2001

PIV Measurements of Channel Flow with Multiple Rib Arrangements

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ABSTRACT OF THESIS

Harald Roclawski

The Graduate School
University of Kentucky
2001
PIV Measurements of Channel Flow with Multiple Rib Arrangements

ABSTRACT OF THESIS

A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science at the University of Kentucky

By
Harald Roclawski
Lexington, Kentucky
Director: Dr. Jamey Jacob, Assistant Professor of Mechanical Engineering
Lexington, Kentucky
2001
Abstract of Thesis

PIV Measurements of Channel Flow with Multiple Rib Arrangements

A model of a gas turbine blade cooling channel equipped with turbulators and a backward facing step geometry was examined. Up to four turbulators oriented cross-stream and inclined 45° to the flow direction were mounted in the channel. The blockage ratio $b/H$ of the turbulators and the height $h/H$ of the backward facing step was 0.125 and 0.14 respectively. The number of turbulators as well as their size was varied. In a preliminary investigation, hot-wire and pressure measurements were taken for three different Reynolds numbers (5,000, 12,000, 18,000) in the center plane of the test section. Subsequently, particle image velocimetry (PIV) measurements were made on the same geometries. Results of PIV measurements for a Reynolds number range of $Re_b=600$ to 5,000 for the turbulators and $Re_h=1,500$ to 16,200 for the backward facing step are presented, where Reynolds numbers are based on turbulator height $b$ and step height $h$, respectively. Plots of the velocity field, vorticity, reverse flow probability and RMS velocity are shown. The focus is on the steady flow behavior but also the unsteadiness of the flow is discussed in one section. Also reattachment lengths were obtained and compared among the various turbulator arrangements and the backward facing step geometry. It was found that the flow becomes periodic after three or four ribs. For one turbulator, a very large separation region was observed. The magnitude of the skin friction factor was found to be the highest for two ribs. If the first rib is replaced by a smaller rib, the skin friction factor becomes the lowest for this case. Compared to the backward facing step, the flow reattaches earlier for multiple turbulators. A dependency of reattachment length on Reynolds number was not observed.

Keywords: PIV (Particle Image Velocimetry), turbulator, channel flow, gas turbine, turbine blade cooling

Harald Roclawski
August 24, 2001
PIV Measurements of Channel Flow with Multiple Rib Arrangements

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THESIS

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Chapter 1

Introduction

The turbine section of a gas turbine engine is often characterized by the high temperatures it experiences under normal operating conditions. In order to increase the thermal efficiency $\eta_{th}$, designers of gas turbine engines attempt to achieve higher turbine inlet temperatures to increase the thermal efficiency $\eta_{th}$. Assuming an ideal gas, $\eta_{th}$ can be computed from equation 1.1 for gas turbines

$$\eta_{th} = 1 - \frac{1}{r_p^{k-1/k}}$$

where $r_p$ is the pressure ratio equal to $\left(\frac{p_{inlet}}{p_{exit}}\right)$ and $k$ is the ratio of specific heats. An optimum pressure ratio $r_{popt}$ can be found as

$$r_{popt} = \left(\frac{T_{inlet}}{T_{ambient}}\right)^{k/(2(k-1))}$$

It can be seen that the optimum pressure ratio increases with increasing turbine inlet temperatures $T_{inlet}$. Therefore, from equations 1.1 and 1.2, one can conclude that the thermal efficiency of gas turbine engines increases with increasing turbine inlet temperatures. Even a fractional improvement in performance can offer significant savings (see, for example, Mayle [1]). (An overview of gas turbine engines as well as the derivation of equations 1.1 and 1.2 can be found in Appendix A.)

The theoretical limit of turbine inlet temperatures is the melting point of the material, though in practical terms, the actual limit is lower. Today, the turbine inlet temperatures are about 1700 K for civilian engines and up to 1800 K-1900 K (Müller[2]) in engines for military utilization.

Currently available materials for turbine blades are unable to withstand long periods of exposure to these high temperatures while maintaining structural integrity, even with thermal barrier coatings (TBC), implying need for active cooling strategies as discussed in the next section applied.
1.1 Gas Turbine Blade Cooling

Several different cooling methods are available for turbine blade cooling, including film-, jet impingement-, transpiration- and internal forced convection cooling (Bathie [3]). The coolant can be air or water. Air is chosen most often, since it is readily available. Usually only the first high-pressure turbine stages need to be cooled.

Impingement cooling is a form of convection cooling. The coolant is brought radially through a center core of the blade, then turned normal to the radial direction, and passed through a series of holes so that it impinges on the inside of the blade, usually just opposite from the stagnation point. It is a very effective method for cooling local areas such as the leading edge of the blade. It can be easily adapted to the stationary blades. If there is sufficient space, it can also be applied to rotating blades.

Film cooling is an external cooling method. It involves the injection of cooling air from the inside of the blade through little holes in the blade surface into the boundary layer of the hot gases. The blade surface is covered with a thin film of cooling air. It can be used for local cooling, where air is only injected through holes at the stagnation point or for full-coverage cooling, where the whole blade is covered with small holes. The pressure of the cooling air must be high enough so that it can be injected into the boundary layer.

Another external cooling method is transpiration cooling. It is the most efficient cooling technique but is still in the development stage. In this case, the blade consists of a porous skin connected to a coolant chamber. The air is forced from the coolant chamber through the porous skin into the boundary layer, where it forms a cooling film around the whole blade. For effective cooling, the pores need to be small. Currently the pore size is on the order of 0.01-0.05 mm (Lakshminarayana [4]). Because of the small size, the pores have the tendency to become blocked by dirt, carbon and other particles. These problems, as well as oxidation problems, will decrease the cooling efficiency during the life cycle. From the aerodynamic efficiency viewpoint it is the least efficient and the most difficult one from the manufacturing and maintenance point of view.

The work presented in this thesis focuses on the flow in internal cooling passages. A sample of an internally convection cooled gas turbine blade is shown in Fig. 1.1. Gases are used in most applications since it is already available. Withdrawn from the last stages of the compressor and bypassing the combustion chamber, the coolant flows through several cooling passages equipped with turbulator ribs to maintain turbulent flow and thus enhance heat transfer. It enters the circuit from the hub and ejects at the trailing edge or blade tip. For simplicity, a straight rectangular channel with turbulator ribs on one wall is investigated in this thesis.

A more effective way of cooling turbine blades is to use a liquid coolant, since it has a much
higher specific heat, a higher heat transfer capability and provides the opportunity for evaporative cooling. This method may be applied to stationary power plants only since the size of the necessary heat exchanger for an aircraft application would be too large. But even in stationary power plants difficulties do exist in transferring the liquid to and from the rotating blades. Furthermore, the system has to be simple, leak-proof and corrosion has to be avoided.

Three different modes of heat transfer apply to internal convection cooling. The first mode is the heat transfer from the hot gas to the blade. Then the heat is transferred by conduction within the blade material. Heat transfer by convection from the blade material to the internal cooling flow is the last mode. It can be calculated from

\[ Q = h_c A_c (T_b - T_c) \]  

(1.3)

where \( h_c \) is the blade to coolant heat transfer coefficient, \( T_c \) the bulk temperature of the coolant, \( T_b \) the temperature of the blade and \( A_c \) the area exposed to the coolant. The effectiveness of convection cooling is limited by the size of the cooling passages inside the blade and the amount of cooling air available. Another disadvantage of convection cooling is that it is not possible to cool the trailing edge of the blade since it is too thin. Since air has a low specific heat, large internal surface areas and high velocities are desirable for effective cooling.

Another key factor in this interior heat removal process is maintaining turbulent flow. At the same time this results in greater pressure losses, which means that more turbine work is required to power the compressor. So there must always be a tradeoff between cooling circuit heat transfer effectiveness and pressure loss in turbine blade design studies. A significant portion of the design effort is devoted to this problem. It is especially difficult for both experimental and computational fluid dynamics (CFD) modeling because of the highly complex geometries of interior cooling air circuits, the turbulent flow, and in the case of experiments, repeating the physics encountered during high rates of blade rotation.

Since the introduction of turbine blade cooling, considerable improvements in increasing turbine inlet temperatures have been made. Fig. 1.2 shows the variation of turbine inlet temperatures over recent years. The first cooling technique applied was convection cooling. Since the introduction of turbine blade cooling (1960), turbine inlet temperatures have increased by 750 K. It can be seen that with a combination of the discussed cooling techniques very high turbine inlet temperatures can be achieved. The dashed line shows the expected trend with new materials and the application of transpiration cooling.
1.2 Previous Work

Previous experimental research in turbine blade cooling has been primarily focused on the external cooling effects (e.g., Wang et al. [5]). This is due to the relative ease of the external measurements as compared to the complex apparatus or gross simplifications required for internal duct measurements, particularly when it applies to measurements of the flow field. Previous research efforts on internal cooling have been limited in scope, typically focusing on a single aspect of the multivariant problem.

Bunker and Metzger [6] examined the local heat transfer from internal impingement cooling using temperature sensitive paint. General relations showed increased heat transfer with increased jet Reynolds number.

Bohn et al. [7] numerically and experimentally examined trailing edge cooling in turbine blades. The experiments were conducted in a scaled test rig and showed anisotropic turbulence profiles resulting in non-symmetrical coolant distribution. The numerical predictions compared reasonably well with the experimental data.

Çakan et al. [8] investigated the flow and heat transfer in a straight ribbed cooling channel. LDV measurements were taken at Re=30,000 based on the hydraulic diameter of the channel. Ribs with a blockage ratio of 10% were mounted on a single and in another case on two opposite walls of the channel. In the latter, it was found that the flow is symmetrical. The rib spacing was varied in steps of three from six to twelve. Separation zones before and after the ribs were observed. They also showed that the flow is three dimensional around the rib. A cross stream secondary flow cell was observed. From pressure measurements they found that the pressure coefficient $C_p$ increases before the ribs and decreases after the rib resulting in a large form drag. No significant difference in turbulence intensity was found when the rib spacing was varied. Regions of high heat transfer were found in the vicinity of the reattachment location and before the ribs.

In a more recent paper, Çakan and Arts [9] studied the flow in a similar test section but this time by means of PIV. Reynolds numbers investigated were 6,500 and 30,000 based on the hydraulic diameter of the channel. The rib blockage ratio of the turbulator was increased to 0.133. They found that the flow through the ribbed channel can be characterized by a series of accelerations, decelerations with separation, reattachment and redevelopment due to the sudden changes in cross-section. The ribs induce a separation and recirculation bubble. The flow reattaches at $X/e = 4.5$ (Re=6,500). Comparing both studies, no Reynolds number dependency of reattachment distance was found. Upstream of the ribs, the flow impinges on the rib, moves to the sidewalls of the channel and produces two vortices. Behind the rib a similar motion occurs due to the recirculation region; in the spanwise flow direction, two counter rotating secondary flow cells are observed.
The flow in a straight, rectangular channel with ribs on two opposite walls was investigated by Liou et al. [10] by means of LDV. The Reynolds number based on the channel hydraulic diameter was 33,000. The ribs were perforated and the effect of the rib open area ratio was investigated. Like Çakan and Arts they found a periodic accelerating and decelerating flow behavior. In contrast to the previous paper, only one secondary flow cell is observed in the spanwise flow direction. Furthermore, it was discovered that the reattachment length downstream of a rib pair is shorter than in the case of a backward-facing step. The maximum heat transfer rate was found to be dependent upon a critical range of the open area ratio governed by whether the flow treated the ribs as permeable or impermeable. In another paper by Liou et al. [11], they found out that the flow shows periodic behavior after passing three or four ribs.

A PIV Investigation of the flow in a rectangular channel with a 45° rib arrangement and a 180° bend was done by Schabacker and Bölcs [12] at Re= 45,700 based on hydraulic diameter and a rib height equal to 0.1 hydraulic diameters. Two counter rotating vortices in the spanwise flow direction were observed. The development length to achieve a fully developed flow condition is longer for the case of a 45° rib arrangement. Furthermore, the 45° ribs prevent the development of zones of recirculating flow in the upstream outer corner of the bend, and the curvature-induced secondary flows are reweakened in this section of the channel. Compared to a smooth channel, the flow recovers faster from the bend effect.

Results for the case of a stationary and rotating, rectangular, ribbed channel with a 180° bend were obtained by Servouze [13] using LDV. The flow conditions were Re=5,000 (based on hydraulic diameter), Ro=0.33 and a rib aspect ratio of 10. Ro is the Rossby number defined as $Ro = \Omega D_h/U_b$ where $\Omega$ is the angular velocity, $D_h$ the hydraulic diameter of the channel and $U_b$ the bulk velocity in the channel. In the stationary case a periodic accelerating and decelerating flow behavior was found. In contrast to other papers, secondary flow structures (vortices) in the spanwise flow direction were not observed.

Iacovides [14] performed an LDA study on the flow in a ribbed channel with a 180° bend. He investigated a stationary case at a channel Re=100,000 and two rotating cases at $Ro = \pm 0.2$ for the same Reynolds number. The rib-height to duct diameter ratio was 0.1. They also observed a periodic flow behavior. Because of the ribs, turbulence increases at the bend entry and an additional separation bubble over the first rib interval downstream of the bend exit is formed. Nevertheless, in agreement with Schabacker and Bölcs, it is claimed that the flow recovers faster from the bend effect in a ribbed channel. Especially the separation bubble along the inner wall is reduced.

Lastly, Hwang and Lai [15] examined laminar flow within a rotating multiple-pass channel with bends from a computational standpoint. Rotation was found to have a large impact on the wall
friction factor. Validations were only made with stationary experiments, however.

1.3 Experiments

The majority of past experiments on turbine blade cooling have been single point measurements, not allowing a full field analysis. This point has proven to be particular difficult when trying to validate numerical simulations which benefit from full field data. The maturation of modern optical field measurements have allowed these hurdles to be overcome, though any experiments must require extreme care in their formulation. The two papers by Çakan et al. [8] and [9] discussed above are an good example of the trend to full field measurements. After doing their first investigation with laser doppler velocimetry (LDV), particle image velocimetry (PIV) was used in the more recent paper.

The objective of this thesis is to obtain fundamental data of turbulator geometries which are used for maintaining turbulent flow and heat transfer enhancement in internal gas turbine blade cooling channels. Furthermore, one should be able to use the data obtained for validation of CFD (Computational Fluid Dynamics) codes. PIV results for ribs of equal sizes with sides $b = 24.5$ mm are presented. Rib arrangements of 1, 2, 3, and 4 turbulators are examined with equal separation distances of 152 mm in multiple turbulator runs. The ribs were oriented cross-stream and inclined $45^\circ$ to the flow direction. Only one wall is equipped with turbulators. In one case, alternating turbulators of different sizes were placed into the test section. For comparison, PIV measurements were also taken for a backward facing step geometry. For the case of two ribs, the flow was also observed between the turbulators. Prior to these experiments, in a preliminary investigation, turbulators were examined by means of hot-wire anemometry and static pressure measurements. All measurements were made along the test section centerline downstream of the step and last rib. Due to the seeding of the flow which is necessary for the PIV measurements, a filter had to be installed at the exit of the test section, resulting in a smaller pressure drop across the channel and therefore lower Reynolds numbers compared to the hot-wire and pressure measurements taken in the preliminary study. Table 1.1 provides an overview of measurements taken for the various geometries and Reynolds number ranges.
<table>
<thead>
<tr>
<th>Geometry</th>
<th>PIV</th>
<th>HWA</th>
<th>Pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>BFS</td>
<td>$Re=18,000-173,000$</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>$Re_b=1,700-16,000$</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Turbulator (Downstream of last rib)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>N=1</td>
<td>$\perp$, $45^\circ$, $Re=6,000-50,000$</td>
<td>$\perp$, $Re=85,000-188,000$</td>
<td>$\perp$, $Re=85,000-188,000$</td>
</tr>
<tr>
<td></td>
<td>$Re_b=600-5,000$</td>
<td>$Re_b=8,000-18,000$</td>
<td>$Re_b=8,000-18,000$</td>
</tr>
<tr>
<td>N=2</td>
<td>$\perp$, $Re=6,000-50,000$</td>
<td>$\perp$, $Re=85,000-188,000$</td>
<td>$\perp$, $Re=85,000-188,000$</td>
</tr>
<tr>
<td></td>
<td>$Re_b=600-5,000$</td>
<td>$Re_b=8,000-18,000$</td>
<td>$Re_b=8,000-18,000$</td>
</tr>
<tr>
<td>N=3</td>
<td>$\perp$, $45^\circ$, $Re=6,000-50,000$</td>
<td>$\perp$, $Re=85,000-188,000$</td>
<td>$\perp$, $Re=85,000-188,000$</td>
</tr>
<tr>
<td></td>
<td>$Re_b=600-5,000$</td>
<td>$Re_b=8,000-18,000$</td>
<td>$Re_b=8,000-18,000$</td>
</tr>
<tr>
<td>N=4</td>
<td>$\perp$, $Re=6,000-50,000$</td>
<td>$\perp$, $Re=85,000-188,000$</td>
<td>$\perp$, $Re=85,000-188,000$</td>
</tr>
<tr>
<td></td>
<td>$Re_b=600-5,000$</td>
<td>$Re_b=8,000-18,000$</td>
<td>$Re_b=8,000-18,000$</td>
</tr>
<tr>
<td>N=2</td>
<td>$\perp$, $Re=8,000-52,000$</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>variable size</td>
<td>$Re_b=800-5,200$</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Turbulator (Upstream of last rib)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\Delta X=2$</td>
<td>$\perp$, $Re=6,000-50,000$</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>$Re_b=600-5,000$</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>$\Delta X=5$</td>
<td>$\perp$, $Re=6,000-50,000$</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>$Re_b=600-5,000$</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

$Re$ - Reynolds number based on hydraulic diameter  
$Re_b$ - Reynolds number based on turbulator height  
$Re_h$ - Reynolds number based on step height  
$\perp$ - turbulator oriented normal to the free stream  
$45^\circ$ - turbulator oriented in $45^\circ$ to the free stream  
$\Delta X=3$ - rib spacing $x/b=3$  
$\Delta X=6$ - rib spacing $x/b=6$

Table 1.1: Experiments
Figure 1.1: Typical turbine blade cooling circuit (after Han et al. [17]).
Figure 1.2: Development of Turbine Inlet Temperatures (after Lakshminarayana [4]).
Chapter 2

Preliminary Investigation

In a preliminary investigation, a model of a turbine blade cooling channel was examined by means of hot-wire anemometry and static pressure measurements. Up to four turbulators with a blockage ratio of \( b/H = 0.125 \) were mounted on plates which could be installed into the test section. Although this simplifies the exchange of the ribs, it turned out to be a major drawback for the pressure measurements since the floor surface was not completely smooth at the union (joint) of the plates. It was always the pressure tap immediately downstream of the intersection of two joined plates from which bad data was obtained. Using Chauvenet’s Criterion, the bad data points were omitted in the plots. To obtain a better understanding of the flow, flow visualization was used [16]. A thin steel wire \((d = 0.009\)”) was covered with oil, which because of surface tension beads up to form droplets. Heated by an electric current, the oil droplets then evaporate. The smoke of each droplet forms a streamline. The Reynolds number based on the wire diameter was kept below \( \text{Re}=50 \) to prevent vortex shedding off the wire. A sample picture of the streamlines visualized by the smoke-wire technique is shown in Fig. 2.1. Unfortunately, it was not possible to visualize the flow downstream of the turbulator since the smoke was not drawn in the recirculation region behind the ribs and it rapidly diffuses because of turbulence. But it can be seen that the wind tunnel produces a nice laminar flow. It was tried to inject smoke from the inside of a bar between the turbulators, but these attempts were not successful.

2.1 Hot-Wire Anemometry (HWA)

2.1.1 Principle of Hot-Wire Anemometry

The principle of hot-wire anemometry is based on convection cooling of a heated body. A thin wire, being one resistor of a Wheatstone bridge and heated by an electrical current, is placed into
the flow. Being cooled by the flow, the wire changes its resistance. A servo amplifier keeps the bridge balanced by controlling the current through the wire so that the resistance is kept constant. The heat transfer process can be regarded as convective heat transfer from a cylinder, which is also a function of the freestream velocity $U$. Therefore, the voltage output of the bridge can be directly related to the flow velocity.

Although it is an intrusive measurement technique, HWA is a powerful tool in examining turbulence because of a high frequency response up to 11 kHz resulting in a high temporal resolution. Because of the small size of the sensor, a large spatial resolution can be obtained as well. It is possible to measure all three velocity components simultaneously with a three-component hot-wire sensor so that instantaneous velocity data can be provided. A major drawback of HWA is that only the velocity magnitude can be measured, if a single component hot-wire is used, such as in the case here. Also, reverse flow (anti-parallel to the free stream) measurements cannot be accurately measured without a change in the setup.

### 2.1.2 Setup

For the hot-wire measurements, a single-component TSI 1218-1.5 Standard Boundary Layer hot-wire probe (Fig. 2.2) capable of wall measurements within 0.3 mm of the surface was used. 10,000 points were recorded at a sampling rate of 10 kHz. The hot-wire probe was mounted on a traverse and could be moved in x- and y-direction.

Measurements were taken up to 8 ribsizes in the x-direction and 2.28 ribsizes in the y-direction in the center plane of the testsection. From the plots of the velocity profiles (e.g. Fig. 2.9), it can be seen that it would have been desirable to increase the measurements in y-direction, but this was limited by the range of the traverse. The number of ribs was varied form $N=1$ to 4. The voltage output was recorded with a Iotech WaveBook 163 data acquisition box and transferred to a PC. As an example, the voltage as a function of time of a single data point is plotted in Fig. 2.3

To find the relationship between voltage output and velocity, the hot-wire has to be calibrated. Static pressure measurements at several tunnel speeds were taken at the test section entrance. Then the velocity was calculated from equation 2.1.

$$v = \sqrt{\frac{2 \Delta p}{\rho_{air}}} \quad (2.1)$$

Together with the corresponding voltage outputs of the hot-wire system, a calibration curve was plotted as shown in Fig. 2.4. For plotting the velocity profiles as shown in Figs. 2.8 and 2.9, the mean value of the velocity at each point was calculated. RMS (root mean square) velocity was calculated from equation 3.10 which is discussed in the next chapter.
2.2 Pressure Measurements

For the static pressure measurements, pressure taps with a diameter of 1.07 mm (17 gauge) were placed 25.4 mm apart downstream of the last turbulator in the center plane of the test section. In addition, three pressure taps were placed before the last turbulator. The spacing between the pressure taps was 25.4 mm, so that pressure data could be obtained up to $x/b = 14.5$ downstream of the last turbulator. The setup is shown in Fig. 2.5.

The pressure taps were connected to a differential pressure transducer with a range of ±0.5 inches of $H_2O$. As in the HWA measurements, the voltage output was measured with the WaveBook data acquisition box. 4000 points were recorded at a sampling rate of 500 Hz. A calibration curve shown in Fig. 2.6 was generated to convert voltage into pressure. The linear relationship between pressure and voltage output can be seen. Figure 2.7 shows the pressure fluctuations at a single point.

2.3 Results

Hot-wire results of mean velocity and RMS velocity for $Re_b = 8,000$ and $Re_b = 12,500$ are shown in Figs. 2.8 to 2.11 for 1, 2, and 4 rib arrangements, respectively. Due to continuity, the flow has to accelerate when passing a rib, resulting in a lift-off (separation) behind the rib. It can be seen that the large lift-off for the 1 rib case decreases as additional turbulators are added upstream. For the case of 4 ribs at $Re_b = 8,000$, the flow reattaches in the examined region. For 1 turbulator, reattachment cannot be seen. Increasing the velocity ($Re_b = 12,500$) results in a larger lift-off. The RMS plots in Fig. 2.10 and Fig. 2.11 show an extremely high level of turbulence intensity downstream of the turbulator in the 1 rib case which continues to increase beyond the reattachment location. Added ribs decrease the RMS intensity, particularly at locations near reattachment and post-reattachment where RMS is nearly constant along the profile. Increasing velocity results in an increased RMS intensity.

Plots of the pressure coefficient $C_p$ are shown in Figs. 2.12, 2.13 and 2.14 for 3 values of $Re_b$ and all 4 turbulator geometries. As a reminder, measurements were taken always behind the last rib at the same downstream location and additional turbulators were added upstream. This is the reason why the static pressure does not decrease with the number of ribs as one would expect when additional turbulators would have been added downstream of the first rib resulting in a measurement region further downstream. The pressure coefficient was calculated from

$$C_p = \frac{\Delta p - \Delta p_{\text{ref}}}{1/2 \rho_{\text{air}} U_\infty}$$  \hspace{1cm} (2.2)
The reference pressure $\Delta p_{\text{ref}}$ was measured at the entrance of the test section.

In general, it can be seen that the static pressure increases before the rib because of the impinging flow. Due to the flow acceleration on top of the rib and the total pressure loss through the channel, a low pressure region behind the ribs can be observed. The resulting force is called form drag. This observation is consistent through the whole Reynolds number range and agrees with the work of Çakan and Arts [8] as discussed in §1.2.

It can be seen from the plots, that $C_p$ is always positive in the 1 rib case in contrast to 2, 3 and 4 ribs where it is negative before the ribs. As will be seen later from the PIV investigation, a recirculation area before the rib (for 2, 3 and 4 ribs) is the reason for the negative value of $C_p$. Whereas for 1 rib, the flow impinging on the rib leads to a higher pressure than the reference pressure. One can see that the form drag is the largest for 1 rib and the smallest for 2 ribs. The form drag for 3 and 4 ribs is about the same. For 2 ribs and $Re_b=12,500$ and $Re_b=17,625$, a large recirculation area seems to be present in front of the last rib, since the pressure coefficient $C_p$ is lower before the rib than downstream. In this case, the pressure gradient before the last rib is the greatest. Behind the turbulator, a similar pressure gradient for 2, 3 and 4 ribs can be seen. Whereas for the 1 rib arrangement, the low pressure region behind the rib is very large so that compared to the multiple turbulator arrangements one can expect a larger recirculation area and longer reattachment length. Concluding the preliminary investigation, one can see that the 3 and 4 rib cases are virtually identical. In fact, Liou et al. [11] have shown that the flow becomes periodic after 3 or 4 turbulators. Because of the strong pressure gradient of the 2 rib arrangement, it would be interesting to examine the flow behavior between two ribs for this case. Also the flow behind one rib needs to be further investigated. For these reasons and to obtain a better understanding of the whole flow field, all cases investigated here as well as the flow between two ribs were examined further using PIV.
Figure 2.1: Flow visualization using the smoke-wire technique.
Figure 2.2: TSI 1218-T1.5 Standard Boundary Layer Probe.

Figure 2.3: Sample hot-wire data at a single data point.
Figure 2.4: Calibration curve for hot-wire measurements.

Figure 2.5: Placement of pressure taps.
Figure 2.6: Calibration curve for pressure transducer.
Figure 2.7: Pressure fluctuation at a single point.
Figure 2.8: Velocity profiles for $Re_b=8,000$. 
Figure 2.9: Velocity profiles for $Re_b=12,500$. 
Figure 2.10: RMS velocity for $Re_b=8,000$. 
Figure 2.11: RMS velocity for $Re_b=12,500$. 
Figure 2.12: Pressure coefficient for $Re_b=8,000$. 
Figure 2.13: Pressure coefficient for $Re_b=12,500$. 
Figure 2.14: Pressure coefficient for $Re_b=17,625$. 
Chapter 3

Experimental Setup

3.1 Windtunnel and Test Section

The wind tunnel arrangements for the backward facing step and turbulator experiments are shown in Figs. 3.1 and 3.2, respectively. Both test sections were installed in a low-turbulence open-circuit blow-down wind tunnel. A 7.5 hp motor powers a radial fan at the inlet. A vibration damper, flow straightener, and turbulence dampening screens precede the nozzle which has a contraction ratio of 6.7:1. The maximum test section velocity is 35 m/s with an open exhaust and approximately 10 m/s when a filter is installed to capture seeding particles as described below. The test sections were made of acrylic and have a channel height $H$ of 0.2 m (203 mm) and width $D$ of 0.4 m (406 mm). The hydraulic diameter was calculated as $D_H=271$ mm. The backward facing step has a step height $h$ of 28.6 mm. For the PIV measurements, the back wall and the floor were painted black to reduce reflection when the channel is illuminated by the laser sheet. The turbulator test section is arranged so that multiple square ribs can be placed in different locations in the channel. Only one wall was equipped with turbulators. In contrast to the preliminary investigation, the turbulators were mounted on the floor of the test section resulting in a completely smooth surface. The last rib was geometrically fixed at a location downstream of the test section entrance of $x/b = 27$. The maximum blockage ratio of the turbulators $b/H$ was 0.125.

3.2 Particle Image Velocimetry (PIV)

3.2.1 Principle of Particle Image Velocimetry

Particle Image Velocimetry benefits from the advances in the development of digital cameras and frame grabbers which have been made in the last 10 years. Prior to this, images were recorded
on photographic plates and evaluated by an optical Fourier transformation (Raffel [18]). This rather
time consuming process has been sped up by the introduction of digital recording and processing
techniques as discussed below. Technically, one can distinguish between PIV and DPIV (Digital
Particle Image Velocimetry). Today, DPIV is commonly referred to as PIV.

To obtain PIV measurements, the flow has to be homogeneously seeded with tracer particles.
It is assumed that the particles are small enough to move with the local flow velocity (see Appendix
B). A plane within the flow is illuminated twice within a short time interval by a laser sheet. The
light from each pulse scattered by the tracer particles is recorded by a CCD sensor on separate
frames. Analyzing one image pair, it is possible to identify the path a particle has traveled. Knowing
the time delay between two pulses, one can then calculate the velocity. The time interval between
two pulses has to be adjusted according to the mean flow velocity and the magnification of the
camera lens. If the time interval is too short, the speckle of the particles might overlap. On the
other hand, particles might have traveled out of the laser sheet or interrogation region if the delay
between the two pulses is too long. In both cases it is not possible to identify two identical regions
of particles and compute the displacement length.

For a single pair of PIV images, velocity fields are determined as follows. First, a pair of images
is divided into regions called interrogation areas. Figure 3.3 shows a typical PIV image. The image
color is inverted for clarity. The squared box indicates the size of the interrogation area. Here a
32 × 32 pixel interrogation area was used for an image size of 1008 × 1018 pixels. Each subregion is
transformed into Fourier space via a Fast Fourier Transformation (FFT). Then, the sub-windows
are shifted and their Fourier transforms cross-correlated until the correlation is found. In general,
the cross-correlation function is given by

\[ R(x, y) = \sum_{i=-K}^{K} \sum_{j=-L}^{L} I(i, j)I'(i + x, j + y) \]  

(Raffel [18]), where \( I \) and \( I' \) are the intensity values of the image pair. The cross-correlation function
produces a signal peak when the images align with each other, since the sum of the product of the
pixel intensities will be larger than elsewhere.

To reduce computation time, the convolution theorem (equation 3.2), which states that the
cross-correlation of two functions is equivalent to a complex conjugate multiplication of their Fourier
transforms, is used in actual applications,

\[ R \leftrightarrow \hat{I}\hat{I}' \]  

(3.2)

where \( \hat{I} \) and \( \hat{I}' \) are the Fourier transforms of the image intensities \( I \) and \( I' \) respectively.

Once the correlation is found, the Fourier transformations are converted back into real space.
The necessary displacement for correlating the interrogation areas is regarded as the particle displacement. But actually it is not the particle displacement which is computed, but the displacement of the interrogation area. To increase the accuracy, a predictor-corrector algorithm (Raffel [18]) was used. Once the matching interrogation areas are found, these areas are divided into smaller windows which are shifted inside of the interrogation area and cross-correlated, resulting in a more accurate velocity vector as shown in Fig. 3.4.

To increase the accuracy of measurements close to the surface, image parity exchange (IPX) developed by Tseui and Savaş [19] is used. The image is extended across the interface by mirroring and reversing it so that the velocity across the interface becomes,

\[ u_{\text{ext}}(\xi, \eta) - u_{\text{int}}(\xi) = -[u(\xi, -\eta) - u_{\text{int}}(\xi)] \] (3.3)

where \( u_{\text{ext}} \) and \( u_{\text{int}} \) is the extended velocity and interface velocity, respectively. \( \xi \) and \( \eta \) are abscissa and ordinate of the orthogonal coordinate system of the surface. In the present thesis, the interface is fixed \( (u_{\text{int}}=0) \). Figure 3.5 shows that with IPX, the interrogation areas can be extended across the boundary, allowing a calculation of the velocity vector at the wall. The red vector indicates the velocity vector closest to the wall which can be obtained without IPX if one defines the center of the interrogation area as the origin of the vector. Because the velocity field was reflected and reversed, velocities should cancel out at the interface.

In contrast to other flow measurement techniques like Laser Doppler Velocimetry (LDV) and hot-wire anemometry (HWA), which only allow single point measurements, the primary advantage of PIV is that one can obtain instantaneous data of the whole flow field. As in LDV, PIV is a nonintrusive flow measurement technique. With PIV, the spatial resolution, limited by the size of the CCD, is large, whereas the temporal resolution depending on capture rate of the camera is rather low compared to HWA and LDV. As an example, in Fig. 3.6, hot-wire and PIV data are plotted for a single point in a time interval of 1 second. Although the Reynolds numbers do not match up exactly, it can be seen that PIV is not suitable for measuring turbulence.

### 3.2.2 Setup

The schematic of the PIV system is shown in Fig. 3.7. For PIV, the laser sheet was generated by a 25 mJ double-pulsed Nd:YAG laser with a maximum repetition rate of 15 Hz per laser head. The CCD camera used is a 10-bit dual channel Kodak MegaPlus ES:1.0. It is a high-speed, high-resolution digital camera with a full-frame interline transfer CCD. The CCD is shown in Fig. 3.8.

The CCD consists of an array of 1008×1018 pixels, resulting in a spatial resolution of over one million pixels. Each pixel is divided into a light sensitive and a masked storage area, having a
total area of \( 9 \mu m \times 9 \mu m \). After the pixels are exposed to light, the charge is shifted first to the vertical storage area and then to a horizontal shift register. Because of this interline arrangement of the storage area, the transfer can take place in about \( 1 \mu s \). Due to the fast read out, the camera can capture 30 images per second. The disadvantage of the interline transfer is the reduced fill factor due to the additional storage sites right next to the light-sensitive area. The fill factor can be improved from 20% up to 60% using a microlens placed on the chip, which focuses the incident light on the photosensitive areas only. The readout of the sensor is in progressive-scan mode. It means that the lines are read out sequentially, conserving the integrity of the image. The camera is also equipped with an electronic shutter, allowing exposure times as short as \( 127 \mu s \). Figure 3.9 shows the timing diagram for the laser pulses together with the camera frame rate. In order to minimize the time delay between to laser pulses, the first pulse is placed at the end of a frame grabbing interval, whereas the second pulse takes place at the beginning of the following interval. With the present hardware configuration, a time delay as short as \( 1 \mu s \) would be possible. In the present study, pulse separations varied from \( 100 \mu s \) to \( 1 \) ms based upon the tunnel velocity.

Uniform seeding was accomplished using either zinc stearate or talc injected at the fan inlet; a 1 micron filter at the tunnel exhaust was required to capture the particles and thus reducing the maximum tunnel velocity during PIV. The mean particle diameter was \( 100 \mu m \). A predictor-corrector algorithm as described above with an interrogation area of \( 32 \times 32 \) was used to generate displacement vectors and velocity gradients. For each PIV run, at least 200 images were recorded for processing resulting in a minimum of 100 vector and vorticity fields from which to generate statistics.

### 3.3 Post Processing

#### 3.3.1 Velocity & Other Velocity Related Quantities

Velocity fields are calculated as part of the PIV algorithm and are scaled accordingly; the 2-D velocity gradients are determined spectrally and do not suffer from typical numerical differentiation problems (Scholl [20]). The output from the PIV algorithm is the tensor, \( C_{ij} \),

\[
C = \begin{pmatrix}
    u & v \\
    \partial u/\partial x & \partial u/\partial y \\
    \partial v/\partial x & \partial v/\partial y
\end{pmatrix}
\] (3.4)

Thus, continuity, vorticity, and rate of strain can be calculated using the appropriate gradients as follows:

\[
\text{continuity} = \partial u/\partial x + \partial v/\partial y
\] (3.5)
\[ \omega = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y} \] (3.6)
\[ \epsilon = \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \] (3.7)

In the case of equation 3.5, this quantity should be identically zero for a perfectly two-dimensional incompressible flow. Any instance of non-zero values can be attributed to either error in the PIV (which should be on the order of 1-5%) or three-dimensional effects. Thus, this can be used as a determination of whether the flow is in fact truly two-dimensional. Vorticity (equation 3.6) and strain rate (equation 3.7) are expected to have finite non-zero values in a non-uniform flow field, however.

Circulation can be computed from either velocity via
\[ \Gamma = \oint u \cdot dl \] (3.8)
or vorticity via
\[ \Gamma = \int \omega \cdot dA \] (3.9)

In general, use of equation 3.8 is more accurate around tight vortical structures than equation 3.9 since peak vorticity is often under-predicted in PIV.

### 3.3.2 RMS Velocity

RMS (root mean square) velocity is calculated by
\[ \text{rms} = \frac{1}{N} \sqrt{\sum (u')^2} \] (3.10)

where \( N \) in this case is the number of acquired velocity fields and \( u' \) is the unsteady velocity component as given by
\[ u = U + u' \] (3.11)

where \( U \) is the time-averaged velocity. This is a very “rough” estimate of the turbulence intensity of the flow. As a caveat, it should be noted that since the PIV measurements are taken at a nominal sampling rate of 10 Hz, any measure of turbulence will be underresolved as mentioned in section 3.2.1. It does provide accurate measurements of the variation in the velocity over a moderate time scale.

### 3.3.3 Reverse Flow Probability (RFP)

Reverse flow probability, or RFP, is determined by examining the percentage of time a vector is facing upstream (Spazzini [21]). For \( RFP = 1 \), the vector is facing upstream regardless of the
value of the vertical velocity component and for $RFP = 0$, the vector is facing downstream 100% of the time. Mathematically, this can be written for a single point as

$$RFP = \frac{1}{N} \sum_{1}^{N} \frac{u}{|u|}$$  \hspace{1cm} (3.12)

where $N$ is the total number of realized images. This is useful in determining both reattachment and the unsteadiness of the flow. A value of $RFP = 0.5$ would indicate maximum unsteadiness by this criterion. This can be used to determine where the flow is most often reattaching at the wall.

### 3.3.4 Skin Friction

Skin friction is determined by the value of the velocity gradient at the wall as given by

$$\tau_w = \mu \frac{du}{dy}|_{y=0}$$  \hspace{1cm} (3.13)

where $\mu$ is the dynamic viscosity of air. This is non-dimensionalized by $(1/2)\rho U_\infty^2$ to provide skin friction coefficient $C_f$;

$$C_f = \frac{\tau_w}{(1/2)\rho U_\infty^2}$$  \hspace{1cm} (3.14)

A value of $C_f = 0$ indicates that $\frac{du}{dy}|_{y=0} = 0$. This can be used to determine where the flow is most often reattaching at the wall. It can also be utilized to determine the wall skin-friction drag per unit width by using

$$C_D = \frac{1}{L} \int_{0}^{L} C_f dx$$  \hspace{1cm} (3.15)

where $L$ is the length of the wall along the direction of the measurement plane.
Figure 3.1: Range of PIV measurements for backward-facing step geometry.

Figure 3.2: Range of PIV measurements for channel with ribbed walls.
Figure 3.3: Typical PIV image. The box indicates the size of the interrogation area. (Field is inverted.)

Figure 3.4: Predictor-corrector algorithm.
Figure 3.5: Image Parity Exchange.

Figure 3.6: Comparison of temporal resolution of PIV and HWA.
Figure 3.7: Particle Image Velocimetry.
Figure 3.8: Progressive scan interline transfer CCD layout (after Raffel [18]).

Figure 3.9: Timing diagram for full-frame interline transfer CCD and double-pulsed laser.
Chapter 4

Results

Results are presented for PIV measurements for the backward facing step, cross-stream turbulator and inclined turbulator geometries in §4.1, §4.2 and §4.3, respectively. A total of 106 runs were made with an associated 11,824 images, which has a size of approximately 23.6 GB. For convenience, figures are organized by geometry and are not always referred to in sequence.

4.1 Backward Facing Step (BFS)

Backward facing step data was taken for Reynolds numbers of $Re_h=1,500$, $Re_h=3,800$, $Re_h=9,300$ and $Re_h=16,200$ based on the step height $h$. For the backward facing step and the turbulator experiments, the wind tunnel was run at the same fan speeds. Because of the larger pressure drop only the cases $Re_h=1,500$ and $Re_h=3,800$ for the BFS were in the Reynolds number range of the turbulator data and thus can be compared with some confidence. The conversion between $Re_h$ (based on step height) and $Re$ (based on hydraulic diameter) can be obtained from table 4.1.

<table>
<thead>
<tr>
<th>Backward Facing Step</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Re$</td>
</tr>
<tr>
<td>$Re_h$</td>
</tr>
</tbody>
</table>

Table 4.1: Reynolds numbers for backward facing step experiments.

A total of 16 runs were made and 3584 images were recorded. In the following sections, time averaged results are discussed. Plots of the velocity field, vorticity, reverse flow probability and RMS velocity were made. Furthermore, the reattachment length was obtained from the skin friction coefficient $C_f$ and the reverse flow probability $RFP$. For the plots being discussed here, an overlapping PIV region further downstream was examined. Considering the amount of overlapping,
both regions were arranged so that the results are shown in one plot.

Figures 4.1, 4.5, 4.9 and 4.13 are plots of the velocity fields. Leaving the top of the step, the flow lifts off and separates causing a region of reverse flow behind the step. For $Re_h = 1,500$ and $Re_h = 16,200$, the upward flow right behind the step displaces the free stream flow slightly upwards. From the plots of vorticity (Figs. 4.2, 4.6, 4.10 and 4.14), it can be seen that this is also true for $Re_h = 3,800$ and $Re_h = 9,300$, since the shear layer is displaced slightly upstream when the flow leaves the top surface of the step. The shear layer is displaced closer to the floor and vorticity increases, the higher the Reynolds number. Increasing the Reynolds number, the flow reattaches earlier for $Re_h = 3,800$, $Re_h = 9,300$ and $Re_h = 16,200$ compared to $Re_h = 1,700$. Values for vorticity (Figs. 4.4, 4.8, 4.12 and 4.16) are quite low. However, a large turbulent wake can be seen behind the step.

The reverse flow probability is plotted in Figs. 4.3, 4.7, 4.11 and 4.15. As already observed from the velocity fields, a large region of reverse flow is present behind the step.

The reattachment length can be computed from skin friction at the wall and reverse flow probability. As a reminder, for $RFP = 1$ the velocity vector is facing 100% of the time upstream in contrast to $RFP = 0$ where it is always facing downstream. A value of $RFP = 0.5$ indicates a maximum of unsteadiness and reattachment at the wall. The skin friction coefficient at the wall was calculated from equation 3.14. For $U_\infty$ the value of the time averaged velocity entering the PIV area was used. The derivative $\frac{\partial u}{\partial y}$ at the wall was computed by the PIV algorithm with the image parity exchange method. It was not possible to match the curve of the skin friction factor for the upstream and downstream PIV regions so that it is continuous at the intersection (Fig. 4.17). But still, the reattachment length can be obtained from the reverse flow probability (Fig. 4.18). In the plots, the x-axes is non-dimensionalized by the turbulator height for better comparison. The flow reattaches at $x/b \approx 6$, $x/b \approx 4.7$, $x/b \approx 5.2$ and $x/b \approx 5.2$ for $Re_h = 1,500$, $Re_h = 3,800$, $Re_h = 9,300$ and $Re_h = 16,200$ respectively.

### 4.2 Channel with Ribs Oriented Normal to the Flow Direction.

A total of 56 runs were made and 6272 images were recorded for the various rib geometries. As in the backward facing step, the velocity of the fan of the wind tunnel was kept constant as additional ribs were added. The Reynolds numbers were calculated based on the mean velocity of the flow entering the PIV area. In some cases, different Reynolds numbers were obtained. This way it can be seen that the placement of ribs into the test section results in a different pressure drop. In general, the pressure drop is higher as more turbulators are added. Therefore, one should be aware of that the Reynolds number is not identical but similar in the cases described below. The channel
was equipped with up to four ribs of equal size oriented normal to the flow direction. The spacing between the turbulators was $x/b = 5$. For each rib configuration, the flow was examined behind the last rib at four different Reynolds numbers based on the rib size ranging from $Re_b=600$ to $Re_b=5,000$. Table 4.2 shows at which Reynolds numbers (based on hydraulic diameter and ribsize) the flow was observed.

<table>
<thead>
<tr>
<th>Turbulator</th>
<th>$Re=6,400$</th>
<th>$Re=19,200$</th>
<th>$Re=48,000$</th>
<th>$Re=61,800$</th>
</tr>
</thead>
<tbody>
<tr>
<td>N=1</td>
<td>$Re_b=600$</td>
<td>$Re_b=1,800$</td>
<td>$Re_b=4,500$</td>
<td>$Re_b=5,800$</td>
</tr>
<tr>
<td>N=2</td>
<td>$Re=6,400$</td>
<td>$Re=18,100$</td>
<td>$Re=41,600$</td>
<td>$Re=53,300$</td>
</tr>
<tr>
<td></td>
<td>$Re_b=600$</td>
<td>$Re_b=1,700$</td>
<td>$Re_b=3,900$</td>
<td>$Re_b=5,000$</td>
</tr>
<tr>
<td>N=3</td>
<td>$Re=6,400$</td>
<td>$Re=18,100$</td>
<td>$Re=37,300$</td>
<td>$Re=53,300$</td>
</tr>
<tr>
<td></td>
<td>$Re_b=600$</td>
<td>$Re_b=1,700$</td>
<td>$Re_b=3,500$</td>
<td>$Re_b=5,000$</td>
</tr>
<tr>
<td>N=4</td>
<td>$Re=5,300$</td>
<td>$Re=18,100$</td>
<td>$Re=37,300$</td>
<td>$Re=52,300$</td>
</tr>
<tr>
<td></td>
<td>$Re_b=500$</td>
<td>$Re_b=1,700$</td>
<td>$Re_b=3,500$</td>
<td>$Re_b=4,900$</td>
</tr>
</tbody>
</table>

Table 4.2: Reynolds numbers for turbulator experiments.

It was found, that for the 1 rib case, the reattachment point was not in the recorded region. Therefore, the measurements were repeated further downstream. Furthermore, measurements behind the last turbulator in a two rib arrangement with alternating rib size at the same Reynolds number range were taken. A turbulator with a rib blockage ratio of $b/H = 0.0625$ was placed $x/b = 5$ upstream of a rib with $b/H = 0.125$.

For the same Reynolds number range and rib size, the flow between two ribs with a spacing of $x/b=2$ and $x/b=5$ was observed. The case with two ribs was chosen, because of the results from the preliminary hot-wire anemometry and static pressure analysis.

For each case 112 pairs of images were recorded. In the next section, time averaged results of these image pairs are presented. Vector plots of the flow field are shown. Vorticity and reverse flow probability (RFP) were calculated. Also plots for RMS velocity are shown, although PIV is not a suitable tool for turbulence measurements. From the analysis of the skin friction coefficient, the reattachment length was obtained and compared to the results of RFP.

Following the section of the steady results, the unsteadiness of the flow is examined. The circulation $\Gamma$ and the shedding frequency was analyzed.

### 4.2.1 Steady Analysis

Examining the vector plots of the velocity magnitude, it can be seen, that in general, for all rib arrangements a recirculation area is present behind the last turbulator (Figs. 4.19, 4.23, 4.27,
4.31, 4.35, 4.39, 4.43, 4.47, 4.51, 4.55, 4.59, 4.63, 4.67, 4.71, 4.75 and 4.79). For 1 rib, a very large recirculation region is present and a strong lift-off occurs behind the rib (e.g., Fig. 4.19). The flow does not reattach in the examined region except for $Re_b=1,800$.

In the case of 2, 3 and 4 ribs, the lift-off and the recirculation areas are smaller so that the flow reattaches in the examined PIV region. The upward flow right behind the rib which was also observed for the backward facing step displaces the free stream flow slightly upwards (e.g., Fig. 4.31). Due to the lift-off, a large turbulent wake is observed behind the ribs. This can be shown from the plots of RMS velocity (Figs. 4.22, 4.26, 4.30, 4.34, 4.38, 4.42, 4.46, 4.50, 4.54, 4.58, 4.62, 4.66, 4.54, 4.58, 4.62 and 4.66). (The values for the RMS velocity are quite low, which is because of the low temporal resolution of PIV. For analyzing RMS velocity, the HWA measurements in the preliminary investigation should be regarded.)

Increasing the number of ribs up to 4 at constant Reynolds number results in a small down-stream displacement of the center of the recirculation area and a decreasing lift-off. On the other hand, if the velocity is increased from $Re_b \approx 600$ and the number of ribs is kept constant, the center of the recirculation region moves closer to the backside of the turbulator and the bottom surface of the test section for $Re_b \approx 1,800$ and 3,500. For $Re_b \approx 5,000$ it moves back to the $x/b$ location for $Re_b \approx 600$.

As in the backward facing step experiments, an overlapping PIV region further downstream was examined. Again, both regions were arranged so that the results are shown in one plot. From the vorticity plots (Figs. 4.20, 4.24, 4.28, 4.32, 4.36, 4.40, 4.44, 4.48, 4.52, 4.56, 4.60, 4.64, 4.52, 4.56, 4.60 and 4.64), the redeveloping boundary layer after reattachment downstream of the rib can be seen very nicely for the 2, 3 and 4 rib configuration. In general, a region of high vorticity is observed starting at the top surface of the rib. The shear layer expands further downstream. For 1 and 2 turbulators it extends out of the PIV region. In the other cases it starts diffusing within the observed region. The exceptionality of the 2 rib arrangement can be seen, since the vorticity values are much smaller and the shear layer is much broader compared to the 1, 3 and 4 rib arrangement. A high lift-off of the shear layer is seen when the channel is equipped with just one turbulator, while it gets displaced closer to the bottom surface when additional ribs are added.(e.g., Fig. 4.20 and Fig. 4.68).

In all vorticity plots, a region of vorticity is seen in the reverse flow region at the bottom surface. It will be seen later from the plots of the skin friction coefficient that these regions correspond to low skin friction. For the 1 rib case, this region is very large and expands into the downstream PIV region. A slight upstream displacement of the shear layer, as already observed from the velocity fields, can be seen.
The reattachment lengths were obtained from the reverse flow probability (RFP) and the skin friction coefficient at the wall. In the plots of the skin friction coefficient and the reverse flow probability, the data for the rib arrangement with alternating turbulator size was added. This case, however, will be discussed later. From the skin friction plot in Fig. 4.83, it can be seen that the flow reattaches in the case of 2, 3 and 4 ribs at the same location ($x/b = 3.2$). This plot demonstrates also that the flow does not reattach in the first PIV region observed for the channel equipped with 1 turbulator. Therefore, in a separate figure, the downstream area was included in the plots (Figs. 4.84 and 4.89).

As already seen from the preliminary investigation and the vorticity plots, the 2 rib case shows a different behavior. Behind the turbulator the skin friction decreases until it reaches a local minimum at $x/b = 0.5$. From there it increases until $x/b = 0.8$ from where it decreases again up to a value 0.032 at $x/b = 1.7$. This minimum is about 3 times higher than the values for the other turbulator configurations. After reattachment the range of values for $C_f$ is similar to the 2, 3 and 4 rib cases. The 1 rib case shows an alternating structure as well. The skin friction plots for 3 and 4 turbulators look similar. The skin friction coefficient decreases until reaching a minimum at $x/b = 1.9$. Then it increases almost continuously. The similar behavior of these arrangements was also found in the preliminary investigation.

It is interesting to note, that except for the 1 rib case, skin friction reaches a minimum at almost the same downstream location $x/b \approx 1.9$. From the vorticity plots, it was already seen that high vorticity regions of the reverse flow correspond to these minima in skin friction coefficient. For the 1 rib case, this region of vorticity was larger than for the other cases, which explains the different behavior in the skin friction plots.

Comparing these results with Fig. 4.88 (a plot of the reverse flow probability at the wall) for the same Reynolds number, it can be seen that RFP is maximum at the same location where the lowest values of skin friction were observed. In agreement with the observation from Fig. 4.83, it can be seen that for the 2, 3 and 4 turbulator cases the flow reattaches at the same point. Again, a similar behavior is seen for the 3 and 4 rib arrangement. It can also be seen that in the 1 rib case, the region of reverse flow is much larger and the flow does not reattach in the examined region.

Increasing the Reynolds number to about $Re_b \approx 1,800$, the 2 rib case becomes similar to the 3 and 4 rib arrangements (Fig. 4.85). But still the skin friction coefficient is the lowest before and the highest after reattachment. The reattachment locations are now spread out in the sequence 2, 4, 3 ribs and 1 rib at $x/b = 2.9$, $x/b = 3.2$, $x/b = 3.4$ and $x/b = 3.9$ respectively. In this case flow over one turbulator reattaches in the examined area, as well. All configurations reach a minimum skin friction at $x/b \approx 1.8$. As before, the same observations can be obtained from the reverse flow
probability (Fig. 4.90) for this Reynolds number.

For the next higher Reynolds number \( (Re_b \approx 4,000) \) which was observed (Fig. 4.86), the 2 rib case differs from the others yet again. The magnitude of skin friction is again significantly higher as compared to the other cases. For 1 rib, reattachment cannot be observed in this region. Now the order of reattachment is 3, 4, 2 ribs at \( x/b \approx 3.1 \) for 3 ribs and \( x/b \approx 3.5 \) for the 4 and 2 rib case. The plot of the reverse flow probability (Fig. 4.91) for this Reynolds number agrees with these results. The 1 rib case does not reattach in the observed area.

Finally for the highest Reynolds number range \( (Re_b \approx 5,000) \) observed (Fig. 4.87), the flow for the 2, 3 and 4 rib arrangements reattaches at the same point as already seen in Fig. 4.83. But now the reattachment location is a little bit further downstream at \( x/b \approx 3.3 \). Similar to the other Reynolds numbers, the skin friction factor is low for the 2 rib case before reattachment, while the 3 and 4 rib case are very similar. For 1 turbulator, reattachment cannot be observed. In general, it can be said that the skin friction coefficient \( C_f \) decreases with increasing Reynolds number.

To obtain the reattachment length for the 1 rib case, the downstream region was included in the skin friction and reverse flow probability plots. As well as for the backward facing step, the plots are not continuous at the intersection of the two PIV areas. But from looking at the RFP plots (Figs. 4.21, 4.25, 4.29 and 4.33), it can be seen that the flow reattaches at \( x/b \approx 8.2 \), \( x/b \approx 3.2 \), \( x/b \approx 6.2 \) and \( x/b \approx 7.2 \) for \( Re_b=600, Re_b=1,800, Re_b=4,500 \) and \( Re_b=5,000 \) respectively.

In all turbulator cases discussed the 3 and 4 rib configuration showed similar behavior. In fact, Liou et al. [11] have shown that the flow shows a periodic behavior after three or four ribs.

### 4.2.2 Alternating Ribsizes

For the alternating rib configuration, the Reynolds number range was \( Re_b=800 \) to \( Re_b=5,200 \). A smaller turbulator was placed upstream of a rib of double size. Compared to the 2 rib case with turbulators of equal size, these Reynolds numbers are higher, due to a smaller pressure drop in the test section. Looking at the velocity profiles (Figs. 4.93 to 4.105), it can be seen that the flow lifts off at the edge of the rib resulting in a large recirculation area. For a Reynolds number of \( Re_b=800 \) and \( Re_b=5,200 \) the center of this recirculation area remains at the same position, while it moves closer to the turbulator for the Reynolds numbers \( Re_b=1,800 \) and \( Re_b=4,000 \). This behavior was already observed for the cases with turbulators of equal size. The flow seems to reattach in the observed PIV area for all cases.

The vorticity plots show similar results to the turbulator of equal size. The shear layer starts also at the top of the rib and extents into the next PIV region. The start of the redeveloping boundary layer can be seen in all cases except for \( Re_b=800 \). A turbulent region in the reverse
flow area is observed as well. It can be seen that the vorticity increases with increasing Reynolds number.

One significant difference can be observed. In contrast to the 2 rib case with equal sized ribs, the shear layer is not as broad. Examining the skin friction coefficient $C_f$ (Figs. 4.83, 4.85, 4.86 and 4.87) it can be seen that also the skin friction is much lower. It is the lowest for all tested rib arrangements, except for $Re_b=800$. The profile is similar to the one for 3 and 4 turbulators. For all Reynolds numbers, the location of reattachment is between $x/b = 3.5$ and $x/b = 3.7$.

### 4.2.3 Summary of Reattachment Analysis

Fig. 4.142 shows reattachment lengths for backward facing step and turbulators. Compared to the BFS, the flow reattaches earlier for the channel with the ribbed wall except for the case of 1 turbulator. A significant difference between the 1 rib and 2, 3, 4 rib arrangement in reattachment length can be seen. Except for $Re_b \approx 1,800$ the reattachment length for the 1 rib case is almost more than twice as long as for the other cases. In the case of 2 ribs with alternating size, the flow reattaches later than in the case of 2 turbulators of equal size. For all cases no Reynolds number dependency can be observed.

### 4.2.4 Flow between two Ribs Spaced by $x/b=2$

Also the flow between two ribs was examined. The Reynolds number range was $Re_b=600$ to 5,000. The plots of the velocity fields (Figs. 4.110, ref320vl, 4.118 and 4.122) for a rib spacing of $x/b = 2$ show a large vortex covering the whole area between the two ribs in x-direction. It expands up into the free stream in the y-direction. A closeup of this vortex is shown in Fig. 4.109. Behind the second rib, the recirculation area which was observed when examining the flow behind the rib can be seen.

The vorticity plots (Figs. 4.111, 4.115, 4.119 and 4.123) show that a region of high vorticity is present between the turbulators due to the vortex. This region seems to be the largest for a Reynolds number of $Re_b=1,700$. The highest vorticity is found to be on the top surface of the first rib. Compared to the second rib, the vorticity is almost twice as high at this location. This is not true for $Re_b=1,700$. Two areas of positive vorticity are seen at the bottom between the two ribs and on upstream of the second rib. As stated earlier, vorticity increases with increasing Reynolds number. The plots for the reverse flow probability (Figs. 4.112, 4.116, 4.120 and 4.124) also show the existence of the vortex, since a region of high reverse flow probability can be seen below the center of the vortex between the two ribs. The regions of high velocity fluctuations are found to be the same as for high vorticity (Figs. 4.113, ref320rmsl, 4.121 and 4.125).
4.2.5 Flow between two Ribs Spaced by $x/b=5$

Increasing the spacing between the turbulators to $x/b = 5$, the flow impinges on the second rib where it gets reversed (Figs. 4.126, 4.130, 4.134 and 4.138). The reversed flow lifts off approaching the first rib for being reversed again into the free stream direction. This lift off occurs further downstream of the first rib, the higher the Reynolds number. For $Re_b=600$ (Fig. 4.126) and $Re_b \approx 1,800$ (Fig. 4.130), there seem to be three vortices present, which might be due to the unsteady flow behavior. It might actually be one vortex wandering back and forth between the ribs, since in the vorticity plots three centers of vorticity cannot be seen. This will be examined further in the next section, where the unsteadiness of the flow is investigated rather than the time-averaged behavior. Increasing $Re_b$ to 3,900 and 5,000 shows only one vortex located in front of the second rib. In all vector plots, no recirculation region can be seen directly behind the first rib.

The lift-off of the reversed flow can be observed much better in the plots for the reverse flow probability (Figs. 4.128, 4.132, 4.136 and 4.140). After lifting off, the flow impinges on the back of the first rib where it splits up and part of it moves upwards and gets reversed by the main stream again, while a small part is forced downwards resulting in a region of high unsteadiness. The vorticity plots for $Re_b=600$ and $Re_b=1,700$ look almost identical, the higher Reynolds number case having just higher vorticity. A shear layer is present lifting off the first rib and getting enhanced by the second rib again. On the bottom surface, a large region of vorticity with opposite sign can be observed. In contrast to these cases, the vorticity area is only half the size of the rib spacing for $Re_b=3,900$ and $Re_b=5,000$. Also, the lift-off of the shear layer is stronger, resulting in a weaker vorticity enhancement of the second rib.

The very low pressure measured before the second rib can be reasonably explained from these measurements. From the analysis of reattachment length, it was seen that for 1 rib, the flow reattaches at $x/b\approx 6$. For the two rib configuration, the second turbulator was placed at this location. Therefore the flow does not reattach between the turbulators. Only a region of reverse flow is present causing a very low pressure zone before the second rib.

4.2.6 Unsteady Analysis

To show the unsteady behavior of the turbulator flow, circulation was calculated from equation 3.8 for the first PIV region behind the last rib and up to one rib height in the y-direction. Figs. 4.143, 4.144, 4.145 and 4.146 are plots of the circulation calculated for the case of 1, 2, 3 and 4 ribs for a Reynolds number range of $Re_b \approx 600$ to $Re_b \approx 5,000$. In the interval of 9 seconds, high fluctuations of circulation can be observed. As already found in the steady analysis, the 3 and 4 rib case show similar behavior. The magnitude of circulation is about the same. The circulation
is the highest for 2 turbulators. Positive circulation can be seen only in the plots for 1 and 2 ribs. One can observe that the magnitude of circulation increases with increasing Reynolds number.

The shedding frequency was examined by Fourier transformation of the circulation. PIV measurements were taken at 10 Hz. Referring to the Nyquist Theorem, which states that the highest frequency which can be accurately represented is one-half of the sampling rate, a Chebyshev filter was designed removing frequencies higher than 5 Hz. The filter is shown in Fig. 4.147. As can be seen from the frequency plots (Figs. 4.148, 4.149, 4.150 and 4.151), no regular pattern was observed. This is because of the low temporal resolution of PIV as already discussed for RMS velocity. For purposes of purely comparison assuming a Strouhal number of $St=0.2$ (circular cylinder), a characteristic length of 1 rib size and a flow velocity of $U=3$ m/s ($Re_b \approx 5,000$), one can calculate the shedding frequency $f \approx 20$ Hz from equation 4.1.

$$St = \frac{fl}{U} \quad (4.1)$$

This shows that the PIV measurements are probably underresolved since accurate measurements can only be obtained for frequencies lower than 5 Hz.

Also the location of the shear layer changes with time. A flapping can be observed behind the ribs. This is shown in a movie attached to this thesis for the case of two ribs with a spacing of $x/b=2$. 
Figure 4.1: Velocity BFS $Re_h=1,500$.

Figure 4.2: Vorticity BFS $Re_h=1,500$. 
Figure 4.3: Reverse Flow Probability BFS $Re_h=1,500$.

Figure 4.4: RMS Velocity BFS $Re_h=1,500$.
Figure 4.5: Velocity BFS $Re_h=3,800$.

Figure 4.6: Vorticity BFS $Re_h=3,800$.
Figure 4.7: Reverse Flow Probability BFS $Re_h = 3,800$.

Figure 4.8: RMS Velocity BFS $Re_h = 3,800$. 

Figure 4.9: Velocity BFS $Re_h = 9,300$. 

Figure 4.10: Vorticity BFS $Re_h = 9,300$. 
Figure 4.11: Reverse Flow Probability BFS $Re_h=9,300$.

Figure 4.12: RMS Velocity BFS $Re_h=9,300$. 
Figure 4.13: Velocity BFS $Re_h=16,200$.

Figure 4.14: Vorticity BFS $Re_h=16,200$. 
Figure 4.15: Reverse Flow Probability BFS $Re_h=16,200$.

Figure 4.16: RMS Velocity BFS $Re_h=16,200$. 
Figure 4.17: Skin Friction $C_f$ Backward Facing Step
Figure 4.18: Reverse Flow Probability Backward Facing Step
Figure 4.19: Velocity 1 rib $Re_b=600$.

Figure 4.20: Vorticity 1 rib $Re_b=600$. 
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Vorticity

$\text{s}^{-1}$
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<tr>
<td>2 ribs</td>
<td>$Re_b=5,000$</td>
</tr>
<tr>
<td>3 ribs</td>
<td>$Re_b=5,000$</td>
</tr>
<tr>
<td>4 ribs</td>
<td>$Re_b=4,900$</td>
</tr>
<tr>
<td>alternate</td>
<td>$Re_b=5,200$</td>
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$RFP$ at the wall $Re_b=600$. 

1 rib $Re_b=600$
2 ribs $Re_b=600$
3 ribs $Re_b=500$
4 ribs $Re_b=800$
all $Re_b=800$
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<table>
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<th>Conditions</th>
<th>$Re_b$</th>
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<td>2 ribs</td>
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<td>3 ribs</td>
<td>1900</td>
</tr>
<tr>
<td>4 ribs</td>
<td>1700</td>
</tr>
<tr>
<td>all</td>
<td>2500</td>
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Looking at the vector plots (Figs. 4.152, 4.156, 4.160 and 4.164), one can see that in contrast to a single turbulator oriented normal to the flow, a stronger and larger recirculation region is present causing the flow to reattach in the examined PIV region. The center of this recirculation bubble shifts closer to the rib for increasing Reynolds numbers. Except in the $Re_b=5,800$ case, it gets displaced further downstream.

Vorticity (Figs. 4.153, 4.157, 4.161 and 4.165) increases with increasing Reynolds numbers. Starting from the top of the rib, a large shear layer can be seen. The lift-off of the flow is seen clearly. Due to the large vortex, the shear layer is broader compared to the 90° rib and reattaches earlier, so that the start of the redeveloping boundary layer can be seen in all plots.

In contrast to all 90° cases, a small region of reverse flow is seen right at the top of the rib from the plots of the reverse flow probability (Figs. 4.154, 4.158, 4.162 and 4.166). A region of high unsteadiness is seen directly behind the rib and after reattachment on the bottom wall. But no reverse flow is seen behind the rib until to the location of the vortex. This might be due to the three dimensionality of the flow. Schabacker and Bölcs [12] have shown that 45° angled ribs induce a secondary flow. They observe in a channel with ribs on the floor and top wall two counter rotating vortices in the plane normal to the main flow direction. So it might be, that a single rib also induces a secondary motion so that there is out of plane flow behind the rib. To prove this however requires a further investigation taking PIV data of the cross stream movement.
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Vorticity

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Velocity

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Vorticity

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Chapter 5

Conclusions and Future Work

5.1 Conclusions

The first detailed PIV study for flow in a straight, rectangular channel with several turbulator geometries across multiple Reynolds numbers was presented; whereas, previous PIV and LDV studies focused on a single point in parameter space, or at most two values of Reynolds number. New data was obtained for turbulators placed cross-stream and inclined 45° to the flow. In contrast to other studies which focus on the flow between the turbulators, detailed measurements were taken behind the last turbulator to examine how the placement of ribs upstream influences the flow behind the last rib. Results were compared with flow over a backward facing step.

Figure 5.1 is a cartoon of the flow fields observed as described in chapter 4. In all cases, a recirculation area was observed behind the last rib. It was found that the flow shows similar behavior for 3 and 4 turbulators. As mentioned in §1.2, Liou et al. [11] also showed that the flow in a channel with multiple ribs shows a periodic behavior after 3 or 4 ribs. For the two rib arrangement, measurements were also taken between the ribs. A large recirculation region was found before the last rib.

From static pressure measurements, a high pressure zone before and a low pressure zone behind the ribs was observed resulting in a large form drag. From the plots, one can see that the form drag is the highest for 1 turbulator. A negative form drag for two ribs was observed since the pressure downstream of the rib was higher than before the rib. The negative form drag can be explained by Fig. 5.1. From the analysis of reattachment length, it was found that for 1 turbulator the flow reattaches around $x/b \approx 6$. Considering the case of two ribs, the second rib was placed just before this reattachment point. Therefore, for two ribs, the flow does not reattach inbetween resulting in a reverse flow region between the ribs. The form drag becomes positive again when another rib is added upstream, because then the flow reattaches between the second and third rib. One can
assume that a small recirculation region is present before the last rib. This needs to be shown from additional PIV measurements.

Analyzing the skin friction factor $C_f$, it was found that it is almost identical for 3 and 4 ribs. For the 1 rib arrangement, values of $C_f$ are in the range of 3 and 4 ribs, but it shows a large fluctuation. $C_f$ is significantly higher for the two rib arrangement. For this case, another experiment was run replacing the first turbulator with a smaller one. This leads to a large decrease in $C_f$, but an increase in reattachment length. Referring to Fig. 5.1 again, it can be seen that the flow lifts-off after the smaller rib. This lift-off is increased by the second rib causing the flow to reattach later than in the case of equally sized ribs.

Reattachment lengths were computed from plots of the skin friction factor and reverse flow probability. In agreement with Çakan and Arts [9], a dependency of reattachment length on Reynolds number was not observed. It was shown that compared to the backward facing step at similar Reynolds numbers, the flow reattaches earlier for 2, 3 and 4 ribs. This was also observed by Lion et.al. [10].

Differences in the flow pattern could be observed for the cases of 1 rib placed normal and in $45^\circ$ to the free stream. In the latter, a very strong vortex was present behind the rib. Also a small region of reverse flow was seen at the top of the turbulator. The flow seems to reattach earlier. For further comparison, more detailed measurements need to be taken for the $45^\circ$ case. But due to the difficulties in the experimental setup as described in §4.3, a sufficient amount of useful data could not be obtained.

To improve the present study, the Reynolds number should be kept constant for all cases for a more accurate comparison. Since the flow was found to be periodic after three ribs, it would be interesting to examine the flow when a smaller rib is placed upstream of 2 and 3 turbulators and when the pattern of alternating rib sizes is extended. Also, flow measurements between the ribs were made only for two turbulators. So this study could be extended in taking measurements between two ribs when the flow is periodic.

In the literature (e.g., Çakan and Arts [9]) it was found that regions of high heat transfer are present before the ribs and in the vicinity of the reattachment location. Regions of lower heat transfer are observed in the recirculation region behind the ribs. So it would be desirable to minimize the recirculation zone. Considering these observations, it is possible to find high and low heat transfer regions for the flow field examined in this study although no heat transfer measurements were taken. Looking at Fig. 5.1, it can be assumed that regions of high heat transfer are present close to reattachment and the areas before the ribs, whereas regions of low heat transfer are found in the recirculation region. For one rib, a fairly large recirculation area was observed,
whereas the flow reattaches earlier in the 2, 3 and 4 rib case. Comparing the case of variable rib sizes with the two rib case, a significant decrease in skin friction was observed when the smaller turbulator is placed upstream. But on the other hand, the flow reattaches further downstream, so that the case of equally sized ribs is more advantageous from a heat transfer point of view.

5.2 Future Work

In the future, it is planned to include rotation effects (centrifugal forces as well as the Coriolis force) in the measurements, since the flow in an actual gas turbine cooling channel is influenced by these forces. The non-dimensionalized Navier-Stokes equations become then

\[
\frac{\partial u}{\partial t} + \frac{U}{R_o} 2\Omega \times u = -\nabla p - \beta (T_w - T_\infty) \left( \frac{\Omega L}{U} \right)^2 \omega \times \mathbf{k} + \frac{\nu}{Re} \nabla^2 u
\]

where \( R_o, Bu, \) and \( Re \) are the Rossby, buoyancy, and Reynolds numbers, respectively.

Furthermore, the experimental model will be extended to a channel with a 180° bend. A rotating test facility will be designed as shown in Fig. 5.2. The channel is rotated by means of a turntable. The turntable will contain a hollow shaft, which serves as air supply. Before entering the actual test section, the flow will be passed through a honeycomb to gain a uniform laminar flow.

The flow field is observed through a dove prism. The characteristic of a dove prism is that when it is rotated 90°, the image rotates 180°. Rotating the dove prism at half the speed of the turntable, a stationary image will be obtained. Both motors, for the prism and the turntable, will be controlled by a PC to exactly synchronize the rotation. With this setup it will be possible to obtain measurements of a rotating system. The author worked on components of this setup that will be used in future experiments.
Figure 5.1: Cartoon of flow fields.
Figure 5.2: Rotating test facility.
Appendix A

Theory of Gas Turbine Operation

A.1 Gas Turbine Engines

One can distinguish between industrial gas turbines and aero-derivative gas turbines. In power plants, industrial gas turbines are mostly used as peaking units due to their fast startup, the low capital cost and their low efficiency. Because of the rather high temperatures of the exhaust gases, gas turbines are also used in the topping cycle of a combined-cycle power plant. In this case, the hot exhausts are passed through a heat exchanger to generate steam for the bottoming cycle. Besides the application in power plants, industrial gas turbines are also used for driving mechanical equipment. The most common application for gas turbines is as aircraft engines. These are the so-called aero-derivative gas turbines.

The most important parts are the compressor, the combustor and the turbine. A single shaft or multiple shafts connect compressor and turbine, so that the turbine supplies the compressor power. Air is drawn from the surroundings and compressed in the compressor. Entering the combustor, the compressed air is mixed with fuel. The air-fuel mixture is then burned at constant pressure (ideally). Exiting the combustor, the gas expands through the turbine. A disadvantage of gas turbine engines is the high backwork ratio which is defined as the ratio of compressor work to turbine work. Usually more than half of the turbine work is required to power the compressor. Starting the gas turbine, the compressor needs to be driven by an external motor, until the turbine power is sufficient to drive the compressor.

In contrast to industrial gas turbines, where the gas expands to atmospheric pressure to achieve maximum turbine work, in an aircraft engine the gas expands only to such a pressure, so that the turbine work is just sufficient to drive the compressor and auxiliary equipment. The high-pressure gas exiting the turbine is accelerated in a nozzle to produce thrust to drive the aircraft.
A.2 Gas Turbine Cycles

The operation of industrial gas turbine engines can be described by the Brayton cycle. In case of an aircraft engine by the Jet-Propulsion cycle. As mentioned above, the Jet-Propulsion cycle differs from the Brayton cycle only in that the exiting gases are not expanded to ambient pressure.

Gas Turbine cycles can be classified in direct open, indirect open, direct closed and indirect closed cycles. Aircraft engines operate in open cycles. A schematic, an ideal $p$-$V$-and an ideal $T$-$S$-diagram of such a cycle is shown in Fig. A.1 and Fig. A.2.

From point 1 to 2, the gas is compressed isentropically. Heat is added in the combustor at constant pressure from point 2 to 3. Then the gas expands in the turbine from point 3 to 4. Finally, the gas is cooled at constant pressure from 1-4 in the open atmosphere.

In the case of an indirect open cycle, the air receives heat from a primary fluid in a heat exchanger. It is useful for applications with environmental restrictions such as nuclear power plants. It prevents the air from receiving heat directly and releasing radioactivity into the atmosphere. A schematic of an indirect open cycle is shown in Fig. A.3.

In closed cycles, the heat of the exhaust gases is rejected by means of a heat exchanger. Then the gases are compressed back into the combustor. Gases other than air can be used as working fluid. Closed cycles also offer the opportunity of pressurizing the working fluid for reducing the size of turbine and compressor. Similar to open cycles, closed cycles can be divided into direct and indirect cycles.

In the following paragraph, an equation for computing the thermal efficiency of a gas turbine cycle is derived assuming constant specific heats. Referring to Figs. A.1 and A.2 and using the first law of thermodynamics, the turbine work $\dot{W}_T$ can be calculated from

$$\dot{W}_T = \dot{m} \int_{T_4}^{T_3} c_p(T) dT \quad (A.1)$$

regarding the process as adiabatic.

Assuming constant specific heats, equation A.1 changes to

$$\dot{W}_T = \dot{m} c_p (T_3 - T_4) \quad (A.2)$$

Together with the perfect gas relationship for isentropic processes

$$\frac{T_3}{T_4} = \left( \frac{p_3}{p_4} \right)^{(k-1)/k} \quad (A.3)$$

and defining a turbine pressure ratio as

$$r_{pt} = \frac{p_3}{p_4} \quad (A.4)$$
the equation for the turbine work can be written as

$$W_T = \dot{m}c_p T_3 \left( 1 - \frac{1}{r_p^{(k-1)/k}} \right)$$ (A.5)

The compressor work can be obtained from

$$-\dot{W}_c = \dot{m}(h_2 - h_1) = \dot{m}c_p T_2 \left( 1 - \frac{1}{r_{pc}^{(k-1)/k}} \right)$$ (A.6)

using

$$r_{pc} = \frac{p_2}{p_1}$$ (A.7)

and

$$\frac{T_2}{T_1} = \left( \frac{p_2}{p_1} \right)^{(k-1)/k}$$ (A.8)

Then the net work of the cycle $\dot{W}_{net}$ is given by

$$\dot{W}_{net} = W_T - |\dot{W}_c| = \dot{m}c_p(T_3 - T_2) \left( 1 - \frac{1}{r_p^{(k-1)/k}} \right)$$ (A.9)

under the assumption that there is no pressure loss in the combustor, meaning that $r_{pT} = r_{pc} = r_p$. For computing the thermal efficiency

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_A}$$ (A.10)

the heat added ($\dot{Q}_A$) to the cycle has to be calculated from

$$\dot{Q}_A = \dot{m}c_p(T_3 - T_2)$$ (A.11)

Then finally the thermal efficiency of the Brayton and Jet-Propulsion cycle is given by equation A.12

$$\eta_{th} = 1 - \frac{1}{r_p^{k-1/k}}$$ (A.12)

where $r_p$ is the pressure ratio equal to $\frac{p_3}{p_4}$ and $k$ the ratio of specific heats given by:

$$k = \frac{c_p}{c_v}$$ (A.13)

An optimum pressure ratio $r_{p_{opt}}$ can be found since the net cycle work $\dot{W}_{net}$ increases with increasing $r_p$, until it reaches a maximum. For finding $r_{p_{opt}}$, equation A.9 has to be differentiated with respect to $r_p$. Setting the derivative equal to zero, an expression for $T_2$ is found as

$$T_2 = \sqrt{T_1 T_3}$$ (A.14)
Together with \( \frac{T_2}{T_1} = \frac{T_3}{T_4} = r_p^{(k-1)/k} \) it follows that \( T_2 = T_4 \). Then, \( r_{p_{opt}} \) is found as

\[
r_{p_{opt}} = \left( \frac{T_3}{T_1} \right)^{k/(2(k-1))}
\]

(A.15)

It can be seen that the optimum pressure ratio increases with increasing turbine inlet temperatures \( T_3 \). Therefore, from equations A.12 and A.15, one can conclude that the thermal efficiency of gas turbine engines increases with increasing turbine inlet temperatures. Even a fractional improvement in performance can offer significant savings (see, for example, Mayle[1]). Although this equation was derived under the assumption of specific heats, the same trend applies for variable specific heats (El-Wakil[22]). Today, the turbine inlet temperatures are about 1700 K for civilian engines and up to 1800 K-1900 K (Müller[2]) in engines for military utilization.

Currently available materials for turbine blades are unable to withstand long periods of exposure to these high temperatures while maintaining structural integrity, even with thermal barrier coatings (TBC), implying need for active cooling strategies.
Figure A.1: Direct Open Brayton Cycle

Figure A.2: a) $p-V$-diagram b) $T-s$-diagram
Figure A.3: Indirect Open Brayton Cycle
Appendix B

Particle Motion in a Fluid

In PIV measurements, the numerical algorithms are tracking the movement of groups of particles, which are assumed to be traveling with the fluid motion. As can quickly be seen, if the particles are moving at a different velocity than the fluid, the measurement is in error. Somerscales has examined this problem using classical techniques [24]. Assuming the seed particles are spherical and obey Stokes Law, the equation of motion for a particle in dimensionless form is

$$\frac{du_p}{dt} + au_p + c \int_{t_0}^{t} \frac{du_p/d\tau}{(t - \tau)^{1/2}} d\tau = au_f + b \frac{du_f}{dt} + c \int_{t_0}^{t} \frac{du_f/d\tau}{(t - \tau)^{1/2}}$$  \hspace{1cm} (B.1)

where

$$a = \frac{18\nu}{[\rho_p/\rho_f + 1/2]d_p^2}$$  \hspace{1cm} (B.2)

$$b = \frac{3}{2[\rho_p/\rho_f + 1/2]}$$  \hspace{1cm} (B.3)

$$c = \frac{9}{[\rho_p/\rho_f + 1/2]}(\nu/\pi)^{1/2}$$  \hspace{1cm} (B.4)

This is known as the Bassett-Boussinesq-Oseen equation. The $1/2$ is the added mass. For small $\nu$ and large $\rho_p/\rho_f$, the history terms may be neglected by setting $c = 0$. This is known as a type I approximation. A type II approximation consists of $c = 0$ plus removal of the added mass term in $a$ and $b$. If the pressure gradient is neglected as well, $b = 0$, then this is a type III approximation and the equation reduces to

$$\frac{du_p}{dt} = (18\nu \rho_f / d_p^2 \rho_p)(u_f - u_p)$$  \hspace{1cm} (B.6)

where $u_f$ is a constant. If $\rho_p/\rho_f$ is small, then this is a poor approximation (e.g., for hydrogen bubbles in water, $\rho_p/\rho_f \approx 10^{-4}$ while for air bubbles in water, $\rho_p/\rho_f \approx 10^{-3}$). Integrating this with $u_p(0) = 0$,

$$u_p = u_f [1 - e^{-(18\nu \rho_f / d_p^2 \rho_p)t}]$$  \hspace{1cm} (B.7)
As $\rho_p/\rho_f$ decreases, then the difference between the particle velocity and fluid velocity increases.

For the Pliolite particles, with $\rho_f/\rho_p = 0.95$ and $d \approx 450 \, \mu m$, equation B.7 becomes

$$\frac{u_p}{u_f} \approx 1 - e^{-100 s^{-1}t}$$

(B.8)

For the silver-coated hollow glass spheres,

$$\frac{u_p}{u_f} \approx 1 - e^{-2 \cdot 10^4 s^{-1}t}$$

(B.9)

Obviously, the smaller particles follow the flow much better despite the greater density difference. In this study, talc particles with $d \approx 100 \, \mu m$ were used. Then equation B.7 becomes

$$\frac{u_p}{u_f} \approx 1 - e^{-67 s^{-1}t}$$

(B.10)
Bibliography


Vita

Harald Roclawski was born on January 5, 1977 in Trier, Germany. After graduating from the Max-Planck Gymnasium in Trier, he joined the University of Applied Sciences of Trier in 1997 where he studied Mechanical Engineering and worked as a student assistant. He graduated with the Vordiplom (Bachelor of Science) in 1999 and continued his studies for one year until he received a Fulbright Scholarship for studying Mechanical Engineering at the University of Kentucky in Lexington, KY.