Numerical Investigation of the Three-Dimensional Temperature Field in the Near Hole Region of a Film Cooled Turbine Vane

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ABSTRACT

The increased severity of the thermal environment of high pressure turbine blades and vanes requires accurate calculations for the successful design of these parts. In this paper, the prediction of the temperature field in the near-cooling-hole region on a film cooled turbine vane is presented. The surface distribution of the heat transfer coefficient and the film cooling effectiveness on the vane in presence of one or several film cooling injections is obtained from boundary layer calculations and via experimental correlations. Cooling jet coalescence is taken into account as well as the main parameters governing this physical phenomenon. The internal boundary conditions result from available correlations. The study was conducted on two different configurations: a flat plate including an injection through two rows of holes and a turbine vane including three injections through two rows of holes on the suction side. Thermal computations using a three-dimensional finite element code yield strong temperature distortions and high temperature gradients around the injection zones. The study also indicates that the three-dimensional temperature field just downstream of the injections becomes two-dimensional when jet coalescence takes place. The influence of one or several obstructed injection holes on the temperature field is studied; important effects are observed when the main flow temperature is high.

NOMENCLATURE

- D = hole diameter
- Rd = heat transfer coefficient without film cooling
- hf = heat transfer coefficient with film cooling
- L = hole length
- L1, L2 = adimensional parameters
- m = blowing rate
- Nu = Nusselt number
- Re = Reynolds number (based on chord length)
- S = lateral hole spacing
- Se = effective slot width (total surface of injection divided by the height of the vane)
- T = temperature
- Tu = free stream turbulence intensity
- X = abscissa. Distance downstream of injection hole
- Z = ordinate. Lateral distance from hole center
- a = exponent
- n = local film cooling effectiveness = n(X,Z)
- n = average film cooling effectiveness as a function of ordinate = n(X)
- np = average film cooling effectiveness predicted using the superposition principle = np(X)

INTRODUCTION

The increasing turbine inlet pressures and temperatures in advanced engines call for more and more effective cooling of high-pressure vanes, the most thermally stressed parts. One cooling process widely used today consists of injecting one or several cold films on the pressure and suction sides through cooling holes. The cold film formed downstream of injections results from the confluence of the cold jets from the discrete holes progressively mixed with the hot mainstream gases. These areas are characterized by a three-dimensional flow leading to
an irregular distribution of film cooling effectiveness and heat transfer coefficients. Heat transfer study of a turbine vane injection zone depends on these external boundary conditions, on the internal boundary conditions and on the injection geometry. All these elements tend to generate a three-dimensional temperature field which can lead to important local temperature distortions. The main objective of this paper is to propose a method for the prediction of the three-dimensional temperature field. The method used is described in the Figure 1 block diagram. It consists in establishing the external and internal boundary conditions and performing the conduction computation after having selected the discretization field.

Conduction was calculated on two different configurations with a material featuring properties usually characterizing metal turbine vanes. The first configuration corresponding to a flat wall represents a partial injection only. The boundary conditions applied are representative of the operation of a turbine vane. Only the curvature effect (the curvature being in practice progressive over the airfoil profile) was not considered for the computation of conduction. The second configuration is based on a suction side portion of an airfoil with three successive injections. A three-dimensional temperature field was calculated and the effect of a blocked hole on the temperature field was shown.

Each step is detailed in the following paragraphs with the corresponding assumptions.

**Boundary layer and heat transfer coefficient without a film**

Determining external boundary conditions on an airfoil begins with the calculation of the boundary layer and external heat transfer coefficient. This problem, even when injections are not considered, is far from being solved because the major difficulty lies mainly in determining the transition. It is essential to know the boundary layer condition over the whole profile, as this condition - apart from generating aerodynamic losses - also results in a more or less high heat transfer coefficient. SNECMA has developed a finite difference boundary layer code, presented by GUYON and ARTS (1989), adopting the Mc Donald and FISH transition model, completed by the ADAMS and JOHNSTON curvature model for transition computation. A mixing length type model is used. The results presented in Figure 2 indicate a good agreement between the heat transfer coefficient

**EXTERNAL BOUNDARY CONDITIONS**

The knowledge of film effectiveness and the heat transfer coefficient downstream of a hole injection is best obtained by computing three-dimensional flows. It is very complex to study the interaction of discrete jets with the boundary layer and the free stream flow and then the thermal boundary layer behavior. Theoretical approaches to this problem are in progress - in particular at SNECMA - with encouraging results. However, these methods are still in the research phase. In an attempt to provide a method for use before these comprehensive theoretical tools are available, this paper describes another approach to obtain a local distribution of film effectiveness and the heat transfer coefficient. External boundary conditions are established in three successive steps. They are:

1) - computing boundary layer and heat transfer coefficient distribution without a film
2) - computing average film cooling effectiveness and the average heat transfer coefficient downstream of injections from experimental correlations
3) - finally, investigating the local distribution of these parameters near and downstream of injection areas.

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obtained experimentally and that computed over the suction side of a high-pressure turbine vane. Experimental results come from the tests conducted in the C72 short duration, compression tube facility at the Von Karman Institute. A very noticeable change in the transition position is to be noted as a function of the Reynolds number value. Satisfactory results are also obtained on the pressure side as illustrated in Figure 3. A new method is being developed at SNECMA for computation in the laminar separation bubbles area, where high gradients exist, KLINK and KUNTZ (1990).

Fig 3 : Example of boundary layer computation on a HP vane - Pressure side Tu = 4%  - From GUYON and ARTS (1989)

Average film effectiveness and heat transfer coefficient

Film effectiveness and heat transfer coefficient are derived from experimental correlations obtained through tests on a linear cascade in the C72 short duration, compression tube facility at the Von Karman Institute. The tested vane incorporates a two-row injection with a hole spacing equal to three diameters and arranged on the pressure or the suction side. This vane is instrumented with thermocouples so as to measure the time-related variation of wall temperature over the test duration. The test rig and the analysis method are accurately described by ARTS and BOURGUIGNON (1989).

The main parameters governing the physical phenomenon, i.e. downstream Mach number, Reynolds number, turbulence intensity and blowing rate, have been taken into account. These tests provide the effectiveness and the Hf/H0 ratio, where Hf is the heat transfer coefficient with a film and H0 the heat transfer coefficient without a film, obtained either by boundary layer computation, or by tests on a solid vane without a film. The values are given as a function of the abscissa on the profile starting from the location of injection. Much work has already been conducted on the analysis of films injected on an airfoil, aiming at establishing an effectiveness law and a Hf/H0 ratio law. The experience shows the importance of selecting relevant parameters when establishing experimental correlations. In the case of turbine vanes, correlations of the following type have been sought:

- $\eta = n_0/(L1)^a$ for the film cooling effectiveness, where $n_0$ is the effectiveness at the location of injection.
- $L1$ is a function of the distance between the injection starting point and the considered position, blowing rate $m$ and effective slot width $S_e$.
- $a$ is a function of free stream turbulence intensity $Tu$ and Reynolds number $Re$.

Correlations allow correct integration of the effects of the main parameters previously listed. These effects are described in detail by ARTS and BOURGUIGNON (1989) for pressure side injection configurations. Some general remarks can be made concerning the suction side injection configurations. Effectiveness is practically the same for all blowing rates but the decrease along the profile becomes faster and the Hf/H0 ratio higher as the blowing rate increases. Boundary layer transition is affected by the film for some injection locations, the transition being moved forward compared to a configuration without film. Increasing the turbulence ratio causes a decrease in effectiveness which is more pronounced when the injection occurs close to the leading edge.

Therefore, for a given case, average film effectiveness and average heat transfer coefficient will be computed using these correlations obtained with four suction side configurations and four pressure side configurations. When studying an injection at a location on the profile other than the reference injections, a linear interpolation is performed between the results of the two adjacent injections. However this assumption can only be valid if both reference injections are affected by physical parameters in a similar way. Otherwise the results applying to the most similar reference case shall be retained.

Local film effectiveness and heat transfer coefficient

Information pertaining to the local distribution of film effectiveness and heat transfer coefficient issues from studies by several authors on injection through one or several rows of holes or injection through a single hole. In the latter case, extrapolating to configurations with one or two hole rows is carried out using the superposition principle, thoroughly described by GOLDSMITH (1970). This principle applies to film effectiveness computation in the presence of several successive injections. The author provides interesting data on the superposition principle with respect to its validity. The effectiveness predicted using this principle and the actual effectiveness obtained on a row of holes in the same test conditions have been compared for different blowing rates (Figure 4). For
m values not exceeding 1, a good agreement can be noted. Beyond 1, deviations are recorded, which increase with m and only reduced abscissa. It can generally be observed that the superposition principle underestimates the film effectiveness at high blowing rates.

Fig 4: Comparison of the laterally averaged film cooling effectiveness for a row of holes with that predicted by superimposed single hole measurements from GOLDSTEIN (1970)

Film effectiveness and heat transfer coefficient resulting from hole injection depend on geometric parameters such as lateral hole spacing, row spacing, curvature and injection angle, and on flow-related parameters such as blowing rate. Each parameter has been analyzed, separately. In this case, it is difficult to predict boundary conditions in a new configuration not necessarily corresponding to those used for basic studies. A determining assumption was therefore retained to compute boundary layers for the geometries presented in this paper. It assumes that effects dissociate from one another and remain the same whatever the geometry.

Boundary conditions are then obtained from a reference configuration, to which a series of corrections are applied as a result of the differences in the parametric values of the studied geometry versus the reference geometry. This approach provides local film effectiveness and heat transfer coefficient distributions. These variables are then averaged laterally and the result is compared with the values obtained by the above-mentioned correlations. If there is any difference, then the level of local distributions is corrected by a constant factor. This adjustment is all the more justified since the three-dimensional temperature field was studied on a vane profile identical to that used to establish the experimental correlations. It could be stated on this occasion that the correction factor varies very little with abscissa X and that it is therefore practically the same over the whole studied area.

A few essential characteristics pertaining to effectiveness and heat transfer coefficient extracted from the literature are dealt with in the paragraphs hereafter.

Local film effectiveness close to an injection

The typical film effectiveness obtained downstream of a hole is illustrated in Figure 5. The overall profile is maintained over a large range of blowing rate values, only amplitudes are different. The effectiveness, which is high downstream along the hole centerline for Z/D not exceeding 0.5, decreases to tend towards an asymptotic, laterally uniform value. The overall effectiveness becomes very low beyond 30 to 40 diameters downstream of injection.

Fig 5: Axial film cooling effectiveness distribution for injection through a single hole at an angle of 35° with main flow \( U = 61 \text{ m/s}, \ m = 1.0 \) - from GOLDSTEIN (1970)

FOSTER and LAMPARD (1979) analyzed the effect of the injection angle. Effectiveness varies downstream of the injection depending on the injection angle and blowing rate, but it tends to a given value, as soon as X/D exceeds 40. Effectiveness quickly becomes more uniform at large injection angles. These authors also performed an experimental study on the effect of hole spacing on film cooling effectiveness. At high blowing rate values, low-amplitude variations, slightly increasing with hole spacing, are recorded. This effect is much more pronounced as m decreases. For Z = 0 maximum effectiveness values vary little but effectiveness between two holes decreases as spacing increases.

ITO (1976) studied the effect of curvature on film cooling effectiveness. It can be observed that effectiveness is better on a convex wall than that on a flat wall, which in turn is better than that on a concave wall.

The influence of the boundary layer thickness is well known: effectiveness generally decreases as the boundary layer thickness increases, whatever the blowing rate, see FOSTER and LAMPARD (1979) and GOLDSTEIN (1970).

The blowing rate variation always results in the same type of evolution as far as effectiveness is concerned. This m function is ascending up to m = 0.5, then descending. At low blowing rates, it decreases with the corrected X/D abscissa, whereas for m greater than 1, effectiveness passes through a maximum which moves downstream as m increases.

Local heat transfer coefficient close to an injection

The paper presented by GOLDSTEIN and TAYLOR (1962) is particularly rich in information on this subject. The local heat transfer coefficient distribution was recorded from flat plate tests, whose principle is based on the existence of an analogy between heat transfer coefficient and mass transfer coefficient. The effect of injection on the...
heat transfer coefficient is expressed as a heat transfer ratio with and without injection. Figure 6 accurately describes results generally recorded as isovalues over the studied surface. Several regions can be distinguished:

- Region A, upstream of injection holes, where heat transfer is barely affected by the presence of jets. Only a slight heat transfer coefficient decrease at low blowing rates and a slight increase at high blowing rates can be observed.

- Region B, intermediate hole region, where the effectiveness is small, except at high blowing rates, for which the heat transfer coefficient increases by about 10%.

- Region C, immediately downstream of hole and of relatively small size (about 0.3D). The Hf/Ho ratio is slightly higher than 1. It can be explained by the physical phenomenon of jet separation due to the injection angle.

- Region D, lateral hole region, where the Hf/Ho ratio is high. Mainstream interaction with the jet explains this result. The extent of this area is a function of the blowing rate and maximum Hf/Ho values are about 2.5 to 3.

- Region E, downstream of the holes, where the Hf/Ho ratio is high.

- Finally, region F, recorded at high blowing rates, where Hf/Ho values higher than 1.25 are observed.

These tests also demonstrate that the Hf/Ho ratio distribution rapidly becomes uniform for X/D values higher than 4.

The Hf/Ho ratio distribution depends on jet reattachment. In this case, the author conducted the study on a plate with one injection through one row of holes with a hole spacing equal to three diameters. The jet hit the surface before coalescence with adjacent jets occurred. This distribution may be considered reproducible for other hole spacings, provided jets do not mix before reattachment. This applies as soon as hole spacing exceeds two diameters.

**Fig 6 : Regions of high and low mass-transfer coefficient around injections holes from GOLDSTEIN and TAYLOR (1982)**

**INTERNAL BOUNDARY CONDITIONS**

These conditions cover the heat transfer coefficient on the internal surface and the heat transfer coefficient inside the holes of the studied area. The coolant flows radially within the internal cavity and feeds all injection holes. In this configuration, as the cavity section varies little and does not feature any turbulence promoters, the convective heat transfer can be taken as that of a flat plate, and constant on the whole internal surface.

The heat transfer coefficient inside the holes is also calculated from experimental correlations. For injection holes, the l/D ratio is generally small (2 to 3) significantly increasing the heat transfer coefficient level. The short l/D values result in heat transfer up to 100% greater than for fully developed flow. The experimental correlation is of the form: \( Nu = f(Re, Pr, l/D) \) where \( l \) is the distance along the holes measured from coolant side.

**NUMERICAL RESULTS**

The thermal equilibrium is obtained by solving the heat equation for the considered geometry. For the complex three-dimensional geometries to be studied, this equation cannot be solved analytically. Therefore numerical techniques based on approximation methods suitable for a discretized field must be applied. The code used is a three-dimensional finite-element code based on a variational principle.

**Flat plate**

An analysis was done for the temperature field determined by injection through two staggered rows of circular holes, with hole and row spacings equal to three diameters. The injection was considered only partially taking one half-hole in each row, since the hole pattern ensures symmetrical conditions and the upstream boundary conditions were assumed to be uniform laterally. The injection angle is 55° and the hole diameter is 0.5 mm. The thickness of the plate is 1 mm and the conductivity of the material, depending on temperature, is of the order of 30 W/m°C. Discretization by finite elements was carried out in the injection zone. Most elements are quadrangular and defined by eight basic nodes. Higher-level elements featuring intermediate nodes are used to represent curved hole edges. The size of elements varies according to their location and their edge dimensions range from 0.3 to 1.5 mm. The upstream and downstream boundary conditions and zero flux condition, and were far removed from the hole area. The external face boundary conditions are representative of those prevailing on vane profiles for a given injection location. Local film cooling effectiveness and heat transfer coefficient distributions are illustrated in Figure 8. The mainstream temperature is 200°C and the coolant temperature 500°C.

Three calculation cases showed the effects of the injected film and the convection phenomenon inside the holes on the temperature field. In all cases, an upstream film was assumed to be uniformly laterally applied to the external surface. A second film injected from modelled holes was considered for the second and third calculations.

The first case is the temperature field analysis with no injection in the holes. Heat transfer coefficient inside the holes is zero and the external
boundary conditions are illustrated in Figure 7 as a function of the reduced abscissa X/D. Heat transfer coefficient and coolant temperature were constant over the whole internal surface and were equal to 1500 W/m²°C and 500°C respectively. The resultant temperature field (Figure 9a) is then two-dimensional, except for the injection area, which is three-dimensional due to the presence of holes. The hottest zone is located downstream of the hole, where effectiveness is the lowest. Zero flux conditions on upstream and downstream surfaces result in the presence of a heat flux within the plate thickness only, thus providing isotherms perpendicular to these surfaces.

This condition applied on the upstream side affects the temperature field in the hole area more than the one applied on the downstream side because it is the nearest geometrically. In the injection area, isotherms are oriented differently, whereby evidencing that a heat flux circulates upstream from the downstream region of the plate, where effectiveness is the lowest. Isotherms show the strongest concentration in this region and perpendicularity to the wall does not exist as a convective heat transfer prevails on the internal surface. Although this configuration is not realistic considering the high temperature levels, it may be taken as a basis to measure the effect of injection on the temperature field.

The second calculation case provides the temperature field of the part by considering only the effect of the injected film from the holes and the upstream film on external boundary conditions. A zero heat transfer coefficient in the holes was taken. External surface boundary conditions are given in Figure 7 for average film cooling effectiveness and heat transfer coefficient values and in Figure 8 for the local distribution of these parameters downstream of the injection holes. In this case, film cooling effectiveness is obtained by superposing both films. The selected blowing rate is 1. So the average effectiveness is 0.51 and the Hf/Ho ratio equals 2 immediately downstream of the second row of holes; then both values decrease. As illustrated in Figure 9b, the overall temperature level is lower. The metal temperature reduction reaches 105°C in the downstream portion of injection. Paradoxically the highest temperature is observed at the holes. This results from the significant increase of heat transfer coefficient in this area, which, in this case, predominates over the existing effectiveness. This is even more pronounced at the second hole, where upstream effectiveness is practically zero, whereas the heat transfer coefficient is already increasing. The temperature field quickly becomes two-dimensional again downstream of injection, because jet coalescence is beginning and conduction within the plate tends to homogenize the temperature laterally. Another reason can explain this result. A high effectiveness level downstream along the hole centerline is concomitant with a high heat transfer coefficient due to jet reattachment. This results in a convective flux balance at a determined metal temperature, close to that obtained between two holes, where film cooling effectiveness is actually lower but which is also characterized by a lower heat transfer coefficient. However this overall convective flux balance on the external surface differs from that obtained by two-dimensional computation. This is illustrated in Figure 10 which plots differences between the two-dimensionally computed temperature where the data is averaged in the pitchwise direction and the three-dimensionally computed, laterally-averaged temperature over the external flat plate surface. The results show the gain brought by three-dimensional modelling. Differences reach 20°C, with the highest values near the holes, a region where film cooling effectiveness and heat transfer coefficient pitchwise variation are the strongest. Then differences decrease and they are small in the downstream portion of the studied area.

The last calculation includes the convective heat transfer in the holes. Heat transfer coefficients are generally very high. This in particular results from the inlet effect appearing when coolant passes from a large cross-section cavity to small holes. Considering injection conditions, the heat transfer coefficient with the inlet effect is 8000 W/m²°C. Figure 9c shows how important this factor may be. The temperature field is strongly modified. It is less uniform at the holes and the high temperature gradient within the thickness indicates that the ejected heat flux is high. The result is a considerably reduced temperature at this location. The average temperature decrease is 105°C in the injection area and the convection effect in the holes affects the regions upstream and downstream of this area. This effect is probably underestimated, since the heat transfer coefficient values selected for the internal surface at the hole inlet are not representative of actual values. In reality, the heat transfer coefficient is not uniform over the whole internal surface. Flow acceleration at hole inlet modifies this value noticeably. A comparison was also made with the equivalent two-dimensional computation. The latter was performed applying average boundary conditions and simulating the convection effect in the holes in the injection region through conduction elements featuring heat sinks. Equivalent
The effects of blowing rate and hole spacing, two significant parameters, have been included for this type of geometry. The blowing rate influence ($m = 0.5, 1.3$ and $1.5$) is presented in Figures 11a and 11b. At the external wall, results give temperature differences compared to the reference case temperature as a function of the X/D reduced abscissa. The values obtained by three-dimensional computations are average values along the lateral axis $Z$, the reference case being that with a blowing rate equal to unity. Two configurations have been envisaged. The first one is concerned with the effect of external surface boundary conditions only, the heat transfer coefficient in the holes remaining constant. Figure 11a shows that only the area downstream of injection is affected by this change. Differences, which are practically nil up to the first row of holes, increase more and more downstream. They reach $1.65, -2.9$ and $-2.14 \%$ respectively for $m = 0.5, 1.3$ and $1.5$, i.e. a temperature difference of about $17, 29$ and $22^\circ C$. It seems that an optimum blowing rate ($m = 1.3$) exists to ensure plate cooling. For higher $m$ values, film cooling effectiveness increases less than the heat transfer coefficient, which results in higher metal temperatures. In addition to the characteristics of the first configuration, the second one also includes
the heat transfer coefficient variation in the holes as a function of cooling flowrate. Internal surface boundary conditions are assumed to be constant. Figure 11b shows changes in $\Delta T/T$ differences observed mainly on the upstream portion and in the injection area. In this region, temperature decreases with the blowing rate. Significant variations are recorded reaching $4.5\%$ for $m = 1.5$, i.e. about $45^\circ C$.

Hole spacing influence is presented in Figure 12. This figure is based on hole spacing values equal to $4D$ and $6D$ for a constant blowing rate ($m = 1$). In these conditions the flowrate is reduced by $25\%$ and $50\%$ respectively compared to the initial configuration. Film cooling effectiveness and heat transfer coefficients were recalculated using the superposition principle from values given for one single hole. Increasing the hole spacing leads to a slower jet coalescence and a noticeable decrease of the average effectiveness. The temperature increase is even more pronounced in hole vicinity. The convection effect in the holes decreases in the same proportions as the number of holes. The temperature field is very sensitive to this parameter. Differences reach $9\%$, i.e. about $90^\circ C$, when hole spacing changes from $3$ to $4$ diameters and $17\%$, i.e. about $170^\circ C$, when hole spacing is doubled. Local increases can be even higher.

Fig 12 : Hole spacing influence

### Vane geometry

The study was conducted on a turbine vane suction side portion. It incorporates three successive injections over an airfoil height of $15\,\text{mm}$. The three injections consist of two rows of holes with a hole spacing of $3$, $6$ and $3.5\,\text{mm}$ and an injection angle of $90$, $55$ and $40$ degrees respectively. Hole diameter is $0.5\,\text{mm}$. The type of spatial discretization used is the same as for the flat plate and includes about $3000$ conduction elements. The last injection has been simulated over the whole height by conductive elements featuring heat sinks. A two-dimensional representation has been considered downstream of the injection. A local distribution of external boundary conditions has been chosen from the leading edge to the portion upstream of the third injection. These conditions become uniform very quickly downstream of the first injection, because hole spacing is small and injection angle is large. A zero flux condition has been applied to the upstream surface at the leading edge. In practice this condition is justified by the pattern comprising two injections of the same geometry located symmetrically about the leading edge. The downstream surface also includes a zero flux condition. Its position has been determined by a preliminary two-dimensional computation on the whole vane giving the place where the temperature gradient exists within the wall thickness only. The same principle was applied to delimit the wall between cavities. Cooling flowrates at the operating point represent $0.45$, $0.52$ and $1.85\%$ respectively of the mainstream for the three injections, the blowing rate being equal to $2.5$ at the first injection and $1$ at the other two. Internal heat transfer coefficient and coolant temperature are constant for the two cavities. Heat transfer coefficients inside the holes are a function of the emitted flowrate and hole geometry. They are equal to $11000$, $8000$ and $7500$ $\text{W/m}^2\text{K}$ respectively.

The temperature field obtained in these conditions is shown in Figure 13a. Hot spots appear at the leading edge and downstream of the second injection. The leading edge is one of the most critical area to be cooled on a vane, because effectiveness is zero at this point and the heat transfer coefficient is high at the stagnation point. In fact, convection in the holes of the first suction side injection and those of the first pressure side injection contributes to cool this zone efficiently. The temperature field remains uniform laterally, except for the area comprised between the second and the third injection. Periodic variations of $25$ to $30^\circ C$ amplitude are recorded. This results from the wise hole spacing at the second injection. Injections are clearly characterized by convection in the holes, more pronounced at the first injection, as the heat transfer coefficient in the hole is higher and hole spacing smaller. Temperature gradients within the wall thickness are larger there. Local values may reach $90^\circ C/mm$. The gradient within the thickness, at the third injection simulated by conductive elements with heat sinks, is of the same order of magnitude as for the first two injections. The flux discharged through the wall between cavities is not negligible, as shown by the isotherm profile. Isotherms are concentrated and perpendicular to the wall. The vane curvature affects the temperature field noticeably. Comparison with a flat plate can be qualitative only. In this case, convexity favors convective heat transfers over the external surface, which is more extended than the internal surface. This element contributes to reduce the temperature gradient within the wall thickness when compared to a flat plate with similar internal and external boundary conditions. A two-dimensional computation was also applied to this geometry. Like the third injection, the first and second ones were represented using conductive elements with heat sinks. Average boundary conditions were selected as a function of the abscissa. Comparisons were made between the temperature obtained through this computation and the average temperature obtained through three-dimensional computation. Figure 14 shows the deviations obtained in this configuration are smaller than those obtained with a flat plate. For injection E1, it can be explained by the fact that the deviation is almost exclusively due to the different types of geometric...
modelling applied, the local boundary conditions being close to the average boundary conditions. For injection E2, hole spacing has been increased, thus reducing the differences between the two models.

\[ T = T_i - T_e \]

\[ T_g = 202700 \]

Blowing rate: \( m = 2.5 \) \( f_{dr} \)

\( E_1 = 1 \) for E2 and E3

\( T_i = \) Temperature given by two dimensional calculation

\( T_e = \) Temperature given by three dimensional calculation

Fig 13: Temperature field (in °C) of a blade section side part

CONCLUSION

The process used to determine external boundary conditions consists in performing calculations in several steps, which are:

- computing boundary layer and heat transfer coefficient without a film,

- establishing film cooling effectiveness and average heat transfer coefficient as a function of abscissa \( X \),

- determining the local effectiveness and heat transfer coefficient distribution.

This process mainly relies on a series of experimental results and on the application of the superposition principle. Currently available values will be completed later on with the results of numerical flow modelling for this type of configuration.

Three-dimensional modelling of a vane incorporating film injections through holes allowed the calculation of an accurate temperature field and an understanding of the thermal behavior close to the holes. This modelling shows that cooling is mainly provided by convection in the holes for the near-hole area and by the film downstream of the injection. Although the temperature field rapidly becomes two-dimensional downstream of the injections, this modelling provides a better temperature prediction than two-dimensional modelling, since the flux balance is taken into account integrally over the whole geometry.

A calculation has been performed on a vane geometry, assuming that one hole in the injection was obstructed. Significant effects on the vane temperature field have been observed. In severe operating conditions, the life of such a part may be strongly reduced.
To refine the three-dimensional temperature field, it would be desirable in the future to investigate the local distribution of internal boundary conditions. This point will essentially apply to the area located at the hole inlet.

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