guédez (2011, p. 77). Predictably, the MK model was also better at predicting the performance than the AMDC model in AXIAL, which is the same result seen in Wei (2000). Concerning the non-standard models validated in AXIAL referred to as MK-S and MK-MSL1, their estimated performance came very close to that of the normal MK model for most parameters, whereas the last non-standard model MK-MSL2 model showed a big difference when compared to the MK model. The use of this special secondary loss parameter (MSL2) in conjunction with the standard MK model does not seem to improve the performance estimation significantly, rather it contributes to overestimation of many parameters, a trend seen also in Guédez (2011, pp. 104-106).

The majority of the loss models in TML came closer to the reference data than the models in AXIAL. Except for the KO-based models, the models also all had a good curve shape. Although the design PR results were underestimated, the results at elevated PR showed very good accuracy for many of the MK-based models, much better than any of the models in AXIAL. More importantly, it was not simply the estimated efficiency which showed a high degree of accuracy at elevated PR, rather, the estimated mass flow and aero power were equally accurate for a pair of the models, although only at and above the design speed. However, below design speed this same pair of models provided significantly less accurate results in terms of
mass flow and aero power. This complex behaviour, exhibited exclusively at elevated PR, was most probably the result of the constraints imposed by estimated choking of the stage. This behaviour would need to be analysed at a deeper level in order to increase the understanding, and ultimately the reliability, of the simulated performance.

The effect of PR on the simulation accuracy proved significant with substantially better agreement with exp. data under the elevated PR compared to the design PR. The work by Dahlquist (2008, pp. 84-85) showed a similar trend, with the estimated efficiency complying less well with reference data for lower PR. However, in this case lower PR resulted in overestimation rather than underestimation of the efficiency as was the case in the present work. More concretely, in the present work, under the elevated PR the total loss of the CC and AMDC-family systems was decreased as compared to design PR mainly due to a loss in profile and/or secondary losses in the stator and rotor. However, since the accuracy of the estimated performance parameters was greatly increased under the elevated PR condition, it would appear as though the much lower losses at elevated PR were at least partly attributable to the design PR losses themselves being greatly overestimated. Thus, it is clear that the PR can have an effect on the accuracy of the efficiency estimation of the 1D software tools and their loss models.

The inability to correctly predict turbine efficiency for low PR might be a major shortcoming of some loss models. Moustapha & Kacker (1990, p. 271) pointed out that the model by Ainley & Mathieson (1951) suffered from this problem, and it is also worth pointing out that the loss system by Kacker & Okapuu (1982) was only validated against seven turbine configurations with a PR below 1.5 out of a total of 33 turbines used for this purpose, with the lowest PR turbine validated having a PR of 1.33. Out of these seven turbines only two had average aspect ratios of 1.5 or lower. This can be compared to the KTH Test Turbine stage investigated in the present work with its low design PR of 1.23 and very low average aspect ratio of stator and rotor of about 1.0. The modest number of turbines of very low PR and aspect ratio used for loss model testing in the previously cited work by Kacker & Okapuu (1982) does not necessarily explain the effect of low PR on the accuracy of performance estimations, however, it could indicate that loss models in general are not as thoroughly validated against these types of configurations.

The AXIAL software, which proved the less accurate of the two tools, actually does recommend that the blading used for simulation have an aspect ratio of at least 2 and preferably higher. Thus, one might speculate that AXIAL has been designed and configured for simulation of much bigger turbine systems at the expense of accuracy for small 1-stage turbines such as stage 4b.
In terms of the impact of PR on the estimated efficiency, the analysis of the dependence of the predicted loss coefficients on PR in Chapter 4.3.1 suggested that overestimated rotor losses could be responsible for the majority of the efficiency underestimation. However, to gain a better understanding of the stator/rotor loss distribution a deeper analysis of the degree of reaction (DoR) parameter would be necessary. The DoR parameter is a good indicator of the dynamic stage behaviour since it relates measured enthalpies at three axial stations: stator inlet, rotor inlet and rotor outlet. This parameter turned out to be greatly underestimated at both design and elevated PR (see Chapter 4.1), which indicates an unbalance in the stator/rotor enthalpy drop of the stage when compared to experimental data. Inevitably, this unproportionality will affect the aerodynamic losses of both blade rows. However, it is impossible to truly understand the impact of this stator-rotor work imbalance on the stage loss distribution without analysis of the flow parameters at the point between stator TE and rotor LE, and unfortunately experimental data for this station was not available to use for validation. Thus, this would be an interesting area of investigation for future work in order to better understand whether or not the stator and rotor are equally responsible for the underprediction of stage efficiency.

It is perhaps no coincidence then, that 1D software tools often support the application of correction coefficients, such as loss-component scaling factors, to allow for the optimisation of the loss model to better fit the particular turbine simulated. Using this kind of approach, it would undoubtedly have been possible to achieve more accurate simulation results both using AXIAL and TML, particularly concerning the design PR case, however, this type of optimisation is outside the scope of this paper which purpose is purely to validate the veracity of the standard loss systems. Albeit, it would certainly be interesting to see these capabilities of the 1D software in question evaluated as well.

Concerning the comparability of the AXIAL and TML software it should be pointed out that if in fact both software were to have implemented the loss models in the same manner, and were supplied with the same geometry parameters, the simulation of the MK (AXIAL) and MK-A (TML) loss models ought to have given the same results since they were based off of the same basic loss and deviation models. However, the fact that the results of the two software showed great discrepancies is an indicator of the difference in implementation of the loss models themselves. It would appear as if though the AXIAL software is dependent on many proprietary correlations for predicting losses in addition to the original loss models that it utilise as its basis, the roughness parameter would be an example of one such proprietary parameter. The TML software on the other hand appears to be based straight off of the loss models themselves, and
does not utilise the same type of proprietary parameters. Thus, if the desire were to study the performance of the original loss models themselves, it would be highly advisable to utilise the TML software tool in favour of AXIAL.

**Geometry Adjustments and Parameter Selection in AXIAL**

The impact of the geometry correction coefficients applied in AXIAL showed that the modification of stator throat and outlet area in essence had a very limit effect on the total stage performance, altering the estimations by under ±0.1% in terms of efficiency and under ±0.2% in terms of mass flow. The decision to use a secondary loss factor of 0.8 for the stator to account for the effects of endwall contouring had a greater impact on efficiency and mass flow than the geometry correction coefficients, increasing the overall efficiency of the stage at design point by between 1.0 – 1.2%, the mass flow by about 0.5% and the aero power by 1.7 – 2.0% depending on loss model. The decision to assume a standard roughness value (1.6μm) for both stator and rotor decreased the efficiency of the stage by between 0.8-1.6% compared to using zero roughness, and the decision to use the default Re limit of 1e30 as opposed to the nominal value increased the estimated efficiency of the BSM model with 1.5 – 2.5% and the CC model by 0.1 – 0.6% depending on PR used. The simulation of impact of the stator lean angle indicated that it only contributed to an efficiency increase of about 0.05% at the design point. It was surprising to observe that the AXIAL software did not seem to be able to handle stator lean angles in excess of 15°, at least not for the KTH Test Turbine stage 4b modelled in the software.

All in all, it turns out that from this set of AXIAL parameter choices, the only ones that made a substantial difference to the simulation were the chosen roughness, Re limit and secondary loss factor of the stator. If the roughness value of the blades would have been set to zero instead of standard roughness, a significant increase of estimated efficiency would have been seen, however this would probably not have been a realistic setting. On the other hand it is unknown how exactly AXIAL handles roughness input, since the majority of loss models implemented in AXIAL do not base their calculations on this parameter for determining profile and/or secondary losses. This is evident in the TML software, which does not require any roughness value to be defined. The same questions surround the Re limit in AXIAL, which is recommended to be set to infinity (1e30) by AXIAL, although in reality it should be much lower to comply with the nominal values of the specific loss models. Thus, it seems as though AXIAL
does not completely rely on the standard loss model that it bases its calculations on, but rather also implements proprietary correlations for its performance simulations.

**Purge Flow**

The results of the purge flow simulation performed with TML were promising, although a trend of overestimated losses could be seen throughout. The agreement with test data was better for the design PR than the elevated PR. The one estimated parameter that did not agree well with the others was the total to static efficiency, which was overestimated rather than underestimated.

The results of the TML purge flow simulation can be compared to those obtained by Dahlqvist & Fridh (2015). The results for estimated total to static efficiency by Dahlqvist & Fridh (2015) using their Entropy Relation (as described in Chapter 3.1.6) are considerably more accurate than the ones obtained from TML in the present work, with all of their estimated total to static efficiency points being within 1% of reference data. The accuracy of their method is also higher for the elevated PR than the design PR, whereas the trend was the opposite for the simulation using TML. The underestimation of the loss from purge seen in TML at the design PR is similar in degree to that obtained using the Simple Correlation in the work by Dahlqvist & Fridh (2015), although at the elevated PR the Simple Correlation instead predicted an overestimation of losses whereas TML still underpredicted the results.

In conclusion, it appears as though the purge flow simulations performed with TML showed the correct general trends although the detrimental effects of purge flow were overestimated. This suggests that the use of a loss correction coefficient for purge flow applied to the TML software could potentially enable proper downscaling of estimated losses and thereby achieve better agreement with exp. data of the KTH stage 4b.

**Additional Turbines**

The performance simulation of the additional stages by Wei (2000) and Ewen et al. (1973) gave greatly varying results. The simulation of the KTH legacy stage as described by Wei (2000) gave relatively good results for the AXIAL models, and even better for the TML models. What should be pointed out concerning this simulation is that it was performed using the same design PR and inlet aero conditions as the simulation of KTH stage 4b, however in this case the simulation did not suffer from the same significant underestimation. In fact, in this case the agreement with
reference data at design PR was just as good as that of stage 4b at elevated PR. This suggests that the geometry of the KTH legacy stage is perhaps more favourable for 1D simulation, or alternately, that some geometrical features of the KTH stage 4b are unsuitable for this type of simulation, and that the unfavourable effects of these features are suppressed at higher PR. Considering that the KTH stage 4b has a number of complicated geometrical features that the legacy stage does not have, such as lean angle of the stator, endwall contouring and contracted annulus between stator and rotor, these could all potentially play a role in the accuracy of the performance prediction.

Of course, it could simply be a case of inaccurate exp. data in the case of the KTH legacy stage, or discrepancies in the way the reference data was collected in the two cases. It should also be mentioned that, except for the PR, the exact operating parameters were not known for the exp. data collected from the KTH legacy stage, and thus the inlet temperature and pressure used for the simulation of said stage in the present work were best approximations based on the experiments conducted on the similar-in-design KTH Test Turbine stage 4b. Hence, this could also have contributed to higher degrees of uncertainty in simulation of the stage performance.

The simulation of the stage by Ewen et al. (1973) was not as successful as in the case of the previously mentioned KTH legacy stage. The very bad cohesiveness of the simulation and the exp. data for the majority of the VR interval can have a number of explanations. It is possible that some feature of the stage geometry was incorrectly extracted from the original report which thus provided the 1D software tools with erroneous data, and it could also be a problem related to the interpretation of the velocity ratio (VR) of the original report which was of an unusual form. However, the exceptionally low blade height and aspect ratio of the blading of the stage should also be taken into account, it could be that the loss models have problems handling this kind of blading. Although the loss models by e.g. Kacker & Okapuu (1982) and Moustapha & Kacker (1990) take the effects of low aspect ratio into consideration, it is not clear just how well they can handle such exceptionally low aspect ratios at off-design conditions as is the case for the stage by Ewen et al. (1973).
Chapter 6

Conclusion

The main objective of this work was to validate the 1D design tools 'AXIAL' by Concepts NREC and 'TML' by GKN Aerospace when simulating the performance of the KTH Test Turbine Stage 4b at different PR and investigate the significance of loss model selection, while simultaneously analysing the impact of special geometrical features of the stage, as well as specific parameter selection in the 1D software. Additionally, the objective was also to investigate and validate the ability of the TML software tool to simulate the stator-rotor cavity flow known as purge. This work proved the ability of the software tools to simulate the performance of the stage to within a reasonable degree of accuracy when under higher PR. At the low PR however, the results were below satisfactory from a validation stand point. Thus, the success of the performance estimation turned out to be heavily dependent on the PR used in the simulation, maybe even more so than the choice of loss model. However, it is clear that loss model selection does indeed have a significant impact on the results. The deviation model should not be overlooked either, since it was shown that in TML the same loss model could give greatly varying results based on what kind of deviation system it depended on. This work also proved the ability of TML to simulate purge flow, although the validation proved that the losses associated with this type of cooling were overestimated.

All in all, the results of this work suggest that 1D simulation could still be a viable tool for preliminary turbine design and simulation in competition from more advanced solutions such as CFD. However, in order to reach a significant degree of accuracy using 1D simulation methods it is of importance to have good knowledge of the specific 1D software tools used in terms of which loss model is the most appropriate to utilise, as well as the impact of the boundary conditions on
the simulation accuracy. Additionally, a good understanding of which is the most appropriate parameter selection in the 1D software is also of importance. Unless all of these aspects are well understood the accuracy of the 1D simulation could turn out unsatisfactorily low.

The next step in achieving better 1D results using the tools here reviewed would be to investigate the loss model customisation features available in the software. Particularly AXIAL offers the designer a great deal of freedom in terms of loss model tweaking capability, such as the possible application of scaling factors to the individual loss components. The TML software also has similar abilities, although far from as elaborate as AXIAL. The fact that these capabilities do exist as a standard component of the software tools is indicative of the nature of 1D loss models: their accuracy is completely dependent on the similarity of the test subject to the real world turbine systems the models themselves were modelled after, and thus, no 1D simulation will ever be perfect without the necessary degree of tuning and customisation.
Bibliography


## List of Figures

1.2-1 Blade geometry (Reproduced from El-Batsh, 2006, Fig. 1) .......................... 5
1.2-2 Turbine blade incidence terminology (Reproduced from Moustapha & Kacker, 1990, Fig. 1) ................................................................. 5
1.2-3 Turbine stage simplified geometry (Reproduced from Vogt, 2007). ................. 5
1.2-4 Stage velocity triangles (Reproduced from Vogt, 2007). .............................. 5
1.2-5 Sign convention for blade and flow angles of stator and rotor blades. .......... 6
1.2-6 Turbine Stage Velocity Diagrams (Reproduced from Dixon & Hall, 2010, Fig. 4.3) ................................................................. 7
1.2-7 Mollier Diagram for a Turbine Stage (Reproduced from Dixon & Hall, 2010, Fig. 4.4). ................................................................. 7
1.3-1 Airfoil and boundary-layer parameters (Reproduced from Stewart et al., 1960, Fig. 1) ................................................................. 11
1.3-2 Cascade vorticity as predicted by classical secondary flow theories (Reproduced from Sharma & Butler, 1987, Fig. 1) ................................. 12
1.3-3 Cascade endwall flow structure (Reproduced from Sharma & Butler, 1987, Fig. 2) ........................................................................ 13
1.3-4 Tip leakage and passage vortices at the tip endwall with a clearance (Reproduced from Sjolander, cited in Lampart, 2009, Fig. 1). .................. 14
1.3-5 Flow over a shrouded tip seal (Reproduced from Denton, 1993, Fig. 29). .... 15
1.3-6 Velocity profiles at and downstream of a simulated turbine blade trailing edge (Reproduced from Denton, 1993, Fig. 25). .............................. 16
1.3-7 Structure of supersonic trailing edge flow (Reproduced from Denton & Xu, 1990, Fig. 1). ................................................................. 16
1.4-1 KTH Test Turbine 1-stage configuration 4b. .................................................. 18

2.1-1 Profile loss coefficients for conventional section blades at zero incidence. \((t/c = 20\text{ per cent; } Re = 2 \times 10^5; M < 0.6)\) (Reproduced from Ainley & Mathieson, 1951, Fig. 4). ................................................................. 22
2.1-2 Positive stalling incidences of cascades of turbine blades. \((Re = 2 \times 10^5; M < 0.6)\) (Reproduced from Ainley & Mathieson, 1951, Fig. 7). ................................................................. 23
2.1-3 Variation of profile loss with incidence for typical turbine blading (Reproduced from Ainley & Mathieson, 1951, Fig. 6). .............................. 24
2.1-4 Secondary losses in turbine blade rows (Reproduced from Ainley & Mathieson, 1951, Fig. 8). ................................................................. 26
2.1-5 Effect of trailing-edge thickness on blade loss coefficients (Reproduced from Ainley & Mathieson, 1951, Fig. 9). ................................................................. 27
2.1-6 Comparison of design point efficiencies of 25 turbines (Reproduced from Dun- ham & Came, 1970, Fig. 1). ................................................................. 30
2.1-7 Inlet Mach number ratio for nonfree-vortex turbine blades (Reproduced from Kacker & Okapuu, 1982, Fig. 6). .................................................. 31
2.1-8 Combined effect of leading edge shock on inner end wall flow and the channel next to it (Reproduced from Kacker & Okapuu, 1982, Fig. 7). ................. 32
2.1-9 Trailing edge loss [energy] coefficient correlated against the ratio of trailing edge thickness to throat opening (Reproduced from Kacker & Okapuu, 1982, Fig. 14). 34
2.1-10 Comparison of predicted efficiency with experimental efficiency of 33 turbines (Reproduced from Kacker & Okapuu, 1982, Fig. 19 and 20). .......................... 35
2.1-11 Comparison of Ainley and Mathieson’s correlation with turbine rig data (Reproduced from Moustapha et al., cited in Moustapha & Kacker, 1990, Fig. 5(b)). 36
2.1-12 Effect of incidence on profile losses (Reproduced from Moustapha & Kacker, 1990, Fig. 6). .................................................................................. 37
2.1-13 Effect of incidence on secondary losses (Reproduced from Moustapha & Kacker, 1990, Fig. 7). .......................................................... 38
2.1-14 Evaluation of Ainley and Mathieson profile incidence loss correlations (left) and improved profile incidence loss correlation (right) (Reproduced from Moustapha & Kacker, 1990, Fig. 8 and 11). ................................................................. 39
2.1-15 Evaluation of Ainley and Mathieson secondary incidence loss correlation (left) and improved secondary incidence loss correlation (right) (Reproduced from Moustapha & Kacker, 1990, Fig. 12 and 15). ........................................ 39
2.1-16 Evaluation of the profile loss correlation of Moustapha & Kacker (1990) (left) and the new correlation by Benner et al. (1995) (right) at off-design incidence (Reproduced from Benner et al., 1995, Figs. 10, 12). ....................... 41
2.1-17 Profile loss ratio against Reynolds number effect (Reproduced from Craig & Cox, 1970, Fig. 3). ................................................................. 42
2.1-18 Lift parameter, $F_L$ (Reproduced from Craig & Cox, 1970, Fig. 4). ......... 43
2.1-19 Basic profile loss (Reproduced from Craig & Cox, 1970, Fig. 5). ............... 43
2.1-20 Secondary loss-aspect ratio factor (Reproduced from Craig & Cox, 1970, Fig. 17) 44
2.1-21 Secondary loss-basic loss factor (Reproduced from Craig & Cox, 1970, Fig. 18). 44
2.1-22 Shrouded efficiency loss (Reproduced from Craig & Cox, 1970, Fig. 21). .... 45
2.2-1 Relationship between gas outlet angles and $\cos^{-1}(o/s)$ for straight-backed blades operating at low Mach numbers (Reproduced from Ainley & Mathieson, 1951, Fig. 5). ......................................................... 46

3.1-1 Representation of the KTH Test Turbine 1-stage configuration in AXIAL by Concepts NREC and TML by GKN Aerospace. Stator contours in red, passage dimensions in blue, stage in- and outlet marked by dashed lines, and unaccounted geometrical features in green (real geometry). ......................... 49
3.1-2 KTH Test Turbine stator and rotor geometries. ............................................. 50
3.1-3 Blade stacking (Reproduced from Harrison, 1992, Fig. 1). .............................. 51
3.1-4 Stator lean angle geometry definition in AXIAL by Concepts NREC, as illustrated in the software. ......................................................... 52
3.1-5 Stator lean angle geometry convention, illustrating the difference between LE and TE definitions. View of axial plane at stage inlet looking downstream. ..... 53
3.1-6 Stator with contoured endwall, KTH Test Turbine stage 4b. ......................... 54
3.1-7 Blade geometry stations, as depicted in AXIAL by Concepts NREC. .............. 54
3.1-8 Illustration of actual evaluated geometry (red lines) and real geometry (dashed line). ................................................................................ 54
3.1-9 Stator blade approximate throat location from TE. ........................................ 55
3.1-10 Illustration of throat location, with unaccounted throat area due to contoured endwall geometry marked in yellow. ................................................................. 55
3.1-11 Illustration of stator throat plane, simplified as if it were completely axial, with the area unaccounted for due to the contoured endwall geometry marked in yellow. ................................................................. 56
3.1-12 Picture of stage midsection geometry between stator and rotor. .................... 58
3.1-13 Illustration of actual evaluated geometry (red and blue lines) and real geometry (dashed green lines). ................................................................. 58
3.1-14 Correction for TE passage area of stator due to contraction of midsection. Axial view of stator TE looking upstream towards inlet. ............................... 59
3.1-15 Stator-rotor cavity with purge flow visible. ................................................. 60
3.1-16 The one-dimensional mixing model (Reproduced from Hartsel, 1972, Fig. 2). ... 61
3.2-1 Stator and rotor geometries of the KTH Test Turbine legacy stages 1 and 2 (Reproduced from Wei, 2000, Fig. 3.1.1). ................................................. 64
3.3-1 Small aerodynamic turbine rig schematic (Reproduced from Ewen et al., 1973, Fig. 2). ................................................................. 66
3.3-2 Stage by Ewen, Huber and Mitchell (1973) stator and rotor geometries (Reproduced from Ewen et al., 1973, Fig. 12). ........................................ 67
4.1-1 Normalised total-to-static efficiency at design PR (AXIAL). ............................ 76
4.1-2 Normalised total-to-static efficiency at design PR (TML). ............................ 76
4.1-3 Normalised mass flow at design PR. Normalised to the point of maximum experimental efficiency (4800 RPM, VR 0.51). .................................................. 77
4.1-4 Aero power at design PR (AXIAL). .............................................................. 77
4.1-5 Aero power at design PR (TML). .............................................................. 77
4.1-6 Normalised stage work output ($\Delta h_{tt}$) at design PR. ................................ 78
4.1-7 Enthalpy based degree of reaction at design PR. ........................................... 78
4.1-8 Pressure based degree of reaction at design PR. .......................................... 78
4.1-9 Normalised total-to-static efficiency at elevated PR, enlarged view (AXIAL). .... 81
4.1-10 Normalised total-to-static efficiency at elevated PR (TML). ......................... 82
4.1-11 Normalised total-to-static efficiency at elevated PR, enlarged view (TML). .... 82
4.1-12 Normalised mass flow at elevated PR. Normalised to the point of maximum experimental efficiency (9800 RPM, VR 0.55). ........................................... 82
4.1-13 Mach number at stator exit, the point of the highest absolute velocity of the stage, for the loss models in TML at elevated PR. ........................................... 83
4.1-14 Illustration of relation between mass flow and Mach number, with the jump in mass flow coinciding with the abs. velocity decreasing below Mach 1. .......... 83
4.1-15 Aero power at elevated PR (AXIAL). ......................................................... 83
4.1-16 Aero power at elevated PR (TML). ......................................................... 84
4.1-17 Aero power at elevated PR, enlarged view (TML). ........................................ 84
4.1-18 Normalised stage work output ($\Delta h_{tt}$) at elevated PR. ........................... 84
4.1-19 Enthalpy based degree of reaction at elevated PR. ...................................... 85
4.1-20 Pressure based degree of reaction at elevated PR. ...................................... 85
4.1-21 Difference in estimated total to static efficiency between elevated and design PR. ................................................................. 87
4.1-22 Difference in estimated mass flow between elevated and design PR. ............ 87
4.1-23 Difference in estimated aero power between elevated and design PR. .......... 87
4.1-24 Difference in estimated enthalpy based degree of reaction between elevated and design PR. ................................................................. 88
4.1-25 Difference in estimated pressure based degree of reaction between elevated and design PR. ................................................................. 88
4.1-26 Difference in estimated CC stator (a) and rotor (b) loss coefficients between elevated and design PR (AXIAL). ................................. 89
4.1-27 Difference in estimated MK stator (a) and rotor (b) loss coefficients between elevated and design PR (AXIAL). ................................. 90
4.1-28 Difference in estimated MK-A stator (a) and rotor (b) loss coefficients between elevated and design PR (TML). ................................. 90
4.1-29 Comparison of efficiency and mass flow estimation accuracy at design PR at VR 0.48 (a) and elevated PR at VR 0.54-0.55 (b). ................................. 91
4.2-1 CC and MK stator loss coefficient distribution over the complete VR range at design PR (AXIAL). ................................................................. 92
4.2-2 MK-A and KO-A stator loss coefficient distribution over the complete VR range at design PR (TML). ................................................................. 93
4.2-3 Comparison of stator profile loss coefficient distribution over the complete VR range at design PR. ................................................................. 94
4.2-4 Comparison of stator secondary loss coefficient distribution over the complete VR range at design PR. ................................................................. 94
4.2-5 Comparison of stator trailing edge loss coefficient distribution over the complete VR range at design PR. ................................................................. 95
4.2-6 Comparison of stator incidence loss coefficient distribution over the complete VR range at design PR. ................................................................. 96
4.2-7 Craig & Cox (CC) rotor loss coefficient distribution over the complete VR range at design PR (AXIAL). ................................................................. 97
4.2-8 Moustapha & Kacker (MK) rotor loss coefficient distribution over the complete VR range at design PR (AXIAL). ................................................................. 97
4.2-9 MK-A rotor loss coefficient distribution over the complete VR range at design PR (TML). ................................................................. 98
4.2-10 KO-A rotor loss coefficient distribution over the complete VR range at design PR (TML). ................................................................. 99
4.2-11 Comparison of BSM and MK loss models, displaying relative difference of loss coefficient distribution over the complete VR range (AXIAL). ................................................................. 99
4.2-12 Comparison of rotor profile loss coefficient distribution over the complete VR range at design PR. ................................................................. 100
4.2-13 Comparison of rotor secondary loss coefficient distribution over the complete VR range at design PR. ................................................................. 101
4.2-14 Comparison of rotor tip clearance loss coefficient distribution over the complete VR range at design PR. ................................................................. 102
4.2-15 Comparison of rotor trailing edge loss coefficient distribution over the complete VR range at design PR. ................................................................. 102
4.2-16 Comparison of rotor incidence loss coefficient distribution over the complete VR range at design PR. ................................................................. 103
4.2-17 Visualisation of efficiency (dark lines) dependence on rotor incidence angle (bright lines) at design PR. ................................................................. 103
4.2-18 Visualisation of CC and MK rotor loss coefficient distribution at design PR as a function of rotor incidence angle (AXIAL). ................................................................. 104
4.3-1  Total loss coefficients of the loss models in the AXIAL and TML software at the operating point of maximum efficiency, distributed on stator and rotor fractions. Design PR: \( PR_{ts} \) 1.24, ≈5000 RPM, VR 0.54; Elevated PR: \( PR_{ts} \) 2.08, ≈9400 RPM, VR 0.54.  
4.3-2  Stator/rotor loss distribution (%) at design PR compared to elevated PR for the individual loss models in the AXIAL and TML software at the operating point of maximum efficiency, distributed on stator and rotor fractions. Design PR: \( PR_{ts} \) 1.24, ≈5000 RPM, VR 0.54; Elevated PR: \( PR_{ts} \) 2.08, ≈9400 RPM, VR 0.54.  
4.3-3  Difference in total loss coefficients between elevated PR and design PR for the individual loss models in the AXIAL and TML software at the operating point of maximum efficiency, distributed on stator and rotor fractions. Design PR: \( PR_{ts} \) 1.24, ≈5000 RPM, VR 0.54; Elevated PR: \( PR_{ts} \) 2.08, ≈9400 RPM, VR 0.54.  
4.3-4  Turbine stage velocity triangles at design PR at the operating point of maximum efficiency as predicted by the MK-A loss model in TML (5000 RPM, VR 0.54).  
4.3-5  Turbine stage velocity triangles at elevated PR at the operating point of maximum efficiency as predicted by the MK-A loss model in TML (9430 RPM, VR 0.54).  
4.4-1  Individual effect of stator correction coefficients for throat and outlet area on estimated efficiency, using CC and BSM loss models.  
4.4-2  Combined effect of stator correction coefficients for throat and outlet area on estimated efficiency, using CC and BSM loss models.  
4.4-3  Individual effect of stator correction coefficients for throat and outlet area on estimated mass flow, using CC and BSM loss models.  
4.4-4  Combined effect of stator correction coefficients for throat and outlet area on estimated mass flow, using CC and BSM loss models.  
4.4-5  Effect of stator secondary loss coefficient factor (0.8) on estimated total-to-static efficiency (a) and mass flow (b) using CC, BSM and MK loss models.  
4.4-6  Effect of stator secondary loss coefficient factor (0.8) on estimated aero power (a) and degree of reaction (b) using CC, BSM and MK loss models.  
4.4-7  Effect of standard (1.6\( \mu \)m) vs. zero roughness on estimated efficiency, using CC, BSM and MK loss models.  
4.4-8  Effect of roughness on estimated efficiency at design and elevated PR, using CC and BSM loss models.  
4.4-9  Effect of roughness on mass flow at design and elevated PR, using CC and BSM loss models.  
4.4-10 Effect of roughness on the rotor loss coefficients of the BSM loss model at design and elevated PR.  
4.4-11 Effect of default vs. nominal Re limit on estimated efficiency, using CC, BSM and MK loss models.  
4.4-12 Effect of Re limit on estimated efficiency at design and elevated PR, using CC and BSM loss models.  
4.4-13 Effect of Re limit on mass flow, using CC and BSM loss models.  
4.4-14 Effect of Re limit on the rotor loss coefficients of the BSM loss model at design and elevated PR.  
4.4-15 Effect of Re limit on the rotor loss coefficients of the CC loss model at design and elevated PR.
4.4-16 Effect of real stator lean angle on estimated efficiency, as compared to zero lean angle, for CC and BSM loss models at design PR. ................................................................. 119
4.4-17 Effect of stator lean angle on estimated efficiency, as compared to zero lean angle, for CC and BSM loss models at design PR. ................................................................. 119
4.4-18 Effect of stator lean angle on stator loss distribution, as compared to zero lean angle, for the BSM loss models at design PR. ................................................................. 120
4.4-19 Effect of stator lean angle on rotor loss distribution, as compared to zero lean angle, for the BSM loss model at design PR. ................................................................. 121
4.4-20 Effect of stator lean angle on stator spanwise static pressure distribution for the BSM loss model at design PR. ................................................................. 121
4.4-21 Effect of stator lean angle on spanwise rotor incidence angle for the BSM loss model at design PR. ................................................................. 122
4.5-1 Effect of purge flow on estimated total-to-static efficiency at design PR (4380 RPM, VR 0.48) and elevated PR (10350 RPM, VR 0.60). ................................................................. 123
4.5-2 Effect of purge flow on estimated total-to-total efficiency at design PR (4380 RPM, VR 0.47) and elevated PR (10350 RPM, VR 0.60). ................................................................. 123
4.5-3 Effect of purge flow on estimated inlet mass flow at design PR (4380 RPM, VR 0.47) and elevated PR (10350 RPM, VR 0.60). ................................................................. 124
4.5-4 Effect of purge flow on estimated aero power at design PR (4380 RPM, VR 0.47) and elevated PR (10350 RPM, VR 0.60). ................................................................. 124
4.5-5 Effect of purge flow on estimated enthalpy based degree of reaction at design PR (4380 RPM, VR 0.47) and elevated PR (10350 RPM, VR 0.60). ................................................................. 124
4.5-6 Effect of purge flow on estimated pressure based degree of reaction at design PR (4380 RPM, VR 0.47) and elevated PR (10350 RPM, VR 0.60). ................................................................. 125
4.5-7 Comparison of normalised total-to-static efficiency for varying purge flow rates at design and elevated PR using the MK-C2V loss model. ................................................................. 125
4.5-8 Comparison of normalised inlet mass flow for varying purge flow rates at design and elevated PR using the MK-C2V loss model. ................................................................. 126
4.5-9 Mach number at stator exit, the point of the highest absolute velocity of the stage, for the different purge flow rates at elevated PR. ................................................................. 127
4.5-10 Comparison of normalised aero power for varying purge flow rates at design and elevated PR using the MK-C2V loss model. ................................................................. 127
4.5-11 Comparison of enthalpy based degree of reaction for varying purge flow rates at design and elevated PR using the MK-C2V loss model. ................................................................. 128
4.5-12 Comparison of pressure based degree of reaction for varying purge flow rates at design and elevated PR using the MK-C2V loss model. ................................................................. 128
4.6-1 Total to total efficiency of the KTH Legacy Stage (AXIAL). ................................................................. 129
4.6-2 Total to total efficiency of the KTH Legacy Stage, enlarged view (AXIAL). ................................................................. 130
4.6-3 Total to total efficiency of the KTH Legacy Stage (TML). ................................................................. 130
4.6-4 Total to total efficiency of the KTH Legacy Stage, enlarged view (TML). ................................................................. 130
4.6-5 Total to total efficiency of the stage by Ewen et al. (1973) (AXIAL). ................................................................. 131
4.6-6 Total to total efficiency of the stage by Ewen et al. (1973) (TML). ................................................................. 132
List of Tables

1.4-1 KTH test turbine data (Dahlqvist & Fridh, 2015, Sodergard et al., 1989). 19
1.4-2 KTH test turbine air supply system data (Sodergard et al., 1989). 19

3.1-1 Parameters of stator and rotor in the KTH Test Turbine stage 4b. 50
3.1-2 1D simulation inlet aero parameters: total-to-static pressure ratio, total pressure at inlet, total temp. at inlet, operating range and corresponding velocity ratio. 62
3.1-3 1D purge flow simulation inlet aero parameters: total-to-static pressure ratio, total pressure at inlet, total temp. at inlet, operating range and corresponding velocity ratio. 62
3.1-4 1D purge flow mass fractions (MFR). 63
3.2-1 KTH Test Turbine legacy stage 1 geometry. 63
3.2-2 Parameters of stator and rotor in the KTH Test Turbine legacy stage. 64
3.2-3 1D simulation inlet aero parameters of the KTH Test Turbine legacy stage. 65
3.3-1 Geometry of turbine stage detailed by Ewen et al. (1973). 66
3.3-2 Parameters of stator and rotor in the turbine detailed by Ewen et al. (1973). 67
3.3-3 1D simulation inlet aero parameters of the turbine described in Ewen et al. (1973). 68
3.4-1 AXIAL loss models, deviation models and their abbreviations as used in this report. AM represents the deviation model by Ainley and Mathieson described in Chapter 2.2. 69
3.5-1 TML loss models and deviation models used for simulation, where AM represents the deviation model by Ainley and Mathieson, and \( \cos^{-1}(At/An) \) and \( \cos^{-1}(o/s) \) are simplifications of the same model, all described in Chapter 2.2. 71
3.5-2 TML loss models and deviation models used for purge flow simulation. 72

4.3-1 Turbine stage velocity triangle data at design and elevated PR. 108

A.1-1 Geometrical parameters of stator and rotor in the KTH Test Turbine. 152
A.2-1 Geometrical parameters of stator and rotor in the KTH Test Turbine Legacy Stage. 153
A.3-1 Geometrical parameters of stator and rotor in the stage by Ewen et al. (1973). 154

B.1-1 KTH Test Turbine experimental performance data. 155
B.1-2 KTH Test Turbine inlet aero experimental data. 156
B.1-3 KTH Test Turbine experimental performance data at elevated pressure ratio. 156
B.1-4 KTH Test Turbine inlet aero experimental data at elevated pressure ratio. 157
B.1-5 KTH Test Turbine purge flow experimental performance data. 157
B.1-6 KTH Test Turbine inlet aero experimental data for purge flow at design pressure ratio. 157
<table>
<thead>
<tr>
<th>Table</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>B.1-7</td>
<td>KTH Test Turbine purge flow experimental performance data.</td>
<td>158</td>
</tr>
<tr>
<td>B.1-8</td>
<td>KTH Test Turbine inlet aero experimental data for purge flow at design pressure ratio.</td>
<td>158</td>
</tr>
<tr>
<td>B.2-1</td>
<td>KTH Test Turbine legacy stages experimental data, extracted from Figs. 4.1.1 and 4.2.1 of the report by Wei (2000).</td>
<td>158</td>
</tr>
<tr>
<td>B.3-1</td>
<td>Ewen, Huber and Mitchell (1973) stage experimental data, extracted from Fig 2 of the report by Ewen et al. (1973).</td>
<td>159</td>
</tr>
<tr>
<td>C.1-1</td>
<td>AXIAL loss models, deviation models and their abbreviations as used in this report. AM represents the deviation model by Ainley and Mathieson described in Chapter 2.2.</td>
<td>160</td>
</tr>
<tr>
<td>C.2-1</td>
<td>TML loss models and deviation models used for simulation, where AM represents the deviation model by Ainley and Mathieson, and ( \cos^{-1}(At/An) ) and ( \cos^{-1}(o/s) ) are simplifications of the same model, all described in 2.2.</td>
<td>161</td>
</tr>
<tr>
<td>C.2-2</td>
<td>TML loss models and deviation models used for purge flow simulation.</td>
<td>161</td>
</tr>
</tbody>
</table>
## Appendix A

### Blade Data

#### A.1 KTH Test Turbine Stage 4b

<table>
<thead>
<tr>
<th>Parameter</th>
<th>KTH Test Turbine Stage 4b</th>
<th>Stator</th>
<th>Rotor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hub radius at LE [mm]</td>
<td></td>
<td>177.5</td>
<td>177.5</td>
</tr>
<tr>
<td>Hub radius at TE [mm]</td>
<td></td>
<td>177.5</td>
<td>177.5</td>
</tr>
<tr>
<td>Tip radius at LE [mm]</td>
<td></td>
<td>211.5</td>
<td>204.0</td>
</tr>
<tr>
<td>Tip radius at TE [mm]</td>
<td></td>
<td>205.0</td>
<td>211.6</td>
</tr>
<tr>
<td>Span at LE ($h_{LE}$) [mm]</td>
<td></td>
<td>34.0</td>
<td>27.5</td>
</tr>
<tr>
<td>Span at TE ($h_{TE}$) [mm]</td>
<td></td>
<td>26.5</td>
<td>34.1</td>
</tr>
<tr>
<td>Nominal flow turning [$^\circ$]</td>
<td></td>
<td>108.1</td>
<td>128.7</td>
</tr>
<tr>
<td>Blade inlet angle ($\beta$) [$^\circ$]</td>
<td></td>
<td>-33.5</td>
<td>58.3</td>
</tr>
<tr>
<td>Blade outlet angle ($\beta$) [$^\circ$]</td>
<td></td>
<td>74.6</td>
<td>-70.4</td>
</tr>
<tr>
<td>Blade LE lean angle [$^\circ$]</td>
<td></td>
<td>12.2</td>
<td>0.0</td>
</tr>
<tr>
<td>Number of tip seals [-]</td>
<td></td>
<td>-</td>
<td>4</td>
</tr>
<tr>
<td>Tip seal clearance [mm]</td>
<td></td>
<td>-</td>
<td>0.5</td>
</tr>
<tr>
<td>Equivalent tip seal clearance [mm]</td>
<td></td>
<td>-</td>
<td>0.279</td>
</tr>
<tr>
<td>Number of blades</td>
<td></td>
<td>42</td>
<td>58</td>
</tr>
<tr>
<td>Chord ($C$) [mm]</td>
<td></td>
<td>34.7</td>
<td>26.6</td>
</tr>
<tr>
<td>Stagger angle [$^\circ$]</td>
<td></td>
<td>46.0</td>
<td>-22.1</td>
</tr>
<tr>
<td>Aspect ratio ($h_{TE}/C$) [mm]</td>
<td></td>
<td>0.764</td>
<td>1.284</td>
</tr>
<tr>
<td>Solidity at midspan (RMS radius) ($C/s_{TE}$) [-]</td>
<td></td>
<td>1.213</td>
<td>1.255</td>
</tr>
<tr>
<td>Stator-rotor spacing [mm]</td>
<td></td>
<td>10.0</td>
<td></td>
</tr>
<tr>
<td>Blade roughness (RMS) [$\mu$m]</td>
<td></td>
<td>1.6</td>
<td>1.6</td>
</tr>
</tbody>
</table>

**Table A.1-1:** Geometrical parameters of stator and rotor in the KTH Test Turbine.
### A.2 KTH Test Turbine Legacy Stage

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Stator</th>
<th>Rotor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hub radius at LE [mm]</td>
<td>178.0</td>
<td>177.5</td>
</tr>
<tr>
<td>Hub radius at TE [mm]</td>
<td>178.0</td>
<td>177.5</td>
</tr>
<tr>
<td>Tip radius at LE [mm]</td>
<td>202.0</td>
<td>202.5</td>
</tr>
<tr>
<td>Tip radius at TE [mm]</td>
<td>202.0</td>
<td>202.5</td>
</tr>
<tr>
<td>Span at LE (h_{LE}) [mm]</td>
<td>24.0</td>
<td>25.0</td>
</tr>
<tr>
<td>Span at TE (h_{TE}) [mm]</td>
<td>24.0</td>
<td>25.0</td>
</tr>
<tr>
<td>Nominal flow turning (^\circ)</td>
<td>74.5</td>
<td>106.5</td>
</tr>
<tr>
<td>Blade inlet angle (\beta) ([^\circ])</td>
<td>0.0</td>
<td>43.0</td>
</tr>
<tr>
<td>Blade outlet angle (\beta) ([^\circ])</td>
<td>74.5</td>
<td>-63.5</td>
</tr>
<tr>
<td>Optimum incidence ([^\circ])</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>Blade lean angle ([^\circ])</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>Radius of curvature of suction side surface between LE and LE throat [mm]</td>
<td>5.5</td>
<td>10</td>
</tr>
<tr>
<td>Radius of curvature of suction side surface between TE throat and TE [mm]</td>
<td>155</td>
<td>67</td>
</tr>
<tr>
<td>Radius of LE (r_{LE}) [mm]</td>
<td>2.0</td>
<td>1.5</td>
</tr>
<tr>
<td>Thickness of TE (t_{TE}) [mm]</td>
<td>0.253</td>
<td>0.266</td>
</tr>
<tr>
<td>Wedge angle at LE ([^\circ])</td>
<td>60</td>
<td>56</td>
</tr>
<tr>
<td>Wedge angle at TE ([^\circ])</td>
<td>9</td>
<td>18</td>
</tr>
<tr>
<td>Shroud axial chord [mm]</td>
<td>-</td>
<td>29.6</td>
</tr>
<tr>
<td>Shroud thickness [mm]</td>
<td>-</td>
<td>1.85</td>
</tr>
<tr>
<td>Number of tip seals [-]</td>
<td>-</td>
<td>2</td>
</tr>
<tr>
<td>Tip seal clearance [mm]</td>
<td>-</td>
<td>0.5</td>
</tr>
<tr>
<td>Equivalent tip seal clearance [mm]</td>
<td>-</td>
<td>0.3737</td>
</tr>
<tr>
<td>Number of blades</td>
<td>42</td>
<td>64</td>
</tr>
<tr>
<td>Chord ((C)) [mm]</td>
<td>40.5</td>
<td>26.4</td>
</tr>
<tr>
<td>Axial chord ((C_x)) [mm]</td>
<td>24.8</td>
<td>24.7</td>
</tr>
<tr>
<td>Stagger angle ([^\circ])</td>
<td>52.3</td>
<td>-20.8</td>
</tr>
<tr>
<td>Aspect ratio ((h_{TE}/C)) [mm]</td>
<td>0.593</td>
<td>0.947</td>
</tr>
<tr>
<td>Pitch at midspan (RMS radius) ((s)) [mm]</td>
<td>28.48</td>
<td>18.69</td>
</tr>
<tr>
<td>Solidity at midspan (RMS radius) ((C / s_{TE})) [-]</td>
<td>1.42</td>
<td>1.41</td>
</tr>
<tr>
<td>Maximum blade thickness ((t_{mx})) [mm]</td>
<td>7.9</td>
<td>8.9</td>
</tr>
<tr>
<td>Throat width ((o)) [mm]</td>
<td>7.60</td>
<td>7.51</td>
</tr>
<tr>
<td>Stator-rotor spacing [mm]</td>
<td>12.0</td>
<td></td>
</tr>
<tr>
<td>Blade roughness (RMS) [\mu m]</td>
<td>1.6</td>
<td>1.6</td>
</tr>
</tbody>
</table>

| Table A.2-1: Geometrical parameters of stator and rotor in the KTH Test Turbine Legacy Stage. |
### A.3 Stage by Ewen et al. (1973)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Stator</th>
<th>Rotor</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hub radius at LE [mm]</td>
<td>110</td>
<td>110</td>
</tr>
<tr>
<td>Hub radius at TE [mm]</td>
<td>110</td>
<td>110</td>
</tr>
<tr>
<td>Tip radius at LE [mm]</td>
<td>126.5</td>
<td>126.5</td>
</tr>
<tr>
<td>Tip radius at TE [mm]</td>
<td>126.5</td>
<td>126.5</td>
</tr>
<tr>
<td>Span at LE ($h_{LE}$) [mm]</td>
<td>16.5</td>
<td>16.5</td>
</tr>
<tr>
<td>Span at TE ($h_{TE}$) [mm]</td>
<td>16.5</td>
<td>16.5</td>
</tr>
<tr>
<td>Nominal flow turning [$^\circ$]</td>
<td>74.8</td>
<td>106</td>
</tr>
<tr>
<td>Blade inlet angle ($\beta$) [$^\circ$]</td>
<td>0.0</td>
<td>34.0</td>
</tr>
<tr>
<td>Blade outlet angle ($\beta$) [$^\circ$]</td>
<td>74.8</td>
<td>-72.0</td>
</tr>
<tr>
<td>Optimum incidence [$^\circ$]</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>Blade lean angle [$^\circ$]</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>Radius of curvature of suction side surface between LE and LE throat [mm]</td>
<td>41</td>
<td>5.4</td>
</tr>
<tr>
<td>Radius of curvature of suction side surface between TE throat and TE [mm]</td>
<td>171</td>
<td>33</td>
</tr>
<tr>
<td>Radius of LE ($r_{LE}$) [mm]</td>
<td>2.28</td>
<td>0.99</td>
</tr>
<tr>
<td>Thickness of TE ($t_{te}$) [mm]</td>
<td>1.02</td>
<td>0.635</td>
</tr>
<tr>
<td>Wedge angle at LE [$^\circ$]</td>
<td>17</td>
<td>38</td>
</tr>
<tr>
<td>Wedge angle at TE [$^\circ$]</td>
<td>8</td>
<td>10</td>
</tr>
<tr>
<td>Shroud axial chord [mm]</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Shroud thickness [mm]</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Number of tip seals [-]</td>
<td>-</td>
<td>1</td>
</tr>
<tr>
<td>Tip seal clearance [mm]</td>
<td>-</td>
<td>0.254</td>
</tr>
<tr>
<td>Number of blades</td>
<td>20</td>
<td>44</td>
</tr>
<tr>
<td>Chord ($C$) [mm]</td>
<td>51.3</td>
<td>22.0</td>
</tr>
<tr>
<td>Axial chord ($C_x$) [mm]</td>
<td>30.5</td>
<td>16.5</td>
</tr>
<tr>
<td>Stagger angle [$^\circ$]</td>
<td>53.5</td>
<td>-41.4</td>
</tr>
<tr>
<td>Aspect ratio ($h_{TE}/C$) [mm]</td>
<td>0.322</td>
<td>0.75</td>
</tr>
<tr>
<td>Pitch at midspan (RMS radius) ($s$) [mm]</td>
<td>37.24</td>
<td>16.93</td>
</tr>
<tr>
<td>Solidity at midspan (RMS radius) ($C/s_{TE}$) [-]</td>
<td>1.378</td>
<td>1.300</td>
</tr>
<tr>
<td>Maximum blade thickness ($t_{mx}$) [mm]</td>
<td>7.5</td>
<td>3.8</td>
</tr>
<tr>
<td>Throat width ($o$) [mm]</td>
<td>9.93</td>
<td>6.73</td>
</tr>
<tr>
<td>Stator-rotor spacing [mm]</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Blade roughness (RMS) [$\mu$m]</td>
<td>1.6</td>
<td>1.6</td>
</tr>
</tbody>
</table>

**Table A.3-1:** Geometrical parameters of stator and rotor in the stage by Ewen et al. (1973).
Appendix B

Experimental Data

B.1 KTH Test Turbine Stage 4b

B.1.1 Performance Simulation Data

Design Pressure Ratio

Table B.1-1: KTH Test Turbine experimental performance data.

<table>
<thead>
<tr>
<th>Point</th>
<th>Speed (RPM)</th>
<th>Veloc. Ratio ($\nu_{t,s}$)</th>
<th>Efficiency ($\eta_{t,s}$)</th>
<th>Mass Flow (kg/s)</th>
<th>Torque (Nm)</th>
<th>Power (kW)</th>
<th>Reaction (Pressure)</th>
<th>Reaction (Enthalpy)</th>
<th>Turbine Constant ($m^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2607</td>
<td>0.278</td>
<td>Conf.</td>
<td>1.795</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.035</td>
<td>0.038</td>
<td>0.00730</td>
</tr>
<tr>
<td>2</td>
<td>2819</td>
<td>0.302</td>
<td>Conf.</td>
<td>1.779</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.051</td>
<td>0.054</td>
<td>0.00725</td>
</tr>
<tr>
<td>3</td>
<td>3829</td>
<td>0.409</td>
<td>Conf.</td>
<td>1.724</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.109</td>
<td>0.116</td>
<td>0.00707</td>
</tr>
<tr>
<td>4</td>
<td>3891</td>
<td>0.416</td>
<td>Conf.</td>
<td>1.722</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.113</td>
<td>0.121</td>
<td>0.00705</td>
</tr>
<tr>
<td>5</td>
<td>4452</td>
<td>0.477</td>
<td>Conf.</td>
<td>1.679</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.144</td>
<td>0.153</td>
<td>0.00694</td>
</tr>
<tr>
<td>6</td>
<td>4510</td>
<td>0.482</td>
<td>Conf.</td>
<td>1.686</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.145</td>
<td>0.154</td>
<td>0.00692</td>
</tr>
<tr>
<td>7</td>
<td>4806</td>
<td>0.513</td>
<td>Conf.</td>
<td>1.670</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.160</td>
<td>0.170</td>
<td>0.00686</td>
</tr>
<tr>
<td>8</td>
<td>5150</td>
<td>0.552</td>
<td>Conf.</td>
<td>1.654</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.176</td>
<td>0.187</td>
<td>0.00682</td>
</tr>
<tr>
<td>9</td>
<td>5195</td>
<td>0.555</td>
<td>Conf.</td>
<td>1.656</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.177</td>
<td>0.188</td>
<td>0.00681</td>
</tr>
<tr>
<td>10</td>
<td>5626</td>
<td>0.603</td>
<td>Conf.</td>
<td>1.637</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.195</td>
<td>0.206</td>
<td>0.00674</td>
</tr>
<tr>
<td>11</td>
<td>6053</td>
<td>0.648</td>
<td>Conf.</td>
<td>1.633</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.210</td>
<td>0.222</td>
<td>0.00670</td>
</tr>
</tbody>
</table>
Table B.1-2: KTH Test Turbine inlet aero experimental data.

<table>
<thead>
<tr>
<th>Point</th>
<th>Pressure Ratio ($PR_{ss}$)</th>
<th>Static Inlet Pressure (kPa)</th>
<th>Total Inlet Pressure (kPa)</th>
<th>Total Inlet Temp. (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.231</td>
<td>124.62</td>
<td>125.27</td>
<td>302.80</td>
</tr>
<tr>
<td>2</td>
<td>1.230</td>
<td>124.28</td>
<td>124.92</td>
<td>301.35</td>
</tr>
<tr>
<td>3</td>
<td>1.231</td>
<td>124.33</td>
<td>124.33</td>
<td>302.97</td>
</tr>
<tr>
<td>4</td>
<td>1.231</td>
<td>124.20</td>
<td>124.20</td>
<td>301.73</td>
</tr>
<tr>
<td>5</td>
<td>1.229</td>
<td>123.03</td>
<td>123.60</td>
<td>302.63</td>
</tr>
<tr>
<td>6</td>
<td>1.231</td>
<td>123.22</td>
<td>123.80</td>
<td>301.87</td>
</tr>
<tr>
<td>7</td>
<td>1.231</td>
<td>123.23</td>
<td>123.80</td>
<td>302.55</td>
</tr>
<tr>
<td>8</td>
<td>1.230</td>
<td>123.14</td>
<td>123.70</td>
<td>302.34</td>
</tr>
<tr>
<td>9</td>
<td>1.231</td>
<td>123.28</td>
<td>123.84</td>
<td>302.87</td>
</tr>
<tr>
<td>10</td>
<td>1.229</td>
<td>123.36</td>
<td>123.90</td>
<td>302.73</td>
</tr>
<tr>
<td>11</td>
<td>1.230</td>
<td>123.65</td>
<td>124.19</td>
<td>302.59</td>
</tr>
<tr>
<td>Average</td>
<td>1.230</td>
<td>123.56</td>
<td>124.14</td>
<td>302.40</td>
</tr>
</tbody>
</table>

Elevated Pressure Ratio

Table B.1-3: KTH Test Turbine experimental performance data at elevated pressure ratio.

<table>
<thead>
<tr>
<th>Point</th>
<th>Speed (RPM)</th>
<th>Veloc. Ratio ($\nu_{ss}$)</th>
<th>Efficiency ($\eta_s$)</th>
<th>Mass Flow (kg/s)</th>
<th>Torque (Nm)</th>
<th>Power (kW)</th>
<th>Reaction (Pressure)</th>
<th>Reaction (Enthalpy)</th>
<th>Turbine Constant ($m^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>5300</td>
<td>0.299</td>
<td>Conf.</td>
<td>3.671</td>
<td>Conf.</td>
<td>0.088</td>
<td>0.112</td>
<td>0.00619</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>6221</td>
<td>0.352</td>
<td>Conf.</td>
<td>3.596</td>
<td>Conf.</td>
<td>0.116</td>
<td>0.146</td>
<td>0.00619</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>7366</td>
<td>0.416</td>
<td>Conf.</td>
<td>3.589</td>
<td>Conf.</td>
<td>0.154</td>
<td>0.191</td>
<td>0.00615</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>8334</td>
<td>0.471</td>
<td>Conf.</td>
<td>3.552</td>
<td>Conf.</td>
<td>0.182</td>
<td>0.224</td>
<td>0.00612</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>8568</td>
<td>0.484</td>
<td>Conf.</td>
<td>3.555</td>
<td>Conf.</td>
<td>0.189</td>
<td>0.232</td>
<td>0.00612</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>9699</td>
<td>0.548</td>
<td>Conf.</td>
<td>3.554</td>
<td>Conf.</td>
<td>0.216</td>
<td>0.263</td>
<td>0.00610</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>9800</td>
<td>0.554</td>
<td>Conf.</td>
<td>3.551</td>
<td>Conf.</td>
<td>0.218</td>
<td>0.265</td>
<td>0.00609</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>10715</td>
<td>0.606</td>
<td>Conf.</td>
<td>3.555</td>
<td>Conf.</td>
<td>0.235</td>
<td>0.284</td>
<td>0.00608</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>11480</td>
<td>0.649</td>
<td>Conf.</td>
<td>3.547</td>
<td>Conf.</td>
<td>0.249</td>
<td>0.300</td>
<td>0.00605</td>
<td></td>
</tr>
</tbody>
</table>
TABLE B.1-4: KTH Test Turbine inlet aero experimental data at elevated pressure ratio.

<table>
<thead>
<tr>
<th>Point</th>
<th>Pressure Ratio ($P_{R_{in}}$)</th>
<th>Static Inlet Pressure (kPa)</th>
<th>Total Inlet Pressure (kPa)</th>
<th>Total Inlet Temp. (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.064</td>
<td>211.62</td>
<td>213.40</td>
<td>335.12</td>
</tr>
<tr>
<td>2</td>
<td>2.053</td>
<td>207.65</td>
<td>209.39</td>
<td>335.09</td>
</tr>
<tr>
<td>3</td>
<td>2.061</td>
<td>208.18</td>
<td>209.91</td>
<td>334.56</td>
</tr>
<tr>
<td>4</td>
<td>2.059</td>
<td>206.83</td>
<td>208.53</td>
<td>334.31</td>
</tr>
<tr>
<td>5</td>
<td>2.060</td>
<td>207.02</td>
<td>208.73</td>
<td>334.43</td>
</tr>
<tr>
<td>6</td>
<td>2.061</td>
<td>207.67</td>
<td>209.37</td>
<td>334.67</td>
</tr>
<tr>
<td>7</td>
<td>2.060</td>
<td>207.79</td>
<td>209.49</td>
<td>334.44</td>
</tr>
<tr>
<td>8</td>
<td>2.059</td>
<td>208.51</td>
<td>210.21</td>
<td>334.47</td>
</tr>
<tr>
<td>9</td>
<td>2.060</td>
<td>209.04</td>
<td>210.73</td>
<td>334.56</td>
</tr>
<tr>
<td>Average</td>
<td>2.060</td>
<td>208.26</td>
<td>209.97</td>
<td>334.63</td>
</tr>
</tbody>
</table>

B.1.2 Purge Flow Leakage Simulation Data

Design Pressure Ratio

TABLE B.1-5: KTH Test Turbine purge flow experimental performance data.

<table>
<thead>
<tr>
<th>Point</th>
<th>Speed (RPM)</th>
<th>Veloc. Ratio ($\eta_{ts}$)</th>
<th>Efficiency ($\eta_{ts}$)</th>
<th>Total Flow (kg/s)</th>
<th>Leakage (%)</th>
<th>Leak- age (kg/s)</th>
<th>Torque (Nm)</th>
<th>Power (kW)</th>
<th>Reaction (Pressure)</th>
<th>Reaction (Enthalpy)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4394</td>
<td>0.470</td>
<td>Conf.</td>
<td>1.773</td>
<td>0.00</td>
<td>0.0000</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.135</td>
<td>0.143</td>
</tr>
<tr>
<td>2</td>
<td>4379</td>
<td>0.469</td>
<td>Conf.</td>
<td>1.779</td>
<td>0.98</td>
<td>0.0174</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.145</td>
<td>0.154</td>
</tr>
<tr>
<td>3</td>
<td>4379</td>
<td>0.469</td>
<td>Conf.</td>
<td>1.790</td>
<td>2.03</td>
<td>0.0363</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.156</td>
<td>0.165</td>
</tr>
<tr>
<td>4</td>
<td>4380</td>
<td>0.469</td>
<td>Conf.</td>
<td>1.819</td>
<td>4.93</td>
<td>0.0897</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.186</td>
<td>0.196</td>
</tr>
</tbody>
</table>

TABLE B.1-6: KTH Test Turbine inlet aero experimental data for purge flow at design pressure ratio.

<table>
<thead>
<tr>
<th>Point</th>
<th>Pressure Ratio ($P_{R_{in}}$)</th>
<th>Static Inlet Pressure (kPa)</th>
<th>Total Inlet Pressure (kPa)</th>
<th>Total Inlet Temp. (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.230</td>
<td>128.06</td>
<td>128.68</td>
<td>302.87</td>
</tr>
<tr>
<td>2</td>
<td>1.230</td>
<td>127.93</td>
<td>128.54</td>
<td>302.78</td>
</tr>
<tr>
<td>3</td>
<td>1.230</td>
<td>127.90</td>
<td>128.51</td>
<td>302.91</td>
</tr>
<tr>
<td>4</td>
<td>1.230</td>
<td>128.39</td>
<td>128.97</td>
<td>302.51</td>
</tr>
<tr>
<td>Average</td>
<td>1.236</td>
<td>128.07</td>
<td>128.67</td>
<td>302.76</td>
</tr>
</tbody>
</table>

Elevated Pressure Ratio
### Table B.1-7: KTH Test Turbine purge flow experimental performance data.

<table>
<thead>
<tr>
<th>Point</th>
<th>Speed (RPM)</th>
<th>Veloc. Ratio ($\nu_{t_{\text{is}}}$)</th>
<th>Efficiency ($\eta_{\text{ts}}$)</th>
<th>Total Flow (kg/s)</th>
<th>Leakage (%)</th>
<th>Torque (Nm)</th>
<th>Power (kW)</th>
<th>Reaction Pressure (kPa)</th>
<th>Reaction Enthalpy (kJ/kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>10531</td>
<td>0.603</td>
<td>Conf.</td>
<td>3.657</td>
<td>0.00</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.236</td>
<td>0.286</td>
</tr>
<tr>
<td>2</td>
<td>10531</td>
<td>0.603</td>
<td>Conf.</td>
<td>3.695</td>
<td>1.11</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.246</td>
<td>0.297</td>
</tr>
<tr>
<td>3</td>
<td>10531</td>
<td>0.602</td>
<td>Conf.</td>
<td>3.717</td>
<td>2.10</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.254</td>
<td>0.305</td>
</tr>
<tr>
<td>4</td>
<td>10530</td>
<td>0.603</td>
<td>Conf.</td>
<td>3.740</td>
<td>2.71</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.260</td>
<td>0.312</td>
</tr>
<tr>
<td>5</td>
<td>10528</td>
<td>0.603</td>
<td>Conf.</td>
<td>3.747</td>
<td>3.52</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.268</td>
<td>0.319</td>
</tr>
<tr>
<td>6</td>
<td>10528</td>
<td>0.603</td>
<td>Conf.</td>
<td>3.783</td>
<td>4.22</td>
<td>Conf.</td>
<td>Conf.</td>
<td>0.274</td>
<td>0.326</td>
</tr>
</tbody>
</table>

### Table B.1-8: KTH Test Turbine inlet aero experimental data for purge flow at design pressure ratio.

<table>
<thead>
<tr>
<th>Point</th>
<th>Pressure Ratio ($PR_{\text{ss}}$)</th>
<th>Static Inlet Pressure (kPa)</th>
<th>Total Inlet Pressure (kPa)</th>
<th>Total Inlet Temp. (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.068</td>
<td>211.26</td>
<td>212.99</td>
<td>326.85</td>
</tr>
<tr>
<td>2</td>
<td>2.070</td>
<td>211.83</td>
<td>213.54</td>
<td>325.60</td>
</tr>
<tr>
<td>3</td>
<td>2.072</td>
<td>210.54</td>
<td>212.25</td>
<td>326.47</td>
</tr>
<tr>
<td>4</td>
<td>2.071</td>
<td>211.29</td>
<td>212.99</td>
<td>325.90</td>
</tr>
<tr>
<td>5</td>
<td>2.063</td>
<td>210.59</td>
<td>212.28</td>
<td>326.83</td>
</tr>
<tr>
<td>6</td>
<td>2.067</td>
<td>211.12</td>
<td>212.81</td>
<td>326.64</td>
</tr>
<tr>
<td>Average</td>
<td>2.085</td>
<td>211.11</td>
<td>212.81</td>
<td>326.38</td>
</tr>
</tbody>
</table>

### B.2 KTH Test Turbine Legacy Stage

#### Table B.2-1: KTH Test Turbine legacy stages experimental data, extracted from Figs. 4.1.1 and 4.2.1 of the report by Wei (2000).

<table>
<thead>
<tr>
<th>Point</th>
<th>Efficiency at PR=1.18-1.23</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Stage 1</td>
</tr>
<tr>
<td></td>
<td>$\nu_{t_{\text{is}}}$</td>
</tr>
<tr>
<td>1</td>
<td>0.198</td>
</tr>
<tr>
<td>2</td>
<td>0.396</td>
</tr>
<tr>
<td>3</td>
<td>0.473</td>
</tr>
<tr>
<td>4</td>
<td>0.531</td>
</tr>
<tr>
<td>5</td>
<td>0.535</td>
</tr>
<tr>
<td>6</td>
<td>0.555</td>
</tr>
<tr>
<td>7</td>
<td>0.577</td>
</tr>
<tr>
<td>8</td>
<td>0.630</td>
</tr>
<tr>
<td>9</td>
<td>0.682</td>
</tr>
<tr>
<td>10</td>
<td>0.737</td>
</tr>
</tbody>
</table>
## B.3 Stage by Ewen et al. (1973)

Table B.3-1: Ewen, Huber and Mitchell (1973) stage experimental data, extracted from Fig 2 of the report by Ewen et al. (1973).

<table>
<thead>
<tr>
<th>Point</th>
<th>$\nu_{t_{s_1}}$</th>
<th>$\eta_{tt}$</th>
<th>$\nu_{t_{s_2}}$</th>
<th>$\eta_{tt}$</th>
<th>$\nu_{t_{s_1}}$</th>
<th>$\eta_{tt}$</th>
<th>$\nu_{t_{s_2}}$</th>
<th>$\eta_{tt}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.561</td>
<td>0.745</td>
<td>0.526</td>
<td>0.762</td>
<td>0.500</td>
<td>0.769</td>
<td>0.547</td>
<td>0.811</td>
</tr>
<tr>
<td>2</td>
<td>0.679</td>
<td>0.812</td>
<td>0.574</td>
<td>0.789</td>
<td>0.569</td>
<td>0.812</td>
<td>0.640</td>
<td>0.860</td>
</tr>
<tr>
<td>3</td>
<td>0.714</td>
<td>0.833</td>
<td>0.643</td>
<td>0.826</td>
<td>0.619</td>
<td>0.842</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>4</td>
<td>0.849</td>
<td>0.867</td>
<td>0.713</td>
<td>0.854</td>
<td>0.700</td>
<td>0.869</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>
## Appendix C

### 1D Software Loss Models

#### C.1 AXIAL

<table>
<thead>
<tr>
<th>Loss Model</th>
<th>Deviation Model</th>
<th>Abbrev.</th>
<th>AXIAL Input</th>
</tr>
</thead>
<tbody>
<tr>
<td>Craig/Cox</td>
<td>AM</td>
<td>CC</td>
<td>CRAIG+COX</td>
</tr>
<tr>
<td>Benner/Sjolander/Moustapha</td>
<td>AM</td>
<td>BSM</td>
<td>AMDC+KO+MK+BSM</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>AM</td>
<td>MK</td>
<td>AMDC+KO+MK</td>
</tr>
<tr>
<td>Ainley/Mathieson, Dunham/-Came</td>
<td>AM</td>
<td>AMDC</td>
<td>AMDC_OLD</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>Sheglyaev</td>
<td>MK-S</td>
<td>AMDC+KO+MK/DEV_SHEGL</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>AM</td>
<td>MK-MSL1</td>
<td>AMDC+KO+MK/ S_AMDC_LIM</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>AM</td>
<td>MK-MSL2</td>
<td>AMDC+KO+MK/ S_AMDC_LIM_0375</td>
</tr>
</tbody>
</table>
### C.2 TML

**Table C.2-1:** TML loss models and deviation models used for simulation, where AM represents the deviation model by Ainley and Mathieson, and \( \cos^{-1}(At/An) \) and \( \cos^{-1}(o/s) \) are simplifications of the same model, all described in 2.2.

<table>
<thead>
<tr>
<th>Loss Model</th>
<th>Deviation Model</th>
<th>Abbrev.</th>
<th>TML Input</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kacker/Okapuu</td>
<td>AM</td>
<td>KO-A</td>
<td>2/2</td>
</tr>
<tr>
<td>Kacker/Okapuu</td>
<td>McDonald</td>
<td>KO-M</td>
<td>2/3</td>
</tr>
<tr>
<td>Kacker/Okapuu</td>
<td>Islam</td>
<td>KO-I</td>
<td>2/5</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>AM</td>
<td>MK-A</td>
<td>3/1</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>McDonald</td>
<td>MK-M</td>
<td>3/3</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>Islam</td>
<td>MK-I</td>
<td>3/5</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>( \cos^{-1}(At/An) )</td>
<td>MK-C1</td>
<td>3/6</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>( \cos^{-1}(o/s) )</td>
<td>MK-C2</td>
<td>3/2</td>
</tr>
<tr>
<td>Moustapha/Kacker with Volvo Tip Clearance Correction</td>
<td>( \cos^{-1}(o/s) )</td>
<td>MK-C2V</td>
<td>3/7</td>
</tr>
</tbody>
</table>

**Table C.2-2:** TML loss models and deviation models used for purge flow simulation.

<table>
<thead>
<tr>
<th>Loss Model</th>
<th>Deviation Model</th>
<th>Abbrev.</th>
<th>TML Input</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kacker/Okapuu</td>
<td>( \cos^{-1}(At/An) )</td>
<td>KO-C1</td>
<td>2/6</td>
</tr>
<tr>
<td>Kacker/Okapuu</td>
<td>( \cos^{-1}(o/s) )</td>
<td>KO-C2</td>
<td>2/2</td>
</tr>
<tr>
<td>Kacker/Okapuu with Volvo Tip Clearance Correction</td>
<td>( \cos^{-1}(o/s) )</td>
<td>KO-C2V</td>
<td>2/7</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>( \cos^{-1}(At/An) )</td>
<td>MK-C1</td>
<td>3/6</td>
</tr>
<tr>
<td>Moustapha/Kacker</td>
<td>( \cos^{-1}(o/s) )</td>
<td>MK-C2</td>
<td>3/2</td>
</tr>
<tr>
<td>Moustapha/Kacker with Volvo Tip Clearance Correction</td>
<td>( \cos^{-1}(o/s) )</td>
<td>MK-C2V</td>
<td>3/7</td>
</tr>
</tbody>
</table>