THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

95-CTP-90

A ROTOR BLADE COOLING DESIGN METHOD FOR HEAVY DUTY GAS TURBINE APPLICATIONS

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ABSTRACT

The increase in gas turbine performance allows for very high total inlet temperatures even in heavy duty applications. Therefore, an accurate rotor blade cooling design is necessary to obtain very high reliability and efficiency. This paper presents a simple and fast procedure to study global cooling systems and to evaluate a blade temperature distribution in a cooled rotor cascade. The blade cooling design is developed by a modular approach: it allows one to select any combination of several cooling techniques.

NOMENCLATURE

A = section
b = rib height
c_p = pressure constant specific heat
d = diameter
d_h = hydraulic diameter
e^* = roughness Reynolds number
f = friction coefficient
F_c = centrifugal force
H = heat exchange coefficient
k = thermal conductivity
h = specific enthalpy
m = mass flow rate
Nu = Nusselt number
P = pressure
q = exchanged heat
R = gas constant
r = radial coordinate, radius
s = s-coordinate
Ra = rotational Rayleigh number
Re = Reynolds number
Ro = Rossby number
St = Stanton number
T = temperature
T_{adw} = adiabatic wall temperature
T_{cin} = inlet coolant temperature
u = velocity
X = pin fin pitch in coolant flow direction

Subscript

o = nominal state
c = coolant
b = blade
g = hot gas
w = wall
max = maximum velour

Greek

β = thermal expansion coefficient
μ = viscosity
ρ = fluid density
Ω = rotation speed
Φ = cooling efficiency

INTRODUCTION

In a real gas turbine cycle, one could demonstrate that the efficiency and specific power are positively influenced by the increase in the fire temperature. This increase must correspond to materials of higher resistance for nozzles and rotor blades in the first turbine stages of high temperatures. But the material resistance is usually not enough.

In previous works (Stecco and Facchini, 1988; 1989a; 1989b; 1989c), the authors have already shown that the use of blade cooling reduces the performance advantages of the increased maximum temperature; by improving the design for cooling systems, together with the materials' heat resistance, these losses

Presented at the 1995 ASME Cogen-Turbo Power Conference
August 23-25, 1995 - Vienna, Austria

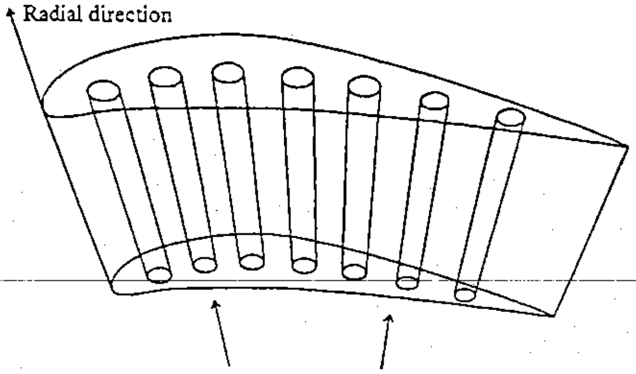

FIGURE 1: TUBES COOLING SYSTEM

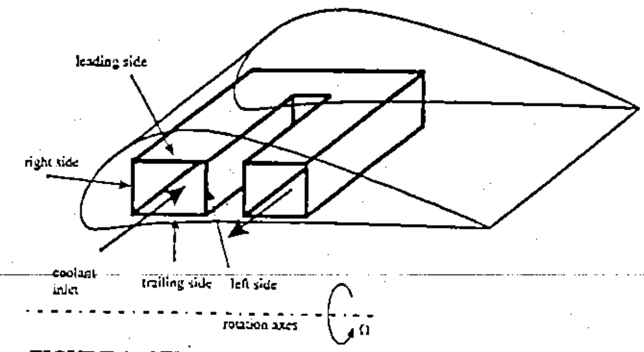

FIGURE 2: SERPENTINE MULTIPASS COOLING SYSTEM

can be limited, thereby raising the maximum temperature levels, and obtaining real increases in the efficiency and the specific power of gas turbine plants.

Several researchers have studied the elementary blade cooling techniques. The most interesting works for our research are those of Florschuetz et al., 1981, Florschuetz and Isoda, 1983; Florschuetz et al., 1984, Florschuetz and Tseng, 1985, (impingement); Han et al., 1985, Zhang et al., 1993, Mochizuki et al., 1994, (multipass); Armstrong and Winstoley, 1988, (ribs and pin fins); L'Ecuyer and Soechting, 1985, (external film cooling).

A typical cooling system combines different solutions both inside and outside of the blade (Carcasci et al., 1994) to allow for cooling and geometry requirements. The determination of the surface blade temperature distribution is the critical goal that leads to good design and reliability.

Therefore the cooling system studies also requires an evaluation about the heat conduction in the metal blade and the hot gas/blade surface heat transfer.

Following this way, the authors have already studied stator blade cooling problems in the heavy-duty gas turbine field, realizing a code, called COOL, already used by an industry (Carcasci et al. 1994). This previous work is based on a meridian blade section study and considers obviously only the Newtonian reference frame effects; every cooling techniques is studied by a one-dimensional simulation and the combination of these techniques is realized by a modular approach. Two-dimensional study (on a blade-to-blade section) of blade stationary heat conduction and of hot gas/blade surface heat transfer (based on Navier-Stokes equations) are also performed.

The approach used for internal cooling rotor simulation follows the way of the stator simulation, but in this case, it is developed in the radial direction. So the coupling with other two-dimensional simulations (again on the blade-to-blade plane but repeated at various radius value) allows to do a simplified three-dimensional approach. These considerable simplifications are justified by the heavy duty applications goal of the developed simulation procedure.

ROTOR COOLING MODEL

Elementary cooling techniques

The code COOL, developed for the nozzle blades, has been extended at the rotor blade, and for this case, the internal cooling systems described are:

- Radial Tubes: a simple cooling system which consists of a series of a radial ducts. The exhaust coolant of each tube is discharged at blade tip (figure 1).
- Serpentine passage: a more complex cooling passage configuration, which consists of a sequence (multipass) of ducts, alternatively, in centrifugal or centripetal direction covered (figure 2). The turbulence promoters (ribs) effects are also considered.
- Pin fins: a series of small cylinders with their axes orthogonal to the radial coolant flow

It is possible to consider other cooling systems already treated in stator blade simulations (Arnone et al., 1993; Carcasci et al., 1994), such as internal impingement and external film cooling.

Non-Newtonian reference frame equations

The effects of no-Newtonian reference frame, typical of the rotor blade, must be considered. The more relevant influence is due to:

- Centrifugal force, which heavily modifies the flow condition
- Coriolis force, which determines secondary flows. They modify the internal heat transfer
- Buoyancy force (low influence), which is determined by the centrifugal mass force field when there is a high density gradient in the coolant flow.

The modified fundamental equation of stationary and one-dimensional flow inside the cooling ducts are:

1) mass continuity equation

$$d(\rho \cdot u \cdot A) = 0 \tag{1}$$

2) momentum equation

$$d\left(p+\rho\cdot u^2\right)=\left[-\frac{\tau}{A}\pm\rho\cdot\Omega^2\cdot r\pm\frac{d(\rho\cdot r)}{dx}\cdot\Omega^2\right]\cdot dx \qquad (2)$$

3) energy equation

$$d\left(h+\frac{u^2}{2}\right)=\left(q\pm\Omega^2\cdot r\right)\cdot dx \qquad (3)$$

4) state equation

$$p=\rho\cdot R\cdot T \qquad (4)$$

where:

h is enthalpy (**h**=$\mathbf{c_p}$·(**T**-$\mathbf{T_o}$))

q is the heat flux ($q=\dfrac{4\cdot H\cdot(T_w-T)}{\rho\cdot u\cdot d_h}$), which depends on the heat transfer coefficient **H**, which can be determined by experimental correlations. τ is the skin friction ($\tau=\dfrac{1}{2}\cdot f\cdot\rho\cdot u^2\cdot d$), which depends on **f** and can be calculated by correlations too.

In conclusion, there are four equations and four unknowns (ρ, **p**, **T**, **u**).

In these equations, two terms represent the no-Newtonian reference frame effects:

The term ($\pm\Omega^2\cdot r\cdot dx$), in equations 2 and 3, depends on centrifugal force effects. This force favors the flow if it is centrifugal, and increases the total outlet pressure.

The term ($\pm d(\rho\cdot r)\cdot\Omega^2$), in equation 2, is due to buoyancy phenomena, which influences only total pressure.

The sign "+" (plus) refers to the centrifugal direction of the flow, and the sign "-" (minus) refers to the centripetal direction.

The heat transfer coefficient determination depends to particular cooling systems. The consideration of non Newtonian reference frame phenomena, requires the particular dimensionaless numbers introduction:

- $Ro=\dfrac{\Omega d}{u}$ Rossby number: ratio between Coriolis force and Newtonian force
- $Ra=\dfrac{\Omega^2\cdot r\cdot\beta\cdot\rho^2\cdot c_p\cdot d_h^3\cdot\Delta T_w}{\mu\cdot k}$ Rotational Rayleigh number: ratio between buoyancy force and viscous force.

Radial Tubes

In this case (figure 1) the flow inside the radial ducts is always centrifugal (from hub to tip of the rotor blade). The solution is conditioned by a low efficiency ϕ value and requires a high coolant mass flow rate, and so it is used when there is a fairly low fire temperature. With respect to the stator case, modifications to the heat transfer coefficient correlations are necessary too. These are due to Coriolis forces, which induce a secondary flow vortex and buoyancy; Morris and Ayhan already in 1984 presented a simple correlation:

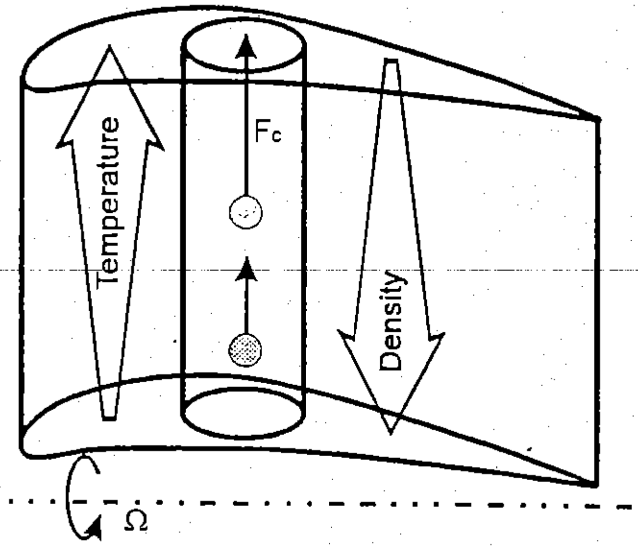

FIGURE 3: BUOYANCY FORCE SCHEME

$$Nu=0.022\cdot\mathrm{Re}^{0.8}\cdot\left(\frac{Ra}{\mathrm{Re}^2}\right)^{-0.186}\cdot Ro^{0.33} \qquad (5)$$

The heat transfer coefficient decreases with the Rayleigh number (buoyancy effects), and it increases with the Rossby number (Coriolis force effects). In the centripetal flow direction, the non Newtonian reference frame effects are considered opposing (Morris and Ayhan, 1984):

$$Nu=0.036\cdot\mathrm{Re}^{0.8}\cdot\left(\frac{Ra}{\mathrm{Re}^2}\right)^{0.112}\cdot Ro^{-0.083} \qquad (6)$$

To increase heat transfer, the tubes section area is usually selected to obtain a very high flow velocity (with Mach number near to one).

Serpentine passage (multipass)

This is a typical solution for a rotor cooling passage, because it permits a higher cooling efficiency and a lower coolant flow rate than simple radial tubes. Obviously in this case a more complex blade development procedure is necessary and this is recommended only with a more relevant hot gas temperature. The present heat transfer study is limited (figure 2) to the outward and inward branches of the duct. In the curve zones, only pressure loss are considered. However regard to the heat transfer coefficient, some experimental results (Mochizuki et al., 1994) report a great turbulence increase and so a cooling effect improvement.

The non Newtonian reference frame effects depend on serpentine zones. In the outward duct, the centrifugal forces are favorable to flow motion, but the buoyancy, due to the same force field, can offer resistance to motion when a very high density gradient occurs in the coolant and determines the buoyancy in the opposite direction to the mass force field (figure 3). The opposite situation is related to inward ducts, where the

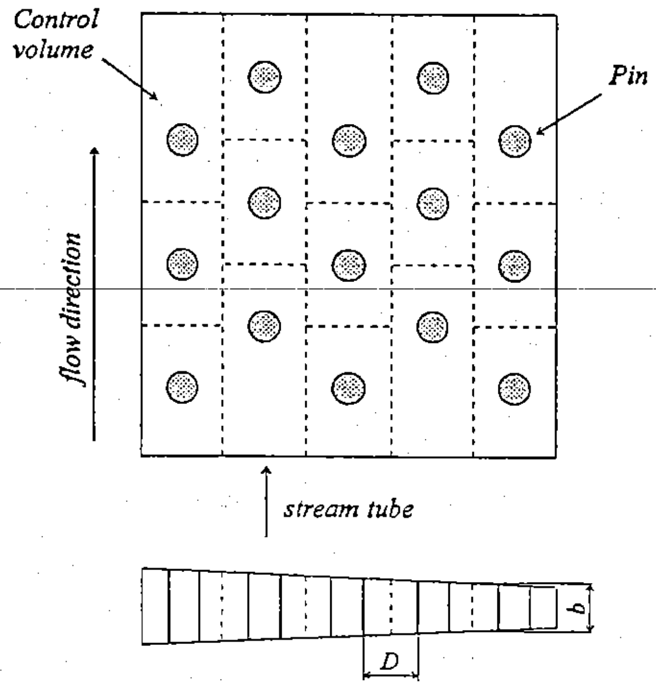

FIGURE 4: PIN FINS COOLING SYSTEM SCHEME

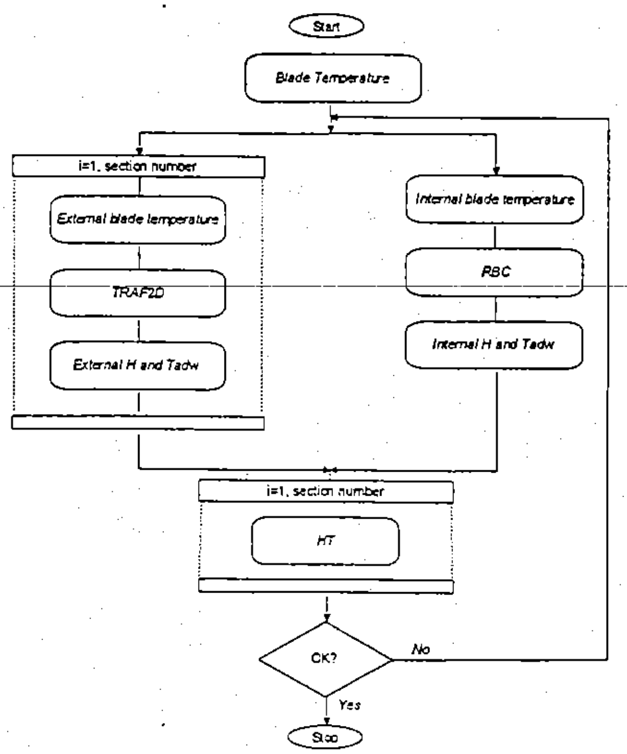

FIGURE 5: FLOW CHART CALCULATION

centrifugal forces are opposed to motion and, in the contrast, the buoyancy effects are ever favorable. As in the tubes, the heat transfer coefficient correlations depend on the Coriolis force and buoyancy effects.

Often in the serpentine passages, the turbulence promoters, *ribs*, are reported. The introduction of ribs allows a heat transfer coefficient increase, which depends on relative incidence (α) between ribs and coolant flow (Han et al., 1985).

The heat transfer coefficient determination is obtained through the $\mathbf{H_r}$ parameter correlation below reported (Han et al., 1985):

$$H_R = \left(\frac{P/e}{10}\right)^{0.14} \cdot 2.83 \cdot \left(\frac{\alpha}{90°}\right)^{0.3} \cdot \left(\bar{e}^{+}\right)^{0.28} \qquad (7)$$

where $\left(\bar{e}^{+}\right)$ is the Reynolds roughness number, and it depends on skin friction $\mathbf{f}$ and the $\mathbf{H_R}$ parameter depends on Stanton number which just permits the heat transfer determination.

The mean velocity in the serpentine is lower than in the tubes solution, because the passage section area is much higher, so a reduction in consistent pressure loss is obtained.

Pin Fins

The use of pin fins, in the internal ducts of gas turbine cooled blades, allows an increase of the turbulence and the heat transfer area, coupled with an higher blade stress resistance. For this reason the pin fins are introduced in the rear blade part, near the trailing edge zone (figure 4). More than two or three rows of pin fins are used and so a simple one-dimensional approach it is not correct because the various "stream tubes" (defined by pin fin rows) area is not equal in rear blade zone (figure 4). In this case, the one dimensional approach is related to each "stream tube", and an orthogonal (with respect to the radial flow direction) flow equilibrium is imposed for a calculation closure. The flow direction can be centrifugal or centripetal, but the pin fins localization usually involves the centrifugal one or, in some cases, a connection to axial ejection coolant at trailing edge can be introduced.

The correlations choice for the heat transfer coefficient determinations it depends on pin fins geometry. Particular influence is related to b/D parameters (see figure 4) and two different correlations are reported in literature: for short pins, the Metzger correlation (Metzger et al, 1981, 1982, 1986) can be used:

$$Nu_D = 0.135 \cdot \mathrm{Re}_D^{0.69} \cdot \left(\frac{X}{d}\right)^{-0.34} \qquad (8)$$

for long pins, the Faulkner correlation (Faulkner, 1971):

$$Nu_D = \left[0.023 + \frac{4.143 \cdot e^{\left(-3.091\frac{d}{Y} - 0.89\left(\frac{Y}{b}\right)^{0.5075}\right)}}{Re_D^{0.2945}}\right] Re_D^{0.8}\, Pr^{1/3} \qquad (9)$$

CALCULATIONS CODES

The relationships presented above for a selection of techniques and geometric of rotor cooling conditions, has been used for a calculation code development. The previous COOL

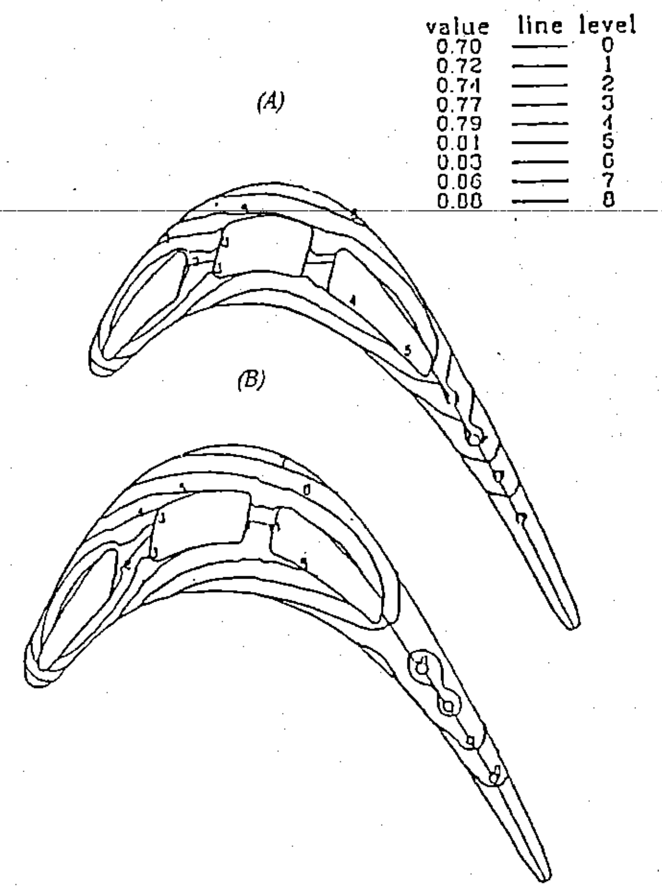

FIGURE 6 BLADE COOLING GEOMETRY WITH TUBES FOR THREE RADIUS: (A) TIP SECTION, (B) MEAN SECTION, (C) HUB SECTION.

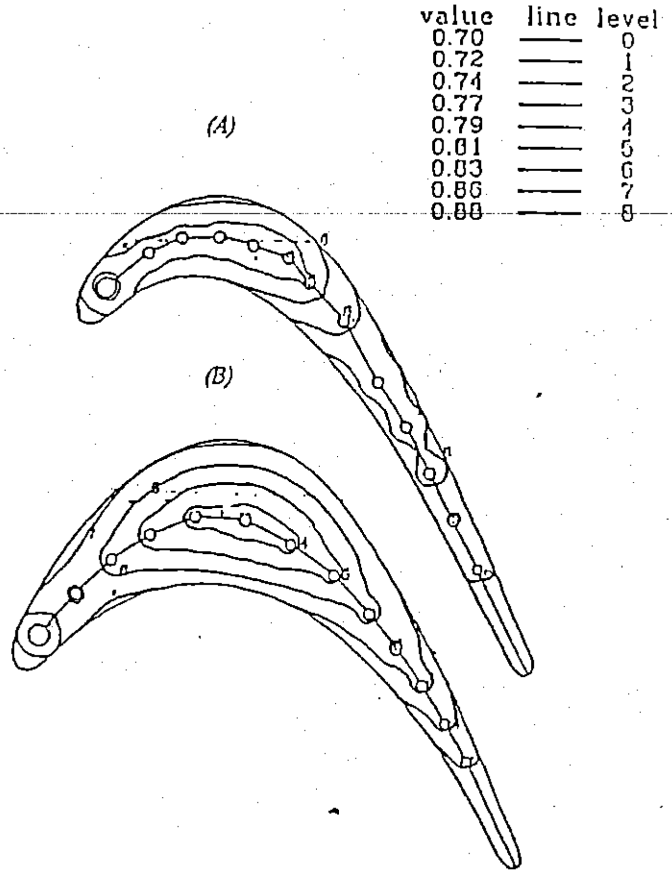

FIGURE 7 BLADE COOLING GEOMETRY WITH MULTIPASS SERPENTINE FOR THREE RADIUS: (A) HUB SECTION, (B) MEAN SECTION, (C) TIP SECTION.

code (Arnone et al., 1993; Carcasci et al., 1994) developed for a stator blade cooling simulation, has been modified, and a new COOL version for a rotor blade cooling has been developed.

The approach for the systems cooling description is modular, therefore it permits to define any possible combination of cooling techniques. The model is based on the linearization theory, which has been applied by the authors in the energy systems simulations field (Carcasci and Facchini, 1995). The main program manages modular subroutines on the bases of a linear matrix, which defines:linearized internal characteristics (for example $\Delta p=k_1 m+k_2$)

connection modules: continuity equations of pressure and mass flow rate

input data: boundary conditions (i.e.: outlet static pressure, inlet total pressure and total inlet temperature)

The good flexibility and the easy of enlargement by adding other cooling techniques or other correlations are the most interesting characteristics of the code.

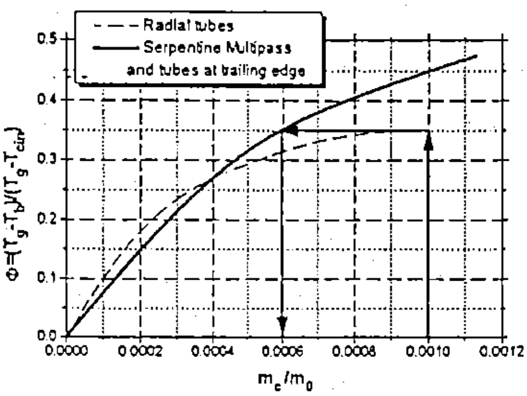

FIGURE 8 EFFICIENCY WITH RADIAL TUBES AND SERPENTINE MULTIPASS COOLING SYSTEMS

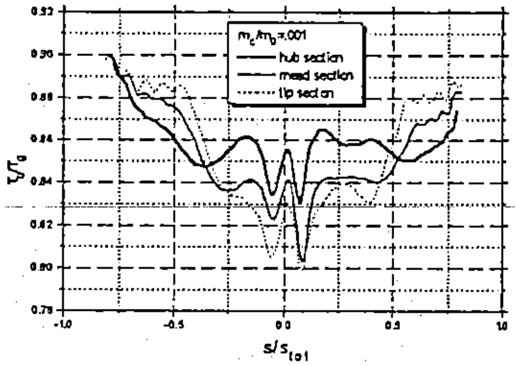

FIGURE 9: EXTERNAL BLADE TEMPERATURE IN THREE DIFFERENT RADIUS WITH TUBES COOLING SYSTEM

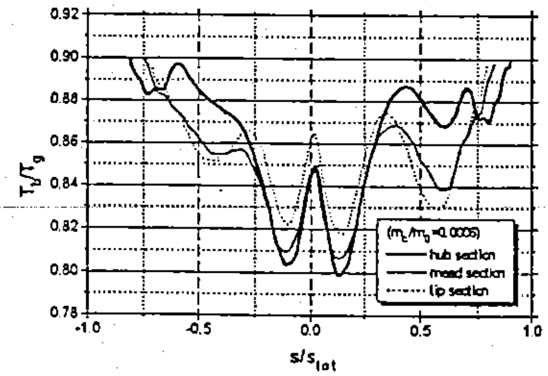

FIGURE 11: EXTERNAL BLADE TEMPERATURE IN THREE DIFFERENT RADIUS WITH SERPENTINE MULTIPASS COOLING SYSTEM

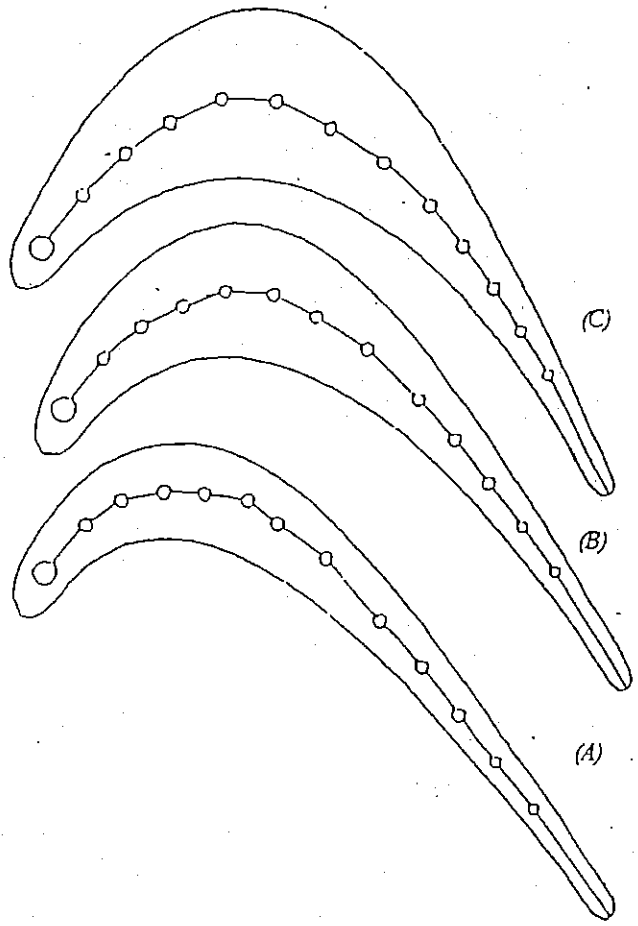

FIGURE 10: TEMPERATURE CONTOURS LEVELS WITH TUBES FOR TWO RADIUS: (A) TIP SECTION, (B) HUB SECTION.

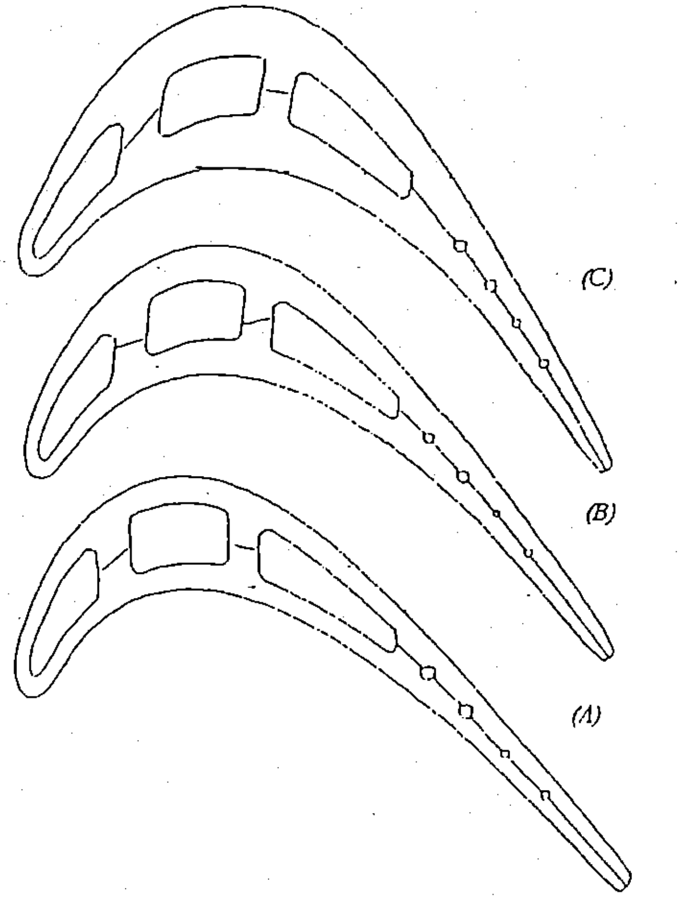

FIGURE 12: TEMPERATURE CONTOURS LEVELS WITH SERPENTINE MULTIPASS FOR TWO RADIUS: (A) TIP SECTION, (B) HUB SECTION.

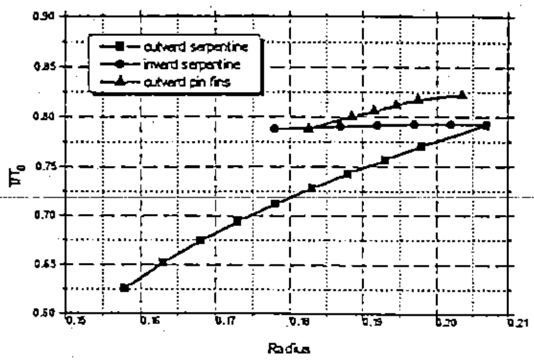

FIGURE 13: COOLANT TEMPERATURE

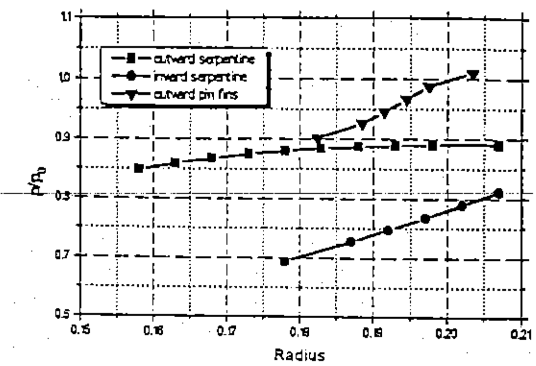

FIGURE 14: COOLANT PRESSURE

A solver of Navier-Stokes equations, called TRAF2D, has been used for a two-dimensional flow simulation in the blade-to-blade section. The TRAF2D code, developed by A. Arnone (Arnone, 1992), is based on a fourth-order Runge-Kutta integration scheme and on the Baldwin-Lomax turbulence model. The external heat transfer coefficient distribution has so been evaluated by this code and for a global blade study, more calculations are carried out at various radius lengths. An other code, called HT, is dedicated to the heat conduction simulation in the blade. This code is two dimensional (orthogonal section to the radius) and it has already been developed (Arnone and Sestini, 1992; Carcasci et al. 1994) for the cooled stator simulation and also in this case, more calculations are carried out at various radius lengths.The calculation procedure is shown in figure 5 where the global flow chart is presented. After a blade temperature initialization, the simulation starts with the internal (COOL) and external (TRAF2D) heat transfer coefficient and adiabatic wall temperature T_{adw} calculation (Arnone et al., 1993; Carcasci et al. 1994). The HT solutions permits the new blade temperature definition and, if the convergence is not verified, the procedure restarts. The COOL simulation is developed in the radial direction, and the TRAF2D and HT approach are related to a radial section, the shown procedure (figure 5) can almost be defined as a simplified three dimensional approach.

For reduction of calculation time, the TRAF2D code simulation is first carried out at various external blade temperature levels; so a simple interpolation substitutes the TRAF2D simulation in the flow chart (figure 5). In this case, on a PC 8486-33 Mhz, to execute all cycles until convergence, only 30 minutes are necessary. It is possible to execute this calculation on the power station, too in few minutes.

RESULTS

The simulation procedure has been applied to a preliminary design study for a rotor blade cooled by multipass serpentine passages with pin-fins. This rotor is designed for a small heavy-duty gas turbine: the PGT2 model manufactured by Nuovo Pignone S.p.A..

The first stage of new design is a substitution of the traditional radial tubes (figure 6) by a serpentine passage in the front part of the blade (figure 7) and so in this study we not consider the optimization of the blade rear part.

The old cooling system performance is shown in figures 8 where it is possible to see that the design efficiency value (Φ = 0.35) requires a great coolant mass flow rate and it appear very difficult an further improvement of Φ parameter.

Figure 9 shows the external blade temperature distribution at three radius values (the S/S_{tot} parameter is referred to suction side for positive values and to pressure side in the other case); the hub section results the most critical in the front part of the blade (abs(S/S_{tot}) < 0.5) and this fact would be dangerous for the blade resistance and reliability. An other representation of the blade temperature distribution is shown in figure 10 where its course in two blade meridian section are reported; it clearly appears that above hub section problem depends on the high blade thickness.

The serpentine passage studied for a cooling system redesign, shown in figure 7, is compound by three radial ducts, two centrifugal and only one centripetal. The introduction of pin fins, is limited to the last duct, moreover the ribs for the other ducts are provided. The course of the multipass cooling efficiency (figure 8) is better than tubes; in fact it achieves the same design efficiency value with the 40% coolant mass flow rate reduction.

Moreover the front part of the blade is more cooled (figure 9) and particularly in hub section too.

For the cooling of the rear part of blade, a few small tubes have been considered, even for a multipass solution. This fact is in agreement with the above mentioned target of this work so the performances increase appears very important because, in this case would be possible the use of axial tubes for trailing edge coolant ejection. Figure 12 shows the temperature distribution obtained with this solution: the contours levels appears more uniform than the tubes one in the front part of the blade.

The COOL simulation also permits the evaluation of coolant characteristics in the various ducts of the serpentine. The course of the temperature is shown in figure 13 where one notes the higher increase in the first ducts, in contrast with the almost-constant course in the centripetal one. The pressure distribution (figure 14) also depends on the direction of ducts flow and it slowly increases in the centrifugal duct and it heavily reduces in the centrifugal one.

CONCLUSIONS

The new version of COOL code, coupled with the TRAF2D and HT codes, permits the design of a rotor cooled blade using an approach which considers non Newtonian reference frame effects and appears almost three dimensional.

The presented procedure is very fast and is compatible with a personal computer or a small workstation.

The definition of the blade cooling system is modular and it is possible to combine the various elementary cooling systems as well as the new cooling system addition.

Interesting applications in the optimization of the rotor cooled blade of a small heavy duty gas turbine are presented.

ACKNOWLEDGMENTS

The authors would like to thank the Prof. E. Carnevale for his valuable contributions to this work, as well as to many others. They would also like to thank the Ing. E. Benvenuti and the Ing. U. Corradini from Nuovo Pignone S.p.A. for the valuable advice and useful discussions.

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