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INVESTIGATION OF FILM-COOLING EFFECTS ON THE FLOW OF A TRANSONIC TURBINE BLADE IN A STEAM TEST RIG

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Abstract

The influence of film-cooling on the aerodynamic performance of a turbine cascade is investigated at a steam test rig. It is shown to what extend superheated steam as working fluid enables the imitation of the original gas-turbine flow. Moreover it is described which defects are made because of the different substance properties of the flow media and which characteristics of the flow can be well imitated.

Introduction

Investigations on the flow of turbines are furthermore required for the efficiency improvement of modern power plants. A possibility to increase the performance of gas turbines is to increase the turbine inlet temperature. During the last years the inlet temperatures went up very fast and exceeded the material weight limits of the turbine blades. This fact led to different methods of protection by cooling. A proven method is the film-cooling. A protecting film of cold air is blown out of the blade and flows between the hot mainstream and the blade surface.

From the thermodynamical point of view film-cooling is a very effective protection of the blade but from the aerodynamical point of view the penetrating cold-air jet is a disturbance of the main flow which inevitably leads to higher losses. Therefore it is an important aim to find possibilities of predicting and minimazing losses caused by the effects of film-cooling.

Within the scope of an AG Turbo project at Siemens KWU a new numerical procedure for the calculation of film-cooling effects is developed and the results should be supported by experimental investigations. A condition for unrestricted transferability of the results is the correspondence of the numerical and experimental flow field. This correspondence is expressed by the equality of so-called similarity coefficients. This contribution explains to what extend the coefficients of the steam test rig correspond to those of the real turbine and which consequences are caused by the disparity.

Test facility

Figure 1 shows the steam test rig of the Siemens KWU flow laboratory which was up to now expendet for investigations on uncooled turbine cascades and other power plant components. The present cascade is a scaled (scaling factor 1.29) version of the first stage stator cascade of the V84.4 gas turbine. It contains 3 film-cooled blades between 2 additional blades on each side. The steam supply is a closed loop system with wet or superheated steam as working fluid. Table 1 shows the operating range of the steam test rig.

p_{t0}:	$\leq 2.0 \cdot 10^5\ N/m^2$
T_{t0}:	$\leq 500\ K$
Tu_0:	$\leq 1\%$
Ma_2:	≤ 2.0
Re_2:	$\leq 1 \cdot 10^6$
$\dot{m}_\infty$:	$\leq 3.4\ kg/s$

Table 1: Possible range of flow parameters

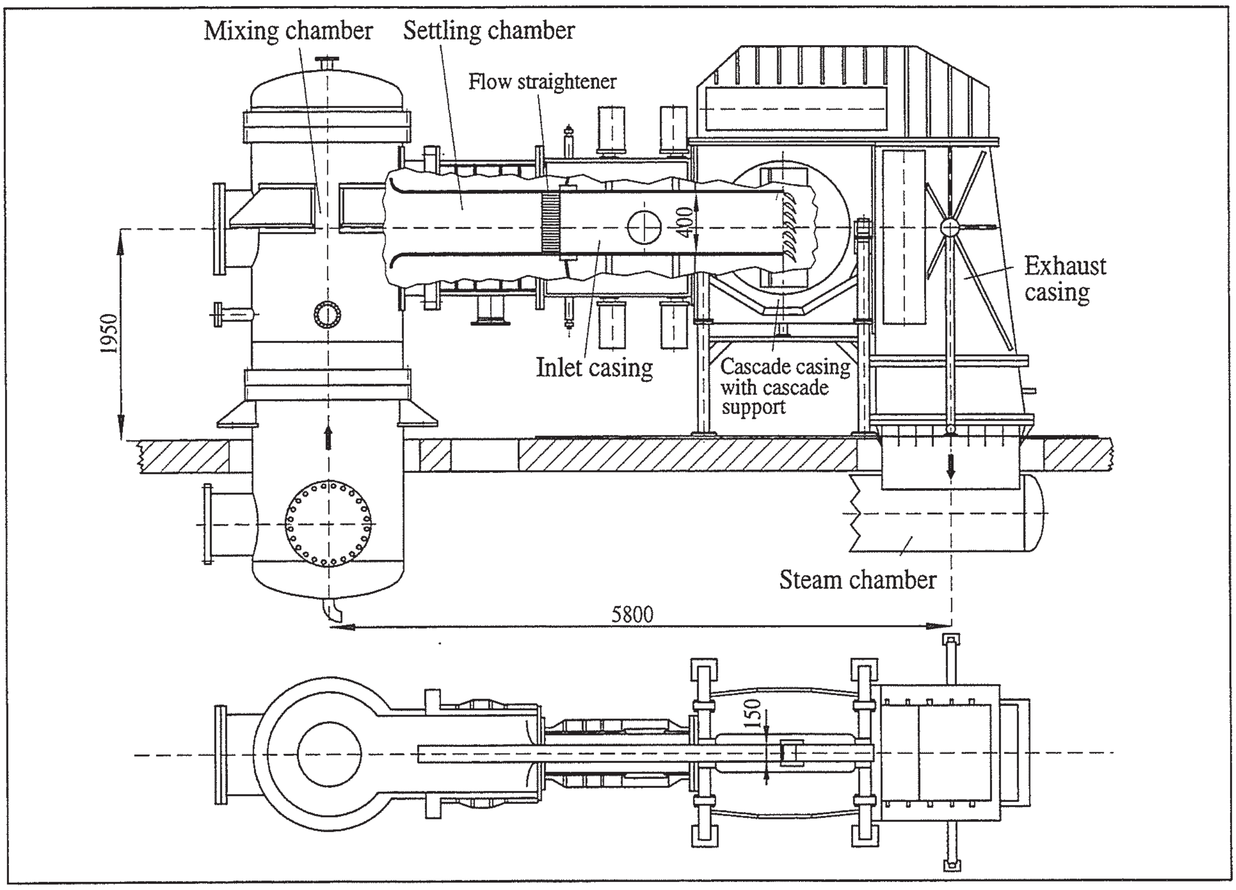


Fig. 1: Steam test rig

For the investigations of the cooled turbine blades it was necessary to enlarge the test rig facility. The cooling steam was splitted up from the main steam and in conformance with the required conditions cooled down and regulated. An additional connection to the three film-cooled blades guaranteed a balanced supply. Figure 2 shows the investigated blade.

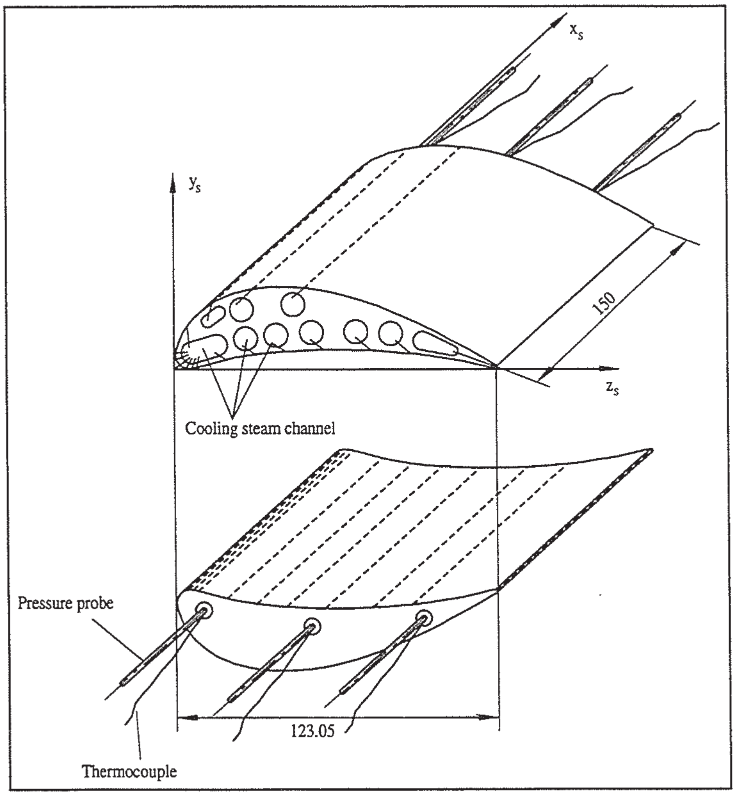
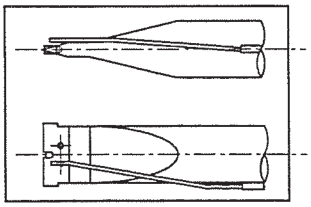


Fig.2: Investigated blade

Measurement equipment

The flow field in front of the cascade is observed by a total pressure probe and a number of static pressure holes. The inlet flow angle is measured with a 2-hole-wedge probe. A NiCr-Ni thermocouple enables the determination of the inlet temperature.



In the outled flow field a traversable 5-hole-wedge-probe is installed. It is fixed at a constant angle and the outled angle is inquired by the pressure difference of the two wedge-pressures.
Especially for the film-cooling investigations the outled pressure probe is equiped with a special calibrated NiCr-Ni thermocouple. In this way it is possible to determine the kinetic efficiency loss by comparing the inlet and outlet enthalpie. The cooling steam conditions are determined by pressure and temperature probes inside of the cooling steam channels (look fig. 2).

Derivation and calculation of the decisive similarity coefficients

Besides the preparation of the steam test rig it was required to inquire the comparability of the numerical and the expected experimental results. Therefore it was usefull to compare the similarity coefficients of the calculated and experimentally investigated flow and explain the errors caused by the defects of the coefficients.
Table 2 shows the different conditions of the numerical and experimentally investigated flow fields.

		Numerical/ Original	**Experimental**
Geometry:			
Chord Length l_{Bi}	[mm]	95.76	123.05
Span h	[mm]	116.7	150
Pitch t	[mm]	70.07	90.1
Flowparameter:			
Mainstream:			
Working Fluid		hot gas	superheated steam
Gas constant R	[kJ/(kgK)]	0.2925	0.4615
Temperatur T_H	[K]	1723	500
Pressure p_H	[bar]	15.69	0.35
Isentropic Exponent κ		1.295	1.305
Density ρ_H	[kg/m^3]	3.113	0.151
Dynamic Viskosity η	[kg/(ms)]	$56.63 \cdot 10^{-6}$	$16.18 \cdot 10^{-6}$
Specific Heat c_p	[kJ/(kgK)]	1.2306	1.975
Thermal Conductivity λ	[W/(mK)]	$81.92 \cdot 10^{-3}$	$33.28 \cdot 10^{-3}$
Cooling fluid:			
Working Fluid		Compressor Air	Superheated Steam
Gas constant R	[kJ/(kgK)]	0.2872	0.4615
Temperatur T_K	[K]	671	345 - 450
Pressure p_K	[bar]	16.32	0.364 - 0.580
Isentropic Exponent κ		1.3659	1.3279
Density ρ_K	[kg/m^3]	8.4	0.22 - 0.35
Dynamic Viskosity η	[kg/(ms)]	$33.43 \cdot 10^{-6}$	$12.00 \cdot 10^{-6}$
Specific Heat c_p	[kJ/(kgK)]	1.0722	1.3279

Table 2: Different flow field conditions

The similarity coefficients arise from the three dimensional consideration of the flow field:

Constancy of mass:

$$\frac{\partial p}{\partial t}+\frac{\partial(\rho\cdot u_i)}{\partial x_j}=0 \tag{1}$$

Constancy of impuls:

$$\begin{aligned}\rho\cdot\left(\frac{\partial u_i}{\partial t}+u_j\cdot\frac{\partial u_i}{\partial x_j}\right)&=-\frac{\partial p}{\partial x_i}+\mu\cdot\left(\frac{\partial^2 u_i}{\partial x_j\partial x_j}+\frac{1}{3}\cdot\frac{\partial^2 u_j}{\partial x_i\partial x_j}\right)\\&\quad+\frac{\partial u}{\partial T}\cdot\left(\frac{\partial T}{\partial x_j}\cdot\frac{\partial u_j}{\partial x_i}+\frac{\partial T}{\partial x_j}\cdot\frac{\partial u_i}{\partial x_j}-\frac{2}{3}\cdot\frac{\partial T}{\partial x_i}\cdot\frac{\partial u_j}{\partial x_j}\right)\end{aligned} \tag{2}$$

Constancy of energy:

$$\rho\cdot c_p\cdot\left(\frac{\partial T}{\partial t}+u_j\cdot\frac{\partial T}{\partial x_j}\right)-\left(\frac{\partial p}{\partial t}+u_j\cdot\frac{\partial p}{\partial x_j}\right)=\frac{\partial\lambda}{\partial T}\cdot\frac{\partial T}{\partial x_j}\cdot\frac{\partial T}{\partial x_j}+\lambda\cdot\frac{\partial^2 T}{\partial x_j^2}+\mu\cdot\Phi \tag{3}$$

with Φ (Dissipation):

$$\Phi=-\frac{2}{3}\cdot\left(\frac{\partial u_j}{\partial x_j}\cdot\frac{\partial u_k}{\partial x_k}+\frac{\partial u_j}{\partial x_k}\cdot\frac{\partial u_k}{\partial x_j}\right) \tag{4}$$

and the indizes i,j,k=1,2,3 and $x_1, x_2, x_3 = x, y, z$.

For the inquiry of the similarity coefficients the superheated steam is treated as an ideal gas:

$$p=\rho\cdot R\cdot T \tag{5}$$

By transforming these equations and observation of the geometric conditions (Fig. 3) the following similarity coefficients have to be inquired.

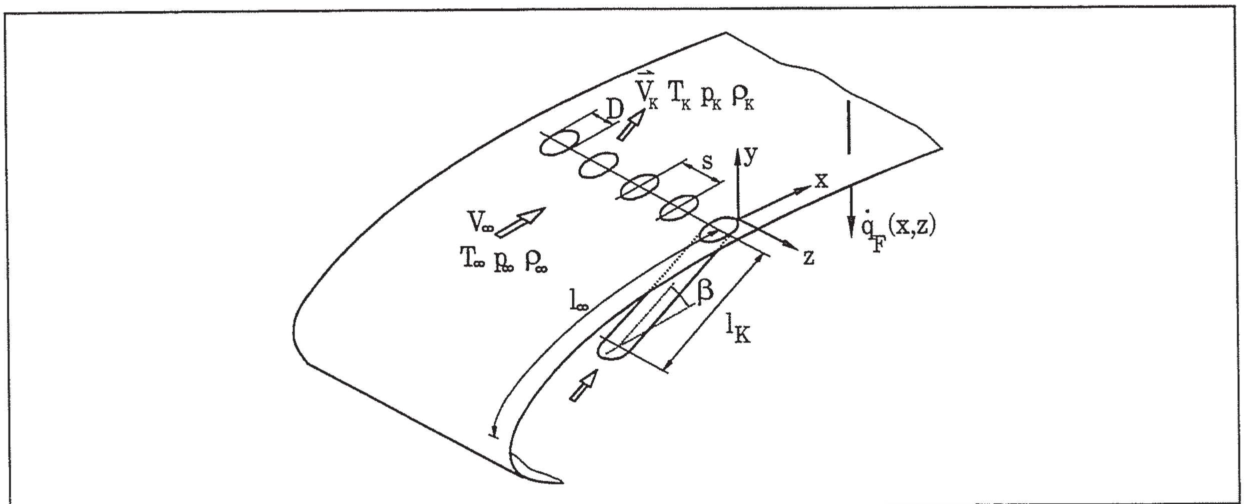


Fig. 3: Fragment of a film-cooled turbine blade with the geometric conditions

Aerodynamic coefficients:

$Ma_\infty = \frac{c_\infty}{\sqrt{\kappa \cdot R \cdot T_\infty}}$ Mach number of mainstream

$Re = \frac{\rho_\infty \cdot c_\infty \cdot l_{Bi}}{\eta_\infty}$ Reynolds number of mainstream

$I = \frac{(F \cdot \rho \cdot c^2)_K}{(F \cdot \rho \cdot c^2)_\infty}$ Ratio of coolant-to-mainstream impuls

$M = \frac{(c \cdot \rho)_K}{(c \cdot \rho)_\infty}$ Coolant-to-mainstream mass flux ratio evaluated at the film hole exit (blowing ratio) or mass flow rate

Thermodynamic coefficients:

$\pi_T = \frac{T_\infty - T_K}{T_\infty}$ Ratio of temperatures

$Pr = \frac{c_{p\infty} \cdot \eta_\infty}{\lambda_\infty}$ Prandtl number

$\kappa = \frac{c_{p\infty}}{c_{p\infty} - R}$ Isentropic exponent

$\pi_\lambda = \frac{\lambda_\infty - \lambda_K}{\lambda_\infty}$ Ratio of thermal conductivities

$\pi_{c_p} = \frac{c_{p\infty} - c_{pK}}{c_{p\infty}}$ Ratio of specific heats

The calculation of these coefficients leads to the results mentioned below:

- By adjusting the pressure ratio at the test rig it is possible to choose the Mach number of mainstream:

$$Ma_\infty = \sqrt{\frac{2}{\kappa - 1}\left(\left(\frac{p_{t\infty}}{p_\infty}\right)^{\frac{\kappa-1}{\kappa}} - 1\right)} \tag{6}$$

- The inquiry of the Reynolds number shows the problem which is caused by using steam as working fluid. Because of the low density of superheated steam it is impossible to reach as high Reynolds numbers as in the real turbine. The ratio arised to:

$$\frac{Re_{STR}}{Re_{Turb}} = 0.15$$

An increase of the experimental pressure level would improve this ratio but because of the following considerations it was possible to perform the investigations at the lower pressure

level and in spite of the low ratio of Reynolds number it is possible to compare the numerical and experimental results.
Remaining examinations show that the efficiency of a turbine cascade is independent from the Reynolds number if it is greater than $Re_\infty \approx 5.5 \cdot 10^5$ (see Fig.4).
The Reynolds number for the investigations at the steam test rig would be about $Re_\infty \approx 4.5 \cdot 10^5$ so comparisons are possible to be made.

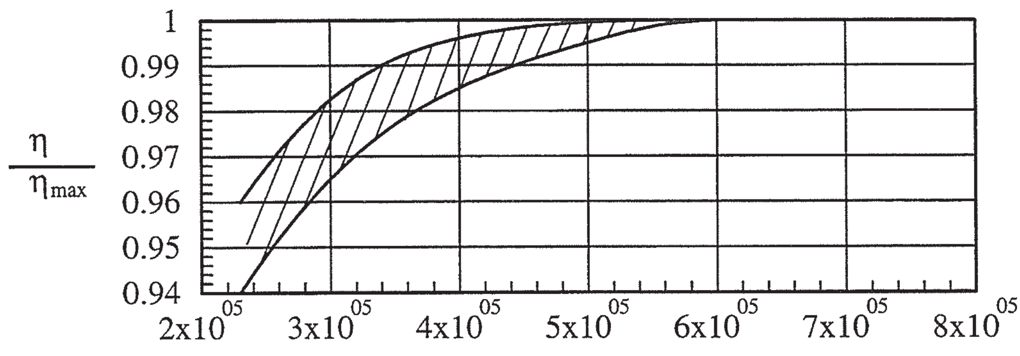


Fig. 4: Influence of the Reynolds number on the efficiency

Additionally the state of the boundary layer has a great influence on the mixing behaviour of main and cooling stream. The Reynolds number on the other hand has no influence on the mixing behaviour if the boundary layer is turbulent. At the investigated turbine blade the shower-head makes the boundary layer overturn from laminar to turbulent. So one can establish that the Reynolds number has only a minor influence on the experimental results. Experiments at different Reynolds numbers confirmed this consideration.

- The ratio of coolant-to-mainstream impuls was varied during the experimental investigations at a range of $0.9 \leq \frac{I_{STR}}{I_{Turb}} \leq 3$ for the simulation of different penetration depths.

- Comparable to the ratio of impuls the coolant-to-mainstream mass flux ratio was varied from 0.9 to 1.3. By variing the pressure ratio of main to cooling steam this was possible without any problems.

Apparent differences between the coefficients arise at the thermodynamic coefficients. The calculation of the coefficients mentioned above leads to the following results:

- $\frac{Pr_{STR}}{Pr_{Turb}} = 0.89$

- $\frac{(\pi_T)_{STR}}{(\pi_T)_{Turb}} = 1.97$

A manipulation of this ratio is limited by the temperature limitations of the insulating material of the test rig and the demand of superheated steam for the cooling steam. So the maximum temperature of mainstream is about 500 K and the minimum temperature of cooling steam about 345 K.

– Ratio of isentropic exponents:

$$\frac{\kappa_{STR}}{\kappa_{Turb}} = 1.008$$

– Ratio of thermal conductivities:

$$\frac{(\pi_\lambda)_{STR}}{(\pi_\lambda)_{Turb}} = 0.86$$

– Ratio of specific heats:

$$\frac{\left(\pi_{c_p}\right)_{STR}}{\left(\pi_{c_p}\right)_{Turb}} = 2.54$$

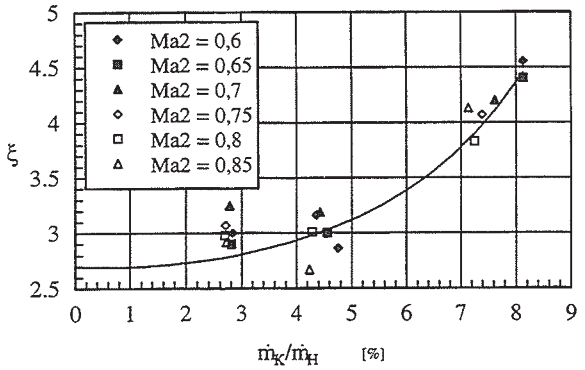
These results show that the investigation of the aerodynamic behaviour (e.g. local outlet angle, local outlet velocity and pressure, kinetic and total pressure loss) is possible. The thermodynamical behaviour (e.g. local temperatur distribution or film-cooling effectiveness) on the other hand can not be imitated realistically at the steam test rig.

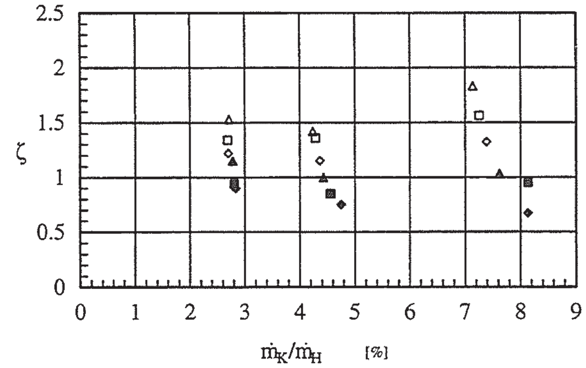
Results

The first stage stator cascade of the Siemens KWU gas turbine V84.4 was investigated at the steam test rig. During the experimental course different parameters were varied and the influence on the turbine flow recorded. The following graphs show a small fragment of the gained results.

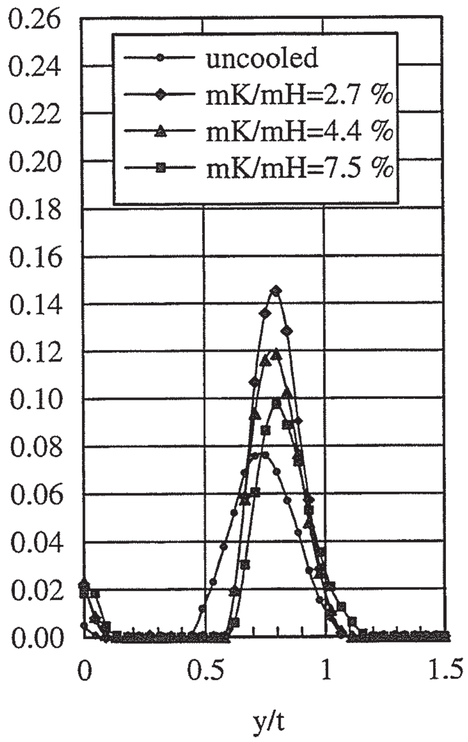
Influence of the blowing ratio on the kinetic energy loss and total pressure loss coefficient:

$$\xi = 1 - \frac{\left(\dot{m}_K + \dot{m}_H\right) \cdot c_2^2}{\dot{m}_H \cdot c_{2,is}^2 + \sum_{i=1}^{N}\left(\dot{m}_{K,i} \cdot c_{K,is,i}^2\right)} \qquad \zeta = \left(1 - \frac{p_{t2}}{p_{t1}}\right) \qquad (7)$$

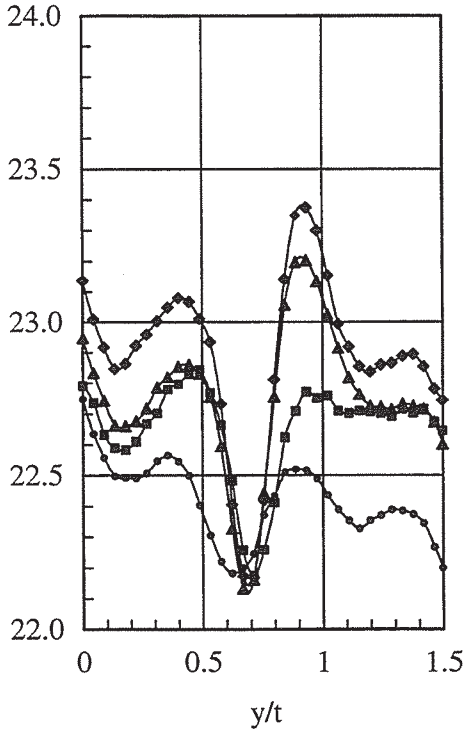




Influence of the blowing ratio on the local outlet pressure and outlet angle:



$(p_{t1}-p_{t2y})/(p_{t1}-p_2)$ at $Ma_2=0.75$



local outlet angle β_2 at $Ma_2=0.75$

Summary

The steam test rig was intended to be used for investigations on film-cooled turbine blades. This intend is only reasonable on condition that the flow of the turbine cascade at the steam test rig is simmilar to the flow inside of the original turbine. The similarity coefficients enable a check of it.

The contribution mentioned above shows that on the one hand the aerodynamic similarity coefficients are almost identically but on the other hand the thermodynamic coefficients diverge. That means that investigations on the aerodynamic behaviour of a film-cooled cascade certainly are feasible. The thermodynamic procedure on the other hand can not be reflected realistically.

So there is a new possibility for investigations of the aerodynamic flow behaviour at film-cooled turbine blades.The results can be used for better understanding of the performance inside of turbine cascades and comparison with numerical results.

Nomenclature

c, u	velocity
c_p	specific heat at constant pressure
D	diameter
Ma	Mach number
p	pressure
R	gas constant
Re	Reynolds number
t	time
T	temperature, pitch
Tu_0	turbulence level
x, y, z	cartesian coordinates
α	inlet angle
β	outlet angle
η	dynamic viscosity
κ	isentropic exponent
λ	thermal conductivity
ρ	density
ξ	kinetic energy loss
ζ	total pressure loss

Indizes

K	cooling fluid
t	total
Turb	original turbine
STR	steam test rig

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