THE ROLE OF ROTOR TIP CLEARANCE ON THE AERODYNAMIC INTERACTION OF A LAST GAS TURBINE STAGE AND AN EXHAUST DIFFUSER

Reinhard Willinger
Hermann Haselbacher
Institute of Thermal Turbomachinery and Power Plants
Technical University of Vienna
Austria

ABSTRACT

An investigation of the aerodynamic interaction between a last gas turbine stage and an exhaust diffuser is presented. Special attention is given to the influence of the rotor tip gap on this interaction. Flow measurements downstream of a linear cascade of turbine blades with tip gap have been performed in a low speed cascade wind tunnel. The geometry of the cascade corresponds to the tip section of a gas turbine of fairly recent design. Five tip gaps lying in the typical range were investigated. The two essential results are the leakage loss and the underturning in the end wall region. The flow field of the turbine cascade supplied the inlet boundary condition for the subsequent numerical investigation of the flow field in an annular diffuser. The geometry of the annular diffuser is based on dimensions of exhaust diffusers of some heavy duty and aeroderivative gas turbines. The result of the investigation is the diffuser pressure recovery factor as a function of the gap of the upstream cascade. The results from the cascade measurements and the diffuser computations have then been coupled by means of an interaction model. For gaps of practical interest, specific work and efficiency of the last gas turbine stage followed by an exhaust diffuser are independent of the rotor gap.

NOMENCLATURE

\[ R_s \] isentropic degree of reaction
\[ Re \] Reynolds number
\[ s \] blade spacing
\[ u \] circumferential speed
\[ w \] relative velocity
\[ x, y, z \] coordinates
\[ y^+ \] nondimensional wall distance
\[ z \] number of turbine stages
\[ \alpha \] absolute flow angle
\[ \beta \] relative flow angle
\[ \gamma \] stagger angle
\[ \epsilon \] turbulent rate of dissipation
\[ \eta \] efficiency
circumferential direction
\[ \theta \] cone angle
\[ \nu \] hub tip radius ratio
\[ \xi \] loss coefficient
density
\[ \rho \] density
\[ \tau \] radial gap width
\[ \psi_s \] isentropic loading coefficient

Subscripts
\[ D \] diffuser
\[ g \] casing
\[ i \] ideal
\[ MS \] midspan
\[ n \] hub
\[ P \] profile
\[ s \] isentropic
\[ S \] secondary
\[ Sp \] tip leakage
\[ u \] circumferential direction
\[ x \] axial direction
\[ 0 \] stator inlet plane
\[ 1 \] stator outlet plane, rotor inlet plane
\[ 2 \] rotor outlet plane, diffuser inlet plane
\[ 3 \] diffuser outlet plane
\[ 2D \] two-dimensional

Presented at the International Gas Turbine & Aeroengine Congress & Exhibition
INTRODUCTION

In axial turbomachines with unshrouded blades, the losses caused by the tip leakage flow are usually a major part of the entire losses. For example, this loss source is responsible for as much as one third of the total loss in an axial turbine (Denton, 1983). Usually, the tip clearance losses are diminished by reducing the tip gap. The minimum gap is dictated by the operational safety of the turbine. An exception seems to be the last turbine stage followed by an annular diffuser. The performance of the turbine stage is influenced by the interaction of the flow field at the rotor trailing edge plane and the pressure recovery of the diffuser. At increasing gap, not only the tip leakage losses but also the kinetic energy of the casing boundary layer at the diffuser inlet increase due to the tip leakage vortices. The enhanced diffuser pressure recovery lowers the back pressure of the upstream turbine stage. The decrease of the back pressure increases the specific work of the turbine. Under these circumstances, an optimum tip gap is expected to exist.

The aerodynamic interaction between an LP-steam turbine and an axial-radial diffuser was investigated experimentally by Zimmermann and Stetter (1993a, 1993b). The three stage turbine was a 1/4.2 downscaled model of a steam turbine. The three gaps covered the range of 4.38 to 12.47% of the chord. The lowest value corresponds to the design gap. The flow field in the diffuser with an area ratio of AR=1.39 was measured with pneumatic pressure probes. At the design gap, the diffuser pressure recovery was low, due to the flow separation at the outer casing wall. At increasing gap, the diffuser pressure recovery increased. However, the improvement of the diffuser performance could not offset the deteriorated turbine efficiency induced by the tip leakage flow.

The influence of blade wakes and swirl upon the flow through a diffusing, S-shaped duct has been investigated experimentally and numerically by Dominy et al. (1996). The meridional streamline curvature produces a cross-passage pressure gradient. Secondary flows resulting from the cross-passage pressure gradient and the blade wakes were clearly evident. However, the influence of blade wakes and swirl upon the overall diffuser performance was surprisingly small.

Due to the high costs of experimental facilities and their operation, the application of CFD to the turbine stage/diffuser-combination becomes more and more attractive. However, the application of full Navier-Stokes solvers to the entire configuration is prevented by the limitation of current computer resources. So the flow fields in the two components are computed separately and their interaction is considered by an interaction model. Benim et al. (1995) computed the flow field in a LP-steam turbine followed by an exhaust diffuser. Special attention was given to the non-symmetry of the flow field at the rotor trailing edge plane. This non-symmetry is induced by the non-symmetry of the exhaust hood geometry. The rotor tip gap was taken into account, but its value was not varied. The turbulence behavior of the flow field was considered by the standard $k/\varepsilon$-model.

The present paper supplies an engineering design tool for the aerodynamic interaction of a last turbine stage and an exhaust diffuser. Measurements made downstream of a linear turbine cascade with tip gap supply the tip leakage loss coefficient and the flowfield at the trailing edge plane. The flow in an annular diffuser was computed with a Navier-Stokes solver. The inlet boundary conditions were adopted from the cascade measurements. The results from the cascade measurements and the diffuser computations are coupled by means of an interaction model. The investigation supplies the performance of the turbine stage/diffuser-combination as a function of the rotor tip gap.

DIFFUSER GEOMETRY

Commonly, the exhaust diffusers of axial gas turbines are of the annular type. The hub is either cylindrical or conical with a small cone angle. The outer casing wall is conical, with an appropriate angle to achieve the desired area ratio. To support the bearing on the hot end, struts are usually arranged in the diffuser flow path. The blockage, caused by these struts, is often compensated by a special shaping of the outer wall of the diffuser. Further, splitters are often located at the diffuser exit, to guide the flow from the horizontal direction to the vertical direction of the chimney. For simplicity, all these deviations from the simple geometry are neglected and the exhaust diffuser is approximated by an annular diffuser with straight walls at hub and casing, respectively. The geometry of an annular diffuser with straight walls can be described by four parameters (inlet hub tip radius ratio $\nu_2$, hub cone angle $\theta_h$, casing cone angle $\theta_g$, nondimensional diffuser length $L/h_2$). Tab. 1 presents a summary of diffuser geometries of a number of modern gas turbines. All of them are heavy duty gas turbines, with the exception of the GE LM2500 and the MAN-GHH FT8 which are of the aeroderivative type. For comparison purposes, the area ratio $AR$ is included in Tab. 1. Values for the cone angles are only included for diffusers with straight walls. The parameter $z$ is the number of turbine stages (or the number of stages of the power turbine in the case of the multispool aeroderivative gas turbines). The symbol in the last column indicates the type of shrouding of the last rotor blade row, with "s" for shrouded and "u" for unshrouded blades, respectively.

<table>
<thead>
<tr>
<th>Turbine</th>
<th>$\nu_2$</th>
<th>$\theta_h$</th>
<th>$\theta_g$</th>
<th>$L/h_2$</th>
<th>$AR$</th>
<th>$z$</th>
<th>Shrouding</th>
</tr>
</thead>
<tbody>
<tr>
<td>ABB GT8</td>
<td>0.53</td>
<td>-</td>
<td>-</td>
<td>4.78</td>
<td>2.06</td>
<td>3</td>
<td>s</td>
</tr>
<tr>
<td>ABB GT11</td>
<td>0.53</td>
<td>6.5</td>
<td>14</td>
<td>6.00</td>
<td>2.99</td>
<td>5</td>
<td>u</td>
</tr>
<tr>
<td>ABB GT13</td>
<td>0.54</td>
<td>7</td>
<td>14</td>
<td>5.85</td>
<td>2.75</td>
<td>5</td>
<td>u</td>
</tr>
<tr>
<td>ABB GT13E</td>
<td>0.50</td>
<td>-</td>
<td>-</td>
<td>7.13</td>
<td>2.68</td>
<td>5</td>
<td>u</td>
</tr>
<tr>
<td>GE LM2500</td>
<td>0.56</td>
<td>-</td>
<td>-</td>
<td>4.50</td>
<td>1.71</td>
<td>5</td>
<td>s</td>
</tr>
<tr>
<td>GE MS9001B</td>
<td>0.63</td>
<td>8</td>
<td>8</td>
<td>4.71</td>
<td>1.88</td>
<td>3</td>
<td>s</td>
</tr>
<tr>
<td>GE MS9001E</td>
<td>0.65</td>
<td>8</td>
<td>8</td>
<td>5.57</td>
<td>2.00</td>
<td>3</td>
<td>s</td>
</tr>
<tr>
<td>MAN-GHH FT8</td>
<td>0.68</td>
<td>8</td>
<td>6</td>
<td>6.52</td>
<td>1.97</td>
<td>4</td>
<td>s</td>
</tr>
<tr>
<td>SIEMENS V84.3A</td>
<td>0.59</td>
<td>0</td>
<td>12.5</td>
<td>2.82</td>
<td>1.92</td>
<td>4</td>
<td>u</td>
</tr>
</tbody>
</table>

Table 1: Diffuser geometries of heavy duty and aeroderivative gas turbines

The performance of a diffuser is mainly influenced by its geometry. Following Sovran and Klomp (1967), it can be assumed that of the four geometry parameters mentioned above, the cone angles of hub and casing play only a minor role. Area ratio and nondimensional diffuser length retain as primary parameters. Apart from the geometry, the diffuser performance is strongly influenced by the boundary layer blockage $B_2$ at the inlet. $B_2$ is the blocked area, which results from the displacement thicknesses of the inlet boundary layers, divided by the inlet area. Sovran and Klomp (1967) also introduced the so-called diffuser performance chart. In this diagram, the area ratio of a diffuser is plotted versus its nondimensional length, with the diffuser recovery factor as a parameter. Fig. 1 shows this performance chart for annular diffusers. Also included are the geometries of the diffusers of the gas turbines from Tab. 1. As can be seen from Fig. 1, most of the diffuser geometries are grouped near the $C_{pD}^*$-line. The $C_{pD}^*$-line is the location of diffusers with maximum pressure recovery factor, at prescribed nondimensional length. It can be concluded that the area ratios of exhaust diffusers are designed for prescribed diffuser length, meeting the demand of maximum pressure recovery factor. Usually, it is aimed to install a short diffuser, which results in minimum costs. Based on these considerations, the following diffuser geometry is selected for the further investigations:
• Hub tip radius ratio at diffuser inlet $v_2 = 0.55$
• Cone angle at the hub $\delta_1 = 0^\circ$
• Cone angle at the casing $\theta_2 = 8^\circ$
• Nondimensional diffuser length $L/h_2 = 5$
• Area ratio $AR = 2.05$

The geometry of this diffuser lies at the CD-line (Fig. 1). The pressure recovery factor is $C_{PD} = 0.6$ for an inlet blockage of $B_2 = 0.02$ and an inlet Reynolds number of $Re_{D2} = 6.10^5$. The diffuser Reynolds number is defined with the freestream velocity at diffuser inlet and the difference from casing and hub radius at the inlet. The value of $C_{PD} = 0.6$ should be compared with the pressure recovery factor of the ideal diffuser, which only depends on the area ratio and is equal to

$$C_{PDi} = 1 - \frac{1}{AR^2} = 0.76. \quad (1)$$

Figure 1: Diffuser performance chart of Sovran and Klomp (1967). The symbols (○) indicate diffuser geometries of the gas turbines from Tab. 1

**MODELING**

An aero-thermodynamic model for the interaction of a gas turbine and an exhaust diffuser was presented by Farokhi (1987). He concluded that about 1% gain in thermal efficiency results from the elimination of the last rotor tip clearance flow. The corresponding increase in thermal efficiency of a modern gas turbine plant due to enhanced diffuser pressure recovery is less than 1%. A similar investigation was conducted by von Rappard (1977). His model focuses primarily on the role of the rotor exit swirl on the diffuser performance. It should be noted that the strength of the exit swirl depends on the load of the turbine. The interaction model presented in this paper is distinguished by a uniform set of parameters for the performance description of turbine stage and diffuser.

**Rotor Tip Leakage Losses**

The starting point for the modeling of the rotor tip leakage losses is a turbine stage consisting of a stator and a rotor blade row. It is assumed that tip leakage losses are only present at the unshrouded rotor blades. Due to an efficient sealing, the stator blade row experiences no tip leakage losses. It is further assumed that the total loss in a cascade can be divided into profile, secondary and tip leakage losses. The expansion in the turbine stage is illustrated in the enthalpy/entropy-diagram in Fig. 2. The specific work of the turbine stage is given by the change of the specific stagnation enthalpy

$$a_u = \left( h_0 + \frac{c_0^2}{2} \right) - \left( h_2 + \frac{c_2^2}{2} \right) = \left( h_1 + \frac{c_1^2}{2} \right) - \left( h_2 + \frac{c_2^2}{2} \right) \quad (2)$$

and by the Eulerian equation of turbomachinery

$$a_u = u(c_{u1} - c_{u2}). \quad (3)$$

In equation (3), constant blade speed $u$ is assumed. The tangential components of the absolute velocities can be taken from the velocity triangles of the turbine stage (Fig. 3). In the rotor blade row, profile and secondary losses are described by the aerodynamic efficiency $\eta''$ and by their loss coefficients (profile loss coefficient $C_p''$, secondary loss coefficient $C_{sp''}$), respectively

$$\eta'' = \frac{w_{2s}^2}{w_{2s}^2} = 1 - \frac{\Delta h''_{p} + \Delta h''_{s}}{\frac{w_{2s}^2}{2}} = 1 - (\xi''_{p} + \xi''_{s}). \quad (4)$$

The rotor tip leakage losses are quantified by the tip leakage loss coefficient

$$\xi''_{sp} = \frac{a''_{sp}}{w_{2s}^2}. \quad (5)$$

The enthalpy differences $\Delta h''_{p}$, $\Delta h''_{s}$ and $a''_{sp}$ are illustrated in Fig. 2 and $w_{2s}$ is the isentropic rotor exit velocity in the rotating frame of reference. Using Eq. (2) to (5), the tip leakage loss divided by the specific work of the turbine stage follows as

$$a''_{sp} = \frac{-\frac{\Delta h''_{sp}}{2}}{2 \left[ (\frac{w_{2s}}{u}) \cos \alpha_1 - \sqrt{C \sqrt{1 - (\xi''_{p} + \xi''_{s}) \cos \beta_2 - 1}} \right]. \quad (6)$$

In equation (6), the stage parameters isentropic loading coefficient

$$\psi_s = \frac{2 \Delta h_s}{u^2}. \quad (7)$$

Figure 2: Enthalpy/entropy-diagram of the turbine stage
and isentropic degree of reaction

\[ R_s = \frac{\Delta h''}{\Delta h'' + \Delta h'''} = \frac{\Delta h''}{\Delta h'} \]  (8)

as well as the abbreviation

\[ C = 1 + \left( \frac{\alpha_1}{u} \right)^2 - 2 \left( \frac{\alpha_1}{u} \right) \cos \alpha_1 + R_s \psi_s \]  (9)

are used.

Figure 3: Velocity triangles of the turbine stage

Influence of the Diffuser

The expansion in the turbine stage (02) and the subsequent compression in the exhaust diffuser (23) are illustrated in the enthalpy / entropy-diagram of the turbine stage/diffuser-combination in Fig. 4. The performance of the diffuser is quantified by the so-called pressure recovery factor, which is defined as the static pressure rise divided by the dynamic pressure at diffuser inlet

\[ C_{PD} = \frac{p_3 - p_2}{\frac{1}{2} \rho_2 C_{2s}^2} \]  (10)

It is assumed that a change in the diffuser flow or diffuser geometry increases the pressure recovery factor from \( C_{PD} \) to \( C_{PD}' \), by an increment \( \delta C_{PD} \). If the ambient pressure \( p_3 \) and the dynamic pressure at the diffuser inlet are assumed as constant, the enhanced pressure recovery factor is

\[ C_{PD}' = \frac{p_3 - p_2}{\frac{1}{2} \rho_2 C_{2s}^2} = C_{PD} + \delta C_{PD} = \frac{p_3 - p_2}{\frac{1}{2} \rho_2 C_{2s}^2} + \frac{p_2 - p_2^*}{\frac{1}{2} \rho_2 C_{2s}^2}. \]  (11)

Now, the expansion in the turbine stage is from 0 to 2* and the subsequent compression in the enhanced diffuser from 2* to 3* (Fig. 4). The increase of the diffuser pressure recovery factor causes an increase of the difference between the ambient pressure and the static pressure at the diffuser inlet. For fixed ambient pressure \( p_3 \), the static pressure at the diffuser inlet decreases from \( p_2 \) to \( p_2^* \). Since the diffuser inlet plane is identical to the rotor trailing edge plane, this results in an increase of the specific work output of the turbine stage of

\[ a_D = h_2 - h_2^* = \eta''(h_2 - h_2^{**}) = \left[ 1 - (\xi'' + \xi''') \right] (h_2 - h_2^{**}). \]  (12)

Since the expansion from 2 to 2* is isentropic and the pressure difference \( p_2 - p_2^* \) is small, the enthalpy difference can be approximated by

\[ h_2 - h_2^{**} \approx \frac{1}{\rho_2} (p_2 - p_2^*). \]  (13)

When the stage parameters formerly defined are used, this results in the increase of the specific work output of the turbine stage \( a_D \) caused by the enhanced diffuser performance divided by the specific work of the turbine stage

\[ \frac{a_D}{a_u} = \frac{\delta C_{PD} \left[ 1 - (\xi'' + \xi''') \right]^2 \sin^2 \beta_2 C}{2 \left( \frac{\alpha_1}{u} \right) \cos \alpha_1 - \sqrt{1 - (\xi'' + \xi''') \cos \beta_2 - 1}}. \]  (14)

The subsequent sections of the paper describe the determination of both parameters for a practical situation.
CASCADE MEASUREMENTS

Test Cascade and Test Section

The geometry of the cascade under investigation is summarized in Fig. 5. The blade profile corresponds to the tip section of a moving blade row of a gas turbine of fairly recent design. The blades have constant chord and no twist. Following Bindon (1987), the tip gap of the moving blade row of a gas turbine varies from 1.5% to 2.5% of the chord. In the present investigation, this range is covered by four tip gaps from 0.8% to 3.6% of the chord (Tab.2). The cascade wind tunnel of the Institute of Thermal Turbomachinery and Power Plants operates in the pressure mode. Air is supplied by an axial blower with variable inlet guide vanes. The test cascade, consisting of six blades, can be turned to adjust the inlet flow angle. In the present case, the inlet flow angle was fixed at \( \beta_1 = 90^\circ \). Since the inlet flow field should show boundary layer parameters typical for a turbine blade row, there was no boundary layer bleeding.

<table>
<thead>
<tr>
<th>Chord Length ( c ) = 182.2 mm</th>
<th>Blade Span ( h ) = 150 mm</th>
<th>Blade Spacing ( s ) = 108 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aspect Ratio ( h/c ) = 0.823</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Solidity ( ch = 1.687 )</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Stagger Angle ( y = 51^\circ )</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

![Figure 5: Summary of cascade geometry](image)

Table 2: Summary of tip gaps

<table>
<thead>
<tr>
<th>( \gamma/y/s = )</th>
<th>( \gamma/y/s = )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.85</td>
<td>1.03</td>
</tr>
<tr>
<td>1.40</td>
<td>1.70</td>
</tr>
<tr>
<td>2.50</td>
<td>3.03</td>
</tr>
<tr>
<td>3.50</td>
<td>4.37</td>
</tr>
</tbody>
</table>

Instrumentation

The inlet flow field was investigated in a plane 1.385 axial chord lengths upstream of the cascade. A three hole pressure probe was traversed in \( x \)-direction at six different positions in \( y \)-direction. The pressure probe was of the cobra type with head dimensions of 0.8 by 2.4 mm. During the downstream measurements a pitot tube (diameter 3 mm) was positioned at midspan to obtain the total pressure of the undisturbed inlet flow. The static pressure was measured by a row of pressure taps (diameter 2 mm) positioned 1.065 axial chord lengths upstream of the cascade. Additionally, a hot wire probe (DANTEC P11) was used to get information on the turbulence intensity of the inlet flow field. The temperature of the inlet flow field was measured via a Pt-100 resistance thermometer. A five hole pressure probe was traversed 0.071 axial chord lengths downstream of the trailing edge plane. This probe was of the conical type with a diameter of 3 mm and a total cone angle of 60°. An exception was the traversing closest to the end wall. It was assumed that in this region the spanwise component of the flow can be neglected. So the three hole probe was used for this traversing. Both probes were nulled in yaw. The pressure probes had been calibrated at the operating Reynolds number to minimize potential calibration errors. Typical estimated measurement uncertainties are \( \pm 2\% \) of the inlet dynamic pressure for the stagnation pressure and \( \pm 1\% \) for pitch and yaw angles, respectively. The static pressure at the end wall was measured by a row of pressure taps (diameter 2 mm). The 23 pitchwise and 11 spanwise traverses produced a grid of a total of 253 measuring points, with smaller spacings in the regions of supposedly high gradients of the flow quantities. These are the near wall region and the regions immediately behind the trailing edges. All pressure differences were measured with piezoresistive pressure transducers (HONEYWELL). The analogue pressure signals were converted to digital form via a HEWLETT-PACKARD HP 3852A data acquisition system. The system was controlled using LabWindows (NATIONAL INSTRUMENTS) on an IBM-compatible PC.

Operating Conditions and Inlet Flow Field

![Figure 6: Cascade inlet velocity profiles at various pitch positions, \( z/c_x = -1.385 \)](image)

The magnitude of the undisturbed velocity at the inlet plane was about \( w_{1,M_S} = 32 \) m/s. Since this corresponds to an upstream Mach number of approximately \( M_{1,M_S} = 0.1 \), the flow field is essentially incompressible. The blade Reynolds number, defined with the undisturbed inlet velocity and the chord length, was \( Re_{1,M_S} = 3.63 \times 10^5 \). The velocity distribution of the inlet flow field at various pitch positions in the measuring plane 1.385 axial chord lengths upstream of the cascade is shown in Fig. 6. The gap side end wall is located at \( z/h = 0 \), the blades are mounted at \( z/h = 1.0 \). The velocities are made dimensionless by the undisturbed upstream velocity \( w_{1,M_S} \). To improve the clarity of the illustration the curves obtained from the different traverses are staggered in the direction of the abscissa. The velocity distribution is symmetrical with respect to a plane \( z/h = 0.5 \), with a broad region of constant velocity magnitude. The end wall boundary layers are clearly...
visible. The parameters of the gap side end wall boundary layer at y/s=0.694 are summarized in Tab. 3. The boundary layer shape factor of $H_{12}=1.34$ is slightly lower than the typical value of $H_{12}=1.4$ for turbulent boundary layers without pressure gradient. The turbulence intensity was about 5% in the region of constant velocity. The relatively high turbulence level in the freestream region is a consequence of a turbulence grid which is located in front of the nozzle of the cascade wind tunnel.

<table>
<thead>
<tr>
<th>Boundary Layer Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Boundary layer thickness</td>
<td>δ_99</td>
</tr>
<tr>
<td>Boundary layer displacement thickness</td>
<td>δ_1</td>
</tr>
<tr>
<td>Boundary layer momentum thickness</td>
<td>δ_3</td>
</tr>
<tr>
<td>Boundary layer shape factor</td>
<td>$H_{12}$</td>
</tr>
</tbody>
</table>

Table 3: Summary of the inlet boundary layer parameters, $x/c=-1.385$, $y/s=0.694$

Measurement Results

In this section, some important cascade exit pitchwise mass averaged flow quantities are presented. Further details of the measurement results are summarized in Willinger and Haselbacher (1997). In Fig. 7, the pitchwise mass averaged flow angle $\beta^+_2$ is plotted as a function of the spanwise coordinate. In the region $z/h > 0.3$, the mass averaged flow angle corresponds closely to the two-dimensional value of $\beta^+_2 = \beta^+ = 38.9^\circ$. This value was determined by a method of Traupel (1977). Overturning near the end wall, caused by the passage vortex, is visible for the zero tip gap case. With increasing gap, the overturning in the near wall region, caused by the tip leakage vortex, increases. Moving away from the end wall, the downstream flow undergoes slight overturning and finally reaches the direction of the two-dimensional flow.

Figure 7: Pitchwise mass averaged flow angle $\beta^+_2$

Fig. 8 shows the ratio of the pitchwise averaged axial velocity $w_{2x}$ to the inlet velocity at midspan $w_{1,MS}$ as a function of the spanwise coordinate. The end wall boundary layer in the case of zero gap is comparable to the inlet boundary layer. As the tip gap is opened, a region of high axial velocity is established near the end wall. The maximum velocity increases with increasing gap. This increase is caused by a decreased flow turning in the gap region. As the gap is increased, the near wall flow tends to turn to the axial direction (Fig. 7).

Figure 8: Pitchwise averaged axial velocity $w_{2x}/w_{1,MS}$

Fig. 9 shows the pitchwise averaged total pressure coefficient $C_{pt2}$. The region $z/h > 0.3$ is dominated by a constant pressure loss. This pressure loss, which is equal to the profile loss, is quite independent of the gap. At zero gap, the rise in loss near the end wall is caused by the passage vortex and the end wall friction. The losses in the tip leakage vortex region increase with increasing gap.

Figure 9: Pitchwise mass averaged total pressure coefficient $C_{pt2}$

DIFFUSER COMPUTATION

Governing Equations

The numerical investigation of the diffuser flow field was founded on the governing equations for steady axisymmetric incompressible turbulent flow. To take into account the turbulent behavior of the flow field, the $k$-$\varepsilon$-turbulence model with the standard empirical constants was used (Lauder and Spalding, 1974). Due to the fact that this turbulence model is of the high Reynolds number type, it is only valid in flow regions which are dominated by turbulent viscosity rather than molecular viscosity. The wall function method with special wall elements was used to model the flow in the vicinity of the hub and the casing walls (Haroutunian and Engelman, 1991).

Numerical Method and Finite-Element Mesh

The set of governing equations, consisting of continuity, momentum, $k$- and $\varepsilon$-transport equations, was solved numerically by the finite-element method (FDI, 1993). Fig. 10 shows the finite-element mesh
with 3721 nodes and 1020 elements, respectively. The computation domain was discretized by 9-node quadrilateral elements with biquadratic interpolation of the various quantities. An exception is the pressure degree of freedom the interpolation of which is of the bilinear type. In the direction perpendicular to the diffuser axis, the mesh was stretched geometrically to resolve the high gradients near the solid walls. The thickness of the elements adjacent to the solid walls was adjusted to satisfy $10 \leq y^+ \leq 100$, with $y^+$ as the nondimensional thickness of the wall elements.

Figure 10: Finite-element mesh

Boundary Conditions

In the case of an axial turbine stage with exhaust diffuser, the flow field at the rotor trailing edge plane in the stationary frame of reference is identical to the inlet flow field of the diffuser. The transformation from the velocity field $\vec{v}$ in the rotating frame of reference to the velocity field $\vec{\varepsilon}$ in the stationary frame of reference is given by

$$\vec{\varepsilon} = \vec{u} + \vec{w}, \quad (17)$$

with $\vec{u}$ as the vector of the circumferential speed. Due to the passage and tip leakage vortices the flow field at the trailing edge plane of a turbine rotor is highly three-dimensional. Furthermore, these vortices as well as the blade wakes are rotating relative to the diffuser. This means that the diffuser inlet boundary condition is time dependent and an unsteady computation is required. It is well known that the computation of three-dimensional, unsteady, viscous flow fields is very time consuming. Due to this fact, the pitchwise averaged velocities adopted from the cascade measurements have been used as the inlet boundary condition. In the present investigation, there is no swirl at the rotor trailing edge plane. Since the swirl depends on the load of the turbine, the assumption of no swirl means that the design point of the turbine stage is investigated. However, some amount of swirl is present in the trailing edge flow field influenced by the tip leakage jet (Fig. 7). Due to limitations of the available hot wire anemometry system, it was not possible to measure the turbulence intensity of the flowfield at the cascade trailing edge plane. So the turbulence intensity of the cascade inlet freestream flowfield ($T_u=5\%$) was used at the diffuser inlet. Assuming isotropic turbulence, the turbulent kinetic energy can be calculated from the velocity and the turbulence intensity via

$$k_T = \frac{3}{2} \left( T_u u_w M_S \right)^2. \quad (18)$$

The rate of dissipation was determined from

$$\varepsilon_T = C_\mu \frac{k_T^3}{0.01}. \quad (19)$$

This means that an characteristic lengths scale of 1% of the blade pitch was used. No-slip boundary conditions were applied to the hub and to the casing wall, respectively. The set of elliptic equations needs a boundary condition at the outlet plane as well. It is possible to specify either the velocity or the surface stress vector. At the outlet boundary, which is located at the diffuser outlet plane, the surface stress vector was set to zero in a weighted sense. In the case of high Reynolds numbers, this so-called traction free condition is equivalent to a prescribed static pressure distribution. This outlet boundary condition matches the constant ambient pressure at diffuser exit. The derivatives of $k$ and $\varepsilon$ with respect to the axial direction were set to zero at the outlet plane.

Computational Results

For the evaluation of the numerical method, the diffuser flow field with constant inlet velocity distribution and wall boundary layers has been computed. The velocity distribution in the inlet boundary layers was approximated by the $1/7$th-power law. The inlet blockage was varied in the range of $B_2=0.007$ to 0.05. Fig. 11 shows the computed pressure recovery factor versus the inlet blockage. The pressure recovery factor decreases with increasing inlet blockage. The computational method predicts the expected influence of the inlet boundary layer thickness on the diffuser performance. As the blockage decreases, the kinetic energy in the inlet boundary layer increases and the diffuser performance is enhanced. For comparison purposes, the measured pressure recovery factor for the inlet blockage of $B_2=0.02$ is also included in Fig. 11 (Sovran and Klomp, 1967). The agreement between computation and measurement is quite good.

Figure 11: Diffuser pressure recovery factor $C_{PD}$ as a function of the inlet blockage $B_2$

In a subsequent step, the boundary conditions, adopted from the cascade measurements have been applied to the diffuser inlet. The boundary layer at the hub remains as the $1/7$th-power law. Due to the pitchwise averaging of the quantities at the cascade trailing edge plane, the diffuser inlet flowfield near the casing appears as an annular wall jet. The intensity of this wall jet increases with increasing gap. In Fig. 12, the diffuser pressure recovery factor versus the relative rotor gap of the upstream turbine stage is plotted. The dependency of the pressure recovery factor is rather weak. It is assumed that the low intensity of the wall jet is responsible for this observation. The pressure recovery factor increases in the range of $\tau/c=1.5$ to 2.5% and is rather constant at narrower and wider gaps, respectively.
cascade with large tip gaps. At the same time, the wall jet due to the
turbines of De Cecco et al. (1995) on a geometrically similar turbine
develops at the same time, the growth of the tip leakage losses begins to stagnate. 

Starting from zero gap, the specific work of the turbine stage 
a = aT/C of the specific work aE is divided by the specific work of the turbine 
stage followed by the exhaust diffuser (+ Variation of 

turbine stage and an exhaust diffuser has been presented. Special 
attention has been given to the effect of the tip gap on this interaction. 
The tip clearance losses and the flowfield at the trailing edge 
plane have been adopted from measurements in a linear cascade wind 
tunnel. Due to the low Mach numbers, the flow has been treated as 
incompressible. In contrast to a compressor rotor, the relative motion 
between rotor blade and casing decreases the tip leakage flow in a 
turbine rotor. In the linear cascade experiment, the influence of the 
relative motion has not been taken into account. A comparison between the rotor tip leakage flow in a 1.5 stage turbine and the corresponding 
linear cascade is given by Kaiser and Bindon (1997). In general, the trends in their results agree fairly well with the linear 
cascade data although the effect of relative motion results in a modified gap discharge coefficient. Another discrepancy between the linear 
turbine cascade and the rotating blade row lies in the pressure distribution in spanwise direction. A radial pressure gradient is established in a rotating blade row. This influences not only the flow in the blade channels (secondary flow in the blade boundary layers) but also the flow in the downstream diffuser. As investigated experimentally by Kruse and Quest (1980), separation of the hub boundary layer in the annular diffuser downstream of a turbine stage can occur.

In a subsequent step, the flowfield at the cascade trailing edge 
plane was used as the inlet boundary condition for the numerical 
investigation of the flow in an annular diffuser. To avoid an expensive 
unsteady computation, the pitchwise averaged quantities have been 
computed. The re-energizing of the casing boundary layer is caused by 
an annular wall jet. This is a crude approximation, since the diffuser 
inlet flow field produced by a real turbine stage consists of rotating 
passage and tip leakage vortices, as well as blade wakes. The computation 
supplies a weak increase of the diffuser pressure recovery factor 
as the tip gap is opened.

The results from the cascade measurements and the diffuser computations were then coupled by means of an analytical interaction model. Although the present investigation is based on some approximations, it is able to describe the reaction of the annular diffuser on the upstream turbine stage. For tip gaps of practical interest, specific work and efficiency of the turbine stage/diffuser-combination are independent of the rotor gap. This means that the common rule to make the tip gap as tight as possible is no longer valid for a last gas turbine stage. However, the reaction of the improved diffuser influences the pressure ratio of each turbine stage in a multistage environment.

CONCLUSIONS

An investigation of the aerodynamic interaction between a last gas turbine stage and an exhaust diffuser has been presented. Special attention has been given to the effect of the tip gap on this interaction. The tip clearance losses and the flowfield at the trailing edge plane have been adopted from measurements in a linear cascade wind tunnel. Due to the low Mach numbers, the flow has been treated as incompressible. In contrast to a compressor rotor, the relative motion between rotor blade and casing decreases the tip leakage flow in a turbine rotor. In the linear cascade experiment, the influence of the relative motion has not been taken into account. A comparison between the rotor tip leakage flow in a 1.5 stage turbine and the corresponding linear cascade is given by Kaiser and Bindon (1997). In general, the trends in their results agree fairly well with the linear cascade data although the effect of relative motion results in a modified gap discharge coefficient. Another discrepancy between the linear turbine cascade and the rotating blade row lies in the pressure distribution in spanwise direction. A radial pressure gradient is established in a rotating blade row. This influences not only the flow in the blade channels (secondary flow in the blade boundary layers) but also the flow in the downstream diffuser. As investigated experimentally by Kruse and Quest (1980), separation of the hub boundary layer in the annular diffuser downstream of a turbine stage can occur.

In a subsequent step, the flowfield at the cascade trailing edge plane was used as the inlet boundary condition for the numerical investigation of the flow in an annular diffuser. To avoid an expensive unsteady computation, the pitchwise averaged quantities have been applied. The re-energizing of the casing boundary layer is caused by an annular wall jet. This is a crude approximation, since the diffuser inlet flow field produced by a real turbine stage consists of rotating passage and tip leakage vortices, as well as blade wakes. The computation supplies a weak increase of the diffuser pressure recovery factor as the tip gap is opened.

The results from the cascade measurements and the diffuser computation have then been coupled by means of an analytical interaction model. Although the present investigation is based on some approximations, it is able to describe the reaction of the annular diffuser on the upstream turbine stage. For tip gaps of practical interest, specific work and efficiency of the turbine stage/diffuser-combination are independent of the rotor gap. This means that the common rule to make the tip gap as tight as possible is no longer valid for a last gas turbine stage. However, the reaction of the improved diffuser influences the pressure ratio of each turbine stage in a multistage environment.

The results from the cascade measurements and the diffuser computation have then been coupled by means of an analytical interaction model. Although the present investigation is based on some approximations, it is able to describe the reaction of the annular diffuser on the upstream turbine stage. For tip gaps of practical interest, specific work and efficiency of the turbine stage/diffuser-combination are independent of the rotor gap. This means that the common rule to make the tip gap as tight as possible is no longer valid for a last gas turbine stage. However, the reaction of the improved diffuser influences the pressure ratio of each turbine stage in a multistage environment.
REFERENCES


