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Investigation of the steam-cooled blade in a steam turbine cascade

Dieter Bohn¹, Jing Ren¹, Karsten Kusterer²

(1. Institute of Steam and Gas Turbines, RWTH Aachen University,
Templergraben 55, D-52056 Aachen, Germany;

2. B&B AGEMA, Gesellschaft für Energietechnische Maschinen,
und Anlage GmbH, Jülicher Strasse 338, 52070 Aachen, Germany)

Abstract: With the increasing demand for electricity, an efficiency improvement and thereby reduced CO₂ emissions of the coal-fired plants are expected in order to reach the goals set in the Kyoto protocol. It can be achieved by a rise of the process parameters. Currently, live steam pressures and temperatures up to 300 bars and 923 K are planned as the next step. Closed circuit steam cooling of blades and vanes in modern steam turbines is a promising technology in order to establish elevated live steam temperatures in future steam turbine cycles.

In this paper, a steam-cooled test vane in a cascade with external hot steam flow is analyzed numerically with the in-house code CHTflow. A parametric analysis aiming to improve the cooling effectiveness is carried out by varying the cooling mass flow ratio. The results from two investigated cases show that the steam cooling technique has a good application potential in the steam turbine. The internal part of the vane is cooled homogeneously in both cases. With the increased cooling mass flow rate, there is a significant improvement of cooling efficiency at the leading edge. The results show that the increased cooling mass flow ratio can enhance the cooling effectiveness at the leading edge. With respect to trailing edge, there is no observable improvement of cooling effectiveness with the increased cooling mass flow. This implies that due to the limited dimension at the trailing edge, the thermal stress cannot be decreased by increasing the cooling mass flow rate. Therefore, impingement cooling configuration at the trailing edge might be a solution to overcome the critical thermal stress there.

It is also observed that the performance of the cooling effective differs on pressure side and suction side. It implicates that the equilibrium of the cooling effectiveness on two sides are influenced by a coupled relationship between cooling mass flow ratio and hole geometry. In future work, optimizing the hole geometry and cooling steam supply conditions might be the solutions for an equivalent cooling effectiveness along whole profile.

Key words: steam cooling; steam turbine; conjugate heat transfer

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Nomenclature

A	area, m ²	p	pressure, Pa
$\underline{A}$, $\underline{B}$, $\underline{C}$	Jacobian matrices	Pr	Prandtl number
E, F, G	flux vectors (body fitted coord.)	Re	Reynolds number
E, F, G	flux vectors (Cartesian coord.)	T	temperature, K
H	vane height, mm	U	vector of conservative variables
J	cell volume	e_{tot}	specific total energy
L	axial chord length, m	p	pressure
Ma	Mach number	q	heat flux vector
		q_x	x -component of heat transfer rate per area, W/m ²

s	vane surface coordinate, m
u, v, w	velocities in Cartesian coordinates, m/s
V	normal velocity at inlet, m/s
y	distance from wall
y^+	nondimensional wall coordinate
Greek letters	
α	incidence angle, ($^{\circ}$)
λ	thermal conductivity, W/(m $\cdot$ K)
η_c	cooling efficiency
ρ	density, kg/m ^{3}
μ	dynamic conductivity
ϕ	mass flow ratio
τ	shear stress tensor, N/mm ^{2}
τ_{ij}	components of shear stress tensor, N/mm ^{2}
Indices	
c	cooling
fl	fluid
h	hot
is	isentropic
ref	reference
s	solid
t	total
w	wall
x, y, z	partial differentiation (Cartesian coord.)
ξ, η, ζ	partial differentiation (body fitted coord.)
1	at inlet
2	at outlet
Superscripts	
n	actual time step
Abbreviations	
LE	leading edge
PS	pressure side
SS	suction side
TE	trailing edge
CCT	Conjugate Calculation Technique

Introduction

Today the market for electricity generation is dominated by combined cycle power plants based on natural gas. However, in the near future the demand for coal-fired power plants will continue to expand due to the indigenous deposits of coal in large developing countries like China and India. In Europe, coal or lignite will also play an important role in the foreseeable future, and although the demand for new plant will be limited, there will be a substantial market for up grading or replacement of existing plants.

For Germany the current predictions assume that 20% more electrical power will be generated in 2020 using about the same amount of fuel as today and that over 50% of the power will come from coal and lignite fired steam power plant. Due to unfavorable C/ H ratio of coal, concerning CO $_2$ emissions, the only way to keep them low is to increase the efficiency. The increase of thermal efficiency of the steam cycle can be achieved by a rise of the process parameters. It is related to raising the temperature, as well as the pressure of the steam generated in the boiler.

Currently, live steam pressures and temperatures of up to 300 bars and 923 K are planned as a next step. This means an increase of 50~ 100 K in comparison to plants presently under construction. Such a large increase will only be possible if all thermally high loaded components of the plant are made of new materials and/or new technologies for cooling and thermal insulation are applied.

To withstand these advanced operating conditions, materials with improved high temperature capability are required. For this reason a series of R&D projects started in recent years in Europe, the USA and Japan aiming at the qualification and testing of new materials for advanced steam parameters at the same time^[1].

Cooling of the thermally high loaded components in steam turbines is considered an applicable technology in order to allow steam cycles with elevated live steam temperatures. In gas turbines, the steam cooling technique attracts a lot of interest from both manufactures and engineers. Based on the knowledge of steam cooling technology in gas turbines, this technique becomes now more and more attractive for steam turbine application as well.

Apart from its favourable fluid properties, process steam used as cooling fluid provides several benefits. In a steam turbine, steam can be provided ‘inexpensively’ at favourable supplying conditions of pressure and temperature levels. Furthermore, the amount of cooling steam is not high because only moderate temperature reduc-

tions on the vane or blade will be necessary with respect to material properties. Due to high cooling potential of steam with respect to its fluid properties, convective cooling in closed circuits can be established using simple straight radial geometric cooling configurations.

The main focus of the former investigations has been the usage of the steam cooling technique in the gas turbine. Nomoto et al.^[2] have performed extensive experimental studies on vane steam cooling in a gas turbine. In their cooling efficiency studies, they have pointed out that under the high pressure steam fluid conditions the inner convection cooling achieves enough cooling efficiency.

Bohn et al.^[3] have investigated the steam cooling technique in the gas turbine experimentally and numerically. The results show that, in order to reach a defined cooling efficiency level, medium steam pressure supply might be advantageous in comparison to a high pressure level in the supply. Sufficient cooling of the trailing edge becomes problematic for steam cooling applications in the given configuration.

Further, Bohn et al.^[4] have investigated the application of steam cooling technique in steam turbine. It is shown that the steam cooling effectiveness is quite acceptable with a minor cooling mass flow rate. This work presents the further investigations on the application of the steam cooling technique on the steam turbine.

The thermal load of a steam turbine's vane is reduced by a convective steam cooling technique. The hot steam conditions are set 923 K and 300 bars, which are the planned parameters of the steam turbine in the next step. A parametric analysis aiming to improve the performance of the steam cooling technology in the steam turbine is carried out by varying the mass flow rate of the cooling fluid. The in-house CFD solver CHTflow with the Conjugate Calculation Technique (CCT) is used to carry out the numerical simulations.

1 Configuration and geometric modelling

1.1 Test configuration

Figure 1 shows the geometrical configuration for a test rig. It consists of a rectangular duct containing a three vane cascade. The central vane can be cooled convectively by supplying steam to 22 straight radial cooling passages.

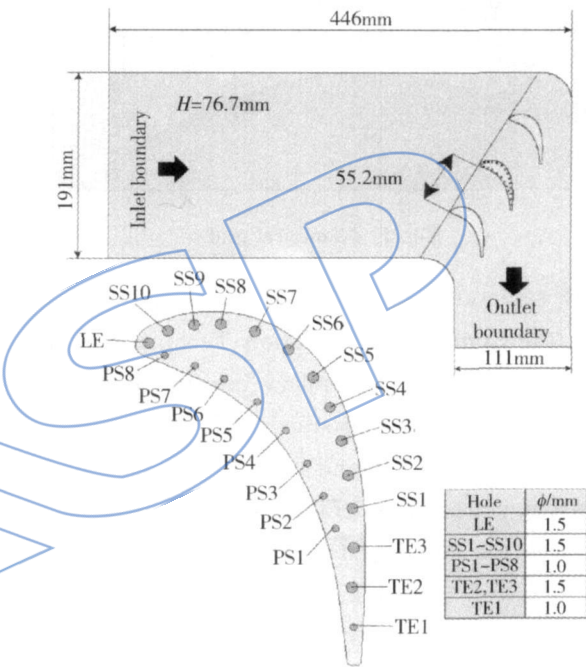


Fig. 1 Test configuration

The arrangement of the cooling passages within the cooled vane is illustrated in a mid cross sectional cut on the bottom left side of Fig. 1, while the corresponding diameters of the cooling passages are listed in the table on the bottom right side of Fig. 1. In the area of the trailing edge, where a minimum wall thickness must be considered, the cooling passage (TE1) has a diameter of just 1 mm.

1.2 Numerical grid

Figure 2 shows the 3D numerical grid used to solve the discretised governing equations. Due to the requirements of the numerical method ("Conjugate Calculation Technique") the hot steam path, the cooling flow passage and the solid body of the central vane are composed of a

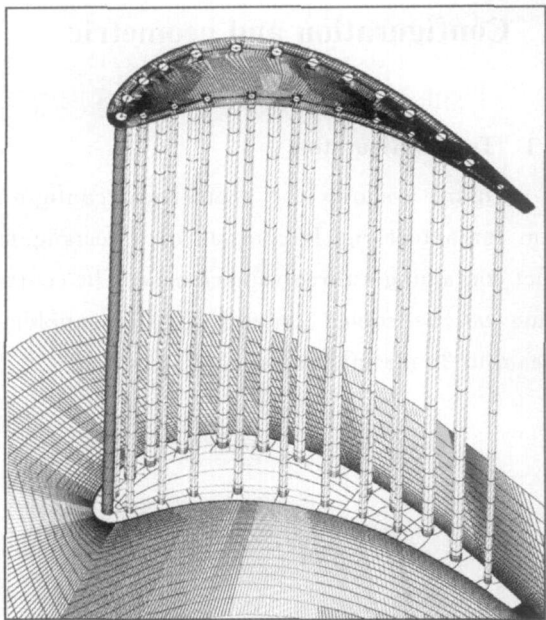


Fig. 2 Numerical grid

fully structured numerical multi-block grid. Using the Conjugate Calculation Technique, there is no need for any boundary information about heat transfer characteristics at fluid-solid interfaces. Due to the demand for exact calculation of the temperature gradients at all solid - fluid interfaces, the non dimensional wall distance y^+ of the first cell row is kept at an acceptable level for a conjugate calculation approach ($y^+ < 1$).

$$y^+ = y \frac{\sqrt{\tau_w \rho_w}}{\mu_w}$$

(1)

Such a high resolution of the thermal boundary layer plus the need for including the hot gas path, cooling flow passage and the solid body of the central vane rises the number of grid points up to 2.8 million. Divided into 172 structured blocks, the whole computation domain is separated into several tasks, which are calculated separately on a PC-LINUX-cluster making use of the parallel structure of the code.

1.3 Boundary and steam-supply conditions

In the presented investigations, the boundary conditions of the hot steam are set to the planned inlet parameters of the HP steam turbine. The cooling steam parameters refer to those of the outlet steam from the HP steam turbine.

The inlet temperature of the hot steam is

set 923 K, which is the hot steam inlet temperature expected for the HP steam turbines in the next step. Table 1 shows the parameters conditions of the hot steam flow.

Table 1 Hot steam boundary conditions and characteristic numbers

Parameters conditions	
Temperature/ K	923.0
Velocity (inlet) / (m/s)	80.0
Static pressure (outlet) /bar	293.5
Re (inlet)	2.89×10^7
Ma (inlet)	0.11

Table 2 Supply conditions of the cooling steam

Parameter	Case1	Case2
Static pressure(inlet) /bar	70.0	70.0
Temperature (inlet)/K	723.15	723.15
Re(inlet, $Re_{inlet, s}$)	3.55×10^4	3.58×10^5
Pr(inlet)	0.990 4	0.990 4
Ma(inlet)	0.045	0.458
Mass flow ratio/ %	0.068	0.34

Table 2 shows the supply conditions of the cooling steam. In order to show the influence of the cooling steam condition on the cooling effectiveness, two cases are defined for the supplied cooling steam. The supply conditions are listed in Tab.2 for case 1 and case 2 respectively.

The mass flow ratio is defined as the ratio of area, density and velocity of the cooling flow in the vane and the hot flow in one passage.

$$\varphi = \frac{A_c \rho_c V_c}{A_h \rho_h V_h}$$

(2)

Due to the changed boundary conditions of the cooling steam, the mass flow ratio of 0.068% and 0.34% result in case 1 and case 2 respectively. This means that the total amount of cooling steam in the case 1 is only a minor part of the mass flow in the complete cycle. In case 2, the total amount of the cooling steam is expected to be a normal one.

2 Conjugate calculation technique and validation

2.1 Numerical method

The conjugate calculations are preformed

with the CFD solver CHTflow, which has been developed in the Institute of Steam and Gas Turbines at RWTH Aachen University. It works on the basis of an implicit finite volume method combined with a multi-block technique. The physical domain is divided into different blocks for fluid flow and solid body regions. The full, compressible, three dimensional Navier-Stokes equations are solved in the fluid blocks. The Fourier equation is solved in the solid body blocks.

The governing equations for the conservative variables in arbitrary, body-fitted coordinates $\xi\eta\zeta$ with the fluxes normal to the $\xi\eta\zeta = \text{const.}$ cell faces are described for the fluid flow as

$$U_t + E_\xi + F_\eta + G_\zeta = 0 \quad (3)$$

with

$$U = J \cdot (\rho, \rho u, \rho v, \rho w, \rho e_{\text{tot}}) \quad (4)$$

$$E = J \cdot (E \cdot \xi + F \cdot \eta + G \cdot \zeta) \quad (5)$$

with e. g.

$$E = \begin{pmatrix} \rho u \\ \rho u^2 - \tau_{xx} \\ \rho uv - \tau_{xy} \\ \rho uw - \tau_{xz} \\ (\rho e - \tau_{xx})u - \tau_{xy}v - \tau_{xz}w + q_x \end{pmatrix} \quad (6)$$

The fluxes F and G are obtained analogously. q is the heat flux vector, which contains the Fourier heat flux. Making use of a Newton method leads to the following approximation of the conservation equations in implicit form.

$$\frac{\Delta U}{\Delta t} + (A^n \Delta U)_\xi + (B^n \Delta U)_\eta + (C^n \Delta U)_\zeta = - (E_\xi^n + F_\eta^n + G_\zeta^n) \quad (7)$$

Upwind discretization is used for the inviscid fluxes. With respect to numerical diffusion, a Godunov type flux-differencing is employed. In order to achieve the third order accuracy, van Leer's MUSCL-technique is used^[5]. The viscous fluxes are approximated using central differences. The resulting system of linear equations is solved by a Gauss-Seidel point iteration scheme. The closure of the Reynolds averaged equations is provided by the Baldwin-Lomax algebraic eddy-viscosity turbulence model^[6]. Due to the smooth internal walls, the model can also be ap-

plied for the calculation of the internal turbulent flow. The model has been chosen because of its low computation time and the good experiences in other calculations for a similar convective cooled vane^[7]. These results have shown that the Baldwin-Lomax mode performs very well in conjugate flow calculations without large flow separation.

Principally, both steam flows are treated as ideal gases. However, a real gas correction has been implemented based on steam tables for real steam conditions (IAPWS-IF97, Wagner et al.^[8]).

The coupling of fluid blocks and solid body blocks is achieved via a common wall temperature resulting from the equality of the local heat fluxes passing the contacting cell faces (Conjugate Calculation Technique (CCT)). Due to the high resolution of the boundary layer including the viscous sub layer (the dimensionless wall distance of the first cell is about $y^+ = 1$ or smaller), the local heat fluxes are determined with sufficient accuracy and the calculated wall temperature serves as an internal boundary condition for the coupling procedure (e. g.)

$$T_w = \frac{\frac{\lambda}{\lambda_0} T_s + T_\Pi}{1 + \frac{\lambda}{\lambda_0}} \quad (8)$$

where T_w , T_s and T_Π are the temperature on the wall, in the solid cell near the wall and in the fluid cell near the wall.

2.2 Code validation

A validation of the code can be found in Bohn et al.^[9,10]. Further applications and validation of the code have been presented for a convection-cooled turbine guide vane with radial cooling passages. To demonstrate the performance of the coupling procedure, results from Bohn et al.^[11] are shown in Fig. 3. The calculated and experimentally determined temperatures and pressures along the blade surface of a turbine nozzle guide with ten straight radial cooling passages (Mark II test case, Hylton et al.^[12]) supplied with air, are shown.

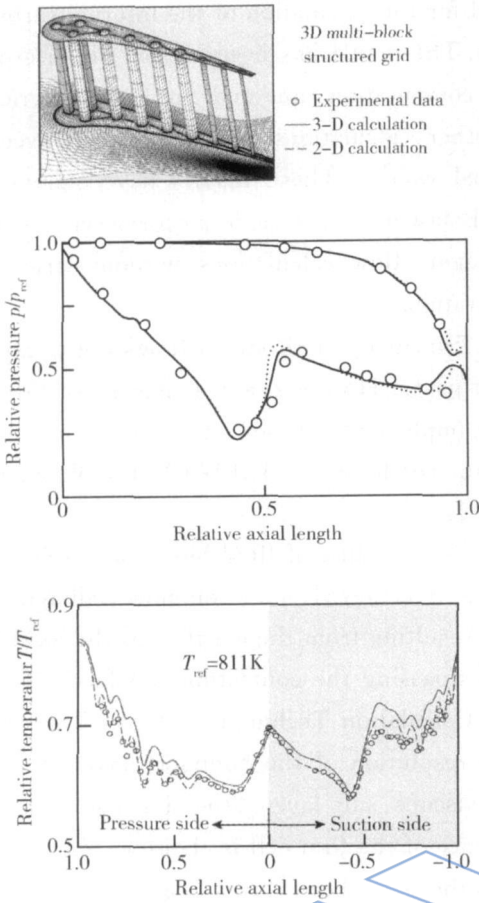


Fig. 3 Code validation temperature distribution(Mark II test case)

Due to the high heat transfer on the fluid side a local temperature maximum can be observed in the vicinity of the stagnation point. The decrease in fluid temperature together with a higher heat transfer in the blade (due to a smaller distance of the channels from the surface) leads to a rapid decrease in surface temperature along the suction side and the pressure side. Further along the pressure side there is a continuous increase in temperature. In contrast to the pressure side, along the suction side the following temperature distribution occurs. Starting from the local temperature minimum mentioned above, the strong shock leads to a steep increase in temperature. The deceleration of the flow results in a thickening of the boundary layer and transition to turbulent flow. These two phenomena lead to a rapid increase in surface temperature.

In the 2D calculation, the maximum local

differences are smaller than 2% over the whole surface. 3D calculations of the test case (Fig. 3, Bohn et al^[7]) have shown somewhat greater deviations between calculation and experiment. This is partly because the application of adiabatic endwall conditions neglects the heat flux from the passage through the endwalls leading to an increase of the vane surface temperatures. Meanwhile, while the 2D calculations benefits from the experimentally determined heat transfer coefficients inside the cooling holes and the medium temperature of the cooling fluid in the cooling passage, the coupled heat transfer of the internal flow and the vane in the 3D case is calculated by the code without additional heat transfer conditions. Thus, the 3D results are reached by the solution of the full conjugate problem (internal flow & vane & outer flow). Concerning the minimum use of boundary conditions, the 3D results can be stated as very good.

The results clearly show that the developed code with CCT is able to predict the thermal load of the blade with a sufficient degree of accuracy. The mutual influence of the cooled blade surface and the main stream is simulated correctly.

3 Results and analyses

3.1 Thermal analysis

As a result of the conjugate calculation approach, the surface temperatures and the temperature field in the vane solid body can be determined. Figure 4 shows the three dimensional temperature distribution in the flow passage and on the vane surface in case 1, in which the cooling mass flow ratio is about 0.068%. It is clear that the external surface temperature of the steam-cooled vane is lower than the surrounding hot steam. With the cooling steam taking up heat on its way from the inner sidewall to the outer sidewall, the external surface temperature rises from the hub to the tip. Therefore, the cooling effectiveness is stronger at the hub than the tip.

Figure 5 shows the three dimensional temperature distribution in the flow passage and on the vane surface in case 2, in which the mass

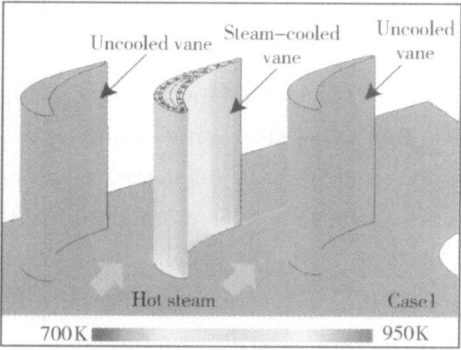


Fig. 4 3D temperature distribution in the passage and on the vane surface in case 1

flow ratio of the cooling steam is increased to 0.34%. It is clear that a strong cooling steam potential leads to significantly lower temperatures on the mainpart of the vane surface. It indicates that the cooling effectiveness can be enhanced in the main part of the vane surface with an increased cooling mass flow ratio. The external temperature varies more strongly from hub to the tip than in case 1.

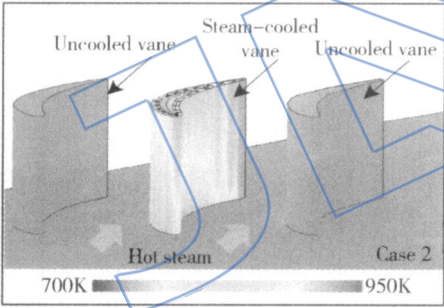


Fig. 5 3D temperature distribution in the passage and on the vane surface in case 2

Figure 6 illustrates the internal vane temperatures on the midspan for case 1 and case 2. In case 1, the temperature of the main part of the solid body is lower than 870 K and quite homogeneous. At the leading edge and the trailing edge, the temperatures increase to 910 K, which is only 13 K lower than the external hot steam. The temperature along the profile reaches around 900 K.

With the increased cooling mass flow ratio in case 2, the core temperature is decreased to 800 K. The main part of the vane reaches a lower temperature of about 820 K. Meanwhile, the

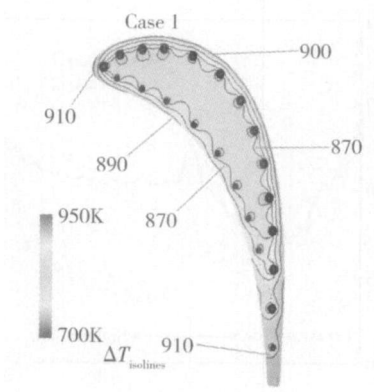


Fig. 6 Internal temperature distribution on the midspan of the steam cooled

Temperature of the leading edge decreases to 890 K, which is 32 K lower than the external hot steam and 20 K lower than case 1.

It means that the cooling effectiveness at the leading edge could be improved by increasing the cooling mass flow ratio. However, the temperature at the trailing edge remains 910 K in both case 1 and case 2. It means that the increase of the cooling mass flow ratio cannot weaken the thermal stress at the trailing edge.

Figure 7 shows the external surface temperature distributions along the profile in case 1 and case 2. Although the external surface temperatures will rise along the blade height due to the temperature increase of the cooling fluid, the temperature distributions on the midspan in Fig. 7 are representative of the distributions from the hub to the tip.

Both curves in Fig. 7 describe the general features of the steam-cooled vane, which is also similar to what happens in the steam-cooled vane of gas turbines. When the hot steam attacks the

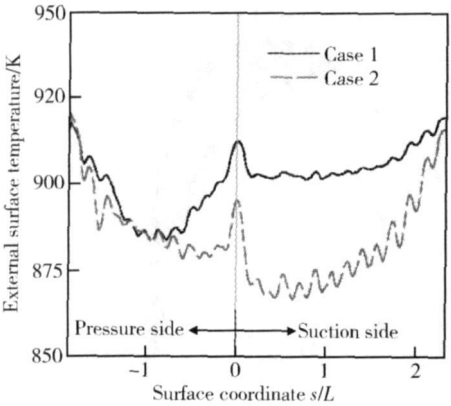


Fig.7 External surface temperature distribution on the midspan of the steam-cooled vane

leading edge of the steam-cooled vane, it impinges on the surface and creates a region of high heat transfer due to a sharp temperature gradient. A local temperature maximum can be observed in the vicinity of the stagnation point. The decrease in fluid temperature together with a higher heat transfer in the blade (due to a smaller distance of the channels from the surface) leads to a rapid decrease in surface temperature along the suction side and the pressure side. Further along the two sides there is a continuous temperature increase.

In case 1 (shown as the black line in Fig. 9), the temperature distribution on the pressure side looks like a valley while it is quite flat on the suction side. After the local minimums, the temperatures increase again towards the trailing edge. They reach the local maximums at the leading edge and the trailing edges. The averaged

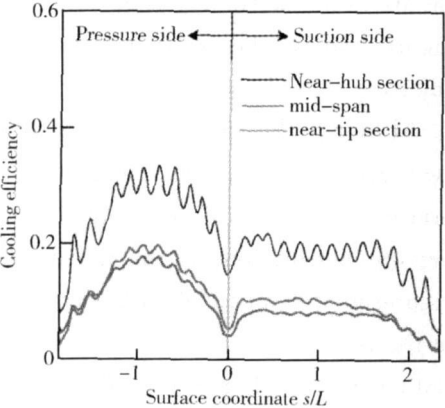


Fig. 8 Cooling efficiency distribution on the steam-cooled blade for case 1

temperature on the suction side is higher than on the pressure side.

In case 2 (shown as the red line in Fig. 9), the increased mass flow ratio causes a decrease of the external surface temperature along the main part of the vane. It is more significantly enhanced at the leading edge and the suction side while quite slightly on the pressure side. The averaged temperature on the suction side is lower than on the pressure side.

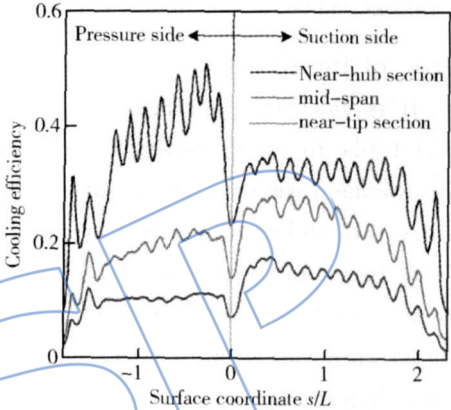


Fig. 9 Cooling efficiency distribution on the steam-cooled blade for case 2

The temperature distribution in case 1 and case 2 show different heat transfer characteristic on the suction side and pressure side. The cooling effectiveness on the pressure side is stronger than the suction side in case 2. It is vice versa in case 1. It implicates that an equilibrium thermal stress along both sides may be pursued by coupled relationship between the hole geometry and cooling mass flow ratio.

3.2 Cooling efficiencies

In order to investigate the cooling efficiency of the steam-cooling configuration in the test rig, Equation 9 is used for the calculation of the local cooling efficiency distributions

$$\eta = \frac{T_h - T_w}{T_h - T_c} \tag{9}$$

T_w is the local wall temperature, which is here a result of the conjugate calculation. Due to such a definition, the local efficiency distributions on the cutting planes show an alternating behavior in comparison to the wall temperature

distributions.

Figure 8 shows the cooling efficiency distribution on the steam-cooled vane in the investigated steam turbine cascade for case 1 on three radial sections. It can be observed that the cooling efficiency decreases when the cooling fluid passes from the hub to the tip. This is mainly because of an increase in the temperature of cooling fluid. The minimum of the cooling efficiency at the near hub sections both the leading edge and the trailing edge is approximately 8%. But at the mid-span and the near tip section the cooling efficiencies decrease to 5%.

If we consider that the mass flow ratio in this case is only about 0.068%, such a cooling effectiveness is acceptable. Nevertheless, with respect to the vane leading edge and trailing edge improvements are recommended. An increase of the cooling steam would be no problem for the given configuration, since the Mach number of the cooling steam at the inlet is only about 0.045.

In order to investigate the influence of the cooling mass flow ratio on the cooling effectiveness, the mass flow ratio is increased to 0.34% in case 2. The distributions of the cooling efficiency at three sections from the hub to the tip are shown in Figure 9. The cooling efficiency decreases gradually from the hub to the tip. This means the cooling fluid maintains an even cooling potential along the cooling flow passages.

Figure 10 compares the cooling efficiency distribution on the midspan of the steam-cooled vane in the investigated cases. It can be observed that with the increased cooling mass flow rate there is a significant improvement of cooling efficiency at the leading edge. It means that the increased cooling mass flow ratio can enhance the cooling effectiveness at the leading edge. With respect to trailing edge, there is no observable improvement of cooling effectiveness while increasing cooling mass flow. This implies that due to the limited dimensions of the trailing edge, the thermal stress cannot be weakened by increasing the cooling mass flow rate. Therefore, an im-

pingement cooling configuration at the trailing edge might be a solution to overcome the critical thermal stress there.

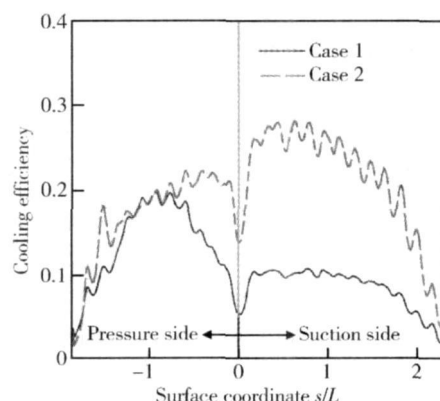


Fig. 10 Cooling efficiency distribution on the steam-cooled blade in a steam turbine (mid span)

It is also observed that the cooling efficiency differs on pressure side and the suction side. This implicates that the equilibrium of the cooling effectiveness on the pressure side and suction side are influenced by the coupled relationship between cooling mass flow ratio and the hole geometries. In future work, optimizing the hole geometry and cooling steam supply conditions might be helpful for finding the solutions of an equivalent cooling effectiveness along the blade.

4 Conclusion

A steam-cooled test vane in a cascade with hot steam flow has been analyzed numerically by applying the Conjugate Heat Transfer method. The boundary conditions of the hot steam are set the planned inlet parameter of the H P steam turbine in the next step. A parametric analysis aiming to improve the properties of the steam-cooling technology in steam turbines is carried out by varying the mass flow rate of the cooling fluid. The numerical solver CHTflow with the Conjugate Calculation Technique (CCT) is used to carry out the numerical simulations.

The results from two investigated cases show that the steam cooling technique has a good application potential in the steam turbine. The internal part of the vane is cooled homogeneously in both cases. While the cooling mass

flow rate increases, a significant improvement of cooling efficiency at the leading edge is observable. Therefore, it is concluded that the cooling effectiveness at the leading edge could be enhanced by increasing the cooling mass flow ratio. However, there is no observable improvement of cooling effectiveness with the increased cooling mass flow. So, the thermal stress at the trailing edge cannot be weakened by increasing the cooling mass flow rate. The impingement-cooling configuration at the trailing edge might be a solution to overcome the critical thermal stress there.

Both investigated cases show the nonequilibrium thermal stress on the pressure side and the suction side. The results show that the equilibrium of the cooling effectiveness on the pressure side and suction side are influenced by the coupled relationship between cooling mass flow ratio and the hole geometry. In future work, the coupled influence of the hole geometries and the cooling steam supply conditions might be helpful to pursue an equilibrium thermal stress along the blade.

Acknowledgments

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