EXPERIMENTAL DETERMINATION OF THE FLOW FIELD IN THE TIP REGION OF A LP-STEAM TURBINE

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Abstract

Pneumatic multihole probes were applied to determine the local flow field in the tip region of a LP-steam turbine. For three tip clearances measurements were carried out varying the operating point of the turbine from full to partial load each.

The maxima of the tip clearance flow and the resulting jet flow in the adjacent axial-radial diffuser show a strong dependency on the operating point, the width of the tip gap, and the axial distance to the rotor. In contrast, the radial extension of the jet flow is primarily influenced by the operating point and limited to the upper 10% of the flow channel. The leakage flow rate and the interaction with the main flow are increasing when widening the tip clearance which leads to a significant global efficiency deficit. In order to prevent a smooth main flow the design is changed by casing treatment upstream of the running wheel yielding a positive overlap of the gap. Its effect on the flow is investigated and compared to the other configurations of the turbine.

1 Introduction

A safe operation of steam and gas turbines requires a minimum of tip clearance between the rotating running blades and the stationary wall casing. This is due to centrifugal forces acting on the blades and different thermal expansion of blades and casing during start and shutdown procedures.

There are two effects of the tip clearance flow. First, the mass flow through the tip gap eludes the energy conversion process of the turbine. Second, it induces further losses due to its interaction with the secondary flow of the cascade as well as with the boundary layers of the blades and the casing. This interaction strongly influences the entire flow regime in the tip region of the running blades and, in the case of a last stage, in the adjacent axial-radial diffuser. Especially in LP-turbines, that kind of losses counts for a substantial part of the overall losses (Keller, 1980).

In connection with the secondary flow inside the rotating flow channel, the flow over the blade tip from the pressure to the suction side forms a highly complicated, three-dimensional flow pattern. It leads to a reduced deflection in the cascade and to a change of the flow-separation characteristics in the peripheral zones. The active pressure gradient in circumferential direction is reduced in the tip region and an interference appears with the upper passage vortex. Furthermore, the flow over the tip produces the leakage vortex which rotates in opposite direction to the upper passage vortex. The tip clearance flow and its affiliated vortex are diminished in the case of a turbine, since the relative motion of the wall is opposite to the clearance flow thereby preventing the material inside the boundary layer from finding its way from the pressure to the suction side (Cordes, 1963).

As a result of the enormous pressure difference in axial direction across the blade tip of cantilevered and highly twisted running blades of last stages, an almost axial flow develops through the tip gap in form of a jet flow, which evokes a significant supersonic region downstream of the wheel. The expanding clearance flow supplies the boundary layer of the adjacent diffuser with its kinetic energy. This energy allows to realize a stronger deceleration of the flow and therefore to recover more outlet energy of the last stage. The supersonic region is limited by a shock caused by the back pressure which is dependent on the condenser pressure. At the point where the shock wave hits the solid wall, the boundary layer thickness increases and the flow may even separate when the boundary layer cannot cope with the pressure gradient (Maier and Wachter, 1988).

For the last decades numerous investigations have been undertaken in that field. In order to conceive the influence of the clearance flow on the overall efficiency of a turbine and to supply engineers with a basis of design criteria, diverse experiments have been carried out on test machines (Lakshminarayana, 1970; Bammert et al., 1968; Traupel, 1966; Dejc and Trojanovskij, 1973; Kofský and Nusbaum, 1967). This way of proceeding has led to empirical or semi-empirical approximations of loss coefficients, which are highly dependent on the blade geometry and the conditions of the machine. Therefore, the results of the equations differ in a wide range. On the other hand, parameter studies with simplified cascade test rig help to examine the physical phenomena inside the tip gap and to come up with flow models (Bindon, 1988; Bindon and Morphis, 1988; Yaras et al., 1989; Moore and...)

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Tilton, 1987; Yamamoto, 1988). Using these models, numerical calculations are applied to determine the three-dimensional, viscid flow through a stage including the tip clearance (Moore et al. 1989; Wadia, 1985). Since these investigations on models only represent idealized cases of the real flow conditions, measurements in test machines are still imperative to detect the influences of relative motion as well as of changed angles of incidence to the next stage for example.

The test rig for LP-steam turbines at the University of Stuttgart represents an effective facility to carry out flow measurements in the tip region of the last stage. Besides the detailed results of the flow parameters the overall efficiency of the turbine is obtained from the global data. In the present paper probe measurements are described for three gap widths varying the operational range from partial to full load each. Additionally, a change in the design has been made by means of casing treatment which led to a positive overlap of the gap. It allows to influence the wall boundary layer of the incoming flow of the running wheel and consequently to determine the possible reduction of the clearance flow.

2 Test Facility

The test rig for LP-steam turbines at the University of Stuttgart is thoroughly described in former publications (Wachter and Eyb, 1982; Stetter et al., 1992). The flow investigations in the tip region of the last stage were carried out on a downscaled model of a modern steam-power turbine with an exit area of 10 m² in the original. The facility in the scale of 1/4.2 was set up in a three-stage configuration allowing incoming flow conditions to the last stage which resemble quite close the real flow in original turbines. Figure 1 shows a longitudinal-section of the test facility and some operational data.

The hub ratio of the last stage is 0.49 and the length of the highly twisted running-blades 232 mm. The geometrical data in the tip region of the last stage and at the inlet of the adjacent axiradial diffuser are described in Fig. 2. A higher diffuser efficiency is achieved by reducing the divergence of the flow channel in the axial gap to 15°. Further downstream, the flow in the diffuser experiences a strong deflection into radial direction towards the condenser which is located underneath the turbine.

In order to ensure a safe operation of the turbine the indispensable radial gap is set to 2.3 mm at shut-down. The gap is designed with a smooth, conical surface representing a negative overlap. During the investigations, material of the inner side of the casing wall was removed to increase the tip clearance width in two steps to 4.9 mm and 6.55 mm.

![Figure 1: Longitudinal section of the last-stage test rig](image)

![Figure 2: Geometrical data in the tip region of the last stage](image)

**Figure 2**: a) Geometrical data in the tip region of the last stage  
b) Radial clearance at the last stage  
c) Casing treatment upstream of the running wheel

### Table 1: Blade-tip design conditions at \( H_{rel} = 0.9 \)

<table>
<thead>
<tr>
<th>configuration</th>
<th>( \delta ) [mm]</th>
<th>( \Delta L ) [mm]</th>
<th>( \tau_1 ) [%]</th>
<th>( \tau_2 ) [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.3</td>
<td>-2.3</td>
<td>0.99</td>
<td>4.38</td>
</tr>
<tr>
<td>2</td>
<td>4.9</td>
<td>-4.9</td>
<td>2.11</td>
<td>9.33</td>
</tr>
<tr>
<td>3</td>
<td>6.55</td>
<td>-6.55</td>
<td>2.82</td>
<td>12.47</td>
</tr>
<tr>
<td>4</td>
<td>6.55</td>
<td>+1.0</td>
<td>2.82</td>
<td>12.47</td>
</tr>
</tbody>
</table>

### Table 2: Investigated turbine configurations

The turbine condition is characterized by the volumetric-flow number \( \varphi = c_{ax}/u_m \) where \( c_{ax} \) represents the axial flow velocity calculated from the equation of continuity and \( u_m \) the circumferential velocity in the mid-section. The flow measurements were carried out at three turbine conditions, which are at the design point \( (\varphi = 0.45) \), at partial load \( (\varphi = 0.30) \), and at full load \( (\varphi = 0.56) \). The condenser level, the temperature at the turbine inlet, and the rotational speed were kept constant at 65 mbar, 150°C, and 12,600 rpm respectively.
3 Measurement Techniques

Pneumatic probes are appropriate tools to determine the three-dimensional flow field of last stages with transonic velocities (Stetter et al., 1992). The flow measurements are performed by means of four-hole wedge probes and three-hole cobra probes. Their measurement planes are sketched in Fig. 3. Up- and downstream of the running wheel the probes are radially guided in traversing devices over 5% and 20% of the dimensionless channel height respectively. The devices, which are sealed against steam and vacuum, are mounted on the movable diffuser ring. Once the flow direction is found by zero balancing the lateral probe pressures, data processing with an interpolation procedure, which uses the probe's calibration data, becomes an easy task in order to obtain information about the flow vector.

The wedge probe meets best the three-dimensional flow in the last stages by its asymmetric shape in radial direction, Fig. 4. The sharp wedge makes it very sensitive to the flow direction as well as suitable for high Mach numbers, because the lateral holes will not be hit by a compression shock caused by the wedge. The thermocouple, which is integrated in the probe head, allows to determine the local density and finally the absolute flow velocity.

Close to the wall and in the gap itself, miniaturized cobra probes are applied. In the case of an attached flow, the vector can be defined in a two-dimensional plane which is parallel to the contour surface. Thus, the probe head is inclined to the outer diffuser contour by an angle of 15° and 20° respectively, Fig. 4. Regarding the size of the probe head, no thermocouple is integrated. Hence, this type of probe directly yields no velocity values.

Both probes are calibrated over a range of Mach numbers from 0.3 to 1.6 yielding pressure and temperature coefficients for every adjustment. Experiments were carried out in the calibration tunnel to investigate the interactions between the cobra probe and solid walls. Due to type and size of the probe, this error is of small order and can be neglected.

In order to get an idea of the clearance flow and the heavily expanding jetflow in the adjacent diffuser, 36 taps are installed along the outer wall to detect the static pressure distribution at steady-state flow conditions of the turbine.

To prevent intrusion of steam into holes and tubes of all the pressure lines (taps and probes), continuous purging by atmospheric air is used. The resulting pressure increase in the tubes is calculated and taken into account during data processing (Jarosch et al., 1980).

Besides the data for flow-field measurements, other quantities have to be obtained in order to define the condition of the turbine and the overall efficiency. The total mass flow through the machine, the enthalpy drop between turbine inlet and outlet, the effective power, the losses due to friction in the bearings, and the condenser pressure are of special interest.

4 Experimental Results

The following diagrams include a comparison of the experimental results between the four turbine configurations.

4.1 Static pressure distribution at the outer diffuser contour

Figure 5 illustrates the static pressure distribution along the outer diffuser contour from the trailing edge of the guide-vane passing the running blade towards the diffuser outlet. The data are given as a ratio of static and condenser pressure versus the dimensionless diffuser-contour length. In all diagrams a slightly increasing pressure is found downstream of the guide-vane which is caused by the conical opening of the flow channel. The pressure level strongly depends on the load of the turbine and the gap width. The following acceleration starts in the gap but is mainly initiated by supersonic expansion and local acceleration which results from the strong deflection of the flow out from the axial into the radial direction.

Figure 3: Measurement planes of the wedge probe (WP) and the cobra probe (CP)

Figure 4: 4-hole wedge probe (WP) (top) – 3-hole cobra (CP) (bottom)
The gap between the blade tip and the solid wall of the turbine begins close to the trailing edge of the running blades. Hence, the supercritical regime begins close to the trailing edge of the running blades. The supercritical regime, characterized by extremely low pressure values (with minima less than 50% of the condenser pressure at full load), shows a strong pressure increase caused by a compression shock. However, due to the interaction of the boundary layer and the shock wave the pressure curve at the wall does not follow with a distinct jump but with a gradual increase.

While the upstream pressure levels are diminished with rising gap widths at the same loads, the pressure minima at the outlet of the turbine are increasing and also move downstream into the diffuser. A comparison of the cases with the same clearances (ΔL = -6.55 mm and +1 mm) yields a significant difference in the pressure upstream of the leading edge of the running blade caused by the sudden enlargement of the flow channel due to the casing treatment. When the flow is entering the tip gap, the pressure recovers again and both variations show similar distributions on the same levels.

4.2 Flow parameters in the tip region

Former investigations at the last stage indicate the necessity to take measurements in radial as well as in circumferential direction to get representative values of the entire flow field (Stetter et al., 1992). In contrast, the flow quantities of the following measurements show the independence of the probe position in circumferential direction in the tip region of the running blades, i.e., the wakes of the guide-vanes are mixed out downstream of the running wheel in the peripheral zones. Thus, these measurements were only performed on a radial traverse.

In order to get an impression of the local flow field in the tip region for the different configurations, Figures 6 and 7 illustrate the radial distribution of the Mach number and the absolute exit angle of the stage in the measurement plane 3. The further development of the flow in axial direction (plane 5) is shown in Figures 8 and 9.

4.2.1 Mach number distribution

All configurations show a stronger clearance flow connected with a more powerful jetflow when the load of the turbine is being increased. This is caused by the rising enthalpy drop and therefore by a higher pressure ratio at the gap. A rise of the velocity maximum is also found, when enlarging the width to 4.9 mm (ΔL = -4.9 mm) and corresponding turbine conditions are compared to each other. In contrast, they almost stay constant when taking the next step to ΔL = -6.55 mm.

Moving downstream from the gap exit into the diffuser, a distinct reduction of the velocity peaks of the jetflow is observed. In contrast, the changes in the velocity distribution of the main flow are of smaller order. The clearance flow influences the main flow up to 90% of the channel height, while the radial extension is only dependent on the load but not on the width of the tip gap. The interactions of the jetflow with the adjacent diffuser wall and the main flow on the other side are responsible for the reduction of the velocity in axial direction. On either side, the jetflow experiences a deceleration which is caused by friction effects at the wall and by vortex generation between the jetflow and the main flow due to differences in the velocities. This phenomenon leads to the reduction and the smoothening of the curves.

Besides the reduction of the flow, a radial displacement of the velocity peaks is visible between plane 3 and 5. When proceeding downstream, the maxima move from the wall into the interior of the diffuser. The reason for this development is given by an act of balancing between the velocity gradients on either side of the jetflow. While the gradient close to the wall exceeds the gradient on the side to the main flow in plane 3, they are almost of the same strength in measurement plane 5. Due to the stronger gradient, the shear stress and therefore the friction drag is found to be more intense. Close to the wall the

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Figure 5: Static pressure distribution along the outer diffuser contour (running wheel, casing treatment)

More details of the expansive phenomenon are gained by considering the critical pressures \( p^* \) (marked by the corresponding dark symbols), which are calculated from the total pressure of probe measurements in the plane upstream of the running blades. In all cases, the supercritical regime begins close to the trailing edge of the running blades. Hence, the gap between the blade tip and the solid wall of the outer casing can be regarded as a convergent nozzle allowing the flow to accelerate up to sonic speed. Due to the supercritical pressure ratio, the supersonic expansion continuous right behind the outlet of the gap up to a location where the back pressure is reached. The end of the supersonic region, characterized by extremely low pressure values (with minima less than 50% of the condenser pressure at full load), shows a strong pressure increase caused by a compression shock. However, due to the interaction of the boundary layer and the shock wave the pressure curve at the wall does not follow with a distinct jump but with a gradual increase.

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reduction of the flow velocity in the jetflow is higher than on its side to the main flow thereby leading to the mentioned radial displacement of the peak.

4.2.2 Flow angle distribution
Due to supersonic expansion and interaction of the clearance flow with secondary flow, a significant rise ($\varphi = 0.45$ and $\varphi = 0.56$) as well as a clear decrease ($\varphi = 0.30$) of the exit angle are observed in the tip region downstream of the running wheel. Inside the clearance gap, the theoretical direction of the steam is given by the exit flow direction of the guide-vane (about 18°) which is equal to about 162° in the system of the running-wheel. Due to the high pressure ratio across the running wheel, the initiated supersonic expansion causes a reduction of the flow deflection, i.e. a deflection towards the axial direction. When superpositioning the directions of the jetflow and the main flow in the cases of design and full load in the diffuser inlet region, the rise of the exit angles up to 120° in rotational direction is found close to the wall. In contrast, the exit angle of the jetflow is diminished at partial load, since there is already a strong momentum in rotational direction in
Comparing the angles of all configurations, the levels of the main flow as well as the peak values in the jetflow show an upward tendency with increasing clearance width. Depending on the turbine condition, a rise up to 25° in the main flow and 10° in the jetflow is notable comparing sequent configurations. The mixing process in axial direction leads to a smoothening of the curves towards the wall. However, at full load the supersonic expansion is not finished yet in plane 5 which is also recognized in the distribution of the Mach numbers. Hence, a distinct rise of the exit angle is still found in that point of operation.
and different gap widths, an efficiency can be extrapolated, which defines the efficiency at zero gap. The difference to that efficiency at the width \( r_a \) yields the clearance loss coefficient \( \xi_a \). This coefficient is calculated from the measurement results and compared to relations which are given in the literature (Bammert et al., 1968; Traupel, 1966; Dejc and Trojanovskij, 1973), Fig. 12. These relations are derived from simplified calculations of the mass-flow rate through the gap. Since Traupel’s diagrams are limited to \( r_a < 0.03 \), a value is only depicted at the original tip gap by extrapolating his diagrams. A fairly good agreement is found between the data according to Bammert and Traupel. They are mainly depending on angles and velocities upstream and downstream of the running wheel which are taken from the probe measurements. Dejc’s relation requires the degree of reaction in the mid-section of the blades and two general constants for tip clearances with negative overlap and smooth surface. These data give higher loss coefficients for all widths. In contrast, lower values are obtained from the measurements.

4.3 Efficiency of the turbine

From the global data of the experiment the isentropic enthalpy drop from the entry to the condenser, the output, and consequently the total efficiency \( \eta \) of the turbine can be calculated.

Figure 11 displays the efficiencies of all investigated configurations versus the volumetric flow number. According to the expectations, the best efficiency is obtained by the original tip clearance with the maximum in the design point at \( \varphi = 0.45 \). When widening the width, a drop of the efficiencies is being observed which can be approximated by a relation of \( \Delta \eta / \Delta r_a = 2 \% \). This drop is caused by the mass flow through the gap, although the resulting jetflow improves the flow conditions in the adjacent diffuser by increasing the energy of the boundary layer along the outer diffuser wall. Comparing the configurations with a negative overlap \( \Delta L \), the parallel curves indicate the independence of the tip clearance losses on the turbine load.

No improvements are noticed in the case of \( \Delta L = +1 \) mm. At partial load, the positive overlap even leads to a decrease of efficiency.

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Summary

At the test rig for models of LP-steam turbines at the University of Stuttgart, the flow field in the tip region of the last stage was examined. With the help of pneumatic multihole probes, the highly three-dimensional flow field was determined varying the tip clearance as well as the turbine condition from full to partial load.

1. Inside the tip gap, the flow is accelerated up to sonic speed in axial direction. Close to the trailing edge a supersonic expansion follows which leads to a powerful jetflow in the adjacent axial-radial diffuser.
2. The velocity maxima of the jetflow are highly dependent on the clearance, the load of the turbine, and the axial distance to the running wheel.
3. A strong momentum in rotational direction is found in the flow in the tip region of the running wheel.
4. The radial extension of the jetflow is restricted to the upper 10% of the flow channel and independent of the clearance width.
5. The decrease in efficiency of the turbine with increasing gap width can be characterized by the relation of $\Delta \eta / \Delta \tau = 2\%$. It is caused by the higher leakage flow across the last stage, whereas the flow in the adjacent diffuser is improved at the same time.
6. No improvements in efficiency can be stated by means of casing treatment (small positive overlap of the radial clearance).

The global data of the investigations confirm the strong influence of the radial gap width on the efficiency of the turbine, while the detailed results of the flow parameters can be used in comparison to modern calculation results.

Nomenclature

| CF  | clearance flow |
| CP  | cobra probe    |
| $h$ | blade length   |
| $H_{rel}$ | dimensionless channel height (0. = hub, 1. = tip) |
| $L_{rel}$ | dimensionless length of outer diffuser contour |
| Ma  | absolute Mach number |
| MF  | main flow      |
| $p$ | pressure       |
| $s$ | chord length   |
| WP  | wedge probe    |
| $\alpha$ | absolute flow angle |
| $\beta$ | tip clearance |
| $\tau$ | gap ratio: $\tau = \delta / h$ |
| $\xi_{sp}$ | clearance loss coefficient |
| $\Delta L$ | overlap |
| $\eta$ | total efficiency of the turbine |

Subscripts

| stat | static |
| cond | condenser |